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Dynamic Loadings Due to Waves and Ship Motions

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ABSTRACT

After consideration of the ways that a ship's hull can fail, a review is first given of the major longitudinal bending loads -- with particular attention to the present status of knowledge of wave bending moments. Correlations are given between predicted trends, using both theory and model tests, and full-scale observations.

Vibratory responses, both transient (slamming and whipping) and cyclic (springing), are discussed, as well as loadings resulting from uneven thermal gradients. Finally, consideration is given to ways of improving design load standards on the basis of present knowledge. Further needed research and the future application of reliability theory is discussed.

INTRODUCTION

Our understanding of wave loads on ship hulls has increased greatly in recent years as the result of extensive research in a number of related areas -- model tests to determine both quasi-static and vibratory wave loads, techniques for the theoretical calculation of wave-induced shears and bending moments, collection of full-scale ship stress data, and the collection and analysis of ocean wave records. These studies have been made necessary by the drastic changes in merchant ship characteristics, particularly the larger size of bulk carriers and the higher speeds of general cargo vessels, plus the development of new types such as LNG carriers for which different load problems arise.

Perhaps it is appropriate at this time to attempt to assess the new technology available today as it can be applied to new ship designs that are now or soon will be on the drawing boards. On the one hand such an assessment should show how research has provided tools to help solve today's design problems; on the other hand it will indicate areas for further research effort needed to meet current and future problems.

The study of non-vibratory wave-induced response of the hull girder began with a pioneering project sponsored by the Hull Structure Committee, SNAME, at the Davidson Laboratory and reported in 1954 (1). A model of a T-2 tanker, jointed amidships, was subjected to head and following seas and the fluctuating bending moment measured. (First mode vibration of the jointed hull was also identified and recorded). Since this experimental work preceded any known analytical treatment of the subject, it was with some surprise that the experimenters noted a reduction in bending moment from the values calculated by conventional quasi-static methods, as shown in Figure 1. (This reduction was later found to have been exaggerated at certain speeds because of dynamic effects in the moment measurements (2)).

The analytical treatment of ship motions and wave loads by Korvin-Kroukovsky (3) and his associates followed quickly. The bending moment was shown to be the result of integrating hydrodynamic and inertia (D'Alembert) forces over the ship length as illustrated by Figure 2, reproduced from reference (4). The work explained the reduction in dynamic wave bending moments on the basis of two factors: the well-known "Smith effect", which accounts for the pressure reduction in a wave crest and increase in a trough resulting from the orbital motion of wave particles, and a second effect of comparable magnitude resulting from ship-wave interaction (damping and added mass).

Further research has established that pitching motion, per se, has a relatively small effect on wave bending moments. But heaving is of greater significance, as shown for example in the photograph published as Figure 25 of reference (1). Here the model is shown in sagging condition with the load waterline completely out of the water over the entire length of the ship. (Static buoyancy

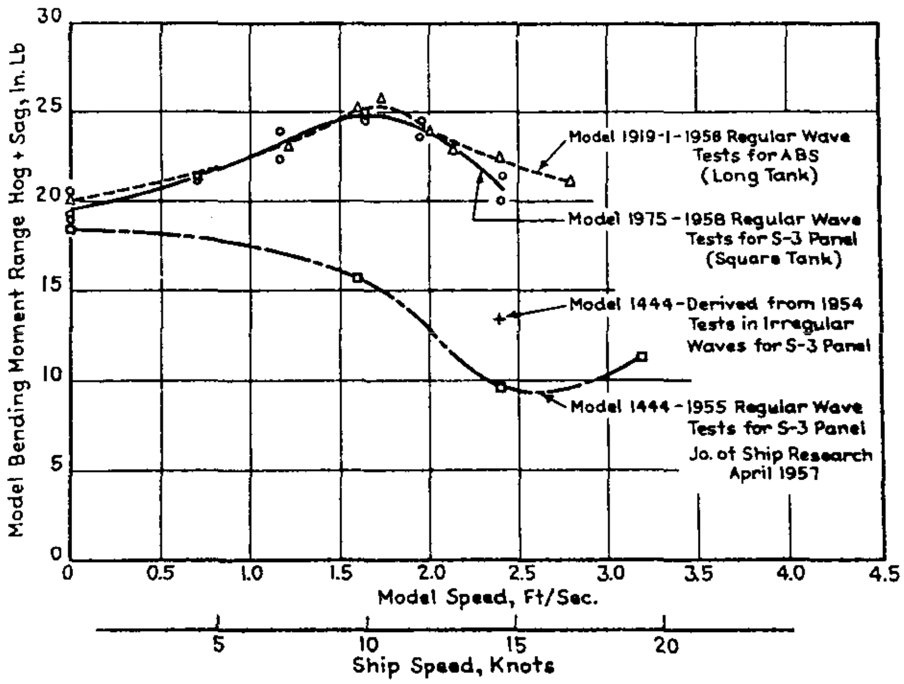


Figure 1. Vertical bending moments of 1/105 scale models of T2-SE-A1 tanker. Head waves of model length and height = L/48.

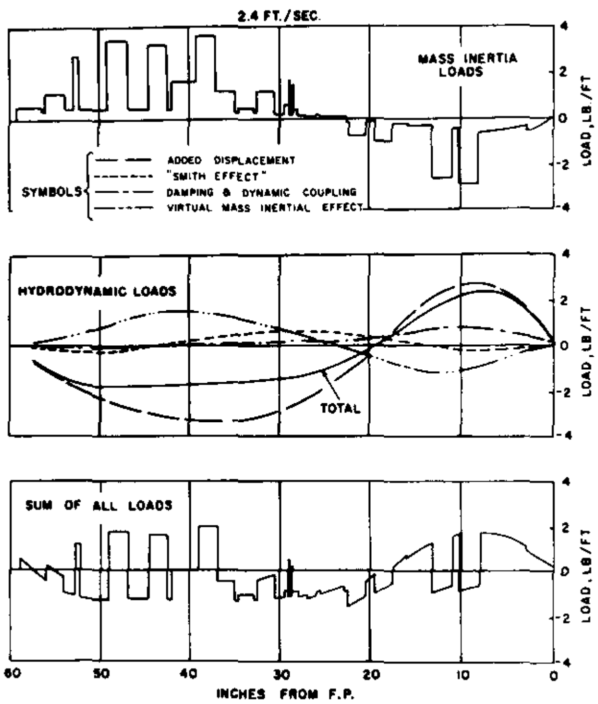


Figure 2. Longitudinal distribution of loads on T2-SE-A1 model in regular model length waves, crest at bow.

is clearly less than half the normal displacement).

Further experimental work established that the wave-induced bending moment is not basically a resonance phenomenon. For example, experiments by Dalzell (5) showed that when data for a wide range of model speeds are plotted on the basis of wave length they collapse into a fairly narrow band as indicated by Figure 3. In other words, the geometrical relationship between wave and ship -- or "ship/wave matching" -- is of prime significance.

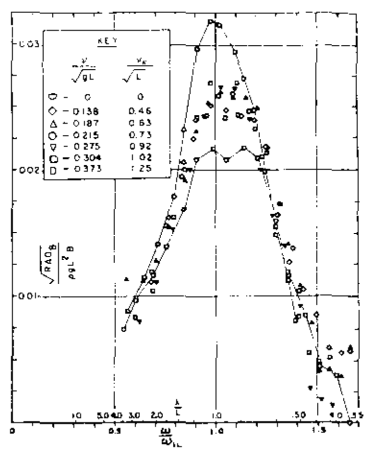


Figure 3. Destroyer bending moment amplitude response in head seas vs. wave frequency ratio (derived from tests in irregular head seas).

Other research, both theoretical and experimental, has been extended to include lateral bending and torsion in oblique seas (6)(7). Excellent agreement has been obtained generally between theory and model tests, with the exception of high-speed ships in following seas. The theory breaks down when the encounter frequency approaches zero, and further developments are needed (8). Comparisons between theory and experiment will be presented later.

The principle of superposition has been applied to the prediction of bending moments in irregular short-crested seas, as defined by their directional spectra. This procedure yields short-term statistics which can be integrated over sea condition to obtain long-term predictions (9).

Full-scale statistical data on wave bending moments have been collected over periods of 2 to 3 years in the form of stresses or strains (10)(11). Fortunately, it has been found that a ship's hull, even though a built-up box girder rather than a homogeneous beam, follows the simple beam theory quite well, provided that areas of stress concentration are specially considered. Consequently, measured stresses can be interpreted as external bending moments, with the help of simple dockside "calibrations", and can be compared with theoretical predictions. Actually, this has proved to be the only way to make correlations between full-scale and model (or theoretical) data. It is never possible to obtain a complete enough picture of the sea condition at a particular time to make a direct comparison.

So as knowledge accumulates on ocean waves and on ship responses to them, the conclusion becomes inescapable that a probabilistic approach is the only feasible one in the long run. The waves themselves can only be described in such terms and hence the resulting wave loads can best be described statistically. Eventually a rational, as well as practical, design approach will be developed for everyday use. Meanwhile, it will be shown later in this paper that the probability approach is of immediate practical usefulness on a comparative basis.

Another conclusion from recent research is that other hull loadings than simple wave bending must be taken into account in design. Figure 4 gives the typical variation in midship bending stress for a tanker, showing variations in still water loads and diurnal thermal stresses as well as wave bending. Furthermore, there are vibratory effects resulting from impact and high-frequency wave excitation that must be taken into consideration.

Accordingly, the next section will deal with the critical loads -- as a means of clarifying the loads to be considered and combined. Later sections will deal with still water and wave bending moments in more detail.

CRITICAL LOADS

Before discussing hull loads in detail it is necessary to consider the different ways that the ship structure can suffer damage or fail. Caldwell (12) considers ultimate failure as the complete collapse by buckling of the compression flange and simultaneous tensile failure of the tension flange.

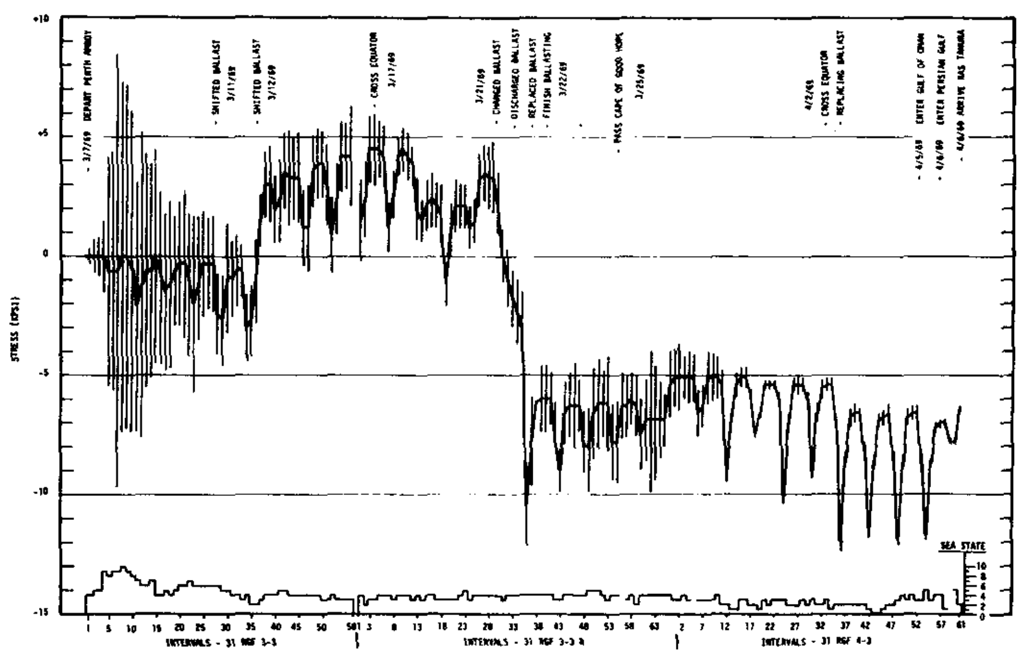


Figure 4. Typical voyage variation of midship vertical bending stress, S.S. R.G. Follis.

