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Shear Buckling of Isotropic and Orthotropic Plates
A Review

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CORRECTION

1. *Page* 2 Equation (1)

Read:— $D\left[\frac{\partial^4 W}{\partial x^4}+2\frac{\partial^4 W}{\partial x^2 \partial y^2}+\frac{\partial^4 W}{\partial y^4}\right]+2\tau d\frac{\partial^2 W}{\partial x \partial y}=0$

2. *Page* 6 Equation (11)

Read:— $Nxy\, b^2/4\,(D_2 D_3)^{\frac{1}{2}}=11{\cdot}71$

3. *Page* 6 Equation (12)

Read:— $Nxy\, b^2/4\,(D_1 D_2^3)^{\frac{1}{4}}=8{\cdot}3$ (simply—supported)

4. *Page* 6 Equation (13)

Delete full stop from exponential, i.e., read:— $\exp(ihy/b)$

5. *Page* 7 Table 1

Final value in θ column and first value in C_a column (Simple Support) should be ∞ (infinity), as at top of C_a column (Fully Clamped)

6. *Page* 8 Line 10

Read:— θ varies between 1 and ∞

7. *Page* 10 Line 16

Read:— edge restraint parameter $\bar{\varepsilon}=4S_0 b/D_2$

8. *Page* 13 Line 3

Read:— (equations (15) and (16): $=\pi^2 K_s/4$)

9. *Page* 17 Reference 46 — Insert Author's name A. Krivetsky

10. *Page* 18 Reference 48 — Delete Author's name.

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Shear Buckling of Isotropic and Orthotropic Plates
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Summary.

A review is presented of the shear buckling of isotropic and orthotropic plates with a detailed consideration of the latter. An extensive bibliography is given and the details of the analyses in these papers are also discussed briefly in order to illustrate the development of theoretical and analytical technique. The relevant theoretical information for shear buckling and for the determination of buckling under combined stress systems including shear is presented graphically in the form of data sheets.

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*Replaces A.R.C. 31 348.

†This work formed part of a study on "The Development of Corrugated Web Shear Structures" performed for British Aircraft Corporation under Contracts BAC/R41W/30/NEW/LWS Min. Tech/KF/50/03/CB. 23a.

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1. *Introduction.*

There is now a considerable body of literature on the subject of shear buckling of plates and it is the purpose of this paper to review briefly this literature, and to present a suitable set of design charts for buckling criteria for various panel configurations, boundary conditions, loading cases, etc.

Isotropic panels are considered initially since it was for this type of panel that the original early analyses were performed. Charts will be presented for buckling under shear alone and for shear in combination with other stress systems.

Whilst an appropriate anisotropic plate theory has been available for some years very few studies of the shear buckling of other than 'special orthotropic' plates appear to have been published. For such plates the principal elastic axes are orthogonal to the plate geometric axes. Charts will be presented for buckling under shear alone and for shear in combination with other stress systems. Some studies of 'general orthotropic' plates will also be discussed.

No claim is made for completeness in this review but sufficient papers are reviewed to illustrate the development of theoretical formulations and analytical technique.

2. *Shear Buckling of Isotropic Plates.*

The first exact solution for the shear buckling of an infinitely long isotropic plate was given in Ref. 1, from the governing differential equation

$$D\left[\frac{\partial^4 W}{\partial x^4}+2\left[\frac{\partial^4 W}{\partial x^2\,\partial y^2}\right]+\frac{\partial^4 W}{\partial y^4}\right]+2\,\tau\,d\frac{\partial^2 W}{\partial x \partial y}=0 \tag{1}$$

By assuming a modal form for the buckling displacements of

$$W = Y_{(y)} \exp{(ikx/b)} \tag{2}$$

where Y is an unknown function in the transverse (y) direction which satisfies the boundary conditions on the long edges, it was possible to obtain by an iterative technique an exact solution for the critical shear stress τ_{cr} and the critical longitudinal wavelength parameter k.

Results were given in Ref. 1 for simply supported and clamped long edges and the respective buckling coefficients K_s were 5·35 and 8·98, where

$$\tau_{cr} = K_s\,\pi^2 D/db^2 = (N_{xy})_{cr}/d \tag{3}$$

N_{xy} = shear stress resultant

d = plate thickness

b = plate width (in y direction)

$D = Ed^3/12(1-\nu^2)$ = plate rigidity

Ref. 2 developed the analyses of Ref. 1 to allow for plates having equal elastic restraints against rotation along the long edges and Fig. 1 shows the dependence of K_s on λ/b, the ratio of the buckle half wavelength to the plate width, b, for various values of the edge restraint parameter ε. The values $\varepsilon = 0$ and $\varepsilon = \infty$ correspond respectively to the extreme cases of simply supported and clamped long edges. The curves shown in Fig. 1 were obtained from a few exact results supplemented by those resulting from an alternative energy method of analysis. $\varepsilon = 4\, S_o\, b/D$ where S_o is the stiffness per unit length of the elastic restraining medium or the moment required to rotate a unit length of the medium through one-fourth of a radian.

For situations in which the value of ε is dissimilar along opposite long edges Ref. 2 proposed the approximate formula

$$K_s = [K_1\, K_2]^{\frac{1}{2}} \tag{4}$$

where K_1 and K_2 are the values of K_s for equal values of ε_1 and ε_2 respectively. Thus when one long edge is clamped and the other is simply supported equation (4) yields $K_s = 6{\cdot}92$ whereas the exact value is 7·07.

For a rectangular plate of finite length with equal restraints along all four edges Ref. 2 presents some approximate curves—Fig. 2—for $\varepsilon = 0$ and $\varepsilon = \infty$ for plate aspect ratio (b/a) from 0 to 1 (i.e. infinitely long to square). The two curves were obtained from numerical values given in Refs. 3 and 4 and although not exact were believed to give good engineering approximations. For intermediate values of ε it was suggested that the $\varepsilon = 0$ curve be used as it offers a small, but acceptable, degree of conservatism.

Ref. 5 contains a study of the shear buckling of finite length isotropic plates with simply supported edges. An energy type analysis was adopted which showed the differences resulting from the assumption of symmetric or antisymmetric buckle patterns. This subject has been examined in much greater detail in Ref. 6 and has shown the importance in energy type analyses of proving convergence of the results. The results confirmed that whilst symmetric buckling is usually critical antisymmetric buckling can be more critical in the range of plate aspect ratios (b/a) of 0·3 to 0·5 (approx.).

Ref. 7 showed how the Lagrangian multiplier method could be used to ensure that the boundary conditions at the plate edges could be satisfied exactly by an assumed buckling mode. The method also enabled upper and lower bounds to be obtained for the critical buckling stress and in Ref. 8 results were presented for a plate clamped on all four edges. Thus the method enables a realistic assessment to be made of the accuracy of the results.

Ref. 9 gives fairly complete numerical results for finite plates with clamped edges. The method adopted was to substitute into equation (1) a linear combination of modal deflection functions which satisfy the boundary conditions. All the terms in equation (1) are then expanded into a Fourier series and by examining the coefficients of like terms stability criteria were obtained.

Using an energy method Ref. 10 presents results for a finite plate with one pair of opposite edges clamped and the other pair simply supported. Ref. 11 discusses this case and others and comparisons are drawn with the results of Refs. 9, 10. The significance of symmetric and antisymmetric modes is again noted.

Ref. 11 actually provides results for a plate with all edges simply supported, for the case of all edges clamped, for a plate clamped on two opposite edges and simply-supported on the other pair, and for the case of one edge clamped and the remaining three edges simply supported. These results and others are presented in Fig. 3 which was given in Ref. 12.

3. *Buckling of Isotropic Plates Under Combined Stress Systems Including Shear.*

Ref. 4 gave results for the buckling of a rectangular plate with simply supported edges when subjected to combined shear and bending stresses. Curves were plotted for values of a/b within the range 0·5 to 1·0 and it was shown that a close approximation to these curves is given by the interaction curve

$$R_s^2 + R_b^2 = 1 \tag{5}$$

$$\text{where} \quad R_s = \tau/\tau_{cr}, \quad R_b = \sigma_b/\sigma_{b.cr}$$

For combined shear and normal stress Ref. 13 presents results for a plate with simply supported edges and, for a square plate, the corresponding interaction curve is

$$R_s^2 + R_c = 1 \tag{6}$$

where $R_c = \sigma_c/\sigma_{c.cr}$

In equations (5) and (6), R_s is defined as the ratio of the critical shear in a combined stress system to the critical shear load considered separately and R_b, R_c have similar definitions.

Further consideration to the combined effects of shear and compression in infinitely long plates with various boundary conditions is given in Ref. 14. This paper is one of a series in which it is shown how a set of dimensionless stability functions can be used to analyse the local buckling of a variety of typical thin walled structures in combined compression and shear. These stability functions are obtained by considering the relative rotations and edge moments (distributed sinusoidally) along the edges of an infinitely long plate. One significant result shown in Ref. 14 is that equation (6) is correct to within 0·2 per cent for all values of R_s and R_c. It was also found that the axial buckle half-wavelength λ varies almost linearly with σ_c and the equation

$$\lambda = \lambda_s[1 - R_c] + \lambda_c R_c \tag{7}$$

has an error less than 0·7 per cent.

For combined shear, bending and transverse compression Ref. 15 presents results for rectangular simply supported panels of infinite length. The curves obtained are shown as Fig. 4. Ref. 16 also deals with this topic and Ref. 17 lists references and results for a number of edge conditions.

For a rectangular plate under combined biaxial compression, bending and shear Ref. 18 presents a novel method of analysis based on an approximate variational method. Results are presented for values of $a/b = 1$, 2 and 3 with boundary conditions of simple support, rigid clamping and an intermediate situation in each case along the edges $y = o, b$, but with simple supports at $x = o, a$. Due allowance is taken of the flexural and torsional stiffness of the edge members at $y = o, b$. For a simply supported square plate on all four edges Fig. 5 presents typical interaction curves which follow the approximate relationship

$$R_s^2 + R_b^2 + R_c = 1 \tag{8}$$

This compound result could be deduced from equations (5), (6).

Ref. 19 presents corresponding results for a simply supported rectangular plate under combined biaxial compression, bending, shear and a uniaxial sinusoidal direct stress distribution which is assumed to represent the in-plane thermal stresses in the plate.

Results of buckling tests for combined shear and uniaxial compression are given in Ref. 20.

4. *Shear Buckling of Isotropic Plates Stiffened by Discrete Transverse Stiffeners.*

It is seen from Figs. 2 and 3 that the value of K_s increases as a/b decreases. Therefore, the buckling strength of an isotropic plate can be increased by dividing it into a number of smaller panels by means of transverse stiffeners. Similarly, longitudinal stiffeners have an equivalent effect by reducing the effective width b of each sub-panel (Ref. 21). Thus in equation (3) τ_{cr} increases as b decreases.

Ref. 22 considers the problem of a long panel of total width $b = (N+1)\bar{b}$ supported at its longitudinal edges $y = o$, $y = b$ and also at a series of N intermediate equidistant simple supports distance $\bar{b}$ apart under compression and shear. Using the stability functions derived in Ref. 14 results are obtained for three different combinations of boundary conditions on the edges $y = o$, b, viz. both edges clamped, both edges simply supported, one edge clamped and the other simply supported. From all three sets of edge conditions the criterion for buckling can be generalised as a function of a parameter r where $r = N, \alpha, 2N+1$ respectively for the above three sets of edge conditions. The results which confirmed equations (6) and (7) are summarised below.

r	1	2	3	4	5	6	7	8	∞
$\tau = 0; \sigma_{cr}$	5·41	4·71	4·42	4·27	4·19	4·14	4·11	4·09	4·00
$\sigma = 0; \tau_{cr}$	7·07	6·20	5·85	5·67	5·57	5·51	5·47	5·44	5·34

The increases in buckling stress of an infinitely long plate when subdivided into an array of smaller panels has been studied by many authors, e.g. Refs. 23–27 and Ref. 12 contains a useful review of this particular problem.

