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Thin-Walled Structures 41 (2003) 91–107
THIN-WALLED STRUCTURES
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Thin-walled structures as impact energy absorbers

W. Abramowicz *

Warsaw University of Technology, Institute of Vehicles, Narbutta 84, Warsaw, Poland

Abstract

The key structural components of the majority of transportation vehicles are designed as thin-walled components. During a crash event, a number of structural components must sustain abnormal loadings in order to meet stringent integrity requirements. At the same time other components must dissipate impact energy in a controlled manner that limits the deceleration of a vehicle to a required safety limit. The present paper focuses on the crushing mechanics of thin-walled components. The analysis method is based on the Superfolding Element (SE) concept, which originates from experimentally observed folding patterns of crushed shell elements. The paper presents milestones of the underlying theory of plastic shells and basic design considerations that are coupled with the SE-based predictive techniques in a CAE software. The paper also presents basic examples of the design process of typical energy absorbing components.

Keywords: Crushing mechanics; Large plastic deformations of shells; Energy absorbing structures; Design for crash

1. Introduction

The majority of structural components of sea, land and air vehicles are designed as thin-walled structures. During decades of growth of the transportation industry a variety of design rules and recommendations were established that help practicing engineers in the design of robust structures. Virtually all the early design codes deal with standard loading conditions encountered during the standard operation of a vehicle. In terms of structural analysis this means that the corresponding compu-

* Tel.: +48 22 723 8418; fax: +48 22 723 84 03.
E-mail address: wa@impactdesign.pl (W. Abramowicz).

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doi:10.1016/S0263-8231(02)00082-4

tational models are restricted to the elastic analysis typically limited to infinitesimal deformations.

In the early 1960s automotive safety regulations opened a new, demanding area of engineering analysis in all sectors of transportation industry. Apart from the standard requirements, thin-walled structures are now designed to sustain abnormal loadings encountered during various kinds of accidents. In general, a vehicle designed for crash (a crashworthy vehicle) must meet integrity and/or impact energy management requirements. For example, in the case of car, plane or train accidents, the structure of passenger(s) compartments are required to sustain crash loading without excessive deformations that compromise safety of passengers. At the same time other structural components must dissipate the kinetic energy of a vehicle while keeping the deceleration level below the tolerable limit. Several structural assemblies, like a side panel of a passenger car, must meet both requirements simultaneously.

In the case of low speed accidents, for example, a ship grounding on a narrow rock, the dissipation of impact energy is not a key issue. The energy absorption capacity of a ship hull is typically much too small to stop the grounding ship. So that designers focus on the integrity aspect, that is to say how to minimize the damage to the hull structure and resulting secondary damages like oil spillage etc.

New design demands set forth by the safety standards initiated rapid development of the design and simulation tools for crash. In the early days of crashworthiness experimentation and the resulting experimental recommendations were perhaps the most important and trusted tools in the design departments. At the same time the numerical tools based on the non-linear Finite Element (FE) approach were lagging behind industry expectations and it is only in the late 1980s when they get mature enough to be used in an actual design process. In parallel with FE codes, another simulation technique, referred to as kinematic or macro element approach, was developed. This technique originates from the pioneering work of Alexander [1] who has solved the problem of progressive collapse of axially compressed cylindrical tubes by assuming the kinematically admissible deformation field on the basis of experimental observations rather then theoretical assumptions. The method suggested by Alexander [1] allows for the determination of the basic parameters of the crashing process by means of relatively simple formulas. In addition, it offers a deeper insight into the folding process itself—a necessary prerequisite for the development of design oriented simulation tools. The present paper is focused on the kinematic approach to the crushing mechanics and its application to the calculation and design of thin-walled components for optimal impact energy dissipation.

2. The analysis method

Several common features characterize the deformed pattern of crushed thin-walled components as explained in Fig. 1. The plastic crushing process develops following the elastic or elastic–plastic buckling and is characterized by localization of plastic deformations in relatively small parts of a structure. Plastic deformations are localized in narrow hinge lines where most of the plastic deformation takes place

