Sandwich panels with cold-formed thin facings

Sandwich Panels with Cold-Formed Thin Facings

Panneaux sandwich à parements en tôle mince formée à froid

Sandwichplatten mit dünnwandigen kaltgeformten Verkleidungen

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SUMMARY

Sandwich panels with thin-walled cold-formed facings and rigid foamed insulating core are becoming more and more popular as enclosures for system buildings. In this paper, the structural behavior, including flexural stresses/deflections, axial stability, and thermal stresses, is presented, summarizing more than a decade of research. Methods used are analytical (boundary-valued approaches), numerical (finite-strip, finite-layer, finite prism approaches) and experimental (full-scale testings). Key equations are formulated, and results by different methods are compared.

RÉSUMÉ

Les panneaux sandwich à parements en tôle d'acier et âme en mousse isolante rigide deviennent de plus en plus utilisés comme enveloppes de bâtiment. Dans cet article est présenté le comportement structural des panneaux sandwich à la flexion, à la compression axiale et sous contraintes thermiques, résumant plus de dix ans de recherche. Les méthodes utilisées sont analytiques, numériques et expérimentales. Les équations déterminantes sont formulées et les résultats des différentes méthodes sont comparés.

ZUSAMMENFASSUNG

Sandwichplatten bestehend aus dünnwandigen kaltgeformten Stahlverkleidungen und einer versteifenden Schaumisolierung werden als Fassadenelemente für Fertigteilbauten immer beliebter. In dieser Arbeit wird das strukturelle Verhalten solcher Platten unter Biegebeanspruchung, Axialstabilität und Temperaturbeanspruchung dargestellt. Damit wird mehr als ein Jahrzehnt der Forschung auf diesem Gebiet zusammengefasst. Die verwendeten Untersuchungen basieren auf analytischen (Grenzwertanalysen), numerischen (finite Streifen-, finite Schichten-, finite Prismen-Analyse) und experimentellen (ausführliche Tests) Methoden. Die hauptsächlichen Gleichungen werden angegeben und die Resultate der verschiedenen Methoden verglichen.

1. INTRODUCTION

Sandwich construction has been widely applied in aircraft and structural engineering since before the Second World War. The structural analysis of sandwich panels with thin flat facings has been investigated as early as the 1940's, particularly for aeronautical applications[1-3]. However, research and development of architectural sandwich panels with formed facings (Fig. 1) began only in the early 1970's, pioneered by Chong and his associates. These panels are becoming popular due to their superior structural efficiency, mass productivity, insulation qualities, transportability, fast erectability, prefabricatability, durability, and reusability. The formed facings serve two purposes: architectural appearance and structural stiffness. In 1972, Chong and Hartsock [4,5] presented a method to predict the localized wrinkling instability of such panels. Subsequently, Chong and his associates have investigated flexural behavior [6,7,8] and thermal stresses [9,10,11] for both simple and continuous spans, axial stability[8,12,13] and vibration [14]. The classical references for flat-faced sandwich panels are given by Allen [1] and Kuenzi [15], and ASCE conducts regular literature surveys [16] including sandwich panels under the composite construction heading.

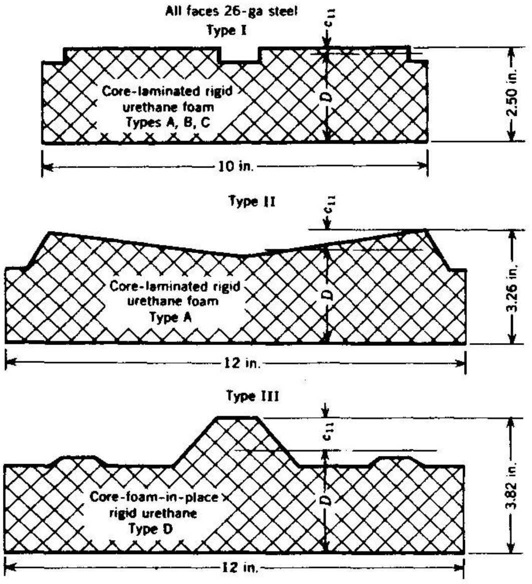

Fig. 1. Panel Geometry [5]
(Note: 1 in = 25.4 mm)

2. FLEXURAL BEHAVIOR

The flexural behavior of sandwich panels subject to bending is investigated analytically [7,17], experimentally [7], and numerically [8].