The theoretical results of Ref. 23 have shown the fallacy of the assumption that continuous plates having equal finite bays buckle in shear as if each bay were simply supported at the intermediate supports. In fact, the increase in buckling stress due to continuity at the supported edges of square sub-panels is of the order of 25 per cent of that increase which would have resulted by clamping the edges. The increase appears to be maximum for square sub-panels and negligible for sub-panels of very small or very large aspect ratio.

Ref. 24 presents a solution based on the Lagrangian multiplier method for the case of transverse stiffeners with zero torsional stiffness whereas Refs. 26 and 27 remedy this deficiency.

The analyses of Ref. 24 for transverse stiffeners of low bending stiffness recognise that the buckling behaviour of the combined plate plus stiffeners may be considered to correspond to that of an orthotropic plate. Curves are presented which compare well with experimental data.

Few analyses have been noted in the literature of finite difference approximations of the governing differential equation for shear buckling; or of finite element analyses. One example of the latter is contained in Ref. 28. The isotropic plate was divided into only four panels per side, each panel being assumed to have constant torque. Thus the twisting moment distribution which should have been a smooth curve was represented by four 'steps'. In view of the close association between panel twisting and shear buckling it is hardly surprising that the calculation was, in fact, inaccurate. Clearly, however, an improved accuracy should result from a much smaller mesh size.

5. *Shear Buckling of Orthotropic Plates.*

The theory and differential equations of bending of anisotropic plates were established by Huber[29] and the governing differential equation for shear buckling of a 'general orthotropic' plate is

$$D_1 \frac{\partial^4 W}{\partial x^4} + 2D_3 \frac{\partial^4 W}{\partial x^2 \partial y^2} + D_2 \frac{\partial^4 W}{\partial y^4} + 2N_{xy} \frac{\partial^2 W}{\partial x \partial y} + 4D_{13} \frac{\partial^4 W}{\partial x^3 \partial y} + 4D_{23} \frac{\partial^4 W}{\partial x \partial y^3} = 0 \tag{9}$$

where the last two terms are a measure of the orthotropic coupling resulting from the fact that the principal elastic axes are not orthogonal with the plate geometric axes.

For a 'special orthotropic' plate $D_{13} = D_{23} = 0$ and this is the case which has received, quite naturally, the most attention.

The problem of the stability of orthotropic plates due to shear was apparently first examined by Bergmann and Reissner[30], who considered a plate infinitely long in the x-direction and they also neglected the bending rigidity in that direction. The governing differential equation used was,

$$2D_3 \frac{\partial^4 W}{\partial x^2 \partial y^2} + D_2 \frac{\partial^4 W}{\partial y^4} + 2N_{xy} \frac{\partial^2 W}{\partial x \partial y} = 0; D_1 = 0 \tag{10}$$

and using a similar method of analysis to Ref. 1—see Section 2—the following exact solution was obtained for a plate simply supported along the long edges.

$$N_{xy} b^2 \quad 4(D_2 D_3)^{\frac{1}{2}} = 11{\cdot}71 \tag{11}$$

A similar analysis was presented in Ref. 31 with $D_3 = D_1$ and with $D_1 \neq 0$. The corresponding results for simply supported and clamped edges as $D_1 \rightarrow 0$ are

$$\begin{aligned} N_{xy} b^2 / 4(D_1 D^{32})^{\frac{1}{4}} &= 8{\cdot}3 \quad \text{(simply-supported)} \\ &= 15{\cdot}2 \quad \text{(clamped)} \end{aligned} \tag{12}$$

The full differential equation for a special-orthotropic plate was used in Ref. 32 and the method of Ref. 1 followed. If the assumed mode has the form

$$W = \exp(ikx/b) \sum_{r=1}^{4} C_r \exp(ih.y/b) \tag{13}$$

then substitution into the equation (9) with $D_{13} = D_{23} = 0$ results in the following characteristic equation in h, for which there are four roots,

$$h^4 + 2\mu k^2 h^2 - 2\tau_1 kh + \rho k^4 = 0 \tag{14}$$

with

$$\mu = D_3/D_2, \rho = D_1/D_2, \quad \tau_1 = N_{xy} b^2/D_2$$

Seydel[32] presents a general method of solution allowing for elastic rotational restraint along the long edges of the plate at $y = o, b$ but only presents results for the two extreme cases—fully clamped and simply supported. These are reproduced in Table 1 below and in Fig. 6 with

$$K_s = 4C_a/\pi^2 \text{ or } K_s = 4C_b/\pi^2$$

It should be noted that a plasticity factor η is included in Fig. 6 to allow for inelastic stresses at buckling This factor which is included also in later figures will be discussed more fully in Section 7.

TABLE 1

Shear Buckling Factors for Infinitely Long Plates

θ	Simple Support		Fully Clamped	
	C_a	C_b	C_a	C_b
0	∞	11·71	∞	18·59
1/5	26·4	11·8	42·15	18·85
1/2	17·25	12·2	28·15	19·9
1	13·17	13·17	22·15	22·15
2	10·8	15·25	18·75	26·55
3	9·95	17·2	17·55	30·45
5	9·25	20·65	16·6	37·1
10	8·7	27·45	15·85	50·05
20	8·4	37·65	15·45	69·1
40	8·25	52·25	15·25	96·5
∞	8·125		15·07	

$$\theta = (D_1 D_2)^{\frac{1}{2}}/D_3$$

If the θ values of the orthotropic plates of infinite length lie between 0 and 1 the critical shear load is found from the formula

$$N_{xy}b^2 \big/ 4(D_2 D_3)^{\frac{1}{2}} = C_b \tag{15}$$

and the appropriate values of C_b are shown enclosed in a box in Table 1.

If the θ values lie between 1 and ∞ the critical shear load is found from the formula

$$N_{xy}b^2/4(D_1 D_2^3)^{\frac{1}{4}} = C_a \tag{16}$$

and the appropriate values of C_a are shown enclosed in a box in Table 1.

It should be noted that the assumption of Ref. 31, viz. $D_3 = D_1$ leads to $\theta \rightarrow \infty$ as $D_1 \rightarrow 0$ and the corresponding values for C_a from Table 1 are 8·125 (simply supported) and 15·07 (fully clamped) whereas the results from Ref. 31 are somewhat higher viz. 8·3 and 15·2.

Ref. 33 contains further discussion by Seydel of the shear buckling of infinite length orthotropic plates and he also suggests simple formulae to calculate the values of D_1, D_2, D_3 for a simple sine-wave type of corrugation. Results of experimental investigations when compared with the tabulated results of Table 1 justify the use of the simple formulae

$$\begin{aligned} D_1 &= (l_o/s)\,(Ed^3/12\,(1-\nu^2)) \\ D_2 &= EI_{\sigma} \\ D_3 &= (s/l_o)\,(Gd^3/6) \end{aligned} \tag{17}$$

where l_o is the subtended length of a half sine wave of actual length s and I_x is the second moment of area per unit length in the direction of the corrugation.

Ref. 34 presents an extension to Ref. 5 to give the buckling of a simply supported rectangular orthotropic plate and there is further discussion in Ref. 35.

Thus if,

$$\beta_a = (b/a)(D_1/D_2)^{\frac{1}{4}} \leqslant 1 \text{ and } \theta \geqslant 1 \tag{18}$$

the critical shear load is again given by C_a where C_a is now dependent on θ *and* β_a as is shown in Fig. 7. These curves are only approximate and are based on the results of Ref. 33 which showed various stability curves for different modal assumptions and suggested a lower bound based on these curves. $1/\theta$ is plotted to enable a linear interpolation as θ varies between 1 and ∞. If the given plate should yield $\beta_a > 1$ the notation for the two rectangular sides should be exchanged and the appropriate value of β_a should be taken, i.e. by putting $D'_1 = D_2$, $D'_2 = D_1$, $b' = a$, $a' = b$ and equation (17) becomes

$$N_{xy}(b')^2/4(D_1[D'_2]^3)^{\frac{1}{4}} = C_a \tag{16a}$$

For $\theta < 1$ one takes

$$\beta_b = (b/a)(D_3/D_2)^{\frac{1}{2}} \tag{19}$$

and the critical shear load is again given by C_b. No results are given by Seydel.

It should be noted that a Galerkin type modal analysis was made in Ref. 34 as distinct from the 'exact' analyses of previous papers. Modes of the form

$$W = \sum_m \sum_n A_{mn} \sin(m\pi x/a)\sin(n\pi y/b) \tag{20}$$

were used and to examine the convergence characteristics an increasing number of modes m, n are taken. Thus for an isotropic plate ($\theta = 1$) the following convergence was shown to the exact solution for an infinitely long plate, viz. $C_a = 13{\cdot}17$ (Table 1)

m	n	C_a	% error
3	3	14·68	11·65
4	2	14·15	7·5
5	2	13·90	5·6
5	3	13·28	0·88
5	5	13·23	0·53
9	3	13·22	0·41

The problem of stability of orthotropic plates has been treated in Ref. 36 and Ref. 37 and provide useful summaries of available knowledge up to 1935.

In Ref. 38 approximate solutions are obtained for the buckling of clamped edged finite plates and infinite strips of orthotropic material under shear, using the Rayleigh-Ritz method. The same approximate deflection functions are assumed as in Ref. 9, viz.

$$W = \sum_m \sum_n A_{mn} X_m Y_n \tag{21}$$

where

$$X_m = \left(\frac{x}{a}\right)^3 - 2\left(\frac{x}{a}\right)^2 + \left(\frac{x}{a}\right) + \left(-1\right)^m \left[\left(\frac{x}{a}\right)^3 - \left(\frac{x}{a}\right)^2\right] - \frac{1}{m\pi}\sin\frac{m\pi x}{a}$$

and Y_n is similar.

Smith[38] considers his energy method to be more accurate than Iguchi's (Ref. 9) series method although his preliminary algebra is more tedious. Analyses are made with m, n as variables and it is shown that solutions only exist in two distinct ranges, i.e. when $(m+n)$ even and $(m+n)$ odd. Ref. 38 states that the case when $(m+n)$ is even gives the lower critical shear loads. That this may not necessarily be true in general has been shown in Ref. 6 for isotropic panels where $m+n < 10$. In Ref. 38 $m+n < 6$.

If the results of Ref. 38 are presented in the notation of Seydel[34,35] the curves of Fig. 8 are obtained. The accuracy of these results appears to be reasonable since Ref. 38 has demonstrated convergence by evaluating stability determinants of various orders.

Ref. 39 is a most comprehensive report on the present subject. A general orthotropic theory is developed and an application to a plywood plate is discussed. The differential equation (9) is analysed for an infinitely long plate by the method of Ref. 1 for simply supported and fully clamped edges. The equations which result for the general orthotropic plate under shear are quoted and seen to be only slightly more complicated than for the special-orthotropic plate.

An alternative, approximate method of solution based on minimum energy considerations is also given in Ref. 39 for the infinitely long orthotropic plate. The assumed buckle modes are:

$$W = A \sin(\pi y/b) \sin(\pi [x - \phi y]/L) \tag{23}$$

for simply supported longitudinal edges, and

$$W = B[1 - \cos(2\pi y/b)][\sin(\pi[x - \phi y]/L)] \tag{24}$$

for clamped longitudinal edges.

These functions represent infinitely long buckled surfaces with the half wave length L, whose straight nodal lines run diagonally to the longitudinal axis x and form an angle with the y axis, the tangent of which is ϕ. The critical shear load is minimised with respect to the two free values L, ϕ.

Although in general the nodal lines are not straight, sufficiently exact buckling loads were expected for the infinitely long, general-orthotropic plate.

The results for several important cases of 'general-orthotropic' plates were presented particularly for an angle of 45 degrees between the principal axes and geometric axes. It was shown that by suitable orientation of this angle a substantial increase in the critical shear load can be obtained.

Ref. 40 gives elastic constants for corrugated core sandwich plates which are typical orthotropic structures.

