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Semi-Probabilistic Approach to the Design of Marine Structures

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Abstract

A complete statistical approach to ship primary safety is not possible at present. The paper briefly reviews semi-probabilistic methods and presents safety index and partial safety factor data for eighteen merchant ship and five naval designs ranging in length from 90 to 330 m. One ship from each group is analysed in greater depth for illustration and to demonstrate the scope for improved design. This shows that for merchant ships there is plenty of scope for weight and cost savings, especially for larger ships. However, for transversely framed ships below 120 m length current rules can lead to surprisingly low safety levels in bottom structure. Safety levels are suggested for the design of merchant and naval ships which fall into two distinct zones. Comparisons are made with those for offshore and submarine structures. Shortcomings are identified and suggestions for further advances are offered.

Nomenclature

C	capability (strength)
D	demand (loading)
$\bar{C}_u$	mean capability
C_k , D_k	characteristic capability, demand
D_e , $\bar{D}_e$	mode, mean of extreme total demand
$f(x)$, $F(x)$	probability density, distribution
F	scalar safety factor
H_e , $H_{1/3}$	effective, significant wave height
L	length BP
M_w , M_s	wave still water moment (demand)
M_u	ultimate hull girder moment (capability)
p , p_f	probability, of failure
p_c , p_{cr} , p_s	collapse, classical, yield pressures for domes
s	standard deviation
S_ζ	wave spectrum
T_1 , ω	wave period, frequency
Z	total section modulus
v	coefficient of variation (s/mean)
α	$1+\zeta$ correction factors
β_f	safety index
γ_c , γ_f , γ_m	partial factors for consequences, forces, material
γ_o	overall partial safety factor
λ	wave length, slenderness parameter
$\bar{\lambda}$	reduced slenderness parameter
κ	slenderness safety coefficient
ψ	post-buckling factor
ϕ	compression strength parameter σ_u/σ_y
ρ	knock-down factor σ_E/σ_{cr}
σ_{cr} , σ_E	classical, imperfect elastic collapse stress
σ_s , σ_w	still water, wave bending stress
σ_u	ultimate average compression strength of most critical gross-panel
σ_y , $\bar{\sigma}_y$	nominal, mean yield strength
θ	central safety factor $\bar{C}_u/\bar{D}_e$
ζ_c , ζ_s , ζ_y	systematic errors in compression strength, section effects, yield strength

Subscripts:

o , s	objective, subjective uncertainties
n	nominal values
C , D	capability, demand

1 BACKGROUND

Marine structures are thin stiffened shells with randomly disposed fabrication imperfections and material properties and subjected to sea loads. It therefore should follow without argument that if one is interested in designing with adequate but not excessive safety, to some optimised combination of weight and cost, a rational approach to design must be statistically based and should take account of three types of error or uncertainty:

- Random
- Systematic
- Blunders

These latter arise from negligence or as a result of circumstances not previously envisaged as a possible cause of failure and are unquestionably the cause of the vast majority of accidents and perhaps of 85-90% of failures (total collapse). They arise mainly from human errors which should be reduced by good supervision and independent checking in design and construction (especially where new types of structure, materials and methods of fabrication are being used), by engaging well qualified staff and by reducing the time, economic, political and other pressures under which the design is done(1). There seems to be little direct relation between the occurrence of blunders and the formal margins of safety such as the probability of failure or the more conventional safety factor. By their very nature blunders defy formal treatment beyond exercising the human and organisational precautions mentioned above. No further mention will be made of blunders.

We can, however, consider random and systematic errors, although the latter are often overlooked. For convenience they will be divided where possible into OBJECTIVE and SUBJECTIVE components. This distinguishes between strength or other variables for which statistical data can be collected and properly understood and those where objective knowledge is lacking and where their assessment requires appreciable experience and judgement, for example, with analysis assumptions in both loading and response.

It is convenient to classify safety concepts according to their degree of sophistication. Ref.(2) to which this paper is really an extension (so detailed arguments will not be repeated here) identifies three levels:

(a) First moment methods - essentially the "safety factor" approach where a worst design load or "demand" D is related to a similarly dimensioned single valued limiting "capability" C of the structure by a scalar quantity F so

$$D \leq C/F \qquad (1)$$

and very often maximum stress, or "equivalent stress", is still used as a basis for comparison even where it can be shown to be a poor index of limiting or failure load.

(b) Second moment methods - where D and C are assumed to be independent random variables which can be represented as shown in Fig. 1, and that it is possible to estimate their second moment statistical properties of the means and standard deviations. The safety margin is still scalar and the methods are therefore hybrid or "semi-probabilistic" in combining conventional determinism with statistics. Two such methods are:

(1) Safety index - assuming C and D to be uncorrelated:

$$\beta_f = \frac{\bar{C}_u - \bar{D}_e}{\sqrt{s_C^2 + s_D^2}} = \frac{\theta - 1}{\sqrt{\theta^2 v_C^2 + v_D^2}} \qquad (2)$$

and the approach dates from 1955(3) but was first applied to ships in 1974(4).

(2) Partial safety factors:

$$\gamma_c \gamma_f D_k \leq C_k/\gamma_m \quad \text{or} \qquad (3)$$
$$\gamma_o = C_k/D_k = \gamma_c \gamma_f \gamma_m$$

This second method (hereafter referred to as PSF) is sometimes referred to as Level-1 method because common usage really makes little if any reference to statistical properties and merely relates "nominal" values of design loads and lowest strength and notional strength models and/or lower-bound results. The method was first applied to ships in 1978(3).

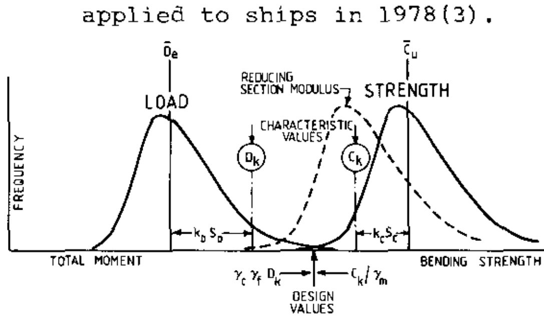

Fig.1. Frequency distribution of load and strength with main statistical parameters

(c) Full statistical approach - in which probabilities of failure in all likely modes are added using purely statistical data and methods

$$p_f = p\{C < D\}$$

If C and D are uncorrelated random variables and both are time invariant

$$\left.\begin{aligned} p_f &= \int_0^\infty \{F_C(x)\} f_D(x)\,dx \text{ , or} \\ &= 1 - \int_0^\infty \{F_D(x)\}\, f_C(x)\,dx \end{aligned}\right\} \qquad (4)$$

where f(x) and F(x) are probability density and distribution functions respectively for the C and D curves. After much research by many and encouragement from the ISSC(5) the approach was first applied to ships in 1972(6,7).

The author believes firmly in the level-3 approach (which is the only one able to combine all modes of failure) but, recognising the formidable difficulties, the Level-2 methods have been advocated as an interim stage (2) for much the same reasons as advanced by Mansour(4).

Since Ref.(2) concentrated on the Safety index approach rather more attention is paid in the paper to the PSF level-2 approach, with some mention of its current level-1 treatment from which most partial factors are derived, for example, in offshore codes. The PSF format, albeit mainly in level-1 format using nominal material properties and "lower bound" formulations, is proving to be acceptable and suited to the design requirements which benefit directly from improved knowledge.

In conjunction with limit state design the use of suitable PSFs will result in greater economy and more consistent safety or reliability than, for example, elastic stress factor design. Both are proven facts for Steel Bridge design. Probabilistic calibration of the draft BS 5400 Part 3 shows(8), and I quote Baker(9), "that components of bridges designed to the existing standard BS 153 vary by as much as 10 orders of magnitude in their notional failure probability, and that by introducing a partial factor format the following can be achieved simultaneously:

- the average safety levels implied by design to BS 153 can be maintained;
- the safety of the least reliable components as designed to BS 153 can be considerably improved;
- average savings of about 6% in the total amount of steel used can be achieved.

This might seem remarkable but is fairly easily demonstrated". It was claimed seven years ago that some 30% of the structural weight could be saved in large ships (Faulkner, ISSC, 1973). This claim can now be substantiated very easily and will be illustrated. Some of this saving comes of course from the more realistic modelling of the limiting conditions as well as from the more rational choice of load factors, but much of it has been there to realise for a long time. The situation in warships is quite different and both communities have much to gain simply by implementing present knowledge with confidence, based on successful previous designs.

