The International Design Technology Conference, DesTech2015, 29th of June – 1st of July 2015, Geelong, Australia

Design Strategy For Selecting Appropriate Energy Absorbing Materials And Structures: Data Library And Customised Selection Criteria

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Abstract

As current determination of appropriate selection of energy absorbers is tedious and complicated, a data library and design template is developed, which hinges on the maximal energy density to stress ratio. This is the optimum point of any energy absorber, at which most energy is absorbed per unit volume at minimal reaction force per unit area. This optimum point is therefore defined by the optimal energy density and the optimal stress. From the former, the optimum absorber thickness is calculated considering the contact area and actual impact energy. From the latter, the peak deceleration is calculated considering the contact area and mass. Optimum thickness and peak deceleration are compared to the design constraints and unfeasible solutions are excluded. The remaining feasible solutions are ranked based on smallest optimum thickness, peak deceleration or absorber mass. Applications of the design template are exemplified by design cases.

Keywords: impact absorber design, foams, energy absorbing materials, data library, customised design selection, energy density to stress ratio, design constraints, helmet design

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1. Introduction

The optimal selection process of energy absorbers as design materials hinges on the knowledge of the ‘shoulder point’ location on the energy density ($W$) – stress ($\sigma$) diagram, and the solid modulus of the absorber’s material [1]. The ‘shoulder point’ corresponds to the maximal ratio of $W$ to $\sigma$, commonly misnamed as efficiency [2]. ‘Efficiency’ denotes an energy ratio, whereas the quantity of the ‘$W/\sigma$ ratio’ is strain ($\varepsilon$). If the $\sigma$-$\varepsilon$ loading curve is superelastic with a rectangular shape, then $W/\sigma$ corresponds to the full $\varepsilon$; if the loading curve is Hookean elastic (sawtooth shape), then $W/\sigma$ corresponds to half-$\varepsilon$ [2]. The rationale of having absorbers work at the ‘shoulder point’ (in the worst case) is to maximise the absorbed energy and minimise the applied stress, i.e. to determine $(W/\sigma)_{\text{max}}$. Exceeding the strain at $(W/\sigma)_{\text{max}}$ leads to densification and increased $\sigma$, and therefore to a decrease of $W/\sigma$.

A typical example is given by [1]; specifically by designing the absorber of a car headrest. The current open-cell polyester foam design (relative density $R$ of 0.06 and thickness of 0.17 m) is tested. The contact area between rest and head is approximated to 0.01 m$^2$; the mass of the head is assumed to be 2.5 kg (should actually be 5 kg); and the maximal admissible deceleration of the head is 50 g. The maximal contact stress $\sigma$ is therefore 125 kPa, and the maximum $W$ absorbed, taken from the $W$-$\sigma$ curve, is approximately 4500 J/m$^3$ (according to Figure 8.15b of [1]). This design point is beyond the shoulder curve on the $W$-$\sigma$ diagram. The maximum admissible impact velocity is therefore 2.5 m/s. Increasing the density of polyester foam from $R = 0.06$ to $R = 0.2$ improves energy absorption at the ‘shoulder point’, yet is too heavy. Due to ineffective design, the modulus of the solid material is increased by a factor of 13, and polyethylene foam is selected, thereby reducing $R$ to 0.03, yet further improving energy absorption.

As this method is rather complicated, the aim of this research is to provide an alternative by establishing a material library, by calculating the worst-case impact energy, by defining the maximum admissible design thickness, and by ranking the suitable (conforming) materials and structures according to customisable selection criteria such as largest $(W/\sigma)_{\text{max}}$, smallest absorber thickness, density, mass, and smallest deceleration and impact force.

2. Materials and methods

2.1. Materials

The following materials and structures were tested: foams such as ARTi-lage (www.artilage.com), Berkeley foams (UC-Berkeley), poron (www.poroncushioning.com), Solyte (www.asics.com), Speva (www.asics.com), polyethylene foam (www.dow.com), polyurethane foams (erapol.com.au, www.joyce.com.au, www.dow.com), EPS foam (expanded polystyrene), and D3O foams (Aerodynamic Decell; www.d3o.com); structures such as cardboard (www.cartonpallet.com), Skydex (50D, 55D, Skydex Pad; www.skydex.com), ‘MASS’ shock absorber (shockabsorbingmaterial.co.uk, plastic-castle.co.uk), and poron cushioning structures (www.poroncushioning.com). A selection of energy absorber structures is shown in Figure 1.

2.2. Methods

The absorbers were tested with an Instron material testing machine (load cell 50 kN) at two different crosshead speeds (500 mm/min, 5 mm/min). From the stress $\sigma$ - strain $\varepsilon$ curves, the following parameters were obtained:
\((W/\sigma)_{\text{max}}\) and \(W, \sigma, \text{ and } e\) at \((W/\sigma)_{\text{max}}\) according to [2] (i.e. \(W_{\text{opt}}, \sigma_{\text{opt}}, \text{ and } e_{\text{max}}\)), as well as the viscosity parameter \(\eta\) according to the method of [3]. \(\eta\) was required for extrapolating the parameters to higher impact speeds. If the density was unknown, the material was weighed and the effective volume \(V\) determined from the projected area \(A\) and the maximum thickness \(d\).