Ref. 41 presents results of analyses to determine the buckling under combined stresses of orthotropic plates formed from flat plates with integral waffle-like stiffening. The theory of Ref. 35 was used and the input data of the effective flexural rigidities D_1, D_2, D_3 were calculated for a wide range of waffle stiffening configurations including many in which the waffle was skew to the plate axes. Correlation with experiment was good and gave confidence in the theory. Theoretical results are given for finite plates with the short edges clamped and the long edges simply supported in a form identical to that of Figs. 7 and 8—see Fig. 9.

A comparison of the relative effectiveness of the various stiffening configurations showed that the $-60° +60°$ pattern is the most effective for shear but that a $-45° +45°$ orientation has the most universal application.

Ref. 42 contains a direct application of previous papers and is largely based on Ref. 35. The paper contains results of analyses for combined stress systems. For combined compression and shear, for a variety of orthotropic configurations the result obtained was approximately $R_c + R_s^2 = 1$ as originally obtained for isotropic plates in (6).

The results of an experimental investigation are reported in Ref. 43 for multiweb beams with corrugated webs with two types of connection between the web and the skin. It was established that this connection affects the structural efficiency of the corrugated web beams.

The effect of restrained warping on the buckling load of corrugated webs (i.e. of orthotropic plates) is discussed in Refs. 44, 45. Ref. 44 gives criteria for long clamped corrugated webs with complete restraint against warping along the edges. The calculations were made for a specific corrugation shape (square wave) and they indicate that the effect of restrained warping on buckling may be considerable. The analyses of Ref. 45 are simple extensions of Ref. 44 in which edge support conditions along the edges are allowed to vary between the extremes of fully clamped and simply supported. The appropriate governing differential equation used in Refs. 44, 45 is (cf. equation (9)).

$$D_1 \frac{\partial^4 W}{\partial x^4} + 2D_3 \frac{\partial^4 W}{\partial x^2 \partial y^2} + D_2 \frac{\partial^4 W}{\partial y^4} - E\Gamma \frac{\partial^6 W}{\partial x^2 \partial y^4} + 2 N_{xy} \frac{\partial^2 W}{\partial x \partial y} = 0 \tag{25}$$

Values of C_a (see equation (16)) are plotted in Fig. 10 against the wavelength parameter $\bar{\lambda}/b$ for various values of the edge restraint parameter $\varepsilon = 4S_o b/D_2$ and are seen to be similar to those obtained in Ref. 2 and plotted in Fig. 1.

$$\bar{\lambda} = (\lambda/b)(D_2/D_1)^{\frac{1}{4}} .$$

Fig. 10 was obtained assuming that $D_3 = \Gamma = 0$. Fig. 11 shows the effect on the buckling coefficients of changes in the bending-torsion parameter $4E\Gamma/b^2 (D_1 D_2)^{\frac{1}{2}}$ for both clamped and simply supported edge conditions, together with typical values of Γ for three common types of corrugations. Considerable increases in C_a are indicated with increases in the above parameter particularly for clamped plates. For the $\bar{\varepsilon} = 0$ curve the increase in C_a is associated with the restrained warping within the plate due to variable torque, since the imposed boundary conditions are such that for simply supported edges in-plane warping along the edges is unrestrained. For the clamped edges $\bar{\varepsilon} = \infty$ both effects are present yielding a larger increase in C_a.

A comprehensive review of buckling of orthotropic plates for various geometries, loading cases, etc., is contained in Ref. 46.

For shear buckling charts are given in terms of $K_s = 4C_a/\pi^2$ or $4C_b/\pi^2$, for

1) Infinitely long simply supported plates:
   (Results as in Table 1—Fig. 6)
2) Infinitely long clamped plates
   (Results as in Table 1—Fig. 6)
3) Finite length simply supported plates
   (Results as in Fig. 7)
4) Finite length plates simply supported on long edges; clamped on short edges. (Results as in Fig. 9).

The interpretation of these charts for the case of $\beta_a > 1$ was described earlier (equation (16a)).

Ref. 46 also suggests formulae to calculate values of D_1, D_2, D_3 for arbitrary configurations of corrugated orthotropic plate. The results for D_1, D_2 agree with those in equation (17) but a quite different expression is obtained for D_3 viz.

$$D_3 = \left[\nu \frac{l_o}{s} + \frac{l_o + s}{2\, l_o (1+\nu)} \right] \frac{Ed^3}{12} \tag{26}$$

Refs. 47 and 48 contain details of analyses for sandwich panels, employing some of the previously quoted references. The many charts presented include some in which account is taken of the shear rigidity of the core.

Ref. 49 presents formulae by which the elastic constants of corrugations may be determined.

Ref. 50 presents formulae based on a minimum energy procedure, for the instability of 'general orthotropic' plates under biaxial compression, shear and the dynamic aerodynamic forces of a fluttering panel. Pronounced effects of the angle of orthotropicity were shown for flutter (in the absence of external in-plane loads). Similar results for in-plane loads are not presented but comparable effects are anticipated.

The Kantorovich method as applied to the static stability of orthotropic plates in the presence of lateral static aerodynamic forces is discussed in Ref. 51. Unfortunately, the method as presented cannot be applied to shear buckling problems as the corresponding integrals with the N_{xy} term have zero value. However, an alternative approach using the Galerkin Method is described.

Ref. 52 gives a brief review of previous work relevant to the shear buckling of general orthotropic plates and attempts to apply the method of Refs. 7, 8 to rectangular plates with various combinations of edge boundary conditions. It was hoped that upper and lower bound results would be obtained for all combinations. Whilst satisfactory results were obtained for isotropic plates, and for some particular general orthotropic plates, with at least three clamped edges, only upper bound loads could be found for other edge conditions (e.g. simply supported edges).

It should perhaps be mentioned that Ref. 53 found an exact series solution to the buckling of a rectangular plate with clamped edges which resulted in a lower bound when the series was truncated and Ref. 54 has demonstrated the convergence of this technique. The technique of Ref. 7 has been shown however, to yield a more rapidly converging method.

Refs. 55–58 have also contributed to the understanding of orthotropic plate buckling.

6. *Buckling of Orthotropic Plates under Combined Stress Systems.*

Whilst very few analyses appear to have been made of this topic there is every reason to anticipate that the approximate interaction formulae deduced for isotropic plates will also apply to orthotropic plates. In fact, results quoted in Section 5 from Ref. 42 confirm this, at least for combined compression and shear.

In order to make this present review as complete as possible and because Ref. 46 in particular does not appear to have had a wide publication the opportunity is taken to include from Ref. 46 and elsewhere a full set of curves for compression and bending for various boundary conditions to supplement those already included for shear. See Figs. 12–15.

It is suggested that the overall interaction formula to be used should be the same as for isotropic plates, viz. equation (8),

$$R_s^2 + R_b^2 + R_c = 1\,.$$

7. *Plasticity Effects.*

Ref. 46 recommends that the plasticity coefficient $\eta = E_T/E$ be inserted into the buckling criteria for direct and shear stresses as in Fig. 6 et al where E_T is the tangent modulus although for shear there is some evidence that the secant modulus should be used, i.e. $\eta_S = E_S/E$.

For the shear case, Ref. 46 suggests, on the basis of Ref. 59, that the critical shear stress is equivalent to a compressive stress given by

$$\sigma_e/\eta = \tau\sqrt{3}/\eta\,;\ \sigma_e = \tau\sqrt{3} \qquad (27)$$

Therefore for a given value of τ_{cr}/η the corresponding value of σ_e/η enables both σ_e and η to be found. From this value of σ_e the design value of τ_{cr} is obtained from equation (27). The applicability of this approach to orthotropic panels of differing geometries is uncertain but thought to be adequate for design purposes.

Ref. 4 refers to experimental results for aluminium alloy plates which showed good agreement with theory for shear buckling if the plasticity reduction factor G_S/G is used (G_S = secant shear modulus).

Ref. 60 quotes results which show that the plasticity reduction factor is best given by E_S/E taken at $\sigma = \tau\sqrt{3}$ as in equation (27). This approach is however slightly optimistic. Ref. 61 adopts this approach in a modified Ramberg-Osgood equation

$$e = (f/E) + (f_N/ME)(f/f_N)^M \quad (28)$$

or

$$\tau/E_S = (\tau/E)[1 + (\tau\sqrt{3}/f_N)^{M-1}/M] \quad (29)$$

8. *Related Topics.*

There are various engineering applications of orthotropic plates in the form of corrugated structures, e.g. aircraft wing webs; decking, wall sheeting and roofing members of buildings, etc., and there is consequently a growing body of literature related to various problems such as shear stiffness of such structures, optimum design under various loadings, etc. In particular the application of corrugated structures in aircraft wings to attempt to relieve thermal stresses has necessitated significant research and development programmes to analyse, manufacture and test these structures for quite complex loadings and environments.

These topics are considered in Refs. 61–68 and the importance of a correct knowledge of the buckling characteristics to some of these studies is clear. An optimum design study in Ref. 61 for shear buckling of a particular triangular corrugation panel shows an optimum vertex angle of about 97° 32′. This increases as the shear stress at buckling failure becomes plastic.

9. *Conclusions.*

It can be deduced from the various graphs presented in this review that

I For infinitely long plates in compression the buckling stress is independent of the orientation of the 'special orthotropic' axes with respect to the geometric axes of the plate.

II For short plates in compression a higher buckling stress is obtained with the larger flexural rigidity in the direction of the load.

III For plates under shear higher buckling stresses result when the larger flexural rigidity is in the the direction of the shorter side than when it is in the direction of the longer side.

It is concluded from this review that the problem of shear buckling of orthotropic plates has a considerable literature and that the main cases, e.g. various boundary conditions, structural configurations, have been analysed.

LIST OF SYMBOLS

a	Plate length
$b\,;\bar{b}$	Plate width; sub-panel width (Section 4)
C_a, C_b	Shear buckling coefficients (equations (15)) and (16) $= \pi^2 K_s/4$)
d	Plate thickness
D	Isotropic plate rigidity $= Ed^3/12\,(1-\nu^2)$
D_1, D_2, D_3 D_{13}, D_{23}	Orthotropic plate rigidities (equation (9))
e	Direct strain (equation (28))
E	Youngs modulus
f	Direct stress (equation (28))
G	Shear modulus
h	Buckle parameter (equation (13))
H	Corrugation dimension (Fig. 11)
I_∞	Second moment of area per unit length (equation (17))
k	Longitudinal wavelength parameter (equations (2) and (13))
K_s	Shear buckling coefficients (equation (3))
K_1, K_2	Values of K_s corresponding to ε_1, ε_2 different values of ε
l_o	Subtended length of a half sine wave of developed length, s
L	Half-wave length (equation (23) and (24))
m, n	Number of buckle half waves in x, y directions (equation (20))
M	Ramberg-Osgood parameter (equations (28) and (29))
N_{xy}	Shear stress resultant $= \tau d$
r	Parameter in Section 4
$R_b\,; R_c\,; R_s$	$\sigma_b/\sigma_{b.cr}\,;\ \sigma_c/\sigma_{c.cr}\,;\ \tau/\tau_{cr}.$
s	Developed length of half sine wave
S_o	Stiffness per unit length of the elastic restoring medium (Sections 2, 5)
W	Plate transverse deflection
x, y	Plate co-ordinate axes
X_m, Y_n	Assumed deflection functions (equations (21) and (22))
$\varepsilon : \bar{\varepsilon}$	Edge restraint parameters $= 4S_o b/D\,;\ 4S_o b/D_2$
$\tau\,;\tau_1$	Shear stress, $N_{xy}/d\,;\ N_{xy}b^2/D_2$ (equation (14))
$\sigma_b\,;\sigma_c$	Direct stress due to bending; compression

λ; $\bar{\lambda}$	Buckle half wave-length; $\lambda[D_2/D_1]^{\frac{1}{4}}$
θ	$[D_1 D_2]^{\frac{1}{2}}/D_3$
μ	D_3/D_2
ρ	D_1/D_2
η	Plasticity factor (Section 7)
β_a; β_b	$(b/a)(D_1/D_2)^{\frac{1}{4}}$; $(b/a)(D_3/D_2)^{\frac{1}{2}}$ (equations (18) and (19))
ν	Poissons ratio
ϕ	Angle of nodal lines to y axis
Γ	Warping restraint parameter (Section 5)