1.1 More About Partial Safety Factors

It is beyond the scope of this paper to rehearse in detail the many aspects of PSFs (see Refs.2,3,8-12). Their use now in many civil engineering codes follows the general principles described in the ISO 2394 of 1973. For multiple loads (type i) equation (3) is frequently expressed in the form

$$\gamma_c \sum_i \gamma_{fi}\, D_{ki} \leq C_{ki}/\gamma_m \qquad (5)$$

where the partial factors are γ_f related to force uncertainties, γ_m to material and fabrication factors and γ_c to the nature of the structure and the economic and social consequences of failure (often omitted or considered to be subsumed in the other factors). The RHS of eq.(5) may often involve interaction equations in which case it has been shown(11) to be more consistent to incorporate γ_f and γ_m in the individual terms. Moreover, each of the i type loads (e.g. live, dead, etc.) can give rise to individual j type forces on elements of the structure. The general formulation then requires a double summation

$$\gamma_c \sum_j \{\gamma_{mj} \sum_i \gamma_{fi} D_{ki}/C_{kj}\}^{nj} \leq 1 \qquad (6)$$

This may look formally complex, but it is in essence what is implied in the more advanced recent offshore codes(12). Further guidance on the use of interaction equations and other design calculations to achieve greater consistency and to reduce unnecessary errors has just been provided(13).

Finally, the following ranges provide some idea of the emerging values of the PSFs in current use for abnormal or extreme conditions

γ_f	1.0 to 1.5
γ_m	1.0 to 1.35
γ_c	1.0 usually
γ_D	1.5 to 2.0

Such consistency does not exist in ships.

2 COMPARISONS OF MERCHANT SHIP AND WARSHIP DESIGNS

For about twenty-five years there have been occasional papers which compare the wave loading experienced by cargo ships, tankers and destroyers (see, for example, Refs. 14 and 15). But, only in the last decade have these comparisons extended to structural safety in the two communities(2,7,16). The debate following Ref. 2 was extensive, and many contributors questioned the truth of the key finding that there are far greater reserves of strength against upper deck compressive collapse in merchant ships than in warships. But not one single contributor produced any objective evidence which substantiated their own beliefs or which could test in any numerate way the validity of the paper. Many mentioned the omission of fatigue, transverse strength and other modes of failure. In an attempt to break through this inertia of thought and acceptance this present paper therefore examines some of the differences and omissions of previous work more closely. Of the possible modes of failure

- Excessive yielding (not initial yielding)
- Buckling collapse
- Fatigue fracture (not minor cracking)
- Brittle fracture

the author has again quite uncompromisingly chosen inelastic compressive buckling collapse for reasons which will not be repeated in detail(7,11,17). It is certainly the most likely mode of failure for upper decks in warships, and in many merchant ships(17) and governs the structural weight, efficiency and cost in all ships(17,18). With level-1 or level-2 methods it is crucial to identify and analyse the most likely mode of failure.

The most popular argument against taking advantage of weight saving in primary structure of merchant ships is the possibility of fatigue becoming important. No one has demonstrated recently that it is so, and comparisons with much more highly stressed naval designs suggest the argument is correct in principle but false in practical terms(16). Moreover, the debate and reply to Ref. 2 itself demonstrated that low stress levels, for example to prevent fatigue, are not necessarily linked with high margins against compressive buckling.

Regarding transverse strength, of course, transverse structure is important in maintenance of longitudinal strength. However, this is a problem to be tackled directly by appropriate safety margins for transverse structure rather than by justifying high longitudinal strength levels to a degree that is bound to be somewhat arbitrary. In fact, except in way of double bottoms, the interaction between transverse loads and longitudinal strength has been found to be small in single-skin warships(19) and it is believed the same is true for merchant ships(17).

Ref. 20 has paved the way by identifying and defining Strength Reduction and Load Magnification factors f_s and f_l which correspond to γ_m and γ_f respectively. Their product γ_o (assuming γ_c is subsumed in γ_m and γ_f) has now been quantified for the primary strength of 23 naval and merchant ships, and this will now be presented. The derivation narrative will be terse, since it is based on previous work properly referenced. Bending moments are used for C and D to obviate non-linear problems.

2.1 Analysis for Merchant Ships

Mansour's data for 18 merchant ships (12 tankers, 3 cargo ships, 2 bulk carriers, 1 oil-ore carrier) ranging in LBP from 158.5 m (520 ft) to 328 m (1076 ft) has been used(4). The sea description was simplified by using long-crested head seas and Pierson-Moskowitz spectra for fully developed seas. A typical Middle East to North America mission profile was chosen for 16 of the ships, as in Fig. 2, based on Ref.21.

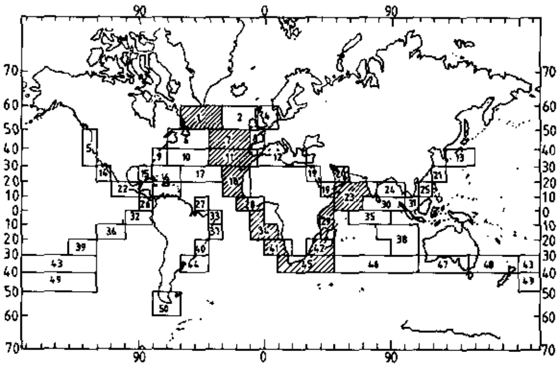

Fig.2. Assumed ships route

For tanker No. 12 the mission profile included Marsden squares 1,2,4 (North Atlantic), 12 (Mediterranean), 23,30,31, 25,21 (Middle East to Far East) to facilitate comparison with a naval frigate(7). Ship no. 16 was a MARINER class cargo ship wholly operating in the North Atlantic to study the effect of a more extreme environment(6). The ships were assumed to be underway for 300 days a year over 20 years of service at 20 knots (or the maximum for the available power in high sea states) and to be fully loaded for half the time and in half-load for the remaining life.

To obtain the "demand" curve the wave response was evaluated using an MIT strip theory program and added to the still water moment to give a total load. The mean value of the extreme wave load $\bar{D}_{ew}$ was obtained from a long-term analysis which linearly summed the mean values of response for a range of differing frequencies and appropriate significant wave heights weighted for times operating in each sea area(21). The objective variance of the wave bending moment s_D^2 was obtained from the r.m.s. less the square of the mean value. The procedure has been fully described(6,7).

The strength or "capability" mean was taken simply as:

$$\bar{C}_u = \sigma_y Z \qquad (7)$$

where the nominal yield was taken as 207 N/mm² (13.4 tsi) and strength coefficient of variation was assumed to be $v_C = 0.13$ throughout.

The above describes the procedure adopted in Ref. 4 to obtain the four statistical parameters $\bar{D}_e$, $\bar{C}_u$, s_D and s_C from which the central safety factor θ and safety index β_f were determined using eq.(2). However, the raw data are not provided in Ref. 4 and so θ and β_f were measured from Figs. 12 and 9, and v_D was estimated using r.m.s. values derived from Figs. 6, 7 and 8 of Ref. 4. Then if γ_o is the overall partial safety factor it follows that (see Fig. 1):

$$\gamma_o = \frac{C_k}{D_k} = \theta\{\frac{1 - k_C v_C}{1 + k_D v_D}\} \qquad (8)$$

For 5% characteristic values and assuming normal distributions (no serious error) the k's may be taken as 1.645. Fig. 3 is a plot of the values of β_f and γ_o so derived.

The question mark against ships 12 and 16 are considered to be the correct plots if $v_C = 0.13$ as stated in Ref. 4. (The left hand plotted positions correspond to $v_C = 0.11$ and 0.09 respectively, which were the values derived in the original references 7 and 6. It seems likely, therefore, that these values were carried through Ref. 4 inadvertently). The full line is a mean curve faired through the points and the dotted curve is obtained by eliminating θ from equations (2) and (8) and taking the maximum positive root for θ, viz.

$$\gamma_o = \{\frac{1 - k_C v_C}{1 + k_D v_D}\}\ \theta\text{, where}$$

$$\theta \doteqdot \frac{1 + \beta_f (v_C^2 + v_D^2 - \beta_f^2 v_C^2 v_D^2)^{\frac{1}{2}}}{1 - \beta_f^2 v_C^2} \qquad (9)$$

and taking $v_C = 0.13$ and $v_D = 0.12$.