2.3. Mathematical procedure

From the incident speed \(v_0\) and the impacting mass \(m\), the impact energy \(E\) was calculated

\[
E = 0.5v_0^2 m
\]

(1)

The contact area \(A\) had to be estimated from the curvature of the materials and structures adjacent to the absorber and from the stiffness of the absorber. Selection of the worst case, i.e. the smallest possible contact area, accounted for the safety factor. As

\[
W = \frac{E}{V} = \frac{E}{dA}
\]

(2)

The optimal thickness \(d\) of the absorber was calculated from

\[
d_{\text{opt}} = \frac{E}{W_{\text{opt}} A}
\]

(3)

The maximum deceleration \((-a_{\text{peak}})\) of the mass was determined from

\[
-a_{\text{peak}} = \frac{\sigma_{\text{opt}} A}{m} = \frac{F_{\text{peak}}}{m}
\]

(4)

As the forces on either side of the absorber are identical (force equilibrium), different masses on either side result in different decelerations. Therefore, if the incident mass is \(m_1\) but the mass to be protected is \(m_2\), then the mass in Eqn (4) is selected as \(m_2\).

2.4. Design constraints

The two design constraints (Figure 2) are related to 1) the maximum space \(d_{\text{max}}\) available for housing the energy absorber, and to 2) the maximum admissible deceleration \((-a_{\text{max}})\) or impact force \(F_{\text{max}}\). The design conditions are therefore \(d_{\text{opt}} \leq d_{\text{max}}\), and \(-a_{\text{peak}} \leq -a_{\text{max}}\).

2.5. Data library and calculation template

An MS Excel design template was generated which listed all parameters required, i.e. absorber identification code, density, available thickness, \((W/\sigma)_{\text{max}}, \sigma_{\text{opt}}, W_{\text{opt}}, e_{\text{max}}\). The template contained 88 different energy absorbers. The specific input for absorber selection was \(v_0\) and \(m\), as well as \(d_{\text{max}}\) and \(-a_{\text{max}}\). After calculating \(d_{\text{opt}}\) and \(-a_{\text{peak}}\) of each absorber, unfeasible design solutions were eliminated from the two design conditions. The remaining feasible solutions were ranked according to their viability under different selectable design conditions: largest \((W/\sigma)_{\text{max}}, \) smallest \(d_{\text{opt}}, \) smallest \(|a_{\text{peak}}|, \) smallest \(\rho, \) and smallest absorber mass (= \(dA\rho\)). Costs per unit volume was not fully available for all absorbers, as such this parameter was not yet factored into the design template.
3. Case reports

The results obtained from the design template are exemplified in terms of case reports.

3.1. Case 1 – helmet design

The manufacturer’s design constraint is a total thickness of 20-22 mm including a comfort layer. The international test standard prescribed by the relevant sports associations is a deceleration of $<300 \text{ g}$ if a mass of 5 kg is dropped from a height of 1.5 m. The resultant incident speed of the dropped mass is $5.425 \text{ m/s}$ and the impact energy amounts to 73.58 J, if the contact area is circular with a diameter of 100 mm. The only suitable solution is provided by EPS foam with a density of about $70 \text{ kg/m}^3$, resulting in a thickness of 15 mm, which meets the manufacturer’s requirements and leaves a space of 5-7 mm for the comfort layer. The peak deceleration amounts to $260 \text{ g}$, which is in accordance with the standard. A denser ‘MASS’ shock absorber ($\rho = 470 \text{ kg/m}^3$; MASS impact body armour) would require a thickness of 22 mm, which can be achieved when using two layers of this damper ($d = 10.7 \text{ mm}$). An additional comfort layer would increase the design constraint. However, the peak acceleration would be reduced to $180 \text{ g}$.

A local sports association proposed that the standard should be changed to a drop height of 2.5 m (which represents the injury mechanism more closely) and a deceleration of $<250 \text{ g}$. This proposal increases the kinetic energy to 122.6 J, and increases the required EPS thickness to 24.5 mm. The peak deceleration, unaffected by the increase in material thickness, would, however, be too high by 10 g. An increase of the EPS’ density would not solve the problem, as an increase in density also increases the stress and therefore the deceleration. A material with a larger solid modulus than EPS would therefore be necessary. A compromise of a 2 m drop height and deceleration of $<300 \text{ g}$ results in a material thickness of 20 mm.

3.2. Case 2 – protection of electronics components

Electronic components are potted with a low density ($600 \text{ kg/m}^3$), lightweight epoxy resin. The resulting spherical potting compound has a diameter of 35 mm and a mass of maximally 40 g. The worst-case impact speed is $14 \text{ m/s}$, corresponding to a drop height of 10 m, and the impact energy is $3.9 \text{ J}$. The effective contact area is $0.001 \text{ m}^2$. The space available for damping is 10 mm. The deceleration should be as small as possible, although the sensor components can survive a shock of 10,000 g. The best solution is the denser ‘MASS’ shock absorber ($\rho = 470 \text{ kg/m}^3$) with a required thickness equaling the actual damper thickness (10 mm). The deceleration is 2800 g.