Suffices

b	Bending
c	Compression
s	Shear
S	Secant
T	Tangent
cr	Critical

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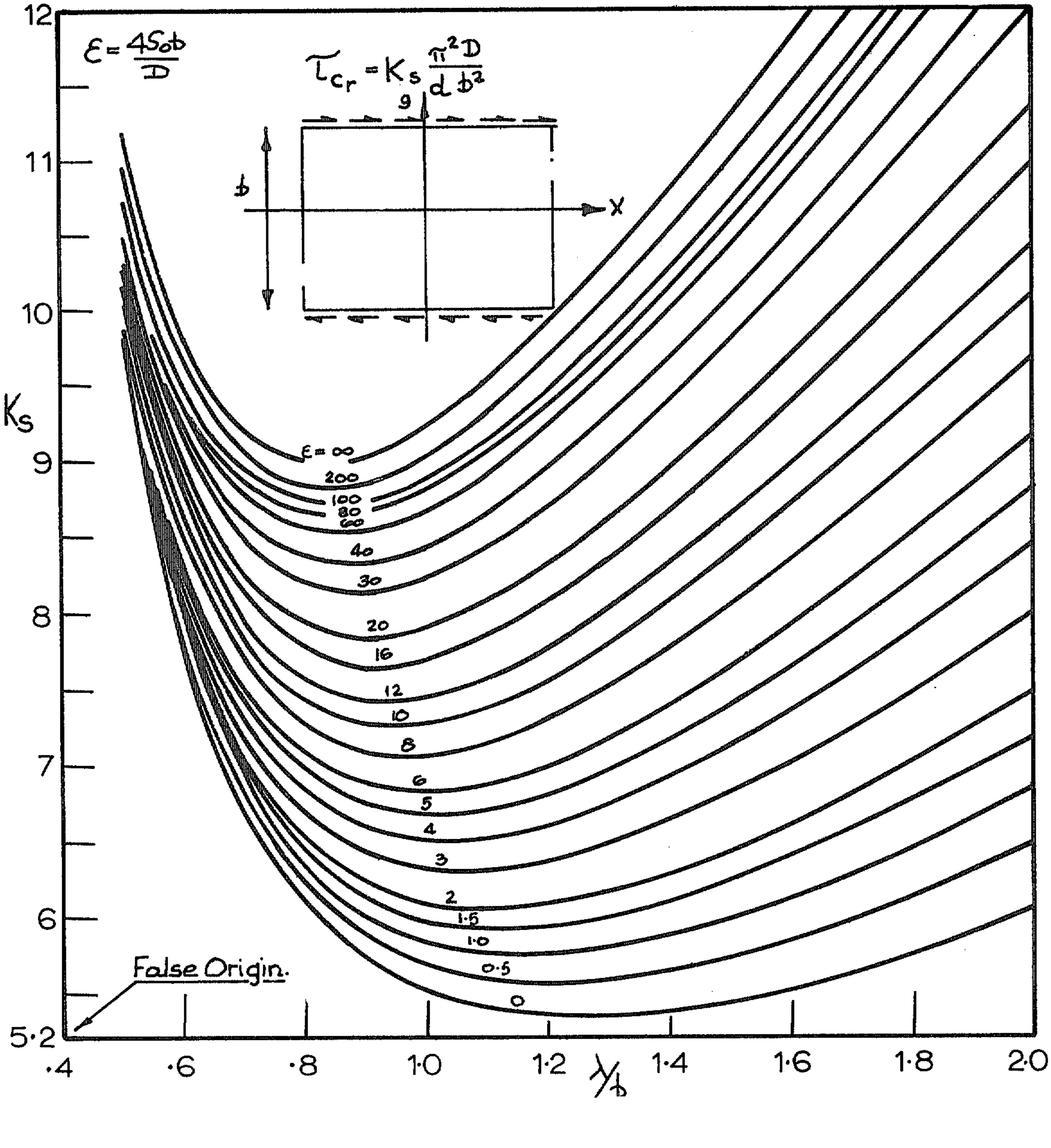

FIG. 1. Chart giving values of K_S in equation for critical shear stress for an infinitely long flat plate with equal restraining loads along the parallel edges. (b = plate width). (Ref. 2).

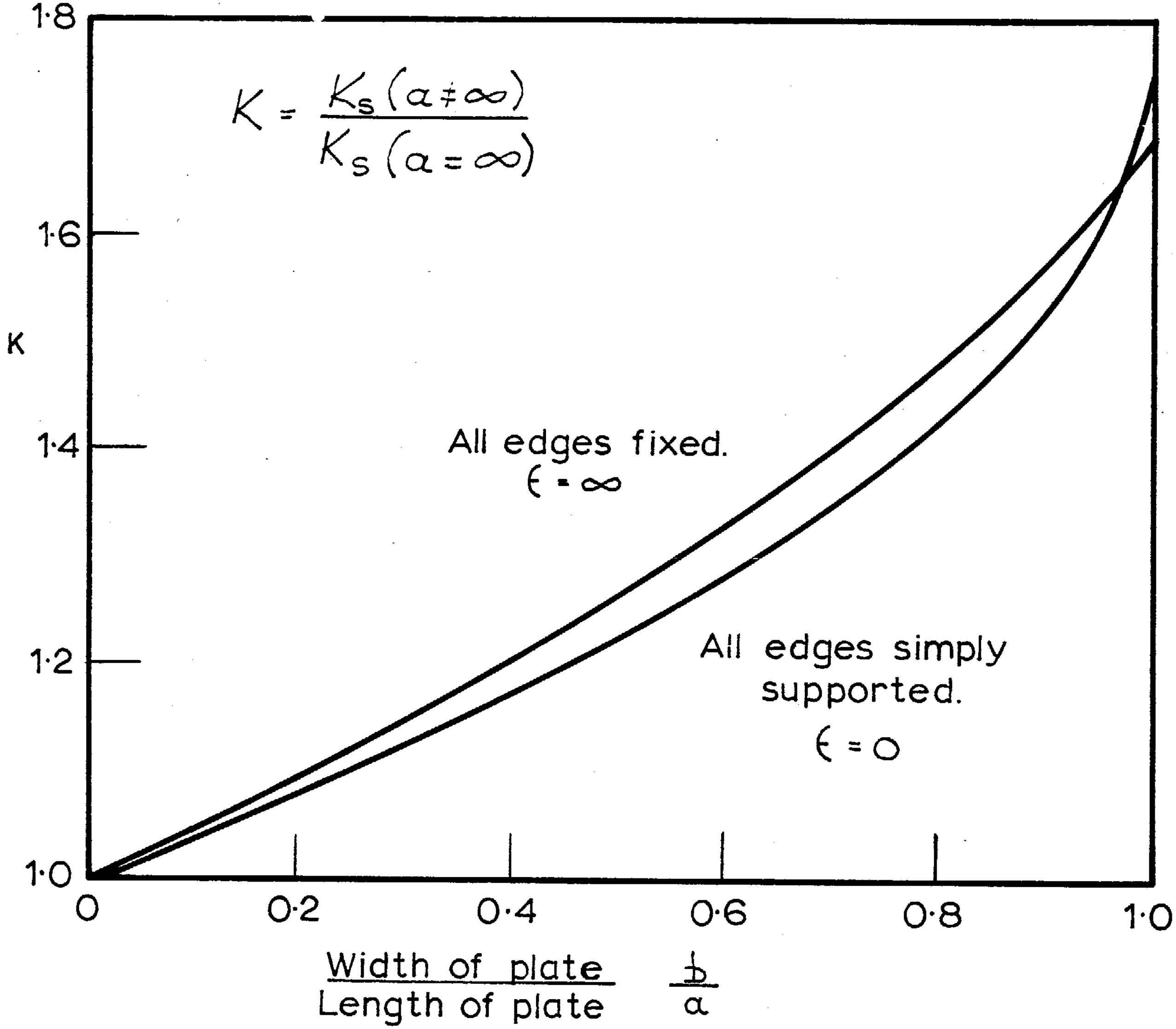

FIG. 2. Variation of K with ratio b/a (Ref. 2).

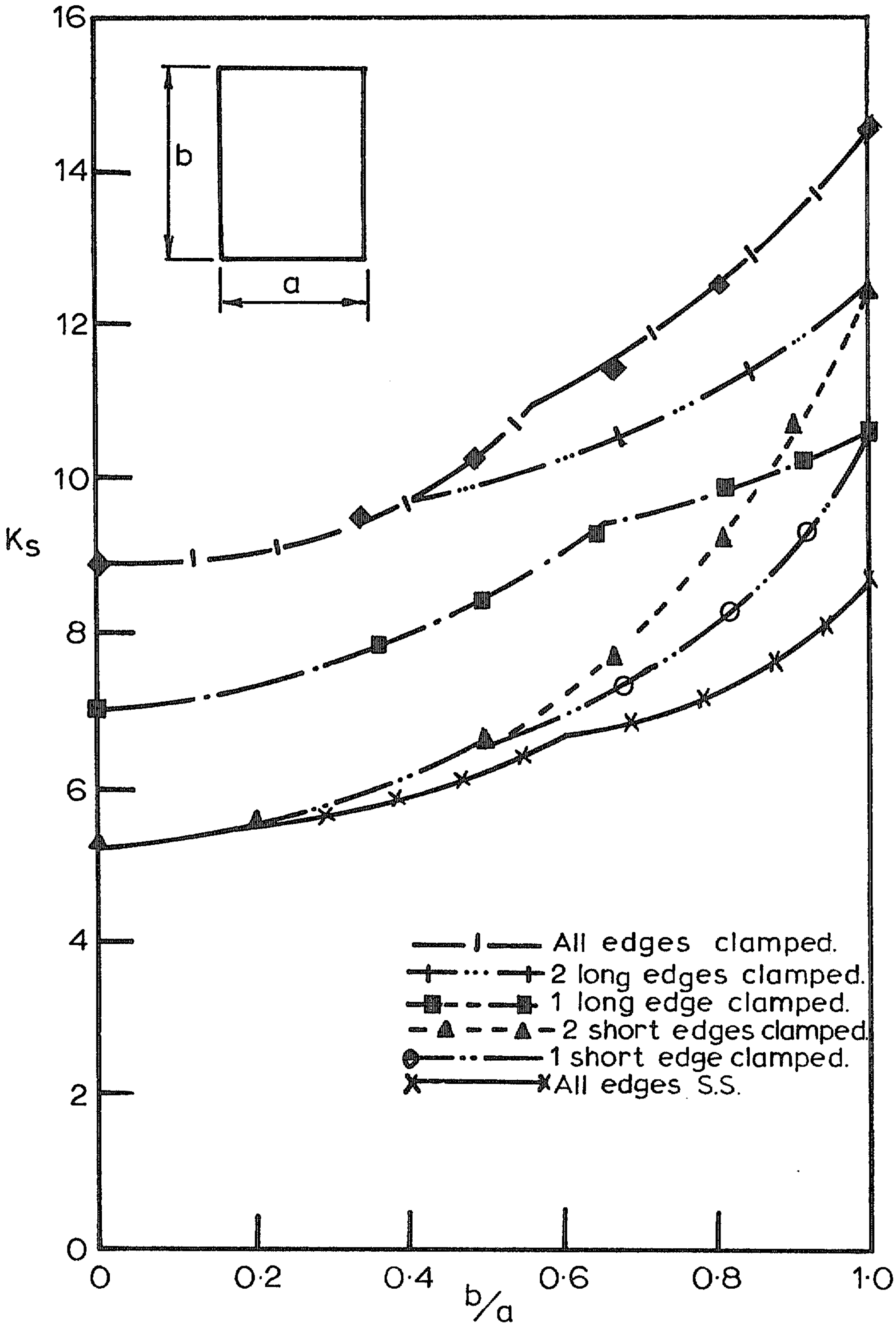

FIG. 3. Shear buckling parameter for isotropic plates with various edge support conditions. (Ref. 12).

