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.

Journal of Ship Research, Vol. 19, No. 3, Sept. 1975, pp. 155–163

Shear Stresses in Bulk Carriers Due to Shear Loading

M. A. Shama[1]

A brief note is given on various components of the longitudinal vertical shearing force. The stillwater component is examined with particular emphasis on the effect of local cargo loading and the mechanism of shear load transmission. The main factors affecting the wave-induced and dynamic components are indicated and an approximate method is given for estimating the impulsive dynamic component. A method is then given for calculating the shear stress distribution over a typical section of a bulk carrier. The ship section is idealized by a simplified structural model comprising closed and open cells. The structural model retains all the geometrical properties of the original section. Two numerical examples are considered to examine the effect of ship section parameters on shear stress distribution. It is shown that: (i) High shear stresses may be developed in the side shell plating. (ii) The variation of ship section parameters has a negligible effect on the maximum shear stress and may have a significant local effect. (iii) The shear carrying capacity of a given ship section could be easily estimated. Alternatively, for a given shearing force, a "shear coefficient," representing shear capability, could be estimated.

Introduction

BULK CARRIERS have lately experienced a continuous increase in size [1],[2] and ships up to 260,000 dwt have been designed and built. The historical background of the structural design of bulk carriers is given in references [2,3]. Also, the general structural and operational problems associated with mammoth ships have been discussed by Conn in [4].

Bulk carriers, however, experience unique problems which result mainly from the particular structural configuration and loading of these ships (alternate hold loading arrangement). Reference [5] discusses part of these structural problems.

This paper deals with the problem of shear stress distribution in a typical ship section of a bulk carrier. The ship section is idealized by a simplified configuration so as to reduce the laborious calculations associated with shear flow distribution. The idealized structure affects neither the magnitude nor the distribution of shear flow around the top wing tanks, hopper tanks, and side shell. Using this simplified structural model, the effect of variation of the relevant geometrical parameters on the shear flow distribution is investigated. The importance of calculating shear stresses in the hull girder is emphasized, particularly for determining the maximum allowable shearing force for a given ship section.

The problem of shear stress distribution in mammoth tankers has been investigated by Shama [6], who emphasized the contribution of longitudinal bulkheads to the longitudinal vertical shearing force. In bulk carriers, the longitudinal vertical shearing force is carried by the side shell plating, top wing tanks, and hopper tanks. The side shell, therefore, may carry a high proportion of the longitudinal vertical shearing force. Consequently, shear stresses in the side shell plating may reach unfavorable values, especially when the alternate hold loading system is used. Shear buckling, or high values of combined stresses, may also occur in some panels in the side shell plating. Adequate measures should be taken, therefore, to prevent instability and high stresses.

The longitudinal vertical shearing force, at any section, along a ship steaming in waves is the vectorial sum of the stillwater component, wave-induced component, and the dynamic component. A brief note on each of these components is also given in the paper. An approximate method is given in the Appendix for estimating the impulsive dynamic shearing force component. The latter is deduced from the dynamic bending moment distribution along the ship length.

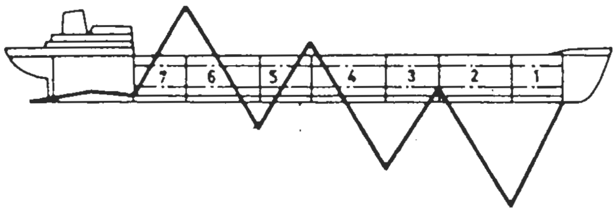

Fig. 1 Shear force distribution

Because of the high rigidity of the double bottom and transverse bulkheads, the shearing force carried by the side shell plating should be corrected for local cargo loading, particularly when the cargo loading is carried in alternate holds. This correction is discussed in the paper together with the factors affecting its magnitude.

Longitudinal vertical shearing force

The longitudinal vertical shearing force, at any section along a ship length, could be resolved into a stillwater component, a wave-induced component, and a dynamic component. This section treats briefly each of these components.

Stillwater component (*FS*)

The conventional calculation of longitudinal vertical shearing force and bending moment, along the ship length, gives results pertinent only to the general behavior of the idealized ship girder, without due regard to the effect of local cargo loading; see Fig. 1. For ships having a single bottom, such as oil tankers, the effect of local loading may not be very significant, whereas for ships having a double bottom, such as bulk carriers and OBO ships (ore bulk oil), the effect of local loading may become very significant and may affect the longitudinal strength of the whole ship, especially when the alternate hold loading arrangement is used and particularly when the transverse bulkheads are of double skin [7].

[1]Associate professor, Naval Architecture Department, Faculty of Engineering, Alexandria University, Alexandria, Egypt.

[2] Numbers in brackets designate References at end of paper.

Manuscript received at SNAME headquarters April 9, 1974.

