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Structural Response and Computer-Aided Design Procedure

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ABSTRACT

A general review of computational techniques employed in predicting structural response and the utilization of computers in the design of ships is presented in this paper. The availability, applications, restrictions and merits of some popular computer programs are discussed. Emphasis is placed on the correlation between theoretical and experimental results, the deviation of stress distribution from linearity, and structural optimization techniques. Points of interest are illustrated by graphs and examples. Recommendations for future items of research are also presented.

INTRODUCTION

The purpose of this paper is to present a survey of several aspects of structural analysis and design of ship structures. Existing material is presented from many sources; this accounts for the use of different unit systems and different symbols. Many portions are adapted from references, which should be consulted for additional details and background.

This paper consists of three parts. The first part deals with structural response and includes computation techniques, correlation of theoretical and experimental results, influence of initial distortions and examples of computer results interpretation. The second part deals with computer-aided design procedures and includes ship characteristics and stability, design loads, structural configuration and scantlings, structural optimization and computer graphics. The third part presents some conclusions and recommendations, giving a summary of the current status and recommended research projects.

STRUCTURAL RESPONSE

Computation Techniques

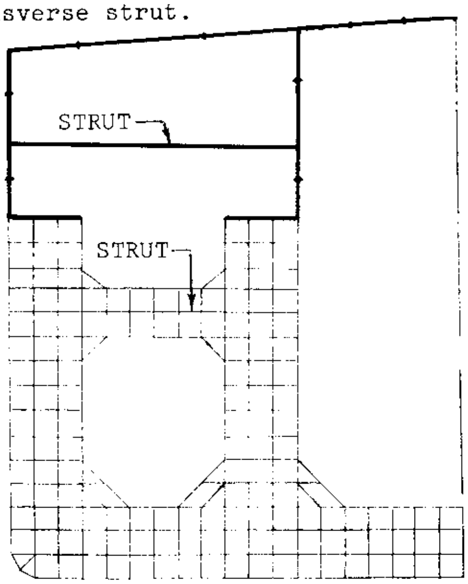
Background. The advent of electronic digital computers in conjunction with the mathematical formulation of matrix methods has resulted in a vast spectrum of commercially available computer programs capable of performing structural analyses of varying degrees of complexity.

Obviously there is an inherent hierarchy among these computer programs or systems softwares. At first there were the simple beam-rod programs, offering tremendous flexibility to the user as far as describing his problem and determining his mode of analytical solution (stiffness against flexibility methods, finite differences, etc.), capable of handling large frames, trusses or grid structures. More experienced users of these systems were able, with some ingenuity, to extend the capabilities of these programs to handle certain problems involving plates.

The success which these earlier releases achieved spurred an even more overwhelming interest by the various engineering disciplines toward the use and capabilities of the matrix formulation of engineering problems. An even greater interest was induced when certain United States government agencies realized that the programs then available were incapable of handling their more sophisticated analyses. Regular conferences and seminars were held and led to widespread dissemination of information on topics including the development of general matrix formulation theory for static and dynamic structural analyses, buckling problems, transfer matrix formulation, finite difference schemes and the finite element method.

Beam-Frame Approximations. The earliest structural analysis computer programs, such as STRESS and the initial versions of STRUDL, were limited to truss and beam elements to represent a complex structure. This required the idealization of plating by a combination of rods and beams, using an effective width of plating associated with stiffeners, taking into account considerations of buckling and shear lag. One method used to calculate areas of truss elements with stiffness equivalent to that of the plating being modeled was developed by Hrennikoff, (1)and provided approximate formulas for representation of plates loaded in-plane and out-of-plane by the use of equivalent truss panels or lattices.

The first plate elements made available in computer programs were membrane elements, so that an approximate method still had to be used to represent bending plates. Nowadays it may still be useful and efficient to represent a portion of plating by an equivalent beam, as illustrated in Figure 1. In this case, for a detailed local analysis of the lower portion of the web frame, beam elements can be used to model the upper portion, including the transverse strut.



BEAM-FRAME APPROXIMATION

Figure 1

Present day computer programs allow the user to model a structure much more realistically, but the beam-frame approximations generally provide good results and may be used for a quick and simplified analysis or preliminary design.

Finite Element Techniques. Some of the larger finite element computer programs developed in the United States are briefly described in this section in alphabetical order. A much more extensive status review of available programs is provided in (2).

1. ANSYS (Engineering ANalsis SYStem) is a general purpose finite element computer program with an element library of some sixty finite elements (beams, plane stress elements, axisymmetric solid elements), seventeen of which have nonlinear formulations. The program is capable of performing static and dynamic structural analyses as well as analyses involving fluid flow and heat transfer.

Its nonlinear capabilities include small strain plasticity, creep induced by thermal changes and irradiation, swell induced by irradiation, large deflections and buckling. The dynamic capabilities include eigenvalue-eigenvector extraction, steady-state harmonic response, and linear and nonlinear transient response. Materials may be isotropic or anisotropic and may have temperature dependency.

For static analyses, the program makes use of the wavefront method of solution, thus making the program unrestricted as far as bandwidth size is concerned. For dynamic analyses, the consistent mass matrix and explicit quadratic integration routines are employed. Available plotting software packages interface with this system for geometry, stress, displacements, and temperatures.

2. DAISY (Displacement Automated Integrated SYstem) is a general purpose finite element displacement method structural analysis computer program, with sixteen active structural elements, including beam, torsion, plane stress/strain, thin shell, and thick plate, three-dimensional solid, composite material and isoparametric elements.

At the American Bureau of Shipping, ABS/DAISY has been extensively employed in the review design and analysis of large tankers, high-speed container ships, LNG vessels, general cargo ships, barges and bulk carriers with structures ranging in size from local detailing to an entire ship hull.

The primary components of the DAISY system are its preprocessor programs, the DAISY finite element program and its postprocessor programs. The three-dimensional preprocessor system consists of a series of programs which can generate data for all or a portion of a ship hull structure. The two-dimensional preprocessor system is used to generate automatically planar structures such as web frames, girders or brackets. Users capable of developing their own computer programs to generate input data or who may have existing data-generating routines may easily integrate their programs into the system. This feature of the DAISY system provides the user with a maximum amount of flexibility to perform a wide range of analyses.

Some of the special DAISY features are a substructuring capability, automatic assignment of nodal degrees of freedom, singularity detection, symmetric and antisymmetric load combinations, automatic generation of buoyancy and tank loads, etc. DAISY can also be used without pre-processors and contains many user-oriented subroutines to allow maximum flexibility in the use of the program.

Further developments in DAISY include the additional capability of dynamic and buckling analyses.

The postprocessor system consists of programs to display the DAISY calculated results, including printout of the entire results and/or selective output and plotting of nodal displacements and stress contours on an incremental plotter. Postprocessors may also be developed by the user to interface with the DAISY results.

3. EASE 2 (Elastic Analysis for Structural Engineering) is a structural analysis computer program, with fewer elements for the discretization of the entire structure than found in other general purpose programs available today. EASE 2 has the capability of analyzing structural models with temperature-dependent material.

4. MARC (Marc Analysis Research Corporation) is a general purpose finite element program for elastic analysis and for nonlinear static analysis of structures with large displacements. The element library contains two- and three-dimensional elements and plate and shell elements. The program is particularly oriented toward solving elastic-plastic and creep problems. Plasticity behavior is based on the theory of isotropic, elastic-plastic, time-dependent materials with a von Mises yield criterion, isotropic or kinematic strain hardening, temperature-dependent elastic properties and equivalent yield stress. Creep behavior is based on a von Mises flow criterion with isotropic behavior described by an equivalent creep rate law specified by the user. The program uses the tangent modulus method for plasticity and an iterated initial strain method for creep calculations.

5. NASTRAN (NAsa STRuctural ANalysis) is a general purpose computer program designed to determine the elastic structural responses to a wide range of loading conditions utilizing the finite element displacement method. The program is applicable to most linear and some nonlinear systems and can generate static responses to concentrated and distributed loads, thermal expansion, and enforced deformations (such as camber and boundary displacements); dynamic responses to transient loads, steady-state harmonic loads, and random excitation; determination of real and complex eigenvalues for use in vibration analysis, dynamic stability analysis and elastic stability analysis, etc.

NASTRAN is highly user-oriented because of the very systematic organization of the program. Errors are usually detected before an actual attempt at assembling the structural stiffness matrix. Because of the program's modular structure and a restart feature, it is possible to restart the program from intermediate stages of the analysis procedure.

The element library of NASTRAN has twelve elements but each element on its own is so general that it is equivalent to several "entirely different" elements in other programs.

6. SAP IV (Structural Analysis Program) is a general purpose finite element structural analysis program capable of performing static and dynamic analyses of linear systems. SAP IV has an element library containing elements for three-dimensional truss and beam, plane stress and plane strain, three-dimensional solid, thick shell, thin plate, thin shell, boundary, and pipe problems.

SAP IV in its dynamic analysis mode utilizes the mode superposition or direct integration techniques and will perform the calculation of frequencies and mode shapes by either the determinant search or subspace iteration method, depending on the size of the problem.

7. STARDYNE (STAtic and DYNamic Structural Analysis System) is a finite element structural analysis computer program consisting of a series of compatible structural engineering programs designed to analyze linear elastic structural problems. The program uses the stiffness method and is capable of analyzing a wide range of static, dynamic and stability problems.

The static structural analysis capabilities of STARDYNE include applied element and nodal loading, specified displacements, automated thermal analysis, inertia loading, and combined loading cases. The dynamic structural analysis capabilities cause STARDYNE to extract eigenvalues and eigenvectors for any desired frequency range, and to compute the generalized weights, participation factors, and internal forces on elements associated with each mode.

STARDYNE can find solutions to a free-free system if the applied forces are self-equilibrating. The program internally renumbers nodes in order to minimize the bandwidth.

8. STRUDL II (STRUctural Design Language) is a subsystem of the Integrated Civil Engineering System, ICES.

STRUDL II offers a Problem Oriented Language, POL, which allows the user to instruct the system through simple words or phrases as to which procedures should operate on the data.

Member design facilities for framed structures are also included in STRUDL II. The reinforced concrete portion provides for the proportioning of beams, flat slabs and columns. The output consists of member dimensions and main longitudinal reinforcement. The member may also be completely specified, in which case its adequacy is checked against flexure, shear bond and deflection criteria.

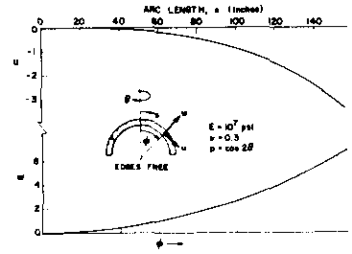
STRUDL II contains a steel member selection procedure based on standard rolled sections and the AISC code, and provides a procedure for the nonlinear analysis of frames, plates, and shallow shells. Other features included are linear buckling analysis, dynamic analysis, and a frame optimization procedure. STRUDL II is easily interfaced with commercially available plotting packages.

9. ASKA, PASSAGE and SESAM are some of the other large finite element programs developed abroad.

Other Approaches. Other approaches to the solution of structural problems include finite difference methods and classical solutions, with extensive use of matrix methods and numerical integration techniques. Many of the programs use a combination of techniques and methods. A fairly comprehensive listing and description of existing computer programs are given in (2). The method of finite differences can be subdivided into the equilibrium method (stresses in the equilibrium equations are expressed as differences at finite intervals) and the energy method (strains in the energy equations are expressed as differences at finite intervals).

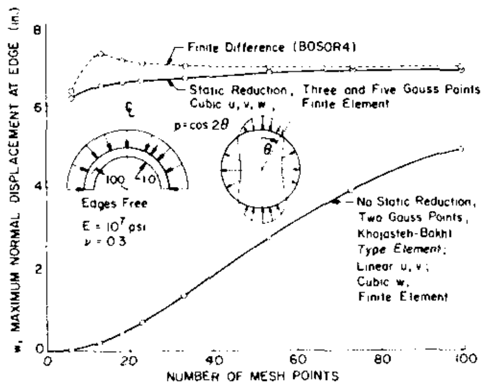
A comparison of results obtained by the finite element and finite difference methods for the case of an axisymmetric shell is illustrated in Figures 2 and 3, adapted from (2), which show the results of a convergence study involving a free hemisphere pinched by a cos 2θ pressure distribution. This rather ill-conditioned problem is a very good test of various methods of discretization. The problem is ill-conditioned because small forces cause large displacements. Thus, the predicted reference surface strains are very small differences of relatively large numbers. The dotted line in Figure 3 is obtained with use of a half-station finite difference energy method, which is equivalent to a finite element method based on linear functions for the tangential displacements u and v and a quadratic function for the radial displacement w.

Classical methods of solution involve the use of equations governing the elastic behavior of the structure, and can be used for simple idealized cases where the equations are known and the solutions can be easily obtained.



Meridional and normal displacements at θ = 0 of a hemisphere with a free edge submitted to pressure p = cos 2θ

Figure 2



Comparison of convergence of finite element method with finite difference energy method

Figure 3

Comparison of Computer Programs. The intent here is not to give a comparison as such of the various systems, since different agencies are inclined to have specialized applications or interests in their applications as far as the programs are concerned; therefore any such attempt would undoubtedly be biased to some extent. A correlative study of the most commonly used systems is given in order to present a better picture of their relative capabilities. No attempt is made to compare the running or execution times on the various computer systems, since no two systems will be identical in their available core allocation, program subroutines or problem formulation. Any such comparison would therefore be misleading, until some standardization is established.

Table I shows the availability of twelve analysis options in the various finite element programs previously described. Most of this information was adapted from (2).

TABLE I - ANALYSIS OPTIONS AVAILABLE WITH PROGRAMS

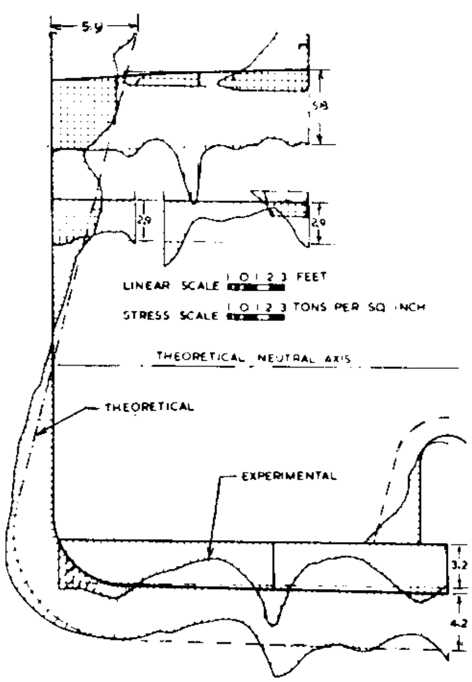
Analysis Options	ANSYS	DAISY	EASE 2	MARC	NASTRAN	SAP IV	STARDYNE	STRUDL II
1. Small displacement	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
2. Large displacement	Yes	No	No	Yes	No	No	No	Yes
3. Incremental plasticity	Yes	No	No	Yes	No	No	No	No
4. Creep	Yes	No	No	Yes	No	No	No	No
5. Temperature-dependent material	Yes	No*	Yes	Yes	No	Yes	No	No
6. Natural frequencies, mode shapes	Yes	No*	No	Yes	Yes	Yes	Yes	Yes
7. Transient response	Yes	No	No	Yes	Yes	Yes	Yes	Yes
8. Data generation	Yes	Yes	Yes	Yes	No	Yes	Yes	Yes
9. Graphic displays	Yes	Yes	Yes	Yes	Yes	No	Yes	Yes
10. Multielement library	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
11. Thermal effects	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
12. Bifurcation buckling	Yes	No*	No	Yes	Yes	No	No	No

* Under development

Correlation of Theoretical and Experimental Results

Purpose. The need to make certain simplifying assumptions in developing theories and formulas implies a certain degree of uncertainty in the accuracy of the results, the range of their applicability and the limitations beyond which they cannot be extrapolated. One way to quantify this uncertainty is to conduct experiments to prove, disprove or amend theoretical results.

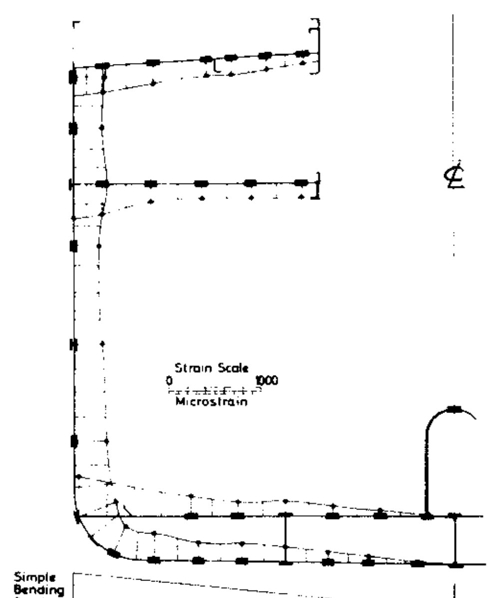
Some typical examples of the correlation between theory and experiments are presented in the following sections, with results from both full-scale and model tests on static, dynamic and thermal loadings.



Full-Scale Distribution of Longitudinal Stress at Center Test Section.

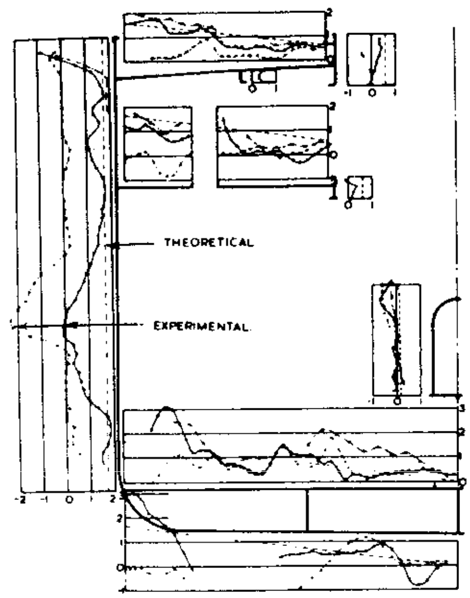
Figure 4

Static Loadings. Measurements taken on a model of a cargo vessel (3) provide a comparison with theoretical calculations and full-scale measurements. Figure 4 shows theoretical and experimental distributions of full-scale longitudinal stresses for a hogging loading condition. The stress distributions are in broad agreement, with some detail differences such as a greater loss of effectiveness of the bottom plating in the ship than the model. Figure 5 shows a similar comparison for longitudinal strains due to horizontal bending and the agreement between values calculated from elementary beam bending theory and those based on measurements is reasonable. The strain distribution of Figure 5 shows some loss of effectiveness of the side shell plating, and the strain distribution due to horizontal bending of the actual ship is shown for comparison in Figure 6.



Longitudinal Strains at Center Test Section

Figure 5

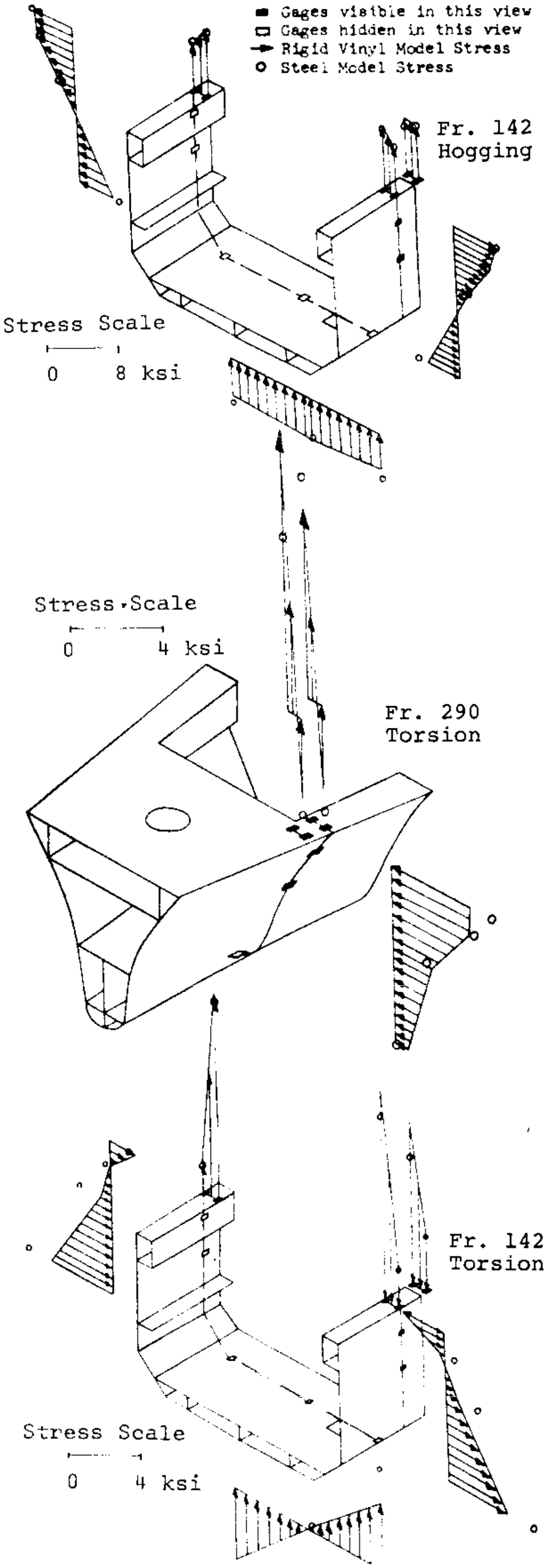


Full-Scale Distribution of Stress at Center Test Section.

Figure 6

A direct comparison between a vinyl structural model of a containership and its steel counterpart under equivalent static load conditions was undertaken (4), and the results show favorable correlation. Some of these results are shown in Figure 7. The same information can be retrieved from the experimental programs of the steel model and the rigid vinyl model. The use of rigid vinyl as the modeling material reduces construction efforts, improves the representation of complex structural shapes and details, and offers reduction of experimental efforts due to ease of handling and convenient load magnitudes. On the other hand, the stress range in which it can be used is limited, the room temperature and humidity must be carefully controlled, modeling of structural joints must be done with care in areas which may affect strain gage results and it has the undesirable characteristic of creep. Differences between the rigid vinyl and steel structures at joint discontinuities indicate that steel model welds may have a stiffening effect on the hull skin as recorded by nearby gages. Gluing may have an effect on the fixity of the vinyl model joint.

Previous model experiments have established that longitudinal stresses are proportional to the bending moment at the frame under examination, regardless of the moment distribution over the rest of the structure. However, this convenience is not available for torsional investigations, since warping stresses are a function of the torsional moment distribution over the entire structure, as shown by the results on Frame 290.



Comparison of Stresses

Figure 7

Full scale experiments to measure the stresses in actual structure on board large tankers subjected to static and dynamic loadings were conducted to correlate the measured values with those obtained from analytical calculations, and the correlation proved to be excellent (5). Some typical results for two ships are shown in Figures 8 through 11, giving the variations of stress between the base and the test conditions for static loadings.

