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On Axial Load Diffusion into a Thin-Walled Reinforced Cylindrical Shell

by

E. H. Mansfield, Sc.D., F.R.Ae.S., A.F.I.A.S.

LONDON: HER MAJESTY'S STATIONERY OFFICE

1963

FOUR SHILLINGS NET

U.D.C. No. 531.259.2 : 539.384.4

C.P. No. 644
August, 1962

ON AXIAL LOAD DIFFUSION INTO A THIN-WALLED REINFORCED CYLINDRICAL SHELL

by

E.H. Mansfield, Sc.D., F.R.Ae.S., A.F.I.A.S.

SUMMARY

An approximate analysis is developed, based on the concept of the incomplete tension field, for investigating the load-diffusion characteristics of a cylinder with very thin skin reinforced with stringers.

Replaces R.A.E. Tech. Note No. Structures 318 - A.R.C.24,414.

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1 INTRODUCTION

When a concentrated axial load is diffused into a cylinder with very thin skin reinforced with stringers, the shears developed in the skin will cause buckling at a low stress. After the skin has buckled, its shear carrying ability changes and depends to an increasing extent on the transfer of tension loads along the direction of the buckles, i.e. a tension field develops. An assessment of the strength of the structure and the rate of load diffusion must therefore take account of the buckled state of the skin. After buckling, the stress at any point in the cylinder will vary in a non-linear manner with the applied load, although the structure may be everywhere elastic. A rigorous determination of the varying stress pattern is out of the question, but an approximate solution may be found by using the concept of the incomplete tension field. A pure tension field would exist in a strip between stringers if the skin carried no compressive stresses, and the behaviour of such an idealised skin may be accurately predicted. The actual behaviour of the skin may then be estimated by introducing an empirical tension field factor which depends on the degree to which the pure tension field conditions have developed.

2 LIST OF SYMBOLS

E = Youngs modulus

G = shear modulus

R = radius of cylinder

t = skin thickness

a = frame pitch

b = stringer pitch

b' = circumferential distance between stringers

A = section area of stringer

n = number of equally spaced stringers

x,y = Cartesian axes in plane of adjacent stringers, x measured along stringer-direction

u,v = displacements associated with x,y

α = angle that pure tension field waves make with stringer

X,Y = Cartesian co-ordinates making angle α with x,y

U,V = displacements associated with X,Y

$\varepsilon_x, \varepsilon_y, \gamma_s$ = direct strains and shear strain associated with x,y

γ_{XY} = shear strain associated with X,Y

σ_X = direct stress in pure tension field

τ_s = shear stress adjacent to stringer in pure tension field

$\sigma_{n,s}$ = stress normal to stringer in pure tension field

$\sigma_{p,s}$ = stress parallel to stringer in pure tension field

k	=	tension field factor
σ_n	=	actual panel stress normal to stringer
τ	=	actual shear stress in panel adjacent to stringer
γ	=	actual shear strain in panel adjacent to stringer
σ_p	=	actual direct stress in panel measured parallel to stringer
τ'	=	component of τ with pre-buckling characteristics
τ_{cr}	=	value of τ to cause buckling
$\varepsilon_{x,cr}$	=	value of ε_x to cause buckling
τ_{eff}		defined by equation (24)
$\bar{\tau}_s, \bar{\gamma}, \bar{\varepsilon}_x, \bar{\sigma}_{n,s}, \bar{\sigma}_{p,s}$	=	non-dimensional terms defined by equation (17)
N_r	=	radially directed loading/unit length on a stringer
M	=	bending moment in stringer due to N_r
L_m^-	=	tensile stringer load to left of station m
L_m^+	=	tensile stringer load to right of station m

3 PURE TENSION FIELD IN A STRIP

The analysis is given here for a pure tension field in a longitudinal strip in a cylinder. The edges of the strip, where in practice the stringers would be, are assumed to be under a uniform compressive strain. The curvature of the sheet is introduced by regarding the strip as an initially flat strip under a uniform lateral compressive strain, which will depend on the stringer pitch and the cylinder radius. The problem is now to determine the stress pattern in the strip when subjected to a uniform shear strain. The notation is shown in Fig.1.

With reference axes Ox, Oy the displacements are given by

$$\left.\begin{aligned} u &= y\gamma_s + x\varepsilon_x \\ v &= y\varepsilon_y \quad . \end{aligned}\right\} \qquad (1)$$

It is convenient to relate these displacements to axes OX, OY parallel and normal to the direction (at present unknown) of the tension field waves. Thus

$$\left.\begin{aligned} U &= u \cos\alpha + v \sin\alpha \\ V &= v \cos\alpha - u \sin\alpha \end{aligned}\right\} \qquad (2)$$

and

$$\left.\begin{aligned} x &= X \cos\alpha - Y \sin\alpha \\ y &= Y \cos\alpha + X \sin\alpha . \end{aligned}\right\} \qquad (3)$$

From equations (1), (2) and (3) the strains referred to the X, Y axes are given by

$$\left.\begin{aligned}
\gamma_{XY} &= \gamma_s \cos 2\alpha + (\varepsilon_y - \varepsilon_x) \sin 2\alpha \\
\frac{\partial U}{\partial X} &= \tfrac{1}{2}\gamma_s \sin 2\alpha + \tfrac{1}{2}\varepsilon_y (1 - \cos 2\alpha) + \tfrac{1}{2}\varepsilon_x (1 + \cos 2\alpha) \\
\frac{\partial V}{\partial Y} &= -\tfrac{1}{2}\gamma_s \sin 2\alpha + \tfrac{1}{2}\varepsilon_y (1 + \cos 2\alpha) + \tfrac{1}{2}\varepsilon_x (1 - \cos 2\alpha).
\end{aligned}\right\} \qquad (4)$$

Now these strains are principal strains, so that the angle α may be determined from the condition that γ_{XY} is zero, whence

$$\tan 2\alpha = \frac{\gamma_s}{(\varepsilon_x - \varepsilon_y)} . \qquad (5)$$

The stress along the direction of the waves is given in terms of ε_x, ε_y, γ_s by eliminating α from equations (4) and (5). Thus

$$\begin{aligned}
\sigma_X &= E \frac{\partial U}{\partial X} \\
&= \tfrac{1}{2}E \left[\left(\varepsilon_x + \varepsilon_y\right) + \left\{\gamma_s^2 + \left(\varepsilon_y - \varepsilon_x\right)^2\right\}^{\frac{1}{2}} \right] .
\end{aligned} \qquad (6)$$

Similarly, the strain normal to the tension field waves is given by

$$\frac{\partial V}{\partial Y} = \tfrac{1}{2}\left[\left(\varepsilon_x + \varepsilon_y\right) - \left\{\gamma_s^2 + \left(\varepsilon_x - \varepsilon_y\right)^2\right\}^{\frac{1}{2}} \right]$$

which is necessarily negative because ε_x and ε_y are negative.

