

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 Carriers 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., (Egypt-2002), Shama, M. A. and Abdel-Nasser, Y.

ANALYSIS OF SHEAR STRESSES IN BULK CARRIERS

M. A. SHAMA*

Naval Architecture Dept, Faculty of Engineering, Alexandria University, Egypt

(Received 11 February 1975)

Abstract—A method is given for calculating the shear stress distribution over a ship section of a bulk carrier. The method is programmed for the Alexandria University IBM computer. The ship section is considered with minimum idealization to the double bottom structure so as to obtain realistic values of the shear stresses.

Several ship section configurations are studied and the results are analysed. A summary of the analysis is given together with a numerical example. A simplified expression is developed for calculating an approximate value of the maximum shear stress in the side shell plating. The results of this expression compare favourably with computer results.

The necessary conditions to safeguard against shear buckling and yielding of side shell, hopper and top wing tanks are specified.

It is shown that: (i) The side shell plating between the hopper and top wing tanks may be subjected to high shear stresses since it carries about 35% of the total shearing force. (ii) The hopper and top wing tanks may be subjected to unfavourable stress conditions.

INTRODUCTION

The increase in ship size has brought to light several structural and operational problems[1]. Some of these problems were considered hitherto trivial, when ship sizes were not far from conventional, but are becoming more and more significant with the increase of ship size. One of the main structural problems associated with large ships is the unproportionate increase of shearing forces. Bulk carriers, in particular, are prone to be subjected to high shearing forces, particularly when the cargo is carried in alternate holds. Ref.[2] examines the static and dynamic components of the shearing forces in bulk carriers, together with an approximate method for calculating the shear stress distribution over a simplified section of a bulk carrier, the emphasis being placed on the stress in the side shell plating.

In this paper, a method is given for calculating the shear stress distribution over a typical section of a bulk carrier. The method is programmed for the Alexandria University IBM 1620 computer. The ship section is considered with minimum idealization to the double bottom structure so as to obtain realistic values of the calculated shear stresses. The double bottom is idealized by a 5-cell box structure. Any other double bottom arrangement could be easily converted into a 5-cell configuration.

Because of the limited capacity of the IBM 1620 computer, the program is divided into two parts. The first part computes the geometrical properties of the ship section in addition to all the data required for shear flow calculation. The second part is based on the results of the first part and computes the shear flow and shear stress distribution over the ship section. The shear carrying capacity of the side shell is also computed in the second part of the program. The computer program is used to study the shear stress distribution and shear carrying capacity of several ship section configurations and sizes. The results of this study are analysed and a summary is given in the paper together with a numerical example.

A simplified expression is developed to give an

approximate value for the maximum shear stress in the side shell plating which is the critical region insofar as shear stress is concerned.

The necessary conditions giving adequate strength against shear buckling and yielding, for side shell, hopper and top wing tanks, are examined and specified.

1. SHEAR FLOW DISTRIBUTION

(a) Structural idealization

A typical section of a bulk carrier is shown in Fig. 1. The idealized section is shown in Fig. 2. It is clear that errors of structural idealization should be negligible. The numbering scheme used in the computer program to simplify the computations, is shown in Fig. 2.

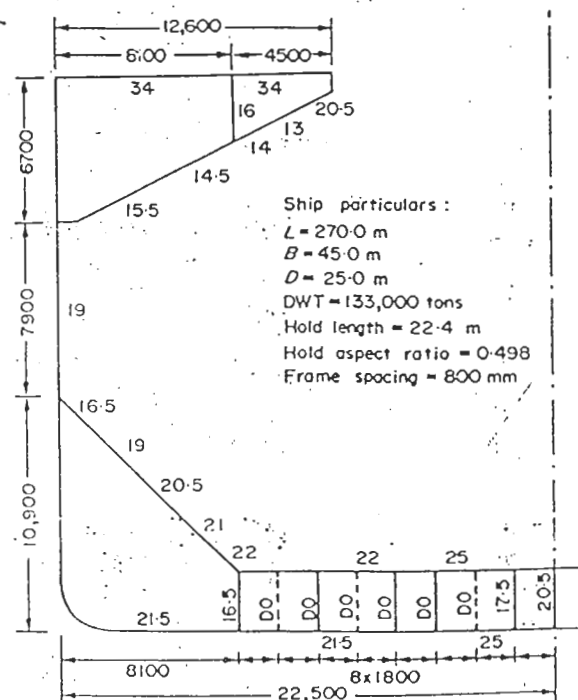


Fig. 1. A section of a bulk carrier.