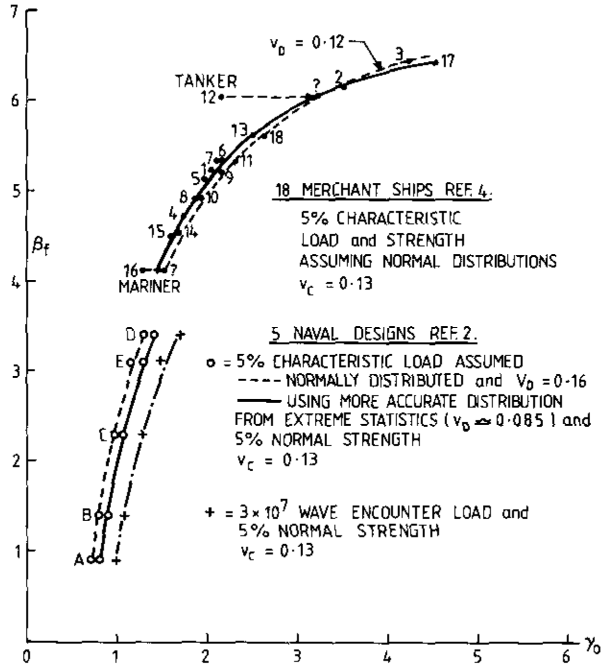

Fig. 3. Theoretical safety indices β_f overall partial safety factors γ_o.

This latter value $v_D = 0.12$ would therefore appear to be a reasonable value for the objective component of the wave load uncertainty appropriate for these merchant ships and the method of calculation.

The scatter in Fig. 3 in γ_o is particularly high and the mean values and c.o.vs of both safety parameters using the corrected values for ships 12 and 16 are:

Parameter	mean	cov %
β_f	5.3	12
γ_o	2.4	37

The β_f values are of course identical with those plotted against ship length in Fig. 9 of Ref. 4 and Fig. 5 of Ref. 2.

2.2 Analysis for Naval Ships

A similar theoretical analysis was conducted for the 5 naval designs of Ref. 2 ranging in length from 91.4 m (300 ft) to 153.9 m (505 ft). There would seem to be some very significant differences in both loading and strength models, but this will be discussed later.

The assumed ship life was 25 years with 120 days a year underway and operating entirely in the North Atlantic in Marsden squares 1 - 2, 6 - 11, 16 - 18.

The ISSC (1967) two-parameter spectrum was used and short crested seas were assumed with a $\cos^2$ spreading function - which seems to be justified by recent measurements(22). Directionality was derived from ships log data assuming the predominant waves came from the same source (direction) as the wind. A SCORES linear strip theory was used which calculates the vertical mean square wave-induced bending moments amidships at a number of heading and speeds in a variety of seas of unit wave height. These responses are weighted for time in sea areas and summed in the usual way using the environmental data(21). The so-called "long-term" wave bending moment probability distribution used in much of the Ship Structure Committee work(23,26)is illustrated in Fig. 4 for naval design B, a 110 m (360 ft) frigate. Her speed is 9.3 knots and hogging still water moment M_S = 34.3 MNm (11,290 tonf ft) and elastic section modulus for the upper deck is Z = 1.054 m^3 (5,360 in^2 ft).

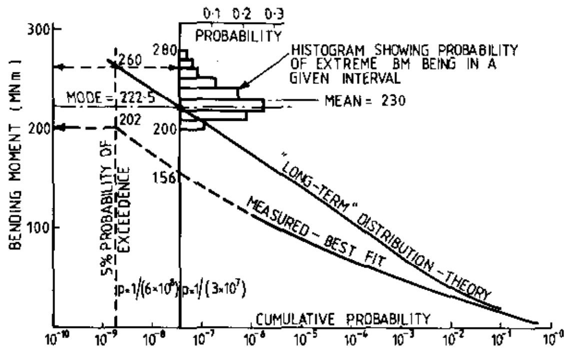

Fig. 4. Wave bending moment probabilities for naval design B

However, it should be noted that evaluating the mean and variance of a demand based upon this type of long-term distribution would not only yield different sets of safety measures from those above (as pointed out in Ref. 2) but based as it is on a probability per cycle it is incompatible with a capability curve that is not related to cycles of stress. We therefore require the probability distribution of the highest bending moment ever likely to be experienced in the ship's expected lifetime. This has been evaluated as described in Ref. 2 and is superimposed on Fig. 4 as an objective histogram at a cummulative probability of $1/(3 \times 10^7)$, there being about 3×10^7 wave cycles of 8.5 sec. mean encounter period in the vessel's lifetime of 3,000 days at sea. By taking first and second moments of this histogram, and by adding a $(\Delta M_W)^2/12$ variance correction for the interval width, gives the mean value and objective variance of the extreme wave moment. This leads to (see Appendix A & Refs. 24,25):

$$\bar{M}_W = 229.9 \text{ MNm (75,710 tonf ft)}$$

$$S_{OW} = 16.5 \text{ MNm (5,440 tonf ft)}$$

Thus the mean value of the total extreme sagging moment and its objective c.o.v. are:

$$\bar{D}_e = 229.9 - 34.3 = 195.6 \text{ MNm (64,410 tonf ft)}$$

$$v_{oD} = (16.6/195.6) \times 100\% = 8.5\%$$

This is in fact the average value of the objective load uncertainty for the 5 naval designs (range 7.0 to 9.5%) and it will be seen to be appreciably lower than the 11-12% average value for the merchant ships based on long-crested seas. To this has to be "added" the subjective uncertainty. A lengthy discussion(2) suggested that for the naval designs this was unlikely to be less than 15%, close to Lewis's suggested 14.9%(26). Some feeling for this unexplained scatter may be gained from Fig.5(27) which compares the ISSC two-parameter spectrum with a number of measured wave spectra for 13-14 m high waves. For ships of L = 90 to 150 m the value $\omega T_1/2\pi$ of most interest lies between 0.8 and 1.2.

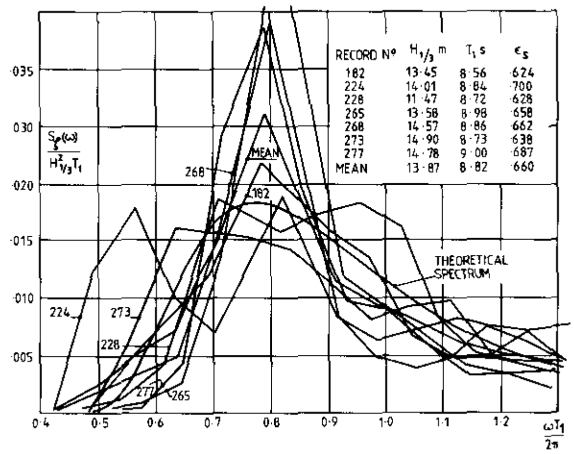

Fig.5. Comparison of spectral shape variation - from SSC-268

Thus the total loading uncertainty would for these severe seas be at least:

$$v_D = \sqrt{8.5^2 + 15^2} = 17.2\%$$

but a lower value of v_D = 0.16 was adopted to enable direct use of data from Ref. 2.

The ultimate bending model eq.(7) adopted for the merchant ships would be quite inadequate for the 5 naval designs with their low safety levels and slender scantlings giving rise to nominal compression strength parameters ϕ_n varying from about 0.53 to 0.74. Equations (17) and (18) and the data in Table III of Ref. 2 were used. In essence this model may be expressed as:

$$\bar{C}_u = \bar{M}_u = \phi_y \sigma_{yn} Z \alpha_y \alpha_{cd} \alpha_s \quad (10)$$

where σ_{yn} is the "nominal" yield strength used in the design and:

$$\phi_y = f_n(\phi_n, \alpha_y) \quad (11)$$

where ϕ_n is from design formulations and codes, α_y is a correction factor for the systematic error in yield strength as determined from sampled data, α_{cd} is a systematic correction to compression design codes, and α_s allows for cross-section effects. These have been described fully in Ref. 2 (see also the discussion at page 25), but a slightly modified model will be presented later in this paper. v_C was taken as 0.13 as before(2) and the effect of variations in this and in v_D will be examined parametrically.

Then, assuming normal distributions and applying equations (8) and (9), and using β_f (or θ) from Table VI of Ref. 2, the overall PSF γ_o for 5% characteristic loads and strengths was evaluated for the 5 naval designs and is plotted in Fig. 3 as points on the dotted line. Also shown as the full line are the more accurate values of γ_o using the properly derived extreme wave moment with a 5% probability of exceedence, which can be approximately derived as illustrated in Fig. 4 and discussed in Ref. 2. But this, however, assumes v_C is given only by the objective uncertainty of about 8.5%, which is why the PSF appears higher in spite of the skewness of extreme p.d.f. toward the higher values.