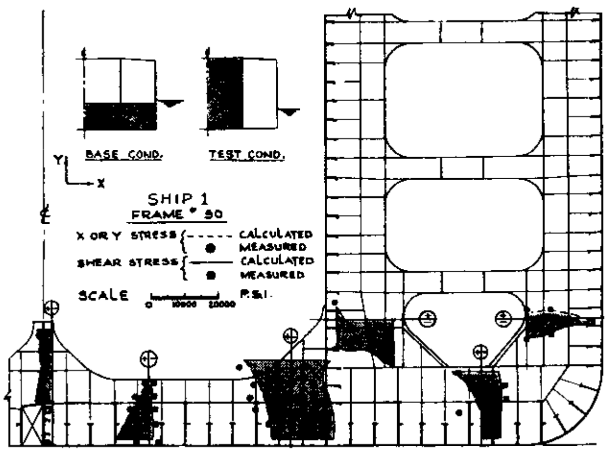


Figure 8

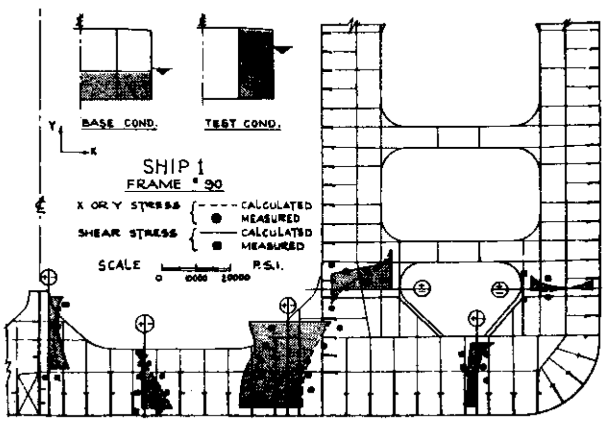


Figure 9

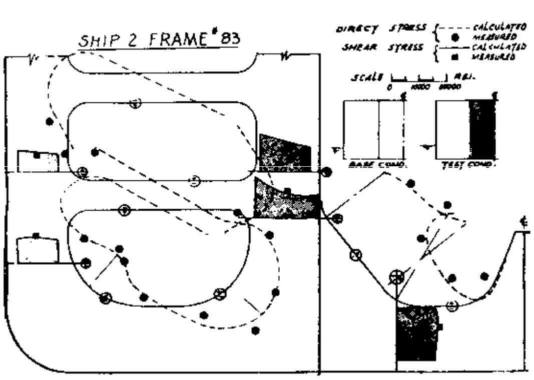


Figure 10

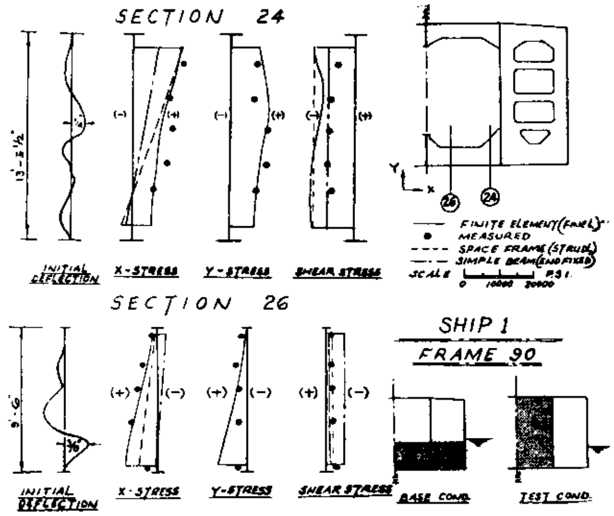


Figure 11

Some of the conclusions drawn from these experiments show the stress characteristics in certain areas and can be summarized as follows:

1. The computed stresses in the lower parts of the supporting structures agree very well with the experimental results, particularly in the highly stressed regions. A few measured stress points deviate from the computed values. These are considered to be influenced either by the proximity of openings, or by initial deflections of an appreciable magnitude.

2. Measured values in the upper parts of the supporting structures are generally low, except in areas of stress concentration such as the corners of the deck and vertical transverses and at the ends of struts. Because the stress values are usually small in the upper portion of the hull structure, most of the investigations have concentrated on the bottom and lower side structure which is usually subject to greater loading and correspondingly higher stresses. By paying attention to details, such as fitting of more rosette gages and a more exhaustive investigation of the effects on the hull girder of temperature changes which are significant in this region, it may be possible to obtain readings which will compare favorably with the computed stress values.

3. Discrepancies caused by neglecting items such as initial deflection, change of temperature gradients along the hull, etc. tend to decrease as the applied loads are increased.

4. As expected, initial deflection affects the distribution of the direct stresses, but would appear to have little effect on the distribution of shearing stresses.

5. Since it is generally assumed that the web portion of the member absorbs the total shearing load, it was expected that the shear measurements in these locations would be less than calculated, with the difference accounted for by virtue of the relatively thick flange and shell plating carrying some of the shearing load.

6. Stress values computed by the finite element and space frame methods correlate with measured values better than the values obtained by the simple beam with end-fixity approximation; a comparison of the former two methods indicates better agreement for the finite element method, which also has the capability of calculating the direct stress perpendicular to the long axis of the structural member.

Bearing in mind the basic objective of the experimental work, correlation of calculated and measured stress values, Figure 11 is of particular interest, because it shows the results of calculations made by the three methods mentioned above. The distortion of the web frame as built is smaller than shown in Figure 11, because the vertical scale has been compressed. Actually, in the case of section 26, the initial deflection is only 3/8" near the bottom. As should be expected, the recent and more advanced methods do show better agreement as is indicated in the next three views. In paricular, this is true with respect to the stresses measured in the horizontal direction. However, in this connection, it is particularly important to note that, where reasonable agreement does not occur, it is usually influenced by the built-in distortion which renders the stress level indeterminate. It should be noted that only the finite element method takes into account the direct stress in the vertical direction, as shown in the section "Y-Stress", and that there is good agreement between calculated and measured values.

Figure 11 illustrates not only the results of improved computation methods, but also the reasonably close values between calculated and measured values for a known static loading condition.

In the comparison of computed and measured stress values as indicated in comment number 1, good agreement was obtained in way of the lower portion of web frames, particularly in the highly stressed regions. Reference has also been made to the inability to compensate for matters such as residual distortions due to welding, other minor unfairness of structure and minor differences in material thicknesses as manufactured. Herein lies one of the outstanding observations associated with this type of experimental work, namely that, owing to the several indeterminate factors involved, some of which may be of a built-in nature, it would appear at the present time that reliance could be placed on instrumentation results only when the loads are such as to induce appreciable stresses in the members involved. A lower limit approximating 8,000 - 10,000 psi would appear to be in order for this purpose.

Dynamic Loadings. Wave-excited main hull vibration results from the interaction between a ship's hull and the sea. In relatively small ships, slamming or bow flare effects seem to be the cause of excitation. In most such cases, the vibration originates from synchronization of the natural frequency of hull vibration with a region of the wave energy spectrum which has a relatively high energy content. The vibration at the natural frequency of the two-node vertical mode of vibration of the ship's hull is a phenomenon known as "springing".

Theoretical calculations for a large tanker in ballast were performed (6), using the ISSC single-parameter spectrum with a predominant period of 6.5 seconds, and the statistical stress response to irregular waves is tabulated below.

TABLE II-STRESS STATISTICS

Statistic	Approx. duration (minutes)	C_n	α=0(head seas)				V=16.0 knots				
			V=11	13	15	17	α=0	10	20	30	40
σ 1/3		4.0	0.44	0.56	0.18	0.15	0.28	0.33	0.42	0.52	0.51
σ 1/10		5.1	0.56	0.72	0.23	0.65	0.36	0.42	0.53	0.67	0.65
1 in 100	4	6.45	0.71	0.91	0.30	0.82	0.45	0.53	0.68	0.84	0.82
1 in 500	20	7.35	0.81	1.04	0.34	0.94	0.51	0.60	0.77	0.96	0.94
1 in 1000	40	7.71	0.85	1.09	0.35	0.99	0.54	0.63	0.81	1.01	0.99

σ = stress amidships, peak to peak, ton/in^2
α = ship heading angle relative to waves, degrees(=0 for head seas)
V = ship speed, knots
C_n= statistical constant

Measurements taken on the same ship over two days indicated about forty hours of vibration present, with sea states ranging from Beaufort numbers 3 to 6. Maximum peak to peak amidship stress values were mostly in the range of 0.5 to 1.8 ton/in^2, with a maximum of 2.25 ton/in^2 reached several times. Since the measured records contained about 100 cycles, these stresses should be compared with the stress statistic of 1 in 100, ranging from 0.30 ton/in^2 at 15 knots and $\alpha = 0$ to 0.91 ton/in^2 at 13 knots and $\alpha = 0$.

The three main factors that should be taken into consideration in the above comparison as having a great influence on the correlation are the wave energy spectrum (excitation), the distribution of the rate of change of added mass (hydrodynamic damping) and the forcing function. For continued studies of comparisons between theoretical and experimental values, a more thorough investigation of the following items is required:

1. hydrodynamic forces exerting influence upon the ship hull and exciting its overall vibration.

2. definition of damping ratio of elastic hull vibrations.

3. combination of ordinary wave-bending moments and vibration moments.

4. longitudinal distribution of added mass.

5. effect of inclusion of a three-dimensional end-correction factor to the usual two-dimensional strip theory.

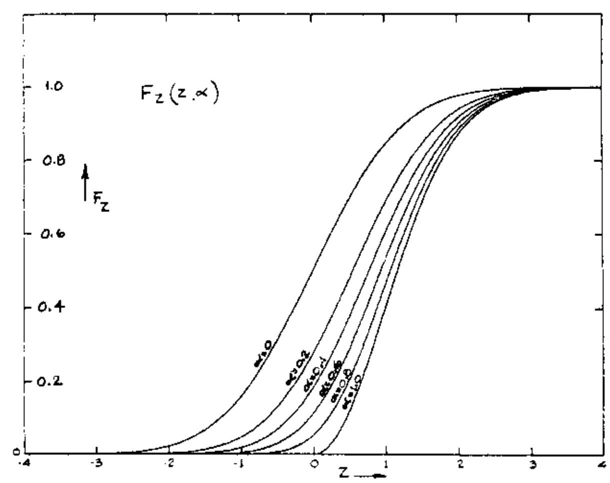
6. determination of wave energy spectrum at its high frequency end.

7. applicability of model experiments to the prediction of actual stress levels.

8. damping distribution and exciting force distribution along the ship's hull.

About 800 recordings of longitudinal stresses amidships were taken on a 200,000 DWT tanker, and 70% of the recordings contained vibratory stresses. Spectral analysis of these data indicated that the assumption of a narrow energy spectrum for springing is justified. The largest vibratory stress measured in the deck amidships was 2.2 ton/in^2 peak to peak, corresponding well with the largest values measured on the above ship (2.5 ton/in^2). In the ballast condition, the probability of getting hull vibrations seemed independent of both Beaufort number and wave direction. In the full load condition, the probability was found to be larger in head seas than in beam seas. The vibratory stresses were also generally larger in the ballast condition.

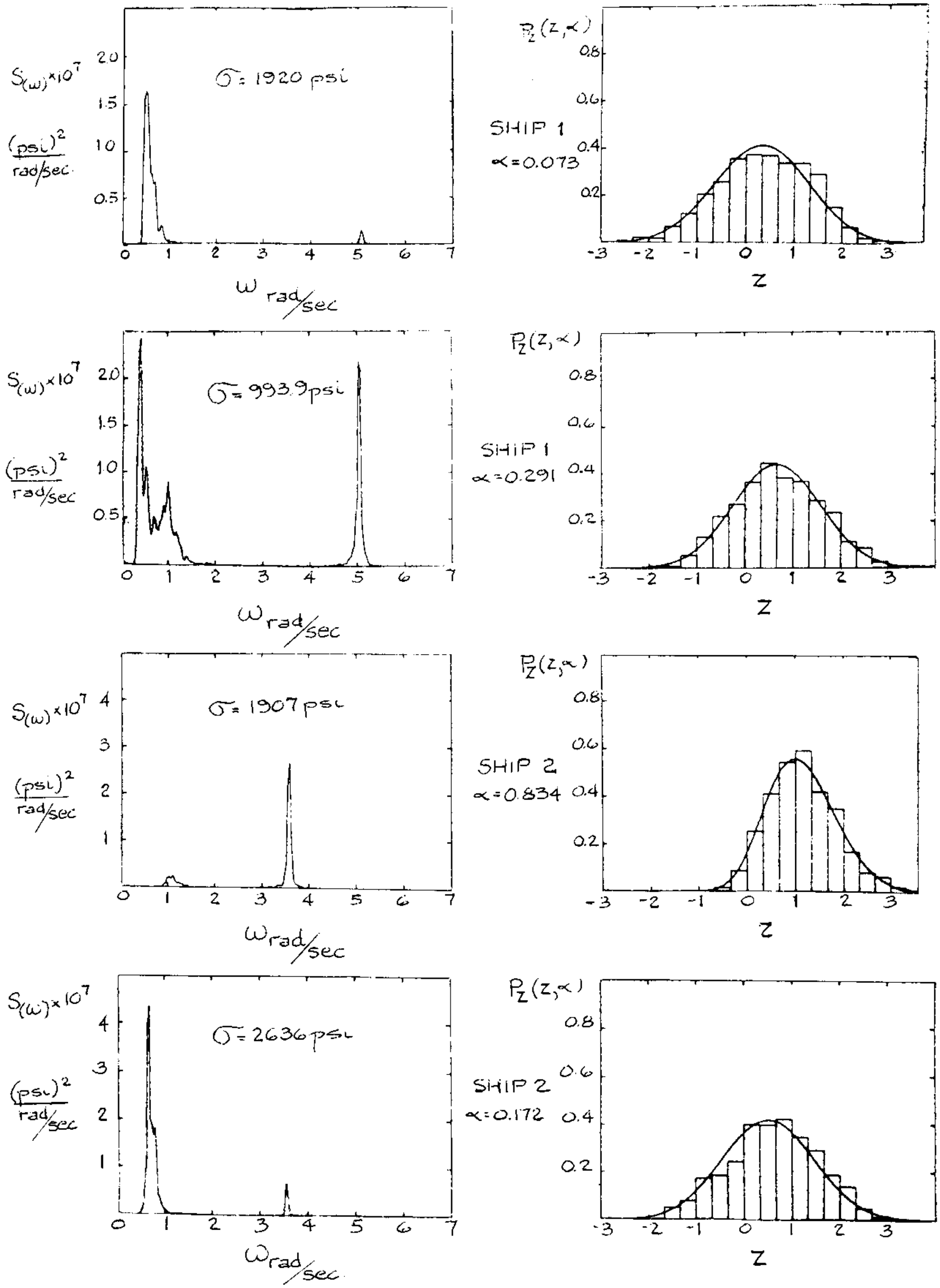
Another study of correlation between theory and experiment for springing stresses was conducted for two Great Lakes vessels (7). The midship longitudinal stress due to springing was theoretically assumed to follow a Gaussian distribution. Because the springing stresses are narrow-band, the peaks follow a Rayleigh distribution. If we define a non-dimensional variable z equal to the peak stress divided by the variance, the probability that z exceeds a certain value can be calculated by analytical integration and is plotted in Figure 12 for various values of a spectral width parameter α.



Cumulative distribution function $F_Z(Z,\alpha)$

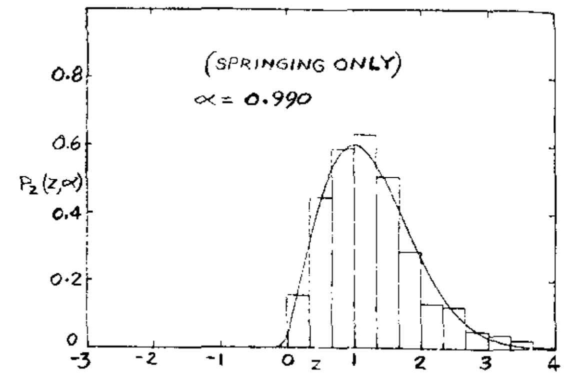
Figure 12

An experimental investigation was carried out to determine how well the theoretical predictions agreed with actual peak distributions of measured springing stress records. Some of the complete stress records for two different ships are shown in Figure 13, and the springing stress histogram for one ship is shown in Figure 14. From this data, it appears that the springing stresses manifest themselves as a narrow-band Gaussian phenomenon and therefore are quite highly correlated statistically. The measured peak distributions were found to agree quite well with the theoretical probability density functions. Further studies of springing correlation are being conducted at Webb Institute and at the Massachusetts Institute of Technology, with emphasis on theoretical developments.



SPECTRA AND HISTOGRAMS FOR STRESS RECORDINGS

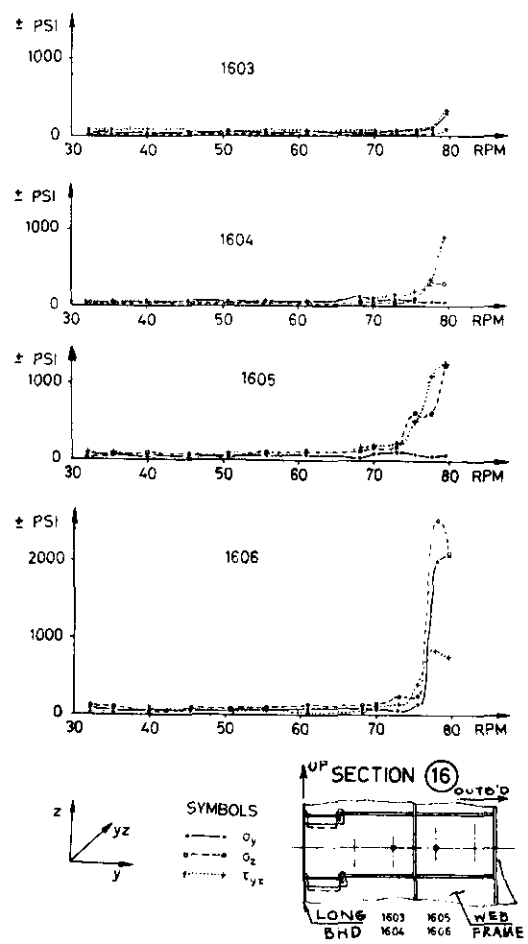
Figure 13



Ship 2 - Histogram For Stress Recording

Figure 14

A large tanker was instrumented for measurement of static, dynamic and vibration stresses (8), and the results of stresses due to propeller-excited vibration are shown in Figure 15. These stresses were measured on a typical webframe adjacent to the longitudinal bulkhead at about one-third depth; they were small (within the range of calculated stresses) and showed no potential vibrational problems.



Vibration Run Propeller Excitation

Figure 15

Thermal Loadings. Theoretical-experimental correlations for thermal loadings were conducted on an asphalt tanker, carrying cargo with temperatures up to 300° F.(9), (10).

The magnitude of thermal stresses developed in the hull structure is governed by the restraints provided by surrounding structure and by the non-uniform temperature differences in the hull due to the ship's internal and external environment. In order to determine the hull girder stresses in the vessel, a sufficient portion of the structure must be considered to reflect both structural restraint and the thermal effects on the hot cargo and the lower temperature of the external environment. The three-dimensional hull structure from Frame 56 (about 23 feet forward of amidships) to the forward perpendicular of the vessel was considered in the finite element analysis, including the region of the maximum total hull girder stress as well as sufficient surrounding structure to accurately represent the thermal stress behavior.

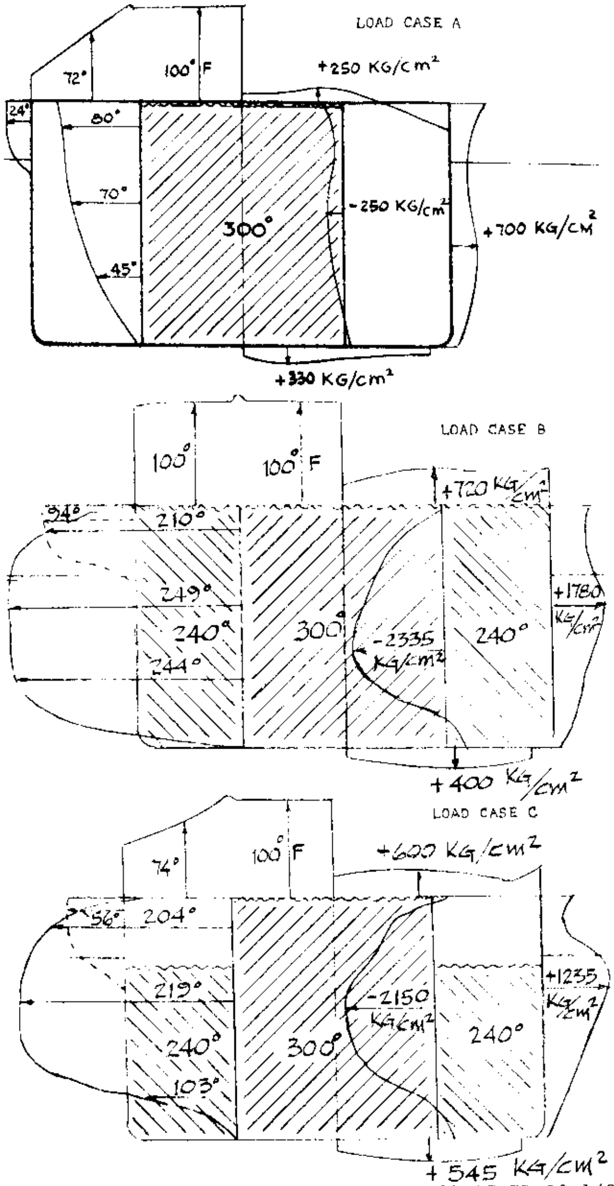
The theoretical results, as expected, indicate that the thermal stresses increase with decreasing deck temperature, since the temperature differences between bulkhead and deck, and bulkhead and bottom will increase. The thermal stresses in the side shell plating near the waterline decrease with decreasing deck temperature, due to decreasing vertical temperature gradient in the side plating.

Some of the thermal loads and calculated stresses are shown in Figure 16. The measurements were taken with strain gages at various locations, and some of the comparative results are shown in Figures 17 through 19.

The correlation between theoretical results and measured stresses is good, bearing in mind the large number of unknowns and idealizations utilized in any theoretical study.

For the port and starboard side shells, Figures 17 and 18, excellent correlation shows clearly a peak of stresses at the round of the sheer strake and the peak of stresses due to the steep temperature gradient between freeboard and underwater shell.

For the deck at frame 54, Figure 19, the correlation is not very good, but theoretical stress distribution is with non-buckled longitudinal bulkheads, and since the bulkheads are buckling in reality, the measured compressive stress peaks over the longitudinal bulkheads are less marked. It is likely that the structural discontinuities on the deck also have some disturbing effect.



THERMAL LOADS AND LONGITUDINAL STRESS AT FR 62 1/2

Figure 16

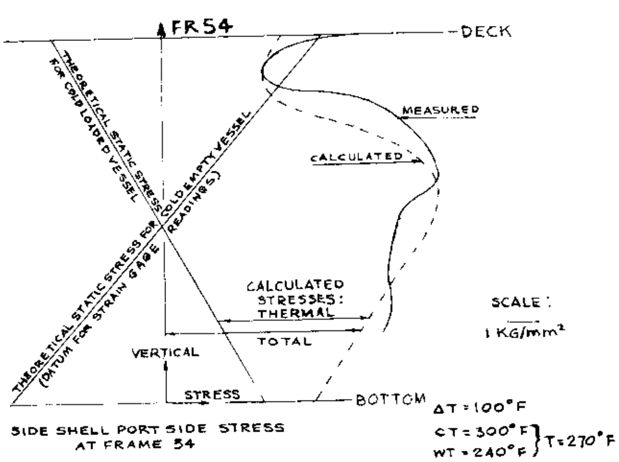


Figure 17

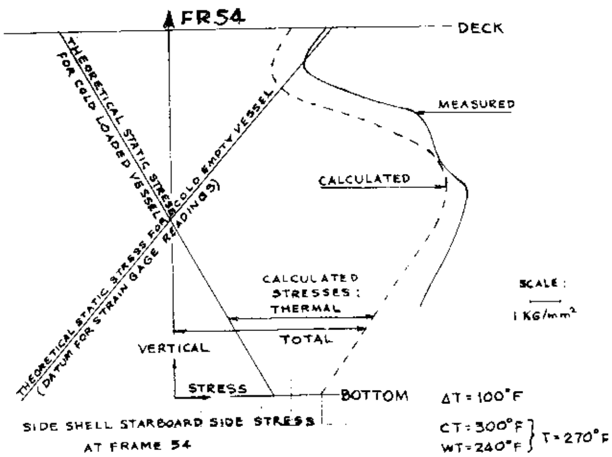


Figure 18

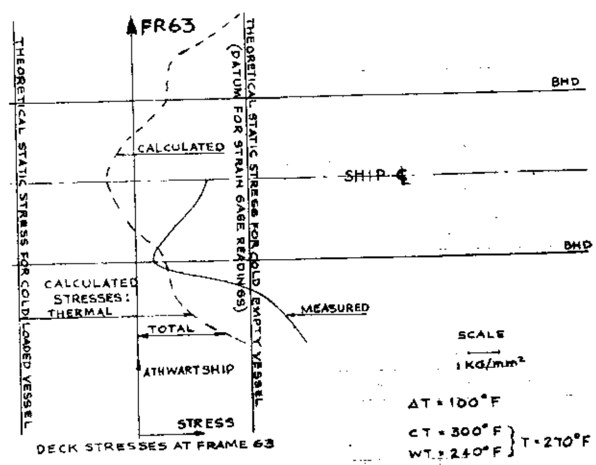
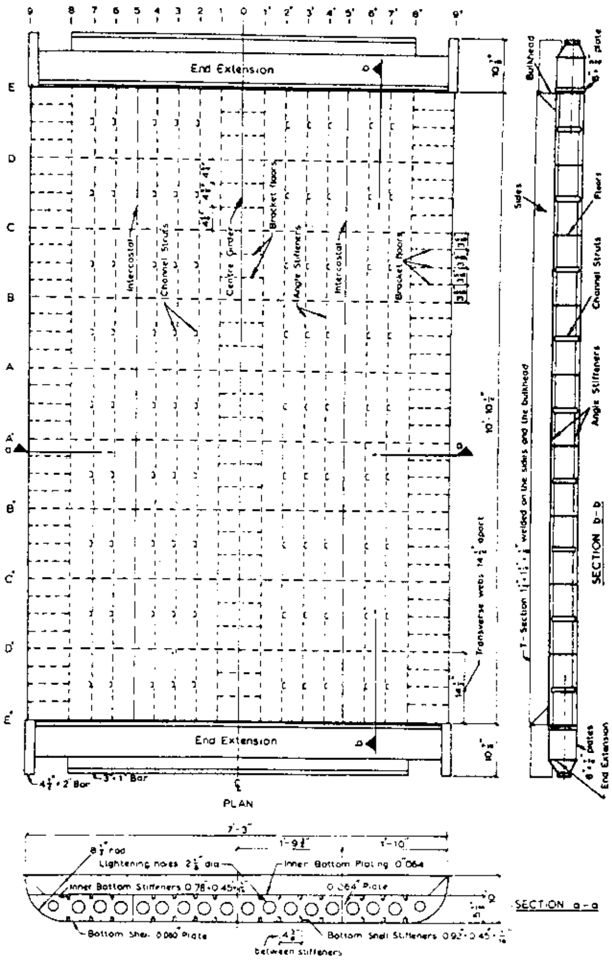


Figure 19

For the measurements on the deck forward of the mid-house, the correlation is slightly better than in the midship section area and the effect of structural discontinuities is negligible.

