1- Publications in Ship Structural Analysis and Design (1969-2002)

1- " Effect of Variation of Ship Section Parameters on Shear Flow Distribution, Maximum Shear Stresses and Shear Carrying Capacity Due to Longitudinal Vertical Shear Forces", European Shipbuilding, Vol. 18. (Norway-1969), Shama, M. A.,

2- "Effect of Ship Section Scantlings and Transverse Position of Longitudinal Bulkheads on Shear Stress Distribution and Shear Carrying Capacity of Main Hull Girder", Intern. Shipb. Progress, Vol. 16, No. 184, (Holland-1969), Shama, M. A.,

3- "On the Optimization of Shear Carrying Material of Large Tankers", SNAME, J.S.R, March. (USA-1971), Shama, M. A.,

4- " An Investigation into Ship Hull Girder Deflection", Bull. of the Faculty of Engineering, Alexandria University, Vol. XII., (Egypt-1972), Shama, M. A.,

5- "Effective breadth of Face Plates for Fabricated Sections", Shipp. World & Shipbuilders, August, (UK-1972), Shama, M. A.,

6- " Calculation of Sectorial Properties, Shear Centre and Warping Constant of Open Sections", Bull., Of the Faculty of Eng., Alexandria University, Vol. XIII, (Egypt-1974), Shama, M. A.

7- "A simplified Procedure for Calculating Torsion Stresses in Container Ships", J. Research and Consultation Centre, AMTA, (EGYPT-1975), Shama, M. A.

8- "Structural Capability of Bulk Carries under Shear Loading", Bull., Of the Faculty of Engineering, Alexandria University, Vol. XIII, (EGYPT-1975), Also, Shipbuilding Symposium, Rostock University, Sept. (Germany-1975), Shama, M. A.,

9- "Shear Stresses in Bulk Carriers Due to Shear Loading", J.S.R., SNAME, Sept. (USA-1975) Shama, M. A.,

10- "Analysis of Shear Stresses in Bulk Carriers", Computers and Structures, Vol.6. (USA-1976) Shama, M. A.,

11- "Stress Analysis and Design of Fabricated Asymmetrical Sections", Schiffstechnik, Sept., (Germany-1976), Shama, M. A.,

12- "Flexural Warping Stresses in Asymmetrical Sections" PRADS77, Oct., Tokyo, (Japan-1977), Intern. Conf/ on Practical Design in Shipbuilding, Shama, M. A.,

13- "Rationalization of Longitudinal Material of Bulk Carriers, Tehno-Ocean'88, (Jpan-1988), Tokyo, International Symposium, Vol. II, A. F. Omar and M. A. Shama,

14- "Wave Forces on Space Frame Structure", AEJ, April, (Egypt-1992), Sharaki, M., Shama, M. A., and Elwani. M.,

15- "Response of Space Frame Structures Due to Wave Forces", AEJ, Oct., (Egypt-1992). Sharaki, M., Shama, M. A., and Elwani. M. H.

16- "Ultimate Strength and Load carrying Capacity of a Telescopic Crane Boom", AEJ, Vol.41., (Egyopt-2002), Shama, M. A. and Abdel-Nasser, Y.

EFFECTIVE BREADTH OF FACE PLATES FOR FABRICATED SECTIONS

M. A. Shama

With the continuing trend towards larger ships the use of fabricated sections, as opposed to rolled sections, for plate stiffeners has increased. In addition the economics of the fabrication process are such as to favour the asymmetrical section. Until recently there was no differentiation between symmetrical and asymmetrical sections when direct calculations were employed for the determination of scantlings. However, strain measurement on actual vessels has indicated disagreement between predicted and measured stresses in certain regions associated with asymmetrical sections. This has prompted a separate study of this phenomenon by Dr Shama, B.Sc., Ph.D., MRINA, of the Structures Section of the Research and Technical Advisory Services of Lloyd's Register of Shipping, from which this article has been abstracted.

In the elastic stress analysis of ship structures using the finite element method (FEM), the face plates of fabricated girders, transverses, brackets, etc, are normally idealised by bar or beam elements (Ref 1). For narrow symmetrical face plates the sectional properties of the idealised member can be computed directly from the face plate dimensions, assuming that it is fully effective. On the other hand, when dealing with offset or curved face plates, the sectional properties of the idealised member should be based on its actual contribution to the load carrying capacity of the structure (Ref 2). Using the full area of the face plate for calculating the sectional properties of the idealised member could lead to serious errors, particularly in the curved regions. By using the concept of 'effective breadth', (Ref 3), these errors can be avoided.

These remarks apply also to the elastic analysis of plane and space frames as well as grillage structures, where errors in the deflections and hence computed forces, moments and stresses may arise, particularly when asymmetrical sections are present. Much improved accuracy can be achieved when the sectional properties utilised in an analysis represent the effective section of each member.

In the following analysis, a method is given in the Appendix for calculating the effective breadth ratio for asymmetric face plates in terms of the relevant parameters of the member together with the torsional properties of the section. For symmetrical curved face plates, a design curve based on the method given in Ref 4 for determining the effective breadth ratio is presented. The validity of these methods was examined by using the finite element method and model testing.

Effective breadth concept

The fall off in normal stresses across the width of a face plate can result from shear lag, cross bending, torsion loading or any combination of these factors depending on the case under consideration. With straight members, shear lag effect in symmetrical face plates or flexural warping in asymmetrical face plates is the main cause of the fall off in normal stresses. However, in curved members, cross bending effect is responsible for the fall off in normal stresses.