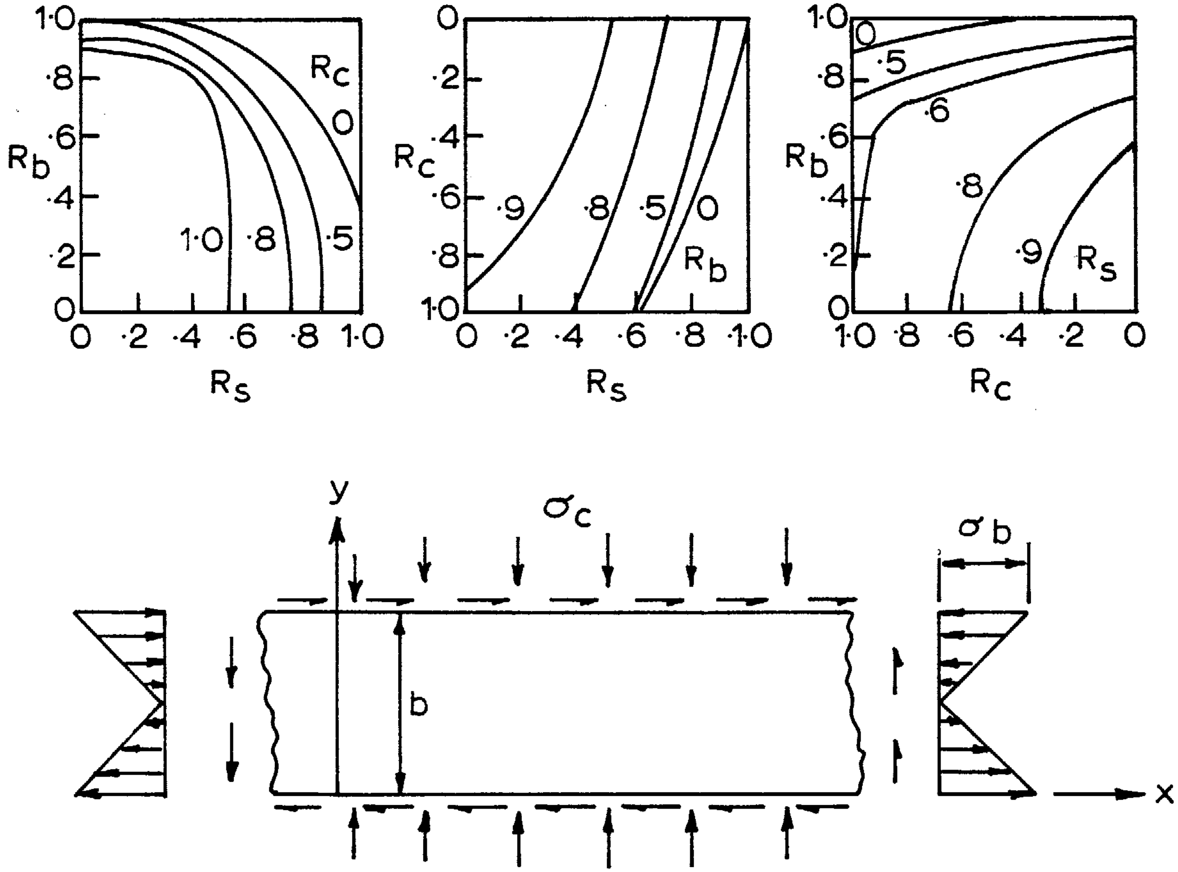

FIG. 4. Interaction curves for long S.S. flat plates under various loading conditions. (Ref. 15).

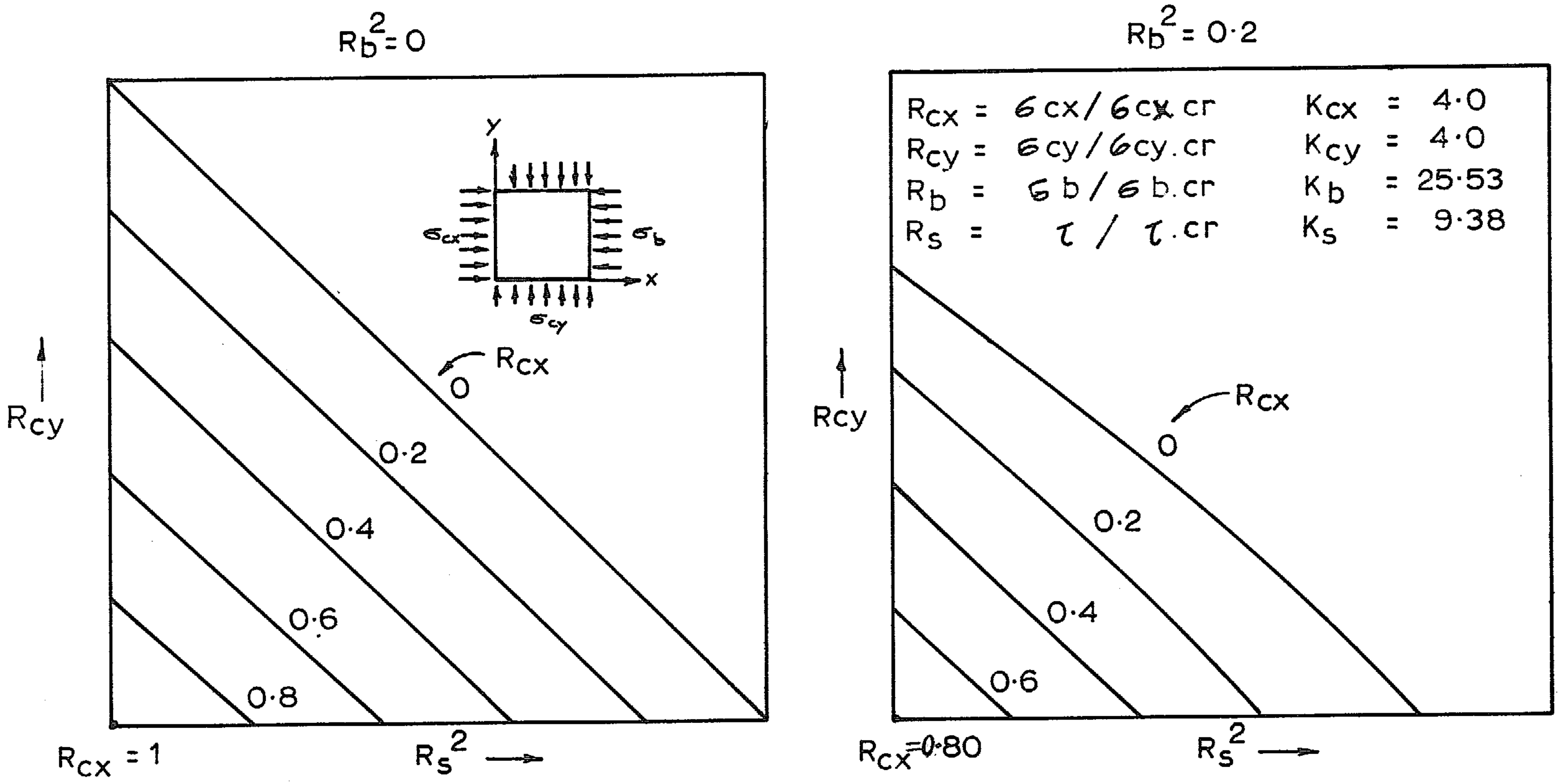

FIG. 5. Buckling interaction curves for simply supported square plate. (Ref. 18).

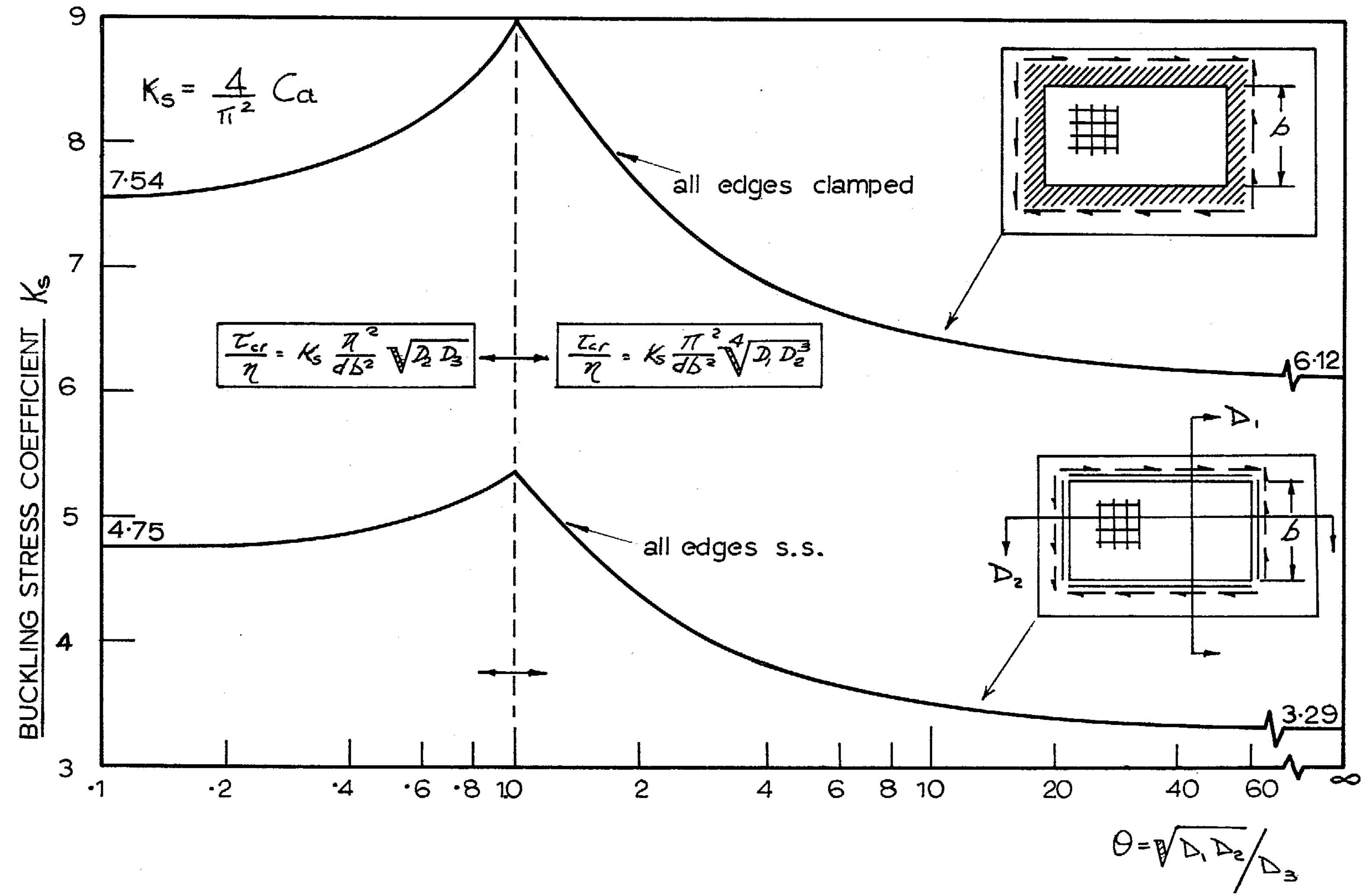

FIG. 6. Buckling coefficients for infinitely long orthotropic plates in shear. (Ref. 46).

