COMPRESSIVE BUCKLING OF FLAT RECTANGULAR METALITE TYPE
SANDWICH PLATES WITH SIMPLY SUPPORTED LOADED
EDGES AND CLAMPED UNLOADED EDGES

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A theoretical solution is obtained for the problem of the compressive buckling of flat rectangular Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges. The solution is based upon the general small-deflection theory for flat sandwich plates developed in NACA TN No. 1526. Good agreement is found between the present results and those of Forest Products Laboratory Rep. No. 1583.

A comparison of computed and experimental buckling stresses of sandwich plates with balsa-wood cores and with cellular-cellulose-acetate cores indicates reasonable agreement between theoretical and experimental results.

INTRODUCTION

The increasing use of sandwich materials as a substitute for the more conventional skin-stringer construction in aircraft design makes the problem of analyzing sandwich plates one of great importance. Since sandwich plates cannot be analyzed by ordinary plate theory because of the appreciable effect of low core shear stiffness on deflections, a general small-deflection theory for elastic bending and buckling of flat sandwich plates was developed in reference 1. This theory was extended to include plastic buckling in reference 2 and was applied to the problem of the elastic and plastic compressive buckling of simply supported flat rectangular Metalite type sandwich plates.

In the present paper the elastic compressive buckling of flat rectangular Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges (fig. 1) is investigated. The differential equations of reference 1 are solved to yield a stability criterion giving the elastic-compressive-buckling coefficient implicitly in terms of the plate aspect ratio and the ratio of the plate flexural stiffness to the core shear stiffness. Charts are presented to facilitate the determination of elastic-compressive-buckling loads and an approximate correction for plasticity is outlined.
The results of the present paper are found to be in good agreement with those of the approximate theory of reference 3. The difference between the computed stresses is at most 5 percent, the results of reference 3 being higher. The difference decreases as the core shear stiffness decreases.

A comparison of computed and experimental buckling stresses of sandwich plates with balsa—wood cores and with cellular—cellulose—acetate cores indicates reasonable agreement between theoretical and experimental results.

**SYMBOLS**

- $E_f$: Young's modulus for face material
- $\mu_f$: Poisson's ratio for face material
- $t_f$: face thickness
- $G_c$: shear modulus for core material
- $h_c$: core thickness
- $D$: flexural stiffness per unit width of Metalite type sandwich plate
  \[ D = \left( \frac{E_f t_f (h_c + t_f)^2}{2(1 - \mu_f^2)} \right) \]
- $a$: plate length
- $b$: plate width
- $\beta$: plate aspect ratio $(a/b)$
- $r$: core shear—flexibility coefficient
  \[ r = \left( \frac{\pi^2 D}{b^2 G_c h_c} \right) \]
- $\sigma_{cr}$: critical compressive stress in x-direction
- $k$: elastic—buckling—stress coefficient
  \[ k = \left( \frac{2b^2 \sigma_{cr} t_f}{\pi^2 D} \right) \]
- $N_x$: critical compressive load per unit width
  \[ N_x = 2\sigma_{cr} t_f \]
- $x, y$: coordinate axes (see fig. 1)
- $w$: deflection of middle surface of plate
m number of half waves in buckled-plate deflection surface
in direction of loading

\[
\frac{Q_x}{G_{chc}}, \frac{Q_y}{G_{chc}}
\]
angles between lines originally perpendicular to undeformed middle surface and lines perpendicular to deformed middle surface

Subscripts:

comp computed
exp experimental

RESULTS AND DISCUSSION

The solution of the problem of the compressive buckling of flat rectangular Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges (fig. 1) is obtained herein by means of the differential equations of deformation and equilibrium derived in reference 1. Details of the solution are given in the appendix.

**Stability criterion and buckling curves.**—The stability criterion (equation (All)) derived in the appendix gives the elastic-buckling-stress coefficient \( k \) implicitly in terms of the plate aspect ratio \( \beta \) and the core shear-flexibility coefficient \( r \). Unlike results obtained for isotropic plates with deflections due to shear neglected, the buckling coefficients of Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges depend on Poisson's ratio for the face material.

