INVESTIGATION OF SIMPLE CUBICAL SPACE TRUSSES AS IMPACT ENERGY ABSORBERS

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ABSTRACT

Experimental work on the possible use of cubical space trusses as energy absorbers is described. To this end the crushing behavior of single cubical space trusses welded from mild steel bars was studied experimentally. Test specimens made from bars of several different diameters d and lengths L were crushed axially between parallel plates both statically and by the use of a drop hammer. It was found that the energy absorbed per unit mass of energy absorber decreased with the increase in aspect ratio R=L / D in both the static and the dynamic testing, although the specimens tested dynamically were found to absorb nearly twice the energy absorbed by specimens tested statically.

Using basic relationships, closed-form equations were developed for predicting the crushing force and the energy absorbed by cubical space trusses. It was shown that the prediction of the crushing force is fairly accurate when compared with experimental results.

Keywords: *Collapsible, Crushing Force, Cubical, Energy Absorber, Truss*

1. INTRODUCTION

An energy absorber is a system that converts, totally or partially, kinetic energy into another form of energy. Energy converted is either reversible, like elastic strain energy in solids, or irreversible, like plastic deformation energy. Energy dissipated in plastic deformation of metallic energy absorbers is the absorbing system reviewed in this study. When designing a collapsible energy absorber, one aims at absorbing the majority of the kinetic energy of impact within the device itself in an irreversible manner, thus ensuring that human injuries and equipment damages are minimal. The conversion of the kinetic energy into plastic deformation depends, among other factors, on the magnitude and method of application of loads, deformation or displacement patterns, transmission rates, and material properties [1].

There has been intense research activity on the absorption of impact energy for minimisation of structural damage. The spectrum of activity includes crashworthiness of vehicles like cars, lifts, aircraft, and ships by [2-4], crash barrier design by [5], safety of nuclear reactors [6], collision damage to road bridges [7] and to offshore structures and oil tankers [8]. Other relevant work was presented under book titles: Crashworthiness of Vehicles [9], Structural Crashworthiness [10], Structural Impact and Crashworthiness [11], Metal Forming and Impact Mechanics [12], Structural Crashworthiness and Failure [13], Structural Impact [14], and Structural Crash-worthiness and Failure [15].

Deformable energy absorbers are made of such items as steel drums [16], circular tubes [17], tubular rings [18], three-dimensional tubular systems [19], square tubes [20-22], corrugated tubes [23], multi-corner columns [10], frusta [24], struts [25], honeycomb cells [26], sandwich plates [27] and other shapes such as stepped circular thin-walled tubes [28] and top-hat thin-walled sections [29-31]. These elements were used when filled with liquids, foam [32, 33], wood shavings [34] and sand.

The elements of energy absorbers can be arranged in a variety of geometries. Some of these include, axial crushing of tubes [35], lateral crushing of tubes [36,37], tube inversion [38,39], tube nosing [40] and tube splitting [41].

As an energy absorber, the frustum was first studied by Postlethwaite and Mills [42]. They used Alexander's method (extensible collapse analysis) for rigid-perfectly material cones. Mamalis et al. [24] developed a theoretical model to predict the mean crushing load for axially loaded circular cones and frusta deformed into the diamond mode of deformation. The model was based on the in-extensional model developed by Johnson et al. [43].

Alghamdi [44] introduced two innovative modes of deformations for frusta. The first one is direct inversion, and the other one is outward flattening. Using the ABAQUS finite element program, Aljawi and Alghamdi [45] modeled the collapse of frusta when inverted. Good agreement was obtained between experimental results and theoretical predictions. Aljawi and Alghamdi [46] further investigated the details of the inversion of frusta when crushed axially. Alghamdi et al. [47] presented the details of crushing of spun frusta between two parallel plates. They classified the deformation modes into three modes: 1.) Outward flattening, 2.) Inward inversion and outward flattening, and 3.) Extensible crumpling. They reported that their predictions using ABAQUS were in good agreement with experimental results. Other studies related to frusta included the work of axial crushing of frusta by Gupta and Abbas [48] and Chryssanthopoulos and Poggi [49]; axial crushing of constrained frusta by Singace et al. [50] and El-Sobky et al. [51], re-inversion of aluminum frusta by Alghamdi [52], crushing of un-constrained frusta by Alghamdi et al. [53] and inversion of constrained frusta by Aljawi et al. [54].

In what follows we report on experimental work on the possible use of cubical space trusses as energy absorbers.

