

# ENERGY ABSORBED BY COMPOSITE CONICAL STRUCTURES IN AXIAL CRUSHING

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## ABSTRACT

Recently crashworthiness trend is the use of thin-walled composite impact attenuators in specific vehicle zones, ensuring kinetic energy absorption. The present paper addresses the composite conical structures design using an analytical approach for predicting energy dissipation in a controlled way. A balance of internal energies involved in the absorption and external force yields the average crush load and the total displacement through a numerical method. Comparison between theory and experiments shows the efficiency of the proposed relatively simple model for predicting energy absorption of axially collapsing composite shells.

**Keywords:** Theoretical Analysis, Crashworthy Problems, Energetic Approach, Composite Materials.

## 1. INTRODUCTION

The survivability of driver and passengers in an accident is achieved by the use of thin-walled impact attenuators placed in specific vehicle zones, able to absorb as much as possible the kinetic energy through a progressive deformation. The crashworthiness design requires to know the crushing force and the final deformation of such structures when subjected to axial loading in order to pass specific homologation tests.

From the numerous experimental studies, it is generally accepted that thin-walled tubes offer the most weight efficient solution for crashworthy aspects. The collapse behaviour of cylindrical and conical shaped shells of round cross sections has received attention in the recent years, because of their possible application to the design for crashworthiness of automotive vehicles. Moreover composite materials are now used in automobiles because they promise to be far more efficient than conventional materials [1]. A representative pioneering work to study the crush behaviour of composite tubes submitted to axial load has been done by Thornton *et al.* [2]. They studied the behaviour of various composite tubes, taking into account different fiber types, lay-ups and thickness to diameter ratio. Further their experimental results showed that rectangular and square sections are less effective in energy absorption than circular ones. Hull *et al.* [3] discussed failure mechanisms for composite tubes in detail and commented upon the influence of geometry and material compo-

sition on structural performance. Farley and Jones [4] studied also the effect of crushing velocity on the energy absorbing characteristic of composite tubes with different lay-ups. In the Mamalis *et al.* book [5] it is summarized many of their results done on composite tubes with different sections under different loading conditions.

Based on the experimental observations by Mamalis *et al.* [5] thin-walled structures under axial loading can deform in four different modes: deformation confined at impact wall (Mode I), longitudinal crack progression (Mode II), centrally confined circumference crack (Mode III), large hinge progressive folding (Mode IV). The challenge of design is to arrange the column of material such that the destructive zone can progress in a stable manner (Mode I) due to the large amount of crush energy absorption. The mean load of crushing becomes an important factor to estimate while choosing a material and geometry for an impact energy absorbing application. Recently many experimental studies have been combined with numerical analysis to predict the final deformation of such structure for various applications: progressive damage in braided composite tubes [6], structural components of a Formula 1 racing car [7], crash-boxes for automotive application based on advanced thermoplastic composite [8], BAR Honda rear impact structure [9], certification of the composite rimp energy absorber for a star Mazda series [10], composite helicopter cruciform elements [11], composite frontal crash-box for a Formula SAE car

[12]. Instead very few authors analysed the collapse mechanism of composite shells from the theoretical point of view [5, 13, 14, 15, 16], due to the difficulty to model analytically the brittle behaviour and heterogeneity of these composite structures. In particular, Mamalis *et al.* model the crumpling and bending process of thin-walled components of fibre-glass materials with different shapes, taking into account the energies involved in total axial crushing. Velmurugan *et al.* instead adopt a simpler analytical approach considering only the first load cycle. Each analysis simplifies the energy formulation using some experimental evidences.

The present study addresses an analytical and numerical investigation on the failure mechanism, pertaining to the stable mode of collapse (Mode I), of thin-walled composite conical tubes subjected to axial loading in order to predict the mean loads and total displacements during collapse. The analysis is based on previous research [5, 13, 14, 15, 16, 17] with the attempt to eliminate some simplifications dictated by experimental evidence and improve the modeling from the mathematical point of view. The theoretical modeling identifies the main energy contributions responsible for internal absorbing and balancing them to the work done by the external load. During the crushing, after the initial peak, the load oscillates around a mean load. This is due to the arise of a main circumferential intrawall crack, the splaying of the material strips, the formation of two lamina bundles bending inwards and outwards and the generation of a triangular debris wedge of pulverized material. Comparison between theory and experiments concerning loads and crushing is good, indicating therefore that the proposed theoretical model is an efficient strategy for predicting the energy-absorbing capability of the axially collapsing composite shells, despite the complexity of the phenomenon.

