Mechanical properties and design of sandwich materials

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Sandwich materials consisting of a low density core with stiff skins offer considerable potential for weight saving in panel applications, where the main loads are flexural. Sandwich materials of interest for car and van body panels, seat shells, etc, include steel/plastic laminates, integral skinned plastic foams and glass fibre-reinforced polyester skins with foamed plastic cores. In this paper, basic design formulae for the flexural stiffness and strength of such sandwich materials are reviewed and a method for designing optimum sandwich structures for least weight or cost is given. Mechanical property data are presented on a range of sandwich materials of potential interest for vehicle panel applications. It is then shown how use of the least-weight design method enables core and skin thicknesses to be determined and gives a means of improving the flexural properties of existing sandwich constructions.

Key words: *sandwich materials; design calculations; flexural stiffness; strength; weight saving; automotive components*

Sandwich materials consising of a low density core with stiff skins offer considerable potential for weight saving in panel applications, where the main loads are flexural. For aerospace applications, new materials such as carbon fibre-reinforced epoxy skins with aluminium honeycomb cores have been developed along with relevant design methods. Because of their high cost and difficulties in fabrication, such materials are not appropriate for use in panel applications in car and van bodywork or container vehicles. Sandwich materials of interest for vehicle applications are integral skinned plastic foams for interior components such as air filter covers, seat shells and floor panels, and steel/plastic laminates or glass-fibre reinforced plastics (GRP) with foamed plastic cores for low-weight body panels with good surface finish. Double skinned metal or thermoplastic panels with foamed plastic cores are also of interest for the energy absorbing or thermal insulation properties provided by the core material. This paper is concerned with the development of simplified design methods for such materials and with the provision of relevant materials property data to enable optimum sandwich constructions to be selected.

In aerospace sandwich structures, design analysis is carried out using finite element and other computer-aided design methods in which details of the stress distribution through the core and skins of the structure are examined in terms of the applied loads, see for example Reference 1. Such detailed analysis methods are not appropriate for vehicle applications because of their complexity and because design criteria are less critical. For these applications a simpler design approach may be used in which the *materials design* is considered independently from the *structural design.* The materials design problem is concerned with the calculation and optimization of the sandwich stiffness and strength properties in terms of skin and core properties. Details of the sandwich construction are then neglected in the structural design problem which is concerned with calculating the effect of support conditions and loads on a panel of known stiffness and strength. Thus the structural design problem can follow traditional design methods for loaded panels.

In this paper attention is restricted to the materials design of sandwich panels under flexural loads. A brief discussion is given of the classification of sandwich materials for design purposes based on the definition of a critical panel length. This is followed by basic design formulae for the flexural stiffness and strength of sandwich materials and a method for designing optimum sandwich materials for least weight or cost. Mechanical property data are then presented on a range of sandwich materials of potential interest for vehicle panel applications. These include a steel/plastic laminate, thermoplastic structural foams and GRP/plastic foam sandwiches. Classification data are presented along with a comparison of measured and

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predicted stiffness and strength properties. Next, the design formulae for least-weight design of a sandwich material are demonstrated and least-weight constructions for several combinations of material determined. These least-weight sandwich materials are then compared with 1 mm thick steel sheet to show the potential weight savings that are possible.

DESIGN METHODS FOR SANDWICH MATERIALS

Classification of sandwich materials

An ideal structural sandwich material consists of a thick sheet of lightweight material, with a thin sheet of much stiffer, stronger material attached to each face. The skin resists in-plane forces and bending moments, whilst the core resists transverse shear forces in the panel. The core must be stiff enough in shear to prevent the skins sliding over each other, and in compression to keep the skins the correct distance apart. The design methods needed to analyse sandwich panels under transverse loads depend on the relative importance of core shear deformations during bending. In this paper, sandwich materials are classified into two types depending on the core flexibility. When the core is *stiff in shear*, shear deformations in bending are negligible and classical bending theory may be used for sandwich panels, with an appropriate value for the flexural rigidity. When the core is *flexible in shear*, shear deformations may be significant and contribute to panel deflections during transverse loading. In this case classical bending analysis for beams and panels requires modification to include thickness shear effects.