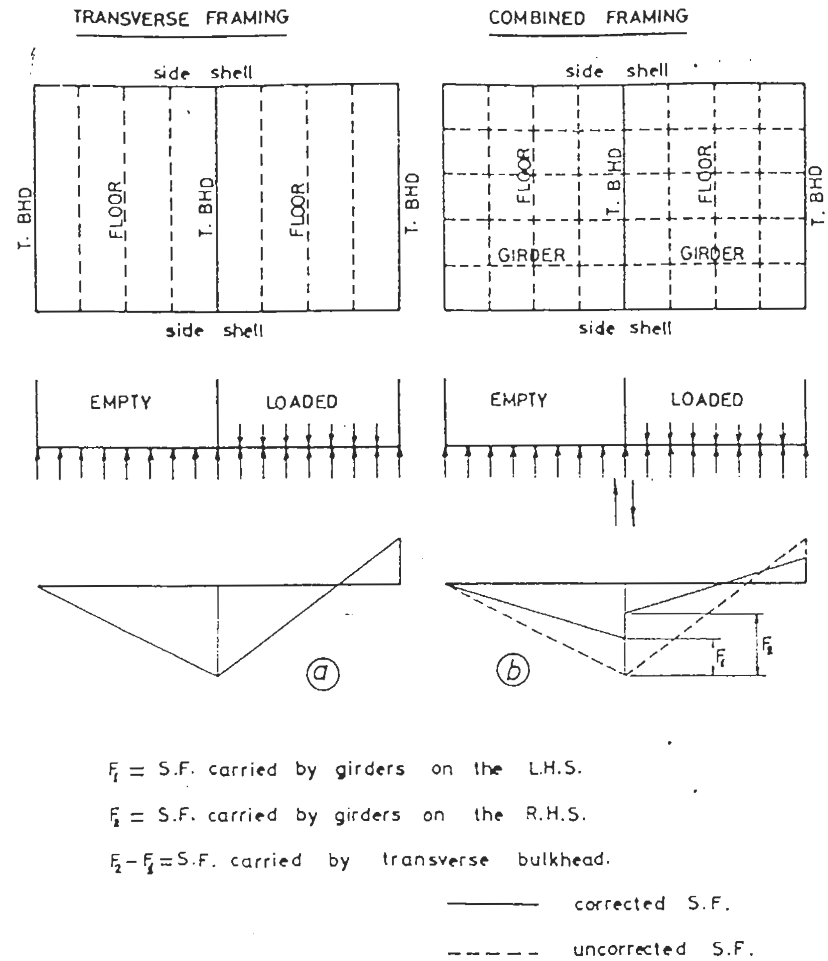

Fig. 2 Shear force distribution

The double bottom structure of a general bulk carrier is normally stiffened by a combined system of framing comprising longitudinal girders and solid floors. The stiffness and rigidity of these members may be increased when the ship is designed for carrying ore [8]. The attachments of the double bottom to both transverse bulkheads and side shell are normally designed in the form of a closed box-girder (a stool for the transverse bulkheads and a hopper tank for the side shell).

The net local loading on the double bottom is the difference between the cargo loading (grain, ore, . . . or ballast) acting on the tank top and the upward water pressure acting on the outer skin. This net loading could reach high values especially when the alternate loading system is used. The double bottom structure will transmit this local loading to the side shell and transverse bulkheads in the form of shear forces and bending moments. The mechanism of load transmission depends mainly upon the construction of the double bottom, that is, the stiffness and rigidity in the longitudinal and transverse directions, the construction of the attachments between the double bottom and both side shell and transverse bulkheads, the shear area of transverse bulkheads, the hold aspect ratio, and the loading condition (uniform or in alternate holds). For relatively short holds (that is, of aspect ratio = 0.5), nearly 80 percent of the net loading is carried by transverse bulkheads as shear forces [1]. These shear forces are transmitted to the side shell as concentrated shear forces presenting the local corrections to the general distribution of the shear force diagram obtained from longitudinal strength calculation. The mechanism of load transmission is best explained by Fig. 2. It is shown that, for a transversely framed double bottom, the shear loading is entirely carried by the side shell, and the effect of local loading is almost nonexistent. When longitudinal girders are used in the construction of the double bottom, the shear forces carried by the side shell will be reduced by the contribution of these girders to the local net loading. The proportion of the local shear forces carried by the transverse bulkheads could be estimated by solving the double bottom as a grillage structure [9].

Therefore, the conventional stillwater shearing force distribution, along the ship length, should be corrected for local loading. Normally, these corrections will reduce the high shearing forces associated with the alternate hold loading system.

Wave-induced component (*FW*)

This component results from the distribution of the forces of support throughout the length of a ship during her passage among waves. The variability of this component results from the variabilities of the sea state and ship parameters.

The sea state is influenced by various factors, among them the fetch, wind velocity, direction and duration. It is possible, however, to represent any particular sea state by its energy spectrum [10,11]. The mathematical formulations of the wave spectra are normally given in terms of some observable properties of the sea, such as the average wave height and average wave period.

The variability of *FW* results also from the variabilities of the main ship parameters affecting her responses. These parameters are

- main ship dimensions (particularly ship length)

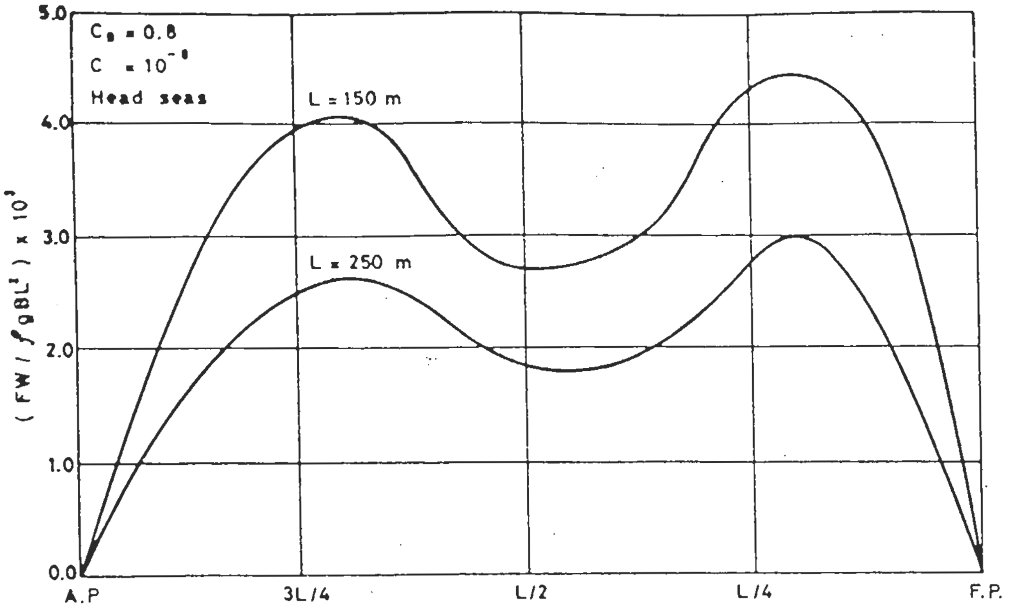

Fig. 3 Distribution of largest expected vertical shearing force over length of ship

- form coefficients
- ship speed
- angle between ship course and direction of wave advance
- weight distribution along ship length.

