It is our objective here to obtain some idea as to the probability that, if a slam occurs, the amplitude of total stress (or equivalent bending moment) will be increased over the wave bending stress by either slamming or whipping. The above probability is to be evaluated at the midship section, because the problem of local stress is not included in our immediate objective. We note that it is not our intention to give a probability analysis of the occurrence of slamming, this having been treated elsewhere, as discussed in Chapter V.

PRELIMINARY ANALYSIS

It was decided to make use of available stress records of the S.S. Wolverine State. The choice of records to be used was made on the basis of the following criteria:

a. approximately the same environmental (sea) conditions,
b. clear stress traces in which the slam was well defined,
c. approximately the same ship speed.

This selection was of critical importance to the analysis, and the consistent results indicate that a good choice of records was made (103).

This analysis was then combined with another similar recent analysis (2) to obtain as complete a statistical sample as possible, comprising approximately 65 data points.

From the selected records the following were obtained:

(a) histograms of phase of slam inception relative to peak wave bending stress (tensile in deck, i.e., hogging condition). (See Fig. 12),

(b) histograms of slam stress amplitudes (See Fig. 13),

(c) histograms of whipping stress amplitudes (See Fig. 14),

(d) empirical solution of the damping of the high frequency (two-noded) stresses. (A representative time trace is given in Fig. 15).

In item (d) it was postulated that the damping is dependent on two frequencies, but only the 2-noded mode was considered; all other modes were lumped.

The empirical solution of the damping function g(t) was:

$$g(t) = a_1 \; e^{-c_1 \omega_s t} + a_2 \; e^{-c_2 \omega_s t}$$

where ω_s = frequency of two-noded hull girder vibration = 10.389 $\mathrm{sec.}^{-1}$

