ANALYSIS OF PRECISION SANDWICH STRUCTURES UNDER THERMAL LOADING

Wilfried Elspass[1]
Manfred Flemming[2]

Institute of design and construction methods (IKB)
Swiss Federal Institute of Technology (SFIT) Zurich
Raemistr. 101, CH-8092 Zurich

Abstract

Composite sandwich structures must be carefully designed to meet very close thermal distortion tolerances such as those required for structures like comunication satellite antennas or reflectors of terrestrial systems.

To analyze and optimize the design of high precision sandwich structures it is necessary to know the complete thermoelastic material property data of all components. For that reason analytical and finite element models have been developed to determine the thermoelastic constants of honeycombs as an orthotropic homogeneous material.

As honeycomb is regarded as a homogeneous orthotropic material, it can be modelled with the knowledge of all 9 elastic constants by use of solid elements in finite element analysis. This procedure simplifies the analysis of sandwich structures thus improving the accuracy concerning the in-plane stiffness (in-plane = in the xy-plane) of the core material.

Nomenclature

A	area [mm^2]
c	cell size [mm]
E	Young's modulus [N/mm²]
f	focal length [mm]
F	force [N]
G	shear modulus [N/mm²]
I	moment of inertia [mm⁴]
l	length [mm]
N	force [N]
Q	transverse force [N]
R	radius [mm]
t	thickness [mm]
T	temperature [°C]
u,v,w	displacements [mm]
cte	coefficient of thermal expansion [μm/m °C]
ε	strain [%]
η	local coordinate
ν	Poisson's ratio
ρ	mass density [kg/m³]
σ	normal stress [N/mm²]
τ	shear stress [N/mm²]
ξ	local coordinate
ϕ	angle between honeycomb webs [°]

Introduction

Spacecraft components require the use of lightweight materials that are dimensionally invariant with respect to temperature changes, to achieve and maintain dimensional accuracy. Advanced composite materials even in combination with metallic honeycomb material result in sandwich components which meet the requirements of high stiffness and minimum weight while providing the capability of achieving a near-zero coefficient of thermal expansion (CTE) over quite a wide temperature range. These properties are most useful to future, high frequency communications satellite antenna reflectors, where deviations from the ideal contour must be held to fractions of a millimeter under all operating environmental conditions of space.

To predict the thermoelastic behavior of a sandwich component with high accuracy, it is necessary to consider all relevant components and their influence on the overall behavior. Figure 1 shows the complexity of a sandwich from this point of view.

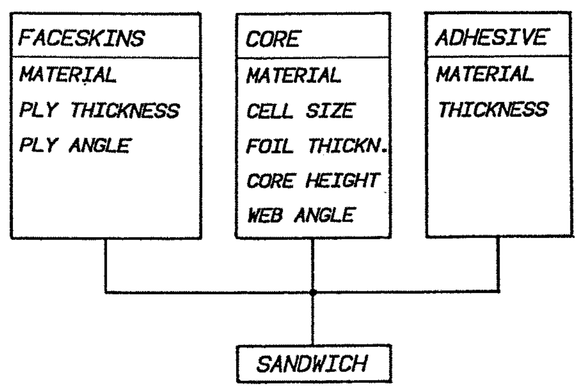


Figure 1. Components of a sandwich plate with faceskins of composite material

[1] Ph.D., Head of structural analysis group

[2] Professor, Head of instute of design and construction methods

Honeycomb and sandwich models

Analytical honeycomb model

To determine the in-plane stiffness of the honeycomb, first of all, a simple analytical model on the basis of the classical beam theory was developed. Because of the symmetry of the honeycomb geometry only a portion of the structure has to be considered. Figure 2 represents the geometry parameters of the honeycomb.

