

NO. 1-W

**CASE FILE
COPY**

Case Only

NACA TN 1910

**NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS**

TECHNICAL NOTE 1910

**SHEAR BUCKLING OF INFINITELY LONG SIMPLY SUPPORTED
METALITE TYPE SANDWICH PLATES**

By Paul Seide

Langley Aeronautical Laboratory
Langley Air Force Base, Va.



Washington

July 1949

FILE COPY

To be returned to
the files of the National
Advisory Committee
for Aeronautics
Washington, D. C.

A D D E N D U M

NACA TN 1910

SHEAR BUCKLING OF INFINITELY LONG SIMPLY SUPPORTED
METALITE TYPE SANDWICH PLATES

By Paul Seide

July 1949

The Forest Products Laboratory has pointed out that the expression for M_{st} in equation (A10) (p. 11) can be simplified to

$$M_{st} = \frac{st \left\{ 1 + r \left(s^2 + \frac{1}{\beta^2} \right) \right\}}{(s^2 - t^2) (1 + s^2 \beta^2)^2}$$

The absence of Poisson's ratio μ_f from this simplified expression indicates that the numerical results of the present paper are valid for any value of μ_f .

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 1910

SHEAR BUCKLING OF INFINITELY LONG SIMPLY SUPPORTED
METALITE TYPE SANDWICH PLATES

By Paul Seide

SUMMARY

A theoretical solution is obtained for the problem of the shear buckling of infinitely long simply supported elastic Metalite type sandwich plates. An approximate correction is suggested for the determination of the critical shear stresses of plates that buckle in the plastic range.

INTRODUCTION

The problem of determining the critical compressive stresses of flat rectangular Metalite type sandwich plates has recently received much attention in the literature (references 1 to 6). Little has appeared, however, on the stability of sandwich plates under conditions of loading other than compression. A loading that is very important in aircraft design is that of shear. The shear buckling of infinitely long simply supported Metalite type sandwich plates (fig. 1) has been treated in reference 7. The analysis, however, is based on assumptions that render it intrinsically approximate and is limited to the derivation of stability equations, without any presentation of numerical results.

In the present paper the problem treated in reference 7, the shear buckling of infinitely long simply supported Metalite type sandwich plates, is solved by means of the more accurate sandwich-plate theory of reference 8. The Rayleigh-Ritz energy method is used to obtain approximate values of elastic critical shear stresses for various values of the ratio of plate flexural stiffness to core shear stiffness. For two limiting values of this ratio the critical stresses are within 1 percent of exact values, and it may reasonably be concluded that the critical stresses for intermediate values of the ratio are also within 1 percent of exact values. An approximate correction to the elastic stresses is outlined for the determination of the critical shear stresses of plates that buckle in the plastic range.

Shortly after the work of the present paper was completed, reference 9, which is on the same subject as the present paper, was received at the Langley Laboratory. Approximate assumptions were made in that analysis as to the behavior of sandwich plates, and an approximate deflection function was used. The numerical results, however, agree very closely with those of the present paper and are discussed in more detail in the section entitled "Comparison with other Solutions."

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 $\left(\frac{E_f t_f (h_c + t_f)^2}{2(1 - \mu_f^2)} \right)$
b	plate width
r	core shear-flexibility coefficient $\left(\frac{\pi^2 D}{b^2 G_c h_c} \right)$
τ_{cr}	critical shear stress
k	elastic shear-buckling-stress coefficient $\left(\frac{2b^2 \tau_{cr} t_f}{\pi^2 D} \right)$
N_{xy}	critical shear load per unit length ($2\tau_{cr} t_f$)
x, y	coordinate axes (see fig. 1)
w	deflection of middle surface of plate

$$\frac{Q_x}{G_c h_c}, \frac{Q_y}{G_c h_c}$$

angles between lines originally perpendicular to undeformed middle surface and lines perpendicular to deformed middle surface

λ wave length of buckle

$$\beta = \frac{\lambda}{b}$$

V potential energy

$a_n, b_n, c_n, d_n, e_n, f_n$ } Fourier coefficients

$i, j, l, m, n, p, q, s, t$ } integers

RESULTS AND DISCUSSION

Elastic Shear-Buckling-Stress Coefficients

The elastic shear-buckling-stress coefficients of infinitely long simply supported Metalite type sandwich plates (fig. 1) are given in figure 2 for Poisson's ratio of the face material equal to 1/3. Shear-buckling-stress coefficients k are plotted as a function of the core shear-flexibility coefficient r . The values of k were computed from the approximate stability equations derived in the appendix and are summarized in table 1. From a comparison of the approximate values and known exact values of k for r equal to 0 and 1.0, it seems reasonable to conclude that the approximate values of k for intermediate values of r are accurate to within 1 percent of exact values.

As the core shear-flexibility coefficient increases, the shear-buckling-stress coefficients approach those for compressed simply supported plates (fig. 2, reference 5). The two groups of coefficients are equal for values of r equal to or greater than unity and are given by

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

This phenomenon is explained in the appendix.