Influence of Initial Distortion

Initial Distortions. Initial distortions always exist in fabricated structures due to welding contraction and to fabrication imperfections. They have a significant effect on the structural behavior of plating. Studies have been made on the behavior of a typical double bottom structure, with tests on a one-eighth scale steel model, Figure 20, and an overall analysis using orthotropic plate theory with shear deformation (11). Initial relative distortions due to welding were present over the central region of the outer shell, as shown in Figure 21. The ratio of distortion to plate thickness was of the same order as occurs in practice.



Principal Dimensions of Double-Bottom Model

Figure 20

A comparison of theoretical and experimental values for the overall behavior for a uniformly distributed normal load is given in Figure 22; it shows generally good agreement, with the differences attributed mainly to local deformation and approximations in the computation of loss of effectiveness. For local behavior of outer shell panels, Figure 23 shows the initial deformations and inplane loads used in the theoretical solutions, based on contours measured prior to any testing (Figure 21). A comparison of theoretical and experimental values for the same uniformly distributed normal load is shown in Figures 24 and 25 for bending stresses and membrane stresses respectively.

The most significant source of error is that the initial deformation of adjacent panels is not antisymmetric as assumed in the theoretical solution. This is apparent in the behavior of panel 2, where the effects of the much larger deformations in panel 3 have swamped the flexural behavior of panel 2 to such an extent that the theoretical solutions for deflections and bending stresses are of the wrong sign. This factor also probably contributed to the lower measured values of transverse membrane stress in panels 1 and 2. In the latter case, the panel on the opposite side of the center girder had suffered significantly greater initial deformation.

A second major source of error was the restriction in the theoretical solution with respect to the symmetry of initial deformations within a panel; this applies mainly to panel 3 in which the effect of concentration of initial deformation toward the floor A' end, which can be seen in the contours, was reflected in the lower measured membrane stresses in this region.

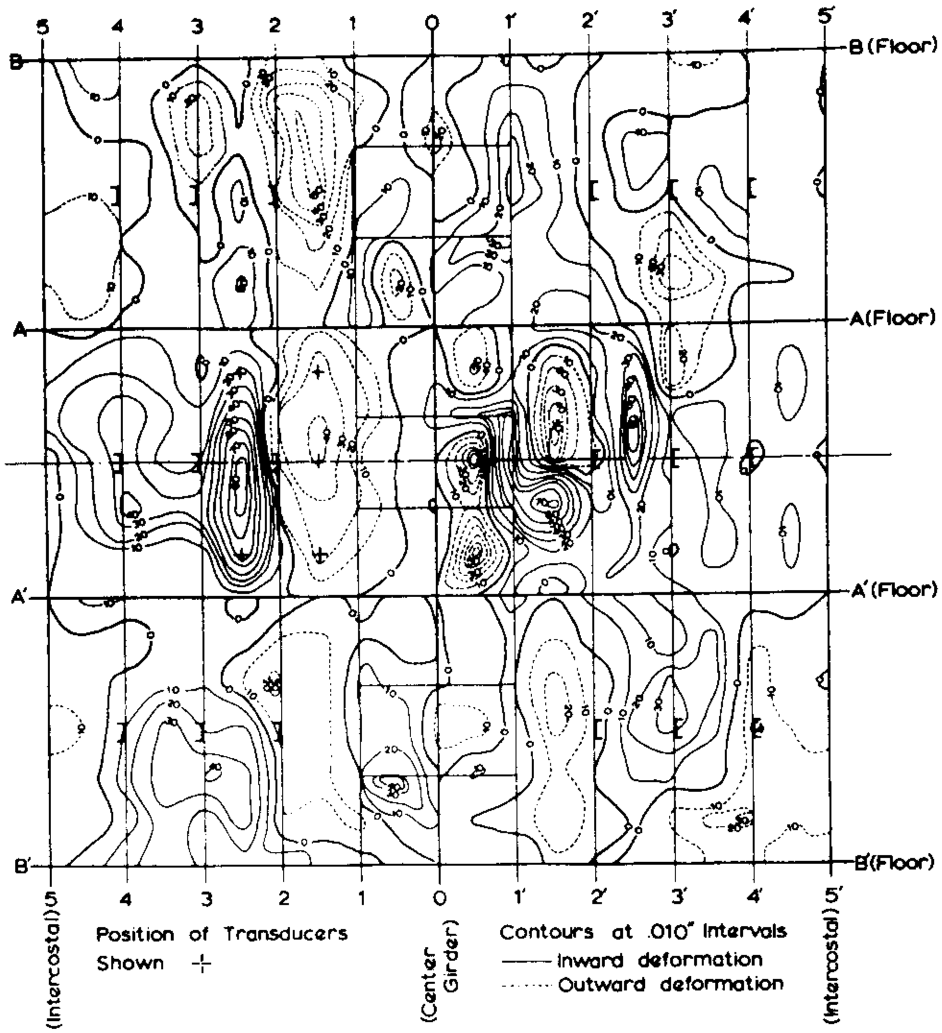
The measured model deflection agrees closely with that calculated by orthotropic-plate theory, provided that allowance is made for shear deflection and for loss of plate effectiveness due to shear lag and out-of-plane deflection. The following comparison shows the deflections relative to an arbitrary value of 100 assigned to the measured deflection:

1.	Measured deflection	100
2.	Calculated deflection neglecting shear deformation and loss of effectiveness	63
3.	Calculated deflection including shear deflection, but neglecting loss of effectiveness	79
4.	Calculated deflection including shear deflection and loss of effectiveness	105

Panel stresses can be estimated by large deflection theory, using in-plane boundary forces derived from the overall analysis. Owing to irregularities in initial deformations and to uncertainties in boundary conditions, the agreement with the measured stresses is less satisfactory than that for overall behavior, differences of the order of 30% being noted at many locations. If the overall stresses are calculated according to 4. above, then noticeably better agreement is found by using large deflection theory for determining the panel bending stresses rather than by using small deflection theory.

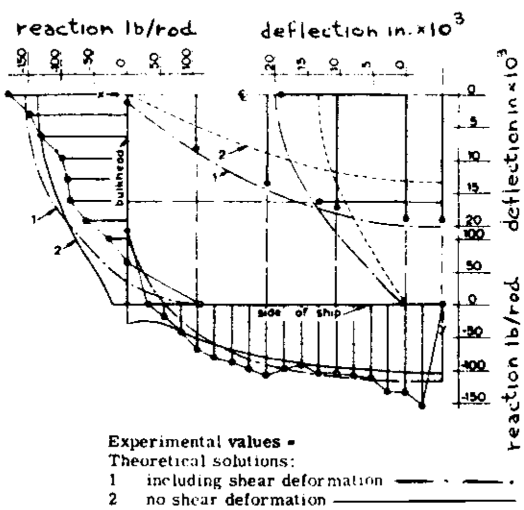
The model was also tested to failure, and it failed at an applied inplane longitudinal compressive stress of 10.4 ton/in^2 combined with a distributed normal pressure of 13.5 lb/in^2 on the bottom plating; this combination corresponded respectively to 1.7 and 1.2 times the load calculated to cause equivalent surface stress yielding of the outer shell plating according to analyses incorporating large and small deflection plate theory.

Contours Are in Mils



Contours of Initial Lack of Flatness over Center Panel of Outer Shell

Figure 21

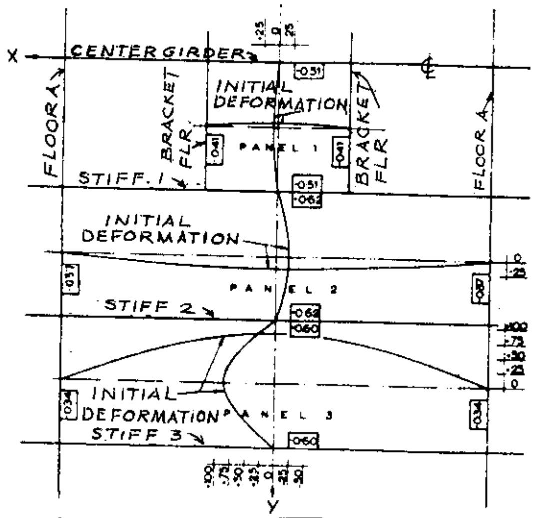


Deflections and Reactions

Figure 22

The following ultimate load comparisons show the loads relative to an arbitrary value of 100 assigned to item 1. below.

1.	Load to cause surface yield as determined experimentally	100
2.	Observed failure load	200
3.	Load to cause surface yield as determined by orthotropic plate theory neglecting shear deformation and loss of plate effectiveness, with panel stresses given by small deflection theory	170
4.	Load to cause mid-plate yield in case 3. if local plate bending is neglected	218
5.	Load to cause surface yielding as determined by orthotropic plate theory including shear deformation and loss of plate effectiveness, with panel stresses given by large deflection theory	115

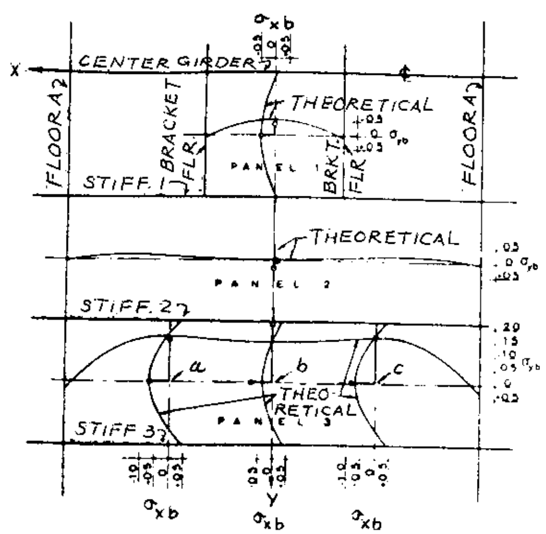


Initial deformation in inches × 10^3 (positive inwards)

Mean in-plane edge stresses, shown $\boxed{-0{\cdot}60}$, in tons/in^2 computed from overall solution including shear deformation (1) for q = 1 lb/in^2

Initial Deformation and Loading used in Outer Shell Panel Analysis

Figure 23



Stresses in Tons/in^2
Experimental values shown •

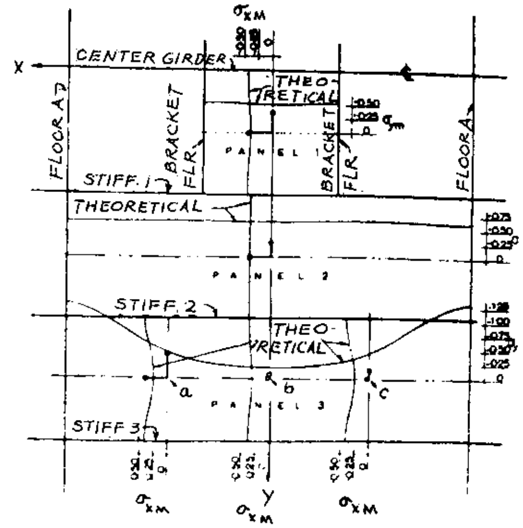
Bending Stresses at Outer Surface of Outer Shell Panels

Figure 24

6. Load to cause mid-plate yield in case 5. if local plate bending is neglected 210

It should also be pointed out that the ratio between yield load and failure load cannot be expected to be the same for all bottom structures.

A comparison with an overall analysis using a finite element model was carried out, with good agreement. However, generally available programs do not allow for local imperfections, so that the local stress results will not have much relevance.



Stresses in tons/in^2
Experimental values shown

Membrane Stresses in Outer Shell Panels

Figure 25

In a fabricated structure, distortions are usually caused by the manufacturing process, and in modern ship structures they are generally associated with residual welding stresses.

Ignoring these stresses, an initially deformed plate loses stiffness immediately as load is applied. Thus the efficiency of a plate, defined as the ratio of the total contraction of a perfectly plane plate to that of a deflected plate, is continually reducing as load increases, even before buckling occurs. This loss in efficiency is small until buckling loads are approached, particularly in plates which are longer than they are wide. Even for wide plates, it has been shown, using an elastic analysis, that, in general, initial deflections do not lower the efficiency of plating significantly, unless the plating is very thin. It has been suggested that the initial deflection should not exceed 0.3t if significant loss in efficiency is to be avoided (12).

An investigation for simply supported square plates in aircraft structures (13) showed the following:

1. As expected, the effects of initial deflection upon buckle growth and effective width are most marked near the theoretical flat-plate critical stress.

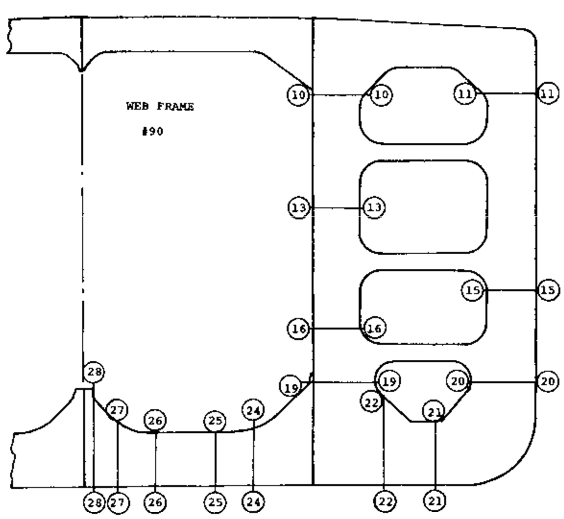
2. At stresses well below the critical stress, the behavior of the plate is very much the same as for an initially flat plate.

3. The effective width is at all values of stress less than that of an initially flat plate.

If we assume, pessimistically, that for a long simply supported plate the initial deflections occur in asymmetrical waves whose half lengths are equal to the plate width (that is, they are in the lowest buckling mode), then the foregoing square-plate conclusions will apply also to the long plate. Loss of effectiveness of plating is approximately proportional to the square of the initial deflection. With the random ripples, which usually occur in welded ships, or with one single lobe, the loss in effectiveness is appreciably reduced and is small in longitudinally stiffened ships. The effect of initial deflection on the maximum end load that a plate can carry has been examined recently, and the experimental results show good agreement with the pertinent theories and demonstrate an appreciable reduction in maximum load capacity, even for very small initial deflections. For example, an initial deflection of only t/20 in a mild steel plate having b/t = 50 appears to lower the maximum average plate stress by about 15%. (t is the thickness and b the width of the plate panel).

Measurements taken in Great Britain in 1965 on typical areas of frigate bottom plating in dry docks (14) show that the average deflection was 0.30t or 0.005b in the least fair frigate, and 0.11t or 0.0024b in the best. Maximum deflections were generally about three times these values, with the very occasional large local depression of about 1.5t in the worst frigate near a welded seam.

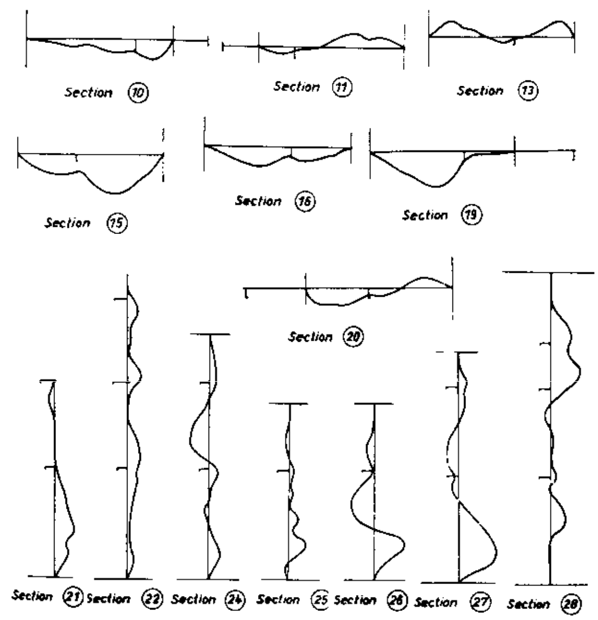
Another investigation of stress distributions in a large tanker (8) showed differences between stresses calculated by a finite element computer program and stresses deduced from strain measurements on the actual ship. Location of the various sections and some of the measured deformations are shown in Figures 26 and 27 respectively. Stress measurements at a typical section are shown in Figure 28. The correlation between theoretical and experimental results was not altogether satisfactory, due to insufficiently narrow mesh patterns in the two-dimensional finite element model and, more significantly, due to the initial deformations which occurred during construction.



LOCATION OF SECTIONS

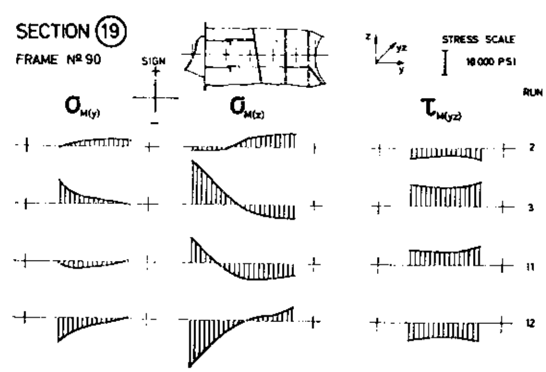
Figure 26

Predeformation Scale is
50 Times Drawing Scale



MEASURED PREDEFORMATION WEB FR.90

Figure 27



Measured Stresses

Figure 28

Influence on Buckling Strength. The influence of fabrication imperfections on the buckling strength of a structure is difficult to assess, due to the unpredictable nature of imperfections. At the present time, it seems reasonable to assume that a small initial deflection of a plate panel, say less than one half of its thickness, would not change the elastic buckling behavior of the panel. A similar conclusion can be drawn for a structural member or section. However, the load-carrying capacity in the post-buckling state of a plate panel with initial deflections will be significantly reduced if material yielding is chosen as a failure criterion. This can be seen from the stress redistribution in a deformed web plate shown in the previous section.

If the initial deflection is larger than the thickness of the plate, the deflection will increase with the increase of the compressive loads and the procedure of calculating buckling strength is no longer applicable.

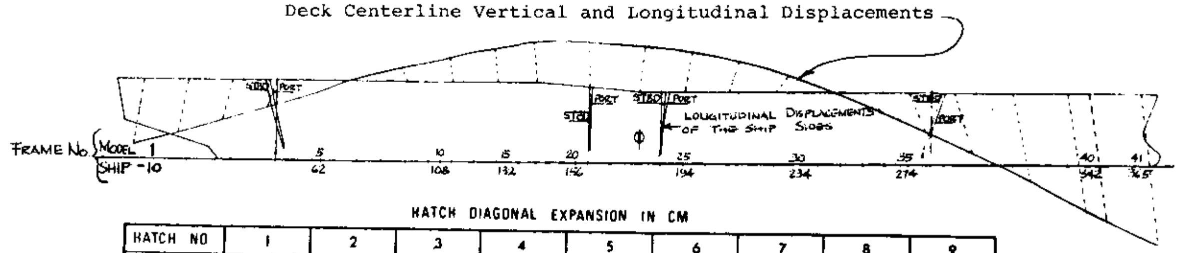
Influence on Structural Reliability. It is usually implied in any specification or design code that safety factors are introduced to provide adequate coverage against structural failure. The safety factors in most design rules have been developed in an evolutionary manner over relatively long periods of time, assuming an ideal structure which usually has not been optimally designed. Where fabrication imperfections exist, the anticipated safety factors will be lower but still seem satisfactory in most cases (because of overdesign rather than efficient design). In the classical approach to ship design, the beam, column and plate theories were used in a piece-by-piece design of the hull girder. These methods did not allow for a simultaneous occurrence of failure modes. Now with the introduction of automated ship design procedures, many constraints applied in the past are waived and the hull structure can be designed by more powerful tools. Thus, ship designers are more apt to lean toward structural optimization than before, and the applicability of the existing safety factors is very much in question. The answer to this question is in the use and development of reliability analysis. An important factor in the reliability analysis is the determination of the level of failure and type of failure mode. The reliability analysis may be developed for predicting yielding, large deflections, cracking, instability, collapse, dynamic response, etc. The same is true for material strength properties, such as fatigue life, ultimate strength and creep, which are being described probabilistically. In other words, it is important to know the true factor of safety involved in the design of ships subject to certain probabilistic loadings and design conditions. A rational procedure in allocating the structural cost is to consider the probability of the occurrence of, and the likely damage cost associated with, each failure level.

Interpretation of Computer Results - Examples

Overall Hull-Girder Responses. The entire hull structure of a container vessel was analyzed by the DAISY system of finite element computer programs (15). The ship was placed in oblique quasi-static regular waves and subject to combined vertical, lateral and torsional loads. Deck displacements are shown in Figure 29, where the top picture shows the vertical and longitudinal displacement components of the main deck centerline and the longitudinal displacement of the ship's side lines at selected frames. In the first curve, the vertical component of the displacement is due to pure longitudinal vertical plane bending of the ship hull girder. The longitudinal component is due to both vertical and torsional deformation of the hull. The second group of curves shows the resultant displacement due to torsional warping and lateral bending deformation of both deck side lines of the ship. It is clear that the longitudinal displacements of both sides are almost negligible near midship.

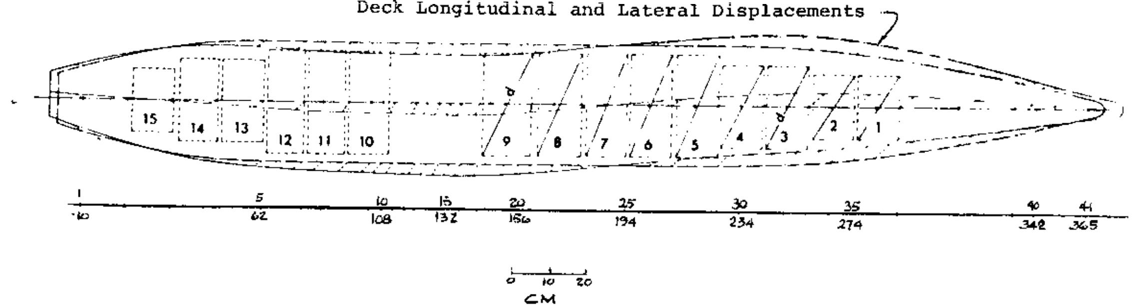
The bottom picture shows the displacements for the upper deck at the centerline and the ship's sides. The distortion of hatch diagonals has been calculated, and the initial diagonal lengths for the idealized structure are tabulated. The maximum distortion is found to be at the second hatch opening forward of the engine room. The deformation gradually decreases toward the forward hatch.

Figure 30 shows the computed longitudinal and shear stresses in the deck and side shell platings between Fr. 188-192 due to wave-induced vertical moment and shearing force. The top diagram shows the distribution of loadings along the length of the vessel. It is interesting to note that the longitudinal stresses computed by means of both the finite element techniques and the elementary beam theory are in good agreement. This seems to confirm the validity of the simple beam approach for calculating the hull girder bending stresses for this type of vessel. The agreement is not as good for shear stress distributions, which may be attributed to local bending not accounted for in the simple beam approach.