For a given ship and sea state, FW could be determined either experimentally [12,14] or by calculation [15,16]. Using wave statistics, it is possible to obtain the long-term distribution function of FW. Figure 3 is obtained from reference [17] and shows the distribution of the largest expected vertical wave-induced shearing force for a probability of exceedence of 10^{-8}. This technique has been used successfully for the prediction of bending moments—among several other ship responses—experienced by a ship over her service life [18]. The problem, however, is outside the scope of this paper and would require a separate investigation.

Dynamic Component, (FD)

This component could be either impulsive, resulting mainly from slamming [19–21], or nonimpulsive, resulting from the vibration of the hull girder [22]. Its magnitude is influenced by ship's speed, heading, length, draft, trim, sea condition, and weight distribution. These factors affect the amplitudes of pitching and heaving, which affect the magnitude of the dynamic component.

The impulsive component increases directly as the area under the bow that appears above the water surface increases. Therefore, for ships having a full form in the bow and running at a shallow draft in a severe sea condition, the impulsive component may be rather significant. This is the case of a bulk carrier, or an oil tanker, in the ballast condition.

The impulsive and nonimpulsive shearing force components could be studied using methods similar to those given in references [21,22]. This problem, however, is beyond the scope of this paper. Nevertheless, an approximate estimate of the impulsive component could be obtained from the dynamic bending moment component, as given in Appendix 1. It is shown that the impulsive shearing force component could be estimated by:

$$(F_{\mathrm{Dyn}})_{\max} \cong 0.3\ B^2 \cdot \frac{h}{\lambda}$$

where

B = ship breadth
h, λ = resonant wave height and length
F_{Dyn} = dynamic shearing force

Total shearing force (F)

The total vertical shearing force, at any section along ship length, is the vectorial sum of the stillwater, wave-induced, and dynamic components. Using this total vertical shearing force, the shear stress distribution around a typical ship section of a bulk carrier could be calculated.

Shear flow distribution

Structure idealization

Since the main objective of this study is a qualitative shear flow distribution around a typical section of a bulk carrier (see Fig. 4), then the local shear flow distribution within the double bottom structure may be deleted for the sake of simplicity. This could be achieved by replacing the complex structure of the double bottom by an equivalent plate structure; see Fig. 5. The idealized structure should retain the same geometrical properties of the whole ship section; namely, total sectional area, shear area, position of neutral axis, and second moment of area.

The longitudinal stiffening members are not included in the calculation of shear flow distribution, as this would have a negligible effect on the results of the parametric study [23]. These members, however, should be taken into account in any quantitative study. This could be simplified by using equivalent thicknesses instead of actual thicknesses, as follows:

$$(t_e)_j = t_j + \sum_{i=1}^{n} a_{ij}/l_j \qquad (1)$$

where

$(t_e)_j$ = equivalent plating thickness of element j
t_j = original plating thickness of element j
a_{ij} = area of stiffening member i within element j
l_j = length of element j

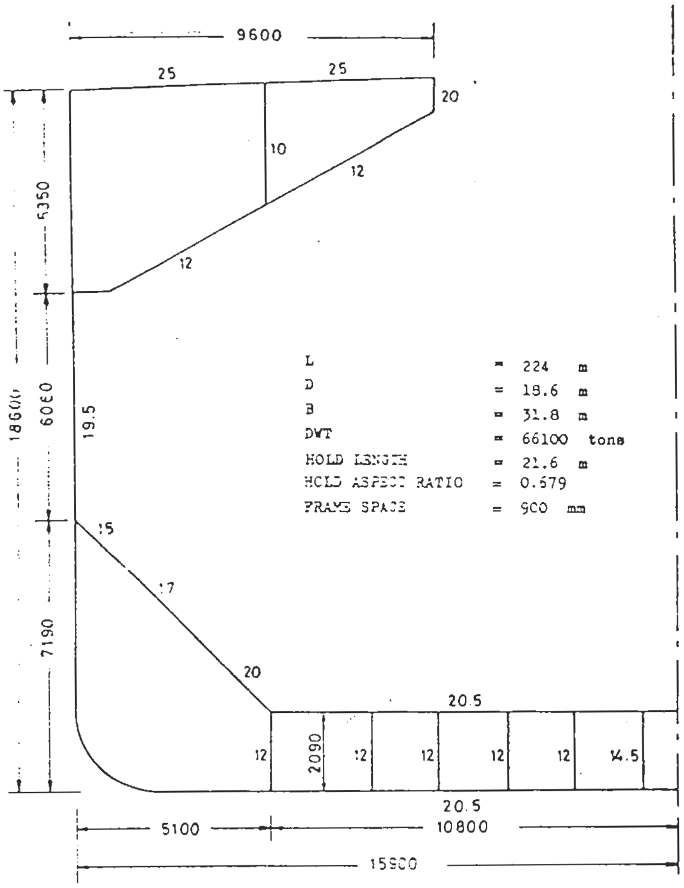

Fig. 4 Section of bulk carrier

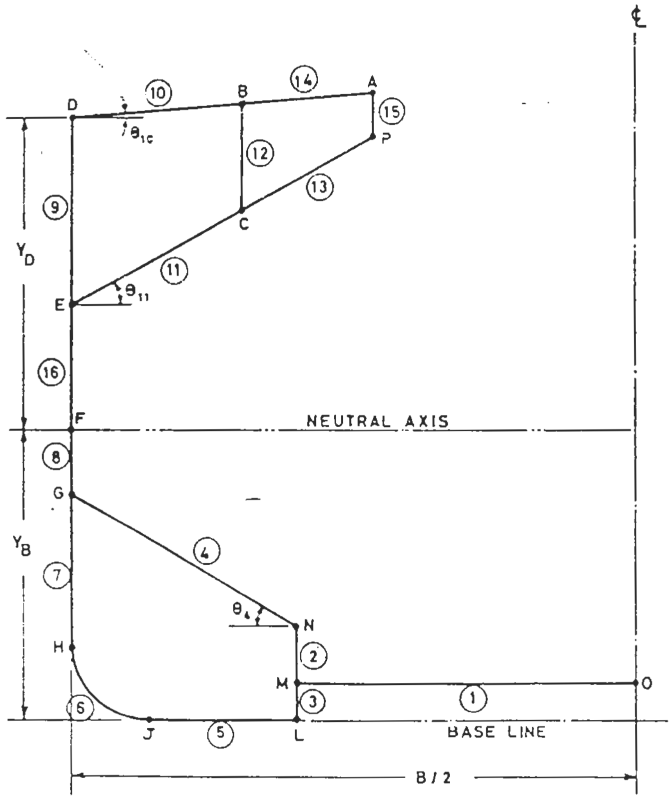

Fig. 5 Idealized structure

The thickness of the idealized single bottom is given by:

$$t_{IB} = t_T + t_B + \frac{1}{l_B} \sum_{j=1}^{r} a_j \tag{2}$$

The distance of the idealized single bottom from the baseline, that is, h_1, is given by:

$$h_1 = \frac{t_T}{t_{IB}} \cdot H \tag{3}$$

where

t_T, t_B = mean thickness of tank top and bottom, respectively
t_{IB} = total thickness of idealized single bottom
H = mean depth of double bottom
$\sum_{j=1}^{r} a_j$ = total area of longitudinal girders within length l_B of double bottom
l_B = length of idealized single bottom

The calculation of shear flow distribution around an actual ship section, including the multicell box-structure of the double bottom, will be studied in a future publication.

Method of calculation

The calculation of shear flow distribution around a ship section of a bulk carrier is based on the method given in reference [23].

In the absence of any torsional loading, the ship section, or any part of it, does not exhibit any torsional deformations. Therefore, the shear flow distribution could be calculated by assuming a certain shear flow distribution, and the resulting torsional deformations are corrected by a set of correcting shear flows. The superposition of the assumed and correcting shear flows gives the correct shear flow distribution. This method is summarized as follows:

(i) *Assumed shear flow.* The shear flow in members *OM*, *MN*, *AP*, *AB*, and *BC* is assumed to be nil, that is

$$q_{ij} = 0, \quad ij = OM,\ MN,\ AP,\ AB, \text{ and } BC$$

The assumed shear flow distribution around a ship section of a bulk carrier is as shown in Fig. 6.

The contribution of each element of the section to the shear flow distribution is calculated by the general shear flow expression

$$q_r = \frac{F}{I} \cdot A_r \cdot \bar{Y}_r, \quad r = 1, 2 \ldots n \tag{4}$$

where

q_r = contribution of member r to shear flow distribution
A_r = sectional area of member r
$\bar{Y}_r$ = distance of centroid of A_r from neutral axis of ship section
I = total second moment of area of ship section
F = total vertical shearing force

If member m is bounded by nodes i, j, then the shear flow in member m is given by:

$$(q_{ij})_m = \frac{F}{I} \cdot \sum_{r=1}^{m-1} A_r \cdot \bar{Y}_r \tag{5}$$

$$(q_{ji})_m = \frac{F}{I} \cdot \sum_{r=1}^{m} A_r \cdot \bar{Y}_r \tag{6}$$

$$= (q_{ij})_m + q_m$$

where

$(q_{ij})_m$ = shear flow in member m from i to j
$(q_{ji})_m$ = shear flow in member m from j to i

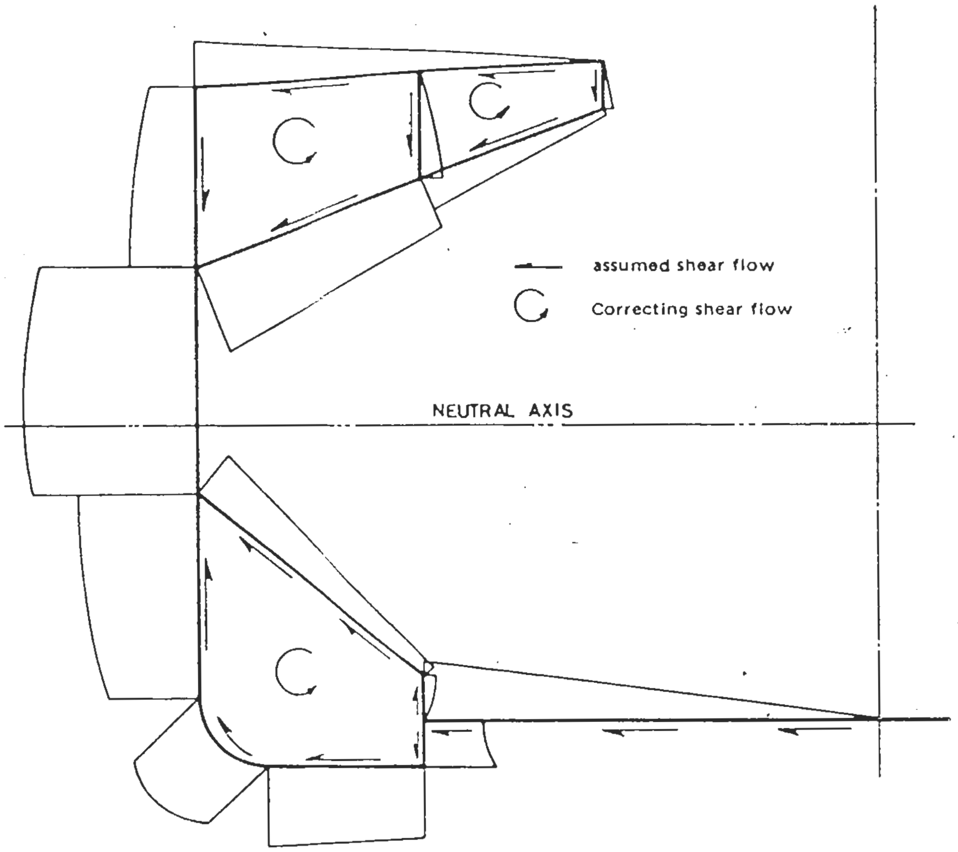

Fig. 6 Assumed shear flow distribution

Nomenclature

a_i = sectional area of stiffener i

A_r = sectional area of member r

A_O, A_I = enclosed area of outer and inner cells of top wing tank

A_H = enclosed area of hopper tank

B = ship breadth

C_B = block coefficient

a, b, c = coefficients

F = total shearing force ($F = FS + FW + FD$)