3.3. Case 3 – design of neck protection for batsmen

Following the fatal injury of Phillip Hughes, the need for neck protection of batsmen is evident and overdue. In this specific design exercise, the design constraints are intended to be explored in terms of what design thickness and peak force can be achieved by appropriate absorber selection. The cricket ball hit Hughes’ neck at a speed of 107 kph ($29.8 \text{ m/s}$ [4]). The worst-case scenario would be a bowling speed of 161 kph, which, however, decreases after impact on the pitch. Therefore, presuming a speed of 150 kph ($41.67 \text{ m/s}$) is used for calculating the impact...
energy of a cricket ball with 160 g mass, resulting in 139 J. In the case of a soft and deformable damping material, the worst-case contact area corresponds to the frontal area of the ball (0.004 m²). This rules out stiffer foams such as denser EPS, PE, and Skydex dampers. The only flexible material suitable for this purpose is the denser ‘MASS’ shock absorber, however for sufficient protection 8 layers would be required (total of 80 mm), which makes the protective material too stiff. The impact force would be 450 N if the neck tissue was rigid.

Increasing the contact area to 0.01 m², by stiffening the damping material (either with a stiffer foam or with an external rigid shell), offers three solutions:

• less dense EPS (35 kg/m³), 65 mm thick, impact force 600 N,
• Skydex 50D, 3 layers, 63 mm, impact force 655 N, and
• Solyte65, 67 mm, impact force 645 N.

Increasing the absorber thickness by twofold, i.e. seven layers (135 mm) of cardboard (honeycomb sandwich panels) would reduce the impact force to 200 N.

The actual impact force of Hughes’ injury could not be calculated as the rebounding ball was deflected by his left shoulder [4], however the force was larger than 700 N (force estimated from incident and rebound speed if the flight path was not obstructed by the shoulder [4]). Furthermore, it has to be considered that the impact energy was only 71 J at a speed of 107 kph, i.e. half that of when at a speed of 150 kph. From Eqns (3) and (4) it is evident that only half the absorber thickness is required at an impact energy of 71 J, whereas the impact force remains the same.

3.4. Case 4 – car headrest design

For this specific case, the design example outlined in the introduction is used. In Table 8.5 of [1], the maximum safe impact velocity when using polyethylene foam amounts to 9.3 m/s (33.5 kph). The head mass used in [1] is corrected to 5 kg. The impact energy therefore results in 216 J. The design constraints are set to a design thickness of close to 0.17 m and a maximal deceleration of <50 g. The best solution is provided by cardboard (Figure 1), 0.19 m thick (10 layers), with a deceleration of 49 g – all this considering that the mass doubled from assumed 2.5 kg to actual 5kg. The deformation of cardboard is not recoverable and has to be replaced once deformed. An additional comfort layer is required at the contact surface with the head.

4. Discussion

In most high-speed applications, EPS foam but also the ‘MASS’ shock absorber performs best. It is therefore not surprising that most bicycle helmets are historically made from EPS foams.

The new Masuri helmet, with a ‘stem guard’ made of foam and a honeycomb pattern of a stiffer material [5], is too small for protecting the entire neck. The stiffness of the honeycomb structure might be ideal for that small size, in order to transfer the impact force from the stem guard to the helmet (partially). When enlarging the absorber area, then the force would be transferred to the neck directly, and the absorber has to be designed properly.

As seen from Case 4, the data library and design template returned honeycomb cardboard as the best solution, which stands in contrast to the polyester foam suggested in [1]. This is not only due to the fact that the head mass was incorrectly assumed in [1], but also as our data library comprises 88 foams and shock absorbers so far, as of the submission date of this paper. The library is about to be expanded through continuing research.

Any design resulting from the data library and design template has to be validated by testing. However, the template acts as an efficient tool in narrowing down the suitable foams, effectively speeding up the testing process.

References

[1] Gibson L., Ashby M. Cellular Solids: structure and properties, 2nd edition, Cambridge: Cambridge University Press; 1999.
[2] Fuss FK. The design strain and dead mass of energy absorbing materials and structures: mathematical principles and experimental determination. Procedia Engineering 2015; in press
[3] Fuss FK. Nonlinear visco-elastic materials: stress relaxation and strain rate dependency, in: Nonlinear Approaches in Engineering Applications, Eds: Dai L, Jazar RN; Springer, New York, pp 135-170, 2012.
[4] Fuss FK. Video analysis of Phillip Hughes’ neck injury. Report for the NSW Police; 2015.
[5] Phillip Hughes tragedy prompts British firm Masuri to redesign cricket helmets. MailOnline, available: http://www.dailymail.co.uk/sport/cricket/article-2949700/Phillip-Hughes-tragedy-prompts-British-firm-Masuri-redesign-cricket-helmets.html