SERIAL NO. **SSC-62**

REVIEW

of

PAST STRUCTURAL STUDIES RELATED TO THE SHIP AND SHIP COMPONENTS AND FOR DETERMINING LOADS AND STRAINS ON SHIPS AT SEA

by

J. H. EVANS

Massachusetts Institute of Technology

and including as appendices

A. "Waviness in the Bottom Shell Plating of Ships with All-Welded or Partially Welded Bottoms"

by

H. E. Jaeger
H. A. Verbeek

B. "Recent Developments in the Study of Longitudinal Strength"

by

James Turnbull

Review Prepared for

NATIONAL RESEARCH COUNCIL'S
COMMITTEE ON SHIP STRUCTURAL DESIGN

Advisory to

SHIP STRUCTURE COMMITTEE

Division of Engineering and Industrial Research
National Academy of Sciences - National Research Council
Washington, D. C.

December 15, 1953

SERIAL NO. **SSC-62**
BuShips Project NS-731-034

SHIP STRUCTURE COMMITTEE

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ADDRESS CORRESPONDENCE TO:

SECRETARY
SHIP STRUCTURE COMMITTEE
U. S. COAST GUARD HEADQUARTERS
WASHINGTON 25, D. C.

15 December 1953

Dear Sir:

The enclosed report entitled "Review of Past Structural Studies Related to the Ship and Ship Components and for Determining Loads and Strains on Ships at Sea" by J. H. Evans, Massachusetts Institute of Technology, is one of a group prepared for the Committee on Ship Structural Design to assist it in assessing the present state of knowledge of the motions of and stresses in ships at sea and of the structural aspects of brittle fracture. These reports have materially assisted in determining areas in which research directed toward the elimination of brittle fracture in welded steel merchant vessels may be most successfully undertaken.

Other reports in this series, SSC-63 and SSC-65, have recently been published.

The report is being distributed to those individuals and agencies associated with and interested in the work of the Ship Structure Committee.

Very truly yours,

K. K. Cowart

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

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under

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I. SUMMARY

1. Basic heart of plate stress distribution across a ship cross section complies well with simple beam theory but with local deviations sometimes evident in such locations as fore and aft stiffener attachments to plating.

2. While both riveted and welded ships experience occasional structural difficulties, they have been more numerous and severe in welded ships. In welded vessels cracks appeared both to initiate and to propagate more readily.

3. Poor welding workmanship, poor design details, inadequate material or physical and metallurgical notches do not appear to satisfactorily provide the full explanation of welded ship failures.

4. Plating panels which are unfair in the unloaded condition of the ship are more prevalent in welded construction than in riveted. When loaded, the stress sustained by such panels depart from the stress distribution predicted by the simple beam theory and cause a lack of uniformity of stress that may contribute to crack initiation and crack propagation.

5. A means of estimating ship bending moments making possible more precise evaluation of the variable dynamic nature of the loading is desirable. Shock loading design criteria are particularly necessary.

II. STATIC TESTS

A. Introduction

Static experiments on ships in still water have had as their primary objective justification of the validity of simple beam theory when applied to the complex ship hull girder as represented by the longitudinally continuous material of the "midship section." Varying, of course, with the availability of ships, personnel and financial support, measurements of longitudinal ship deflection and strains in a girthwise plane (generally in the region of maximum bending moment i.e. about amidships) have been measured for known applied external loadings. Stresses inferred from the measured strains then permit the effective section modulus and moment of inertia to be deduced.

Measured deflections when related to the second integral of M/EI afford another means of checking the assumed values of the product EI. This procedure led some earlier investigators to the conclusion that to reconcile measured and calculated values of deflection, a reduced value of the modulus of elasticity must be used in place of that usually associated with steel. However, widespread rivet slip and structural hysteresis as arguments in favor of a reduced modulus are little in evidence. Recent investigations lend support to the now generally accepted view that any lack of agreement is due to an erroneous evaluation of the effective moment of inertia. For example, some of the material usually included in the calculation of the section modulus may not be fully effective. This view is generally confirmed by the value of the moment of inertia inferred from strain analysis.

The first extensive static experiments were carried out on the 200-ft. transversely framed British destroyer "Wolf"*[1] in 1903, and while in many respects setting the pattern for subsequent work, many questions were left unanswered principally because of the scanty strain data taken below the vessel's neutral axis. In 1930 two identical 310-ft. transversely framed U. S. destroyers, "Preston" and "Bruce",[15,17] were observed while being loaded in sagging and hogging respectively. Within the past three years another British destroyer, the 355-ft. longitudinally framed "Albuera"[48] was similarly loaded in hogging. These vessels were of riveted construction and were loaded by being supported on piers in drydock as the water level and internal weight distribution were varied. The "Preston", "Bruce" and "Albuera" were loaded

*Salient features of all the ship data referred to are contained in Table I and figures 7 through 19 pp. 44 through 55.

to destruction when complete buckling of deck or bottom structure occurred.

Except on the "Wolf", for which only fore and aft strains were measured, multiaxial plane strains were determined enabling principal stresses in magnitude and direction to be calculated. In all these cases, stress distribution was found to be in generally good agreement with classical beam theory even for extended ranges of loading.

Transversely framed, dry cargo or passenger type vessels of more than one deck have also been similarly studied by imposing known bending moments up to substantial magnitudes by adding and shifting weights while the ships were afloat in still water. Tankers, with their fine internal subdivision simplifying ballast shifting arrangements are ideal for such experiments and several of them have been so investigated.

The results obtained to date indicate that generally good agreement exists between stresses calculated from measured strains and the stresses predicted by the beam theory. This appears to be true even though the vessel has one or many full-width decks; whether it has corrugated, plane, or no longitudinal bulkheads; whether it is transversely, longitudinally or combinationally framed; or whether it is riveted or welded.

An interesting series of similar static tests have been performed by the British on identical cargo ships, "Ocean Vulcan"(43) and "Clan Alpine"(44) and on identical tankers, "Neverita"(36,37) and "Newcombia".(38,39) One vessel in each pair was riveted while the other was predominantly all welded. The purpose was to determine if there were any differences in structural response that might be attributed to the method of construction. As in other tests, good agreement was found between stresses computed from measured strains and those predicted from beam theory. In these investigations, however, more detailed strain measurements were made permitting an assessment of localized stresses. These results indicate that in specific areas there are discrepancies in the heart of plate stress (which will be discussed later).

For numerous cases, calculated and observed ship deflections were in good agreement and, once more, in the "Neverita"-"Newcombia" and "Ocean Vulcan"-"Clan Alpine" the differences, while possibly real, were nevertheless small. In fact, it may be concluded that the service deflections of ships built to existing standards of scantlings and frame spacing may be quite accurately predicted from calculations involving known bending moments by assuming all longitudinally continuous material fully effective, using the usual value of Young's modulus and taking into account shear deflections and insuring that thermal effects are minimized.

With the probable exception of the "Ocean Vulcan"-"Clan Alpine", no static tests appear to have been made of vessels in other than the upright position.

B. Effects of Initial Unfairness of Plating

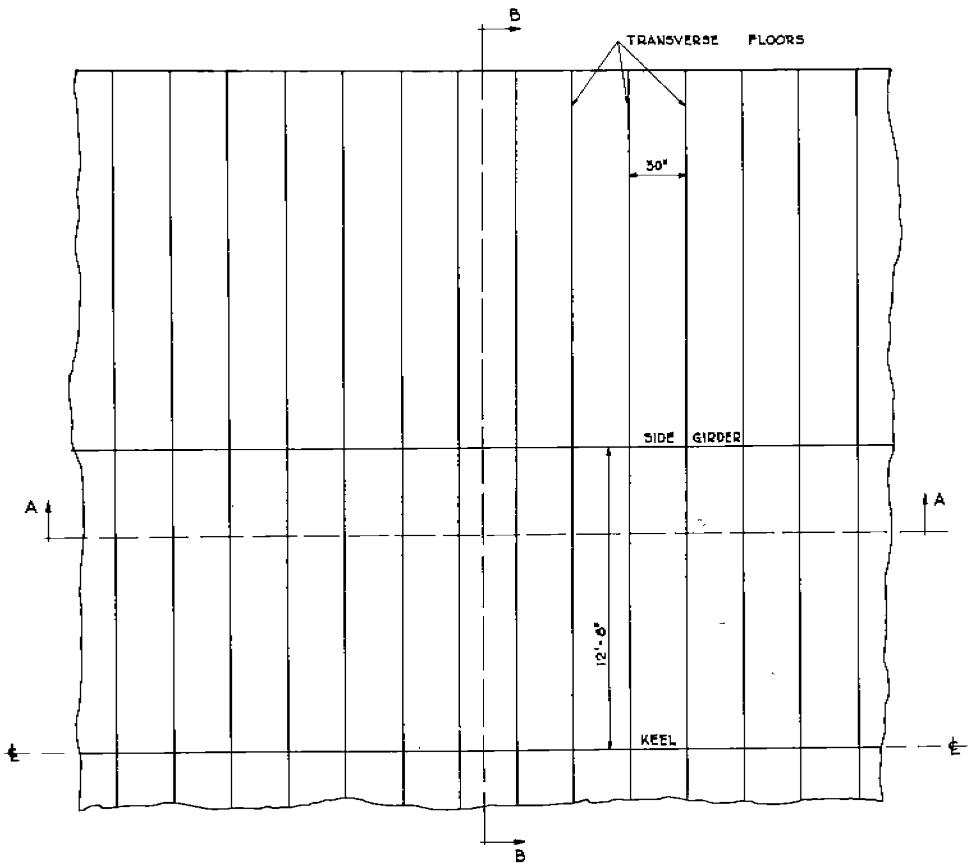
Taking note of the aforementioned small deflection differences of the four British vessels suggests the following comments. In the longitudinally framed ships the slightly greater deflection of the riveted ship may perhaps be laid to minor accommodations in some of the riveted joints. In contrast, in the transversely framed ships even though some corresponding, localized rivet slip undoubtedly occurred in the riveted ship, the greater deflection was found in the welded ship. This may have been due to the initially greater panel unfairness in the transversely framed welded ship. That such panel unfairness did exist was borne out by careful surveys. The British studies also appear to indicate that initial plate unfairness is greater in transversely framed ships than in longitudinally framed ships since the observed hull deflection in the transversely framed riveted ship exceeded the observed deflections in both longitudinally framed ships.

As another example of the possible magnitude of initial plate unfairness, in one panel of the "Philip Schuyler" (a Liberty Ship) it was observed to be 0.31 in.[31,32]

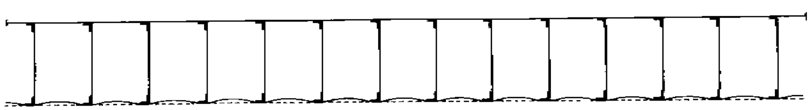
Earlier studies have also attributed greater than expected hull deflections to initially unfair plating. Increasing hogging bending moments in the "Cuyama",[16] caused an apparent increase in the vessel's stiffness. This is possibly due to the known initially unfair deck plating rather than to rivet slip particularly in view of the moderate magnitude of the loads imposed and the increasing evidence from other tests that rivet slip plays no significant part.

Nevertheless, ship deflection is not the primary aspect of the matter. Should local pating unfairness be appreciable, the curved fibers of the plating will not carry their predicted magnitudes of either tensile or compressive load. Naval architects have probably been more aware of the reduced load carrying capacity of initially bowed plating in compression than they have been in tension. For increasing tensile loading in plating with initial curvature, the unfairness must, of course, decrease and more of the material take its full share of load. However, if the initial unfairness is beyond some limiting value, even large ship bending moments will not be sufficient to substantially strain the mid panel, midthickness fibers even though at the surface the bending stresses may be three times the midthickness stress. Disproportionately high stresses, substantially constant through the plate thickness, would then be the rule at the longitudinal stiffening members. These high stresses may initiate cracking, especially in the presence of load alternations, which may be augmented by shock loadings due to slamming and/or residual welding stresses. (Figure 1) (Between differing degrees of initial bulging there may be in one a greater tendency toward crack initiation but a lesser tendency for crack propagation while in another the reverse may be the case under the variable ship bending moment.) It is worthy of note that bottom plating strain measurements, especially in the "Ocean Vulcan"-"Clan Alpine" comparison, point out the larger local fluctuations from beam theory stresses in the welded ship for both hogging and sagging. (See Figure 2) This substantiates the foregoing argument since the initial plate unfairnesses of the welded ship were generally about twice those of the riveted ship. Based upon such measurements and practical observations, these differences in plating unfairness are considered typical for the two methods of fabrication.

