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

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SUMMARY

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
μ_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 $\left(\dfrac{E_f t_f (h_c + t_f)^2}{2\left(1 - \mu_f^2\right)}\right)$
a	plate length
b	plate width
β	plate aspect ratio (a/b)
r	core shear-flexibility coefficient $\left(\dfrac{\pi^2 D}{b^2 G_c h_c}\right)$
σ_{cr}	critical compressive stress in x-direction
k	elastic-buckling-stress coefficient $\left(\dfrac{2b^2 \sigma_{cr} t_f}{\pi^2 D}\right)$
N_x	critical compressive load per unit width $(2\sigma_{cr} t_f)$
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

$\frac{Q_x}{G_c h_c}$, $\frac{Q_y}{G_c h_c}$ 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 (A11)) derived in the appendix gives the elastic-buckling-stress coefficient k implicitly in terms of the plate aspect ratio β 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

$$k = \frac{1}{r} \qquad (1)$$

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 $\left(\mu_f = \frac{1}{3}\right)$ 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

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

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 semiempirically 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 semiempirical 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 agreement between computed and semiempirically determined flexural stiffnesses 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 COMPRESSIVE 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

$$\left.\begin{aligned} &\mu_x = \mu_y = \mu_f \\ &D_x = D_y = (1 + \mu_f)D_{xy} = \tfrac{1}{2} E_f t_f (h_c + t_f)^2 \\ &D_{Q_x} = D_{Q_y} = G_c h_c \end{aligned}\right\} \tag{A1}$$

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

$$\left.\begin{aligned} &\frac{N_x}{D}\frac{D}{G_c h_c}\frac{\partial^2}{\partial x^2} w - \frac{\partial}{\partial x}\frac{Q_x}{G_c h_c} - \frac{\partial}{\partial y}\frac{Q_y}{G_c h_c} = 0 \\ &\frac{\partial}{\partial y}\left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2}\right)w - \frac{1 + \mu_f}{2}\frac{\partial^2}{\partial x \partial y}\frac{Q_x}{G_c h_c} - \left(\frac{1 - \mu_f}{2}\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} - \frac{G_c h_c}{D}\right)\frac{Q_y}{G_c h_c} = 0 \\ &\frac{\partial}{\partial x}\left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2}\right)w - \left(\frac{1 - \mu_f}{2}\frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial x^2} - \frac{G_c h_c}{D}\right)\frac{Q_x}{G_c h_c} - \frac{1 + \mu_f}{2}\frac{\partial^2}{\partial x \partial y}\frac{Q_y}{G_c h_c} = 0 \end{aligned}\right\} \tag{A2}$$

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 $x = 0,a$

$$w = M_x = \frac{Q_y}{G_c h_c} = 0 \tag{A3a}$$

and at $y = \pm\frac{b}{2}$

$$w = \frac{Q_x}{G_c h_c} = \frac{\partial w}{\partial y} - \frac{Q_y}{G_c h_c} = 0 \tag{A3b}$$

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

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

<u>Solution of differential equations</u>.- 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 $Q_x/G_c h_c$ and $Q_y/G_c h_c$ are then taken in the form

$$\left.\begin{aligned} w &= \sin\frac{m\pi x}{a}\sum_i A_i \cosh\frac{\pi N_i y}{b} \\ \frac{Q_x}{G_c h_c} &= \cos\frac{m\pi x}{a}\sum_i B_i \cosh\frac{\pi N_i y}{b} \\ \frac{Q_y}{G_c h_c} &= \sin\frac{m\pi x}{a}\sum_i C_i \sinh\frac{\pi N_i y}{b} \end{aligned}\right\} \tag{A5}$$

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_i, B_i, C_i, and N_i:

$$kr A_i - \frac{a}{m\pi} B_i + N_i \frac{\beta}{m} \frac{a}{m\pi} C_i = 0 \tag{A6a}$$

$$N_i \frac{\beta}{m}\left[\left(N_i \frac{\beta}{m}\right)^2 - 1\right]A_i + \frac{1 + \mu_f}{2} N_i \frac{\beta}{m} \frac{a}{m\pi} B_i + \left[\left(\frac{\beta}{m}\right)^2 \frac{1}{r} + \frac{1 - \mu_f}{2} - \left(N_i \frac{\beta}{m}\right)^2\right]\frac{a}{m\pi} C_i = 0 \tag{A6b}$$

$$\left[\left(N_i \frac{\beta}{m}\right)^2 - 1\right]A_i + \left[\left(\frac{\beta}{m}\right)^2 \frac{1}{r} + 1 - \frac{1 - \mu_f}{2}\left(N_i \frac{\beta}{m}\right)^2\right]\frac{a}{m\pi} B_i - \frac{1 + \mu_f}{2} N_i \frac{\beta}{m} \frac{a}{m\pi} C_i = 0 \tag{A6c}$$