2.1 Analytical Studies

The basic assumptions are: Young's moduli of the faces are large compared to that of the core; and adequate adhesion exists between the faces and the core. The total moment, M, is equal to the sum of the moment due to composite action, M_c, and the moment due to bending of the faces about their own centroidal axis, M_o. Incorporating flexural and shear deformations, the resulting governing differential equation is:

$$\frac{d^4y}{dx^4} - C_y \frac{d^2y}{dx^2} = -\frac{C_y}{E_1 I} M + \frac{dV}{dx}\frac{1}{E_1 I_o} \qquad (1)$$

where, C_y = panel properties [7]; E_1 = Young's modulus of face 1; I, I_o = composite moment of inertia and moment of inertia of faces, respectively, normalized with respect to E_1; V = shear force. Integrating twice,

$$\frac{d^2y}{dx^2} - C_y y = -\frac{C_y}{E_1 I}\iint M dx^2 + \frac{M}{E_1 I_o} + C_1' x + C_o \qquad (3)$$

C_1' and C_o can be determined from boundary conditions [7,17]. In actual cases, M is a known function and Eq. 3 can be put in a generalized form:

$$\frac{d^2y}{dx^2} - C_y y = C_4 x^4 + C_3 x^3 + C_2 x^2 + C_1 x + C_o \qquad (4)$$

The solution is:

$$y = D_4 x^4 + D_3 x^3 + D_2 x^2 + D_1 x + D_o + F_1 e^{\sqrt{C_y}x} + F_2 e^{-\sqrt{C_y}x} \qquad (5)$$

To find the coefficients, F_1 and F_2, boundary conditions [6,7] are used; e.g.

for simple spans under symmetrical loads: at $x = \frac{L}{2}$: $\frac{dy}{dx} = 0$; and at $x = 0$: $y = 0$; in which L denotes the length of the span. The results are

$$F_2 = \frac{D_4 \frac{L^3}{2} + 3D_3 \frac{L^2}{4} + D_2 L + D_1 - D_o C_y^{1/2} e^{\sqrt{C_y}L/2}}{C_y^{1/2}(e^{\sqrt{C_y}L/2} + e^{-\sqrt{C_y}L/2})} \tag{6}$$

$$F_1 = -D_o - F_2 \tag{7}$$

D_o, D_1, D_2, D_3, and D_4 can be expressed in terms of C_y, C_o, C_1, C_2, C_3, and C_4. For the stress in face 1

$$S_{11} = \frac{-(M - E_1 I_o \frac{d^2y}{dx^2})}{a_1 D} - E_1 c_{11} \frac{d^2y}{dx^2} \tag{8}$$

where, a_1 = cross-sectional area of Face 1; c_{11} = distance from neutral axis of face 1 to outside fiber; D = distance between neutral axes of faces 1 and 2, and

$$\frac{d^2y}{dx^2} = 12D_4 x^2 + 6D_3 x + 2D_2 + C_y(F_1 e^{\sqrt{C_y}x} + F_2 e^{-\sqrt{C_y}x}) \tag{9}$$

Stress at other locations in the profile of face 1 can be found similarly using other values in place of c_{11}. For the shear stress carried by the core

$$S_{sc} = (V - E_1 I_o \frac{d^3y}{dx^3})/(bD) \tag{10}$$

in which $$\frac{d^3y}{dx^3} = 24 D_4 x + 6D_3 + C_y^{1.5}(F_1 e^{\sqrt{C_y}x} - F_2 e^{-\sqrt{C_y}x}) \tag{11}$$

b = width of the panel.

