Chapter 1  The macro element method in crashworthiness calculations

the state-of-the-art and the future

by
Wlodek Abramowicz
(Impact Design, Europe)

Chapter 2  Macro Element Fast Crash Analysis of 3D Space Frame

Kenji Takada
Honda R&D Co., Ltd. Tochigi R&D Center

Wlodek Abramowicz
Impact Design, EUROPE

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Chapter 3  Using Visual Crash Studio for crash model simulation

A pole crush model example from the SMARTBATT project

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Introduction

The VCS is a software for pre design of crashworthy structures. Macro elements methodology of VCS is very fast. Thanks to that a user can make a number of concepts of a structure and test it in minutes which stay in opposition to software based on finite elements where user needs hours or days for one concept. VCS software has no direct software competitors. In VCS program user defines a concept for complex crashworthy structures, and then designs details in another software. VCS and software based on FE Methodology complement each other providing an overall design environment by combining fast pre design assessments with later detailed calculations.

The benefits of using VCS

VCS is especially useful at early stages of the design process when fast assessment of various structural configurations substantially speeds up product development process. It is an all-in-one environment for early design, which means that the user may easily test, modify and optimize the cross sections as well as complex structures. Compared to a FE model the VCS Macro element model gives similar results in substantially shorter period of time (seconds). The constantly improving modeling tools for 3D view enable the user to easily modify the tested construction.

VCS effectively combines complex macro element solutions at the level of individual elements with object oriented implementation of the global equilibrium solvers. Two major areas of application of the VCS are design and calculation of thin walled members for optimal crash performance and flexible crash simulation of structural sub-assemblies and complete space frame structures. It should be noted that early predecessor of VCS the CAE program Crash Cad was capable of calculation and design at the cross-section level only. Now the enhanced functionality of this program is included as one of several dedicated design modules of VCS.

The design of a structural member at the cross sectional level is especially important at the pre design and early design stages when the proper shape and optimal dimensions of a member are sought and the design concept undergoes frequent modifications. The macro element approach is especially attractive at this level as the calculation routines require as input only overall dimensions of the cross-section, and tensile characteristic of the material while the calculation process takes only few seconds on a standard PC. Consequently, designer can examine a wide range of cross sectional topologies and run several parametric studies within only few hours of work.

The design and flexible crash simulation of structural sub-assemblies and complete space frame structures made in VCS gives results of exceptionally good agreement (with respect to both crushing characteristics and deformation shape) compared to the FE models simulations. The time needed for the VCS simulation is however substantially shorter. For example the macro element
model (rear frame) consisted of 45 nodes and 56 macro elements required 30 sec. CPU on a 2GHz Windows PC.

The macro element approach is extensively used in the crashworthiness community for fast analysis and preliminary design of energy absorbing structures. Recent developments in macro element methodology confirm robustness and accuracy of macro element models in crashworthiness simulations.

**Industrial Visual Crash Studio and Crash Cad users (in historical order):**

VOLVO, BMW, FIAT, AUDI, PSA, ROVER, LOTUS, HONDA, HONDA AMERICAS, MAZDA, MITSUBISHI, TOYOTA, DAIHATSU, HINO, NISSAN, ISUZU, CORUS (BRITISH STEEL), ALCOA, SUMITOMO, ALCAN, DLR, SIEMENS, DUARTE, SAIC UK, VOESTALPINE, GELLY, LKR RAMSHOFEN, HYUNDAI GERMANY
Chapter 1

The macro element method in crashworthiness calculations – the state-of-the-art and the future

by Wlodek Abramowicz
(Impact Design, Europe)

Abstract

Successful design of a crashworthy structure involves assessment of several simulation models at various levels of simplification and accuracy. Typically a design process starts from numerous analyses of semi-analytical and experimental models. The more mature is the structure concept the more detailed simulation models are needed. The macro element models take over the main load of simulation work at the intermediate level of the design process when basic dimensions of the structure undergoes frequent modifications.

This paper presents concise overview of the existing macro element technology and presents range of new elements developed for car, train and ship industries. These elements are implemented in the framework of novel finite element formulation that is capable of handling both standard finite elements as well as a variety of macro and user defined elements. Finally, number of examples is presented to illustrate potential of the new approach and indicate directions of further research.

Introduction

The macro element method is derived from the kinematic method of plasticity and energy method of classic elasticity. The leading idea of this method is to assume the kinematics of deformed continua rather than calculate it from classic equations of equilibrium. The macro element method is especially successful in the field of large deformations of thin sheets and thin walled elements made of variety of materials. The typical deformation pattern of thin elements is apparent in many areas of human activity.
Figure 1 Folding patterns of fabric on Michelangelo tempera *Holy Family* and folded roof of a car. Perhaps the most spectacular illustration of this fact is noticeable in paintings and sculptures of Michelangelo. The painting (tempera) in Figure 1 shows typical configuration of fabric’s folds and wrinkles formed under the gravity forces. This pattern is compared with plastic folds and wrinkles typical for crashed metallic sheet – in this case deformed roof of a small passenger car.

It took several centuries before this apparent feature of thin sheets was utilized in practical engineering calculations. The first kinematic solution directly related to large distortions of thin shells subjected to crash loading is due to Alexander [Error! Reference source not found.], and was published in 1960. The leading idea of Alexander’s approach is to assume a class of shape functions close to the experimentally observed deformation pattern of a shell (or segment of the shell). Alexander considered progressive folding of cylindrical shells now widely known as concertina pattern, Figure 2. The solution to the problem was obtained by postulating inextensibility of the shell in the axial direction and global minimum condition with respect to the length of the plastic folding wave, denoted as 2H in

Figure 2. The novel approach involved only few lines of elementary calculations and rendered simple expressions for the length of the plastic folding wave and energy absorption capacity of the shell.

Figure 2 Concertina folding of cylindrical tube and corresponding kinematic computational model due to Alexander, [Error! Reference source not found.]

Significant advances in the kinematic method of plasticity were made in early eighties of the last century. At that time number of papers addressed most of the unresolved issues present in early solutions. The following short description of milestones of the emerging macro element methodology focuses on important practical results but does not pretend to give a full review of the relevant literature.

The first key modification to early kinematic solution addressed the problem of underestimated level of energy absorption. The theoretical solution for effective crushing distance due to Abramowicz [Error! Reference source not found.] allowed for the estimate of the height of completely squeezed plastic lobe in axially compressed prismatic elements. The effective crushing distance concept was then extensively validated experimentally by Abramowicz and Jones [Error! Reference source not found.]. The next important
contribution was the introduction of the energy equivalent stress measure [Error! Reference source not found.] that allowed for reliable estimate of a representative stress level in crushed shells.

Works due to Kecman [Error! Reference source not found.] published in late 70 and early 90 of the last century showed a practical way of deriving semi-empirical solutions to the bending collapse of rectangular and square sections. Kecman’s solutions required number of empirical parameters to describe bending crushing characteristic but at the same time showed a way of seamless connections of classic elastic solutions (up to the point of peak bending moment) with kinematic results for deep collapse response.

Development of more general approach to the crushing of thin walled structures was made possible by works of Wierzbicki, Jones and Abramowicz. The important contribution of these authors was to isolate a typical buckling or folding pattern in folded structural elements and describe its mechanical response. At that time the general folding mechanism was referenced as Basic Folding Mechanism (BFM) and is now considered to be a first prototype of the Super Folding Element [Error! Reference source not found.].

Further advances in the method should be attributed to works done at MIT in middle eighties of the last century and summarized in eight Volumes of the Manual of Crashworthiness Engineering. [Error! Reference source not found.]. The manuals summarized the formulation of the method and documented number of analytical solutions to various crushing mechanisms in thin shells. The latter research was a starting point for the development of the Superfolding Element (SFE) concept that was subsequently implemented in specialized programs for design and calculation of energy absorbing structures. Since early 90 of the last century the CAE programs based on the macro element concept are used in routine daily engineering practice in virtually all branches of transportation industry.

The paper summarizes the current formulation of the macro element method and presents recent advances in the generalization of the method to complex 3D structures.

The Macro Element Method

The general methods of structural mechanics like beam and shell theories or finite element method are formulated at the level of differential equations of equilibrium and are applicable to a wide class of structures subjected to variety of boundary and loading conditions and constitutive relations. In contrast to these methods the macro element formulation is dedicated to narrow class of structural elements or even to a single type of a structural element (or its representative part). Consequently, sharp restrictions are imposed onto the type of an analyzed structure and type of boundary and loading conditions.