Plastic Shear Buckling Stress

When buckling occurs in the plastic range, the stresses computed from the elastic shear-buckling-stress coefficients of figure 2 are too large and must be corrected for plasticity effects. An approximate plasticity correction may be obtained by means of the theory of reference 10. In that paper the plasticity corrections for solid isotropic plates have been derived and apply with no error to sandwich plates with an infinite core shear modulus ($r = 0$). If these corrections are used for sandwich plates with finite core shear stiffness ($r > 0$), the results are conservative and are probably accurate enough for most design purposes.

In figure 3 the plastic shear buckling stresses for plates of 24S-T3 aluminum alloy are plotted against the corresponding elastic shear buckling stresses. This curve is another way of presenting the computed correction curve of figure 1 of reference 10. Plates that buckle in the plastic range may be analyzed by first computing the elastic shear buckling stress from figure 2 and then entering the curve of figure 3 with this stress to obtain the corresponding plastic shear buckling stress. Similar curves for materials other than 24S-T3 aluminum alloy may be plotted by use of the equations on the upper half of page 12 of reference 10.

Comparison with Other Solutions

The results given in reference 9 apply to plates with faces for which the resistance to bending about their own middle surface is not negligible. The face bending stiffness is shown, however, to have only a small effect on the shear-buckling-stress coefficients if the core shear flexibility coefficients are less than 1.0. In this range of values of r the shear-buckling-stress coefficients of reference 9, computed from the theory of reference 7 which is known to yield results lower than exact values, agree very closely with those of the present paper. The present results are slightly high, as is characteristic of energy solutions, but the excellent agreement with the results of reference 9 indicates the accuracy of both solutions.

When r is greater than 1.0, the bending stiffness of the faces may have a large effect on the shear-buckling-stress coefficients. The results of the present paper in this range of values of r are conservative and apply only to plates with faces that are very thin in comparison with the core. Sandwich plates having practical dimensions, however, will probably not have core shear-flexibility coefficients greater than 1.0.

CONCLUDING REMARKS

The elastic shear-buckling-stress coefficients of infinitely long simply supported Metalite type sandwich plates have been computed for Poisson's ratio of the face material equal to $1/3$. An approximate correction for the effects of plasticity has been outlined. This correction is theoretically conservative and is probably accurate enough for most design purposes.

Langley Aeronautical Laboratory
National Advisory Committee for Aeronautics
Langley Air Force Base, Va., May 20, 1949

APPENDIX

DETERMINATION OF CRITICAL SHEAR STRESSES OF
INFINITELY LONG SIMPLY SUPPORTED METALITE
TYPE SANDWICH PLATES

Energy expression.— The potential-energy expression for flat sandwich plates is given by equation (16) of reference 8. For plates subjected only to shear and with the physical constants of Metalite type sandwich plates given by equation (A1) of reference 5, the energy expression becomes

$$\begin{aligned}
 v = & \frac{D}{2} \iint \left(\left[\frac{\partial}{\partial x} \left(\frac{\partial w}{\partial x} - \frac{Q_x}{G_c h_c} \right) + \frac{\partial}{\partial y} \left(\frac{\partial w}{\partial y} - \frac{Q_y}{G_c h_c} \right) \right]^2 \right. \\
 & - 2(1 - \mu_f) \left\{ \frac{\partial}{\partial x} \left(\frac{\partial w}{\partial x} - \frac{Q_x}{G_c h_c} \right) \frac{\partial}{\partial y} \left(\frac{\partial w}{\partial y} - \frac{Q_y}{G_c h_c} \right) \right. \\
 & \left. \left. - \frac{1}{4} \left[\frac{\partial}{\partial y} \left(\frac{\partial w}{\partial x} - \frac{Q_x}{G_c h_c} \right) + \frac{\partial}{\partial x} \left(\frac{\partial w}{\partial y} - \frac{Q_y}{G_c h_c} \right) \right]^2 \right\} \right. \\
 & \left. + \frac{G_c h_c}{D} \left[\left(\frac{Q_x}{G_c h_c} \right)^2 + \left(\frac{Q_y}{G_c h_c} \right)^2 \right] \right) dx dy \\
 & + N_{xy} \iint \frac{\partial w}{\partial x} \frac{\partial w}{\partial y} dx dy \tag{A1}
 \end{aligned}$$

Boundary conditions.— Because the plates considered are infinitely long, 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 periodic in the x -direction over a wave length λ . (See fig. 1.) Furthermore, the functions chosen for w , $\frac{Q_x}{G_c h_c}$, and $\frac{Q_y}{G_c h_c}$ must satisfy the simple support conditions of no moment,

no middle-surface deflection, and no rotation parallel to the boundary, due to shear deformations, of any line in the boundary perpendicular to the middle surface. These conditions are given by the following equations: At $y = 0, b$

$$w = M_y = \frac{Q_x}{G_c h_c} = 0 \quad (A2)$$

where M_y is given by equation (6b) of reference 8 as

$$M_y = -D \left[\frac{\partial}{\partial y} \left(\frac{\partial w}{\partial y} - \frac{Q_y}{G_c h_c} \right) + \mu_f \frac{\partial}{\partial x} \left(\frac{\partial w}{\partial x} - \frac{Q_x}{G_c h_c} \right) \right] \quad (A3)$$

Fourier series for the middle-surface deflections and the shear angles are therefore taken as

$$\left. \begin{aligned} w &= \sum_{n=1}^{\infty} \left(a_n \sin \frac{\pi x}{\lambda} + b_n \cos \frac{\pi x}{\lambda} \right) \sin \frac{n\pi y}{b} \\ \frac{Q_x}{G_c h_c} &= \sum_{n=1}^{\infty} \left(c_n \sin \frac{\pi x}{\lambda} + d_n \cos \frac{\pi x}{\lambda} \right) \sin \frac{n\pi y}{b} \\ \frac{Q_y}{G_c h_c} &= \sum_{n=1}^{\infty} \left(e_n \sin \frac{\pi x}{\lambda} + f_n \cos \frac{\pi x}{\lambda} \right) \cos \frac{n\pi y}{b} \end{aligned} \right\} \quad (A4)$$

These equations satisfy the boundary conditions (A2) term by term.