2.2. Experimental

The two test panel profiles (Types I and III) are shown in Fig. 1. Strain gages were mounted at the highest parts of the profiles (No. 1) at sixth, third, and midspan. At midspan, gages were also located at the bottom of the grooves of the laminated panels and at different elevations of the foam-in-place panel (gages Nos. 2 and 3). The panels were tested in a suction box. The 1,220 mm long samples were mounted on 25.4 mm diameter pipe supports with a 1,140 mm span. The formed faces were facing up. The box was covered with a 0.15 mm polyethylene film which was taped to the sides of the box. Deflection was measured by means of dial indicators located over one support and at sixth, third, and midspan. Suction was applied by a vacuum pump through a hole in one side of the box and controlled by means of a small damper. Pressure was measured with a water manometer.

2.3. Theoretical vs.Experimental Results

Theoretical computations were made on a computer. The properties of the faces were calculated from conventional methods assuming the full cross-sectional areas of the faces were effective. The core thickness, D_c, applies more directly to panels with thick faces rather than to formed faces of light gage sheet metal. Therefore, the mean core thickness was used. Computed deflections are in good agreement with the experimental values. The moduli are reasonable, and the shapes of the computed deflection/span curves are in good agreement with the experimental. In conclusion, within the buckling loads, the methods outlined herein provide a reasonably accurate method of calculating the deflections and stresses in sandwich panels with formed faces. There is good agreement between experimental and theoretical values. The theory is applicable to combinations of formed and flat faces, and to faces of different or the same material, subjected to flexural loading.

2.4. Numerical Method

The finite strip method and the finite prism method are especially efficient in analyzing prismatic members such as these architectural sandwich panels, whereas it would be prohibitive in cost and time to apply the finite element method [18]. The pioneering papers on the finite strip method were written by Cheung on plate bending problems [19,20] and had been applied in the analysis of flat-faced sandwich plàte [21]. The finite prism method was described by Cheung, et al. [22]. The various applications of the finite strip method, the finite prism method, and in addition to the finite layer method are discussed comprehensively in a paper by Cheung [23].

In this article, the finite prism method, in combination with the finite strip method, are used. Results are compared with those obtained analytically by Hartsock and Chong [7] for flexural bending. The general equilibrium matrix equation is

$$S\delta = F \tag{12}$$

where, S = stiffness matrix; F = load vector; and δ = displacement vector.

2.4.1 Finite-Strip-Prism Model for Sandwich Construction

Thin faces are approximated by the finite shell strip, whereas the weak core is modelled by the finite prism. The detail formulation for the above matrices is given in texts [21], and only germane points are outlined here.

2.4.1.1. Finite Strip for Modelling the Thin Face

Both in-plane and bending action of the strip are considered, and the shape functions for the simply supported strip (Fig. 2) are given as

$$u = \sum_{m=1}^{r} [1 - \bar{x}, \bar{x}] \begin{bmatrix} u_1 \\ u_2 \end{bmatrix}_m \sin \frac{m\pi y}{L} \tag{13}$$

$$v = \sum_{m=1}^{r} [1 - \bar{x}, \bar{x}] \begin{bmatrix} v_1 \\ v_2 \end{bmatrix}_m \cos \frac{m\pi y}{L} \tag{14}$$

$$w = \sum_{m=1}^{r} [(1 - 3\bar{x}^2 + 2\bar{x}^3), (x - 2\bar{x}x + \bar{x}^2 x), (3\bar{x}^2 - 2\bar{x}^3), (\bar{x}^2 x - \bar{x}x)] \begin{bmatrix} w_1 \\ \theta_1 \\ w_2 \\ \theta_2 \end{bmatrix} \sin \frac{m\pi y}{L} \tag{15}$$

where $\bar{x}$ = x/b and b, L are the width and length of the strip, respectively.

In the linear analysis, the in-plane and bending terms are decoupled, and the stiffness matrices for each action can be formed separately. The in-plane and bending stiffness matrices of the bending strip are given by Cheung [21].

