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LOAD CRITERIA FOR SHIP STRUCTURAL DESIGN

WEBB INSTITUTE OF NAVAL ARCHITECTURE

PREPARED FOR
NAVAL SHIP ENGINEERING CENTER

MAY 1973

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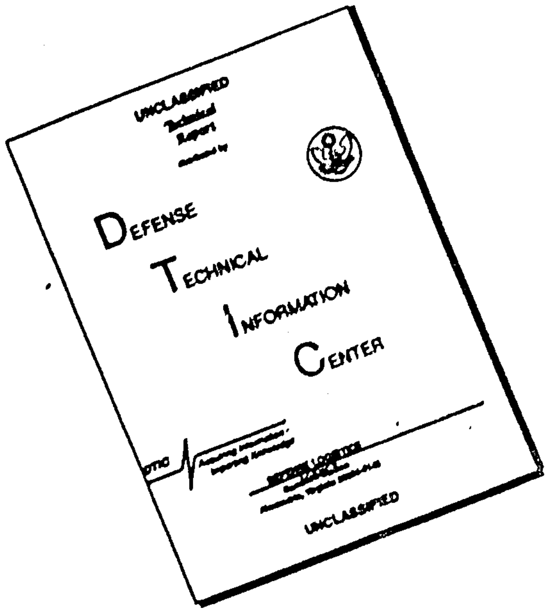

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SR 198
18 JUL 1973

The development of a rational procedure for determining the loads which a ship's hull must withstand is a primary goal of the Ship Structure Committee program. In the last several years, considerable research activity has been devoted to theoretical studies on the prediction of hull loads and to measurement of response both on models and on ships at sea.

This report describes a first effort into the synthesis of the results of these diverse projects into a rational design procedure.

Comments on this report would be welcomed.

W. F. Rea III

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Rear Admiral, U. S. Coast Guard
Chairman, Ship Structure Committee

A

Unclassified
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Webb Institute of Naval Architecture	2a. REPORT SECURITY CLASSIFICATION Unclassified
	2b. GROUP

3. REPORT TITLE

LOAD CRITERIA FOR SHIP STRUCTURAL DESIGN

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Final Report

5. AUTHOR(S) (First name, middle initial, last name)

Edward V. Lewis, Dan Hoffman, Walter M. Maclean, Richard van Hooff and Robert B. Zubaly

6. REPORT DATE	7a. TOTAL NO OF PAGES	7b. NO OF REFS
May 1973	107	130

8a. CONTRACT OR GRANT NO N00024-71-C-5372	9a. ORIGINATOR'S REPORT NUMBER(S) SSC-240
b. PROJECT NO	
c.	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)
d.	

10. DISTRIBUTION STATEMENT

The distribution of this document is unlimited.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	Naval Ship Systems Command

13. ABSTRACT

Consideration is given to the critical loads on ships' hulls, as indicated by possible modes of structural damage and/or failure. An ultimate load criterion is then set up involving the following bending moments:

Quasi-static wave-induced, vertical and lateral combined.
Still water, including effect of ship's own wave.
Dynamic loads, including slamming, whipping, and springing.
Thermal effects.

The determination of each of these loads is discussed in detail, and the need for further clarification of dynamic loads is brought out. Methods of combining these loads, all expressed in probability terms, are considered.

A criterion for cyclic loading is discussed, involving the prediction of the expected number of combined loads of different levels, as well as the expected shifts of mean value. A criterion for brittle fracture is also discussed.

Attention is given to estimating an acceptable probability of failure for use in design. Finally, calculations of loads are carried out for a typical cargo ship, the S.S. Wolverine State. The loads are then combined in accordance with the proposed ultimate load criterion and compared with the standards under which the ship was designed.

DD FORM 1 NOV 65 1473

Unclassified
Security Classification

14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Bending moments						
Wave loads						
Still water loads						
Thermal effects						
Cyclic loads						
Dynamic loads						
Combining loads						
Design of hull girder						

SSC-240

Final Technical Report

on

Project SR-198, "Load Criteria"

to the

Ship Structure Committee

LOAD CRITERIA FOR SHIP STRUCTURAL DESIGN

by

Edward V. Lewis
Dan Hoffman
Walter M. Maclean
Richard van Hooff, and
Robert B. Zubaly

Webb Institute of Naval Architecture

under

Department of the Navy
Naval Ship Engineering Center
Contract No. N00024-71-C-5372

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U. S. Coast Guard Headquarters
Washington, D.C.
1973

ABSTRACT

Consideration is given to the critical loads on ships' hulls, as indicated by possible modes of structural damage and/or failure. It is recognized that of particular importance is the possibility of damage in the form of compression buckling or plastic flow in tension of one or both flanges which could lead to ultimate failure. Another mode of failure is by fatigue, which is important because cracks may occur which must be repaired before they propagate to a dangerous extent. A third mode of failure is brittle fracture, which is particularly difficult to deal with but can be minimized by control of material quality and use of the customary "fail-safe" approach by using crack arresters. Finally, the possibility of shear and/or torsional buckling requires consideration.

Hence, an ultimate load criterion is set up involving the following bending moments:

Quasi-static wave-induced, vertical and lateral combined.

Still water, including effect of ship's own wave.

Dynamic loads, including slamming, whipping, and springing

Thermal effects.

The determination of each of these loads is discussed in detail, and the need for further clarification of dynamic loads is brought out. Methods of combining these loads, all expressed in probability terms, are considered.

A criterion for cyclic loading is discussed, involving the prediction of the expected number of combined loads of different levels, as well as the expected shifts of mean value.

A criterion for brittle fracture is also discussed.

Attention is given to estimating an acceptable probability of failure for use in design. Finally, calculations of loads are carried out for a typical cargo ship, the *S. S. WOLVERINE STATE*. The loads are then combined in accordance with the proposed ultimate load criterion and compared with the standards under which the ship was designed.

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I. INTRODUCTION

RATIONAL DESIGN

For many years the goal of truly rational design of ship structures has been discussed, and a great deal of research bearing on this objective has been carried out. The concept was described, for example, in an early planning document of the Ship Structure Committee (1)*, and since the establishment of the International Ship Structures Congress (I.S.S.C.) in 1961 it has been regularly discussed on a worldwide basis by Committee No. 10, Design Philosophy. Although this report is intended only to indicate progress to date, it is hoped that it will assist in the advance toward the ultimate achievement of rational design of the main hull girder.

The concept of rational design involves the complete determination of all loads on the basis of scientific rather than empirical procedures, in order that uncertainties may be reduced to a minimum. This approach carries with it the idea that the response of the structure can also be accurately determined and that arbitrary large factors of safety, or "factors of ignorance," can be avoided. The concept is consistent with the modern approach to structural design that considers the "demand" upon and "capability" of the structure. In short, instead of insuring that a simple calculated design stress is below the ultimate strength of the material by an arbitrary factor of safety, an attempt is made to determine the demand of all loads acting on the structure and then the capability in terms of load-carrying ability -- the load the structure can withstand without failure. Of course, this approach requires a definition of failure, which may be a serious buckle, a major crack, complete collapse, or a tensile failure (Chapter II). The concept of rational design of a ship hull is believed to be consistent with a probabilistic approach, which has already been found to be essential for dealing with random seaway loadings. Both demand and capability can be expressed in terms of probabilities, and a satisfactory design is then one in which the probability of failure is reduced to an acceptably low value. The problem of determining local loads or stresses for detailed structural design is much more complex and is not discussed here.

This particular report deals only with the demand -- or loading -- on the hull girder, but an attempt has been made to formulate it in a manner that is consistent with the above approach. In due course, with the cooperation of the ship structural designer, it is anticipated that a rational design procedure will evolve (2).

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*Numbers in parentheses refer to References listed at the end of this report.

It is not intended to minimize the importance of the conventional empirical approach to ship structural design which has served the designer, builder and operator well through the years. But there is currently a substantial need for a fully rational approach because of such new maritime developments as larger ships, faster ships, unusual hull configurations (such as the catamaran), and new materials. Complete and comprehensive load criteria can facilitate the extrapolation of ship design into new configurations, using new concepts and materials.

LONG-RANGE PATTERN OF LOAD VARIATION

It may be useful at this point to describe the typical long-range pattern of load variations on typical merchant ships as background for the detailed discussion of the various types of loads in subsequent chapters.

For completeness, we should perhaps begin with the construction of the ship on the building berth. Strictly speaking the only loads present are those induced by the weight of the structure itself. However, there are residual stresses in the plating and locked-in stresses due to welding, often of considerable magnitude and sometimes sufficient to lift the bow and/or stern off the keel blocks. The locked-in stresses are of particular concern where they may exist in combination with other stresses at a weld defect or notch and under certain conditions could help to produce a brittle fracture. For other types of failure it seems reasonable to consider them to be of minor significance to longitudinal strength, since they tend to be eliminated by "shakedown" or adjustment in service. That is, an occasional high longitudinal wave bending load -- in combination with other loads -- may be expected to cause local yielding in any of the high residual stress region. Upon determination of this high wave load the structure will tend to return to a condition of reduced residual stress.

During launching a high longitudinal bending moment may occur, but this is usually calculated and allowed for by the shipyard. During outfitting a continual change of still water shear and bending moment can be expected as various items of machinery and outfit are added. The longitudinal still water bending moment on the ship can always be calculated, but the midship stress will probably not correspond exactly to this calculated value because of possible built-in hog or sag residual stresses, and departures of the hull behavior from simple homogeneous beam theory. In short, the ship is never in a simple no-load condition nor even in a condition where the absolute value of even the longitudinal bending moment is exactly known. Such a built-in bending moment will not be considered in this report since it is believed that changes in load while the ship is in service are of primary significance.

In general the still water hull loadings vary quite slowly. When a ship is in port there are gradual changes in the bending moments, shears, and perhaps the torsional moments as cargo is discharged and loaded, fuel oil and stores are taken aboard, etc. During the voyage there are even more gradual changes in mean loadings as fuel is consumed, and ballast is added or shifted. Typical changes of this kind are shown in Figs. 1 and 2(3). Finally, at the end of voyage changes resulting from cargo discharging and loading, plus possible fuel oil and ballast changes, will again modify the bending moments, shearing forces and torsional moments. The loading changes in port may be considerable and depend on the nature and quantities of cargo carried on various legs of the voyage. These changes do not show up in Figs. 1 and 2 because the recording equipment zero was customarily readjusted at every

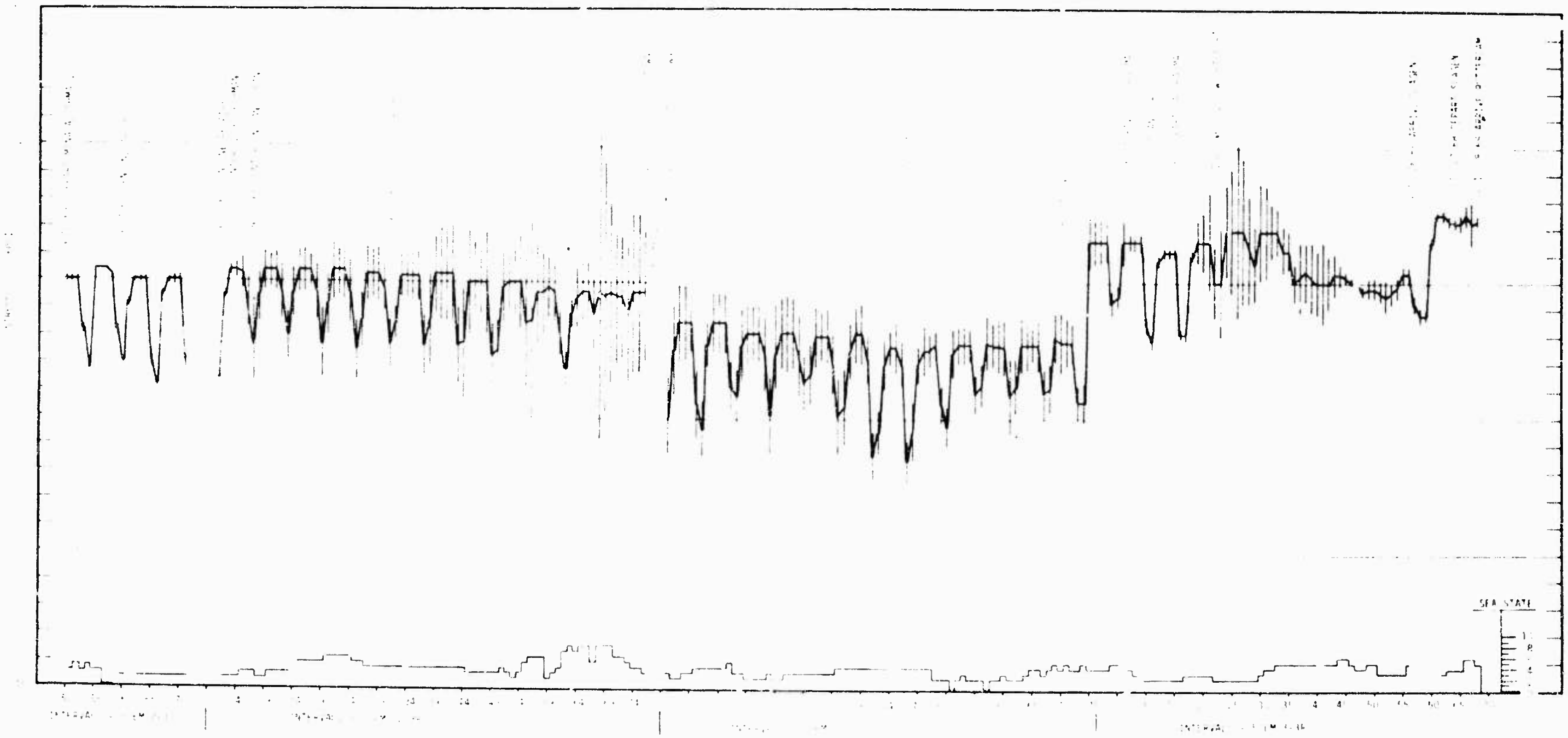

FIGURE 1 - Typical Voyage Variation of Midship Vertical Bending Stress, *S. S. ESSO MALAYSIA*, Loaded Condition (3)

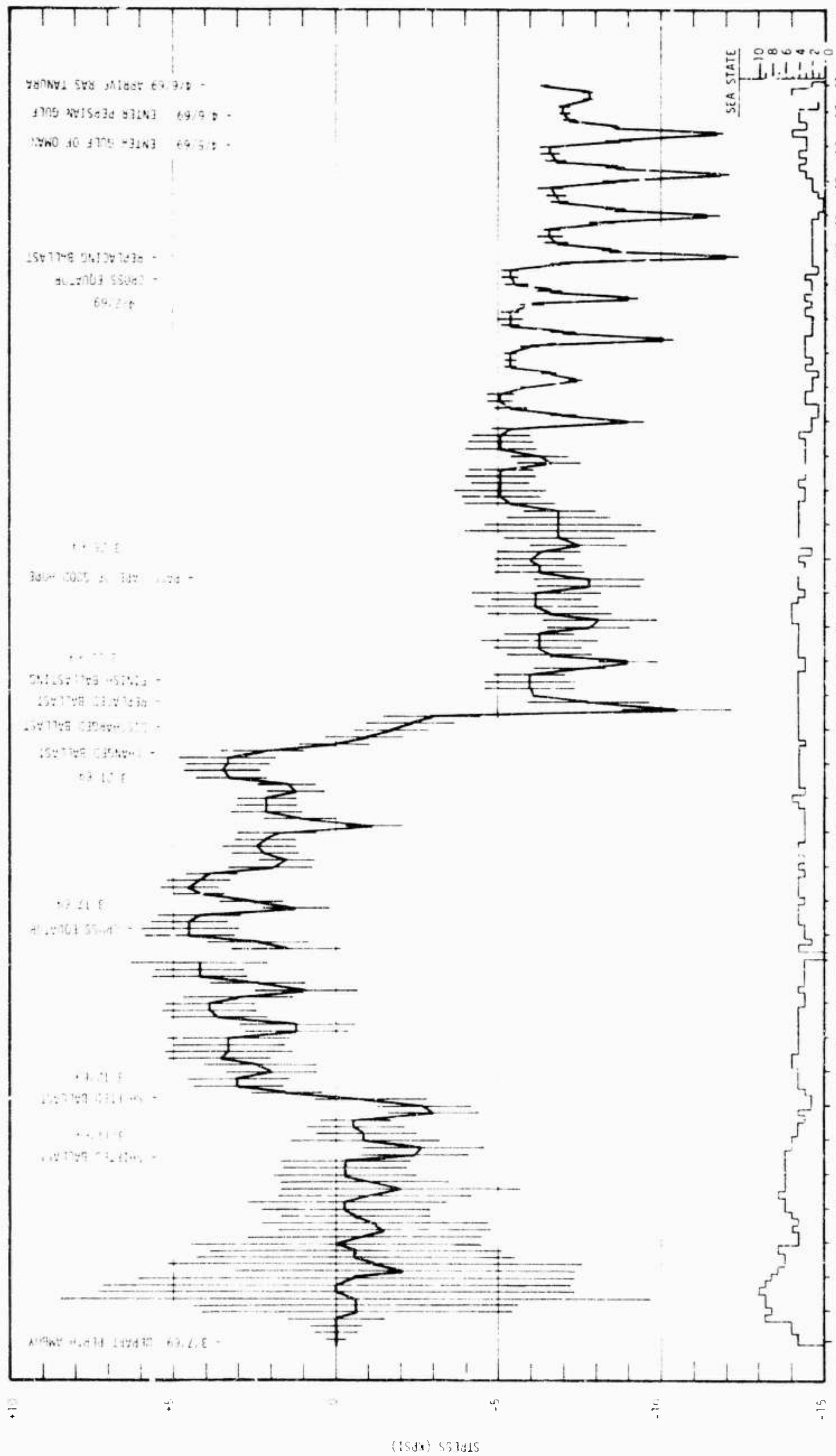

FIGURE 2 - Typical Voyage Variation of Midship Vertical Bending Stress, *S. S. R. G. FOLLIS* (3)

port visit. As explained in an SSC report, "This capability is necessary to prevent the dynamic stress range from exceeding the limits of the instrumentation system" (4).

When the ship gets under way to go to sea, the first new hull loading to be experienced -- especially if the ship is a high-speed vessel -- is the sagging bending moment induced by the ship's own wave train. This longitudinal bending moment is a function of ship speed, and will be superimposed with little change onto other bending moments (5).

Another load variation results from diurnal changes in air temperature* and in radiant heating from the sun. The effect is clearly shown in Figs. 1 and 2. Such thermal stresses can be explained on the basis of irregular or uneven thermal gradients, which can perhaps be considered as the "loads." In general, if a beam is subject to heating that produces a uniform thermal gradient from top to bottom it will deflect and there will be no resulting stresses. But, if the gradient is not uniform, stresses will be induced. In the case of a floating ship, the temperature of all the steel in contact with the water will be at the nearly uniform water temperature, and there will be very little change from day to night. But the portion of the hull above water will usually be at a different temperature that changes continually and depends on both the air temperature and the amount of sun radiation (extent of cloudiness, duration of sunlight, altitude of sun at noon). In respect to the latter factor, the color of the deck is important also. There is usually a marked change in stress in the vicinity of the waterline, especially on the sunny side of the ship, but from the point of view of longitudinal strength the temperature change of the weather deck -- in relation to the underwater hull temperature -- is significant.

Another large load at sea is that induced by the encountered waves (Fig. 3). This load usually varies in an irregular fashion with an average period of 5-10 seconds, depending on the ship. Not only is there irregularity in wave-induced loads from one cycle to the next, but there is a pronounced variation in average level with ship heading and with weather changes during a voyage and from one season to another. The irregularity of these loads is, of course, due to the irregularity of the waves at sea. However, the baffling irregularity of ocean waves has yielded to modern analytical techniques. This was explained by Dr. Norbert Wiener, who developed the necessary statistical techniques for another purpose. "How could one bring to a mathematical regularity the study of the mass of ever shifting ripples and waves ....?," he wrote (6). "At one time the waves ran high, flecked with patches of foam, while at another, they were barely noticeable ripples ..... What descriptive language could I use that would portray these clearly visible facts without involving me in the inextricable complexity of a complete description of the water surface. This problem of the waves was clearly one for averaging and statistics .." In time Wiener evolved his mathematical tool, spectrum analysis -- a means of breaking down complex patterns into a large number of measurable components.

In recent years wave-induced bending moments have been extensively studied, so that a good statistical picture is beginning to emerge. Research over a number of years (4) (7) has provided a bank of statistical stress data on four cargo ships in several services. Using some of these data it has been found (8) that two different mathematical models can be used to extrapolate such results to much longer

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*Gages were temperature compensated.

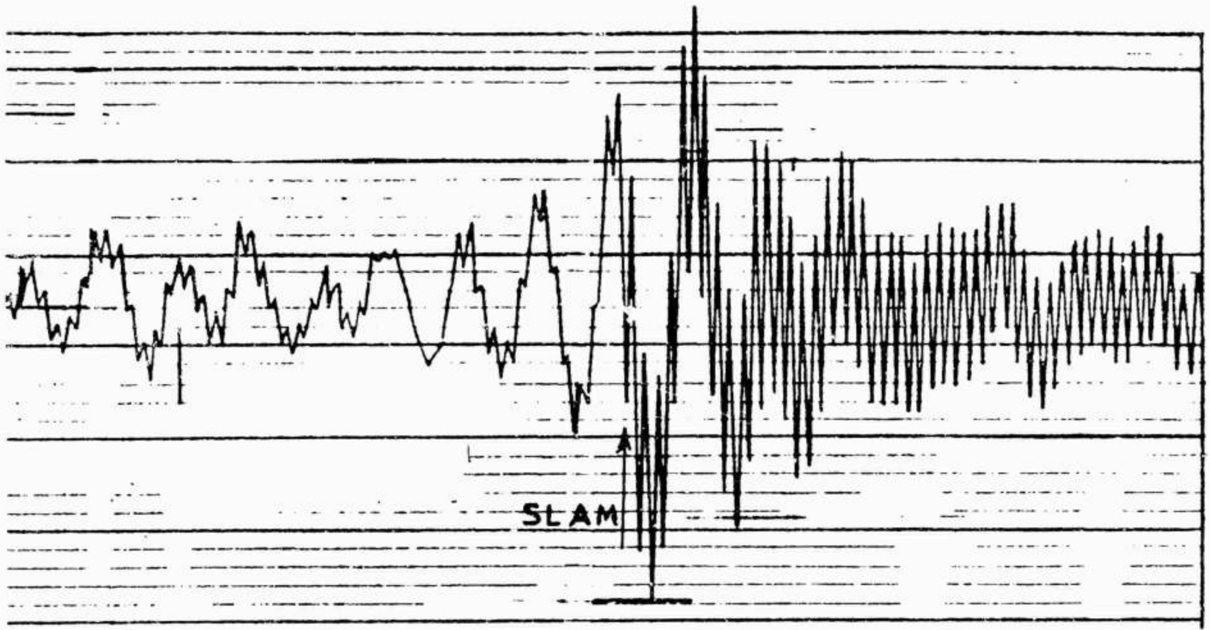

Figure 3 - Typical Record of Midship Vertical Bending Stress with Slamming, M.V. *FOTINI L.*

term probabilities. Furthermore, it has been shown that using the same mathematical models -- combined with model tests in regular waves and ocean wave spectra -- short-term (9) and long-term trends (10) can be predicted with a precision that depends only on the reliability of the data. At the same time, computer programs have been developed for applying ship motion theory to the calculation of loads in regular waves as a substitute for model tests.