However, it is clear that a considerably less severe damage would be a serious matter, as indicated by such factors as necessity for major repairs, interference with normal ship operation and non-watertightness.

Hence, for our purpose we may define damage as a structural occurrence *that interferes with the operation of* the ship to the extent that withdrawal from service for repair is required, such as:

- Excessive local hull deflection: buckling and/or permanent set.
- Fatigue cracking.
- Brittle fracture, minor.

Failure is a severe damage that endangers the safety of the ship:

- Collapse of the hull girder.
- Extensive brittle fracture.

Considering the various types of damage (or failure) in more detail, the first, excessive hull deflection is a rare occurrence, except locally, and complete failure or collapse is even rarer. This suggests that conventional standards of strength are generally adequate -- *in fact, they may be excessive.* Loads that can combine to threaten hull failure are still water bending moments, wave-induced bending moments (quasi-static), vibratory (high frequency) loads, and thermal effects (13).

Second is the possibility of fatigue cracking, which seldom constitutes failure but is important for two reasons: Fatigue cracks, which are fairly frequent, can grow to the point that they must be repaired, and fatigue cracks are notches that under certain circumstances can trigger rapid propagation as brittle fracture. Nibbering notes (14), "It is a favorable circumstance that fatigue cracks propagate very slowly in ship's structures". Cyclic loads to be considered include the same loads as mentioned above, with widely varying periodicities and mean values.

Brittle fracture, which was a serious problem with early welded ships during World War 2, was long ago brought under control by insuring satisfactory "notch-toughness" of shipbuilding steel, as well as by eliminating severe design stress concentrations and by improving welding techniques, inspection, etc. However, brittle fracture can and does occur, and therefore the philosophy has been one of "fail-safe" design. Crack arresters, consisting of rivetted seams or strakes of steel having lower transition temperature are provided as standard practice. These have proven effective in limiting crack propagation and thereby restricting brittle fracture to a minor damage rather than a hull failure problem.

It can be argued then, that since fatigue cracking does not threaten the life of the ship and *brittle fracture* can be controlled, the primary criterion of rational ship structural design should be one of ultimate strength -- avoiding excessive deflection through buckling or plastic flow (15). Accordingly, ultimate strength and the corresponding bending loads will be given particular attention here. As mentioned above, these consist of:

Still water bending moments
Wave-induced bending moments (quasi-static)
Vibratory bending moments
Thermal effects

STILL WATER LOADS

Although this paper is concerned primarily with wave loads, it is important to consider still water loads because they provide widely-varying mean levels about which the wave loads vary. This is clearly shown in Figure 4 by the variations within a single voyage, while even larger variations can be found from one voyage to the next return voyage (e.g., loaded and ballasted).

Although the longitudinal bending moment in still water is easy to calculate for any number of loading conditions, it sometimes receives attention only in the loading manuals prepared by the shipbuilder for a few hypothetical conditions which may or may not be used for guidance in ship operation. For the purposes of a rational hull design standard, there have been two approaches:

(1) Make calculations for all extreme conditions of loading possible in the ship's lifetime and design for the largest hogging and sagging conditions expected.
(2) Set up reasonable, attainable conditions of loading and establish maximum allowable hogging and sagging moments. Then provide guidance information, or a computer program to calculate bending moments, that will insure that the limits are never exceeded.

The first approach has been tactily assumed, if not explicitly adopted and vigorously followed, in the design of most general cargo vessels. The second approach has been adopted for tankers, modern container ships such as the SL-7 (16), and for the Great Lakes bulk carriers.

A third approach is a statistical or probabilistic one, where typical conditions such as full load, ballast, light load, etc. are established. Calculations are then made -- which can be verified by service records -- of both the average value of bending moment and the standard deviation for each basic condition. This approach is desirable because ships are never loaded exactly in accordance with the designer's loading manuals, and it is consistent with an overall probabilistic approach to design (13).

An effect peculiar to high-speed ships is the bending moment created by the ship's own wave. At a speed corresponding to a Froude No. of about 0.2, the crest at bow and hollow amidships will produce a significant sagging moment. This can be estimated from a model test wave profile and included with the still water bending moment.

WAVE BENDING MOMENTS

Research has brought about a drastic change in the way wave bending moments are formulated for design. It is not long since a calculation of the static moment for a ship poised on an L/20 wave -- both in sagging and hogging conditions -- sufficed. It had been recognized, however, that as ships increased beyond 350-400 feet in length this simple standard was unrealistic. The Naval Architect's crude way of adjusting for increasing length was to assume an increase in allowable stress. Although such a procedure is not logical nor consistent with Civil Engineering practice (which would consider a variation in load with constant allowable stress) the resulting strength standards were reasonable so long as ships did not increase drastically in size, say up to 600-700 feet.

Dimensional considerations show that if the design wave length goes up in proportion to ship length, wave bending moment $\propto L^4$. If a typical allowable stress formula is used, $\sigma \propto L^{1/3}$, it is easy to show that this is equivalent to assuming a wave height $\propto L^{2/3}$ (and constant allowable stress). As a matter of fact, some time ago the Navy introduced a design wave height $\propto L^{1/2}$ as being more realistic, but allowable stress was not assumed to be constant. In comparing these trends with more recent work later on, it should be noted that they included an allowance for still water as well as wave loadings.

Although either of the above design approaches has been found to be acceptable for ships up to 600-700 feet, the rapid increase in tanker size after World War II raised serious questions regarding the extrapolation of longitudinal strength standards to ever larger and larger vessels. Fortunately, as previously noted, new research techniques had become available, including model test techniques to measure wave bending moments, theoretical methods of calculating motions and bending moments in both regular and irregular waves, new data on wave patterns in spectral form, and full-scale ship stress collection programs. These new approaches have been shown (13) to fit together in a consistent probabilistic picture of wave loadings on ships, which eventually will undoubtedly be merged with a similar probabilistic picture of structural capability to produce a wholly rational design technique consistent with modern reliability theory. This trend for the future will be discussed at the end of this paper. Meanwhile, however, it is important to point out how the probabilistic approach has already contributed to the determination of practical standards for design waves of ships of ever-increasing size.

In the early 60's the ABS suspected that the extrapolation of current design standards to tankers in the range of 800-900 feet was leading to excessively severe requirements, which penalized such vessels. Hence, comparative calculations to determine the trend of wave bending moments with ship size were started at Webb Institute and have continued ever since with ABS support. Similar work has been done by other classification societies.

A Webb report to the ABS in 1963 showed (17), on the basis of a single severe storm sea spectrum, that on a comparative basis, design wave height for full tankers increased slowly above a ship length of 600 feet and tended to level off at 1000-1100 feet. This showed that the tentative assumption of wave height = $0.6\,L^{0.6}$ was unnecessarily high above 600 feet. See Figure 5.

Subsequent work at Webb under the guidance of a special Panel on Larger Vessels of the ABS Naval Architecture Committee made use of wave spectra of different levels of severity, considering the frequency of occurrence of each in the North Atlantic. Although this work was based on probabilistic predictions for different sizes of geometrically similar ships, results were expressed simply in terms of effective or design wave heights as a function of ship length. Results were comparative rather than absolute, since the probability level for design wave height trends was selected to correspond to values of wave height known from experience to be satisfactory at 600-foot length -- including still water bending moments. See Figure 5. It may be seen that there is a definite tendency for the design wave height to level off or drop above a length of about 1100 feet.

In the work on which Figure 5 is based, effective wave height, h_e, is not a direct measure of an observed height of the sea. Rather it is a measure of the external wave bending moment to be used in design. It is defined as the height of a trochoidal (or sinusoidal) wave whose length is equal to that of the ship, which by conventional static

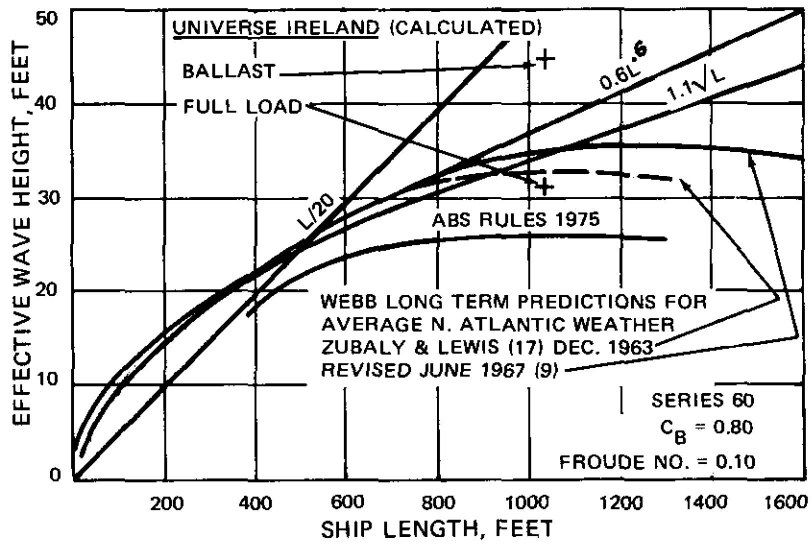


Figure 5. Comparison of Effective Wave Height Formulations.

bending moment calculation (Smith effect excluded) gives a bending moment (hog or sag) equal to that experienced by the ship in an irregular sea. Thus, if $\bar{h}$ is the wave height used in a static calculation,

$$\frac{\text{Static Wave B.M.}}{\bar{h}} = \frac{\text{Irregular Wave B.M.}}{h_e}$$

or

$$\frac{h_e}{\bar{h}} = \frac{\text{Irregular Wave B.M.}}{\text{Static Wave B.M.}}$$

Representing the static wave bending moment amplitude (hog or sag) by an equation,

$$BM_S = c\ \rho\ g\,\bar{h}\ BL^2\ c_w$$

the coefficient c depends on the wave form and the hull form of the ship. Hence, c has a convenient physical interpretation in terms of conventional wave bending moment calculations made by naval architects. L is length, B is breadth, c_w is waterplane coefficient, ρ is mass density and g is the acceleration of gravity.

Substituting the above expression for static wave bending moment, $\bar{h}$ cancels out, and

$$h_e = \frac{\text{Irregular Wave B.M. Ampl.}}{c\,\rho\,g\ BL^2\ c_w}$$

Since the irregular wave bending moment above is continually varying from one sea condition to another, it must be defined in statistical terms. It could be, for example, the value expected to be exceeded once in the lifetime of the ship (N approximately 10^8). As a matter of fact, the Webb curves in Figure 5 correspond closely to N = 10^8 for wave bending only.

The problem of specifying a design h_e is complicated by the question of factor of safety and allowable stress. In simple terms, one may either use some sort of an average high bending moment in association with a low allowable stress (large safety factor) or an extreme, rare value of bending moment with a higher allowable stress. The curve adopted in the new 1975 ABS Rules is also shown in Figure 5. It is lower than the Webb curves because all still water loads are excluded. This is reasonable because the h_e values are to be used in conjunction with an allowable stress that is well below the yield point of steel and hence allows a sizable margin of safety.