2.4.1.2 Finite Prism Modelling the Weak Core

Referring to Fig. 2, it can be seen that a suitable set of displacement functions for a straight prism is

$$u = \sum_{m=1}^{r} C_k\, U_{km} \sin \frac{m\pi y}{L} \qquad (16)$$

$$v = \sum_{m=1}^{r} C_k\, V_{km} \cos \frac{m\pi y}{L} \qquad (17)$$

$$w = \sum_{m=1} C_k\, W_{km} \sin \frac{m\pi y}{L} \qquad (18)$$

where C_k is the shape function of the isoparametric element (Fig. 3) for interpolation in the x-z direction. The shape functions for the isoparametric quadrilateral with 4 corner nodes (ISW4 model [8]) are

$$C_i = \frac{1}{4}(1 + \eta_i\eta)(1 + \xi_i\xi) \qquad (19)$$

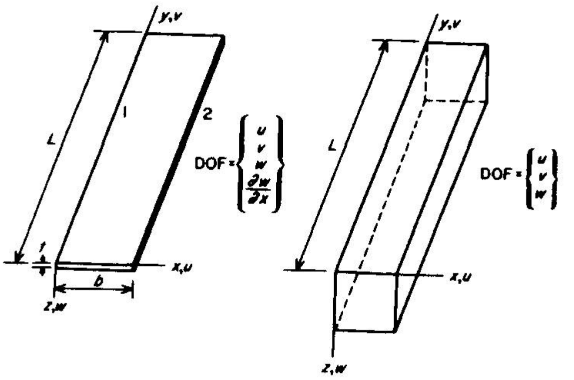

Fig. 2. Rectangular strip and prism [8]

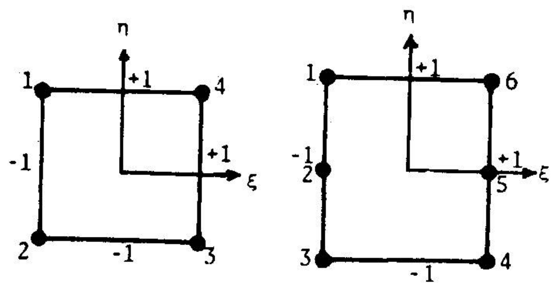

Fig. 3. ISW4 and ISW6 models [8]

The shape functions for the isoparametric quadrilateral with 6 nodes (ISW6 model [8]) are

$$\text{Corner Nodes:} \quad C_i = \frac{1}{4}\eta_i\eta(1 + \eta_i\eta)(1 + \xi_i\xi) \qquad (20)$$

$$\text{Mid-Side Nodes:} \quad C_i = \frac{1}{2}(1 + \xi_i\xi)(1 - \eta^2) \qquad (21)$$

ξ and η are localized coordinates of x and z. The stiffness matrix is given as

$$^pS_{ijmn} = \int {}^pB^T_{im}\, {}^pD\, {}^pB_{jn}\, d(vol) \qquad (22)$$

where $^pB_{im}$ = the gradient matrix [8]

and ^pD = the elasticity matrix for isotropic materials. (23)

The geometric transformation from the natural coordinate (x-z) to the local coordinate (ξ-η) can be carried out as in the standard finite element method, and the stiffness matrix can be obtained accordingly.

2.4.1.3 Coupling of the Thin Face and the Weak Core

The thin face is assumed to be in contact with the weak core throughout the loading history; hence, the stiffness matrices can be assembled easily. With the above choice of harmonic series, the stiffness matrices are zero for unequal m and n, and they are decoupled. The static equilibrium equation is

$$^sS^p_{mm}\, {}^s\delta^p_m + {}^sS^B_{mm}\, {}^s\delta^B_m + {}^pS_{mm}\, {}^p\delta_m = F_m \qquad (24)$$

where ${}^{s}S^{P}_{mm}$ is the in-plane stiffness matrix of the bending strip; ${}^{s}S^{B}_{mm}$ is the bending stiffness matrix of the bending strip; ${}^{p}S_{mm}$ is the stiffness matrix of the finite prism; ${}^{s}\delta^{P}_{m}$ and ${}^{s}\delta^{B}_{m}$ are the in-plane and bending interpolating parameters of the bending strip; ${}^{p}\delta_{m}$ is the interpolating parameters of the finite prism; δ_m is the total interpolating parameters; F_m is the loading vector.