Finally, oceangoing ships experience dynamic loads, the most troublesome of which result from impact (slamming) and the vibratory response (whipping) that follows it (Fig. 3). In general these loads are transient and therefore are difficult to deal with statistically. They are superimposed on the previously mentioned loads. Both full-scale measurements (11) and theoretical studies (12) (13) have been carried out on slamming and whipping, and these have clarified but not solved the problem. Shipping of water on deck and flare immersion are other sources of transient dynamic loading.

Recent attention has been focused on another dynamic phenomenon, springing, which under certain conditions seems to be excited more-or-less continuously in flexible-hulled ships, without the need for wave impact. Considerable progress has been made toward solution by means of theoretical and experimental studies (14).

All of the above loads will be discussed in detail in subsequent chapters.

HULL LOAD CRITERIA

In general treatises on structural design (15) two types of loading are usually distinguished: controllable and uncontrollable. In the first case one can specify design loads with instructions to insure that these are never exceeded. An example is a highway bridge designed on the basis of a posted load limit. In the second case, usually involving natural forces, one must make a statistical analysis and endeavor to design on the basis of the expected loads, with no limitation on the structure or its operation.

In the design of ships, still water loads are generally controllable and wave loads are not. If calculations of typical conditions of loading indicate that excessive still water bending moments might occur, specific operating instructions may be issued to make sure that certain limits are not exceeded. The possibility has been discussed of specifying limiting wave bending loads, as well -- somewhat in the same manner that wing loads on an aircraft are limited by requiring certain performance restrictions. Such a limit on wave loads for ships could only be applied if special instrumentation were available to advise the officer on watch when and if the limiting bending moment is reached, since there is no way for him to judge this loading unaided. Furthermore, he must have guidance information at hand that will enable him to take steps to reduce the bending moment if it should approach the safe limit.

Dynamic loads are partially controllable, since the vibratory response of the hull girder can be felt by the Master on the bridge. By a change in ship speed and/or course he can reduce the magnitude of the exciting forces and thus indirectly reduce the loads to levels that he has found by experience to be acceptable.

In this report a compromise approach has been adopted regarding statistical dynamic wave loads. An effort is made to determine all the loads acting on the ship's hull to provide load criteria from which a satisfactory but economical structural design can be developed. However, to guard against the possibility of some unforeseen extreme load condition, it is recommended that suitable stress instrumentation be provided as a warning device for added safety (16).

A great deal of research has been done in recent years on the ship hull loadings mentioned in the previous section, much of it in the Ship Structure Committee (SSC) program. Research under other sponsorship has also contributed to an understanding of hull loads, including particularly that supported directly by the U.S. Navy, the Society of Naval Architects and Marine Engineers and the American Bureau of Shipping in this country, and by various organizations in Great Britain, Norway, the Netherlands, and Japan, as reported to the International Ship Structures Congress (I.S.S.C.). A partial bibliography is given at the end of this report (Appendix A).

Some typical loads have received more attention than others, however, leaving gaps in the overall picture. It is the purpose of this report to present a comprehensive and reasonably complete picture of the hull loads and hence load criteria for ship design, with particular emphasis on dry cargo ships. Hence, consideration will be given in the next chapter to identifying the critical loads of interest to the ship structural designer. In succeeding chapters each of the various loads will be discussed in turn, and consideration of typical magnitudes and of procedures for detailed calculations will be included. Finally the problems of combining these loads for hull girder design purposes will be taken up. Where important gaps in our knowledge appear, they will be identified and recommendations made for further research. A numerical example for the S.S. Wolverine State will be presented.

A number of attempts have been made to consider how the available material on loads can be combined and applied to the rational design of ships. Of these, particular mention might be made of the work of Caldwell (17), Aertssen (18), Abrahamsen, Nordenstrøm, and Røren (19), and of Committee 10 of the I.S.S.C. (20).

II. CRITICAL LOADS

INTRODUCTION

Before discussing hull loads in detail it is necessary to consider the different ways that the structure can suffer damage or fail. The object is to investigate structural aspects of the problem only to the extent necessary to be sure that all of the necessary information on loading -- or demand -- will be made available to the structural designer. In short, we must ask, what are the critical loads and how do they combine? Meanwhile, it is hoped that work will continue toward developing a completely rational approach to ship structural analysis and determination of the capability of the structure.

Discussion of critical loads can be facilitated by defining structural failure. Caldwell (17) considers ultimate failure as the complete collapse by buckling of the compression flange and simultaneous tensile failure of the tension flange. 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-watertightedness. As pointed out by C. S. Smith in discussion of (17), "In designing a midship section, the designer should consider the various levels of damage which a hull girder may experience between the limits of initial yield and final collapse, and should attempt to relate each level of damage to an applied bending moment."

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. Failure is then a severe damage that endangers the safety of the ship.*

Further study of the subject of critical loads during this project has resulted in no basic improvement in Gerard's analysis of specific ways in which the hull girder could fail, as given in "A Long-Range Research Program in Ship Structural Design" (1). He considered overall damage by compressive buckling, overall tensile yielding, low-cycle fatigue cracking and brittle fracture. To these should be added combined normal and shear stress buckling, and it is possible to elaborate somewhat on his scheme and in certain respects to obtain more definite statements.

The types of damage that should be considered then in connection with critical loads might consist of any of the following:

Damage

- Excessive hull deflection associated with buckling and/or permanent set.
- Fatigue cracking.
- Brittle fracture, minor or extensive.
- Shear or torsional buckling.

Failure

- Collapse and/or fracture of the hull girder.

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*This is sometimes referred to as "collapse" (20), but we feel that this term connotes buckling failure to the exclusion of tensile failure or permanent set and therefore prefer "failure."

Although only the last is considered to be structural failure, all of these types of damage are important for a longitudinal strength criterion. Clarifying the nature of these potential damages will assist us in providing the necessary information on loads.

The magnitude of elastic hull deflection is usually considered in the design criteria of classification societies, and it will also be discussed in this chapter.

Finally, consideration should be given to other minor effects, such as the forces generated by rudders and anti-rolling fins. And other types of service loading, such as berthing, drydocking, and grounding, which may have direct effects on primary hull girder structure, cannot be overlooked. Local damage to structure that is not part of the main hull girder is excluded from consideration.

An important consideration in structural design is corrosion. However, since this is not a load it will not be considered in this report.

PERMANENT SET AND ULTIMATE FAILURE

We may first consider overall static damage to one of the "flanges" (deck or bottom) in either compression or tension, i.e., buckling or elasto-plastic yielding. The effect of lateral as well as vertical longitudinal bending and torsion must be included here. Consideration must be given to the combined effect of still water bending, wave bending and thermal loads. In addition, a basic question is whether or not the superimposed dynamic effects of high frequency "whipping" following a slam and/or flare entry should be considered, as well as the effect of wave impacts on the side of the ship and continuously excited springing. It is quite possible that the short duration of dynamic bending moments -- and stresses -- limits the amount of permanent set or buckling that they can produce. As noted by Spinelli, "It should be borne in mind that the short time in which the wave moments due to slamming develop their maximum values, and the entity of the total deflection that would be consequent on them, make the probability of its realization extremely scarce" (21).

And in referring to plastic deformation, Nibbering states, "In practice these deflections will not develop the very first time an extreme load of the required magnitude occurs. The time during which the load is maximum is too short, especially when a part of the load is due to slamming" (22). This is a problem in structural mechanics not within the scope of this project, and therefore we shall attempt merely to identify and evaluate dynamic as well as static loads.

Finally, local loads (not due to longitudinal bending) on which all of the above are superimposed must not be overlooked. These include deck loads, cargo loads on innerbottom, liquid pressures within tanks, and external water pressures.

Although there seems to be general agreement on the importance of ultimate strength, involving extensive plastic yielding and/or buckling, there seems to be some doubt as to how to deal with it in design. From the point of view of the present study, however, definite conclusions can be drawn regarding the load information needed for designing against potential damage of this type.

FATIGUE

Second is the possibility of fatigue cracking, which seldom constitutes failure but is important for two reasons: fatigue cracks 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, "It is a favorable circumstance that fatigue cracks propagate very slowly in ships' structures" (22).

The possibility of fatigue cracking is increased by the presence of stress concentrations -- as for example, at hatch corners (23)(24), and it involves consideration of the magnitude of still water bending -- i.e., the shift of mean value -- as well as the range of variation of wave bending moments. For example, a ship may operate with a large still water sagging moment (loaded) on its outward voyage and with a large still water hogging moment (ballast) on its return, and such a large variation in mean value needs to be considered in relation to fatigue. As before, consideration of lateral as well as vertical bending must also be given. Dynamic loads and vibratory stresses may be expected to contribute to the fatigue loading.

It appears that the fatigue loading histories of actual ships show considerable variety. Hence, the objective for this study is felt to be simply to obtain clear statistical or probabilistic pictures of each of the types of loading involved:

1) Probability density of mean still water bending moments, which tentatively and approximately appears to be two normal curves, one representing outbound and the other inbound conditions.

2) Long-term cumulative distribution of wave-induced bending moment, which together with 1) can be interpreted as a low-frequency loading "spectrum."

3) Probability density of high-frequency bending moments associated with dynamic loads (slamming, whipping and springing). The combination of these effects with low-frequency loads is a difficult problem, as discussed in later chapters.

4) Thermal stress conditions, which cause a diurnal change in stress level.

At first glance it appears to be a hopeless task to collect all the necessary statistical data on the various loads for ships of different types and to develop ways of combining them that are not only sound by probability theory standards, but are meaningful from the viewpoint of the mechanics of fatigue and of the properties of the materials used. A short-cut answer, as proposed by Gerard (1) would be simply to design to avoid overall combined loads as listed above that exceed the yield point of the material anywhere in the structure, including areas of stress concentration. Design of the structure on this basis would virtually insure the ship against low-cycle fatigue, but would possibly lead to heavier structure than in present designs. Since fatigue cracks can be detected and repaired, it is not felt that it is necessary to limit stresses to yield point level in this way, Attempts should be made to understand and evaluate all components of cyclic loading.

BRITTLE FRACTURE

Third, is the possibility of failure by brittle fracture. This mode of failure was common in early days of welded ship construction, but has been greatly reduced in recently built ships. It cannot be overlooked in a comprehensive scheme, however. All of the above-mentioned loads apply, including residual and thermal stresses and the notch effect of weld defects. It has been pointed out that a low-cycle fatigue crack can be the initiation point for brittle fracture (24).

It is generally recognized that the following factors are involved in brittle fracture:

(1) Ambient temperature.

(2) Steel characteristics (transition temperature).

(3) Notches or stress raisers, including weld defects.

(4) Stress (or load) level.

(5) Strain rate.

Secondary factors include strain as well as stress fields, corrosion effects, metallurgical effects of welding, structural details that introduce constraint, and residual stresses.

Because of improvements in design and materials, brittle fracture now seldom occurs in actual service. However, it is conceivable that if, as a result of more rational approaches to design, working stress levels are increased we may again have trouble with brittle fracture. Furthermore, it is important to recognize that brittle fracture has been brought under control by careful attention to material qualities, selection and control of fabrication techniques, and inspection at all stages of construction. Diligence cannot be relaxed, especially as new materials, new fabrication techniques, and more rational design procedures are introduced.

Nibbering maintains "that 90% of all ships in the world move regularly and undamaged in conditions where the temperature is lower than the crack-arrest temperature of their steels .... The nominal stresses mostly are so low that with present day quality of design and workmanship brittle fractures cannot initiate" (22).

For design purposes the load information needed is generally the same as for ultimate bending, as discussed in a preceding section, including all dynamic loads, except that only tensile loads need be considered. Rate of application of dynamic loads and ambient temperature conditions should also be specified.

Of the various dynamic loads, it is believed that consideration should be given particularly to the midship stress following a bottom impact slam. Since higher modes than the hull fundamental are involved, the strain rate may be quite high. See Chapter VI for further discussion.

SHEAR AND TORSION

Fourth is the possibility of shear failure in the hull girder "web." Although this is a problem in the design of light naval vessels, it has not been of much concern in more heavily built merchant vessels. This is not to say that shear loading on the side shell or longitudinal bulkheads is unimportant, but rather that other types of side shell loadings probably constitute more severe criteria of satisfactory design. Though there is a possibility that the side shell of merchant ships is excessively heavy, safe reductions in these scantlings can only be made by developing more precise ways of determining the hull girder torsion and shear loadings, as well as lateral loadings due to such aspects of operation as bumping into dock structures, being handled by powerful tugs, etc.

Another aspect of concern here arises from the recent development of large bulk carriers which are frequently loaded only in alternative holds with high density ores. The result of such loading is that large shear and moment variations are experienced along the vessel's length which must be allowed for in the design of hull girder structure. Further definition of this problem area is needed, since it can be expected that large shear and moment, coupled with reduced structural effectiveness of the hull girder material, can lead to combined loadings of critical magnitude.

Torsion is important in relation to both shear and deflection, especially in wide-hatch ships (25). Excessive hatch distortion has become the major area of concern as progressively larger hatch sizes have been employed. Hence, methods need to be established for determining the magnitude of torsional loading as a basis for rational design. In so doing, the influence of transverse shear on torsional deformation, resulting from the unsymmetric nature of the ship's structure, must be included. That is to say, the transverse shear loading must be defined not only as to magnitude but as to effective point of application as well, and it must also be directly related to the torsional loading, since both are developed simultaneously in any particular oblique wave condition.

To provide sufficient information on loads to carry out a satisfactory analysis of torsional stresses it is necessary to know more than simply the torsional moments, since this implies a knowledge of the torsional axis. Hence, for example, in model tests carried out in regular waves at the Davidson Laboratory for the SL-7 research program, the following measurements were made at the critical sections:

Vertical bending

$$\left.\begin{array}{l}\text{Horizontal shear}\\\text{Vertical shear}\\\text{Torque}\end{array}\right\}\quad\begin{array}{l}\text{About arbitrary but}\\\text{known axes}\end{array}$$

Since both amplitudes and phase angles were recorded, this provided the complete information required for a general stress analysis -- provided, of course, that the number of sections for measurement was adequate. Such analysis would, in the case of cellular container ships, probably include the intersections of closed cell systems as well as hatch corners.

It is concluded that shear and torsion need to be considered both as separate load criteria and in combination with other criteria previously discussed.

DEFLECTION LIMITS

Overall hull girder design may be affected by elastic longitudinal deflection. Some of the pertinent factors are:

1. Possible damage to shafting piping systems, etc.
2. Effects of deflection on drafts entering and leaving port.
3. Effects of hull flexibility on natural vibration frequency and hence on springing and whipping stresses.

The question is whether some design criteria should be introduced to limit deflection in service, aside from the possibility of damage or failure of the structure.

Direct effects of abnormal deflections on shafting, piping, etc., could no doubt be provided for in design. Effects on drafts forward, aft and amidships -- hence on load line requirements, bottom clearances, etc. -- could be dealt with by special attention to loading condtions, perhaps with the help of additional arrangements for ballast. But the effect of hull flexibility on dynamic structural response requires further consideration. It has been established by the work of Kline (26) and others that the increase in natural period associated with greater hull flexibility is favorable from the viewpoint of slamming and the vibratory stress, or whipping, that follows a slam. However, such may not be the case for the more continuous vibratory response referred to as springing. Evidence to date suggests that the latter phenomenon is increased by increasing hull flexibility.

In the past deflection has been limited by restrictions on length/depth ratio. For example, the rules of the American Bureau of Shipping require that special consideration be given to any design for ocean service in which L/D is greater than 13. Whether or not such a severe limitation is necessary has never been clearly established, but there can be no doubt it has prevented difficulties from deflection in mild steel ships.

The question of deflection generally arises, therefore, only with consideration of unusual ship proportions or when a material other than ordinary mild steel is to be utilized. For example, a recent Ship Structure Committee report (27) develops tentative criteria for aluminum alloy construction of a bulk carrier, and in addition to specifying section modulus requirements determined by strength considerations it discusses the necessity for a midship moment of inertia value that will limit deflection. It is stated that, "The only guidance in this area at present is the ABS requirement that the hull girder deflection of an aluminum ship shall not be more than 50 per cent greater than that of a 'Rules' steel vessel, while Lloyd's and Bureau Veritas suggest no increase." The report itself does not agree, however. "It is concluded that no limits should be placed on the hull girder deflection of an aluminum bulk carrier, but that the effects of the deflection resulting from normal structural design should be considered in the areas noted above." A study of the report indicates that no consideration was given to the possibility that springing stresses would be aggravated by the increased flexibility (and hence longer natural period of vibration). It is felt that any elimination of deflection limits should be qualified by a provision that a study be made of the possibilities of serious springing.

A similar situation arises when extensive use of high strength steels is made. If full advantage is taken of their higher strength, then greater flexibility and hence the possibility of springing must be considered. A study of design procedures for high strength steels has been made (28), which accepts classification society limits on deflection.

In the present report, in which dry cargo ships are under consideration, deflection is not often a problem. Such ships are volume limited, and hence L/D ratios are quite low. This is especially true of container ships in which there appears to be a trend toward increasing depth in order to reduce the number of containers stowed on open decks. If high strength steels or aluminum is extensively used, a check should perhaps be made of stiffness and vibration frequency. No further detailed consideration of the problem is felt to be necessary for the present purpose of establishing hull load criteria.

A special case of objectionable deflection previously mentioned is the excessive distortion of hatches, resulting from torsional hull moments, which may cause loss of watertightness.

SUMMARY

The consideration of critical loads and hull deflections leads to the conclusion that information on the following loads is needed for rational longitudinal strength design:

Statical still water bending loads, mean values and variation.

Thermal effects on hull girder.

Wave bending loads, both extreme values affecting hull girder damage (or failure) and the cyclic loading picture affecting fatigue, including shear and torsion.

Dynamic loads, both extreme and cyclic, with phase relationships, durations and rates of application.

Each of the above will be discussed in turn in the succeeding chapters.

III. STILL WATER LOADS

INTRODUCTION

There were two aspects of the subject of still water bending moments studied in this research project on load criteria. One was a review of available procedures for calculating such loads and the other was a semi-statistical study of still water loadings on typical ships. Both of the above studies will be reviewed in this chapter.

CALCULATION OF STILL WATER BENDING MOMENTS

Broadly speaking, the techniques available for determining longitudinal static still water bending moments in ship hulls can be classed in two categories:

1. Approximate methods, used primarily in the early design stage for determining required scantlings before detailed lines and weight distributions are known, or used by ship's officers to determine changes in the bending moment caused by variations in the distribution of cargo and consumables aboard the ship.

2. Exact methods requiring detailed hydrostatic data (Bonjean curves) and weight distributions, used primarily by a design agent or shipyard to determine the bending moments expected in service and to produce loading guides for the ship's officers.

Approximate Methods

Many variations of approximate calculation techniques have been presented in the literature on longitudinal bending moments. Descriptions of the methods proposed by W. J. M. Rankine (1866), John (1874), Vivet (1894), Alexander (1905), Suyehiro (1913), and Foerster (1930) are given by Murray in his 1947 paper (29), in which he also develops a simplified method for calculating the bending moment at amidships for vessels of normal form. Mandelli (30) introduced the concept of "influence lines" which makes possible the quick tabular calculation of the midship bending moment for any condition of loading in still water.

He also extended these approximate methods to include calculation of the complete bending moment curve. It is worth noting that all of these methods, as well as the exact calculations, may be used to determine static wave bending moments as well as static still water bending moments. Indeed most of the techniques were developed for the express purpose of determining bending moments in some standard static wave profile.

Most of the tabular forms and graphs included in ship loading manuals and in instructions for the guidance of ships' officers, which they may use to compute midship bending moments, are based on the "influence lines" approach (30). Accuracy of the results is quite good (so long as the influence of trim on the buoyancy distribution is accounted for in the graphical data furnished) because the buoyancy and moment of buoyancy are known exactly for the completed ship. Less accurate calculations using the same techniques are possible without detailed hydrostatics by approximating the influence lines and giving their equations as functions of ship form (31). For the complete still water bending moment and shear curves to be determined without detailed hydrostatics, the Faresi "integral factors method" of approximation is also available (32).

Exact Method

Since the digital computer has come into general use in ship design, the detailed methods customarily used in final ship design have become almost as easy to use as the approximate ones. The latter remain useful only if,

(a) a computer is not available -- as on shipboard (usually),

(b) detailed data are not available -- as in early design.

For other purposes, exact calculations of still water bending moments in various conditions of loading are the rule. The basic method is well known and the principles involved are not at all complicated, but the numerical work is voluminous and tedious if done by hand.

Briefly stated, the still water shear (or static wave shear) at any point in the ship's length is calculated as follows:

$$V_x = \int_0^x W\,dx = \int_0^x w\,dx - \int_0^x b\,dx$$

where V_x = shear force at distance x from bow (or stern),

w = weight per unit length,

b = buoyancy per unit length,

$W = w - b$, load per unit length.

The integrations (summations) are performed from the bow (or stern) to station x.

For determining the bending moment at any point,

$$M_x = \int_0^x \int_0^x W\,dx\,dx = \int_0^x \int_0^x w\,dx\,dx - \int_0^x \int_0^x b\,dx\,dx$$

where M_x = bending moment at station x. The double integrals are the summations of the moments of weight and buoyant forces forward (or aft) of station x, taken about station x as an axis.

It is clear that the above equations can be easily evaluated numerically using a digital computer. In fact, the simplicity of the equations is the reason for the fact that a large number of programs to calculate still water (and wave) bending moment are available to the designer today. For example, nine firms and organizations have furnished abstracts of their bending moment computer programs to the SNAME index, T & R Bulletin No. 4-13 (33). In addition, various shipyards and ship operators are known to have operating programs.