The concept of effective breadth, Fig 1, is widely used in ship structural analysis to cater for this fall in normal stresses. Schade (Refs 3 and 5) has investigated various cases.

In the following analysis, the effective breadth of face plates on fabricated girders, longitudinals, brackets, etc, is investigated. These cases are not covered by Schade's curves since curvature and torsion are the main factors responsible for the fall off in normal stresses across the width of the face plate.

(a) Straight members

These include fabricated longitudinals, stiffeners, girders, etc.

(i) Symmetrical face plates.

The sectional properties of the idealised bar or beam elements can be computed directly from the face plate dimensions as the geometry and scantlings normally used for fabricated symmetrical sections tend to have fully effective face plates. However, should these face plates become exceptionally wide, their effective breadth can be determined using the curves in Ref 3.

(ii) Asymmetrical face plates.

For this case Schade (Ref 5) presented a method and design curves based on shear lag phenomena, showing that the maximum effective breadth ratio was 0·25. Yuille (Ref 6), attributed the fall off in normal stresses across the face plate width to flexural warping stresses. Following Yuille's conception, a method is given in the Appendix for the calculation of the effective breadth ratio for a straight member fully fixed at both ends and subjected to uniform loading. This ratio is given in terms of the torsional characteristics of the section and the length of the member.

(b) Curved members

(i) Symmetrical face plates.

The circumferential stresses in a symmetrical face plate induce radial forces which cause the outer edges of the face plate to deflect radially. This effect is sometimes called 'cross bending', see Fig 2, and has an adverse effect on the effectiveness of such face plates. As a result, high stresses may be induced in both the curved plate and the web near the face plate. Cross bending also induces transverse stresses in the curved face plate which may be higher than the circumferential stresses.

For this type of face plate, Bleich (Ref 7), gave coefficients for the calculation of both transverse stresses and effective breadth ratio. Anderson (Ref 4), carried out a theoretical and experimental investigation for calculating the flexural stresses in curved beams of I and box sections, and gave a method for calculating the effective breadth ratio and transverse stresses. Fig 3 gives a design curve, based on Anderson's method, for determining the effective breadth ratio of curved face plates in terms of the width, thickness and radius of curvature of the face plate.

The effective breadth ratio is given by:

$$\lambda = \frac{1}{\psi} \cdot \frac{\sin\psi \cdot \cos\psi + \sinh\psi \cdot \cosh\psi}{\cos^2\psi + \cosh^2\psi}$$

where $\psi = \dfrac{1{\cdot}28\, b}{\sqrt{R \cdot t_f}}$

b, t_f and R are half width, thickness and radius of curvature of the face plate respectively, see Fig 3.

(ii) Asymmetrical face plates.

For this type of face plate, the effective breadth not only depends upon the width, thickness and radius of curvature of the face plate, but it also depends upon the stiffness of the web and attached plating as well as the torsional characteristics of the section. In this case, cross-bending and flexural warping are the main factors affecting the stress distribution across the width of the face plate.

Since a rigorous theoretical solution of this type of face plate is outside the scope of this study, the effective breadth is examined using the FEM and model tests, from which an empirical design curve is presented, see Fig 4.

Elastic stress analysis using the FEM

Two programs employing the finite element method were used for this investigation:

(i) LR 215 Plane Stress Analysis (Lloyd's Register Computer Program Library)

(ii) ICSL FEM program (Ref 10)

Program (i) is used for the elastic stress analysis of two dimensional plane stress problems. Triangular and rectangular plate elements, together with bar elements are used.

Program (ii) is used for the elastic stress analysis of two and three dimensional plane stress problems. Triangular and quadrilateral plate elements, together with beam elements, are used.

In the above programs, a bar element carries only axial loading, a beam element carries axial, shear and bending loads, a triangular plate element assumes uniform normal and shear stresses and a quadrilateral plate element assumes linear normal stresses and constant shear stress (Ref 9). For both types of plate elements, the stresses are computed at the centroid of the element.

The cases investigated by the FEM were:

(1) Straight member with asymmetrical face plate.

(2) Symmetrical and asymmetrical curved face plates.

Model tests

A series of twelve model tests was carried out at Lloyd's Register of Shipping's Research Laboratory at Crawley. The main objective of these tests was to investigate the validity of the theoretical methods. The results of the tests were as follows:

(a) Symmetrical curved face plates.

Fig 6 shows the normal stress distribution across the breadth of the face plate at two sections for model No. 3. When the width of the face plate was reduced and when a thick face bar was used, the fall off in stress was markedly reduced giving a higher effective breadth ratio.

(b) Asymmetrical curved face plate.

Fig 7 shows the normal stress distribution across the face plate width for three sections of model No. 7. When the width of the face plate was reduced, there was still a marked fall off in stress which presumably results mainly from the flexural warping stresses.

(c) Straight members with asymmetrical face plate.

From the measured stress distribution across the width of the face plate at both midspan and at 0·5 in from the fixed end, the effective breadth ratio was calculated and found to be approximately 0.38 at mid-span. When the bottom and web plating were stiffened by stiffeners and tripping brackets respectively, the effective breadth ratio was increased to about 0·72. In this case, two tripping brackets were fitted within the 36 in span.