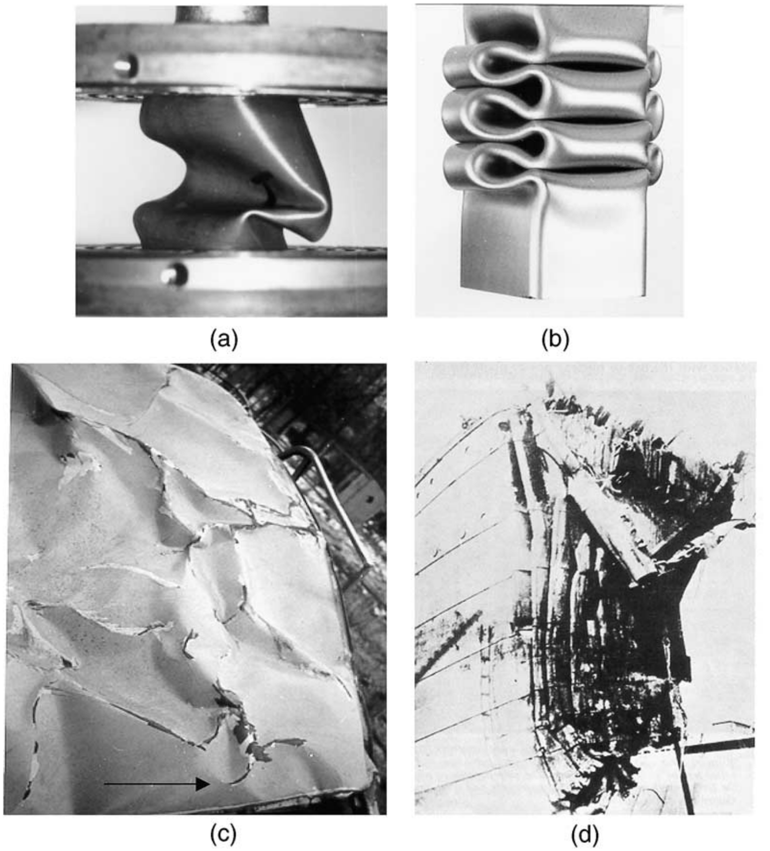

Fig. 1. Representative folding patterns of crushed shell structures are composed of similar basic pattern (A), referred to as a Superfolding Element (SE). Four Superfolding Elements create entire layer of folds in a crushed square column (B). More complex folding patterns like crashed roof of a car (C) or folded bow of a ship (D), [5], are composed of complex combination of SE's.

while the global deformation of a structure results from rigid body motion of undeformed or slightly deformed shell segments. An interesting feature of the crushing process is a 'geometrical similitude'. It has been observed that most of the actual folding patterns of shells can be represented by an assemblage of a single representative folding element, illustrated in Fig. 1A. Deformation of such an element is described by using the concept of a specialized macro element referred to as Superfolding Element (SE).

The cross-sectional dimensions of a SE and its most general deformation mode are shown schematically in Fig. 2. 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, Φ

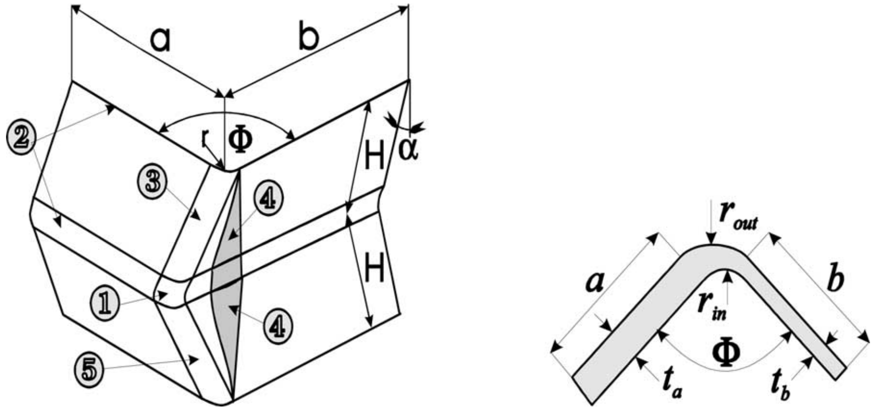

Fig. 2. Idealized deformation mechanisms and cross-sectional dimensions of a Superfolding Element.

3. wall thickness *ta* of the arm of the length *a*
4. wall thickness *tb* of the arm of the length *b*.

The plastic folding of the element involves creation of five different deformation mechanisms. These are (refer to Fig. 2):

- Deformation of a 'floating' toroidal surface.
- Bending along stationary hinge lines.
- Rolling deformations.
- Opening of conical surfaces.
- Bending deformations along inclined, stationary, hinge lines following locking of the traveling hinge line.

The general folding mechanism in Fig. 2 is constructed from two simpler folding modes, illustrated in Fig. 3. These modes are referred to as an asymmetric and sym-

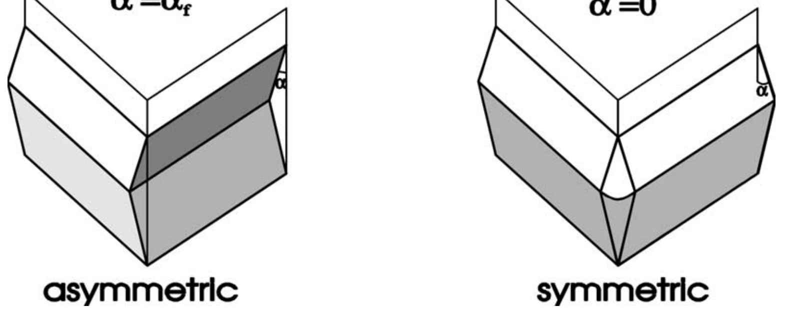

Fig. 3. Two fundamental folding modes of a single Superfolding Element controlled by limit values of the switching parameter $\alpha*$.

metric deformation mode, respectively (the mode shown in Fig. 2 is called an asymmetric mixed mode). A progress of the deformation process in each mode is controlled by a single process or time-like parameter α, $0 \le \alpha \le \alpha_f$, which defines the rotation of a side face of an element from the initial upright position, Fig. 2. At the initiation of the folding process $\alpha = 0$. The process terminates when $\alpha = \alpha_f = \pi/2$. The asymmetric deformation mode is short of the conical surface 4 in Fig. 2. Consequently, the propagating hinge line, 3, controls the entire folding process. The symmetric deformation mode, on the other hand, lacks the propagating hinge line 3 in Fig. 2. In this case local extensional plastic deformations are confined to the conical surface 4 as shown in Fig. 3B. The development of a particular folding mode in Figs. 1 and 3, is controlled by a single switching parameter $\alpha^*_,$ $0 \le \alpha^* \le \alpha_f$.