FD = dynamic shearing force component

FS = stillwater shearing force component

FW = wave-induced shearing force component

F_{Dyn} = dynamic shearing force

H = mean depth of double bottom

h_1 = distance of idealized single bottom from baseline

h = wave height

I = total second moment of area of whole ship section about its neutral axis

K = coefficient

L = ship length

l_j = length of member j

M_{Dyn} = dynamic bending moment

q_r = shear flow of member r

q_{ij} = shear flow from i to j

$(q_{ij})_m$ = shear flow in member m from i to j

$(q_C)_H$ = correcting shear flow in hopper tank

$(q_C)_I, (q_C)_O$ = correcting shear flows for inner and outer cells of top wing tank

T = trim of ship

t = actual plate thickness

t_e = equivalent plate thickness

t_B = total thickness of idealized single bottom

Y_B = distance of neutral axis of whole section from baseline

Y_r = distance of centroid of A_r from neutral axis of ship section

ζ = normalized dynamic bending moment

$\eta = F_{\mathrm{Dyn}} \cdot \frac{L}{2} / M_{\mathrm{Dyn}}$

λ = wave length

θ_{HA} = angle of twist of hopper tank resulting from assumed shear flow

θ_{HC} = angle of twist of hopper tank resulting from correcting shear flow

$(\theta_C)_I, (\theta_C)_O$ = angles of twist of inner and outer cells of top wing tank

ΔS = an elementary length of plating over its perimeter

τ_r = shear stress in member r

$\oint_H$ = integration over closed cell of hopper tank

Φ = shear coefficient

It is to be noted that the continuity equation of the shear flow at any joint must be valid; that is, the shear flow going into a joint must be equal to the shear flow coming out of the same joint.

(ii) *Correcting shear flow*. The assumed shear flow will distort the hopper tanks and the top wing tanks. Since it is assumed, however, that no torsional loading is applied, the angle of twist of the whole section, or any part of it, must vanish. This could be achieved by applying a set of correcting shear flows for each closed cell; see Fig. 6. The magnitude of each correcting shear flow is calculated by equating the angle of twist resulting from the assumed shear flow to that produced by the correcting shear flow.

For the hopper tank, the angle of twist resulting from the assumed shear flow is given by:

$$\theta_{HA} = \frac{1}{2GA_H} \cdot \oint_H q \cdot \frac{\Delta S}{t} \tag{7}$$

The angle of twist resulting from the correcting shear flow $(q_C)_H$ is given by:

$$\theta_{HC} = (q_C)_H \cdot \oint_H \frac{\Delta S}{t} \tag{8}$$

Equating (7) and (8), the correcting shear flow in the hopper tank is given by:

$$(q_C)_H = -\oint_H q \cdot \frac{\Delta S}{t} \Big/ \oint_H \frac{\Delta S}{t} \tag{9}$$

Since it is assumed that there are two cells in the top wing tank, two correcting shear flows $(q_C)_O$ and $(q_C)_I$, for the outer and inner cells, are required. The corresponding angles of twist are given by:

$$(\theta_C)_O = \frac{(q_C)_O}{2GA_O} \cdot \oint_O \frac{\Delta S}{t} - \frac{(q_C)_I}{2GA_O} \cdot \left(\frac{\Delta S}{t}\right)_O^I \tag{10}$$

$$(\theta_C)_I = \frac{(q_C)_I}{2GA_I} \cdot \oint_I \frac{\Delta S}{t} - \frac{(q_C)_I}{2GA_I} \cdot \left(\frac{\Delta S}{t}\right)_O^I \tag{11}$$

The angles of twist, for the outer and inner cells, resulting from the assumed shear flow, are given by

$$\theta_O = \frac{1}{2GA_O} \cdot \oint_O q \cdot \frac{\Delta S}{t} \tag{12}$$

$$\theta_I = \frac{1}{2GA_I} \cdot \oint_I q \cdot \frac{\Delta S}{t} \tag{13}$$

Equating equations (10), (12) and (11), (13), the correcting shear flows $(q_C)_O$ and $(q_C)_I$ can be calculated.

(iii) *Shear stress distribution*. The correct shear flow in member r within the hopper tank is therefore given by:

$$(q_r)_{ij} = (q_{ij})_r \pm (q_C)_H \tag{14}$$

and for the outer and inner cells, of the top wing tank, are given by

$$(q_r)_{ij} = (q_{ij})_r \pm (q_C)_O \tag{15}$$

$$(q_r)_{ij} = (q_{ij})_r \pm (q_C)_I \tag{16}$$

The correct shear flow for the common member between the outer and inner cells of the top wing tank is given by:

$$(q_r)_{ij} = (q_{ij})_r \pm \{(q_C)_O - (q_C)_I\} \tag{17}$$

The sign $\pm$ depends entirely on the direction of the assumed and correcting shear flows.

Having calculated the correct distribution of the shear flow around the ship section, the shear stress in any member is therefore given by:

$$(\tau_r)_{ij} = \frac{(q_r)_{ij}}{t_r} \tag{18}$$

It is to be noted that the shear stress in the side shell plating between the hopper tank and the top wing tank could be calculated directly without computing any of the correcting shear flows.

This method of calculation is programmed in Fortran II for the Alexandria University IBM 1620 computer. A summary of the program is given in Appendix 2.