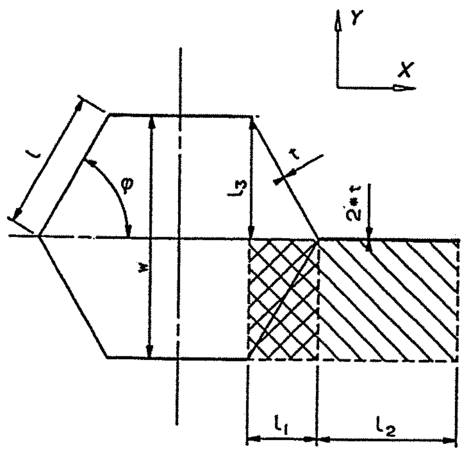


Figure 2. Geometric parameters of the honeycomb

To determine the elastic constants E_x ,E_y ,G_{xy} ,ν_{xy} the problem is reduced to a 2D bending problem. It is assumed that the double thickness webs (2*t) do not bend because of the symmetry of the honeycomb cells. Furthermore the web angle ϕ does not change in the deformed state.

A schematic representation of the analytical model is given in Figure 3.

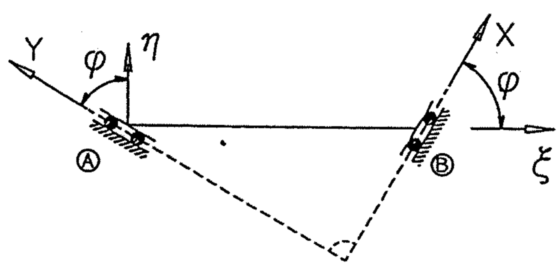


Figure 3. Beam model of the representative portion of the honeycomb

A local cartesian η ,ξ –coordinate system is introduced to formulate the differential equation with respect to the boundary conditions under in-plane loading in x- and y-direction and shear load in the xy-plane.

In the following the procedure to determine E_x is shown. The other constants E_y ,G_{xy} and ν_{xy} are calculated in a similiar way.

Due to the kinematic boundary conditions the displacement u (ξ -direction) and w (η -direction) at supports A and B can be expressed as:

$$w_A = \frac{1}{\tan \varphi} u_A \tag{1}$$

$$w_B = - \tan \varphi \, u_B \tag{2}$$

The force acting in x-direction F_x is reduced with respect to the local coordinate system:

$$F_\eta = - F_x \sin \varphi = N_x \tag{3}$$

$$F_\xi = F_x \cos \varphi = Q_x \tag{4}$$

By integrating the components u and w of the displacement is otained:

$$w = \frac{1}{6} c_1 \xi^3 + \frac{1}{2} c_2 \xi^2 + c_3 \xi + c_4 \tag{5}$$

$$u = c_5 \xi + c_6 \tag{6}$$

Considering the boundary conditions, the constants c_1 to c_6 are determined:

$$c_1 = - \frac{Q_x}{EI} \tag{7}$$

$$c_2 = \frac{1}{2} \frac{Q_x l}{EI} \tag{8}$$

$$c_3 = 0 \tag{9}$$

$$c_4 = - \frac{\frac{N_x l}{EA} \tan \varphi - \frac{1}{12} \frac{Q_x l_0^3}{EI}}{1 + \tan^2 \varphi} \tag{10}$$

$$c_5 = \frac{N_x}{EA} = \varepsilon \tag{11}$$

$$c_6 = \tan \varphi \, c_4 \tag{12}$$

To calculate the x-component of the strain, the displacements must be transformed back to the global xy-coordinate system:

$$U = u \frac{1}{\cos \varphi} \tag{13}$$

$$W = w \frac{1}{\cos \varphi} \tag{14}$$

In addition to the contibution of section 1, the tensile strain of section 2 has to be taken into account:

$$U_{x2} = \frac{F_x}{2\,E\,A}\, l_2 \qquad (15)$$

The equivalent Young's modulus E_x is expressed by

$$E_x = \frac{\sigma_x}{\varepsilon_x} \qquad (16)$$

with

$$\varepsilon_x = \frac{U_{x1}(\xi = l) + U_{x2}}{l_1 + l_2} \qquad (17)$$

and

$$\sigma_x = \frac{F_x}{l_3\ t_c} \qquad (18)$$

With this simple model the behavior of the honeycomb far away and near the faceskins can be investigated. Far away from the faceskins, the single thickness members deform in a cubic shape. However near the faceskins only a tensile deformation of the webs is permitted by the much stiffer faceskins. This restrained state of deformation can be simulated in the analytical model by setting all nonlinear terms in the differential equation to zero (see Figure 4).