Along the edges of the strip the stress σ_X may be resolved into a shear stress and a direct stress normal to the edges. Thus,

$$\begin{aligned}
\tau_s &= \sigma_X \sin\alpha \cos\alpha \\
&= \tfrac{1}{4}E\gamma_s \left[1 + \frac{\varepsilon_x + \varepsilon_y}{\left\{\gamma_s^2 + \left(\varepsilon_x - \varepsilon_y\right)^2\right\}^{\frac{1}{2}}} \right]
\end{aligned} \qquad (7)$$

and

$$\sigma_{n,s} = \sigma_X \sin^2 \alpha$$

$$= \tfrac{1}{4}E \left[2\varepsilon_y + \frac{\gamma_s^2 - 2\varepsilon_y(\varepsilon_x - \varepsilon_y)}{\left\{\gamma_s^2 + \left(\varepsilon_x - \varepsilon_y\right)^2\right\}^{\frac{1}{2}}} \right] . \qquad (8)$$

Across the strip the stress σ_X may be resolved into the shear stress τ_s and a direct stress $\sigma_{p,s}$ parallel to the edges, where

$$\sigma_{p,s} = \sigma_X \cos^2 \alpha$$

$$= \tfrac{1}{4}E \left[2\varepsilon_x + \frac{\gamma_s^2 + \varepsilon_x(\varepsilon_x - \varepsilon_y)}{\left\{\gamma_s^2 + \left(\varepsilon_x - \varepsilon_y\right)^2\right\}^{\frac{1}{2}}} \right] \qquad (9)$$

3.1 Range of validity of the equations

The equations presented so far are based on the tacit assumption that a tension field exists. If, however, the longitudinal edge strain is large and negative (i.e. compressive) and the shear strain γ_s small, it is possible that compressive strains exist everywhere so that a tension field does not develop. The criterion for the existence of a tension field is that σ_X, given by equation (6) is positive; if ε_x and ε_y are both negative, this reduces to

$$\gamma_s^2 > 4\varepsilon_x\varepsilon_y \quad . \qquad (10)$$

If a tension field does not exist

$$\gamma_s^2 < 4\varepsilon_x\varepsilon_y$$

and

$$\sigma_X = 0 \quad .$$

3.2 Expression for ε_y

There are three components contributing to ε_y, first, that due to the curvature of the sheet, second, that due to radial displacements of the stringers and, third, that due to circumferential displacements of the stringers. Of these three components that due to the curvature is probably the most important and it is the only component which is independent of the applied loading.

If there are n stringers equally spaced around the circumference of the cylinder, the curved length of skin between stringers is given by

$$b' = \frac{2\pi R}{n}$$

and the "flat" distance between stringers is given by

$$b = 2R \sin\left(\frac{\pi}{n}\right)$$

$$\approx b'\left(1 - \frac{\pi^2}{3n^2}\right)$$

so that

$$\varepsilon_{y,\text{curve}} \approx \frac{-3\cdot 3}{n^2} . \tag{11}$$

In what follows it will be assumed that this component of ε_y is the only one that is significant: that due to radial displacement being virtually prevented by the rigidity of the stringers, and that due to circumferential displacement being virtually prevented by the rigidity and closeness of the frames.

3.3 Special cases

Consider now some special cases of pure tension fields which exhibit the type of non-linear stress-strain relationship.

In a few problems the term $-\varepsilon_y$ will be sensibly constant and large in comparison with γ_s, and γ_s will be large in comparison with ε_x. Consider therefore the limiting case in which

$$\left.\begin{aligned} \gamma_s^2 &<< \varepsilon_y^2 \\ \text{and} \qquad \varepsilon_x &= 0 . \end{aligned}\right\} \tag{12}$$

Substituting in equations (7) and (8) and omitting terms of higher order gives

$$\left.\begin{aligned} \frac{\tau_s}{E\varepsilon_y} &= \left(\frac{\gamma_s}{2\varepsilon_y}\right)^3 \\ \text{and} \qquad \frac{\sigma_{n,s}}{E\varepsilon_y} &= -\left(\frac{\gamma_s}{2\varepsilon_y}\right)^4 , \end{aligned}\right\} \tag{13}$$

or, regarding τ_s as the independent variable,

$$\left.\begin{aligned} \frac{\gamma_s}{\varepsilon_y} &= -2\left(\frac{\tau_s}{-E\varepsilon_y}\right)^{1/3} \\ \text{and} \qquad \frac{\sigma_{n,s}}{E\varepsilon_y} &= -\left(\frac{\tau_s}{-E\varepsilon_y}\right)^{4/3} . \end{aligned}\right\} \tag{14}$$

Equation (14) is useful in showing the type of non-linear behaviour existing in a pure tension field in an initially curved strip with simple boundary conditions. The restriction on the range of validity of equation (14), embodied in equation (12), is equivalent to the fact that the angle α must be small. When

$$\gamma_s >> \varepsilon_x - \varepsilon_y \tag{15}$$

the wave angle α tends to $\frac{1}{4}\pi$ and equations (7) and (8) reduce to the results for a simple Wagner tension field, namely

$$\left.\begin{aligned} \tau_s &= \tfrac{1}{4}E\gamma_s \\ \text{and} \qquad \sigma_n &= \tfrac{1}{4}E\gamma_s . \end{aligned}\right\} \tag{16}$$

In practice the variation of γ_s and σ_n with τ_s varies between the limiting conditions represented by equations (14) and (16).