Thus, it appears, that in addition to the factors

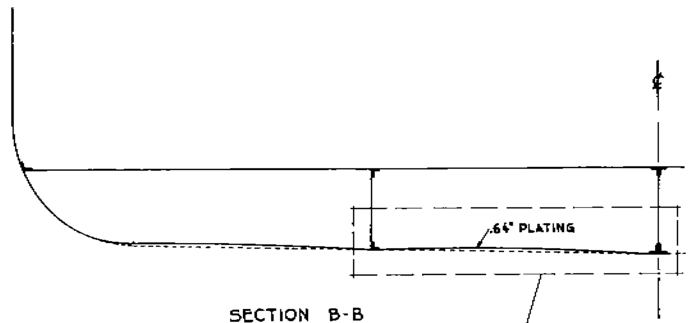


INNER BOTTOM STRUCTURAL ARRANGEMENT
OF THE "PHILIP SCHUYLER"



SECTION A-A

(NOTE: THE UNLOADED PANEL DEFLECTIONS INDICATED FOR ILLUSTRATION WERE NOT NECESSARILY PRESENT IN EVERY FRAME SPACING)



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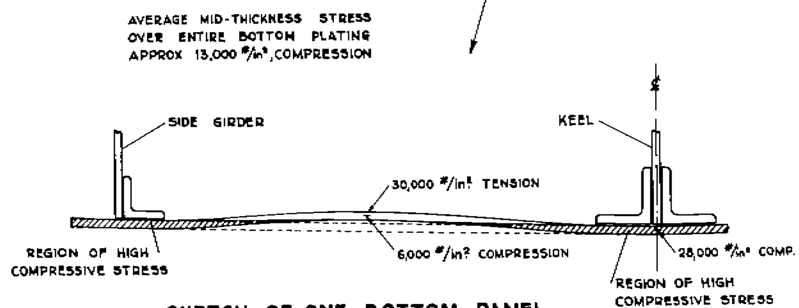
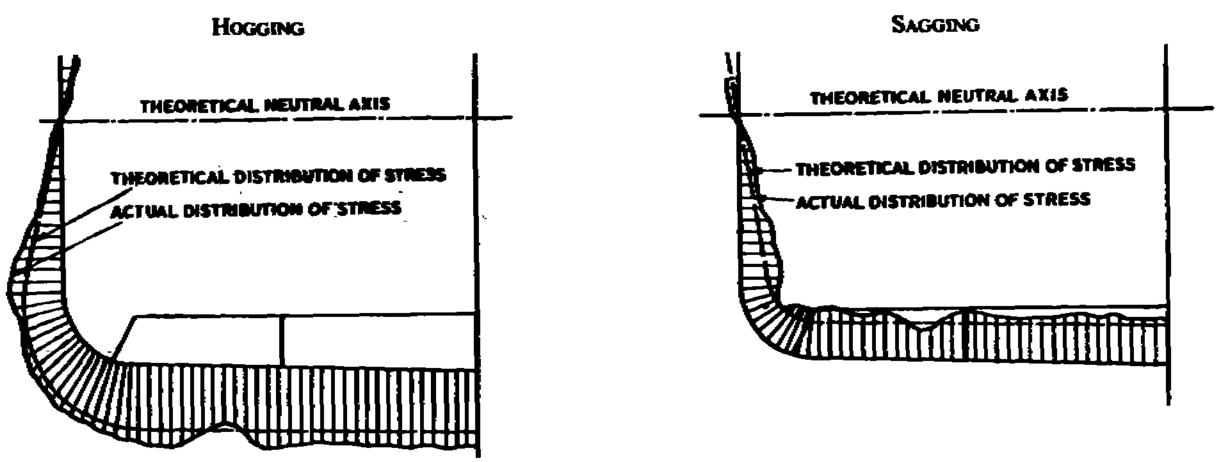


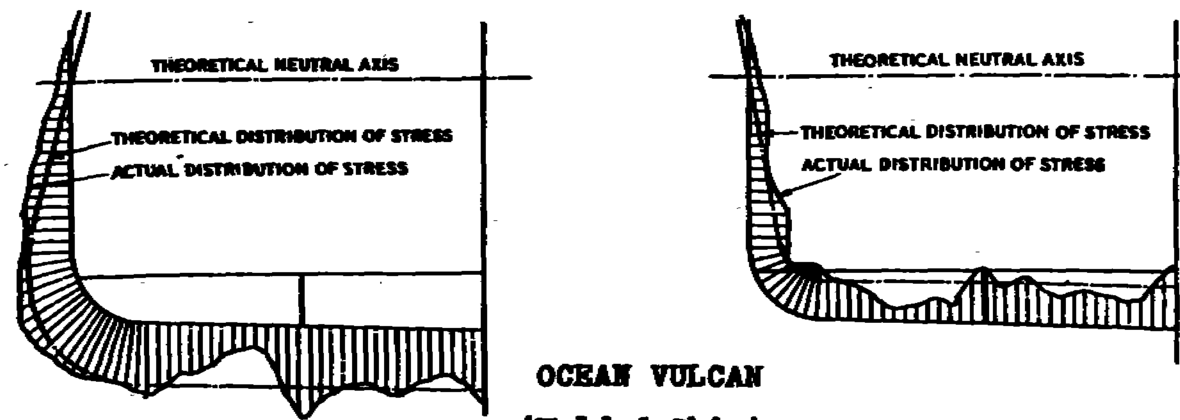
SKETCH OF ONE BOTTOM PANEL
WITH OBSERVED LONGITUDINAL STRESSES
(NOT TO SCALE)

ABOVE STRESSES WERE OBSERVED WITH A HULL GIRDER HOGGING MOMENT 80% OF THE DESIGN BENDING MOMENT

FIGURE 1



CLAN ALPINE
(Riveted Ship)



OCEAN VULCAN
(Welded Ship)

Distribution of longitudinal mid thickness stresses for bottom shell plating near amidships (Reference 49)

Figure 2

generally considered as contributing to the less satisfactory structural performance of the welded ships of wartime construction versus riveted ships, there may well be the added factor of initially unfair plating.

To date, there is still room for suspicion that the whole explanation for the difference in structural performance between riveted and welded ships has not been found. This is an all-important question and it cannot be satisfactorily explained by laying blame entirely on the steel, the welding workmanship or the design details. Emphasis should therefore be made to explain more fully the difference in performance resulting from the two fabrication processes, since there is certainly no evidence that these processes have any effect upon the external loadings of the ships. More evidence as to the characteristics and behavior of unfair plating may contribute to the present hypotheses in explaining welded ship failures.

That welded plating generally requires more care and remedial treatment than riveted is reasonable and well known and is due to thermal distortions accompanying the welding process. These then may occur on account of welding at seams and butts of plating and in way of the plating--stiffener connections through closing up of the angle including the fillet weld and axial contraction along it. (Figure 3). Incidentally, the predominantly

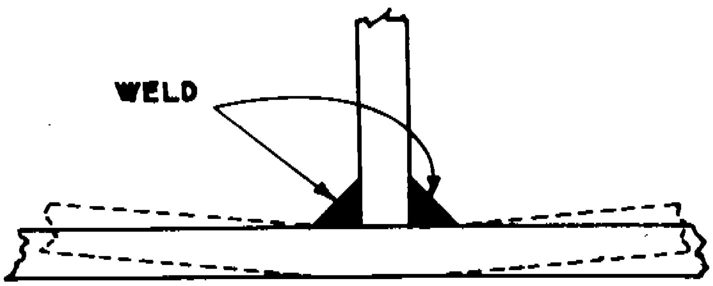


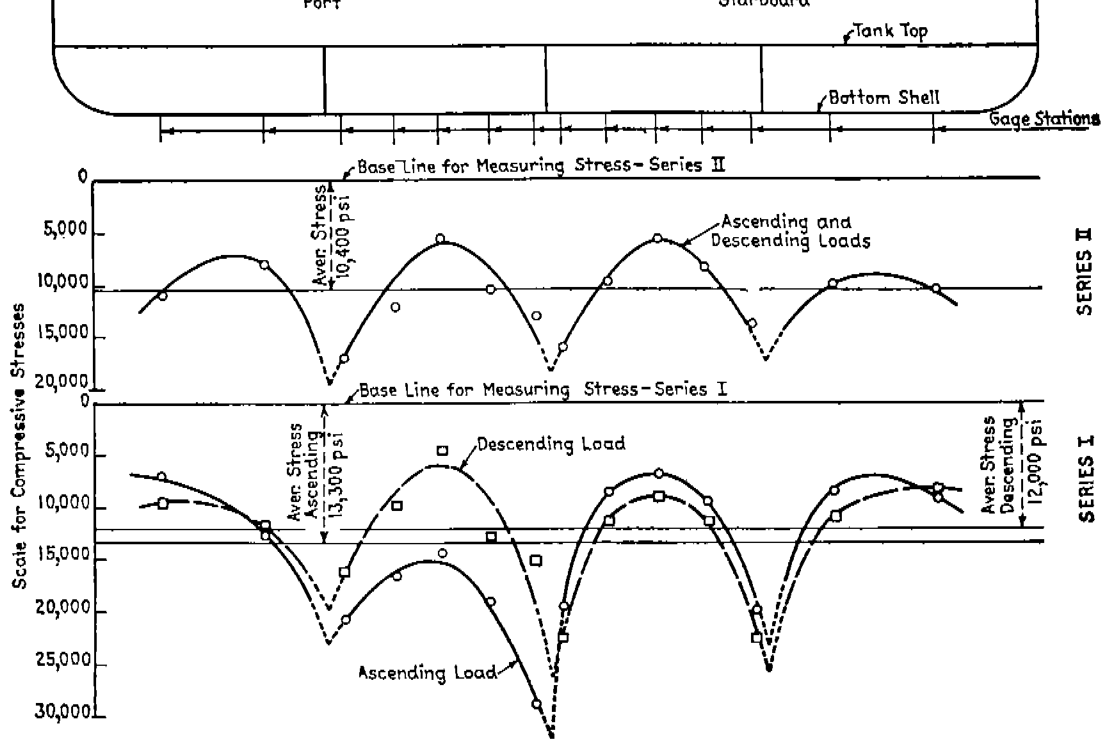
FIGURE 3

welded vessels on which the plating surveys so far quoted here were performed, namely, "Philip Schuyler", "Neverita" and "Ocean Vulcan" had riveted connections of plating to frame. It is interesting to conjecture as to the magnitudes of initial plating unfairness had these joints been welded as was the case with many of the Liberty ships and T2 tankers.

