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Dynamic lateral crushing of empty and sandwich tubes

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Abstract

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against the experimental observations. Detailed deformation features and energy absorption characteristics during the crushing process were identified. It was found that increasing the compression velocity leads to an increase in the total internal plastic energy dissipation for both empty and sandwich tube. The propagation of plastic bending in the form of a dynamic rolling hinge with a radius is the main mechanism of energy dissipation, as opposed to the low velocity impact which involves stationary plastic deformation zones.

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1. Introduction

Sandwich structures have attracted much attention in automotive and aerospace fields due to their higher strength-to-weight ratio, better impact resistance and associated energy absorption capacity [1]. Recent research on sandwich structures subjected to impact or blast loading has enhanced their potential in a wider range of crashworthiness applications than ever [2-4]. As typical energy absorbers, thin-walled tubes are usually crushed plastically under several loading conditions, such as axial crushing, lateral crushing, axial inversion and folding [5, 6]. The load-deformation curves, especially under lateral compression, are desirable for energy absorption because of higher efficiency with plateau force over long stroke [6]. As a result, thin-walled tube systems are widely used as efficient energy absorbers in engineering fields [1]. On the other hand, cellular materials such as metallic foams have recently emerged as a novel material in energy-absorbing devices owing to their high specific stiffness and specific strength compared with traditional metals such as steels. Researchers have studied the axial crushing behaviour of foam-filled tubes [2, 7], which have full advantages of both the thin-walled tubes and metallic foams. It was found that by combining both the two structures, the energy absorption capacity is improved [8, 9], though not necessarily weight-effective. Meanwhile, work on the lateral crushing behaviour of aluminium foam-filled tubes [10] showed that the foam filling advantage still existed under the transverse loading condition, in terms of better specific energy absorption.
Thus, similar energy absorption enhancement might be expected for sandwich tubes under lateral crushing.

Investigations on the lateral crushing of empty tubes have been extensively conducted by many researchers [5, 6, 11-14]. All the investigations demonstrate that the crushing of those tubes involves plastic bending, which may be idealised as plastic hinges to model the lateral collapse of tubes. Because of the strain localization around the plastic hinges, it may not be structurally efficient to dissipate energy under this condition. Therefore, to further improve the energy dissipating performance, tubes with metal foam sandwiched might be an alternative structure. Previous research on the quasi-static responses has already identified this improvement in the crush strength and energy absorption capability [15]. The behaviour of sandwich tubes under dynamic lateral compression has been less reported in the literature. Moreover, metallic foams can be strain-rate sensitive [16, 17] and have a higher plateau stress and a higher energy dissipation capacity under impact or blast loadings, compared with the quasi-static loading case.

In this paper, the dynamic response of empty and sandwich tubes under lateral compression is investigated. The effect of velocity is examined experimentally and by using finite element analysis. A series of tests on empty tubes and sandwich tubes were performed by using an Instron VHS8800 High Rate Test System, which enables a single shot test to be conducted at a constant crosshead speed, up to 10 m/s for compression. The specimens were placed on the bottom platen and then they moved upwards together, until they collided with the top platen, with very little change in the speed during the whole crushing process. The dynamic force and deformation histories were recorded simultaneously by a high-speed data acquisition package. Finite element code ABAQUS/Explicit was then employed to analyse the response of the tubes under higher velocities, up to approximately 100 m/s. Based on the deformation modes from FE analysis, a critical compression velocity was defined above which the tube deformation is localised around the impact end. A comparative study of the load-deflection curves under quasi-
static and impact tests is described and the effect of velocity on the total energy absorption is studied.

2. Dynamic compression experiments

2.1. Material properties of specimens

Fig. 1 shows the specimens used in the tests. The sandwich tubes were manufactured by assembling together the individual components, i.e. the inner tube, the outer tube as well as the aluminium foam core. The aluminium tubes were made of AA6060-T5 and the length was fixed at 50 mm for each specimen. The stress-strain curves of the aluminium were obtained from the standard uniaxial tensile tests [18], from which the average yield stress $\sigma_y$ was 150 MPa. The density of tubes $\rho_s$ is 2760 kg/m$^3$. Cylindrical tubes of aluminium foam were cut from an initial ALPORAS® aluminium foam block (400 × 700 × 2400 mm), with the nominal relative density of 9%, supplied by Gleich Ltd., Germany. The material properties of the foam core are the same as those reported in [17]. For bonding, a two-component thixotropic epoxy liquid adhesive (FORTIS AD825) was used to paste, separately, the two monolithic aluminium tubes with the aluminium foam core. Four empty tubes and seven sandwich tubes were fabricated and tested separately in the INSTRON VHS8800 machine and their details are listed in Table 1. The definitions of geometric parameters in the tests are shown in Fig. 2.