This parameter defines a configuration at which symmetric mode takes over the control of the folding process. If $\alpha^* = \alpha_f$, the folding of a SE is controlled by an asymmetric mode alone while the case $\alpha^* = 0$ corresponds to a purely symmetric mode, see Fig. 3. For $0 \le \alpha^* \le \alpha_f$, both mechanisms are involved in the folding process: folding starts as an asymmetric mode and continues up to the point where the moving hinge line 3 is locked within an element. At this point the conical surface 4 starts to grow. An actual value of the switching parameter, $\alpha*$, depends on both the input parameters, $\{C, t, \Phi\}$, and constraints imposed onto deforming faces of a SE. In the case of an unconstrained or standing alone SE the asymmetric mode of deformation, $\alpha* = 0$, is predominant for right angle and acute elements, $\Phi \le \pi/2$, while the symmetric mode controls the folding process of obtuse elements with the central angle, Φ larger then 120 degrees, approximately. In the intermediate range of central angles both modes coexist while the fractional contribution of each mode to the total energy dissipation depends on the central angle, Φ, and the width to thickness aspect ratio, C/t. The folding modes of a standing alone SE are referred to as natural folding modes.

An SE that models single plastic buckle in an entire folding layer, see e.g. Fig. 1B, is constrained by neighboring elements. Kinematic constraints are imposed onto the element either by the deformation of one or two arms of the element in a predefined direction or by constraining the deformation of the element's corner line. The former case is typical for an assemblage of elements, which model the deformation of columns with closed cross-section. In this case the requirement of circumferential continuity of the deformation field may induce in some elements deformation modes different than the natural folding mode. Similarly, constraints imposed onto the corner line may change the natural deformation mode of an element. During the deformation of 'X' and 'Y' sections the continuity conditions imposed onto the common corner line of all contributing flanges prevent the development of asymmetric modes.

2.1. The solution procedure

The solution procedure to the crushing problem, formulated in terms of the kinematic method, is typically obtained in three steps. First, the deformation pattern (displacement or velocity field) of a structural member or a representative segment of structural member (macro element) is postulated on the basis of experimental observations. Typically the postulated deformation pattern is determined with accuracy to a vector, β, of free geometrical parameters (note that β does not depend on time-like parameter α). For example, in the case of an SE in Fig. 2 the geometry of the element is defined with accuracy to the length of the plastic folding wave, $2H$, average rolling radius, r, and switching point parameter $\alpha*$. All these parameters must be found as a part of the solution procedure. Once the deformation field is postulated, the functionals corresponding to the rate of internal energy dissipation and total energy dissipation are defined in terms of the adopted theory of plastic shells (the above functionals are defined with accuracy to the vector β). In the last step of the solution procedure the 'optimal' values of the vector β are defined by means of an appropriate minimum condition. Readers interested in more detailed discussion of the above procedure are referred to the base publication on this subject: [2,3,4,7].

2.1.1. The rate of internal energy dissipation and total energy dissipation

The rate of internal energy dissipation in a deformed shell element results, in general, from the continuous and discontinuous velocity fields, [4,7].

$$\dot{E}_{\text{int}} = \int_S (M_{\alpha\beta}\dot{\kappa}_{\alpha\beta} + N_{\alpha\beta}\dot{\varepsilon}_{\alpha\beta})dS + \sum_{i=1}^{n}\int_{L^i} M_o^i[\dot{\theta}_i]dl^i \tag{1}$$

In Eq. (1) S denotes the current shell mid surface, n is the total number of plastic hinge lines, Li is the length of the ith hinge while $[\dot{\theta}_i]$ denotes a jump of the rate of rotation across a moving hinge line. Components of the rate of curvature and rate of extensions tensors are denoted, respectively, as $\dot{\kappa}_{\alpha\beta}$ and $\dot{\varepsilon}_{\alpha\beta}$ while $M_{\alpha\beta\beta}$ and $N_{\alpha\beta}$ are the corresponding conjugate generalized stresses.

Consider, as an example, the deformation of an SE illustrated in Fig. 1. In the SE the assumed deformation filed is composed of axisymmetric shells (cylinders and toroids). In addition, inextensibility is assumed in the hoop direction. Consequently, the expression for the rate of internal energy dissipation is

$$\dot{E}_{\text{int}} = \int_S N_o\dot{\varepsilon}_1 dS + \sum_{i=1}^{n}\int_{L^i} M_o^i[\dot{\theta}_i]dl^i \tag{2}$$

where $\dot{\varepsilon}_1$ is the rate of straining in a principal direction, tangent to the shell's mid surface and equals to a corresponding component of the rate of deformation tensor $\dot{\varepsilon}_1 = d_{11}$. The strain rate component perpendicular to the mid surface does not contribute to the internal energy dissipation due to specific form of the Tresca yield condition used in the present calculations. Integrating Eq. (2) in the interval $0\leq \alpha\leq\alpha_f$ renders the expression for energy dissipation in a single SE

$$E_{\text{int}}(\beta) = \int_0^{\alpha^*} {}_{(1)}\dot{E}_{\text{int}}d\alpha + \int_{\alpha^*}^{\alpha_f} {}_{(2)}\dot{E}_{\text{int}}d\alpha \tag{3}$$