Solutions of the stability criterion for Poisson's ratio equal to 1/3 are presented in figure 2. The elastic-buckling-stress coefficient is plotted against the plate aspect ratio for different values of the core shear-flexibility coefficient. As the core shear stiffness decreases, the decreasing wave length of buckle lessens the effect of the clamped unloaded edges on the plate buckling strength and the buckling curves approach the curves obtained in reference 2 for plates simply supported on all edges. This phenomenon also occurs as the aspect ratio of the plate decreases. When the core shear-flexibility coefficient is equal to or greater than unity, the wave length of buckle is infinitely small. In this case, as for simply supported Metalite type sandwich plates (reference 2), the buckling-stress coefficient is determined by the shear modulus of the core and is given by

\[
k = \frac{1}{r}
\] (1)
This last result is a consequence of the assumption, implied by the theory of reference 1, that the plate faces are so thin that they can be treated as membranes having a negligible stiffness in bending about their own middle surface. If the flexural stiffness of the faces were taken into account, the wave length of buckle would not become infinitely small. The buckling-stress coefficients given by equation (1), however, hardly differ from those of a more exact theory, for plates having practical dimensions.

In figure 3 the compressive-buckling coefficients of infinitely long Metalite type sandwich plates with clamped unloaded edges \((\mu_r = \frac{1}{3})\) are compared with the buckling coefficients of infinitely long isotropic sandwich plates with simply supported edges. As was noted in the discussion of figure 2, the buckling coefficients of the clamped plates approach those of the simply supported plate as the core shear-flexibility coefficient increases, the two being equal for values of \(r\) greater than unity.

**Comparison with approximate solution.**—The results of the more approximate theory of reference 3 agree very well with those of the present paper. Buckling-stress coefficients computed from equations (38) to (45) of reference 3 or from equations (1), (8), and (9) of reference 4 are at the most 5 percent higher than those obtained from the curves of figure 2, the error decreasing with decreasing core shear stiffness. The approximate stability equation of references 3 and 4 may be written for Metalite type sandwich plates, when the notation of the present paper is used, as

\[
k = \frac{16 \left( \frac{\beta^2}{3 \, m^2} + \frac{1}{2} + \frac{3}{16} \frac{m^2}{\beta^2} \right)}{1 + \frac{r}{1 + \frac{4}{3} \, \frac{\beta^2}{m^2}} \frac{16 \left( \frac{\beta^2}{m^2} + \frac{1}{2} + \frac{3}{16} \frac{m^2}{\beta^2} \right)}{1 + \frac{4}{3} \, \frac{\beta^2}{m^2}}}
\]

(2)

Reference 2 indicates that the theory of reference 3 was equivalent to that of reference 1 for the problem of the compressive buckling of simply supported plates. The results of the present paper indicate further that the simplifying assumption of reference 3 applies with little error in problems involving other support conditions. This assumption states that any line in the sandwich core that is initially straight and normal to the middle surface of the core will remain straight after deformation of the panel but will deviate from the direction of the normal to the deformed middle surface by an amount that is proportional to the slope of the plate surface, the proportionality factor being the same throughout the plate.
Correction for plasticity.— Because of the complexity of the stability criterion and the number of parameters involved, no attempt was made to extend the solution to include buckling in the plastic range. An approximate correction for plasticity is suggested by the results of references 5 and 6 from which, for long plates with edges elastically restrained against rotation, the ratio of the plastic buckling stress to the elastic buckling stress can be seen to be approximately independent of the magnitude of the elastic restraint. When the results of reference 2 for simply supported plates are used, curves of plastic buckling stress plotted against elastic buckling stress may be obtained for various values of plate aspect ratio and core shear-stiffness parameter. The appropriate curve is then entered with the elastic buckling stress obtained by means of figure 2 to get the approximate buckling stress of a plate with simply supported loaded edges and clamped unloaded edges. The curves for infinitely long plates can be used with little error for plates having any aspect ratio.

The results of this method agree closely with those obtained by using the procedure suggested in references 3 and 4: that the elastic modulus be replaced by a reduced modulus everywhere it appears in equation (2).