*Associate Professor.

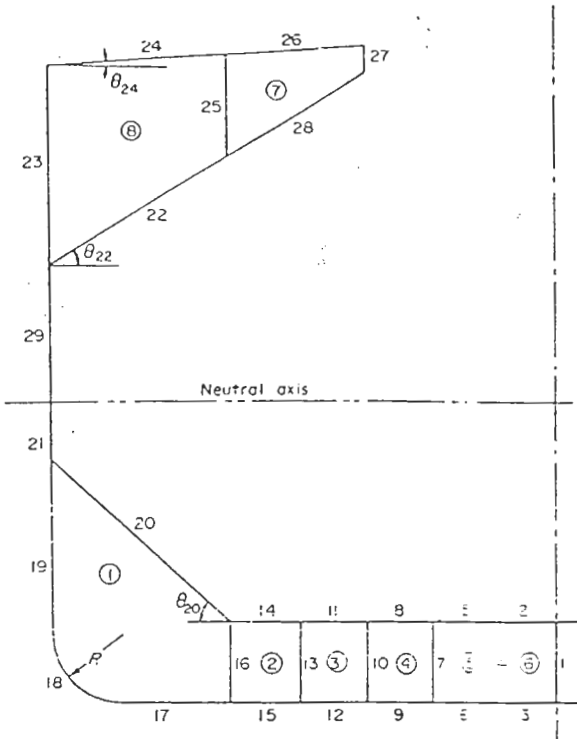


Fig. 2. Numbring scheme

$$I = \sum_{i=1}^n \left\{ a_i \cdot Z_i^2 + \frac{a_i}{12} (L_i \cdot \sin \cdot \theta_i)^2 \right\} \quad (2)$$

where, a_i = sectional area of member i , $a_i = T_i \cdot L_i$. L_i and T_i are the length and thickness of member i .
 θ_i = inclination of member i
 Z_i = distance of centroid of a_i from neutral axis.

To simplify the computational procedure, the assumed shear flow distribution is calculated at both ends of each member. Thus, the assumed shear flow distribution over the ship section is given by:

$$(AQ) = \begin{pmatrix} AQ_{11} & AQ_{12} \\ AQ_{21} & AQ_{22} \\ \dots & \dots \\ AQ_{r1} & AQ_{r2} \\ \dots & \dots \\ AQ_{n1} & AQ_{n2} \end{pmatrix} \quad (3)$$

where n = number of members.

(ii) *Correcting shear flow.* The set of correcting shear flows are obtained separately for the lower and upper parts of the ship section. The upper part is assumed to be composed of two closed cells, which is the normal practice for large ships. Therefore, two correcting shear flows are required. For the lower part, it is assumed that the double bottom is composed of five closed cells, excluding the hopper tank. Any other structural configuration of the double bottom could be easily reduced to the idealized 5-cell structure.

The correcting set of shear flows are given by:

$$\{QC\} = [D^{-1}] \cdot \{P\} \quad (4)$$

where $\{QC\} = \{QC_1, QC_2, \dots, QC_N\}^T$
 N = number of cells

$$[D] = \begin{bmatrix} D_{11} & D_{12} & & & & & & 0 \\ D_{21} & D_{22} & D_{23} & & & & & \\ & D_{32} & D_{33} & D_{34} & & & & \\ & & D_{43} & D_{44} & D_{45} & & & \\ & & & D_{54} & D_{55} & D_{56} & & \\ 0 & & & & D_{65} & D_{66} & & \end{bmatrix} \quad (5)$$

$$D_{ij} = -\frac{L_{ij}}{T_{ij}} \quad (6)$$

i and j are the numbers of any two adjacent cells.

$$D_{ii} = \sum_{r=1}^m \frac{L_r}{T_r}, \text{ for cell } j \quad (7)$$

i to m = total number of members in cell j .

$$\{P\} = \{P_{11}, P_{22}, \dots, P_{NN}\}^T \quad (8)$$

and

$$P_{jj} = \oint_j AQ \cdot \frac{dS}{T} = \sum_{r=1}^m QM_r \cdot \frac{L_r}{T_r}, \text{ for cell } j \quad (9)$$

QM_r = mean shear flow in member r of cell j .
 Solving equation (4), a set of correcting shear flows are obtained.

