PHILOSOPHICAL TRANSACTIONS OF THE ROYAL SOCIETY A
MATHEMATICAL, PHYSICAL & ENGINEERING SCIENCES

On the Estimation of Bending and Shear Stresses in Beamlike Ships Travelling in a Seaway

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On the estimation of bending and shear stresses in beamlike ships travelling in a seaway

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Hydroelasticity studies of beamlike structures have been undertaken for the past 15 years or more. Investigations, dating as far back as 1984, have revealed the occurrence of proportionality between bending moment and shear force at certain regions of the hull, considering the symmetric motions and distortions.

Although such behaviour can be observed, in principle, in all beamlike structures, its pattern depends on various parameters associated with the type, structure and loading of the beamlike hull. In this paper, the occurrence of proportionality and its implications are examined for a variety of ships, such as container ships, bulk carriers, etc., travelling in a seaway.

1. Introduction

A ship is a complex and complicated structure designed to fulfil a particular function economically, taking into consideration a multitude of concepts such as strength, resistance, stability, manoeuvrability, etc., and built to survive in a hostile, non-uniform and unpredictable random environment. In general, unlike other modes of transportation – rolling off the assembly line – ships have no prototypes. Every ship is designed on paper and one has no means of assessing its success until the vessel is fully operational and remains so throughout its life without any major overhaul.

It is, therefore, only fair to say that the design and construction of ships is a difficult task. Experience has had and will continue to play a significant role in the conception and execution of such an undertaking. An outsider might, nevertheless, be amazed at the relative simplification of a naval architect's task by the existence of empirical formulae, rules and regulations covering almost every aspect of naval architecture. Inadvertently, one cannot deny the success of such a procedure, particularly in the time conscious era in which we live. On the other hand, one cannot overlook its failures either as a mere blip or as a statistic within the bounds of acceptable risk as loss of human life and environmental pollution ensue from such failures. We need only examine recent records to realize the existence of an abundance of disasters such as the loss of the bulk carrier Onomichi Maru, the disappearance of the oil–bulk–ore (OBO) Derbyshire and the failures suffered by her sister ships (Bishop *et al.* 1990*b*).

This paper is not, however, intent on describing postmortem examinations of disasters and failures at sea. Neither does it propose to describe the cure to the problem in the form of novel or improved empirical formulae, rules and regulations relating to the strength of ship hulls. It aims to describe and analyse the structural

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behaviour of different types of beamlike ships, namely a containership, a bulk carrier and an OBO, at sea. Scarcity of relevant data prevents, at the present, the extension of the analysis to other types of ships, such as tankers, passenger vessels, ferries and naval ships. The paper is only concerned with the symmetric (vertical) bending of hulls and the relevant loads. It must be noted that only the global behaviour of ship hulls are considered. The study shows that, despite all ships considered being of comparable dimensions and operating in similar environments – therefore governed by the same equations of motion – they display different characteristics of stress distribution along their hulls. The results tend to strengthen our conviction that designing for the strength of the midship section leaves much to be desired, for some ships, at any rate.

2. The analytical background

Hydroelasticity theory (Bishop & Price 1979) allows for the inherent flexibility present in ship hulls and thus estimates loads on the hull – such as bending moments, shear forces and ensuing stresses – arising from the application of external agents, for example waves. The calculation of these loads is based on fundamental principles and the method is, unlike the rigid body analysis, physically correct as the stresses are a direct result of the hull undergoing distortions. For beamlike hulls, the method uses the Timoshenko beam theory and a two-dimensional potential flow analysis (strip theory). The seaway is represented, in the frequency domain, by wave spectral density functions. From these a random seaway can be generated, in the time domain, by combining a large number of sinusoidal waves with random phasing. Transient excitation due to slamming can also be included using unit impulse response functions and appropriate input (slamming force) descriptions. For the symmetric motions and distortions, the method has been applied to a variety of ships with success.

During the investigation of the loss of the bulk carrier Onomichi Maru (Bishop *et al.* 1985*a*, *b*) a hitherto unnoticed relationship between the calculated bending moment and shear force records came to light. It transpired that at either end of this hull – for, approximately, 20% of the hull length – the bending moment and shear force curves were proportional to each other. To be more precise, at the aft end the bending moment and shear force records were mirror images of each other, while at the forward end they coincided with each other. This proportionality gradually disappeared as one approached amidships and this effect was discussed and previously illustrated. It was known that this vessel had lost her bow – broken at a position 85% of the length from stern – while travelling in a moderate to gale force sea condition. In addition photographs of the actual damage indicated contributions due to bending as well as shear. Therefore, the concept of principal stress appeared to be the only suitable way of representing the resultant global stresses on the structure. The use of principal stresses, rather than direct stresses arising from bending moment only, introduced two additional peaks in the stress curve along the hull, namely in the vicinity of 20% and 80% of the length from the stern, as well as in the vicinity of the midship section. These results were unaffected by the inclusion of slamming. This stress distribution coupled with the phasing of bending and shear stresses at the ends indicated to us the possibility of failure due to fatigue at about 85% of the length from the stern.