2.2. Setup of dynamic experiments

All the lateral crushing tests were performed at a room temperature of 25 ºC. The dynamic tests were performed using a High Rate Test System (INSTRON VHS8800). A photograph of the experimental setup is shown in Fig. 3. The top platen was fixed and the finishing position of the bottom platen was controllable. This system provides a feedback mechanism by its FastTrack™ VHS8800 controller package to adjust the drive profile of the hydraulic system according to the experimental condition. After some initial trials with several iterations, an
almost constant loading rate could be achieved for the subsequent experiments. In our tests, for the bottom platen, a constant velocity of 1.0 m/s with a sampling rate of 50 kHz was used for low rate compressions and a constant velocity of 10.0 m/s (which was the maximum possible) with a sampling rate of 500 kHz for high velocity compression tests. The load history was measured by Kistler® load cell (Type 9071A) without data filter. The displacement history was measured by a linear variable differential transformer (LVDT) with data filter with a cut-off frequency of 1000Hz. During the initial tests, the empty tubular specimen was placed either on the bottom platen or top platen, in order to explore the difference of these two arrangements. Subsequently, for all the tests with empty and sandwich tubes the specimen was placed on the bottom platen and then moved upwards together with the base platen. Double-sided tapes were used to stick the specimen on the bottom platen during the movement. A high-speed camera was used to track the deformation profile of the specimen during crushing.

The tube’s contact forces with the top and the bottom platens are quite different in the presence of a tube’s inertia. To study the possible difference, two kinds of impact methods were explored for the empty tubes. Impact method I was to attach the specimen to the bottom platen and then move together at a velocity, as sketched in Fig. 2(a). The other, impact method II, was to fix the specimen with the top stationary platen, as depicted in Fig. 2(b). The load cell was mounted in the top platen and the displacement of the bottom platen was measured. The force-displacement curves of both the methods are plotted in Fig. 4, for specimens ET01 and ET02, respectively. The two methods result in different forces in the initial stage, but later the two forces were almost identical. For the second method, at the early crushing phase of small deflection, the magnitude of contact force was very small compared with that of the first one. This delay in experiencing the force by the load cell was due to the inertia of the tube. Also, a drop in the force after the initial small peak might indicate elastic bouncing of the tube from the top platen. As the displacement developed further, the tube wall started to interact with the top platen again, rendering a steeper increase of the force in the subsequent phase. This bouncing-
back was also detected from the high speed photography. For all the tests reported here, method I was used.

3. Experimental results

3.1. Empty tubes

Fig. 5 shows the load-deflection results of several empty tubes (ET01, ET03 and ET04) from the quasi-static and dynamic tests with a crushing velocity of 10 m/s. Apart from the initial difference at the early deformation phase, the magnitudes of the crushing strength are nearly the same under each loading case. From the deformation profile recorded by the camera, the deformation modes of the empty tubular specimens were almost the same when they were subjected to a constant velocity. The progressive collapse has three phases, i.e., initial tube wall collision, steady dynamic collapse and unloading phase. In the first phase, the contact force had large fluctuations probably due to the elastic effect during the collision of the tube wall. The fluctuations were damped by the plastic deformation of the tube, in a mode similar to that in quasi-static compression [15]. Following this initial phase, a slight increase in the compressive strength of the plastic collapse phase was found, which commenced with a series of very small peaks and troughs. To avoid possible damage to the apparatus, the crushing process was stopped before the self-contact of inner surface of the tube and this caused an unloading phase in the load-deflection curve.

3.2. Sandwich tubes

To improve the energy dissipation of sandwich tubes, it is important for the plastic deformation to occur extensively in the protective structure. From the analysis on the quasi-static lateral compressive responses of sandwich tubes, the crushing strength depends on the collapse patterns [15]. Similar to the quasi-static case, three types of collapse patterns were observed in dynamic crushing. There were simultaneous collapse pattern I, plastic collapse
pattern II with foam fracture and sequential collapse pattern III, respectively. Some photographs of the deforming specimens showing those collapse patterns, together with finite element results to be discussed next, are illustrated in Figs. 6 to 8.

