

Structural Optimization of Thin-Walled Tubular Structures for Progressive Buckling Using Compliant Mechanism Approach

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ABSTRACT

This investigation introduces a methodology to design dynamically crushed thin-walled tubular structures for crashworthiness applications. Due to their low cost, high-energy absorption efficiency, and capacity to withstand long strokes, thin-walled tubular structures are extensively used in the automotive industry. Tubular structures subjected to impact loading may undergo three modes of deformation: progressive crushing/buckling, dynamic plastic buckling, and global bending or Euler-type buckling. Of these, progressive buckling is the most desirable mode of collapse because it leads to a desirable deformation characteristic, low peak reaction force, and higher energy absorption efficiency. Progressive buckling is generally observed under pure axial loading; however, during an actual crash event, tubular structures are often subjected to oblique impact loads in which Euler-type buckling is the dominating mode of deformation. This undesired behavior severely reduces the energy absorption capability of the tubular structure. The design methodology presented in this paper relies on the ability of a compliant mechanism to transfer displacement and/or force from an input to desired output port locations. The suitable output port locations are utilized to enforce desired buckle zones, mitigating the natural Euler-type buckling effect. The problem addressed in this investigation is to find the thickness distribution of a thin-walled structure and the output port locations that maximizes the energy absorption while maintaining the peak reaction force at a prescribed limit. The underlying design for thickness distribution follows a uniform mutual potential energy density under a dynamic impact event. Nonlinear explicit finite element code LS-DYNA is used to simulate tubular structures under crash loading. Biologically inspired hybrid cellular automaton (HCA) method is used to drive the design process. Results are demonstrated on long straight and S-rail tubes subject to oblique loading, achieving progressive crushing in most cases.

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INTRODUCTION

Thin-walled tubular components are extensively used as structural members in the majority of transportation vehicles because of their low cost, good energy absorption capability and relatively low density. They have the ability to absorb the kinetic energy of the impacting body in the form of plastic

deformations, hence protecting the structure and passengers involved. These structures can be used in various loading conditions such as axial crushing, bending, oblique impact, transverse loading, among others. Tubular structures show significant energy absorption for long strokes in an axial crushing mode and hence, present an attractive option in crashworthiness designs.

A great deal of research has been done to study the axial crushing of thin-walled tubes since the pioneering work of Pugsley [1] and Alexander [2] in the 1960s. Later work from Abramowicz and Jones on both the static and dynamic responses of tubes focused on theoretical and experimental studies [3,4,5,6]. The concept of a superfolding element was introduced by Wierzbicki and Abramowicz [7] to better understand the mechanics of the crushing of thin-walled structures. With the advancement in computing power and the numerical implementation of finite element methods in the last two decades, considerable work has been done in creating equivalent numerical models for tube crushing and their validation with experimental tests [8,9,10,11].

Various experimental and numerical studies revealed three dominant modes of deformation during the axial crushing of thin-walled structures: progressive buckling, global or Euler-type buckling, and dynamic plastic buckling [12]. Of these three, the progressive buckling mode is desired for crashworthy designs because of its efficient energy absorption and better force-displacement behavior. In progressive buckling, crushing starting at one end (often the end close to the impact) and progressing systematically toward the other end of the structure is preferred as it utilizes the maximum possible material for plastic deformation without jamming. Collapse initiators, also known as triggers, stress concentrators, or imperfections, can be used to 1) initiate a specific axial collapse mode, 2) stabilize the collapse process, and 3) reduce the peak force during the axial crush. Many researchers came up with ideas for introducing buckle initiators or surface patterns to enhance energy absorption and buckling behavior [13]. Crush zones in the structure have been used to enforce progressive buckling [14]. Chamfering and other triggering mechanisms have been investigated on quasi-static axial crush of square aluminum tubes [15]. The effect of triggering dents on the energy absorption capacity of axially compressed aluminum tubes has been also investigated [16]. An overview of techniques of geometric and material modifications to improve the buckling behavior and energy absorption characteristics of thin-walled tubes under axial crushing can be found in a review article by Yuen and Nurick [17].