Preliminary investigations on the loss of the OBO Derbyshire confirmed the above results (Bishop *et al.* 1984*b*). Further investigations on a containership (Bishop *et al.*

Table 1. *Principal dimensions of the ships*

	ship A	ship B	ship C
L, length/m	216.4	281.94	281.0
B, beam/m	31.7	44.2	32.26
D, depth/m	17.3	24.99	24.6
Δ, displacement/t	66874	192065	67150

1990*a*) and a frigate demonstrated that this phenomenon was not a peculiarity of the two bulkers – rather a feature of all beamlike hulls – and was independent of the type of motion, i.e. static, sinusoidal, transient or random. The overriding question is, therefore, whether all ships should experience similar failures. The answer may lie in the fact that although the three sensitive regions identified above exist in all beamlike ships, their level of significance may vary in different ships and, even, loading conditions on the same ship. There is also the possibility that environmental conditions may affect this behaviour (Aksu *et al.* 1990).

This investigation will focus on symmetric bending moment $M(x,t)$ and shear force $V(x,t)$ realizations excited in beamlike hulls travelling in random long-crested seaways. The principal stress records are determined from the corresponding bending and shear stress records through the following relationship:

$$2\sigma_{1,2}(x,t) = \sigma(x,t) \pm [\sigma^2(x,t) + 4\bar{\tau}^2(x,t)]^{\frac{1}{2}}, \tag{1}$$

where the bending and shear stresses, say on the deck, are calculated as

$$\sigma(x,t) = M(x,t)/(I_y(x)/z(x)), \tag{2}$$

$$\tau(x,t) = V(x,t)/kA(x) \tag{3}$$

and

$$\bar{\tau}(x,t) = \kappa(x)\,\tau(x,t). \tag{4}$$

Here x denotes the longitudinal distance from stern, $I_y(x)$ the second moment of area, $z(x)$ the distance from the centroid to a position within the section, i.e. deck, bottom, etc., $kA(x)$ the effective shear area, $\kappa(x)$ a shear stress distribution factor within the section and t the time.

3. Numerical analysis

3.1. *Main particulars of the ships*

Three merchant vessels were selected for this investigation. Ship A is a bulk carrier (Bishop *et al.* 1985*a*, *b*), ship B is an OBO carrier (Bishop *et al.* 1984*a*) and ship C is a containership (Bishop *et al.* 1990*a*). The main particulars of these vessels are summarized in table 1. Sketches of the dimensionless mass per unit length, second moment of area, effective shear area and section modulus at the deck are shown in figure 1.

3.2. *Main particulars of the analysis*

The vessels were discretised using 20 sections for ship A and 50 sections for ships B and C. Six distortion modes were included in the analysis for ships A and C; while 12 distortion modes were required for ship B, due to its mass distribution. The adequacy of the number of distortion modes included in the analysis was verified using the still-water bending moment and shear force calculated from the net load (weight–buoyancy) and modal analysis. The primary results relating to mode shapes,

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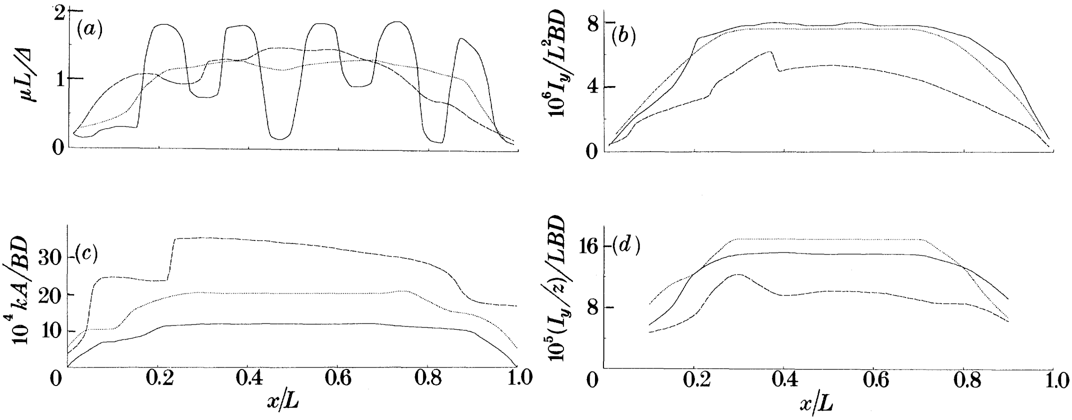


Figure 1. Non-dimensional (*a*) mass per unit length (Δ/L), (*b*) second moment of area (L^2BD), (*c*) effective shear area (BD) and (*d*) deck section modulus (LBD). Expressions in brackets indicate the factors of non-dimensionalization. ·····, Ship A; ——, ship B; ------, ship C.