Generally, for the sandwich tubular structure with collapse pattern I, both the inner and outer tubes deformed simultaneously in the plastic collapse. The foam core was less deformed, as illustrated in Fig. 6(a). In some cases, especially when thick foam was involved, ductile fracture of the foam core occurred during the crushing stage, resulting in collapse pattern II, as shown in Fig. 7(a). A distinct crack can be seen near the lower portion of foam core at the displacement of 20 mm. It is worth noting that a similar phenomenon was revealed in the previous experiments with foam-filled tube [10].

When the foam was even thicker, the inner and outer tubes deformed at different time, unlike the previous patterns. In the beginning the inner tube did not deform at all. Noticeable deformation in the inner tube took place only after a certain displacement, indicating a sequential collapse pattern III, as described in Fig. 8(a). This collapse pattern occurred with the specimens of the thickest foam, in both the quasi-static loading and dynamic impact (SWT02, SWT05 and SWT06, for instance). Because of the inherent large variation in the size, shape and distribution of foam cells, slightly non-symmetric deformation occurred in this collapse pattern.

3.3. Comparison between quasi-static and dynamic crushing of sandwich tubes

Figs. 6 to 8 show a comparison of force-displacement curves between the dynamic and quasi-static compression for each deformation pattern. For specimens such as SWT03 with the collapse pattern (I) shown in Fig. 6(a), the crush strength subjected to dynamic impact is lower than in quasi-static situation. A drop of dynamic collapse force was found near the displacement of 10 mm, which was due to the failure of the adhesive joint. The adhesive might be vulnerable to failure under dynamic situation because of the large intensive stress and heat. The force-displacement curve of dynamic case was very close to that of a sandwich tube without glue.
under quasi-static compression.

For specimens with collapse pattern II (SWT01, SWT04 and SWT07) as shown in Fig. 7(a), the collapse forces under dynamic loading \((v = 10 \text{ m/s})\) have the same magnitude as those fully bonded specimens under the quasi-static compression. However, the collapse force under dynamic compression at \(v = 1.0 \text{ m/s}\) is lower than that in the quasi-static loading. This phenomenon indicates that the failure of two adhered surfaces is sensitive to the impact at the beginning of the test, regardless of the compression velocity. The enhancement of crushing strength is more significant as the compression velocity increases.

For specimens with collapse pattern III as shown in Fig. 8(a), sandwich tubes exhibited a larger collapse strength under dynamic compression at \(v = 10 \text{ m/s}\) than that under quasi-static loading. When the normalized displacement becomes larger than 0.2 small amount of foam fracture occurred, leading to the reduction of curve slope. Similar trend was observed for the collapse force under compression velocity of 1.0 m/s. When the displacement was larger than the radius of the outer tube, the foam core became densified and the inner tube started to deform, resulting in a sharp increase of force.

4. Finite element analysis

Experimentally the maximum velocity of dynamic compression is limited to 10 m/s for this machine. In order to investigate the behaviour of tubes at higher velocity and to further understand the mechanics and mechanisms of plastic deformation, a finite element (FE) analysis was conducted. Fracture was not considered in the simulation.

4.1 Dynamic response of empty tubes

4.1.1 Deformation modes and critical velocity

The empty tube was modelled as a 4-node doubly curved shell element S4R, with reduced integration. The Simpson thickness integration rule with five integration points along the thickness direction was adopted. The actual stress-strain curve from tensile tests for the tube was...
used and no strain rate effect was considered, because aluminium is not so strain rate sensitive. The tube and the bottom platen were given an initial velocity, which for the platen was maintained constant throughout each test.

Fig. 9 shows deformed profiles of an empty tube (Specimen ET01: $D = 99.81$ mm, $t = 1.92$ mm) at three values of deflection (25, 50 and 75 mm), subjected to a velocity of 10 m/s and 50 m/s, respectively. When the compression velocity was 10 m/s (Fig. 9(a)), the deformation mode was very much the same as that of a typical quasi-static case. But for $v = 50$ m/s, plastic deformation initially took place around the impact region of the tube, with little bending deformation in the remaining part. As the tube was further compressed by the platen, large plastic deformation was extended to the whole tube.