The first step in evaluating the integrals is the "balancing" of the ship for a specific weight distribution, involving calculation of the displacement and the longitudinal center of buoyancy by integrating the area under the sectional area curve and taking first moments. The correct mean draft and trim can then be determined by trial and error. The only difference between the process for still water or for a static wave is the profile of the waterline, i.e., a straight line in the first case or a specified mathematical wave shape in the latter. The second step is evaluating the integrals for as many values of x as may be needed.

Another mathematically convenient program to calculate the above is one based on a mathematical description of the hull which requires very little input information and limits the numerical integration to the minimum necessary. Such a program has been developed at Webb Institute based on the use of mapping coefficients to describe the two-dimensional ship section (34). In this new method the computations are programmed to give the bending moment and shear force anywhere along the hull for any draft and trim (or for any mathematically-defined wave profile). Hence, comprehensive investigation of a wide range of still water loadings is possible with a short computing time. The program, designated WTS 130, is presently being documented.

Electrical Methods

We have learned about two electric instruments for calculating ship longitudinal stresses and/or bending moments:

Loadmaster - Kockums

Lodicator - Götaverken

The former is particularly good because it shows visually a graph of the bending moment distribution along the ship's length. One can see immediately the effect of a change in load on the bending moment curve.

STILL WATER BENDING MOMENT TRENDS

A pilot study has been made of still water bending moments for three ships of different types: a containership, a supertanker and a bulk ore carrier. The objective was to obtain enough actual still water bending moments for each ship in the outbound and inbound loading conditions to evaluate their statistical distributions, including mean values and standard deviations for outbound and inbound voyages separately.

Sources of Data

Ideally, the proposed program of calculations would require complete voyage loading diagrams for many voyages of each ship, plus detailed hydrostatic data, including Bonjean curves. The hydrostatic properties are always made available to the ship owner by the designer or shipyard, and voyage loading diagrams of one kind or another are in common use by ships' operating personnel. However, the availability and adequacy of such loading diagrams or tables for the purpose of calculating bending moments varies considerably among different operators.

In many cases, especially for tankers and ore carriers in ballasted condition, loading data are not sufficiently detailed to permit accurate assessment of bending moments. The total amount of ballast is usually recorded, but its actual distribution is left to the judgment of the ship's officers, who are not required to record the quantities allocated to each ballast tank. Nor are records of ballast shifts at sea during tank cleaning operations retained. Therefore, significant variations in still water bending moment (SWBM) may actually occur which cannot be calculated from recorded voyage data. To a lesser degree there are similar omissions of certain items in the loaded conditions as well, usually for items whose influence on bending moment is small. By contrast, rather complete loading information was obtained for a containership for both outbound and inbound voyages.

Calculations Made to Date

The number of voyages for which loading information (complete or not) were obtained is as follows:

Containership New Orleans (Seattle - Alaska)

59 outbound voyages }
60 inbound voyages } SWBM's could be calculated.

Tanker Esso Malaysia

2 loaded voyages }
1 ballasted departure } SWBM's could be calculated.

Ore carrier Fotini L.

5 loaded voyages - incomplete data, but SWBM could be estimated.
7 ballasted voyages - data too incomplete to estimate SWBM.

Because of the paucity of reliable data on actual voyages of the latter two vessels, their loading manuals were consulted for "standard" loading conditions. The still water bending moments in these standard conditions were therefore included in the results discussed below. Since these loading conditions were intended for the guidance of shipboard personnel, they should be representative of actual practice.

Techniques and Results

As a first step toward a statistical description of the still water bending moments (SWBM's), histograms have been prepared showing the frequency of occurrence of different values of SWBM's for outbound and inbound (or ballasted) voyages. The "maxima" plotted in the histograms are the maximum values along the length of the ship, which generally occur near, but not necessarily exactly at, amidships. The many-peaked ore carrier bending moment curves required special treatment, as described below. Additional actual voyage data would be required if we were to proceed any further in the analysis of the tanker and ore carrier. Special notes regarding the methods used in the calculations follow.

Containership New Orleans. The owner furnished detailed calculations of midship bending moments in static L/20 wave profiles, hogging and sagging, for the actual cargo loading conditions of 59 outbound (Seattle to Anchorage) and 60 inbound (Anchorage to Seattle) voyages. Two modifications were made to the calculations furnished in order to arrive at the SWBM's:

1) The still water bending moment was approximated as the mean of the static wave bending moments in hog and sag, and was determined to be either hogging or sagging.

2) The resulting approximate SWBM was adjusted because the consumables (fuel and fresh water) assumed in the wave BM calculations were "burned out" for the outbound voyages and "full" for the inbound voyages, rather than the conditions actually listed for the given voyages. Actual tankages were therefore substituted in the calculations to adjust them to actual conditions. Results of the adjusted SWBM calculations are shown in Fig. 4.

Tanker Esso Malaysia. Two loaded voyages and one ballasted condition were available, the SWBM's having been calculated by the owner using his own computer program. Sufficient data to do the same for other runs were not obtainable. The other three load and three ballast conditions indicated in Figure 5 are "standard" conditions from the loading manual. Each standard condition represents either a departure or arrival condition, whichever has the larger SWBM.

Ore ship Fotini L. All SWBM's plotted in Figure 6 are from the loading manual, since available data on actual voyage loadings were insufficient to calculate actual SWBM's. When the vessel carried heavy ore in holds 1, 3, 5, 7, and 9, with the remaining holds empty, the bending moment curve has many peaks, the highest peak often occurring relatively far from amidships. Instead of one, there are several maxima in these cases. The upper plot of Fig. 3 shows the value of the highest peaks of the SWBM curves occurring within the midship 20% of length. The lower plot shows the highest peaks occurring within the midship 40% of length. It is seen that in a number of cases the peak value occurring outside of the midship 20% of length is higher than that within. In other cases, there are no significant peaks at all within the midship 20% of length.

It is tentatively concluded that for containerships a single distribution curve for still water bending moments can be established in design for a particular service. In the case of bulk carriers, two distribution curves are usually required -- one for loaded and one for ballast condition.

CLASSIFICATION SOCIETY LIMITS

It should be noted that although classification societies do not in general base hull girder strength standards on a direct addition of still water and wave bending moments, still water moments are taken into account.

For example, the current Rules for Building and Classing Steel Vessels of the American Bureau of Shipping, 1972, require an increase in deck section modulus if the maximum still-water bending moment in the governing loaded or ballasted condition is greater than

$$s\ c\ f\ B\ (C_b + 0.5)$$

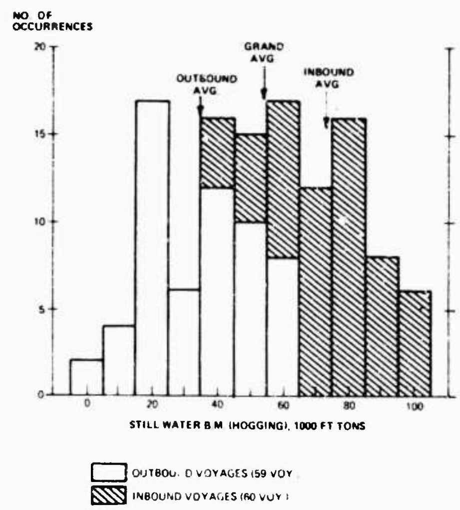


FIGURE 4 - Histogram of Still-Water Bending Moments Containership *NEW ORLEANS*

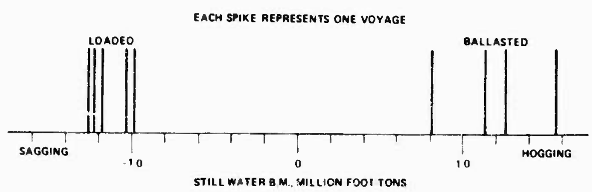


FIGURE 5 - Typical Still-Water Bending Moments, Tanker *ESSO MALAYSIA*

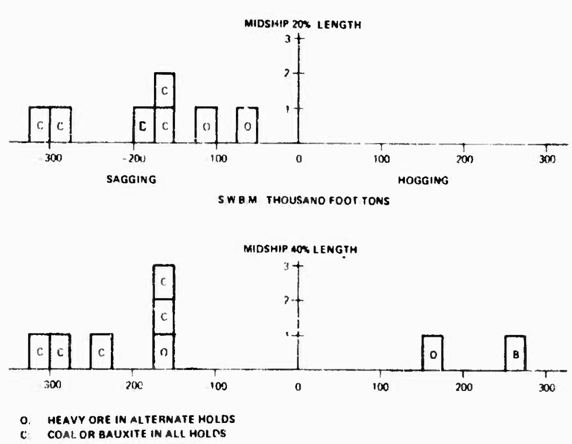


FIGURE 6 - Typical Still-Water Bending Moments, Ore Carrier, *FOTINI L.* Each box represents one voyage

where s and f are coefficients depending on length (see below), B is breadth in ft., C_b is block coefficient (not less than 0.68), and c = 1.00, except for oil carriers, where it is 1.03.

Length, ft.	f	s for oil carriers	s for others
600	594	3.78	4.25
800	1175	3.78	4.25
1000	1921	3.90	4.38

The trend of still water bending moments with ship length on the basis of the above is shown in Fig. 7.

INSTRUMENTS

Our study of still water loadings showed that in many ships a wide range of bending moments can be experienced in service. Hence, a shipboard instrument for quickly calculating still water bending moments can be an important adjunct to safe ship operation. A recommendation is made by Lloyd's Register: "In order to guard against high stresses being imposed through an unsatisfactory cargo or ballast loading, the Society recommends that an approved instrument or other means of determining the suitability of loading be placed on board" (35).

Various instruments of this type are available, as discussed at the beginning of this chapter.

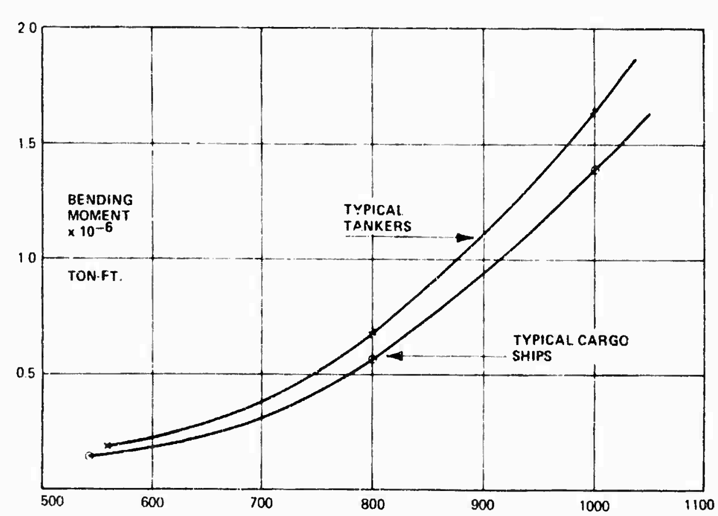


FIGURE 7 - Trends of Still-Water Bending Moment, Maximum Value by ABS Rules (1972) Requiring no Addition to Section Modulus

SHIP'S OWN WAVE

When a ship proceeds at appreciable speed in calm water, as previously noted, a wave pattern is generated which causes the ship's own bending moment. This may be a hogging moment in the case of a full ship pushed to high speed, but in fine, fast ships -- where the effect is most pronounced -- it is a sagging moment created by the bow and stern waves.

Results of systematic model tests by Vossers give the trend of bending moments due to a ship's own wave over a range of block coefficients and speeds (36).

Model tests with destroyers have shown (37) that when waves are encountered the effects of the ship's own wave and of the ocean waves are superimposed with very little interference.

SUMMARY

It has been shown that still water bending moments can readily be calculated by available techniques in the design stage. But relatively little data in statistical form are available for actual service loadings of various ship types, particularly in ballast conditions.

IV. WAVE LOADS

INTRODUCTION

The problem of wave-induced loads on a ship at sea is that of determining successive conditions of dynamic equilibrium of forces and moments acting in and on an elastic body moving in the irregularly disturbed interface of two different media. This problem can be simplified by considering external loads only, on the underwater part of the ship, which is considered to be a rigid body in an ideal fluid. Motions and other ship responses in waves are regarded as linear functions of wave height, and both the irregular waves and the irregular responses can be considered as the sum of many sinusoidal functions. Hence, the analysis begins with the study of harmonic oscillations of a rigid body, moving at forward speed on the surface of an ideal fluid under the action of regular gravity waves.

Though in principle the ship motion problem has been solved for three-dimensional cases (38)(39), the analytical solution is limited to forms such as a sphere or an ellipsoid. In view of this, a less rigorous strip theory solution has been developed which is suitable for long, slender bodies, where each cross section of the ship is considered to be part of an infinitely long cylinder. Hence, a series of individual two-dimensional problems can be solved separately and then combined to give a solution for the ship as a whole. The idea was originally introduced by Korvin-Kroukovsky (40) and has since been endorsed, criticized and improved by many authors (41)(42)(43).

The main drawback of the strip theory is that it neglects the mutual interactions between the various cross sections, which are of particular importance for certain frequency ranges, depending on the size of the body. Hence in waves that are either very long or very short relative to a ship the theoretical justification of strip theory is somewhat questionable. This statement is particularly applicable to lateral motions, since the hydrostatic restoring force is small or non-existent under these circumstances.

In spite of the above reservations, the basic strip theory has been found to be very satisfactory for heave and pitch motions and bending moments in head waves (40), and it is the only suitable method for numerical computation. Modifications have included the use of "close-fit" methods, which have led to a significant improvement in the computation of the sectional added mass and damping coefficients for all but the simplest sections. An additional major contribution to the theory has been the inclusion of all the forward speed terms in the equations of motion in order to satisfy the symmetry relationship proved by Timman and Newman (44). All the modified strip theories developed in the past two to three years (45)(46) have practically identical forward speed terms. Extension of the theory to oblique waves, to lateral motions, torsional moments, and lateral bending moments has also been achieved, as shown in (39)(43). Finally the use of close-fit mapping techniques and strip theory for determining the distribution of pressures over the hull has been demonstrated (47)(48).

Since we are concerned with successive conditions of dynamic equilibrium, it should be noted that a complete solution of the problem of wave loads and bending moments cannot be obtained without first determining the motions.

LOADS IN REGULAR WAVES

Basic Theory

In order to evaluate the state of development of ship motion and load calculation in waves a short analysis of the basic approach of most investigators will first be given. The mathematical formulation of the problem, i.e., a ship advancing at constant mean speed with arbitrary heading into regular sinusoidal waves, can be presented in most general form by defining the velocity potential so as to satisfy the assumptions of the ideal fluid, linearized theory. At this initial stage no strip theory assumption is required. The time-dependent part of the potential can be decomposed into three components representing the potentials due to incident wave, defraction and the mode of motion considered, as in the original theory by Korvin-Kroukovsky (40). However, an additional time-independent term due to steady forward motion of the ship has been added in more recent theories (43).

Once the formulation of the component potentials is completed, the hydrodynamic forces and moments acting on the hull can be determined. Using the Bernoulli equation the pressures in the fluid are defined and expanded in a Taylor series about the undisturbed still water position of the hull. Ignoring steady pressure terms at first,the linearized time-dependent pressure on the hull can be formulated and integrated over the hull surface. The hydrodynamic forces and moments can be obtained in two superposable parts: those associated with a wave passing a restrained ship (excitation) and those acting on a body forced to oscillate in calm water.

In order to obtain a numerical solution the application of strip theory approximations are necessary for the integration of the sectional exciting and motion-related forces over the length of the ship. These sectional forces involve two-dimensional added mass, damping and displacement terms. The speed-dependent coefficients are expressed in terms of a speed-independent variable, which is evaluated by means of strip theory, and of a speed-dependent term which is obtained from a line integral along the waterline as given by Stoke's theorem.

Hence, the main difference between the original strip theory in (40) and the more recent "new" methods is in the formulation of the problem. In (40) the strip theory assumptions were applied in the initial formulation, and the forward speed effect was only introduced in certain terms. In the "new" theories the assumptions with regard to strip theory were made after the general terms for the coefficients in the equations of motion were determined, including the forward speed terms.

In addition to the above, the theory presented in (43) includes end terms in the coefficients associated with the aftermost sections, which are not usually included in the strip theory and are claimed to be important for bluff bodies. These terms are independent of the strip theory assumptions.

Using either the old or the new approach, the formulation of hydrodynamic forces and moments permits the equations of motion to be solved and the amplitudes and phase angles of motion determined. Then the longitudinal distribution of all forces -- including those that are dependent on the motions and forward speed -- can be evaluated and shearing forces and bending moments calculated for any instant in the motion cycle, usually at midship. In general the solutions for two instants of time suffice to determine the amplitudes and phase angles of these quantities.

In general, design calculations, full-scale data collection, and model tests concentrate on conditions amidships. This is a sound procedure, particularly for collecting statistical data on different ships at sea. But some consideration must also be given to the longitudinal distribution of loads (48A). There are two questions to be answered by suitable trial calculations or experiments:

1. Are bending moments ever significantly higher at any other section than midships?

2. Over what ship length do midship bending moment values extend before they begin to taper off significantly?

It is felt that these questions can be answered by a limited number of systematic calculations, using the technique just described, on the basis of regular waves of various lengths, without the need for collection of long-term data or calculation of irregular wave responses.

Computation in Regular Waves

The preceding section treated the overall problem of ship motions, shear forces and bending moments in a seaway, indicating the need for a strip theory solution if numerical computations are required. In the following, the specific computer programs available for the above computations and the theories associated with them will be discussed.

When referring to an available computer program indication will be given of the degree of availability in terms of whether the program is public property or whether it is of proprietary nature.

The two programs in the first category are "SCORES," developed by Oceanics, Inc. under SSC Project SR-174, and the MIT program developed under sponsorship of the Maritime Administration. Though the basic equations of motion are identical for the two programs, the scope is somewhat different. SCORES calculates the vertical and lateral motions and loads and torsional moment, while MIT is limited to the vertical longitudinal plane only, but includes additional information such as approximate mean added resistance in waves. SCORES is documented in a recent Ship Structure Committee report (49) and the MIT program in two MIT reports (50)(51).

Although the basic equations of motion for pitch and heave are identical in both of the above programs, there are slight differences in the coefficients of the equations, as well as in the excitation forces and moments. Both programs are based on coefficients originally derived in (40) and later modified slightly by (41). The only difference in the coefficients is in the restoring force coupling terms which are corrected in the MIT program to account for the fact that the origin of the ship coordinate system is taken at midship rather than at the more conventional location at the center of gravity. The programs are therefore virtually identical for vertical motion and loads when the center of gravity is near midships. It should be noted that the routines used to calculate the two-dimensional added mass and damping coefficients are also identical in both programs and are based on (42).

Note should be made of the fact that the MIT program includes a special routine to handle bulb sections which cannot be properly mapped by the commonly used "Lewis" routine (52). However, both programs lack a general routine to handle any shape, such as can be obtained by "close-fit" techniques. SCORES includes the motion and loads in the lateral plane for which the two-dimensional properties are calculated according to (46).

An additional program available to the public through the NSRDC is the "YF-17." This program is limited to motions in the longitudinal vertical plane and is based on the same equations and coefficients as the above two programs. The main feature of this program is its close-fit subroutine which allows an exact mapping of any required section and therefore is of particular value for ships of non-conventional section shapes. The YF17 is the most advanced available program at NSRDC. A new program based on the theory in (43), which includes ship motions and loads in three dimensions, as well as a new strip theory approach, is not yet available for public use and is now being tested and modified. In the case of certain types of ships, such as Naval destroyers or very full tankers, a combination of elements from two or three of the above programs may prove to yield the best results.

It should be noted that all of the above programs supercede the original Davidson Laboratory program based on the theory in (40). In addition to the latter, several less-known programs, generally not available to the public, also exist. The University of California has a computer program called "SEALOAD" for the calculation of dynamic loads (pressures) at discrete points on the hull of a ship for a range of frequencies, speeds and headings. It is based on motion in the vertical and the lateral directions. A similar program for calculating the transverse as well as longitudinal pressure distribution on the hull due to motion in head seas is available at Webb Institute (47).

The above survey does not include programs presently in use outside the U.S.A., mainly because of lack of information with regard to the details. Such programs exist in Japan (46), Germany (43), the Netherlands (41), Norway (43), and U.S.S.R., but are generally not available for distribution, although specific runs can usually be purchased.

For most of the purposes of this project the SCORES program (Oceanics) is the only suitable one available. Consequently, it has been adapted to the Webb computer facilities, and is now regularly in use. It will be a simple matter to upgrade this program in scope or in detail as new developments appear. Each section is supposed to be amenable to a conformal transformation from which the hydrodynamic coefficients can be derived for vertical and lateral oscillations. The particular coefficients used are:

Vertical	oscillations:	Grim (42)
Lateral	"	Tasai (46)

The hydrodynamic forces are then obtained for each strip after the motions are calculated. The integration of the difference between hydrodynamic forces and gravity forces yields the shear force and hence the bending moment for a particular position of the wave along the length of the ship. The results for all wave lengths and headings are the response amplitude operators as a function of the wave circular frequency per unit wave height.

Comparison of Theoretical and Experimental Bending Moment

A comprehensive comparison between calculations by program SCORES and model basin test results in regular waves has been made by Oceanics, Inc. (53). Results are presented graphically for the cases listed below, covering a range of wave headings in all cases.

	Load Conditions	Speeds	Bending Moment Vertical	Bending Moment Lateral	Torsion	Shear
Wolverine State (54)	2	2	x	x		
Series 60, Block 0.80 (55)	1	1	x	x	x	x
Series 60, Block 0.70 (36)	1	4	x	x		
T-2 tanker (56)	1	5	x	x	x	

The following conclusions were drawn from the graphical comparisons:

> "The comparison between calculations of vertical and lateral bending moments and the experimental results for the Wolverine State indicates generally very good agreement. This holds for both loading conditions, both speeds, and over the range of wave angle and wave length. The experimental results shown for lateral bending moment in head and following seas, where lateral motions and loads should be zero as in the calculations, are regarded as indicative of the possible error, or range of discrepancy, to be expected between calculations and experimental results. These loads are believed to arise in the model tests due to its free-running, but rudder controlled, condition. That is, the model may undergo small lateral motions, with rudder corrections to keep course, which leads to the measured lateral bending moments.
>
> "The comparison for the Series 60, block 0.80 hull for vertical and lateral bending moments indicates excellent agreement, in general ... The agreement for torsional moments is only fair and indicates excessive response at roll resonance conditions. The agreement for the shear forces is quite good, in general, with the exception of some deviation in lateral shear at 110° wave angle. However, the shear forces are generally small at midships, and should really be investigated at the quarter-length points."