During an actual crash event, the tubular structure will seldom be subjected to pure axial loads. For example, thin-walled structures are subjected to both axial forces and bending moments in an oblique crash. A phenomenon of onset of global bending or Euler-type buckling was observed in thin-walled tubes under oblique impacts if the load angle was higher than a critical value [18, 19]. This onset of global bending severely reduces the energy absorption capability of tube structures. In the methodology presented here, the compliant mechanism design approach is used for thickness-based (topometry) design of thin-walled square tubes under axial compression [20]. The ability of compliant mechanisms to transfer motion and forces from an input load location to the desired points in the structure is utilized to achieve the desired buckle zones in the axial member. By suitably

defining the output port locations and desired displacement directions, progressive buckling can be initiated at the desired locations. Moreover, using this method, thin-walled structures can be designed to show progressive buckling even in cases of oblique impact at angles higher than the critical value at which bending collapse dominates the axial collapse and leads to poor energy absorption. To perform thickness-based topometry design of mechanisms, the HCA-based method for compliant mechanism synthesis [21] is modified by using thickness as the design variable. Base line assessment is performed to set benchmark for critical oblique angle and the optimum force-displacement curve. The benchmark is used for comparison with the results obtained from the proposed design methodology.

PROBLEM STATEMENT

During pure axial crushing of thin-walled square tubes, the maximum peak force F_{max} appears when the buckling starts in the structure for the first time, and after that, the force oscillates between local peaks and minimum loads as shown in Figure 1. Each pair of peaks is associated with the development of a wrinkle or buckle. Usually, these wrinkles or buckles develop sequentially from one end of a tube so that the phenomenon is known as progressive buckling. Designers often ignore the oscillations in the force-displacement behavior and use a mean value F_{mean} as an indicator of the energy absorption capability.

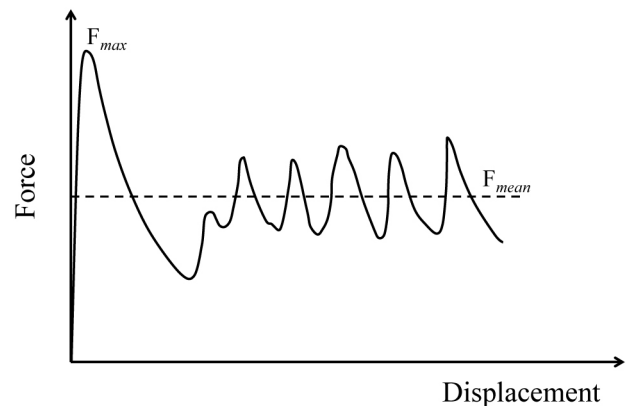


Figure 1. Typical force-displacement behavior for an axially crushed thin-walled square tube.

The problem of the present work is to design the thin-walled structure in order to invoke a symmetric mode of collapse by defining output ports on the two parallel faces pointing inward. During axial crushing of thin-walled square tubes, it is often desired that the folding or collapse start at the front end (end closer to the impact) and progresses systematically toward the other end of the structure to utilize the maximum possible material for plastic deformation without jamming. Also, the progressive buckling from the front end to the rear end helps in protecting important components close to or behind these energy absorbing

structures. For example, the damage for low intensity (~ 7 -10 mph) frontal impacts in automotive vehicles can be restricted to the bumper-beam and crash-box (hence saving the front frame-rail behind them) if the crash-box collapses progressively without jamming. Thin-walled square tubes, however, do not buckle progressively from the front end to the rear end in all cases. The buckling behavior is dependent on many factors such as loading conditions, geometry, imperfections and asymmetries in the structure, but most importantly, by the direction of the load. While axial load may trigger progressive buckling, oblique impact may cause Euler-type buckling.