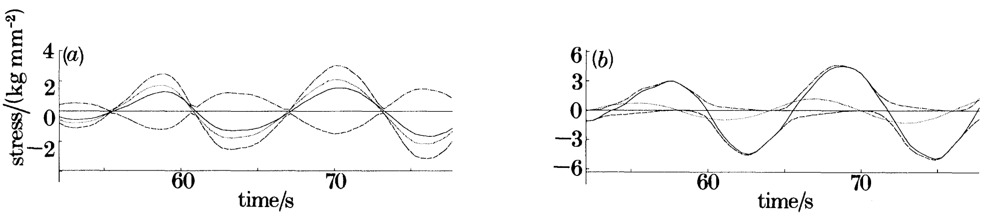


Figure 2. Portions of stress realizations for ship B at (*a*) 0.15 L and (*b*) 0.50 L from stern. ——, σ; ·····, τ; —·—·—·, σ_1; ------, σ_2.

natural frequencies, resonance frequencies, etc., are not discussed as these are published elsewhere. The vessels travel in the same long-crested head seas, defined as moderate gale force, with forward speed 10 knots for ships A and B and 20 knots for ship C. The seaway is described by an ISSC wave spectrum with significant wave height $h_{\frac{1}{3}} = 5.5$ m. The characteristic period T_1 is taken, initially, as 11 s. The duration of the simulation is 24.5 min. For these three-deep-draughted vessels it was shown that slamming did not take place in the prescribed seaway.

3.3. *Representation of the results*

By definition σ_1 is a positive stress (tensile) while σ_2 is a negative stress (compressive). In contrast bending and shear stresses can be positive or negative. Portions of the stress realizations experienced by ship B are shown in figure 2*a*,*b* at positions 0.15 L and 0.50 L from the stern respectively. These figures illustrate the diverse behaviour of the stress distributions along the hull. Therefore, finding a suitable measure illustrating this diversity is not a straightforward process due to the nature of the random processes representing the stresses (Price & Bishop 1974). To begin with, this measure has to reflect adequately the behaviour of the principal as well as the bending and shear stress realizations since the relative significance of the former to the latter is sought. It has to be representative of the whole random process; and last, but not least, it has to be relatively easy to obtain and comprehend.

The bending and shear stress realizations, particularly in the absence of transients, have zero or negligible mean values; while the principal stress realizations have non-

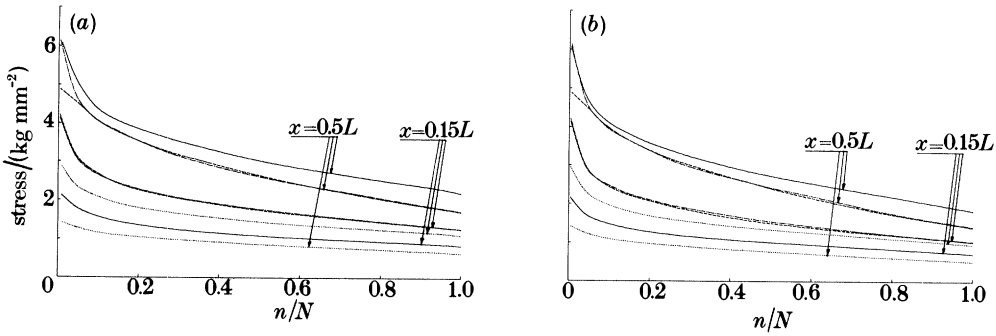


Figure 3. Average statistics of peak magnitudes for ship B at 0.15 L and 0.50 L from the stern using (*a*) average sampling and (*b*) gradient methods. (N is the total number of peaks and n indicates the first n highest peaks.) ——, σ; ·····, τ; —·—·—·, σ_1; ------, σ_2.

zero mean values. Therefore use of the mean value for the entire random process is unsuitable. The mean square values, on the other hand, are more suitable but not very useful on their own. These two statistical values can be used to obtain the root mean square (r.m.s.) value of the entire random process. This, however, by definition will produce smaller values for the principal stresses compared with bending stresses. Other statistical properties such as average $\frac{1}{10}$, or significant values (average $\frac{1}{3}$) tend to enhance the influence of large peaks or transients, when present.