All the major steps of a typical macro element formulation are illustrated in this section on the example of Superbeam Element, an essential element of the CAE computer programs discussed later in this paper. Readers interested in more detailed discussion of the formulation of the method are referred to the base publication on this subject. [Error! Reference source not found.].
In the kinematic method the assumed deformation of a structure is defined in terms of space–time shape functions postulated on the basis of experimental observations. The basic building block of macro elements that describe crushing response of prismatic thin walled members is the Superfolding Element (SFE), Figure 3. The SFE describes crushing behavior of a segment of corner line of the prismatic section subjected to arbitrary crash loading. The cross–sectional dimensions of a SE and its most general deformation mode are shown schematically in Figure 3. The initial geometry of a SE is defined by four parameters:

1. total length, \( C \), of two arms of a SE, \( C = a + b \),
2. central angle, \( \Phi \)
3. wall thickness \( t_a \) of the arm of the length \( a \)
4. wall thickness \( t_b \) of the arm of the length \( b \)

The plastic folding of the element involves creation of five different deformation mechanisms. These are:

1. Deformation of a ‘floating’ toroidal surface.
2. Bending along stationary hinge lines.
3. Rolling deformations.
4. Opening of conical surfaces.
5. Bending deformations along inclined, stationary, hinge lines following locking of the traveling hinge line 3.

![Diagram of a SuperFolding Element](image)

Figure 3 Nomenclature and basic dimensions of the SuperFolding Element.

The solution to the crushing response of a SuperBeam Element, Figure 4, is obtained in three steps. First, the two ends of a prismatic beam, referred to as deformable cells, are discretized into SuperFolding Elements. The height of each cell corresponds to the length of the plastic folding wave, \( 2H \), in a given cross–section. Therefore, the shortest SBE element that could be defined in the element is of the length of two plastic folding waves, \( 4H \). Longer elements are defined by assuming that the two deformable cells are separated by an elastic/plastic beam element, Figure 4.
Figure 4 The SuperBeam Element modeling concept. Crushing response of a single layer of folds is described by the deformable cell element of the height equal to the length of the plastic folding wave, 2H. Two deformable cells separated by elastic/plastic core form the longer version of SuperBeam element.

In the second step, the axial crushing response of the SBE is determined for all kinematically admissible deformation histories, $h_i$, that include loading reloading and reverse loading processes. The rate of energy dissipation for axial crushing of deformable cells is calculated based on the assumed folding geometry. The general form of the governing rate equation is

\[
\dot{E}_{int} = \int_S N_i \dot{e}_i dS + \sum_{i=1}^{n} \int_{l_i} M_i \dot{\theta}_i \, dl
\]

where $N_i$ and $M_i$ are, respectively, the fully plastic membrane force and bending moment, $L_i$ is the length of the plastic hinge line. Associated with each hinge line is the discontinuity of angular velocity defined by the jump function $[\dot{\theta}_i]$. The continuous deformation fields, $S$, correspond to the toroidal and conical surfaces 1 and 4 in Figure 3. The corresponding rate of deformation, $\dot{e}_i$, associated with continuous deformation is defined from the assumed geometry of contributing SFE.

The rate equation, Eq. 1, is determined with accuracy to four free geometric parameters: the length of the plastic folding wave, 2H, average rolling radius, $r$, and a ‘switching point’ parameter $\alpha^*$, which controls amount of tensile deformations in conical surfaces 4 in Figure 3. All these parameters are found from minimization of the functional:

\[
P_m = \frac{t^2}{4} \left( \sigma_o N (\bar{C}_1) A_1 \frac{r}{t} + \sigma_o M (\bar{C}_3) A_3 \frac{C}{H} + \sigma_o M (\bar{C}_3) A_3 \frac{H}{r} + \sigma_o M (\bar{C}_4) A_4 \frac{H}{t} + \sigma_o N (\bar{C}_3) A_3 \frac{2H}{\delta_{eff}} \right)
\]

\[
\text{Eq. 2}
\]

where $A_i = A_i(\Phi, \alpha^*)$, $i = 1, 2...5$

Physically, the governing functional, Eq. 2, defines the mean force in axial crushing of the beam element. Once the unknown geometrical parameters are determined, the instantaneous axial
crushing force is defined by equating rates of external loading, \( \dot{E}_{\text{ext}} \equiv P(\delta) \dot{\delta} \) and rate of internal dissipation defined in Eq. 1. The resulting formula is:

\[
P(\delta, \dot{\delta}, \dot{h}) = \frac{\dot{E}_{\text{int}}}{\dot{\delta}}
\]

where \( \dot{\delta} \) is the rate of axial crushing and \( \dot{h} \) stands for deformation history.

Figure 5 Distribution of normal stresses in a square SBE subjected to axial compression and bending, Error! Reference source not found.\[.

The last step of the solution procedure involves definition of bending and torsion moments and interaction (hyper) surfaces for deformed end cells of the SBE. The solution to the bending problem is based on the analysis of normal stresses distribution at the boundary of deformable cells, Error! Reference source not found.] Error! Reference source not found.\[. An example of the stress distribution in the post-collapse range is illustrated in Figure 5 for a square beam bent along major axis. The distribution shown in Figure 5 is constructed by considering asymmetric crushing modes of contributing SFE. Once this is done the resulting bending moment is defined by integrating the stress profile along the cross-section line, \( L \):

\[
M(\theta, \dot{\theta}, \dot{h}) = \int_L \sigma(\theta, \dot{\theta}, \dot{h}, l) \, dl
\]

Like in the case of axial crushing the bending moments are determined for all kinematically admissible bending histories, \( \dot{h} \). Following similar lines the torsion crushing characteristics of the beam are found. When all crushing characteristics for all contributing cross-section forces are determined the interaction hyper surface is constructed. In general the interaction surface has the form, Error! Reference source not found.\[:

\[
\left( \frac{P}{P_{\max}} \right)^{a_1} + \left( \frac{M_y}{M_{y_{\max}}} \right)^{a_2} + \left( \frac{M_z}{M_{z_{\max}}} \right)^{a_3} + \left( \frac{T}{T_{\max}} \right)^{a_4} = 1
\]

where \( P_{\max}, M_{y_{\max}}, M_{z_{\max}} \), and \( T_{\max} \) are, respectively, maximal values of axial force bending moments and torque. The constants \( a_i \) are functions of initial geometry, material properties and deformation
history and must be defined separately for each cross-section. Schematic illustration of a 3D axial force–bending moment–torque interaction surface is shown in Figure 6.

Figure 6 Schematic illustration of the evolution of the interaction surface in the deep collapse range
For each deformation step the position of interaction surface is uniquely defined by the deformation history, $h$. Actual values of cross-section forces on the other hand are functions of current rates of deformation and are defined from the standard normality rule.

Object Oriented Formulation

For many years application of the macro element method was limited to specific structural elements, like concertina folding of circular tube, or simple structural elements, like prismatic members subjected to elementary quasi static crushing loading. The limitation comes from the relative complexity of a solution to the dynamic response of macro elements and resulting difficulty in implementation of macro elements in standard FE programs. The desired break-through solution followed the widespread of Object–Oriented Programming (OOP), which inspired development of object oriented formulation of the generalized finite element method, Error! Reference source not found.]. This section provides for a short overview of the basic concepts of the new Object Oriented Formulation of the generalized finite element method.

The OOP is an effective way of programming solutions to complex non–linear problems. Central to the successful OOP is the unambiguous definition of all entities, referred to as objects, contributing to a given model. In the case of FE–like approach, intuitive definition of key objects is straightforward. ‘Nodes’ are the objects that govern global equilibrium of the model while ‘Elements’ are the objects that impose loading onto nodes as a result of motion of ‘Nodes’. The relation of these two key objects to the governing Newton equation for a representative node is illustrated in Figure 7.
Figure 7 Relationship between the equation of motion and Node–by–Node/Element–by–Element technique.

The mass of the node $M$ represents reduced inertia of all the elements connected to the node while the vector of internal forces, $f^{\text{int}}$ represents a combined action of all the loadings inserted onto the Node by connected elements. The external forces, either ‘given forces’ or body forces, are defined separately. It should be noted that once the resultant internal force vector is defined (as either a constant force or a given function of node’s kinematics) the time stepping solution to the above equation do not require any knowledge on the type or number of elements connected to the node. Consequently, the iteration step is performed on the set of nodes only. In the case of explicit time stepping routine this operation is referred to as a node–by–node method. Once the iteration step is completed, the internal state of all the contributing elements is updated based on the current kinematics of the node set. In FE nomenclature, this procedure is referred to as a constitutive update. It is obvious that the constitutive update is performed in the element–by–element manner. The above general description of the simulation paradigm does not address particulars of the communication between two key types of objects or particulars of the time stepping routines. Two additional objects responsible for these tasks: the ‘Interface’ and ‘Iterator’ objects are defined later in this section.