In this paper attention is restricted to the stiff core materials and data are presented to show that the sandwich materials of interest for vehicle applications fall into this category. For these materials conventional structural design methods based on conventional beam, plate or shell theory, which neglect through-thickness shear effects, are valid to a good approximation. For a more rigorous analysis, modified design analysis methods which include core shear flexibility are available,[2] and these methods are necessary for certain aerospace sandwich materials with very stiff skins.

A convenient method of classifying materials is to consider their behaviour in a three-point beam bend test. Consider a beam of length l, with flexural rigidity D and shear stiffness Q, simply supported at the ends and subjected to a transverse central load P. When deflections due to both bending and shear are taken into account, the centre point deflection w is given by:[2]

$$w = \frac{Pl^3}{48D}\left[1 + \frac{12D}{l^2Q}\right] \qquad (1)$$

where the second bracketed term gives the contribution to the beam deflection from shear. It can be seen that if $D/l^2Q \ll 1$, Equation (1) reduces to the classical beam bending relation in which shear effects are neglected. In sandwich materials these shear deformations arise from the core. In order to classify materials, we define the core to be *stiff in shear* if:

$$D/l^2Q < 0.01 \qquad (2)$$

and *flexible in shear* if:

$$D/l^2Q > 0.01 \qquad (3)$$

It is apparent that the relative importance of shear depends on beam length l, and a critical length l^* may be defined from Equations (2) and (3):

$$l^* = 10(D/Q)^{1/2} \qquad (4)$$

such that when beam length $l > l^*$, shear effects are small.

A series of three-point bend tests on sandwich beams of different lengths l enables both D and Q to be measured and hence the critical length l^* to be determined. This procedure has the advantage of classifying sandwich materials without having to measure skin and core properties separately. If the measured centre point deflections w for several beams of length l and given load P are plotted in the form w/Pl against l^2, Equation (1) shows that the results should be a straight line with gradient $1/48D$ and intercept $1/4Q$. Alternatively, a plot of w/Pl^3 against $1/l^2$ should again give a straight line with gradient $1/4Q$ and intercept $1/48D$. Thus D, Q and hence the critical length l^* may be determined. In order to use classical beam and plate design formulae for sandwich panel analysis, typical panel length dimensions are then required to be greater than the critical length l^*. Classification data on a range of materials obtained by these methods will be presented below.

Calculation of stiffness and strength

Consider the symmetric three-ply sandwich beam shown schematically in Fig. 1. The beam has width b, thickness h and consists of a central core with thickness h_c, and skins of thickness h_s, so that $h = h_c + 2h_s$. The core and skin Young's moduli along the beam axis are E_c and E_s respectively and it is assumed that transverse strains arising from differences in Poisson's ratio are negligible. The core and skin densities are denoted by ρ_c and ρ_s respectively.

If ρ is the beam density, consideration of the total mass of the beam gives:

$$bhl\rho = bl(h_c\rho_c + 2h_s\rho_s)$$

whence

$$\rho = \rho_c v + \rho_s(1 - v) \qquad (5)$$

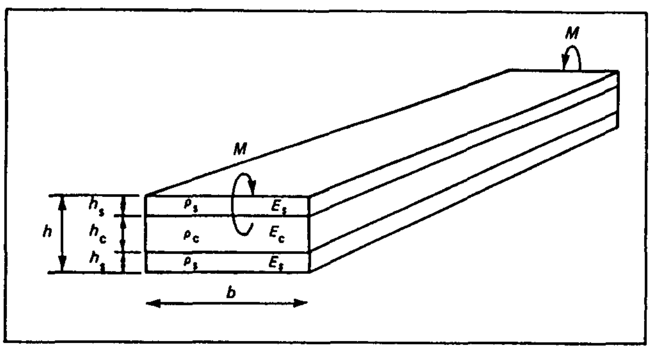

Fig. 1 Schematic of sandwich beam

where the core/sandwich thickness ratio parameter v is introduced:

$$v = h_c/h \tag{6}$$

Using a similar 'rule-of-mixtures' argument to the beam tensile stiffness, the tensile modulus of the sandwich material E_t is given by the formula:

$$E_t = E_c v + E_s(1 - v) \tag{7}$$

which shows that the skins and core contribute to the tensile stiffness in proportion to their thicknesses.