Two integrals on the right hand side of Eq. (3), defined through the switching parameter $\alpha*$, correspond to the contribution of asymmetric and symmetric modes, respectively. In practical calculations it is convenient to define following expressions for the membrane and bending contributions to the internal energy dissipation

$$
\begin{aligned}
E^N_{\text{int}} &= \int_0^{\alpha} d\alpha \int_S N_0 \dot{\varepsilon} dS = \sigma^N_0(\bar{\varepsilon}) \int_0^{\alpha} d\alpha \int_S t\dot{\varepsilon} dS \\
E^M_{\text{int}} &= \sum_{i=1}^{n} \int_{L^i} M^i_o [\dot{\theta}_i] dl^i = \sum_{i=1}^{n} {}_{(i)}\sigma^M_0(\bar{\varepsilon}) \frac{t}{4} \int_0^{\alpha} d\alpha \int_{L^i} [\dot{\theta}_i] dl^i
\end{aligned}
\tag{4}
$$

where σ^N_o and σ^M_o denote, respectively, an average level of the flow stress in the entire crushing process. This stress is referred to as an *energy equivalent stress* and is discussed in more detail later in this section.

2.1.2. External loading and global minimum condition

The internal energy dissipation, Eq. (2), is determined with accuracy to a vector of free geometrical parameters of the process, β. Typically, the set of 'optimal' parameters, β^o, is defined from the postulated global minimum condition; see, for example, Abramowicz [3]

$$\beta^o = \min_{\beta}\left[\frac{\Delta E^*_{\text{int}}}{\Delta\delta}\right] \tag{5}$$

where $\Delta\delta$and ΔE_{int} are, respectively, a global deformation measure and the internal dissipation, pertinent to the given crushing process. For example, in the case of axial deformation of a SE in Fig. 2, $\Delta\delta$ corresponds to the length of a local plastic folding wave, *2H*, while internal dissipation is given by Eqs. (3) and (4). In the latter case Eq. (5) reduces to

$$\beta^o = \min_{\beta}\left[\frac{\Delta E^*_{\text{int}}}{\Delta\delta}\right] = \min_{\beta}[P_m] \tag{6}$$

where P_m is referred to as a mean crushing force and corresponds to an average energy dissipation per unit shortening of a SE.

Once the internal energy dissipation is determined, the external loading is found by equating the rate of internal energy dissipation to the rate of external work. In the shell theory the rate of external work is expressed in terms of global cross-sectional forces, *P*, and moments, *M*

$$\dot{E}^o_{ext} = \mathbf{P}\dot{\delta} + \mathbf{M}\dot{\theta} \tag{7}$$

and in general consists of six contributions. In practical applications the number of generalized forces *P* and *M* is reduced to one or two, refer to Wierzbicki [7] and Abramowicz [2].

2.1.3. The energy equivalent stress measure

Determination of proper flow stress measures is crucial for accuracy of all calculations based on the macro element approach. The material properties enter the energy functionals, Eq. (4), via the equivalent stresses σ_o^M and σ_o^N. These stresses are defined, respectively, as [2]

$$\sigma_0^N(\bar{\varepsilon}) = \frac{1}{\bar{\varepsilon}}\int_0^{\bar{\varepsilon}} \sigma(\varepsilon)d\varepsilon \sigma_0^M(\bar{\varepsilon}) = \frac{1}{\bar{\varepsilon}^2}\int_0^{\bar{\varepsilon}} \sigma_0^N(\varepsilon)\varepsilon d\varepsilon \tag{8}$$

and correspond to an average level of the plastic flow stress in regions subjected to quasi-static uniaxial tension/compression or bending, characterized by a representative strain $\bar{\varepsilon}$ (in most cases $\bar{\varepsilon}$ corresponds to maximal strain in a given region). In Eq. (8) $\sigma_v(\cdot)$ corresponds to a standard quasi-static tensile characteristic. In the case of dynamic processes the visco-plastic constitutive relation, $\sigma(\varepsilon, \dot{\varepsilon})$, is postulated as a product of two contributions

$$\sigma(\varepsilon,\dot{\varepsilon}) = \sigma_v(\varepsilon,\dot{\varepsilon}_v)\gamma(\varepsilon,\dot{\varepsilon}) \tag{9}$$

where $\sigma_v(\cdot)$ corresponds to the standard quasi-static tensile characteristic of a given material, determined at the constant strain rate, $\dot{\varepsilon}_v$, (10^{-3} [1/*s*]$\leq \dot{\varepsilon}_v \leq 10^{-4}$ [1/*s*]) while the dynamic factor$\gamma(\cdot)$describes strain rate effects (the stress and strain measures, used in Eqs. (8) and (9) correspond, respectively, to the Cauchy stress σ and logarithmic strain ϵ).