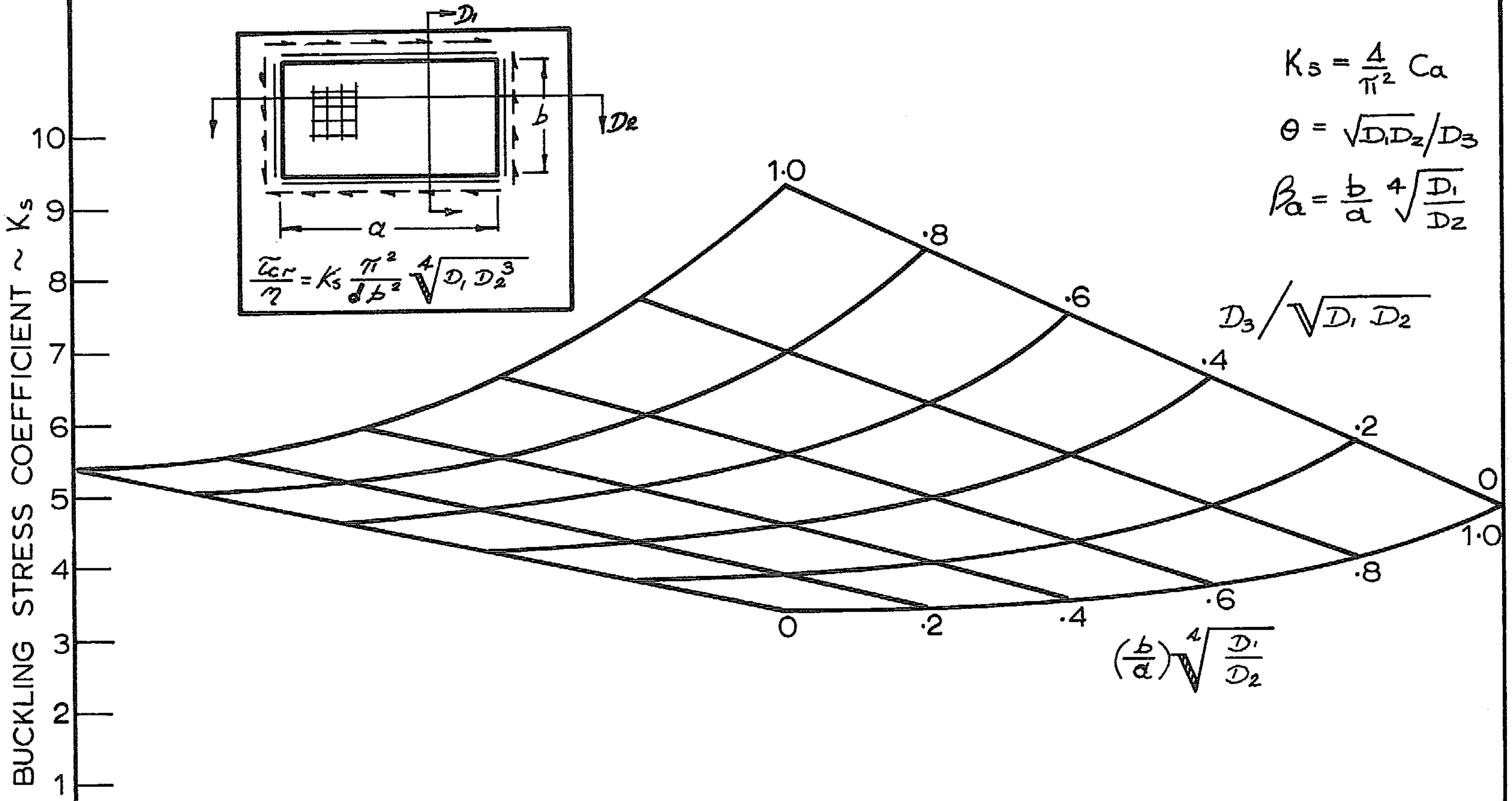

FIG. 7. Buckling coefficients for orthotropic plates in shear ~ all edges simply supported. (Ref. 46).

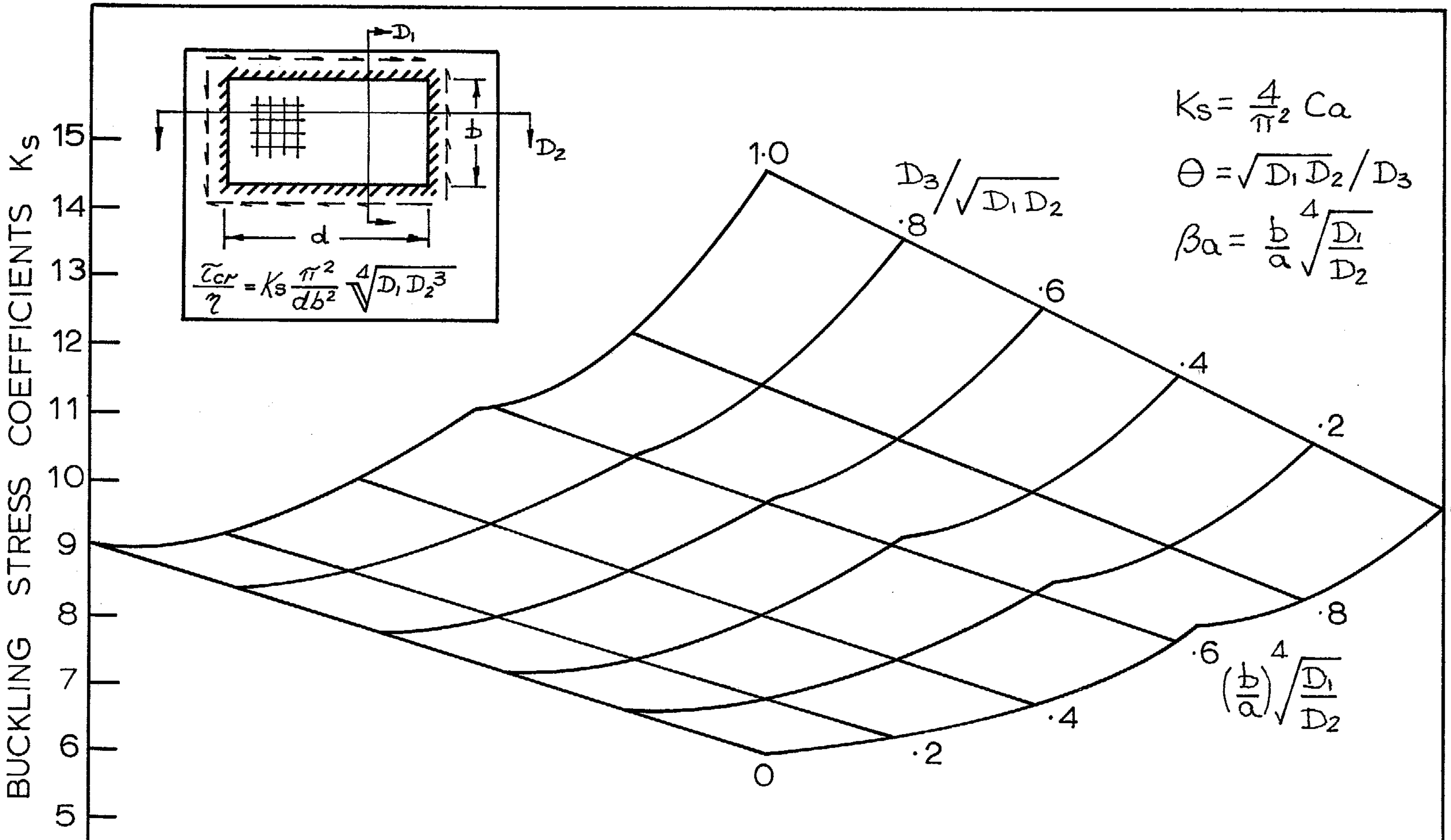

FIG. 8. Buckling coefficients for orthotropic plates in shear ~ all edges clamped. (Ref. 38).

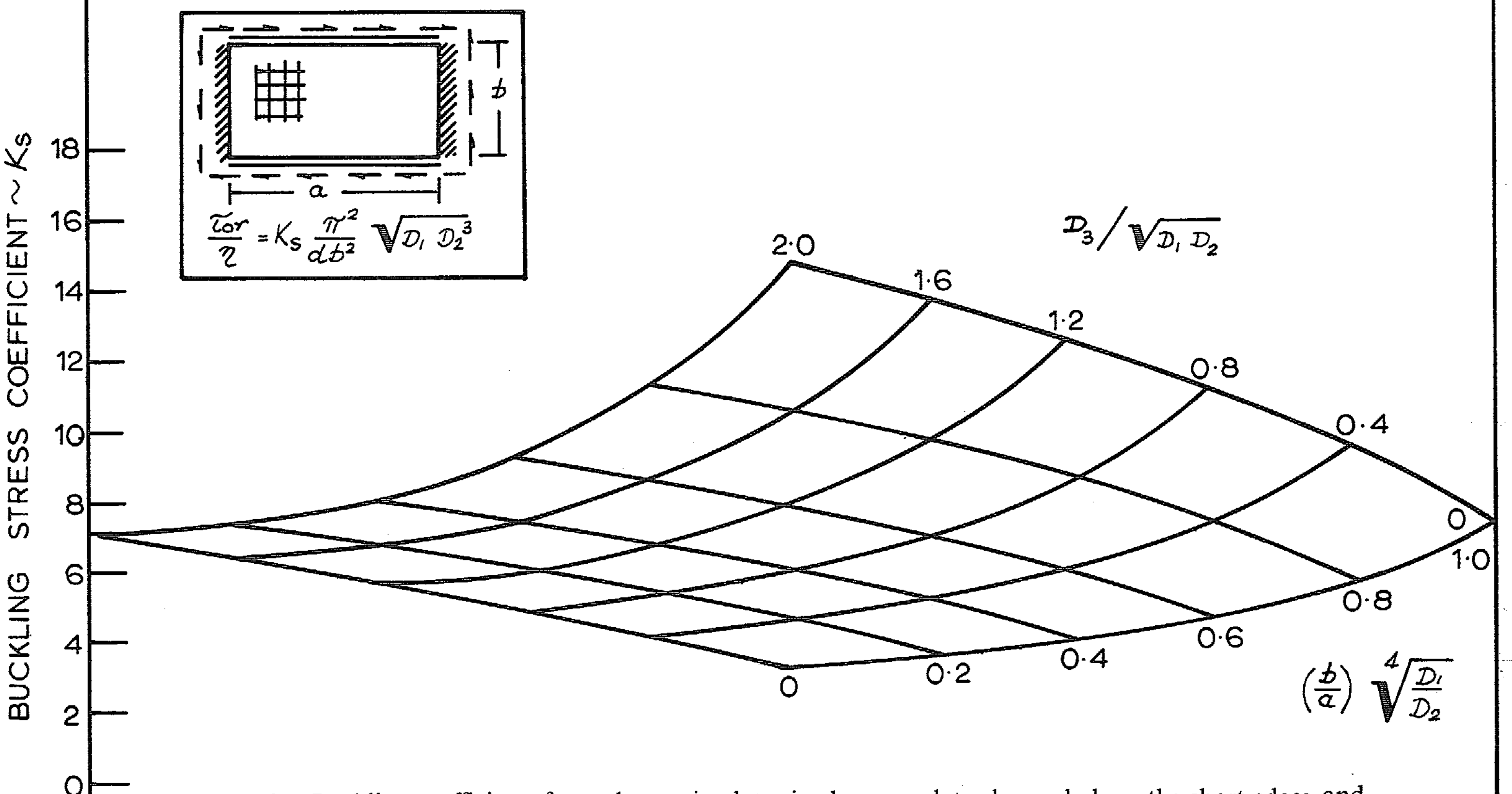

FIG. 9. Buckling coefficients for orthotropic plates in shear ~ plate clamped along the short edges and simply supported along the long edges. (Ref. 46).