Fig. 10(a) shows the energy dissipation against deflection for an empty tube (ET01) compressed at various velocities. It shows that when the compression velocity is less than a certain value, say 20 m/s, energy dissipated by the structure is not affected by velocity significantly. This is because the deformation mode remains the same as that for the quasi-static case, with most plastic deformation localised around the four “plastic hinges”. For higher velocity, considerable increase in energy absorption was observed. This could be attributed to the fact the plastic deformation started from the impact region and then gradually propagated to the whole tube, as explained in the next section.

To further quantify the possible existence of a critical velocity for the dynamic mode change as shown in Fig. 9, a tube was divided into four equal segments, one quadrant for the proximal (front) end, one for the distal (back) end, and one each for the two sides (Fig. 10(b)). The plastic energy for each quadrant was then evaluated from FEA and this broadly indicated the energy distribution within the tube. The effect of compression velocity on the energy distribution for tubes with different values of $D/t$ was then studied. The energy absorbed by the front quadrant and back quadrant, normalised by the total energy, is plotted in Fig. 10(b). It is striking that, for a given value of $D/t$, there exists a narrow band of velocity, below which the front and back
quadrants dissipate the same amount of energy. Above this velocity, the front dissipates relatively significant amount and the back dissipates less and less. This is due to the fact that at low velocity the deformation mode was very much the same as quasi-static ones, involving four stationary hinges. But for the high velocity impact, inertia plays a significant role and deformation started from the impact face and then propagated in a plastic “rolling hinge”, as explained later. This region for critical velocity seems to change with $D/t$.

For each value of $D/t$, the value of critical velocity was defined by locating the intersection of two straight lines, one for low velocity and the other for high velocity region. To study the effect of material and geometry on the critical velocity ($v_{cr}$), two dimensionless groups were chosen, $\rho_y v_{cr}^2 / \sigma_y$ and $t/D$, where $\sigma_y$ is the yield stress of the material. Their relationship is plotted in Fig. 11, on a double-logarithmic scale. From the best fitted straight line, we have

$$\frac{\rho_y v_{cr}^2}{\sigma_y} = 0.41 \left(\frac{t}{D}\right)^{0.94}$$

(1)

Since the exponent of the above equation is very close to 1, it is reasonable to approximate it to 1. Hence, an empirical equation from the plots is obtained,

$$\frac{\rho_y v_{cr}^2}{\sigma_y} = 0.3 \frac{t}{D}$$

(2)

The above relationship demonstrates that the dimensionless critical velocity squared is a constant for a given tube and it is linearly proportional to $t/D$. Hence, the critical velocity is rewritten as

$$v_{cr} = \sqrt{0.3 \cdot \frac{t}{D} \cdot \frac{\sigma_y}{\rho_y}}$$

(3)

4.1.2 Dynamic deformation mechanism- rolling hinges

Fig. 10(a) indicates that the energy absorbed in the dynamic case when velocity was 100 m/s can be as high as five times that of the static case. It is therefore natural to explore the
mechanisms of such high energy dissipation under high velocities. As the empty tube is relatively thin, two possible mechanisms exist: plastic bending and in-plane stretching/compression in the circumferential direction. At a given instant (and therefore displacement), the complete profile of the deforming tube was obtained from FE and from that the curvature around the circumference was computed, and so was the mid-plane circumferential strain. Figs. 12 and 13 plot the curvature distribution and in-plane strain distribution, respectively, at three values of displacements and for both $v = 10$ m/s and 50 m/s. Fig. 12(a) shows that for low velocity impact ($v = 10$ m/s), at the both proximal end ($s = S/(\pi D) = 0$) and distal end ($s = 0.5$), the curvature changes from the initial $20$ m$^{-1}$ to zero (becoming flat) and this flattening zone extended as deflection increased. At the mid-range of the tube ($s = 0.25$), the curvature increased monotonically, with the plastic deformation confined within the same size (or even getting smaller) of 4 to 6 thicknesses.