2.1.4. Crushing response of a single Superfolding Element and an assemblage of elements

The explicit expression for the mean crushing force, P_m, corresponding to a complete folding of a single SE is defined by substituting expressions for contributing energy dissipation mechanisms in Fig. 2 into the governing energy functionals, Eqs. (3) and (4). The final form of the governing expression is (a complete derivation of the following result is given by Abramowicz [2,4]),

$$P_m = \frac{t^2}{4}\{\sigma_o^N(\bar{\varepsilon}_1)A_1\frac{r}{t} + \sigma_o^M(\bar{\varepsilon}_2)A_2\frac{C}{H} + \sigma_o^M(\bar{\varepsilon}_3)A_3\frac{H}{r} + \ + \sigma_o^M(\bar{\varepsilon}_4)A_4\frac{H}{t} + \sigma_o^N(\bar{\varepsilon}_5)A_5\}\frac{2H}{\delta_{eff}} \tag{10}$$

The five terms in parenthesis on the right hand side of Eq. (10) correspond, respectively, to fractional contributions to the total energy dissipation resulting from five elementary deformation mechanisms, identified in Fig. 2. The five factors, *Ai i* = 1,2...5 result from the surface-time integration of functionals defined in Eqs. (3) and (4). Factors. A_2 and A_4, are readily calculated as a closed-form functions of geometrical parameters. The remaining factors A_1, A_3 and A_5, are functions of elliptic integrals and must be calculated numerically. The meaning of other variables in Eq. (10) is explained in the following section.

The crushing response of an assemblage of SE is calculated by summing up fractional contributions of all active SE. For example in the case of progressive axial crushing of prismatic columns (compare Fig. 2B) the governing equation of the problem is

$$P_m = \sum_{i=1}^{J} \frac{t_i^2}{4}\{\sigma_0^N(\bar{\varepsilon}_1)A_1^i\frac{r}{t_i} + \sigma_o^M(\bar{\varepsilon}_2)A_2^i\frac{C_i}{H} + \sigma_o^M(\bar{\varepsilon}_3)A_3^i\frac{H}{r} + \sigma_o^M(\bar{\varepsilon}_4)A_4^i\frac{H}{t_i} + \sigma_o^N(\bar{\varepsilon}_5)A_5^i\}\frac{2H}{\delta_{eff}} \qquad (11)$$

where the summation is extended over the J contributing SE. It is tacitly assumed that the column is made of one material, so that all average stresses are calculated on the basis of a single constitutive relation. Each element, however, may have different geometrical dimensions: Ci, Φi and ti, $i = 1, 2, ...J$. Since, all elements in a given layer of folds deform with the same length of the folding wave, $2H$, there is only one 'H' parameter in the governing equation. In order to simplify the calculation routines it is also assumed that values of the two other free parameters, i. e. the rolling radius r and the switching parameter α^*, are also the same in all contributing elements.

Parameters corresponding to the equilibrium of a SE or an assemblage of SE's are determined from the set of three nonlinear algebraic equations, resulting from the minimum condition, Eq. (6)

$$\frac{\partial P_m}{\partial H} = 0; \frac{\partial P_m}{\partial r} = 0; \frac{\partial P_m}{\partial \alpha *} = 0$$

In general the set of above algebraic equations has no closed-form solution and therefore an iteration procedure is needed in order to determine crushing response of an arbitrary structure. It should be noted however that the most CPU expensive iteration procedure does not depend on the number of SE in the model. Indeed, the equation that governs crushing response of a single element, Eq. (10), differs form its counterpart for an assemblage of SE's. Eq. (11), only by the summation procedure that must be done prior the minimization of the governing functional. This specific feature of the SE formulation for progressive crushing results in an inexpensive numerical algorithm capable of analyzing even very complex structures in seconds on a standard PC.

3. Effective folding modes of thin-walled prismatic members

The primary energy absorbing members of a vehicle are typically designed as an assemblage of thin-walled prismatic segments. From the point of view of the crash energy management, the quantity of interest is the total amount of energy that a given element can absorb. The energy dissipation is defined as:

$$E = \int_0^{\delta_{max}} P(\delta)\ d\delta \tag{12}$$

where $P(\delta)$ is an instantaneous crushing force of a given member, δ and $\delta_{\max}$ are, respectively, the current and maximum attainable crush distance while P_m is the mean crushing force. A good design is one in which not individual components of Eq. (12) but the product of P_m and $\delta_{\max}$ is maximized. The mean crushing force depends primarily on the gauge thickness t, average flow stress of the material σ_o, Eqs. (8) and (9), and the number of corners in a given member [4]. It is relatively easy to adjust the above parameters (t and σ_o) so that the required level of the crushing force P_m is reached. However, it is much more difficult to maintain the required average force level over the entire available crushing distance $\delta_{\max}$. Therefore, the problem of optimum design against crash is equivalent to the requirement that a steady, progressive collapse pattern is maintained in a column throughout the entire crushing process. In successfully designed columns, the folding pattern shall rapidly converge to a stable, repeatable mode. However, the regular symmetric folding process is often disturbed in incorrectly designed members and a structure may undergo a premature bending collapse. Elementary examples of only some typical design flaws encountered in thin-walled columns are illustrated in Figs. 4 and Fig. 5.