For the Series 60, block 0.70 hull form and the T-2 tanker,

> "In general, the agreement is fairly satisfactory, considering the factors involved in the experimental comparison. With regard to this point, consider the double peak calculated vertical bending moment response for the T-2 tanker at 120° wave heading and 1.65 fps model speed. While the corresponding experimental data do not indicate such a response, similar double peaked responses for vertical bending are confirmed by experimental results for Wolverine State, full load and the Series 60, block 0.80 hull. The greater resolution of the test data due to testing at more wavelength conditions for these latter cases tends to produce such results, thereby limiting the utility of the experimental points for the T-2 tanker as a complete measure of bending moment variation" (53).

Bending moment calculations for the Wolverine State carried out by means of Program SCORES are compared with experimental results in (53).

Another available comparison between theory and experiment is for the tanker Universe Ireland (57). Model tests were carried out at the Davidson Laboratory (58) and calculation using a modified SCORES program were made at Webb Institute, both under the sponsorship of the American Bureau of Shipping. Vertical bending moment results for the head seas case in both full-load and ballast conditions indicate excellent agreement between theory and experiment over the range of wave lengths tested. Similarly good results were found for 150°, 30°, and 0° headings, full load.

For the 60° and 120° headings, the comparison is not as good as for the above-mentioned cases. Although similar trends are maintained, the differences in magnitude at some wave lengths are rather large. A possible explanation of the above discrepancies is the variation between the actual and desired heading angle for the free running model in oblique waves. It has been previously shown (59) that deviations of up to 10° in the heading angle can occur for certain wave lengths. This effect will be most pronounced for the 60° and the 120° cases, and any small change in heading angle for these cases may result in a large shift in the response curves, as proved to be the case.

An extensive comparison was made between Series 60 model results obtained at Wageningen and strip theory calculations (60). The vertical bending moment comparisons for head seas covered block coefficients of 0.60, 0.70, and 0.80, and for $C_B = 0.70$ three values of L/B, three values of L/H, three values of gyradius, four forebody section shapes, and four heading angles (1 case only). Agreement with the basic Korvin-Kroukovsky strip theory was excellent except for some higher speed cases and cases of large gyradius and extreme V forebody.

Extensive comparisons are given by Faltinsen (61) for an 0.80 block coefficient hull for which Wageningen model tests were available (55). Using the basic Korvin-Kroukovsky theory (on which SCORES is based), agreement was good except at 50° and 130° heading angles -- as in the above results for Universe Ireland. However, using the more refined procedure developed by Salvesen, Tuck and Faltinsen (43), vertical bending moment results were good at all angles. The refined theory generally gave better results for vertical shear also.

Since comparisons between model tests and theory showed generally good agreement in vertical bending and shear, the theoretical calculations are believed to be satisfactory for the present stage of development of load criteria, especially when it is remembered that the integration of results over a range of ship-wave headings averages out the result and reduces any errors considerably. But further refinements in SCORES are needed along the line of Salvesen (43) and greater precision in determining lateral bending and torsional moments is needed.

Rudder Forces

Kaplan and Raff point out that "since the lever arm of the rudder forces is large for moments at midships, it appears that rudder forces can significantly affect the lateral bending and torsional moments. To the extent that the use of the rudder affects the overall ship motion response in oblique seas, the vertical bending moment also can be influenced, but to a much smaller degree" (53). Rudder effects are not included in SCORES, but they are taken into account by Grim and Schenzle (62). However, the rudder effect is not separated out so that its relative importance can be assessed. They also included the effect of anti-rolling fins on torsional moment, which produced a significant reduction. It is felt that rudder effect can be neglected insofar as vertical wave bending moments are concerned, but it should be given further attention with reference to lateral bending and torsion.

LOADS IN IRREGULAR WAVES

The extension of regular wave results to short-crested irregular seas, by means of the superposition principle, was accomplished by St. Denis and Pierson (63), on the assumption that both the irregular waves and the ship short-term responses are stationary stochastic processes. The computer calculation of ship response was obtained some time ago, using model test data as inputs (64). In the case of all the above-mentioned computer programs, however, irregular wave capabilities are incomplete. The MIT and YF-17 programs have been extended to long-crested seas as represented by the Pierson-Moskowitz one-parameter sea spectrum family, and statistical response parameters can be computed from which various seakeeping events are determined. The SCORES program has been extended by Oceanics, Inc., to include response to short-crested seas, as well as a limited option for the use of a two-parameter spectrum formulation (dependent on both wave height and period). These are features which are considered to be absolutely essential in order to avoid erroneous results which may be obtained otherwise, such as near-zero longitudinal bending moment in beam seas. Furthermore, it has been previously shown (65) that superposition of responses to irregular waves should not be limited to a single spectrum for each sea condition (i.e., a one-parameter family), but a group of spectra should be used in order that both the **mean** response and its standard deviation can be determined for each range of wave height. Such programs are available using as input spectra either ordinates of actual spectra obtained for the North Atlantic or tables of probability of occurrence of certain combinations of wave height and period, which are substituted in the ISSC two-parameter spectrum formulations. These programs, designated as WTS 120 and WTS 121, respectively, were recently used to extend the SCORES program at Webb as additional options available to the user.

The present status of wave data will be reviewed in a subsequent section.

In the numerical exmmple given in Chapter IX, the Webb "H-family" of spectra, based on wave spectra from (66), is used **in** conjunction with WTS 120. For each spectrum the standard spreading function was applied:

$$S(\mu) = \frac{2}{\pi} \cos^2 \mu$$

where μ represents the wave direction angle. The RMS response corresponding to each wave spectrum was obtained from,

$$(RMS)^2 = \int_{-\pi/2}^{\pi/2} \int_{0}^{\infty} S(\mu) \quad S(\omega) \ | \ R\ (\omega)|^2 \ d\omega \ \ d\mu$$

Then the mean m_j and standard deviation s_j of RMS values within each wave height group j were determined.

LONG-TERM COMPUTATIONS

The final step of ship loading response calculations is to extend the available calculations for limited periods of time in specific irregular sea conditions to long-term predictions, covering the lifetime of a ship or a fleet of ships. The mathematical model and program WTS 62 were described in detail in (10). The only additional required input to this program is a distribution of sea conditions for the particular ocean area of interest.

The mean m_j and the standard deviation s_j for each wave height group, together with the probability of occurrence of that group p_j, give us the necessary parameters to obtain the long-term distribution:

$$P(x_o) = \sum_{j=1}^{5} p_j \int_o^{x_o}\int_o^{\infty} p(x_o;\ RMS)\ p(RMS;\ m_j,\ s_j)\ d\ RMS\ dx_o$$

where $p(x_o;\ RMS)$ = Rayleigh distribution of x_o, for a particular value of RMS, and $p(RMS;\ m_j,\ s_j)$ = Normal distribution of RMS. The technique is applied to the shear forces, and to vertical, lateral and combined bending moments.

It is believed that the incorporation of the above programs, and the various modifications to SCORES, have produced the most comprehensive program presently available in this country for use in wave load aspects of ship design.

It should be noted that the long-term probability density function can be expressed as follows:

$$p(x_o) = \sum_{j=1}^{5} p_j \int_o^{\infty} p(x_o;\ RMS)\ p(RMS;\ m_j,\ s_j)\ d\ RMS$$

This function is of importance for two applications, in particular:

1. For combining with the probability density function for still water bending moment. (See Chapter VIII).

2. For combining with the probability density function for structural capability to determine the probability of failure (31).

The present status of wave data is reviewed in the next section.

WAVE DATA

Wave data available for calculation of bending moments have been obtained in two ways: by actual measurement of the surface elevation over a period time and by observation of wave heights and periods. Each of these sources will be reviewed in turn.

Wave Records

The largest source of data available under the first category is the NIO wave records collected by means of Tucker wave meters on British Weather Ships over the past 18 years in the eastern part of the Atlantic Ocean. The records are available in the form of strip charts. A typical example of one type of analysis which can be performed on such data is given in (67). The analysis is limited to three basic parameters, the peak-to-trough mean height, the mean zero-crossing period and the mean-crest period. Other parameters, such as the significant wave height, or the spectral width parameter can be approximated from the above measured parameters.

Strip chart wave data, though basically suitable for any type of further analysis, are in practice rather expensive to transfer to the frequency domain. However, in order to extract maximum information out of such records a spectral analysis should be performed in order that the results can be given in a form representing the distribution of wave energy of the irregular wave components over frequency, i.e., an energy spectrum. This constitutes the ideal form of wave data presentation, for ideally if one is to predict the performance of a ship under certain environmental conditions the actual spectrum, or a family of spectra representing different possible wave heights at that location, should be used.

Very few sources of wave records reduced to spectrum form are available in quantities adequate for statistical sampling. The main source of data is (66) and (68). The former is a biased sample of 460 records selected from the North Atlantic weather stations representing mostly fully-developed storm conditions. The latter constitutes a sample of 307 records all taken at one station at midday over a period of 12 years, using a stratified sampling procedure. These records were selected at Webb Institute from National Institute of Oceanography (Great Britain) records, with the assistance of Professor Pierson of New York University, and were spectrum analyzed by the National Research Council (Canada). Correlation of results with Beaufort No. and other analyses are being carried out currently at Webb Institute.

An area for which more measured wave data are needed is the North Pacific. It is hoped that several projects now under way will produce some useful data. Other sources of ocean wave spectra are Scott (69), Fukuda (70), and Yamanouchi (71).

An additional parameter often associated with spectra is the normalized average period slope parameter, K. The value of K is an indication of the deviation of the actual average period of the spectrum from that of a fully developed Pierson-Moskowitz spectrum and was originally introduced in (72). By defining K, as well as the significant wave height and period, some further indication as to the nature of a spectrum is provided. A typical distribution of K values against wave height was recently given in (73) based on 307 records collected at Station INDIA in the North Atlantic.

A limited amount of data has been obtained by means of a buoy measuring wave slope (74). Analysis of the records permitted the determination of directional spectra that give the distribution of wave component directions as well as frequencies.

It must be recognized that shoaling water has a significant effect on wave patterns. The waves tend to become steeper as they travel in from deep to shallow water. Although this effect is noticeable only in water depths of magnitude comparable to wave length, there are a number of ocean areas where the effect is significant for large, modern ships. Of particular importance is the continental shelf at the entrance to the English Channel. In general, the relatively shallow waters of the North Sea, Irish Sea and English Channel are characterized by short, steep seas. Some wave data including representative spectra are available from British sources (75)(76).

Wave Observations

A much more common way of describing the world oceans, because of its relative simplicity and low cost, is by means of the observed heights and periods of the waves and their frequency of occurrence. Both the wave height and period can be estimated to varying degrees of accuracy by seagoing peronnnel. The most reliable data of this type are those available in reports of trained observers on board weather ships, as well as at various permanent weather stations around the world.

However, the most common source of observed data is the reports of various ships sailing throughout the world's oceans. The most comprehensive such collection of data is given in (77), which covers most of the ocean areas. Other sources are (78)(79)(80)(81)(82)(83)(84). Similar, more localized tables are available for specific areas of interest such as the Great Lakes (85), Cape of Good Hope, (86) etc. The information is usually given in tabular form showing the probability of occurrence of the various possible combinations of wave height and period. An alternative form of presentation is by means of a histogram.

Figure 8 gives a comparison of the results given by various sources, mostly for the North Atlantic. Local data for the vicinity of the Cape of Good Hope is also included, in order to show how a local condition may differ from commonly used data. This ocean area is of considerable interest because large tankers regularly traverse it between the Persian Gulf and Europe, and long, heavy swells are known to occur there at times. More data are needed, but the new information helps to round out the picture.

The interpretation of observed wave height and period in terms of the significant wave height and the average period is not easy. It has been found, however, that for the untrained observer a simple approximate relation between the observed wave height H_v and the significant wave height $H_{1/3}$ can be established (87). Similar relationships can also be established for trained observers. When wave records are available it is customary to refer to the period as the average zero-crossing period.

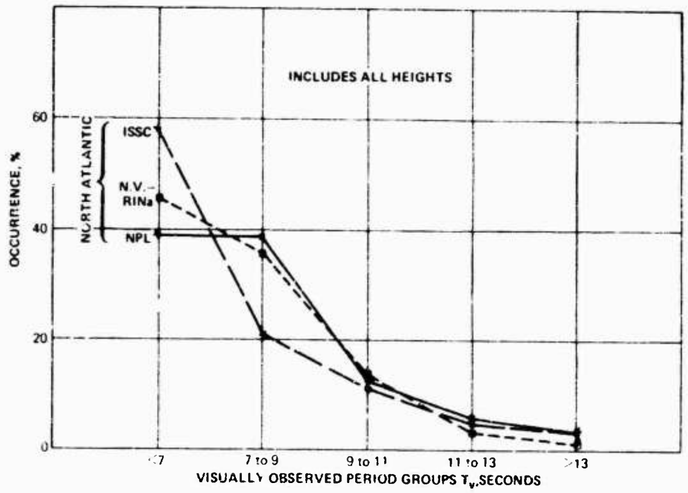
Mathematical Formulations for Spectra

In order to make use of observed wave data for predicting ship motions or wave-induced loads, it is essential to make use of some idealized spectrum formulation. The description of the sea by means of two basic observed characteristics, i.e., wave height and period, can be achieved by means of a mathematical expression of the spectrum in terms of these two parameters. The basic parameters are mathematically defined in terms of the area under the spectrum and the moments of the area. Several such formulations are available, and the most common one is usually referred to as the ISSC spectrum formulation (88).

For each pair of values for wave height and period a spectrum is defined in terms of the spectral ordinates at discrete values of the frequency, ω. Hence, for a matrix of wave heights and periods a family of spectra can be obtained for which the probability of occurrence of each spectrum can be defined and applied as a weighting factor. Though the mathematical formulations do not completely describe the variability of spectral shape, they can be used for long-term predictions of bending moments with reasonably satisfactory results (10).

Wind Parameter

In the absence of any of the above information, i.e., the actual spectra or observed wave heights and periods, the only way to describe the sea is by means of the wind speed, which can be considered to be the single most important factor in generating waves. It is apparent that wind speed alone is not adequate for an exact description of the sea, since other factors such as duration, fetch, tempera-



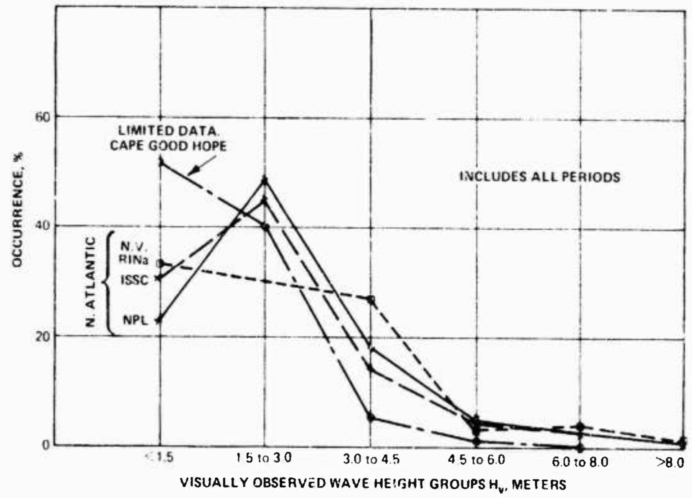


FIGURE 8 - Comparison of Wave Statistics: Observed Periods and Heights

tures, etc., are important. However, wind speed or Beaufort No. is a relatively easy parameter to measure and is commonly available in most weather charts. Relations between wind speed and wave height have been suggested in several references such as (69)(71). It is always meaningful to relate the wind speed to a mean significant wave height, so long as the standard deviation is also given. One should realize, however, that in lakes or gulfs such data are less meaningful because of the limited fetch conditions which prevail.

The relationship between wind speed and wave period is much less significant and not so easy to define. However, if the significant wave height is available some typical relationships between it and the wave period can be given for specific ocean zones.

FUTURE DEVELOPMENTS

The short survey presented here indicates that a great deal of progress has been made in recent years in the theory of ship behavior in waves, and a variety of computer programs is available to apply this theory to specific problems. However, these programs are only partly coordinated, and only a beginning has been made in showing how different programs can be combined for design use in the area of wave loads. It is felt that further work should be done in documenting, evaluating and improving individual programs, in comparing results in specific cases -- among programs, and between programs and experimental results -- and in determining how different programs can best be combined for various problems and for different types of ships.

An overall view of the problem of determining wave loads discloses that experimental and theoretical aspects have been brought to an advanced stage of development, but that the biggest gap is in our knowledge of ocean waves. Actual wave records that can be subjected to spectral analysis are available for only a limited number of locations on only a few routes, leaving a great uncertainty regarding all of the major trade routes of the world. The collection of more wave records is believed to be of primary importance.

V. DYNAMIC LOADS

INTRODUCTION

Dynamic hull loads may be classed as:

a) steady-state,

b) transient.

Steady-state dynamic loads are the result of such exciting systems as propeller-hull interaction, machinery-hull transmission and seaway component excitation of hull girder response, as in "springing." Transient dynamic loads result from forward bottom slamming, bow flare immersion and submergence, carrying green seas aboard and wave impacts on structural members during rough sea operation.

The transient loadings will be considered first. There are a multitude of possibilities for such transient loadings, and the generated impacts on the ship's exposed surfaces often result in structural damage of considerable severity. The impact loadings are generally most severe in the forward portions of a ship's structure where relative velocities between ship and sea are highest. These impact loads arise from the short-time exchange of momentum between ship and sea, and the damage results when the local elastic energy absorption capability is inadequate. Energy absorbed locally is distributed through the structure in the form of dynamic response, "whipping," etc., and dissipated through damping, structurally, by cargo, etc. The question of concern is what loading is to be used in design. There are as yet no established criteria for assessing structural loadings arising from impacts of the sea on ship structure. Consequently, a real need exists for the documentation of resulting damages and the conditions leading thereto, so that future evaluations of probable loads would be possible.

Unlike quasi-static bending moments, dynamic loads acting on the hull girder cannot be considered independently of the structural response of the hull. In the first place, the dynamic magnification factor depends on the natural hull frequency as well as the nature and duration of the exciting force. Secondly, the response depends on the damping, which involves structural as well as hydrodynamic terms. Thirdly, the extreme structural response of the hull which is of particular interest -- plastic buckling or permanent set -- requires time and the absorption of large amounts of energy. As mentioned in Chapter II, the duration of a slam load may be too short to cause buckling of a ship's deck or bottom amidships. This is probably not true, however, of the longer-duration flare-immersion type of slamming which in some ships may contribute significantly to hull girder damage or failure.

Hence, the capability of the structure to withstand a dynamic load or stress may be increased by some factor which is dependent on the rate of application and duration of the load, as well as on the structural properties of the hull. Pending resolution of this structural design problem, it is important that techniques for determining and specifying the magnitude, duration and rate of application of the dynamic loads be dealt with. The problem of phase relationships between dynamic and static loads will be discussed in Chapter VI.

SHIPPED WATER

Consider first the miscellaneous loadings arising from taking aboard green seas over the forecastle, poop and even at boat deck levels and higher. Such loads have resulted in the shearing off and washing away of outfit items, deck houses and bulwarks. More seriously, hatch covers have been stove-in, fo'c'sle decks collapsed and deck cargoes torn free of their lashings and washed overboard (89). There are no known published evaluations of the sea forces associated with these actions, though classification societies and cargo survey bureaus have revised scantling, stowage and other requirements on the basis of reported damage. Most loadings of this type are sufficiently random in occurrence, both in time and space, that their importance is primarily local, even though the hull girder may on occasion be excited in a manner which will superimpose significant stress variations upon those arising from static and quasi-static hull girder flexural, torsional and shear loads. Better documentation of shipping water damages and evaluation of the loadings that cause them would allow assessment of such loads for purposes of setting criteria.

Loading of the hull girder as well as local structure resulting from bow submergence into oncoming waves can cause, in addition to severe local damage, significant dynamic loading of the hull girder and generate structural response of such magnitude and duration that the stress generated can significantly augment the levels experienced in main structural elements of the ship. The problem of green seas impacting the ship and being carried over the bow has received attention by investigators in Europe, the U.S.A. and Japan (90). Hoffman and Maclean (91) have shown that in order to predict the event, knowing the ship's relative motion operators and the sea condition is sufficient. Kawakami (92) has shown that carrying green seas over the bow can generate whipping stresses equal in magnitude to as much as 40%-70% of the sagging bending moment in regular waves. Thus, the prediction of the event and the magnitude of the loading response of the hull girder appear amenable to treatment by combined experimental and theoretical means. There has been, however, no full-scale confirmation of experimental findings, or vice versa. Consequently, there is not at present an established procedure for evaluating the magnitude and frequency of hull loading resulting from carrying green seas aboard at the ship's bow. Though experimental work by Kawakami (92) showed that full-scale fore deck loadings of 40-70 lb/in^2 could be experienced, generalization of these findings has not yet been attempted.

In view of the fact that these loadings can result in the generation of significant hull girder responses which are of sufficient duration to cause superposition on the quasi-static loading of the structure, it is recommended that further work be carried out in this area with a view to establishing full-scale experience and combined experimental-theoretical clarification for design.

FLARE IMMERSION

Slamming of the forward bottom and bow flare surfaces give rise to large loads on the hull girder of sufficiently short duration that high-frequency hull girder dynamic stress variations are generated of such magnitude that they may significantly augment the maximum stress levels experienced by main structural elements. The slam-generated forces developed on the ship forward bottom surfaces are generally the result of high pressures acting over relatively small areas of the flatter bottom plating; significant force duration is generally less than 100 milli-seconds. The forces developed by the immersion of the bow flare are generally the resultant of lower pressures acting on relatively larger surface areas of the hull. The duration of these forces is generally greater, but usually of less than a second with significant magnitude. These latter forces are of importance primarily to vessels with large bow flare, such as some aircraft carriers and some recently designed commercial vessels with large bow flare and overhang.

The generated pressures and resulting forces developed on the bow flare are susceptible to computation on the basis of available theory. The initial effect as the bow flare enters the water is a non-linear addition to the quasi-static wave-induced bending moment. A step-wise evaluation of the exciting force for the calculation of vibratory hull girder response, as discussed below, can be used to compute the initial bow flare immersion effect. The duration of this bending moment is sufficiently long that it should probably be added to the other moments involved in ultimate strength.