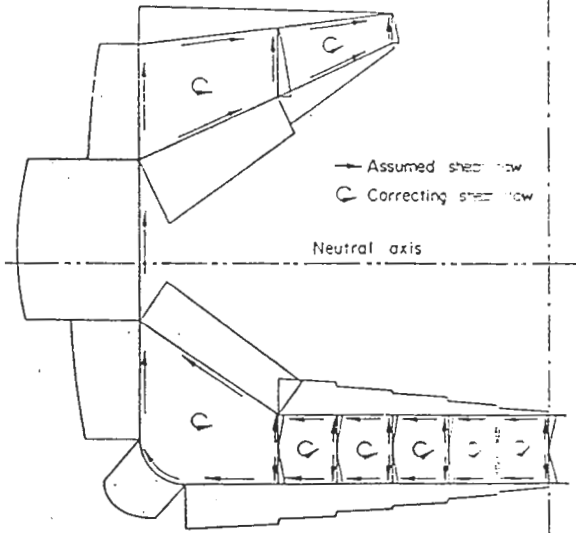


Fig. 3. Assumed shear flow distribution.

(b) *Method of analysis*

The method of analysis is given before in Ref. [2]. However, a brief summary of the method using matrix notation, is given here:

(i) *Assumed shear flow.* The assumed shear flow distribution is shown in Fig. 3. To make possible this distribution, the shear flow is assumed zero at certain points in the idealized ship section. The assumed shear flow, AQ , at any point, is given by:

$$AQ_{ji} = \frac{F}{I} \cdot A_i \cdot Y_i \quad (1)$$

where AQ_{ji} = assumed shear flow at point i of member j .

F = shearing force.

A_i = sectional area outside point i .

Y_i = distance of centroid of A_i from neutral axis of section.

I = second moment of area of ship section, and is given by:

(ii) *correct shear flow*. The correct shear flow distribution over a typical section of a bulk carrier is obtained by superimposing the set of correcting shear flows to the assumed shear flow distribution. Thus, the correct shear flow for any member in a closed cell is given by:

$$Q = AQ \pm QC \quad (10)$$

where Q = correct shear flow.

Hence, for cell r , the correct shear flow in member j , at point i , is given by:

$$(Q_{ji})_r = (AQ_{ji})_r \pm (QC)_r \quad (11)$$

For members that are not part of any closed cell, such as the side shell between the hopper and top wing tanks, the correct shear flow is the assumed one.

(c) *Computer program*

The above method of calculation is programmed in FORTRAN II for the Alexandria University IBM 1620 computer. A subroutine is especially developed for solving the six simultaneous equations of the correcting shear flows in the double bottom structure.

Because of the limited capacity of the computer, the program is divided into two parts as follows:

Program I

(i) Data—main ship particulars. —geometry and scantlings of ship section.

(ii) Results—position of neutral axis. —second moment of area of ship section about neutral axis. —data relevant to shear flow calculation.

Program II

(i) Data—shearing force. —results of program I.

(ii) Results—assumed shear flow. —correct shear flow. —shear stress distribution. —shear carrying capacity.

The shear carrying capacity of any vertical, or inclined, member is calculated as follows:

$$(SF)_j = \int_j Q_j \cdot dS \quad (a)$$

where dS = elementary length of member j .

Q_j = shear flow distribution over member j .

$(SF)_j$ = shear force carried by member j .

The integration of equation (a), over the length of member j , is carried out numerically using Simpson's first rule. The required intermediate values of shear flow are generated in the program.

It should be mentioned here that the given method is very general and could be efficiently used for the analysis of numerous types of multi-cell box-girders subjected to shear loading. Although the emphasis has been placed on bulk carriers, the computer program could also be used for the analysis of other types of ships such as OBO carriers (Oil/Bulk/Ore), container ships, floating docks, etc.

A simplified block diagram of the computer program is shown in Fig. 4.