In order to calculate the flexural rigidity D, assume that classical bending theory may be applied to the sandwich beam. Thus cross-sections which are plane and perpendicular to the neutral axis of the unloaded beam remain so when bending takes place. Also assume that the beam is narrow so that transverse stresses parallel to the faces may be neglected. It can then be shown that D is given by:

$$D = E_c I_c + 2E_s I_s \tag{8}$$

where I_c and I_s are the moments of inertia of the core and skin cross-sections about the centroidal axis of the beam. With the notation of Fig. 1, $I_c = bh_c^3/12$ and $I_s = bh_s^3/12 + (h_c + h_s)^2 bh_s/4$, which gives the result:

$$D = bE_s[h_s^3/6 + h_s(h_c + h_s)^2/2] + bE_c h_c^3/12 \tag{9}$$

Let E_f be the effective flexural modulus of the sandwich beam, then:

$$D = bh^3 E_f/12 \tag{10}$$

After some algebraic manipulation, Equation (9) may be expressed in the more simple and useful form as a formula for E_f:

$$E_f = E_c v^3 + E_s(1 - v^3) \tag{11}$$

Several modes of failure may occur in sandwich beams subjected to flexural loading, depending on sandwich geometry and core and skin properties. Two important failure modes are tensile (or compressive) failure of the skin material at the beam surface, where stresses in flexure are highest, and by skin wrinkling at the compressive face. Assuming that the skin and core materials are linearly elastic to failure, flexure beam theory gives the strain e at a distance z from the neutral axis of a sandwich beam subjected to a bending moment M as:

$$e = Mz/D \tag{12}$$

where D is given by Equation (9) or (10). In the case of a symmetric sandwich, in which the skin material fails at the beam surface $z = \pm h/2$, the surface strain reaches the skin material failure strain e_u at the ultimate bending moment M_u, given from Equation (12) by:

$$M_u = 2De_u/h \tag{13}$$

For linearly elastic skins we may set $e_u = \sigma_u/E_s$, where σ_u is the skin material failure stress, and using Equation (10) obtain:

$$M_u = bh^2 \sigma_u E_f/6E_s \tag{14}$$

In Equation (14) σ_u is the lowest of the skin material tensile or compressive strength and the skin wrinkling stress. The latter may be estimated from a buckling analysis, and for thin skins is given approximately by:[2]

$$\sigma_u = \tfrac{1}{2} E_s^{1/3} E_c^{2/3} \tag{15}$$

Design formulae for other modes of failure such as core shear failure or skin/core debonding are also discussed by Allen.[2]

Design of sandwich materials

In the design analysis of sandwich panels, the basic formulae — Equations (10), (11) and (14) — are used for calculating the flexural rigidity D and ultimate bending moment M_u in terms of sandwich geometry. The critical length l^* is also estimated from Equation (4) or measured in a series of beam bending tests. Provided that typical panel dimensions are greater than l^*, we may use standard design formulae (see, for example, Roark and Young[3]) or computer programs for plates and beams to determine quantities such as panel deflections and bending moments. The maximum deflections may then be compared with imposed limiting deflections and the maximum bending moments with the ultimate bending moment M_u for the sandwich material.