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

$$\left.\begin{aligned}
N_1 &= \sqrt{\frac{2}{(1 - \mu_f)r} + \left(\frac{m}{\beta}\right)^2} \\
N_2 &= \frac{m}{\beta}\sqrt{1 - \frac{kr}{2} + \sqrt{k\left(\frac{\beta}{m}\right)^2 + \left(\frac{kr}{2}\right)^2}} \\
N_3 &= \frac{m}{\beta}\sqrt{1 - \frac{kr}{2} - \sqrt{k\left(\frac{\beta}{m}\right)^2 + \left(\frac{kr}{2}\right)^2}}
\end{aligned}\right\} \tag{A7}$$

Expressions for the coefficients B_i and C_i in terms of A_i are found by solving equations (A6a) and (A6b). This procedure gives

$$\left.\begin{aligned}
B_i &= \frac{m\pi}{a} \lambda_i A_i \\
C_i &= \frac{m\pi}{a} N_i \frac{\beta}{m} \gamma_i A_i
\end{aligned}\right\} \tag{A8}$$

where

$$\lambda_i = \frac{\left(N_i \frac{\beta}{m}\right)^2 \left[\left(N_i \frac{\beta}{m}\right)^2 - 1\right] - kr\left[\left(\frac{\beta}{m}\right)^2 \frac{1}{r} + \frac{1 - \mu_f}{2} - \left(N_i \frac{\beta}{m}\right)^2\right]}{\frac{1 - \mu_f}{2}\left[\left(N_i \frac{\beta}{m}\right)^2 - 1\right] - \left(\frac{\beta}{m}\right)^2 \frac{1}{r}}$$

$$\gamma_i = \frac{\left(N_i \frac{\beta}{m}\right)^2 - 1 + \frac{1 + \mu_f}{2} kr}{\frac{1 - \mu_f}{2}\left[\left(N_i \frac{\beta}{m}\right)^2 - 1\right] - \left(\frac{\beta}{m}\right)^2 \frac{1}{r}}$$

and

$$i = 1,2,3$$

Equations (A5) may then be written as

$$\left.\begin{aligned}
w &= \left(A_1 \cosh \pi N_1 \frac{y}{b} + A_2 \cosh \pi N_2 \frac{y}{b} + A_3 \cosh \pi N_3 \frac{y}{b}\right)\sin \frac{m\pi x}{a} \\
\frac{Q_x}{G_c h_c} &= \left(\lambda_1 A_1 \cosh \pi N_1 \frac{y}{b} + \lambda_2 A_2 \cosh \pi N_2 \frac{y}{b} + \lambda_3 A_3 \cosh \pi N_3 \frac{y}{b}\right)\frac{m\pi}{a} \cos \frac{m\pi x}{a} \\
\frac{Q_y}{G_c h_c} &= \left(N_1 \frac{\beta}{m} \gamma_1 A_1 \sinh \pi N_1 \frac{y}{b} + N_2 \frac{\beta}{m} \gamma_2 A_2 \sinh \pi N_2 \frac{y}{b}\right. \\
&\quad \left. + N_3 \frac{\beta}{m} \gamma_3 A_3 \sinh \pi N_3 \frac{y}{b}\right)\frac{m\pi}{a} \sin \frac{m\pi x}{a}
\end{aligned}\right\} \quad (A9)$$

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:

$$\left.\begin{aligned}
&A_1 \cosh\frac{\pi N_1}{2} + A_2 \cosh\frac{\pi N_2}{2} + A_3 \cosh\frac{\pi N_3}{2} = 0 \\
&\lambda_1 A_1 \cosh\frac{\pi N_1}{2} + \lambda_2 A_2 \cosh\frac{\pi N_2}{2} + \lambda_3 A_3 \cosh\frac{\pi N_3}{2} = 0 \\
&N_1\left(1 - \gamma_1\right)A_1 \sinh\frac{\pi N_1}{2} + N_2\left(1 - \gamma_2\right)A_2 \sinh\frac{\pi N_2}{2} \\
&\qquad + N_3\left(1 - \gamma_3\right)A_3 \sinh\frac{\pi N_3}{2} = 0
\end{aligned}\right\} \qquad (A10)$$