The scatter in Fig. 3 in β_f is particularly high and the mean values and c.o.v.s of the two safety parameters for the 5 ships are tabled below. For a larger population the c.o.v.s may be slightly smaller, but the means would probably be similar.

Parameter	mean	cov %
β_f	2.2	48
$\gamma_o(5\%)$	1.1	24

The safety parameters are between 40-45% of those tabled in Section 2.1 for Merchant Ships.

Also included in Fig. 3 as the chain-dotted curve is the corresponding PSF using the most probable (mode) lifetime load as D_k as given in Table III of Ref. 2. It was there argued (section 3.3.1.2) that, although in theory there was a 63% chance of exceeding this theoretical modal value of the extreme, the actual probability (in most designs), based on measured strains, is only about 2%. It may therefore provide an acceptable characteristic load.

Measured values for design B(2,15, 22) provide a basis for a best-fit curve which has been incorporated in Fig. 4 and it will indeed be seen to be well below the theoretical curve. It is pointed out, however, that this measured curve can only be an approximation since the data are derived from "return period" graphs of the sort presented in Ref. 15. The return period is the average period between exceedences of stresses (or moments) of a certain level and so some assumptions have been made in order to superimpose the results on the long-term cumulative probability plot. Accuracy is nevertheless considered to be sufficient for the present discussion, and the values accord with those given in the discussion of Ref. 2.

2.3 Analysis of Bottom Structure

Similar approximate analyses have been carried out for the bottom structure of a typical longitudinally stiffened cargo ship using current "rule" scantlings and for naval design B. The cargo ship is 160 m long (525 ft) with a N. Atlantic mission profile (to be comparable with ship 16). In both designs the extreme load pressure effects were approximately allowed for. With similar load and strength uncertainties as those assumed for the deck structure (0.12, 0.13) the safety margins are:

Ship	β_f	$\gamma_o(5\%)$
Cargo ship	4.2	1.8
Naval design B	1.8	1.0

Of greater interest, perhaps, are the results of two similar analyses for transversely framed ships - the WOLVERINE STATE of length 151 m (496 ft) and a fictitious but current "rule design" naval ship of length 120 m (394 ft). The former belonged to a class of cargo ships extensively studied for the Ship Structure Committee and included service strain measurements which picked up significant slam induced values ≈ 50 N/mm² (3.22 tsi). This class of ship operated in the N. Atlantic and are slightly smaller than the MARINER ship 16. The analysis is presented in Appendix B (which also includes an analysis for the transversely framed deck). Three things stand out:

a) Outer bottom safety measures are very much lower than for the deck, with likely values lying between $\beta_f = 1.7$ and 2.1 and $\gamma_o = 0.9$ and 1.0

b) There are large variabilities in safety measures and these arise mainly from calculation assumptions concerning strength formulations and still water loads

c) The range of still water moments is quite significant and can affect safety levels appreciably, especially in low safety designs

Transverse framing is allowed in, for example, Lloyd's Register current rules for ships of length 120 m or less. The structure would generally be cheaper to build and often weight is less critical than in deck structure. A "rule design" naval ship (120 m) was therefore examined which was close in specification to a current design(32). A double bottom structure was adopted with transverse frames at 700 mm (27.5 in) spacing, 11mm thick IB and 18 mm OB which are acceptable within the rules (but 5% of the plate thickness was ignored to allow for corrosion, mill-scale, rolling tolerances, etc.). The lifetime (3×10^7) extreme hogging compression stress in the OB is 158 N/mm^2 (10.2 tsi) which leads to the following results according to whether mean compression failure stress (σ_u in N/mm^2) is assessed by Refs. 29 or 30:

Ref.	OB $\bar{\sigma}_u$	Inner Bottom $\bar{\sigma}_u$	β_f	γ_o
(29)	130	35	-0.5	0.6
(30)	226	76	1.8	0.9

The still water bending stress in the OB is 72 N/mm^2 (4.7 tsi) compression and so it follows that the IB would undoubtedly be in a state of near collapse even before the ship left harbour. This of course ignores the "hard spot" and other effects(73,74) and the strain limitation imposed by the greater strength of the OB but it is unlikely that the current rules make even implicit allowance for these factors. The actual design was 122 m long and some longitudinal stiffening was introduced(32).

2.4 Comparative Remarks

Fig. 3 for upper deck safety displays a very large range and suggests certain important points for consideration:

a) Even the least safe merchant ships have higher β_f and γ_o values than for the safest naval design, and the differences in their mean safety levels are orders of magnitude apart (in terms of probability of failure p_f)

b)/

b) Indeed the two groups of ships fall into two quite distinct zones which may be categorised as:

I over-safe, and
II low-safety

It can be shown(2,4) that at the upper end of zone I (tanker 12) one order change in p_f is approximately equivalent to changes in β_f of about 0.5 and in γ_o of about 1.0, but toward the lower end of zone II (naval design B) the corresponding changes are approximately 1.0 and 0.2 respectively, indicating a distinct reversal in sensitivities

c) In spite of the different assumptions made the "line up" of the curves through the merchant and naval ships is good, and supports the basis argued for the many comparisons made in Ref. 2 which will not be repeated here

d) The two zones rather suggest that PSF γ_o may be a preferred basis for design of merchant ships because of the large spread in γ_o, whereas β_f may be preferred in naval designs where the lack of sensitivity of safety to γ_o is more obvious

e) Whatever the choice, the author sees no reason to depart from the advice given in Ref. 2 concerning appropriate safety levels to use in design, especially in view of the low safety implicit in certain bottom structures. This was:

β_f = 3.0 for merchant ships

β_f = 1.5 for naval designs

However, values of β_f in the range 4.0 to 4.5 may be more acceptable to classification societies until greater confidence is gained. Such values imply a γ_o of just under 1.0 for naval designs (values less than unity exist in supposedly successful ships) and γ_o = 1.5 to 1.7 for merchant ships within the assumptions made in the analysis.

f) The adoption of even these safety measures in merchant ships (clearly conservative when compared with certain bottom structure safety levels) would result in appreciable weight saving (see later).

It should be stressed that the safety measures discussed are "notional" in the sense that they depend upon the assumptions made in the analysis, e.g. mission profile, type of loading analysis, c.o.v. of load and strength.

Two other interesting observations regarding upper deck safety arise from the results tabled at the beginning of section 2.3 for a modern cargo ship and a rule naval design B. The increased safety levels for the latter compared with the actual design supports the contention of unnecessary conservatism in present rules, and this may also be borne out by the safety level for the modern cargo ship being somewhat greater than for the MARINER ship 16. One would have hoped with recent reductions in section modulus requirements (which dominate the strength model eq(7) for the merchant ships) that these oversafe safety margins would have reduced. However, one should perhaps not read too much into the result for one fictional ship.

Finally, it appears that in merchant ships whilst the safety levels of transversely stiffened decks is no less than for longitudinally stiffened decks, the same cannot be said of the strength of bottom structure. The results in Appendix B indicate low safety and a great sensitivity to assumptions concerning compression strength and still water loads. The last observation in section 2.3 also suggests present rules for 120 m (394 ft) ships could lead to very low safety, if not to unsafe, bottom structure designs. There is therefore strong evidence to suggest that the treatment of deck and bottom structure in current rules can be quite inconsistent as regards safety.

3 PARAMETRIC STUDIES

Following the pattern of Ref. 2 the variation of the safety parameters was examined over a credible range of the variables. This was done directly from eqs.(8) and (9). For example, for the three values of safety index suggested, that is, 1.5, 3.0 and 4.5, Table I shows the variation in the 5% PSF for values of v_C from 0.10 to 0.15 and v_D 0.10 to 0.20 assuming the distributions are approximately normal. Also shown is the corresponding variation in Central Safety Factor for one value of $v_C = 0.125$.