c_1 = damping constant for two-noded mode = 0.00513

c_2 = damping constant for all other modes = 0.1540

a_1 = 0.8

a_2 0.2

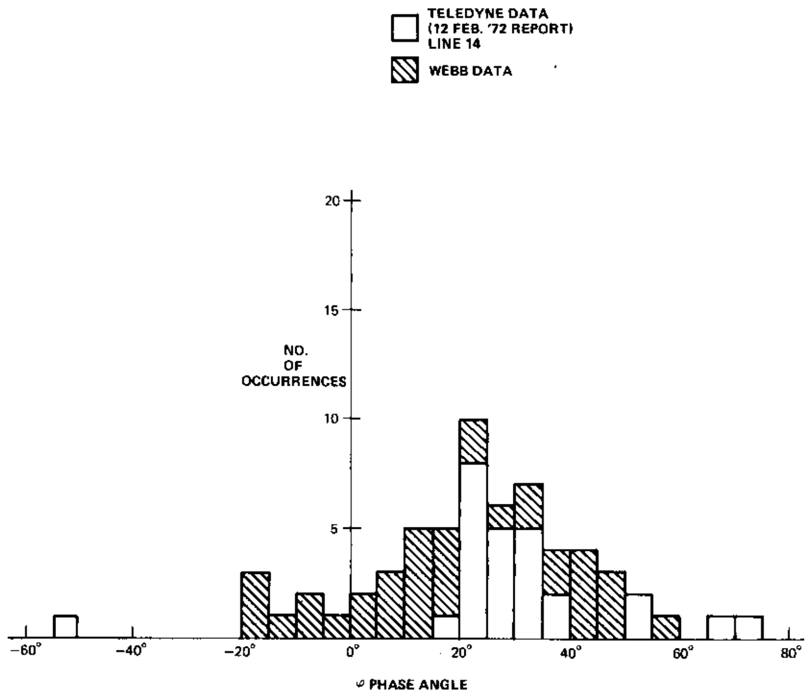

FIGURE 12 - Distribution of Slam Phase Angles,
S. S. WOLVERINE STATE

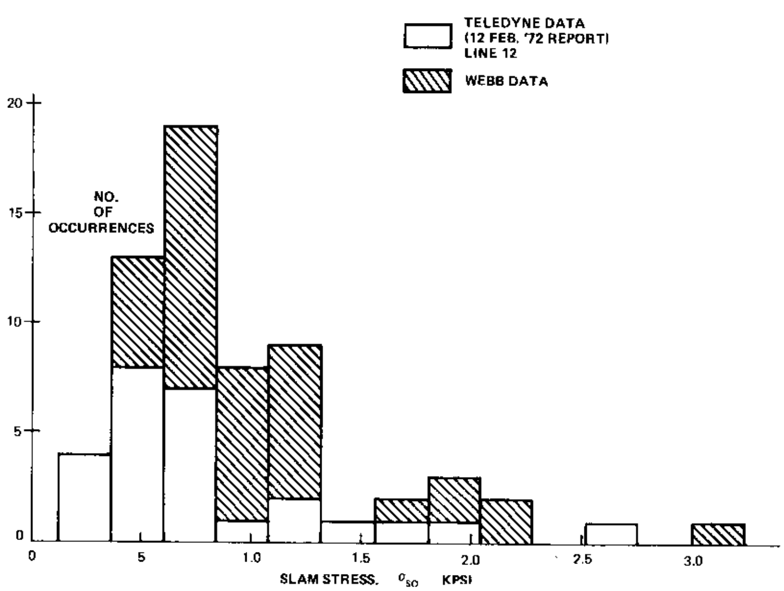

FIGURE 13 - Distribution of Slam Stress,
S. S. WOLVERINE STATE

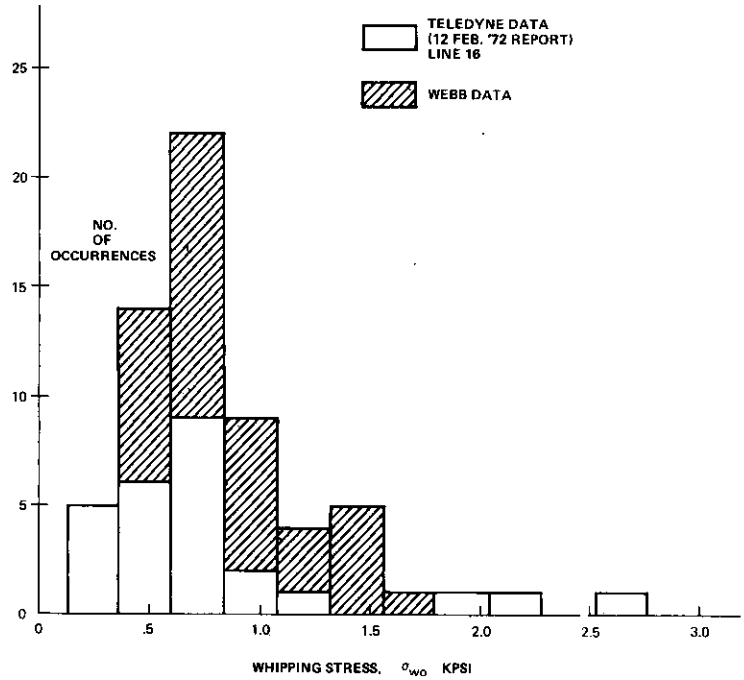

FIGURE 14 - Distribution of Whipping Stress, *S. S. WOLVERINE STATE*

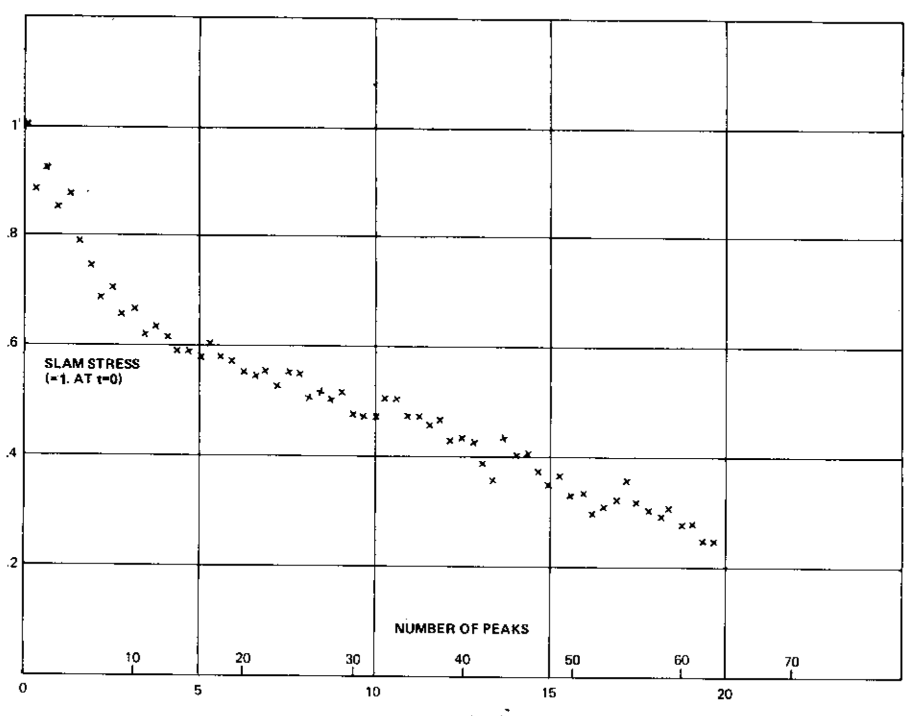

FIGURE 15 - A Typical Decay Curve of Whipping Stress

DISCUSSION

From the histograms and figures the following can be deduced:

1. The phase angle seems to be normally distributed with a range of $-56° \leq \phi \leq +72°$, relative to t = 0 (peak hogging wave stress). This implies that the phasing is a random Gaussian process, which is a reasonable assumption since slamming occurs only if certain conditions are satisfied, namely bow out of water, sufficiently high impact velocity, etc.

2. The slam stress seems to follow a distribution of a type with lower limit $\neq 0$. It has been stated (103) that the type of distribution is exponential. Although this is not entirely confirmed by our data (Fig. 13), we followed the findings in the above-mentioned report because the sample reported there was much larger than the sample of 65 we were able to obtain for this particular study.

3. The histogram for whipping stress is more regular in shape, and we notice that the dispersion is far less than in the case of slamming. Since the frequency of the vibrations associated with whipping is relatively high, we may directly superimpose the whipping stress as presented in the histogram on the compressive wave stress. Therefore, the mean increase over the wave stress is also the mean of the whipping stress, namely $\bar{\sigma}_w = 0.876$ KPSI, in this case. See Fig. 14. This is not the case with the slamming stress which we will consider next.

4. The histogram of that part of the slam stress which is additive to the maximum tensile stress shows two things (Fig. 16):

(a) The distribution of addition and non-addition is approximately 50 - 50%. Therefore, of all the slams the mean stress increase over the wave bending stress is approximately 0.

(b) If we take that half of the number of cases in which the slam stress does increase the total stress at midship, then the mean increase is, $\bar{\sigma}_s = 0.13$ KPSI, which is indeed very small compared to the mean increase due to whipping.

We may therefore conclude that whipping is of relatively greater importance than slamming for this ship.

5. The evaluation of this sample would not be complete without consideration being given to the separate relationships of dynamic stresses to the tensile and compressive wave-induced deck stresses. We observed that the slam stresses are associated with the tensile wave-induced deck stresses, and high whipping stresses are associated with the compressive stresses. Therefore it is useful to define the following ratios (see Fig. 11):

$$r_s = \frac{\sigma_{so}}{\sigma_{ho}}$$

$$r_w = \frac{\sigma_{wo}}{\sigma_{ho}}$$

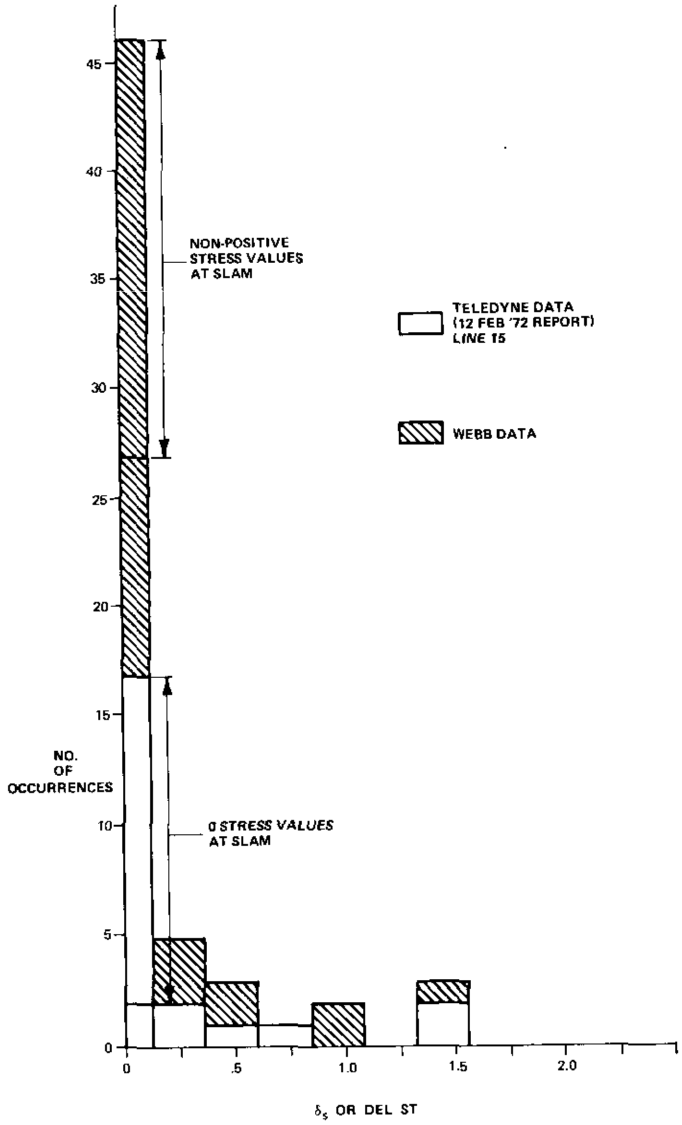

FIGURE 16 - Histogram of Slam Stress Additive to Wave Stress

and consider the histograms of these two ratios (Figs. 17 and 18). Both histograms show reasonably regular distribution of the ratio, considering the limited size of the sample. Without going into an analysis of what specific distribution they fit, it is gratifying to see that the shapes are similar to the shapes of the individual slam stress and whipping stress distributions, as shown in Figs. 13 and 14..

6. Our last consideration is a study of the correlation between slam stress and tensile stress. In Fig. 19 we have plotted the slam stress σ_{so} against the tensile wave-induced stress σ_{ho}. We conclude that there is no correlation, and therefore we may consider these stresses to be independent.