**Node objects**

When both forces and moments act on a node, the node equilibrium is governed by the Newton–Euler equations of motion

\[
\text{Eq. 6} \quad m\ddot{r} = F \\
\text{Eq. 7} \quad J \dot{\omega} = M - \omega J \omega
\]

where $F$ and $M$ are the forces and moments exerted on the node, $m$ is the node mass, $\omega$ is the angular velocity, $J$ is the inertia tensor, and $\ddot{r}$ is the nodal acceleration in the global coordinate system. Note that the rotary motion in Eq. 7 is defined in the local coordinate system. When updating node orientation in space, angular velocity $\dot{\omega}$, is transformed to the global coordinates and the instantaneous increment in angular position $\Delta \theta$ is calculated from Eq. 8:

\[
\text{Eq. 8} \quad \Delta \theta = \omega \Delta t
\]

The updated orientation of the node is defined by three unit vectors of the co–rotational coordinate system coupled with each node, Eq. 9.
\[
\begin{align*}
\tilde{i}_n &= \tilde{i}_{n-1} + (a \times \tilde{i}_{n-1}) \sin(\Delta \theta_n) + 2[a \times (a \times \tilde{i}_{n-1})][1 - \cos(\Delta \theta_n)] \\
\tilde{j}_n &= \tilde{j}_{n-1} + (a \times \tilde{j}_{n-1}) \sin(\Delta \theta_n) + 2[a \times (a \times \tilde{j}_{n-1})][1 - \cos(\Delta \theta_n)] \\
\tilde{k}_n &= \tilde{i}_n \times \tilde{j}_n
\end{align*}
\]

**Interface objects**

The functionality of the interface object is explained in Figure 8 and Figure 9. The cross-section loading in beams is defined at the centroid by summing forces and moments parallel and perpendicular to the cross-section plane (bending and torsion moments, axial and shear forces). The object capable of handling this functionality is referred to as 'Interface'.

![Figure 8 Two beam elements connected at a Node.](image)

Orientation of the interface is defined by the 3D coordinate system \{\alpha_i\} co-rotational with the interface plane. The origin of the coordinate system, referred to as a node of the beam element, marks the point of application of cross-section forces. When two beams meet together at the Node object, marked as J in Figure 8 and defined by its own co-rotational coordinate system \{\alpha_i\}, the beam nodes, marked as I in Figure 9, do not necessarily coincide with each other. The cross-section planes of both elements have, in general, different orientation in space, and are located at certain distance apart. However, following the basic assumption of the beam theory they are 'rigidly' connected with the Node object. Therefore, the reduction of forces from interfaces to the Node as well as retrieving of kinematic variables in the interface coordinate system is done easily by simple vector transformations between two 3D coordinate systems. This procedure is much simpler than standard method involving matrix transformations.

![Figure 9 The concept of the interface object.](image)
The element concept

The general term ‘element object’ refers to any finite part of a solid or structure. It is assumed that the response of an element is uniquely defined by the kinematics of interfaces defined on that element.

![Figure 10](image)

Figure 10 The internal mechanics of a beam element is uniquely defined by the kinematics of two interfaces.

For example, in the case of a beam element in Figure 10 the dynamic response of the ‘element’ is uniquely defined by the kinematics of two interfaces regardless of the employed beam theory. In the object oriented formulation elements are divided into two basic groups: Finite Elements (FE) and Macro Elements (ME). The division is functional rather then formal. The name Finite Element applies to all classic elements that are defined and updated following standard FEM matrix paradigm. These elements can be implemented directly as ‘encapsulated objects’ with only minor re-coding of standard FE algorithm. All remaining elements are referred to as Macro Elements and can be defined by any algorithm, expression, separate program and/or any rule that can be cast into the form of a computer code.

The CAE software based on the macro element concept

The number of programs based on the Macro Element concept is rather small in comparison to large offer of classic FE programs. The most renowned Macro Element programs are closely related to the crashworthiness of cars (Crash Cad and Visual Crash Studio) and tanker/ships crashworthiness (DAMAGE). This section describes basic concepts implemented in the program Visual Crash Studio (VCS) – a successor of Crash CAD. The VCS is probably the most advanced software for design of energy absorbing sections and simulation of crushing response of arbitrary space frames. The VCS is widely used in the automotive industry worldwide.

The Visual Crash Studio program

Design of energy absorbing structures is achieved in several steps, usually in a highly iterative design/calculation process. Objectives of each design step are achieved by usage of dedicated computational tools typically different for each step. The selection of an appropriate tool depends on the complexity of the problem and availability of suitable software or other predictive techniques such as engineering formulas, empirical data or handbook – type information. The major problem prohibiting effective usage of various techniques in a single design/calculation environment is the lack of compatibility between simulations tools. The difficulty of combining various discretization/solution techniques into single software is not only a technical problem of different input/output organization. The primary problem lies deep in the generic formulation of the FE and semi-analytical or macro element methods. The calculation algorithms of these methods are
frequently inherently different. The first successful combination of macro element and flexible FE-like simulation environment is achieved in the program Visual Crash Studio (VCS). Error! Reference source not found. VCS effectively combines complex macro element solutions at the level of individual elements with object oriented implementation of the global equilibrium solvers. This section describes two major areas of application of the VCS. These are design and calculation of thin walled members for optimal crash performance and flexible crash simulation of structural sub-assemblies and complete space frame structures. It should be noted that early predecessor of VCS the CAE program Crash Cad was capable of calculation and design at the cross-section level only. Now the enhanced functionality of this program is included as one of several dedicated design modules of VCS.

Crashworthiness analysis at the level of individual members
The design of a structural member at the cross-sectional level is especially important at the pre-design and early design stages when the proper shape and optimal dimensions of a member are sought and the design concept undergoes frequent modifications. The macro elopement approach is especially attractive at this level as the calculation routines require as input only overall dimensions of the cross-section, and tensile characteristic of the material while the calculation process takes only few seconds on a standard PC. Consequently, designer can examine a wide range of cross-sectional topologies and run several parametric studies within only few hours of work. The simplicity of cross section modeling by means of macro elements is illustrated in Figure 11 for a typical hat cross-section made of aluminium alloy.

![Input data screen of VCS with a discretized cross-section of a hat member defined by major overall dimensions of the cross section, position of spot-welds and material characteristic.](image)

Design of cross sections for axial crush
Calculations at the cross-sectional level are the first essential step in all Superbeam calculations. This section shows how basic calculations are used to predict crushing response of a single prismatic member.
One of the most sensitive parts of the design at the level of a single prismatic section is concerned with progressive folding during a head-on collision. Among all possible deformation modes of prismatic sections the progressive folding absorbs most of the impact energy. At the same time, this folding mode is the one most difficult to obtain in a real-world design. Development of progressive folding requires simultaneous completion of several conditions.

These are:

- The cross-section geometry must be properly designed in order that the local deformation of a section in each plastic lobe can be accommodated without internal contacts and penetrations. In addition, the deformation of each plastic lobe must be compatible with the deformation of its closest neighbour,
- Spot welds (rivets or laser weld-line) must not interfere with the local plastic deformation of a section,
- The section must be properly “triggered” through the introduction of correctly designed hoop dents which guarantee the development of a proper folding mode and reduce the peak load to such a level that the potentially unstable plastic deformations are induced only in the region of triggering dents and finally
- The boundary and loading conditions (stiffness of joints, loading direction) are kept in the range that guarantees the predominantly axial loading of the section.

The first three conditions, pertinent to the level of a single member, must be met at the design stage of a given member while the last condition must be checked at the level of full crash simulation of a car. This task requires an interaction of component level design routine with simulation of the crash response of complete model of the space frame.

Typically cross section optimization is done iteratively in several optimization steps. At each step the program provides the user with information on the design errors at the level of single Superfolding element (that models crushing behavior of individual corners of prismatic members) and at the level of the whole cross section. The cross section analysis module lists necessary corrections to the cross-sectional geometry up to the point when the section can collapse progressively without internal contacts and/or penetrations. This stage of the design requires fine-tuning of central angles, widths of side faces and appropriate geometry of spot welding. An example of initial, bad design of a rocker panel and final, correct design are shown in Figure 12. It transpires from Figure 12 that both cross-sections are quite similar and the decision on the correctness of the design is impossible without a detailed analysis of the folding mechanics.
Figure 12  Initial (bad) and final (correct) geometries of a rocker panel optimized for axial crash through a proper selection of central angles and widths of side faces. Red bullets on the first sketch mark flows in the crashworthy design – in this case improper central angles.

The final stage of cross-section design is concerned with appropriate triggering mechanism. An example of triggering dents, designed on the basis of macro element folding analysis, is shown in Figure 13 for hexagonal columns with flanges. Introduction of triggering dents in members designed for axial crash is necessary in order to promote a desired progressive folding pattern and reduce the peak force below the level, which is likely to induce global, Euler – type buckling of a column. A triggering of columns is especially important for complex cross-sections that develop a large number of natural folding modes. Usually only few of these modes are likely to converge to the desired progressive folding pattern while other modes lead to a premature bending of a column.