Although the Webb curves in Figure 5 were derived for full cargo vessels, the ABS curve is assumed to apply to all ships, regardless of form and fullness. Although there is a waterplane coefficient factor in h_e and a block coefficient adjustment in the rule formula for required section modulus, calculations based on Vossers' model tests suggested that there should be more than one curve of h_e; the curve for 0.60 block coefficient should be slightly lower than 0.80 block (17). Further calculations show that in the full load condition large, full modern tankers actually show a lower trend of h_e because of their extremely large draft (which means a very high wave attenuation or "Smith" effect). However, the ballast conditions show a somewhat higher trend than finer cargo ships, confirming the earlier calculations. This suggests that particular attention should be given to the ballast conditions in design. See calculated points for Universe Ireland in Figure 5. (Full load draft is 81.7 feet and ballast draft is 30.5 feet).

On the other hand, a better solution might be to require deeper ballast drafts when encountering heavy seas. In the case of the Universe Ireland above, the forward draft was 28.5 feet, which almost meets the ABS minimum for reducing bottom plating thickness. Indications are that either a greater draft should be required or an increase of h_e would be called for.

An opposing trend of ballast drafts should be noted as a result of the desire of IMCO to require the use of clean ballast only. A recent paper on the subject (18) considers the advantages and disadvantages of lighter ballast drafts but makes no mention of a possible increase in longitudinal wave bending moment -- other than that resulting from slamming. It is recommended that the effect of ballast draft on wave bending moment be given further attention.

CORRELATIONS BETWEEN CALCULATIONS AND FULL-SCALE

The question arises as to how meaningful are these calculated trends of design bending moment (or effect wave height) as shown in Figure 5, in relation to real ships. The answer can only be given in statistical terms; i.e., we can compare the long-term predicted wave bending moment with that calculated from observed stresses. Such correlations have been made for the following ships, for which points in Figure 6 have been compared with some of the curves taken from Figure 5.

Wolverine State	(9)
California Bear	(19)
Fotini L.	(11)
R.G. Follis	(11)
Idemitsu Maru	(11)
Esso Malaysia	(11)
Universe Ireland	(11)

The points plotted represent an extrapolation of the recorded data to cover a ship's lifetime ($N = 10^8$).

In the case of the Universe Ireland the extrapolation of measured stress data given in (11) has since been revised as described below. In view of the fact that the long-term curve for actual weather (Figure 23 of (11)) seemed inexplicably low, the first point for consideration was a recheck of the extrapolation used. Accordingly, data for Universe Ireland and Esso Malaysia from Figure 15 of (11) were replotted in Figure 7 of this paper, showing the extrapolation above Beaufort 8 that was used in the original analysis.

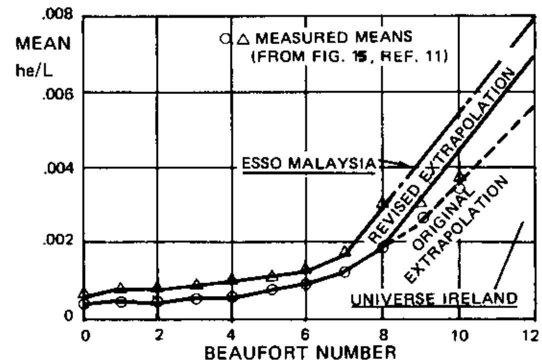


Figure 7. Measured and Extrapolated Bending Moment Coefficients vs. Wind.

Reference to Figures 10 and 11 of (11) showed that for these two ships the total number of points (records) above Beaufort 8 was only 44 for Esso Malaysia and 50 for Universe Ireland. (Table 3 (11) is in error for the former ship). This is a very small sample for both ships, and it seemed quite likely that the waves encountered in these high Beaufort numbers happened to be higher for one ship than the other. Hence, it seemed that it would be a safer prediction to retain the Esso Malaysia extrapolation curve of Figure 7 and to modify

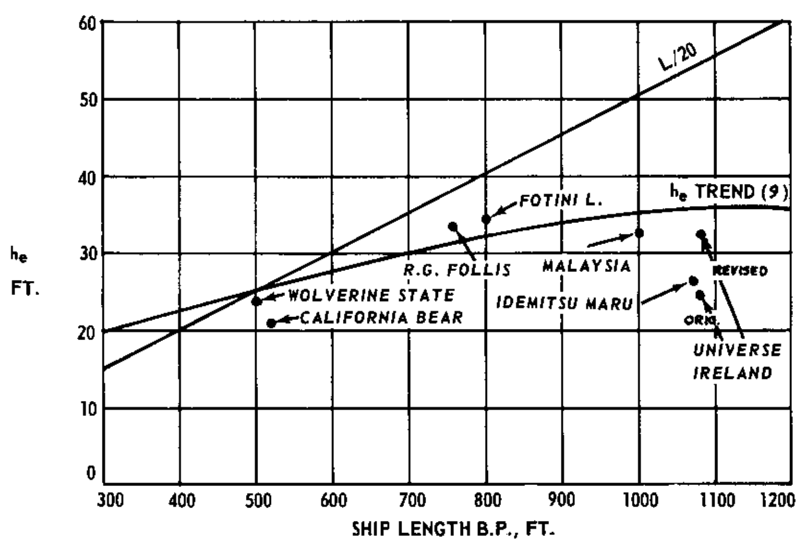


Figure 6. Long Term Distribution of Bending Moment in North Atlantic Weather.

the curve for the Universe Ireland as shown, which represents an increase of about 20%. At the same time, Figure 15 (11) showed that the standard deviations at Beaufort 8 and above were assumed as follows:

Esso Malaysia	1.4×10^{-3}
Universe Ireland	1.0×10^{-3}

It seemed to be more reasonable to assume the same value of 1.4×10^{-3} for both ships.

Accordingly, the long-term calculations were rerun for the Universe Ireland in actual weather, and plotted in Figure 8 (Figure 23 revised), which also shows the original extrapolations for all the ships. It will be noted that at log N = 8, the stress is increased from 16.0 to 18.7, an increase of 17%. This is believed to be a safer curve to use than that given in the original paper.

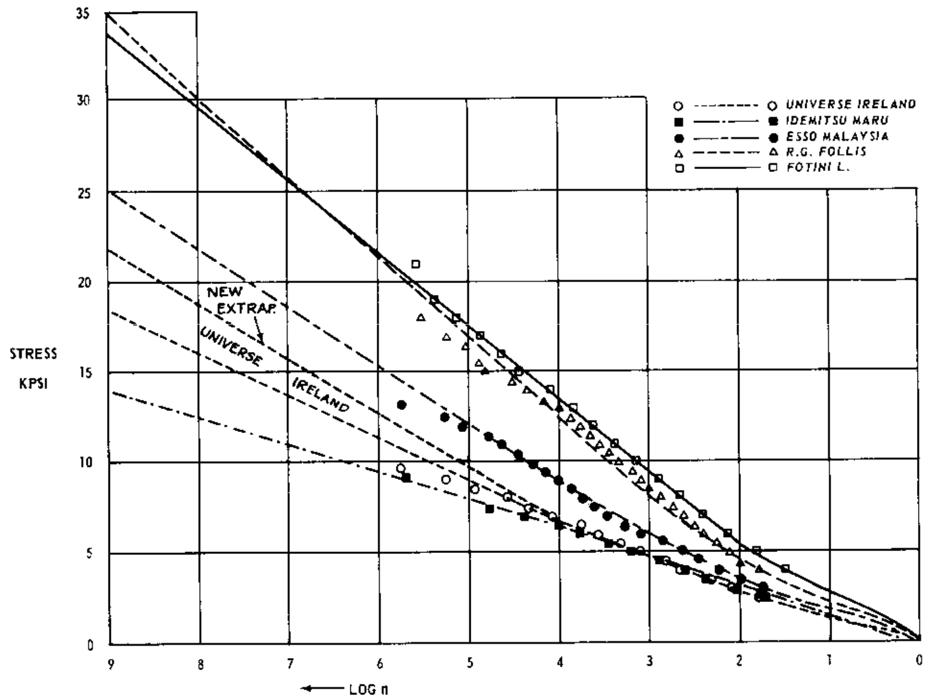


Figure 8. Long Term Distribution of Stresses in Actual Service. (Figure 23 of (11))

Subsequent to carrying out the above revised calculations, long-term stress data were obtained from Lloyd's Register on several large ships in the same service. See Table I following. These data were converted to bending moment coefficients and plotted in Figure 9, along with the original and revised curves for the Universe Ireland. In comparing these results, it should be borne in mind that the British data may include the effects of springing and whipping, since the records were not filtered as in the case of Universe Ireland. The Myrina was known to experience significant springing stresses (20). Also Lloyd's method of extrapolation may be different from that used here.

Table I
Ship Particulars

Ship	Length ft.	Breadth ft.	Depth ft.	Draft ft.
Esso Malaysia	1062	154.8	77.8	60.4
Esso Northumbria*	1143	170.0	84.0	65.4
Myrina*	1050	155.0	77.0	58.0
Universe Ireland	1135	175.0	105.0	81.4

*Ship Nos. 47 and 43, respectively, of BSRA program (21).

It is concluded that the revised long-term curve in Figure 8 for the Universe Ireland appears reasonable for the actual weather experienced.

CORRELATIONS BETWEEN MODELS, THEORY AND FULL-SCALE

Another type of correlation of interest is between predictions based on model tests and/or theory and full-scale trends. A previously unpublished correlation for the Universe Ireland will first be presented. The correlation involves first a comparison of model and theoretical response operators, followed by a comparison of long-term predictions with full-scale statistical results.

A sample comparison is given in Figures 10 and 11 for head seas only

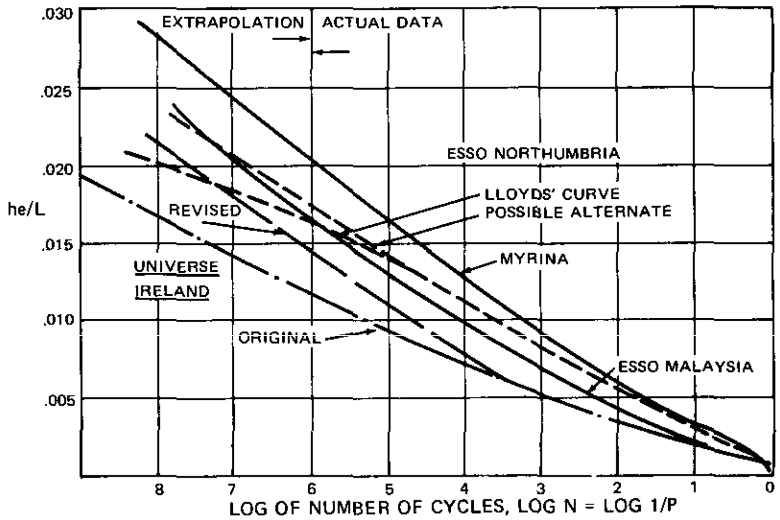


Figure 9. Long Term Extrapolations for Several Ships.

between response operators obtained from Davidson Laboratory model tests (22) and by means of strip theory calculations using program SCORES (23) as modified and extended at Webb. Other headings are given in (24). It is shown that in general agreement is very good, especially when it is considered that the calculations involved in determining short and long-term trends tend to average out any small differences in response operators. The long-term calculations discussed in this section made use of the calculated response amplitude operators.

It is important to note again the large difference in response between the full load and the ballast conditions. In large, modern ships with drafts far greater than other vessels, the Smith effect correction causes a significant reduction in bending moment in the deep loaded condition. Consequently, the ballast condition may be the governing one for design.

The next problem was to obtain suitable wave data for the ship's route, and to make reasonable assumptions regarding the corresponding wave spectra, in order that short-term bending moment predictions and hence long-term trends could be determined. Several sources of wave data were used, and results of calculations are presented in the following paragraphs.