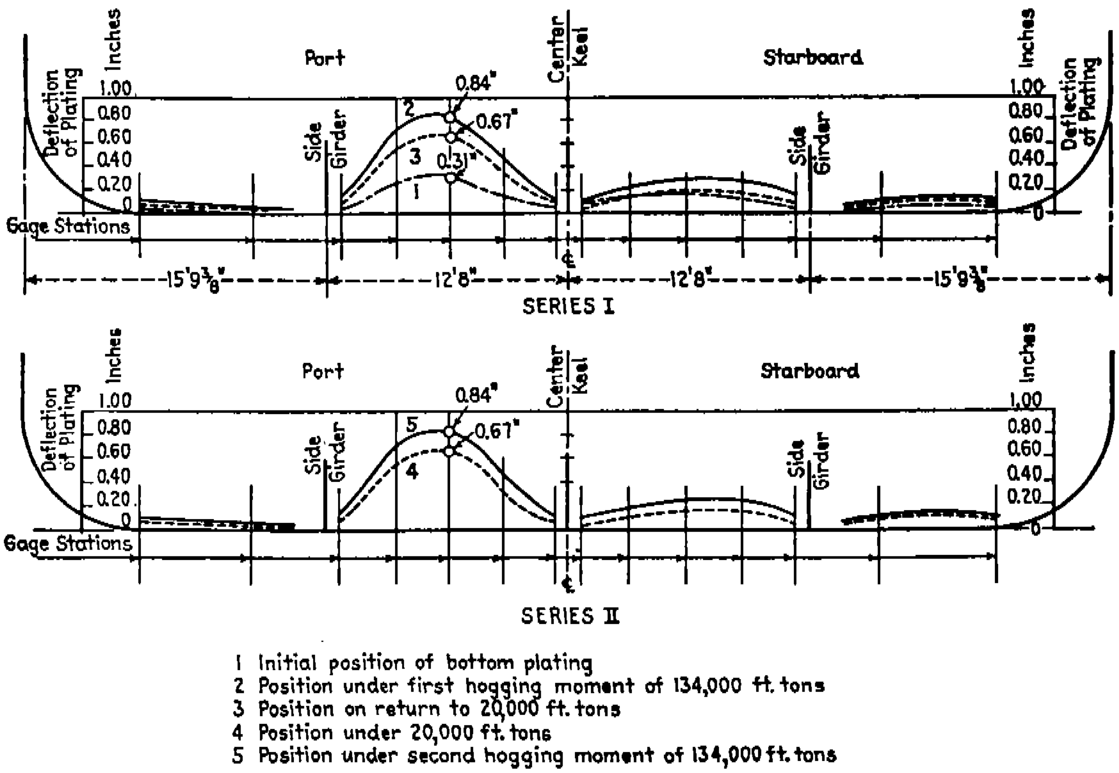
What is the allowable limit of unloaded plating unfairness? The limit, which insures against crack formation at the longitudinal panel supports under the maximum loading anticipated, is the ultimate criterion. However, it seems that excessive unloaded panel deflections may be attained not only as a result of construction techniques but by growth under water pressure and/or hogging and sagging loadings. Professor H. E. Jaeger of Delft University, and a panel chairman of the Netherlands Shipbuilding Research Association, has presented data (see Appendix A) on the growth of plating unfairness as an outcome of a recent survey of the plating of some 36 wartime, American built vessels, mostly

Liberty and Victory ships. He has observed permanent plating panel deflections of as much as 1 1/4-inch. The curvature of the bottom plating of the "Ocean Vulcan" was found to have increased during the bending tests to such a degree that fairing and additional stiffening became necessary. Previously, one panel of the "Philip Schuyler" was observed to take on increased permanent deflection under the imposed moderate hogging moments. (Figure 4)

Unfairness in the ship's compression flange, if severe and extensive enough, will decrease the section modulus even in the tension flange and so may raise the stress there in fair plating. Unfairness in both tension and compression areas naturally compounds the evils. The report of the Board of Investigation to Inquire into the Design and Methods of Construction of Welded Steel Merchant Vessels in 1946 stated that buckles were involved in very few of the casualties and, in no case, were they considered responsible for endangering the vessel, and were hence not analyzed in the report. On the other hand Dr. G. Vedeler, Managing Director of the Norwegian Bureau Veritas (Classification Society), in a recent article[51], suggests the possibility of buckling of the deck contributing to, if not causing, the failure of 6 transversely framed tankers which broke in two, not by



Longitudinal Membrane Stresses for Hog Moment of 134,000 Foot-Tons



Deflection Profile of Bottom Plating
Midway Between Two Frames Amidships

Bottom Plating Behavior on the PHILIP SCHUYLER (Reference 32)

Figure 4

brittle fracture, but by tearing in the bottom plates. It is not reported whether the ships were of welded or riveted construction, yet the implication is clear that ships may break up in a variety of ways and for diverse reasons. Here too, however, it is not unreasonable to suspect unfair plating as an extenuating contributor to structural disaster.

It may be argued that the eccentricity of loading at a riveted shell butt lap is an equally serious condition which has not proven critical. However, reflection will show that such eccentricity while causing some stress variation through the thickness of the plating nevertheless does not raise the over-all stress level in the plate. It is to be noted that in way of riveted butt laps, stiffness of the panel is increased with consequently less panel unfairness. Furthermore, the effect of the eccentricity is limited in extent to no more than about 25% of the plating in a girthwise direction between two frames because of the required staggering of riveted butts.

Although offering a smaller statistical sample, the satisfactory performance of ships built since 1945 gives reason for optimism.[53] This apparent improvement may be due to the returned pride of craft resulting in improved ship fitting and, therefore, less unfair plating. In addition, more suitable materials have been specified and greater vigilance of welded construction is exercised in design and fabrication.

The ultimate answers to be sought, if unfair plating is indeed a critical factor in the problem of welded ship failures, as seems likely, should settle the questions of how much initial deflection is admissable to limit stress and to prevent growth of unfairness. Performing the major portions of all seam and butt welds from the inside surface of the shell so as to set up an initial panel bulge in opposition to the water pressure load, and providing greater width of landing than the mere web thickness of the stiffener to back up the plating may be effective remedies. Turnbull[49] and others have advocated longitudinal framing in preference to transverse for the deck and bottom stiffening. Indeed it appears highly desirable that initial unfairness in the skin of the ship be limited by whatever means practicable. It may well be that ships over about 700 feet in length, because of their thicker plating, may not be as subject to critical bulging. At the other extreme, small vessels, despite their thin plating and consequent disposition toward washboarding, may not suffer since for those under about 200 feet in length the shell thicknesses are based more upon lateral panel loadings than hogging and sagging loads which are comparatively small.

Finally, it should be pointed out that further and more detailed analyses should undoubtedly be conducted in order to prove conclusively that the inferences drawn herein are valid and merit experimental verification.

C. <u>Applicability to Recent Structural Failures</u>

In the light of recent past experience, it is evident that answers should be sought to explain not only differences in structural performance between riveted and welded ships but between some welded ships and other welded ships.

That stress or stress history are important in cases of brittle fracture is indicated by the high incidence of such failures originating in portions of the structure with the highest basic stress levels (viz. decks and bottom).

Unusually high basic stress at time of fracture is probably not required as seen from those cases known to have occurred when levels of nominal, computed stress were only moderate.

Evidence of past high stress levels would be exhibited in plastic deformation and/or macroscopic cracks which may have characteristics sufficient to initiate brittle fracture under conditions of low temperature.

Plating unfairness bringing about a reduction in ship section modulus and a peaking of stress is generally larger in welded than in riveted ships. Furthermore, it is sufficiently variable from ship to ship and from panel to panel to encompass many shades of difference in structural performance between welded ships of the same class.

The improved structural performance of welded Liberty ships with improved details,[53] permitting them to be compared favorably with partially riveted Liberty ships may simply indicate that no one or two factors alone but several are necessary to initiate brittle fracture. The elimination of one such factor, in the form of improvements to hatch-corner design details, for example, may have been sufficient to reduce the likelihood of structural distress to a tolerable limit in the all welded ships. (Nevertheless, the stresses in localized areas of welded ships may still have been in excess of those in similar locations of riveted vessels, presuming the riveted ships to have been conservatively stressed.)

By far, the highest incidence of fracturing in the Liberty ships occurred at the hatches or the vessel's sides in the neighborhood of the Upper Deck, amidships.[52] Hazarding a reconstruction of a typical failure on the basis of the foregoing results in the following illustration.

The largest standard, calculated values of ship bending moment take place when the vessel is at, or near, its full load condition. For vessels with machinery amidships, as in the Liberty ships, the greater moment occurs almost invariably under the hogging condition which puts the deck in tension and the bottom in compression.

Any unfair bottom plating would reduce somewhat the section modulus and thus increase slightly the general stress level even in the upper deck. This unfairness may have increased progressively in service.

Panels of deck plating are bounded by the transverse deckbeams and, in the direction of the tensile stress, by the vessel's sides and the hatch side girders. The cumulative effect of high stress peaks at side and hatch due to plating unfairness being augmented locally by residual welding stresses and in the presence of a sharp discontinuity, such as a hatch corner or sheer strake cut-out, may be sufficient to initiate a crack. This crack, under auspicious conditions of temperature and loading, may subsequently spread in the characteristically brittle manner.

It may well be significant that the British sister ships of the Liberty ships, ships such as "Ocean Vulcan" and "Clan Alpine", have an additional longitudinal stiffening member supporting the main deck panels between the hatch coaming and the ship's side, thus limiting unloaded panel bulges and non-linearity of stress distribution.

If, in the upper side shell area, the panel dimension in the direction of the tensile stress is subject to greater increase for a bowed panel than for a plane one, under equal loads, then the deck stresses in a ship with such bowed panels would very likely be larger than those of a second ship with plane side shell panels. The second ship here is meant to typify the Bethlehem-Fairfield group of Liberty ships with riveted shell seams. These consequently have fairer plating and an effectively reduced panel dimension because of the lapped joint. Bottom plating of these riveted ships should likewise be more effective. The panels should also have greater shear carrying capacity from the instability aspect because of the smaller effective panel dimension.

In tankers, the distribution of cargo is subject to such wide variations that equally large hogging and sagging bending moments are possible. Unfairness of plating in these normally longitudinally framed ships is not as great as in those transversely framed. Nevertheless, this condition may conceivably add a share of stress concentration to that existing at such structural discontinuities as bilge keels and at ends of longitudinals at transverse bulkheads in T-2 tankers.

On the other hand, since plating unfairness is apparently less a factor in longitudinally framed ships than in those transversely framed, it seems not unlikely that riveted crack arrestors will prove less beneficial in T-2 tankers than in Liberty ships for example. The implication here is that crack arrestors may have alleviated the occurrence of brittle fracture largely through fairing the plating and thereby reducing stress concentrations.

III. DYNAMIC TESTS

In addition to the questions regarding the response of a ship to its service loadings, there are of course those relative to the loads themselves. The rational approach toward the ultimate in efficiency of structural design under widely variable external loadings lies in proceeding with caution to reduce scantlings and strength systematically through a series of similar designs until signs of that ultimate being reached are apparent from the latest design's showing a weakness in service. Under such a procedure riveted as well as other ships may experience some structural distress. More precise means of predicting loadings and thence strength at all points in the ship structure must be found in order for a more certain, synthetic approach to be possible. The assumption of the ship poised statically on a wave of length equal to its own and of height one twentieth the length (based on early observations) has long been the standard basis of structural comparison. The somewhat less realistic proportions this presumes for the larger vessels is taken into account by allowing higher calculated stresses therein.

Various early attempts to determine the actual nature and magnitude of external and inertia forces suffered from the lack of remote reading instrumentation for simultaneously recording the input of numerous sources. Strain readings in particular were difficult to obtain. In most cases, the investigators had to be content simply with stress peaks in strategic locations in company with visual observations of wind and waves.

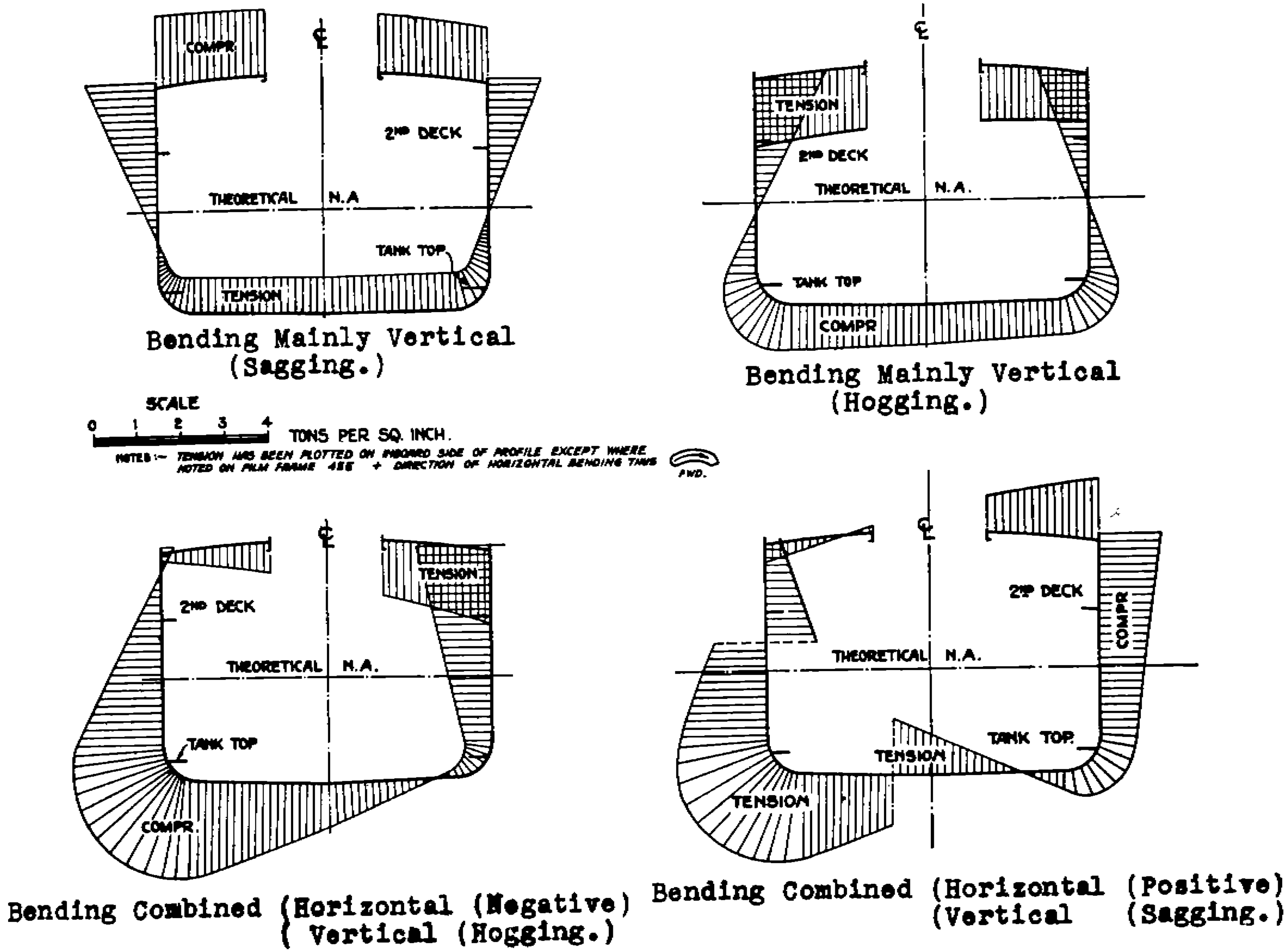
The first really notable investigation into this matter was made on the tanker "Cuyama"[14,16] in which the vessel was calibrated by means of the strains created in the deck under the imposition of known bending moments in still water. The vessel was then sent to sea and the service bending moments inferred. Only moderate seas were encountered and the data taken was insufficient to afford a breakdown into loading components such as pitching, heaving, etc.