The maximum value of the stresses during one realization is a convenient way of illustration, although it represents an extreme. Statistical averages of all the peak magnitudes occurring in a realization are representative as well as placing all stresses on the same footing. Such values are also more relevant in describing the cyclic stressing of the structure. Identification of peaks, however, may pose some practical problems. Sampling a realization at regular intervals, based for example on the zero upcrossing period, is a simple and effective way of obtaining the peaks. However, this method might not produce the whole peak sample space, particularly when transients are present. Identification of the peaks using the gradient of a random realization, on the other hand, may produce a clutter of results due to the presence of undulations. The effects on the peak magnitude average statistics are shown in figure 3*a*,*b* using average sampling and gradient methods respectively for ship B at positions 0.15 L and 0.50 L from the stern. These curves represent the average 5% ($\frac{1}{20}$), 10% ($\frac{1}{10}$), 15%, ..., 100% (mean value) peak magnitude statistics. The differences between the two methods are small, particularly towards the ends of the hull due to more orderly stress realizations, a result of the proportionality between bending and shear stresses present in these regions.

The peak statistics presented in the remainder of the paper are obtained using the gradient method. It should be emphasized that the statistics relating to the principal stresses are derived from the corresponding principal stress realizations. One should refrain from using equation (1) for operations involving statistical properties.

3.4. *Stresses experienced in a seaway*

Figure 4*a*–*c* show the distribution of the maximum stresses along ships A, B and C experienced in the same random seaway (i.e. ISSC wave spectrum, $h_{\frac{1}{3}} = 5.5$ m, $T_1 = 11$ s and same random phase distribution). Figure 5*a*–*c* show the distribution of the r.m.s. values of all the peak magnitudes along ships A, B and C. The shear stress

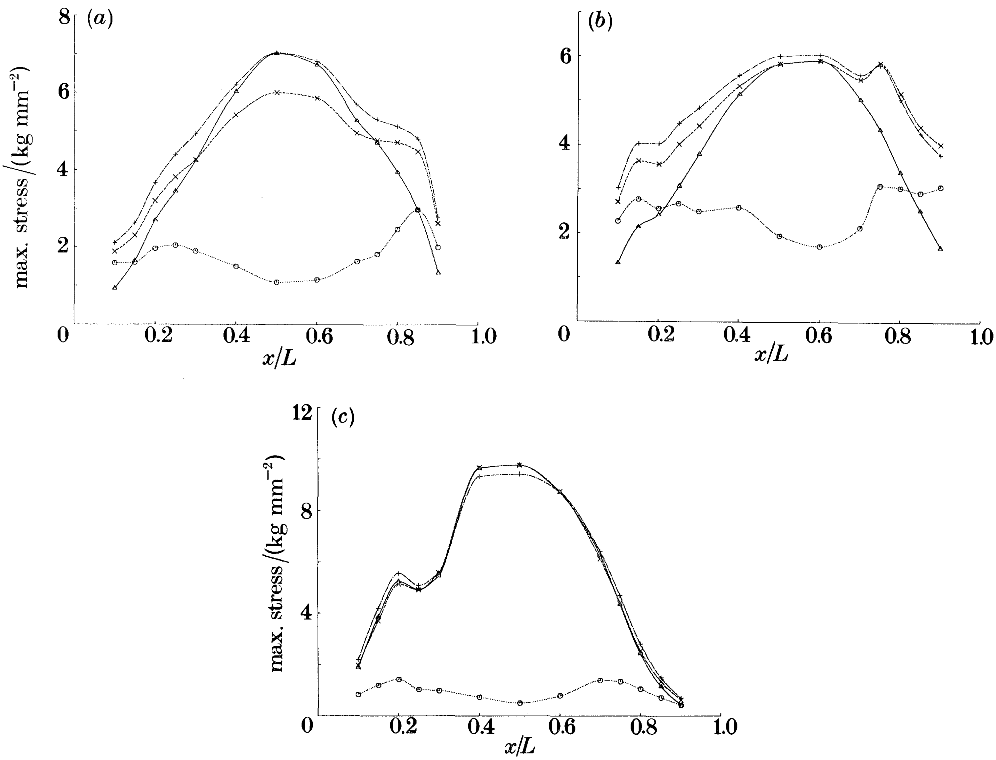


Figure 4. Maximum stress magnitudes experienced in a seaway (ISSC wave spectrum $h_{\frac{1}{3}} = 5.5$ m, $T_1 = 11$ s) for (*a*) ship A, (*b*) ship B and (*c*) ship C. ——, σ; ·····, τ; —·—·—·, σ_1; - - - - - -, σ_2.

distribution factor $\kappa(x)$ was taken to be 1 throughout the hull. The variations of the stresses along the hull show similar trends whether one examines the maximum peak or the r.m.s. of the peaks, as well as the $\frac{1}{10}$ and $\frac{1}{3}$ average values which are not shown here.

The shear stresses, in general, exhibit maxima near the ends of each hull and a minimum in the midships region. The position and sharpness of these peaks (or troughs), however, vary from hull to hull. For example, in ship A the shear stress has a large, and reasonably sharp, peak at 0.85 L from the stern while the peak at 0.25 L from the stern is rather flat; however, for ship B the shear stress maxima are rather flat at either end but extend over a longer portion of the hull; for ship C, on the other hand the magnitude of the shear stress is considerably smaller from the bending stress.