It will be seen that for a given β_f the PSF γ_o is barely sensitive to v_D and only increases significantly with v_C for the larger, and therefore less important values of β_f. For $\beta_f = 1.5$, γ_o decreases with increasing values of v_C and v_D as one might intuitively expect, but only very slightly. However, for the two higher safety indices γ_o increases as both v_C and v_D increase. This behaviour may seem strange on first sight, but of course to maintain the same safety index as v_C and v_D increase, the central safety factor θ also has to increase, and this is illustrated for one value of v_C in Table I. γ_o is more sensitive to v_C because it will be seen

TABLE I Variation in the 5% Partial Safety Factor γ_O and Central Safety Factor θ for three values of Safety Index β_f and a range of v_C and v_D

β_f / v_D	γ_O values $v_C = 0.10$	$v_C = 0.125$	$v_C = 0.15$	θ $v_C = 0.125$
$\beta_f = 1.5$:				
0.10	0.89	0.88	0.86	1.28
0.125	0.88	0.86	0.85	1.31
0.15	0.87	0.85	0.84	1.34
0.175	0.86	0.84	0.83	1.37
0.20	0.86	0.84	0.82	1.40
$\beta_f = 3.0$:				
0.10	1.12	1.16	1.24	1.71
0.125	1.12	1.16	1.22	1.76
0.15	1.12	1.16	1.21	1.82
0.175	1.13	1.16	1.21	1.88
0.20	1.14	1.16	1.21	1.94
$\beta_f = 4.5$:				
0.10	1.44	1.67	2.08	2.45
0.125	1.46	1.67	2.06	2.53
0.15	1.47	1.67	2.04	2.62
0.175	1.49	1.68	2.03	2.72
0.20	1.51	1.69	2.03	2.83

from eq.(9) that the term $\beta_f v_C$ in the denominator exerts a dominant role especially for the larger values of β_f. β_f itself will be seen from eq.(2) to be more influenced by v_C than by v_D due to the effect of θ in the denominator. Physically, of course, for a given c.o.v. S_C will always be greater than S_D for $\theta \geq 1$, so the greater effect of v_C than v_D on safety is understandable, if perhaps surprising, at first sight.

Parametric studies were also carried our to gain some insight into the effect on safety of different zones of operation heading and short-crested sea analysis. The findings are:

a) Operating in the northern N.Atlantic zones 1 and 2 gives rise to 5-10% higher bending loads than for mid-Atlantic zones 6 and 7; operating on a worldwide mission profile reduces bending by 5-10%.

b) Either of these two changes is about equivalent to one order of magnitude change in the probability of failure for the low-safety naval designs, and only about one-third of an order of magnitude change in p_f for high safety ships such as Tanker 12. It follows that defining the mission profile as accurately as possible is important in naval designs but is unimportant for merchant ships.

c) Using short-crested, as distinct from long-crested, analysis reduces the extreme wave bending moment by about 20% in the naval designs which is equivalent to 2 or 3 orders **in** p_f - about 1 order change in p_f might be expected for the merchant ships.

d) Taking account of the ship's heading is not important but may be so with long-cresed analysis.

These findings should be used for guidance only. Although it may be tempting to stay with the greater simplicity of long-crested analysis, most important problems in ship motion and stressing are realistically concerned with short-crested seas. Then the coherence between the ship motion and the exciting wave forces is not one, and attempting to relate motions (and stresses in particular) with point wave observations presents special difficulties.

4 ADDITIONAL REMARKS ON UNCERTAINTIES

Ref. 2 debated the topic fairly fully and so the present remarks are confined to new data and analyses.

4.1 Wave Loading

Wave height. Section 2.2 referred to Fig. 5 which compares seven measured wave spectra for severe seas from the N. Atlantic with the ISSC two-parameter spectrum. (Mathematical spectral formulations are nevertheless required because of the limited availability of such wave data). Defining "bias" as the measured means ÷ the theoretical spectrum a statistical analysis of the measured data of interest for the naval designs yields:

$\frac{\omega_1}{2\pi}$	0.8	0.9	1.0	1.1	1.2
Bias	1.35	0.92	0.83	0.81	0.91
cov %	36	22	38	27	39

Over this range the average bias = 0.96 and cov = 30.5%. Although these data are measured and therefore might be regarded "objective" they nevertheless cannot be used with any certainty. We may assume the non-dimensional plotting ensures that a single line eliminates variations in the spectra resulting from small differences in $H_{1/3}$ and T_1. Hence the difference between each of the seven spectral shapes and the single mean theoretical spectrum represents actual variations in shape(27) and would naturally lead to a scatter in the response spectra as well as in the RMS values. Even though the mean spectrum is not appreciably different from the theoretical line (except in the mid range), this fact is of little significance in relation to the suitability of the theoretical spectrum to represent sea conditions of this severity. It would be wrong therefore to assume an additional random uncertainty of 30.5% and it was a matter of judgement that about half this value (15%) was taken as the subjective uncertainty for the naval designs as in section 2.3 leading to a total uncertainty of at least 17%, and very likely nearer 20%(2) for extreme loads.

Because there are fewer larger waves it is likely that this uncertainty will be reduced for longer ships, and this seems to be confirmed by Figs. 49-51 of Ref. 27 which present short-term bending moment responses in terms of mean RMS and standard deviations for three ships. From this data the cov's are:

Ship	L(m)	cov %
WOLVERNINE STATE	151	25
SEA-LAND SL-7	268	18
UNIVERSE IRELAND	330	12

The WOLVERINE STATE is small (but marginally larger than the naval designs), the SL-7 is a fast large container ship, and the UNIVERSE IRELAND is a very large tanker. Assuming, as a matter of judgement, a subjective uncertainty of one half times 20% say (= 10%) and "adding in" the objective uncertainty of 11-12% gives a total v_D of 15-16% for extreme loads.

Length of records. The significance of this is discussed in Ref. 7 (and no doubt also at this Symposium). It can be seen from the above table that whilst the average bias (systematic error) is negligible, this is largely because of the one high value at $\omega_1/2\pi = 0.8$. For the other values a systematic error of 10-20% would appear reasonable, and is in line with the relatively short period over which the data were collected. When using Ref. 21 (7 year data collection) directly a systematic error of about this same order should be considered for fully operated ships over 20-25 year lives. However, it is best to allow for this explicitly in long-term analyses(7).

Ship speed and form. Fig. 6 from Ref. 14 shows the results from experiments with a large tanker and destroyer models at realistically low heights for the wave heights prevailing. The closed circles are the results from experiments at unrealistically high speeds.

It will be seen that speed appreciably increases the wave-induced sagging

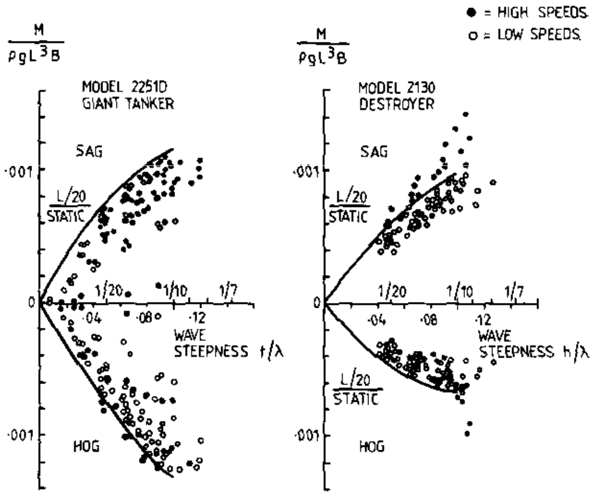

Fig.6. Experimental wave bending moment results from SSC-156

moment for the destroyer form, and does cause some increase in hogging stress in both the tanker and destroyer. These are certainly dynamic effects probably caused by slamming, and will be discussed in the next section.

The full lines drawn in Fig. 6 are upper bounds for the open circles which are the realistic wave-encounter responses for the models. They illustrate the non-linearity of bending response with wave height, which is thought to be mostly attributable to ships form and in which case statically derived form corrections can be applied to ship theory bending response, as described earlier. The reduction in wave hogging moments is quite noticeable for the destroyer and suggests there may be a limiting upper value (such as upper deck edge effects). Non-linearity wave moment uncertainties have also been detected in recent sea trials(33).

Full-scale measurements and slamming. Ref. 2 referred to measured strains in naval designs following the lines of that first reported 15 years ago(15). The reply to the discussion(2) gave the latest revised naval data and demonstrated three things:

a) Linear strip theory generally but not always overpredicted bending moments - by typically 30% - and an example is illustrated for naval design B in Fig. 4.

b) But there was no consistency in this and the "scatter" if considered as random had a cov of around 20%.

c) The revised data applied to Fig. 2 of Ref. 2 would have provided ample justification for an effective wave height approach to extreme wave moments in naval designs given by:

$$H_e = L/10 \text{ for } L \leq 100 \text{ m}$$
$$= 10 \text{ m for } L > 10 \text{ m}$$

Some of the large discrepancy in (a) can be attributed to the notional nature of the environmental data used for the comparisons (the ships' logs were not used because they are incomplete). Designs D and A ran into particularly stormy weather which is reflected in their high measured wave induced moments giving rise to stress amplitudes (123 and 211 N/mm^2 respectively), much higher than even the highest range of stress ever recorded on merchant ships. There are also substantial doubts as to how best to convert measured strains into moments in naval ships because of structural slenderness, section modulus, and other uncertainties. These are some of the reasons why prediction of bending moments and comparisons with theory are generally much less satisfactory than, for example, comparisons of ship motions(22,23). Indeed the whole situation for naval ships seems to be far from satisfactory, although the gaps in our knowledge are slowly closing. For merchant ships the situation is a little better(33).