## 2. THEORETICAL MODELLING

During the crushing of a composite conical structure under axial impact, after the initial peak, the load oscillates around a mean load  $P$ . The first sharp drop in the load is due to the formation of a main circumferential intrawall crack of height  $h$  at the top end parallel to the axis of the shell wall. As the deformation proceeds further, the externally formed fronds curl downwards with the simultaneous development, along the circumference of the shell, of a number of axial splits followed by splaying of the material strips. The post crushing regime is characterized by the formation of two lamina bundles bent inwards

and outwards due to the flexural damage; they withstand the applied load and buckle when the load or the length of the lamina bundle reaches a critical value. At this stage, a triangular debris wedge of pulverized material starts to form; its formation may be attributed to the friction between the bent bundles and the platen of the hammer mass. The theoretical model takes into account only the first cycle of progressive crushing (Mode I), because the total deformation is characterized by successive cycles that are repeated with a similar trend. This is an acceptable hypothesis for circular frustum with angles less or equal to  $10^\circ$ , where the typical load-deflection curves for Mode I is characterized by a progressive fluctuation around a mean value. Fig. 1 shows this behaviour for three different angles, namely  $0^\circ$ ,  $5^\circ$  and  $10^\circ$ .

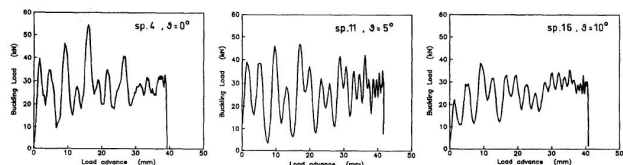


Fig. 1: Axial load-deflection curves for specimens with different angles

The idealized model of the crush zone is shown in Fig. 2, where  $R$  is the mean radius,  $H$  is the height and  $T$  is the thickness of the shell.

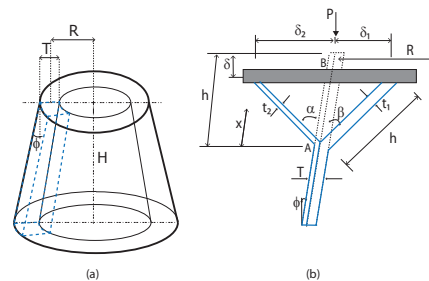


Fig. 2: Conical shell model: (a) shape before loading, (b) idealized crush zone configuration

In order to simplify the deformation mechanism, the following assumptions have been adopted: the internal and external fronds maintain a constant length equal to  $h$ ; the transition between straighten and bended zone is sudden, so the central crack is placed in  $A$  as shown in Fig. 2; the elastic energy associated with the first impact phase was not considered, because very low respect to the other contributions.

It has been observed in earlier studies [16] that energy is absorbed in four principal modes during the formation of crush zone in progressive crushing of cylindrical tubes: work required for bending of petals ( $W_b$ ), work required for petal formation ( $W_h$ ), work required for circumferential delamination ( $W_c$ )

and energy dissipated due to friction between the debris wedge and fronds and between fronds and platen ( $W_f$ ). Follow in detail the expressions used for the various energy contributions.

### 2.1 Bending energy

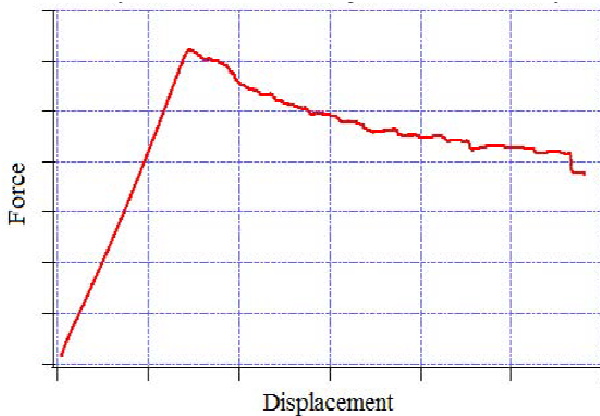
As the crushing process initiates, fibres bend both inside and outside the shell radius. Let  $t_1$  and  $t_2$  be the thickness of the fibre layers bending inside and outside the shell radius and  $\alpha, \beta$  the bending angles. By construction  $t_2$  is equal to  $T - t_1$  and the angle  $\beta$  is  $\alpha - 2\phi$ . Assuming that the fibre layers are perfectly plastic during bending, the work required to bend the fibre inside and outside the shell radius can be expressed as