2. EXPERIMENTAL

A number of cubical elements, shown schematically in Figure 1, were manufactured by welding from mild steel bars of diameter *d* and length *L*.

The specimens were tested using a universal testing machine (UTM) to simulate static testing, and a drop hammer facility (DHF) for dynamic testing conditions. In the samples used for UTM testing, the L was varied from 36 mm to 160 mm, *d* was 6 or 8 mm, such that aspect ratio $L / d = R$ varied from 6 to 20. The material was hot rolled mild steel rods in both cases, with yield strength S_y = 325 MPa and ultimate strength S_{ut} =455 MPa.

The DHF facility was used for dynamic loading, where eighteen of the cubical space trusses were used. Rod diameters were 6, 8 or 10 mm, L varied from 40 to 180 mm, such that aspect ratio R varied from 4 to 18.

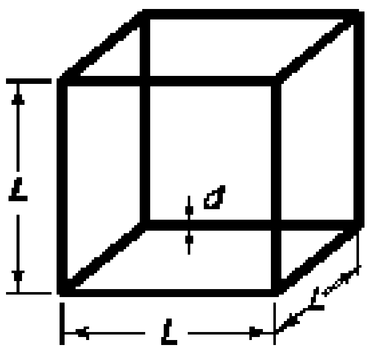

Figure 1: Schematic drawing of a cubical space truss

3. UTM TESTING

The results of crushing of single cubical space trusses between two parallel plates are summarised in Figures 2 and 3. As it would be expected, the energy absorbed per unit mass, as computed from the results of experiments, is observed to decrease with the increase in aspect ratio. This is due to limited plastic hinges being formed during the crushing process. Also, as the aspect ratio increases the deformation mode changes form plastic progressive bucking to overall Euler-type deformation. For very large aspect ratios (say 50) it would be expected that Euler buckling would take place with only one plastic hinge. The maximum instability force (P_{max}) is observed to decrease with the increase in aspect ratio, and this is attributed to the easiness of initiation of plastic hinges as the length of the column increases.