For the case of $v = 50$ m/s, Fig. 12(b) shows a complete different mechanism with two significant features. First, at the proximal end the curvature finally became to zero and this “flattened” zone was increasing with deflection, which appeared the same as that for the low velocity case. However, the material ahead of this zone had undergone a positive change of curvature first before it then changed back to zero curvature. Let us look at a small segment around position $s = 0.15$, for example. The initial curvature would be approximately $20$ m$^{-1}$, corresponding to a radius of 99.8 mm. At displacement of 25 mm, the segment was bent further to a curvature of $46$ m$^{-1}$. As displacement proceeded, the segment started unbending with a reducing curvature, until it reached almost zero at displacement of 75 mm. There was practically little deformation at the distal end ($s = 0.5$) and, for small displacement, even mid-segment ($s = 0.25$). The second feature is that, as displacement proceeded, the position of peak curvature was propagating and the peak value increased. The maximum curvature of $52$ m$^{-1}$ occurred at $s$ equal to 0.18 when $\delta = 25$ mm, but it was $72$ m$^{-1}$ at $s = 0.21$ when $\delta = 50$ mm. At this instant, the distal end started to flatten. When the displacement was 75 mm, the flattened
distal end had extended considerably and the peak curvature was about 100 m\(^{-1}\) occurring at \(s = 0.28\), which is five times the initial curvature.

For the possible in-plane deformation, Fig. 13(a) shows that there was tension around the contact regions, which propagated with increasing deflection, but compression around the mid-segment \((s = 0.25)\), for low velocity impact. The peak value was around \(9 \times 10^{-4}\) for compression and \(6 \times 10^{-4}\) for tension. For high velocity impact, there was a very small tension zone at proximal end, followed by a compression zone of a peak strain \(30 \times 10^{-4}\) and then another tension zone and compression zone. The distribution and peak values did not seem to change with the increasing displacement.

In order to assess the contribution of energy from bending and stretching/compression, the magnitudes of strain should be compared. Using Kirchhoff’s hypothesis, the mean strain corresponding to curvature \(\kappa\) would be \(\kappa t/4\). This gives a value of \(480 \times 10^{-4}\) for a peak curvature of 100 m\(^{-1}\). This is 16 times the peak compressive strain at the mid-surface and hence should be the dominant energy dissipation mechanism.

The above described curvature evolution may also be qualitatively observed from the profiles shown in Fig 9. In essence, the dynamic deformation mechanism at high velocity involves a travelling plastic hinge which invokes bending of a strip to a curvature and then unbending to another curvature. Here this is called dynamic rolling hinge in order to emphasize the important feature of unbending, as opposed to the conventional travelling hinge which could involve bending only (and unbending also, though). The bending energy associated with such a rolling energy can be worked out by considering an arc section with two plastic hinges \((B\ and\ C)\) in Fig. 14. The rolling hinge \(BC\) has a radius \(R_0\) and the strip before bending and after final unbending has a radius of \(R_1\) and \(R_2\), respectively. It is easy to understand the rolling process if one imagines that the material is flowing from \(AB\) to \(BC\), and at \(B\) the strip is bent to radius \(R_0\) (< \(R_1\)). When the strip moves to point \(C\), unbending occurs and the strip’s radius becomes \(R_2\) (> \(R_0\)). For a strip of unit length and unit width, the energy absorbed by bending the material at \(B\)
and unbending at $C$ are given, respectively, by,

$$W_B = m_p \cdot \left(\frac{1}{R_0} - \frac{1}{R_1}\right), \quad W_C = m_p \cdot \left(\frac{1}{R_0} - \frac{1}{R_2}\right)$$

where $m_p = \frac{\sigma_f t^2}{4}$ is the fully plastic bending moment per unit width. The total energy dissipated by plastic bending when the unit strip passes this rolling hinge is equal to the sum of the two. For a strip which is flat both before and after bending, this total energy is $2m_p/R_0$. In the present case, $1/R_2$ is approximately zero. This equation shows that the total plastic energy is almost inversely proportional to the radius of the rolling hinge, $R_0$. A small value of $R_0$ would need a large amount of energy for the hinge to roll forward. Indeed, an infinitely sharp crease (which has a zero radius) would not roll forward, as it would need an infinite amount of energy to do so. In the present case, low velocity impact involves stationary plastic deformation zones (or hinges). But the high velocity impact is associated with dynamic rolling hinges, which may not only have higher curvatures than their static counterparts, but also travel and spread over a larger area. It is highly plausible that the observed large difference in energy dissipation may be attributable to this fundamental difference in the deformation mechanisms between the static and dynamic cases. Nevertheless, further analytical models based on impact mechanics should be established to accurately explain the fundamental mechanics of the deformation modes observed.