Of greater interest, perhaps, are the recent measurements of slam-induced stresses since this still remains the biggest unknown for primary bending in fast ships having full forward form and/or bow flare. The early measurements on a dry-cargo ship(23) have recently been augmented by those in a fast container ship(34) and in two naval frigates(22). These trials have shown that slam-induced vibratory stresses can indeed coincide with the maximum wave-encounter sagging moments as was feared. Their magnitude in naval design C(22) is about 50 N/mm^2 (3.2 tsi) compression in the upper deck. It has been estimated that this addition to wave encounter stress would reduce the overall PSF based on a lifetime D_k and 5% strength for design C by about 0.5 which is equivalent to at least two orders of magnitude in probability of failure terms. Of course the wave encounter stresses were not in either ship anywhere near their extremal values when these slams occurred, but it is known that damage has occurred in merchant and naval ships. New theoretical methods are emerging, for example, the UCL modal analysis(31), which should lead to joint probability functions for combining wave-encounter and slamming effects. What the captain does is another matter, but the designer should attempt to define and cater for the worst realistic or credible operational extreme combination.

4.2 Hull Girder Strength

Whilst there are many random errors in the variables contributing to strength the writer believes there are systematic errors which in most cases are largely ignored, but which are very significant and rather more important than random errors. Indeed, it is quite likely that were it not for the presence of these systematic errors (most of them estimated conservatively) there would have been several upper deck failures in naval designs. (The recent failures in upper decks of certain container ships arise for quite different reasons which have little to do with overall safety and will not be discussed). Therefore, as indicated in section 2.2 it is quite inadequate to use eq.(7) which was adopted for ultimate bending strength for merchant ships. The suggested strength formulations are:

$$\bar{M}_u = \phi_y \sigma_{yn} Z \alpha_y \alpha_{cd} \alpha_s \quad (10)$$

where
$$\left.\begin{aligned} \phi_y &\simeq 1 - \alpha_y + \alpha_y \phi_n \\ \alpha_y &= \bar{\sigma}_y / \sigma_{yn} \end{aligned}\right\} \quad (11)$$

and
$$\left.\begin{aligned} \sigma_{cd} &= \sigma_{c1} + \zeta_{c2} - \zeta_{c3} \\ \zeta_{c2} &= 2\zeta_{c2o}(1 - \phi_n) \end{aligned}\right\} \quad (12)$$

Defining the terms, $\phi_y = \bar{\sigma}_{uy}/\bar{\sigma}_y$ the compression strength parameter corrected for the expected systematic error in yield strength, and σ_{yn} is the nominal acceptance value of yield strength, and $\phi_n = \sigma_{un}/\sigma_{yn}$ is the associated nominal compression strength parameter using the nominal yield strength; σ_{cd} is a systematic correction for compression design codes having up to three components which are $\sigma_{c1} \geq 1$ a constant value, ζ_{c2} $(=\alpha_{c2}-1)$ a positive addition factor varying with structural slenderness, and ζ_{c3} $(=\alpha_{c3}-1)$ a positive reduction factor varying with structural slenderness; α_s allows for redistribution through the ship cross-section following initial collapse of the weakest gross-panel in the section(18,35, 36) - it was 1.13 for the naval design with the lowest residual or reserve strength, and was sometimes much more. Typical conservative values assumed for these designs are:

$$\alpha_y = 1.1\ ,\ \alpha_{c1} = 1.05\ ,\ \zeta_{c2o} = 0.1\ ,$$
$$\zeta_{c3} = 0\ ,\ \alpha_s = 1.15$$

The three components of α_{cd} are represented in Fig. 7 as a plot against the nominal slenderness parameter and they will now be explained more fully. The derivation of eq.(11) and (12) is outlined in Ref. 2 (and its discussion p 25).

A typical example of α_{c1} is the use of lower-bound buckling curves. For

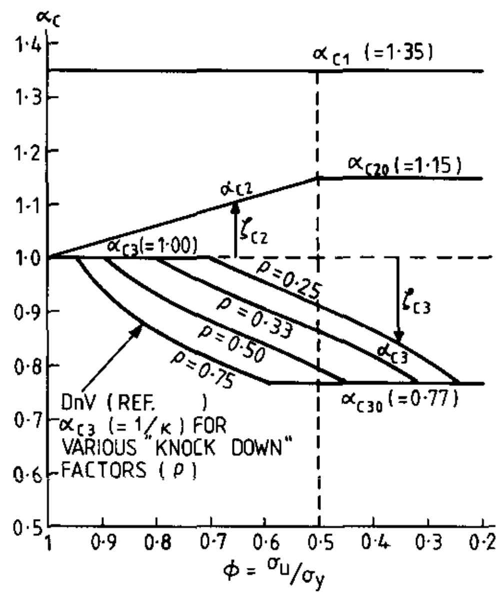

Fig. 7. Types of systematic errors in buckling codes

example, the value of 1.35 shown in Fig.7 corresponds to the lower bound correction when using statistical methods for assessing the safety of pressurised end closures(37) - see also Fig. 8. If owners and designers are serious about less conservative more efficient structural design then in many buckling determined structures there is attractive scope for guaranteeing something above lower bound values by way of control of imperfections once appropriate numerical techniques have been established and backed up by experiments to cover the likely range of imperfections.

α_{c2} (or ζ_{c2}) is a variant of α_{c1} to allow some correction for the unnecessary pessimism implicit in design formulations for certain slender structures over the range of nominal ϕ = 0.5 to 1.0 (where α_{c2} must be 1.0 to avoid $\bar{\sigma}_u$ exceeding $\bar{\sigma}_y$). This corresponds to a slenderness range of $\lambda = \sqrt{2}$ to zero. The use of single-bay test data for compression strength is an example of such an error, and analysis of actual grillage test data when compared with such widely used formulations suggests ζ_{c2o} lies in the range 0.1 to 0.2 and is often closer to 0.2.

A good example of α_{c3} (or ζ_{c3}) is the slenderness safety coefficient κ in certain offshore codes(12). See eq.(13) later. It varies from 1.0 to 1.3 for shell structures and is shown for a typical range of knock-down factors in Fig. 7. It is intuitively attractive, perhaps, to cater for the greater sensitivity to shape imperfections expected in slender structures. But its use has been challenged when using a lower-bound approach to design(37) and recent numerical studies(38) not only support this contention but go further by suggesting that the effects of practical shape imperfections, as given for example in codes, are negligible at the slender end but still significant at the stocky end of the slenderness scale. This is a reversal of the effect of κ which has also recently been illustrated in ring stiffened cylinder structures(13) where it has been suggested κ should not be used, that is, take $\alpha_{c3} = 1.0$. The use in the same offshore codes of the post-buckling factor Ψ (an inverse α_{c1}) has also been challenged(13) and a re-appraisal of all safety coefficients is recommended. This reference also examines the use of the so-called reduced slnderness parameter $\bar{\lambda}$ and other formulations for inelastic effects in stiffened shell structures since most structures fail elasto-plastically.

To illustrate the range of slenderness parameters for the deck structures examined in the five naval designs, the following results may be of interest:

$$A_s/bt = 0.11 \text{ to } 0.32$$

$$\beta = (b/t)\sqrt{\sigma_{yn}/E} = 1.93 \text{ to } 4.03$$

$$\lambda = (\ell/\pi r_c)\sqrt{\sigma_{yn}/E} = 0.40 \text{ to } 0.85$$

$$\sqrt{E/\sigma_{yn}} = 26 \text{ to } 30$$

This gives a nominal range of ϕ_n from 0.4 to 0.64, with corrected values used for assessing the safety parameters from 0.54 to 0.74(2). The range of slendernesses in the merchant ships is much smaller (ϕ close to 1.0) and could clearly be relaxed in many places in view of the more than adequate safety incorporated. This would save construction costs.