Discussion of results

1. Straight member with asymmetrical face plate.

From Fig 5 the effective breadth ratio is calculated for the two conditions of web thickness and the result is compared with the corresponding values calculated according to the Appendix as follows:

t_w mm	Effective breadth ratio	
	F.E.M.	Proposed method
15	0·3	0·256
25	0·31	0·262

The result indicates that the method given in the Appendix underestimates the effective breadth ratio by about 5%. Also, it is shown that increasing the web thickness by 67% does not improve the effectiveness of the face plate.

Hence the use of the full sectional area of the face plate in calculating the sectional properties of the member underestimates the maximum stress in both web and face plate. These errors result mainly from the additional flexural warping stresses.

In reference 8, four tests were carried out on stiffened plating using different sizes of angle stiffeners. From these test results, the effective breadth ratio was found to vary between 0.6 and 0.75. Using the method given in the Appendix, the effective breadth ratio at mid-span for these members varies between 0·62 and 0·83. From these results, it is shown that for rolled asymmetrical sections, the fall off in normal stresses in the flanges due to flexural warping stresses is much less than the fall off in the large fabricated sections. Furthermore, it is shown that the method given in the Appendix gives good results for rolled asymmetrical sections.

For fabricated sections, reinforcement of web plating seems to be a major factor for improving the effectiveness of the face plate provided the bottom plate is also stiffened. Model test results showed that the effective breadth ratio at mid-span increases by 100% when the web is stiffened by tripping brackets.

The computed effective breadth ratio, using the method given in the Appendix is 0.25 for both the fixed end section and mid-span. This result agrees with the measured value for the fixed end, which gives $\lambda = 0{\cdot}265$ but is about 13% less at mid-span. This discrepancy at mid-span results mainly from the bending of the web and the face plate out of their planes.

The model tests showed that the effective breadth ratio is increased by increasing the thickness of the face plate and/or the radius of curvature or by reducing its width.

It was also shown that, in general, there is good agreement between the measured effective breadth ratio and the corresponding values given by Fig 3, as shown in Table 1.

2. Curved members with asymmetrical face plates.

The model tests results showed that, in general, the effectiveness of curved face plates is very low.

The results in Table II demonstrated that increasing the thickness or reducing the width of a face plate only marginally improves its load carrying capacity.

The effective breadth ratio calculated from the results of the FEM for the structure of model test No. 7 was 0·26, which is in good agreement with the test result. From the results, it is evident that the flexural warping stresses are more responsible than the cross bending effect for the reduction of the effectiveness of asymmetrical curved face plates. The cross bending effect is sharply reduced because of the bending of web plating.

Model No.	Effective breadth ratio	
	Test value	Theoretical value
1	0·6	0·57
2	0·84	0·925
3	0·7	0·685
4	0·94	0·89
5	0·93	0·86
6	1·0	0·985

Table I. Effective breadth ratio for symmetrical curved face plates

Test No.	Effective breadth ratio	
	Test value	λ_c
7	0·277	0·410
8	0·412	0·650
10	0·330	0·575
11	0·400	0·880
12	0·588	0·980

Table II. Effective breadth ratio for asymmetrical curved face plates λ_c = effective breadth ratio obtained from Fig 4

In the elastic stress analysis of ship structures by the FEM, the face plates of straight and curved members can be idealised by bar or beam elements provided that the sectional properties of these idealised members are based on their effective breadth, as derived from the relevant design curve or the Appendix.

In order to improve the effectiveness of curved face plates the following condition should be satisfied:

$b \leqslant \sqrt{R \cdot t}$ for symmetrical face plates

The effectiveness of asymmetrical face plates is greatly improved if web and plate bending is reduced. This can be achieved by fitting effective tripping brackets.

Although service experience to date with asymmetrical sections has been satisfactory, this study indicates that this experience may not necessarily be applicable to larger fabricated sections. The initial advantages in the economy of fabrication of asymmetrical sections may be outweighed by a shorter fatigue life as a consequence of the higher maximum stress compared with the symmetrical section. This aspect is now the subject of further study.

Appendix

In the following analysis, the effective breadth of an asymmetrical face plate is determined using the following assumptions:

1. The fall off in normal stresses across the width of the face plate results mainly from the flexural warping stresses.
2. The stiffness of the web and attached plating is relatively large.

The flexural warping stresses are calculated from the torsion equation for a uniform member (Ref 12) as follows:

$$M_t = C\,\frac{d\phi}{dx} - C_1\,\frac{d^3\phi}{dx^3} \quad \ldots\ldots (1)$$

where C and C_1 are the torsional and warping rigidities respectively. ($C = G \cdot J_t$, $C_1 = E \cdot J_w$).

The torque M_t is given by:

$$M_t = M_o + qex \quad \ldots\ldots (2)$$

where M_o = torque at the beam end

e = distance between the applied load and shear centre.

The general solution of equation (1) is as follows:

$$\phi = \phi_o + \frac{\phi_o'}{K}\sinh Kx + \frac{\phi_o''}{K^2}(\cosh Kx - 1) + \frac{M_o}{C}\left(x - \frac{\sinh Kx}{K}\right) + \frac{qex^2}{2C} \quad \ldots\ldots (3)$$

$$\text{where } K = \sqrt{C/C_1} \quad \ldots\ldots (4)$$

ϕ_o, ϕ_o' and ϕ_o'' are constants of integration and could be determined from the end conditions.