Stability determinant.— Substitution of equations (A4) in equation (A1) and integration between the limits $x = 0, \lambda$ and $y = 0, b$ yields, after some simplification,

$$\begin{aligned}
\frac{8}{\pi^4} \frac{\lambda^3}{bD} V = & \sum_{n=1}^{\infty} \left\{ \left[(1 + n^2\beta^2)a_n - d_n' - n\beta e_n' \right]^2 + \left[(1 + n^2\beta^2)b_n \right. \right. \\
& + c_n' - n\beta f_n' \left. \right]^2 - 2(1 - \mu_f) \left[n\beta(a_n - d_n')(n\beta a_n - e_n') \right. \\
& + n\beta(b_n + c_n')(n\beta b_n - f_n') - \frac{1}{4}(2n\beta a_n - n\beta d_n' - e_n')^2 \\
& \left. \left. - \frac{1}{4}(2n\beta b_n + n\beta c_n' - f_n')^2 \right] + \frac{\beta^2}{r}(c_n'^2 + d_n'^2 + e_n'^2 + f_n'^2) \right\} \\
& + \frac{16}{\pi} \beta^3 k \sum_p \sum_q \frac{pq}{p^2 - q^2} a_p b_q
\end{aligned} \tag{A5}$$

where

$$\frac{c_n'}{c_n} = \frac{d_n'}{d_n} = \frac{e_n'}{e_n} = \frac{f_n'}{f_n} = \frac{\lambda}{\pi}$$

$$\beta = \frac{\lambda}{b}$$

k shear-buckling-stress coefficient $\left(\frac{b^2 N_{xy}}{\pi^2 D} \right)$

r core shear-flexibility coefficient $\left(\frac{\pi^2 D}{b^2 G_c h_c} \right)$

p, q integers such that $p + q$ is odd

The undetermined Fourier coefficients $a_n, b_n, c_n, d_n, e_n,$ and f_n must now be chosen so as to make the potential energy V a minimum. This condition requires that

$$\frac{\partial V}{\partial a_n} = \frac{\partial V}{\partial b_n} = \frac{\partial V}{\partial c_n} = \frac{\partial V}{\partial d_n} = \frac{\partial V}{\partial e_n} = \frac{\partial V}{\partial f_n} = 0 \tag{A6}$$

where $n = 1, 2, 3, \dots$, or, when the indicated operations are performed, that the coefficients satisfy the following set of simultaneous equations:

$$(1 + n^2\beta^2)a_n - \left(1 + \frac{1 - \mu_f}{2} n^2\beta^2 + \frac{\beta^2}{r}\right)d_n' - \frac{1 + \mu_f}{2} n\beta e_n' = 0 \quad (A7a)$$

$$n\beta(1 + n^2\beta^2)a_n - \frac{1 + \mu_f}{2} n\beta d_n' - \left(\frac{1 - \mu_f}{2} + n^2\beta^2 + \frac{\beta^2}{r}\right)e_n' = 0 \quad (A7b)$$

$$(1 + n^2\beta^2)b_n + \left(1 + \frac{1 - \mu_f}{2} n^2\beta^2 + \frac{\beta^2}{r}\right)c_n' - \frac{1 + \mu_f}{2} n\beta f_n' = 0 \quad (A7c)$$

$$n\beta(1 + n^2\beta^2)b_n + \frac{1 + \mu_f}{2} n\beta c_n' - \left(\frac{1 - \mu_f}{2} + n^2\beta^2 + \frac{\beta^2}{r}\right)f_n' = 0 \quad (A7d)$$

$$\begin{aligned} &(1 + n^2\beta^2)^2 a_n - (1 + n^2\beta^2)d_n' - n\beta(1 + n^2\beta^2)e_n' \\ &+ \frac{8}{\pi} \beta^3 k \sum_l \frac{n l}{n^2 - l^2} b_l = 0 \end{aligned} \quad (A7e)$$

$$\begin{aligned} &(1 + n^2\beta^2)^2 b_n + (1 + n^2\beta^2)c_n' - n\beta(1 + n^2\beta^2)f_n' \\ &- \frac{8}{\pi} \beta^3 k \sum_m \frac{nm}{n^2 - m^2} a_m = 0 \end{aligned} \quad (A7f)$$

where $n = 1, 2, 3, \dots$ and l and m are integers such that $n + l$ is odd and $n + m$ is odd.

The criterion for the shear buckling of infinitely long simply supported Metalite type sandwich plates is determined from the condition that the Fourier coefficients have values other than zero; that is, that the plate buckles. The determinant of the coefficients of these terms in equations (A7) must then be equal to zero. The determinant obtained from equations (A7) is rather unwieldy, however, and may be simplified.