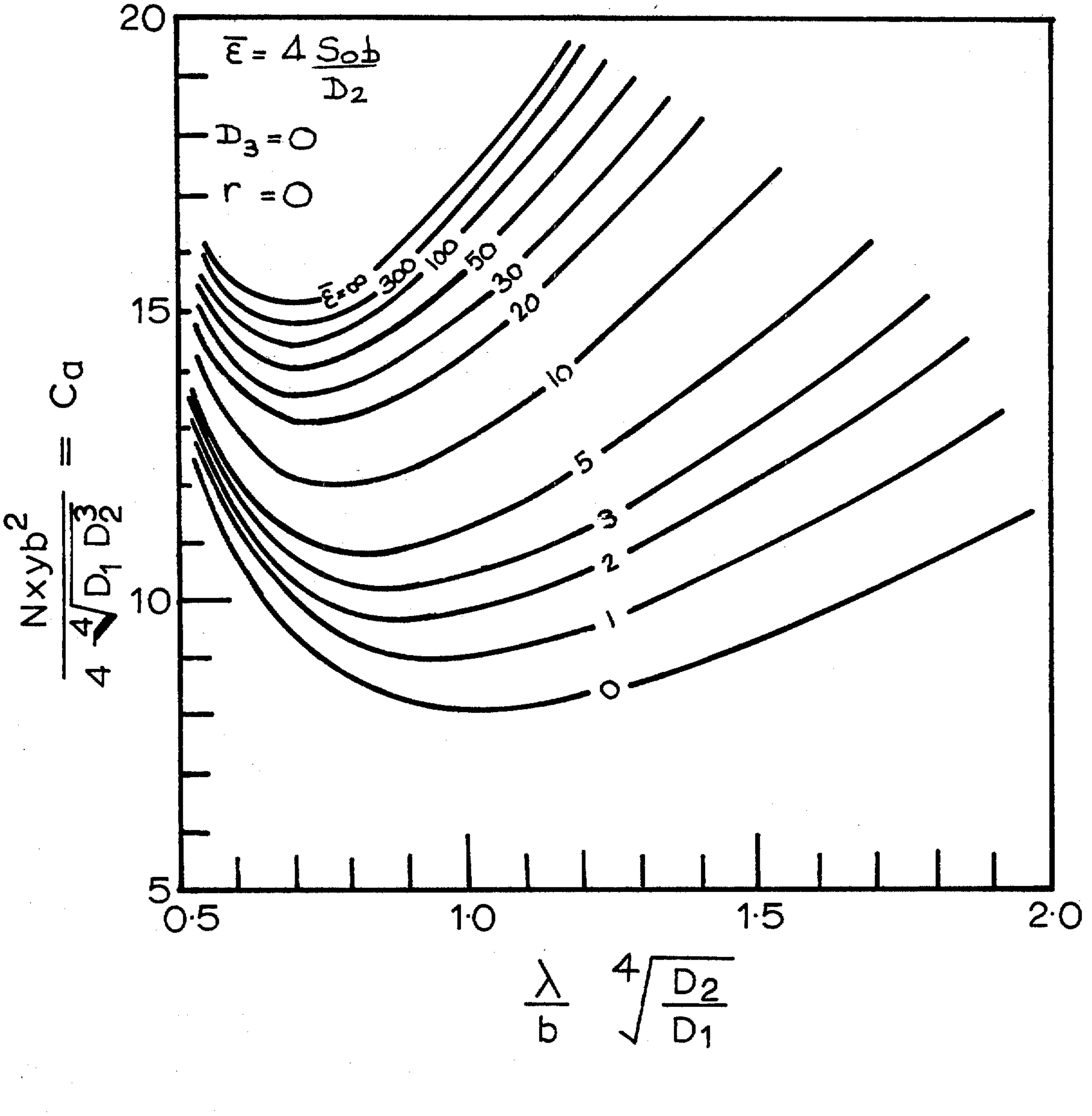

FIG. 10. Shear buckling coefficients of infinitely long corrugated plates with non-deflecting edge supports of variable torsional stiffness. (Ref. 45).

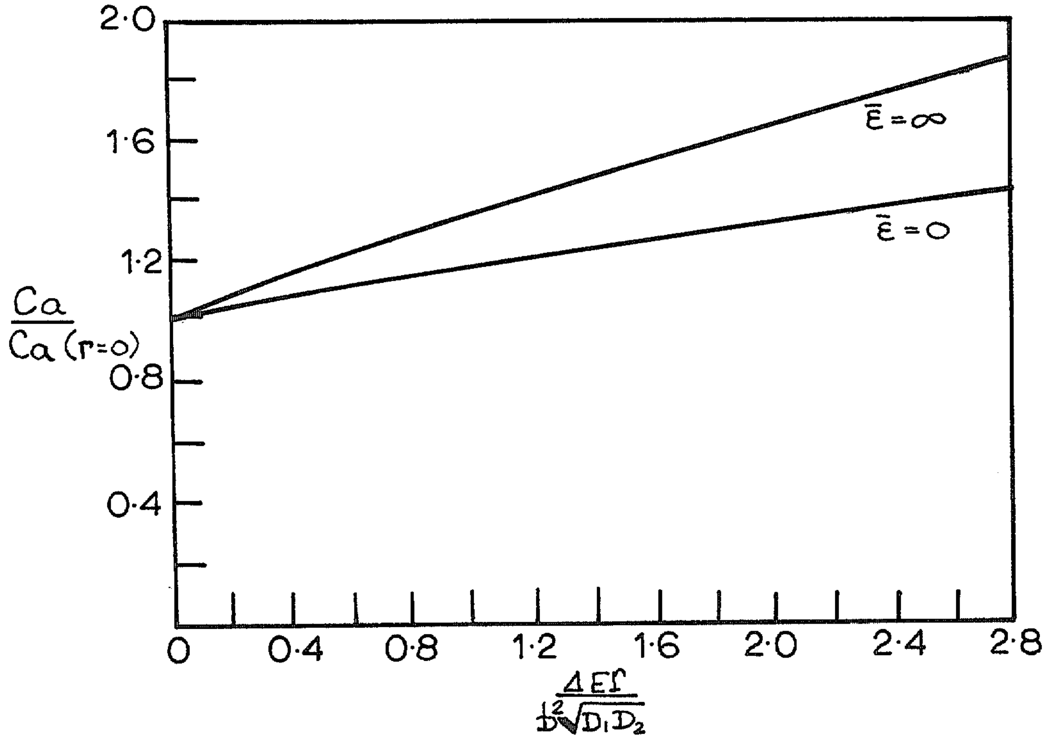

FIG. 11a. Effect of restraint of warping on the buckling strength of corrugated plates in shear.

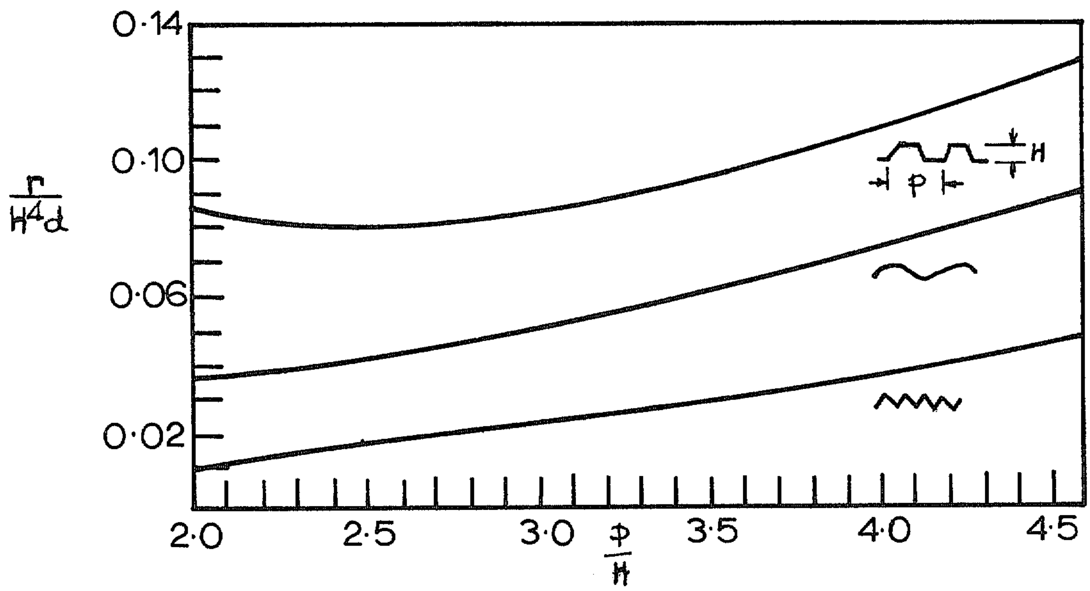

FIG. 11b. Bending torsion parameters for three common types of corrugation. (Ref. 45).

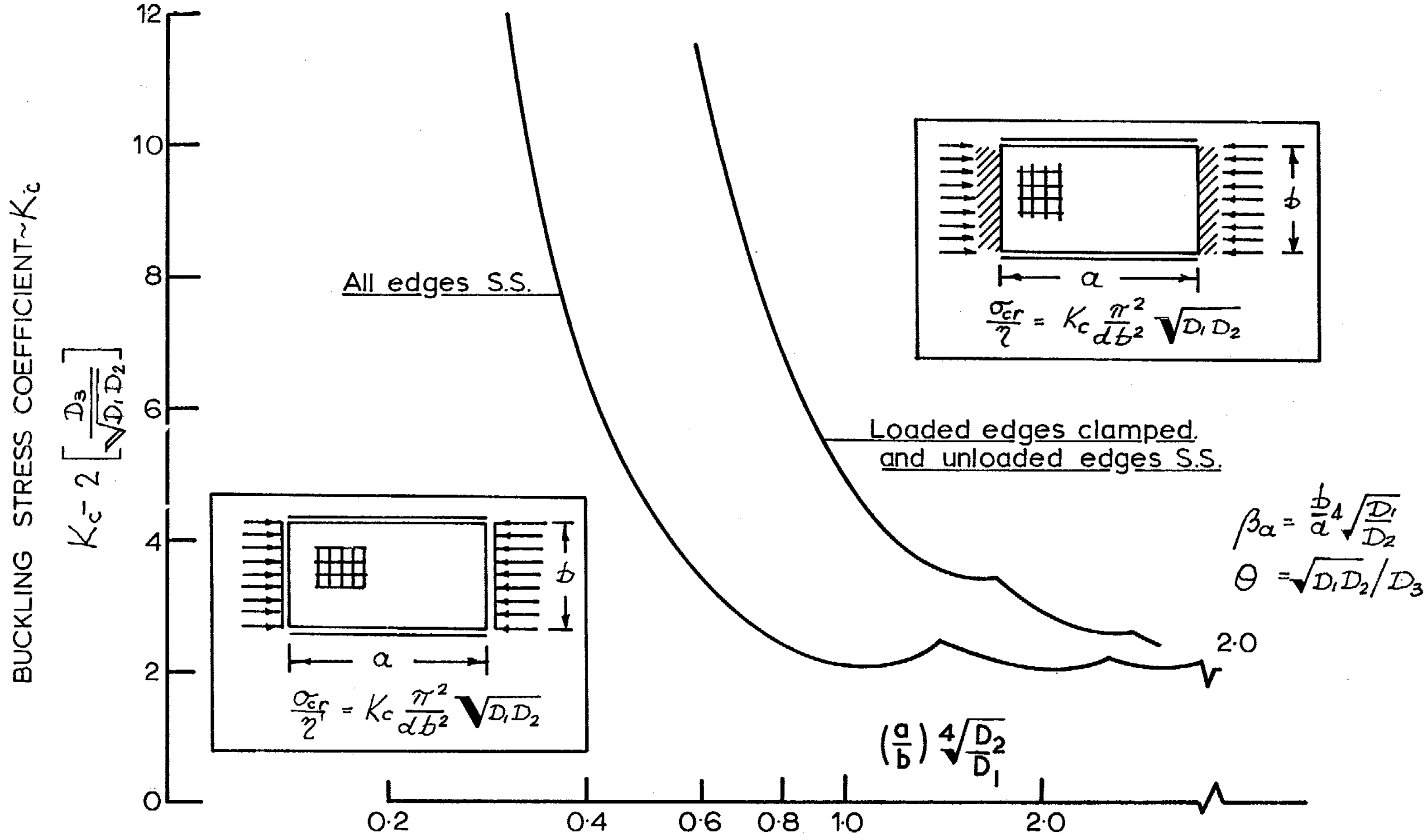

FIG. 12. Buckling coefficients for orthotropic plates in compression. (Ref. 46).