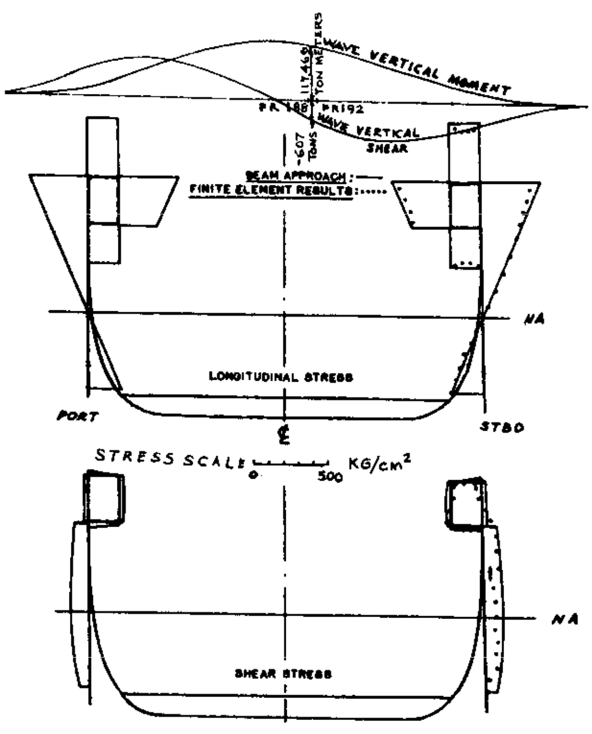
HATCH DIAGONAL EXPANSION IN CM

HATCH NO	1	2	3	4	5	6	7	8	9
d	2048.197	2046.749	2494.283	2494.189	2969.781	2969.800	2965.790	3036.431	3036.469
EXPANSION	0.382	0.634	0.896	.125	1.510	2.056	1.989	2.413	1.695



OVERALL DISPLACEMENTS OF DECK AND FRONT HATCHES DIAGONAL EXPANSIONS

Figure 29



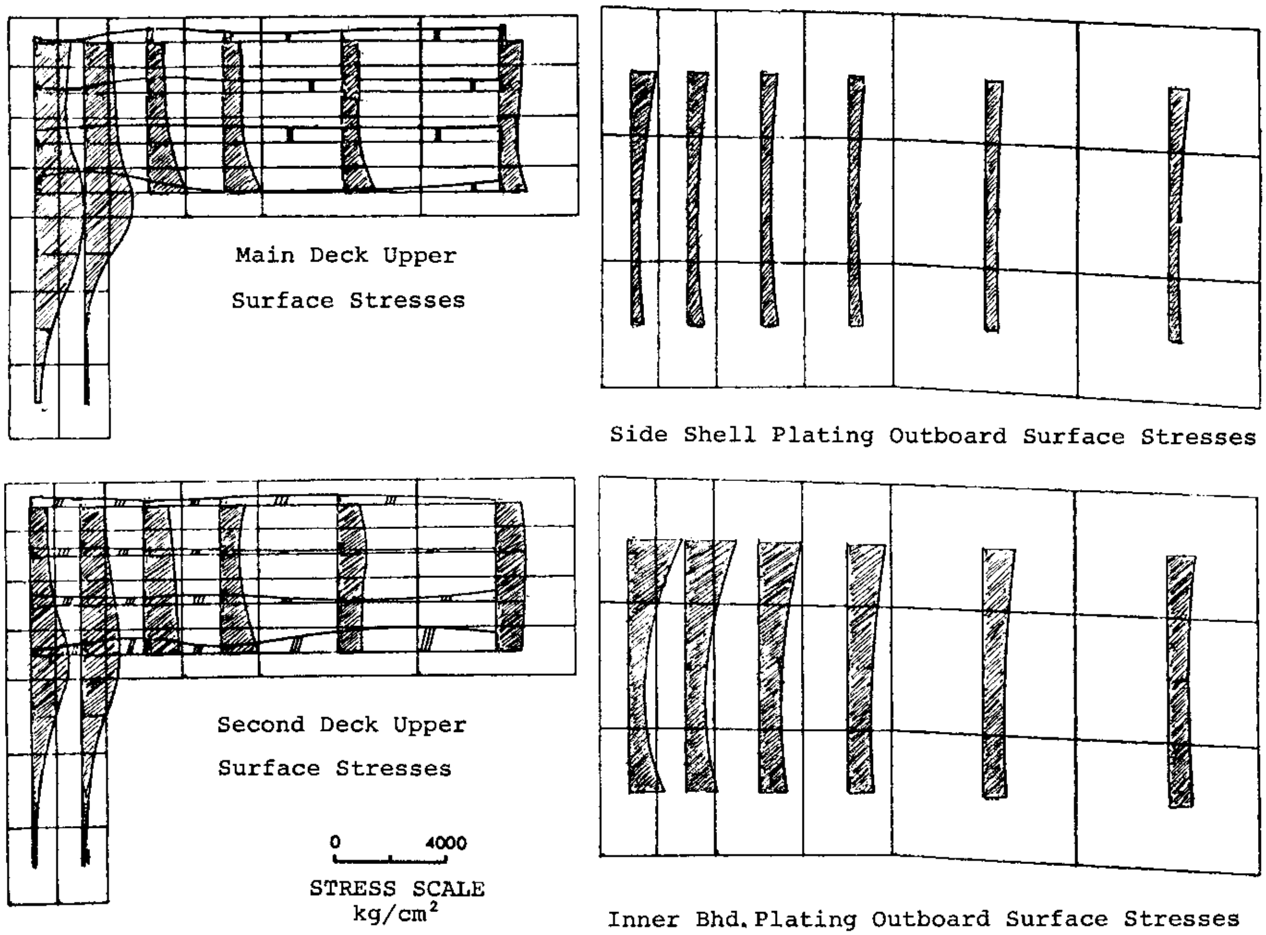
STRESS DISTRIBUTIONS DUE TO WAVE INDUCED VERTICAL MOMENT AND SHEARING FORCES AT FR 190

Figure 30

Figure 31 shows the longitudinal stress distribution of a substructure along the port and starboard sides. The restraint produced by the engine room housing is the cause of high stress magnitudes in the port wing box in the deck area, where the stresses on the inner bulkhead plating are higher than those of the shell plating; for the starboard wing box, the opposite is true. This is attributed to the effect of secondary lateral bending in the wing boxes. The results of the fine mesh model confirm this observation and enable us to detect the region of highest shear stress near the hatch corner circular cutout.

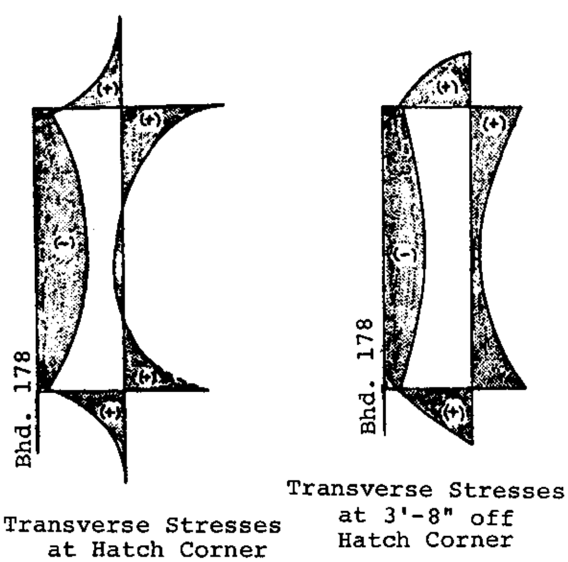
Figure 32 shows the stress distribution around two sections of the transverse box, obtained as the resultant of symmetrical and anti-symmetrical components. Figure 33 shows the antisymmetric transverse stress components at a section of the transverse box, together with a straight line approximation.

The DAISY system was also used to analyze a vessel carrying liquid cargo. In such vessels, stress increases in the hull girder plating, in addition to the normal expected secondary or tertiary stresses, are sometimes caused by a particular loading pattern. Stress concentrations usually occur where ballast tanks are either full or empty and are surrounded by empty or full cargo tanks. Figure 34 shows a loading pattern with the vessel in a heavy ballast condition on a sagging wave and the corresponding stress patterns. In the bottom plating outside the loaded ballast tank, section I-I, the stress distribution is almost linear. Inside the ballast tank, section II-II, there is an almost 25% increase in longitudinal stress in way of the bilge, due to the horizontal pressures of the end bulkheads of the ballast tanks causing an additional local tension in the vicinity of the bilge.



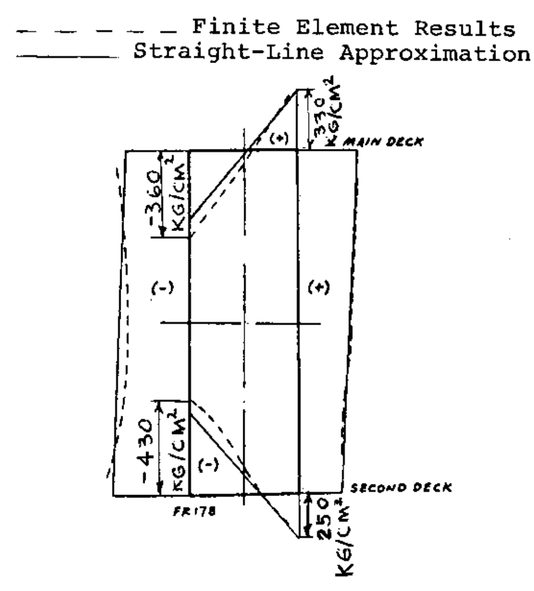
LONGITUDINAL AND TRANSVERSE STRESSES SUBSTRUCTURE FRS 142-150, PORT SIDE

Figure 31



TRANSVERSE BOX AT FR 178, PORT SIDE

Figure 32



ANTISYMMETRIC STRESSES ALONG TRANSVERSE BOX AT FR. 178 - PORT SIDE

Figure 33

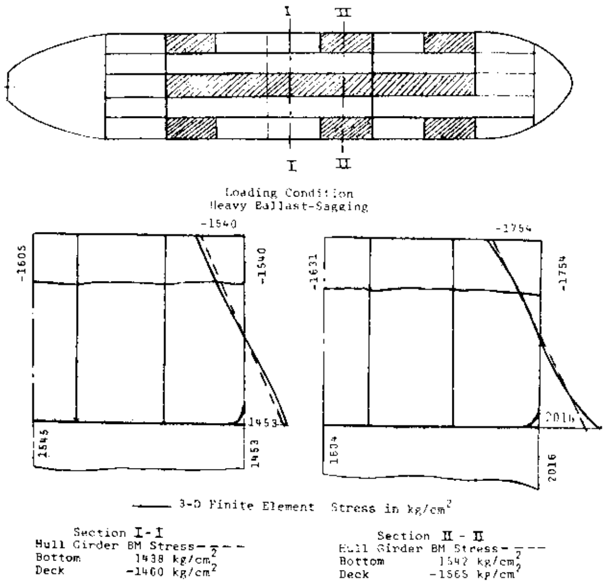


Figure 34

<u>Stress Concentrations</u>. Stress concentrations or stress raisers are usually due to local irregularities in the structure, discontinuities or abrupt changes in relative stiffness between adjacent structural members. The corresponding maximum stress intensities can be evaluated using finite element structural analysis, photoelastic analysis and/or measurements. Stress concentrations can be divided into three categories:

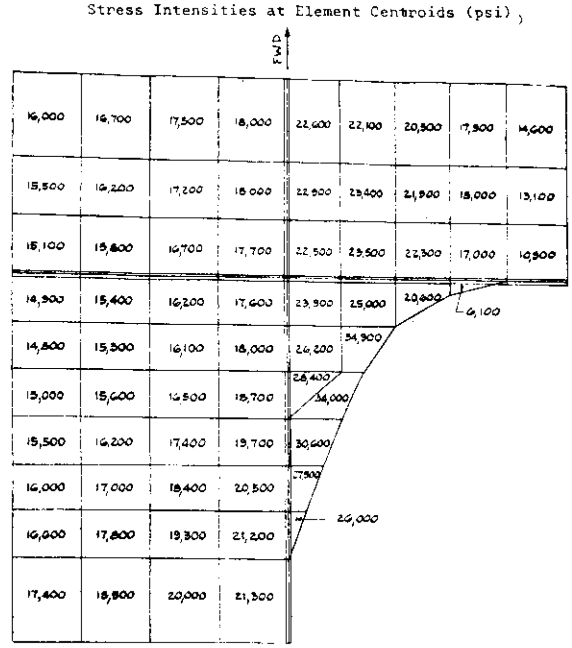
1. Stress concentrations due to imperfect fabrication.
2. Stress concentrations resulting from unavoidable structural discontinuities.
3. Stress concentrations introduced by improper design of connection details.

Stress concentrations due to imperfect fabrication are introduced during fabrication (misalignment, surface scratches, notches in plate edges or circular shafts, built-in stresses, etc.) or are caused by local corrosion or local damages (caused by collision, etc.). These stress concentrations can usually be corrected by preventive maintenance or elimination of the contributing factors.

Stress concentrations resulting from a design constraint or product condition have been generally ignored, since their repetitive use and application has shown that under static loading conditions no danger of rupture or failure can be expected. High stress concentrations in this category may occur at the edges of rivet or bolt holes, keyways in circular shafts or fillet radii in circular shafts with a change in diameter. They should be investigated when fatigue is involved, as in the case of machinery components.

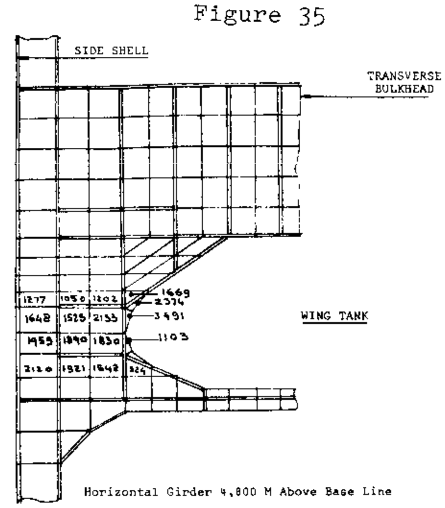
Figure 35 shows a typical example of stress concentration around a hatch corner opening of an LNG carrier, occurring as part of a necessary design condition. The results obtained from a finite element analysis show a stress concentration factor of about 2.5, with a maximum stress intensity of 34,900 psi, which is the yield strength of the material. (The actual stress will be lower due to plastic yielding). Since the nominal stress in that region is governed by the overall characteristics of the hull girder, a local increase in plate thickness would not result in any significant reduction in stress. The only way to decrease this stress concentration is to reduce the nominal stress in the deck plating, which would mean a substantial increase in material to increase the deck section modulus.

Stress Intensities at Element Centroids (psi)



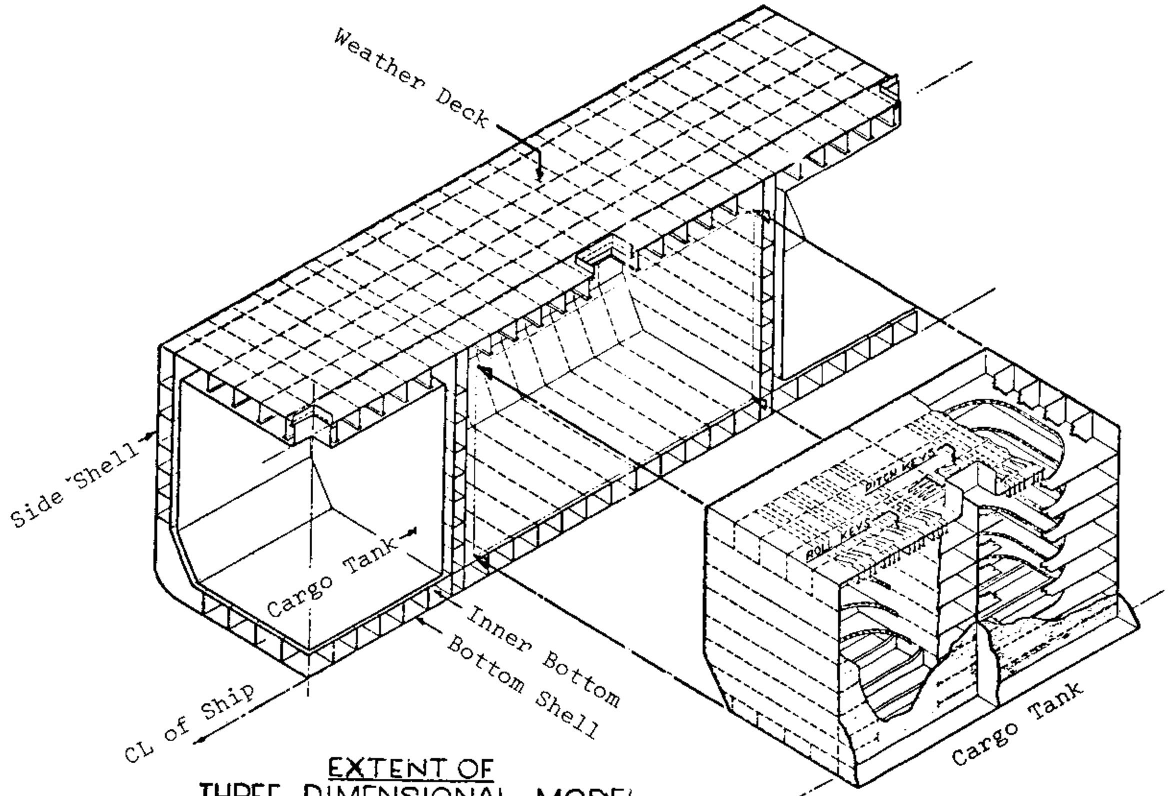
Stresses around hatch corner

Figure 35



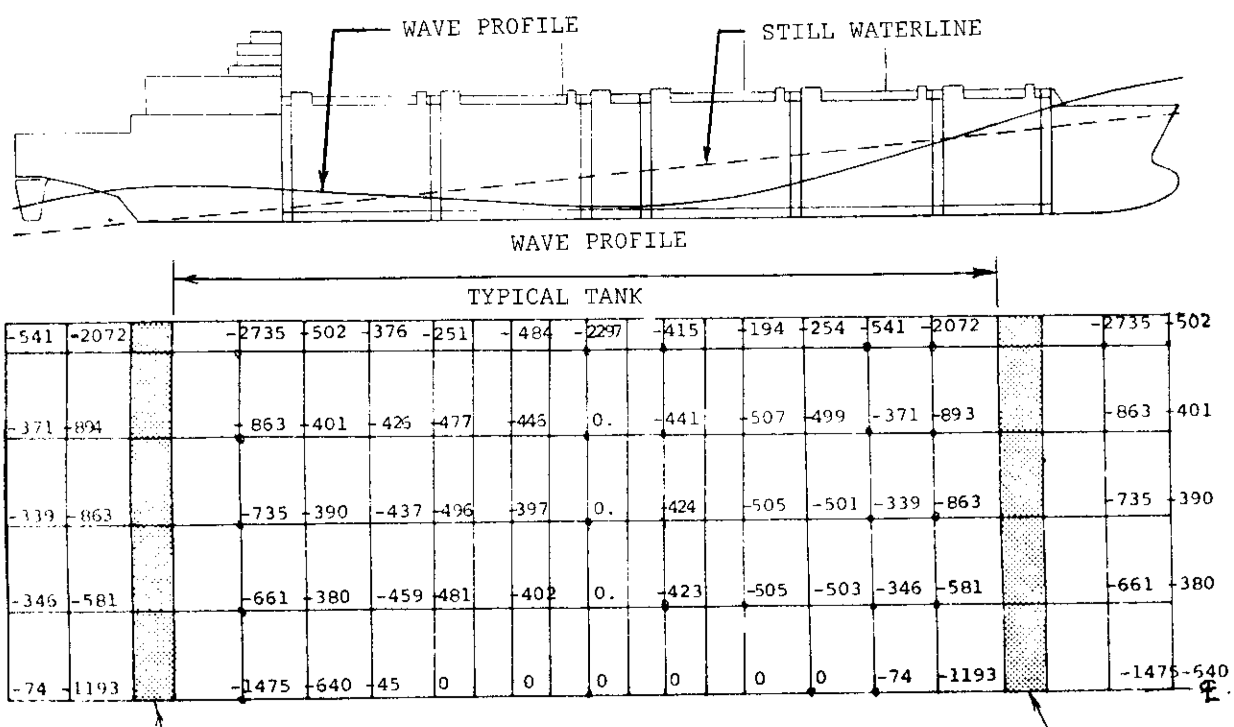
Stress Intensities at Element Centroids (kg/cm²)

Figure 36



EXTENT OF
THREE DIMENSIONAL MODEL
FOR INDEPENDENT TANK L.N.G.
CARRIER

Figure 37



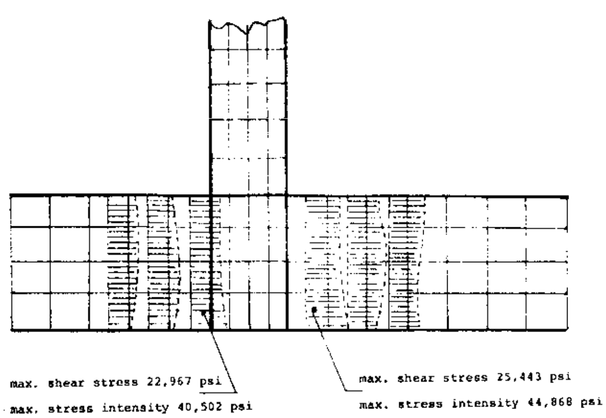
MODEL NODAL LOADS IN KIPS

Figure 38

Practical experience with a great number of vessels has shown that, with proper fabrication of the radius corner, avoiding notches by grinding the plate edges smooth will result in satisfactory service without failure. It should be pointed out that for ductile metals the probability of failure is much smaller than for brittle materials.

Another typical detail in ship structures, where stress concentrations are present is shown in Figure 36, with a stress intensity of 3491 kg/cm^2 partly due to the geometry of the triangular finite element. A stress concentration factor of about 2 is to be expected for such details.

Stress concentrations in this category are sometimes caused by non-uniform, unfavorable load distribution on certain major components of a ship structure. A typical case of an LNG carrier with independent tanks, Figure 37, shows an exploded view of the ship structure with the tank removed. The design of the containment system included a centerline longitudinal bulkhead and a complete transverse bulkhead at midspan of the tank. Figure 38 shows a seagoing condition for the fully loaded vessel encountering a head-on wave in sagging and the corresponding load distribution on the inner bottom of the vessel. It clearly shows that the inner bottom and tank bottom are not in contact in the middle of hold, and load concentrations are apparent at the corners and at both ends of the centerline bulkhead of the independent tank. Figure 39 shows the stress concentrations caused by these loads imposed on the centerline girder of the vessel. Since the maximum stress intensity of 44,860 psi is mainly due to excessive shear stress, an increase in plate thickness will reduce the stresses almost proportionally.



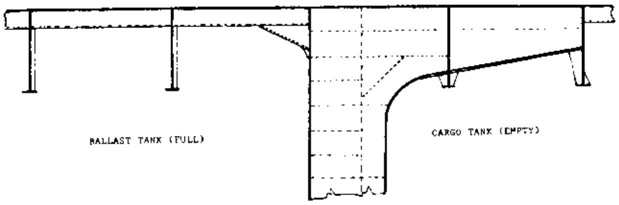
MAXIMUM STRESSES IN CENTERLINE BULKHEAD

Figure 39

Stress concentrations introduced by improper design of connection details are exemplified by Figure 40, showing the structural detail of a shell longitudinal connection to a horizontal oiltight bulkhead girder. This detail has been suspect for some time because of so-called "nuisance cracks".

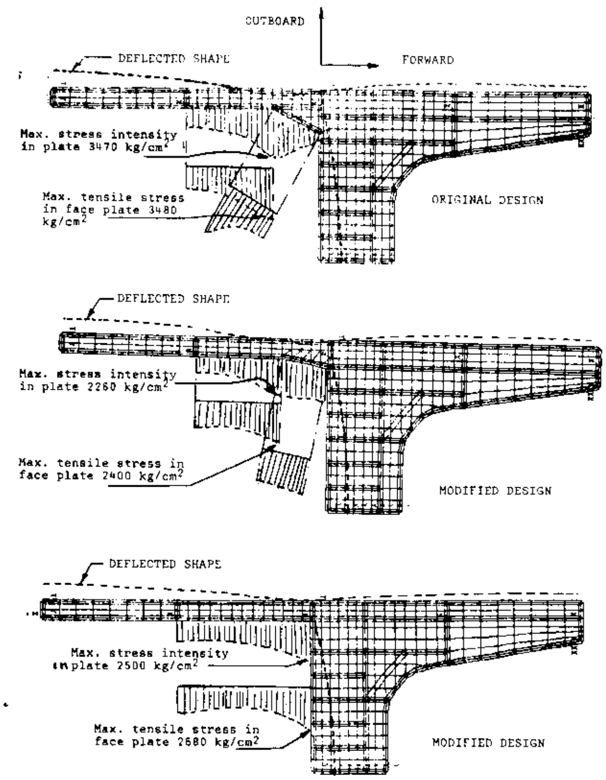
A detailed investigation using finite element analysis shows that, in most cases, these stress concentrations can be eliminated with minor design changes, and therefore belong in a separate category. The stress concentrations are due to local load concentration from horizontal girder loadings, improper bracket detail and discontinuity of face plates.