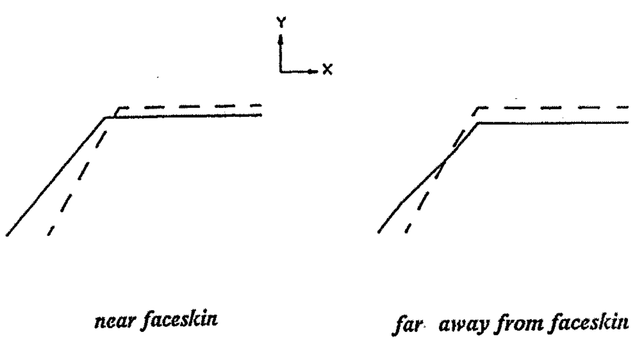


Figure 4. Deformed shapes of honeycomb

Sandwich finite element model

Because the core in-plane stiffness is mainly a geometrical one (i.e. heavily depends on the boundary conditions (faceskin stiffness), and to consider the influence of the core height t_c on the in-plane elastic constants of the honeycomb), a detailed 3D-model of a sandwich was developed.

The symmetry about the xy-plane only requires half of the structure to be idealized while suppressing the corresponding degrees of freedom in z-direction in the symmetry plane. In this model the webs and faceskins are idealized by using thin shell elements.

It should be noted that the stiffness of the adhesive layers are taken into account in that of the faceskins by assuming the adhesive as an additional layer of the faceskin laminate. The concave fillet weld between the webs and the faceskins is modeled by truss element.

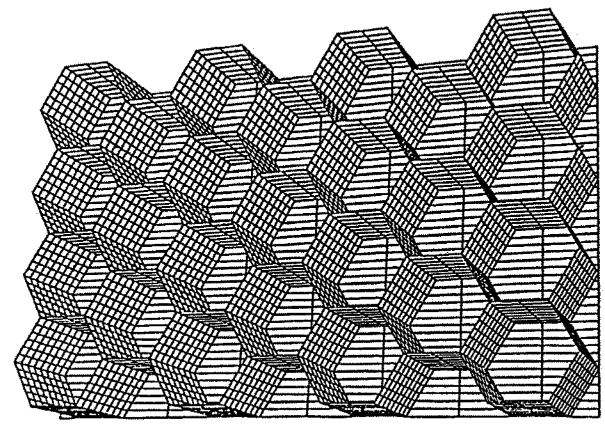


Figure 5. Detailed 3-D finite element model

The postprocessing of the results, i.e. the calculation of the equivalent elastic constants is also done by using a program file routine. The approach is completely the same as for the analytical model.

Results

In Table 1 on page 4 the results are summarized for a honeycomb type 3/16-5056-.001 [3] with a core thickness of $t_c = 1\,mm$ and a rigid faceskins. Regarding the in-plane elastic constants in the xy-plane the influence of the faceskins is obvious. The Young's modulus E_x for the honeycomb near the faceskins is increased by an factor of 3000 compared with the behavior far away from it. This increase of stiffness is forced by the restrained bending deformation of the webs. Also the Poisson's ratio decreases from $\nu_{xy} = 0.99$ to $\nu_{xy} = 0.28$ at the same time. Moreover the Young's modulus in direction of the double thickness webs E_x is about 35 % higher than the corresponding in y-direction for the elastic honeycomb behavior near the faceskins. At this point it can already be stated that there must be a dependency of the core thickness on the in-plane elastic constants.

Also it can be stated that the results of the analytical and the finite element model showed a good correspondence.