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

$$\begin{aligned}
&N_1\left(1 - \gamma_1\right)\left(\lambda_2 - \lambda_3\right)\tanh\frac{\pi N_1}{2} + N_2\left(1 - \gamma_2\right)\left(\lambda_3 - \lambda_1\right)\tanh\frac{\pi N_2}{2} \\
&\qquad + N_3\left(1 - \gamma_3\right)\left(\lambda_1 - \lambda_2\right)\tanh\frac{\pi N_3}{2} = 0
\end{aligned} \qquad (A11)$$

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:

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

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TABLE 1

EXPERIMENTAL AND COMPUTED DATA FOR SANDWICH PLATES WITH END-GRAIN BALSA-WOOD CORES

$\left[E_f = 9.9 \times 10^6 \text{ psi}; G_c = 19{,}000 \text{ psi}\right]$

(1) t_f (in.)	(2) h_c (in.)	(3) a (in.)	(4) b (in.)	(5) a/b	(6) D_{comp} (lb-in.)	(7) r	(8) k	(9) N_{comp} (lb/in.)	(10) σ_{comp} (psi)	(11) N_{comp} (lb/in.) (a)	(12) σ_{comp} (psi) (a)	(13) D_{exp} (lb-in.)	(14) N_{exp} (lb/in.)	(15) σ_{exp} (psi)	(16) Error (percent)
0.012	0.255	33.02	39.95	0.826	4765	0.0061	7.28	215	8.95	---	----	4401	165	6.87	30.2
.013	.257	33.03	39.88	.828	5279	.0067	7.25	238	9.15	---	----	4624	176	6.77	35.1
.013	.251	33.02	39.98	.825	5047	.0065	7.26	231	8.88	---	----	4326	164	6.31	40.7
.012	.252	33.02	39.90	.827	4706	.0061	7.28	212	8.83	---	----	4383	165	6.87	28.5
.012	.255	23.02	35.95	.642	4765	.0075	6.73	245	10.21	---	----	4416	217	9.04	12.9
.012	.259	23.03	35.82	.642	4909	.0077	6.73	254	10.58	---	----	4418	181	7.54	40.3
.012	.251	23.02	35.90	.641	4623	.0074	6.73	238	9.92	---	----	4323	205	8.54	16.1
.013	.251	23.03	35.94	.640	5047	.0081	6.71	259	9.96	---	----	4294	205	7.88	26.3
.011	.259	19.03	28.83	.660	4467	.0108	6.62	351	15.95	---	----	4119	275	12.50	27.6
.012	.249	19.02	28.81	.660	4553	.0114	6.60	357	14.87	---	----	4223	304	12.67	17.3
.012	.244	19.01	28.80	.660	4381	.0112	6.61	345	14.37	---	----	4088	290	12.08	18.9
.011	.252	19.01	28.82	.659	4238	.0105	6.65	335	15.23	---	----	4212	275	12.50	21.8
.014	.246	16.01	23.83	.674	5272	.0196	6.40	586	20.93	---	----	4517	457	16.32	28.2
.012	.251	16.02	23.84	.671	4623	.0168	6.48	521	21.70	---	----	4512	423	17.62	23.1
.013	.246	16.02	23.84	.671	4858	.0180	6.43	543	20.88	---	----	4330	423	16.27	28.3
.011	.251	16.02	23.84	.671	4206	.0153	6.51	476	21.64	---	----	4332	406	18.45	17.2
.013	.255	14.03	20.82	.673	5201	.0245	6.30	747	28.73	730	28.1	4514	584	22.46	24.8
.013	.249	14.03	20.89	.672	4971	.0238	6.30	709	27.26	702	27.0	4440	528	20.31	33.0
.011	.257	14.00	20.80	.673	4401	.0206	6.40	643	29.23	627	28.5	4320	544	24.73	15.3
.013	.251	14.03	20.94	.670	5047	.0238	6.30	653	25.11	653	25.1	4618	528	20.31	23.6
.013	.253	12.02	18.98	.633	5124	.0292	6.18	868	33.38	806	31.0	4506	632	24.31	27.5
.013	.252	12.02	18.87	.636	5085	.0294	6.17	870	33.46	806	31.0	4426	571	21.96	40.9
.013	.247	12.02	18.95	.634	4895	.0287	6.19	833	32.04	785	30.2	4070	571	21.96	37.2
.014	.248	12.02	18.90	.635	5272	.0309	6.11	891	31.82	840	30.0	4334	571	20.39	47.0
.013	.248	11.02	16.87	.653	4933	.0362	6.00	1025	39.42	883	34.0	4380	667	25.65	32.3
.013	.253	11.01	16.88	.652	5124	.0369	5.99	1063	40.88	896	34.5	4571	711	27.35	25.9
.014	.250	11.01	16.94	.649	5435	.0393	5.93	1108	39.57	952	34.0	4568	667	23.82	42.8
.013	.253	11.00	16.98	.647	5124	.0365	6.00	1053	40.50	894	34.4	4411	711	27.35	25.5

[a]Corrected for plasticity.