In the classic experiments conducted by him on the "San Francisco" in 1934,[17,18,19,20,21] Professor Schnadel was fortunate in meeting waves and bad weather in the extreme. By measuring pressures on the ship's bottom sufficient to delineate longitudinal buoyancy distribution and comparing it with the known distribution of mass he was able to deduce vertical accelerations for comparison with those measured. The buoyancy and weight curves permit solution for shearing force and bending moment values and, finally, stresses based on the simple beam theory which again were compared with those derived from measured strains. Similarly, ship deflections deduced from bending moments were compared with deflections measured. As a result, Schnadel concluded that the dynamic effect upon ship bending was less severe in hogging than in sagging (a view inclined to be accepted by the "Ocean Vulcan" investigators) and he suggested use of an L/25 wave height for the standard hogging condition with no change from the L/20 height for sagging.

Following the same approach, great quantities of similar but more extensive data have recently been taken on the "Ocean Vulcan".[42,45] In this case, loadings and responses for torsional and horizontal bending in addition to the longitudinal bending case were determined. The most significant parts of the data are as yet unpublished but some findings have been made available. For example, maximum horizontal longitudinal bending moments approaching 50% of the vertical bending moments have been measured, and while these maxima apparently do not occur simultaneously, considerably augmented stresses in way of the bilge and deck edge (over the upright prediction) are assured, as was to be expected. (See Figure 5.)

Subsequent tests on the "Ocean Vulcan" and its riveted sister ship the "Clan Alpine" have compared their performance under identical loads comparable to those found at sea but imposed under the controlled conditions of still water floatation. They are reported in References 43 and 44 which are as yet unobtainable. Vibration experiments have also been made.(46) These have shown a tendency toward larger vibration amplitudes in the welded ship suggesting greater structural damping in the riveted ship also lesser stiffness in welded ship.

To date few experimental investigations have been carried out to asses the importance of "slamming", that phenomenon whereby the forward end of a vessel receives a transient impulsive loading from impact with the sea creating hull vibrations thereby sending elastic stress waves thru the ship. These stress components superimpose on other stresses and may on occasion contribute to fracture even in areas remote from the point of impact. Several investigators have expressed opinions as to the importance of "slamming", for example, Laws(10), Schnadel (17,21,26), and Bull and Baker(42). The increment of stress occasioned by this source in fibers distant from the ship neutral axis may commonly reach $\pm$ 1 1/2 tons/in.^{2}



OCEAN VULCAN

Typical Stress Distributions (Reference 42)

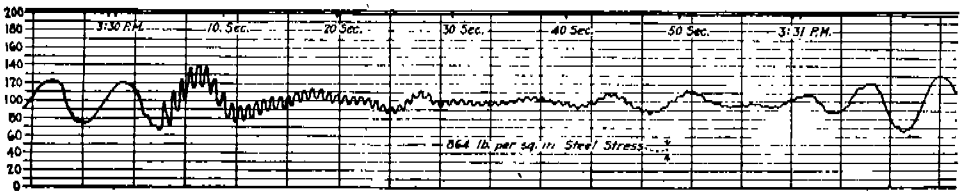
Figure 5

(See Figure 6).

More definitive information as to the shapes, sizes, periodicity and frequency of occurrence of waves that may be encountered are required. A statistical attack seems the most promising one. Collecting such data from a ship is difficult and may be inaccurate because of the oscillation of the reference platform.

Rational design procedures must eventually be formulated which permit the estimation of stress ranges within ships of varying size, speed, fineness of underwater and above water form, and mass distribution when under the influence of vertical and horizontal bending, torsion and fore and aft compression. Stress ranges taken on one particular ship, while interesting, can serve only as verification of the theory or to tentatively evaluate practical constants. To build up a sufficient mass of data to make possible a statistical analysis including all the variables above seems out of the question, but observations made on a reasonable number of ships may yield sufficient information in which satisfactory design criteria may be based. Preliminary experiments with scale models should prove valuable in this regard.

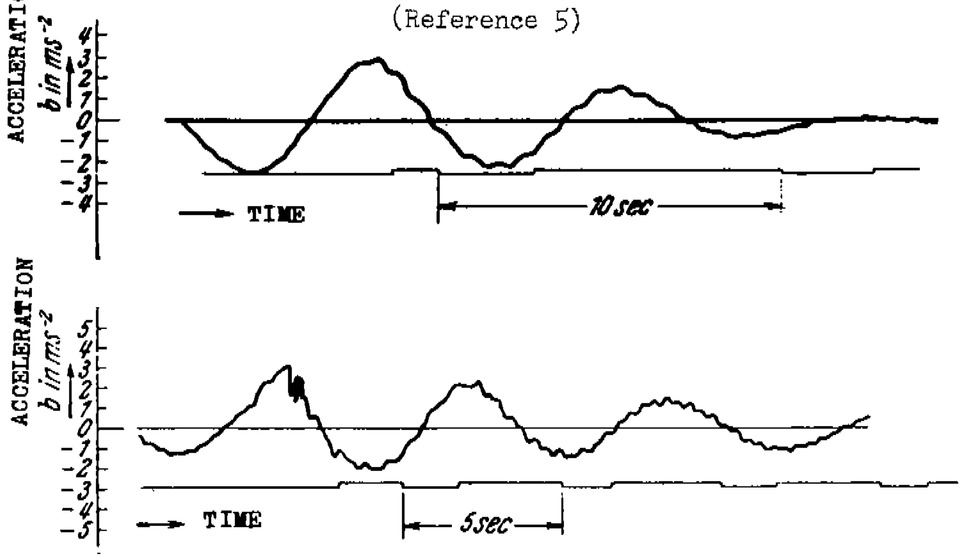
More quantitative data on the slamming phenomenon are also essential. Because of the very short period of the stress augment due to this cause and the infrequency of



WESTBORO

Stress from strain measurement just below uppermost deck, presumably amidships.

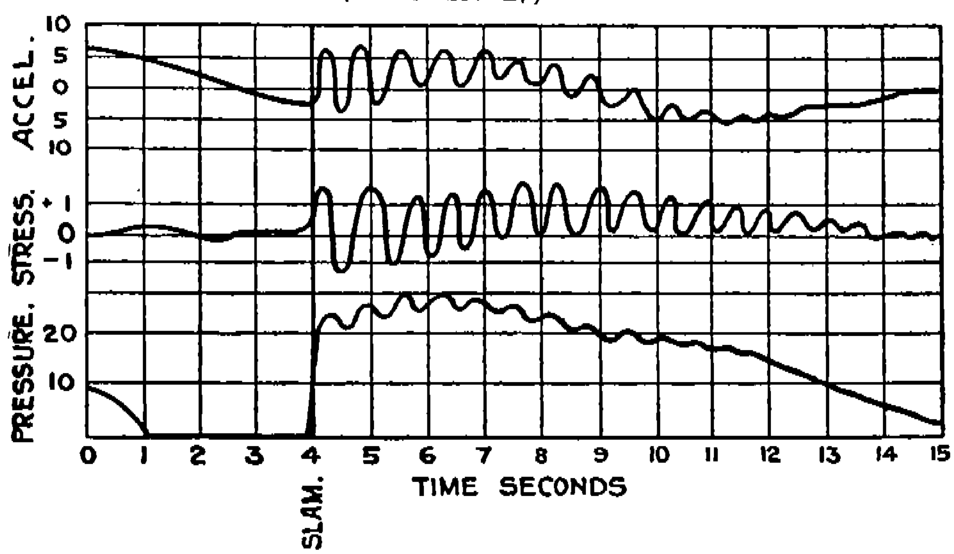
(Reference 5)



SAN FRANCISCO

Accelerations measured on deck at extreme forward end

(Reference 17)



OCEAN VULCAN

Accelerations and stress from strain measurements at deck, amidships, in ft/sec^2. Pressure measurement at bottom, forward, in feet of head.

(Reference 42)

SPECIMENS OF OBSERVED SLAMMING DATA

FIGURE 6.

its recurrence, instrumentation difficulties must certainly be anticipated. A theoretical approach here too should reduce the amount of data necessary to provide a satisfactory solution and design evaluation.

IV. RECOMMENDATIONS

It is suggested that the following studies are of importance:

1. To determine stress distributions and overall strain in plating of varying unfairness and aspect ratio when loaded in tension, shear and compression.

2. To study the growth of unloaded deflection in panels subject to lateral and/or compressive loadings with determination of upper limit for initial values to prevent increase of unfairness.

3. To make further surveys of unloaded plating panel deflections including composite built Liberty ships. Deck panels should be included.

4. To study fabrication procedures to minimize built-in panel unfairness.

5. To study brittle crack propagation in plating panels of varying unfairness.

6. To make sufficient observations of ocean waves to permit statistical prediction of period, length, height and frequency of occurrence.

7. To determine influence of "slamming" on ship behavor.

8. To develop rational design criteria based on observations whereby stresses can be predicted for variations of ship and wave characteristics.

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TABLE I

TABULATION OF FULL SCALE SHIP TEST DATA

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Data: Location	Notes	Ref.
1903	WOLF	200' Destroyer. Single deck	Transverse framing, 20" spacing. All riveted	Known hogging and sagging B.M. imposed on vessel by changing water level	Fore and aft strains	Stromeyer (mechanical) strain gage, 20" G.L.	30' aft of amidships, also about amidships only few gages below N.A. In general, stresses read on both sides of plating.	Reduced effective E determined for I incl. all longitudinally continuous material less rivet holes on tension side of N.A. Stress distribution generally in accord with beam theory. Max. deflection about 3". Vessel returned to original form upon unloading. Subsequently Hoffman reasoned thin plating in compression buckled reducing effective Ship I. E used would then be that of material.	1. 6. 9.
					Ship Deflections	Battens on ship read from dock	11 positions fore and aft		
				Vessel in seaway Moderately rough weather	Fore and aft strains	Stromeyer strain gage,	30' aft of amidships. 3 locations; keel and deck, port and starboard	Calibration of ship in dock via strains the means of inferring B.M. in a seaway. Sagging moments greater than hogging. Keel stresses greater than deck. Very small stresses set up by rolling and those during pitching and rolling less than for pitching alone. Actual stresses in rough weather much less than those calculated for standard L/20 wave e.g. keel, 5.38 vs. 7.14; deck, port, 2.88 vs. 5.30; deck, starboard, 2.14 vs. 5.30 tons/in^2. Period of encounter about 6 - 8 seconds at fastest speed viz. 13.2 knots. Stresses highest at highest speeds.	
1913	ANCON			Vessel in seaway. Full load outward bound; 2/3 load homeward Weather fair		Scissors gage, 6" G. L.	Deck at quarter points; port side and at break in bulwark rail	Strains greater with increasing pitch. No estimate of bending moment or weight distribution data given	2.
				Loading cargo	Fore and aft strains 20" G. L.	Telescopic gages	Deck and rail at after quarter point, port and starboard	Strains due chiefly to temperature variations; little if any due to loading cargo.	
1913	NEPTUNE	Collier	Transverse framing	During launching, fitting out and docking	Ship deflections	Transit sights	Several stations fore and aft	No bending moment or weight distribution data given	3.
	ORION	"	Longitudinal framing						
	JASON	"	Longitudinal framing						
1915	ULYSSES	Collier		On the ways, during launching, fitting out and docking	Ship deflections	Transit sights	Several stations fore and aft	No bending moment or weight distribution data given. Considerable question as to the effects of temperature variations on the ship deflection.	4.
	ACHILLES	"							
	ATLANTIC	Freighter							
1918	FAITH	Cargo Ship	Longitudinal framing Reinforced concrete	Vessel in seaway. one voyage. Moderately rough weather	Fore and aft strains	Recording extensometers	8 points	Maximum range of stress 3.1 tons/in^2.	5.
1919	WESTBORO	410' - 6" Cargo Ship Three decks	Transverse framing	Vessel in seaway. one voyage.	Fore and aft strains	Recording extensometers	Up to 22 points simultaneously in a transverse section admidships.	Maximum range of stress about 3.6 tons/in^2. 2000#/in^2 stress increase due to pounding indicated.	