2. NUMERICAL EXAMPLE

The computer program is used for calculating the shear flow distribution for several ship section configuration and sizes. Figures 5 and 6 give the assumed and correct

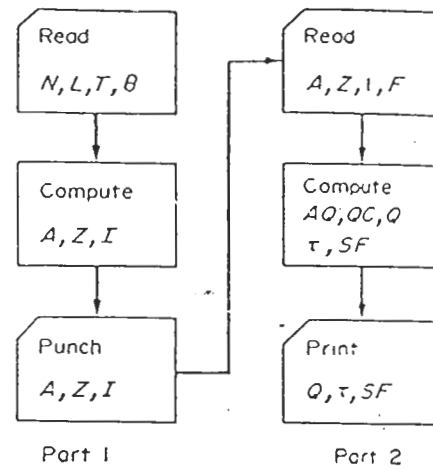


Fig. 4. Block diagram of computer program.

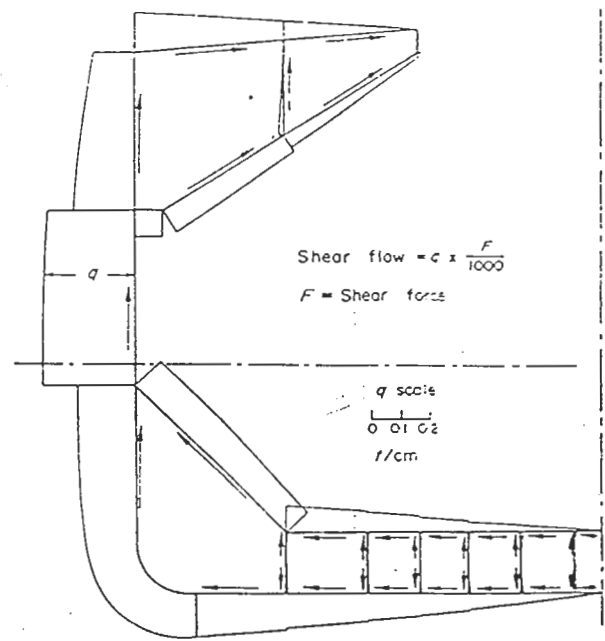


Fig. 5. Assumed shear flow distribution

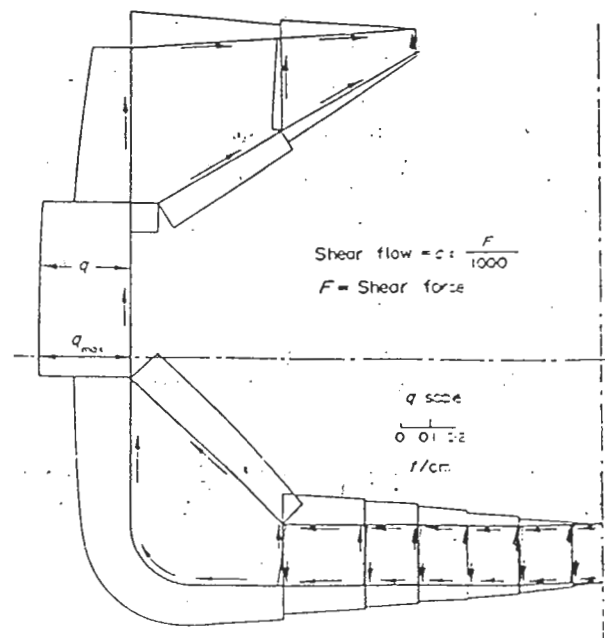


Fig. 6. Correct shear flow distribution.

shear flow distributions, respectively, for one ship section of a bulk carrier having the following particulars:

L.B.F.	=	224	m
D_m	=	18.6	m
B_m	=	31.8	m
hold length	=	21.6	m
hold aspect ratio	=	0.679	
floor spacing	=	1800	mm
frame spacing	=	900	mm
Dwt	=	66100	tons

3. ANALYSIS OF RESULTS

(a) Shear flow distribution

The shear stress distributions over the various ship sections investigated, indicate that the maximum shear stress occurs in the side shell plating between the hopper and top wing tanks. These tanks may also be subjected to high shear stresses. Therefore, unfavourable stress conditions may be developed in these tanks because of the additional high stresses induced by local loading and hull girder bending.