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TABLE 2

EXPERIMENTAL AND COMPUTED DATA FOR SANDWICH PLATES WITH CELLULAR-CELLULOSE-ACETATE CORES

$[E_f = 9.9 \times 10^6 \text{ psi};\ G_c = 3,500 \text{ psi}]$

① t_f (in.)	② h_c (in.)	③ a (in.)	④ b (in.)	⑤ a/b	⑥ D_{comp} (lb-in.)	⑦ r	⑧ k	⑨ N_{comp} (lb/in.)	⑩ σ_{comp} (psi)	⑪ D_{exp} (lb-in.)	⑫ N_{exp} (lb/in.)	⑬ σ_{exp} (psi)	⑭ Error (percent)
0.012	0.247	33.00	39.82	0.829	4484	0.0323	6.65	186	7.75	4818	167	6.95	11.5
.013	.247	33.04	39.82	.830	4895	.0352	6.59	201	7.73	4673	167	6.42	20.4
.012	.248	33.03	39.88	.828	4519	.0323	6.65	186	7.75	4644	140	5.83	32.9
.013	.246	32.44	39.86	.813	4858	.0350	6.59	199	7.65	4709	176	6.77	13.0
.012	.249	23.02	35.88	.641	4553	.0400	5.90	206	8.58	4769	212	8.83	-2.8
.012	.249	23.01	35.84	.642	4553	.0401	5.90	206	8.58	4932	222	9.25	-7.2
.012	.246	23.02	35.88	.641	4449	.0396	5.93	202	8.41	4699	212	8.83	-4.8
.013	.243	23.02	35.98	.639	4746	.0425	5.85	212	8.14	4685	193	7.42	9.8
.013	.244	19.00	28.82	.659	4783	.0665	5.40	307	11.81	4774	348	13.38	-11.7
.013	.245	19.02	28.85	.659	4820	.0666	5.40	309	11.88	4778	301	11.58	2.6
.013	.248	19.01	28.85	.658	4933	.0674	5.39	315	12.11	4860	301	11.58	4.6
.013	.247	19.01	28.85	.658	4895	.0672	5.39	313	12.04	4898	301	11.58	4.0
.013	.243	16.02	23.84	.671	4746	.0969	4.92	406	15.61	4796	379	14.58	7.1
.013	.241	16.00	23.84	.671	4672	.0962	4.93	400	15.38	4806	379	14.58	5.5
.013	.238	16.00	23.84	.671	4562	.0951	4.95	392	15.08	4764	406	15.61	-3.4
.013	.245	16.00	23.85	.670	4820	.0974	4.90	410	15.77	4914	447	17.19	-8.3
.014	.250	14.01	21.13	.663	5435	.1374	4.40	529	18.89	4826	503	17.96	5.2
.014	.257	14.00	20.80	.673	5727	.1450	4.30	561	20.04	5165	558	19.93	0.6
.013	.256	14.00	20.88	.670	5240	.1323	4.45	528	20.31	4906	483	18.58	9.3
.013	.254	12.01	18.88	.636	5162	.1610	4.10	587	22.58	4688	554	21.31	6.0
.012	.253	12.01	18.83	.637	4694	.1475	4.29	560	23.33	4517	571	23.79	-1.9
.013	.251	12.02	18.87	.636	5047	.1592	4.13	578	22.23	4656	473	18.19	22.2
.013	.251	12.01	18.93	.634	5047	.1583	4.15	577	22.19	4549	522	20.08	10.5
.014	.249	10.99	16.57	.633	5394	.2220	3.50	678	24.21	4785	618	22.07	9.7