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Reference numbers refer to the Bibliography.

TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Data: Location	Notes	Ref.
1926		235' Collier 390' Freighter 400' Tanker 440' Tanker	Transverse framing. " " Longitudinal framing.	Loading and discharging cargo.	Fore and aft strains	Mechanical strain gages. 24"-36" G.L.	Region of uppermost deck (one exception)	Bending moments calculated. Stress calculated from observed strain and modulus of 13,000 tons/in². Virtual section modulus calculated from strains and bending moments above. Strains observed at no more than four positions. Bending moments relatively small. No temperature compensation.	7.
	LONDON MARINER KENMORE	450' Express Freighter 363' "		Vessel in seaway. one voyage. Weather fine to moderate.	Wind velocity and period of wave encounter.	Anemometer and stop watch.		No weight distribution data.	
					Pitch and roll angles and periods.	Bubble level.			
					Ship speed.	Log and stop watch.			
					R. P. M.	Stop watch			
					Fore and aft strains. (Maximum values)	Mechanical strain gage. 24"-36" G.L.	Uppermost deck and sheer stroke.		
	SAN TIRSO SAN FRATERNO	420' Tanker 420' Tanker		Vessel in seaway. Fairly rough weather. Waves about 35' high.	Same as for London, Mariner, and Kenmore except no speeds or R.P.M.	Same as for London, Mariner and Kenmore.	Fore and aft strains taken in deck, sheer strake, bilge, bottom and keel; all near amidships.	No more than 4 strain gages used simultaneously. Maximum stress range 10.5 tons/in² vs. 12.7 tons/in² for standard wave, i.e., about 6.7 tons sagging, 6.0 tons hogging.	
				Undocking or discharging cargo.	Ship deflections.	Telescope sight.	Several positions fore and aft on deck.	Account taken of shear deflection in calculation.	
1927		Car float	Transverse framing, 24" spacing, plus 6 full depth longitudinal girders.	Thermal	Hull deflections.	Transit sights.	Deck, port and starboard	Data taken throughout one day. Temperature compensation formula for deflection proposed.	8.
				Hogging moment imposed in still water	Fore and aft strains.	2 mechanical strain gages, 20" G.L.	40 stations, mostly amidships. Some fore and aft in stringer plate.	For ship I with all material considered fully effective except rivet hole area in tension, calculated stresses 10% greater than determined via strain measurements. Strain gages not located to insure determination of heart of plate stresses. Fairly good agreement between calculated and observed deflections when corrected for temperature.	
					Hull deflections	Transit sights.	Deck, port and starboard		
1929		425 Shelter deck cargo vessel.	Transverse framing.	Vessel in seaway. one voyage. outward bound in ballast; homeward bound, loaded Weather severe.	Pitching and rolling.	Pendulums		Maximum angles; pitch, 9°; roll, 19°.	10.
					Uniaxial strains	Recording mechanical strain gage.	Various locations and directions.	Tests performed primarily to evaluate local effects rather than general behavior. Determination of racking showed negligible amounts. Transverse frame deflections measured. Some random service stress and estimated wave proportion data given. Transverse stresses small but stress concentrations in way of discontinuities and importance of transient impacts considered to be appreciable.	

TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Data: Location	Notes	Ref.
1929	GOTTINGEN	Cargo Ship	Transverse Framing	Vessel in seaway.	Fore and aft strains.		At break in bulwark, forward of amidships.		11.
					Wave profile.				
					Pitch and roll.				
1930	ODIN	Cargo Vessel Two decks	Transverse framing. All riveted.	Vessel in seaway. Sea calm	Fore and aft strains.	Dial gage extensometers.	Across upper and second decks, in side shell between decks and just below neutral axis.		12.
					Ship deflections.	Telescope sight.	Forward side of bridge.		
1930	CUYAMA	455' Tanker	Transverse framing. 27" spacing. keelsons and side stringers. All riveted.	Known static bending moment in still water.	Ship deflections.	Telescope sight and cantilever extending 70' aft of bridge.	On deck.	Known B. M. and inferred stress in deck gives section modulus 4% less than standard calculation. From calculated position of neutral axis and measured deflection, E = 24,000,000 lbs/in^2. Average B. M. over selected span assumed constant in relating B. M. to deflection for EI. No correction for shear deflection. Maximum B. M. - 129,000 to + 125,000 tons ft. or about ½ range of design B. M. Section modulus values increase slightly with B. M. No temperature corrections since readings taken at approximately constant temperature.	13. 16.
					Fore and aft strains.	3 Extensometers 300" G. L.	Upper surface of deck only.		
				Vessel in seaway for 50 days. weather mostly fine.	Fore and aft strains.	3 Extensometers associated with reostat type.automatic recorder.	Upper surface of deck only.	B. M. for sea conditions inferred by comparing strains at sea with those under known.B. M. in still water. Approximate maximum seaway B. M. encountered; 56,000 tons ft. sag and 73,000 tons ft. hog. Normal deep load still water. B. M. 120,000 tons ft. sag. Design B. M. (total), 234,740 tons ft. Seaway hogging moments greater than sagging. Buckling occurred in deck plating. For most part, only maximum values of strain measured. Instrument activated thus for 9½ minutes then arranged to give continuous readings for ½ minute and released.	14. 16.
1930	PRESTON	310' Destroyer single deck	Transverse framing, 21" spacing All riveted.	Known sagging B.M. imposed on vessel by changing water level in dock and ballasting.	Multi-directional strains.	Portable mechanical gages. 10" G. L.	Girthwise around vessel at 3 stations; one at each quarter point and one amidships.	Vessels tested to ultimate load. PRESTON deck and longitudinals buckled at B. M. of 44,000 tons feet. Corresponding inferred stringer plate stress, 13.7 tons/in^2. (Design B.M., 12,300 tons ft.) BRUCE bottom plating buckled at B.M. of 36,000 tons ft. Corresponding inferred stress, 10.35 tons/in^2. (Design B.M., 18,600 tons ft.) Close agreement between fiber stresses as calculated and as inferred from strain measurements. Beam theory stress distribution valid. Therefore, I calculation with no rivet hole deductions, all continuous structure effective and full allowance for lightening holes justified. At ultimate loads, 3% apparent reduction in I probably due to plating buckling. Heart of plate stresses found by measuring strains at plate surface and on buttons of known thickness. Not such good agreement between inferred and calculated shear stresses. Observed and calculated total deflections based on E of material and I as above in good agreement. Shear deflection 8-10% of total deflection. All readings and tests made at night so limiting temperature difference to 8°.	15. 27.
					A few fore and aft strains.	Dial gage with mechanical magnification. 300" G. L.			
					Ship deflections.	Theodolite	Keel and deck edge, port and starboard.		
					Transverse change of shape.	Dial gages.	Amidships.		
1931	BRUCE	Identical to PRESTON	Identical to PRESTON	Same as PRESTON but hogging.	Same as PRESTON.	Same as PRESTON.	Same as PRESTON.		

TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Location	Notes	Ref.
1934	SAN FRANCISCO	430' Cargo-passenger vessel. Three decks superstructure 25% of vessel's length.	Transverse framing. All riveted.	Vessel in seaway. Hamburg-Panama Canal - Vancouver and return. Very rough weather.	Shape, length and height of waves.	Electric contacts on shell lighting lamps simultaneously photographed.	Stations 66' apart fore and aft; 12"-16" apart vertically.	Buoyancy distribution determined from bottom pressures and ship weight distribution altered to suit, the difference being the acceleration component due to ship oscillation. These accelerations checked by accelerometer. Buoyancy and virtual weight curves yield bending moments and stresses for comparison with those measured. (Strains read on both surfaces of plating.) Similarly, deflections deduced from bending moments compared with those measured. Transverse strains taken into account by increasing E 2.5% for influence of transverse stiffening. Hogging bending moment reduced but sagging bending moment increased by dynamic forces. Therefore, ship hove to worst reasonable condition for hogging but steaming ahead worse for sagging. Maximum stresses when ship among waves of ship's length. Then maximum values in deck, 9250 lbs/in^2 tension and 13100 lbs/in^2 compression. (Compressive stress includes 2700 lbs/in^2 in transient pounding loading.) Good agreement between deduced hogging bending moments and strains. Not so good for sagging. Satisfactory agreement between observed deflections and those calculated including shear deflection and dynamic effects. Maximum pitching angles + 12° to - 10°. Maximum angle of roll 20°. Maximum heaving acceleration 8.2 ft/sec^2. One wave encountered 610' x 54' in which ship center of gravity oscillated 39.4'. Pounding shocks noted.	17. 21. 26. 20.
						Stereo-photographs			19.
					Pressure on ship's bottom.	Flexible diaphragms operating scratch recorders.	8 stations.		
					Accelerations, rolling and pitching angles.	Recording, pendulum type accelerometers and gyroscopes.	About 5 positions throughout length of ship; port and starboard (accelerations)		18.
					Ship deflections	Continuous photographic record of 8 light sources.	Light sources spaced over major portion of ship's length, uppermost deck, port.		
					Strains fore and aft and at 45°.		A few positions on uppermost deck, mostly on stringer plate, port, and at a point somewhat forward of amidships and below the neutral axis.		
					Deflection of deck and bottom plating panels.				22.
1935-36	BEAVERBRAE	495' Cargo Vessel superstructure 25% of vessel's length. Two complete decks.	Transverse framing.	Vessel in seaway. Two voyages. London-Halifax and return. Rough weather.	Uniaxial strains (Fore and aft in deck)	Portable strain meter 3" G. L.	Superstructure house ends.	Zero strains for vessel in drydock. Readings taken during loading and at sea whence strains represented aggregate imposed values. Length of longest waves encountered about half ship's length. Decreasing period of encounter increased transient peaks of stress. Therefore, with longer waves, (up to ship's length), mean range of stress higher but ratio of maximum to mean stress ranges probably lower because encounter period more nearly synchronous with vessel's pitching period thus reducing pounding and shipping of water. Reducing speed analagous. Transient stress peaks up to about 1 3/4 tons/in^2 in excess of mean peaks. Maximum range of stress on deck amidships, clear of discontinuities, 5 tons/in^2. Maximum angle of roll, 30°. Maximum angle of pitch, 7°.	23.
						4 dial gage extensometers. 72" G. L.	On upper deck in way of house front forward of amidships and abreast of house, amidships.		
1935	DEWEY	334' Destroyer Forecastle. 33% ship's length	Longitudinal framing	Vessel in seaway	Strains (Fore and aft?)	At 50 stations by recording scratch gages of latch-key type. Maximum and minimum values over one or two pitching cycles taken simultaneously at 18 stations with Huggenberger gages.	Two transverse sections; amidships and 38% vessel's length from bow.	Scratch gage data inconsistent. Only small bending loads encountered. Maximum stress range about 4.5 tons/in^2. Rather definite evidence that neutral axis lies above its calculated position. Little information given.	24.
					Ship deflections	Photographing row of lights	Lights extending over 55% vessel's length		
1936	FLUSSER	334' Destroyer Forecastle 33% ship's length		Vessel in seaway	Strains, mainly fore and aft but at two points in side plating strains read at 45° and 135° to neutral axis.	Recording scratch gages of latch-key type and one rosette of mechanical gages. photographically recorded.		Strain data not extensive and of very small magnitudes although including some for shear. Not feasible to do more than follow range of stress variation. Little information given.	
				Known static bending moments in still water	Strains			Bending loads small. Very little information given.	
					Ship deflections				