Simplification of stability determinant.— Equations (A7a) to (A7d) may be solved to give expressions for d_n' and e_n' in terms of a_n , and for e_n' and f_n' in terms of b_n . These expressions are

$$\begin{aligned} \frac{d_n'}{a_n} &= -\frac{c_n'}{b_n} \\ &= \frac{1}{n\beta} \frac{e_n'}{a_n} \\ &= \frac{1}{n\beta} \frac{f_n'}{b_n} \\ &= (1 + n^2\beta^2) \frac{\frac{1 - \mu_f}{2}(1 + n^2\beta^2) + \frac{\beta^2}{r}}{\left(1 + \frac{1 - \mu_f}{2} n^2\beta^2 + \frac{\beta^2}{r}\right)\left(\frac{1 - \mu_f}{2} + n^2\beta^2 + \frac{\beta^2}{r}\right) - \left(\frac{1 + \mu_f}{2} n\beta\right)^2} \end{aligned} \quad (A8)$$

where $n = 1, 2, 3, \dots$. Substitution of equations (A8) in equations (A7e) and (A7f) then yields

$$\begin{aligned} (1 + n^2\beta^2)^2 \left\{ 1 - \frac{(1 + n^2\beta^2) \left[\frac{1 - \mu_f}{2}(1 + n^2\beta^2) + \frac{\beta^2}{r} \right]}{\left(1 + \frac{1 - \mu_f}{2} n^2\beta^2 + \frac{\beta^2}{r}\right)\left(\frac{1 - \mu_f}{2} + n^2\beta^2 + \frac{\beta^2}{r}\right) - \left(\frac{1 + \mu_f}{2} n\beta\right)^2} \right\} a_n \\ + \frac{8}{\pi} \beta^3 k \sum_l \frac{nl}{n^2 - l^2} b_l = 0 \end{aligned} \quad (A9a)$$

$$\begin{aligned} (1 + n^2\beta^2)^2 \left\{ 1 - \frac{(1 + n^2\beta^2) \left[\frac{1 - \mu_f}{2}(1 + n^2\beta^2) + \frac{\beta^2}{r} \right]}{\left(1 + \frac{1 - \mu_f}{2} n^2\beta^2 + \frac{\beta^2}{r}\right)\left(\frac{1 - \mu_f}{2} + n^2\beta^2 + \frac{\beta^2}{r}\right) - \left(\frac{1 + \mu_f}{2} n\beta\right)^2} \right\} b_n \\ - \frac{8}{\pi} \beta^3 k \sum_m \frac{nm}{n^2 - m^2} a_m = 0 \end{aligned} \quad (A9b)$$

where $n = 1, 2, 3, \dots$ and $n + l$ is odd and $n + m$ is odd.

Equations (A9) may be divided into two independent groups. One group of equations involves only the Fourier coefficients

$$a_2, a_4, a_6, \dots ; b_1, b_3, b_5, \dots$$

whereas the other involves only the Fourier coefficients

$$a_1, a_3, a_5, \dots ; b_2, b_4, b_6, \dots$$

The determinants of these two groups of equations are identical. In the final simplification of the stability determinant, therefore, only one of the groups of equations need be considered.

Those equations (A9a) that involve only Fourier coefficients of the first kind may be solved to give the coefficients a_2, a_4, a_6, \dots in terms of the coefficients b_1, b_3, b_5, \dots . These expressions are then substituted into the appropriate equations (A9b) to yield

$$b_n + \left(\frac{8}{\pi} \beta^3 k\right)^2 \sum_i \sum_j M_{ni} M_{1j} b_j = 0 \tag{A10}$$

where both M_{ni} and M_{1j} are of the form

$$M_{st} = \frac{st}{(s^2 - t^2)(1 + s^2 \beta^2)^2} \frac{1}{1 - \frac{(1 + s^2 \beta^2) \left[\frac{1 - \mu_f}{2} (1 + s^2 \beta^2) + \frac{\beta^2}{r} \right]}{\left(1 + \frac{1 - \mu_f}{2} s^2 \beta^2 + \frac{\beta^2}{r}\right) \left(\frac{1 - \mu_f}{2} + s^2 \beta^2 + \frac{\beta^2}{r}\right) - \left(\frac{1 + \mu_f}{2} s \beta\right)^2}$$

and

$$n = 1, 3, 5, \dots$$

$$i = 2, 4, 6, \dots$$

$$j = 1, 3, 5, \dots$$

Solution of stability determinant.— The stability determinant to be solved is as follows:

	j = 1	j = 3	j = 5
n = 1	$1 + \left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{1i}M_{i1}$	$\left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{1i}M_{i3}$	$\left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{1i}M_{i5}$
n = 3	$\left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{3i}M_{i1} + \sum_{i=2,4,6,\dots}^{\infty} M_{3i}M_{i3}$	$1 + \left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{3i}M_{i3}$	$\left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{3i}M_{i5}$
n = 5	$\left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{5i}M_{i1} + \sum_{i=2,4,6,\dots}^{\infty} M_{5i}M_{i3}$	$\left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{5i}M_{i3} + \sum_{i=2,4,6,\dots}^{\infty} M_{5i}M_{i5}$	$\left(\frac{8\beta^3 k}{\pi}\right)^2 \sum_{i=2,4,6,\dots}^{\infty} M_{5i}M_{i5}$
...
			= 0 (All)