For a uniform bar fixed at both ends, the end conditions are:

at $x = o$ and $x = l$

$\phi = \phi' = o$

Hence $\phi_o = \phi_o' = o$

Due to symmetry $M_o = -\dfrac{qel}{2}$

Hence equation (3) becomes:

$$\phi = \frac{qel}{KC}\left[\frac{\sinh \frac{Kx}{2} \cdot \sinh \frac{K(l-x)}{2}}{\sinh \frac{Kl}{2}} - \frac{Kx(l-x)}{2l}\right] \quad \ldots (5)$$

where q = uniform load per unit length.

The flexural warping stresses are given by (Ref 12).

$$\sigma_w = E \cdot \omega \cdot \frac{d^2\phi}{dx^2} \quad \ldots\ldots (6)$$

where ω = principal sectorial co-ordinate

$$\text{and } \frac{d^2\phi}{dx^2} = \frac{qel}{2C}\left[\frac{2}{l} - \frac{K \cosh\left(Kx - \frac{Kl}{2}\right)}{\sinh \frac{Kl}{2}}\right] \quad \ldots (7)$$

The bending stress in the face bar due to lateral loading is given by:

$$\sigma_b = \frac{M_y (I_y \cdot Z - I_{yz} \cdot y)}{I_y \cdot I_z - I_{yz}^2} \quad \ldots\ldots (8)$$

where I_z, I_y, I_{yz} are the moments and product of inertia about the y and z axes, see Fig 8.

However, for the normal dimensions of asymmetrical sections used in ship construction, the angle between the y and z axes and the principal axes is relatively small. Consequently, the bending stress could be approximately given by:

$$\sigma_b = \frac{M_y \cdot Z}{I_y} \quad \ldots\ldots (9)$$

where M_y = bending moment

$$= q\left(\frac{l^2}{12} - \frac{lx}{2} + \frac{x^2}{2}\right)$$

Hence, using equations (6), (7) and (9), the effective breadth ratio at any position x is given by:

$$\lambda_x = \frac{\sigma_b + \frac{1}{2}[(\sigma_w)_i + (\sigma_w)_o]}{\sigma_b + (\sigma_w)_i} \quad \ldots\ldots (10)$$

where $(\sigma_w)_i$, $(\sigma_w)_o$ are the flexural warping stress at the inner and outer edges respectively.

The mean effective breadth ratio is given by:

$$\lambda_m = \frac{2}{l}\int_0^{l/2} \lambda_x \, . \, dx \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (11)$$

At the fixed end, ie at $x = 0$, equation (6) becomes:

$$\sigma_w = \frac{Eqe}{C}\left[1 - \frac{Kl}{2\tanh\frac{Kl}{2}}\right] . \, \omega \qquad \ldots\ldots\ldots\ldots\ldots (12)$$

and equation (9) becomes:

$$\sigma_b = \frac{ql^2 . Z}{12\, I_y}$$

Hence, the effective breadth ratio is given by:

$$\lambda_F = \frac{\dfrac{Zl^2}{12\, I_y} + \dfrac{1{\cdot}3e}{J_t}\left(1 - \dfrac{Kl}{2\tanh\frac{Kl}{2}}\right)(\omega_i + \omega_o)}{\dfrac{Zl^2}{12\, I_y} + \dfrac{2{\cdot}6e}{J_t}\left(1 - \dfrac{Kl}{2\tanh\frac{Kl}{2}}\right) . \, \omega_i}$$

where ω_i, ω_o = principal sectorial co-ordinates for the inner and outer edges

J_t = torsion constant of the section.

NOMENCLATURE

B = total width of a symmetrical face plate
b = width of an asymmetrical face plate or $\frac{1}{2}$ width of a symmetrical face plate
b_e = effective breadth
C = torsional rigidity of a section ($C = G\, J_t$)
C_1 = warping rigidity of a section ($C_1 = E\, J_w$)
E = elastic modulus
e = distance of the shear centre from the web
FEM = finite element method
G = shear modulus
I_y, I_z = second moment of area about the y and z axes respectively
I_{yz} = product of inertia about the y and z axes
J_t = torsional constant of a section
J_w = warping constant of a section
K = $\sqrt{C/C_1}$
l = length of member
M = bending moment
M_t = torque
P = a force vector
q = uniform load
R = radius of curvature
S = spacing between stiffeners or longitudinals (S = effective breadth of plating)
t = thickness
t_f, t_w = thickness of face bar and web respectively
x, y, z = co-ordinate axes
Z = distance of face bar from centroidal axis
σ = stress
σ_b = bending stress
σ_i, σ_o = stress at the inner and outer edges of the face plate respectively
σ_w = flexural warping stress
$(\sigma_w)_i, (\sigma_w)_o$ = flexural warping stress at the inner and outer edges of the face plate respectively
φ = angle of twist
ω = principal sectional co-ordinate
λ = effective breadth ratio
ψ = non-dimensional coefficient $\psi = \frac{1{\cdot}28b}{\sqrt{Rt_f}}$