However, in order to make best use of sandwich materials, guidelines are needed for the materials design problem in which core and skin thicknesses are specified. Note that the flexural strength and flexural modulus E_f for a sandwich material are lower than those for a homogeneous beam of the stiffer skin material. The main advantage of a sandwich construction arises because the low density core material enables the panel thickness h to be increased for the same weight. It is then the increase in h^3 in Equation (10) which offsets the lower flexural modulus and results in a higher sandwich panel flexural rigidity D. Similarly, the h^2 term in Equation (14) puts up the limiting bending moment M_u. Increasing the core thickness to enhance flexural properties also increases panel weight and attention may need to be given to least-weight design. The choice of sandwich geometry for a least-weight panel is considered by Johnson[4] and design charts are given for optimizing core thickness ratio.

For a sandwich material with specified skin and core moduli, strengths and densities, the main design variables become the total thickness h and the core thickness ratio, $v = h_c/h$. In terms of h and v we have:

$$h_c = hv, \quad h_s = h(1 - v)/2 \tag{16}$$

Johnson[4] considers the design of a least-weight sandwich material for a given flexural rigidity D or ultimate bending moment M_u. It is shown that in both the stiffness and strength limited cases there is an optimum choice of the core thickness ratio v.

In the stiffness limited case it is found that the panel weight/unit area, W, given by:

$$W = h\rho = h[\rho_c\nu + \rho_s(1-\nu)]$$

from Equation (5), is minimized for a sandwich material of given flexural rigidity D, when the thickness ratio ν is given by:

$$\nu = \left[\frac{1-\rho_c/\rho_s}{1-E_c/E_s}\right]^{1/2} \tag{17}$$

To obtain the least-weight sandwich construction for a given flexural rigidity D, ν is first calculated from Equation (17) in terms of skin and core moduli and densities. The sandwich flexural modulus E_f is then given by Equation (11), and the required panel thickness is determined from Equation (10) written in the form:

$$h = (12D/bE_f)^{1/3} \tag{18}$$

in terms of the required flexural rigidity/unit width, D/b. Equation (14) may then be used to determine the ultimate bending moment for the sandwich material.

In the strength limited case, it is found that the optimum core thickness ratio ν satisfies the cubic equation:

$$\nu^3 - 3\nu^2/(1-\rho_c/\rho_s) + 2/(1-E_c/E_s) = 0 \tag{19}$$

For given values of ρ_c/ρ_s and E_c/E_s, the root ν in the range $0 < \nu < 1$ may be determined graphically or by numerical computation. In the strength limited case, ν is obtained from Equation (19) and the sandwich thickness h is then given from Equation (14) in terms of the required ultimate bending moment/unit width, M_u/b:

$$h = (6M_u E_s/b\,\sigma_u E_f)^{1/2} \tag{20}$$

In order to facilitate the design method, Equations (17) and (19) for the least-weight core thickness ratios may be put onto a design chart, see Fig. 2. Using the figure it is possible to read off the optimum core thickness ratios ν for given values of the density ratios ρ_c/ρ_s and modulus ratios E_c/E_s. The continuous lines on Fig. 2 refer to the stiffness limited values from Equation (17) and the broken lines are the corresponding strength limited values derived from the appropriate root of the cubic Equation (19).