Equation (All) cannot be solved exactly and approximate methods must be used to obtain values of the elastic shear-buckling-stress coefficient k. The method used in the present paper is to approximate the infinite stability determinant by finite subdeterminants, in effect to use an infinite number of Fourier coefficients a_n , d_n , and e_n , but only a finite number of Fourier coefficients b_n , c_n , and f_n . A first approximation is given by the first order determinant as

$$\left(\frac{8}{\pi} \beta^3 k\right)^2 = - \frac{1}{\sum_{i=2,4,6,\dots}^{\infty} M_{1i}M_{i1}} \tag{A12}$$

The second approximation is given by the second-order determinant as

$$\begin{aligned} & \left(\frac{8}{\pi} \beta^3 k\right)^4 \left[\left(\sum_{i=2,4,6,\dots}^{\infty} M_{1i} M_{11} \right) \left(\sum_{i=2,4,6,\dots}^{\infty} M_{3i} M_{13} \right) \right. \\ & \quad \left. - \left(\sum_{i=2,4,6,\dots}^{\infty} M_{1i} M_{13} \right) \left(\sum_{i=2,4,6,\dots}^{\infty} M_{3i} M_{11} \right) \right] \\ & + \left(\frac{8}{\pi} \beta^3 k\right)^2 \left(\sum_{i=2,4,6,\dots}^{\infty} M_{1i} M_{11} + \sum_{i=2,4,6,\dots}^{\infty} M_{3i} M_{13} \right) + 1 = 0 \quad (A13) \end{aligned}$$

The shear-buckling-stress coefficient k is a function of the parameters r and β . For a given value of r the correct value of β is that which yields the lowest value of k . In the computations made for the present paper, therefore, the parameter β was varied until a minimum value of k was obtained for a given value of r . The results of the computations made with equations (A12) and (A13) are summarized in table 1. The values of k for $r = 0$ and $r = 1.0$ obtained from the second approximation are seen to agree excellently with the known exact values. The exact value of k for $r = 0$ was obtained from reference 11 and agrees with the second approximation to two decimal places. The exact value of k for $r = 1.0$ was obtained as a result of physical reasoning, which is explained in the next section, and is only 1 percent lower than the second approximation. It may reasonably be concluded that the approximate values of k for the intermediate values of r are also within 1 percent of the corresponding exact values.

Stability criterion when $r > 1$.— A shear buckling mode that is possible for Metalite type sandwich plates is shown in figure 4. The plates may buckle with straight nodal lines inclined at an angle of 45° to the direction of the shear stress and with an infinitely small wave length. If an element of plates between consecutive nodal lines is isolated (fig. 4), it is seen to be subjected to a compressive stress which is equal in magnitude to the critical shear stress. The element is therefore equivalent to a very short compressed simply supported plate for which the buckling load is given by equation (1) of reference 6 as

$$k = \frac{1}{r} \quad (A14)$$

The computations of the present paper indicate that the wave length of buckle rapidly decreases with increasing r so that the shear-buckling-stress coefficients approach those of compressed simply supported plates. When r is equal to or greater than 1.0 the shear-buckling-stress coefficients are given exactly by equation (A14).

As was noted in reference 6, this result is a consequence of the assumption implied by the sandwich-plate theory of reference 9, that the stiffness of the plate faces in bending about their own middle surface is negligible. The plate is then free to buckle as shown in section A-A of figure (4); that is, buckling is due to the core shearing in planes perpendicular to the faces, with no interference due to bending of the faces.

REFERENCES

1. March, H. W.: Effects of Shear Deformation in the Core of a Flat Rectangular Sandwich Panel. Rep. No. 1583, Forest Products Lab., U. S. Dept. Agric., May 1948.
2. Bijlaard, P. P.: On the Elastic Stability of Sandwich Plates. I. Koninklijke Nederlandsche Akad. Wetenschappen. Reprinted from Proc., vol. L, no. 1, 1947, pp. 79-87.
3. Plantema, F. J.: Some Investigations on the Euler Instability of Flat Sandwich Plates with Simply-Supported Edges. Rep. S.337, Nationaal Luchtvaartlaboratorium, Amsterdam, June 1948.
4. Reissner, Eric: Finite Deflections of Sandwich Plates. Jour. Aero. Sci., vol. 15, no. 7, July 1948, pp. 435-440.
5. Seide, Paul, and Stowell, Elbridge Z.: Elastic and Plastic Buckling of Simply Supported Metalite Type Sandwich Plates in Compression. NACA TN 1822, 1949.
6. Seide, Paul: Compressive Buckling of Flat Rectangular Metalite Type Sandwich Plates with Simply Supported Loaded Edges and Clamped Unloaded Edges. NACA TN 1886, 1949.
7. Bijlaard, P. P.: On the Elastic Stability of Sandwich Plates. II. Koninklijke Nederlandsche Akad. Wetenschappen. Reprinted from Proc., vol. L, no. 2, 1947, pp. 186-193.
8. Libove, Charles, and Batdorf, S. B.: A General Small-Deflection Theory for Flat Sandwich Plates. NACA TN 1526, 1948.
9. Plantema, F. J., and de Kock, A. C.: The Elastic Overall Instability of Sandwich Plates with Simply-Supported Edges. Rep. S.346, Nationaal Luchtvaartlaboratorium, Amsterdam, Dec. 1948.
10. Stowell, Elbridge Z.: Critical Shear Stress of an Infinitely Long Plate in the Plastic Region. NACA TN 1681, 1948.
11. Southwell, R. V., and Skan, Sylvia W.: On the Stability under Shearing Forces of a Flat Elastic Strip. Proc. Roy. Soc. (London), ser. A, vol. 105, no. 733, May 1, 1924, pp. 582-607.