The objective of this work is to find the thickness distribution of a thin-walled structure such that progressive buckling is achieved even when the structure is subjected to oblique impact. This design problem is approached following principles of design of compliant mechanisms. Given the loading conditions, the problem formulation is as follows

find	thickness distribution
maximize	mutual potential energy
subject to	mass constraint
	thickness bounds
	constitute equations

(1)

Compliant mechanism design using topology optimization methods is currently an established approach [22,23,24,25,26,27,28]. The use of compliant mechanisms design using topology optimization (material distribution) has been also explored in energy absorbing structural concept design [29, 30]. This work introduces the concept of topometry optimization of thin-walled structures following compliant mechanism design approaches in order to achieve progressive buckling. The design methodology adopted in this work is described in the following section.

DESIGN METHODOLOGY

The objective of a compliant mechanism is to efficiently transfer forces and motion from an input actuation location (referred to as input port I/P) to the desired output locations (referred to as output ports O/P) in a structure. Depending on the choice among force, motion or a combination of both to optimize, an appropriate objective function is defined. Alternatively, these objectives can be expressed in terms of geometric advantage GA and mechanical advantage MA defined

$$GA = \frac{d_{out}}{d_{in}} \text{ and } MA = \frac{F_{out}}{F_{in}} \quad (2)$$

where d_{in} and d_{out} represent the displacement at I/P and O/P, respectively, and F_{in} and F_{out} represent the force at I/P and the reaction force at O/P, respectively. In order to obtain such forces, linear springs of stiffness k_{in} and k_{out} are (virtually) attached to I/P and O/P, respectively. Figure 2 illustrates the

problem's layout. For a given F_{in} the output displacement d_{out} can be determined, from which $F_{out} = k_{out}d_{out}$.

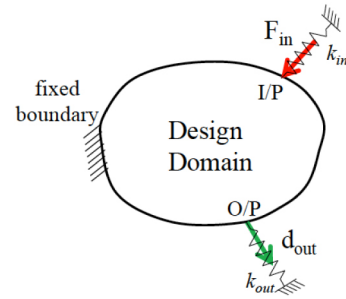


Figure 2. Structural optimization set-up in a compliant mechanism design problem using the “dummy load” method.

For a tubular structure, I/P is prescribed at the contact nodes with a rigid wall. The location of O/P corresponds to the desired buckling trigger location. This work proposes that such location may be prescribed according to the designer's criterion. However, it may be naturally assigned by the wavelength λ of the progressive buckling corresponding to an ideal axial crushing condition (Figure 3).

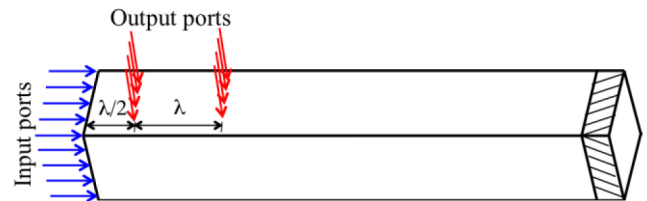


Figure 3. Location of input and output ports for a squared tubular structures following the wavelength λ corresponding to the progressive buckling after an ideal axial crushing condition.

For a given input displacement d_{in} , the objective is to maximize d_{out} , i.e., maximize MA . Assuming a fictitious “dummy” unit input force vector F_d that acts only O/P in the direction of the output displacement, d_{out} can be expressed as the scalar product

$$d_{out} = [0 \cdots 0 \quad 1 \quad 0 \cdots 0] \begin{bmatrix} \vdots \\ d_{out} \\ \vdots \end{bmatrix} = F_d^T U_1 \quad (3)$$

where U_1 is the displacement vector corresponding to the input load. The product $F_d^T U_1$ is termed mutual potential energy MPE or the complementary virtual work [31]. For a discrete finite element model, MPE can be expressed as the sum of element quantities. This is

$$MPE = \mathbf{F}_d^T \mathbf{U}_1 = \sum_{i=1}^n \sigma_{d_i}^T \epsilon_{1_i} \quad (4)$$

where σ_{d_i} and ϵ_{1_i} are the element stress and strain vector corresponding to the dummy and the input loads, respectively.