[3] 3/16: cell size in fractions of an inch; 5056: type of aluminum alloy; .001: nominal foil thickness in inches

Tests of sandwich specimen measuring the in-plane Young's moduli confirmed the above results within a deviation of 8 %.

Table 1. Elastic constants of honeycomb 3/16-5056-.001		
	far away from faceskin	near faceskin
E_x [N/mm^2]	0.127	415
E_y [N/mm^2]	0.127	267
E_z [N/mm^2]	996	996
ν_{xy} []	0.99	0.29
ν_{xz} []	0.30	0.30
ν_{yz} []	0.30	0.30
G_{xy} [N/mm^2]	0.87	131
G_{xz} [N/mm^2]	192	192
G_{yz} [N/mm^2]	287	287

Referring to this type of honeycomb the influence of the following parameters on the honeycomb elastic constants was quantified:

- faceskin stiffness $E_f\,t_f$
- core height t_c
- angle between webs ϕ

The honeycomb parameters cell size and foil thickness can be summarized in the parameter core weight density. It is clear that the increase of core weight density means a proportional increase of honeycomb stiffness.

Concerning the faceskin stiffness this parameter was changed within technical reasonable limits while keeping the core unchanged. The limits represent the stiffness of CFRP faceskin material available now (from high tensile T-300 to ultra high modulus GY-70 laminate). The faceskins are assumed to have a quasi isotropic lay-up. As a basic result it turned out that the in-plane stiffness constants of the honeycomb with a faceskins stiffness E_f = 260 kN/mm ares about 35 % higher than one with E_f = 20 kN/mm (Figure 6).

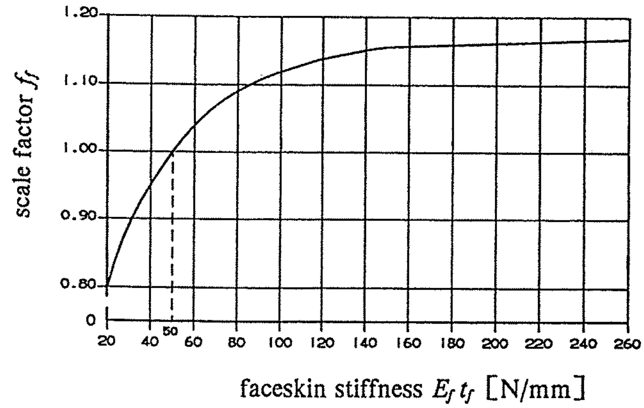


Figure 6. Scale factor concerning faceskin stiffness

Another important influence parameter is the web angle ϕ. In practical work one will never find a perfect honeycomb with ϕ = 60 °. The analysis shows that a change of $\Delta\phi$ = 5° results in a maximum change of 30 % referring to the Poisson's ratio ν_{xy} of the ideal geometry.

To evaluate the effect of the core thickness this parameter was varied within a range 2 mm < t_c < 50 mm while keeping the faceskins unchanged. It turns out that the faceskins only restrain a narrow edge zone of the honeycomb, i.e. only up to a thickness of approximately t_c = 5 mm the honeycomb core fully contributes to the overall (smeared) lateral stiffness of the sandwich. For increasing core thickness the equivalent stiffness contribution of the honeycomb drops as the honeycomb can deform without restrain. Therefore a honeycomb with a thickness t_c = 50 mm only contributes 28 % of the in-plane stiffness of one with a thickness t_c = 5 mm (Figure 7).

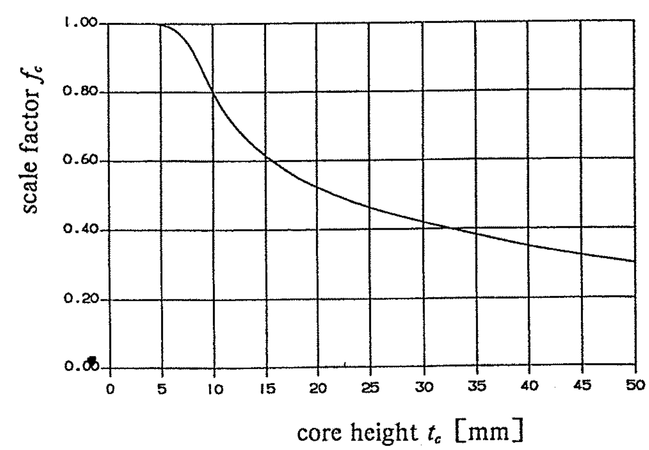


Figure 7. Scale factor concerning core thickness

The deformation of one cell of a honeycomb with t_c = 30 mm is shown in Figure 8 on page 5.