The superscript s indicates strip variables and p , those of the finite prism. Also, the load vector (F_m) of the out-of-plane loading (q is the load intensity) can be expressed in terms of the displacement parameters of the prism, that is

$$F_m = \int q\, C_k\, W_{km} \sin \frac{m\pi y}{L}\, d(vol) \tag{25}$$

2.4.1.4. Numerical Examples

(a) In Fig. 4, the depth of corrugation of face 1, Y, is a variable to investigate the influence due to the degree of forming. The properties of this panel are: Young's modulus of thin faces = 1000.0 units; Poisson ratio of thin faces = 0.0; thickness of thin faces= 0.02 units; and Young's modulus of core = 1.0 units. The ISW6 model is used in the analysis, and the results for bending are depicted in Fig. 5. The results compared favorably with Hartsock and Chong's analytical solutions [7].

(b) A commercial type of formed-face sandwich panel (Type II, Fig. 1) is investigated. The properties and dimensions are:
Properties of Core: E_c = 4685 kN/m^2; ν_c = 0.177.
Properties of Top and Bottom Faces: E_f = 203,550 MPa; ν_f = 0.267; t_f = 0.51 mm.
Span = 1143 mm; Loading = 4.79 kN/m^2
The midpoint deflection (4.47 mm) is compared with Hartsock and Chong's analytical solution [7] of 4.55 mm, which are very close.
The formed face sandwich panel has been modelled by the finite-prism-strip model. Such method saves considerable computing or experimental cost and effort and can easily be applied for the analysis of formed face panel. The ISW6 model is more accurate than that of the ISW4 and is recommended for such analysis.

3. AXIAL STABILITY

3.1 Numerical Analysis

The buckling of sandwich panels due to axial compression is investigated using the finite-strip-prism model as described in Article 2.4. Buckling is usually preceded by wrinkling instability [4]. Similar to Eq. 12, the equilibrium matrix equation for a stable loaded system is

$$S\delta + G\delta = F \tag{26}$$

where G = geometric stiffness matrix.

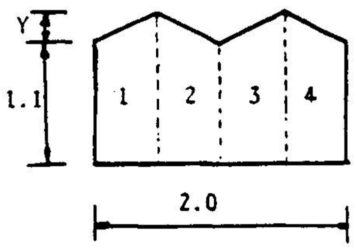

Fig. 4. Mesh details [8]

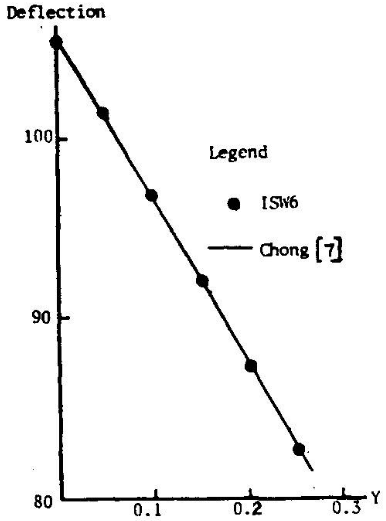

Fig. 5. Deflection vs. Y [8]

The stability analysis of the system is equivalent to finding a set of nontrivial solutions to the set of homogeneous equations (a generalized eigen-value problem) of the form

$$(S + \lambda G^*)\delta = 0 \qquad (27)$$

where G^* is the geometric stiffness matrix corresponding to the fixed preimposed stress distribution ${}^o\sigma_{ij}$. The finite-strip-prism model has been described in Eqs. 13-22.

In the stability analysis of strut, it is obvious that instability can only arise due to the action of axial stress ${}^o\sigma_{yy}$. The geometric stiffness of the strip element can then be shown to be [8]

$$G^*_{mn} = \begin{bmatrix} G^P_{mn} & 0 \\ 0 & G^B_{mn} \end{bmatrix} \qquad (28)$$

in which, $$G^P_{mn} = t\,{}^o\sigma_{yy} \int g^T_{um}\, g_{un}\, d(\text{area}) + t\,{}^o\sigma_{yy} \int g^T_{vm}\, g_{vn}\, d\,(\text{area}) \qquad (29)$$

$$G^B_{mn} = t\,{}^o\sigma_{yy} \int g^T_{wm}\, g_{wn}\, d(\text{area}) \qquad (30)$$