5 COMPARISONS WITH OTHER MARINE STRUCTURES

Most of the impetus for semi-probabilistic methods for marine structures has been provided by offshore codes in conjunction with the so-called limit state approach to design. A version of the safety eq.(5) in one-dimensional form is(12):

$$\gamma_f D_k \leq \frac{C_k \Psi}{\gamma_m \kappa} \qquad (13)$$

On the basis of the bias ratio for the 100-year wave to the annual wave being 1.3 this value is frequently taken to be the value for γ_f for extreme environmental loads(39,12). For say a 25-year life with mean annual wave encounters of 5×10^6 then $n \simeq 125 \times 10^6$ lifetime encounters and from Ref. 39:

$$\text{Bias} = 1 + \log 100/\log n = 1.25$$

This seems somewhat high compared with values nearer to 1.1 for more recent designs and so a mean value of 1.18 will be assumed. Taking $v_D = 0.30$ (values between 0.25 and 0.35 have recently been suggested following the work of Kim and others used in Ref. 40) and assuming the above analysis approximately refers to the 5% characteristic extreme environmental load, it follows for normal distributions that the mean load:

$$\bar{D} = D_k/(1 + 1.645\, v_D) = 0.67\, D_k$$

For moderately slender compression members a value of 1.24 has been suggested (39) for $\bar{C}_u/C_k$ which, as expected, is slightly lower than $\gamma_m\, \alpha_c = 1.15 \times 1.1 = 1.27$ for yielding. Hence it follows that the central safety factor is approximately given by $\theta = (1.24 \times 1.18)/0.67 = 2.18$.

Taking $v_C = 0.17$ as in (39) and $v_D = 0.30$ as above, and assuming normal distributions, it follows that the safety index and 5% overall PSF are approximately:

$$\beta_f = 2.5 \;,\; \gamma_o = 1.6$$

A range of alternative approximations gave $\theta = 2.0$ to 2.2 and provided values of $\beta_f = 2.3$ to 2.8 and γ_o 1.5 to 1.8.

Whatever the assumptions, it will be seen that these safety indices are appreciably lower than for the upper deck of any of the 18 merchant ships and fall in the middle of the naval design band in Fig. 3. On the other hand, the overall partial safety factor range is greater than that for the naval designs, and slightly greater than for some of the merchant ships. This apparent anomaly, of course, arises because of the appreciably higher covs in offshore structures, which, as judged some years ago, are about twice those applicable to extreme loads and response in ships(41). The results also lie in a zone where a preference for the safety index or the PSF approach is not clear, and in view of the greater acceptance of the latter to most engineers it is naturally adopted.

Similar studies for submersibles(37) show that end closure safety indices range from 3.4 to 3.8 with overall 5% PSFs ranging from 1.7 to 1.9. The ring stiffened cylindrical pressure hull of submarines also have similar safety levels. Two things are noted also from the reference:

a) the relatively high safety indices arise in particular because of the low standard deviation of the load, which is closely controlled in operation

b) for the ring stiffened elements the concept of "guaranteed strength" is acceptable (with a safety factor of 1.5), but as can be seen from Fig. 8 there is a significant probability that tests on a practical structure may fall below any lower bound curve drawn as a result of test data from a relatively small sample.

For these reasons a PSF or even a safety factor approach to design is entirely adequate. Direct comparison with surface ships results is again not so easy, but in this case is due to the low covs pertaining.

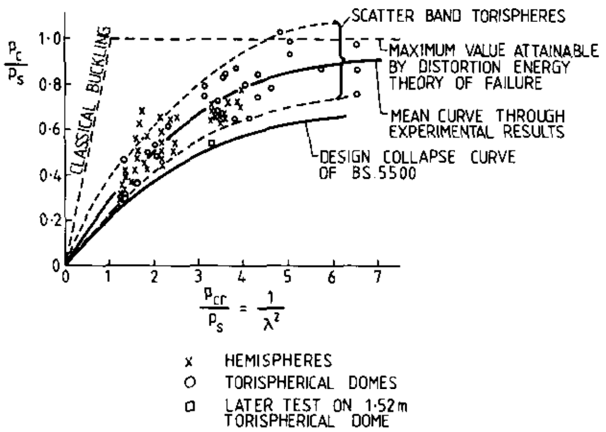

Fig. 8. Results of collapse tests on domes

6 REDESIGNED SHIPS

It is not of course practical to redesign in detail the hull cross-section, but it is possible to re-evaluate a section modulus Z to the requirements recommended in section 2.4. Subscripts r will be used to denote the revised (and reduced) values for the merchant ships 12 and 13. The loading will be assumed to be unchanged, but the ultimate moment will be reduced as shown by the dotted line in fig. 1.

Using the simple strength model eq.(7) it follows from $\bar{D}_e$ remaining unchanged that the reduction factor r is:

$$r = \frac{Z_r}{Z} = \frac{\bar{C}_{ur}}{\bar{C}_u} = \frac{\theta_r}{\theta} \qquad (14)$$

If we assume that v_D and v_C remain unchanged (the latter being determined largely by yield strength variation), then we can determine θ_r from the revised safety index β_f using eq.(9) and hence r from (14). The revised γ_{or} is then given by eq.(8). This has been evaluated for ships 12 and 13.

Ship 12 is a very safe 92,600 tonnes displacement tanker of LBP = 236 m studied extensively. Her safety parameters are β_f = 6.03, 5% γ_o = 3.24 and θ = 4.94. With the same cov assumptions as before the reduced section modulus and parameters for the safety indices recommended in section 3 are:

β_{fr}	γ_{or}	θ_r	r
4.5	1.73	2.63	0.53
3.0	1.17	1.78	0.36

If 5% PSF is used as a design criteria the corresponding values are:

γ_o	β_f	θ_r	r
2.0	4.95	3.05	0.62
1.5	4.01	2.29	0.46

Of course, some of the structural steel weight does not contribute to overall strength and so the potential for weight saving may be less than is indicated by (1 - r). But even taking only two thirds of this value it can be seen that this design has the potential for 30-40% weight saving, which justifies the more modest "20-30% or perhaps more" claim made eight years ago(42). The steel mass for this design would be around 12,300 tonnes, so taking say 35% of this represents a saving of 4,300 tonnes. Presumably the saving in money would also be significant and be measured in millions of dollars.

When this tanker was designed in 1964 the under deck flat bar longitudinals had d/t ratios of 21 which would only ensure compressive strength in the range 0.72 to 0.79 of yield. In 1975 the rules required ratios not more than 16.5, and nowadays this is even smaller to give compression tripping strengths close to yield. Thus over little more than a decade such upper decks have increased in compression strength by perhaps as much as 20-25%. The reduction in required section modulus over this same period has not even matched this demonstrable increase in strength, much less taken advantage of the greater knowledge acquired.

The oil-ore carrier ship 13 was analysed in a similar way, being selected from the middle range of safety of Fig. 3. The potential saving in this case (on the same basis) is over 20%, which again is quite significant. There is scope for weight saving in the naval designs by closing down the spacing of longitudinal stiffeners, but of course the construction cost in this case would certainly increase.

Stress levels would increase inversely with weight saving, and so greater attention to structural details would be advisable to minimise fatigue damage. The cost involved in achieving this would be a small percentage of the overall saving. Much more highly stressed naval ships have little or no fatigue problems.

Finally, although construction cost alone has been mentioned, when savings in weight of the order suggested above are possible then there should also be scope for appreciable saving in fuel bills throughout the ship's life, as suggested recently(43).

7 SUMMARY AND SUGGESTIONS

I have avoided the word recommendation in the heading of this closing section because the important ones have already been made in section 2.4 and their implications are illustrated in section 6. It is therefore merely a suggestion that, if ship owners are seriously interested in reducing total costs, they examine the scope for this carefully in the light of this study. In this respect they may wish to instigate studies of their own, hopefully involving the Classification Societies who otherwise seem to have very little motivation to act independently. The more important broad conclusions and suggestions are now summarised based on extreme lifetime load conditions:

1) The safety levels in merchant and naval ships lie respectively in two distinct zones:

   I over-safe
   II low-safety

   The mean values for merchant ships are β_f = 5.3 and γ_o = 2.4 whereas the corresponding values for naval designs are 2.2 and 1.1 for the safety index and the 5% PSF; and yet there is greater uncertainty in both loading and strength of naval ships

2) The scatter in results is vast and ships with high conventional safety factors can nevertheless be the least safe

3) Deck and bottom structure safety levels vary considerably, and certain transversely framed bottom structures permitted within existing rules could be in a state of near collapse before leaving harbour

4) Unifying the approach to strength and adopting β_f values in the range 3.0 to 4.5 (5% γ_o 1.2 to 1.7 approximately) would provide scope for structural weight savings of 20-40% whilst retaining more than adequate safety against jack-knifing

5) The safety parameters and especially the overall PSF γ_o are very much more sensitive to variability in strength than to variability in load

6) However, the cov for extreme load is generally underestimated and values of 15% for most merchant ships, and perhaps 20% for naval ships, are more realistic than those used in previous studies

7) The ultimate bending strength model adopted in merchant ships is very crude and is only acceptable because the hull is over-designed; by contrast much more sophisticated modelling is necessary when considering safety of naval ships

8) The safety levels in offshore structures are typically in the ranges β_f = 2.3 to 2.8 and γ_o 1.5 to 1.8, with load and strength covs of about 30% and 17% respectively - noticeably higher than those for ships; and incidentally offshore codes are changing quite rapidly

9) The use of "nominal" values, e.g. for yield strength, presents their own problems, as discussed, and can mask the true safety picture unless such systematic errors are accounted for

10) Errors arising from different assumptions and calculation approaches can be very significant, and any sound reliability approach should attempt to identify and take some account of these errors.