3.4 Use of non-dimensional parameters

The assumption was made in Para.3.2 that ε_y was constant and given by equation (11). This being so, it is convenient to introduce the non-dimensional parameters:

$$\left.\begin{aligned} \bar{\tau}_s &= \tau_s/(-E\varepsilon_y) &= 0{\cdot}3\ n^2\tau_s/E \\ \bar{\gamma}_s &= \gamma_s/(-\varepsilon_y) &= 0{\cdot}3\ n^2\gamma_s \\ \bar{\varepsilon}_x &= \varepsilon_x/(-\varepsilon_y) &= 0{\cdot}3\ n^2\varepsilon_x \\ \bar{\sigma}_{n,s} &= \sigma_{n,s}/(-E\varepsilon_y) &= 0{\cdot}3\ n^2\sigma_{n,s}/E \\ \bar{\sigma}_{p,s} &= \sigma_{p,s}/(-E\varepsilon_y) &= 0{\cdot}3\ n^2\sigma_{p,s}/E . \end{aligned}\right\} \tag{17}$$

Equations (7), (8) and (9) then assume the simpler forms:

$$\bar{\tau}_s = \tfrac{1}{4}\bar{\gamma}_s\left[1 - \frac{1 - \bar{\varepsilon}_x}{\left\{\bar{\gamma}_s^2 + \left(1 + \bar{\varepsilon}_x\right)^2\right\}^{\frac{1}{2}}}\right] \tag{18}$$

$$\bar{\sigma}_{n,s} = \frac{\bar{\gamma}_s^2 + 2(1 + \bar{\varepsilon}_x)}{4\left\{\bar{\gamma}_s^2 + \left(1 + \bar{\varepsilon}_x\right)^2\right\}^{\frac{1}{2}}} - \frac{1}{2} \tag{19}$$

$$\bar{\sigma}_{p,s} = \tfrac{1}{2}\bar{\varepsilon}_x + \frac{\bar{\gamma}_s^2 + 2\bar{\varepsilon}_x(1 + \bar{\varepsilon}_x)}{4\left\{\bar{\gamma}_s^2 + \left(1 + \bar{\varepsilon}_x\right)^2\right\}^{\frac{1}{2}}} \tag{20}$$

and the condition for the existence of a tension field is that

$$\bar{\gamma}_s^2 > -4\bar{\varepsilon}_x .$$

4 THE INCOMPLETE TENSION FIELD

The analysis so far is based on the assumption of a pure tension field. From a practical standpoint it is convenient to assume that the shear stress τ can be divided into a pure tension field part τ_s and a simple shear part τ' where

$$\left.\begin{aligned} \tau_s &= k\tau \\ \text{and} \qquad \tau' &= (1-k)\tau \end{aligned}\right\} \qquad (21)$$

For a cylinder under pure torsion (i.e. $\varepsilon_x = 0$ before buckling) Kuhn and Griffith[1] give the following empirical formula for the fraction k (valid for $\tau \geq \tau_{cr}$):

$$k = \tanh\left[\left(0\cdot5 + 300\,\frac{ta}{Rb}\right)\log_{10}\left(\frac{\tau}{\tau_{cr}}\right)\right]$$

where τ_{cr} is the shear stress at the onset of buckling. In the present problem, the strain ε_x is not necessarily zero before buckling and it is preferable to define k by

$$k = \tanh\left[\left(0\cdot5 + 300\,\frac{ta}{Rb}\right)\log_{10}\left(\frac{\tau_{eff}}{\tau_{cr}}\right)\right] \qquad (22)$$

where τ_{eff} includes (albeit in an empirical way) the effect of longitudinal strains on initial buckling. Now the interaction curve for initial buckling of a curved strip under shear and longitudinal compression is given approximately by

$$\left(\frac{\tau}{\tau_{cr}}\right)^2 + \frac{\varepsilon_x}{\varepsilon_{x,cr}} = 1 \qquad (23)$$

and accordingly τ_{eff} will be defined by

$$\frac{\tau_{eff}}{\tau_{cr}} = \left\{\left(\frac{\tau}{\tau_{cr}}\right)^2 + \frac{\varepsilon_x}{\varepsilon_{x,cr}}\right\}^{\frac{1}{2}} \qquad (24)$$

so that at the onset of buckling $\tau_{eff} = \tau_{cr}$ and $k = 0$. Graphs for determining τ_{cr} and $\varepsilon_{x,cr}$ are reproduced from Ref.2 and shown in Figs.2 and 3. A graph for determining k has been reproduced from Ref.1 in Fig.4. Equation (19) is, of course, only valid for $\tau_{eff} \geq \tau_{cr}$. When $\tau_{eff} < \tau_{cr}$ the skin has not buckled and k is zero.

4.1 Shear stress-strain relation

From equations (7) and (21) the actual shear strain γ is related to the actual shear stress τ by the approximate equation

$$\begin{aligned} \gamma &= \frac{\tau'}{G} + k\gamma_s \\ &= (1-k)\tau/G + k\gamma_s \end{aligned}$$

which may be written in the non-dimensional form:

$$\left.\begin{aligned} \bar{\gamma} &= 2\cdot 5\ \bar{\tau}(1-k) + k\bar{\gamma}_s \\ \text{where}\qquad \bar{\gamma} &= \gamma/(-\varepsilon_y)\ . \end{aligned}\right\} \qquad (25)$$

Equation (25) has been plotted in Fig.5 for various values of $\bar{\varepsilon}_x$, for the particular case in which

$$\left.\begin{aligned} t &= 0\cdot 2 \text{ in.} \\ a &= 24 \text{ in.} \\ b &= 6 \text{ in.} \\ R &= 60 \text{ in.} \\ E &= 30 \times 10^6 \text{ lb/in}^2\ . \end{aligned}\right\} \qquad (26)$$

This stress-strain relationship cannot be expected to be very reliable in the region of small $\bar{\gamma}$ and large negative $\bar{\varepsilon}_x$ (corresponding to a highly buckled skin) for the tension-field approach will underestimate the shear stiffness of the skin. Accordingly it is suggested that the linear relationship represented by the line OA be used in this region. From a practical standpoint this particular region is not important.