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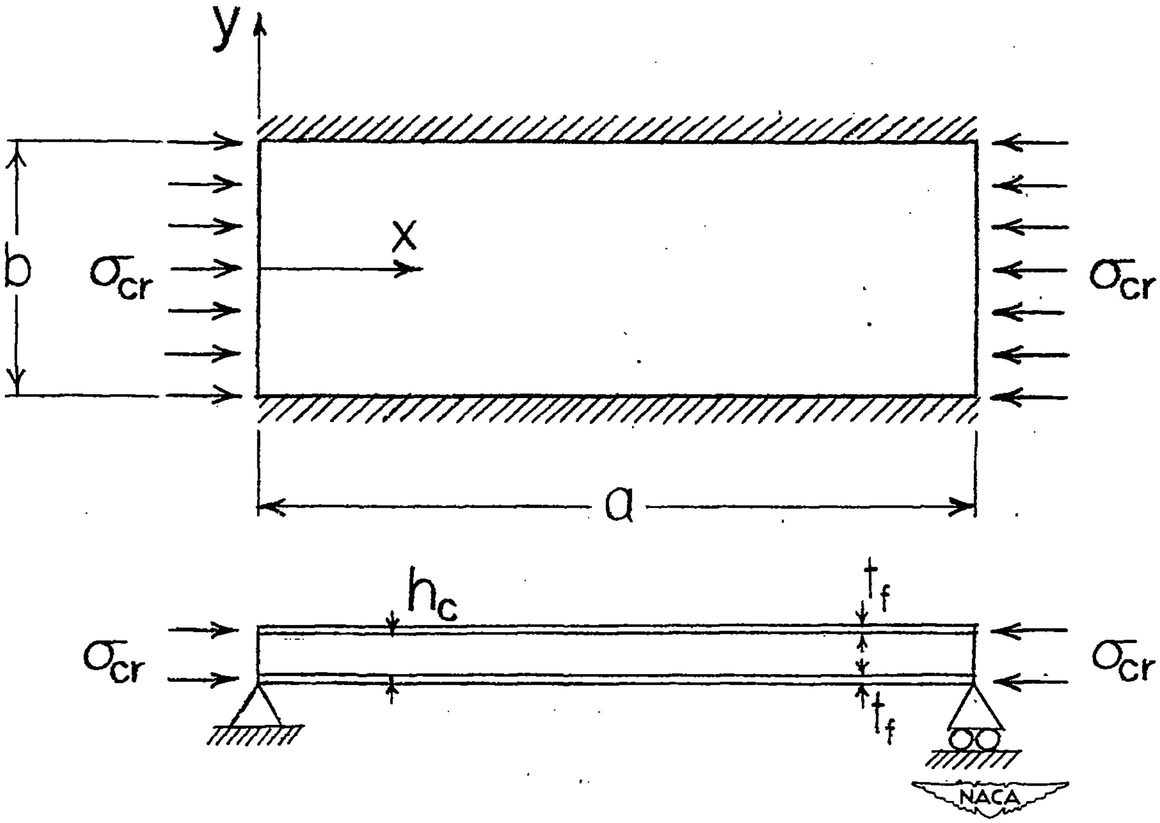

Figure 1.- Metalite type sandwich plate with simply supported loaded edges and clamped unloaded edges.