TABLE I
 SHEAR-BUCKLING COEFFICIENTS FOR INFINITELY LONG SIMPLY
 SUPPORTED METALITE TYPE SANDWICH PLATES

$$\left[\mu_f = \frac{1}{3} \right]$$

r	k_{xy}		
	First approximation	Second approximation	Exact solution
0	5.56	5.34	5.34
.05	4.80	4.55	----
.1	4.23	3.95	----
.2	3.42	3.11	----
.4	2.48	2.14	----
.6	1.94	1.58	----
1.0	1.34	1.01	1.00


 NACA

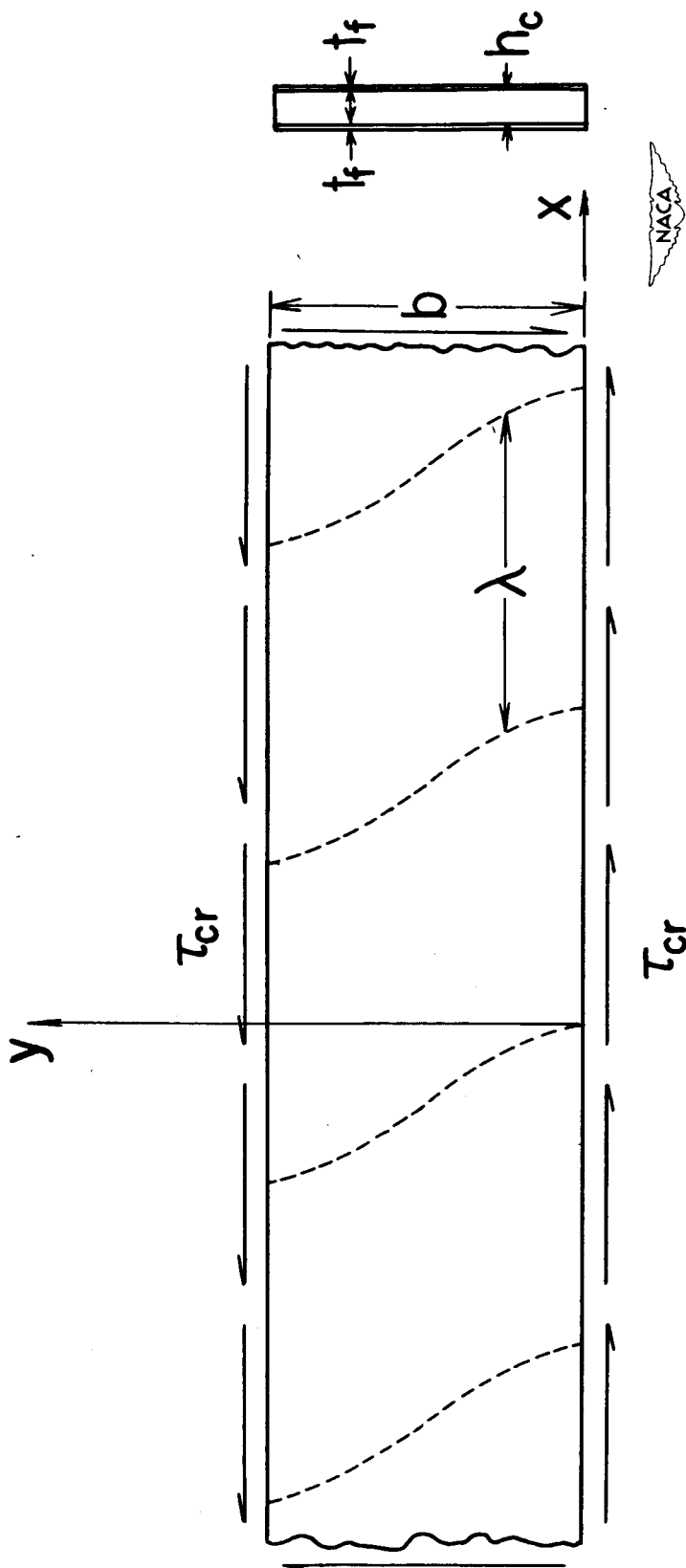


Figure 1.— Infinitely long simply supported Metalite type sandwich plate under shear.