4.2 Actual stress normal to stringer

The actual direct panel stress normal to the stringers is given approximately by

$$\sigma_n = k\sigma_{n,s}\ . \qquad (27)$$

4.3 Actual stress parallel to stringer

The actual direct panel stress measured parallel to the stringers is given approximately by

$$\sigma_p = k\sigma_{p,s} + (1-k)\ E\varepsilon_x\ . \qquad (28)$$

This equation shows that in a well developed tension field ($k \to 1$) σ_p will be tensile despite the fact that ε_x may be negative. Such tensions in the skin must be equilibrated by compressions in the stringers, a point discussed in detail in Section 5. It will be noted that across any section of the cylinder the total compressive load in the stringers may exceed the total applied compressive load.

4.4 Radial loading on the stringers

The component of direct stress normal to a stringer and in the plane of adjacent stringers is given by equation (27). Now successive planes between adjacent stringers meet at an angle $2\pi/n$, and the skin in each plane therefore exerts a radially directed distributed loading on each of the adjacent stringers given by

$$N_r = \left(\frac{\pi kt}{n}\right)\sigma_{n,s} \qquad (29)$$

and the total radially directed loading on a stringer will be the sum of two such terms. Although such a loading is not in itself large, its effect on the bending stresses in the stringers is not negligible. For example, if the term $\sigma_{n,s}$ remained sensibly constant over a number of stringer and frame pitches, the stringers are effectively clamped at each frame (from symmetry) and the maximum bending moment in each stringer occurs at the frames and is given by

$$M = \left(\frac{\pi a^2 kt}{12n}\right) \sigma_{n,s} . \tag{30}$$

5 APPLICATION TO REINFORCED CYLINDER

The analysis so far determines, in an approximate manner, the sheet stresses in a long strip in terms of the edge displacements. In applying these results to panels bounded by stringers and frames it is necessary to make further assumptions, namely

(i) the stress distribution in each panel is constant,

(ii) the stress distribution depends only on the average strains in each panel, and hence only on the corner displacements of the panel,

(iii) direct forces in the stringers vary linearly between frames,

(iv) discontinuities in σ_p across a frame are transmitted via the frame equally to the adjacent stringers which maintain longitudinal equilibrium,

(v) discontinuities in σ_n across a stringer are transmitted via the lateral rigidity of the stringers to the frames, but no account is taken of the resulting change in frame strain.

With these assumptions in mind it is appropriate to consider the equilibrium position of a typical station O surrounded by four panels. The skin thicknesses and stringer areas may vary from panel to panel but are assumed constant in each panel. The numbering of the surrounding eight stations is shown in Fig.6.

The average shear strain and the average longitudinal strain in each of the four panels is then expressed in terms of the u-displacements of the surrounding stations:

$$\left.\begin{aligned} \gamma_1 &= (u_3 + u_4 - u_5 - u_o)/2b \quad , \quad \gamma_2 = (u_2 + u_3 - u_o - u_1)/2b \\ \gamma_3 &= (u_5 + u_o - u_6 - u_7)/2b \quad , \quad \gamma_4 = (u_o + u_1 - u_7 - u_8)/2b \end{aligned}\right\} \tag{31}$$

and

$$\left.\begin{aligned} \varepsilon_{x,1} &= (u_3 + u_o - u_4 - u_5)/2a \\ \varepsilon_{x,2} &= (u_1 + u_2 - u_3 - u_o)/2a \\ \varepsilon_{x,3} &= (u_o + u_7 - u_5 - u_6)/2a \\ \varepsilon_{x,4} &= (u_1 + u_8 - u_o - u_7)/2a \end{aligned}\right\} \tag{32}$$

and the average strains in the stringers are given by

$$\left.\begin{aligned} \varepsilon_{x,43} &= (u_3 - u_4)/a \;, & \varepsilon_{x,32} &= (u_2 - u_3)/a \\ \varepsilon_{x,50} &= (u_o - u_5)/a \;, & \varepsilon_{x,01} &= (u_1 - u_o)/a \\ \varepsilon_{x,67} &= (u_7 - u_6)/a \;, & \varepsilon_{x,78} &= (u_8 - u_7)/a \end{aligned}\right\} \quad (33)$$

The shear stress in each panel can now be expressed in terms of the u-displacements by virtue of equations (31) and (25). The panel stress σ_p can likewise be expressed in terms of the u-displacements by virtue of equations (28), (20), (31) and (32).

In considering the equilibrium of the (typical) station 0, it is convenient to regard the surrounding 8 stations as fixed. Further, it is important to distinguish between the stringer loads on either side of a frame for these will generally differ in order to equilibrate differences in σ_p across a frame.

Let therefore,

L_m^- = tensile stringer load to left of station m ,

L_m^+ = tensile stringer load to right of station m .

Equilibrium of the stringer $\overline{50}$ then yields

$$L_5^+ - L_o^- = at\,(\tau_1 - \tau_3) \quad (34)$$

in which, for simplicity, it is assumed that the skin thicknesses in the various panels are the same. The extension to differing thicknesses is straightforward.

Equilibrium of the stringer $\overline{01}$ yields

$$L_o^+ - L_1^- = at\,(\tau_2 - \tau_4) \;. \quad (35)$$

Now

$$\begin{aligned} \tfrac{1}{2}\left(L_5^+ + L_o^-\right) &= \text{average load in stringer } \overline{50} \\ &= AE\varepsilon_{x,50} \;. \end{aligned} \quad (36)$$

Similarly,

$$\tfrac{1}{2}\left(L_o^+ + L_1^-\right) = AE\varepsilon_{x,01} \;. \quad (37)$$

(Variation in stringer section may be readily accommodated in the analysis.)

Eliminating L_5^+ from equations (34) and (36) gives

$$L_o^- = AE\varepsilon_{x,50} - \tfrac{1}{2}\,at\,(\tau_1 - \tau_3) \;. \quad (38)$$

Similarly, from equations (35) and (37),

$$L_o^+ = AE\varepsilon_{x,01} + \tfrac{1}{2}\,at\,(\tau_2 - \tau_4) \; . \qquad (39)$$

The difference in the loads L_o^- and L_o^+ must be equated to the difference in the tensile loads in the skin. Thus

$$L_o^- - L_o^+ = \tfrac{1}{2}\,bt\left(\sigma_{p,2} + \sigma_{p,4} - \sigma_{p,1} - \sigma_{p,3}\right) \; . \qquad (40)$$

Equation (40) determines u_o in terms of u_1, u_2, ... u_8. Repeated applications of equation (40) suffice to solve problems associated with load diffusion into thin-walled reinforced cylinders. The equations are, of course, non-linear and necessitate the use of high powered computer methods.