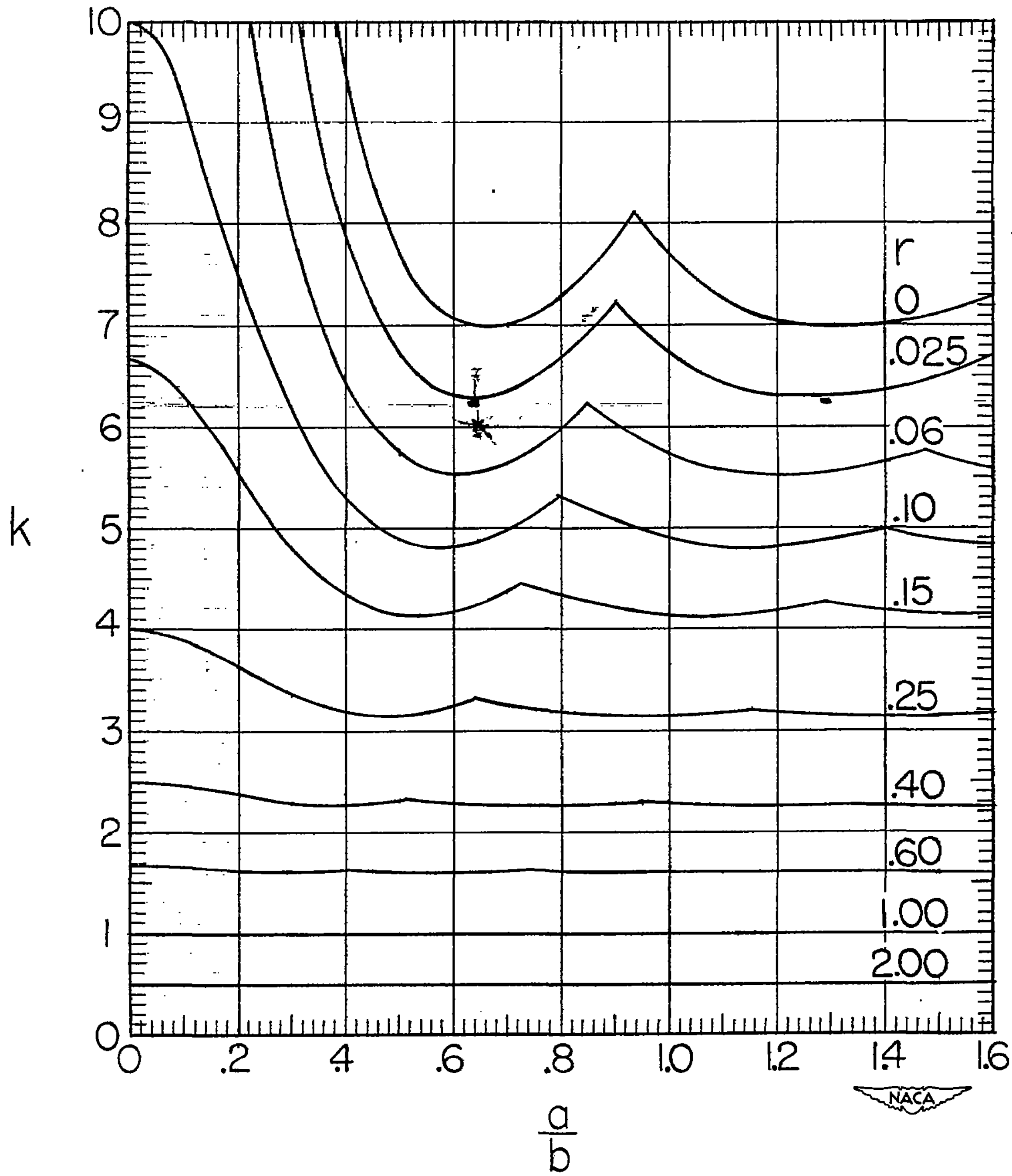

Figure 2.– Elastic–compressive–buckling curves for Metalite type sandwich plates with simply supported loaded edges and clamped unloaded edges $\left(\mu_f = \frac{1}{3}\right)$.