where g_u, g_v, and g_w are slope matrices associated with displacement variables u, v, and w, respectively, and t is the thickness of the strip. They are,

$$g_{um} = [(1 - \bar{x})\,\frac{m\pi}{L} \cos \frac{m\pi y}{L} \;,\quad \bar{x}\,\frac{m\pi}{L} \cos \frac{m\pi y}{L}] \qquad (31)$$

$$g_{vm} = [-(1 - \bar{x})\,\frac{m\pi}{L} \sin \frac{m\pi y}{L} \;,\quad -\bar{x}\,\frac{m\pi}{L} \sin \frac{m\pi y}{L}] \qquad (32)$$

$$g_{wm} = [(1 - 3\bar{x} + 2\bar{x}^3),\ (x - 2\bar{x}x + x^2\bar{x}),\ (3\bar{x}^2 - 2\bar{x}^3),\ (\bar{x}^3 x - \bar{x}x)]\,\frac{m\pi}{L} \cos \frac{m\pi y}{L} \qquad (33)$$

Coupling the faces and the core, the equilibrium equation for the stability analysis is [8]

$${}^sS^p_{mm}\;{}^s\delta^p_m + {}^sS^B_{mm}\;{}^s\delta^B_m + {}^pS_{mm}\;{}^p\delta_m + \lambda\, G_{mm}\,\delta_m = 0 \qquad (34)$$

where G_{mm} = global geometric stiffness matrix; and the rest of matrices have been defined in Eq. 24.

3.2 Numerical Examples

(a) Panel is the same as Example (a) in Article 2.4.1.4. ISW6 model was used, and it compared favorably with solution presented in Allen's book [1]. Results are shown in Fig. 6.

(b) Panel is the same as Example (b) in Article 2.4.1.4. The ISW6 model gives a buckling load of 12.07 kip (53.71 kN).

Again, it is found that the ISW6 model is more accurate than the ISW4 model. For flat-faced

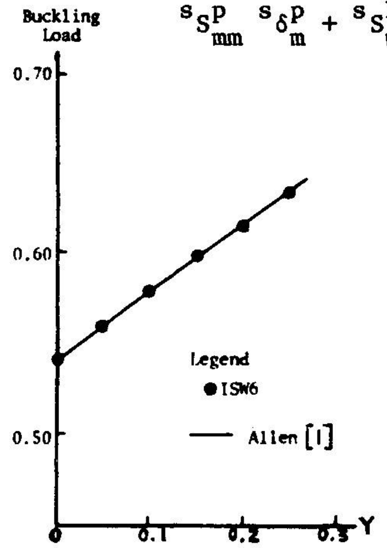

Fig. 6. Buckling load vs. Y.

sandwich panels, the ISW4 model gives satisfactory results.

3.3 Cores with Variable Stiffnesses

Due to uneven curing or other reasons, the sandwich cores may have variable stiffnesses [12] throughout the thickness. In this article, sandwich panels with flat faces and variable core stiffnesses are investigated by the finite layer method [12,21,24]. The formulation is similar to Eqs. 26 - 34.

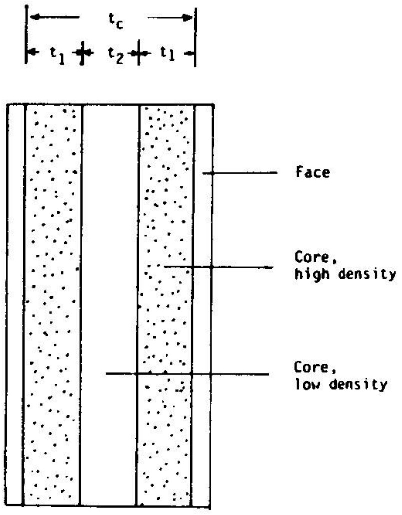

Fig. 7. Variable core densities

A typical sandwich rectangular plate is divided into a number of layers (Fig. 7) which are simply supported at boundaries. The faces of the sandwich plate are assumed to be isotropic and equal in thickness. The stiffness of the core follows the distribution assumption