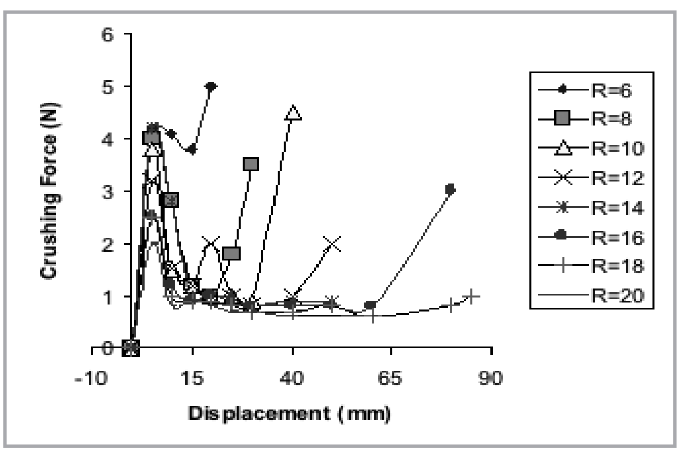

Figure 2: Load displacement curves for trusses with rod diameter d =6mm

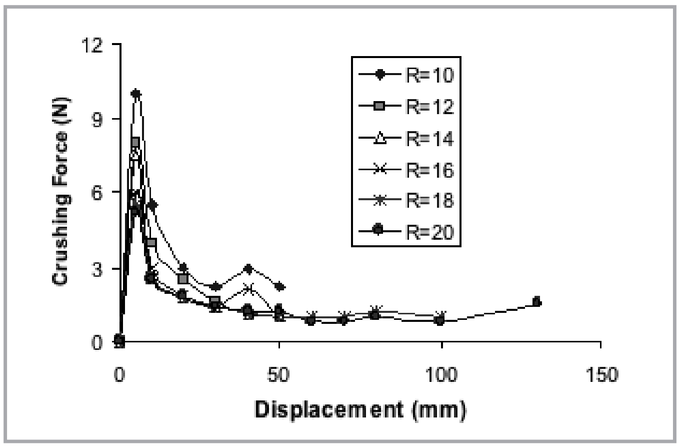

Figure 3: Load displacement curves for rod diameter d=8 mm

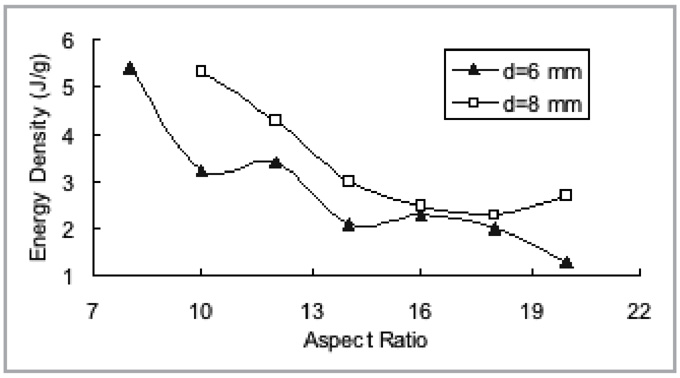

Figure 4: The relation between energy density and aspect ratio

The relation between the energy density and the aspect ratio is shown in Figure 4. As one can see energy density decreases as the aspect ratio increases. This is expected because only limited amount of material participates in plastic deformation irrespective of cubic cell length.

Now to see the history of deformation, load deformation curve for specimen D6R20 is shown in Figure 5, where the crushing force increases from zero to maximum instability value of 21364 N. The elastic response is usually ignored due to the large plastic deformation and a perfectly-rigid perfectly-plastic material model is common to use in energy absorption and plasticity [14]. The curve falls suddenly at high rate with a formation of three plastic hinges for each column one at the top base, another at the lower one and the third at the middle. Figure 6 shows the final shape of the specimen.

4. DYNAMIC TESTING

It would be expected that energy absorbers will find frequent applications in dynamic cases. As an attempt to address the conditions of dynamic cases, an in-house drop hammer facility was used for dynamic testing. The testing program consisted of crushing single cubical trusses between two parallel plates. The mass of the drop hammer was fixed during these tests to 51.5 kg. Thus only the height of the dropping distance was changed due to the change in the mass and deformation pattern of the absorber. Since the DHF is limited to a drop height of 3.75m, multiple impacts were used for absorbers that needed energy more that the potential energy of this height.

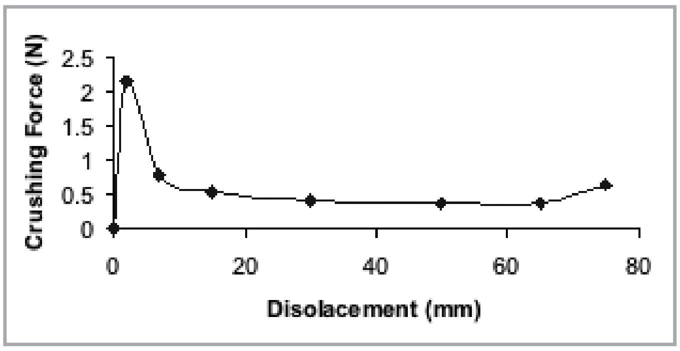

Figure 5: Load displacement curve for specimen D6R20

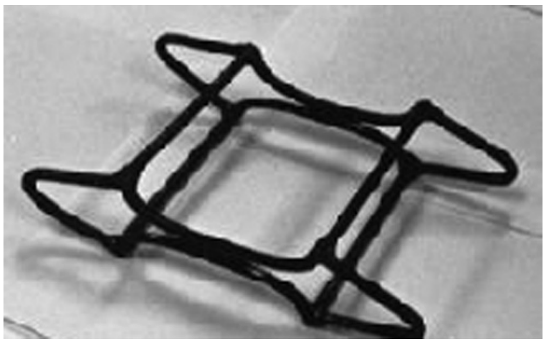

Figure 6: The final shape of specimen D6R20

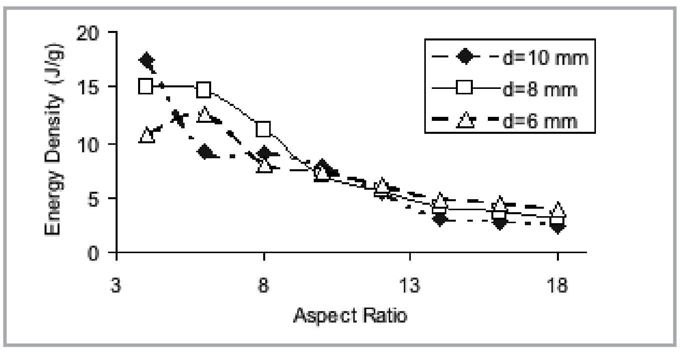

Figure 7: Relation between aspect ratio and energy density in dynamic crushing tests

Figure 7 summarises the relations between aspect ratio and the energy density for the three categories of absorbers, i.e., d = 6mm, d = 8mm, and d = 10mm. As would be expected, the energy absorbed decreases with the increase in aspect ratio because of limited plastic zone.