4.2 Dynamic crushing of sandwich tubes

A crushable foam model was employed for the aluminium foam core in FE analysis for the lateral crushing of sandwich tubes by two rigid platens. The representative FE model in ABAQUS/Explicit is shown in Fig. 15. The element type of both inner and outer tubes was S4R, which was the same as in the previous analysis for empty tubes. The aluminium foam core was modelled as a three dimensional 8-node linear brick element C3D8R, with reduced integration.
Distortion control of foam elements was introduced since large plastic deformation was observed in the tests. The inner surface of outer tube and the outer surface of inner tube were tied to the foam core to model perfect glue without debonding. Surface to surface contacts were applied for the cases without glue and the friction coefficient was set at 0.1. Both platens were modelled as rigid bodies. The top platen, which was at the proximal surface, was fixed during the whole crushing stages. A constant velocity was applied on the bottom platen. To replicate the impact method adopted in previous experiments, an initial velocity was applied to the sandwich tube. The interactions among the rigid platen, outer tube and foam core were considered by adopting a penalty function, as provided in the software package.

Since the metallic foam core was involved in the dynamic analysis, it was necessary to validate the numerical model of foam first. Based on the uniaxial compressive tests and corresponding finite element analysis on the cylindrical foam extrusion, the crushable foam model [19] was confirmed to represent the macroscopic mechanical behaviour of metallic foam [17]. Further numerical simulations were made for the sandwich tubes. A good agreement between the prediction and the experimental result is shown in Fig. 16. Thus, the FE modelling could be used to study the dynamic behaviour of sandwich tubes. More detailed information such as partition of the energy absorption for each component is discussed in the later section and might provide potential support for the analytical models.

The broad agreement in Fig. 16(d) is also noted for the specimen with deformation pattern I. The discrepancy and sudden drop of the crush strength at a displacement of 24 mm were probably due to the failure of adhesive. This may be inferred also from the further comparison with the unglued case, where the load-displacement curve was much closer to that without glue. It implies that the adhesive was probably prone to fail at this point, which led to a reduced load.

4.2.1 Deformation mode of sandwich tubes

From the collapse modes observed in the experiments with sandwich tubes under low and
moderate velocity impact, symmetric behaviour about both vertical and horizontal planes was reported. The deformations at both the lower and upper portion of such structures were symmetric, as depicted in Figs. 6 to 8 for both quasi-static [15] and low velocity crushing. However, similar to the case of a single empty tube, when the compression velocity increased beyond a certain value, the deformation near the proximal and distal surfaces became non-symmetric with respect to the horizontal plane during the high speed crushing process. Fig. 17 shows typical deformation patterns of two different specimens subjected to high velocity impact up to 100 m/s. It can clearly be seen that the severe deformation of tubes was localized at the upper portion of structure, which was near the proximal surface. The lower portion of sandwich remained undeformed even when the stroke reached a value equal to the radius of the outer tube. Fig. 17(a) is for a sandwich tube (SWT03) with thin foam core, identified as collapse pattern I. Both the outer tube and foam core deformed from the beginning of the crushing. Visible deformation of the upper portion of inner tube occurred when the foam core became densified. Nevertheless, the lower portion of the sandwich tube remained undeformed. For a sandwich tube (SWT06) with thick foam core, sequential deformation is also shown in Fig. 17(b), as collapse pattern III. In both the cases when sandwich tubes subjected to high speed impact, crushing propagated from the contact surface into the undeformed layer, which was different from the observation in the tests with low velocity. The rolling hinge as a dynamic bending mechanism presented for empty tubes should be applicable here.

4.2.2 Collapse load of sandwich tubes under high velocity impact

To understand the effect of velocity on energy absorption, the sandwich tubes were dynamically compressed at different velocities, ranging from 10 m/s to 100 m/s. Fig. 18(a) shows the variation of collapse load in specimen (SWT03) for different compression velocities. It is evident that when the compression speed was lower than 20 m/s, the average collapse strength of the sandwich tube having thin foam was almost the same regardless the value of
velocity. Small discrepancy appeared only at the initial crushing process where extensive plastic deformation took place at the upper part of the tubes. For the sandwich tube with thick foam, noticeable difference in the load-deformation curves for each loading velocity existed, as shown in Fig. 18(b). Based on the area underneath each of these curves, the energy absorbed was larger for the specimens with thick foam and small inner tube, compared with those with thin foam.