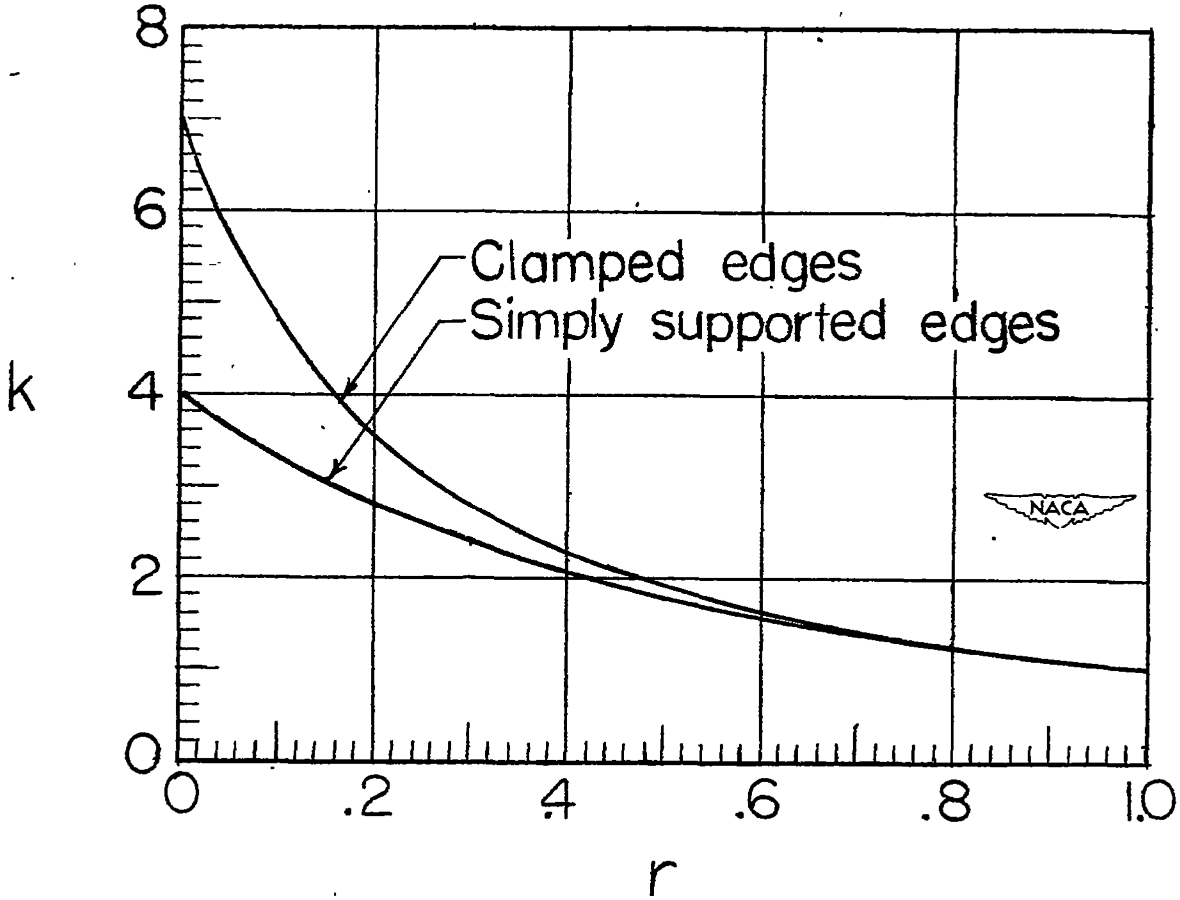

Figure 3.— Comparison of elastic-compressive-buckling coefficients for infinitely long Metalite type sandwich plates with clamped edges $\left(\mu_f = \frac{1}{3}\right)$ and with simply supported edges.

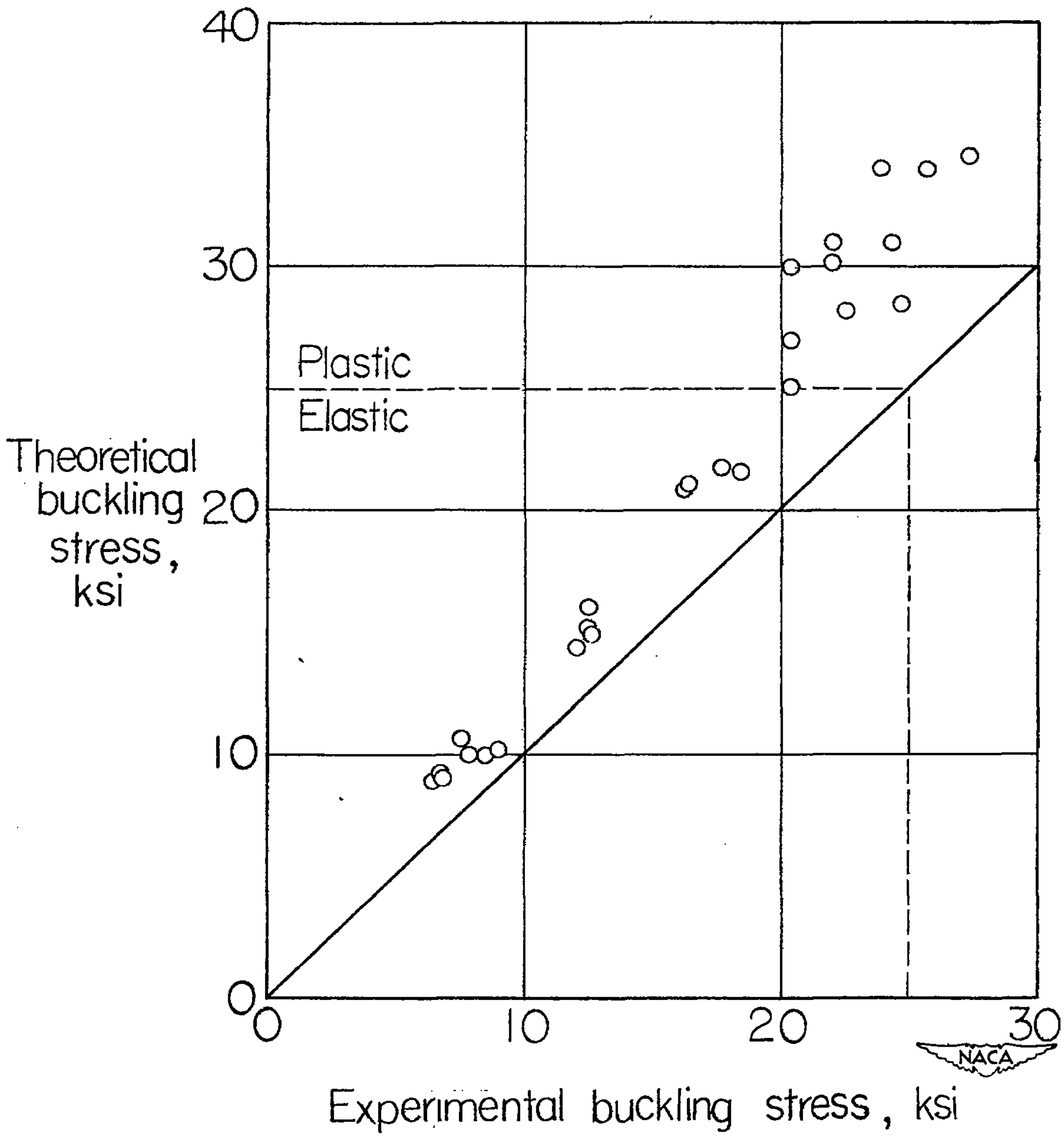

Figure 4.– Comparison of theoretical and experimental buckling stresses for sandwich plates with Alclad 24S-T aluminum faces and balsa-wood cores.