Figure 13  Paper model shows triggering dents in hexagonal column with flanges designed on the basis of VCS calculations.

The importance of proper triggering is further clarified on the example of laboratory experiment on a simple cross-sectional geometry, Error! Reference source not found. The first photograph in Figure 14 shows a long square prismatic column made of mild steel. Such a column has a cross-sectional geometry that guarantees proper folding without internal contacts and does not
contain spot welds that may destroy progressive folding pattern. A properly triggered square column collapses progressively up to the point when the last plastic fold is completed.

(a).

(b).

Figure 14  Progressive collapse of a properly triggered long square column, top, and global bending or irregular folding of untriggered columns, bottom.

For example, the 54x54x1.4 mm square column at the top of Figure 14 a was squeezed over 700 mm and developed 32 symmetric plastic lobes with no sign of any global bending (the completely squeezed column in this figure was initially one-half of the size of the longer column and developed 16 lobes). The total energy absorbed was 25 kJ, which is sufficient to bring to the rest a 1000 kg car traveling with an initial velocity of 25 km/h. On the other hand much shorter, untriggered, columns in Figure 14 b collapsed in bending or deteriorated from progressive folding despite of a carefully controlled loading and boundary conditions at both ends of a column.

Higher order macro elements

The preceding sections discussed several macro elements that describe crushing response of elementary structural elements. These elements are referred to as first order macro elements. Successful usage of these elements in an actual design practice encouraged development of even more complex elements that are capable of modeling and optimization of complete sub-structures and structural assemblies. These elements, frequently containing several first order macro elements, are referred to as higher order macro elements. Currently operational higher order elements are capable of modeling dynamic response of honeycomb barriers, tapered thin-walled elements, semi-rigid bodies and thin walled joints, Error! Reference source not found.). The latter structure is discussed in the following paragraph in order to illustrate the growing potential of the new family of macro elements.

The SuperJoint Element models the connection of several thin-walled beams in a single structural member. A well known and practically important example of a joint is the connection of rocker panel with B-pillar and floor cross member of an automobile body. The macro element that models crushing behaviour of a joint is referred to as SuperJoint Element (SJE) and is developed to
address the problem of optimal joint design and increase modeling potential of macro element models.

The SJE is defined as an assembly of several SuperBeam Elements referred to as legs of a joint. The joint legs are interconnected by a “core” of a joint. The SuperJoint Element offers several options of core modeling at various stages of the design of a structure. The simplest model of a joint assumes infinite rigidity of the core. The rigid core model offers realistic references point for the optimal joint design where core is “stronger” than the legs and do not promote premature collapse of the joint. Elementary examples of “weak” and “strong” joint response are illustrated in Figure 15, which shows that that correctly designed core is “stronger” then connected beams and therefore do not deform under applied crash loadings.

In the case of perfectly rigid core all the beams meet at one node (with appropriate offsets) and retain original beam response; i.e. crushing of one beam is not coupled with the crushing response of remaining beams. In particular this modeling procedure does not account for local denting and lateral crushing of joints, typical for “weak” – badly designed core.

Figure 15 “Weak” and “strong” joint response under axial crashing of vertical beam.

The core of a SuperJoint is subjected to combination of axial loading, bending, torsion and denting deformations. Pre-design of a joint is facilitated in SJE in two steps. In the first step the connecting shells are designed in such a way that the strength of core under beam–like combined loading (axial, bending and torsion) is higher then the strength of contributing legs and consequently local collapse occurs in joint leg(s) rather then in a core. In the second, “fine–tuning”, design step the shell–like denting response of connecting shells and SBE(s) is analyzed in order to account for possible denting deformations and suggest local stiffening of connecting shells and SBE.

The performance of SJE is illustrated on the example of elementary T joint with square cross–section that is fixed in all 6 d.o.f. in horizontal legs and translational d.o.f. of the vertical leg. Quasi–static skewed bending loading is applied to the vertical leg so the leg is bent in a diagonal plane of the square cross section. The slanted connecting shells were modified up to the point when collapse of the joint occurred in the vertical leg rather then in the joint core.
Figure 16  Quasi static bending crushing of elementary T joint. Solid line SJE prediction, broken line FE results.

The results of macro element calculations are shown in Figure 16 together with FE calculations that were run for final design of the joint in order to verify the prediction of calculation/design macro element procedure.

Examples of application

The VCS is relatively new software that integrates complete macro element algorithm with the FE-like simulation environment. Despite that several benchmarking tests were already done at universities and automotive R&D departments in order to verify accuracy of the software and its applicability to the design process. This section summarizes just few representative examples of the application of the macro element method in automotive industry.

The target components of the benchmarking study at Honda R&D, Error! Reference source not found., are illustrated in Figure 17.

Figure 17 Target components of body in white tested in this evaluation program, Error! Reference source not found.

All macro element models were created semi-automatically by using the CAD-macro element program interface. Simulations were run in parallel for both macro element and FE models. The
example of simulation results for rear frame assembly is shown in Figure 18. The exceptionally good agreement of simulation with respect to both crushing characteristics and deformation shape is evident from the graphs in this figure.

![Figure 18](image)

**Figure 18** Quasi static deformation of rear frame. Undeformed FE model and deformed macro element model. VCS – red curve, FE black curve

The 56 km/h frontal crash of the Ford Taurus structure was studied by Lasek at al. Error! Reference source not found.]. The macro element model consisted of 45 nodes and 56 macro elements and required 30s CPU on a 2MHz Windows PC. The macro element simulation results were then compared with standard FE simulation. Results of both simulations are compared in Figure 20 and Figure 19. Global collision parameters: shortening–time and c.g. velocity–time histories are compared in Figure 20. Figure 19 shows deformed configurations of both models in 10 milliseconds intervals.

![Figure 19](image)

**Figure 19** Velocity– time and shortening–time histories for macro element and FE models.
Conclusions

The macro element method, formulated in the 50 and 60 of the last century was used extensively in the mechanics of elastic shells and plates. This method was then generalized to describe crushing response of thin walled components. Rapid development of FE codes significantly limit practical application of the macro element method in nonlinear elastic analysis. At the same time the macro element approach is extensively used in the crashworthiness community for fast analysis and preliminary design of energy absorbing structures. Recent developments in macro element methodology, described in last sections of this paper, confirm robustness and accuracy of macro element models in crashworthiness simulations.

Further development of the method will most probably be guided by industry needs. The effective design/calculation environments are now intensively sought by many automotive manufacturers and most probably this sector of industry would direct further development of macro element method toward seamless integration with other numerical tools. The CAE program VCS is just a first step in this direction and its further development to include classic FE elements is highly feasible.
Chapter 2

Macro Element Fast Crash Analysis of 3D Space Frame

Kenji Takada
Honda R&D Co., Ltd. Tochigi R&D Center

Włodek Abramowicz
Impact Design, EUROPE

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ABSTRACT

The paper discusses recent developments in the macro element methodology. Newly developed macro elements: tapered super beam, thin-walled super joint or deformable barrier allow for simulation of the crash response of space frames in arbitrary crash configurations. The paper discusses underlying modeling concept and calculation methodology used in the development of new macro elements and demonstrates its effectiveness in the calculation/design process. Number of crash simulation examples illustrates accuracy of the macro element method in comparison to time consuming FE calculations.

INTRODUCTION

The macro element method is used for fast analysis and design of thin walled structures subjected to crash loading. The early predecessor of the macro element methodology in CAE is the solution for concertina folding of metallic cylinders published by Alexander at the beginning of 60 of the last century. For more than two decades the macro element method was a subject of intensive university research, which resulted in the development of numerous analytical and semi-analytical solutions for specific crash problems. The effective industrial application of the method was made possible by the development of the Super Folding Element (SFE) method and its implementation in user friendly computer programs in early 90 of the last century. The SFE methodology allows for fast and reliable analysis and design of thin walled prismatic cross-sections for optimal energy dissipation in progressive folding mode. Subsequently the SFE method was generalized to include bending and combined crushing loadings in thin walled prismatic beams. These solutions were implemented in a macro element referred to as the Super Beam Element (SBE). The invention of the SBE together with novel object oriented formulation of the finite element method paved the way to the development of CAE programs capable of predicting crash response of general space frame structures. The methodology of SFE and SBE are described in a number of publications. The present paper focuses on new developments in the macro element methodology. First, the next generalization of the SFE is shown that encompasses torsion crushing response of thin walled beams. This research resulted in the development of subsequent version of the SBE element capable of predicting crushing response of prismatic and tapered beams under combined action of axial force bending and torsion moments.
The subsequent version of the SBE is used to construct macro element model of typical joints of an automobile body. This macro element is referred to as SuperJoint Element (SJE) and is briefly described later in this paper. Finally, number of simulation examples, at the level of components and sub-assemblies of body-in-white is discussed in order to illustrate the range of modeling applicability and simulation accuracy of the macro element method, Fig. 1.