The vibratory response or whipping following flare entry can be determined by analog simulation (93)(94) or by deterministic calculation (95)(96). This short-duration response, generally in the fundamental mode, should be considered in relation to fatigue and brittle fracture. However, there are no known published recommendations concerning the magnitude of bow flare-generated forces that should be considered as design loads either for the hull girder or the local structure forward.

The basic theory of impact for a wedge entering the free surface of a fluid is applicable to bow flare load generation. Such theory, developed by Wagner, von Karman, Pabst and others has been modified by Szebehely, Ochi, and others for application to ship forms in which the body is not stopped by the impact. This theory has been the basic ingredient for all the work mentioned above. To use it for the generation of load information, as per Kaplan's program (95)(96), requires a deterministic approach in which a mathematically defined hull is subjected to an irregular wave pattern derived from a sea spectrum of interest. Kaplan's procedure allows the prediction of ship loads on such a basis and, upon repeating the process for a suitable number of sea conditions, a body of load data could be developed for use in a suitable probability model.

The chief drawback seen in the above approach appears to be the fact that sinkage and trim, as well as dynamic bow wave build-up, should be included if reliable bow immersion results are to be expected. These non-linear effects have been identified in experimental work (91) as vital to reliable prediction of flare submergence. The works of Tasaki (97) and Tasai (98) are also useful here. Further development of the theory should be carried out.

It is believed that a part of a rational design criterion would be first the calculation of initial bow-flare immersion bending moment for one or two representative severe head sea conditions, especially if the ship has considerable flare and operates in a deeply loaded condition. An exhaustive statistical study should not be necessary but simply a long enough irregular sea run to determine the highest value in, say, three to four hours.

The problem of phasing of flare-immersion bending moment appears to be much simpler than that of slamming (See Chapter VI). The bending moment (sagging) gradually increases as the flare enters the water and reaches a maximum at or near the time when the wave bending moment is also a maximum in sagging. Hence, the combined effect of wave bending moment and flare immersion can be approximated by adding the maximum values directly.

Secondly, the vibratory response (whipping) that follows flare immersion should be calculated for ships with large flare by the method developed by Kaplan (96). It is recommended that this approach be developed into a standard procedure for use in relation to design for ultimate strength, fatigue and brittle fracture.

BOTTOM IMPACT

The initial effect of a bottom impact is the generation of large local pressures which may in some cases cause local structural damage. The resulting hull girder response is in the higher modes, as well as the fundamental, and therefore may produce a high initial stress. The whipping that follows will be mainly in the fundamental, since the higher modes are quickly damped.

In respect to loads generated by forward bottom slamming, present capability is restricted to the prediction of peak pressures at a point on the forward bottom centerline of typical hull forms. Based on the work of Ochi (99), the hull form of a ship's forward bottom sections can be assigned coefficients, k, such that for a known impact velocity, V, the peak pressure generated at a section would have a most probable value $p = kV^2$. The coefficients, k, for some typical section shapes have been determined on the basis of regression analysis of 3-dimensional model experiments in waves (100). Using these values, a comparison can be made respecting the slamming propensities of competing hull forms. To do so, the ship response properties must be determined, experimentally or analytically, and the relative velocity spectrum for the critical ship section established. Using this approach, maximum slam-generated pressures in a given sea can be predicted on the basis of probability. Such data may be used for design of local structure if the probability can be prescribed with a satisfactory degree of confidence. Unfortunately, no full-scale assessment of local loading has yet resulted in the establishment of a probability level that could be used as above; a recommendation is made that such be pursued.

It is understood that Dr. Ochi is working on a mathematical description of slam loading which attempts to describe the distribution of pressures in both time and space. With such a load definition various procedures are available for calculating the hull girder response and hence the midship slamming stress (101)(102). However, the requisite load definition is not yet available.

Because of the above situation, it is not possible to prescribe at this time a slam loading of the hull girder, due to forward bottom slamming, which should be used in setting design criteria. On the other hand, the problem can be viewed in another light somewhat more positively. Full-scale strain recording on board the S.S. Wolverine State has captured hull girder slam and whipping stress variations experienced during rough sea operations of the vessel. Evaluation of data (103) obtained in various high sea states has shown that the severity of slam response -- when it occurs -- tends to be somewhat insensitive to sea condition. This can be explained by noting that though slamming frequency and severity are increasing functions of sea state, sea speed and vessel heading, the latter two are under the control of the ship master who will tend to alter course and speed so as to keep the frequency and severity of such sea loadings within reasonable bounds, according to his experience. Fig. 9 shows histograms of midship slam stress variation for the Wolverine State (103), which can be interpreted in terms of equivalent hull girder bending moment.

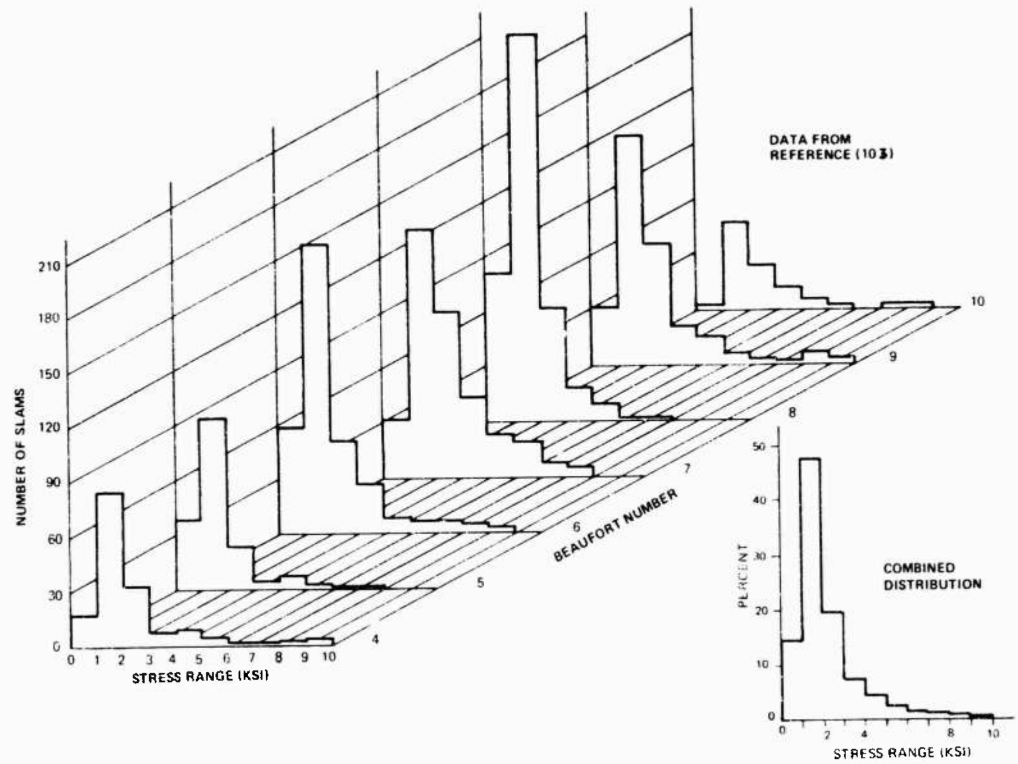


FIGURE 9 - Peak-to-Peak Slam Stress Distributions in Different Weather Conditions, *S.S. WOLVERINE STATE*

Whereas the findings as to the relative level of loading experienced by the S.S. Wolverine State would apply only to that vessel and in its particular trade, it seems apparent that the collection of similar data on a sampling of vessels of other types loaded in different ways should provide a suitable basis for the development of allowances for slam loading of the hull girder. Further, in view of the complexity of the slamming phenomenon, it does not appear that an analytic description of slam loads will be forthcoming in the foreseeable future, and the collection of full-scale slam response data appears to offer the most promise for the establishment of expected hull girder slam loads.

It is recommended, therefore, that stress (strain) recordings presently in hand for other vessels in other trades be reviewed and analyzed as to the frequency of slamming and the magnitude of slam stresses, and that consideration be given to the gathering of similar data on vessels of more recent design to be reviewed on the same basis. Such data can be used for the time being as a guide in estimating dynamic loads and stresses for new ship designs.

Another aspect of the slam load problem to be considered, if one is to determine the combined effect of slamming and quasi-static wave bending on the hull girder, is the phasing of the slamming loads. The particular instant in the wave bending cycle at which a slam occurs determines the extent to which the slam stress-- and the subsequent whipping stress -- add to the total stress magnitude. A study of this subject has been made, on the basis of Wolverine State stress records, and is reported in Chapter VI. This investigation indicates that the phasing -- as well as the magnitude -- of slam loading can be established empirically for different ship types as a basis for design. However, it also shows that the stresses at impact seldom if ever add significantly to the total bending moment. Of much greater importance is the whipping that follows a slam.

For design purposes, the practical problem of determining the transient dynamic loads reduces to two parts:

(a) The prediction of the probability that slamming will occur (over a period such as a 20-minute record).

(b) The distribution of added stress (or equivalent bending moment) to be expected when slamming and whipping occur.

OBSERVED LOADS AND STRESSES

Available published data on actual slamming pressures recorded on ships at sea and on measured midship slamming stresses are scarce. However, a summary of available information on maximum recorded values is given in the accompanying Table I, compiled from the indicated references.

Reference should also be made to a general survey of slamming by Henry and Bailey (104).

SPRINGING

As mentioned in the Introduction, one type of steady-state dynamic effect is known as "springing." This phenomenon has been noticed particularly in Great Lakes bulk carriers (114), but it has also been reported on ocean-going ships (115)(113). A clue to the 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 (2-noded periods of 2 secs. or longer). The apparent explanation is that when the ship is running into comparatively short waves that give resonance with the natural period of vibration, significant vibration is produced. This vibratory respohse may continue over some period of time, gradually fluctuating in magnitude. A corresponding fluctuation in stress amidships is therefore

Table I

MEASURED FULL-SCALE

SLAM LOADS AND MIDSHIP STRESSES

	Pressure p.s.i.	Peak-to-trough Midship Slam Stress, kpsi
USCG Cutter	295 local	-
Unimak (105)	86 ave. panel	-
Dutch Destroyer (106)(107)	100+	11.20
Cargo Ship S.S. Westboro (108)	-	1.90
Cargo Ship S.S. Wolverine State Voy. 288	49	2.60
(109) Voy. 277	69	-
(103)		10.00
Bulk carrier Fotini L (110)	-	15.50
Container ship Manchester City (111)	-	5.00
Cargo ship M.V. Jordaens (112)	-	4.50
	19.	2.04
Container ship Flinders Bay (113)	-	15.60

superimposed on the quasi-static wave bending moment. The springing stress appears to have the characteristics of a stochastic process, but one that is essentially independent of the low-frequency wave bending, which -- as previously noted -- is also treated as a stochastic process.

Figure 10 shows a record of midship stress in which both low and high-frequency stresses are present. Also shown are two records in which first the high-frequency stresses have been filtered out and second in which the low-frequency stresses have been filtered out.

The well-developed strip theory of ship motions has been applied to the springing problem (115). Although motions of a springing ship may 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, results for one ocean-going ship were found to agree quite well with full-scale records (115). Further coordination between theory and experiment has been attempted for Great Lakes ships, including model tests where idealized wave conditions can be provided (116). Subsequent tests with a jointed model of the Great Lakes carrier Stewart J. Cort using connecting beams of varying stiffness, show clearly that the springing bending moment increases with hull flexibility.

When the above calculation procedures have been tested and revised as necessary, a tool will be available for predicting springing stresses in a new ship design. The remaining problem for incorporating springing into the comprehensive ship hull design criteria is to determine the manner in which springing stresses superimpose on the low-frequency bending. This problem has also been solved in principle (117) on the basis of the assumption that the two stochastic processes are independent. This assumption is illustrated by the records in Fig. 10 which show that high amplitudes of the two types of stress do not occur at the same time.

Further work needs to be done as new information becomes available. Meanwhile, the general cargo ship -- which is the principal subject of the present study -- does not seem to be subject to significant springing effects, and therefore they need not be included in the present criteria.

VI. PHASING OF SLAM AND WAVE LOADS

INTRODUCTION

It has been the goal of this project to obtain complete information on all aspects of ship hull loading, particularly loads generated by waves at sea. The first type of load, quasi-static wave-bending moments, has, of course, been focused upon first, and the current state of knowledge is summarized in Chapter IV. Next must be considered the influence of slamming, and whipping after the slam, which are discussed in Chapter V. Finally, in order to determine the combined effect of wave bending and slamming, particularly in reference to brittle fracture, it is necessary to understand the way in which the latter high-frequency load is superimposed on the former. If the slam should occur, for example, when the wave bending moment is near zero, the immediate effect of the slam will not be an increase in the total amplitude of bending moment amidships unless it exceeds the wave bending moment amplitude. On the other hand, if the slam should occur exactly at a peak bending moment it would increase the total by whatever its amplitude might be. Similarly the amount by which the vibratory whipping stress (or equivalent bending moment) following the slam would add onto the subsequent peak of the wave bending moment curve would depend on how rapidly the vibration is damped out.

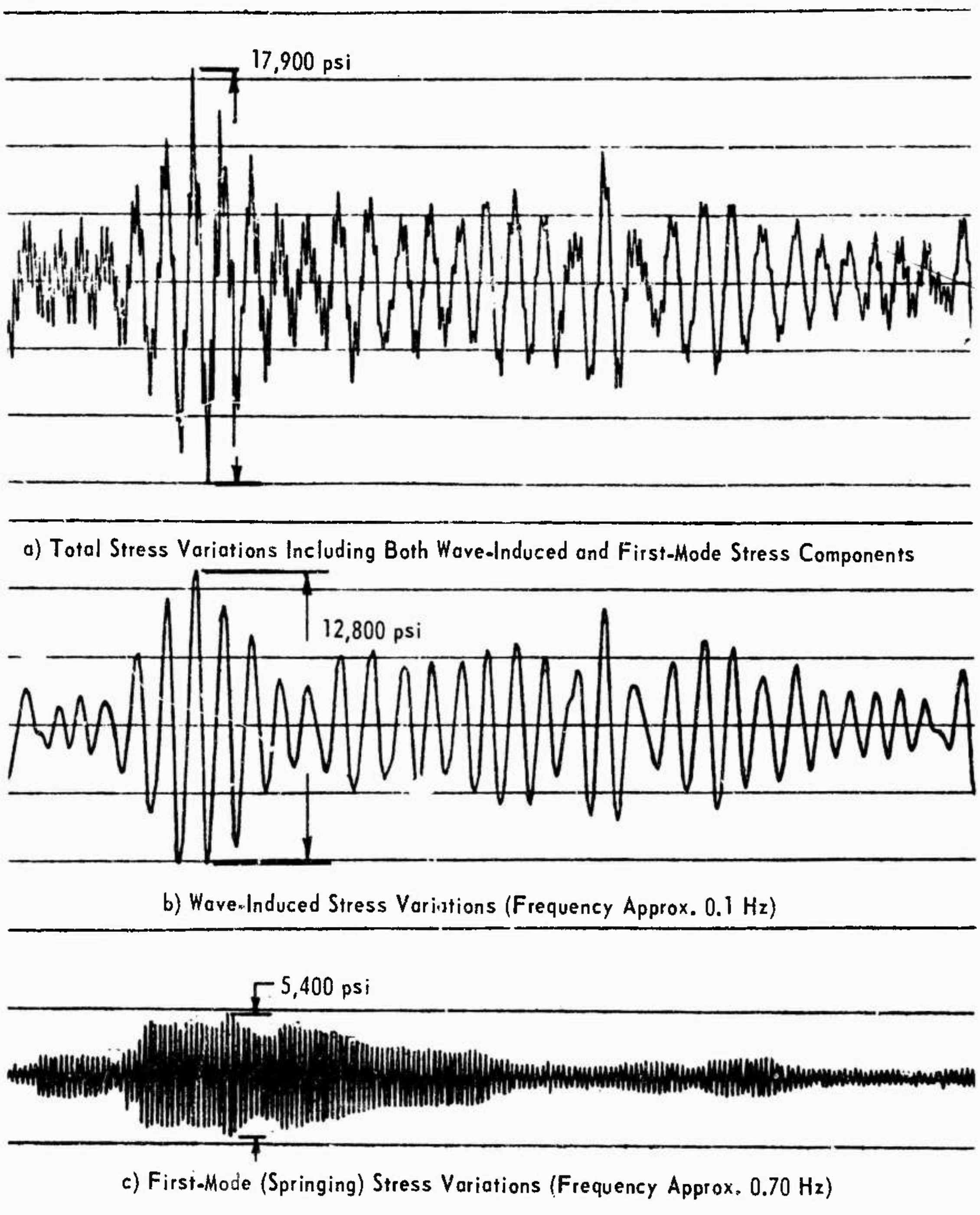


FIGURE 10 - Typical Record of Midship Stress Variation, M.V. *FOTINI L*, Showing Filtered Wave-Induced and Dynamic Stresses (3).

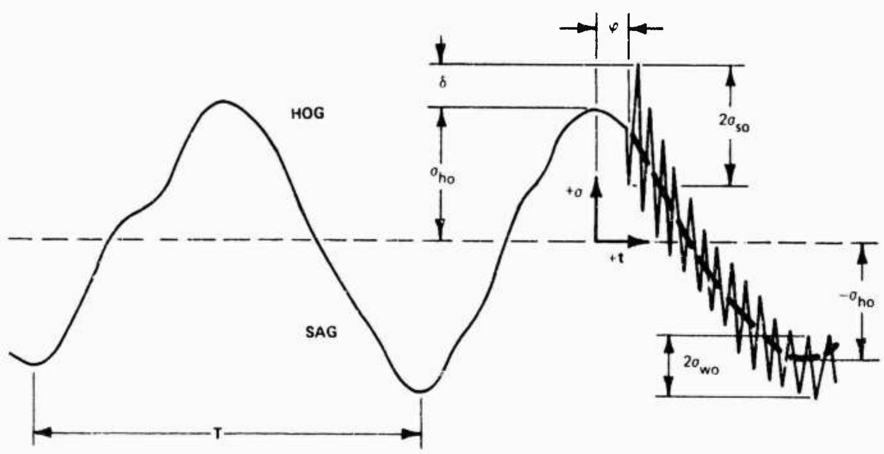
Nothing could be found on this subject in the literature. Hence, it was decided to make a study of the phasing of slam and wave loads for the Wolverine State, since it was possible to obtain stress records of slamming from Teledyne Materials Research, and a reasonably clear picture emerged for this particular ship.

It is realized that is impossible to generalize this result for one ship to apply to all ships, and hence it is believed that similar studies should be made on ships of other types and sizes.

DEFINITIONS

A slam is defined as a transient impact force acting on a ship. We can therefore call the hull girder stress generated by this transient force the slam stress, σ_{so}, by which we mean the first peak (sag) or the second peak (hog) in the slam stress time record. (See Fig. 11). The two-noded hull girder vibration generated by the slam is called whipping. We call the stress generated by this vibration the whipping stress, σ_{wo} . It should be noted that bending moments equivalent to these stresses can be determined for combining with other loads. Wave bending moment is defined as the low-frequency hull girder moment which is experienced at frequencies corresponding to the ship's encounter with the waves in the seaway and produces the wave stress, σ_{ho}.

It should be pointed out here that in this study we are particularly interested



σ_{ho} = WAVE STRESS
σ_{so} = SLAM STRESS
σ_{wo} = WHIPPING STRESS
δ = ADDITIVE PORTION OF σ_{so}
φ = PHASE ANGLE

FIGURE 11 - Definitions of Stresses and Phase Angles Involved in Slamming

in: (a) the second peak of the slam stress time trace -- generally occurring during a hogging wave stress condition -- and (b) the whipping stresses in the subsequent sagging and hogging wave stress conditions. See Fig. 11.