$$\begin{aligned} W_b = W_{b_1} + W_{b_2} &= 2\pi d_c (M_1 \alpha + M_2 \beta) = \frac{\pi}{3} \sigma_0 d_c (\alpha (T - t_1)^2 + \beta t_1^2) = \\ &= \frac{\pi}{3} \sigma_0 d_c (\alpha (t_1^2 + (T - t_1)^2) - 2\phi t_1^2) \end{aligned} \quad (1)$$

where

$$d_c = R - T/2 + t_1 + h \sin \phi \quad (2)$$

is the frustum radius at the crack tip,  $M_1$  and  $M_2$  are the bending moment of the internal and external laminate respectively. As remarked by [18], the collapse moment per unit circumferential length for a conventional material verifies  $M_i = \sigma_0 t_i^2 / 6$  for  $i=1,2$  for  $i=1,2$  and  $\sigma_0$  the ultimate stress in uni-axial tension of the laminate. It is assumed that the bending moment is constant, after the first elastic phase, as the relative bending angle increases. This hypothesis overestimates the energy generated by fibres bending, because during deformation there are interply and intraply fracture propagation which reduce the ability of the folding to support the load. As displayed in Fig. 3, the bending force for a composite laminate is characterized by a first linear elastic region to which a slowly decreasing trend follows.



**Fig. 3:** Bending force of laminate as function of the relative displacement

### 2.2 Hoop energy

Assume that the hoop strain varies linearly between  $A$  and  $B$  in Fig. 2 that is it is null in  $A$  and maximum in  $B$ , and  $x$  indicates the distance from  $A$  to  $B$ . The expression for hoop strain energy in a single crush is

$$\begin{aligned} W_h = W_{h_1} + W_{h_2} &= \int_0^h \sigma_0 |\varepsilon_1| dV_1 + \int_0^h \sigma_0 |\varepsilon_2| dV_2 = \\ &= \sigma_0 \pi h^2 [(T - t_1)(\sin(\alpha - \phi) + \sin \phi) + t_1 |-\sin(\beta + \phi) + \sin \phi|] = \\ &= \sigma_0 \pi h^2 [T(\sin(\alpha - \phi) + \sin \phi) - 2t_1 \sin \phi] \end{aligned} \quad (3)$$

where  $\varepsilon_1$  and  $\varepsilon_2$  are the hoop strain corresponding to the layers bending inside and outside the shell radius, and  $dV_1, dV_2$  are the differential volume for the inside and outside layers. In details,

with  $d_c$  the frustum radius at the crack tip (2).

### 2.3 Crack energy

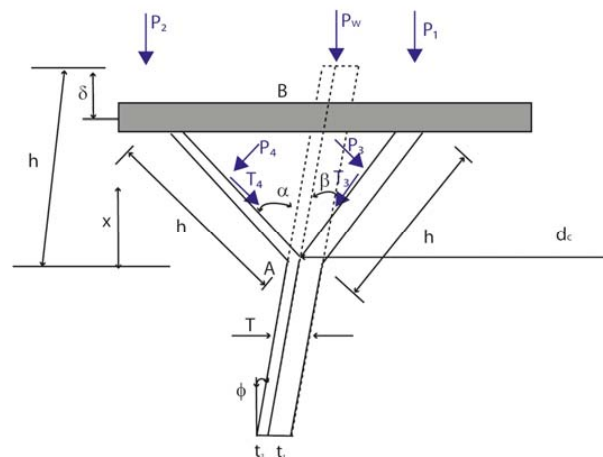
The energy required for circumferentially delamination in a single stroke is

$$W_c = 2\pi h G d_c \quad (4)$$

where  $G$  is the critical strain energy release rate per unit interlaminar delaminated crack area and  $d_c$  verifies (2).  $G$  is determined experimentally through DCB test as per ASTM D5528-01 [19].

### 2.4 Friction energy

After the formation of the internal and external fronds, normal stresses develop on the sides of the debris wedge followed by shear stresses along the same sides due to friction at the interface between the wedge and the fronds. Moreover additional normal and shear stresses develop at the interface between the steel platen and the deforming shell as the formed fronds slide along the interface.