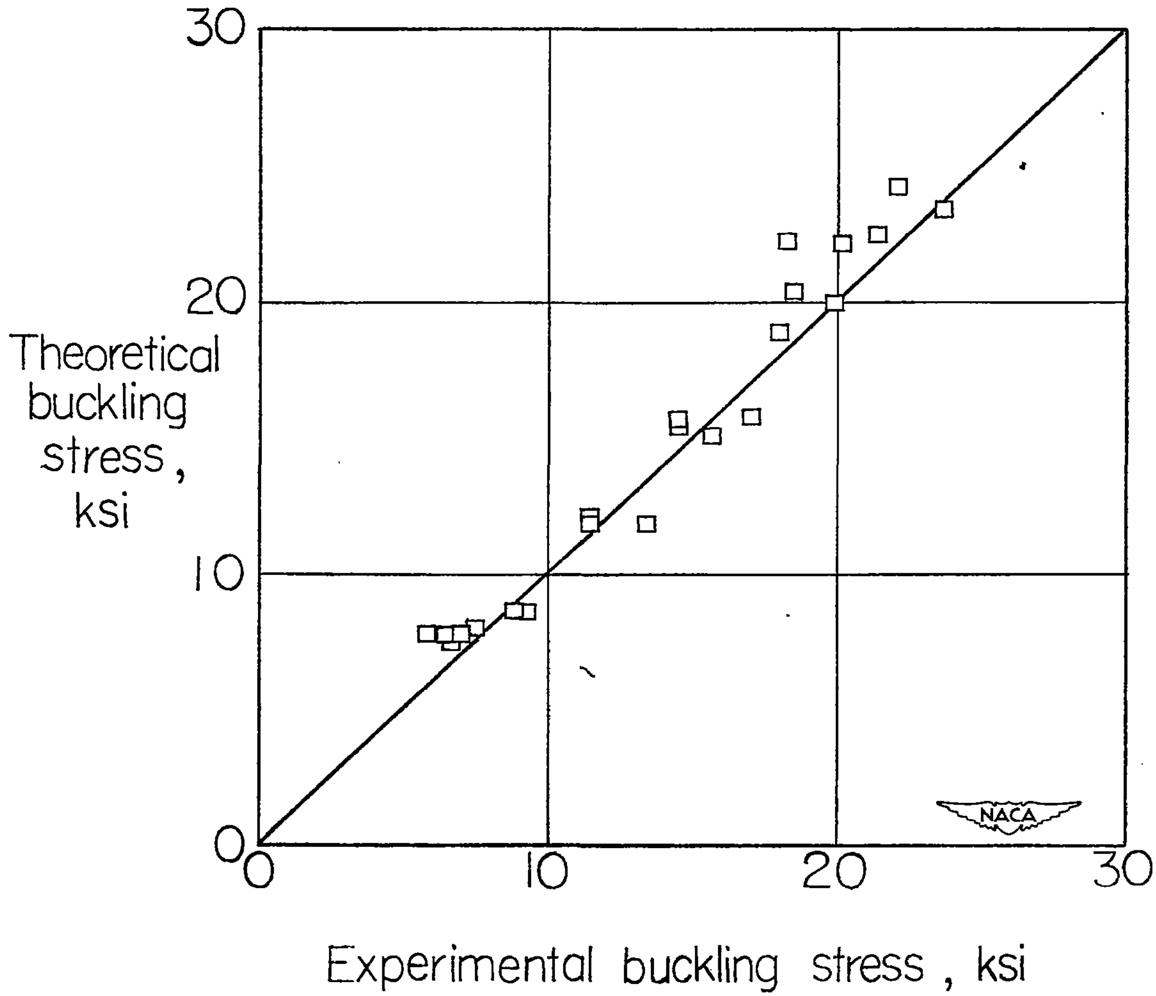

Figure 5.– Comparison of theoretical and experimental buckling stresses for sandwich plates with Alclad 24S-T aluminum-alloy faces and cellular-cellulose-acetate cores.