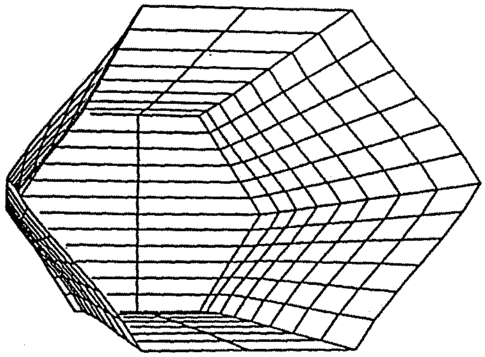


Figure 8. Deformed shape of a 30 mm thick honeycomb

Additional runs loading the sandwich model with a global temperature rise confirmed the CTE's of the sandwich in the xy-plane calculated with laminate theory using the equivalent thermoelastic data of the core. Concerning the resulting CTE of the sandwich in z-direction however, a strong effect of the Poisson's ratio of the honeycomb has to be noticed. It is found that the CTE_z of the sandwich is about 30 % higher than that of the core material. The reason for this effect is the much higher facesheet stiffness (factor 100 - 500) and the less CTE (factor 8 - 30) compared with the core. Under thermal loading the facsheets restrain the in plane expansion of the core where the core has to expand more in the orthogonal direction because of the lateral contraction (Poisson's ratio).

The influence of different parameters on the elastic constants of the honeycomb has been quantified. So it is possible to convert the data of the master core listed in Table 1 on page 4 to any other sandwich configuration.

Prediction of contour accuracy for sandwich panels with parabolic shape

With the knowledge of the detailed set of material constants an improved analysis of sandwich structures can be performed by fe-analysis. Thus a precise prediction of sandwich behavior under such loads as i.e.

- manufacture
- temperature fields
- humidity
- gravity
- acceleration loads

is enabled. Figure 9 shows the element types used for the analysis of a antenna reflector panel with CFRP faceskins and an aluminum honeycomb core. The honeycomb core is modelled by volume elements with orthotropic material law, thus using membrane elements for the faceskins.

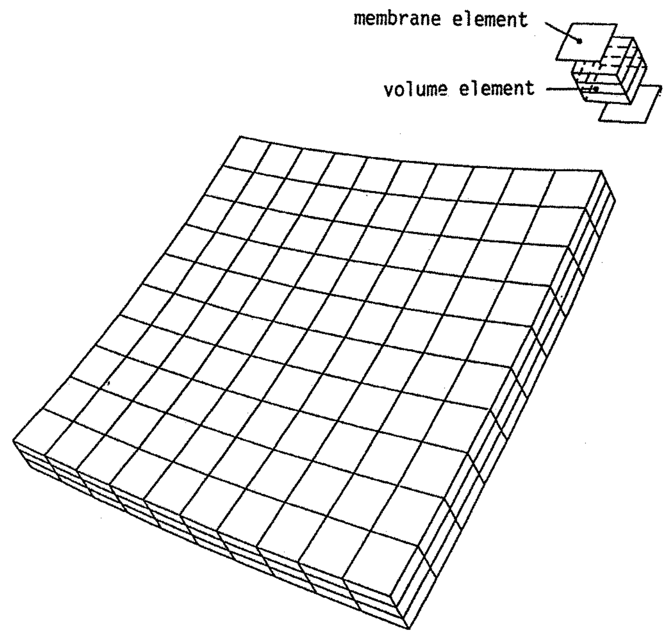
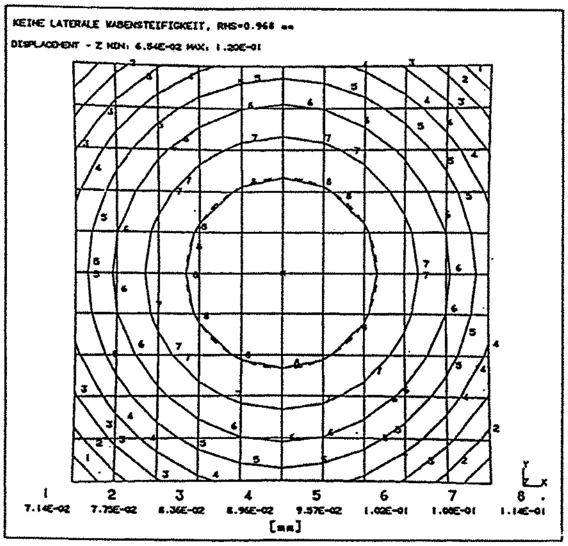