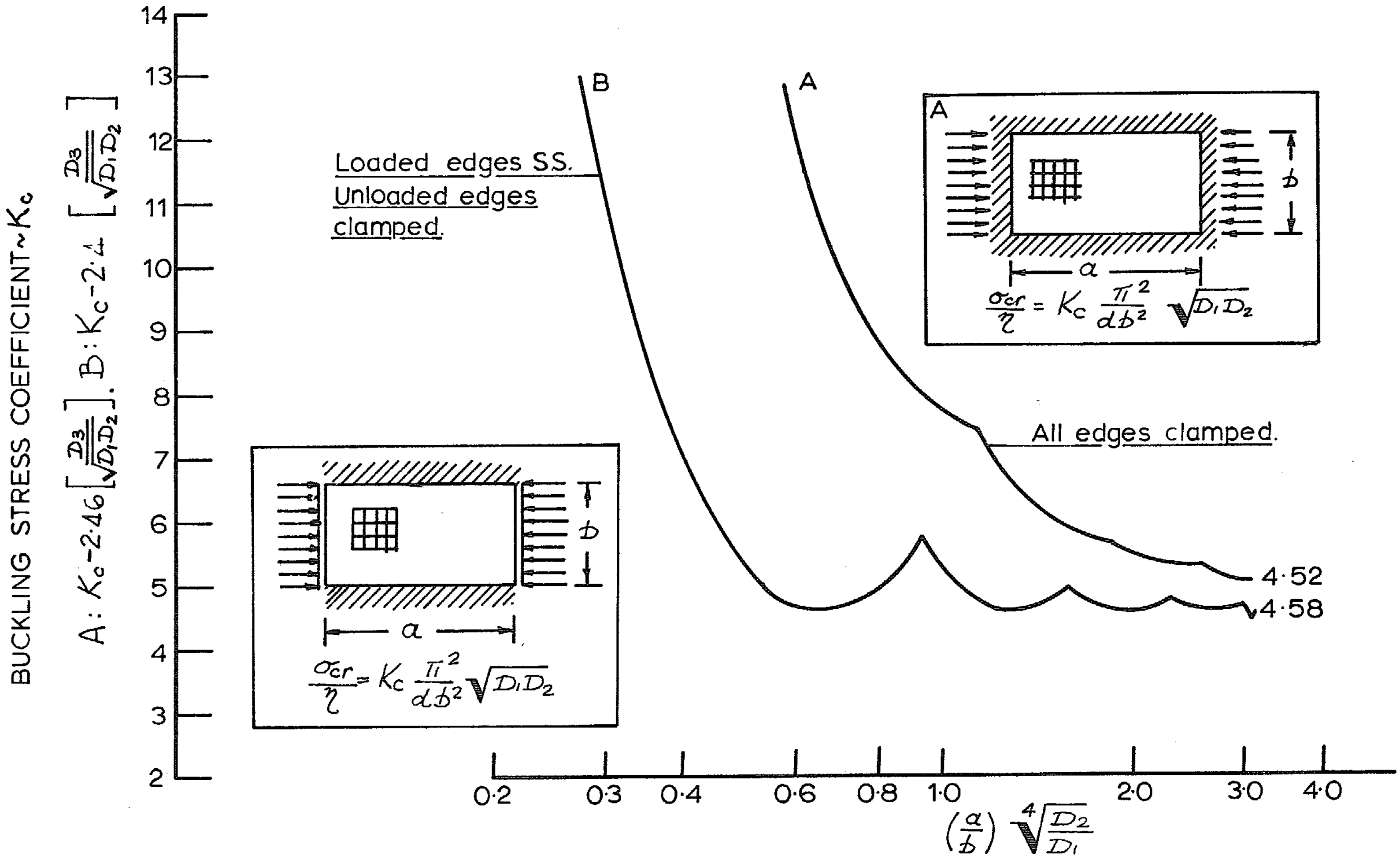

FIG. 13. Buckling coefficients for orthotropic plates in compression. (Ref. 46).

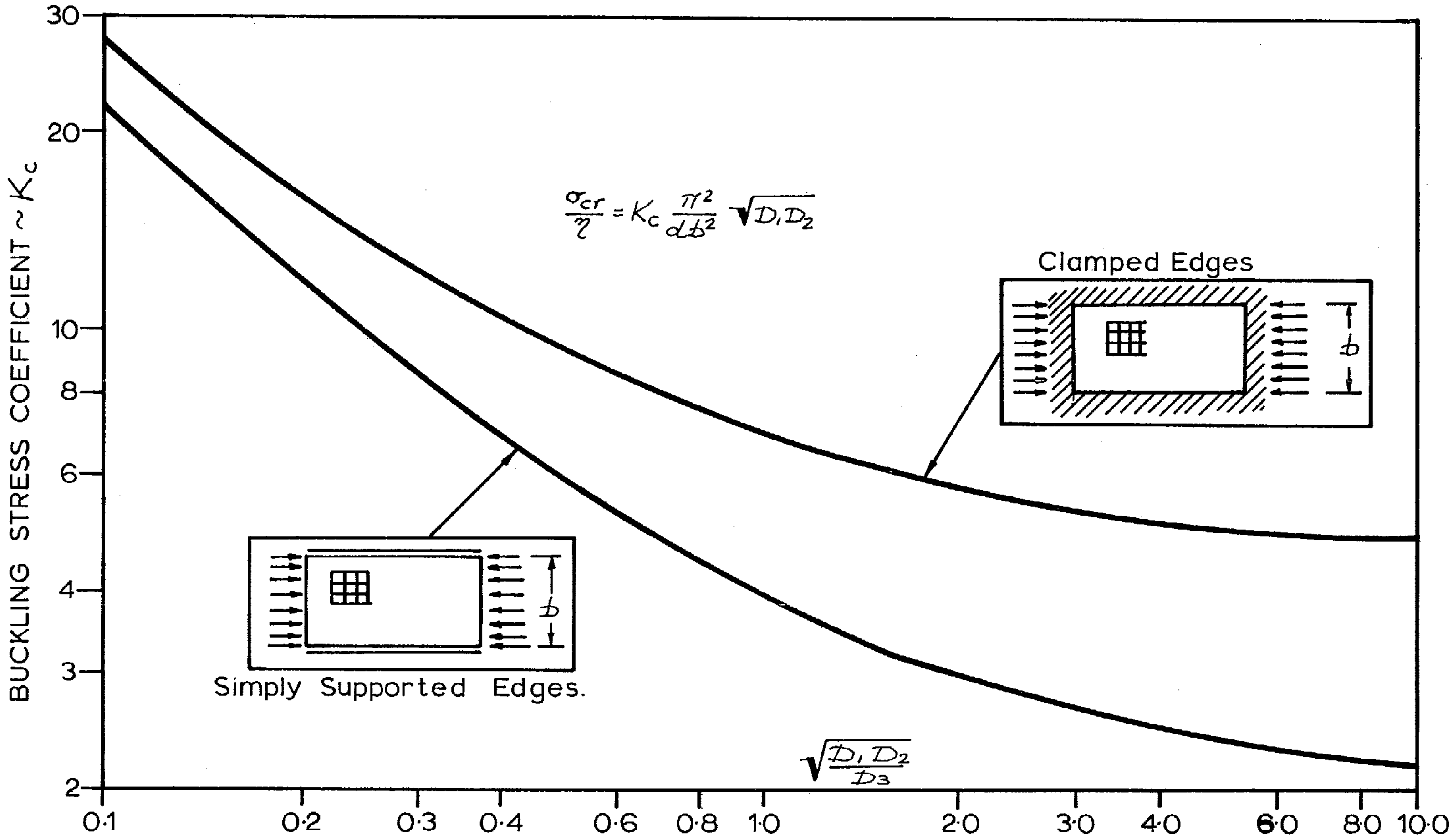

FIG. 14. Buckling coefficients for infinitely long orthotropic plates in compression. (Ref. 46).

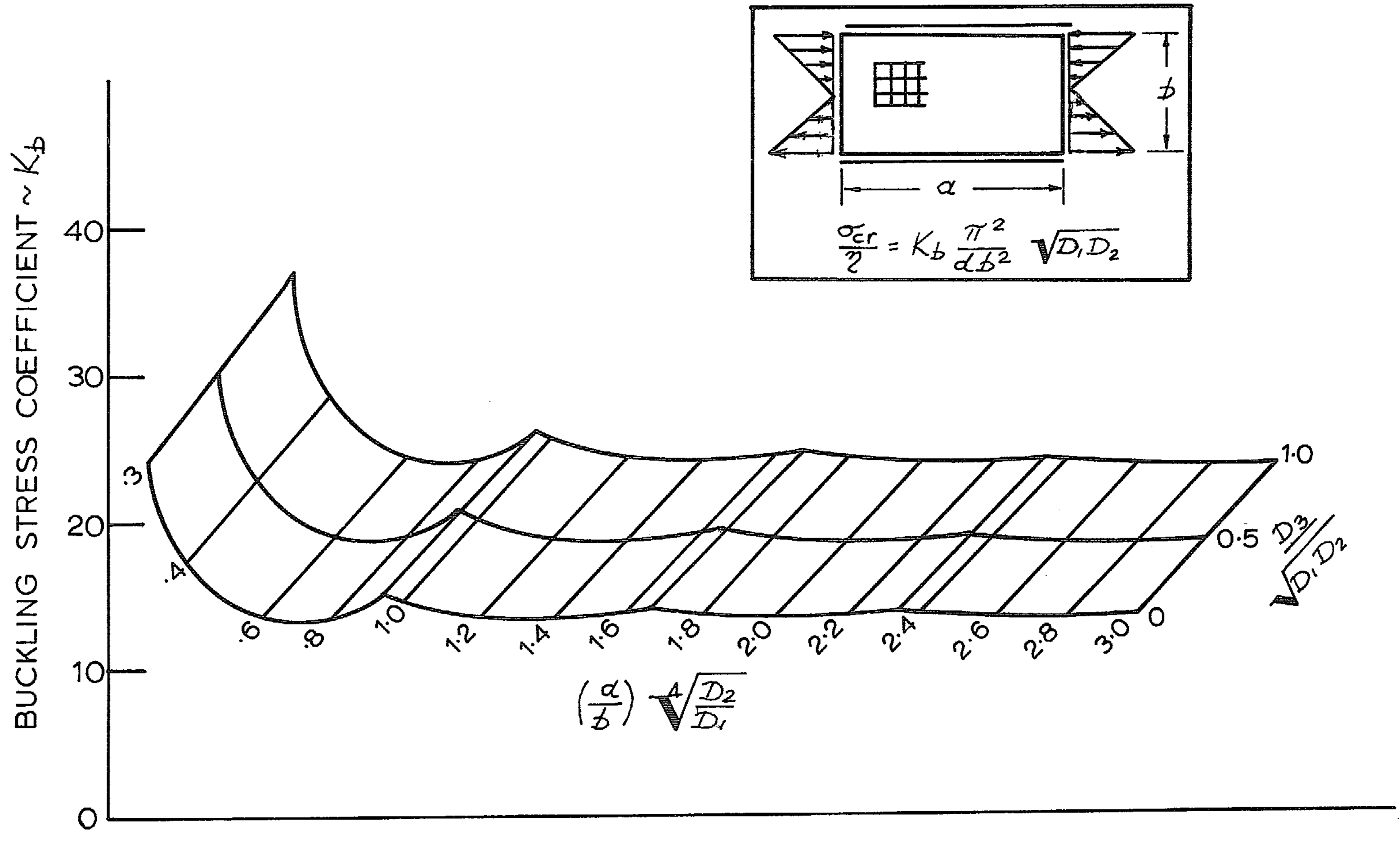

FIG. 15. Buckling coefficients for orthotropic plates under edgewise bending—all edges simply supported. (Ref. 46).

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