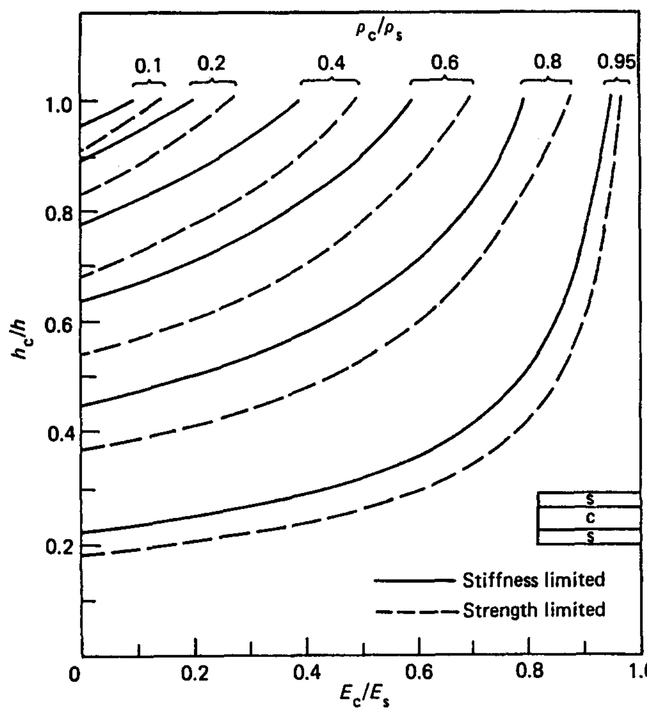

Fig. 2 Design chart for optimum core thickness ratio h_c/h (= ν) for sandwich materials

If ρ_c and ρ_s are taken to be the cost/unit volume of the core and skin materials, then the design formulae given here for the optimum core thickness ratio ν may be used to design a least-cost sandwich material for a given flexural rigidity or flexural strength.

SANDWICH MATERIAL PROPERTIES

Skin and core property data

In order to use the sandwich materials design formulae presented above, materials property data on skin and core materials are needed. Table 1 lists such data for a

Table 1. Mechanical property data on skin and core materials

Material		Density, ρ (kg m^{-3})	Young's modulus, E (GPa)	Shear modulus, G (GPa)	Tensile strength, σ (MPa)
Steel		7900	207	88	250*
GRP fabric		1900	18.8	—	265
GR nylon		1390	7.2	1.3	150
PP		910	1.23	0.45	27*
PP structural foam	Skin	860	1.73	—	36*
	Core	580	0.73	—	—
PU foam		72	0.019	0.005	0.52
		155	0.056	0.016	2.2

*Yield stress

Table 2. Classification data on sandwich materials

Material	Thickness, h (mm)	Flexural rigidity per mm width, D/mm (kN mm^{-1})	Shear stiffness per mm width, Q/mm (kN mm^{-1})	Critical length, l^* (mm)	l^*/h
PP structural foam	8	56	1.6	59	7.4
PC structural foam	8	86	2.8	55	6.9
PPO structural foam	8	80	3.2	50	6.3
RIM PU	3.7	2.7	—	< 35	< 10
GRP/PU sandwich	10.7	820	0.53	390	37
Steel/PP sandwich	0.95	11.9	0.25	69	73
Steel	1	17.3	88	4.4	4.4

range of skin and core materials of potential interest in vehicle applications. Values are given for density ρ, Young's modulus E, shear modulus G and tensile strength σ; these were measured at the National Physical Laboratory (NPL) or collated from materials suppliers' data sheets. The NPL data were obtained under slow loading rates ($\sim$ 20 mm min^{-1}) at 20°C and 50% relative humidity (RH). Note that the properties of the foam core materials are strongly dependent on the degree of foaming, which may be characterized by the density. The table shows typical values for these materials at a particular density. The materials listed are steel for the skin material in steel/plastic laminates; polypropylene (PP) as a typical core material for such laminates; a glass fabric-reinforced polyester (GRP fabric) and short glass fibre-reinforced nylon (GR nylon) as typical reinforced plastics skin materials for use with low density cores such as rigid polyurethane (PU) foam. The PP structural foam is an integral skinned polypropylene foam, and is a typical thermoplastic structural foam fabricated in a single shot injection moulding process. Note that the PP structural foam contains a different grade to that used in the metal/plastic laminate. The GRP fabric and GR nylon have anisotropic stiffness and strength properties; the values quoted refer to a principal fibre direction.