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 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

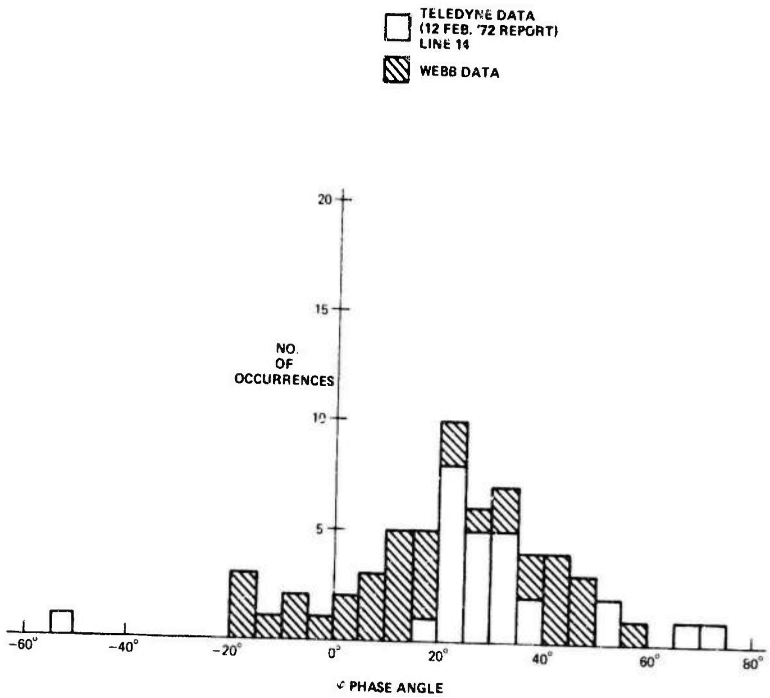


FIGURE 12 - Distribution of Slam Phase Angles, *S. S. WOLVERINE STATE*

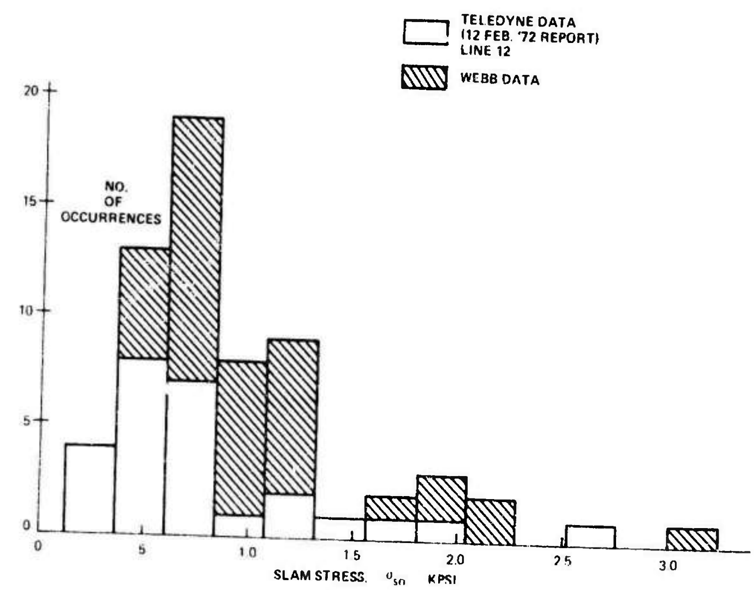


FIGURE 13 - Distribution of Slam Stress, *S. S. WOLVERINE STATE*

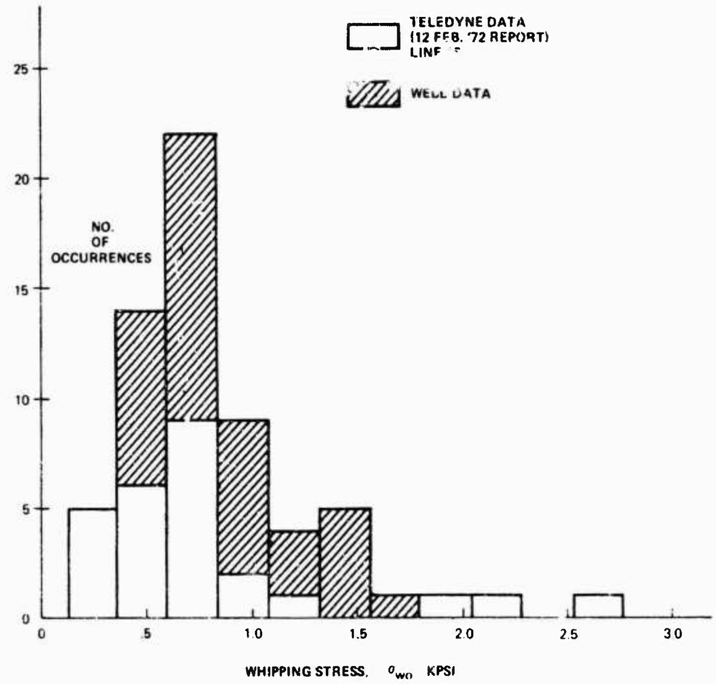


FIGURE 14 - Distribution of Whipping Stress,
S. S. WOLVERINE STATE

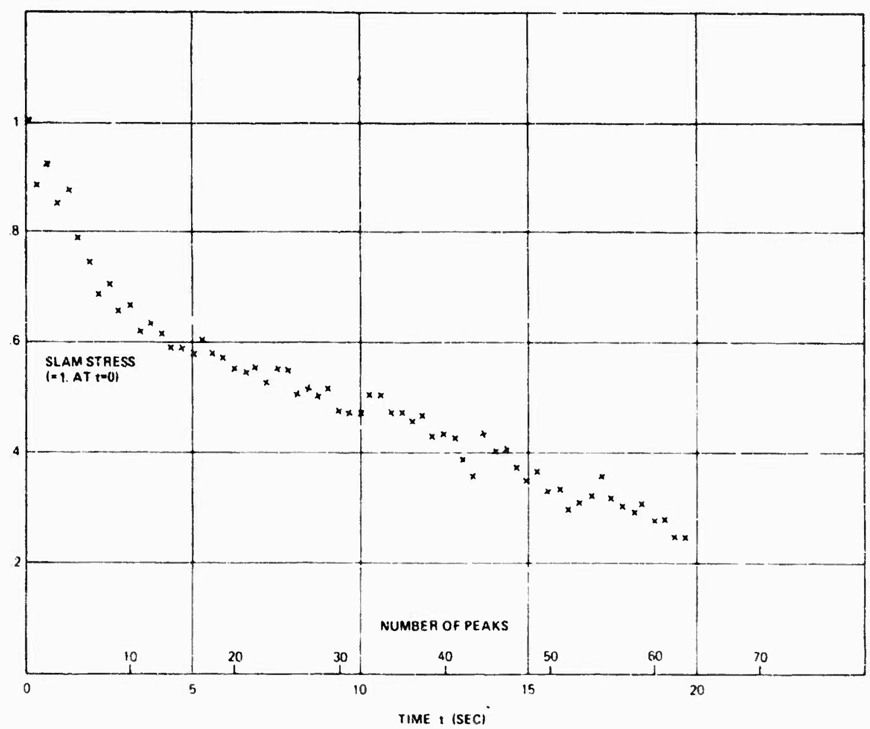


FIGURE 15 - A Typical Decay Curve of Whipping Stress

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 $\overline{\sigma}_w = 0.876$ 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, $\overline{\sigma}_s = 0.13$ 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):

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

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

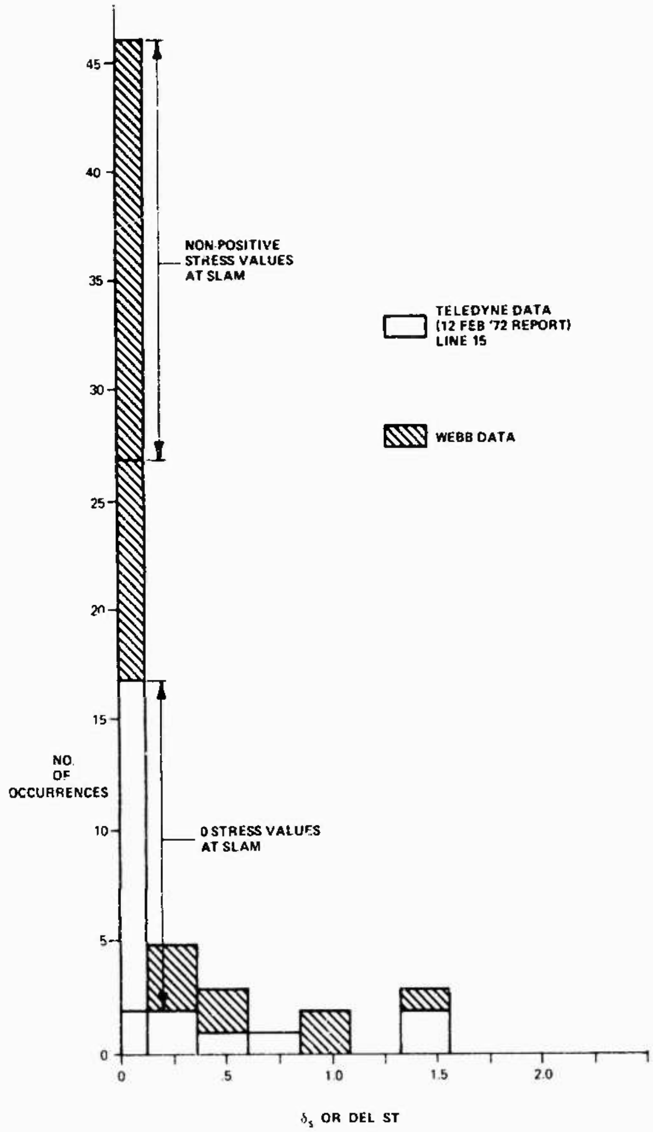


FIGURE 16 - Histogram of Slam Stress Additive to Wave Stress

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.

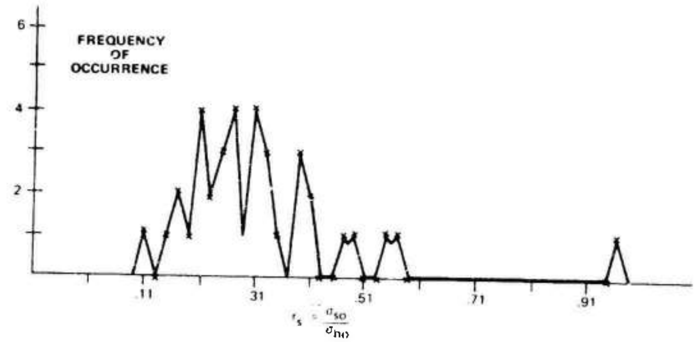


FIGURE 17 - Histogram of the Ratio of Slam Stress to Wave Bending Stress

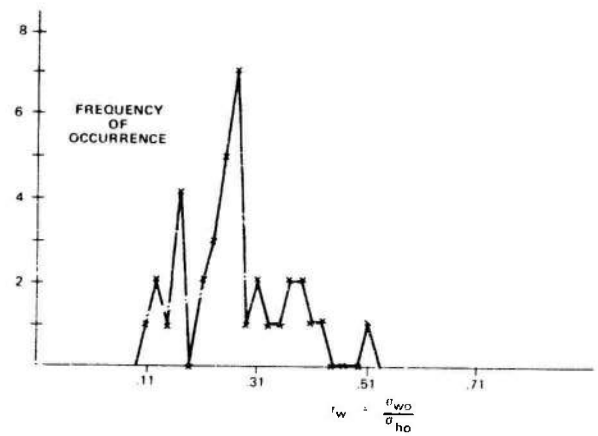


FIGURE 18 - Histogram of the Ratio of Whipping Stress to Wave Bending Stress

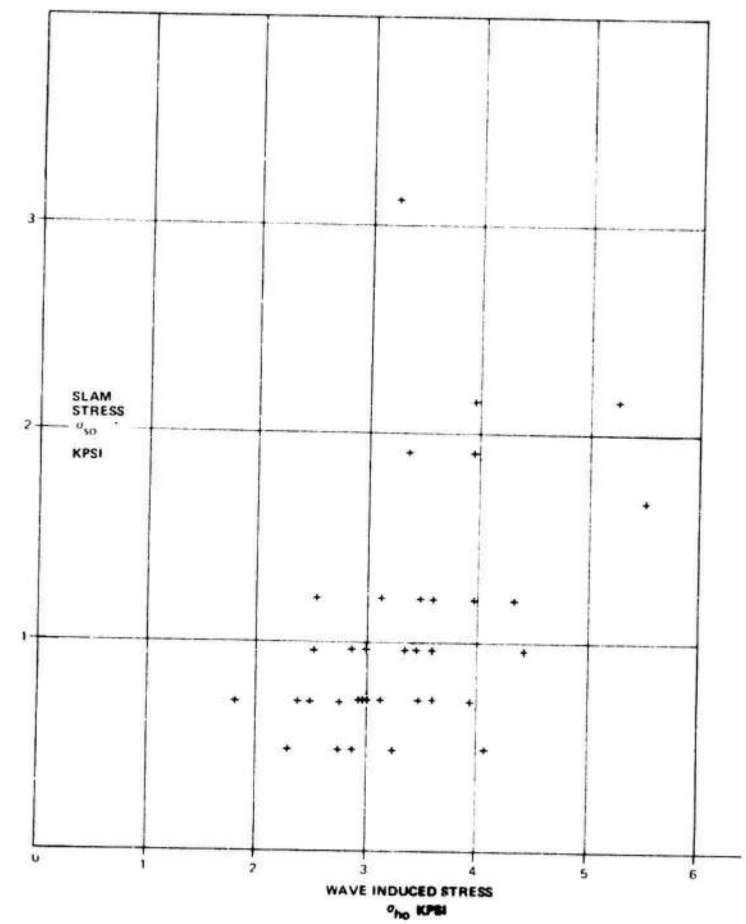


FIGURE 19 - Plot of Slam Stress vs. Wave Bending Stress

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 ΔT = 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 when slamming occurs 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 Wolverine State 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 generalalize 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.

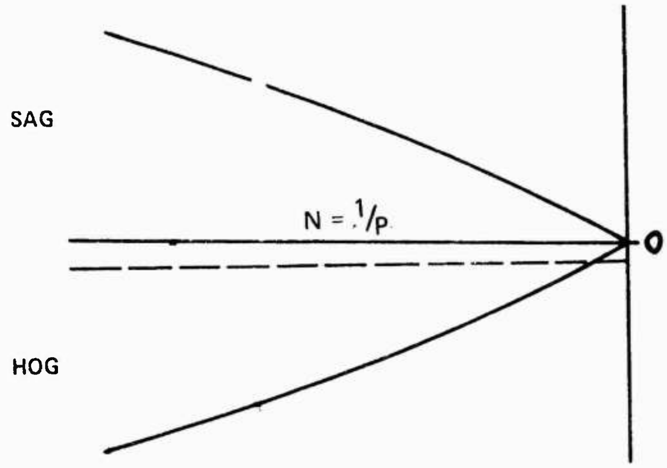


FIGURE 20 - Typical Long-Term Distributions of Wave Bending Moment for Sag and Hog

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 into two classifications -- those that occur at right, 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.

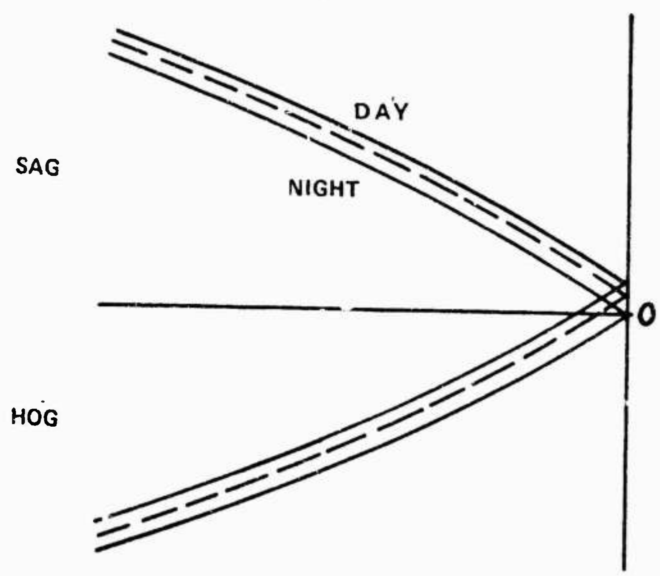


FIGURE 21 - Typical Long-Term Distribution of Wave Bending Moment, Sag and Hog, with Thermal Stress Superimposed

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 0.00[illegible] 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	362	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 previous 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 damage might be of different degrees 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 cracks 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-

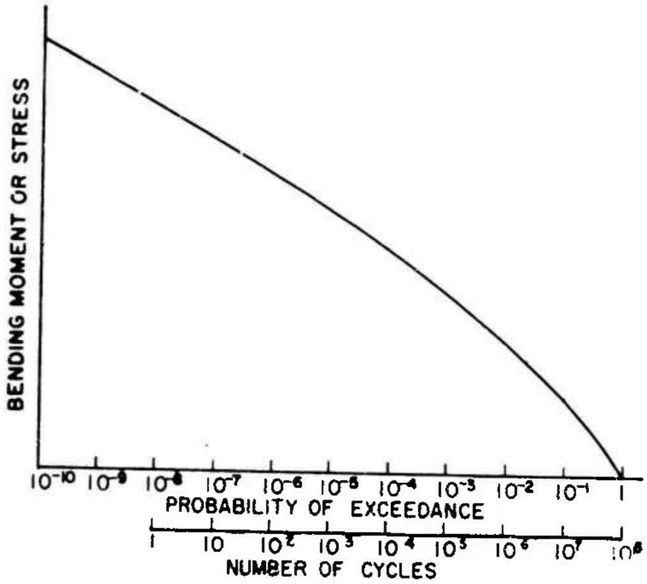


FIGURE 22 - Long-Term Distribution of Bending Moment or Stress, with Reversed Scale Showing Cyclic Loading or Number of Cycles of Each Stress Level in One Ship Lifetime (10^8 Cycles)

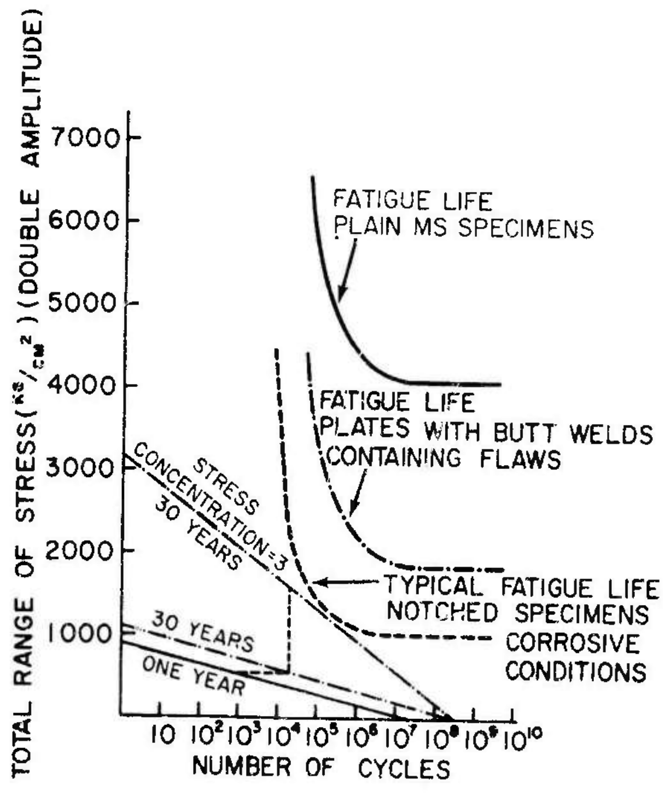


FIGURE 23 - Example of Application of Cyclic Loading Curves to Study of Fatigue (128)

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.

It was decided that the most suitable ship for detailed study was the S.S. Wolverine State. Not only are service wave bending stress data available, but information could be obtained on dynamic stresses and on still water loadings.

The load calculations followed the procedures described in the preceding chapters. Considering first the loads affecting ultimate bending failure, each step will be described in turn. Then the combined effect of all loads will be considered and interpreted in terms of a simple design bending moment for hogging and sagging conditions, outbound and inbound. This load criterion will then be compared with conventional design standards.

Finally, brief consideration will be given to the cyclic loading pattern as a criterion relative to possible fatigue damage, and to a criterion for brittle fracture. For this ship, with relatively heavy shell plating, small hatches, and low length/depth ratio it was felt that criteria of shear, torsion or elastic deflection need not be considered.

STILL WATER LOADS

Estimated Bending Moments

During most of the time that data were collected on the S.S. Wolverine State the ship was engaged in North Atlantic service. Available data on service drafts indicated that the ship usually operated at drafts considerably less than full load, both east and westbound. However, in order to provide a typical numerical example, it was felt to be desirable to assume a fully loaded cargo condition on the outbound voyage and typical light loadings inbound.

Unfortunately, it was impossible to obtain detailed distributions of cargo and liquids for a sufficient number of actual voyages to obtain a statistical picture of still water bending moments. Hence, calculations were made, using the owners' loading manual as a guide, to supplement actual loading data. A normal distribution of still water bending moment in the light cargo condition was constructed as shown in Fig. 24, based on the available loading data and on calculated highest and lowest expected values in a 25-year lifetime. To determine the extremes,

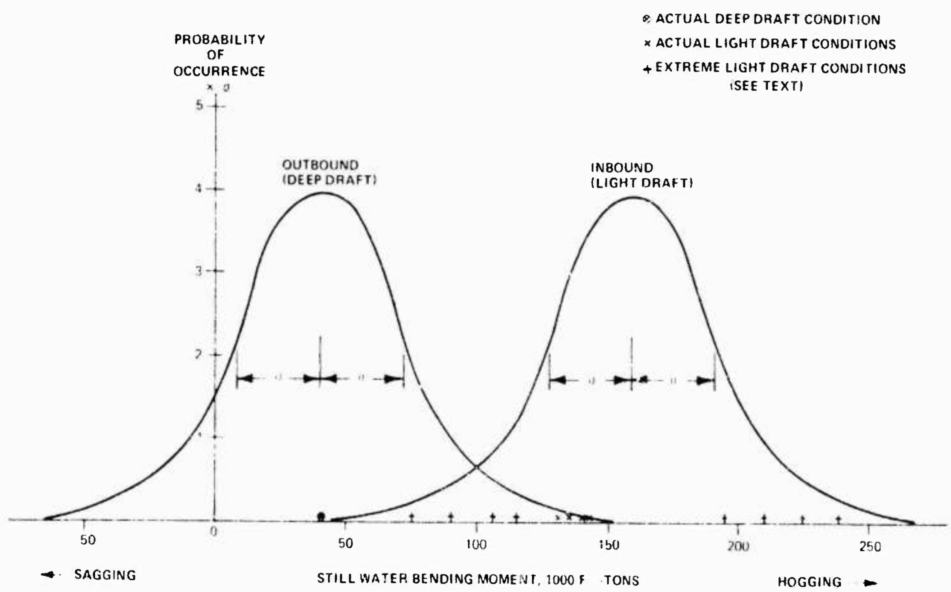


FIGURE 24 - Estimated Distributions of Still-Water Bending Moments, *S.S. WOLVERINE STATE*

the amount of cargo typically carried in the light load condition was imagined to be distributed more and more toward the ends of the ship until an estimated extreme condition was reached. We assigned to the resulting still water hogging bending moment (239,900 ton-ft.) a probability representing one occurrence in 25 years.

For 12 round trips per year, there are 300 crossings in each direction in 25 years. Thus the highest expected single occurrence would have a 1/300 = 0.00333 frequency, which corresponds to $\pm 2.71\ \sigma$, where σ is the standard deviation. A similar approach was used to determine the minimum hogging condition in 25 years (75,100 ton-ft.), which corresponds to $2.71\ \sigma$ below the mean. The mean value for the normal distribution is the average of the highest and lowest calculated values. Thus the mean and standard deviation, and from them the normal distribution, were determined as shown in Fig. 24, along with actual values determined from the loading data.

The mean still water bending moment for the full load condition was calculated from one actual voyage at nearly full load, and it was assumed that the standard deviation would be the same as for the light cargo condition. The results of the calculations are given in Table V. All bending moments are in ton-ft.

TABLE V

STILL-WATER BENDING MOMENTS
S.S. Wolverine State

	Bending Moments, Ton-Ft.	
	Full Load Outbound	Light Load Inbound
Mean	40,000 hog	157,500 hog
Standard deviation	30,400* hog	30,400 hog
10% are below	800 hog	118,300 hog
10% are above	79,200 hog	196,700 hog
Highest in 25 years	122,400 hog	239,900 hog
Lowest in 25 years	42,400 sag	75,100 hog

*Assumed the same as in Light Load Condition.

The above maximum values have been inserted in Table IX, along with extrapolated 1000-ship values and other results to be discussed below.

Classification Society Requirements

It should be noted that although classification societies do not in general base hull girder strength standards on a direct addition of still water and wave bending moments, still water moments are taken into account at the present time. (See Chapter III).

For the Wolverine State the maximum still water bending moment without penalty is calculated by ABS 1972 Rules to be 136,000 ton-ft. It may be seen that this value would be exceeded by the calculated value of 239,900 ton-ft in the hogging condition, inbound. If these conditions were included in the data submitted to the ABS at the time of the design of a similar ship today, some addition to the midship section modulus requirement might be made. At the time the ship was built (1945), however, there was no explicit still water section modulus requirement in the ABS rules.