The mass M of the structure can be also expressed in terms of element quantities as

$$M = \sum_{i=1}^n \rho A_i t_i \quad (5)$$

where ρ is the material density and A_i is the area. Nonlinear finite element analysis is required to account for large displacements, contact, and plastic hardening. Given the crushing loading conditions and the location of I/P and O/P, the design problem statement (1) can be more clearly expressed as

$$\begin{aligned} &\text{find} && \mathbf{t} \\ &\text{maximize} && MPE = \sum_{i=1}^n \sigma_{d_i}^T \epsilon_{1_i} \\ &\text{subject to} && M = \sum_{i=1}^n \rho A_i t_i \leq M_{max} \\ &&& t_{min} \leq t_i \leq t_{max} \\ &&& \mathbf{R} = \mathbf{0} \end{aligned} \quad (6)$$

where $\mathbf{R}$ is the residual in obtaining the structural equilibrium. The input vector load is obtained from the loading condition is used by the optimizer to find the optimum thickness distribution. This corresponds to dynamic event with a given velocity. Sensitivity analysis of geometric nonlinear shell structures under dynamic loading is presented by [32]. Analytical sensitivities for materials nonlinear structures have been considered by [33,32,33,34,35]. Sensitivity analysis for both geometrically and materially nonlinear behavior is reported by [25, 36, 37]. A discussion on geometrically nonlinear material and applications to non-compliant structures and compliant mechanisms is presented by [24, 38]. However, the analytical treatment or numerical approximation of problems involving contact, geometric and material nonlinearities, and dynamic loading conditions, still remains unsolved or intractable. In order to find a solution to (6), this work incorporates the principles of uniform mutual potential energy distribution using hybrid cellular automata (HCA) [39,40,41,42,43].

The hybrid cellular automaton (HCA) algorithm is a structural design methodology inspired by the biological process of bone adaptation [44, 45]. This methodology assumes that cellular automata (CAs) form a structure or design domain, and sensors and actuators within the CAs activate local formation and resorption of material. With a

proper control strategy, this process drives the overall structure to an optimal topology by updating the thickness distribution. Using distributed controlled rules, the problem (6) can be stated as

$$\begin{aligned} &\text{find} && t_i, \mu \\ &\text{minimize} && |\overline{MPE}_i(\mathbf{t}) - MPE^*(\mathbf{t}, \mu)| \\ &\text{subject to} && M = \sum_{i=1}^n \rho A_i t_i \leq M_{max} \\ &&& t_{min} \leq t_i \leq t_{max} \\ &&& \mathbf{R} = \mathbf{0} \end{aligned} \quad (7)$$

where μ is the Lagrange multiplier associated with the mass constraint. In (7) MPE^* is the target value to be achieved by every element in the structure. This target is a function of the thickness distribution $\mathbf{t}$ and μ . In order to preserve a consistent topology design that is independent of the number of cells, an effective element mutual potential energy $\overline{MPE}_i$ defined as a function of the cell's neighborhood size,

$$\overline{MPE}_i = \frac{\sum_{j \in N_i} MPE_j}{|N_i|} \quad (8)$$

where N_i is the neighborhood of the i -th cell defined by

$$N_i = \{j: d(i, j) \leq r\} \quad (9)$$

and $d(i, j)$ denotes the distance between the i -th and the j -th cells. The neighborhood layout depends on the distance r referred to as the range of the neighborhood. Figure 4 shows the CA neighborhood layouts for different ranges.

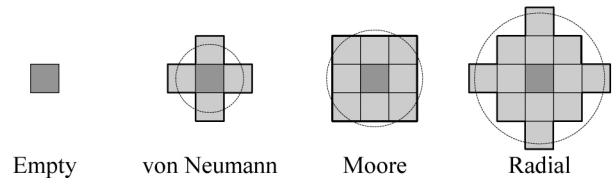


Figure 4. CA neighborhood layouts for different ranges r .

The iterative approach may be achieved using a control-based algorithm [39, 46] or a ratio approach [47]. This work makes use of the latter one.

NUMERICAL EXPERIMENTS

Benchmark Structures

During the practical crash scenario the impact may be at any angle. While designing a structure to sustain such kind of impact, there are geometric constraints that need to be considered. Due to the chassis geometry, tubular structures depict features that trigger undesired buckling modes e.g.,