Classification data

Table 2 gives classification data on a range of sandwich materials in the form of flexural rigidity per mm width (D/mm), shear stiffness per mm width (Q/mm), critical length (l^*) and critical length/thickness ratio (l^*/h). Data are given on integral skinned thermoplastic structural foams of PP, polycarbonate (PC) and polyphenylene oxide (PPO); short glass fibre-filled reaction injection moulded PU (RIM PU); GRP fabric/PU foam sandwich and a standard steel/PP sandwich laminate (Hercules, Liteplate S). Comparative data on a 1 mm thick steel panel are also given. Variable length three-point beam bend tests, as described above, were carried out at NPL to obtain the classification data on all the polymeric sandwich materials. For the structural foams and RIM PU there are no well-defined core and skins, hence this test method is useful in providing a rapid means of obtaining sandwich stiffness properties without separating the core and skins. For the steel/PP laminate the core and skins are well defined and it is possible to determine l^* by calculation. Thus flexural rigidity D was calculated using Equations (10) and (11) with the basic materials data of Table 1 (see also below). Assuming that all the shear deformations take place in the plastics core, Q may be estimated from the relation:

$$Q = bh_c G_c \tag{21}$$

where G_c is the core shear modulus. In a similar way it is possible to use Equations (10) and (21) to obtain the D and Q values given for the steel panel material.

The main point to notice from the table is that all the sandwich materials have much greater critical lengths than the value 4.4 mm obtained for the steel. The structural foams and steel/PP laminate are seen to have critical lengths in the range 50–70 mm, whilst the RIM PU value, which could not be measured accurately, is below 35 mm. It follows that classical design methods for calculating panel flexural response are valid, and shear effects small, provided that panel dimensions are greater than 70 mm. This will be the case for typical panel components in vehicles. However, it is apparent that care is needed with these materials at fixing points and under local loads, where the effective panel length may be small. In these situations shear effects may become significant in increasing panel deflections and thus reducing panel stiffness below the assumed flexural rigidity. The classification data on the GRP fabric/PU foam core sandwich give a critical length of 390 mm for this high performance, low-weight sandwich material. Thus, although it has high flexural rigidity, significant attention would need to be given to shear effects in this type of material if used in small automotive structures.

Because of the wide range of sandwich thicknesses and hence flexural rigidities, the materials in Table 2 are not all equivalent in their flexural properties to a 1 mm steel panel. To allow for this, it is informative to express the critical length relative to the panel thickness; ie, as l^*/h. The integral skinned plastics foams all have $l^*/h < 10$ and are not much higher than the steel value. However, the steel/PP laminate value of 73 and GRP/PU value of 39 indicate that sandwich materials with very stiff skins and low modulus cores may need special design methods for critical areas near load and fixing points.

Measured stiffness and strength properties

In this section the measured stiffness and strength properties of sandwich materials are compared with values predicted from the design formulae

Table 3. Comparison of measured and predicted sandwich properties

Material		Thicknesses (mm) h	h_c	h_s	Moduli (GPa) E_t	E_f	Strength, M_u/b (N mm mm^{-1})	Density, ρ (kg m^{-3})
Steel/PP	Experiment	0.95	0.55	0.2	88	170	—	3720
	Theory				86	167	30.3	3850
GRP/PU foam	Experiment	10.7	8.9	0.9	3.5	8.0	1200	440
	Theory				3.3	8.1	1605	452
PP structural foam	Experiment	8	—	—	0.96	1.25	416	650
	Theory		6.1	0.95	0.96	1.29	280	—

(Equations (7), (10), (11) and (14), thus establishing the validity of these simple formulae for sandwich materials design. The materials considered are a standard steel/PP laminate (Hercules, Liteplate S), a GRP fabric/PU foam sandwich material and a PP structural foam. Table 3 gives core and skin thicknesses along with measured and predicted values of tensile modulus E_t, flexural modulus E_f, density ρ and ultimate bending moment/unit width M_u/b.

The standard steel/PP laminate has well-defined core and skins, whose thickness could be easily measured. Table 3 shows very good agreement between measured and predicted tensile and flexural moduli. Note that the flexural modulus is about 80% of the steel value, whilst the tensile modulus is much lower, being about 40% of the steel value. There is, however, considerable weight saving, with the density being about 50% of that of steel. It was not possible to measure accurately the bending moment at failure of this material because of the difficulty of detecting the onset of yield in the steel skins.