The percentage of the plastic zone decreases with the increase of the mass of the absorber. Thus energy density decreases with size because only limited volume of the absorber participates in the deformation.

It is to be noted that the energy density in dynamic testing is more than that in static testing. In static testing a maximum value of 9.83 J/gm was found whereas a maximum value of 17.45 J/gm is reported here. This is attributed to the dynamic behavior of the steel rods under impact. And this may be considered to be a good advantage of the absorber because it would absorb more energy under dynamic impact, which corresponds to the real case scenario. It may be shown [55] that the average energy density absorbed during dynamic testing is 1.9 times that for static testing. Figure 8 shows a comparison between the absorbed energy density per unit mass for all experiments conducted statically and dynamically. It may be concluded hence that in general the energy absorbed per unit mass in dynamic testing is nearly double that for static testing.

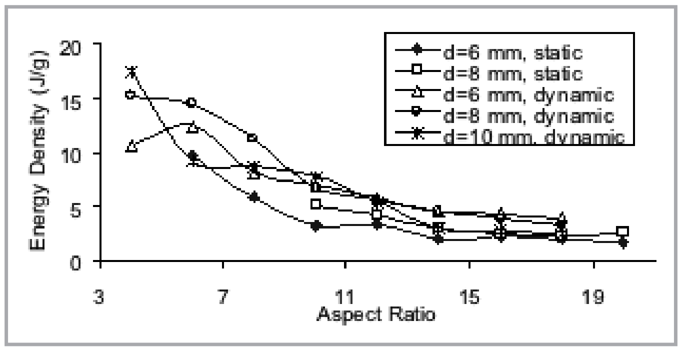

Figure 8: Comparison of the energy density in the two testing cases

5. ANALYSIS

It was noted during static tests that there seem to form three plastic hinges during axial crushing. These hinges are shown in Figures 9 and 10 where the angle of deformation θ is marked

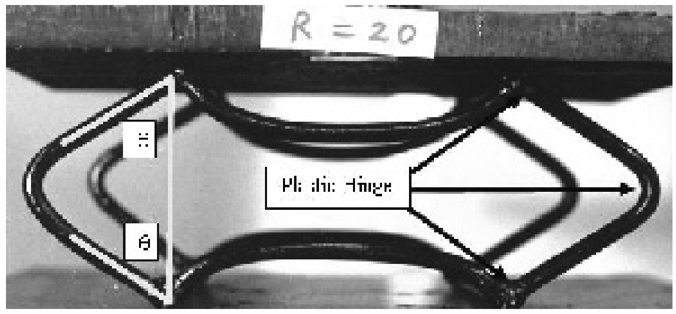

Figure 9: Specimen D6R20 during the static crushing text (at 45mm displacement)

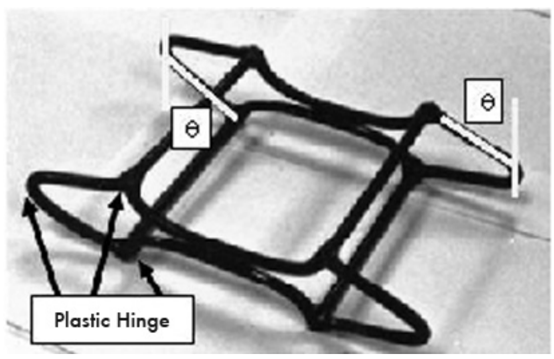

Figure 10: Specimen D6R20 at the end of the static crushing test (after 76mm displacement)

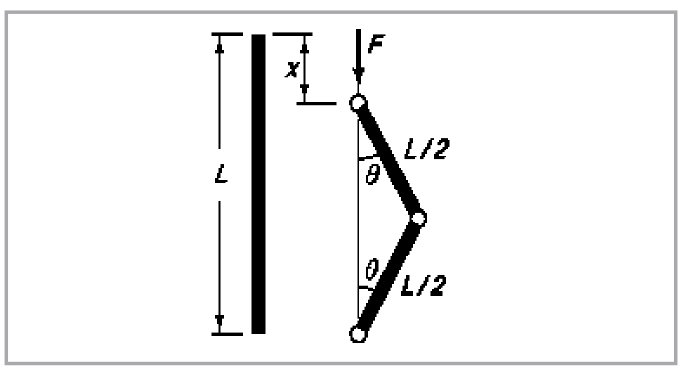

Figure 11: The system of three plastic hinges

The same deformation pattern was observed during dynamic testing. This observation was clearly verifiable during the multiple impact cases.