Comparison of theory and experiment.— In figures 4 and 5 experimental compressive buckling stresses are compared with the buckling stresses computed from the results of the present paper. The experimental stresses are the results of Forest Products Laboratory tests made on sandwich plates with Alclad 24S-T aluminum-alloy faces and end-grain balsa-wood or cellular-cellulose-acetate cores. (See reference 4.) Theoretical stresses in the plastic range are approximate and were obtained by the method described in the previous section. The experimental and computed data are summarized in tables 1 and 2.

Much better agreement exists between theoretical and experimental results for panels with cellular-cellulose-acetate cores than for panels with end-grain balsa-wood cores. (See figs. 4 and 5.) The average discrepancies between theory and experiment for the two types of panels are 5.6 percent and 28.2 percent, respectively.

An explanation for the apparent different behavior of the two types of panels can be found from an examination of the data of tables 1 and 2. A comparison of computed and semiparametrically determined flexural stiffnesses (columns 6 and 13 of table 1 and columns 6 and 11 of table 2) indicates again good agreement for panels with cellular-cellulose-acetate cores and poor agreement for panels with end-grain balsa-wood cores. The semiparametric values of plate flexural stiffness given in reference 4 were computed from the results of tests of sandwich beams cut from the panels. The effective stiffnesses obtained from these tests were corrected, in accordance with the procedure outlined in reference 7 —
a procedure which involves a knowledge of the core shear modulus - to
obtain the flexural stiffnesses listed in column 13 of table 1 and
column 11 of table 2.

In view of the agreement between theoretical and experimental
results for sandwich plates with cellular-cellulose-acetate cores, it
seems reasonable to expect the same agreement for panels with balsa-wood
cores. The semiempirically determined flexural stiffnesses for panels
with balsa-wood cores are, therefore, very likely incorrect. The Forest
Products Laboratory has suggested that the shear modulus assumed for
balsa wood is inaccurate. A lower shear modulus would give good agree-
ment between computed and semiempirically determined flexural stiff-
nesses for the panels with end-grain balsa-wood cores. A lower shear
modulus would also increase the core shear-flexibility coefficients of
the panels so that the computed values of the buckling stresses would
be low enough to agree fairly well with the observed stresses. The
required shear modulus is of the order of 6,000 psi to 9,000 psi, values
which are by no means unusual for balsa wood. (See reference 8.) With
this explanation in mind, reasonable agreement apparently exists between
theoretical and experimental results.

CONCLUDING REMARKS

Charts have been presented to facilitate the determination of
theoretical elastic-compressive-buckling loads of flat rectangular
Metalite type sandwich plates with simply supported loaded edges and
clamped unloaded edges. A correction for plasticity has been suggested.

Reasonable agreement between theoretical and experimental results
is indicated by a comparison of computed and experimental buckling
stresses of sandwich plates with balsa-wood cores and with cellular-
cellulose-acetate cores.

Langley Aeronautical Laboratory
National Advisory Committee for Aeronautics
Langley Air Force Base, Va., March 21, 1949
APPENDIX

DERIVATION OF COMPRESSION BUCKLING CRITERION FOR FLAT
RECTANGULAR METALITE TYPE SANDWICH PLATES WITH
SIMPLY SUPPORTED LOADED EDGES AND CLAMPED
UNLOADED EDGES

Differential equations.— Differential equations for sandwich plates
that may be used to derive the buckling criterion are given on the bottom
of page 13 of reference 1. Seven physical constants (two Poisson's
ratios, two flexural stiffnesses, a twisting stiffness, and two shear
stiffnesses) which must be specified are given in reference 2 for
Metalite type sandwich plates as

\[
\begin{align*}
\mu_x &= \mu_y = \mu_f \\
D_x &= D_y = (1 + \mu_f)D_{xy} = \frac{1}{2} Ef t_f (h_c + t_f)^2 \\
D q_x &= D q_y = G_c h_c
\end{align*}
\]

(A1)