$$E_c t_c = \sum E_i t_i \qquad (35)$$

where E_c is the mean Young's modulus of the core, t_c is the thickness of the core, E_i is the Young's modulus of each layer of the core, and t_i is the thickness of each layer of the core.
The buckling loads of a sandwich square plate with homogeneous core material are studied. The core is polyurethane with a Poisson's ratio of 0.25, and the Young's modulus is 5175 kNm^{-2}, with a density of about 40 kg m^{-3}. The Poisson's ratio and Young's modulus of each steel face are 0.3 and 207 GPa, respectively. Thicknesses of each face and core are 0.508 mm and 12.7 mm, respectively. All of the sandwich plates are divided into five layers (Fig. 7). The buckling load for $L_x = L_y = 51$ cm is 1.31 kN as compared with Allen's result [1] of 1.34 kN.

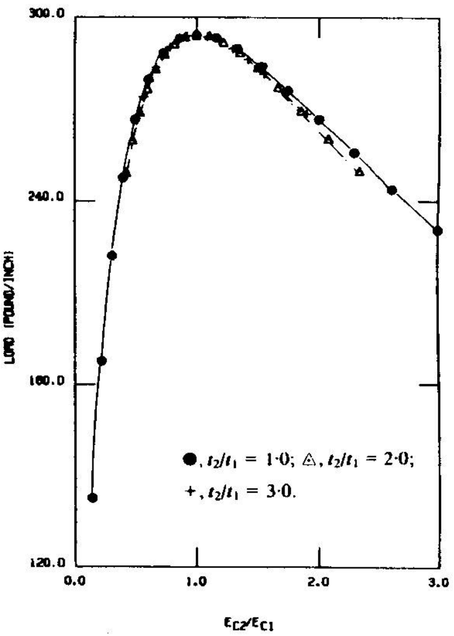

Fig. 8. Buckling load of short panels

For variable core density of square sandwich plate, the results are given in Fig. 8 in which the mean Young's modulus is 5175kNm^{-2}. Similar results are obtained for longer panels, 3 m long, 1 m wide, and 7.7 cm thick.

For variable stiffness of the core of sandwich plates, the optimal results are when the stiffness ratios of the outer layers to the inner layers are equal to 1, which means the core material is homogeneous through the thickness of the core. Thus, it is best to have a homogeneous core in any fabrication process. If the core is cured improperly, the outer layers tend to be different in stiffness compared with the inner layers. The strong outer layer or strong inner layer will reduce the strength of the plate. This article presented quantitative data on the buckling strength due to uneven core stiffnesses. Qualitatively, it can be seen if any part of the core is weakened or strengthened at the expense of the overall core stiffness (Eq. 35), then the buckling strength is lowered. Intuitively, efficient sandwich columns should have stiff faces and weak cores such that the radii of gyration (by the transformed area concept) are maximized.

4. THERMAL STRESSES AND DEFLECTIONS

Due to the superior insulation quality of sandwich panels, they have been used in places of extreme climates. It is not unusual that the temperature difference between the inside and outside wall surfaces may exceed 100°F (55.6°C). Due to the flexural rigidity of cold-formed facings, thermal stresses [10] are present even in simple span conditions.

In this investigation, a series of experiments was conducted to study the stresses and deflection induced when a sandwich panel was exposed to a temperature gradient between the two faces. The sandwich panels were tested for both single and two-span conditions, with the temperature difference between the two faces reaching up to 55.6°C (100°F).

Formulated as an ordinary fourth-order differential equation with suitable boundary conditions, theoretical expressions [10] were derived for deflection, flexural stresses in the facings, and shear stresses in the core. Numerically, finite-prism-strip method [11] is used. Experimental data, numerical analysis, and theoretical predictions are found in reasonable agreement. Due to the lack of space, details of this Article are omitted.

5. CONCLUSIONS

Structural behavior of sandwich panels with cold-formed facings is investigated analytically, experimentally, and numerically. Close agreements among these independent methods show that the results are reliable. Since experiments usually consume much time, theoretical (analytical) and/or numerical analyses are preferred for the designing and optimization of such panels. If necessary, confirmatory tests can be performed.

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