4.3 Discussion on energy absorption of empty and sandwich tubes

From the finite element results on empty tube (ET01) and sandwich tube (SWT03) in Fig. 17 and Fig. 18(a), the ratio of energy dissipation under dynamic impact and that of the quasi-static counterpart is plotted in Fig. 19(a), corresponding to a displacement of 70 mm. It is interesting to note that the points seem to fall into a single curve, whether it is for the empty or sandwich tubes. When the velocity is less than 20 m/s, the ratio of dynamic energy to quasi-static energy is around 1.1. Afterwards, the energy absorption capability under the dynamic crushing increases sharply with the compression speed, and especially it is over five times when the compression velocity reaches 100 m/s. For the empty tubes studied, the dynamic energy enhancement over its static one is obtained, empirically, as follows

\[ E_{d}/E_s = 1 + 38.86v\sqrt{\sigma_y/\rho}^{2.5} \]  

which is also plotted in Fig. 19(a).

The specific energy absorption (SEA), which is the energy absorbed (EA) per unit mass of a crushed structure, is studied. The values of EA and SEA for specimens under dynamic experiments are also given in Table 1, for a displacement of 0.7D. It is shown that sandwich tubes with collapse pattern III is the most weight efficient, with a maximum SEA value of 3.96 J/g. For velocities up to 100 m/s, the values of SEA for typical specimens of comparable dimensions (ET01, SWT03 and SWT06) are plotted in Fig. 19(b). The trend is similar to that shown in Fig. 19(a), as expected. The dynamic effect would lead to an increase in the SEA for
all the specimens. It is also illustrated that for specimen SWT06 with the thickest foam and smallest inner tube, the value of SEA is almost four times that of the corresponding empty tube (ET01). This result demonstrates that sandwich tubes with a thicker foam core and thicker inner tube have better weight efficiency.

To assess the energy dissipation characteristic of sandwich tubes, the total energy absorbed by the sandwich tubes was partitioned for each component of the outer and inner tubes as well as the foam core. Fig. 20(a) shows the strain energy of each component in a sandwich tube (SWT06) with thick foam core subjected to a velocity of 10 m/s. When the dynamic displacement was less than 20 mm, the inner tube did not contribute to energy dissipation process until the crushing layer became severely deformed and stiffened. The impact energy was almost equally dissipated by the outer tube and foam core. Afterwards, all the three components started to dissipate energy. It is noted that at the displacement of 70 mm, the foam core dissipated 56.2% of the total strain energy. For the outer tube and inner tube it was 28.5% and 15.3%, respectively. To further investigate the effect of velocity on energy absorption, a graph of energy dissipation for the tube under different compression speeds is plotted in Fig. 20(b), in terms of the proportion of energy absorbed. The proportion for the foam core is over half of the whole internal strain energy, nearly 60%, and this percentage does not vary with the compression speed. However, for compression velocity up to 50 m/s, the outer tube absorbs more energy than the inner one. On the contrary, when the compression velocity is large, the inner tube is crushed and deformed severely with more strain energy. Overall, for the sandwich tubes with collapse pattern III, the energy absorption is dominated by metallic foam core, up to 60%, which again confirms the foam-filling effect in structural design. Future work is needed to elucidate the mechanics embedded.

5. Concluding remarks

This paper presents both the experimental and finite element results on the dynamic
responses of empty tubes and sandwich tubes by two flat plates. The lateral crushing tests of several empty tubes and sandwich tubes with different geometrical parameters have been carried out using an INSTRON machine. The deformation histories and load-deformation profiles have also been obtained. The energy was dissipated by plastic deformation starting at the section near the proximal surface, which is the impact region. A critical velocity has been identified for mode change and is related to $t/D$ by means of dimensional analysis.

Furthermore, three types of collapse patterns in the quasi-static tests have also been observed in the dynamic crushing experiments on the sandwich tubes subjected to compression velocity up to 10 m/s, resulting in different energy dissipation mechanisms. The dynamic behaviour of sandwich tubes under different velocities has been discussed, in terms of deformation patterns and energy dissipation. Non-symmetric deformation pattern about the horizontal plane has been observed in the case of high velocity impact. The energy absorption characteristics of sandwich tubes under impact crushing were analysed. For both empty and sandwich tubes, significant increase has been shown for the dynamic crushing load and energy absorption over its quasi-static counterpart, for which the inertia effect would be responsible. This can be explained by employing the concept of a rolling hinge to describe the propagation of plastic deformation from the proximal surface. A rolling hinge with a small radius would dissipate a significant amount of energy, which is completely different from a conventional stationary plastic hinge. Future work is needed to establish analytical models in order to explain, by means of impact mechanics, how the impact velocity determines the radius of the rolling hinge.