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Date	Ship	Ship Type	Construction	Loading	Data			Notes	Ref.
					Type	Method	Location		
1937	SAN CONRADO	460' tanker	Combination framing; longitudinal, deck and bottom; transverse sides.	Vessel in seaway. One voyage. Outward bound in ballast; homeward, full. Weather; fairly severe to moderate.	Fore and aft strains	2 dial gage extensometers 30" G.L. and one portable strain gage 3" G.L.	Upper surface of deck plating. During rough weather; within short bridge house on ship center line only.	Maximums, minimums and mean values of strain read, also the time taken for a number of such cycles. Maximum range of stress measured (In bridge house, forward of amidships) 4.3 tons/in^2. Estimated wave dimensions given and related to stresses. Mean range of stress appears to reach a maximum at wave lengths equal to ships length but isolated maxima probably increase with increase of wave proportions. Athwartship strains found to be negligible. Maximum angle of roll, 18° Maximum angle of pitch, 9°	25.
1938	BAGLEY	334' destroyer, forecastle 33% ship's length		Vessel in seaway	Strains	Recording scratch gages of latch-key type.		Time capacity of gages increased over those used previously. Bending moments of significant value obtained. Very little information given.	24.
1939	PHOENIX	600' light cruiser		Vessel in seaway. Weather calm	Strains	Recording scratch gages of latch-key type. Stress counter		Stress counter designed to keep cumulative count of number of times stress exceeded predetermined value. Very little information given.	
1939	DUISBURG	465' cargo vessel 2 complete decks. Superstructure 25% of vessel's length	Transverse framing	Loading cargo	Strains (mostly tri-axial) 7.9" G.L.	Photographic recording of Zeiss "orthotest" instrument dials	7 points throughout ship depth but arranged in two transverse planes aft of amidships 35' apart.	Ship loading proceeded uniformly throughout length. Resulting strains, ship deflections and plate panel deflections very small. Similarly when at sea bending loads very small. Data on thermal straining given.	28.
					Ship deflections	Theodolite	Deflection of point amidships from line of sight 194' long.		
					Plate panel buckling	Dial gages	Various points on shell		
				Vessel in seaway. Weather calm	Ditto for condition loading cargo.	Ditto for condition loading cargo.	Ditto for condition loading cargo.		
1943	SHILOH	503' tanker (T2 type) Bridge superstructure 7% of vessel's length	Longitudinal framing, fluted bulkheads. All welded	Known hogging and sagging B.M. by filling various tanks in still water	Strains, mostly bi-axial at 325 points	Mechanical strain gages 2" and 10" G.L. Electric resistance gages 1" G.L.	Mainly in way of midship section. Also on stringer plate aft, bridge deck fashion plate aft, bridge deck and longitudinal-transverse bulkhead intersection.	Maximum bending moments imposed about equal to vessel fully loaded on standard wave, viz. 279000 tons ft. hogging, and 209000 tons ft., sagging. All strain readings taken at night. Hence, range of structure and air temperatures not greater than 6°. Considerable data unreliable but stress distribution essentially as for simple beam theory. Measured stresses less than calculated by 17% in hogging and 9% in sagging. May be due to discrepancy in computed moment of inertia, computed B.M., accuracy of instruments and methods or reading strains on one side of plate only. No conclusion relative to effectiveness of longitudinal bulkhead in shear. No excessive stresses in deck at transverse bulkheads or in bridge fashion plate aft. Bridge deck stresses less than in upper deck. Notable stress concentrations at longitudinal-transverse bulkhead intersections. In hogging, measured deflection 10% less than calculated. In sagging, fairly good agreement. Calculated deflections are for bending only; excluding shear.	29.
					Ship deflections	Transit sight	Transit on poop, targets amidships and forward		
1943	PURNELL CADILLAC	Identical 595' Great Lakes ore carriers	Combination framing. All welded except riveted side shell seams.	All loaded similarly under one hogging and one sagging B.M. in still water	Strains in rosettes at least in way of local stress concentrations	Dial gages 10' to 48' G.L. Mechanical gages 2" and 10" G.L. Huggenberger gages 1" G.L.	In each midship section plane and various localized areas. Also near after quarter point in side shell of CADILLAC	All readings taken at night. No unusual departure from simple beam stress distribution noted. The combination riveted and welded structure works integrally and in a homogeneous manner. Negligible longitudinal stresses in deck plating between hatches. Local areas of strain measurement included main deck access door opening, cargo hatch opening, hatch coaming and gunwale.	30.
	CHAMPLAIN HUTCHINSON	Identical 605' Great Lakes ore carriers.							

TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Data: Location	Notes	Ref.
1944	PHILIP SCHUYLER	416' cargo ship (Liberty) Two decks superstructure 20% vessel's length	Transverse framing 30" spacing. All welded except shell-frame connection riveted	Known hogging, sagging and torsinal moments applied in still water	Strains in rosettes	About 200 electric resistance strain gages 13/16" G.L. and about 200 mechanical gage stations, 2" and 10" G.L.	In way of #3 hatch, forward of amidships at decks, sides, inner bottom and bottom shell. Also extensive readings in bottom shell amidships and in way of #3 hatch corners.	Strains read only at night. Only one sagging condition of loading. Data taken and discussed for hogging conditions only, maximum of which was 134000 tons ft. or about 80% standard bending moment. Second deck and inner bottom attained 50% theoretical beam theory stresses resulting in 5% reduction in predicted Ship I and correspondingly small increase in predicted deck and bottom stresses. High degree of transverse strain restraint in main deck but not in side shell i.e. biaxial tension in deck but uniaxial in side. Initially bulged plating in bottom brought about peaks and hollows of heart of plate stress under compressive loading. Most severely bulged panel took on permanent set. Thereafter stress distribution in bottom plating became symetrical about the keel but average stress loading accepted by it somewhat reduced apparently to be borne by other members such as vertical keel and side girders. Average stresses in fair agreement with beam theory distribution reaction to tensile loading. Torsion tests showed nothing unusual. Torsinal moment and shear stresses low.	31. 32.
					Plate panel deflections	Sagitta gage 10" G.L.	At all points of strain measurement except hatch corners.	Plate panel deflections measured and associated with strains read on one plate surface only in order to deduce heart of plate stresses.	
					Ship deflections	Transit sights	Surveyor's level rod at 16 stations port and starboard on deck.	Ship deflections read during the day. Calculated shear plus bending deflections agree well with those observed when corrected for temperature. Shear deflection about 15% of total.	
1945	VENTURA HILLS	503' tanker (T2 type) Bridge superstructure 7% of vessel's length	Longitudinal framing, fluted bulkheads	Known hogging and sagging B.M. imposed on vessel in still water	Strains			Range of applied bending moment; 244000 tons ft. sag to 284,000 tons ft. hog i.e. about 100% bending moment on standard wave. Direct and shear stress distribution throughout in good agreement with beam theory. Longitudinally fluted fore and aft bulkheads carry their full share of bending moment and shearing force.	33. 34. 35.
					Ship deflections				
1944	NEVERITA	460' tanker. Bridge superstructure 10% of vessel's length	Combination framing; longitudinal framing in deck and bottom, transverse framing in sides and bulkheads. All welded except shell-frame connection riveted	Various known hogging and sagging B.M. imposed on vessel in still water	Strains, fore and aft and in rosettes	Mechanical gages 5" G.L. Mechanical extensometers 100"-120" G.L. Electric resistance gages 13/16" G.L. Acoustic gages 4 3/4" G.L.	Transverse section amidships, between frames and webs at mid length of tank.	Range of bending moments applied; 158,000 tons ft. hogging to 86,000 tons ft. sagging. Plate panel deflections and surface strains used to determine heart of plate stresses. Electric resistance strain gage results not considered reliable. Stress variation fairly close to beam theory prediction. Transverse stresses 20% of longitudinal stresses noted. High local bending deflections and stresses sufficient for plastic yielding in transversely oriented bottom plating panels. Bending stresses much higher than heart of plate stresses in such panels. With all continuous material included in I and modulus of elasticity taken as that for the steel, calculated ship deflection including bending and shear about 10% greater than measured. Heart of plate stress of 1 ton/in² in deck appears to have been induced by 30° temperature difference between deck and bottom. Temperature range small during taking of data.	36. 37.
					Plate panel deflections	Mechanical arc rise gage, 6" G.L. "Mushroom" slope difference lever system 3" G.L.	Same as for strains	NEVERITA - NEWCOMBIA Comparison Stress distribution and longitudinal deflections in the two forms of construction show no major differences. Local bending stresses in the particular panels examined were in general less in the riveted ship owing to fairer plating and the stiffening influence of riveted overlaps. Stress concentrations around large structural discontinuities approximately the same for the two forms of construction.	
					Ship deflections	Theodolite, water-level tubes and drafts.	About 6 stations fore and aft (except drafts)		
					Thermal effects	Thermometers and thermocouples			
1945	NEWCOMBIA	460' tanker similar to NEVERITA	Similar to NEVERITA but all riveted except deck plating butts, sheer to stringer plate connection, longitudinal bulkheads to bottom plating, longitudinal stringers to side shell and longitudinal bulkheads also various brackets.	Similar to NEVERITA	Same as NEVERITA except no examination for thermal effects			Range of bending moments imposed about 80% range calculated for vessel on standard L/20 wave. Results from electric resistance gages more satisfactory than in NEVERITA. Probably fairly even temperatures throughout ship during observations. Stress distribution fairly close to beam theory. Transverse stresses averaged about 1/6 longitudinal stresses, i.e. about 50% product of Poisson's ratio and longitudinal stress. Little vertical shear carried by other than side shell and longitudinal bulkheads. Average shell plating unfairness about 6% of plate thickness, i.e. negligible. Probably because of this greater fairness than in NEVERITA, local bending stresses less. Plating unfairness has little effect on local bending stresses from lateral water pressure loading. Local bending stresses sometimes greater than heart of plate stresses. Where axial stresses are imposed on plating having initial deflection due to unfairness or lateral pressure, total stresses do not conform to Principle of Superposition. For Ship I and E, as for NEVERITA, total calculated ship deflection about 6% greater than measured. Shear deflection averaged 7% bending deflection.	38. 39.

TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Data: Location	Notes	Ref.
1944-45	NISO	460' tanker. Bridge superstructure 10% of vessel's length	Similar to NEWCOMBIA	Vessel in seaway. One voyage outward bound in ballast. Weather moderate to fairly severe	Direct strains	Extensometer with dial gage and remote reading extensometers with variable induction choke as well as electric resistance gages	On deck amidships and various other locations	Voyage of exploratory nature to determine suitability of various instruments and ranges of values to be expected. Relative merits of each briefly discussed. Maximum angle of roll ± 12°, pitch ± 6°.	40.
					Ship deflections	Ciné camera and target boards	Camera on poop, target boards on forecastle and after end of bridge		
						21' steel trusses for deflection references and hinged 80' trusses. Both types remote reading	On deck forward		
					Water pressure on hull	Diaphragm type pressure gage with remote reading variable inductance choke.	Forward of bridge		
					Wave profile	Series of electrical contacts on ship's side closed by presence of sea water.	One vertical row, amidships		
					Wind forces	Deflecting wind board with remote reading variable inductance choke.			
					Accelerations	Tridirectional accelerometer, using unbonded electric resistance strain gages			
					Roll and pitch angles and periods	Gyroscope and 2 stereoscopic cameras	Cameras port and starboard on bridge.		
1945	FORT MIFFLIN	503' tanker (T2 type) Bridge superstructure 7% of vessel's length	Longitudinal framing, fluted bulkheads (Identical to VENTURA HILLS)	Known hogging and sagging B.M. imposed on vessel in still water	Strains in rosettes	Mechanical gages 10" G.L. Electric resistance gages.	Transverse section about 60' aft of amidships and in way of longitudinal - transverse bulkhead intersection. Generally on one surface only.	Range of applied moments, amidships; 282,000 hogging, 215,000 sagging vs 168,000 hogging and 230,000 sagging for ship fully loaded and on standard L/20 wave. 282,000 tons ft, amidships corresponds to 228,000 ton ft. at test section plus 1,760 tons shearing force. Sagita gage readings with strain readings permit solution for heart of plate stresses. All observations made at night. Remarkable agreement with results from VENTURA HILLS. Effectiveness of two types of bracket at longitudinal-transverse bulkhead intersection evaluated. One 12'-2" x 30" panel of 3/4" plating in bottom examined. Built in deflection 14% thickness. Additional deflection linear with water head reaching 3% additional for 27' head. Any increment due to edge compression loading too small to be measured. Agreement between measured and calculated hull deflections very good. Shear deflection and small temperature correction included. Temperature correction .058" deflection / °F of air temperature for 414' length of ship.	41.
					Plate panel deflections	Sagitta (arc rise) gage 10" G.L.	Transverse section 60' aft of amidships		
						Dial gages	Bottom plating panel		
					Ship deflections	Transit sights	On deck, port and starboard at 5 points on a 414' length of ship		

TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Data: Location	Notes	Ref.
1946-47	OCEAN VULCAN	416' Cargo vessel. Two decks	Transverse framing All welded except shell to frame connections	Vessel in seaway. 8 voyages Gt. Britain to U.S. and return. Westbound in ballast. Eastbound loaded.	Strains fore and aft and girthwise	Electric resistance gages.	Sheer strake and inner bottom on station near amidships, both sides of plating	Data tabulated here recorded synchronously as well as ships speed engine speed and shaft torque. Photographic recording speed for all data; 2 frames/sec. for 18 minutes before reloading. Other data recorded at random included stereoscopic photographs of sea conditions observations with portable instruments of local stresses and accelerations as well as continuous record of stress reversals by statistical strain gage recorder. Water pressure distribution determined from pressure gages and wave profile indicators. Inertia forces determined from known distribution of mass and observed accelerations. Forces from water pressures and accelerations combined to give net forces making possible the computation of bending moments, shearing forces and torsion moments, etc. Most severe waves encountered estimated to be 600' x 35'. Corresponding ranges of values; vertical bending moment 190,000 tons ft., stress in sheerstrake 8 tons/in^2 (to be added to still water values). Horizontal and vertical bending fequently in phase but maximum values of one coincident with minimum values of the other. Torsion moments not over 1/2 ton/in^2 and not coincident with maximum vertical bending moments. Maximum fore and aft axial compression 1/2 ton/in^2. Probable slamming increment $\pm$ 1 1/2 tons/in^2 in sheerstrake. Heaving and pitching stresses negligible if unaccompanied by slamming.	42. 45. 49.
					Water pressure on hull	Photographic recording of meters connected to diaphragm type pressure gages	5-6 points around and below turn of bilge for 12 stations fore and aft		
					Wave profile	Electrical contacts on ships side closed by presence of sea water thus energizing telephone type indicators to be photographically recorded	30 points each side of ship at each of 12 girthwise stations throughout ship's length		
					Wind forces	Wind vane generator output recorded	Deck house top and mast crosstrees.		
					Accelerations	Recording tri axial electric strain gage type accelerometer.	4 Positions disposed fore and aft and athwartships		
					Angles of roll, pitch and yaw and periods	Recording gyroscopes			
1947				2,4 and some 3 node vertical and horizontal vibrations in still water	Critical frequencies, amplitudes and vibration profile	Cambridge low.-frequency and Geiger vibrographs and electric resistance strain gage accelerometers	On upper deck at 23 equidistant stations in ship's length	Records taken for seven differing displacements and loading distributions of ship including damping curves of free vibrations. Amplitudes measured for various magnitudes of exciting forces created by vibration generator at stern. For any particular mode of vibration, approximately linear relationship between maximum value of exciting force and resonant amplitude of vibration. In loaded ship conditions, resonant amplitudes per ton of exciting force do not differ appreciably from those in light ship conditions despite fairly large changes in frequency. Welded ship, OCEAN VULCAN, tends to have larger amplitudes than riveted ship, CLAN ALPINE, suggesting greater structural damping in riveted ship.	46.
				Known hogging and sagging B.M. applied in still water				Some apparent evidence of slightly greater ship deflections in welded OCEAN VULCAN than in riveted CLAN ALPINE. Unfairness of bottom plating between frames clear of longitudinal stiffening in CLAN ALPINE in general, about double that in CLAN ALPINE. Also more prominent stress deviations from beam theory in OCEAN VULCAN. In this and abnormally unfair plating in transversely framed welded ship, surface stresses reached 3 times heart of plate stresses.	43. 49.

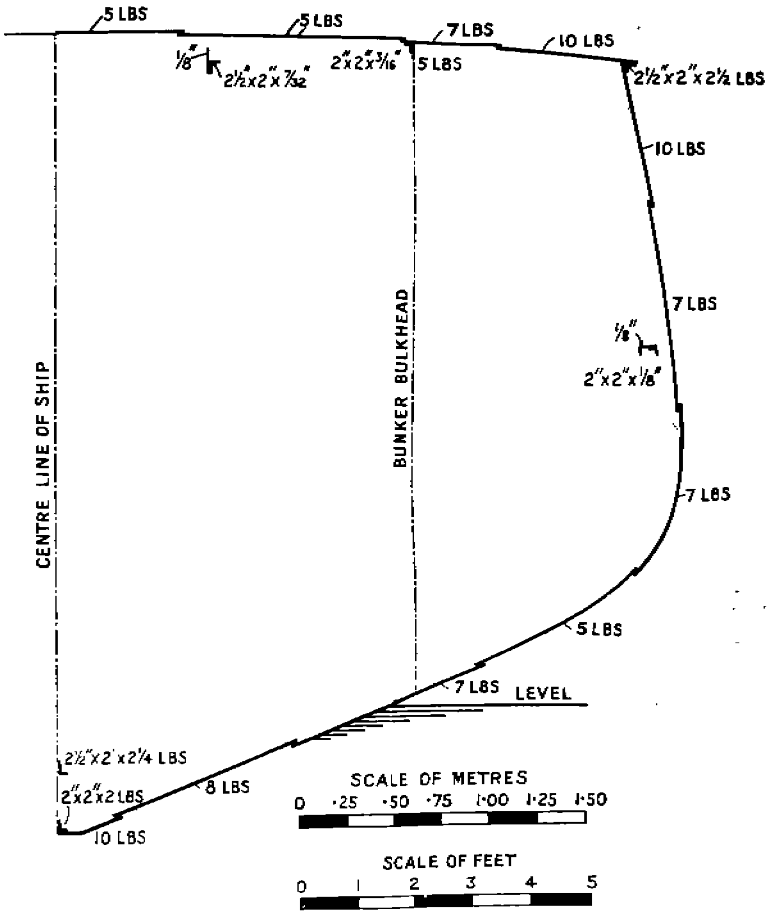
TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data: Type	Data: Method	Data: Location	Notes	Ref.
1947	CLAN ALPINE	416' cargo vessel. Two decks (similar to OCEAN VULCAN)	Similar to OCEAN VULCAN but all riveted	2 and some 3 node vertical and horizontal vibrations in still water	Same as OCEAN VULCAN	Same as OCEAN VULCAN	Same as OCEAN VULCAN	Records taken for four differing displacements and loading distributions of ship including damping curves of free vibrations. (For further details and comparisons of performance see OCEAN VULCAN notes.)	46.
				Known hogging and sagging B.M. applied in still water				(For comparison of performance see OCEAN VULCAN notes.)	44. 49.
1947	PRESIDENT WILSON	590' Passenger vessel. Multiple decks. Aluminum superstructure 35% vessel's length	Transverse framing Riveted shell and deck seams Welded butts and shell-frame connections	Known hogging B.M. only applied in still water	Multi-axial strains	Electric resistance gages 13/16" G.L.	In superstructure, particularly at ends. Generally on both sides of plating	Maximum hogging moment applied, 202,000 tons ft. Sagitta gage readings with short gage length strains permit solution for heart of plate stresses in locations where strains not available for both plate surfaces. Primary objective to determine interaction between the superstructure and hull girder. Stress distribution linear with distance from neutral axis up to uppermost full-breadth deck but decreased thereafter as decks stepped inboard. Further marked reduction in stress in way of aluminum structure. Including longitudinally continuous material up through superstructure second deck in Ship I appears warranted from ship deflection analysis (despite low stresses in this uppermost deck). Shear deflection 20% of total. Conclusion that length of superstructure important in determining its strength contribution.	47.
					Fore and aft strains	Mechanical gage, 60" G.L.	Transverse section amidships, inner bottom to second deck of superstructure		
					Plate panel deflections	Sagitta (arc rise) gages 4" G.L.	A few points in superstructure		
					Ship deflections	Transit sights	Port and starboard on uppermost continuous deck. Deflections at 10 intermediate points along 476' line of sight.		
1949-50	ALBUEBRA	355' Destroyer. Single deck Forecastle 43% vessel's length	Longitudinal framing All riveted	Known hogging and sagging B.M applied in still water	Strains			Constant draft maintained for all loadings also constant shearing force at test section. Maximum bending moments 39,000 tons ft. hogging, 30,000 tons ft. sagging. Corresponding maximum stress of 5¼ tons/in^2 attained. Stress distribution at forecastle break investigated. Very little information given for this loading.	48.
					Ship deflections	Theodolite and 7 targets	Theodolite and targets on upper and forecastle decks.		
				Ship in dock on piers loaded by known hogging B.M. to ultimate load	Strains, generally in rosettes	420 Acoustic gages 5 and 10 cm. G.L.	Transverse section amidships. Both sides of plating or one side plus another on button of known thickness.	Support reaction from piers applied to ship neutral axis. Ultimate failure due to buckling of bottom at stress (in keel) of 17 tons/in^2. Corresponding hogging bending moment at test section 125,000 tons ft. In instrumented side panel mean value of maximum shearing stress, 4.75 tons/in^2 vs. 4.09 calculated from beam theory for 548 ton shearing force increment. Up to moderately large bending moments, observed and calculated deflections agree closely, with shearing deflection included. Measured deflection greater than calculated before any plate wrinkling observed and discrepancy increased with loading. No rivet slip therefore no allowance necessary for rivet holes in Ship I calculation. All longitudinally continuous material was fully effective. Simple beam theory gave correct values for mean direct stresses but slightly underestimated deck edge stress and slightly overestimated deck centerline stress.	
						15 Dial gage type 13½", 15" and 100" G.L.	Transverse section amidships on deck and webs of longitudinals		
						100 Electric resistance gages 1/2" G.L.	Both sides of plating panel at ship neutral axis in region of high shearing force		
					Ship deflections	Theodolite and 7 targets	Theodolite on deck, targets on upper and Forecastle decks		

TABLE I (Continued)

Date	Ship	Ship Type	Construction	Loading	Data Type	Method	Location	Notes	Ref.
1951	CASCO	300' Coast Guard weather ship		Vessel in seaway 32 days				Double amplitudes of pitch up to 20°; roll up to 35°. Maximum stresses not over 1 $\frac{1}{2}$ tons/in^2. Waves about 260' x 15'. Very little information given.	50.