**Fig. 1** Target components of body in white tested in the evaluation program

### LARGE DISTORTIONS OF SBE

This section discusses generalization of the SBE response to include large torsion-generated distortions of thin walled beam. The beam is assumed to be fixed between two parallel planes (this model does not account for warping deformation important in open sections with weak constraints at the ends). When the distance between boundary planes is kept constant the torsion response is referred to as fixed-end-torsion. When plates are allowed to move freely in axial direction, while still remaining parallel, the response is referred to as free-end-torsion. An actual crushing response of the SBE falls in between these two extreme cases illustrated in Fig. 2.

**Fig. 2** Schematic representation of the torsion crushing response of SBE.

The general torsion response of a SBE is divided into three distinct ranges, Fig. 2, discussed in more details in following sections.

### LINER ELASTIC RESPONSE

The linear elastic response is described by classic equations of linear theory of elasticity. The contributing plates of a SBE are subjected to the superposition of three types of elastic deformations: torsion, bending and shear. A plate for which plate centroid coincides with the centroid of a SBE is subjected to pure torsion response. Plates that lie radially with respect to
the SBE centroid undergo combination of bending and twist deformation while plates perpendicular to the rotation radius are subjected to combination of twist and shear deformation. All plates in the intermediate configurations are subjected to the combination of torsion, bending and shear deformations. Contributions of all three deformation mechanisms are given as:

\[
M_s = C \left( \frac{r}{t} \right)^2 \left( \frac{t}{b} \right) \theta_t \tag{1}
\]

\[
M_t = \frac{1}{3} C \left( \frac{t}{b} \right) \theta_t \tag{2}
\]

\[
M_b = 0.65 C \left( \frac{t}{b} \right)^2 \left( \frac{t}{L_o} \right)^2 \theta_t \text{ where } C = G \frac{b^3 t^2}{L_o} \tag{3}
\]

respectively, for shear, torsion and bending contributions. In Eqs.(1 − 3), \( L_o \) is the length of the plate in the direction of SBE centroid, Fig. 4, \( b \) and \( t \) are plate width and thickness, respectively, \( r \) is the distance between centroid of the beam and centroid of the plate and \( G \) is the Kirchhoff modulus. The shear, bending and torsion moments and angles are denoted by subscripts, \( S \), \( T \) and \( B \), respectively. It should be noted that in the case of small deformations the bending and shear angles are given as linear functions of the torsion angle of the SBE, \( \theta_t \). Consequently, the resulting elastic torque is given as a sum of three contributions:

\[
M(\theta_t) = M_s(\theta_t) + M_t(\theta_t) + M_b(\theta_t) \tag{4}
\]

MAXIMAL LOADING

The maximal load carrying capacity is calculated by means of the critical strain method developed by one of the co–authors (WA)\(^9\). The starting point of this approach is the classic effective width formula due to von Karman. In the von Karman approach the width of effective plate strips, \( b_{eff} \) adjacent to unsupported ends of a plate is calculated from classic formula for elastic buckling stress:

\[
\sigma_{cr} = k_s \frac{\pi^2 E}{12(1-v^2)} \left( \frac{t}{b} \right)^2
\]

In Eq.(5), \( E \) and \( v \) are elastic constants while the buckling coefficient, \( k_s \), depends on support boundary conditions on plate edges.

The effective width is defined as a width that corresponds to the buckling stress equal to the yield stress of the plate material, \( \sigma_y \), Eq. 6

\[
\frac{b_{eff}}{t} = \sqrt{\frac{k_s}{E} \frac{t}{12(1-v^2) \sigma_y}}
\]

The von Karman postulate is that at the onset of collapse the entire in plane compressive loading is carried by two strips, adjacent to plate edges, of the total width, \( b_{eff} \). The stress distribution across effective strips is constant and equal to the yield stress. In other words
increasing the plate width above $b_{\text{eff}}$ does not result in any increase of its load carrying capacity. The von Karman postulate, introduced in 40 of the last century was subsequently generalized to other types of loading (bending and combined loading) and verified by huge number of experiments. It now forms a basis of virtually all standards and recommendations for cold formed steel and aluminium sections. The von Karman formula applies to plates that buckle in the elastic range. More narrow plates buckle in plastic range. The plastic buckling is promptly followed by localization of plastic flow and local collapse of a beam and, therefore, it is widely accepted as suitable estimate of maximal load carrying capacity. The plastic buckling stress is conveniently determined from the Stowell–Iliushyn equation:

$$\sigma_{cr}^p = \frac{\pi^2 E_S}{9} \left( \frac{t}{b} \right)^2 \left( 2 + \sqrt{1 + \frac{3 E_T}{E_S}} \right)$$

(7)

where material constants $E_T$ and $E_S$ correspond to secant and tangent moduli of a given material. Both governing equations for buckling stress, Eq. (5) and Eq. (7), can be conveniently expressed in terms of critical strains by dividing both sides by Young and secant module, respectively. The resulting formulas are:

$$e_{\sigma}^c = \frac{\sigma_{\sigma}}{E} = \frac{\pi^2}{3(1-\nu^2)} \left( \frac{t}{b} \right)^2$$

(8)

$$e_{\sigma}^p = \frac{\sigma_{\sigma}^p}{E_S} = \frac{\pi^2}{9} \left( \frac{t}{b} \right)^2 \left( 2 + \sqrt{1 + \frac{3 E_T}{E_S}} \right)$$

(9)

It has been shown in ref.9 that for power-type stress–strain relation:

$$\frac{\sigma(\varepsilon)}{\sigma_y} = \left( \frac{\varepsilon}{\varepsilon_y} \right)^n$$

(10)

The critical strains are given as

$$e_{\sigma}^c = \frac{\sigma_{\sigma}}{E} = \frac{\pi^2}{3(1-\nu^2)} \left( \frac{t}{b} \right)^2$$

(11)

$$e_{\sigma}^p = \frac{\pi^2}{9} \left( \frac{t}{b} \right)^2 \left( 2 + \sqrt{1 + 3n} \right)$$

(12)

respectively, for elastic and plastic buckling. Both equations predict similar level of critical strains. Additionally, very weak dependence on the hardening exponent, $n$, is evident in the plastic buckling formula, compare Fig.3.
Once the critical strain is defined from either of the formula in Eq.11 and Eq.12 the corresponding stress is found from standard stress-strain relation of a given material. The above procedure is fundamental for all SFE applications. It has been generalized to other types of loading and is used throughout all SFE calculation routines. For example, in the case of torsion response the critical strain is given by the formula:

$$\gamma_{cr} = \begin{cases} \frac{0.88}{(1-\nu)} \left( \frac{t}{b} \right)^2 & \text{ssl free} \\ \frac{8.800}{(1-\nu)} \left( \frac{t}{b} \right)^2 & \text{ssl ss} \\ \frac{14.772}{(1-\nu)} \left( \frac{t}{b} \right)^2 & \text{ccl cc} \end{cases}$$

(13)

for various support conditions at unloaded edges. The critical shear strains \(\gamma_{cr}\) are used to find \(\tau_{cr}\) and corresponding contribution to maximal torsion moment \(T_{max}^{\text{plate}}\) for each plate of a SBE. The resulting maximal torque is than calculated by summing up fractional plate contributions:

$$T_{max} = \sum_{i=1}^{\text{plate}} T_{i}^{\text{max}}$$

(14)

**POST-COLLAPSE RESPONSE**

The post-collapse response is found by considering simplified kinematics of twisted SBE. Each rectangular plate of a SBE is divided into two, right angle, triangles. The upper and lower edges of triangles are fixed to rigid boundary plates that remain parallel during the whole deformation process. Consequently, only two edges of each triangle undergo plastic deformation (this significantly simplifies calculations of dissipated energy and minimization procedure). During twisting deformations, Fig. 4, corner points of each triangle travel along a circle of given radius, \(r\), (distance to the centre of rotation) and the two corresponding engineering strains are not difficult to calculate, Eq. 15 and 16:
Fig. 4 Calculation scheme for representative triangular plate subjected to large torsion deformation.

\begin{align}
\varepsilon_L &= \frac{L(\theta) - L_o}{L_o} \\
\varepsilon_S &= \frac{S(\theta) - S_o}{S_o}
\end{align}