6 CONCLUSIONS

A simplified and approximate analysis has been presented for investigating the axial load-diffusion behaviour of a cylinder with very thin skin reinforced with stringers. The thin skin buckles at a low stress and the transfer of load between stringers is thereafter effected primarily by tension fields. The initial curvature of the skin between stringers causes a certain slackness in the tension field, and this results in non-linear elastic behaviour of the skin. Because of this non-linearity the solution of the governing equations presents considerable difficulty. In addition to deriving these equations the Note draws attention to the bending stresses induced in the stringers by radial loads resulting from tension fields in adjacent, non-planar skin panels.

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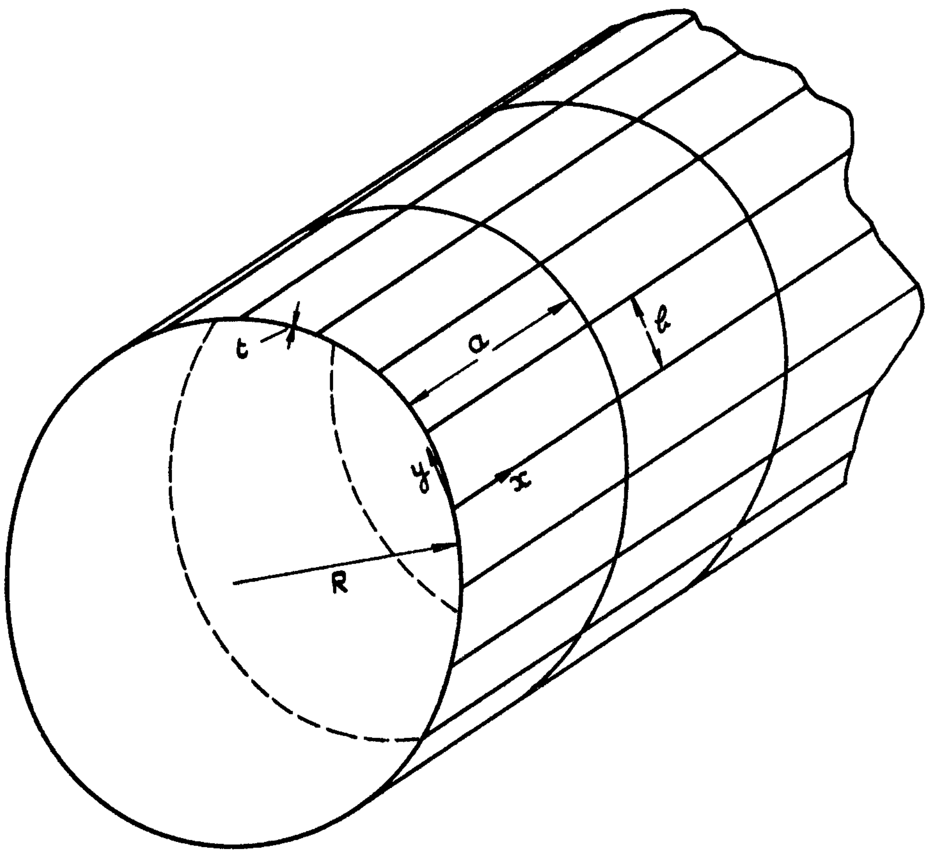
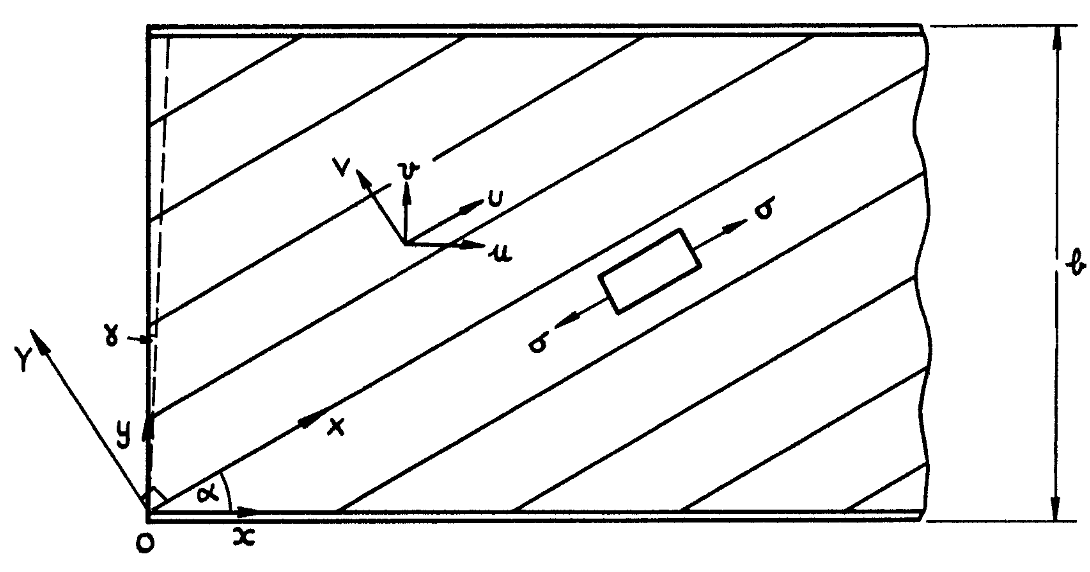
No.	Author	Title, etc.
1	Kuhn, P. Griffith, G.E.	Diagonal tension in curved webs. N.A.C.A. Technical Note No.1481, November, 1947.
2	-	R.Ae.S. Data Sheets Nos. 02.03.18 and 02.01.10.

WT.2078 C.P. 644 K3. Printed in England



(a). CYLINDER SHOWING GENERAL NOTATION.



(b). NOTATION FOR TENSION FIELD STRIP BETWEEN STRINGERS.

FIG.1(a & b). FIGURES SHOWING NOTATION.