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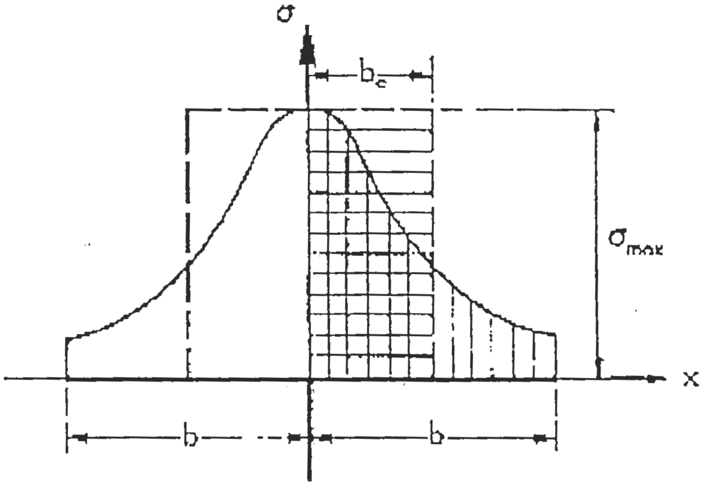
REFERENCES

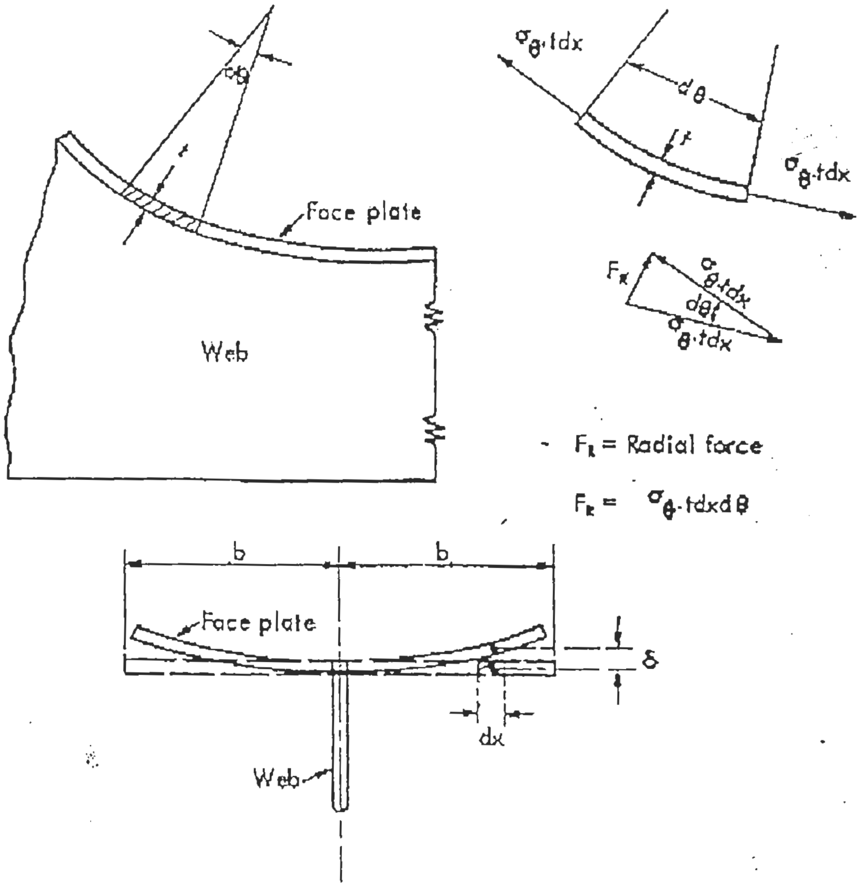
1. ISSC. Report (5): Stiffened panels in three dimensional structures. Tokyo, Japan, 1970.
2. RYUICHI NAGAMOTO, *et al.*: On the strength of the transverse webs and their joints of large tankers. Mitsubishi Heavy Industries Technical Review, Vol. 6, No. 2—1969.
3. SCHADE, H. A.: The effective breadth concept in ship structure design. SNAME, November 1953.
4. ANDERSON, C. G.: Flexural stresses in curved beams of I and box sections. IME, 1950.
5. SCHADE, H. A.: The effective breadth of stiffened plating under bending loads. SNAME, Vol. 59, 1951.
6. YUILLE, I. M.: Application of the theory of elastic bending to the structural members of ships. RINA, April 1960.
7. BLEICH, H. H.: Spannungsverteilung in den Gurtungen Querschnitt. Der Stahlbau, January 1933.
8. VINER, A. C., *et al.*: Lateral loading tests on bulkhead specimens with inverted angle stiffeners. BSRA, RB 2049.
9. ZIENKIEWICZ, O. C., CHEUNG, Y. K.: The finite element method in structural and continuum mechanics. McGraw-Hill Book Co., 1967.
10. Structural Analysis. Reference Manual, Bureau Division, English-Electric-Leo-Marconi Computers Ltd. Publication EM/1004, 1967.
11. TIMOSHENKO, S. P.: Strength of materials, Part II. D. Van Nostrand Co. Inc., 1956.
12. VLASOV, V. Z.: Thin-walled elastic beams. English translation National Science Foundation, Washington, D.C., London, Old Bourne Press, 1961.



b_e = Effective breadth

$\lambda = b_e/b$

Fig. 1 EFFECTIVE BREADTH CONCEPT.



δ = Radial deflection

Fig. 2 CROSS - BENDING EFFECT.