These strains are used to calculate the principal in-plane directions and corresponding principal strains. The third principal strain (corresponding to the thickness variation) is then found from incompressibility condition. Having determined three (engineering) strains the classic equivalent strain of the Theory of Plasticity is calculated as:

\begin{equation}
\varepsilon_{eq,in} = \frac{2}{\sqrt{3}} \sqrt{\ln(\varepsilon_1 + 1)^2 + \ln(\varepsilon_2 + 1)^2 + \ln(\varepsilon_3 + 1)^2}
\end{equation}

The corresponding stress is calculated as an energy equivalent flow stress\(^{(3)}\) on the basis of given stress-strain curve, \(\sigma_{xx} = f(\varepsilon_{eq,in})\). Finally, the strain energy dissipated in a triangular plate is calculated as:

\begin{equation}
E_s = \frac{1}{2} \sigma_{xx} S_o L_o t_o
\end{equation}

Where \(S_o\) and \(L_o\) are the initial lengths of the plate, Fig. 4, while \(t_o\) is the initial plate thickness. The bending energy \(E_b\) is calculated by tracing the changes of the angle between neighboring triangular plates along edges \(S_o\) and \(L_o\) in Fig. 4. The corresponding expression for \(E_b\) does not have a simple analytical form and must be calculated numerically. Finally, the total energy dissipation \(E_t\) in torsion crushing is defined as a function of the initial length \(L_o\) of the plastically deformed zone, Eq. 19.

\begin{equation}
E_t(L_o) = \sum_{i} (E_i' + E_i'')
\end{equation}

The minimization procedure is then run to check for the possibility of creating “plastic torsion waves” of the length \(L_o\). In most practical cases this length is larger than the length of SBE used in space frame models, which means that the entire beam element undergoes large plastic distortions.
THE TAPERED SUPER BEAM ELEMENT AND SUPERJOINT ELEMENT

This section describes two new macro elements: the Tapered SuperBeam Element (TSBE) and the SuperJoint Element (SJE). Fig.5 shows typical B–pillar component discretized into standard SBE and new TSBE and SJE. It is evident from the picture in Fig.5 that the new macro elements are necessary for reliable discretization of automotive body.

![Image showing TSBE, SBE, and SJE](image)

**Fig.5 The B–pillar component discretized into SBE, TSBE and SJE**

TAPERED SUPERBEAM ELEMENT (TSBE)

The TSBE is derived from the existing SBE by introducing deformable cells of different cross sections, Fig. 6 at both ends of a beam element. The crushing response of deformable cells is calculated at the preprocessing level. Consequently, the tapered elements are equally CPU efficient as the existing SBE.

![Image showing Tapered SuperBeam Element](image)

**Fig.6 Elementary example of a TSBE (tapered square cross–section).**

SUPERJOINT ELEMENT (SJE)

The SJE is developed to address the problem of optimal joint design and accuracy of macro element models. The SJE is defined as an assembly of more than two SBE, referred to as legs of a joint. The joint legs are interconnected by a “core” of a joint, Fig. 8. The SJE offers several options of core modeling at various stages of the design of a structure. The simplest model of a joint assumes infinite rigidity of the core. The rigid core model offers realistic references point for the optimal joint design where core is “stronger” than the legs and do not promote premature collapse of the joint. Elementary examples of “weak” and “strong” joint response are shown in Fig.7. Fig.7 b) indicates that correctly designed core is “stronger” then connected beams and therefore do not deform under considered crash loadings. In the case of rigid core all
the beams meet at one node (with appropriate offsets) and retain original beam response; i.e. crushing response of one beam is not coupled with the crushing response of remaining beams. In particular this modeling procedure does not account for local denting and lateral collapse of joints, typical for “weak” – badly designed core.

![Diagram of joint response](image)

**Fig. 7** Elementary example of “weak” and “strong” joint response in the case of axial loading of vertical leg.

The deformable core of a joint is modeled by a number of connecting shells and triangular plate elements, as explained in Fig. 8.

![Diagram of structural model](image)

**Fig. 8** Structural model of a joint core composed of connecting shells

The core of a SJE is subjected to combination of axial loading, bending, torsion and denting deformations. Pre-design of a joint is facilitated in SJE in two steps. In the first step the connecting shells are designed in such a way that the strength of core under beam-like combined loading (axial, bending and torsion) is higher than the strength of contributing legs and consequently local collapse occurs in joint leg(s) rather than in a core. In the second, “fine-tuning”, design step the shell-like denting response of connecting shells and SBE(s) is analyzed in order to account for possible denting deformations and suggest local stiffening of connecting shells and SBE. This design feature will be fully operational in the next version of the SJE.
EXAMPLES OF APPLICATION

New macro elements, described in the preceding section, are implemented in the CAE program Visual Crash Studio\textsuperscript{(5)}. The new elements were extensively tested both in benchmark tests as well as at the level of real world structures. This section reviews representative results of benchmark tests.

LARGE DISTORTION OF PRISMATIC BEAM

The subject of large torsion deformations was studied very carefully both at the level of elementary cross sections as well as at the level of complex members of an actual automobile body. Fig.9 and Fig.10 show comparison of FE and macro element calculations for rhomboidal and rectangular cross sections with fixed and free ends. The correlation of both results is excellent. It should be noted that in Fig.9 the plastic deformation spreads over the entire beam. This situation is typical for beams with fixed ends.

![Figure 9](image1.png)

**Fig.9** The FE versus macro element calculations for large torsion deformation of elementary rhomboidal cross-section.

![Figure 10](image2.png)

**Fig.10** FE versus macro element calculations for large torsion deformation of elementary rectangle section

In contrary, the rectangular cross section in Fig.10 had free ends and consequently deformations are confined to the middle part of the beam. In this case the presence of “torsion plastic wave” is evident. Representative example of macro element robustness at the level of complex cross...
sections is shown in Fig.11. The results shown in this figure correspond to free twisting of a segment of side sill.

Fig.11 Twisting crushing of side sill beam.

ELEMENTARY T–JOINT CONNECTION

This section presents one of the benchmark tests of the SJE. An elementary T joint with square cross-section was fixed in all 6 d.o.f. in horizontal legs. The translational d.o.f. of the vertical legs were also fixed. Quasi-static skewed bending loading was applied to the vertical leg so the leg was bent in a diagonal plane of the square cross section. The slanted connecting shells were modified up to the point when collapse of the joint occurred in the vertical leg rather then in the joint core.

Fig.12 Quasi static bending crushing of elementary T joint

The results of SJE simulation are shown in Fig 12 together with FE calculations that were run for final design of the joint in order to verify the prediction of calculation/design SJE procedure.

CONTACT/IMPACT ALGORITHM

The Visual Crash Studio program that implements the new macro elements discussed in this paper is equipped with dedicated contact/impact mechanism that is compatible with relatively large size of macro elements and does not compromise CPU efficiency of the macro elements.
method. Details of the contact algorithm and its implementation in macro element code are given in Ref.5. This section presents basic benchmarks for the contact algorithm. The hat cross section S-frame with 500 [kg] mass attached to one end hits rigid obstacle with other end with initial velocity of 30 [km/h], Fig. 13.

![Fig.13 Dimension of the thin-walled steel S-frame](image)

Fig.13 Dimension of the thin-walled steel S-frame

Fig.14 shows final configurations of macro element simulation together with FE results for the same model while Fig. 15 provides for comparison of macro element and FE force–time histories.

![Fig.14 Deformation shape of S-frame with initial velocity of 30 [km/h].](image)

Fig.14 Deformation shape of S-frame with initial velocity of 30 [km/h].

A very good correlation of both methods is evident from the graph in Fig.15. Especially important is a good correlation of dynamic peak force that results from the interaction of a barrier with impacting beam. This result shows that macro-element contact/impact algorithm performs as well as standard FE dedicated mechanism. It should be noted that macro element model involves 3 SBE while total CPU cost is below 1 second on 2GHz PC. The FE model consisted of 13000 elements and required 110 seconds on IA64 SPMD machine.

![Fig.15 Correlation of SBE and FEM result for the S-frame](image)

Fig.15 Correlation of SBE and FEM result for the S-frame
CRASH LOADING OF AUTOMOTIVE BODY COMPONENTS

The validation of the new macro elements were done on numerous structural assemblies of a car body illustrated in Fig. 1. In this paragraph the performance of Visual Crash Studio is illustrated on the example of three representative substructures:

- Front side member, Fig. 16.
- Rear frame component, Fig. 17.
- A-pillar sub-structure, Fig. 18.

All macro element models were created semi-automatically by means of the Catia–Visual Crash Studio interface. All simulations were run in parallel by using macro element and FE models. The results of simulations are compared in Fig. 16, Fig. 17 and Fig. 18. The exceptionally good agreement of simulation with respect to the deformation shape and other loading histories validates the robustness and accuracy of macro element models in crashworthiness simulations. The CPU cost of each simulation discussed in this section is below 15 seconds on Pentium4 PC.