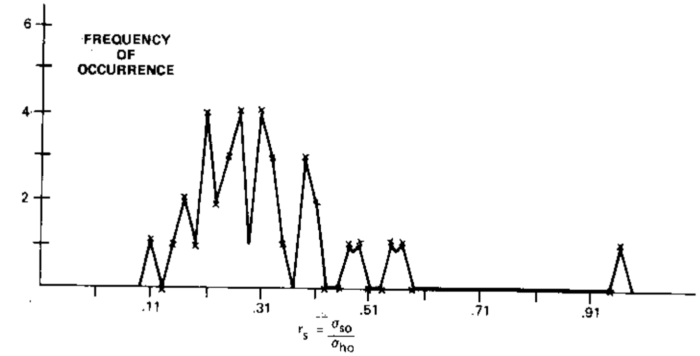

FIGURE 17 - Histogram of the Ratio of Slam Stress to Wave Bending Stress

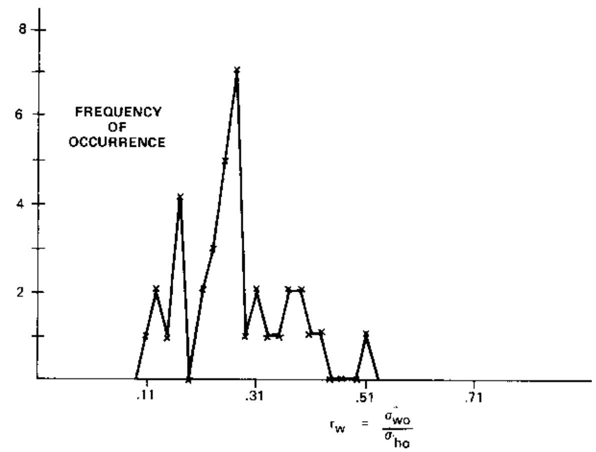

FIGURE 18 - Histogram of the Ratio of Whipping Stress to Wave Bending Stress

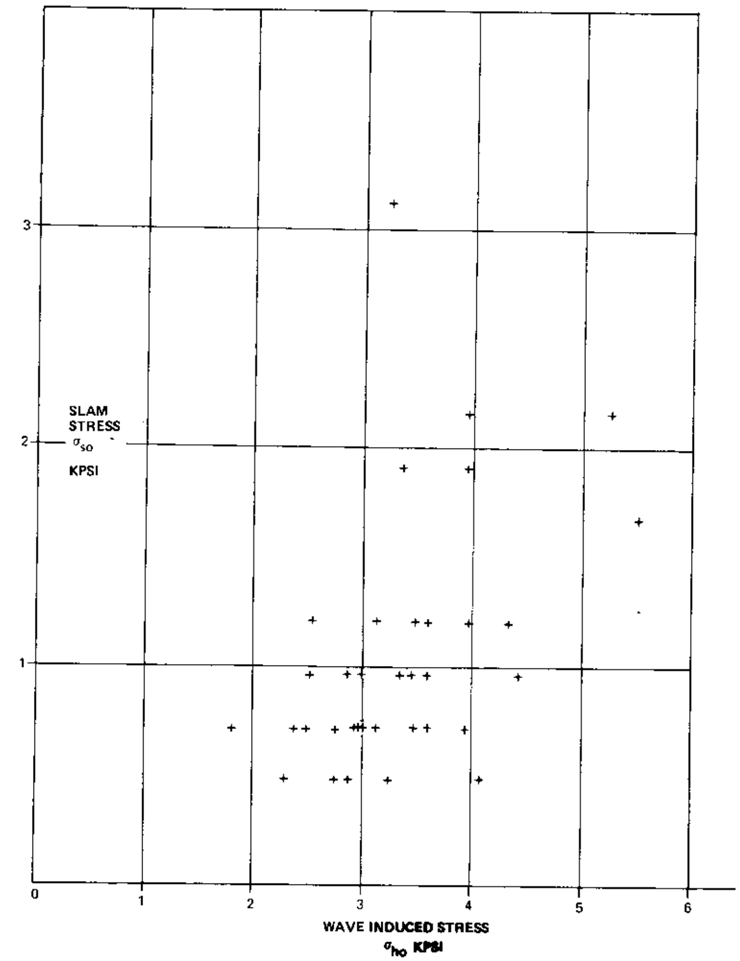

FIGURE 19 - Plot of Slam Stress vs. Wave Bending Stress

FURTHER WORK

A theoretical development has been made on the probability model from which to determine:

a) combined slam stress and wave bending stress, and

b) combined whipping stress (following the slam) and wave bending stress.

Part of this work is incorporated in Chapter IX, but a continuation of the development is recommended.

VII. THERMAL EFFECTS

INTRODUCTION

Records of midship stress obtained on five bulk carriers (3) indicated surprisingly high thermal effects. These showed a consistent diurnal variation, with magnitudes of 3-5 kpsi in some cases. The temperature gradients that produce such thermal stresses may not be, strictly speaking, loads but they are considered to be loads here nevertheless.

Although it often happened that high thermal stresses occurred at times of low wave bending stresses (sunny weather), and vice versa (stormy and cloudy weather), this was not always the case (3). The exceptions are presumably times when a heavy swell was running while the weather was clear.

It should be noted that the thermal stress changes recorded here were overall averages, since they were based on combined port and starboard readings. Because of the effect of local shading it can be expected that even larger thermal stresses would be experienced. However, it can be assumed that such local high thermal stresses can be ignored for the present purpose.

In order to include thermal effects in design calculations two distinct steps are required: estimating the magnitude of the effect under different conditions of sun exposure and estimating the frequency of occurrence of these different conditions in service.

In a discussion of (3) tanker service data were presented which showed a strong correlation between change in sea-air temperature differential and change in stress level. Theoretically there should be no difficulty in calculating one from the other. This chapter describes the application of available theory (119), assuming simplified structure and using estimated temperature changes.

The simplified procedure is applied to the tanker Esso Malaysia first, because records of the measured diurnal stress changes and some temperature data are available. It was assumed that if good agreement between prediction and measurement was found the technique could be used for numerical calculations on Wolverine State in a later chapter.

TEMPERATURE CALCULATIONS

Esso Malaysia logbook data plotted by Breves in his discussion of the Little-Lewis paper (3) indicated air-water temperature differences at noon (when they are maximum) of 10° to 13° F., averaging 11.5° F. over eight days (maximum was 16°). If constant sea water temperature is assumed over any 12 hour period (noon to midnight or midnight to noon) the average diurnal change in air temperature is also

11.5° F. Deck plating would be subjected to this change plus the change due to insolation (i.e., the absorption of radiant heat). The temperature change due to insolation depends on cloud cover and color of deck. As an approximation, Fig. 82 of Principles of Naval Architecture (120)(p. 247) shows maximum differences between air and deck (sun overhead, unshaded) for different colors of deck as follows:

black	50° F.
red	40° F.
aluminum	10° F.
white	10° F.

It was established that the color of the Esso Malaysia deck was a very dark gray, almost black; hence a figure of 40° was used plus the measured average air temperature change of 11° -- giving 51° (say 50°) for ΔT.

A simplified distribution of ΔT is assumed, namely constant over the deck and the sides down to the water line and $\Delta T = 0$ elsewhere (below WL). Longitudinals will have the same ΔT as the plating which they support. Longitudinal bulkheads and associated longitudinals have ΔT decreasing linearly from the deck ΔT at top to $\Delta T = 0$ at the assumed level of oil inside the tanks.

RESULTS OF STRESS CALCULATIONS

Under the assumed conditions the calculated thermal stress at deck edge due to temperature change is about 2000 psi. [1600 at center of deck stringer strake and 2300 at shear strake]. From the measured stresses during the same period of time (Fig. 38 of (3)) the 11 day-night or night-day stress variations in KPSI are as follows (9/18/68 to 9/26/68):

2.3, 2.3, 1.7, 1.7, 1.6, 1.5, 1.7, 1.8, 1.9, 1.8, 1.7.

The average value is 1.8 or 1800 psi.

It is concluded that the approximate calculation is satisfactory. Stresses given elsewhere (119A)(119B) are higher because they include unsymmetrical temperature gradients.

PREDICTING SUN EXPOSURE IN SERVICE

The prediction of voyage average thermal stresses and expected maxima requires also that the frequency of occurrence of different conditions of sun exposure be determined. Source data for such predictions are given in the U.S. Navy Marine Climatic Atlas of the World, Volume VIII (82). Cloudiness is represented by charts of the world's oceans showing for each month of the year:

1. Total cloudiness, with isopleths indicating,

   (a) % frequency of total cloud cover less than or equal to two-eighths,

   (b) % frequency of total cloud cover greater than or equal to five-eighths.