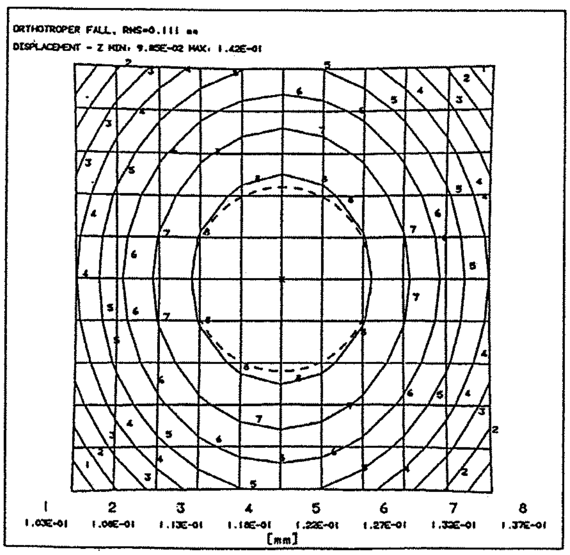
Figure 9. Idealization of a honeycomb sandwich panel

To determine the accuracy of parabolic shaped contour it is required to compare the deformed paraboloid either with the undeformed ideal or with a best fitted paraboloid by changing the paraboloid parameters as location of the apex, orientation of the paraboloid axes or the focal length. So the minimum contour error is obtained. The result is the root mean square of the vertical distances v_i from a point of reference to the deformed contour. To estimate the error, if the in-plane stiffness of the honeycomb core material is neglected, analyses using the above model (see Figure 9) under different thermal loading (global temperature rise of the entire structure, thermal gradient through the thickness of the sandwich) were performed. The essential result is that the neglecting of the core in plane stiffness leads to errors up to 60 % (referring to rms contour accuracy) compared with the real behavior of the core, i.e. having orthotropic in-plane stiffness.

This result can be clarified if the isocontours of displacements in z-direction (in direction of the paraboloid axis) are plotted for both cases (Figure 10 on page 6).



A. no in-plane core stiffness



B. orthotropic in-plane core stiffness

Figure 10. Isocontours of z-displacements for honeycomb core with and without in-plane stiffness

It is obvious that the orthotropic in-plane stiffness of the core changes the undeformed spherical paraboloid to a elliptical in the deformed state. Experimental tests have turned out that the deviation of the calculated and measured thermal deformations of a corresponding sandwich specimen are less than 10 % max.

So it can be concluded that this analysis procedure is a very useful tool to predict the deformation behavior of sandwich components under thermal loading with high accuracy.

References

1. L.J. Gibson, M.F. Asby

   Cellular Solids, Structure and properties, Pergamon Press 1988

2. W. Elspass

   Thermal stable sandwich structures, PhD thesis, Swiss Federal Institute of Technlogy Zurich, 1989