Three columns in Fig. 4 did not collapse axially due to apparent flaws in the cross-sectional geometry. Column A in Fig. 4 has a trapezoidal cross-section with two acute and two obtuse corner elements. The resulting deformation mode is not an axial crushing but flattening of the cross-section. Flattening of the cross-section leads to the reduction of bending stiffness, which is followed by global bending of the column. The difference between acute/obtuse central angles of the cross-section is diminished in column B in Fig. 4. This design change results in the initiation of a

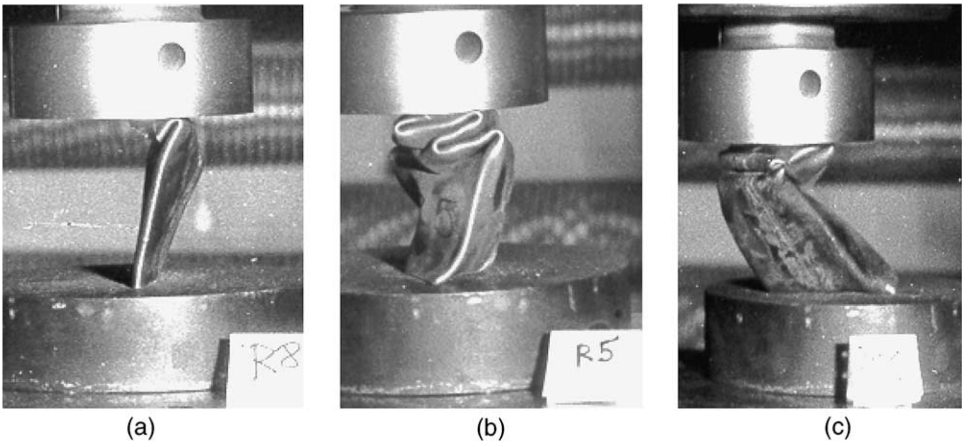

Fig. 4. Global instability of axially compressed prismatic column resulting from incorrect cross-sectional geometry.

Fig. 5. Global bending of non-triggered square columns resulting from the development of symmetric and 'inverted' folding modes indicated by white arrows

desired asymmetric folding mode (refer to Fig. 3) of neighbouring corner elements. However, the resulting plastic folding wave is too bulky to fit inside the deformed column. Consequently the plastic folds crash against each other inside the column and trigger overall bending. Finally, column C in Fig. 4 has a triangular cross-section, which prevents development of the desired asymmetric folding mode illustrated in Fig. 3.

The desired repeatable folding pattern, composed of asymmetric folding modes of corner elements, is frequently induced in square columns as shown in Fig. 1B. However, the initiation of such a desired folding pattern can be disturbed by creation of other modes such as symmetric or inverted mode illustrated in Fig. 5.

Once one of these modes is induced in the folded column during the initiation of the folding process or later in the progressive collapse range, the overall stability of the process is lost and column falls in bending. On the other hand columns in which the desired folding pattern is introduced during the manufacturing process can develop dozens of plastic folds without any tendency to overall bending as illustrated in Fig. 6.

The preceding short overview of the most typical design flows addresses only part of the requirements that must be met by successfully designed column. Development of progressive folding requires simultaneous completion of several other conditions. These are:

- The cross-section topology must be properly designed, so 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 it's closest neighbor,
- 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 'natural' fold-

Fig. 6. Progressive folding of properly triggered prismatic members subject to the predominantly axial crash loading.

ing 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.

4. Computer assisted design of energy absorbing systems

Design of an energy absorbing structure is a highly iterative process that typically requires several calculation/design loops. The objective of each design step is achieved by using specialized computational tools. The selection of an appropriate tool for a given step depends on the complexity of the problem and availability of suitable software. For example, in the pre-design or pre-prototyping stages factors such as the specific energy absorption per unit length or the maximal moment capacity of a cross section are of primary importance. In the case of typical cross sections such information is readily available in handbooks. In the case of more complex, real world members, the desired parameters can be easily calculated by using specialized tools. This section demonstrates how the concept Superfolding Element, implemented in the computer program CRASH CAD, is used at early stages of the design process. It is shown how the synthetic approach to the design process proceeds from selection of a cross-section and its basic dimensions, to the pre-design concept of a structural assembly.

4.1. Crashworthiness of prismatic members

Fast design of prismatic members is remarkably important at the early design stages when the proper shape and optimal dimensions of a member are sought and

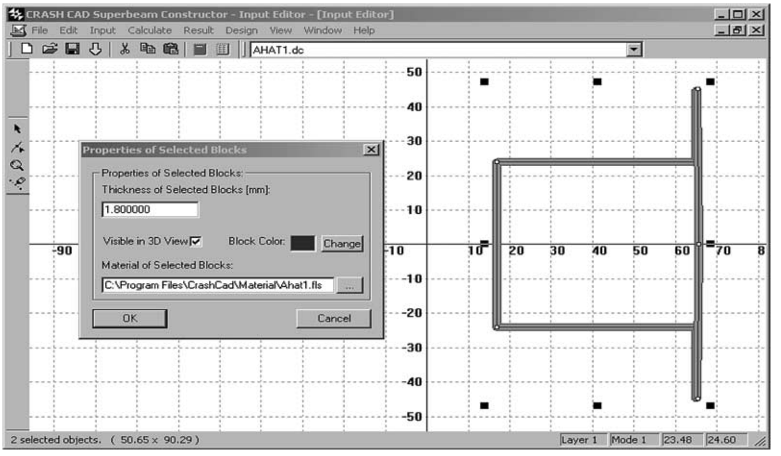

Fig. 7. Input data screen of Crash Cad with a discretized cross-section of a hat member. Only overall dimensions of a section are needed in order to define the Crash Cad input data file.

the design concept undergoes frequent modifications. Usage of specialized dynamic FE codes is not justified at this stage of the design due to the excessive modeling effort and processing time. On the contrary, application of a software based on the Superfolding Element concept is especially attractive at this stage of the design since the input routine of such a program requires only overall dimensions of the cross-section while the processing time is of the order of 1 second on an average PC, Fig. 7. The robustness of the SE-based software follows from the simplicity of SE formulation discussed in the preceding sections.