FORWARD SPEED EFFECT

The bending moment (sagging) induced by the ship's own wave at forward speed in calm water was estimated on the basis of model test data on similar ships (36). At the design speed of 16 knots the result is a sagging bending moment of 2700 foot-tons in the heavy (outbound) condition and 5200 foot-tons in the light (inbound) condition. Thus the forward speed effect reduces the still water bending moments, which are hogging moments in both conditions. The model tests show that the maximum hogging moment due to forward speed occurs when the ship is at very low speed, and the reduction in still water bending moment at normal speed becomes zero.

The above figures have been entered in Table IX, in which a summary of all bending moments is given. Average values have been estimated for the table on the basis of service speed data.

THERMAL EFFECTS

Next an estimate was made of the thermal stresses to be expected under different symmetrical conditions of sea/air temperatures (sun overhead). Since calculations have shown that symmetrical heating of the deck results in compressive stresses in deck and bottom plating of nearly equal magnitude (119), only the changes in deck stress were computed.

The basis for calculating thermal stresses is described in Chapter VII. Calculations were made for five values of diurnal temperature change, representative of typical North Atlantic conditions. Average air temperature changes were determined from ships' logs covering a two-year period. The results were:

Season	Avg. Diurnal ΔT of Air
Winter	10°
Spring	9°
Summer	10°
Fall	7°

Because the seasonal variations were small, differences were ignored and a yearly average of 10° was assumed.

An additional ΔT of 40° was assumed for insolation with full sun, since the deck color is dark gray, as in the Esso Malaysia estimate (see Chapter VII). This figure represents full sun conditions; so adjustments were made for cloudy conditions. Available seasonal and annual cloud cover data for the North Atlantic route (82) are given in Table VI.

TABLE VI

DATA ON CLOUD COVER AND ESTIMATED TEMPERATURE DIFFERENTIALS, NORTH ATLANTIC

Season	Median Factor for Cloud Cover	% of time < 2/8 cover	% of time > 5/8 cover
Winter	6/8	10%	70%
Spring	6/8	20	60
Summer	5/8	20	60
Fall	6/8	20	60
Yearly avg.	6/8	20	60

Factor for Cloud cover	Diurnal ΔT			Estimated Frequency of Occurrence	
	Insolation	Air	Total		
0/8	40°	10°	50°	10%	20%
2/8	30°	10°	40°	10%	
4/8	20°	10°	30°	20%	
6/8	10°	10°	20°	30°	60%
8/8	0°	10°	10°	30°	

Again the seasonal differences were ignored in this table, and the yearly average figures for cloud cover were used. In order to relate the insolation temperature to the cloud cover information, we assumed that the insolation ΔT is directly proportional to the amount of cloud cover, while the air temperature change (10°) remained constant. The resulting diurnal ΔT's for different cloud conditions are also given in Table VI.

The calculated compressive stresses at deck edge for the Wolverine State are given in Table VII, and the calculated stress distribution is shown in Figure 25.

TABLE VII

CALCULATED CHANGES IN STRESS AND EQUIVALENT BENDING MOMENT, S.S. WOLVERINE STATE IN NORTH ATLANTIC SERVICE

Diurnal ΔT	Frequency of Occurrence, Table VI	Deck Compressive Stress, PSI	Corresponding Equiv. Sagging B.M., ton-ft.
50°	10%	2170	47,700
40°	10%	1740	38,250
30°	20%	1300	28,600
20°	30%	870	19,100
10°	30%	430	9,500
Weighted average		1040	22,900

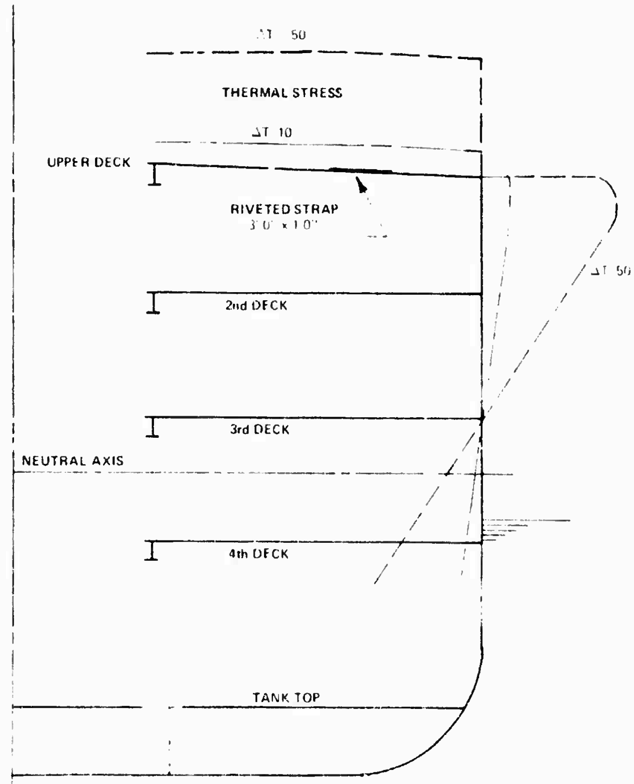


FIGURE 25 - Calculated Thermal Stresses,
S.S. WOLVERINE STATE

Next an "equivalent" sagging bending moment that would produce the same deck stress was computed for each value of diurnal ΔT, as listed in Table VII. The weighted average was then inserted in Table IX as the average sagging value. On the assumption that the bottom compressive stress in hogging would be approximately the same as the calculated deck stress in sagging, an effective hogging bending moment of 22. [illegible] ton-ft. was calculated and entered in the table.

WAVE LOADS

Comprehensive computer calculations were carried out for the S.S. Wolverine State for two conditions:

(1) Full load.
(2) Light load.

The responses covered by these calculations included:

Vertical longitudinal bending moment.
Lateral " " "
Combined vertical and lateral longitudinal bending.
Torsional moments.

The calculations proceeded in the following steps, in accordance with the procedures described in Chapter IV:

Calculation of response amplitude operators at all headings to regular waves.

Comparison of above with model tests.

Prediction of response to irregular short-crested seas described by the H-family.

Estimating long-term cumulative distribution for North Atlantic service, based on the weather statistics given in Table VIII.

TABLE VIII

WEATHER DATA FOR NORTH ATLANTIC

Probability p	Average Significant Height H-$1/_3$ (ft.)	Range of H-$1/_3$ (ft.)
0.8454	10.0	5 - 15
0.1330	20.0	15 - 25
0.0201	30.0	25 - 35
0.0014	40.0	35 - 45
0.0001	48.2	45 and above

The effective vertical bending moment giving the combined effect of vertical and lateral bending was calculated by the method described in Chapter IV. The ratio Z_V/Z_L for the Wolverine State is 0.8446.

The program SCORES (49) was modified to suit the present purpose by appending a memory block to carry over the necessary information, so that it would still be possible to run SCORES in its original form for other purposes. The calculations necessary to give the long-term distribution were appended in the form of a sub-program.

Long-term calculated results are presented in Figs. 26 and 27, along with curves obtained from full-scale statistics for comparison. See also (10). Calculated bending moment results in regular waves are compared with model test results in (53). Values from Figs. 26 and 27 for combined vertical and lateral bending moments have been entered in Table IX.

DYNAMIC LOADS

We come now to the consideration of dynamic loads on the S.S. Wolverine State. For a ship of this type, springing would not be expected, nor is there sufficient flare to produce significant bow flare immersion effects. Stress records confirm that neither of the above effects were experienced significantly in actual service. Furthermore, a student thesis project at Webb with a jointed model of Wolverine State confirmed that only negligible springing stresses could be developed in the model tank.

However, slamming stresses resulting from bottom impact forward, followed by vibratory whipping response, were both expected and found in Teledyne records taken in the light load condition (103)(109).

As explained in a previous chapter, it is theoretically possible to predict the occurrence of slamming if one can specify the level of relative vertical velocity at

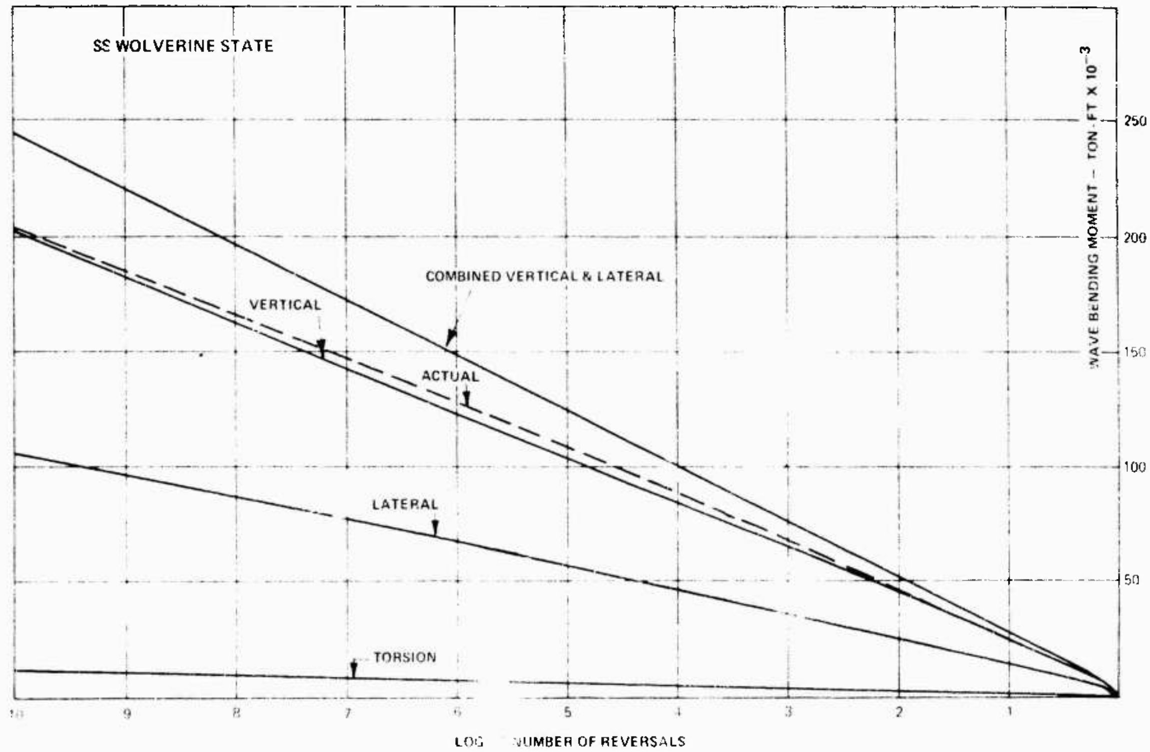


FIGURE 26 - Long-Term Distribution of Bending Moment, Light-Load Condition

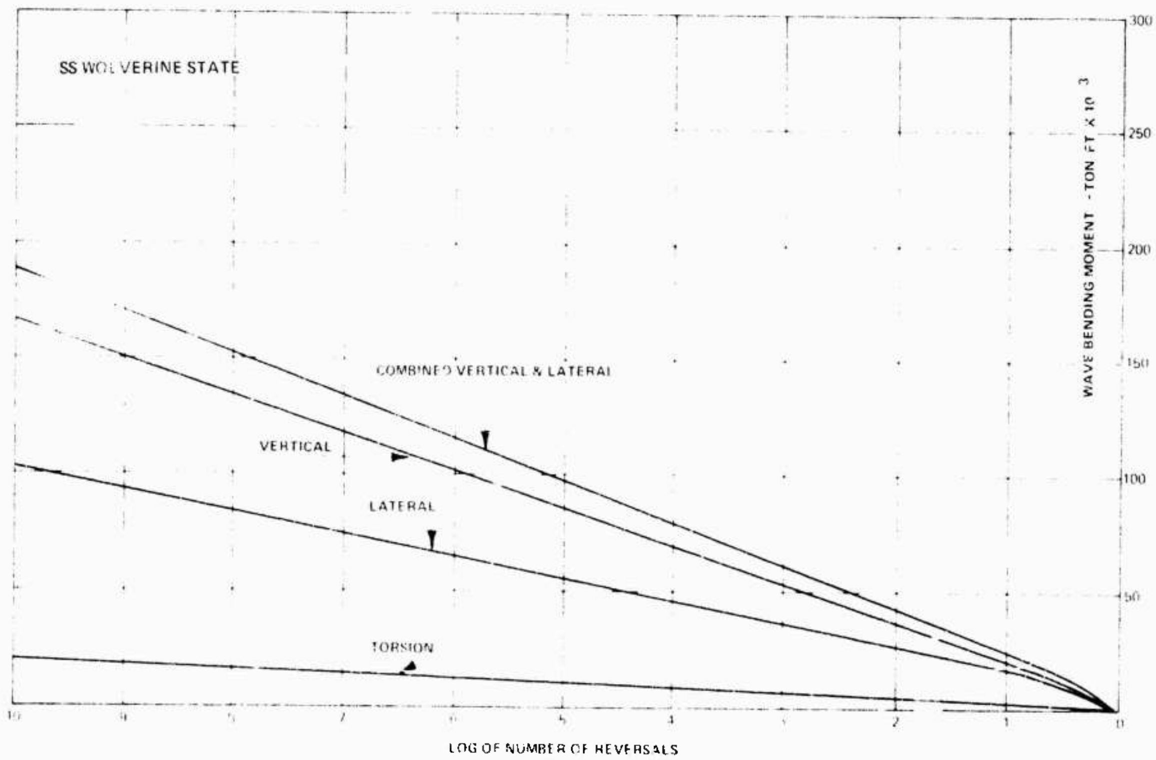


FIGURE 27 - Long-Term Distribution of Bending Moment, Full-Load Condition

which it will occur for the ship in question. Since there are still a number of doubtful questions about this procedure, reference was made in this case to the recorded data on the S.S. Wolverine State. The stress records showed that, over the entire period of data collection, slamming (with stresses exceeding 1.0 kpsi) occurred in 16% of the records. This can be considered to be an indication that in normal operation the probability that slamming will occur in any 20-minute period is 0.16 in the usual light load condition at which the ship operated. (Average draft forward was 16-20 ft.)

In the hypothetical full load condition (forward draft 29 ft.) the probability of slamming is greatly reduced, and for the present purpose was arbitrarily assumed to be 0.

Referring to Chapter VI, the following conclusions may be restated here:

1. the slam stress distribution σ_s is exponential:

$$p(\sigma_s) = a \exp\{-a(\sigma_s - c)\}$$

where $\sigma_s = \sigma_{so}/\mathrm{RMS}$

$c = 0.36$

$a = 1$

2. the distribution of the phase angle ϕ is normal

$$p(\phi) = \frac{1}{\sigma\sqrt{2\pi}} \exp\left\{-\frac{1}{2}\left(\frac{\phi - \mu}{\sigma}\right)^2\right\}$$

where $\mu = 0.406$ radians,

$\sigma = 0.381$ radians.

The theoretical solution to the problem of calculating the distribution of the increase of the slam stress δ_s and the whipping stress δ_w over wave bending stress can be given briefly as follows:

$$p(\delta_s) = \frac{a}{2\sigma^2} \exp\left\{-\frac{a\delta_s}{2\sigma^2}\right\} A(a,\sigma,\mu)$$

where δ_s is the increase due to slamming, and the first two factors express the exponential distribution of additive δ_s, while the third factor A, deals with the truncation due to non-additive δ_s.

The expression obtained for whipping stresses leads to a Rayleigh distribution:

$$p(\delta_w) = 2b\,\delta_w \exp(-b\,\delta_w^2)$$

where δ_w is the whipping stress and b is a function of (a, μ, σ,c). All of the above relationships can also be interpreted in terms of the corresponding effective bending moments.

On the basis of these distributions, using available slam stress data for the Wolverine State, as previously discussed, we arrive at a maximum expected increase due to slamming over the lifetime of the Wolverine State of 6.81 KPSI, corresponding to an effective increase in the sagging bending moment of 133,000 ton-ft.

Similarly the maximum expected w ipping stress was calculated at 6.95 KPSI, corresponding to an eff ctive increase of hogging bending moment of 136,000 ton-ft. These figures have been entered in Table IX, along with figures calculated for 1000 ship lifetimes.

COMBINED LOADS

The results of all the load calculations for the Wolverine S te can be tabulated in the manner shown in the accompanying Table IX. Results are given on the basis of both of the following long-term assumptions (where N is the number of wave bending cycles):

(a) One ship's lifetime: $N = 10^8$ for 25 years; approximately 300 round voyages in North Atlantic service.

(b) The combined lifetimes of 1000 ships ($N = 10^{11}$), i.e., bending moment expected to be exceeded with a probability of 0.001 in a ship's lifetime.

The latter has been suggested as a basis for a rational design criterion with respect to possible ultimate failure of the hull girder.

The dynamic loads associated with slamming have been entered only for the light load condition, under the assumption that slamming seldom if ever occurs when fully loaded.

If all the maximum stresses in Table IX are added directly, an unrealistically high value is obtained, because all of the maxima will probably not occur at the same time. Hence, a reduced total has been calculated, based on combining still water and wave bending by the method of Chapter VIII. The results of this calculation are shown in Fig. 28, using the normal distributions of still water bending moment previously given and the combined vertical and lateral wave bending moment data on which Figs. 26 and 27 are based. The reduced totals in Table X were then obtained by adding the forward speed effect (if any) and one-half the average thermal effect to the values read from the curves in Fig. 28. The dynamic effects of slamming or whipping have been added directly (light load condition), under the assumption that extreme wave bending and slamming are likely to occur at the same time. This assumption is obviously not correct, but it probably does not result in a large error because both effects require rough sea conditions. The expectation that at the same time that slamming gives an increased dynamic load (demand) on the structure the capability also increases is a favorable effect of unknown magnitude.

STRUCTURAL CAPABILITY

Having estimated the probable lifetime combined loading for the Wolverine State, separated into maximum hog and maximum sag, outbound (full load) and inbound (light load), it is of interest to see what these results signify in relation to conventiona design standards. This requires that we consider a "rational" evaluation of capability as well as of demand.

First, it should be noted that if the hull of the S.S. Wolverine State behaved like an ideal girder and tensile considerations governed, the bending moment

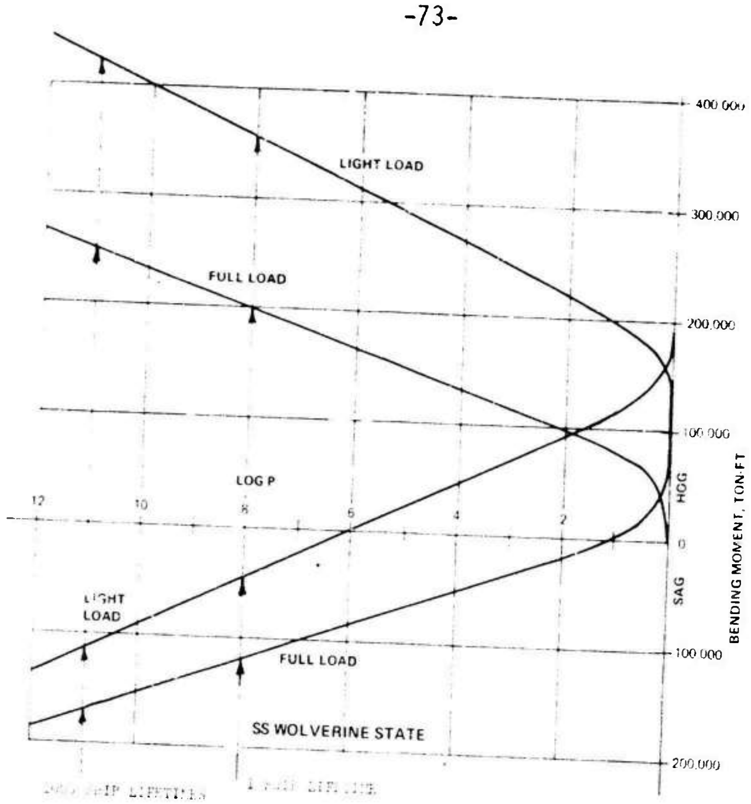


FIGURE 28 - Long-Term Distributions of Combined Bending Moments: Wave Bending (Vertical and Lateral) and Still-Water Bending

TABLE IX

SUMMARY OF LIFETIME HULL LOADS
S.S. WOLVERINE STATE

Showing individual maximum values

	Average		Max. Lifetime		Max. 1000 Ship Lifetimes	
	Hog	Sag	Hog	Sag	Hog	Sag
Full Load - Outbound						
Still water	40,000	-	122,400	42,400	176,800	96,800
Forward speed effect	-	1,000	0	2,700	0	2,700
Thermal effect*	22,300	22,900	46,400	47,700	46,400	47,700
Wave-induced	15,000	15,000	153,000	153,000	207,000	207,000
Dynamic	0	0	0	0	0	0
Light Load - Inbound						
Still water	157,500	-	239,900	-75,100	294,300	-20,700
Forward speed effect	-	2,200	0	5,200	0	5,200
Thermal effect*	22,300	22,900	46,400	47,700	46,400	47,700
Wave-induced			196,000	196,000	268,000	268,000
Dynamic { Slamming	-	63,650	-	133,000	-	170,000
Dynamic { Whipping	31,300	-	136,000	-	168,000	-

*On compression flange only

TABLE X

COMBINED LIFETIME HULL LOADS
S.S. WOLVERINE STATE

Showing probable combined maximums	Bending Moments in Ton-Ft.			
	Max. Lifetime		Max. 1000 Ship Lifetimes	
	Hog	Sag	Hog	Sag
Full Load - Outbound				
Still water }# Wave }	200,000	118,000	252,000	170,000
Forward speed effect, average	0	1,000	0	1,000
Thermal effect, average	11,150	11,450	11,150	11,450
Dynamic	0	0	0	0
TOTALS	211,150	130,450	263,150	182,450
Light Load - Inbound				
Still water }# Wave }	360,000	43,000	423,000	112,000
Forward speed effect, average	0	2,200	0	2,200
Thermal effect, average	11,150	11,450	11,150	11,450
Dynamic { Slamming	-	133,000	-	170,000
Dynamic { Whipping	136,000	-	168,000	
TOTALS	507,150	189,650	602,150	293,650

\# Combined probability.

capability at yield would be simply the product of section modulus by the yield stress of mild steel, 36 kpsi (16 tons/in^2):

Tensile yielding in deck (hogging):
Bending moment = 45,800 x 16 = 732,000 ton-ft.

Tensile yielding in bottom (sagging):
Bending moment = 47,200 x 16 = 753,000 ton-ft.

Of course, the tension flanges would be able to sustain considerably higher loads in conjunction with extensive plastic yielding.