(iv) *Numerical examples*. The foregoing method is used to calculate the shear flow distribution for two bulk carriers having the following particulars:

Ship particulars	Ship No. 1	Ship No. 2
LBP	[illegible]24 m	270 m
B	[illegible].8 m	45 m
D	[illegible].6 m	25.5 m
dwt	66,[illegible]00 tons	133,000 tons
Hold length	[illegible].6 m	22.4 m
Hold aspect ratio	0.[illegible]79	0.498
Frame spacing	[illegible]00 mm	800 mm

The shear flow distribution around a section of each bulk carrier, for a shear force $F =$ [illegible]00 tons, is shown in Figs. 7 and 8. It is clear that the side shell in the vicinity of the neutral axis of the section may have to sustain high shear stresses. Consequently, adequate measures should be taken to ensure that the shell plating between the hopper tank and the top wing tank is strong enough to withstand shear buckling and local loading.

Parametric study

The main parameters investigated were:

- Thicknesses of hopper plate, top wing tank, and the internal partition of the top wing tank.
- Width of hatch opening, that is, the size of the top wing tank.
- Height of the double bottom.

Results of calculations

The main results obtained from the parametric study follow.

(*a*) The magnitude and distribution of shear stresses over the ship section are slightly affected by the variation of any one parameter. The thickness variation of any member has only a significant local effect and does not have any marked effect on the maximum shear stress in the side shell plating.

(*b*) The location of the maximum shear stress depends upon the location of the neutral axis of the whole section. The parametric study indicates that the neutral axis may take any position between 0.4D and 0.46D, with a mean value of 0.43D, that is

$$Y_B \cong 0.43D \tag{19}$$

where Y_B is the distance of neutral axis from the baseline.

(*c*) High shear stresses may be developed in the side shell plating and the hopper plate. The maximum shear stress in the side shell plating occurs at the neutral axis of the whole section and is given by:

$$\tau_{\max} = [illegible] \cdot F \tag{20}$$

where

$\tau_{\max}$ = maximum shear stress in side shell plating
F = total shear force at section under consideration

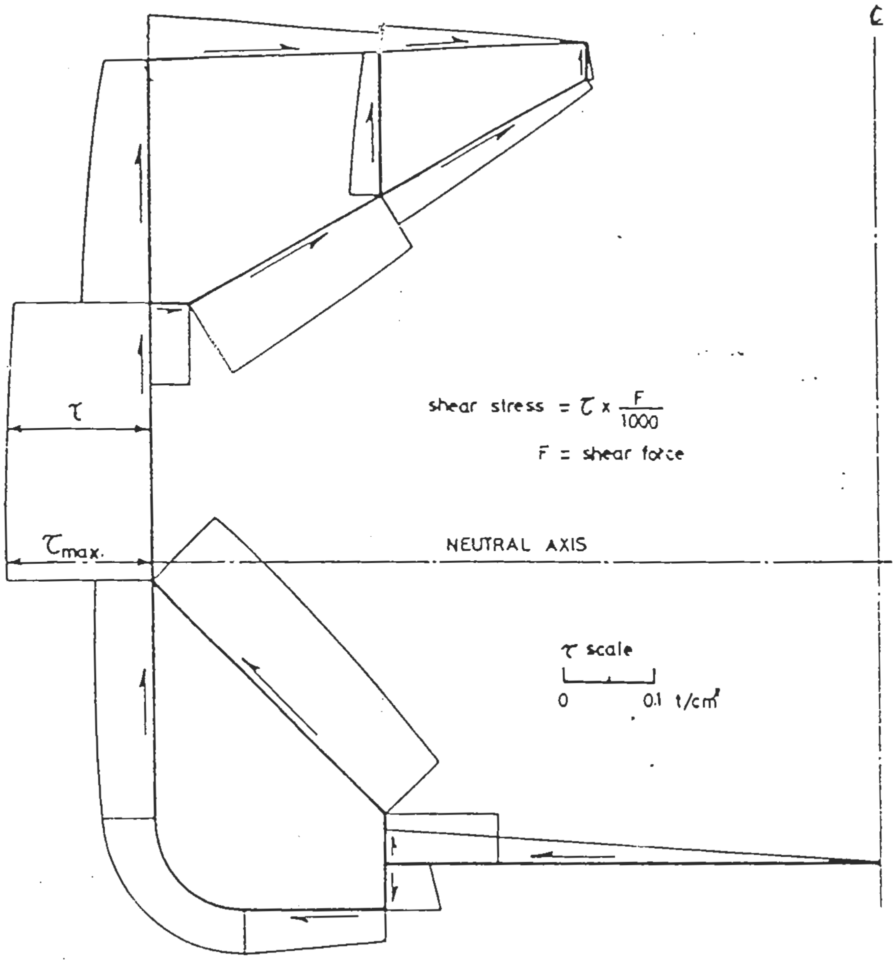

Fig. 7 Correct shear stress distribution (Ship No. 1)

Φ = shear coefficient; it depends on the geometrical configuration of the ship section

$$\Phi = \frac{\sum A \cdot \bar{Y}}{I \cdot t}$$

For the two ships considered in this investigation, the following values of Φ are obtained:

$\Phi = 0.16 \times 10^{-3}$, for Ship No. 1 (66,100 dwt)
$\Phi = 0.12 \times 10^{-3}$, for Ship No. 2 (133,000 dwt)

(*d*) The maximum allowable shear force could be calculated readily when the permissible shear stress is known. The latter could be determined using a stress criterion based either on yielding or on shear buckling [11]. Thus, assuming that τ_{max} = 1.2 t/sq cm, the maximum allowable shear force is given by

$$F_{max} \leqslant \tau_{max}/\Phi$$

that is

For Ship No. 1, $F_{max} \leq 7500$ tons
For Ship No. 2, $F_{max} \leq 10{,}000$ tons

Consequently, the maximum allowable total shearing force could be calculated for any given section of a bulk carrier. Alternatively, the required Φ could be determined, for any bulk carrier, when the required expected total shearing force is known.