The GRP fabric/PU foam sandwich also had clearly defined skins and again reasonable agreement is seen between measured and predicted modulus values. Note that the PU foam core in this sandwich material has a density of 155 kg m^{-3} so that the higher density data from Table 1 are used. Because of the importance of core shear effects in these materials, discussed above, the flexural modulus was obtained from the flexural rigidity D measured in the classification tests and given in Table 2. In calculating the ultimate bending moment M_u, Equation (14) was used. From Equation (15) it was found that the skin wrinkling stress, σ_u, was 195 MPa for this sandwich material. This is below the GRP fabric strength given in Table 1. Failure is therefore expected to occur by buckling of the compressive skin, and a failure stress of 195 MPa was used for σ_u in Equation (14). A three-point beam bend test was carried out on this material to failure and a compressive buckling failure was observed. However, the measured ultimate bending moment M_u was about 25% lower than the predicted value. This is probably due to the difficulty in observing the onset of skin buckling and the fact that Equation (15) gives only an approximate lower bound for the skin wrinkling stress. Note that this GRP/PU sandwich material has very low density compared with the steel/PP sandwich and, because of its thickness, a high ultimate bending moment.

In the integral skinned PP structural foam, it is difficult to measure the skin thickness with any accuracy, since the skins and core merge together. However, core, skin and sandwich densities are readily determined, and these can be used to calculate an effective h_c/h value for use in the design formulae. Rearranging Equation (5) gives:

$$v = h_c/h = (\rho_s - \rho)/(\rho_s - \rho_c) \qquad (22)$$

which enables v to be determined in terms of the densities ρ, ρ_c and ρ_s. Using the density data on PP structural foam from Tables 1 and 3, this gives a value $v = 0.76$. Hence from Equation (16), $h_c = 6.1$ mm and $h_s = 0.95$ mm are the effective core and skin thicknesses for use in the design formulae. Table 3 shows very good agreement between measured and predicted tensile and flexural moduli based on Equations (7) and (11) with $v = 0.76$. Equation (14) now predicts a low value for M_u compared with that measured in a three-point bend test. The calculated M_u value was based on an assumed yield stress for σ_u, ie, M_u is the bending moment at yield, whereas the measured M_u is at ultimate failure, which explains the higher measured value. Compared with the solid PP skin material, the structural foam sandwich retains 72% of its flexural modulus with a weight saving of 25%.

LEAST-WEIGHT SANDWICH MATERIALS

Examples of least-weight design method

The design methods presented above are now illustrated by calculating the stiffness limited least-weight sandwich construction for a range of materials. The calculations are based on using Equation (17) or the design chart (Fig. 2) to obtain the optimum core thickness ratio v for given core and skin density and modulus ratios. It is of interest to compare a range of sandwich materials with an equivalent steel panel. A typical vehicle panel of 1 mm thick mild steel has a flexural rigidity per mm width (D/mm) of 17.3 kNmm as indicated in Table 2. The results of using the design formulae to determine least-weight sandwich geometry for this value of flexural rigidity are given in Table 4.

For the steel/PP laminate, the data from Table 1 give $\rho_c/\rho_s = 0.115$ and $E_c/E_s = 0.0059$; hence, from Equation (17) or reading from Fig. 2, $v = 0.94$. The

Table 4. Least-weight sandwich constructions

Material	Thicknesses (mm)			Stiffnesses		Strength, M_u/b (N mm mm^{-1})	Panel weight, W (kg m^{-2})	Normalized weight
	h	h_c	h_s	D/b (kN mm)	T/b (kN mm^{-1})			
Steel	1	—	—	17.3	207	42	7.9	1
Steel/PP	1.8	1.7	0.05	17.3	24.4	23	2.4	0.30
PP structural foam	5.5	4.1	0.7	17.3	5.4	132	3.6	0.45
GRP/PU foam	4.5	4.3	0.1	17.3	3.9	87	1.0	0.13
GR nylon/PU foam	5.7	5.4	0.15	17.3	2.6	117	1.2	0.16