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manuscript, which encouraged the authors to undertake more detail studies and resulted in the addition of Section 4.1.2 in the revised version. The authors are most grateful to Professor Reid for this.

References


Fig. 1. Photograph of typical specimens for dynamic lateral crushing tests, including empty tubes and sandwich tubes.

Fig. 2. Schematic of a sandwich tube under lateral impact, (a) Impact method I: specimen and bottom platen gain a velocity and then it impacts to the top rigid platen; (b) Impact method II: bottom platen has a velocity and then impacts to the stationary specimen. Insert: illustrating of the outer diameters of inner and outer tubes, $D_i$ and $D_o$, as well as the thicknesses of both tubes, $t_i$ and $t_o$, respectively.
Fig. 3. Experimental setup for dynamic tests.

Fig. 4. Comparison of force-displacement curve from the two impact methods. Specimens ET01 and ET02 were tested using method I and II, respectively.

Fig. 5. Comparison of load-displacement curves of empty tubes between the quasi-static and dynamic crushing ($v = 10$ m/s).
Fig. 6. Illustration of collapse pattern I: (a) deformation history of SWT03; (b) comparison of force-displacement curves between quasi-static and dynamic crushing. Note that the side-load protection devices partially blocked the full view of specimens.
Fig. 7. Illustration of collapse pattern II: (a) deformation history of SWT01; (b) comparison of force-displacement curves between quasi-static and dynamic crushing.
Fig. 8. Illustration of collapse pattern III: (a) deformation history of SWT06; (b) comparison of force-displacement curves between quasi-static and dynamic crushing.
Fig. 9. Deformation profile of an empty tube (ET01: \(D = 99.81\) mm, \(t = 1.92\) mm) under two different compression velocities: (a) \(v = 10\) m/s; (b) \(v = 50\) m/s.

Fig. 10. (a) Plastic energy profile of an empty tube (ET01) under different compression speeds; (b) comparison of plastic energy ratios between quadrants for the distal end \((E_{\text{Distal}})\) and proximal end \((E_{\text{Proximal}})\).

Fig. 11. Double-logarithmic graphic plot of \(\frac{\rho v_c^2}{\sigma_y} \) versus \(t/D\).
Fig. 12. Curvature distribution along the mid-surface of an empty tube (ET01) at different values of displacement. (a) \( v = 10 \) m/s; (b) \( v = 50 \) m/s. Note that \( s = S / (\pi D) = 0 \) corresponds to the proximal end and \( s = 0.5 \) the distal end.

Fig. 13. Mid-surface circumferential strain distribution along the mid-surface of the empty tube at different values of displacement. (a) \( v = 10 \) m/s; (b) \( v = 50 \) m/s. Tensile strain is positive.

Fig. 14. Schematic diagram for a rolling hinge.
Fig. 15. Schematic of FE model subjected to impact loading (specimen: SWT01).

Fig. 16. Validation of FE models against experimental results.
Fig. 17. Deformation history of sandwich tubes under high velocity ($v = 100$ m/s): (a) SWT03 with thin foam core; (b) SWT06 with thick foam core.

Fig. 18. Load-deformation curves of sandwich tubes under different compression velocities: (a) SWT03 with thin foam core; (b) SWT06 with thick foam core.
Fig. 19. (a) Plots of enhancement over its quasi-static counterpart of energy dissipation versus velocity; (b) plot of SEA versus compressive velocity. Note that solid square points are the results of empty tube and other points are the results of sandwich tubes. The equation in solid line is obtained via similar double logarithmic methodology to Fig. 11.

Fig. 20. Energy characteristics for each component of SWT06 subjected to different velocities: (a) energy dissipation; (b) partition of energy dissipation.

Table 1(a). Specimens for empty tubes

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<th>Test</th>
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<th>Thickness $t$ (mm)</th>
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<th>EA (J)</th>
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Table 1(b). Specimens for glued sandwich tubes

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<th>Mass $m$ (g)</th>
<th>Velocity $v$ (m/s)</th>
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