2. Median cloudiness, with the midpoint (50% of observations) of total cloud cover reported in eighths.

In addition, special low cloud data are given, which are not necessary for these calculations.

From the plotted data, it is possible to estimate average cloud cover for any given trade route on a monthly, seasonal or yearly basis. Cloud cover is then related to air-deck temperature difference due to insolation (the 40° F. value stated above, for example) by assuming that the insolation ΔT is directly proportional to the extent of cloud cover. Thus the 40° temperature difference would apply to full sun (cloud cover = 0/8), while total cloud cover (8/8) would indicate $\Delta T = 0$. Intermediate values are assumed to vary linearly. The resulting insolation ΔT's are added to the sea-air ΔT's to determine total ΔT for each cloud cover condition. A weighted average of total ΔT can then be calculated by combining the total ΔT's with their frequencies of occurrence as determined from the Atlas (82).

A sample calculation is shown in Chapter IX, where the method is applied to the Wolverine State.

VIII. COMBINING LOADS FOR DESIGN

GENERAL

As suggested in Chapter II, the primary load criterion is assumed here to be the maximum combined bending moment resulting from the various loads that can cause excessive deflection or failure by buckling or plastic deformation. The discussion of loads in Chapters III - VII indicates that this combined load can best be stated in probability terms, i.e., the overall combined bending moment to be exceeded once in the lifetime of a ship or of a fleet of ships. We are also interested in lesser combined values that may occur more frequently and may cause structural damage without complete failure. Hence, the next section will deal with the combining of static and dynamic loads, referred to as Ultimate Bending Loads. Load criteria for fatigue and brittle fracture will also be discussed in following sections.

ULTIMATE BENDING LOADS

Still Water Loads

The first step in calculating and combining static and quasi-static loads -- and hence determining the primary load criterion -- is to consider the still water bending moments expected in the life of the ship. Typically, as indicated in Chapter III, there will be a distinct difference between outbound and inbound loadings (in some cases there may be three typical conditions -- as for a ship engaged in a triangular service -- or even more). One case may be full load and the other ballast; or there may be different loaded conditions outbound and inbound. However, for each case there will be a variation of bending moment from voyage to voyage, depending on density and distribution of cargo and/or ballast. The object is to estimate the mean and standard deviation of outbound and inbound bending moments over many voyages. On long voyages there may be significant variations in bending moment as fuel is consumed and salt water ballast is added, and therefore a distinction may be necessary between departure and arrival conditions. Hence, probability density curves, showing the probability of different levels of still water bending moment, can be estimated for both outbound and inbound loadings.

Having made the above estimate of separate outbound and inbound still water loadings, it is essential to carry through the remainder of the load calculations separately for each case, since the basic differences in drafts and load distribution will affect wave bending moment, probability of flare immersion, etc.

An estimate should also be made of the bending moment caused by the ship's own wave system at forward speed in calm water, as discussed in Chapter III.

Wave Loads

Coming to wave loads (Chapter IV) response amplitude operators for vertical bending moment, horizontal bending moment (at midship) and vertical shear (at quarter length) must first be calculated for both outbound and inbound conditions (average). These operators should be calculated for a range of wave lengths and headings. However, since for most displacement ships speed is not an important factor for wave bending moment, calculations may be made for one representative rough weather speed. Torsional moments should also be calculated at the midship section.

Next, the family of sea spectra which are to be used as a basis for design must be selected. If a particular ocean route or routes are to be used, then in general the family of spectra will be based on tabulated data on frequency of occurrence of different combinations of observed wave height (significant) and period (average). Or if the design is to be based on North Atlantic service, families of spectra for each of five different average wave heights (66) can be used.

Before proceeding to the bending moments in irregular waves, the problem of combining vertical and lateral (or horizontal) bending moments must be considered. It has been shown that a ship operating in oblique seas is subjected to unsymmetrical bending, so that the stresses measured at one deck edge will usually exceed the mean value. This diagonal bending in an oblique sea can be dealt with as the combination of vertical and lateral bending components (121). Since we are interested first in bending moments rather than stresses, we can calculate an effective vertical bending moment, M_e, that produces the same average stress as the maximum deck edge stress, $\bar{\sigma}$, resulting from combined vertical and lateral bending. If Z_V is the section modulus for vertical bending and Z_L for lateral

$$M_e = \bar{\sigma} \; Z_V$$

$$= \sqrt{M_{wv}^{\;2} + M_{wL}^{\;2}\left(\frac{Z_V}{Z_L}\right)^2 + 2M_{wv} \; M_{wL} \; \frac{Z_V}{Z_L} \; \cos\delta}$$

where M_{wv} and M_{wL} are the vertical and lateral wave bending moments, respectively, and δ is the phase angle between vertical and lateral bending (121).

The accurate way to proceed is to obtain response amplitude operators on the basis of the above for combined vertical and lateral bending and to use these for calculating effective vertical bending moments in irregular waves. This requires that the ratio Z_V/Z_L be known or assumed in the design stage. However, this procedure would not provide an estimate of trends of vertical bending moment alone, for comparison with other ships. Hence, for most cases, it is recommended to:

1. Calculate vertical bending moments M_{wv} in irregular seas, for comparison purposes.

2. Calculate from response amplitude operators for both vertical and lateral bending the values of effective vertical moment, M_e, using a tentative value of Z_v/Z_L.

If Z_v/Z_L should change significantly during the design, calculations for M_e must be repeated.

If there is evidence that maximum bending moments occur at a section significantly different from amidships, a correction factor can be applied to the midship results.

The next step, as discussed in detail in Chapter IV, is to calculate the bending moment response to different sea conditions and hence derive both a probability density function and a long-term cumulative distribution of bending moment coefficient for both vertical bending alone and combined vertical and lateral bending. In a similar fashion the long-term distribution of vertical shear at quarter points and of torsional moment would be calculated.

Thermal Effects

On the basis of reasonable assumptions regarding air and water temperatures, and their diurnal variations for each of four seasons, Jasper's method (119) can be used to calculate thermal stresses in the weather deck. The interest here is in the overall average change in stress across the deck from day to night, rather than local high stresses. Such calculations should be repeated under the assumption that there either is or is not full sunshine. Hence, considering data on the percentage of time that the sun shines for each season on the route in question, a reasonable probability function of thermal stress variation can be constructed. For the present purpose, however, it is felt to be adequate to use a weighted average thermal stress converted to a corresponding effective vertical bending moment.

Since in general there is a tendency for high thermal stresses to occur in good weather (sunshine) when wave bending moments are comparatively small, and vice versa, an attempt might be made to estimate a suitable correlation factor. However, since a heavy swell and bright sun may appear together, it is perhaps best to assume that the effects are independent and additive.

Dynamic Bending Moments

The evidence indicates that springing -- the more or less continuous vibration excited by waves -- is not found in ordinary dry cargo ships or container ships, presumably because of their low length/depth ratios, hence relative stiffness. For example, in the case of the Wolverine State the only vibratory response measured was that associated with slamming. Hence, it is assumed that the only dynamic loading to be considered here is slamming, which causes a damped high-frequency stress variation (whipping) that can be interpreted as an effective superimposed bending moment.

The first step, for both outbound and inbound conditions taken separately, is to estimate the probability of slamming. This can be done on the basis of data on actual voyages of similar ships or calculated by Ochi's method (100).

Next we must estimate the probability function for effective bending moment due to slamming when it occurs. This involves a number of steps, all of which are not at present in satisfactory state of development for routine design use:

1. Estimate local slam pressure and its variation in time and space in a representative severe sea condition -- hence its distribution.