These results, however, indicate clearly that the maximum allowable shearing force does not increase at the same rate as

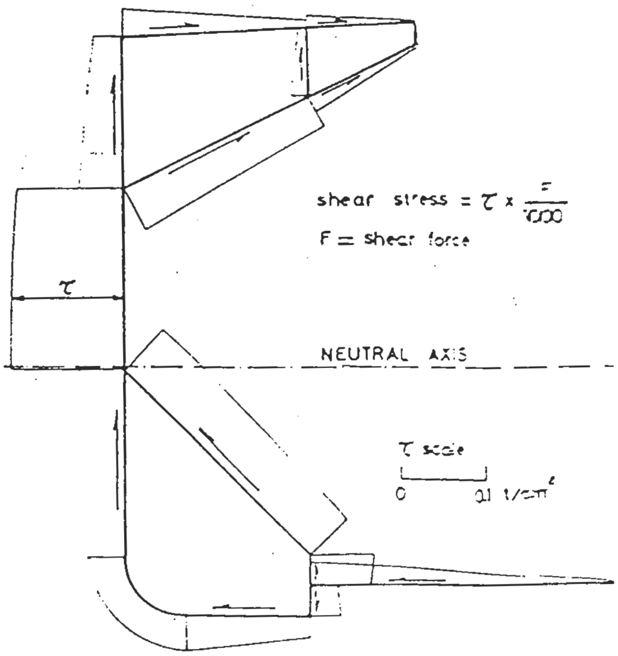

Fig. 8 Correct shear stress distribution Ship No. 2

ship size. Therefore, it is expected that larger ships may be subjected to high shear stresses.

Concluding remarks

- A rational procedure for the structural design of bulk carriers should be based not only on an accurate method of structural analysis but also on a realistic estimate of loading, taking dynamic effects into account.
- The magnitude of the stillwater shearing force component depends upon the structural configuration of the double bottom in addition to the distribution of weight and buoyancy along the ship length.
- The wave-induced shearing force component could be estimated using the same technique adopted for estimating the wave-induced bending moment.
- In the absence of a rational method for calculating the dynamic shearing force component, the proposed approximate method could be used.
- High shear stresses may be developed in the side shell plating of bulk carriers. Therefore, adequate measures should be taken against shear buckling and yielding.
- The shear carrying capacity of a given ship section could be easily estimated. Alternatively, a "shear coefficient" could be determined when the estimated maximum shearing force is known.
- The variation of ship section parameters has a negligible effect on the maximum shear stress and may have a significant local effect.

References

1 Hagiwara, Koichi et al., "Structural Strength of Large Bulk Carriers," *Technical Review*, Mitsubishi Heavy Industries Ltd., Jan. 1970.

2 Murray, J. M., "Standard of Strength in Relation to Bulk Carriers," Lloyds Register of Shipping, No. 24, London, 1963.

3 Murray, J. M., "Large Bulk Carriers," IESS, March 1965.

4 Conn, J. F. C., "Mammoth Ships," IESS, Feb. 1970.

5 Bruland, Thv., "Problems which Face the Shipbuilder and Classification Societies over the Construction of Today's Large Tankers and Bulk Carriers," Det norske veritas, Publication No. 50, Nov. 1965.

6 Shama, M. A., "On the Optimization of Shear Carrying Material of Large Tankers," JOURNAL OF SHIP RESEARCH, Vol. 15, No. 1, March 1971.

7 Kurz, H., "Spannungsmessungen an den Verbanden eines 160,000 tdw—OBO Carriers" *Schiff und Hafen*, Heft 5, 1971.

8 "Lloyds Register of Shipping Rules and Regulations for the Construction and Classification of Steel Ships," London, 1972.

9 Viner, A. C., "Structural Analysis of Bulk Carriers," R.&T.A. Report No. 5081, Lloyds Register of Shipping, London, May 1971.

10 ISSC, Delft, Holland, 1964.

11 ISSC, Oslo, Norway, 1967.

12 Moor, D. I., "Longitudinal Bending Moments in Head Seas," Royal Institution of Naval Architects, April 1967.

13 Murdey, D. C., "On the Double Peaks in Wave Bending Moment Response Curves," Royal Institution of Naval Architects, April 1969.

14 Ivarson, A., "Ore Carrier Model Tests," Project S-15, The Swedish Shipbuilding Research Foundation, Report No. 35, 1963.

15 Korvin-Kroukovsky, B. V. and Jacobs, W. R., "Pitching and Heaving Motions of a Ship in Regular Waves," *Trans. SNAME*, Vol. 65, 1957.

16 Fukuda, J. I., "Theoretical Determination of Design Wave Bending Moments," *Japan Shipbuilding and Marine Engineering*, May 1967.

17 Abrahamsen, E., "Recent Developments in the Practical Philosophy of Ship Structural Design" Det norske Veritas, Publication No. 60, July 1967.

18 Bennet, R., "Results from Full-Scale Measurements and Predictions of Wave Bending Moments Acting on Ships," Swedish Shipbuilding Research Foundation, Report No. 32, 1962.

19 Bell, O. A. and Taylor, K. V., "Wave Excited Hull Vibration Measurements on a 47000 dwt Tanker," *Shipping World and Shipbuilder*, Vol. 191, Feb. 1968.

20 Pedersen, B., "Wave Loads on the Fore-Ship of a Tanker," *European Shipbuilding*, No. 6, 1968.

21 Kumai, T. and Tasai, F., "On the Wave Exciting Force and Response of Whipping of Ships," *European Shipbuilding*, No. 4, 1970.

22 Goodman, R. A., "Wave Excited Main Hull Vibration in Large Tankers and Bulk Carriers," Royal Institution of Naval Architects, 1970.

23 Shama, M. A., "Effect of Ship Section Scantlings and Transverse Position of Longitudinal Bulkheads on Shear Carrying Capacity of Main Hull Girder," *International Shipbuilding Progress*, Vol. 16, No. 184, Dec. 1969.

24 "Standard of Strength for Sea-Going Steel Vessels," Issued by the Register of Shipping of the USSR, 1961.

Appendix 1

Dynamic shearing force

An approximate estimate of the impulsive dynamic shearing force component could be obtained from the available data on the dynamic bending moment component [24]. The distribution of the latter along the ship length is given in reference [24]; see Fig. 9. This nondimensional distribution is based on the assumption that the maximum value occurs at amidships. This is not always the case, as the position of the maximum value depends on the shape of the fore end of the ship and her length, among several other parameters.