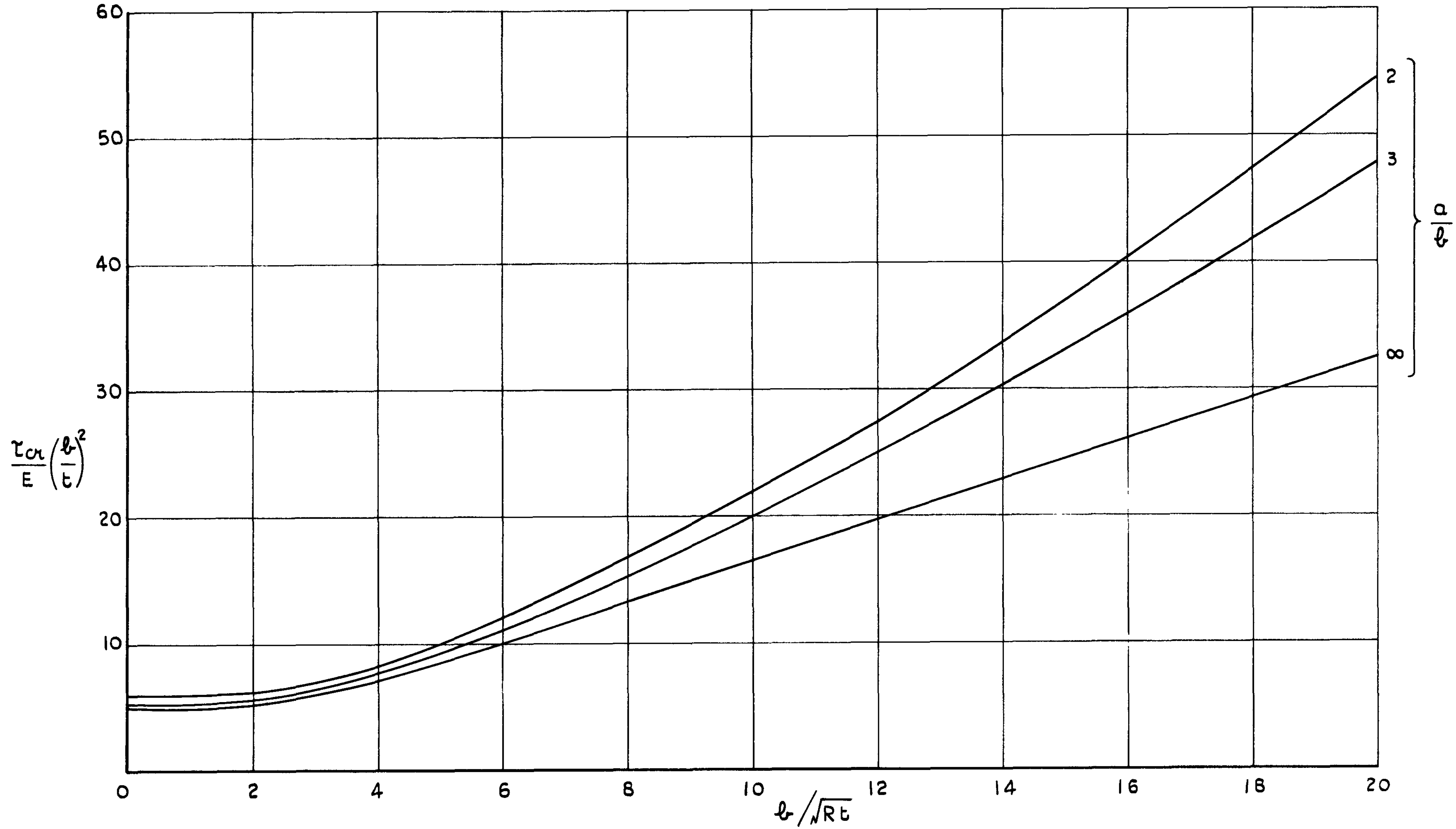


FIG. 2. BUCKLING STRESS FOR CURVED PLATE IN SHEAR.

(FROM R.Ae.S. DATA SHEET No. 02.03.18).