One method made use of Hogben and Lumb wave data (25) on frequency of occurrence of different wave height and period groups for this service. Each combination of observed wave height and period was fitted to a member of the ISSC two-parameter spectrum formulation (26). Short-term and long-term calculations were then carried out for both full load and ballast conditions (27). See curves A and B of Figure 12. If the ship actually encountered severe seas in this comparatively light ballast condition, then the ballast curve should be the basis for design. It may be seen that both curves overestimate the long-term trend, however.

For comparative purposes alternate calculations were made on the basis of the Webb "wave height" family of spectra, following the procedures described in (9) for a weather distribution typical of the North Atlantic instead of the actual ship route. Figure 13 shows the results of short-term calculations -- bending moment coefficient h_e/L (mean rms and standard deviation) as a function of significant wave height. The assumed weather distribution and results of the long-term calculations are shown in Figure 14, where both the load and ballast conditions are shown. It is the results at 10^8 cycles that were plotted in Figure 5 and the ballast condition is seen to be higher than the value in Figure 6 predicted from full-scale measurements. Presumably this is because of the more severe seas the ship would encounter in the North Atlantic.

HYDRODYNAMIC PRESSURE DISTRIBUTIONS

An important aspect of the theory of ship motions has been that not only can it provide a basis for calculating the longitudinal distribution of vertical forces -- hence shear and bending moment -- but it can permit the complete distribution of hydrodynamic pressures over the hull to be determined (28)(29). With the advent of sophisticated finite element techniques of stress analysis, such a detailed definition of hydrodynamic loads at any specific instant in the cycle of ship motion is essential. Work at Webb has been completed for the head sea case and will soon be available for oblique seas as well. A paper on the subject by D. Hoffman and C. Hsiung is in preparation for the STAR Symposium.

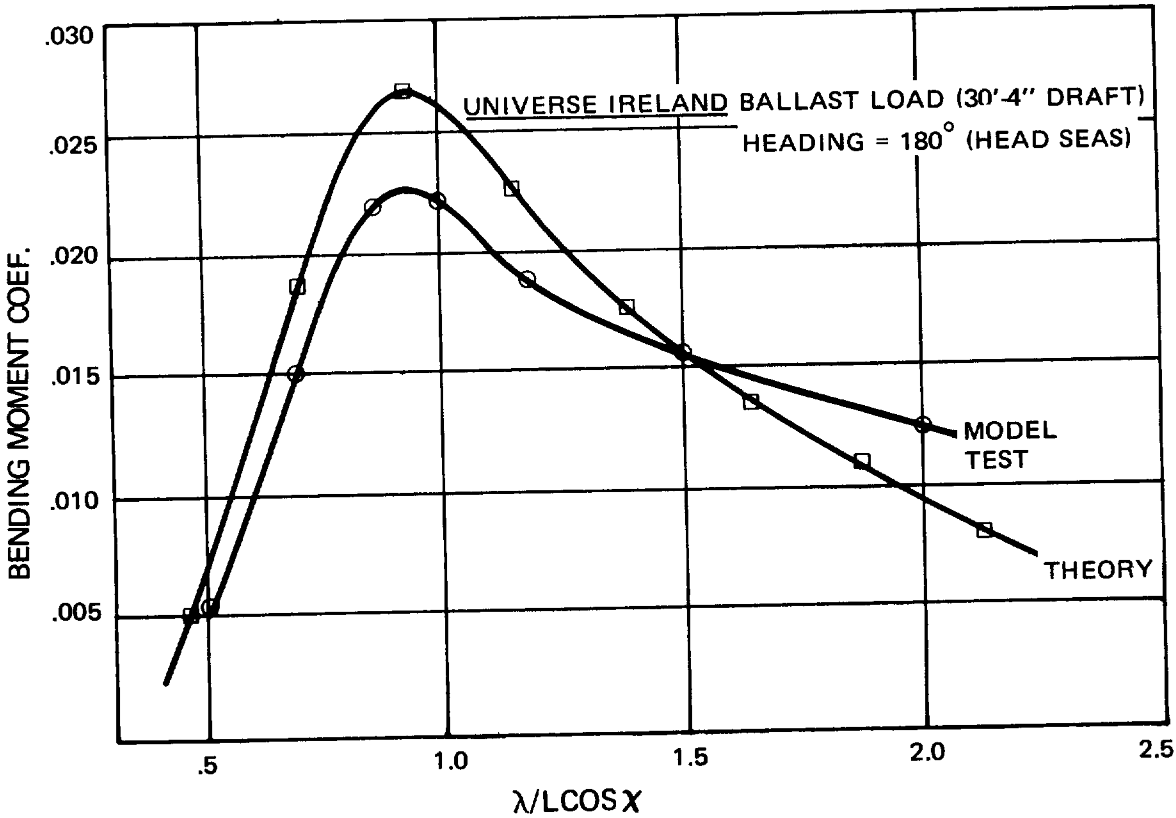


Figure 10. Comparison of Model Tests and Theory.

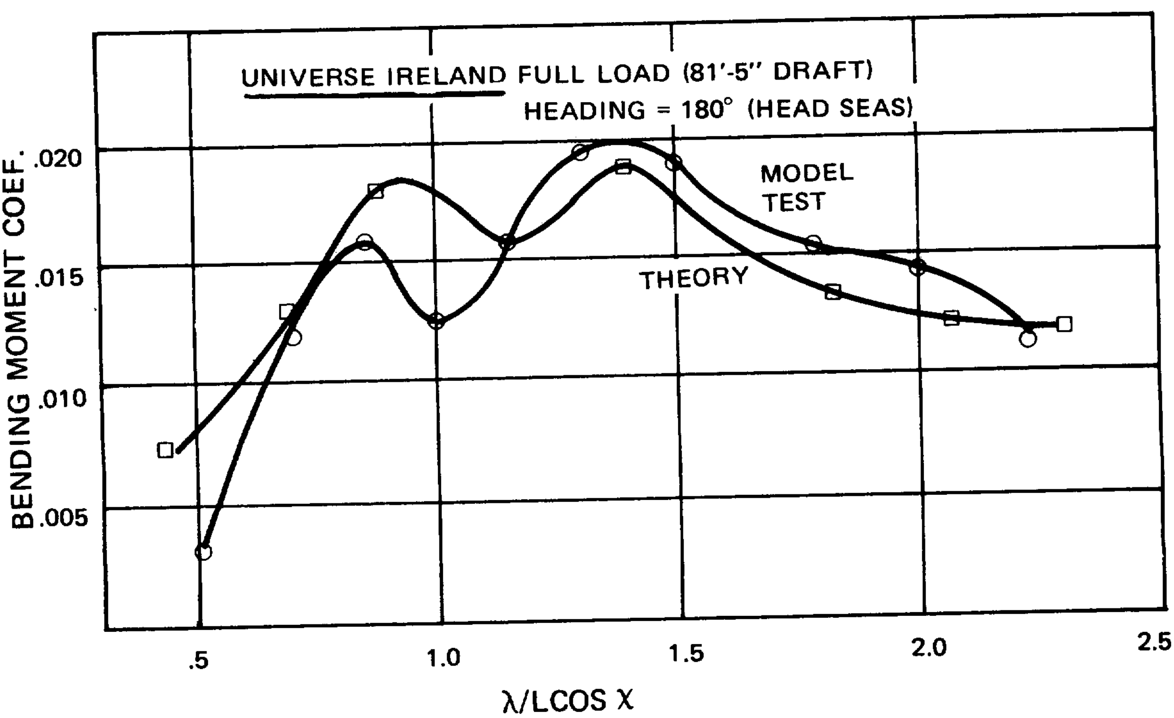


Figure 11. Comparison of Model Tests and Theory

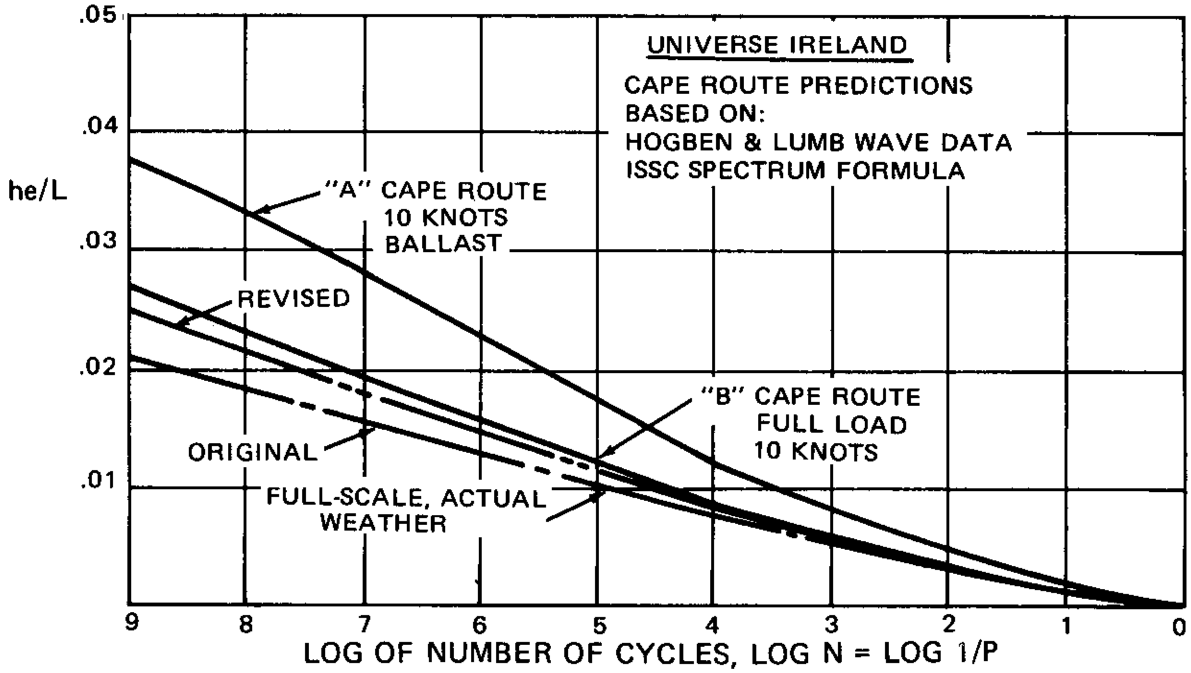


Figure 12. Comparing Long-Term B.M.'s with Predictions Based on Published Wave Data.

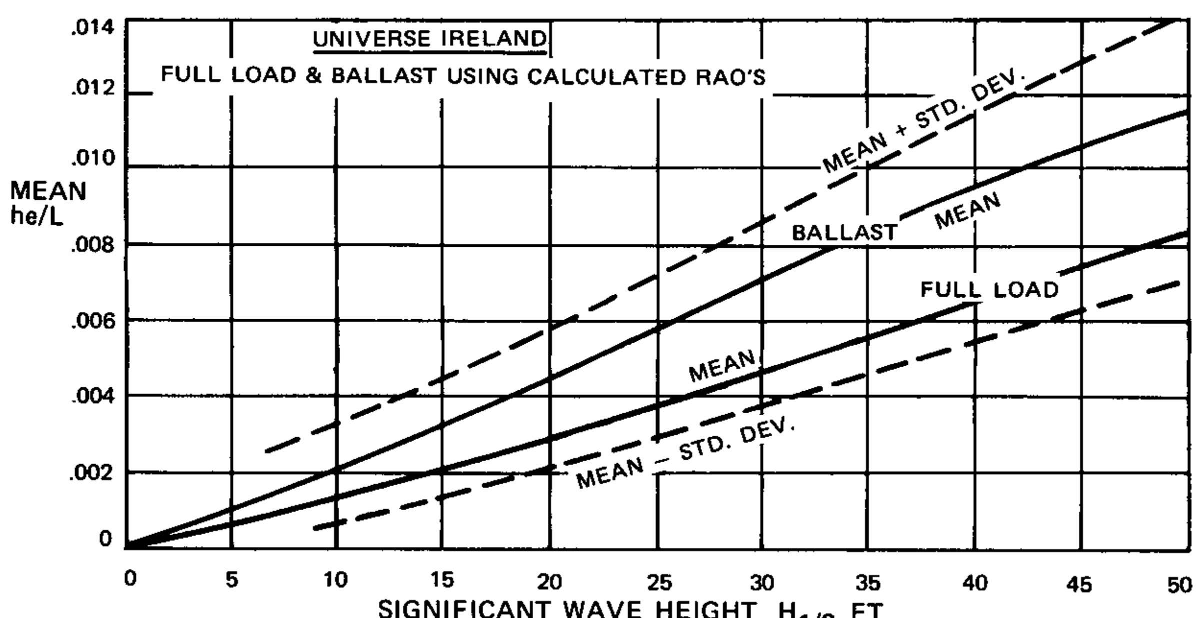


Figure 13. Bending Moment Trends vs. Significant Wave Height.