The critical loading condition analyzed for this particular detail was for the vessel ballasted in a sagging wave with the aft wing ballast tank full. The nominal load or stress in the shell longitudinal midway in the ballast tank would be the sum of the hull girder bending moment stress, the secondary bending stress of the cargo and hydrostatic load on the longitudinal plus a stress due to the local axial force from the horizontal bulkhead girder load. At the connection of the shell longitudinal and horizontal bulkhead girder, additional shear and bending moment are caused by the rotation of the horizontal girder, and some increase in stiffness is required at the connection to the oiltight bulkhead. Figure 41 shows the deflections and stress distributions for three different end connections. Since the stiffness of the horizontal girder is large in relation to the stiffness of the side longitudinal, the rotation and deflection of the horizontal girder in way of the side shell is not influenced by the stiffness of the longitudinal end connection. From the top picture of Figure 41, it can be seen that the larger bracket as originally designed causes more deflection in the shell longitudinal at the bracket toe, resulting in stress concentrations of almost 3500 kg/cm^2. Reducing the size of the bracket as shown in the middle picture and lining up the faceplate with the stiffener on the horizontal girder reduces the maximum stress intensity in the faceplate to 2400 kg/cm^2. The only increase required is a 50% increase in the longitudinal web to reduce the shear stress. The bottom picture shows another solution to the problem, where the stiffener was moved and the web of the longitudinal and face plate were increased locally by 50% to reduce the bending and shear stress, resulting in a maximum tension in the face plate of the side longitudinal of 2680 kg/cm^2, a 25% reduction from the original design. A reduction in the bracket along the horizontal girder is shown to reduce the stress concentrations, while the added material is less than what was used for the original design.



STRUCTURAL DETAIL

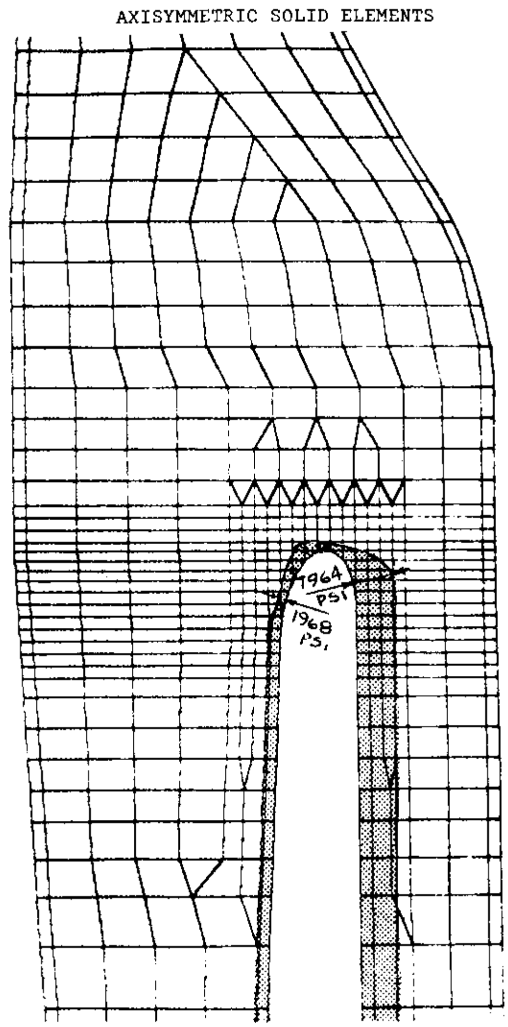
Figure 40



Effect of connection detail on stress concentration

Figure 41

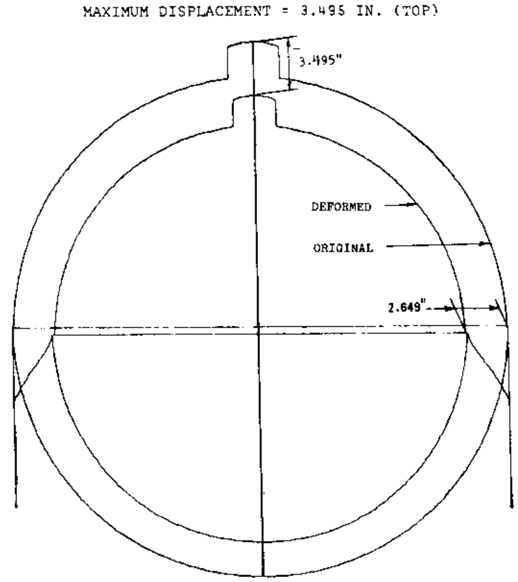
<u>Distribution of Thermal Stresses.</u> A portion of a liquefied natural gas carrier, consisting of the dome, the spherical tank shell and cylindrical supporting skirt, together with one horizontal stiffener near the equatorial ring, has been modeled, using axisymmetric solid elements for a detail investigation (16). A typical section of the fine mesh model near the equatorial ring is shown in Figure 42 for a specific stationary temperature distribution in the tank support, with the thermal stresses in the groove profile, a critical region, shown in the shaded area. The deformation of the entire model due to the temperature differential is shown in Figure 43. A structural analysis of another LNG spherical tank system (17) gives a similar pattern of deformation, as shown in Figure 44.



THERMAL STRESS DISTRIBUTION ON THE EQUATORIAL RING AND SKIRT

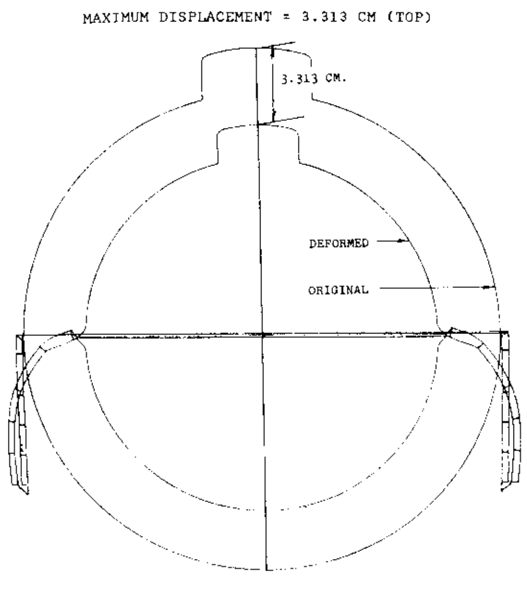
Figure 42

MAXIMUM DISPLACEMENT = 3.495 IN. (TOP)



DEFORMATION DUE TO THERMAL LOADS

Figure 43



DEFORMATION DUE TO THERMAL LOADS

Figure 44

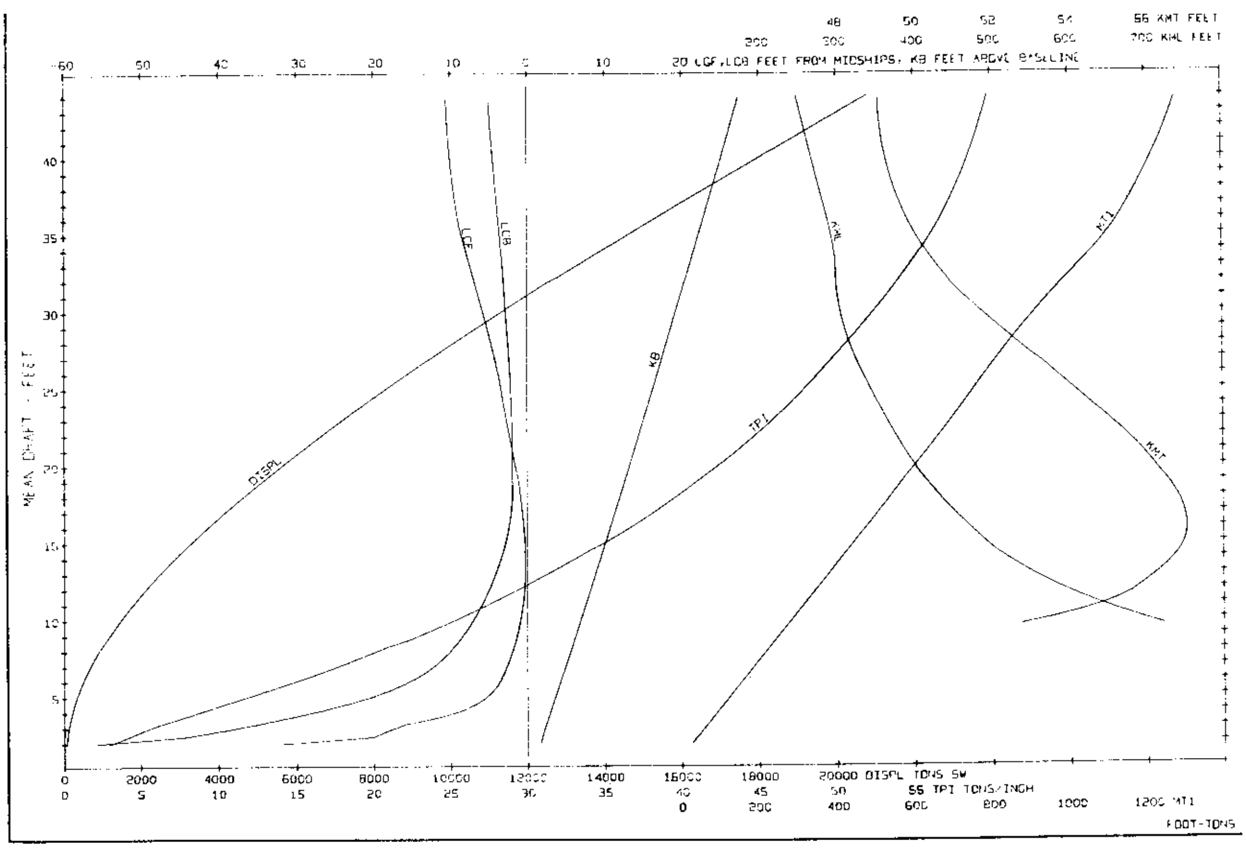
COMPUTER-AIDED DESIGN PROCEDURES

Scope

Computer usage appears in most phases of ship structural design and production. Its sheer "number crunching" powers are fully utilized in engineering applications, such as finite element programs. Its facility for data management and retrieval enabled such computerized systems as AUTOKON, STEERBEAR, CASDOS and others to have been developed for ship production and detailing applications. The ability to produce graphical output via interactive terminals has also received much attention in ship design in recent years. Many papers have been written on the above subjects and it is beyond the scope of this paper to describe in adequate detail the many applications of computer-aided design. Instead, only a few subject areas will be addressed, primarily in the structural design and analysis field.

Ship Characteristics and Stability

Numerous computer programs have been developed over the years to perform various naval architecture calculations such as hull girder shear and bending moment, section modulus, and hydrostatic stability. These calculations all require a description of the hull geometry, usually in the form of offset data. Perhaps the most widely used and comprehensive hull characteristics program is SHCP (Ship Hull Characteristics Program)(18). This



Computer plot of hydrostatic curves

Figure 45

program develops both intact and damage stability characteristics for ship forms by conventional methods. Hull girder shear and bending moments can also be calculated. Computer-generated plots of the body plan, water planes, sectional areas and curves of form, Figure 45, can also be produced.

In addition to comprehensive computer codes such as SHCP, special purpose programs for calculating grain stability, tank ullage, and section modulus are also available to naval architects. As the description of the hull form is a requirement for all of the above calculations, establishment of a common data base for the hull surface geometry is highly desirable.

Design Loads

Static Loadings. The forces acting on a ship consist of its own weight, inertia forces, cargo weight, and sea loads. The ship and cargo weights are well defined. Sea loads are a combination of the hydrostatic pressure of the sea and the dynamic forces resulting from the vessel moving through waves. Ocean waves are difficult to define since they occur randomly in nature. Because of the lack of exact understanding of the nature of real sea waves as well as the vessel behavior in these waves, the design and analysis of ships has traditionally been based on a static calculation. Sea loads were computed for a ship poised statically on a wave profile of its own length, using an empirical wave height that is expected to give stress resultants comparable to what the vessel may encounter in actual operation. In applying this static sea load and performing a finite element analysis, the resultant stresses are considered only as representative stress levels rather than absolute stress values.

Recent advances in the analysis of waves and in defining the response characteristics of a ship in waves has removed many of the difficulties in predicting ship motions in realistic seas. It is now possible to solve the equations of motion of a vessel moving in regular waves with a high degree of accuracy. The response of a vessel in irregular waves can then be considered as the summation of responses to regular waves of different frequencies. A knowledge of the vessel response permits the calculation of the hydrostatic and hydrodynamic pressure distributions along the length of the vessel.

Of greater interest for design purposes is the ability to predict the maximum expected loading on the vessel during its service life. A statistical approach must be taken in order to establish this maximum.

Dynamic Loadings. The dynamic loads experienced by a ship in a seaway can be generally divided into two categories. The first category comprises continuous loads associated with wave-induced forces, bending moments, torsion , and the so-called springing. The second category comprises transient loads in connection with slamming, whipping, etc. A survey and review of various loads imposed on a ship's hull is given in (19) and Professor Lewis' paper presented at this meeting.

The study of wave-induced responses (motion, force, bending moments and torsion) has been made for some time. Theories in this respect can be divided roughly into two groups, i.e. O.S.M. (ordinary strip method) and revised O.S.M. Generally speaking, the differences between these two methods consist of different expressions of wave excitation forces and the inclusion of the end-effect by the revised O.S.M. The computer programs "SCORES", developed at Oceanics, Inc. and sponsored by the Ship Structure Committee (Report SSC-230) and "SEAKEEPING", developed at the Massachusetts Institute of Technology, are two typical examples of the aforementioned methods. Basically, "SCORES" and "SEAKEEPING" are similar in predicting ship responses in regular and irregular waves. However, the scope is somewhat different. "SCORES" with its recent modification calculates, in addition to the other modes, the surge motion approximated by the corresponding result of an ellipsoid. "SEAKEEPING" includes a special routine to handle bulb sections which cannot be properly treated by the commonly used "Lewis" routine, and also computes the non-linear rolling of the ship, as well as the occurrences of slamming, deck wetness and propeller racing.

The basis for predicting the dynamic response in a seaway rests on the assumptions that both the irregular waves and the ship's short-term responses are narrow-banded, that ship responses are time-invariant linear systems, and that the superposition principle is applicable to the prediction of ship responses in irregular seas. With these assumptions, the so-called long-term prediction of ship responses in her lifetime can be made, choosing a certain form of probability density function of short-term ship response characteristics. Many general statistics theories are currently used in the field of naval architecture. In contrast to the narrow-banded process, the technique of extreme values has recently been proposed due to the spectra of many random phenomena observed in practice which cover a certain range of frequencies and often may have several maxima during one cycle, as determined from zero crossings. The above-mentioned principles and procedures in prediction techniques are described in (20).

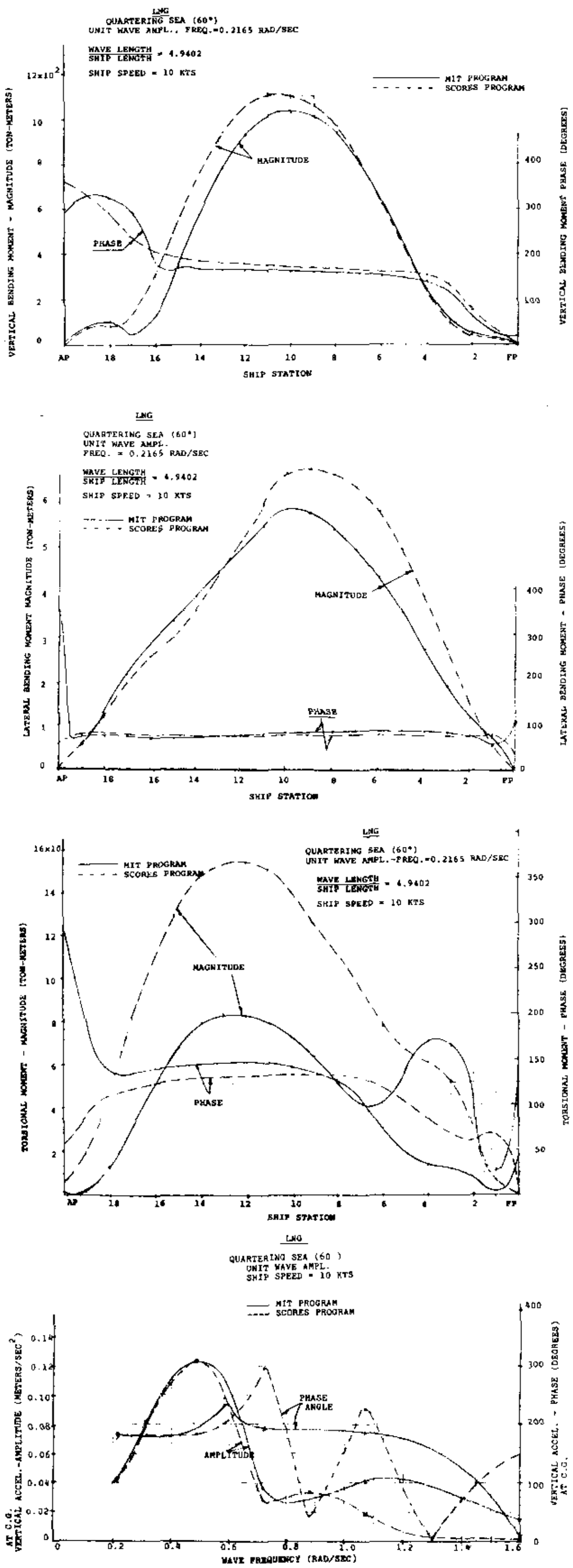


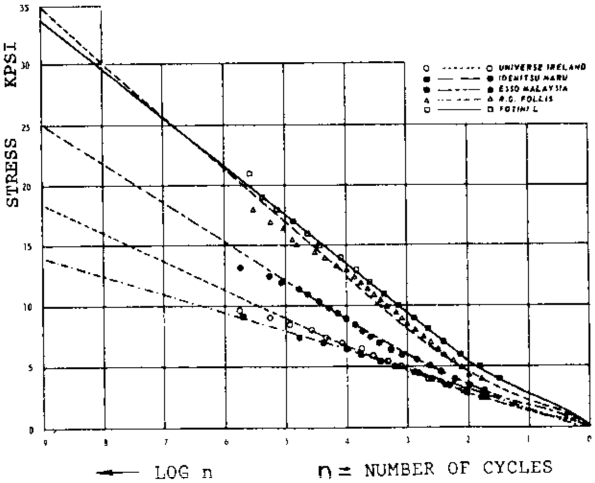
Figure 46

A comparison between two theoretical calculation methods, using the "SCORES" and "SEAKEEPING" computer programs, has been made by the Research and Development Division of the American Bureau of Shipping, with Figure 46 showing samples of comparison. The two different programs yield acceptable results for symmetric motions (but not for asymmetric motions) and generally show better predictions of bending moments and shear forces than of motions.

The spectral density function of waves plays an important role in a ship's long-term response determination. Two different approaches are used: an empirical approach employing an expression for the long-term behavior (Weibull distribution) of the sea as a basis for predictions of long-term behavior of ships at sea, and an approach using the wave spectra representing the sea condition in a certain ocean area, with a normal distribution of short-term Rayleigh parameters in each sea condition.

There are a number of different methods used for long-term predictions. All the classification agencies and regulatory bodies have their own techniques, using the same principles, but with differences in detail procedure. Unlike theoretical predictions of ship responses in regular waves, which can be verified by model tests, the validity of various long-term prediction methods can only be justified by sufficient collection of full-scale measured data. Two such comparisons are shown in Figures 47 and 48. In Figure 47, adapted from (21), data on four tankers and one bulk carrier are compared with an extrapolation based on a probability model assuming a Rayleigh distribution of the measured short-term responses and normal distributions of the Rayleigh parameters in each of five weather groups. When statistical stress data are not available, as in a new design, the SCORES program can be used to calculate response operators from which the short-term responses can be calculated for representative sea spectra. This approach is currently used by the American Bureau of Shipping and represents one of the two typical prediction procedures.

In Figure 48, adapted from (22), data on a cargo ship are compared with a theoretical extrapolation where both the long-term behaviors of the ship and the waves are assumed to follow a Weibull distribution. This kind of approach is another typical technique currently used.



LONG-TERM DISTRIBUTIONS OF STRESS FOR ALL SHIPS IN ACTUAL ALL-WEATHER SERVICE, COMPARED WITH HISTOGRAM DATA POINTS

Figure 47

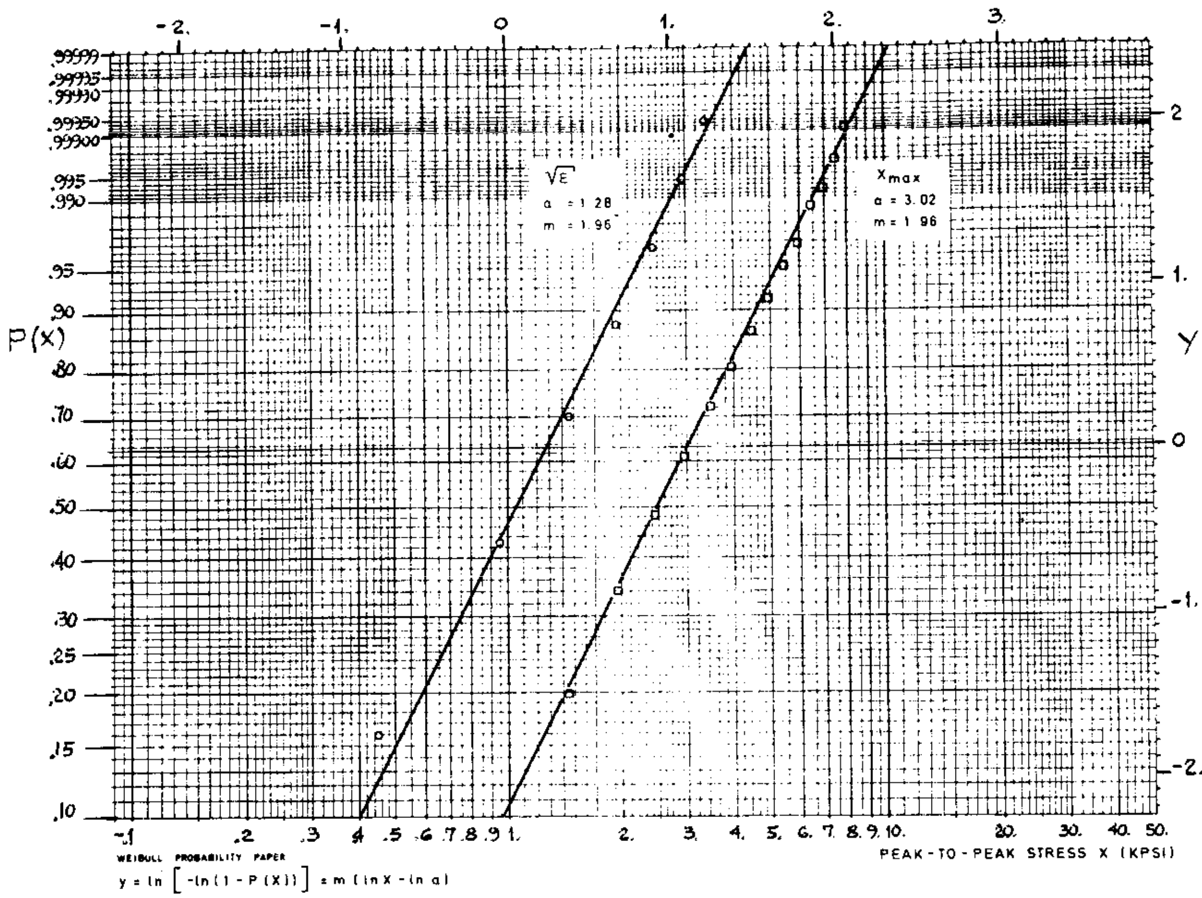
Both approaches fit very well the full-scale data chosen for each comparison. It is quite possible that the Rayleigh-normal distributions may not fit data for small ships and the Weibull distribution may not fit data for large ships as it does for small ones. However, the load prediction should be emphasized for large ships since the design technique for small ships is quite well established.

<u>Pressure Distribution.</u> The recent development of longer, wider ships such as slow-moving bulk carriers and longer, faster ships for the container, LNG and and LPG trades, makes it desirable to use a sophisticated method for both the overall and the local ship structure analyses, in which the ship hulls are characterized by a great number of stiffened plate fields or grillages - the so-called "finite element approach".