For a Metalite type sandwich plate compressed in the x-direction, the
equations of reference 1 are then

\[
\begin{align*}
\frac{N_x}{D} \frac{D}{G_c h_c} \frac{\partial^2}{\partial x^2} w - \frac{\partial}{\partial x} \frac{Q_x}{G_c h_c} - \frac{\partial}{\partial y} \frac{Q_y}{G_c h_c} &= 0 \\
\frac{\partial}{\partial y} \left( \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) w - \frac{1 + \mu_f}{2} \frac{\partial^2}{\partial x \partial y} \frac{Q_x}{G_c h_c} - \left( \frac{1 - \mu_f}{2} \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} - \frac{G_c h_c}{D} \right) \frac{Q_y}{G_c h_c} &= 0 \\
\frac{\partial}{\partial x} \left( \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) w - \left( \frac{1 - \mu_f}{2} \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial x^2} - \frac{G_c h_c}{D} \right) \frac{Q_x}{G_c h_c} - \frac{1 + \mu_f}{2} \frac{\partial^2}{\partial x \partial y} \frac{Q_y}{G_c h_c} &= 0
\end{align*}
\]

(A2)

Boundary conditions.— The boundary conditions that are to be satisfied
by the functions chosen for the middle-surface deflection \( w \) and
the shear angles \( Q_x/G_c h_c \) and \( Q_y/G_c h_c \) are that no middle-surface
deflection occurs at the plate edges, that no point in the boundary is
permitted to move parallel to the edges, that no bending moment exists
along the simply supported edges, and that along the clamped edges the sections making up the boundary do not rotate. These conditions are given by the following equations:

At \( x = 0 \),

\[
w = M_x = \frac{Q_y}{G_c h_c} = 0 \quad (A3a)
\]

and at \( y = \pm \frac{b}{2} \)

\[
w = \frac{Q_x}{G_c h_c} = \frac{\partial w}{\partial y} - \frac{Q_y}{G_c h_c} = 0 \quad (A3b)
\]

The bending moment \( M_x \) is given by equation (6a) of reference 1 as

\[
M_x = -D \left[ \frac{\partial}{\partial x} \left( \frac{\partial w}{\partial x} - \frac{Q_x}{G_c h_c} \right) + \mu F \frac{\partial}{\partial y} \left( \frac{\partial w}{\partial y} - \frac{Q_y}{G_c h_c} \right) \right] \quad (A4)
\]

**Solution of differential equations.** The plate is assumed to buckle symmetrically about the \( x \)-axis and sinusoidally in the \( x \)-direction (fig. 1). Solutions for the middle-surface deflection \( w \) and the shear angles \( \frac{Q_x}{G_c h_c} \) and \( \frac{Q_y}{G_c h_c} \) are then taken in the form

\[
w = \sin \frac{\pi x}{a} \sum_i A_i \cosh \frac{\pi N_i y}{b}
\]

\[
\frac{Q_x}{G_c h_c} = \cos \frac{\pi x}{a} \sum_i B_i \cosh \frac{\pi N_i y}{b} \quad (A5)
\]

\[
\frac{Q_y}{G_c h_c} = \sin \frac{\pi x}{a} \sum_i C_i \sinh \frac{\pi N_i y}{b}
\]

where \( m \) is an integer indicating the number of sinusoidal half waves in the \( x \)-direction and values of \( N_i \) and the coefficients \( A_i, B_i, \) and \( C_i \) are to be determined. Equations (A5) satisfy the boundary conditions (A3a).
Substitution of equations (A5) in equations (A2) yields, after simplification, the following set of simultaneous equations which applies for each set of values of \( A_1, B_1, C_1, \) and \( N_1:\)

\[
krA_1 - \frac{a}{m}B_1 + N_1 \frac{\beta}{m} \frac{a}{m} C_1 = 0 \tag{A6a}
\]

\[
N_1 \frac{\beta}{m} \left[ \left( N_1 \frac{\beta}{m} \right)^2 - 1 \right] A_1 + \frac{1 + \mu_r}{2} N_1 \frac{\beta}{m} \frac{a}{m} B_1
\]

\[
+ \left[ \left( N_1 \frac{\beta}{m} \right)^2 \frac{1}{r} + \frac{1 - \mu_r}{2} - \left( N_1 \frac{\beta}{m} \right)^2 \right] \frac{a}{m} C_1 = 0 \tag{A6b}
\]

\[
\left( N_1 \frac{\beta}{m} \right)^2 - 1 \right] A_1 + \left[ \left( \frac{\beta}{m} \right)^2 \frac{1}{r} + 1 - \frac{1 - \mu_r}{2} \left( N_1 \frac{\beta}{m} \right)^2 \right] \frac{a}{m} B_1
\]

\[- \frac{1 + \mu_r}{2} N_1 \frac{\beta}{m} \frac{a}{m} C_1 = 0 \tag{A6c}
\]