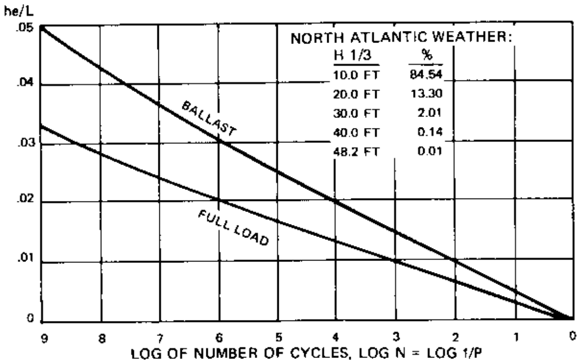


Figure 14. Universe Ireland - North Atlantic Prediction.

VIBRATORY RESPONSE

High-frequency or vibratory responses can as a practical matter be separated from quasi-static responses. For example, Figure 15 shows at (a) a typical midship stress record containing several modes. In (b) the high-frequency response and in (c) the low-frequency response has been filtered out. In (c) only the first mode response is visible, although higher modes might also be present.

The vibratory modes of hull girder response can be considered to be subdivided on the basis of the nature of the excitation into transient and cyclic. The former category is generally described by the terms slamming and whipping, where slamming refers to the initial effect of a wave-ship impact and whipping to the consequent hull vibration in one or more modes. Cyclic responses can be self-excited, as by ship's machinery or propellers, or externally excited by encountered waves. Wave-excited cyclic responses are of particular interest here and are generally referred to as springing.

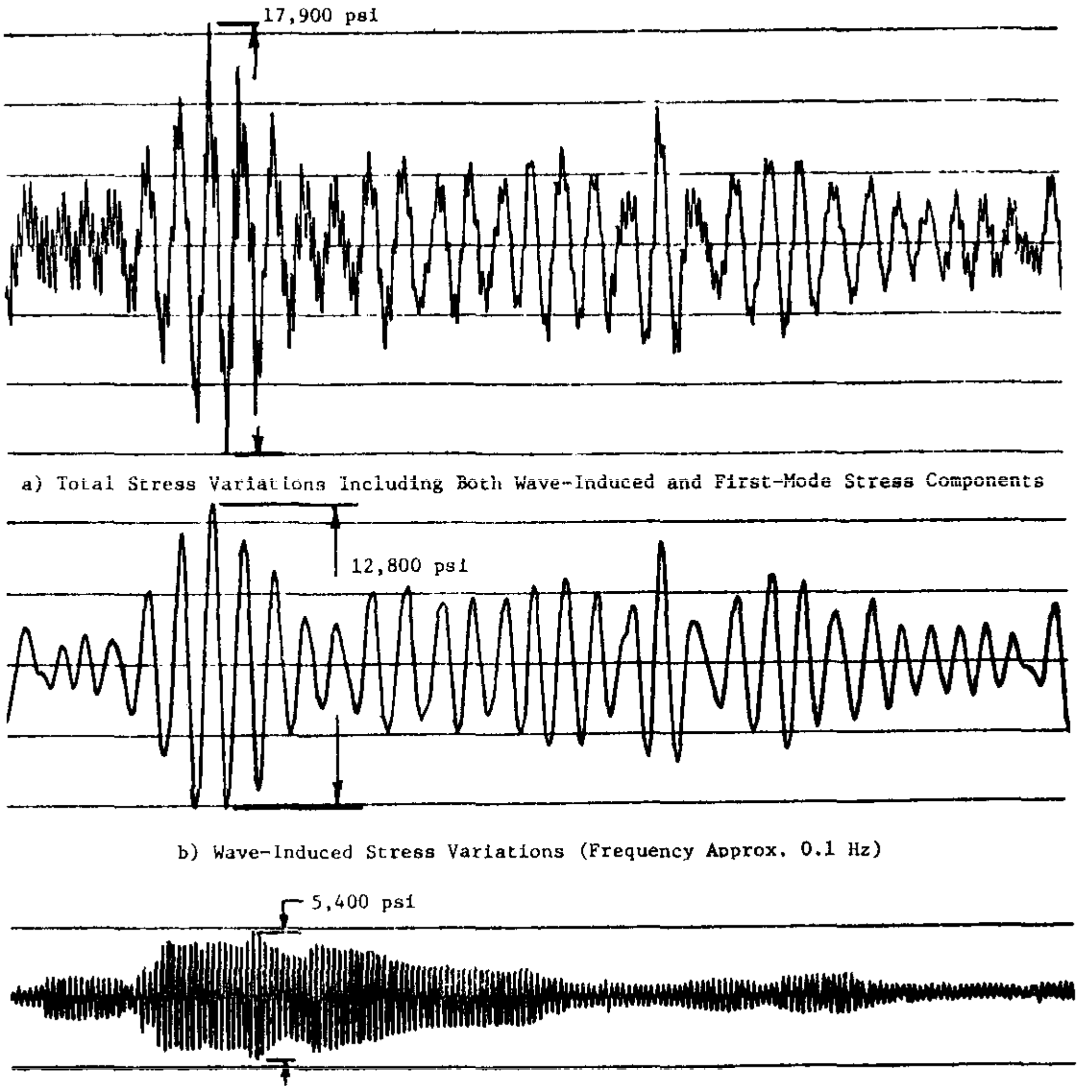
Both the transient and cyclic hull responses can in principle be handled by the theory of vibration of a free-free beam. However, there are more difficulties here than in the case of quasi-static loadings. First of all, the dynamic response of a ship hull does not follow simple beam theory. In the case of a typical cargo ship with double bottom it has been hypothesized (30) that it can be described as a composite beam consisting of the double bottom, having certain elastic properties, and the superimposed hull having other properties. A second problem is that the cargo and other loads carried by the ship seem to behave like sprung masses whose dynamic properties are difficult to compute. Third is the problem of damping, which is twofold: internal, involving the structure and the cargo loads, and external, involving mainly hydrodynamic effects. Both are difficult to calculate, but the former can be determined experimentally on full-scale ships, as by anchor-drop or shaker tests. Hydrodynamic effects are more troublesome to evaluate accurately.

Slamming has been studied extensively but is still far from being completely understood. One aspect adding to the difficulty is that it is to some extent under the control of the shipmaster, since severe slamming can be ameliorated by a reduction in speed and/or change in course. A particular concern is usually fear of local bottom damage that would necessitate drydocking the vessel for repair. This provides a sort of safety valve that limits the magnitude of wave impacts and hence the severity of the hull girder response. In some modern ships with bridge aft it may be difficult to detect bow slamming, but new instrumentation is being developed to assist the ship's officer. On the other hand, if special pains are taken in design to minimize the danger of local damage, through use of thicker bottom plating, local reinforcement, higher strength steel, etc., larger impact loads can be permitted. In this case the ship may be driven harder and subjected to higher dynamic hull girder stresses.

For the ship designer it is important to consider the phasing between the slam-induced loads and those due to quasi-static wave action. Some work by van Hooff (13) indicates that the initial slam response (slamming) seldom adds significantly to the initial sagging bending moment. However, the whipping that follows a large slam will always add to the first hogging moment, and often to subsequent quasi-static peaks.

Another type of transient loading is that associated with flare immersion. In a ship having considerable bow flare not only can a large transient force build up, but it will have a longer duration than a bottom impact, and therefore fundamental beam theory (31) suggests a greater dynamic load factor. An example of such a situation was given in records of hull girder stresses on an aircraft carrier rounding Cape Horn (32). In this case the whipping stresses associated with flare immersion were of the same order of magnitude as the quasi-static wave stresses. See also Figure 15.

Full-scale measurements of slamming and whipping stresses are given by Aertssen (33), Wheaton, et al (34),



a) Total Stress Variations Including Both Wave-Induced and First-Mode Stress Components

b) Wave-Induced Stress Variations (Frequency Approx. 0.1 Hz)

c) First-Mode (Springing) Stress Variations (Frequency Approx. 0.70 Hz)

Figure 15. Typical record of midship stress variation, M.V. Fotini L, showing filtered wave-induced and dynamic stresses.

Bourceau and Volcy (35), Meek (36), and others.

Another source of transient loading that excites vibratory response is the shipping of water on deck forward. In many cases this load may simply be the static head of the water scooped up by the bow, acting downward until it runs off. The duration of this load therefore is relatively long, more like flare immersion than a bottom slam. However, there may be a dynamic component, especially if the ship is moving forward at high speed into head seas. The water in the wave crest will be moving in a direction opposite to the ship and therefore its velocity is additive to that of the ship. Since the bow will normally be pitched downward at the time of shipping water, a sizable dynamic force downward can result. Experimental values of pressures from model tests have been reported by Tasai (37).

Shipping water can be predicted on the basis of the same calculations of relative bow motion used to predict slamming. The only condition in this case, however, is that relative motion exceed the bow freeboard. Such predictions have proved reasonably satisfactory, but are subject to error from the bow wave due to forward speed and from non-linear effects (38).

The whipping that results from shipping water may be more significant than the relatively small increase in hogging moment. It has been calculated and compared with model test results (39). It has also been recorded full-scale by Aertssen (40). Ferdinande (41) discusses a case in which whipping was induced by the emergence of the bulbous bow of an ore carrier.

In the past there has been some question as to whether or not the full magnitude of high-frequency stresses should be assumed to be superimposed on still water and quasi-static wave bending stresses. It appeared possible that the duration might be too short to allow time for the large energy absorption involved in panel buckling or ductile tensile failure. A further question was whether only the fundamental response of the hull girder to slamming impacts need be considered, while the higher harmonics that damp out quickly are ignored.

The answer to both of the above questions seems to depend on the nature of the impact. Bottom slamming is characterized by large hydrodynamic pressures but very short duration, while flare entry -- an increasingly common phenomenon with recent highly flared bows -- is of appreciably longer duration. As previously noted, beam theory (31) shows that when the impact duration is very short relative to the natural period of the structure, the dynamic response is relatively small. But when it is of the same magnitude a magnification factor of 2.0 can be attained.

Hence, the immersion of a wide flared bow is a more serious threat than bottom slamming to the hull girder. This has been clearly demonstrated in a recent paper by McCallum (42) in which three cases of deck and/or upper side shell buckling are reported. (Two other ships had local forecastle damage only.) The ships involved were 20 to 22-knot cargo ships of 430-530 ft. length, with relatively large bow flare. The buckling was clearly the result of a dynamic loading, because the combined still water and quasi-static wave bending was predominantly hogging. Apparently, higher modes of vibration were significant because large bending moments extended far forward of midships. Since the section modulus was reduced at a distance from midships, the resulting stress increased to a maximum at 0.2 L from the F.P.