E = PARAMETER OF SHORT TERM RAYLEIGH DISTRIBUTION OF PEAK-TO-PEAK STRESS X

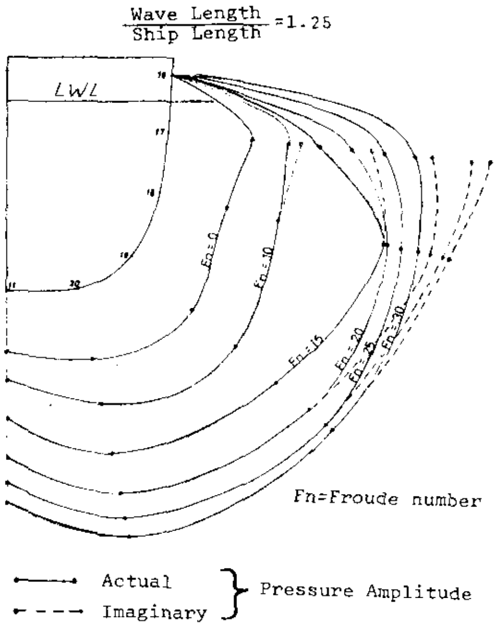
P = PROBABILITY OF OCCURRENCE

a,m = PARAMETERS OF LONG TERM WEIBULL DISTRIBUTION OF $\sqrt{E}$



Long-term distributions of $\sqrt{E}$ and of short-term extremes obtained from full scale data

Figure 48



Dynamic Pressure Distributions
Figure 49

In order to make full use of the advantages of such a mathematical model, the dynamic loads resulting from the waves encountered by the ship in a seaway have to be known, so that the forces at each finite element node on the wetted surface can be approximated. This requires the calculation of pressure distribution on the ship's hull. The study of two-dimensional pressure distributions has been done for many years. More recently, under the sponsorship of the American Bureau of Shipping, Webb Institute developed a technique to determine the pressure distribution, using an extended mapping method, (23), (24), and this distribution is linked with a ship motion program.

All of the aforementioned methods predict the pressure up to the still water-line, due to the linearized theory of infinitesimal waves. Figure 49, adapted from (25), illustrates a typical distribution of dynamic pressures around a forward section 0.15L from the bow of a T-2 tanker model at different forward speeds. It can be seen that the predicted pressure agrees quite well with measured values. Also, in order to make use of theoretically computed results for structural analysis, approximations as shown by the solid lines in Figure 49 have to be used for estimating pressure above the still water line.

Structural Configuration and Scantlings

Classification Rules. Some of the classification societies have developed computer programs based on their rules to assist in the determination of vessel scantlings. A typical computer approach to ship design is ABS/RULESCANT, a system of time-sharing programs that determine scantlings satisfying the ABS Rule requirements (26) for midship sections of oil, ore or bulk carriers. The input typed by the user at a time-sharing terminal consists of the basic scantlings of the vessel, the plate and stiffener scantling values, location of longitudinals, and transverse members. A data base (file) of all the processed ship information is created for use by the many output RULESCANT programs which can be individually selected for execution by the user at the time-sharing terminal.

One of the programs is used to check scantlings of the midship section. The program determines the Rule-required plate thicknesses, stiffener and hull girder section moduli, incremental sectional area required to be added to the deck or bottom to meet Rule requirements, weight per unit length, and the allowable shear stress and shear force. The formulas used by the program in determining the Rule requirements are also listed, as a matter of information to the user. After viewing the results, the user can immediately change any of the input, including the given values of the plating and stiffener scantlings. Then he can rerun the program while at the terminal and obtain answers within minutes of altering the input.

There is also a preliminary design program, which requires input of only a basic definition of the midship geometry. The program then determines the minimum Rule requirements for the midship section. This program can be used to provide quick Rule analysis for preliminary design purposes.

Typical output from RULESCANT follows.

```
RULES FORMULAS USED IN THE PLATE THICKNESS CALCULATIONS

  1: FLAT PLATE KEEL:            PLATE NO.  1
22.19.1 T=0.0003937L(2.6+10/D)  T=      0.745 L=    677.790 D=     52.500
22.19.1 T=0.00331S(0.7DRAFT+0.02(L-164))**1/2+0.1   T=      0.706
 S=     30.000 DRAFT=     38.500 L=    677.790
15.15.2 THTS=(TMS-C)Q+C   THTS=  0.745 TMS=  0.745 C=0.170 Q=1.0000
22.19.3    T=T+0.06  T=       0.805
22.19.7  T=      0.731 REDUCTION=       0.081
22.19.3    T=T+0.06  T=       0.791
```

```
 2: BOTTOM SHELL:                 PLATE NO.  2
22.19.1 T=0.0003937L(2.6+10/D)  T=      0.745 L=   677.790 D=    52.500
22.19.1 T=0.00331S(0.7DRAFT+0.02(L-164))**1/2+0.1  T=      0.706
 S=    30.000 DRAFT=    38.500 L=   677.790
15.15.2 THTS=(TMS-C)Q+C  THTS=  0.745 TMS=  0.745 C=0.170 Q=1.0000
22.19.7  T=      0.675 REDUCTION=      0.075

 3: SIDE SHELL:                   PLATE NO.  3
22.19.1 T=0.0003937L(2.0+21/D)  T=      0.640 L=   677.790 D=    52.500
22.19.1 T=0.00287S(0.7DRAFT+0.02L)**1/2+0.1  T=      0.648
 S=    30.000 DRAFT=    38.500 L=   677.790
15.15.3 THTS=(TMS-C)((Q+2Q**1/2)/3)+C  THTS=  0.648 TMS=  0.648 C=0.170 Q=1.0000
22.19.7  T=      0.675 REDUCTION=      0.075

RULES FORMULAS USED IN THE STIFFENER SECTION MODULUS CALCULATIONS


STIFFENERS   1   2   3   4   5   6   7   8   9  10  11  12  13  14  15  16
22.29.2 SM=0.0041CHSL**2  SM=     86.817 C=1.40 H= 60.500 S=  2.500 L= 10.000
22.29.2 SMCC=.9SM  SMCC=     78.136 SM=     86.817
6.15.3 SM=Q(SM)  SM=       86.817 Q=1.0000
6.15.3 SM=Q(SM)  SM=       78.136 Q=1.0000


STIFFENERS  36  35  34  33  32  31  30  29  28  27  26  25  24  23  22  21
STIFFENERS  20  19  18  17
22.29.2 SM=0.0041CHSL**2  SM=     10.250 C=1.25 H=  8.000 S=  2.500 L= 10.000
22.29.2 SMCC=.9SM  SMCC=      9.225 SM=     10.250
6.15.3 SM=Q(SM)  SM=       10.250 Q=1.0000
6.15.3 SM=Q(SM)  SM=        9.225 Q=1.0000
```

STIFFENERS

BOTTOM LONGITUDINALS

ID	OFF.SM IN**3	REQ.SM IN**3	OFF.SMC IN**3	REQ.SMC IN**3	H FEET	S FEET	L FEET	Q	C
3	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
4	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
5	3923.2	86.8	3666.7	78.1	60.50	2.50	10.00	1.0000	1.40
6	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
7	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
8	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
9	3923.2	86.8	3666.7	78.1	60.50	2.50	10.00	1.0000	1.40
10	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
11	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
12	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
13	3923.2	86.8	3666.7	78.1	60.50	2.50	10.00	1.0000	1.40
14	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
15	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
16	40.2	86.8	39.6	78.1	60.50	2.50	10.00	1.0000	1.40
1	4065.9	86.8	3848.8	78.1	60.50	2.50	10.00	1.0000	1.40
2	40.7	86.8	40.2	78.1	60.50	2.50	10.00	1.0000	1.40

STIFFENER CONTRIBUTION TO HULL GIRDER SECTION MODULUS
MX, MY, IX, IY ARE CALCULATED ABOUT BASELINE
OWN IX IS CALCULATED ABOUT OWN CG PARALLEL TO BASELINE

CORROSION-CONTROL

STIFF	AREA SQ*IN	MX SQ*FT*IN	MY SQ*FT*IN	IX SQ*FT*SQ*IN	IY SQ*FT*SQ*IN	OWN IX SQ*FT*SQ*IN
1	40.14	261.20	0.0	1699.69	0.0	448.34
2	11.20	5.23	28.00	2.44	70.00	0.81
3	11.20	5.23	56.00	2.44	280.00	0.81
4	11.20	5.23	84.00	2.44	630.00	0.81
5	80.28	522.40	802.80	3399.37	8028.00	896.68
6	11.20	5.23	140.00	2.44	1750.00	0.81

STIFF	AREA SQ*IN	MX SQ*FT*IN	MY SQ*FT*IN	IX SQ*FT*SQ*IN	IY SQ*FT*SQ*IN	OWN IX SQ*FT*SQ*IN
7	11.20	5.23	168.00	2.44	2520.00	0.81
8	11.20	5.23	196.00	2.44	3430.00	0.81
9	80.28	522.40	1605.60	3399.37	32111.99	896.68
10	11.20	5.23	252.00	2.44	5670.00	0.81
11	11.20	5.23	280.00	2.44	7000.00	0.81
12	11.20	5.23	308.00	2.44	8470.00	0.81
13	80.28	522.40	2408.40	3399.37	72251.94	896.68
14	11.20	5.23	364.00	2.44	11830.00	0.81
15	11.20	5.23	392.00	2.44	13720.00	0.81
16	11.20	5.23	420.00	2.44	15750.00	0.81
36	32.14	1645.13	80.35	84207.81	200.87	80.88
35	10.87	581.65	54.36	31118.38	271.80	0.91
34	10.87	581.65	81.54	31118.38	611.55	0.91
33	10.87	581.65	108.72	31118.38	1087.20	0.91
32	32.14	1645.13	401.75	84207.81	5021.87	80.88
31	10.87	581.65	163.08	31118.38	2446.20	0.91
30	10.87	581.65	190.26	31118.38	3329.55	0.91
29	10.87	581.65	217.44	31118.38	4348.80	0.91
28	32.14	1645.13	642.80	84207.81	12855.99	80.88
27	10.87	581.65	244.62	31118.38	5503.95	0.91
26	10.87	581.65	271.80	31118.38	6795.00	0.91
25	10.87	581.65	298.98	31118.38	8221.95	0.91
24	32.14	1645.13	964.20	84207.81	28925.98	80.88
23	10.87	581.65	353.34	31118.38	11483.55	0.91
22	10.87	581.65	380.52	31118.38	13318.20	0.91
21	10.87	581.65	407.70	31118.38	15288.75	0.91
20	0.0	0.0	0.0	0.0	0.0	0.0
19	10.87	579.00	461.22	30834.93	19566.06	0.89
18	10.87	575.63	488.19	30476.89	21921.25	0.89
17	10.87	572.25	515.16	30120.95	24410.24	0.89
98	80.28	529.27	3604.92	3489.31	161876.63	16.00
99	80.28	689.41	3781.04	5920.27	178079.75	16.00
TOTAL	868.	18396.990	21216.777	823021.161	709077.024	7034.32

PLATE CONTRIBUTION TO HULL GIRDER SECTION MODULUS
MX, MY, IX, IY ARE CALCULATED ABOUT BASELINE
OWN IX IS CALCULATED ABOUT OWN CG PARALLEL TO BASELINE

CORROSION-CONTROL

PLATE	AREA SQ*IN	MX SQ*FT*IN	MY SQ*FT*IN	IX SQ*FT*SQ*IN	IY SQ*FT*SQ*IN	OWN IX SQ*FT*SQ*IN
1	27.00	0.0	40.50	0.0	81.00	0.01
2	324.61	0.0	7360.13	0.0	208760.19	0.09
3	39.79	299.61	1918.47	2332.59	92494.13	76.43
4	252.18	7083.95	12413.23	226712.06	611019.81	27720.99
5	60.58	2991.29	3104.88	147892.25	138422.13	197.21
6	184.50	9963.00	5581.13	538002.00	175290.31	0.06
7	108.84	5795.75	5061.08	308644.31	236646.38	20.44
8	60.00	300.00	2400.00	2000.00	96000.00	500.00
9	50.63	734.14	2025.21	10986.79	81087.69	341.76
10	43.85	1030.43	1753.92	24511.03	70404.06	295.97
11	40.50	1316.25	1620.00	43051.50	65296.09	273.38
12	43.85	1819.69	1753.92	75813.13	71094.69	295.97
13	48.00	2400.00	1920.00	120255.94	78386.94	256.00
14	77.88	127.27	3161.73	405.34	145082.25	573.11
990	7.31	386.65	365.62	20444.32	18596.17	0.34
TOTAL	1370.	34248.035	50479.817	1521051.268	2088661.820	30551.75

OFFERED VALUES

TOTAL AREA=   4889.918 SQ*IN
TOTAL MX=   116403.694 IN*SQ*FT
TOTAL MY=   0.0   IN*SQ*FT
TOTAL IX=   2402812.291 SQ*FT*SQ*IN
TOTAL IY=   6182457.138 SQ*FT*SQ* IN
N.A. FROM BASELINE 23.805 FEET
N.A. FROM CENTERLINE   0.0   FEET
SM TOP =   83735.750 IN*SQ*FT
SM BOTTOM =   100938.000 IN*SQ*FT
SM PORT =   123649.125 IN*SQ*FT
SM STBD =   123649.125 IN*SQ*FT
WEIGHT PER LENGTH   7.427 TON/FEET
TOTAL LOCAL INERTIA   41402.074

REQUIRED AS OF 1975 RULES

REQUIRED STILL WATER AND WAVE BENDING MOMENTS

6.3.2 MS=CST*L**2*SQRT(L)*B*(CB+0.5) MS=   435329.250 CST=.0002757
L=   677.790 B=   100.000 CB= 0.82000
6.3.2 HE=0.018*L1+11.535 HE=   23.955 L1=   690.000
6.3.2 MW=C2*L1**2*B*HE MW=   652726.563 C2=0.0005723
L1=   690.000 B=   100.000 HE=   23.955
6.3.2 MW=C2*L1**2*B*HE MW=   591700.000 C2=0.0005188
L1=   690.000 B=   100.000 HE=   23.955
6.3.2 MT=MSW+(KB*MW) MT= 1088055.000 MSW=   435329.250
KB= 1.000 MW=   652726.563
6.3.2 FP=10.56-(790.-L)/845. FP=   10.427 L=   677.790
6.3.2 SM=MT/FP SM=   104347.688 MT=   1088055.000 FP=   10.427
6.15.3 SM=Q(SM) SM= 104347.688 Q=1.0000
6.15.3 SM=Q(SM) SM= 104347.688 Q=1.0000
HULL GIRDER SECTION MODULUS REQUIREMENTS
TOP =   104347.688 BOTTOM =   104347.688
Q-TOP =1.0000 Q-BOTTOM =1.0000
MT(OFF)=SMTOP(OFF)*FP MT(OFF)=   873129.875 SMTOP=   83735.750 FP=   10.427
SWBM(OFF)=MT(OFF)-MW SWBM(OFF)   220403.313 MT(OFF)=   873129.875 MW=   652726

TO MAKE OFFERED HULL GIRDER SECTION MODULUS EQUAL TO REQUIRED
SECTION MODULUS-DO THE FOLLOWING:
1. AREA TO BE ADDED TO THE TOP IS   434.630IN**2
THIS IS EQUIVALENT TO AN INCREASE OF THE AVERAGE TOP PLATE THICKNESS
 OF FROM   0.812 TO   1.368 IN
2. AREA TO BE SUBTRACTED FROM (NOT INCLUDING THE F.P.K.) IS   15.468IN**2
THIS IS EQUIVALENT TO A DECREASE OF THE AVERAGE PLATE THICKNESS
 OF FROM   0.750 TO   0.734 IN
MOMENT B.N.A IS   25408.422 SQ*IN*FT FOR HALF THE SHIP
SHEAR THICKNESS IS   1.141 INCHES FOR HALF THE SHIP

SHEAR CALCULATION

1975 SHEAR CALCULATION

FRAME NO. 264 STATION   36.000

USING REQUIRED SHEAR FORCES
6.3.4B FW=K*MW/L1 FW=   1011.166 K= 1.069 MW=   652726.563
L1= 690.000
6.3.4B FSW=5.0*MS/L FSW=   3211.388 MS=   435329.250 L=   677.790
6.3.4A FS=(FSW+FW)*M/(2.*T*I) FS=   3.262 FSW=   3211.388
FW=   1011.166 M=   609802.125 T= 1.141 I=   346004736.000
MAXIMUM STILL WATER SHEARING FORCE= 3211.381 WHEN THE MAXIMUM SHEAR
STRESS= 6.7500 AND ACTUAL SHEARING STRESS INDUCED BY WAVE= 3.2623

FRAME NO. 240 STATION   96.000

```
***USING REQUIRED SHEAR FORCES***
6.3.4B FW=K*MW/L1  FW=      2696.440 K= 2.850 MW=     652726.563
L1= 690.000
6.3.4B  FSW=5.0*MS/L  FSW=       3211.388 MS=   435329.250 L=   677.790
6.3.4A  FS=(FSW+FW)*M/(2.*T*I)  FS=          4.564 FSW=        3211.388
FW=      2696.440 M=    609802.125 T= 1.141 I=      346004736.000
MAXIMUM STILL WATER SHEARING FORCE=  3211.384 WHEN THE MAXIMUM SHEAR
STRESS=  6.7500 AND ACTUAL SHEARING STRESS INDUCED BY WAVE=  4.5643
```

Engineering Analysis. Since the classification rules cannot cover all design aspects in specifics, most of the classification societies will review under special consideration any design supported by rational calculations. Such design may deviate from the published classification rules and yet be accepted if the supporting engineering analysis proves it to be structurally sound. For example, according to the ABS classification Rules (26), alternative arrangements and scantlings will be considered if "they can be shown through ...... a systematic analysis based on sound engineering principles, to meet the overall safety and strength standards of the Rules". The design procedure, in this case, combines both intuition based on past design experience and structural analysis aimed at determining satisfactory structural response.

Nowadays, many computer programs are available for rational engineering analyses (2). These computer programs cover various types of analyses, such as: small displacement, large displacement, incremental plasticity, creep, thermal effects, temperature-dependent material, natural frequencies, mode shapes, transient response and structural instability.

Optimization Techniques

Optimization Techniques. Optimization techniques of ship structures are in their first steps in comparison to ship structure analysis technology. Among the few known optimization methods, optimality criteria and a mathematical programming formulation such as the penalty function approach are most promising at this stage. Some of the areas of development are:

1. Optimization programs based on ABS Rules, such as a web frame structural optimization (27), deal with the weight minimization of a web frame based on stress constraints derived from standard load cases recommended by ABS. Web height, thickness and flange area are design variables in each member of the frame, whereas the effective area of the web flange consisting of the shell, deck or bulkhead plating is treated as fixed user input. A two-dimensional frame analysis program is used in the design procedure.

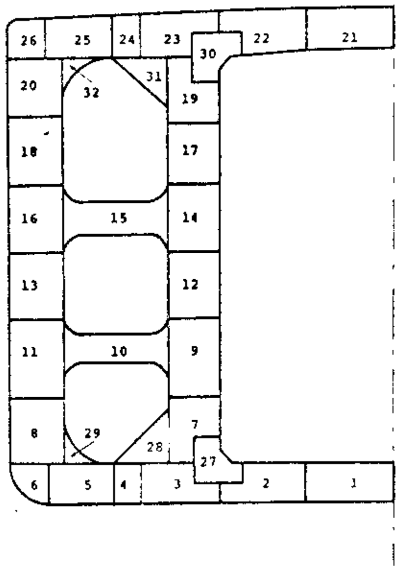
2. Midship section design work being done for the American Bureau of Shipping at the University of Michigan deals with the weight minimization of the hull girder section per unit ship length. About ten design variables are selected to characterize the midship section, such as plate thicknesses of the bottom, deck and side shell, section moduli of the bottom and deck longitudinals and their spacings, stiffener spacing along ship sides and longitudinal bulkheads, and stringer positions.

The ABS Rule requirements are used to act as design constraints, such as maximum and minimum plate thickness, maximum and minimum stiffener size, etc.

3. Other optimization work includes a midship section optimization program (28), oriented toward the design of the longitudinally effective structure of a cargo ship, conforming to the classification Rules established by ABS. The program is capable of handling any structural material in the midship section. Combinations of framing systems (transverse and/or longitudinal) are possible within the panels created by the decks, longitudinal bulkheads, inner bottom, bottom shell, and the side shell sections between decks. Material combinations are varied as zones throughout the ship cross section. In addition to the longitudinally effective material, the program sizes transverse structural framing, consisting of web frames for longitudinally framed sections and transverse deck and/or shell stiffeners for transversely framed sections. Also included in the design are bottom floors and reverse frames. Upon completion of the design for each ship, an estimate is made of the structural weight and of the labor and material cost involved in the construction.

Examples. A study summarized in Reference (29) was performed to investigate the feasibility of designing a web frame that satisfies certain requirements stipulated by an isolated ballast system to be used in tankers.

For the process of optimization, the web frame must be divided into zones of constant thickness, as shown in Figure 50. The extent of this decomposition is arbitrary and depends on the size and availability of steel plates, convenience, ease of construction, etc. A finite element analysis and framework analysis were used in conjunction with von Mises yield criteria for computation of stresses.



Zones of constant thickness

Figure 50

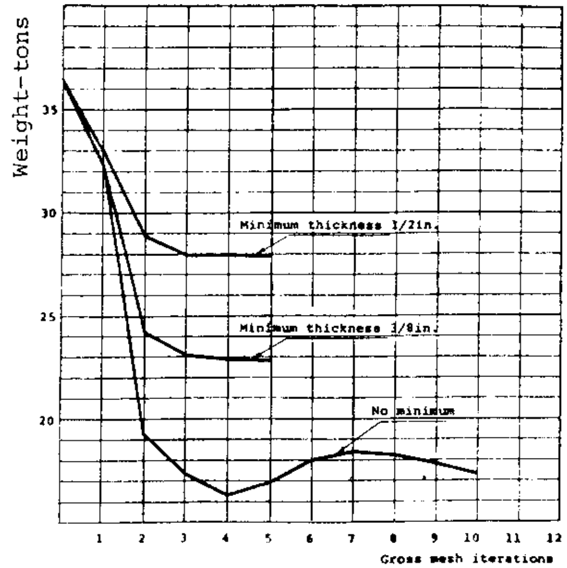
An optimality criterion based on fully stressed design was used to minimize the weight of the web frame with a double iteration procedure developed for the efficient use of the finite element analysis in the optimization program. The shell, deck and bottom platings were not allowed to vary in the optimization procedure, since the thicknesses of these are determined from longitudinal strength requirements and other considerations.

The main conclusion was that it is possible to minimize the weight of the web frame, with the reduction in the weight being dependent on the minimum allowable plate thickness. Three different thickness requirements were used and the corresponding reductions in weight are shown in Figure 51.

The optimization procedure is general and is applicable to any web frame or similar structures. A possible extension of this computer program is to relate the amount of stiffeners required to prevent shear buckling and vibration to the minimum plate thickness of the webs.

Computer Graphics

With the improvement of the general purpose structural programs to analyze large and complex structures economically, the need for efficient methods of checking input data and reviewing output results becomes more pressing. The field of computer graphics satisfies this need by producing visualizations of the structural models and stress



Influence of the minimum thickness on the optimization

Figure 51

patterns. The inherent advantages and disadvantages of the two basic methods of computer graphics, passive and interactive, dictate the usefulness and areas of application of each method.

Passive graphic systems include plotters (flatbed, drum, and electrostatic) and microfilm recorders. The nature of these devices prohibits user interaction and therefore their use is best suited to applications where a user has time to review the resultant plots before making any changes or going on to the next step. The most popular applications include plots of the geometric model (input), deflected shapes and stress contours (output). Since turnaround requirements for these applications can usually be measured in hours, the normal operating procedure is to run these jobs in the batch mode on large computer systems and produce tapes which can subsequently run on the plotters.

Most of the large finite element programs have plotting capabilities as part of the basic program or as separate add-on modules. In addition, there are many general purpose plotting programs that can use the output files generated by most finite element programs, although in some cases interface programs must be written. Input geometrical plots (two-and three-dimensional perspective views) and a wide variety of output plots (deflected shapes, force and moment diagrams, stress contours) form the most generally available plot features.

The basic advantage of the passive system is that large amounts of data can be processed economically. The plotter is the only additional equipment needed, and it is relatively inexpensive ($5,000 - $20,000 range); the actual computer runs necessary to generate the plots can be run in the batch mode and the impact on a large computer system is minimal. The obvious disadvantage is the lack of interaction. Incremental mode plotters are relatively slow (detailed plots can take many hours) but electrostatic plotters can produce hard copies at rates comparable to most reproducing machines.

Interactive graphics systems consist of display consoles, means of entering and editing data (usually CTR's, tablets, keyboard devices), and a computer system to perform the various operations of maintaining the data files, and the calculations needed to produce the plots.