2. Estimate resulting midship stress immediately following the slam and the subsequent whipping stress -- hence their distributions.

3. Translate these stress distributions into effective bending moment distributions.

Only one representative severe sea condition is assumed to be required, because it has been shown that the statistics of slam stress <u>when slamming occurs</u> is relatively insensitive to sea severity (Chapter V) -- in contrast to the frequency of occurrence of slamming. This is because speed is voluntarily reduced as necessary to avoid severe slams. However, one must make a realistic assumption regarding the ship speed in this representative sea condition. The speed is probably determined by the master on the basis of the ship's slamming behavior (since if the ship is slamming it probably is not deeply loaded, and therefore shipping water or propeller emergence are probably not governing factors). The exact criterion for speed might be any one of the following:

- Frequency of slam occurrence.
- Severity of impact (as judged by the sound).
- Amplitude of hull vibration at the bridge.
- A subjective correlation of any or all of the above with cases of past bottom damage on this ship.

Assuming that speed can be estimated for a new design in the sea condition selected, the procedure might be as follows:

1. The local keel pressure can be estimated from calculated motions (relative velocity) and Ochi's data (100). However, the extent of the impact in space and time can at best be only a rough estimate.

2. Kline's method can be used to calculate the resulting midship stresses.

3. The stress distribution can be translated into bending moment distribution by multiplying by the section modulus.

Pending availability of techniques to carry through the above procedures, another approach would be to assume that after extensive sea experience on a class of ships the ultimate criterion of slamming is simply the avoidance of bottom damage. Hence, one could calculate the pressure over a reasonable bottom area that the local structure could sustain without permanent set. Then this load would be used in the calculation of midship slamming stress -- and hence of effective bending moment (items 2 and 3 above).

A third approach would be simply to obtain statistics on the midship slamming stresses that are allowed to occur on ships of different types and to estimate such a distribution for any new design (as given for <u>Wolverine State</u> in Chapter V).

The final step is to determine the probability distribution of the amount by which the slam bending moment adds onto the wave bending moment. This involves the phase relationship between the occurrence of slamming and the wave-induced bending moment. A study of this problem on the Wolverine State indicates that it can be solved for that ship (Chapter VI), but some further development is needed to genera alize the solution to apply to any ship. In general, any slam bending moment increase is a hogging moment.

The result should be long-term distributions of added bending moment due to slamming for different sea conditions, obtained from:

(Probability of Slam) x (Prob. of added bending mt.)

It is possible, in principle, to combine this distribution with that of wave bending moment for consideration of the probability of ultimate failure. Results for the Wolverine State presented in Chapter VI, as well as Aertssen's data (18), indicate that slam stresses are relatively small, however, and for some ships can be neglected.

Of more importance may be the whipping that follows a slam and which will increase the next peak sagging (and hogging) moments. The magnitude of the increase depends on both the phase angle and rate of decay, as well as the slam stress amplitude (assumed to be equal to the whipping amplitude) to the maximum expected wave sagging moment.

It is clear that the determination of dynamic hull loads associated with slamming, which are of importance in relation to the probability of failure, cannot at present be predicted with the precision of the static and quasi-static loadings.

As indicated in Chapter II, another dynamic load that should be included in the ultimate load criterion is that associated with bow flare immersion. For many ships in which the flare is small, and/or the bow freeboard is high, this factor will not be significant. For others, such as the Fotini L reported in (3) or the aircraft carrier Essex discussed in (96) it may be very important and must be considered in addition to bottom slamming.

Accordingly, the design procedure should include the calculation of initial bending moment caused by bow flare immersion for several sea conditions, considering both the non-linear increase in the maximum sagging moment to be assumed for design and the vibratory whipping that follows (5).

Local Loads

Insofar as the primary load criterion is concerned, the principal local load to be considered is that of hydrostatic pressure on the double bottom. As discussed by Evans (122), this results in a bending moment at the middle of each hold and a larger one in way of each bulkhead. The latter implies a significant tensile stress in the bottom plating and compressive stress in the inner bottom, both of which would be superimposed on the longitudinal bending stresses. These local stresses are higher in the vicinity of longitudinal girders in transversely framed bottoms (47) but would be more uniform across the ship in the case of the more common longitudinal double bottoms. Since the bottom pressure is higher when wave crest is amidships than when wave trough is amidships, this effect is greater in hogging than in sagging. Pressures can be calculated on the basis of static head for the present purpose, although methods have been developed for taking into account the dynamic effects of ship motions (47).

COMBINING LOADS

Finally we have the problem of combining the probability distributions of four different loads for both (A) outbound and (B) inbound conditions. See Table II, which lists the various loads and summarizes the steps involved in evaluating each.

The combining of longitudinal bending loads will now be considered in relation to the possibility of damage and/or ultimate failure by buckling or by plastic flow and permanent set. Local loads to be considered in a specific design will not be included here.

Because of the difficulty in establishing a reliable zero base line in recording full-scale ship stresses, it has been customary to present statistical data in terms of peak-to-trough stress values. Thus no separation was attempted of hogging and sagging stresses. However, since for design purposes it is essential to provide independent estimates of sagging and hogging bending moments, wave stresses or moments must be separated into two parts for combining with other loads. This can be conveniently done by assuming that a predicted long-term bending moment curve can be considered to represent either sagging or hogging (with opposite signs). Accordingly, wave bending moments can be represented by two symmetrical long-term curves, one for hogging and one for sagging, as shown in Fig. 20. Taking account of the still water bending moment caused by the ship's own wave results in a base line shift, as shown. In some ships, particularly those with flaring bows and sterns, the bending moment may be non-linear with wave height, and in heavy seas there may be large differences between sagging and hogging bending moments (113). This effect could be evaluated by model tests and an adjustment of the division of total bending moment between hogging and sagging could be made accordingly.

Table II

DETERMINATION OF ULTIMATE HULL GIRDER BENDING LOADS

	Bending Moments	Successive Steps				End Result
I	Still Water	Set up typical conditions with different cargoes &/or ballast arrangements (A) & (B)*		Calculate bending mt. & shear for each	Determine mean and std. deviation	Long-term probability density functions (A) & (B)
II	Forward Speed	Estimate average speeds (A) & (B)		Calculate self-induced bending mts.		Average increase in bending mt.
III	Thermal	Diurnal and seasonal variations in ambient temperatures & cloud cover	Diurnal & seasonal variations in stresses	Corresponding variations in effective B.Mt.	Probability density function for sunny and for cloudy weather	Long-term probability density functions
IV	Wave-Induced	RAO's in regular waves: Vert. B.Mt. Horiz. B.Mt. Shear Torsion	Appropriate sea spectra families (& distr.)	Response to irreg. waves	Probability density functions: Combined H.&V. Bending Mt. Shear at Qtr.Pts. Torsion	Long-term distrs: Combined H.& V.B.Mt. Shear at Qtr. Pts. Torsion
V	Dynamic	Probability of slamming and/or flare immersion	Distribution of slamming and whipping bending mt. when they occur	Distribution of phase angles	Probable addition to bending moment Slamming Whipping	Long-term distr. of added dynamic bending moments

* Note: In each case above, calculations are to be made for both outbound (A) and inbound (B) loading conditions.