The earlier of these findings are hardly creditable and the author suggests they stem from an over-concern in ship classification with section modulus and stress analysis. This could with benefit be replaced by limit strength analyses married to a soundly based semi-probabilistic level-2 safety approach. It does not seem unreasonable with present knowledge to do this now, based on sensible comparisons. Otherwise reliability research might just as well stop as far as ships are concerned.

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Appendix A - Extreme Mean and Variance

The objective mean and standard deviation of the extreme wave histogram in Fig. 4 are obtained from the usual statistical equations for a true population. To facilitate understanding, this population is taken as N = 100 so the number of values in the histogram intervals, designated by x_i $(= M_w)$, is simply $n_i = 100\,p$. Then see (24, 25):

$$\bar{x} = \sum_{i=1}^{N} n_i x_i / N \quad ,$$

$$s^2 = \{ \sum_{i=1}^{N} x_i^2 - N\bar{x}^2 \}/N + (\Delta x)^2/12$$

x_i $(= M_w)$	n_i	First Moment $n_i x_i$	Second Moment $n_i x_i^2$
205	8	1 640	336 200
215	22	4 730	1 016 950
224	27	6 075	1 366 875
235	19	4 465	1 049 275
245	12	2 940	720 300
255	6	1 530	390 150
265	4	1 060	280 900
275	2	550	151 250
	100	22 990	5 311 900

Thus $\bar{x}$ = 22 990/100 = 229.9 MNm

$S^2 = \{5\ 311\ 900 - 100\ (229.9)^2\}/100 + (10)^2/12 = 273.3$

S = 16.5 MNm

The word objective is stressed because the population is exactly calculable within the constraints of the theory adopted. (Indeed, it cannot be sampled by direct measurement and a continuous distribution curve would be more appropriate but possibly less easily understood). For this reason it is appropriate to use N in the denominator for S^2 and not (N - 1) as would be more appropriate for a sampled population of readings. The notion of a finite sample size is, as stated, a fiction in this example and the actual probabilities p = 0.08 , 0.22 .... would usually be used directly in the second column of the table with 1 in the denominator for s^2.

Appendix B - WOLVERINE STATE Safety

Using an approximation based on Ship Department (RN) procedures, the most probable extreme wave bending moments for WOLVERINE STATE with a N. Atlantic service life are:

Sagging M_w = 247,270 tonf ft

Hogging M_w = 197,810 tonf ft

The still water moment is hogging and varies from 157,500 tonf ft in the light condition to 40,000 tonf ft in the laden condition(28). The average is:

Hogging $\bar{M}_s$ = 98,750 tonf ft

The mean extreme total load is then taken as:

$$\bar{D}_e = 1.033\ M_w + \bar{M}_s$$

which takes account of the skewness of the extreme wave loading distribution(2). With deck Z = 41,300 in^2ft and bottom Z = 43,160 in^2ft the mean extreme compression loads in the sagging and hogging conditions are (in stress units and allowing 3% increase to cover loss of Z due to corrosion, rolling tolerances, etc.):

Deck $\bar{D}_e$ = 3.91 tsi

Bottom $\bar{D}_e$ = 7.23 tsi

Had we taken the worst total moments using the laden still water moment for the sagging condition and the light still water for hog, these compression results would have been $\bar{D}_e$ = 5.38 tsi in the deck and 8.63 tsi in the bottom. Thus, assumptions about the value of the still water moment are clearly important and can appreciably affect the notional safety measures for merchant ships.

Deck Safety

The panels are 276 in. wide, 30 in. long and 1.06 in. thick (but 1.03 in. used as just explained). Two assumptions will be used to illustrate the different results which can easily arise. Using Ref. 29 and applying a factor of 1.14 suggested by an analysis of the experiments:

$$\sigma_{cr} = \frac{0.175\ \pi^2 E}{12(1-\nu^2)} \left\{\frac{a}{b}+\frac{b}{a}\right\}^{1.25} \left\{\frac{t}{a}\right\}^{1.5}$$

$$= 7.86 \text{ tsi}$$

$$\bar{\sigma}_u = 1.14 \times \sigma_{cr} = 8.95 \text{ tsi} = \bar{C}_u$$

Then $\theta = 8.95/3.91 = 2.29$ and with $v_C = 0.13$ and $v_D = 0.12$ this leads to:

Safety index $\beta_f = 4.0$

5% PSF $\gamma_o = 1.5$

The presence of deep longitudinals would also add to the strength in providing hard "corners", and if this were allowed for as a 1.15 systematic addition to the strength the above results would have increased to $\theta = 2.63$, $\beta_f = 4.5$ and $\gamma_o = 1.7$.

But, although Ref. 29 is drawn from a comprehensive range of tests, unfortunately the test rig applied uniform stress rather than uniform strain. Therefore, the results may be expected to be conservative and eq.(8) of Ref. 30 was also used for compressive strength. With the geometry specified above this gives $\sigma_u = 13.8$ tsi and hence central safety factor $\theta = 3.53$ and

$$\beta_f = \underline{5.3}\ ,\quad \gamma_o = \underline{2.3}$$

Of course these would be higher values if the 1.15 factor was included, but they would not then be so directly comparable with the deck safety for the 16 ships in section 2.1 of the paper. The above values are identical with the mean values of those in 2.1 - quite fortuitously.

Outer Bottom Safety

The panels are assumed to be 96 in. wide, 30 in. long and 0.78 in. thick (but 0.76 is taken). Using Ref. 29 and ignoring pressure effects for the moment gives $\sigma_{cr} = 7.18$ tsi, $\bar{\sigma}_u = 8.19$ tsi and $\theta = 1.13$, hence:

$$\beta_f = 0.7\ ,\quad \gamma_o = 0.7$$

which is close to the lowest value $\beta_f = 0.5$ estimated previously(2). Including the effects of extreme water pressure with a $1.1\sqrt{L}$ wave gives a pressure $\simeq$ 15 psi. Hence, $pb^4/Et^4 = 1.21$. From Ref. 44 the effect of pressures of magnitude $pb^4/Et^4 = 2$ on plates of aspect ratio 3 and $b/t = 50$ was to increase the weak direction collapse stress by 20-30%. Taking the lower value $\bar{\sigma}_u = 9.83$ tsi and $\theta = 1.36$ hence

$$\beta_f = 1.7\ ,\quad \gamma_o = 0.9$$

If, however, we use the methods of Ref. 30 then ignoring pressure $\sigma_u = 10.66$ psi, $\theta = 1.47$ and hence:

$$\beta_f = \underline{2.1}\ ,\quad \gamma_o = \underline{1.0}$$

The Reference makes no allowance for strengthening effects from pressure. Indeed, it suggests when pressure is present that the plate elements may be weaker when compressed in the weak direction. But, this would be offset by the strengthening effects from the deep longitudinals mentioned earlier, and so pressure effects are ignored and the underlined values for safety are considered to be the most reliable. However, the sensitivity of the results to design calculation assumptions should be noticed for this low safety structure.

Calculation Assumptions

The above analyses indicate the sensitivity of safety measures to design assumptions. Likewise, much of these analyses can be compared directly with results from Ref. 28 and it would be found that there are significant differences throughout the results depending upon assumptions made. As an example, as recently as ten years ago it was customary in some rules to include continuous longitudinal deck girders in the section modulus, whereas it would seem unreasonable to do so now. Even in this transversely framed ship this addition alone adds 7% to the strength predictions. Systematic differences of 20-30% are by no means unusual even between competent analysts and designers. This should be borne in mind when finalising safety measures.

Ref. 28 is also of interest in showing how certain upper deck section modulus requirements have actually increased for this class of ship since it was designed. This apparently haphazard approach to safety seems difficult to justify in the light of the other inconsistencies mentioned, for example, in certain bottom structure requirements.