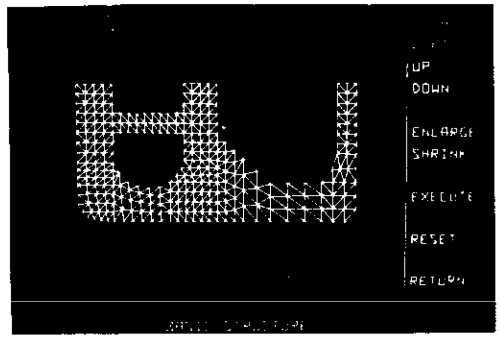
Interactive systems find their greatest use in design work. The designer is able to communicate with the computer, see the results, and make the necessary changes. The earlier systems required either a totally dedicated medium size computer or a large portion (partition) of the resources of a large computer. In recent years, the availability of time-sharing systems and powerful minicomputers has relaxed these requirements.

There is a proliferation of interaction graphic devices, minicomputers, and specialized systems, but the design features and goals are similar. A desirable system should:

1. provide full accessibility to all data through a graphics terminal.
2. use low-cost graphics terminals to allow access to the greatest number of users at the minimal capital outlay.
3. be machine-independent.
4. possess enough flexibility so that it can easily be maintained and expanded in response to user needs.
5. be able to interface with finite element programs.

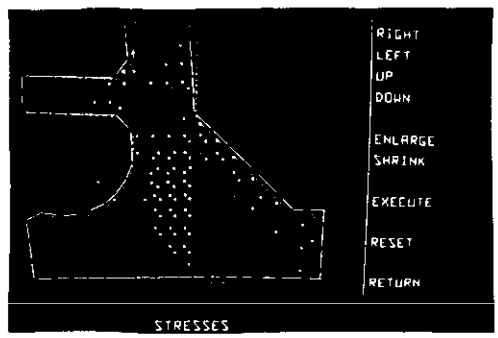
If one cannot afford the luxury of a completely dedicated system, criteria 1. and 2. can best be met with a time-sharing system. Of particular interest, since it satisfies the above criteria and especially since it was designed primarily for ship structures, is the GIFTS (Graphics-oriented Interactive Finite Element Analysis Package for Time Sharing Systems) package (2).

The entire system accesses a Unified Data Base (UDB) which stores all pertinent data on a set of random access files. Each individual module can access and operate on the UDB. After the entire model has been verified, part of the UDB forms the input to a general purpose analysis program (e.g., NASTRAN, DAISY, SAP). The output from the analysis program is then incorporated into the UDB and additional modules can display results. Some of the displays obtained during the various phases of an analysis of the lower portion of a tanker web frame are shown in Figures 52 and 53.



INPUT MODEL

Figure 52



OUTPUT - DEFLECTED SHAPE AND STRESS VALUES

Figure 53

The greatest disadvantage of the interacting graphics system is the cost associated with a dedicated computer system, ranging from $20,000 for minicomputers to a few million dollars for large computer systems. Time-sharing equipment can significantly reduce the initial cost, but on a long-term basis connect charges for such systems can be significant.

CONCLUSIONS AND RECOMMENDATIONS

Summary of Current Status

Structural Response. The analysis of ship structures for static loads or equivalent static loads is well established, and the finite element method of solution provides structural responses of excellent engineering accuracy. New developments in the field of substructures together with the incorporation of new finite elements, such as solids, allow a better structural representation and a significant reduction in manhours per analysis.

Dynamic loads on a ship are not nearly as well defined as static loads, and much work remains to be done in this area. Hydrodynamic coefficients, especially at high wave frequencies, and internal damping characteristics require more investigation. More full-scale data on excitation forces and wave spectra is also needed.

In performing a rational engineering analysis, structural reliability should be taken into account, together with the appropriate extensions beyond elastic theory to fracture mechanics, crack propagation, etc.

Computer-Aided Design. Programs to determine ship characteristics, such as hydrostatic curves of form, stability, etc. have been widely developed and used, and they provide satisfactory results for use in the development of the ship design. However, programs such as ship motion, external pressure, etc., are still in a preliminary stage, with many limitations on their capability.

The trend of classification society rules is to encourage rational computer-aided analysis. Structural optimization is an excellent tool whose applications are expanded in the industry. Computer graphics is another fairly recent and not yet very widely used development, providing significant time savings together with very quick error detection.

Recommended Research Subjects

Based upon the above description of the current status, the following items are recommended for further consideration, study and investigation:

1. excitation (loading) forces induced by propeller and waves including theoretical analyses and experimental measurements.

2. improvement of the strip theory in calculating added mass and damping coefficients by taking three-dimensional effects into account.

3. statistical combined effects of wave-induced and springing vibratory response of ship structure.

4. development of a procedure to assess the reliability of ship structure by means of damage statistics based on survey records, and their incorporation into fracture mechanics analyses and long-term wave load analyses.

5. development of required ship design computer programs to facilitate and expedite the design process.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the valued and much appreciated assistance of members of the Research and Development Division of the American Bureau of Shipping who have assisted in the preparation of this paper, in particular Mr. A. Bakker, Dr. H.H. Chen, Dr. H-Y. Jan, Mr. D. Liu and Mr. M. Wojnarowski.

Special mention must also be given to the Ship Structure Committee, which has sponsored many projects that have been utilized either directly or indirectly by ABS in development of Rules for ship construction or analytical methods for predicting structural response. For example, the ABS ship motion computer program currently in use is essentially the SCORES program, which was funded by the Ship Structure Committee and issued as SSC Report No. 230. Another extremely important Ship Structure Committee activity is the SL-7 research program. A good portion of this program is directed toward gathering ship motion and strain measurements at sea and correlating analytical and model test results. This project is jointly sponsored by Sea Land Services, Incorporated, the Ship Structure Committee and the American Bureau of Shipping.

Finally, thanks must be given to the American Bureau of Shipping for sponsoring the research work described herein, including in-house research as well as research projects assigned to other organizations. A paper such as this could not be possible without access to this information.

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DISCUSSION

J. R. Cheshire, Member

The paper presents a comprehensive survey of different aspects of structural analysis and design of ships' structures, and offers the reader brief details from which he may compare a variety of the larger F. E. computer programs. It is commendable in its scope and compares very favourably with the many other papers of similar nature which have been presented in recent years.

Lloyd's Register's direct interest lies in the discussion and relative merits of the NASTRAN finite element computer program as dealt with in the paper, and the Table below gives corrections to the analysis options shown in Table 1 of the paper. As with many large computer programs, NASTRAN is of dynamic design and a number of facilities not mentioned in Table 1 are contained in the more recent releases.

TABLE 1

ANALYSIS OPTIONS AVAILABLE IN NASTRAN

1.	Small displacements	YES
2.	Large displacements	YES*
3.	Incremental plasticity	YES
4.	Creep	NO
5.	Temperature dependent material	YES
6.	Natural frequencies, mode shapes	YES
7.	Transient response	YES
8.	Data generation	NO
9.	Graphic displays	YES
10.	Multi-element library	YES
11.	Thermal effects	YES
12.	Bifurcation buckling	YES

* First approximation.

Lloyd's Register uses NASTRAN as an integral part of a large engineering system (LR.SAFE), developed for application to the appraisal of ship structures, a description of which is given in Ref.1. The system contains comprehensive data generation and post-processing facilities available for use on main frame computers, together with complementary interactive graphics modules run on a mini-computer. The use of finite element techniques has progressed beyond the stage of the static elastic response examples given in the paper.

In an attempt to demonstrate the range of work undertaken, the results of three analyses using the NASTRAN capability are given in Fig. 1. through Fig. 3. Fig. 1 shows a graphic display of the buckling mode from a set of results of a stiffened floor panel of a double-bottom structure.

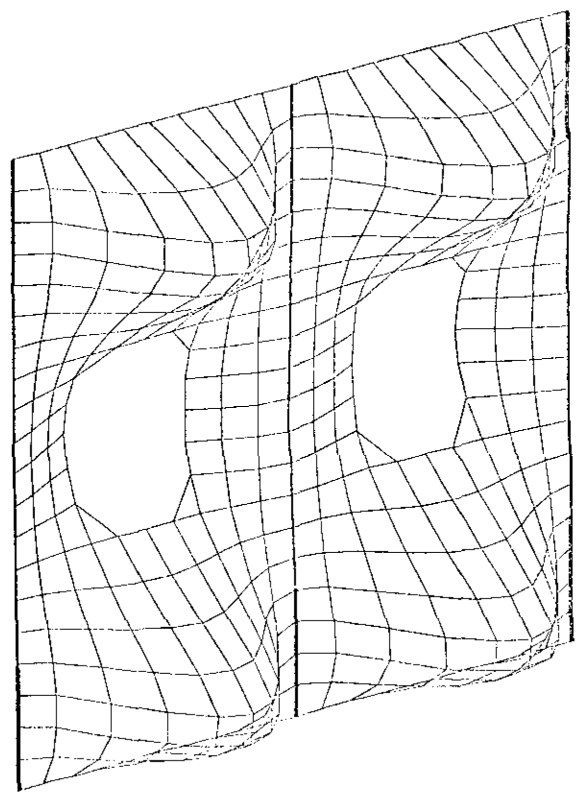


Fig 1 BUCKLING MODE — DOUBLE BOTTOM FLOOR PANELS

The analysis makes use of NASTRAN rigid format 5 in which a stress analysis is performed to obtain the stress state on which the differential stiffness generation is based. Eignvalues are extracted using inverse power iteration.

Fig. 2 shows a comparison of edge stresses around a transverse bulkhead cutout for a longitudinal. The diagram compares the results of a linear analysis using NASTRAN rigid format 1, and a piecewise linear analysis using NASTRAN rigid format 6. In the non-linear or piecewise analysis, the load is applied in increments until the full load intensity is reached. At each increment the stiffness matrix for the nonlinear elements is computed and then added to the basic linear stiffness matrix. The incremented solutions are added to the current solution after each load increment is applied to the structure. In the example shown, five load increments were applied before attaining full load.

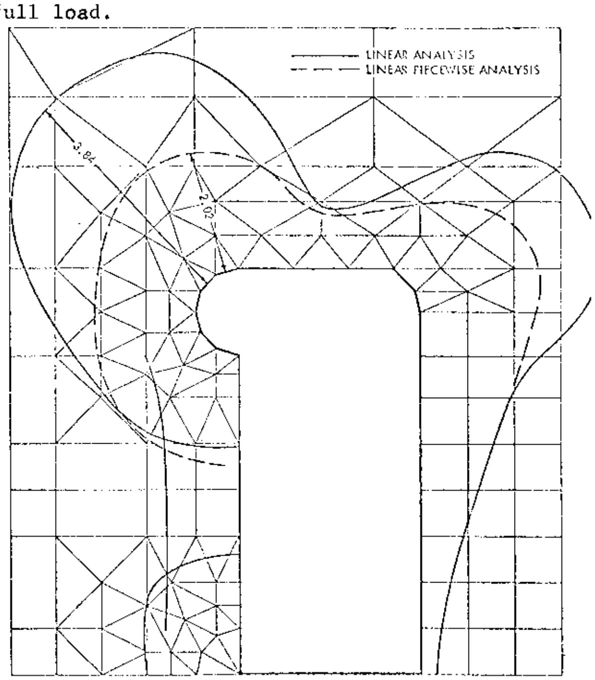


Fig 2 COMPARSION OF EDGE STRESSES

Finally, Fig. 3 illustrates a transient response model in which the bending moments applied to a hull girder by a transient force are estimated from the particular solution of the equation.

$$M\ddot{x} + N\dot{x} + Tx = F(t)$$

where: M = mass matrix

N = damping matrix

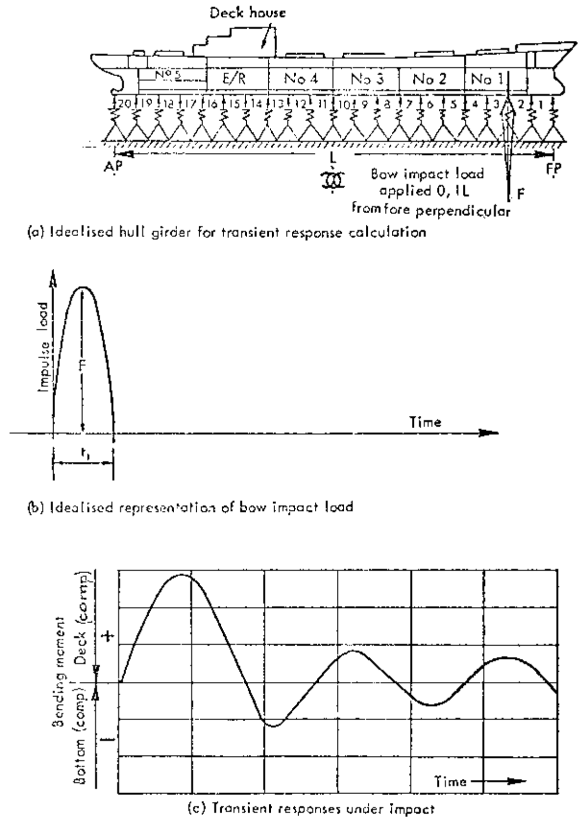
T = stiffness matrix

$F(t)$ = time dependent force vector

$\dot{x}$ = first derivative of displacement

$\ddot{x}$ = second derivative of displacement

The solution to this equation is arrived at by using the NASTRAN rigid format 12, which makes use of modal analysis methods and employs finite difference techniques. Further details of the application shown in Fig. 3 are contained in Ref. 2.



(a) Idealised hull girder for transient response calculation

(b) Idealised representation of bow impact load

(c) Transient responses under Impact

Fig 3 TRANSIENT RESPONSE MODEL

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S. G. Arntson, Visitor

Mr. Stiansen has presented a vivid overview of the current state-of-the-art in computer aided structural design. I am sure the paper will be well received and widely read by structural designers and ship design managers. I would like to address a few comments for the benefit of the latter group, the ship design managers who oversee the structural designer and especially his computer budget.

The discussion of the comparison of results using various computer techniques may lead to the erroneous conclusion that only the more sophisticated programs, such as those utilizing finite element techniques, are adequate. This is not the case at all. All of the programs discussed in this paper, as well as some additional ones, are valid analytical tools. The simple, inexpensive programs have generally been proven to be adequate for the vast majority of design programs at a fraction of the cost of the more expensive sophisticated ones. Simple beam and space-frame programs have been successfully applied for complex appearing problems which, by careful planning and full understanding, have been reduced to simple terms. On the other hand there are design problems, sometimes simple appearing ones, which require the full rigorous treatment of the complex, expensive programs. The main point of this comment is simply that the structural designer needs a wide variety of computer techniques at his disposal in order to handle a typical ship design. He should have the freedom to match the proper design tool with any specific problem in order to insure that it is adequately and economically handled.

Considering the proposal of a reassessment of factors of safety as noted under structural reliability, this is one area which requires the greatest caution. As pointed out in the text, current practices are largely based on ignorance factors but it must also be noted how well they work within the existing scheme of things. Advances in techniques should, and indeed must, be incorporated into designs, however the principal way to fully refine the design process is to insure total confidence in the design criteria and assumptions. The only way to achieve total conficence is to establish controls; material property controls, construction technique controls, fabrication tolerance controls, operating controls, etc. Generally controls beget more control and the cost in time as well as money surely mounts. In many cases, such as the development of submarine pressure hulls and nuclear reactor containment vessels, stringent controls are obviously necessary and well worth the cost. But in the case of most surface ships, especially commercial type hulls, it would appear that there is very little advantage to be gained at what may be great cost. Therefore, it is recommended that any reassessemnt of factors of safety be approached with the utmost prudence and caution.

In the discussion of Static Loading, the author points out that "the resultant stresses (from the static ship balance) are considered only as representative stress levels rather than absolute stress values". This is a critical point which cannot be repeated too often. It is surprizing how often this point is misunderstood or perhaps misrepresented. Just as in the case of factors of safety, it is an ignorance factor which covers a multitude of sins and should never be considered as anything else.

The final point which I would like to take up is an additional advantage which results from the use of computer graphics. Design managers and others who control design budgets often consider the use of passive graphics as an unnecessarily expensive and time consuming luxury which is not justified. It should therefore be pointed out that passive graphics can often be used to insure computer/operator communication, or in other words, as a validation check. The successful use of computers depends upon complete understanding between the machine and human elements; however machine and human logic are not generally the same. Therefore graphics can be used to insure understanding by requiring the machine to interpret the data in pictorial form. As an example, Figure (marked SA-1) indicates two sections of a body plan as drawn by the computer. The offsets for these sections were similar except for one point at the turn of the bilge and both sets would have been adequate for human interpretation. As is evident from the Figure, the computer could not correctly interpret the data and therefore generated a

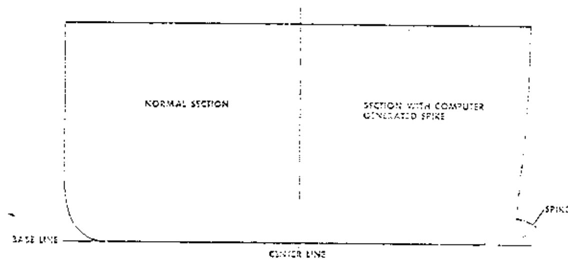

FIGURE (SA-1)

"spike" at the turn of the bilge in the one case. When calculating the area of the section, the computer interprets the data in the same manner thereby introducing an error which is carried through to the buoyancy and bending moment calculations. This particular error is common but is very easily detected by the use of a passive graphics check. Figure (marked SA-2) illustrates an unrelated error which was only detected by use of computer graphics. Again the computer had difficulty in interpreting the body plan although it was able to reproduce each section correctly. By computer logic it generated an excessive amount of buoyancy in the fore body of the ship (noted as a buoyancy spike) which naturally invalidated the entire calculation. The sagging case is shown, but buoyancy spikes were also evident, to lesser extents, in both the still water and hogging cases as well. Standard numerical print-out indicated only that something was amiss (excessive trim by the stern), but did not indicate the nature of the problem. Without a graph check, a great deal of engineering time could have been wasted trying to resolve the problem. Both of these examples are from the same program, however similar ones can occur in almost any program. The main point being that computer graphics are often a simple means of insuring computer-user understanding and of validating the process.

By this paper, Mr. Stiansen has made a needed addition to our general understanding of computer usage in ship structural design.

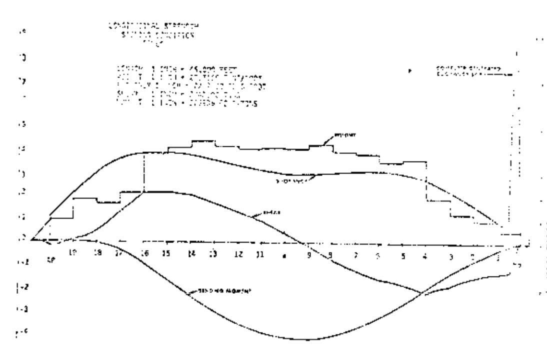

FIGURE (SA-2)

Edward V. Lewis, Honorary Member

My comments apply to only three items in this comprehensive paper. The first is on the type of dynamic response known as springing (pp. N-8, 9). This is a particularly important problem for full-ended, flexible ships -- particularly Great Lakes bulk carriers. As indicated in the paper, studies have been going on for some time at Webb Institute on both experimental and theoretical aspects (under the sponsorship of the U.S. Coast Guard). (30).

The model tests confirmed the statement in the paper that "in most cases, the vibration originates from synchronization of the natural frequency of hull vibration with a region of the wave energy spectrum which has a relatively high energy content". In fact, a regular train of short waves (say 6 inches long and 1/4 inch in height) excited remarkably high vibratory response in a jointed model at the speed for synchronism. After some modifications and refinements to the Goodman theory (6), good agreement was obtained between model tests and theory. Correlations between full-scale "maneuvering tests" and theory was less satisfactory, presumably because of difficulties in obtaining reliable simultaneous wave buoy and stress records and because of doubts regarding full-scale damping factors. It was concluded that for similar, full bulk carriers the non-dimensional springing bending moment in representative severe seas varies mainly with the non-dimensional natural frequency of vertical hull vibration. An important area for further study is recommendation no. 3 (p. N-35), "statistical combined effects of wave-induced and springing vibratory response of ship structure."

Mention is made (p. N-17) of the importance of structural reliability considerations in developing new and improved design standards. It is true that both the loads (demand) and the response of the structure (capability) need to be treated probabilistically, and hence the damage cost determined for each mode of failure. Hence, I would agree with recommendation no. 4 (p. N-35): "development of a procedure to assess the reliability of ship structure by means of damage statistics based on survey records, and their incorporation into fracture mechanics analyses and long-term wave load analyses".

My third point is to support the statement regarding hydrodynamic aspects of the powerful finite element technique of stress analysis (p. N-28). "In order to make full use of the advantages of such a mathematical model, the dynamic loads resulting from the waves encountered by the ship in a seaway have to be known, so that the forces at each finite element node on the wetted surface can be approximated". As indicated, Hoffman's theoretical approach to this problem has been programmed for computer at Webb Institute for the simple head sea case. The programming of the more difficult general case of oblique seas, with motions in five degrees of freedom (pitch, heave, sway, roll and yaw) is nearing completion. Another factor to be accounted for -- not mentioned at this point in the paper -- is the mass inertia effects on cargo and on structural elements as a result of the motions. These refinements should greatly enhance the value of finite element caluclations.

In conclusion, I should like to offer an additional recommendation for further research. This is related to the preceeding comment on the reliability approach and the need for information on structural capability. It would seem to be of great importance to go beyond the detailed determination of stresses to the evaluation of "load-carrying ability" of structure (John Vasta's term). In many cases this will involve buckling strength of critical panels subject to in-plane compressive loads. In other cases it will involve the low-cycle fatigue strength of details having high stress concentrations. The paper shows both a great deal of recent progress and a great many important things still to be done. Hence, it is a valuable and timely piece of work.

Reference

30) Hoffman, D. and van Hooff, R. W., "Feasibility Study of Springing Model Tests of a Great Lakes Bulk Carrier:, International Shipbuilding Progress, March 1973.

Egil Abramhamsen, Member

In regard to your interesting suggestion to a mixed beam-membrane frame idealization, Fig. 1, DnV made some investigations in 1974 on a hybrid frame model. The results were satisfactory from the technical point of view but had little practical consequence as only negligible cost reduction was achieved.

As the authors assumed, the difference between measured and computed stresses in the web frame no. 90 of "ESSO NORWAY", Fig. 8 and 9, must be due to modelling approximation and initial deflections. DnV has performed 3 separate finite element analyses of "ESSO NORWAY" as part of the test procedure for our SESAM-69 program system at varied stages, and we would like to add the following:

Generally speaking, very good correlation between measured and computed stresses was found for all three analyses.

In the last analysis, which was of the order of 24,000 degrees of freedom with quite accurate modelling of openings, etc., the disturbance due to initial deformation was even further revealed. Referring to Fig. 8, the normal stresses plotted at a section through the bracket in the lower part of the wing tank, it should be mentioned that we obtained similar discrepancies regarding the normal stresses in the transverse direction. This was a typical result of the difference between the linear elastic finite element analysis of a mathematically plane structure and the nonlinear but elastic buckling behaviour of the initially deformed unstiffened plate part. The measured normal bending stresses on each side of the plate show that the plate bending stresses due to buckling were about 5 times the membrane stress. It is therefore of interest to learn that workmanship inaccuracies, other sources of imperfections, dynamic load accuracy and nonlinear effects will govern the necessary mesh fineness.

Recommended Research Subjects.

I find the items listed by the authors very appropriate and would like to supply the following information and suggestions regarding possible approaches.

Considering propeller-induced excitation, Det norske Veritas has developed a method and corresponding computer programs for predicting hydrodynamic loading on propeller blades, the amount of cavitation and the pressure impulses on the afterbody. In brief the following procedure is used:

- The input data describe the propeller geometry and the axial and tangential wake field in the propeller plane. A modified lifting surface technique is applied to calculate the mean and fluctuating chordwise pressure distribution on the propeller blades. The instantaneous pressure distribution on the blading is also integrated to give the excitation forces acting on the shaft.
- The amount of cavitation is then calculated using the maximum bubble concept combined with correlation functions from several full-scale observations of cavitation.
- Finally the pressure impulses at various points on the hull are calculated based on additional data describing the hull form and the intermediate results from the previously described calculations. The pressure impulses are found by superposition of pressure impulses from a non-cavitating propeller and from volume pulsation of cavities.

The development of the method and the associated computer program for calculation of propeller-induced excitation has been closely followed up by extensive experiments both in model tanks and onboard ships. These measurements and observations illustrate the the method gives reliable results.

The authors suggest that three-dimensional hydrodynamic effects should be included in computations. I agree that this may be necessary or desirable in some cases, particularly in the case of ships with large dimensions and blunt end forms. In this connection I can mention that one of the techniques applied in Det norske Veritas' ship motion programs (Frank Close-fit Source and Sink Technique) is now extended to a three-dimensional description. The method permits computation of added mass and damping coefficients for any water depth, also at frequencies in the range of vibration.

For ship forms we expect the importance of such calculations to be of primary interest in ship vibration problems (springing, whipping). Evaluations with respect to low-frequency shear forces and bending moments undertaken so far indicate only minor influence of three-dimensional effects.