Ferdinande (43) reported that a higher mode of vibration, which damped out quickly, was responsible for a sharp amplification of the first whipping stress peak of a record from the ore carrier Mineral Seraing.

The relative importance of bow flare impact was confirmed by Aertssen in his discussion of (42). "Two tables in the appendix of my Jordaens paper mention the whipping stresses in main deck amidships .... Curiously enough, because I did not expect it at all, in medium-loaded condition the highest whipping stress was $2 kg/mm^2$, whereas in full-loaded condition the highest whipping stress was $3.6 kg/mm^2$. This means that the whipping stresses were lower in the classic bottom impact of Ochi than in the bow flare impact".

It may be concluded that, although the seriousness of bottom slamming and whipping for longitudinal strength remains uncertain, there is no doubt of the gravity of flare impact effects on high-speed ships. Consequently, Lloyds has adopted special rules applicable to ships having excessive flare forward (42).

The steady-state vibratory effect known as "springing" has been noticed particularly in Great Lakes bulk carriers (44), but it has also been reported on large ocean-going ships of full form (45)(46). A clue to its origin is given by the fact that the Great Lakes bulk carriers are quite shallow in depth and consequently have unusually long natural periods of vertical hull vibration (two-noded periods of 2 sec. or longer). The explanation is that when the ship is running into comparatively short waves which give resonance with the natural period of vibration, significant vibration is produced. This vibratory response may continue over some period of time, gradually fluctuating in magnitude. A corresponding fluctuation in stress amidships is therefore superimposed on the quasi-static wave bending stress. The springing stress appears to have the characteristics of a stochastic process, but one that may be partly independent of the low-frequency wave bending, which is also treated as a stochastic process (47)(48).

The phenomenon of springing has been studied both experimentally and theoretically at Webb Institute of Naval Architecture (49)(50). Using a jointed model of a Great Lakes bulk carrier and running it in very short waves producing resonance with the natural frequency of vibration, large vibratory responses were obtained.

The well-developed strip theory of ship motions has been applied to springing in short waves (46). Although motions of a springing ship may then be very small, the theory provides information on the exciting forces acting on the ship in the short waves that produce springing. Hence, when these forces are applied to the ship as a simple beam the vibratory response can be predicted. Despite the fact that strip theory is not rigorously applicable to such short waves, good agreement was obtained at Webb between theory -- after a number of refinements had been made -- and experiment.

These correlations have confirmed the hypothesis that increasing hull flexibility has an unfavorable effect on springing. They have also shown that increasing fullness is also unfavorable, because the wave excitation comes about primarily from short-wave effects concentrated at the blunt ends of the ship, which are anti-nodes. In the case of a fine hull, the wave forces are relatively small and distributed along the length of the ship.

Proposed new standards of strength for Great Lakes bulk carriers -- applicable also to full oceangoing ships -- are now being developed under the cooperative efforts of U.S. Coast Guard, American Bureau of Shipping, NSRDC, SNAME Panel HS-1, Webb Institute and others.

THERMAL EFFECTS

Records of midship stress obtained on five bulk carriers (11) 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 (11). 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 stress readings. Because of the effect of local shading it can be expected that even larger thermal stresses would be experienced locally on one side of the ship. 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 (11) 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 by means of available theory, assuming simplified structure and using estimated temperature changes. The simplified procedure was applied to the tanker <u>Esso Malaysia</u> first (51), because records of the measured diurnal stress changes and some temperature data were available.

Under the assumed conditions the calculated thermal stress at deck edge due to temperature change was about 2000 psi. (Average of 1600 at center of deck stringer plate and 2300 at sheer strake.) From the measured stresses during the same period of time (Figure 28 of (11)) the 11 day-night or night-day stress variations in KPSI were 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 was concluded that the approximate calculation of 2000 psi was satisfactory. Typical stresses given elsewhere (52) are higher because they include unsymmetrical temperature gradients.

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 <u>Climatic Atlas of the World</u>, Volume VIII (53). 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, ΔT, due to radiant heating of the deck by assuming that the air-deck, ΔT, is directly proportional to the extent of cloud cover. Thus the maximum 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 air-deck ΔT's are added to the sea-air ΔT's (from ship logs or statistical climatic data) 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 <u>Atlas</u> (53). A sample calculation is shown in (13), where the method is applied to the <u>Wolverine State</u>.

Other local thermal stresses to be

considered in design result from the effects of heated or cooled (refrigerated) cargoes.

DESIGN LOAD STANDARDS

It is now well-established that a design bending moment must include at least two separate components -- still water and wave bending moments. Thanks to recent research, the state of our knowledge is quite good regarding techniques and procedures for establishing both of these for modern large and/or fast ships. One exception is that more information is needed on wave data for certain ocean areas, such as the vicinity of the Cape of Good Hope. Attention should also be given to wave bending moments under ballast conditions -- especially if the tendency toward reductions in ballast continues.

The urgent problem at the present time is the prediction of vibratory effects superimposed on the above loads. Two cases are of prime importance:

(a) Springing of full, flexible hulls, such as Great Lakes bulk carriers.

(b) Whipping following slamming (bottom or bow flare) of high-speed vessels.

For the former, research has provided suitable calculation techniques for determing trends (46)(50). But pending completion of research on the development of techniques for reliably predicting slamming and whipping effects in the design stage, it is necessary to rely on empirical data for similar ships in service for the latter effects.

Since whipping is a particular problem for high-speed, fine-hulled vessels, it appears that there are two aspects of hull fullness in relation to longitudinal strength standards, and that these have opposing effects. First is the effect of fullness on quasi-static wave bending. This can be calculated or determined by model tests, and in general a reduction with reducing fullness is indicated. Second is the indirect effect of the increasing speed usually associated with reduction in fullness and the consequent increased possibility of superimposed dynamic loads.

Pending the further development and confirmation of methods for predicting slamming and whipping stresses (54), perhaps the best approach is that recommended by Aertssen (55). He suggests an additon of 60% of the whipping bending moment (hog or sag) to allow for bottom slamming on a medium-speed cargo liner.

For greater generality, Ferdinande has collected data on the ratio s/S, where s is whipping stress and S is wave bending stress, and his results are reproduced in Table II (56). He explains, however, that this ratio does not give directly the addition discussed by Aertssen because the maximum vibratory response may not occur at the same instant as the maximum wave bending.

For example, he presents a record of severe slamming on Dart Europe (33), reproduced in Figure 16. The ratio s/S was 0.82, but the maximum combined wave and springing stress (hogging) exceeded the wave stress by only 48%. Hence, the percentage of the springing stress to be added to the wave bending stress was,

$$0.48 \ / \ (s/S) = 0.48 \ / \ 0.82 = 0.59$$

which agrees well with Aertssen's figure of 60%. Of course, these factors can be applied equally well to bending moments as to stresses.

In other words, a tentative standard for the addition for whipping expressed as a percentage of wave bending moment would be

$$0.60 \times s/S$$

Values of s/S must be obtained from observed stresses on similar ships in service, as tabulated in Table II.

A similar but larger allowance for flare immersion should be added in the case of large flare, as called for in recent new Lloyds regulations (42).

Finally, thermal effects must be either calculated and combined with other loads, or allowed for in the factor of safety. As mentioned previously, research has provided satisfactory calculation methods. It has been recommended that, if thermal effects are to be explicitly allowed for, an average figure (sunny and cloudy weather) be used, since extreme thermal stresses are very unlikely to coincide with extreme wave bending moments.

The authors are firmly convinced that research will ultimately permit wave-induced loads on ship hulls to be incorporated not only into a complete probabilistic picture of hull loads but into a philosophy of design based on so-called reliability theory. This means that the resistance of the hull to applied loads (capability), as well as the loads themselves (demand) will be expressed in probability terms. Hence, the probability of failure of a ship in its lifetime can be determined and the design adjusted to insure an acceptably low value. Such an approach is important not only to insure optimum design of present-day ship types but to provide a basis for the design of new and unusual types of craft that are continuing to appear.

TABLE II

Whipping Stresses (56)

Author	Ship	Whipping stress range, s(MN/m^2)	$\frac{s}{S}$	Remarks
Schnadel	SAN FRANCISCO (dry cargo)	22	0.35-0.5	
Jasper, et al	ESSEX (air craft carrier)	120	> 1	bow flare
Bennet	CANADA (dry cargo)	20		
	MINNESOTA (dry cargo)	38		
Bledsoe, et al	Dutch destroyer	110	1	bow flare
Aertssen	LUKUGA, JORDAENS (dry cargo)	40		assumed
	MINERAL SERAING (ore carrier)	60		extreme
	ROI BAUDOUIN (car-ferry)	20		
Ferdinande	JORDAENS loaded	36	0.30	bow flare + green water
	medium loaded	20	0.31 (avg)	bottom + bow flare
Maclean, Lewis	WOLVERINE STATE (dry cargo, light loaded)	70		bottom
Wheaton, et al	WOLVERINE STATE	31	0.20	bottom
Kosugi, et al	HODAKASAN MARU (dry cargo)	45		
Aertssen	DART EUROPE (container ship)	160	0.82 (max.)	B10 - 11 T_{FP} = 7_m
Meek, et al	FLINDERSBAY (container ship)	120	2.4 (max.)	bow flare
Nibbering	OSSENDRECHT (bulk carrier)	30	0.33 (max.)	B11 - 12 (2 days) ballast, N. Atl.
Bourceau	47,00 dwt tanker	25	0.85	F_{pp} = 7.80 m
		47	0.92	bottom
Ojak	fishery mother ship (L = 165.5 m)	16	0.90	stern slamming
A.S.W.C.	OCEAN VULCAN	24		

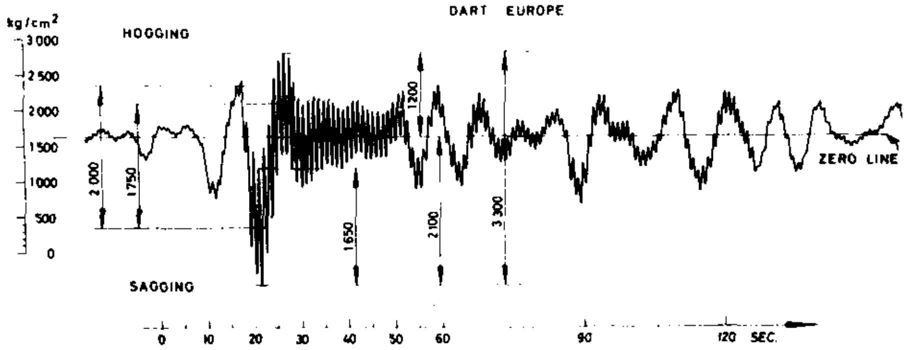


Figure 16. Stress Record Obtained on S.S. Dart Europe Showing Combined Wave-Induced and Whipping Stresses.

Freudenthal has shown (57) how acceptable levels of failure probability can be determined on economic grounds. A trial numerical application of this approach to ships (13) has demonstrated that, from a cost point of view, fatigue damage costs may be approaching the costs of rare hull failures. In other words, the accumulated expenses incurred by nuisance cracking may require that more attention be given to fatigue in design.

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DISCUSSION

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The authors are to be congratulated on the explanations of today's design criteria design methods and tomorrow's problems presented in this paper. Additionally, as the authors note, most of the research upon which their paper is based has been funded by the American Bureau of Shipping over a period of the past 15 years. Even though a classification society depends to a great degree on experience of successful previous designs, in this instance the classification society has recognized the problems presented by the many new designs and has undertaken a significant research program in order to substantiate the changes which have been felt acceptable and necessary for the newer ships of today.