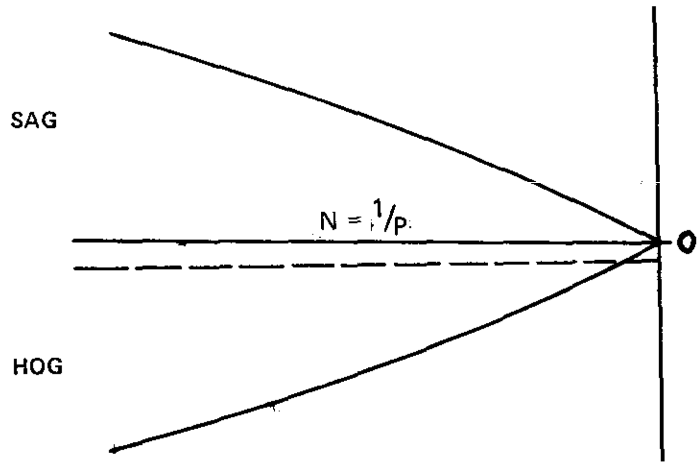

FIGURE 20 - Typical Long-Term Distributions of Wav Bending Moment for Sag and Hog

As previously noted, it is desirable to separate outbound and inbound conditions, since not only are the corresponding still water loadings distinctly different, but in many cases wave bending moments are also different. It has been suggested that for either case it can be assumed that the still water bending moments over a long period of time -- such as a ship's lifetime -- are normally distributed. Since the still water bending moment stays relatively constant for long periods of time -- usually an entire voyage -- it has the effect of periodically changing the baseline about which the wave bending moment varies.

To obtain single long-term curves for hogging and for sagging -- including still water bending -- requires that the wave bending moments be first expressed as a probability density function (instead of a cumulative distribution). The functions for still water and wave bending moments can then be combined on the basis of joint probability, since the two phenomena are independent.

Let y be a random variable describing the wave-induced bending moment (hog or sag). The density function of y will be called $p_w(y)$. Let x be a random variable describing the still water bending moment, with density function $p_s(x)$, which will be assumed to be normal. We are interested in the distribution function for the random variable $z = (x + y)$, which is given by:

$$p(x + y) = \int_{-\infty}^{\infty} p_s(x)\ p_w(y)\ dx$$

or

$$p(z) = \int_{-\infty}^{\infty} p_s(x)\ p_w(z-x)\ dx$$

Since $p_s(y)$ is not known in explicit form, the above integral cannot be evaluated. However, it can be determined numerically for any specific case. (See example in Chapter IX).

Nordenstrøm has discussed this subject in greater detail (122).

We come now to combining the thermal effects, which it has been shown can be interpreted in terms of a zero bending moment at night and an effective sagging bending moment in daytime -- especially if the sun is shining. To simplify the problem of combining loads, we can make the safe assumption that the thermal effects are always present, combining with wave bending. All wave data can be roughly divided intc two classifications -- those that occur at night, with no thermal effects and those that occur in daytime, with thermal effects superimposed. This would lead to two long-term curves, as shown in Fig. 21. We can conclude then that a safe treatment of thermal effects is to shift the base line by one-half the amount of the average total change in effective thermal bending moment.

Finally, local average stresses in inner bottom and bottom shell plating at a bulkhead can be calculated and equivalent constant bending moment increases determined.

This, and any other essentially constant effect, such as the bending moment caused by the ship's own wave system, can be allowed for by additional base line shifts.

Thus, insofar as the possibility of damage or ultimate failure by buckling or permanent set, the complete hull loading picture can be presented in the form of two sets of curves, one for outbound and the other for inbound. (In the case of a ship engaged in a triangular or tramp service, three or more sets of curves might be necessary). Each set of curves would consist of a pair of long-term distributions of bending moment coefficients, one for sagging and one for hogging, including horizontal bending, with the base line shifted to allow for:

Bending moment due to ship's own wave pattern.
Effective bending moment corresponding to thermal effects.
Effective bending moment corresponding to local effects.

With the above picture of expected quasi-static loadings available (demand), the structural designer can in principle estimate the capability of the structure and hence the probability of failure in a lifetime or in the lifetimes of many ships. He can also estimate the probability or expectation of damages that do not constitute failure. Strictly speaking the capability of the structure must also be expressed in probability terms, as explained and developed in the work of Freudenthal (123). Various writers have pointed out that the probability of failure (or damage) can be obtained by mathematical treatment of the overlapping probability density functions of demand and of capability (124)(125)(2). However, for the present purpose we will assume that the capability is deterministic (i.e., standard deviation is zero). The probability of failure is then simply the probability of exceeding a specific limiting bending moment.

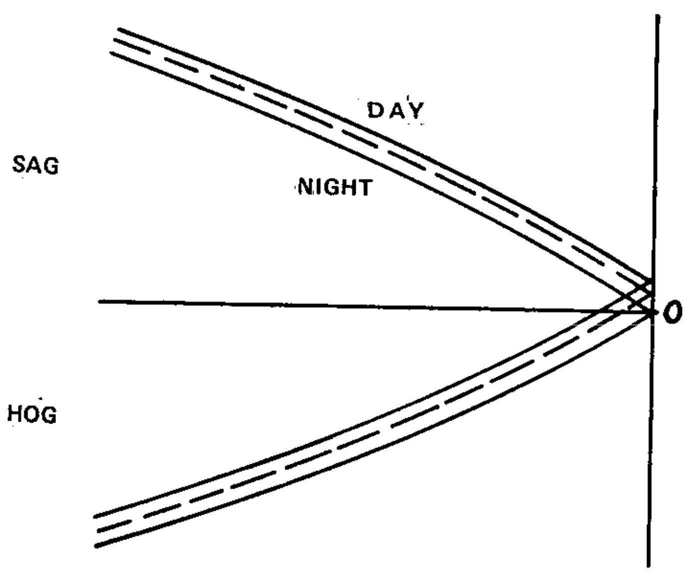

FIGURE 21 - Typical Long-Term Distribution of Wave Bending Moment, Sag and Hog, with Thermal Stress Superimposed

Since maximum stresses resulting from combined static and dynamic loading may or may not be expected to occur simultaneously, a probabilistic model of such joint occurrence needs development; it is conceivable that circumstances could cause simultaneous addition of significant loads of each type, but the probability might be small. Pending the development of a complete model, at least the following alternative load combinations should be determined:

A. Static loading predominant

(1) The highest expected load value due to combined still water and wave-induced bending moment, local loading, thermal gradients, etc., that could cause tension damage.

(2) The magnitude (and frequency) of superimposed dynamic loads occurring at the same time.

B. Dynamic loading predominant

(1) The magnitude of the highest expected dynamic bending moment.

(2) The highest quasi-static tensile loading due to bending moment, local loading, thermal gradients, etc., expected to occur at the same time. Since, as previously noted, the duration of the load has a bearing on the capability of a structure to resist dynamic loads, the duration of such loads should be specified.

It remains to determine an acceptable probability of failure and of damage, which will be discussed in the next section.

PROBABILITY LEVEL FOR DESIGN

The final step in establishing design criteria for ultimate bending is the determination of a probability level to adopt for determining design bending moment. It is necessary first of all to consider the safety of the ship and its crew. The only sound basis for a strength standard in this respect is one based on probability theory. We must be sure that the total risk of structural failure is never greater than society can accept. Nor must the occurrence of structural damage that does not endanger the ship be burdensome to the ship operator either through excessive repair cost or too frequent withdrawal of the ship from service. As progress is made in developing techniques for predicting long-term trends of various loads acting on a ship's hull, along with sophisticated techniques for determining detailed distributions of stresses, the time is approaching when we should decide what risk of structural failure is acceptable. Here the classification societies can be of assistance by analyzing their records to determine the number of failures occurring over the years in ships of different types and sizes and computing the corresponding probabilities that have presumably been considered acceptable. One question is, should we use the highest bending moment to be expected once in a single ship's lifetime or once in the lifetime of a fleet of 10, 100 or 1,000 ships?