The bending (or direct) stresses are expected to have a peak amidships and decrease towards the ends. Ships A and B, to a certain extent, live up to this expectation. Nevertheless, one can easily note the presence of humps in the bending stress at 0.85 L from the stern for ship A and 0.15 L from the stern for ship B. The bending stress distribution on ship C clearly displays two peaks: a large one amidships and a smaller one at 0.20 L from the stern.

It is these trends observed in the variations of the bending and shear stresses along the hulls which shape the behaviour of the principal stresses. Accordingly, for either ship A or B, the principal stresses follow closely the trend of the bending stress in the vicinity of amidships, but diverge from it towards the ends exhibiting larger

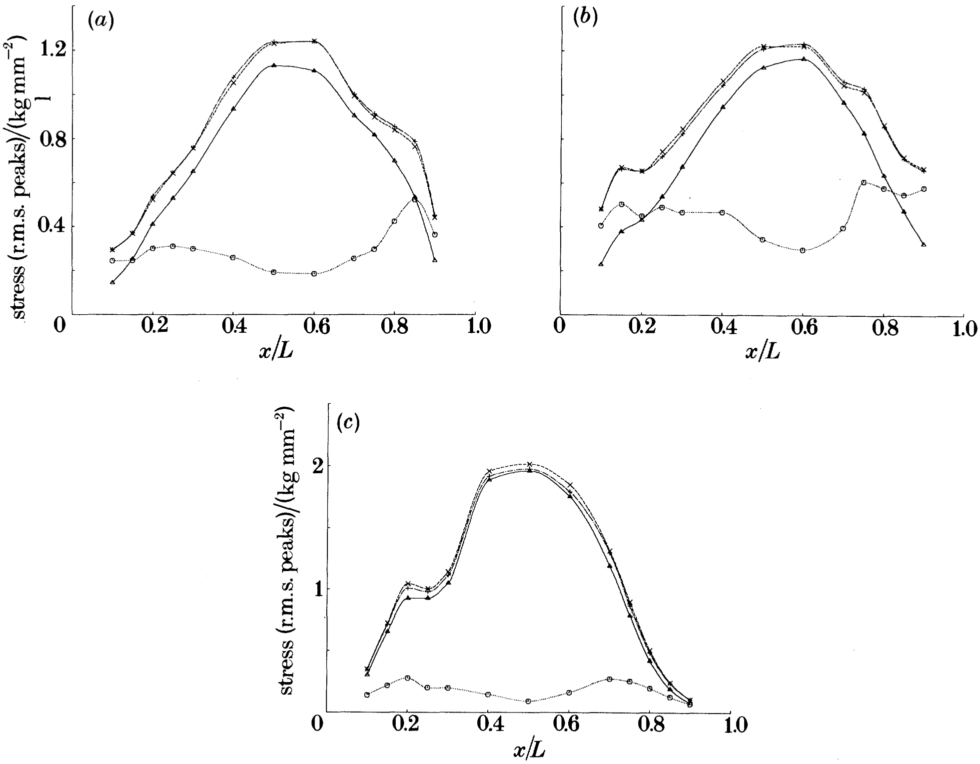


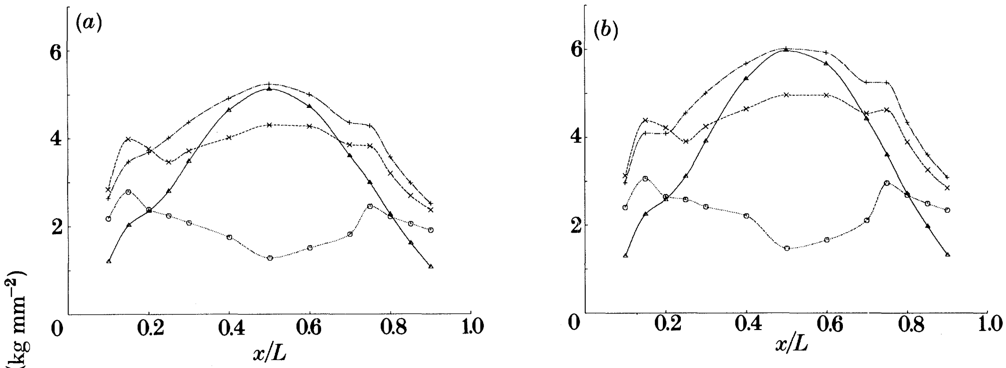
Figure 5. R.m.s. values of the peak stress magnitudes experienced in a seaway (ISSC wave spectrum $h_{\frac{1}{3}} = 5.5$ m, $T_1 = 11$ s) for (*a*) ship A, (*b*) ship B and (*c*) ship C. ——, σ; ·····, τ; —·—·—·, σ_1; ------, σ_2.

magnitudes. However, while one can clearly identify three peaks for the principal stresses for ship B – at 0.20 L, 0.50 L and 0.80 L from the stern – the peak in the aft end for ship A is not as distinct. On the other hand, the principal stresses for ship C follow the bending stress very closely throughout the hull.