Fig. 16 Deformation shape of front side member.

Fig. 17 Deformation shape of rear frame component.

Fig. 18 Deformation shape of A–B pillars assembly.
INTERFACE SYSTEM

Data exchange mechanism between CAD and crash simulation software is an important component of integrated design/calculation environment. Implementation of a link between CAD program Catia and Visual Crash Studio (VCS) is a relatively easy task since both programs are fully object oriented and implement variety of communication options. In the present setup most of the geometrical design is done on the side of CAD software. Initial geometry of cross sections, position of beams and joints, thicknesses and material properties for all elements are defined by the CAD user and written to the XML file compatible with VCS input format. Therefore, all the operations on individual cross-sections, components and sub structures build from individual components can be done in VCS without further interaction with the CAD program. For example the labor cost of preparing the interfacing data for each of the models described in the previous section was below 15 minutes.

CONCLUSION

The paper presents new developments in the macro element method and application of the new elements in daily CAE practice. It has been confirmed that macro element method provides for fast and reliable tool in crashworthiness calculation that can take over a large part of simulation work during the design for crash of automobile body. Apart from exceptional modeling simplicity and CPU efficiency the macro element method provides for a unique calculation /design features that make possible optimal design of crashworthy elements in early stages of the design process. It is demonstrated that plausible usage of macro element programs for basic design and examination of design variants together with final FE simulation of detailed model can significantly speed up the entire design process. In view of the apparent potential of the macro element method the authors would like to encourage further development of macro elements such as large-size plate and shell elements. This planned development would increase the potential of macro element method and would make possible the design and crashworthiness analysis of complete body-in-white.
Chapter 3

Using Visual Crash Studio for crash model simulation

A pole crush model example from the SMATBATT project

by

Peter Luttenberger, Marian Ostrowski, Manuel Kurz, Wolfgang Sinz

USING THE ME METHODE FOR EES INTEGRATION

Design of battery housing and redesign of the surrounding structure, incorporating additional energy absorbers, were the main issues in this stage of the project. In present automotive structures thin walled beam frames are responsible for most of the body strength and crash parameters. Typical FE crash models are very complex in terms of their geometry, number of elements, and, by universality of the FE method, do not directly provide important engineering information (like limit loads of cross-sections, their average crush forces, length of the folding waves, etc.). Computing times and problems with manipulation of the geometry lead to expensive design efforts. Moreover, due to the nonlinear and nonsmooth nature of the crash optimization problem, no practically acceptable topology optimization procedure is known.

To overcome such problems a crash macro element method can be used for engineering first order effect optimization. Its genesis lies in smart observations of nature, caught in mathematical solutions of large, plastic deformations of thin walled parts. The geometry of a structure can be divided into sets of parts, whose cross-sections can be discretized into basic superfolding elements (SFE). Analytical solutions of large deformations of each SFE’s are performed and folding modes of cross-sections and its characteristics for axial, bending and torsional deformations are calculated. Structural interaction curves are being recalculated during simulation time steps. Cross-sections are integrated into superbeam (SBE) characteristics, consisting each of two cross-sections based on deformable cells at each end, and an elastic beam element in the middle (figure 4). Folding propagation can be transferred to the next internal cells generated inside the elastic part, replacing its properties, as well as to neighboring SBE elements.

ME model creation requires greater structural knowledge and experience than use of the FEM. The user has to perform the discretization in a proper way, understanding mechanisms of thin walled crash mechanics and limitations of this method. Current macro elements implemented in this environment can be used for simulating behavior of prismatic or cone shaped beams (vehicle frame members)
and shear panels (roof or floor sheets). Responses of complexly casted parts have to be introduced by use of user defined characteristics, or by equivalent elements in terms of their mechanical characteristics.

Computing times for typical assemblies of full vehicles like cars, buses or trains are about tens of seconds, instead of hours comparing to FE. Users also have access to direct properties of the cross-sections, like their limit loads, average crushing forces or lengths of folding waves, thus simplifying and speeding up the design process. In this paper a Visual Crash Studio (VCS) program was used.

1.1 Creation and validation of the ME model – comparison to FE model

Basing on a shell mesh of the SLC, superbeam elements were created in the VCS (figure 5). All shear panels, like floor, roof sheets as well as the windshield, were modeled by use of equivalent diagonal strips. Nonstructural effects, like behavior of crushed pneumatic wheel, were simulated by user defined elements with predefined characteristics. Typical computing time of the model was around 14 seconds on a single core of modern PC per 100 ms of simulation time. The macro element model has been validated by a comparison to results of the FE model simulations.
Selected results for side pole impact cases are presented in figures 6 and 7. Colours on the macro element model results express the state of each superbeam element: grey—elastic; green—limit load; yellow—post collapse; red—deep collapse (fully formed plastic folds of thin walled surfaces). Deformation plots show good convergence between detailed and simplified models. Velocities are differing in case of lateral crush, due to the not modeled denting effect of central tunnel and side rocker panel. Histories of acceleration show convergence on average accelerations only. It reveals the vibrating nature of the macro element model which is caused by differences in mass granulation and wave propagation and influence of simplifications in the macro element model. For comparison, the number of nodes in the FE model is as much as 954 thousands, while in the macro element model of the SLC only 172 are needed.
Figure 23: Comparison between FE and ME models results for pole crash. Deformations at 100ms of simulation time.
The next section describes the use of an initial packaging data defining battery locations used for modification of the floor subassembly, incorporating battery housings.

1.2 Different Pack locations in the macro element software tool – Faster development process

During the concept phase, several configurations of battery pack locations were tested in the used macro element model software. Among others, two most interesting versions (figure 8), derived from the volume defined in chapter Error! Reference source not found., were selected for presentation in this paper. Version A includes central and side battery pockets plus a rear battery housing. The second pack, namely version B, consists of version A and additional side pockets spread along the whole floor of the passenger compartment. Both packaging concepts use space under second row seats (fuel tank location of the ICE vehicle) for battery storage. The central tunnel was closed from the bottom side, allowing for additional package location (i.e. for power electronics or extra batteries).

Version A incorporates space under first row seats, where lateral battery housing was located. This configuration keeps initial occupant space intact, while version B needs an increase of “H point” locations of all passengers, and movement of the roof surface upwards.

The lateral battery housings were used as structural parts providing support of lateral crash absorbers. This concept was created to improve side and pole impact characteristics, as well as for increased protection of stored batteries, allowing for centered position of the CoG. None of the versions were strictly optimized, being only subjected to engineering optimization based on an iterative process of modification–check loop scheme, what is sufficient for concept assessment.

Figure 24: Comparison between FE and ME models results for pole crash. From the top: velocity [m/s] of node of lower B-pillar on side opposite to impact; acceleration [m/s^2] of node of lower B-pillar on side opposite to impact; penetration [mm] measured between lower parts of B-pillars. All curves displayed as a function of time [ms]. Results from finite elements on left, results from macro elements on right.
1.2.1. Lateral pole crash at B-pillar location

Standard Euro NCAP lateral pole crash simulation, with initial velocity equal to 29 km/h and impact point located near B-pillar, was calculated for both considered versions. Some results are presented in Figure 26, including force–displacement curves for both cases, where reaction force on the impacting pole was presented in function of lateral distance between rocker panels ( intrusion). Lateral crushing (denting) behavior of rockers was not covered in the simulation. As it can be seen in Figure 26 version A achieved a maximum intrusion depth of 200 mm, while for version B it was 230 mm.
1.2.2. Lateral pole crash at front door location

The performance of both versions (A and B), having intrusion at standard B-pillar pole test from 200 mm to 230 mm is considerably better than the original configuration of the SLC, not equipped with side impact energy absorbers. Electric versions are mutually comparable in terms of additional test results. For example in test with the impacting pole moved to the middle of the front doors, intrusions were in range of 175 mm to 210 mm (shown in figure 10).

Battery pack location is influenced by several different conditions, which are mostly multidisciplinary. From a crash engineering scope, distributed lateral impact absorbers have multiple advantages, which may be observed on force-crush plots, where distributed absorbers in version B have better force-displacement curve filling (near idealized square). This provides also an almost equal support of the rocker during crash in different impact positions. However, packaging limitation has not made it applicable for further development steps.

![Figure 26: Results of the pole crash at 29km/h with normative pole location at B-pillar area. Left side shows A, right side B versions results. From the top: isometric view of deformed ME model at 100ms of simulation, bottom view and crash force [kN] versus intrusion depth [mm] plots.](image-url)
1.2.3. Detailed model development based on version A

On the starting point the ICE version of the SLC was discretized into a macro element model. Its floor panel was replaced with battery housings, and proposed structural members represented with simplified cross-sections, with equivalent properties. After the above described concept phase, design version A was selected for further development.