Consequently, the scantlings of side shell plating, hopper and top wing tanks should be adequate enough to sustain yielding and shear buckling.

(b) Shear carrying capacity of section

The contribution of the longitudinal girders, in the double bottom structure, to the shear carrying capacity of ship section is found to be insignificant. This is because of the low shear stress values induced in these girders.

The shear carrying capacity of side shell, hopper and top wing tanks are as follows:

item	%ge of total shearing force
side shell between hopper and top wing tanks	35-38%
side shell between bilge and sheer strikes	74-79%
hopper and top wing tanks	62-65%

The range of variation depends on the geometry and scantlings of the ship section.

4. DESIGN CRITERIA

In order to ensure adequate strength against yielding and shear buckling, the minimum plating thicknesses of side shell, hopper and top wing tanks should be determined from the following conditions:

(a) Side shell

The side shell plating between the hopper and top wing tanks is the critical region insofar as shear stress is concerned. The critical buckling stress of a panel subjected to pure shear loading (see Fig. 7), is given by (3):

$$\tau_{cr} = \frac{E \cdot \pi^2}{12(1-\nu^2)} \left(\frac{T_s}{S} \right)^2 K \quad (12)$$

where S = frame spacing

E = modulus of elasticity.

ν = Poisson's ratio.

T_s = thickness of side shell plating.

The constant K depends on the end conditions of the panel:

(i) For simply supported ends

$$K_s = 5.34 + \frac{4}{\alpha^2}$$

(ii) for fixed ends

$$K_f = 8.98 + \frac{5.6}{\alpha^2}$$

where: $\alpha = b/S$, b being the length of panel.

In order to ensure adequate strength against shear buckling, the minimum thickness of side shell plating could be determined from the condition that the maximum expected shear stress should not exceed the critical value, i.e.

$$\tau_{max} \leq \tau_{cr}$$

Therefore, assuming that:

$$\alpha \geq 6.0, \quad \tau_{cr} = \tau_y = 1.395, \quad K \doteq K_s,$$

and allowing a 50% reduction in τ_{cr} due to tolerances on dimensions and scantlings, residual stresses, permanent out of plane deflections, ... etc., the minimum ratio of T_s/S , given by equation (12), will be:

$$T_s/S \approx \frac{1}{60} \quad (13)$$

On the other hand, in order to ensure adequate strength against yielding, the expected maximum shear stress should not exceed the yield stress of the material, i.e.

$$\tau_{max} \leq \tau_y$$

where τ_y = yield stress in shear $\approx 1.395 t/cm^2$.

A survey of some published data on bulk carriers indicates that:

(i) The critical buckling stress in shear is always higher than yielding shear stress;

(ii) The minimum ratio between the thickness of side shell plating and frame spacing is given by:

$$T_s/S \geq 1/55. \quad (14)$$

The difference between the minimum thickness given by (13) and (14) may be attributed to corrosion allowance and local loading.

(b) Hopper and top wing tanks

The scantlings of these tanks should also have adequate strength against yielding and buckling.

Since the hopper and top wing tanks are subjected to relatively high normal stresses, induced by the bending of hull girder, in addition to the high shear stresses, the minimum thicknesses of plating should be determined from the following two conditions:

(i) Adequate strength against buckling. The condition of critical buckling of a panel of plating under combined shear and normal compressive stresses is given by [4]:

$$\left(\frac{\tau_{max}}{\tau_0} \right)^2 + \frac{\sigma}{\sigma_c} \leq 1.0 \quad (15)$$

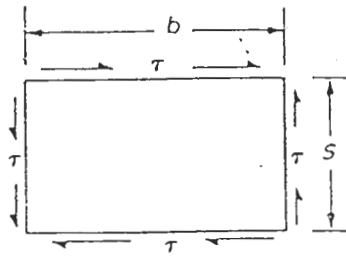


Fig. 7. Shear stress over a panel.

where $\tau_0 = \tau_{cr}$ for a panel under pure shear loading,
 σ = normal compressive stress,
 σ_c = critical compressive stress,

(ii) *Adequate strength against yielding.* This could be achieved using an equivalent stress criterion, such as that given by von Mises[5]:

$$\sigma_e \leq \sigma_y$$

where $\sigma_e = \sqrt{(\sigma_x^2 + \sigma_y^2 - \sigma_x \cdot \sigma_y + 3\tau_{xy}^2)}$
 σ_x = longitudinal stress in the X direction
 σ_y = transverse stress in the Y direction.