Three values of \( N_1, \) for which equations (A5) satisfy the differential equations (A1), are obtained by setting the determinant of the coefficients of equations (A6) equal to zero

\[
N_1 = \sqrt{\frac{2}{(1 - \mu_r)\beta} + \left( \frac{m}{\beta} \right)^2}
\]

\[
N_2 = \frac{m}{\beta} \sqrt{1 - kr^2 + \sqrt{k \left( \frac{\beta}{m} \right)^2 + \left( \frac{kr^2}{2} \right)^2}}
\]

\[
N_3 = \frac{m}{\beta} \sqrt{1 - kr^2 - \sqrt{k \left( \frac{\beta}{m} \right)^2 + \left( \frac{kr^2}{2} \right)^2}}
\tag{A7}
\]

Expressions for the coefficients \( B_1 \) and \( C_1 \) in terms of \( A_1 \) are found by solving equations (A6a) and (A6b). This procedure gives

\[
B_1 = \frac{m \pi}{a} \lambda_1 A_1
\]

\[
C_1 = \frac{m \pi}{a} N_1 \frac{\beta}{m} \gamma_1 A_1 \tag{A8}
\]
where

\[
\lambda_1 = \frac{(N_1 \beta_m)^2 \left[ \left( N_1 \beta_m \right)^2 - 1 \right]}{\frac{1 - \mu F}{2} \left[ \left( N_1 \beta_m \right)^2 - 1 \right] - \left( \frac{\beta}{m} \right)^2 \frac{1}{r}} \left( \frac{\beta}{m} \right)^2 \frac{1}{r} + \frac{1 - \mu F}{2} \left( \frac{\beta}{m} \right)^2 \frac{1}{r} - \left( \frac{\beta}{m} \right)^2 \frac{1}{r} - \frac{1}{r}
\]

\[
\gamma_1 = \frac{(N_1 \beta_m)^2 - 1}{\frac{1 - \mu F}{2} \left( N_1 \beta_m \right)^2 - 1} \left( \frac{\beta}{m} \right)^2 \frac{1}{r} + \frac{1 + \mu F}{2} \frac{k r}{\frac{1}{r}}
\]

and

\[i = 1, 2, 3\]

Equations (A5) may then be written as

\[
w = \left( A_1 \cosh \pi N_1 \frac{y}{b} + A_2 \cosh \pi N_2 \frac{y}{b} + A_3 \cosh \pi N_3 \frac{y}{b} \right) \sin \frac{m \pi x}{a}
\]

\[
\frac{Q_x}{G_{ch_c}} = \left( \lambda_1 A_1 \cosh \pi N_1 \frac{y}{b} + \lambda_2 A_2 \cosh \pi N_2 \frac{y}{b} + \lambda_3 A_3 \cosh \pi N_3 \frac{y}{b} \right) \frac{m \pi}{a} \cos \frac{m \pi x}{a}
\]

\[
\frac{Q_y}{G_{ch_c}} = \left( N_1 \beta_m \gamma_1 A_1 \sinh \pi N_1 \frac{y}{b} + N_2 \beta_m \gamma_2 A_2 \sinh \pi N_2 \frac{y}{b} + N_3 \beta_m \gamma_3 A_3 \sinh \pi N_3 \frac{y}{b} \right) \frac{m \pi}{a} \sin \frac{m \pi x}{a}
\]

\[\text{(A9)}\]

The coefficients \( A_1, A_2, \) and \( A_3 \) must be adjusted so as to make equations (A9) satisfy boundary conditions (A3b). Substitution of equations (A9) in equations (A3b) gives the following set of simultaneous equations:
A_1 \cosh \frac{nN_1}{2} + A_2 \cosh \frac{nN_2}{2} + A_3 \cosh \frac{nN_3}{2} = 0

\lambda_1 A_1 \cosh \frac{nN_1}{2} + \lambda_2 A_2 \cosh \frac{nN_2}{2} + \lambda_3 A_3 \cosh \frac{nN_3}{2} = 0