The paper does not attempt to solve all questions on ship loading. There are a number of questions which remain unsolved by present research, many of which are noted by the authors. In the introduction to the paper, after noting the importance of the "Ship to Wave Matching", the authors go on to state that the ship's hull follows the simple beam theory quite well, provided that areas of stress concentration are specially considered. It should also be noted that besides areas of stress concentration, there were instances in the Great Lakes testing (during the Ryerson tests) wherein two midships strain gauges located on the main deck, one inboard of the other, seemed to change their ratio of load acceptance as the overall stress level increased. When this point is followed up in more detailed tests, it may require a variation in the stress concentration factor according to the intensity of stress experience.

In the same paragraph, the authors finish their observations with the statement that it is never possible to obtain a complete picture of the sea at a particular time to make a direct comparison. This discusser, while agreeing that thus far in ship research this has been the case, would hope that the authors would also have some hope that in the future, a properly conducted test with fully directional seaway observation and instrumentation would be able to obtain a simultaneous picture of the sea condition in order that direct comparisons can be made. Indeed, in the most recent meeting of the SNAME panel HS-1 and its Great Lakes subcommittees, it was held to be a virtual necessity for final understanding of the theoretical work done by Webb Institute and the other researchers that such a test must be carried out. This discusser feels that the instrumentation necessary to achieve such a test is currently available and is merely a matter of funding, timing, and perseverance plus willingness on the ship operators part to make the ship available, even including some short delays for a day or so in order to carry out such tests.

In the section labeled "Critical Loads", the authors begin by considering the ultimate failure as defined by Caldwell with which there can be little argument. However, such a statement does need interpretation to the better understood by all readers. For instance, the Coast Guard would consider a "Class One Failure", any failure of the main hull girder even if only of the tension or compression flange, if it is considered enough to lead to total failure within a short period (i.e. - a few minutes in a storm situation, a few hours in a mild seaway condition, or within one voyage if any significant seaway conditions might occur before the ship could reach safe harbor).

The authors consider that the rarity of total hull failure is an indication that the conventional standards of strength are generally adequate. This discusser finds himself in a position of general agreement to that point in the sentence, but in substantial disagreement when the authors finish the sentence by saying, "in fact, they may be excessive". In this discusser's opinion, the conclusion of this sentence is excessive in its tone. It can be generally accepted in 1975 that there was room in the 1950's for reduction of the hull scantlings from the straight line extrapolation of the section modulus required by an $\frac{L}{20}$ wave. But, it is this discusser's opinion that the discovery of the many other loading phenomena which are just being recognized as significant for ship stress in today's larger ships is in part due to the fact that the previous standard of strength for larger ships has already been reduced from the extrapolated $\frac{L}{20}$ standard over the past 15 years by a series of modifications to the classification society rules and that further steps should be taken extremely cautiously. The authors note the results of the Committee on Larger Vessels and the Webb recommendations during the 1960's.

In the next section under "Still Water Loads", the authors introduce the probabilistic approach stating that calculations can be made and verified by statistical records of the average value of bending moment in typical loading conditions and also the standard deviation for each basic condition. The term "standard deviation" has a rather exact mathematical definition. Yet, the possible variations of loading in the lightly loaded conditions and the subsequent bending moments and stresses are larger in the light conditions especially for the break bulk carrier or a container ship. Therefore, the standard deviation in the mathematical sense might not have a great deal of meaning if that is the way it is used.

The section on "Wave Bending Moments", is perhaps the heart of the authors paper and is certainly of extreme interest to all designers. We are indebted to the authors for the several comments made on Figure 5 and for the introduction of the effective wave height concept. There are one or two points which are not clear to this discusser as to the final use of the wave bending moment as described in the paper. In the second paragraph, the authors label the Navy design wave height as being realistic, but then note that an allowance for still water was included as if it should not have been included. In this discusser's opinion, for many years the section modulus of a ship was set by a single formula supposedly encompassing all loading and there was a multiplier coefficient in the formula which was there for the purpose of providing the design safety factor plus covering all factors of ignorance. It would make things clearer if the authors would redefine the present use of the wave bending moment.

The authors state that the coefficient C depends on the interaction of the wave form and the hull form of the ship. It stands to reason then that the effective wave height H_e will vary if there is a significant variation of the interaction between the wave form and the hull form of the ship. It will also be vulnerable to error if the statistics gathered to develop H_e are from trade routes which are less than the most severe to be encountered in the world. It would be improper, in this discusser's opinion, to utilize an H_e for large tankers based entirely upon readings taken on the Persian Gulf to Europe route even though there are occasionally large wave conditions at the Cape of Good Hope. Similarly, voyages from the Persian Gulf to Japan are along the fringes of the greater Eurasian land mass and not directly across the most exposed latitudes of the Indian and Pacific Oceans. If the statistical base did not also include the expected wave severity from the North Pacific, large tankers intended for a lifetime of service in the Gulf of Alaska and Bering Sea throughout the year might find themselves underdesigned.

The statement that the calculations show that the full modern tankers actually show a lower trend of H_e because of their extremely large draft is also quite interesting. This statement seems proper for the deep draft ship, but there has been recent interest in building VLCCs which will be shallow draft tankers with an $\frac{L}{D}$ of between 15 and 20. How much would the the coefficient in the formula have to be modified for such ships? The authors should be commended for pointing out that in the case of very large ships and bulk carriers in general, the design approach must include an evaluation of the ballast or light load conditions. Indeed, as the authors point out, some ships need to be designed for special light load conditions in order to avoid exceeding their maximum bending moment allowance. The design method then, should be one not based on draft, but one simply based on the form of the ship and the expected severity of service. This further means that PNA needs to be changed in Chapter 4 wherein several formulas for design bending moment are functions L^2, B and T (full load draft).

The discussions entitled "Correlations between Calculations and Full-Scale" seems to support some of this discusser's fears that the modification of design standards may be moving a little too quickly. The plot shown by the authors in Figure 7, appears to this discusser to show that since revised extrapolations became necessary on the data originally utilized from the UNIVERSE IRELAND instrumentation, we may very well be in a position of revising our data extrapolations on all large vessels. The authors note their work suffered at this point from very small samples for both ships.

In the next section entitled "Correlations Between Model, Theory and Full-Scale", the authors again come back to the point that the ballast condition may be the governing one for design. They further indicate in Figure 9 that the present Lloyd's curve may have to be increased if the revised UNIVERSE IRELAND figures are accurate. In reviewing Figure 13, the authors show the UNIVERSE IRELAND in full load and ballast in significant wave heights a mean ratio of the observed effective wave heights to length ratios. They also utilize what appears to be a mathematical calculation of standard deviation once having established this mean. It appears necessary to this discusser to remind the designer that although the ship is in a design significant wave height (of say 35 feet) there will be one in three thousand which will be well over 60 feet. The designer must then realize that mere utilization of the mean plus standard deviation as an idealization of the highest stress that this ship will see might not be a proper evaluation of the ultimate design stress as was stated earlier in the paper.

The section on "Design Load Standards" is a compilation of problems and new types of stress which the authors suggest ought to be taken into account in some overall design load standard. This discusser would agree that the items mentioned by the authors should certainly be taken into account. But, how should these be handled in the voluntary rules which are utilized by vessels which are classed by classification society? Additionally, shouldn't the standard for classification be higher than the minimum federal standard since classed vessels should be an example of a higher degree of design safety? In civil engineering practice, safety factors of 4 and 5 are often used for machinery or structures subject to impact or dynamic loading. Ships have impact, dynamic loading in many different planes, and a continuously changing foundation. It would seem prudent for the ship designer not to throw away all the safety factors at this time.

In conclusion, the design approach delineated in this paper is not just of passing interest to the ship designer. It is critically important since it is the framework of a new method of establishing a structural design standard for ships. It is essential, therefore, that the designer understand not only what is being proposed, but also the limitations and the versatility of the method with regard to new designs and especially the several places indicated by the authors as areas in which research is not completed to the extent a fully confident standard can be set. It is a paper which should at least awaken each responsible ship designer to today's problems.

William H. Buckley, Associate Member

It is difficult to review this excellent paper without becoming aware of the disparity between the current approach to hull girder strength standards and the essentially rational approach to determination of seaway loadings reflected in the authors' research. For example, in discussing the effective wave height formulation vs. ship length the authors state that "The curve adopted in the new 1975 ABS rules is shown in Figure 5. It is lower than the Webb curves because all still water loads are excluded. This is reasonable because h_e values are to be used in conjunction with an allowable stress that is well below the yield point of steel and hence allows a sizeable margin of safety." This discussor is somewhat confused by the implication that the recommended Webb values of h_e contain still water loads since this is not reflected in the definition given of h_e. In any case the specification of ABS design wave height curve appears to be directly influenced by material strength allowables which have no immediate bearing on applied bending moments. As the authors state earlier in the paper "The problem of specifying a design h_e is complicated by the question of factor of safety and allowable stress. In simple terms, one may either use some sort of an average high bending moment in association with a low allowable stress (large factor of safety or an extreme, rare value of bending moment with a higher allowable stress.)" From an academic point of view this question would seem to be resolved most directly by using an effective wave height which when multiplied by BM_S results in the highest bending moment anticipated in service for the ship in question. Further studies to modify maximum bending moment estimates to accommodate empirical strength standards do nothing to enhance our ability to understand or predict the hull girder loads which a given ship will experience in a seaway.

The difficulty of translating rationally estimated hull girder bending moments into values compatible with today's design procedures does not end with the subject of effective wave height. As the paper illustrates in Figure 4, the bending moments seen by the hull are the result of a superposition of still-water, wave-induced, slam-induced, speed-induced, and thermally induced loadings. One would expect to combine these in some rational manner for design purposes. For example, reference 13 of the present paper is an interesting study by Professor Lewis and his co-workers in which the various sources of bending moment were investigated for individual maximums, and as they might act in combination under extreme conditions for the dry cargo ship WOLVERINE STATE. The maximum combined bending moments were then compared to the estimated ultimate strength of the hull girder as developed from the ABS rules and found to agree rather well, the allowable bending moment being slightly larger than the extreme applied bending moment. From a research point of view this suggests that rational assessments of total hull girder bending moments are possible. However, what if tomorrow's research is completely successful in this regard? How will these achievements be translated into useful design standards within the context of today's approach which accounts for loadings due to slamming and temperature effects through artifically reduced allowable stresses? Already in the simpler case of uncombined wave-induced bending moments we have philosophical difficulties. As the research progresses these difficulties will certainly become more extensive.

The practical need for translating research findings into a form which will influence current design standards in a beneficial way is unquestioned. What is questioned, however, is the implicit constraint of the research because current design standards do not deal rationally with many of the realities of hull loading. It would appear that the time has come to acknowledge the fact that the research endeavors reflected in this paper are leading us in a direction which is largely incompatible with the existing approach to establishing hull girder strength levels. This is not to say that the current approach is outdated or that a more rational approach is ready to replace it, or that no effort should be made to influence curpnet design standards, but simply that a more rational approach should be allowed to develop.

How can this be done? Only by the adoption of an additional strength design philosophy which addresses itself directly to the realities of applied loads and actual hull girder strength. If it is accepted as a research tool and permitted to develop it may one day supplement the current approach or even replace it.

In their concluding paragraph the authors state their firm conviction that such a rational approach can be developed. This discussor shares that conviction and suggests that the time has come to define the approach in an explicit manner.

Egil Abrahamsan, Member

I find it reasonable that the authors, after many years of active contribution to the naval research, want to sum up their experience and their own work in a paper like the present one. Being merely a survey, however, the paper conveys little new information.

Some significant works on the problems discussed have not been mentioned, partly leading to a disregarding of real pioneer works in favour of the author's own. (Example: Work on short- and long-term statistics of ship responses). There are also several minor inconsistencies and questionable contentions in the text, but none of these are of such importance that they should be discussed here.