**Fig. 4:** Forces model

The energy dissipated in frictional resistance for a crush distance  $\delta$  is

$$W_f = 2\pi\delta(\mu_1(P_1 + P_2) + \mu_2(P_3 + P_4)) \quad (5)$$

where  $\mu_1$  is the coefficient of friction between frond and platen,  $\mu_2$  the coefficient of friction between the wedge and the fronds and  $d_c$  is the frustum radius at the crack tip (2).  $P_1$  and  $P_2$  are the normal force per unit length applied by the platen to the fronds.  $P_3$  and  $P_4$  are the normal force per unit length applied to the sides of the wedge, as shown in Fig. 4. Due to feasibility  $\mu_1, \mu_2 \in ]0, 1[$  and by construction the deformation is given by

$$\delta = h(\cos\Phi - \cos(\alpha - \Phi)). \quad (6)$$

Static equilibrium at the interface yields

$$P = 2\pi d_c(P_w + P_1 + P_2), \quad (7)$$

where  $d_c$  is the distance between the center of the frustum radius and the crack tip (2) and  $P_w$  is the normal force per unit length applied by the platen to the debris wedge given by

$$P_w = P_3 \sin(\beta + \Phi) + T_3 \cos(\beta + \Phi) + P_4 \sin(\alpha - \Phi) + T_4 \cos(\alpha - \Phi).$$

$T_3$  and  $T_4$  are the frictional forces per unit length developed between wedge and fronds. Assuming that Coulomb friction prevails between the debris wedge and fronds,  $T_i = \mu_2 P_i$  for  $i=3, 4$ . Note that

$$P_i = \sigma_0 h \text{ for } i=3, 4, \quad (8)$$

therefore

$$P_w = 2\sigma_0 h(\sin(\alpha - \Phi) + \mu_2 \cos(\alpha - \Phi)) \quad (9)$$

By (7), (8) and (9)

$$P_1 + P_2 = \frac{P}{2\pi d_c} - 2\sigma_0 h(\sin(\alpha - \Phi) + \mu_2 \cos(\alpha - \Phi)),$$

and due to (6), the friction energy (5) is given by

$$\begin{aligned} W_f &= \mu_1 \delta P + 4\pi\sigma_0 h \delta d_c (\mu_2 - \mu_1 \sin(\alpha - \Phi) - \mu_1 \mu_2 \cos(\alpha - \Phi)) = \\ &= \mu_1 h (\cos\Phi - \cos(\alpha - \Phi)) P + 4\pi\sigma_0 d_c h^2 (\cos\Phi - \cos(\alpha - \Phi)) \cdot \\ &\quad \cdot (\mu_2 - \mu_1 \sin(\alpha - \Phi) - \mu_1 \mu_2 \cos(\alpha - \Phi)). \end{aligned} \quad (10)$$

### 2.5 Mean load P

The total energy dissipated for the deformation of the shell is given by the sum of the bending energy (1), the hoop energy (3), the energy required for circumferentially delamination (4) and friction energy

(10), i.e.

$$W_f := W_b + W_h + W_c + W_f \quad (11)$$

and it is equal to the work done by the external load  $P$  on the crushing displacement  $\delta$  in a single progression, that is

$$W_e := P\delta = Ph(\cos\Phi - \cos(\alpha - \Phi)) \quad (12)$$

Therefore the mean load  $P$  is a function of three variables  $h, t, \alpha$  and depends on three geometric parameters  $\phi, R, T$ , i.e.

$$\begin{aligned} P &= P(h, t, \alpha, \phi, R, T) = \\ &= \frac{\pi}{(1 - \mu_1)h(\cos\phi - \cos(\alpha - \phi))} \left[ \frac{1}{3} \sigma_0 d_c (\alpha(t_1^2 + (T - t_1)^2) - 2\phi t_1^2) + \right. \\ &\quad + 2hGd_c + \sigma_0 h^2 (T(\sin(\alpha - \phi) + \sin\phi) - 2t_1 \sin\phi) + \\ &\quad \left. + 4\sigma_0 h^2 (\cos\phi - \cos(\alpha - \phi)) d_c (\mu_2 - \mu_1 \sin(\alpha - \phi) - \mu_1 \mu_2 \cos(\alpha - \phi)) \right], \end{aligned} \quad (13)$$

with  $d_c$  the frustum radius at the crack tip (2).

Note that the domain of the function  $P$  is given by

$$D = ]0, H] \times ]0, T] \times ]2\Phi, \pi/2 - \Phi], \quad (14)$$

indeed for feasibility the crush length is strictly positive, the thickness of the plies bending outside belong to the interval  $[0, T]$  and the external bending angle  $\alpha$  is larger than  $2\Phi$ , because  $\beta = \alpha - 2\Phi \geq 0$  and the denominator of (13) has to be strictly positive.

## 3 RESULTS AND DISCUSSION

The present approach aims at determining the critical values of the length  $h$ , the thickness  $t$ , and the opening angle  $\alpha$  belonging to the domain  $D$  (14) in which the mean load (13) is minimum. The nonlinear function (13) is not easy to analytically minimize because the mean load gradient is difficult to nullify, so a numerical optimization method is necessary.