Example characteristics, of a thin walled rectangular cross-section, are shown in figure 11. Axial crushing, two plane bending and twisting characteristics are defining the behavior of a superbeam member. Influence of their mutual interactions to resultant response is simulated by the software, however not taken directly into account in the methodology. Fully identical responses between simplified and final members are not needed, limiting it rather to most important parameters like: axial limit load \( (P_{\text{max}}) \), average crushing force \( (P_m) \), length of plastic folding wave \( (2H) \), bending limit torques \( (M_{xx\text{, max}}, M_{yy\text{, max}}) \), approximate character of post collapse bending characteristics and torsion limit load \( (T_{\text{max}}) \).

<table>
<thead>
<tr>
<th>Axial crushing</th>
<th>Bending</th>
<th>Twisting</th>
</tr>
</thead>
</table>

Figure 27: Results of the pole crash at 29 km/h with pole in location at middle of front doors. Left side shows A, right side B versions results. From the top: isometric bottom view of deformed ME model at 100ms of simulation and crash force [kN] versus intrusion depth [mm] plots. An intrusion depth was measured between rockers centroids at point of impact.

Basic structural crash parameters for central tunnel and side pockets of version A design are shown in table 5. Shapes obtained on this level, cannot be treated as direct design cross-sections, but rather as equivalent shapes defining an approximate design envelope, with determined minimal strength parameters for a more detailed structure.

Figure 28: Example characteristics for axial crushing, bending in two principal directions and torsion for a rectangular cross-section.
The next step of the development process incorporates more detailed design of the cross-sections, and interferes with design space modifications, what usually occurs very often at early stages. In this particular example, version A based on floor modules was developed with modifications (figure 12) due to addition of front seats to BIW attachment points and final battery packs dimensions. The lateral absorber body is proposed as a four cell thin walled profile, made of aluminum alloy, connected to the internal side of the rocker. After this stage a first CAD design and finite element design-assessment loop can begin.

Table 1: Properties of the equivalent cross-sections of the A version.

<table>
<thead>
<tr>
<th><em>Central battery housing</em></th>
<th><em>Central Battery Housing 2</em></th>
<th><em>Lateral battery housing</em></th>
<th><em>Rear battery housing</em></th>
<th><em>Side impact absorber</em></th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_{\text{max}} ) [kN]</td>
<td>231.1289</td>
<td>264.2503</td>
<td>247.8043</td>
<td>108.9953</td>
</tr>
<tr>
<td>( P_{\text{m}} ) [kN]</td>
<td>105.947</td>
<td>117.9981</td>
<td>150.2441</td>
<td>65.3936</td>
</tr>
<tr>
<td>( 2H ) [mm]</td>
<td>86.06</td>
<td>108.72</td>
<td>133.98</td>
<td>122.23</td>
</tr>
<tr>
<td>( MAX(+) ) [kN/m]</td>
<td>26.0623</td>
<td>18.6912</td>
<td>50.2565</td>
<td>20.2024</td>
</tr>
<tr>
<td>( Myy ) [kN/m]</td>
<td>13.2843</td>
<td>24.0293</td>
<td>9.9368</td>
<td>8.8619</td>
</tr>
<tr>
<td>( T_{\text{max}} ) [kN]</td>
<td>20.1635</td>
<td>26.5852</td>
<td>27.0841</td>
<td>16.844</td>
</tr>
</tbody>
</table>
2 CONCLUSIONS

This paper has shown that the high simulation speed and reasonable accuracy of a macro-element based simulation can be used to significantly improve the design of a battery pack for a BEV. The method presented in this paper can be applied to different investigation issues (structural optimization, battery integration, fast boundary development). One of them, the battery integration process described above was improved with this method in quality, time and costs.

Some advantages can be listed as:

- **Element/Node Reduction:** 954,000 Nodes in FE against < 200 Nodes in ME
- **Time reduction** from 14 hours (8 cores) by usage of the FE model to approximately 14 seconds on a single core for the ME model
- **This allows a broader view** (~1000 simulations on different concept types with defined crash scenarios) and a better understanding of the possible concepts

An enhanced battery housing was the result.

A significantly speed up of the integration additionally allowed fast engineering optimizations even on the structural components of the BiW (30% less intrusion was reached for pole crash). That is a reasonable way as the focus on battery integration cannot exclude the surrounding structure because of problems in stiffness variations. These parts (front longitudinals, floor panel, side rocker, energy absorbers) should buckle/collapse before the housing is destroyed. Additionally, the view can be on the battery housing when a predefined plastic hinge, acting as a structural fuse, is included to minimize loadings on the relevant housing parts. The results of the method then can be used for integration in the FE model of the vehicle and there further optimized with only one last concept envelope.
The final battery housing envelope described above cannot be seen as the optimum for every vehicle type as some restrictions in the design process and the main focus in this study was only a crashworthy integration of the battery housing.

Table 6 gives examples of the pros/cons:

<table>
<thead>
<tr>
<th>Battery envelope from Figure 29</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>pros</strong></td>
</tr>
<tr>
<td>- Meets the SmartBatt targets</td>
</tr>
<tr>
<td>- High grade of integration into the BiW structure</td>
</tr>
<tr>
<td>- Crash protected battery modules</td>
</tr>
<tr>
<td>- Using upper housing part as “floor panel” gives advantages in weight reduction, cost effectiveness and easy assembly</td>
</tr>
</tbody>
</table>

Table 2: Pros and Cons for the final battery envelope
Chapter 4

Visual Crash Studio NEWS

New features and improvements introduced to VCS in the year 2012 / 2013

VCS 2.1
✓ Area Selection
✓ Create Node Tool
✓ Move Node Tool
✓ Project Node To Line Tool
✓ Create Sphere Tool
✓ Create Cone Tool
✓ Create Box Tool
✓ Create Ruler Tool
✓ Create Ruler From Beam
✓ Ruler Properties
✓ Center Of Gravity
✓ Toggle Global Draw Type
✓ Zoom To Selected Object
✓ Gauges Quick View
✓ Keyboard shortcuts definition
✓ Recent files
✓ Export STL file
✓ Other minor changes

VCS 2.2
✓ Repository
✓ Visual Software Digitizer
✓ Rotate Nodes Tool
✓ Project Node Tool
✓ Ruler tool improvement
✓ Protractor tool improvement
✓ Help
✓ Move Node tool improvement
✓ Create Rigid Body tools
✓ Undo & Redo
VCS 2.3

✓ Cross Section Optimizer
✓ Solution Layers
✓ Orientation tool
✓ Welcome Screen
✓ Merge Proximate Nodes
✓ Copy Cross Section from other Beam
✓ Cross Section Editor – Lateral Crushing & Denting
✓ Cross Section Comparison tool
✓ Import and Units Conversion Utility
✓ Mass of selected Model
✓ Chart wizard – time line
✓ Cross Sections Editor – dimensions
✓ Clean Model
✓ Add new Cross Section in the Solution Explorer tree
✓ Select all Beams with the chosen Cross Section
✓ New ‘Calculate Solution’ icon

Cross Section Optimizer

New optimization tool for Cross Sections
VCS 2.3 provides the new Cross Section Optimizer. The Cross Section Optimizer provides the possibility to optimize any Cross Section of the Solution.

Based on the genetic algorithm, and the calculation algorithms of the VCS Cross Section Editor, the Cross Section Optimazer is a fast and accurate tool for optimization. The Cross Section Optimizer enables the user to effectively look for an optimal Cross Section geometry.

The Cross Section Optimizer gives the possibility to define a number of Optimization Parameters. In the Optimization Parameters section of the Optimizer window you can set the maximum and minimum values for: Bounding Box, Specific Mass, Squash Load, Mean Crushing Force and SEA.

Moreover in the Parameters field you can define the number of Iterations, strenght and coverage of the optimization, as well as the weight of Design Recommendations for the chosen Cross Section. For each optimization parameter you should set the requested weight in [%]. The higher percentage of the weight, the more significant the parameter will be in the scoring calculations.

Similarly to the definition of optimization Parameters, before the launch of the optimization process Optimization Constraints might be defined, and their weight set.

Cross Section Optimizer provides three types of constraints: move constraints, length constraints and angle constraints. When the optimization procedure is finished you can accept the Optimized Cross Section. The Optimized Cross Section will be saved and visible in the Solution Explorer window. It will be automatically named as: ‘initial cross section name (Optimized)’, and will be added to the end of the Thin Walled Cross Section list. You can also save the whole optimization project containing all setting made for the optimization parameters and optimization constraints. The optimization project will be saved and visible in the Solution Explorer window. It will be automatically named as: Optimization of “initial cross section name”, and will be placed on the end of the Solution Explorer tree under the Optimization Projects branch.

For more information see: www.impactdesign.pl/