flexural modulus of this least-weight laminate is given from Equation (11) as $E_f = 36$ GPa. The thickness h to give a flexural rigidity D/mm of 17.3 kNmm is obtained from Equation (18) as $h = 1.8$ mm. Hence Equation (16) gives for the core and skin thicknesses $h_c = 1.7$ mm and $h_s = 0.05$ mm. In order to evaluate other properties of the laminate, the tensile stiffness T, defined by:

$$T = bh\,E_t \tag{23}$$

is also given in Table 4, along with the ultimate bending moment M_u calculated from Equation (14) and the panel weight/unit area W given by the formula presented above.

In a similar manner the least-weight sandwich constructions for PP structural foam, GRP fabric/PU foam and GR nylon/PU foam, for the same flexural rigidity, may be determined using the materials data from Table 1 and these results are also shown in Table 4.

Discussion of weight saving

Table 4 shows that considerable weight savings are possible with sandwich materials designed for an equivalent flexural rigidity as 1 mm steel. The optimum steel/PP laminate is only 30% and the PP structural foam only 45% of the steel weight. The two reinforced plastics panels with the very lightweight PU foam core have weights only about 15% of that of steel. Note also that most of the sandwich panels have ultimate bending moments M_u comparable to or greater than the value at yield in a steel panel. The major disadvantage of these constructions compared with steel is the significant reduction in tensile stiffness T, with the lowest weight panels being less than 2% of the steel tensile stiffness. This could be a significant disadvantage for sandwich panels subjected to both in-plane and bending loads, or under compressive buckling loads.

It is interesting to compare the least-weight sandwiches with the more typical constructions described in Table 3. The main difference is that the least-weight materials have thicker cores and thinner skins. Thus the commercially available steel/PP laminate in Table 3 has 0.2 mm skins on a 0.55 mm core, whilst the optimum laminate requires 0.05 mm skins on a 1.7 mm core. Similarly, the reinforced plastics skins sandwiching the PU foam need only be 0.1 or 0.15 mm. However, the PP structural foam in Table 3 is close to the least-weight construction with a core thickness ratio $v = 0.76$, compared with an optimum value $v = 0.75$.

For applications in vehicles it is probable that these least-weight constructions require skins that are too thin and hence susceptible to damage or corrosion. However, the design procedure given here shows the potential weight savings that are possible with sandwich panels and in practice thicker skins than the optimum may be required. The main reason for the very thin skins shown in Table 4 is that the sandwich materials were designed to be equivalent to 1 mm steel. In applications such as vehicle doors and bonnets, the construction is a double skin panel or a steel skin on a subframe. A plastics sandwich construction to meet the component design specifications would thus be much thicker than the idealized panels in Table 4, with correspondingly thicker skins. The thicker core required in, say, a door would also allow other benefits to be designed into the component, such as impact energy absorption, thermal insulation or vibration damping.

CONCLUSIONS

The classification of sandwich materials for design purposes has been considered in this paper. If core shear thickness is too low then shear deflections during flexural loading are significant. It is shown that a critical panel length may be defined, above which shear effects are small and where design may be based on conventional beam, plate or shell analysis. Data on sandwich materials of interest for automotive applications show that they are generally in this latter category.

Simple design formulae are given for calculating the flexural stiffness and strength of sandwich materials. These are supported by test data on a range of different material combinations. Design formulae are also given for selecting skin and core thicknesses to give a sandwich material of least weight or least cost for a required flexural stiffness or strength. Some least-weight sandwich configurations are given and the weight saving compared with an equivalent steel panel is calculated.

It is shown that sandwich materials have high potential for application in low weight automotive panels and components. Possible problems in their use may be associated with fixing and loading points, where shear effects may be significant and where the stiff skin materials are susceptible to local damage.

ACKNOWLEDGEMENT

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