The above observations can be verified by examining the structural properties of the hulls. It is seen from figure 1*c* that ship C has a comparatively (in fact, absolutely) large effective shear area; hence the effects of the shear force are, practically, negligible. The sharp drop of the bending or principal stresses aft of amidships and the subsequent peak at 0.20 L from the stern are due to apparent reinforcement between 0.25 L and 0.40 L from the stern (see figure 1*b* or *d*); in fact this is the area where the engine room and accommodation unit reside. Ships A and B have effective shear areas of similar magnitude; although figure 1*c* suggests that ship B has the smallest effective shear area, in relative terms. At any rate, the effects of the shear force cannot be neglected for either ship A or B. Examining the section modulus at the deck, in figure 1*d*, we may also note the relative weakness of ship A in the fore region and ship B in the aft region.

To ensure that the results shown in figures 4 and 5 describe patterns depending on the hulls themselves and not the generated seaway the behaviour of ship B was examined by varying the parameters of the seaway. To begin with, the wave spectrum parameters were kept the same and the random phase angles were changed. The results are shown in figure 6*b*. The variation of stresses in these figures are similar

Figure 6



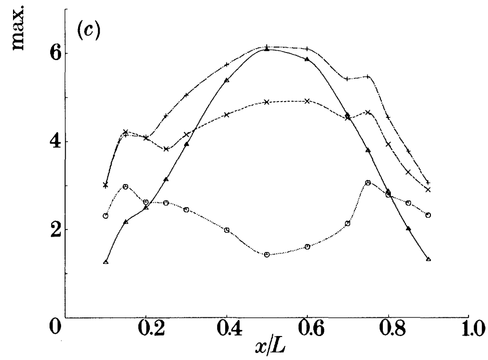


Figure 7

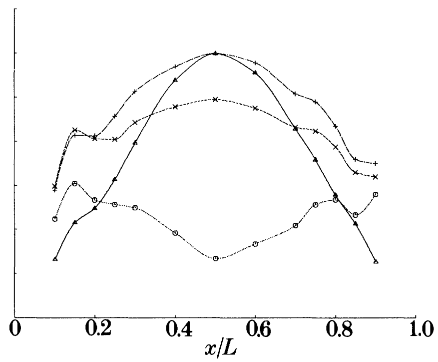


Figure 6. Maximum stress magnitudes experienced in a seaway (ISSC wave spectrum $h_{\frac{1}{3}} = 5.5$ m) for ship B when (*a*) $T_1 = 9.5$ s, (*b*) $T_1 = 11$ s and (*c*) $T_1 = 12.5$ s. ——, σ; ·····, τ; —·—·—·, σ_1; - - - - - -, σ_2.

Figure 7. Maximum stress magnitudes experienced in a seaway (ISSC wave spectrum $h_{\frac{1}{3}} = 5.5$ m, $T_1 = 11$ s) for ship B under a hypothetical uniform mass distribution. ——, σ; ·····, τ; —·—·—·, σ_1; - - - - - -, σ_2.

to the ones shown in figure 4*b*. However, the shear stress distribution, in this case, has sharper peaks – in particular at the aft – and therefore the aft peak of the principal stresses is almost as large as the fore peak.

Subsequently the characteristic period of the wave spectrum was varied, maintaining the same $h_{\frac{1}{3}}$ and random phase angles. The results for $T_1 = 9.5$ s and 12.5 s are shown in figure 6*a* and *c* respectively. Figure 6*a–c* are, fundamentally, similar to figure 4*b* with slightly different magnitudes and peak characteristics. This similarity, demonstrated for the maximum values only, is also observed in the statistical properties such as r.m.s. values of peaks, etc.

We cannot fail noting the marked variations in the mass distribution of ship B, compared with the two other ships, which is due to the preferred manner of loading of OBO carriers (i.e. one hold loaded, the next empty, etc.). To examine the possible influence of this on the stress variations on ship B a hypothetical uniform mass distribution was generated without altering the displacement. The maximum stresses experienced in a seaway described by an ISSC wave spectrum ($h_{\frac{1}{3}} = 5.5$ m and $T_1 = 11$ s) are shown in figure 7. These results are comparable with the ones