Therefore, the limiting shear stress could be determined in terms of the longitudinal and transverse stresses as follows:

$$\tau_{max} \leq \frac{1}{\sqrt{3}} \cdot \sqrt{(\sigma_y^2 - \sigma_x^2 - \sigma_y^2 + \sigma_x \cdot \sigma_y)}. \quad (16)$$

For the special case when:

$$\sigma_y = 0$$

and

$$\sigma_x = 1.5 \text{ t/cm}^2$$

the limiting shear stress in the hopper or top wing tanks is given by:

$$\tau_{Max} \leq 1.08 \text{ t/cm}^2.$$

5. APPROXIMATE FIRMULA

Since the computational procedure is rather lengthy and requires a digital computer, a simplified expression is developed to give an approximate value for the maximum shear stress in the side shell plating of a bulk carrier.

The maximum shear stress in the side shell plating, at the neutral axis, is given by:

$$\tau_{max} = \frac{F}{I \cdot T_i} \cdot \delta \quad (17)$$

where $\delta = \sum_0^{y_B} A \cdot \bar{Y}$

y_B = distance of neutral axis from base line.

From the various ship sections investigated, it is found

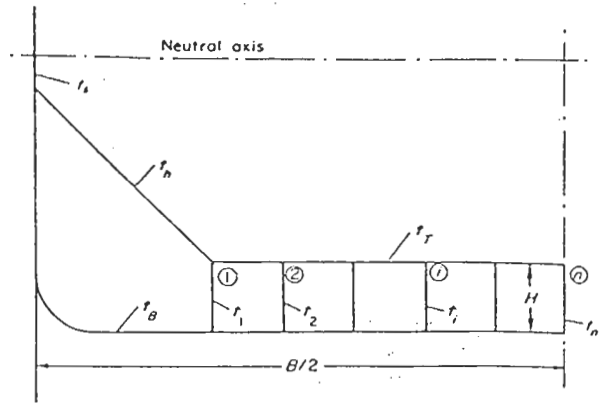


Fig. 8. Idealization of the bottom part.

that:

$$Y_B \approx 0.43 D$$

where D = ship depth.

Hence,

$$\begin{aligned} \delta = & D \cdot T_B (0.171 B + 0.092 D) \\ & + B \cdot (0.0196 B \cdot T_h + 0.1 D \cdot T_T) \\ & + 0.3 D \cdot H \cdot \sum_{i=1}^n T_i \end{aligned} \quad (18)$$

where, B = ship breadth

n = number of longitudinal girders in the double bottom.

The other terms are defined in Fig. 8.

Expression (17) is used to calculate the maximum shear stress for the various ship section configurations and the results are compared with computer results. The difference, in general, does not exceed 1%.

6. CONCLUDING REMARKS

From the various calculations and analysis of results, the following main conclusions are drawn up:

(a) The structural configuration of a bulk carrier is prone to bring high shear stresses in the side shell plating, hopper tanks and in the top wing tanks. The critical zone, insofar as shear stress is concerned, is the portion of the side shell between the hopper and top wing tanks.

(b) The side shell between the hopper and top wing tanks carry about 35% of the total shearing force.

(c) The double bottom does not contribute significantly to the shear carrying capacity of the ship section.

(d) In order to have adequate strength against shear buckling, the frame spacing should not exceed 60 times the thickness of the side shell plating.

REFERENCES

1. J. F. C. Conn, Mammoth ships, *IESS*, (1970).
2. M. A. Shama, Shear stresses in bulk carriers due to shear loading, *J. Ship Res. SNAME*, (1975).
3. S. P. Timoshenko, *Theory of Elastic Stability*, McGraw-Hill, New York (1961).
4. H. Becker and G. Gerard, Handbook of structural stability, Parts 1-5, NACA TECH. Notes, No. 3781-3785, (1957).
5. S. P. Timoshenko, *Strength of Materials*, Part II, D. Van Nostrand, New Jersey, (1956).