In view of generalizing the model discussed in Section 2, considering more than just three independent variables, a numerical algorithm named L-BFGS-B, which stands for Limited memory Broyden-Fletcher-Goldfarb-Shanno method for Bound constrained optimization [20, 21], has been implemented. The purpose of such a method is to minimize a nonlinear function of many variables subject to simple bounds, that is each variable should belong to a specific closed interval. As for every constrained nonlinear optimization strategy, at each iteration a search direction is defined to be the vector leading from the



current iterate to a smaller value belonging to the given set, until an optimization condition is reached. In particular the L-BFGS-B method approximates the Hessian with a limited memory (BFGS) matrix which defines a quadratic model of the given function. Therefore second derivatives of the objective function are not required, and only the knowledge of the gradient is necessary.

In order to evaluate the effectiveness of the idealized model, implemented with this numerical approach, specific cases were analysed. Not having available experimental data, results obtained by Mamalis *et al.* [22, 23] were used. Tables 1 and 2 report the mean crushing loads, the crush length and the errors in percentage between the described method and the experimental results for the geometrical cases taken into account. In particular Table 1 refers to static loading conditions, while Table 2 to dynamic loading ones. As reported in [23] the microfracturing mechanism for the progressive collapse of conical shell subjected to dynamic loading is, in general, similar to that obtained during the axial static collapse; therefore the above proposed theoretical model can be applied also for dynamic conditions modifying only the frictional parameters, such as  $\mu_1$  and  $\mu_2$ . From the tables it is clear as, despite the simplifications adopted, the proposed analysis is able to predict within  $\pm 20\%$  the mean load, which is absorbed for about 55% from frictional effects, for about 37% from fronds bending, for about 6% from hoop strain and for only 2% from crack propagation. Also according to Mamalis *et al.* the distribution of the main energy sources can be estimated with the same order. As mentioned before, the model refers to the first cycle of deformation; therefore the crush length  $s$  can be obtained multiplying the minimum

displacement  $\delta$  for the ratio between the experimental energy to absorb and the minimum energy obtained from the model.

**Table 2:** Geometrical and crushing characteristics during dynamic loading

$R$ [mm]	$T$ [mm]	$\phi$ [°]	$P$ [kN]	Error %	$s$ [mm]	Error %
14.8	2.2	5	15.82	16	63.00	14
14.6	3.1	5	23.79	9	45.39	10
16.1	4.3	5	38.50	8	39.45	7
17.2	5.2	5	51.44	19	26.77	16
16.3	5.7	5	55.76	10	23.67	9
15.4	2.1	10	16.24	2	64.76	2
15.05	2.9	10	23.66	7	43.23	7
18.25	4.7	10	49.88	4	26.88	6
15.5	5.1	10	49.19	1	25.19	1
15.75	6.5	10	68.24	5	21.41	5

#### 4. CONCLUSIONS

A crashworthiness problem was investigated using an analytical approach. In particular the energy absorption of composite conical tubes subjected to axial loading was analysed, defining analytically the external load as a function of three variables and identifying the minimum through a numerical approach. Despite some simplification, the method adopted is able to predict the mean load and the total crushing of circular frustum made of composite, once known parameters of the used material. Only the first deformation cycle, as Velmurugan *et al.*, and a more realistic formulation for the friction energy, as Mamalis *et al.*, have been considered. Differently from other authors no experimental statements for the presented model are assumed, therefore the methodology can be used as the first approach to follow during the design of specific impact attenuators. Forthcoming analysis will eliminate the assumption of a constant crush length both internally and externally to the axial splits, considering also the curvature of the folding.

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**Table 1:** Geometrical and crushing characteristics during static loading

$R$ [mm]	$T$ [mm]	$\phi$ [°]	$P$ [kN]	Error %	$s$ [mm]	Error %
11.9	1.6	5	16.15	12	72.55	14
13.4	2.4	5	28.30	4	60.98	4
15.1	4.3	5	61.75	5	116.26	5
18.55	5.6	5	99.79	15	65.98	13
15.2	4.8	5	70.76	20	52.14	16
15.15	4.3	5	61.92	1	122.25	1
10.45	4.3	5	46.00	10	139.34	11
13.6	5.6	5	77.89	16	135.07	19
12.65	1.7	10	19.18	5	60.95	4
12.8	2.3	10	27.37	12	72.55	14
15.25	4.3	10	66.00	12	104.85	10
11.9	7.4	10	105.11	15	136.81	18
20.1	4.3	10	82.40	4	110.19	4
14.35	5.1	10	77.26	13	134.34	15
9.7	7.4	10	89.91	19	141.80	24
11.65	7.9	10	112.07	16	135.63	19

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