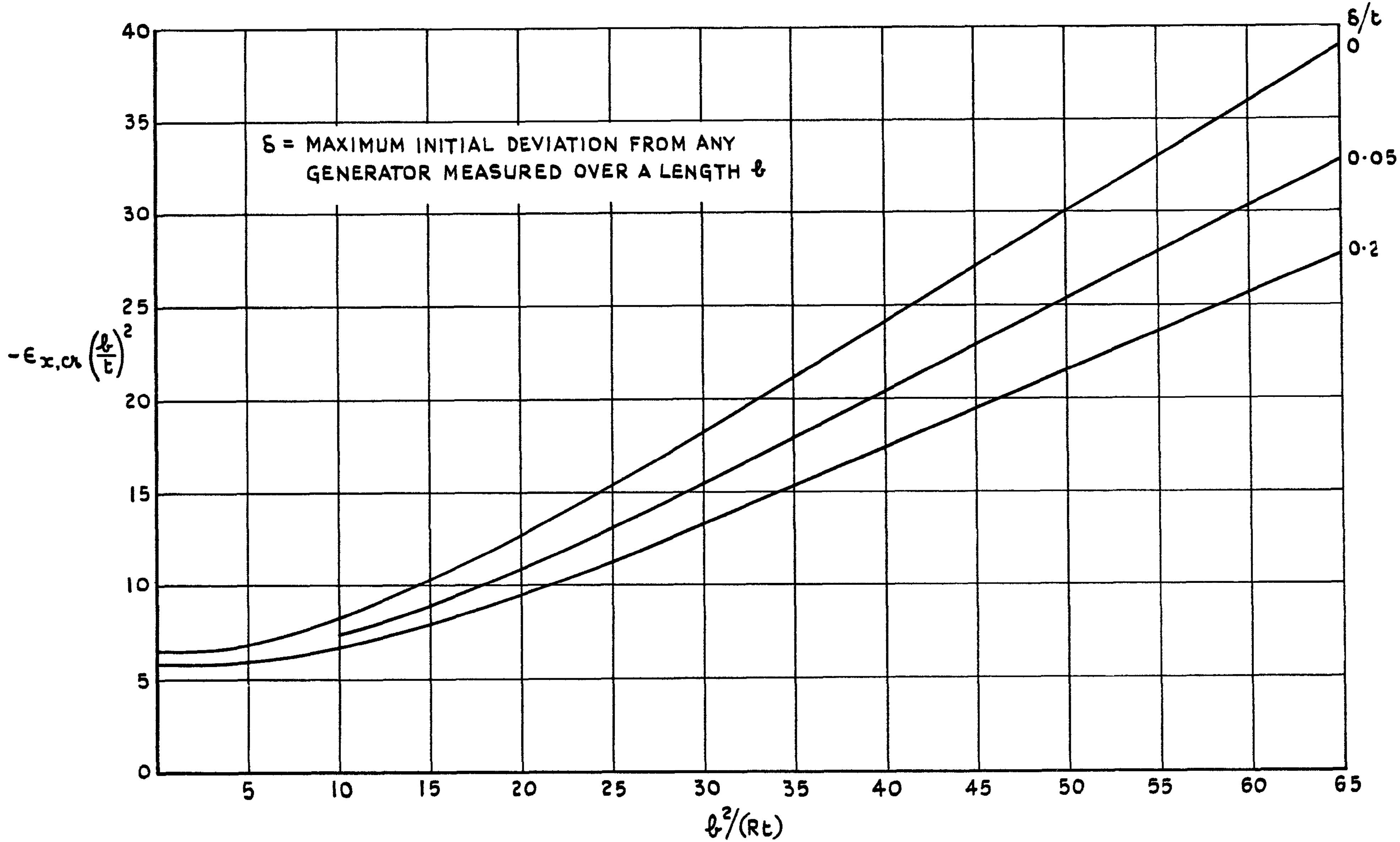
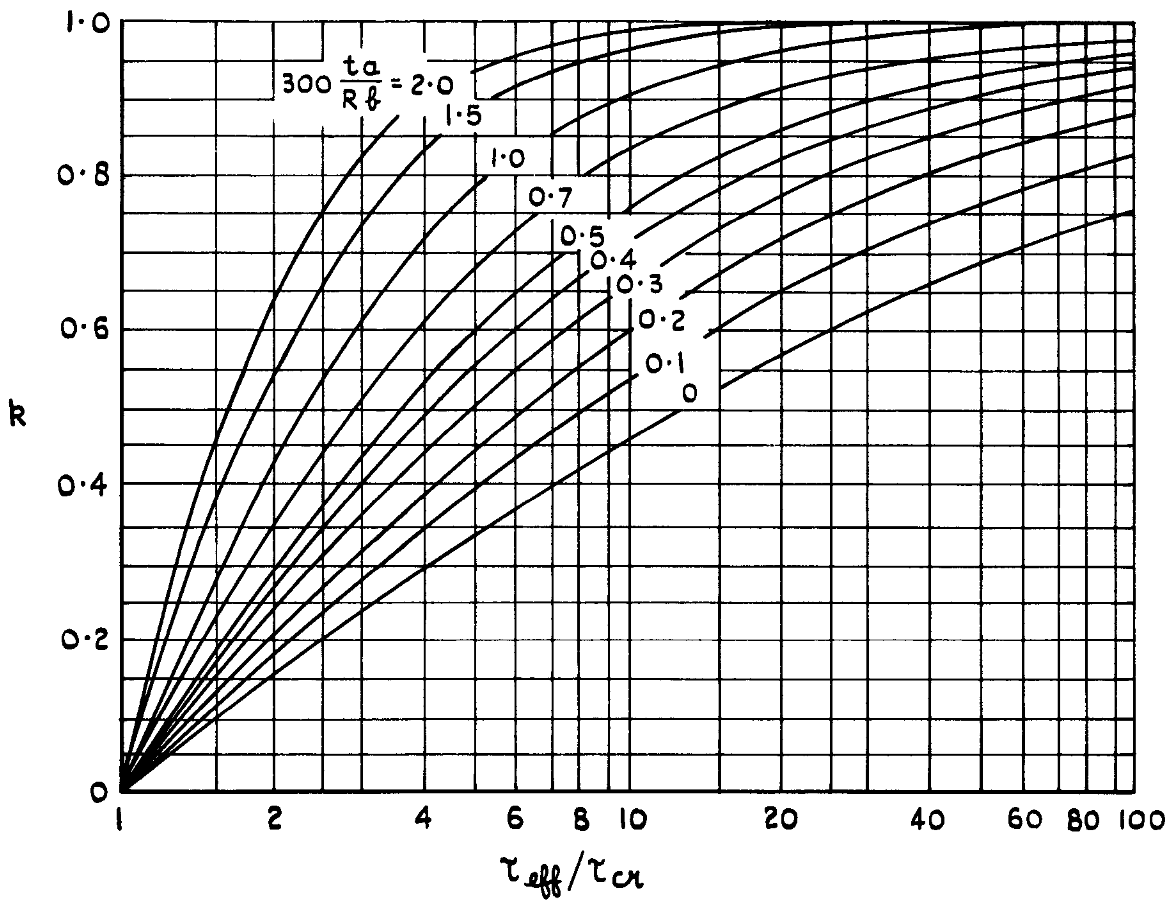


FIG.3. DIRECT STRAIN FOR BUCKLING OF CURVED PLATE IN COMPRESSION.

(FROM R.Ae.S. DATA SHEET No. 02.01.10).