\left\{ \begin{array}{l}
N_1 (1 - \gamma_1) A_1 \sinh \frac{nN_1}{2} + N_2 (1 - \gamma_2) A_2 \sinh \frac{nN_2}{2} \\
+ N_3 (1 - \gamma_3) A_3 \sinh \frac{nN_3}{2} = 0
\end{array} \right. \tag{A10}

The condition that $A_1$, $A_2$, and $A_3$ have values other than zero determines the criterion for stability under compression of flat rectangular Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges. The stability criterion, obtained by setting the determinant of the coefficients of equations (A10) equal to zero, is

$$
N_1 (1 - \gamma_1) (\lambda_2 - \lambda_3) \tanh \frac{nN_1}{2} + N_2 (1 - \gamma_2) (\lambda_3 - \lambda_1) \tanh \frac{nN_2}{2} + N_3 (1 - \gamma_3) (\lambda_1 - \lambda_2) \tanh \frac{nN_3}{2} = 0 \tag{A11}
$$

When the plate shear stiffness is infinite ($r = 0$), equation (A11) reduces, in the limit, to the stability criterion for isotropic plates with deflections due to shear neglected:

$$
\frac{\pi}{2} \sqrt{\frac{m}{\beta}} \left( \sqrt{k + \frac{m}{\beta}} \right) \tanh \frac{\pi}{2} \sqrt{\frac{m}{\beta}} \left( \sqrt{k + \frac{m}{\beta}} \right) + \frac{\pi}{2} \sqrt{\frac{m}{\beta}} \left( \sqrt{k - \frac{m}{\beta}} \right) \tan \frac{\pi}{2} \sqrt{\frac{m}{\beta}} \left( \sqrt{k - \frac{m}{\beta}} \right) = 0 \tag{A12}
$$
REFERENCES


### Table 1

**EXPERIMENTAL AND COMPUTED DATA FOR SAWING PLATES WITH END-GRANUL SAW-AWOOD CORRS**

\[
\begin{align*}
\bar{f}_t &= 9.9 \times 10^6 \text{ psi; } \bar{q}_s = 19,000 \text{ psi} \\
\end{align*}
\]

| \( \bar{f}_t \) (lbs/in.²) | \( \bar{q}_s \) (lbs/in.²) | \( q_s \) (lbs/in.²) | \( k \) | \( n_{comp} \) (lbs/in.²) | \( n_{comp} \) (force) | \( A \) | \( z_{comp} \) (lbs/in.) | \( z_{comp} \) (force) | \( D \) (in.) | \( D \) (force) | \( R \) (in.) | \( R \) (force) | \( \% \text{ Error} \) |
|-----------------|-----------------|-----------------|-----|-----------------|-----------------|-----|-----------------|-----------------|-----|-----------------|-----------------|-----|-----------------|-----------------|-----|
| 0.012 | 0.255 | 33.08 | 39.95 | 0.826 | 4765 | 0.0061 | 7.28 | 215 | 8.99 | --- | 4401 | 185 | 6.87 | 30.1 |
| 0.013 | 0.277 | 33.08 | 39.88 | 0.818 | 4739 | 0.0061 | 7.28 | 230 | 9.15 | --- | 4624 | 176 | 6.77 | 35.1 |
| 0.013 | 0.291 | 33.00 | 39.98 | 0.815 | 5047 | 0.0063 | 7.26 | 231 | 8.88 | --- | 4706 | 174 | 6.31 | 40.7 |
| 0.013 | 0.292 | 33.00 | 39.90 | 0.817 | 4706 | 0.0061 | 7.28 | 212 | 8.83 | --- | 4383 | 165 | 6.87 | 38.5 |
| 0.013 | 0.295 | 33.00 | 39.99 | 0.812 | 4765 | 0.0073 | 6.73 | 245 | 10.21 | --- | 4415 | 217 | 9.64 | 31.9 |
| 0.013 | 0.299 | 33.00 | 39.80 | 0.812 | 4909 | 0.0077 | 6.73 | 254 | 10.58 | --- | 4418 | 212 | 7.58 | 40.3 |
| 0.013 | 0.301 | 33.00 | 39.90 | 0.811 | 4813 | 0.0074 | 6.73 | 238 | 9.90 | --- | 4823 | 209 | 8.34 | 16.1 |
| 0.013 | 0.301 | 33.00 | 39.98 | 0.810 | 5047 | 0.0082 | 6.71 | 259 | 9.96 | --- | 4894 | 205 | 7.88 | 26.3 |
| 0.013 | 0.309 | 33.00 | 39.83 | 0.660 | 4467 | 0.0108 | 6.62 | 351 | 15.95 | --- | 4119 | 275 | 12.55 | 27.6 |
| 0.013 | 0.309 | 33.00 | 39.88 | 0.660 | 4467 | 0.0118 | 6.60 | 357 | 14.87 | --- | 4223 | 304 | 12.67 | 17.3 |
| 0.013 | 0.311 | 33.00 | 39.80 | 0.660 | 4361 | 0.0120 | 6.61 | 365 | 14.37 | --- | 4088 | 290 | 12.65 | 18.9 |
| 0.013 | 0.312 | 33.00 | 39.82 | 0.659 | 4278 | 0.0105 | 6.65 | 335 | 15.23 | --- | 4212 | 275 | 12.55 | 21.8 |
| 0.013 | 0.316 | 33.00 | 39.85 | 0.654 | 4216 | 0.0196 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.318 | 33.00 | 39.83 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.318 | 33.00 | 39.93 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.318 | 33.00 | 39.83 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.321 | 33.00 | 39.90 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.321 | 33.00 | 39.83 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.321 | 33.00 | 39.83 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.321 | 33.00 | 39.83 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |
| 0.013 | 0.321 | 33.00 | 39.83 | 0.651 | 4216 | 0.0193 | 6.40 | 506 | 20.93 | --- | 4517 | 457 | 16.50 | 28.1 |