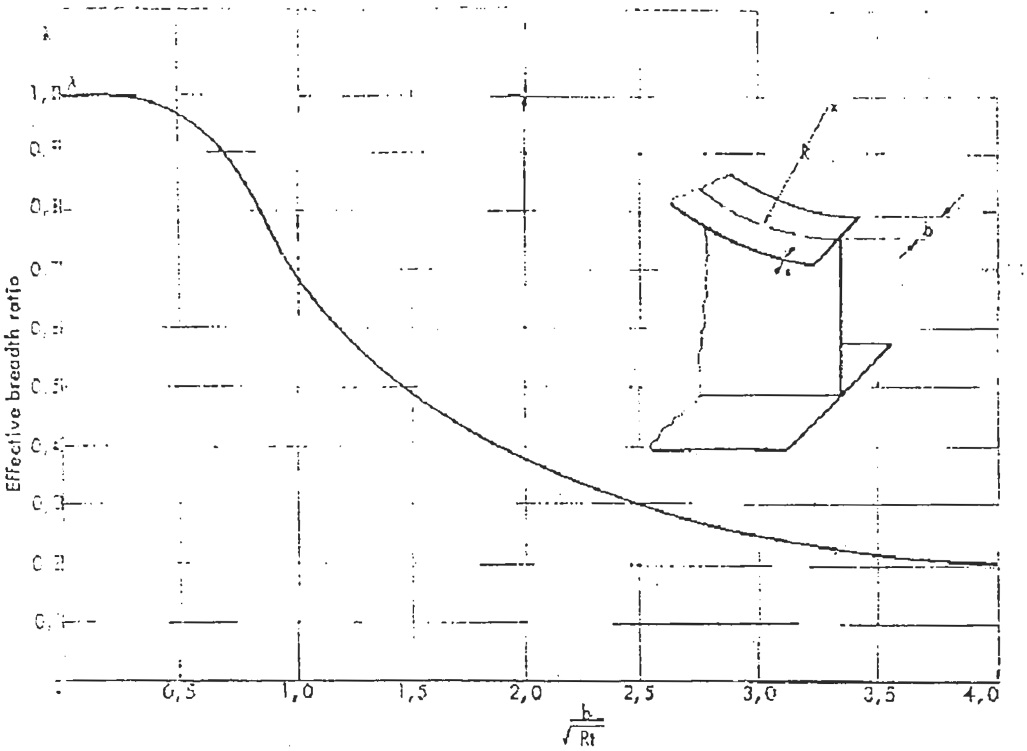


Fig 3 EFFECTIVE BREADTH RATIO FOR SYMMETRICAL CURVED FACE PLATES

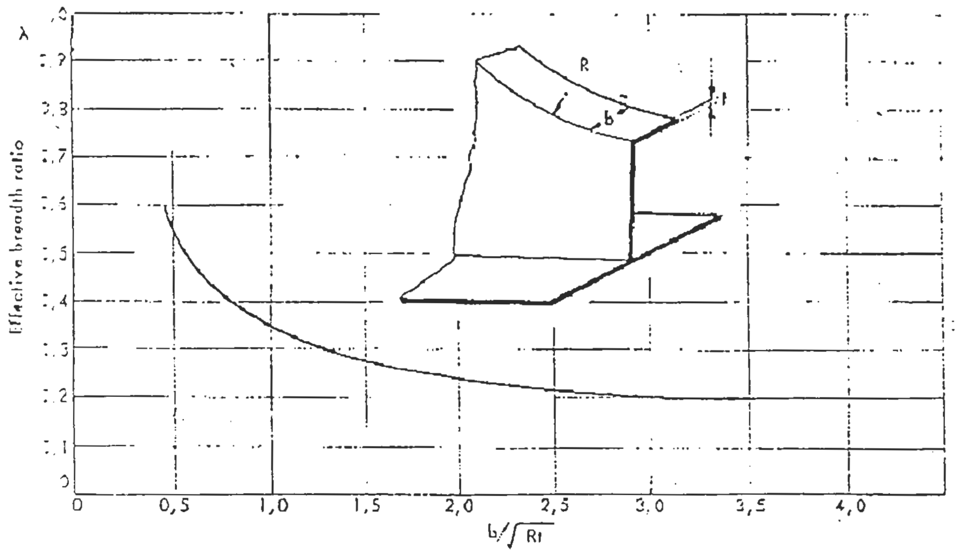


Fig 4 EFFECTIVE BREADTH RATIO FOR ASYMMETRICAL FACE PLATES