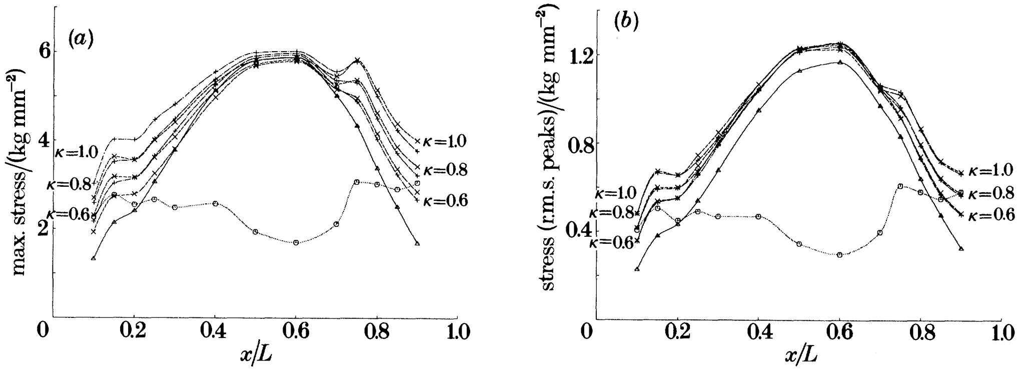


Figure 8. (*a*) Maximum and (*b*) r.m.s. values of the peak stress magnitudes experienced in a seaway (ISSC wave spectrum $h_{\frac{1}{3}} = 5.5$ m, $T_1 = 11$ s) for ship B for $\kappa(x) = 1.0$, 0.8 and 0.6 for all positions along the hull. ——, σ; ·····, τ; —·—·—·, σ_1; ------, σ_2.

shown in figure 6*b* and they exhibit the same pattern of behaviour; the only notable difference being the rather blunt shape of the fore peak of the principal stresses.

The shear stresses calculated according to equation (3) are nominal values experienced at the neutral axis. Away from the neutral axis these values decrease. It is difficult to predict accurately the rate of this reduction as the variation of the shear stress within a section is not known. Nevertheless, the effects of such a reduction are demonstrated, assuming different values for the shear distribution factor $\kappa(x) = 1.0$, 0.8 and 0.6 throughout the hull, for ship B in figure 8*a*, *b* for the maximum and the r.m.s. values of the peak stress magnitudes respectively. From either figure, it can be seen that the peaks in the principal stresses are still present, although with a reduced magnitude, even for the smallest assumed value for $\kappa(x)$. It is also worthwhile noting that the effect of the reduction on the peaks of the principal stresses is more pronounced in the fore end. It must be emphasized that the assumption for $\kappa(x)$ being constant along the hull has been made for simplicity. In reality $\kappa(x)$ will depend on the shape and material distribution within a section.

4. Conclusions

A study has been carried out investigating the stress distributions on a variety of hulls considering the symmetric motions and distortions. The influences of environmental and loading conditions are also examined. This study, although by no means extensive, encompasses the main parameters involved and confirms previous findings corroborated by actual failures.

1. All the ships examined clearly indicate that the midships region should not be the sole characteristic taken into account during the determination of strength-related properties. For the containership, probably due to the double-skin structure, the effects of shear stress are practically negligible. This is definitely not the case for the two bulk carriers. Based on global properties areas of sensitivity are identified within the structure, namely at midships and the regions 15–20 % of the length from either end. The analysis, due to its global nature, cannot take into account any local strength effects. However, it is only fair to assume that any local strength deficiency will have more adverse consequences for these sensitive areas, the end regions in particular, since a reduction in stress values is, in general, stipulated in these areas during design.

2. On the evidence available, the random quality of the seaway and changes in the parameters describing its state do not affect the blueprint of the sensitive areas identified.

3. The effect of changing the load distribution for ship B does not appear to influence the sensitive areas. Nevertheless, more studies involving various, real life, ballast and partly loaded conditions are necessary. Further hypothetical load conditions will only cast a shadow on the validity of the calculations.

4. Past studies indicate that the inclusion of transient (slamming) effects tend only to amplify the obtained results.

5. A reduced shear stress contribution on the deck (or bottom) principal stresses – taking, rather crudely, into account the variations of the shear stress away from the neutral axis – does not alter any of the conclusions.

6. Investigations on other types of hulls should be carried out as a matter of priority.

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Discussion

D. W. Robinson (*Lloyd's Register, London, U.K.*). The paper concentrates on the now proven phenomenon of proportionality of bending moment and shear forces at certain regions towards the ends of ships and goes on to suggest that design procedures for overall hull girder bending strength should be based on principal stresses rather than separate criteria for bending and shear.

Any criticism of existing procedures must be viewed against the success of current practice, where special attention is paid to the sheerstrake, the main structural component where principal stresses may be important.

Bearing in mind that ships, particularly at their ends, are complex three-dimensional structures, how should calculated principal stresses be used for global and local strength assessments, what failure modes would be relevant and how should criteria be set to prevent failure? What steps have been taken to calibrate their theoretical methods against full-scale measurements for the type of ships discussed in the paper?