Consider now a bar of length L and diameter d as shown in Figure 11. We assume that three plastic hinges will be formed on this bar. It is further assumed that there is no change in volume of the rod during deformation. One can write the axial deformation, x as

$$x = L[1-\cos(\theta)] \tag{1}$$

The full plastic hinge developed in the solid circular rod of diameter d can be written as,

$$M_y = S_y \frac{d^3}{6} \tag{2}$$

where S_y is the average yield strength in Pa.

Considering Figure 11, the external moment due to the crushing force F can be written as,

$$M = F \frac{1}{2} \sin \theta \tag{3}$$

Equating Equation (2) (the internal moment) with Equation (3) (the external moment),

$$M_y = F\frac{L}{2} \sin \theta = S_y \frac{d^3}{6} \tag{4}$$

Solving for the axial force F,

$$F = \frac{S_y d^3}{3 \sin(\theta) L} \tag{5}$$

Equation (5) predicts the force required to overcome the plastic deformation of one rod and one full plastic hinge. Thus, to account for the other three plastic hinges at the middle of the rod and the two hinges at the top and bottom sides, one can rewrite Equation (5) to be,

$$F = \frac{4 N S_y d^3}{3 \sin(\theta) L} \tag{6}$$

where N is the number of rods in the cubical rod cell.

One arrives thus at the work done in the plastic deformation,

$$W = M_y \theta \tag{7}$$

or

$$W = S_y \frac{d^3}{6} \theta \tag{8}$$

where the deformation angle (θ) can be calculated using the axial displacement (x),

$$\theta = \cos^{-1}\left(\frac{L-x}{L}\right) \tag{9}$$

Figure 12 depicts a comparison between the experimental results and the theoretical prediction as estimated by Equation (6). The average yield strength and other physical properties of the rods were determined by standard tests.

Note that the theoretical value of the force goes to infinity at small values of crushing angle theta (θ). It must be pointed out that the theoretical prediction is based on a plasticity approach, and that the elastic response is totally ignored. The experimental curve is observed to go up after a deformation of about 65-mm. This is due to the touching of the upper rods with lower ones. It

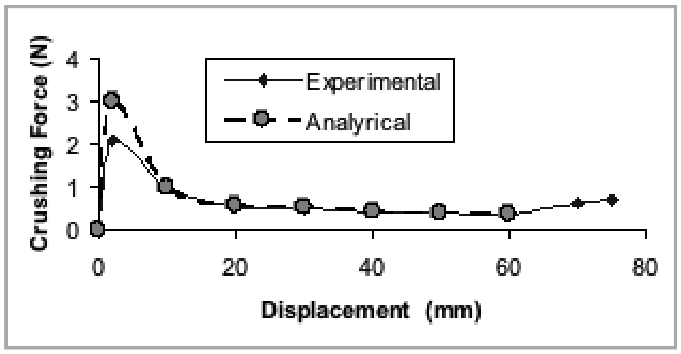

Figure 12: Experimental results and theoretical predictions of axial crushing for d = 6 mm and R = 20

may be further shown [55] that predictions of crushing force for other geometries are also reasonably accurate. Furthermore the work predicted by Equation (8) may be also shown to come out reasonably close to work found in the experiments. ■

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PROFILES

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Abdulghaffar Azhari Aljawi, whose main interests lie in finite element analysis, vibrations and plastic analysis, graduated with top honors as a mechanical engineer from KFUPM (Saudi Arabia) in 1983. He worked in the industry for a while, and then completed his MS (1989) and doctoral studies (1993), both from the University of Michigan at Ann Arbor, Michigan, USA. His doctoral studies were on vibration localization in dual-span axially moving elastic systems. Dr. Aljawi has been involved in a number of research projects involving vibrations, impact and plasticity, energy absorption and robotics. He is well known for his expertise in finite element programs, and especially the ABAQUS package.

Mehmet Akyurt received his BS (1963) and MS (1964) degrees in Mechanical Engineering from the Middle East Technical University in Ankara, Turkey, and his PhD degree (1969) from Purdue University in Lafayette, Indiana, USA. He has a long backlog of experience in the design and development of mechanisms, machinery and equipment. He has participated in a number of research projects and has published extensively. He has developed the software package Al-Yaseer for the computer-aided analysis of mechanisms and machinery.