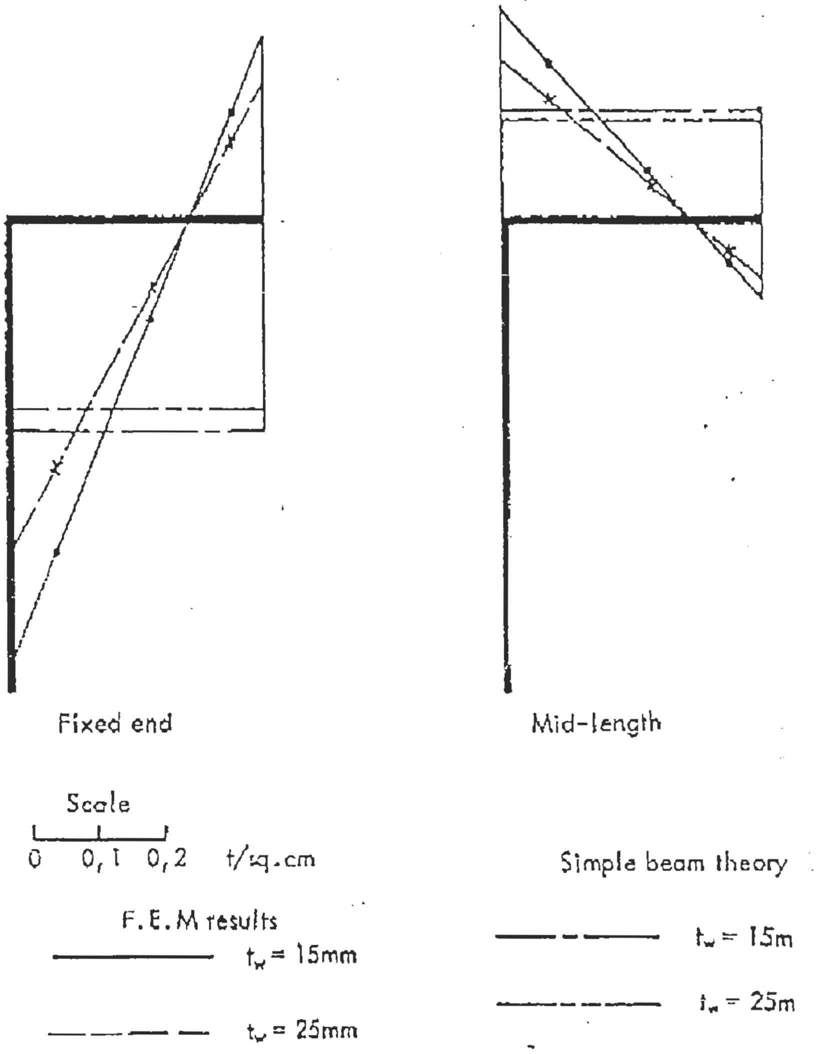


Fig. 5 FLEXURAL STRESS IN THE FACE PLATE

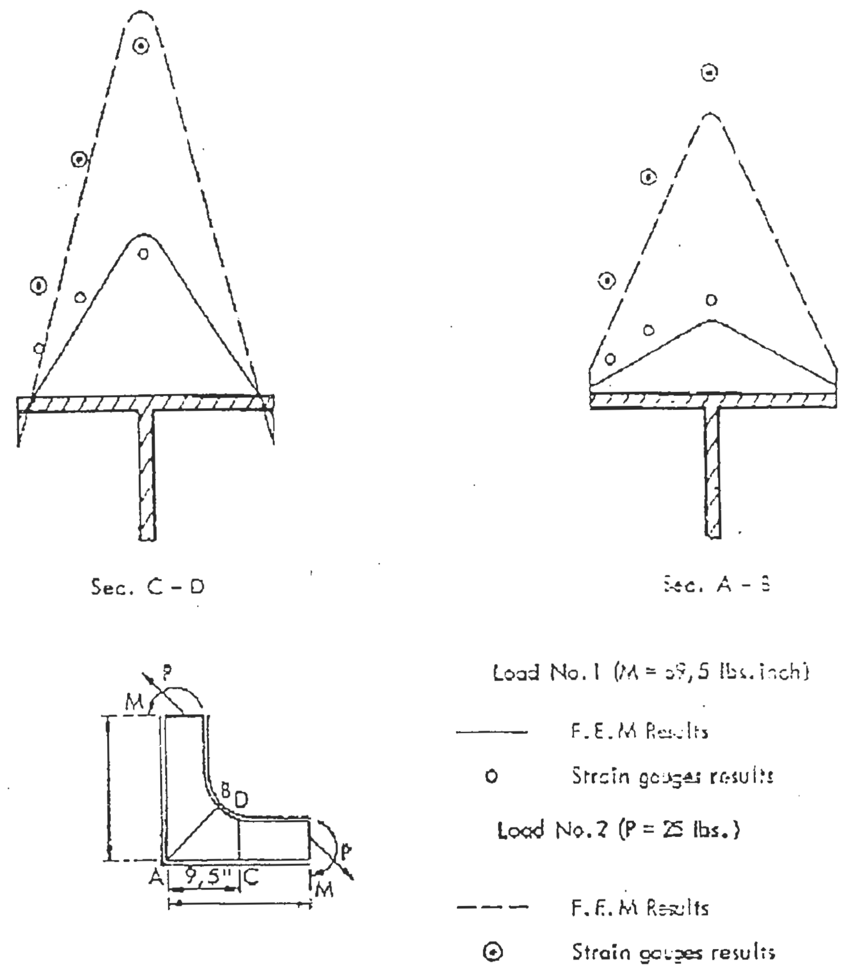


Fig. 6 LONGITUDINAL STRESS DISTRIBUTION IN THE FACE PLATE OF BRACKET (1)