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P. Temarel. In a two- or three-dimensional system principal stresses reflect the true magnitudes of stresses arising in the material. Use of Mohr's circle or the Von Mises criteria is common in many branches of engineering and, perhaps, the time has come for its adoption by the naval architects as well. Establishing separate criteria for bending and shear implies the uncoupling of these actions, contrary to the actual physical phenomenon. Furthermore, as Professor Pedersen and Professor Hylarides suggest, the method can be extended to include the stresses arising from horizontal bending and torsion. The success of a practice is judged by its lack of failures and not the other way round. It is perhaps fortuitous that failures tend to attract more publicity than successes. Our research, at present, identifies the weaknesses in the global strength of ships. Our main concern is the proper evaluation of the dynamic loads experienced by ships at sea. We would welcome any full-scale measurements to verify our findings.

P. T. Pedersen (*The Technical University of Denmark, Denmark*). Dr Temarel has used an empirical formula according to which the shear force is taken as an assumed fraction of the shear force divided by the cross-sectional areas. When he wants to combine bending induced stresses with the shear-induced stresses it is important to take into account also the spatial distribution of these stresses on the cross section. Why didn't he calculate the stress distribution according to the multicell thin-walled beam theory? I believe that such a consistent stress analysis will show that the combined stresses will take their maximum values at the intersections between the deck and the ship sides. This could motivate an extension of our study to include also the horizontal bending stresses and torsion induced shear and warping stresses in his analysis.

Dr Temarel explained that it was necessary to include up to 12 modes in the analysis to get sufficient accuracy. The response type he is dealing with has a relatively low frequency compared with the lowest hull frequency. Couldn't the same results be obtained using a rigid beam theory? Assuming a rigid ship hull it is also possible from simple equilibrium consideration to derive the phase angles between shear forces and bending moments he has observed.

P. Temarel. Lack of detailed structural data on cross sections prevents us from doing anything but the most rudimentary stress calculations contained in this paper. May we also add that we find what he states about the combined stresses at the intersection of deck and sides very interesting. It is possible to derive stress realizations using the rigid body analysis. The relevant software in our possession performs only frequency domain analysis and needs to be extended, to put it lightly, to allow for time simulations. However, we need to draw his attention on the fact that the stress–strain–distortion relationship implies a non-rigid ship. Furthermore, a rigid body analysis will exclude the distortion related resonances. These resonances are significant for ships of comparable size to the ones used in this paper.

D. B. Foy (*London, U.K.*). The studies described by Dr Temarel appear to be applicable to new vessels and those which retain their design strength. About 300 million tons of iron ore are shipped by sea every year, a significant proportion I think in old general purpose bulk carriers in which the bending and shear stresses referred to result in the loss of ship, cargo and crew.

I would like to see some of the resources of Brunel University devoted to a study of general purpose bulkers in the 15–20-year-old age bracket, assuming access can be gained to such vessels. Many will be found to be badly wasted but kept 'in class' by some classification society, kept insured, and kept in service. The general purpose bulk carrier is particularly vulnerable to structural failure when carrying ore as the centre of gravity of the cargo if evenly loaded is too low. Raising that centre of gravity by putting more cargo in some holds and leaving some holds empty will of course produce shear stresses. Professor Price, in his opening address spoke, of the very high incidence of foundering of bulk carriers. Indeed many lives are lost every year as sinking is too rapid to enable a distress message to be transmitted or lifeboats to be launched. Surely research should be directed at the hull condition and the working condition of the 15–20-year-old bulk carrier?

P. Temarel. A systematic study of bulk carriers and other types of ships in various load conditions is necessary. We would welcome any relevant data.

S. Hylarides (*Wageningen, The Netherlands*). To derive an impression of the acceptability of the stress system in a given point of the structure the Mohr's principle is used. For a plain stress system the following formulation is used: $\sigma_{eq} = \sqrt{(\sigma^2+4\tau^2)}$. This expression is nonlinear. As long as we operate in the time domain we can exactly derive the time history of this equivalent stress. However, when statistical predictions are made phase differences are lost and the statistical predictions of the equivalent stress will become very conservative, to my opinion.

Secondly it has to be realized that in the paper only the loads and motions in the vertical plane are considered. Torsional and horizontal deformation leads to additional stresses and have to be accounted for in the prediction of the maximum lifetime stresses in the ship's hull. Statistically they have to be dealt with simultaneously with the other stresses. In my opinion this makes the prediction more conservative. How will the predictions become less conservative?

P. Temarel. The equivalent stress is only part of the expression used for the evaluation of the principal stresses (see (1)). Calculated bending, shear and principal stresses are individual random processes with their corresponding statistics obtained as temporal averages. When one attempts to represent an entire realization by a few temporal averages information is, inadvertently, lost. We have tried to be less conservative in our estimates by considering peak magnitude statistics. It is made clear in the paper that (1) should not be used for statistical predictions.