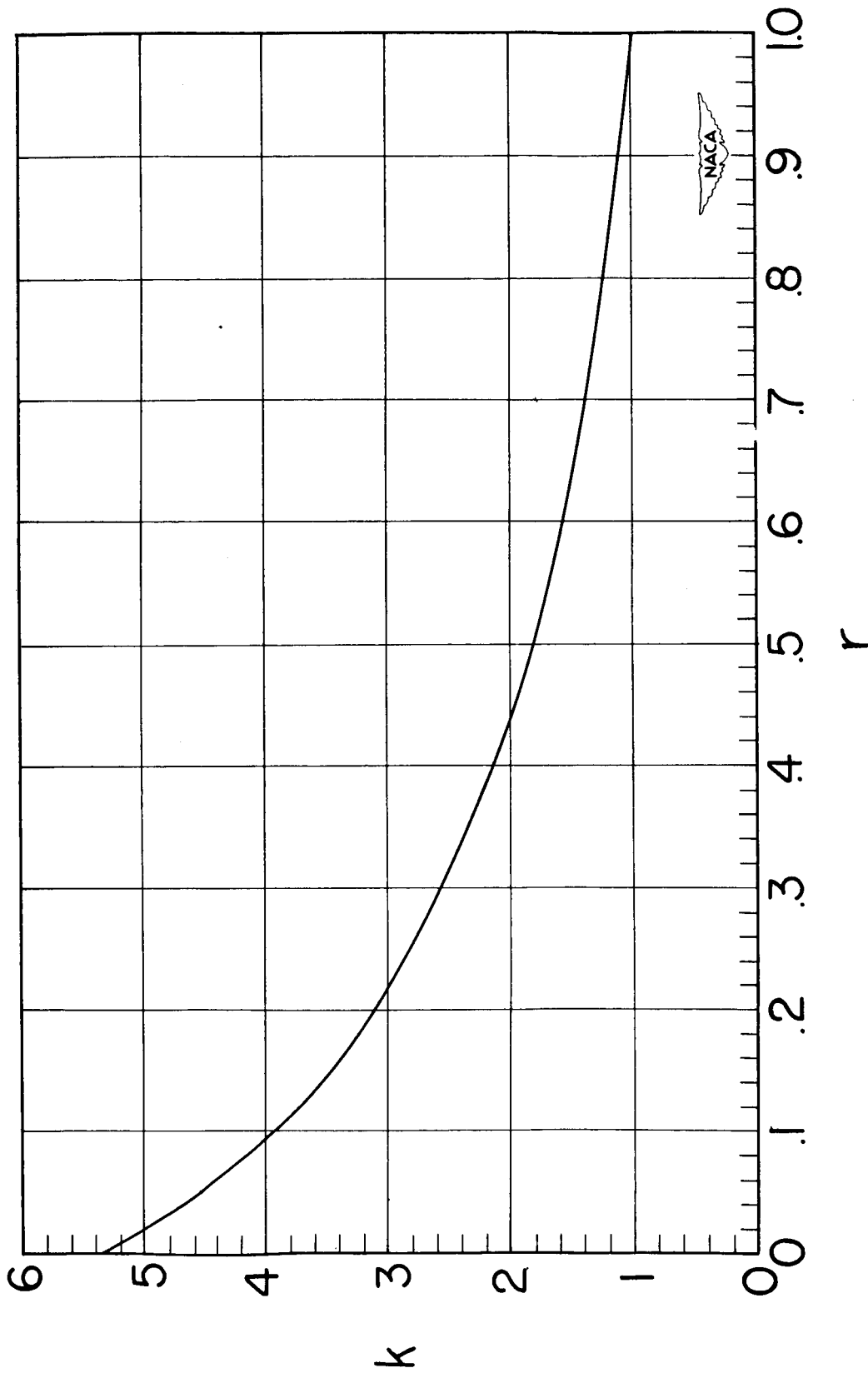


Figure 2.— Elastic shear-buckling-stress coefficients for infinitely long simply supported Metalite type sandwich plates $\left(\mu_f = \frac{1}{3}\right)$.