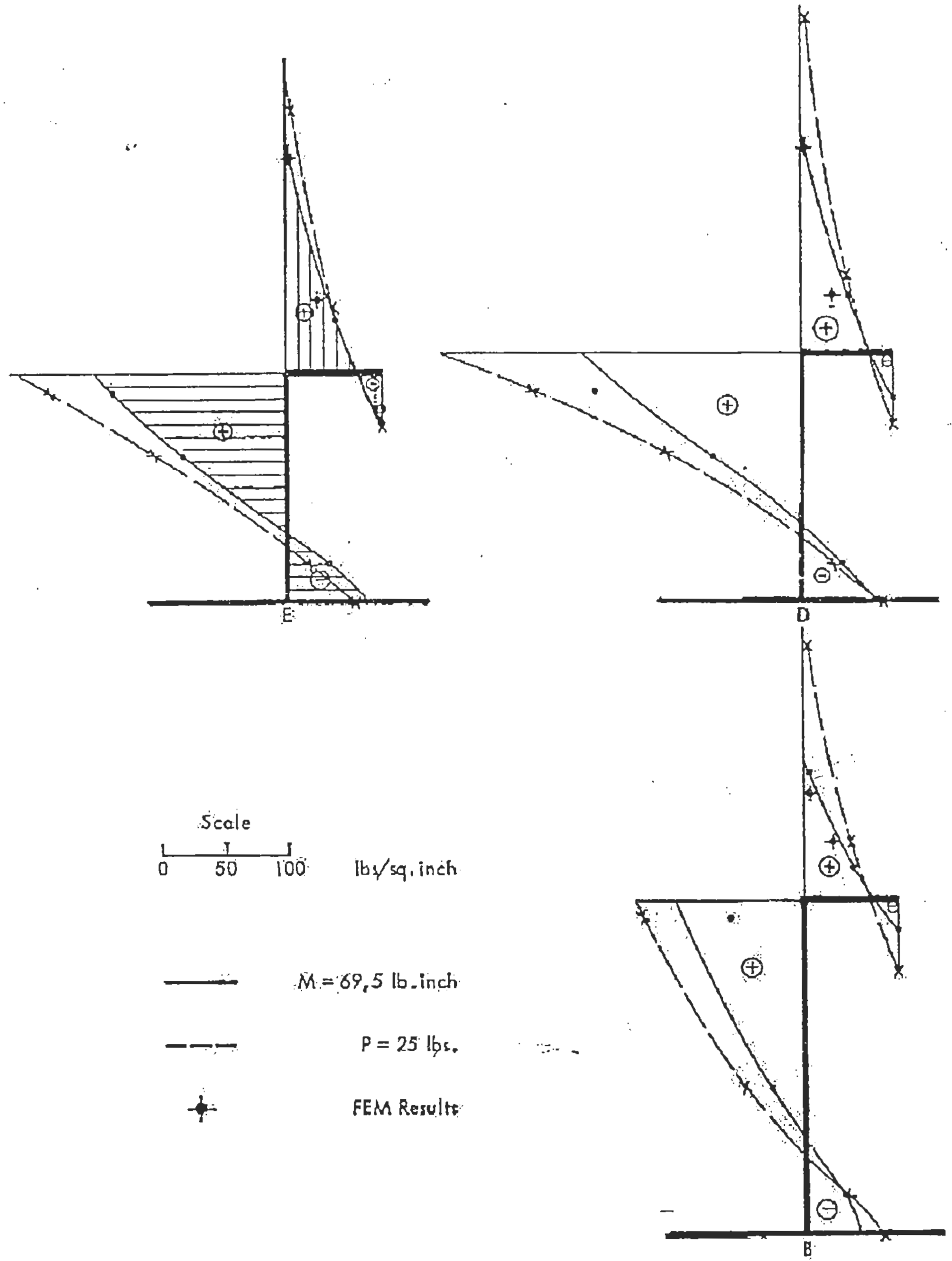


Fig. 7 BENDING STRESS DISTRIBUTION IN MODEL (7)

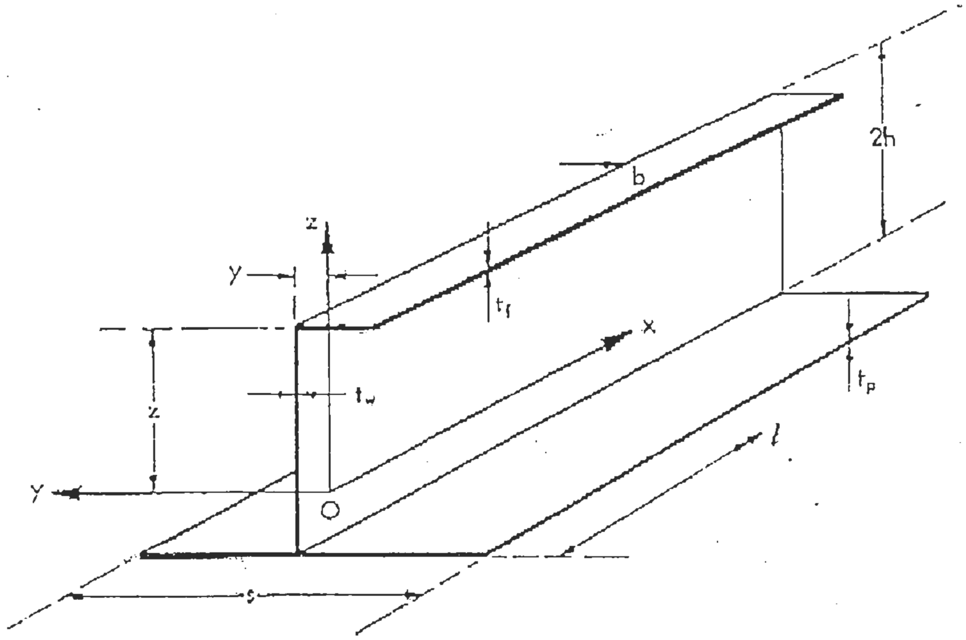


Fig. 8 COORDINATE SYSTEM.