Regarding the subject: "Development of required ship design computer programs to facilitate and expedite the design process", we are presently preparing new rules which explicitely state the design loads and criteria. In this connection we are working on a program package for design and analysis of ship. The program involved will either be rule dependent or used in direct analysis. Many programs are not purely a tool for analysing a given structure, but will calculate the required scantlings according to rule criteria or specified allowable stresses. The package can be used during all stages of the design process and will assist the designer in his everyday tasks. Some programs of primary interest are listed below.

- Section Program.

  This program analyses a given section or calculates the dimensions of all longitudinal elements according to our new rules. Any ship type and arbitrary chosen sections can be handled. The program can also be used for shear flow analysis at any open or closed (multicell) sections.

- Frame Analysis Programs.

  One of these programs has proved to be a very powerful tool for the designer. This is due to advanced method and data generation with excellent accuracy combined with low cost. The program can be used for analysis of the primary stiffening system of any ship type or weight optimization of the transverse web frames applying either fully stressed or SUMT optimization technuqies.

- IMCO Arrangement Design.

  This is also a powerful tool when dealing with large tankers as a SUMT optimization technique is used to find the "best" tank arrangement which fulfill the IMCO regulations.

- Hydrodynamic Calculations.

  Our series of programs for prediction of the ship motion and response in regular or irregular seaways will be used.

All programs may be used separately or integrated with operation on a common database. Connections to our PRELIKON and SESAM-69 system will ensure flexibility and good efficiency.

C. W. Coward, Member

The author has given the reader a good peek into the current approval and associated research activities at ABS. He has done a great job of condensing a very large amount of material into manageable size and yet retained enough detail, examples, etc. to assure that the paper will get much use as a reference document by those of us involved in ship structural design.

My comments will be restricted to one specific area, namely the subject of stress concentrations. Figure 35 of the paper shows an example of calculated stress intensities around a hatch corner. As can be seen in the Figure, local stresses can become quite large. This raises questions as to the acceptability of the design. In such cases, acceptability can be justified, as pointed out in the paper by the fact that, when a local stress reaches the elastic limit, a redistribution of stresses in the area begins and elastic limit stress levels are not exceeded unless the basic stress reaches that level. It can also be pointed out that the calculated stresses represent the worst condition of loading expected during the life of the ship. It can be stated that similar concentration conditions have always existed in various locations on all ships. If the same type of analysis were performed for ships which have experienced no structural failure throughout their life, it is safe to say that stress intensities greater than yield stress would be identified in certain locations.

This type of rationale may satisfy one who has a great deal of structural design experience but it is difficult to convince a concerned ship buyer in this manner. It is also inconsistent with the sophistication involved in developing the loads by ship motions programs and the fine mesh finite element stress analyses uses to identify the stress patterns. The following is offered as a rational method of establishing the acceptability of local, high stress intensities which are certain to emerge as structural analyses become more detailed and comprehensive.

Even though local high stresses tend to be redistributed when the elastic limit is reached, many occurrences could result in failure of the material due to fatigue. Figure 1 is the familiar S-N curve for an ABS steel which might be found in a deck with a hatch opening. Assuming the hatch corner is so configured that the worst loading conditions the ship will ever see induces local maximum and minimum stress intensities of 50 ksi and 5 ksi respectively, the relationship between alternating stress and number of cycles (10^8) currently being used is depicted Figure 2.

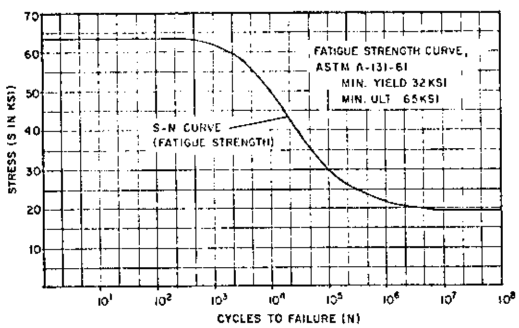

FIGURE 1

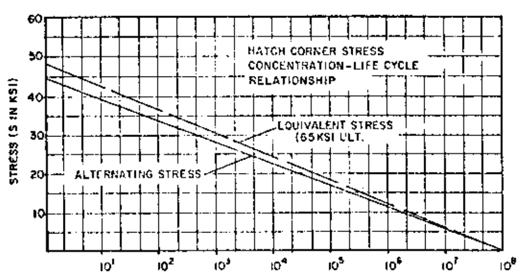

FIGURE 2

From ASME Boiler and Pressure Vessel Code (Sections III and VIII, Division 2), the alternating stress becomes:

$$\text{alt.} = \pm (\sigma\ \text{max} - \sigma\ \text{min}) = \pm 45 \text{ ksi for } N = 10^{\circ}$$

Since the stress at times exceeds the yield stress of the material, an equivalent alternating stress component for zero mean stress must be calculated:

$$\sigma_{\text{equiv}} = \frac{\sigma\ \text{alt}}{1 - \dfrac{\sigma\ \text{min}}{\sigma\ \text{ult}}} = \frac{\sigma\text{alt}}{1 - \dfrac{5}{65}} = \underline{1.08\ \sigma\ \text{alt}}$$

For $N = 10^{\circ}$ and $\sigma\text{alt} = \pm 45$ ksi

$$\sigma_{\text{equiv}} = \pm 48.75 \text{ ksi}$$

The equivalent alternating stress is also plotted in Figure 2.

Using Figures 1 and 2, we can determine, for any number of cycles of loading, both the equivalent level of stress and the fatigue strength. Dividing the fatigue strength by the equivalent stress level results in a factor of safety against fatigue failure for that number of cycles. This, then, is plotted in Figure 3. Unless the curve dips below a factor of safety of 1.0, it may be concluded that for any anticipated conditions of loading during the life of the ship, even though local stress intensities might occasionally become severe, structural failure should not occur.

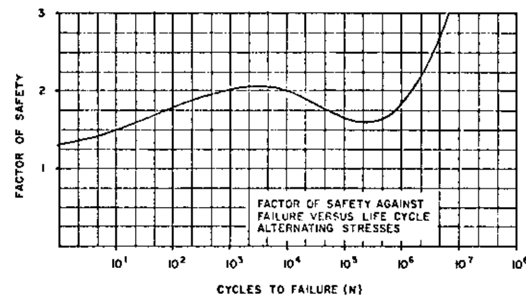

FIGURE 3

It is expected that, in the future, computer programs will handle this type of situation as a normal thing. I submit that this will be necessary to take full advantage of our growing ability to better identify loads and structural responses for complex structures of ships.

We are at an interim point at present. In addition to designing ships to regulatory body rules, we perform the more sophisticated analyses and identify areas of high stress and modify structures accordingly. On the other hand, areas of low stress are not modified unless scantlings exceed those required by regulatory body rules. I do not suggest that we throw away the rules, but we must be willing to develop confidence in our new techniques, be assured that all factors are considered, and continue to move toward accepting the results no matter how they compare with the more simple, empirical methods of the past.

Gordon G. Piche, Member

Mr. Stiansen's paper presents an excellent overview of many facets of the design and analysis of ship's structure. The paper calls out what can be done in such areas as linear and non-linear static stress analysis, dynamic response analysis, thermal stress analysis and initial distortion influences; however, it only hints at the price in manpower, time and money that must be paid in order to carry out these analyses. The old axiom "you don't get something for nothing" was never truer than when applied to structural analysis using the finite element methods.

Programs such ANSYS, NASTRAN, DAISY and MARC are very sophisticated computer programs that require the very latest and largest in computer hardware. The user of these programs must spend many hours learning how to use these programs, in fact, courses are offered for that reason.

After one has become familiar with the program and determined what information is needed, a discretized model of the structure must be created. It is in the creation of the model that the ingenuity of the user must hold forth if a valid solution to the problme is to be obtained. In addition to the creation of the model, loads and constraints must be selected and applied such that the model will respond like the real structure. The preparation of the data to create, constrain and load a large three-dimensional finite-element model is usually measured in weeks and often months. Most general purpose finite element programs have preprocessors which aid in preparing the data necessary for computer solution. The preprocessors can significantly reduce the time for data preparation.

The computers required to do extensive finite element analysis are large and very fast. But even these computers require minutes of central processing time (CPU) to build up and then decompose the stiffness matrix associated with large three dimension problems. Costing anywhere from 15 to 50 cents a CPU second, these problems consume hundreds of dollars quite rapidly.

The result from a finite element analysis is thousands of pieces of information detailing the forces, moments, displacements, rotations and stresses. These numbers must be sorted into usable information, generally graphically. In this area, the post-processors available with the general purpose programs came to the engineer's rescue. Stress contour plots and deflected shape plots allow rapid reduction of the voluminous information.

As is apparent from Mr. Stiansen's paper, there has been a lot of emphasis toward developing pre and post processors for finite element programs in recent years. These processors ease the burden of data preparation and result reduction and make the finite element method more attractive to those who cannot afford a large commitment in manpower, time and money. The Coast Guard found itself in precisely this position.

With manpower ceilings in its' various technical offices, the finite element method could be used for only small and unusual applications. To put the finite element analysis method in the hands of its naval architects and marine engineers, the Merchant Marine Technical Division of the U. S. Coast Guard has begun using the GIFTS system which was created by Dr. Hussein A. Kamel and his group at the University of Arizona. Mr. Stiansen has briefly described the GIFTS program in his paper. I would like to add that our experience with the program has shown that it is quite easy to use, even for the novice, and that the mesh generators, loading and constraint routines and graphic displays reduces to a couple of days problems that would have taken six or eight weeks otherwise.

Interactive graphics is surely the path to follow toward reducing the manhour requirement associated with the FEM. The additional cost for interactive graphics appears to be more than offset by the savings in manhours. GIFTS uses low-cost storage tube terminals in a timesharing environment.

I would tend to disagree with the author's statement "Interactive systems find their greatest use in design work". That may have been true over the last couple of years, but I think more and more interactive graphics will become the preferred method for engineers to interface with finite element programs for all types of applications.

I would also point out that figures 52 and 53 in the paper were not obtained from the GIFTS program. These figures show the use of a refreshed scope terminal, where GIFTS use only a storage tube terminal; however, plots such as this are obtainable from the GIFTS system.

Vladimir Boban, Associate Member

The author with his able associates have provided a comprehensive summary of the many computer programs in use today and his paper contains a large amount of useful information.

The trend in the last decade to build larger and larger vessels necessitated a development of computerized techniques for the analytical approach to ship structural design. The fast pace of development of many programs and techniques initially created some confusion in the industry as to which program is best suited for a particular design problem. The author gives some insight into the programs that are available and their capabilities.

In addition to the listing of programs and their capabilities, the author emphasizes the need for correlation between theoretical and experimental results. Please note, however, that one or several confirmation tests covering longitudinal strength programs can hardly give the complete answer for the long-term statistical prediction. Additional correlation is required to obtain complete confidence in the program.

For commercial reasons it sometimes may be necessary to keep certain data or program technology confidential. Because of this situation it is inevitable that some duplication between Societies will occur. It is hoped, however, that through IACS (International Association of Classification Societies) a greater interchange of information will occur which will be beneficial to the industry. This does not mean that Classification Societies should have exactly the same rules or the same philosophy but they should try to best utilize the information at hand which will satisfy the shipbuilding industry at large, i.e., interchange of information on structural defects.

Papers generally tell how to do something but equally important, and which is often lacking, is that the papers fail to tell one how not to do something. Many details which are very important in the overall structure are superficially treated. Their importance is often misunderstood by the academic community. Because of this, many ship owners are disappointed when they have to pay large repair bills to solve problems which should have been solved initially. Such problems are capable of solution and hopefully will be solved by the appropriate computer programs with the proper attention to details. In practice many minor structural defects are corrected as they appear but at considerable cost and often with no assurance that the same or similar defects will not show up again. To many people these are "nuisance problems" but they forget that to the owners who pay the bill they are anything but a nuisance.

As an example of the foregoing, I refer to the cracks which occurred in the forward most cargo tanks of the first generation VLCC's. These cracks were located in way of the intersection of the side longitudinals and transverse webs. Due to the poor connection detail between the side longitudinal and transverse web the connection failed. This failure resulted in some minor shell cracks which presented a pollution hazard. Due to the urgency of eliminating the pollution hazard these cracks were very expensive to fix. This situation occurred on many VLCC's, which were built in various countries and shipyards and classed with several Classification Societies after about 20 - 24 months of service.

The author also mentions a statistical approach which will hopefully give us a future tool in problem solving. The failing with a statistical approach basically is the reliability of the variables. The number of variables used, the importance of the variable and the number of variable readings have to be considered when analyzing the results of the statistical approach method.

We have often heard that since Classification Societies have their rules and various computer programs, there is no need for an individual computer program investigation. In fact, for special or novel designs a sound engineering approach utilizing computer techniques is a necessity irrespective of what Classification Societies might or might not do on plan review.

A combined effort is required from industry, the universities, the Classification Societies and Ship Structure Committee in order to produce a sound structural design. All should be involved in combining or extracting useful features from existing programs instead of creating more programs for every day use in practical shipbuilding.

Huynh duc Bau, Visitor

The author should be commended and thanked for this very comprehensive review of computational techniques currently under use in ships design.

The paper covers a wide range of topics and it is thus understandable that detailed treatment can not be made of any particular point of interest. However, this discusser would like to see the author's further comments on the following:

Computer programs sophistication

The computer programs mentioned in the paper (Table 1, pg N-5) are quite powerful and sophisticated. The table would presumably be twice as long if one had to compile all available programs on the market. Considering:

a) The present knowledge of design loads

b) The degree of ignorance concerning safety factors

c) The time and dollar cost that one must expect to pay for such complexity and sophistication.

The author's comments are sought regarding whether a common effort should be undertaken by the community of ship structural designers to halt this escalade toward excessive computers accuracy.

Such undertaking as Full Hull Finite Element are no doubt useful and certainly interesting. However, strain guages adequately fitted could have provided, at lesser cost, confirmation of the validity of Navier beam theory (Fig. 4 of the paper).

Furthermore, after a little over one decade of computer aided structural calculations, efforts are still overdue to develop simple, yet accurate enough, design guide lines or criteria. Indeed, with progress being made in the field of optimization and approach to failure, pressure sooner or later will be exerted on Regulatory Bodies for tangible analytical formulations of structural requirements to permit efficient evaluation of structural load carrying capabilities (transverse elements) or economical optimum criteria.

This approach has been adopted by Bureau Veritas concurrently with the direct engineering analysis of modern ship's design. Since 1973, Bureau Veritas Rules and Regulations have been directed toward such aim.

Influence of Structural Reliability (Pg. N-17)

The author quite rightly qualifies the classical approach to ship design as a "piece-by-piece" procedure not allowing for simultaneous occurrence of failure modes. However, at the present State-of-the-Art, Reliability Analysis (full probablistic approach) is still very immature and in almost all cases excludes multimodal failures (impossibility of analytical formulations of load and resistance distributions). The nature of the problems resides not in the approach to the safety factor, but indeed in the formulation of criteria for multimodal failure occurrence. Moreover, Reliability Analysis for multi-members structure (ship's transverse elements) for practical reasons does not, as yet, provide a valid substitute to the classical approach of the weakest link.

Thus, roughly speaking, computer aided design procedures, in improving the computation of design loads and structural response have, in the same time, rendered the formulation of efficient and economical failure criteria somewhat unattainable. The advance in computation should be matched by a better knowledge of resistance distributions.

Interpretations of Computer Result (Pg N-18-N22)

1.- The 25% increase in the longitudinal stress in way of the bilge attributed by the author to local perturbation due to horizontal pressures on the end bulkheads is somewhat surprising.

No pertinent discussion can be made without a better knowledge of the vessel's structure. However, it can be said that for such loading pattern the classical (Navier beam) approach to the Hull girder bending moment ignores the following perturbations:

a) Relative deflection between longitudinal primary members

b) Longitudinal stress due to reactions in way of transverse end bulkheads due to hydrostatic pressures.

c) Horizontal hydrostatic pressures (Poisson effect)

Perturbations (b) and possibly (c) can be of non neglible importance according to the stiffening system used for the transverse bulkheads.

In the case discussed in the paper the above seems not applicable to the bottom longitudinal stress and to midsection (II-II). The authors further explanation would be much instructive, particularly in providing indication of the transverse structure in the ballasted tanks.

2.- The example of the connecting detail (Fig. 40) is very much interesting. This detail has been, time and again, discussed. As mentioned by the author it has been subjected to "Nuisance Cracks". The explanation given is quite clear. This example provides an excellent demonstration of the capability but also limitation of the computer. Indeed, the finite element calculation has provided 2 alternate designs. However, such calculation assumes:

a) A perfect alignment between the longitudinal's face plate and the stiffener allowing thus a theoretical perfect stress flow.

b) No discontinuity at the transverse bulkhead (welding).

Thus Nuisance Cracks can still develop should above details not be perfectly executed, and the author's claim regarding the reduction in weight may be compensated by heavier labor cost.

Would the author please indicate whether the second modified design has been actually built?

2.- Design loads, dynamic loading (Pg. 25) The author's assertion regarding cargo weight should be somewhat tempered in regard to:

a) Liquid cargo motion

b) Bulk/ore cargo pressures

Research and computers calculation have been undertaken but much remains to be done. Bureau Veritas has undertaken full-scale data measurement in this field. The same comment may apply to the vessel's weight transverse distribution regarding rolling calculation.

Would the author please comment further on the remark (Pg. 26) of better accuracy in forces computation. Strip theory (OSM) assumes a 2 - dimensional water flow around non interreacting strips. Thus, the method should lead to higher inaccuracy of computed wave-induced loads as they, more than the vessels motions, are subject to interaction between transverse sections.

Regarding the two procedures for long-term load prediction, would the author please provide further clarification as to:

a) The need to account for non energy linearity in heavy sea state and for spectrum non-narrowness (assymetric shape of large waves) when considering low probabilities of occurrence.

b) Any experimental basis for the Galtonian distribution of long-term wave characteristics (Gaussian short-term wave elevation compounded by a gaussian likelyhood of occurrence) besides the Jasper's 1956 measurement in the North Atlantic.

c) The need of using extreme value approach considering that full-scale data are collected over a short period of time compared to ship's life. Moreover the tail of the gaussian short-term distribution of wave elevation (yielding the Rayleighian distribution) should be modified for the long-term trend since for long-term prediction and ultimate strength design the bulk of the data is of little significance.

Author's Closure

The author thanks the discussers for their interest and for their valuable comments on the various subjects mentioned in the paper. Since the paper is a general review survey-type effort, it could not go more deeply into many of its component subjects, any of which would merit an entire paper.

On the subject of computer programs, I thank Mr. Arntson for his comments and his presentation of the ship designer's viewpoint with regard to computer techniques.

Computer programs have been increasing their capabilities and decreasing their running time and cost to the point where it is better to analyze a more representative, albeit more complex, structure than to introduce simplifying assumptions whose accuracy is bound to be questioned at a later stage of the design. The careful planning and full understanding advocated by Mr. Arntson are necessary for any engineering problem, but a trade-off study between a bigger computer model and the extra engineering time necessary to establish and justify simplifying assumptions may well lead to the choice of the former over the latter.

I thank Mr. Cheshire for his comments and find it encouraging that Lloyd's Register is utilizing the many capabilities of NASTRAN in its computer-aided analysis of ships.

The Analysis options given in Table I of the paper are constantly being expanded; this is a desirable and prgressive feature of present day computer utilization. As Mr. Cheshire points out, NASTRAN capabilities are being expanded and all the other programs show similar trends. The planned and immediate obsolescence of Table I, or any such similar compilation, is a welcome sign of progress and improvement in the field of computer-aided design and analysis.

I thank Mr. Bau for his discussion, and agree with the statement that a table such as Table I could be considerably longer (much more than twice as long) if all the available programs were listed. I must point out, however, that the present knowledge of design loads is increasing, the degree of ignorance concerning safety factors is decreasing and the time and dollar cost for any given degree of computer complexity and sophistication is greatly decreasing. This points to the conclusion that large-scale finite element analyses are a necessary component of a reliable and cost-effective design.

On the subject of reliability, I would like to point out that refinement of the design process as well as material and production controls will not insure total confidence in any analysis. Even if any portion of the design uncertainties is minimized, if not eliminated, the random statistical nature of applied loads, (waves, etc.) will always indicate the need for a reliability approach in structural design.

As has been pointed out, a better knowledge of resistance distributions is indeed necessary for the formulation of efficient and economical failure criteria. Computer techniques can be called upon in the analysis of these resistance distributions just as they are used in improving the calculation of design loads. I therefore agree with Professor Lewis' additional recommendation to extend the reliability approach order to determine the load-carrying ability of the structure.

I thank Mr. Coward for his valuable contribution on the fatigue strength of a highly stressed hatch corner. It is, of course, impossible to eliminate all the structural discontinuities which cause high stress concentrations in a ship. The best technique for checking the adequacy of a highly stressed structural detail is by calculating its fatigue strength, as shown in Mr. Coward's example.

In answer to specific queries about some of the examples of computer results, the higher bilge stresses illustrated in Figure 34 are a result of many structural parameters, but they are mainly caused by the horizontal pressures on the end bulkheads. Several horizontal stringers, extending two web frames beyond the transverse bulkheads, are present in the ballast tanks. The example of figure 40 shows that a computer analysis can be performed for different alignment and continuity patterns with a minimum of changes in the preparation of data. The ship in question was built with the original design, a choice made by the shipyard based on their evaluation of options under the particular applicable circumstances at the time.

On the subject of design loads, it is obvious that some areas such as cargo pressure distribution, liquid cargo motions and weight distribution in rolling are not completely defined and should be further investigated, as has been pointed out. The ship and cargo weights for the case of static loadings must be considered as well defined, however, in relation to the much greater uncertainties present in the evaluation of sea loads and inertia forces.

The questions raised by Mr. Bau on the accuracy of the strip theory and clarification of long-term load predictions go well beyond the scope of this paper, which emphasizes the application of computer programs available in the industry. Neglecting the interaction between adjacent sections is one of the well-known assumptions of O.S.M. strip theory. Although these interactions are important, it is doubtful that they alone will provide a significant improvement to the wave-induced load predictions. Some of the other important factors to be considered are the incorporation of non-linear rolling into the linearized equations of motion and the question of whether the velocity potential is governed by the two-dimensional Laplace equation for a ship in an oblique wave.

The considerations of non-linearity associated with heavy sea states and of the tail of the Gaussian short-term distribution are pertinent to the long-term predictions. However, there is no sufficient full-scale measured data available to evaluate the theoretical predictions. For this reason, either of the long-term prediction methods mentioned in this paper or the suggested extreme-value approach provide only a reference for practical design purposes.

I thank Mr. Boban for his comments, and I agree that greater confidence in the longitudinal strength programs can be obtained by a greater number of correlation tests.

On the subject of classification rules and engineering analysis, I would like to point out in reply to Mr. Boban that many classification societies, and ABS in particular, conduct research investigations utilizing numerous computer programs and techniques, quite apart from the plan review. This is an area where exchange of information and collaboration in the acquisition of measured data would be beneficial to the entire industry. The International Association of Classification Societies (IACS) is working in this general direction of unification.

The owner's design agent is a very important and busy member of the design process. He must know the owner's requirements for a particular design and then translate them into a properly engineered ship which will satisfy these requirements. No classification society can take his place. At best, a classification society can provide design assistance in certain areas of its expertise, such as the development of structures.

I thank Mr. Abrahamsen for his comments and his presentation of some of the research work undertaken by Det norske Veritas. On the subject that he mentions, development of required ship design computer programs to facilitate and expedite the design process, I again emphasize the importance of the computer in changing the ways of design, eliminating cumbersome tables and equations, and enabling classification societies to implement these changes. In the case of ABS, these trends are reflected in the development of the RULESCANT program mentioned in the paper and the extension of the Rules to permit plan approval based on engineering principles.

I thank Commander Piche for his contribution to computer usage experiences at the Coast Guard, specifically with the GIFTS System. In answer to his comments on interactive graphics, I would like to point out that with the increased availability and decreased cost of interactive graphics, its use will expand throughout the industry. However, by its very nature of being a picture version of a conversational process between man and machine, its greatest use is, and should continue to be, for design work. This is not to minimize the application and importance of interactive graphics in other phases of engineering, where its use is also on the increase.

In answer to Commander Piche's specific question, Figures 52 and 53 in the paper are indeed from the GIFTS Program. They were obtained on the PDP 15 System used by Dr. Kamel at the University of Arizona, where the system can operate either on a storage tube terminal or a refresher tube terminal.

Mr. Arntson's example detailing error detection by means of computer graphics is a good typical case history to illustrate the advantages of this technique.

I also thank Professor Lewis for his valuable comments and his information on the latest results of some of the research conducted at Webb Institute.

I appreciate the interest and the contribution of all the discussers.