*Corrected for plasticity.*

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NACA TN 1886
<table>
<thead>
<tr>
<th>( t_p ) (in.)</th>
<th>( h_c ) (in.)</th>
<th>( a ) (in.)</th>
<th>( b ) (in.)</th>
<th>( a/b )</th>
<th>( \bar{D}_{\text{comp}} ) (lb-in.)</th>
<th>( r )</th>
<th>( k )</th>
<th>( \bar{N}_{\text{comp}} ) (psi)</th>
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<th>( \text{Error} ) (percent)</th>
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Figure 1.— Metalite type sandwich plate with simply supported loaded edges and clamped unloaded edges.
Figure 2.— Elastic-compressive-buckling curves for Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges \( \mu_p = \frac{1}{3} \).
Figure 3.— Comparison of elastic-compressive-buckling coefficients for infinitely long Metalite type sandwich plates with clamped edges \((u_F = \frac{1}{3})\) and with simply supported edges.
Figure 4.—Comparison of theoretical and experimental buckling stresses for sandwich plates with Alclad 24S-T aluminum faces and balsa-wood cores.
Figure 5.—Comparison of theoretical and experimental buckling stresses for sandwich plates with Alclad 246-T aluminum-alloy faces and cellular-cellulose-acetate cores.
Plates, Flat – Unstiffened 4.3.1.1

Compressive Buckling of Flat Rectangular Metalite Type Sandwich Plates with Simply Supported Loaded Edges and Clamped Unloaded Edges.

By Paul Seide

NACA TN 1886
May 1949

(Abstract on Reverse Side)

Loads and Stresses, Structural – Compression 4.7.2

Compressive Buckling of Flat Rectangular Metalite Type Sandwich Plates with Simply Supported Loaded Edges and Clamped Unloaded Edges.

By Paul Seide

NACA TN 1886
May 1949

(Abstract on Reverse Side)
Abstract

A theoretical solution is obtained for the problem of the compressive buckling of flat rectangular Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges. The solution is based upon the general small-deflection theory for flat sandwich plates developed in NACA TN No. 1526. Good agreement is found between the present results and those of Forest Products Laboratory Rep. No. 1583.

A comparison of computed and experimental buckling stresses of sandwich plates with balsa-wood cores and with cellular-cellulose-acetate cores indicates reasonable agreement between theoretical and experimental results.

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NACA TN 1886