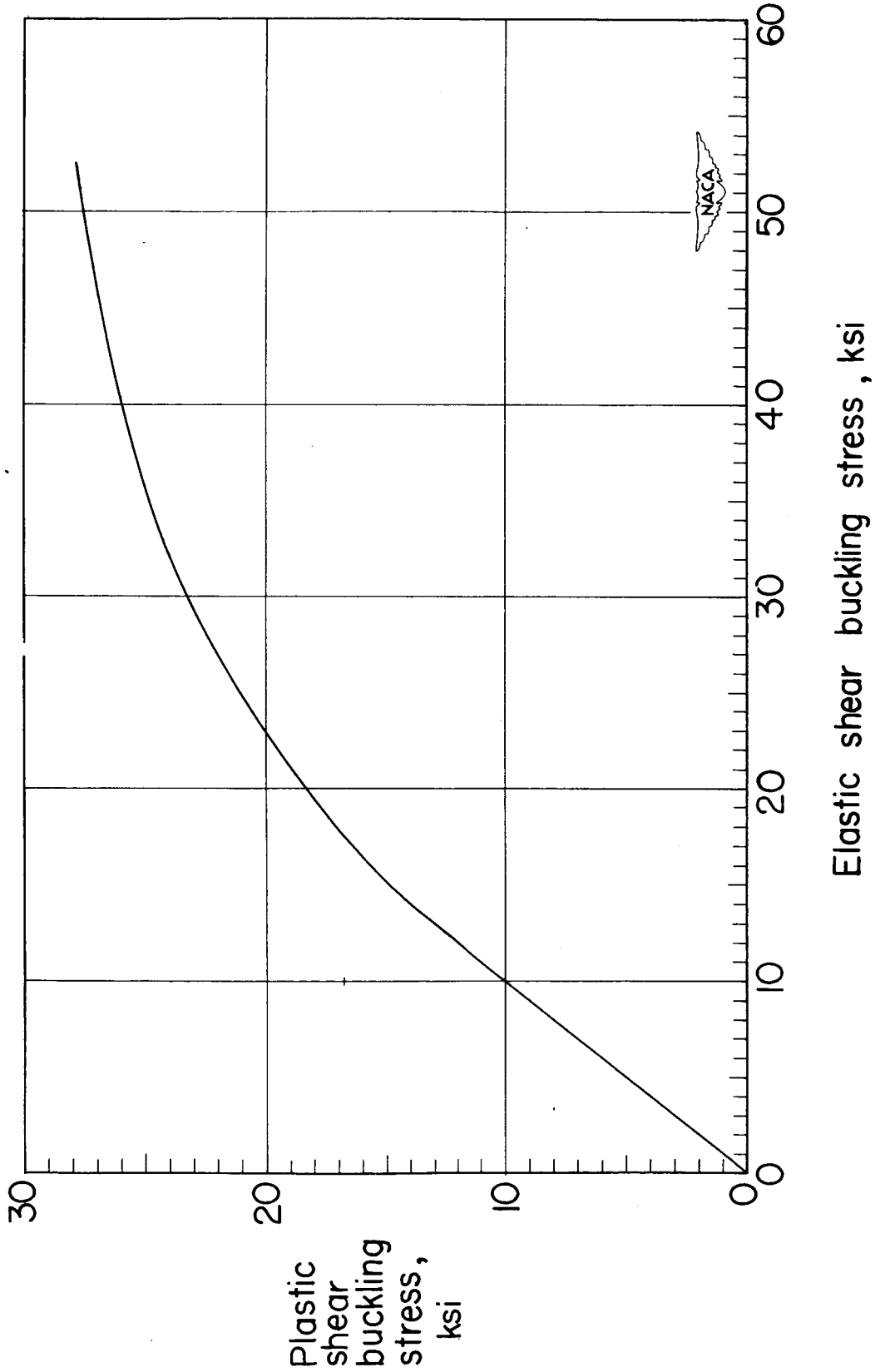


Figure 3.— Relationship between shear buckling stresses for an infinitely long simply supported plate computed from theories of elastic and plastic buckling of 24S-T3 aluminum alloy.

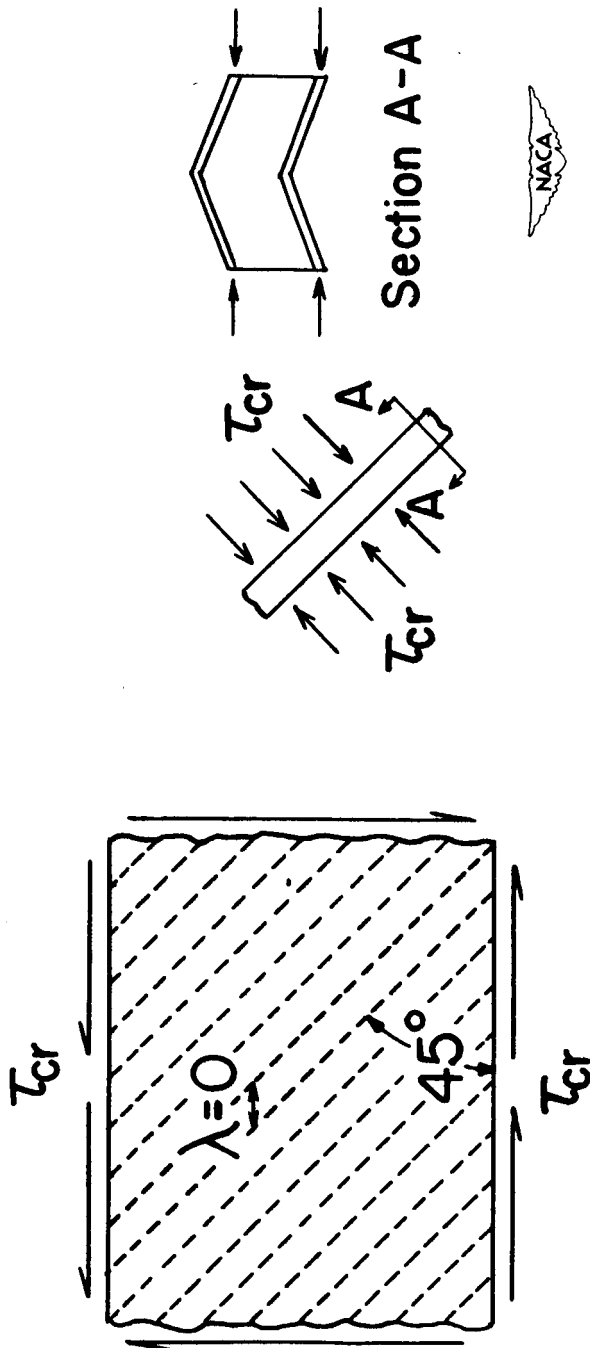


Figure 4.— Buckled shape of plate with core of low shear stiffness.