J. F. Dalzell, in an informal memo to the Ship Research Committee (12 May, 1970) gave a valuable analysis of some published Lloyd's Register data on merchant ship losses (126), which covered 18 years (1949-1966) and 390,000 ship-years of service experience. He assumed that the losses designated "Foundered" (31% of all losses) were cases of complete structural failure, although there were no doubt numerous exceptions. Assuming a 25-year average ship life, he arrived at Table III, showing probability of failure (in a ship's lifetime) for different sizes and types of ships

TABLE III

APPROXIMATE PROBABILITY
OF FAILURE: "FOUNDERING" (126)

Ship Size (Gross Tons)	Ship Type	
	Dry Cargo	Tanker
100- 1,000 T.	0.11	0.006
1,000- 5,000 T.	0.04	—
5,000-10,000 T.	0.02	0.008
10,000 Tons up	0.006	0.006
All Sizes	0.05	

A principal reason for the higher values for dry cargo ships, particularly in the small sizes, is probably the fact that losses from other causes are included -- such as capsizing (low stability, flooding, or cargo shifting) or foundering from flooding (hatch cover failure, leakage, etc.). A figure somewhere between 0.003 and 0.006 would be a reasonable value for the probability of failure that has been tacitly accepted over the past 20 years for large oceangoing ships. In proposing a specific figure for a new design criterion, however, we feel that a more conservative figure should be adopted, and Dalzell's suggestion of 0.001 is tentatively proposed. This implies that merchant ships would be designed with a probability of ultimate -- or catastrophic -- failure of no greater than 0.001, i.e., that a new ship would have a chance of not over one in a thousand of failure during a normal life span. See also discussions by J. F. Dalzell and by M. K. Ochi of (3), and authors' reply.

Although our principal interest is in extreme bending moments that cause complete failure, lesser values of bending moment than occur more frequently in a ship's lifetime are also of interest. As noted by Smith in discussion of (17), "If bending moments were estimated statistically then it would be necessary to specify for each level of damage an acceptable probability of occurrence."

Some statistics are available on ship structural damages from various sources. A particularly valuable study made by Lloyd's Register (35) covers dry cargo ships and tankers (not bulk carriers). It relates the number of cases of fractures in strength deck and shell plating to ship years of service, as shown in the accompanying Table IV. In response to our inquiry to Lloyd's Register regarding the data in Table IV, Mr. G. J. Jensen advised that not all of the cracks were of the brittle kind. Presumably the others were mostly fatigue cracks. "However, at this stage it would neither be possible to indicate the percentages of each type of fracture occurring, nor their seriousness, without re-opening the investigation." Mr. Jensen also stated that of all the fractures tabulated, only one resulted in the loss of a ship. "The World Concord built in 1952 broke in two in a brittle fashion. The fracture was traced to a hard spot caused by welding, in way of one of the bottom longitudinal endings."

Of particular significance are the figures for "occasions per 100 years," which -- except for old tankers -- run at or below 1 per 100 ship-years. This appears to be an acceptable figure, considering that few if any of these fractures actually resulted in the loss of a ship. Considering a ship's lifetime to be 25 years, 1 damage per 100 years would be equivalent to a lifetime damage probability of 0.25 (i.e., 1 damage in the lifetime of 4 ships.)

TABLE IV

FRACTURES IN STRENGTH DECK AND SHELL PLATING

	Tankers		Dry Cargo	
	Old*	New †	Old*	New †
Number of ships at risk *†	889	1362	2240	3039
Total ship years service	3692	4506	9391	10248
Number of ships affected	53	29	88	87
Percentage of ships affected	5.96	2.13	3.93	2.86
Ships affected per 100 years	1.44	0.66	0.94	0.85
Number of occasions	74	30	97	99
Occasions per 100 years	2.00	0.67	1.03	0.97
Number of ships with special steel	326	596	434	1172
Percentage of ships fitted with special steel	37	44	19	39

* "Old" means ships built 1943-1957 inclusive.
† "New" means ships built 1958-1967 inclusive
Years of service counted to end of each period, i.e., 1957 and 1967 inclusive, or to exit from class, whichever is the shorter.

(35) (Reproduced by permission of Lloyd's Register of Shipping.)

A basic approach to determining the probability level to be used in a design criterion is that of "expected loss,"* which has been summarized in convenient form by Freudenthal for application to maritime structures in general (127). It is based on the principle that the best design is the one that minimizes the expected total cost, where the latter consists of the sum of initial cost, failure cost, and damage cost, as explained below. It is very difficult, if not impossible, to assign a dollar value to the passengers and crew, but it will be assumed -- perhaps over-optimistically -- that in this day of efficient communication and life-saving technology a ship may founder without the crew being lost.

A distinction is made between "failure," in which the safety of the ship is endangered or the ship actually breaks in two, and "damage," in which local cracking or buckling of main hull girder elements results in a requirement for repairs to be made. All damages not involving the main hull girder are excluded, since presumably they would not be affected by any change in the main hull structural design.

Expressed as an equation, the total expected cost to be minimized,

$$L = I + p_1 F + (1 - p_1) S p_2$$

where I = initial cost of the ship (or structure)

p_1 = probability of failure (in a lifetime)

F = anticipated total cost of failure (replacement cost + cargo loss + temporary charter of replacement ship + loss of business from customer reactions + cost of pollution or other environmental effects, etc.)

S = anticipated cost of damage or "failure of function of surviving structure" (the "Success cost"), i.e., cost of repairs and of associated costs of damage that does not involve the loss of the ship.

p_2 = the expectation or expected number of such damages.

*This is the so-called Baysian decision rule.

For the case of the design of a ship's main hull structure, we may hypothesize that the probability, p_1, of failure that can lead to the complete loss of the ship is very low. But it might occur on the basis of some combination of extensive buckling and yielding or it might occur primarily in the form of brittle fracture, perhaps preceded by fatigue cracking.

The expectation of other damage, p_2, that would require more or less extensive time out of service for repair depends on any one of the modes of failure previously discussed -- or of a combination of them. In fact, a ship might experience one or more such damages in several modes during its lifetime. Furthermore, such damages might be of different degrèes of severity. Hence, in our case the term Sp_2 should actually be a summation,

$$\sum Sp_2$$

For example, a particular hull design configuration with certain specified scantlings might have a very low probability of one severe buckling or a tensile failure of the main deck. But the probability of local tensile failure or fatigue cracking at a hatch corner occurring several times might be relatively high.

Hence, we come finally to the concept of determining for each failure mode the probability of failure in a lifetime, and for each damage mode the expectation of damage, each of which should be multiplied by the corresponding cost.

In principle the total expected cost, L, can be evaluated for several alternate hull designs and the optimum design determined graphically. The following types of damage should be considered in addition to ultimate failure:

1. Panel buckling that is not immediately dangerous.
2. Excessive yielding.
3. Fatigue cracking.

Because of the many uncertainties involved, brittle fracture is excluded from consideration here.

A sample calculation will be presented in Chapter IX for the the Wolverine State in order to ascertain the significance of the proposed approach.

As noted by the I.S.S.C. Committee No. 10 (20) it should be possible in due course to relate probability of failure to a conventional deterministic load and a factor of safety. This may be a desirable thing to do for general guidance in order to correlate any new approach with the empirical standards that have been successful in the past.

FATIGUE CRACKING

One approach to structural design relative to the secondary criterion of fatigue loading discussed under Critical Loads (Chapter II) was simply to make sure that the probability of exceeding the yield point at critical areas of stress concentration was at an acceptable level. However, this approach is imprecise and may lead to excessive scaltlings. Therefore, it appears that as complete a picture of cyclic loading should be furnished for the use of the structural analyst (and researcher) as possible.

From the fatigue viewpoint the type of loading is one of cyclic load reversal, usually with fluctuating mean load and possible occasional overload at points of stress concentration. It is further complicated by diurnal thermal stress variations. These loads are tabulated below along with the estimated cycles of load reversal for each in a typical ship's lifetime:

Still water	340
Wave Bending	10^7-10^8
Dynamic	10^6
Thermal	7000

The fluctuating mean load is the so-called still water bending moment, discussed in the section on Ultimate Loads. In general the specification of two probability curves, one for outbound (A) and the other for inbound (B) conditions, will provide the information needed for fatigue design. However, one additional item is needed: the time that the ship operates in condition A before changing to B, time operating in condition B, etc. In general both times will be equal simply to one-half the total round voyage time and will be measured in weeks. To be more accurate the effects of consumption of fuel and additions of salt water ballast should be included.