(FROM N.A.C.A. TECH. NOTE No. 1481)

FIG. 4 DIAGONAL-TENSION FACTOR k

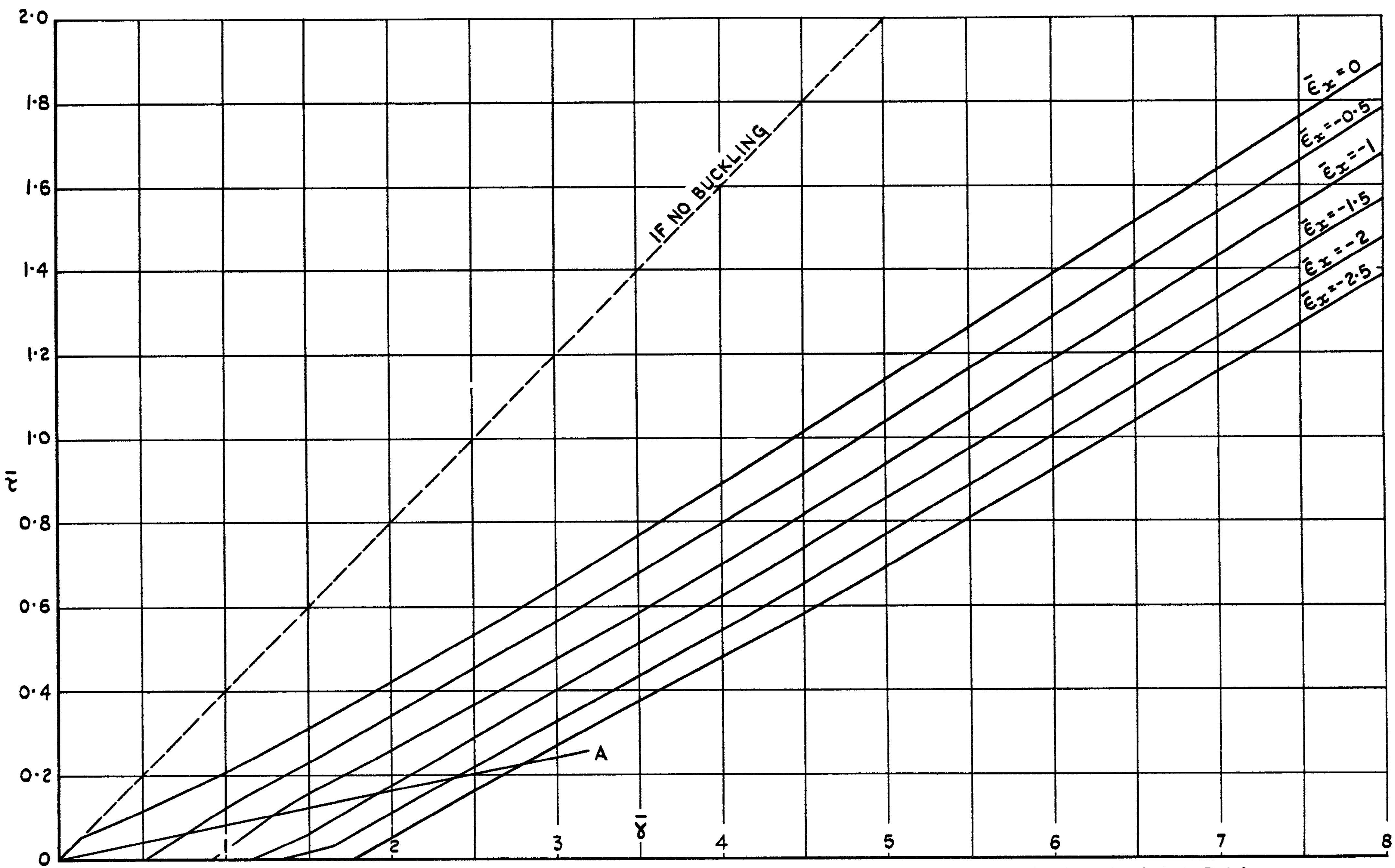


FIG.5. NON-DIMENSIONAL SHEAR STRESS-STRAIN RELATION.

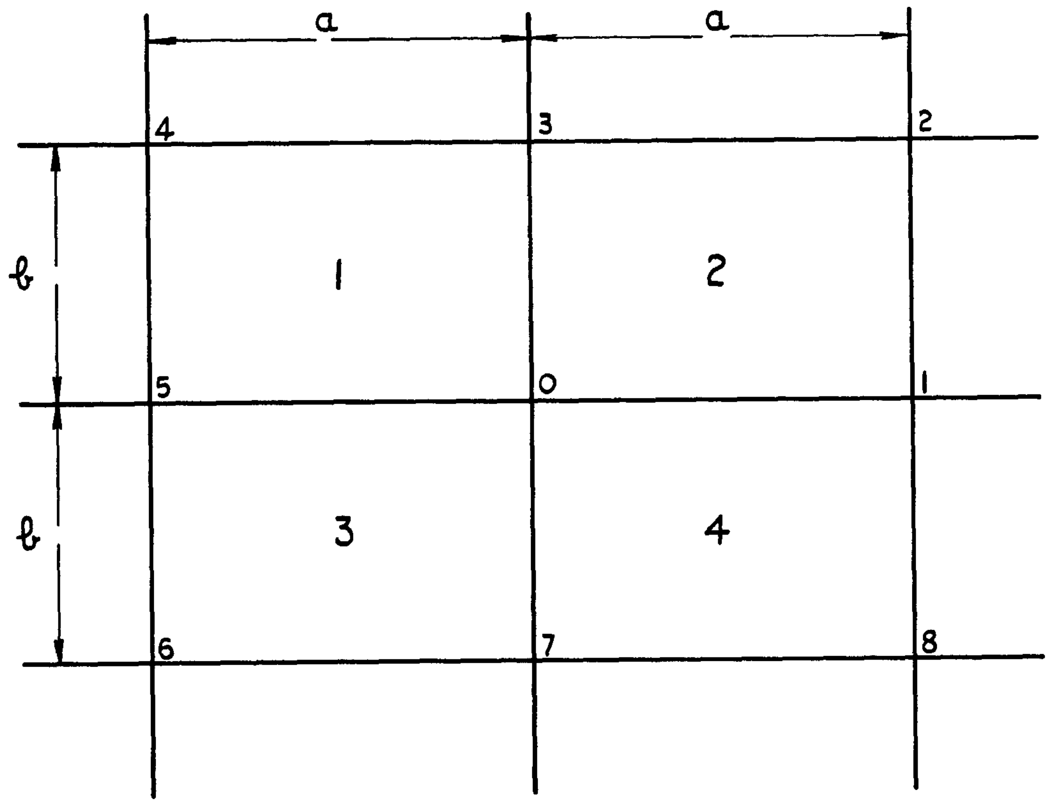


FIG.6. NUMBERING OF PANELS AND STATIONS.

A.R.C. C.P. No.644

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539.384.4

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Royal Aircraft Establishment

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S.O. CODE No. 23-9013-44

C.P. No. 644