In their treatment of wave-bending moments, the authors point out the importance of considering the ballast conditions. The indicated trend is in agreement with a research study recently undertaken at Det norske Veritas for a VLCC. Further investigation of this subject should be undertaken particularly in view of the new IMCO requirements for ballasting.

As to the problem of defining hydrodynamic pressure distributions over the hull surface, this has been included for some years in methods and computer programs applied by DnV. The method, which is based upon application of a source and sink technique, is developed also for oblique seas. Since a year, it has also been extended to enable computation for three-dimensional problems and finite water depth.

In their discussion of vibratory response, the authors express that internal damping can be measured by anchor drop tests. This is not easy to see, since the vibratory response under such tests must be heavily influenced by hydrodynamic damping.

Later in the paper it is stated that good correlation between calculated and measured springing response has been obtained. Some evidence to this effect would be highly appreciated, since a study of references /49/ and /50/ leaves rather the opposite conclusion.

Finally, I would like to add a comment on the bow-flare immersion problem. A method for prediction of such loads has earlier been suggested by Paul Kaplan under his work for the Ship Structure Committee. Following a similar approach we have at DnV recently developed computer routines for computation of bow-impact loads. Similarly to Kaplan's technique we have made a strip modelling of the bow, but allowed the strip orientation to deviate from the vertical. This provides better fulfilling of assumptions for two-dimensional flow, but the model applied still seems to overpredict the loads somewhat.

On the other hand, for this kind of loads there seems to be a great lack of satisfactory data for comparison.

Computed loads and responses for a fast container ship indicate, however, that severe stresses due to bow-flare immersion may occur. Also, it has been shown necessary to evaluate the effect of hull flexibility on computed loads.

R. B. Hulla, Associate Member

There has been an increasing complexity in the structural design process of merchant vessels which appears to have accelerated noticeably in the past few years. To a large measure this situation represents not only the increased research mentioned in the paper, nor the appearance of certain naval ship types, but also the impact of computer technology in our business of ship design.

It is now possible to analyze the structural aspects of a ship design to an ever increasing degree, as evidenced in this paper. However, it must be remembered (and it is too often forgotten) that in order to analyze a structure, one must first have a structure to analyze. The time allocated for the initial design of the basic hull structure seems to be one of the few aspects of preliminary and contract ship structural design which has not changed.

Thus, although the subject paper is a good state of the art review of the dynamic behavior of a ship, some additional guidance and perhaps comment with respect to whether certain loads are worth worrying about would have been helpful. In this regard for example, the question of flare immersion is stated as being important in producing whipping stresses. However, Lloyds current rules seem to indicate this is a problem only for excessive flare in association with vessels in certain limited length and speed ranges, and for these the only guidance is that the basic hull structure will be "specially considered". In order that the knowledge of dynamic loads be fully utilized early in the design of the basic hull structure, the designer must know not only what are potential problem areas, but also whether the problem is likely to be structually significant for the vessel he is designing (i.e., what factors are likely to cause the problem in the first place, and what magnitude of stresses might arise).

Along these lines, it would seem that a better approach, rather than the statistical and probabalistic one, for thermal effects and other secondary effects would be to recognize certain minor or second order effects exist and to account for these in the allowable design stresses. Otherwise a lot of time and effort is used in determining whether the thermal stresses are one or three thousand psi, when the degree of uncertainity in some of the major stresses is double, triple, or even quadruple such values.

The fruits of a large amount of research, including that discussed in this paper, have recently been used in certain modifications in classification society rules. The concept of a wave-induced bending moment for example is now included in the longitudinal strength section of both the American Bureau of Shipping Rules and those of Lloyd's Register of Shipping. However, the concept of a single effective wave height, as noted by the authors, is based on several simplifying assumptions. In view of the experience shown in Figure 5 of the paper for the "Universe Ireland", the wisdom of relying too heavily on such a concept may be questioned. It would be interesting for the authors to compare the 1975 ABS criteria with the "Universe Ireland" data, i.e., if the "Universe Ireland" were designed by 1975 ABS criteria, would it likely be overstressed in the ballast condition.

It is noted that torsional loads are briefly mentioned but not discussed. Considerations of transverse strength and shear stresses are also not explored. The authors' comments on the influence of dynamic loadings in these areas would be appreciated.

Naresh M. Maniar, Life Member

A few minor comments to what is considered a nice summary of the state of the art on the determination of hull girder loadings.

Whether a statistical approach to still-water loadings is appropriate can only be determined after sufficient service records for different types of ships have been examined. The writer's recent experience indicates that information generally found in log books is quite inadequate to establish the experienced load distribution; and extensive Ship Structure Committee effort with cooperation on part of owners will be required to gather the necessary data.

With regards to the authors' note on the bending moments created by the ship's own wave which may be included with the still-water bending moment, it should be stated that a separate calculation would be required for each speed in question.

The authors were careful to note that the "probability approach is of immediate practical usefulness on a comparative basis". It should be stressed that the usefulness will remain on a comparative basis till the naval architect has the necessary input on the values of risk to be associated with the statistical calculations. Of course, the meaning of the risk and its consequences will have to be properly appreciated.

S. G. Stiansen, Member

The authors present a very interesting paper, dealing with the most critical wave-load problem. Congratulations to the authors is in order.

I would like to offer the following comments to this valuable paper:

These comments relate to ABS Rules and are offered as a clarification of the author's paper.

1) The difference between Webb's long-term predictions and the effective wave heights specified in ABS Rules 1975, as shown in Figure 5, may be attributed to the different wave encounted headings used in predictions. Webb's trend was based on head sea conditions only, while ABS's trend was based on all headings.

2) Ship motion and statistical analysis were carried out by ABS for vessels with various block coefficients. Results obtained to date show generally higher effective wave heights for finer ships.

3) It is our current practice at ABS to use ship motion and statistical analysis in determing the maximum dynamic loadings for new types of vessels, such as LNG or LPG carriers.

Authors' Closure

William H. Buckley

First, our statement on p. 6 regarding the difference between the ABS curve in Figure 5 and the Webb curves is misleading. Although still-water loads were formerly included with wave bending moment in the old ABS Rules (before 1975), they are not included in any of the h_e curves shown in Figure 5.

It is true that there is still room for increased rationalization of ship hull strength standards, but the recently adopted ABS 1975 Rules represent a significant step forward. Still-water and wave-bending moments have been clearly separated, and the effective wave height concept adopted. However, because of some uncertainities previously mentioned, the requirements are still tied to past experience. As further research develops new ideas, they can be incorporated within this framework. Meanwhile, it should be noted that there is nothing absolute about the loads -- or h_e values -- plotted. N = 10^8 cycles -- one ship's lifetime -- has been adopted for comparative purposes. Ultimately, a design based on 10^{11} might be more appropriate, with a much smaller factor of safety. At the same time, other loads must be explicitly determined and combined -- as Mr. Buckley states. On the other hand, h_e based on 10^5 or 10^6 with a high factor of safety -- or low allowable stress -- is equivalent. Hence, we do not believe that present and future research are incompatible with the present framework.

Nevertheless, we do agree that rapid progress should continue on developing a completely rational standard of strength. One of the most important gaps at present is in Mr. Buckley's area -- the probabilistic determination of structural capability -- the load that can actually be carried.

E. Abrahamsen

In this survey paper, with emphasis on current practical applications, we have not attempted to cover important parallel developments in other countries. We did note on p. 5, however, that "similar work has been done by other classification societies." It is always necessary to keep in touch with the advanced work being done by Norske Veritas -- as well as by Lloyds, Bureau Veritas, and others -- and we appreciate having reminders of such recent work.

We are interested to note that Norske Veritas is also concerned about the high bending moments for tankers in ballast condition -- and with possible effects of new IMCO ballast regulations.

We agree that internal damping cannot be separated from hydrodynamic damping -- except by theoretical calculations of the latter.

We expect to have a new report on correlations of theoretical and experimental springing on Great Lakes ships available in the near future. Meanwhile, we shall be interested in seeing the new "rational" Norske Veritas Rules when they become available.

William A. Cleary, Jr.

Mr. Cleary is more optimistic than we are regarding the possibility of checking calculations or model tests by full-scale measurements. This is difficult enough for wave-bending moments, but is even more so for springing. We feel that overall statistical comparisons may be not only more flexible but also more satisfactory. As brought out in yesterday's discussion, full-scale research should be primarily verification.

We agree that any further reduction in scantlings would be made with great caution, but as Mr. Cleary says, the danger is probably in some other area -- such as fatigue -- rather than in ultimate failure caused by a single large load.

Regarding still-water loads, we had in mind the exact definition of "standard deviation". There may be two or more "bell-curves", each with a different standard deviation -- one for full load, one for ballast, another for medium load, etc.

As for wave-bending moment -- it definitely should not include the still-water moment. This should be calculated separately.

We do not believe that the change in the Universe Ireland extrapolation means that other large tankers would be similarly affected, because -- as shown in the paper -- the Universe Ireland was out of line with the others.

We note that he agrees with our discussion of the effect of draft on bending moment and has added some important comments.

We agree that safety factors should not be thrown away, but since they are partly "factors of ignorance", they can certainly be reduced as our knowledge grows.

Finally, we are not advocating design for a particular service. This is the reason we showed long-term predictions for the Universe Ireland in North Atlantic service.

R. Hulla

We agree that perhaps not enough time is regularly allocated to ship structural design. On the other hand, it should be possible to make general calcualtions for different ship types and then apply short-cut methods for routine use. Certainly more work is needed in many areas, such as the effect of flare immersion. In the latter case, however, the basic techniques are available -- as noted by Mr. Abrahamsen.

Mr. Hulla questions whether certain stresses are significant enough to treat statistically in overall ship structural design. Taken alone, many of the loads imposed on the ship structure are of low magnitude. Since they vary with time, however, they can increase the maximum stresses in a statistically predictable way. That is, in fact, the strength of the statistical approach as opposed to adjustment of an allowable design stress.

The effective wave-height concept is simply a convenient way to express bending moment and neither this concept nor any assumptions made in defining it are the cause of the high bending moment values predicted for the Universe Ireland in ballast. The same high values would be reached if midship bending stress, for example, were plotted directly.

N. Maniar

We agree with Mr. Maniar's comments regarding the difficulty in obtaining data on still-water loadings and hope that his current project for the Ship Structure Committee will be successful.

His is also correct in noting that the bending moment created by a ship's own wave will vary with speed. It can be either calculated or determined easily by model tests for representative types of ships.

Finally, we also hope that the probability approach will be applied in other than a "comparative" basis in the near future. In principle, the acceptable risk can be determined on the basis of past classification society experience.

S. Stiansen

We appreciate the favorable comments on our paper and clarification of a number of points. It is true that Webb curves of effective wave height in Figure 5 were based on head seas only, which accounts for some of the difference relative to the ABS curve. All recent work at Webb has been based on consideration of all headings, however.

The statement that recent ABS calculations show greater effective wave heights for finer ships is important. It suggests that a different h_e curve for such ships may be called for.

H. Townsend

We agree with the idea that there would be distinct advantages in removing the corrosion allowance from classification formulas for scantlings and making it an explicit add-on quantity. This would eliminate a great deal of uncertainty from the Rules.

J. Boylston

We are glad to have attention called to the economic penalties of "nuisance" cracking, involving both repair costs and indirect costs through removal of a ship from service. Perhaps Mr. Boylston missed the last paragraph of the paper (p. M-18), referring to fatigue damage costs.

E. Haciski

The question of practical criteria for design purposes is an important one. However, we do not feel in a position to make specific proposals at this time. The classification societies have formualted such criteria, of course, and are continually working on improvements in the light of new information as it becomes available.