The cyclic loading consists of the low-frequency wave-induced bending moments and the high-frequency dynamic bending moments previously discussed. Their phase relationship is perhaps of less significance for fatigue than for brittle fracture. At any rate, long-term cumulative distributions of both should be available as part of the load determination for ultimate loading. From these distributions one can obtain cyclic load spectra in the following manner. The reciprocal of the probability is the number of cycles, n. For a ship's lifetime of n_L cycles, a scale of $n_F = n_L - n$ is then constructed on the distribution plot. Then n_F gives the number of cycles expected in the ship's lifetime of any desired level of bending moment. See Figure 22, which deals with wave bending effects only (128).

Finally, information should be provided on the expected diurnal variation of thermal effects, as previously noted.

The above information should provide the data needed by the stress analyst to evaluate the cyclic loading, variation in mean stress, and -- with estimates of stress concentration factors -- the frequency and direction of local stresses. Fig. 23 was developed for the case of constant mean value (128).

The object is to provide a means of estimating cyclic loading that can eventually be balanced by the structural designer against the endurance properties of the structure. Thus he would be able to provide an efficient structure in the design stage that would have an acceptably low probability of fatigue cracking in service. However, it should be emphasized that, since in general the safety of the ship is not threatened by a fatigue crack of a certain maximum length, a higher probability of cracking can be accepted than for ultimate failure of the hull girder.

If the general application of more rational design standards should in time result in reduced hull scantlings, then the incidence of fatigue cracking might increase to an unacceptable level. In this case some modification in strength standards by classification societies might be called for.

BRITTLE FRACTURE

From the point of view of ship structural design the possibility of failure by brittle fracture requires careful consideration both independently of and in combina-

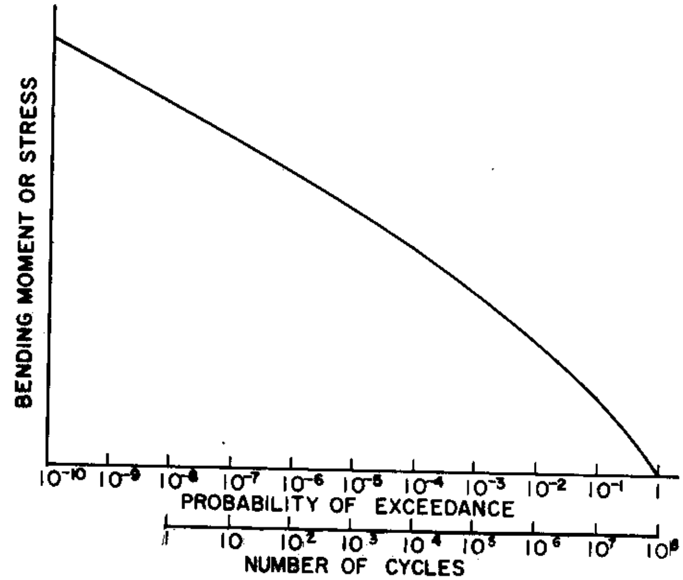

FIGURE 22 - Long-Term Distribution of Bending Moment or Stress, with Reversed Scale Showing Cyclic Loading or Number of Cycles of Each Stress Level in One Ship Lifetime (10^8 Cycles)

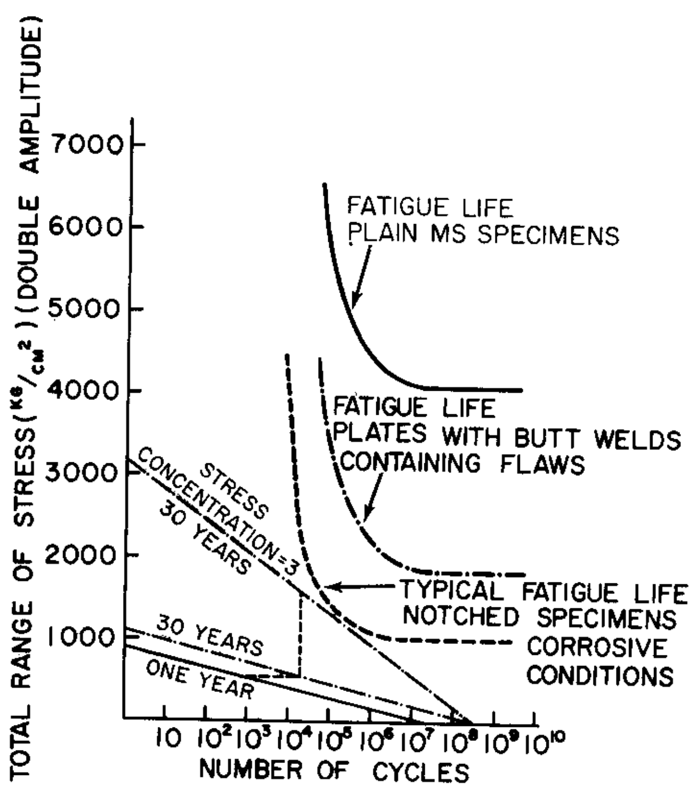

FIGURE 23 - Example of Application of Cyclic Loading Curves to Study of Fatigue (128)

tion with other modes of potential failure. However, the applicable hull loads are essentially the same as those discussed under Ultimate Bending Loads. Hence, the long-term distribution curves developed there, combining still water, wave bending, dynamic, and thermal loadings, should apply to design in relation to avoiding brittle fracture.

Since brittle fracture is a tensile phenomenon, buckling need not be considered. But the structural designer must consider many factors in addition to mean stress levels -- stress concentrations, weld defects, fatigue cracks, ambient temperatures, strain rate, steel qualities, locked-in stresses, and metallurgical effects of the welding process. Most of these factors involve many uncertainties, and therefore it is more difficult to predict a probability of failure for brittle fracture than for failure in ultimate bending.

Ideally, brittle fracture could be avoided if it could be established that the temperature in service never drops below the transition temperature of the steel, considering the steel properties (or chemical composition), including the effects of welding, the plate thickness, the nature of possible weld defects and the severity of local concentration factors in the main hull girder.

At present the above determination is not possible. Furthermore, the determination of dynamic hull loads associated with slamming, which are of importance in relation to the probability of failure, cannot at present be predicted with the precision of the static and quasi-static loadings. Hence, it does not appear feasible at present to adopt a load criterion for brittle fracture, even if that were desirable. Instead it is customary to adopt a fail-safe design procedure, by providing a sufficient number of crack arresters to insure that a single crack is limited in its propagation sufficiently to avoid endangering the ship. These crack arresters can be either riveted seams or strakes of high ductility (notch-tough) steel.

The rational design philosophy would then seem to be as follows:

1. Recognize that with present design standards and material quality brittle fracture seems to be under control, if not entirely eliminated.

2. If design and material standards do not change, brittle fracture need be considered only in maintaining good design, construction and operating practices.

3. The application of new design standards, based on the quasi-static primary load criterion proposed herein, may lead to suggested increases in working stress levels.

4. Such increased working stresses should be accepted only if either calculations show probability of brittle fracture is not increased or more stringent material requirements are introduced.

IX. SAMPLE LOAD CALCULATIONS

INTRODUCTION

It was felt at the outset of this project that a numerical example of hull load determination, using the procedures developed during the project, should be carried out, leading to specific load criteria for design of one type of ship. The objective was twofold: the example would illustrate and explain the procedures developed, and it would give an indication as to how the proposed load criteria compare with conventional standards.