It has been shown in the previous section that progressive collapse of a prismatic member requires careful design of the cross-sectional geometry and introduction of triggering dents that override the initial imperfections ever present in any real-word structure and induce the desired, stable folding pattern. The first step in the cross-section design is the check for internal contact(s). This procedure is illustrated in Fig. 8 where the contact of two propagating plastic hinges is detected in the late

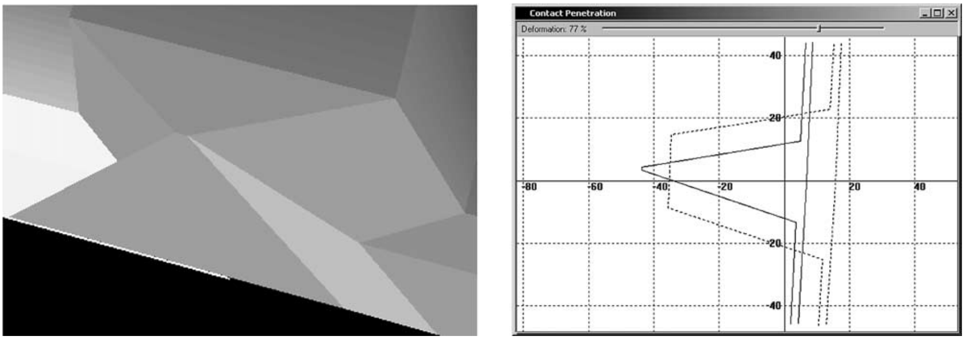

Fig. 8. Predicted collision of propagating plastic hinge lines in a trapezoidal cross-section of rocker panel. The contact occurs at 77% of axial deformation (right picture).

stage the folding process. The contact event reproduced numerically in Fig. 8 is analogous to the folding flaw illustrated in Fig. 4B.

Detection of design flows is relatively easy in the case of simple cross-sections. It becomes more difficult in the case of more complex designs. For example, an early design concept of a longitudinal member of a production car is illustrated in Fig. 9. This cross-section is capable of developing 8 different folding modes each composed of different combination of basic folding mechanisms of contributing SE's. In order to find out a mode suitable for the design of a triggering mechanism all the potential folding modes must be detected and analysed by the program. On the basis of these results various design changes of the cross-section are suggested to the designers. The detected design flows are marked in red on the right hand side of Fig. 9 while suggested design changes are listed in the lower part of the figure. The designer should use the program hints in order to modify the side widths and central angles of the cross-section in such a way that the desired asymmetric folding pattern of all contributing SE's is achieved. Once this is done the program automatically calculates design parameters of triggering dents and spot-weld geometry.

4.2. Crashworthiness of structural assemblies

Crash behaviour of individual prismatic elements must be harmonized with the dynamic response of more complex structural members, frequently referred to as structural assemblies. The main difficulty at this stage of the software development is a construction of simple yet accurate mechanical models of typical assemblies commonly encountered in engineering practice. In the CRASH CAD environment this goal is achieved by using the Superbeam Element concept [2]. The name Superbeam is given to any prismatic column discretized into Superfolding Elements. Speci-

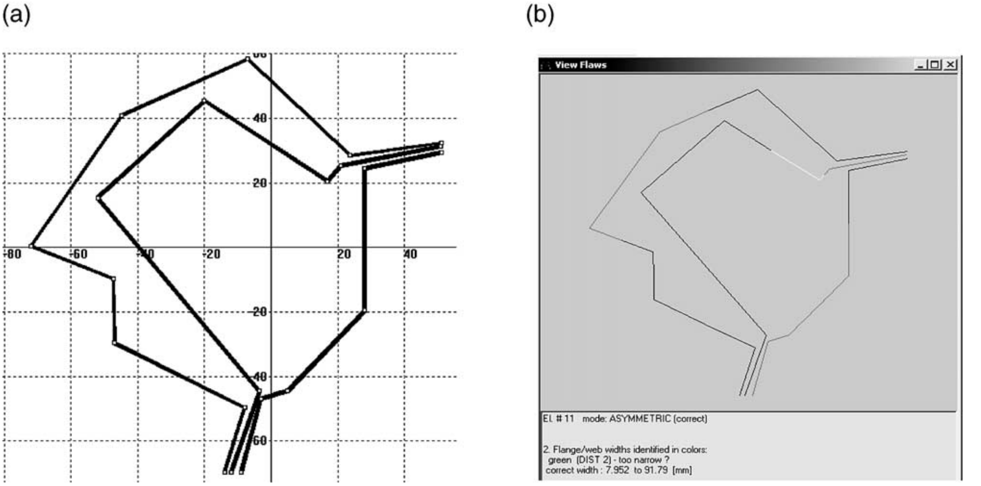

Fig. 9. Cross-section of an early design concept of a longitudinal member (A) and list of design flaws detected during numerical analysis of the folding process (B).