However, it is the compression loading that usually governs the capability of a beam to sustain applied bending moments. An estimate has been made of the hull girder longitudinal bending moment capability of the S.S. Wolverine State, considering the compressive loading on this transversely framed vessel. Both hogging and sagging conditions have been considered and each will be discussed in turn.

It was assumed that the capability of a general cargo vessel's structure will be tested at sea with loading of cargo distributed more or less throughout its spaces. Thus, the 'tween deck plating would be constrained to buckle in a clamped-clamped mode by the cargo. Similarly, the lateral hydrostatic loading on bottom plating will strongly influence growth of plate deformation under in-plane loading. Weather (upper) deck plating, however, would normally not be subject to these constraints.

On the basis of the above, it was found that the plating of the 'tween decks would buckle at about two thirds of yield stress, while the heavier main deck plating would sustain in-plane loading to almost the yield stress. Schultz (129) found that due to plate unfairness the effectiveness of deck plating was about 83% and the unfairness patterns simply grew as loads increased. It was assumed that the capability could be assessed on this basis, no better information being available as to plate effectiveness in the unfair state.

The main deck plating has a nominal buckling stress in excess of 34 ksi, and because of the additional strap installed, it can be expected to sustain nearly yield stresses prior to buckling.

On the basis of the above, it was decided that a fair assessment could be made of moment capability by assuming 83% effective 'tween deck (sagging) or bottom plating (hogging) under compressive post-yield conditions.

Hogging Moment

It was assumed that decks and side plating under tension were fully effective, while bottom plating under compression was 83% effective. On such a basis the neutral axis of the effective structure was computed. Assuming next that the effective material was at compressive or tensile yield stress, the resisting moment of the effective material was computed about the neutral axis of the effective structure.

By this process it was found that the neutral axis would be at a position 24 feet above the base line and that the internal resisting moment of the section would be 753,000 ton-feet in hogging.

Sagging Moment

The buckling stress of the main deck plating was found to be

$$\sigma_c = \frac{k_c \, \pi^2 \, E}{12(1 - \mu^2)} \left(\frac{h}{b}\right)^2 = 34.5 \text{ ksi}$$

where $E = 30 \times 10^6$, thickness $h = 1.06$ in., $b = 30$ in. and Poisson's ratio $\mu = 0.3$. Under such a condition the second deck would have a stress well below buckling conditions and significant structural deformation would therefore have to take place before increased load capability could be generated. Hence, it seems reasonable to assume that all decks are fully effective. The resisting moment for main deck buckling can then be calculated from $M = \sigma Z$, where Z is the section modulus. It seems reasonable to include continuous longitudinal deck girders in the section modulus here, even though it is not customary to do so. Hence, with

$$\sigma = 34.5 \text{ ksi and } Z = 49{,}100 \text{ in}^2\text{-ft.},$$

the resisting moment for main deck buckling is

$$\sigma Z = 755{,}000 \text{ ton-feet.}$$

While it is true that additional moment could be developed in the limit as additional structure is induced until the ultimate load is reached, it appears unsound to consider it a possibility in the same sense as for hogging. The reason for this is that in hogging the double bottom is a primary system of collapse, while in sagging all three decks would have to be involved. However, if the procedure used in evaluation of hogging were used for sagging, considering upper, 2nd and 3rd decks to be 83% effective and at yield stress of 36 ksi, it is found that a load of 826 x 10^3 ton-feet could be sustained. In such a case, the neutral axis of effective material would be 20.95 feet above base line, with the moment computed about it.

Hence, in summary the estimated capability of the hull girder in hogging and sagging can be tabulated below in comparison with the expected maximum values from Table X for the light load condition.

	Capability Max. bending Ton-Ft.	Lifetime Demand Max. bending Ton-Ft.	.001 Probability Demand Max. bending Ton-Ft.
Hogging	753,000	507,150	602,150
Sagging	755,000	189,650	295,650

It is concluded that the structural capability of the Wolverine State exceeds the criterion for ultimate bending moment (demand) by a comfortable margin. This suggests that if fatigue and brittle fracture considerations were ruled out, some scantling reduction would be permissible. However, such a simplification is not permissible -- as proved conclusively by the occurrence of cracks in the upper deck of the ship in service. (See section on Fatigue).

No attempt was made to estimate the capability of the structure to resist short duration dynamic loads, but there can be no doubt that it is considerably higher than the static figures derived above.

CONVENTIONAL STRENGTH STANDARDS

At the time of the design and construction of the S.S. Wolverine State, there was no explicit section modulus requirement in the Rules of the American Bureau of Shipping, although minimum shell scantlings and strength deck sectional area were specified. It was customary to check the midship section design against the Load Line Regulations promulgated by the U.S. Coast Guard, which specified a required deck section modulus to be met by all vessels subject to Load Line assignment.

In this case the required deck section modulus by Load Line Regulations was 37,536 in^2-ft., on the basis of the formula

$$SM = f\ d\ B$$

where f is a factor having a value of 16.03 at ship length of 496 ft., d is the design draft of 32.75 ft. and B is the beam of 71.5 ft. It was customary in the case of ships with machinery aft for the ABS to add 10% to the Load-line required value, which gave a figure of 41,290 in^2-ft. Actual design values for section modulus for the Wolverine State were:

Deck	41,297 in^2-ft.
Bottom	43,161.

In 1961 a riveted doubler was added to the strength (upper) deck at the Owner's option, port and starboard. It was 5.0 ft. wide and 1 inch thick, and since its length of 169 ft. was less than 50% of the ship length it did not officially add to the section modulus. (It was also less than 40%, as required by current Rules). Recent calculations of section modulus, including the deck straps and other continuous longitudinal members omitted from the original calculation (but not deck girders), gave a deck section modulus of 45,800 in^2-ft.

A calculation of required section modulus by the current (1972) ABS Rules gives the following:

Deck	32,100 in^2-ft.
Bottom	33,100

It is impossible to relate the above required section modulus values to bending moment because the implied allowable stress is unknown. However, Arnott in 1939 (131) gave the standard design bending moment M for cargo vessels with machinery amidships as

$$M = \frac{L\Delta}{35}$$

where L is length and Δ is load displacement. Taking a block coefficient of 0.68 (minimum allowed in the Load Line and ABS Rules), the value of Δ for Wolverine State is 22,600 tons, and

$$M = \frac{496 \times 22,600}{35} = 321,000 \text{ ton-ft.}$$

At the time the ship was built this design bending moment included an unspecified still water bending moment, in addition to the wave bending moment. In fact, the 10% addition to section modulus for machinery aft was a still water consideration. The corresponding bending moment would be 353,000 ton-ft., which is smaller than the highest combined value in Table X of 602,500 ton-ft.

Current (1972) ABS Rules are more specific about still water bending moment requirements (see Chapter III). The maximum still water moment without penalty now would be 136,000 ton-ft. for this ship. If one uses the maximum lifetime still water value of 239,900 given in Table IX, this would mean an increase in the required section modulus by ABS Rules (1972) to:

Deck	47,300 in^2-ft.
Bottom	48,700

Since, as indicated above, the calculated section modulus for the ship with deck straps is 45,800 in^2-ft., it may be concluded that the ship would not quite meet present-day ABS Rule standards (1972), if the still water bending moment of 239,900 ton-ft. were considered a possible maximum value.

MINIMUM TOTAL COST CALCULATION

A preliminary and very approximate calculation has been carried out to illustrate the principles discussed in Chapter VIII for a hypothetical cargo ship, similar to the Wolverine State designed to present-day standards. As in Chapter VIII, it is assumed that the structural capability of the hull is deterministic and therefore that the probability of damage or failure is the probability of exceeding specific values of bending moment.

It was estimated that adding (subtracting) 0.2 in. to deck and bottom plating thickness would increase (decrease) the section modulus by 10% and this change would add (decrease) approximately $100,000 to the initial cost of the ship. An increase in section modulus of 10% would be equivalent to a reduction of average stress levels by a factor of 1/1.1 = 0.91, which reference to the long-term bending moment probability curves shows to correspond to a reduction of failure probability by a factor of 1/10. Similarly, a reduction in section modulus would increase failure probability by a factor of 10.

In order to carry out the calculation of total cost, the following assumptions were made:

Initial cost of ship	$15,000,000.
Replacement value at mid-life	8,000,000.
Failure cost F (replacement value + value of cargo + cost of temporary replacement)	20,000,000.

Since the structural capability of the Wolverine State was found to far exceed the demand corresponding to a lifetime probability of failure and loss of 0.001, it will be assumed arbitrarily that after a 10% reduction the section modulus of our hypothetical cargo ship would still provide at least a 0.001 failure probability, and a 20% reduction would provide 0.01 failure probability.

The total cost calculations on the basis of the above are summarized in Table XI.

TABLE XI
CALCULATED TOTAL COST
OF FAILURE

Design	I	p	pF	Total Cost
Basic Ship - 20%	$14,800,000	.01	$200,000	$15,000,000
Basic Ship - 10%	14,900,000	.001	20,000	14,920,000
Basic Ship	15,000,000	.0001	2,000	15,002,000
Basic Ship + 10%	15,100,000	.00001	200	15,100,200

This indicates an optimum somewhere near the ship with 10% reduction in section modulus. Since these figures are very rough and approximate, a number of other assumptions regarding costs and probabilities were also tried. Naturally there were changes in the final Total Cost column, but the general picture did not change significantly. This supports the conclusion in Chapter VIII that an ultimate failure probability of 0.001 is a reasonable tentative figure for a rational design criterion.

On the basis of some crude assumptions then, it appears that if only ultimate strength considerations were involved, some reduction in scantlings would be economically justified. However, it will be shown subsequently that the picture changes when consideration is given to structural damage that requires repair but does not immediately threaten the life of the ship -- as fatigue cracking, for example.

FATIGUE

Cyclic Loading

The long-term probability data discussed and presented in this chapter for the Wolverine State can be used to provide the load spectra (or patterns) which are needed to determine design criteria relative to fatigue. Since the project is concerned primarily with loads, and not with structural response, it will suffice to derive such loading spectra covering:

(a) Wave bending moments.
(b) Dynamic loads.

Results are presented in Fig. 29 for the light load condition, which corresponded to the actual normal operating condition of the ship in service. The wave stress curves are derived from Fig. 26, using actual section modulus (top), in the manner described in Chapter VIII. The dynamic stress curves were obtained by first estimating the histogram of whipping stress cycles for the light load condition (inbound) on the basis of 12 voyages per year, 0.16 probability of slamming (per 20-min. record) and 15 slams per 20-min. period when it occurs. (It was assumed that no slamming occurs in the hypothetical full-load condition, outbound). As previously noted, the techniques for predicting this in advance for a new design are not yet available. The histogram was then integrated to obtain the cumulative curve shown in Fig. 29.

It should be noted that the variation of mean value for the cyclic loading due to simple wave bending is given approximately by the still water loading information previously presented. The variation in mean value for the superimposed dynamic loading is given approximately by the data on wave bending moment (Fig. 26).

Service Experience

The Wolverine State was one of a group of five C4-S-B5 cargo ships built in 1945. A check of damage and repair records started in 1964 revealed for these ships one case of hull girder damage related to heavy weather -- the S.S. Hoosier State in January 1971, enroute from Antwerp to Philadelphia in ballast condition. A typical case of damage is described as follows:

> "Stbd. side crack between frame No. 115 and No. 116, starting at inboard rivet of gunnel bar forward to outboard rivet of deck strap, in a length of approx. 6.5 feet. Outboard rivet at crack arrester. Starts at 8 in. after frame 116 and travels aft. Crossing 117 approx. 7 ft. inboard of the longitudinal girder and travels between frames 117 and 118, 11 in. aft of 9 in. pillar and continues hatch insert approx. 9 in. and then inboard approx. 3 ft. on plate No. UD73. Longitudinal girder broken."

Temporary repairs were effected in the Azores and permanent repairs in New York in February, 1971.

Study of correspondence and the complete files of the Wolverine State and Hoosier State showed that the above type of damage had been a problem with all three of the States Marine C4-S-B5 vessels. In fact, riveted deck straps were installed by the owners in 1961 for the dual purpose of increasing the section modulus and providing additional crack arresters. Since these were war-built ships (1945) it is reasonable to assume that the quality of the steel was below the standards subsequently established. Whether there were fatigue cracks or brittle fractures, the riveted seams appeared to have been successful in limiting fracture propagation.

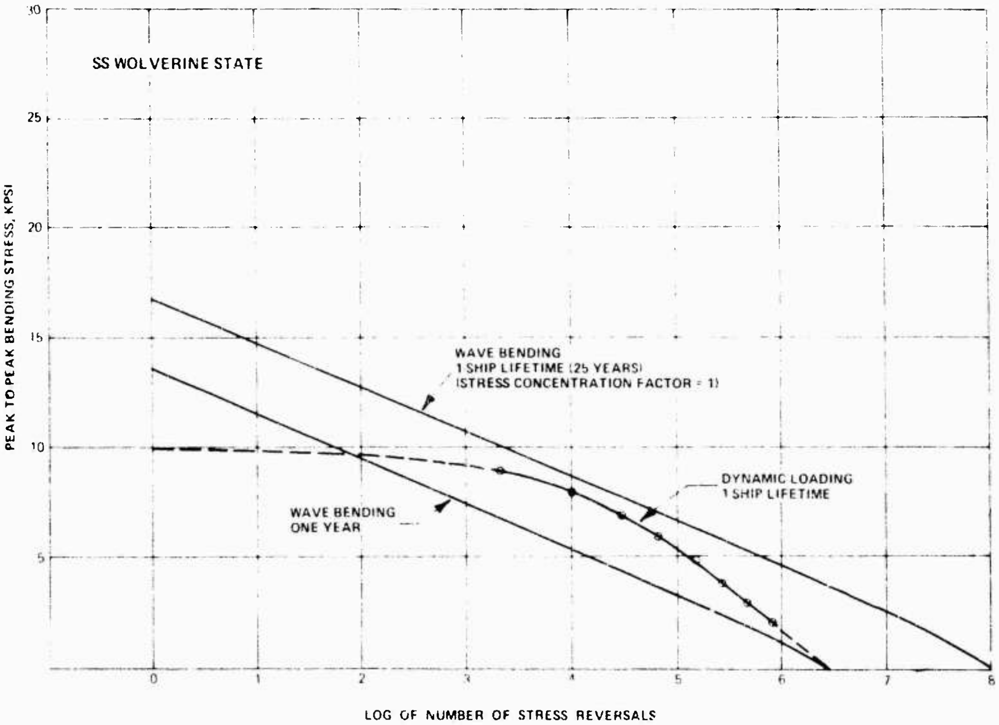


FIGURE 29 - Cyclic Loading "Spectra", *S.S. WOLVERINE STATE*

In a new ship being designed today and to be built with the present steel quality requirements, a lower probability of damage would be expected.

Total Cost with Fatigue Cracking

The total cost calculations previously presented will now be extended to include the effect of fatigue cracking, a type of damage that requires repair but rarely threatens the safety of the ship.

In addition to the previous cost and probability assumptions, it will be assumed that the lifetime expectation of structural damage, q, is 0.25 for the basic ship and that increasing (decreasing) the section medulus by 10% decreases (increases) this expectation by a factor of 1/2 (2). The cost of each damage, S (repair cost + cost of substitute ship + misc.), is assumed to be $1,000,000.

The new calculations are summarized in Table XII.

TABLE XII

CALCULATED TOTAL COST
OF FAILURE AND DAMAGE

Design	I	p	pF	q	(1-p) q S	Total Cost
Basic Ship - 10%	$14,900,000	.001	$20,000	0.5	$250,000	$15,170,000
Basic Ship	15,000,000	.0001	2,000	0.25	125,000	15,127,000
Basic Ship + 10%	15,100,000	.00001	200	0.125	62,500	15,162,500

The table indicates that when fatigue damage, as well as ultimate failure, is considered, the optimum design -- on the basis of some very crude assumptions -- is the basic ship.

It appears then that for a ship such as the Wolverine State, in which the capability far exceeds the demand associated with ultimate failure, attention must be shifted to the nuisance type damage associated with fatigue. The problem is to ascertain whether or not a reduction in scantlings on the basis of ultimate bending consideration would really increase the incidence of fatigue cracking to an uneconomical level.

BRITTLE FRACTURE

As indicated previously, no attempt has been made to establish a load criterion for brittle fracture. However, the loading picture for the Wolverine State should be the same as that obtained under Ultimate Loads, as summarized in Tables IX and X, except that we are concerned only with tensile loads. Hence, the compressive thermal effects need not be included.

Referring to Table X, it may be seen that in the light load condition, when slamming is likely, the largest bending moment would be 602,150 - 11,150 = 591,000 ton-ft. in hogging, which corresponds to a tensile load in the deck. Thus it appears that for this particular ship the principal danger of brittle fracture would be in the hogging condition as a result of large superimposed whipping following a slam. Since it has been shown that the ductile capability of the structure is above 753,000, the actual occurrence of brittle fracture would depend on stress concentrations, steel quality, temperature, and other factors.

X. CONCLUSIONS AND RECOMMENDATIONS

The following are the principal conclusions developed in the current project:

1. Basic techniques are now available for making rational calculations in probability terms of most of the loads acting on the main hull girder of modern merchant ships, including:

   Wave-induced loads.
   Still water loads.
   Thermal effects.

   Further development is needed for the calculation of dynamic loads.

2. Input data for the calculation of loads for ships on various ocean routes is incomplete. In particular, more actual wave records are needed -- even for the North Atlantic routes -- from which to obtain wave spectra for statistical treatment. Actual data on still water loads, particularly in ballast conditions, are also needed for different types of ship.

3. On the basis of the above, a rational load criterion can be set up for modern merchant ships in relation to ultimate failure by buckling or excessive permanent set, with some reservations in regard to dynamic loads.

4. A trial numerical calculation of ultimate bending loads for the S.S. Wolverine State shows that a large margin exists between the "rationally" determined loads and the capability of the hull structure to resist failure by buckling of one of the flanges. The proposed load criterion is less severe than current design standards (ABS), which presumably allow also for fatigue. When similar calculations have been made for a sufficient number of types of ship and checked against conventional empirical standards, it should be possible to adopt a new rational ultimate load criterion for use in the design of even the most unusual ships.

5. The loads affecting fatigue can be expressed as cyclic loading patterns derived from the above item 1, with separate data on the loads having different frequency of variation:

(a) Still water loads (shift of base line).
(b) Diurnal thermal effects.
(c) Low-frequency wave bending loads.
(d) High-frequency dynamic loads.

A fatigue loading criterion appears to be of great importance in design relative to keeping the frequency of occurrence of nuisance cracking at an acceptable level.

6. A load criterion relative to brittle fracture, including dynamic loads, is somewhat uncertain at the present time. However, the determination of the capability of the structure in advance of construction is also uncertain.

It is recommended that further research be carried out on subjects related to the problems of load criteria for ship design. In particular:

1. Obtain many more systematic wave records for important ocean routes that can be spectrum-analyzed and compiled systematically for reference. Of particular importance are the North Atlantic, North Pacific, and areas in the vicinity of the Cape of Good Hope.

2. Obtain systematic data from ship operators on still water loadings on several typical ships over a period of at least six months, from which actual still water bending moments can be calculated.

3. Check and refine available theories for calculating springing loads and stresses.

4. Investigate further slamming and whipping relative to midship bending stresses. Immediate emphasis in this big research area should be:

(a) Obtaining relatively short-term statistical data on the magnitudes of midship slamming stresses that are allowed to occur (by the shipmaster) on ships of different types under different conditions of loading, for guidance in preparing similar new designs.

(b) Obtaining relatively short-term statistical data on the magnitudes of the amount by which the above slamming, and whipping, increase the total combined stress (or bending moment) on ships of different types under different conditions of loading.

5. Investigate further both technical and economic aspects of fatigue damage.

6. Develop further the total cost approach to optimizing design, including failure and damage costs and making use of actual ship damage data.

7. Continue to investigate possible extreme load conditions arising from unusual circumstances, such as shallow water effects on wave spectra, unusually severe local sea conditions (Bay of Biscay), docking loads, wave impacts on side shell, loads due to shipping water on deck, etc.

8. Extend the work on wave loads beyond the determination of midship bending moments to include the determination of pressure distributions over the entire hull surface of a pitching, heaving and rolling ship. Such a detailed picture of hydrodynamic loads is believed to be essential for the application of modern finite element techniques of structural analysis.

9. Encourage parallel research on determining ship structural capability -- and probability of damage and failure -- on a probabilistic basis, considering both quasi-static and dynamic (rapidly applied) loads.

10. Investigate non-linear flare immersion effects on ships with large flare, including further study of dynamic structural response.

11. Continue research on methods for predicting slam loads and phase relations to wave bending.

ACKNOWLEDGMENTS

The authors wish to acknowledge the assistance of Teledyne Materials Research in furnishing additional data beyond that which has been published, of Oceanics, Inc., in connection with problems arising in the use of their computer program for wave loads, and of the American Bureau of Shipping for allowing certain results obtained at Webb under their sponsorship to be included in the report. Useful still water loading data were obtained from Mr. John Ritter of States Marine Lines and from Mr. John Boylston of Sea Land Services. Temperature data for a tanker were provided Mr. L. R. Breves of Esso International. Several part-time research assistants and students have been of assistance in various aspects of the project, including William Garzke, D. Finifter (MIT), Thomas Zielinski (Webb), and J. van Santen (Delft).

The helpful comments and suggestions on drafts of several chapters are greatly appreciated from the following individuals:

Prof. G. Aertssen, formerly University of Ghent, Belgium
Prof. J. Caldwell, University of Newcastle, England
John P. Dalzell, Davidson Laboratory
Mr. D. Faulkner, Ship Department, Ministry of Defence, Bath, England
Dr. R.A. Goodman, Lloyd's Register of Shipping, London
Prof. C.J. Jensen, Lloyd's Register of Shipping, London
M. Jourdain, Institut de Recherches de la Construction Navale, Paris
Prof. A. Mansour, Massachusetts Institute of Technology
Ir. J.J.W. Nibbering, University of Ghent, Belgium
Dr. N. Nordenstrøm, formerly Norske Veritas, Oslo
Dr. M.K. Ochi, Naval Ship Research and Development Center
Dr. Ing. Lorenzo Spinelli, Registro Italiano Navale, Genoa
Mr. K.V. Taylor, Lloyd's Register of Shipping, London

Figs. 1, 2 and 10, from a paper (3) published in the 1971 Transactions of the Society of Naval Architects and Marine Engineers, is included here by permission of that Society.

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APPENDIX A

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