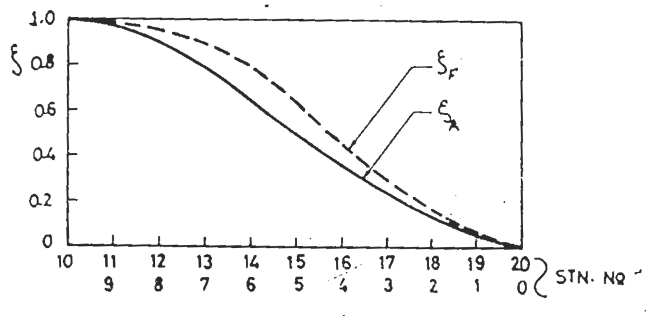

Fig. 9 Distribution of ζ along ship length

By fitting a polynomial expression to the nondimensional distribution, the impulsive shearing force component could be easily obtained as follows:

$$F = \frac{dM}{dX} = \frac{2}{L}\frac{dM}{dx} = \frac{2}{L}\cdot\frac{d\zeta}{dx}\cdot (M_{Dyn})_{max} \qquad (21)$$

This procedure is applied to the forward and aft bodies of the ship as follows:

(i) *Forebody*. The nondimensional distributions of the impulsive dynamic bending moment and shearing force are given by

$$\zeta_F = 1.0 - 0.57\,x^2 - 2.502\,x^3 + 2.414\,x^4 - 0.342\,x^5 \qquad (22)$$

$$\eta_F = -1.14\,x - 7.506\,x^2 + 9.656\,x^3 - 1.71\,x^4 \qquad (23)$$

(ii) *Aft body*. The corresponding distributions for the aft body are given by

$$\zeta_A = 1.0 - 2.66\,x^2 + 0.03\,x^3 + 3.594\,x^4 - 1.956\,x^5 \qquad (24)$$

$$\eta_A = -5.336\,x + 0.09\,x^2 + 14.376\,x^3 - 9.78\,x^4 \qquad (25)$$

where X = distance from amidships

x = normalized distance from amidships $= \dfrac{2X}{L}$

$$\xi = \frac{\text{dynamic bending moment}}{\text{dynamic bending moment amidships}}$$

$$\eta = \frac{\text{dynamic shearing force}}{\text{dynamic bending moment amidships}} \cdot \frac{L}{2}$$

Thus

$$F_{\text{Dyn}} = \eta \cdot \frac{(M_{\text{Dyn}})_{\max}}{L/2} \tag{26}$$

The maximum value of F_{Dyn} occurs at about $L/3$ forward of amidships and at about $0.23\,L$ aft of amidships, and is approximately given by

$$(F_{\text{Dyn}})_{\max} \approx \frac{3}{L}(M_{\text{Dyn}})_{\max} \tag{27}$$

The maximum dynamic bending moment is given by [24]:

$$(M_{\text{Dyn}})_{\max} = K \cdot B^2 \cdot L \cdot \frac{h}{\lambda} \tag{28}$$

where

$$K = 0.2\left[a + b(C_B - 0.6) + c \cdot \frac{T}{0.08}\right]$$

and a, b, and c are coefficients depending on the d/L ratio and Froude number. L and B are length and breadth of ship loadline respectively, while h and λ are the height and length of the resonant wave, that is, a wave having a period equal to the natural pitching period of the ship in still water.

T = trim = aft draft—forward draft
C_B = block coefficient

Therefore, for bulk carriers having zero trim and a Froude number of 0.15, the maximum impulsive bending moment is approximately given by

$$(M_{\text{Dyn}})_{\max} \cong 0.1\ B^2 \cdot L \cdot \frac{h}{\lambda} \tag{29}$$

and the maximum impulsive dynamic shearing force component is approximately given by

$$(F_{\text{Dyn}})_{\max} \cong 0.3\ B^2 \frac{h}{\lambda} \tag{30}$$

The length of the resonant wave depends on ship length, Froude number, and ship draft, among several other factors. It could, however, be approximately given by

$\lambda = 0.64\,L$ for ballast condition
$\lambda = 0.94\,L$ for loaded condition

The height of the resonant wave could be determined either from the available statistical data on sea and ocean waves [10,11] or by using one of the standard formulas adopted by classification societies.

Appendix 2

Summary of computer program

Data Required

N = number of elements
L = length of each element, m
T = thickness of each element, cm
θ = slope of each element, deg
F = shear force, tons
R = bilge radius

Calculations

Geometrical properties

A = sectional area of each element, sq cm
Y = distance of centroid of each element from baseline,
$A \cdot Y$ = moment of area of A about baseline, sq cm · m
AT = total sectional area, sq cm
NA = position of neutral axis
YB = distance of neutral axis from baseline, m
YD = distance of neutral axis from deck at side, m
Z = distance of each element from neutral axis, m
I = second moment of area of whole section about neutral axis, sq cm · sq m
$\Phi = \frac{1}{I \cdot t_s}\sum_{i=1}^{n} A_i \cdot Z_i,\ i = 1, 2 \ldots n$
t_s = thickness of side shell plating
n = number of elements above, or below, neutral axis

Shear flow distribution

AQ = assumed shear flow $= \frac{F}{I} \cdot A \cdot Z$
$D = \oint \frac{\Delta S}{t}$, for each closed cell
$D_{ij} = -\frac{\Delta S}{t}$ for a common member between cells i and j
$P = \oint AQ \cdot \frac{\Delta S}{t}$, for each closed cell
QC_i = correcting shear flow for cell i
$Q_r = AQ_r - QC_{ir}$
= correct shear flow in member r of cell i
τ_r = shear stress $= Q_r/t_r$

Results

Q = shear flow distribution
τ = shear stress distribution
Φ = shear coefficient

Reprinted from JOURNAL OF SHIP RESEARCH, Vol. 19, September 1975, pp. 155-163