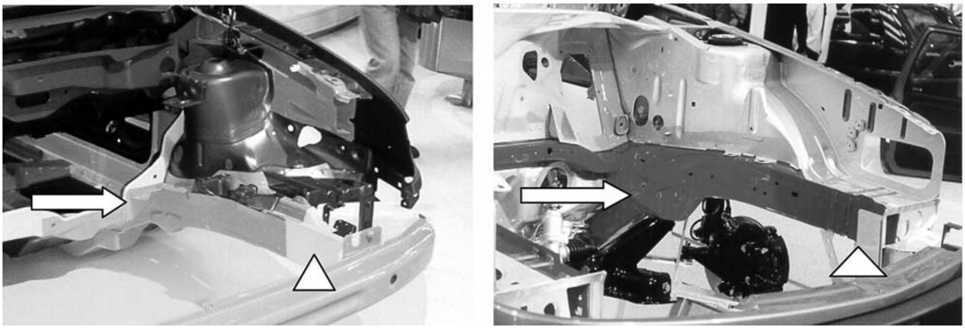

Fig. 10. Energy absorbing structures of the front end of passenger cars. Arrows indicate ‘S’ frame assemblies. Crash boxes are marked by triangles.

alized calculations routines, not discussed in this paper, are capable of predicting crush response of a Superbeam under any combination of cross-sectional forces. Consider as an example the important energy-absorbing component of a car’s body: the ‘S’ frame assembly illustrated in Fig. 10.

The planar ‘S’ frame computational model in Fig. 11 is composed of prismatic segments with an arbitrary cross-section. The entire ‘S’ frame is discretized into only four ‘Superbeam’ elements. Consequently, the input procedure requires specification of just overall dimensions of a frame, as illustrated in Fig. 11.

The ‘S’ frame module calculates the peak force, energy absorption, as well as the whole force-deflection characteristic for a given frame. An example of crash simulation is shown in Fig. 12 together with corresponding experimental data taken from the SAE paper by Ohkami [6].

Typically, ‘S’ frame assembly is not very effective as an impact energy absorber.

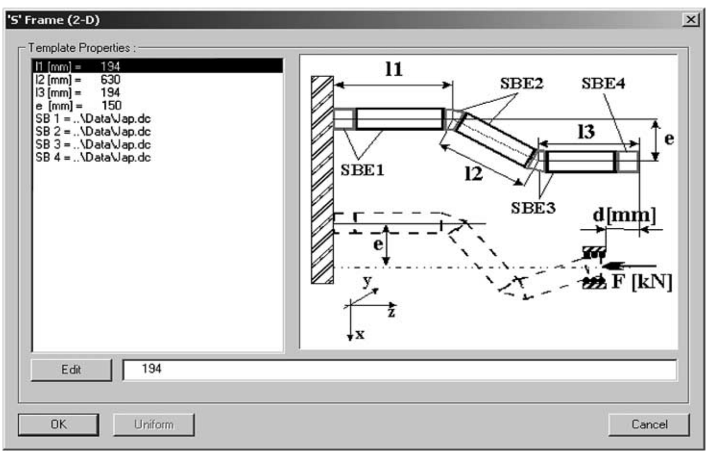

Fig. 11. The input template of ‘S’ frame model. The whole frame is discretized into four Superbeam Elements. The user is asked to specify the dimensions shown in the figure.

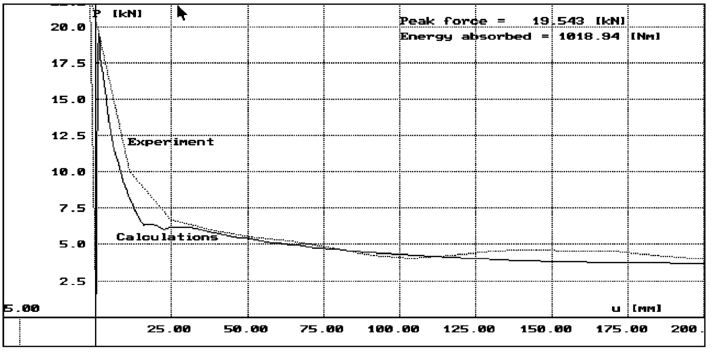

Fig. 12. Comparison of Crash Cad calculations with experimental results. The energy absorption is calculated for the crush of 200 [mm].

In addition crushing characteristic of ‘S’ frame is of softening-type as illustrated in Fig. 12. Therefore, in actual crashworthy design the impacted end of ‘S’ frame is designed as a crash box that must collapse progressively in order to effectively dissipate the impact energy, refer to Fig. 10. This is possible only if the ‘S’ frame offers enough support to the collapsing box so that the maximal loading capacity of the frame is not attained until folding of the crash box is completed. Consequently, the primary goal in the design of a crash box is to reduce the peak force associated with the crash response of the box itself to the level that is below the maximal load carrying capacity if the frame. This goal is achieved by means of the calculation/design procedures discussed in the preceding section.

5. Conclusions

The present paper provides for an overview of basic concepts behind the SE-based predictive routines and established design principles. The limited content of the paper does not allow for a detailed description of modern design/calculation environments used in the transportation industry. Problems such as torsion collapse, crush response of foam filled structures, stability of collapse and complete FE simulations are not covered in this paper. Interested readers are referred to a number of papers and books listed at the end of this paper for more information on these interesting subjects.

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