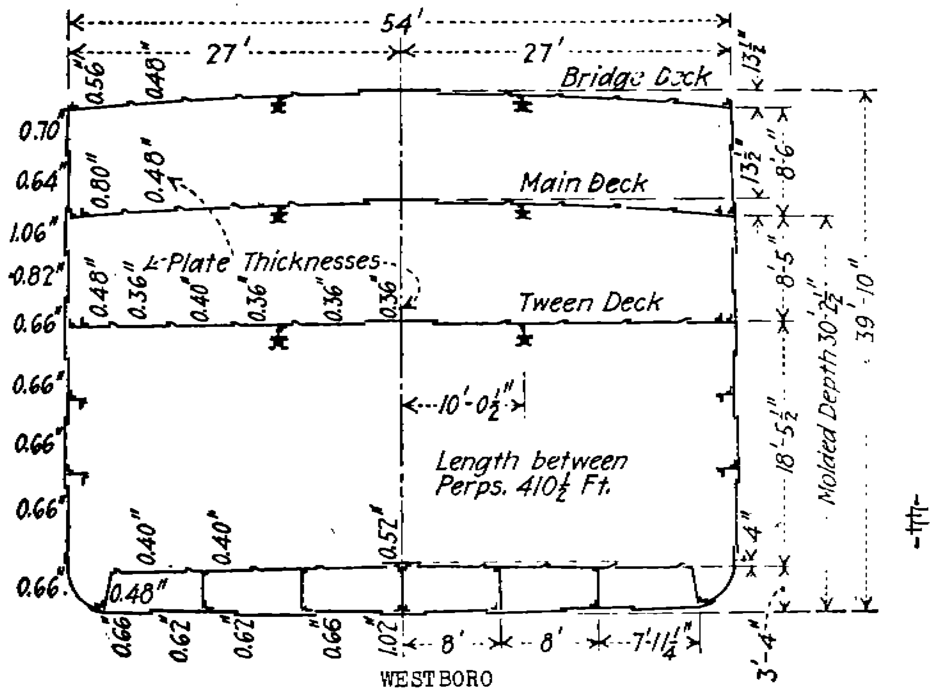
12/5/1952



WOLF

Longitudinally continuous material
(Reference 9)

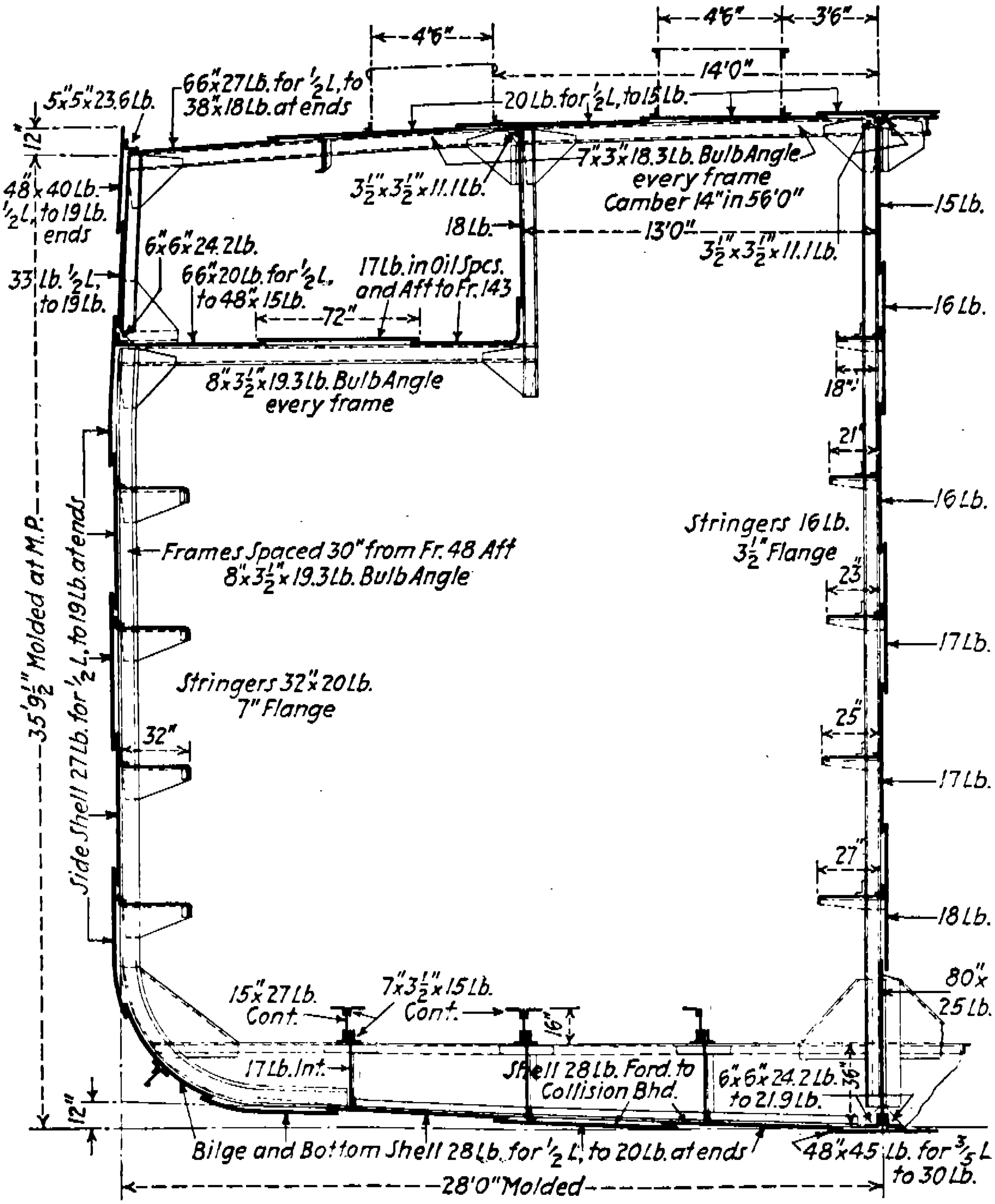
FIGURE 7



WESTBORO

Longitudinally continuous material
(Reference 5)

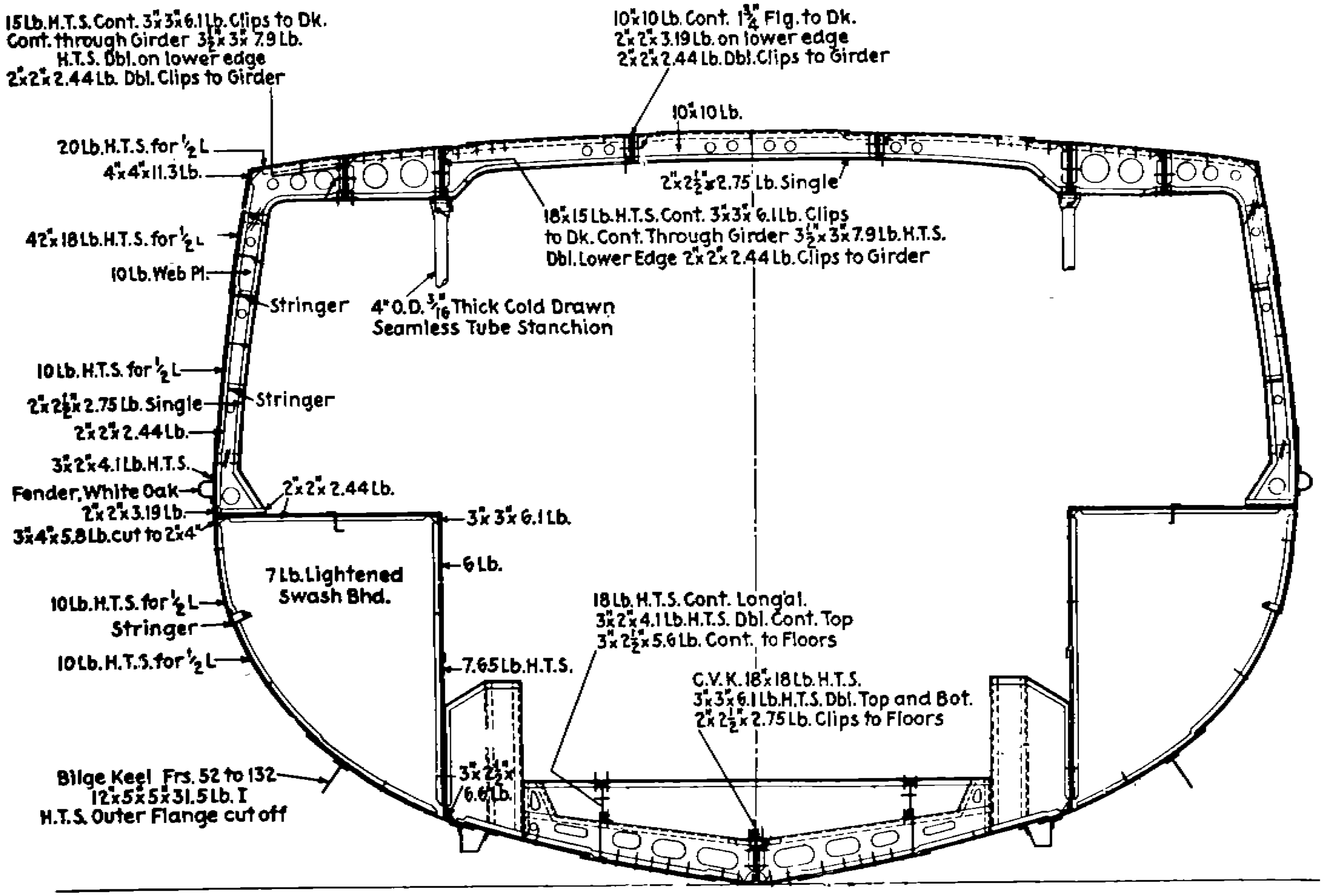
FIGURE 8



CUYAMA

Midship Section
(Reference 16)

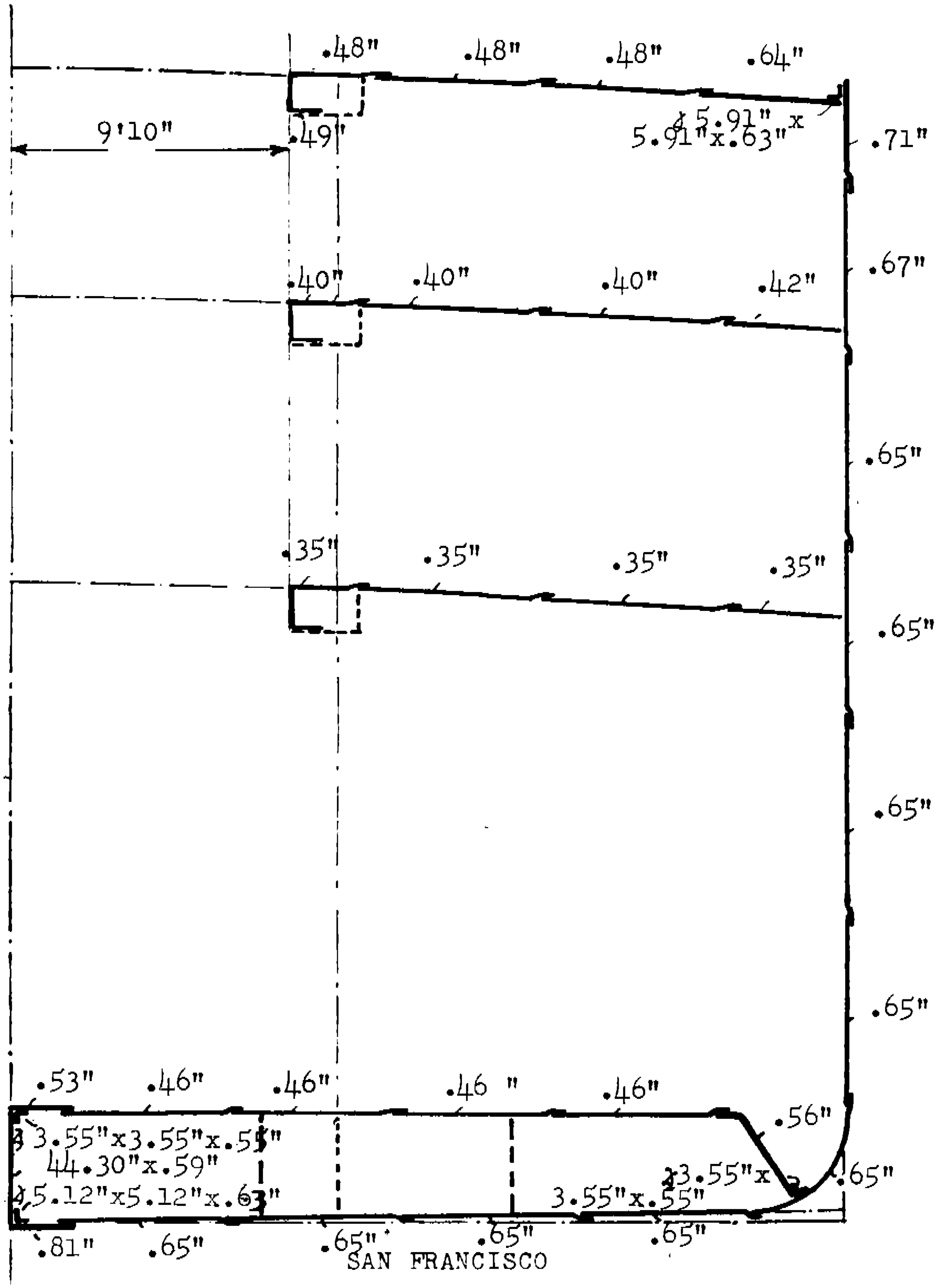
FIGURE 9



PRESTON and BRUCE

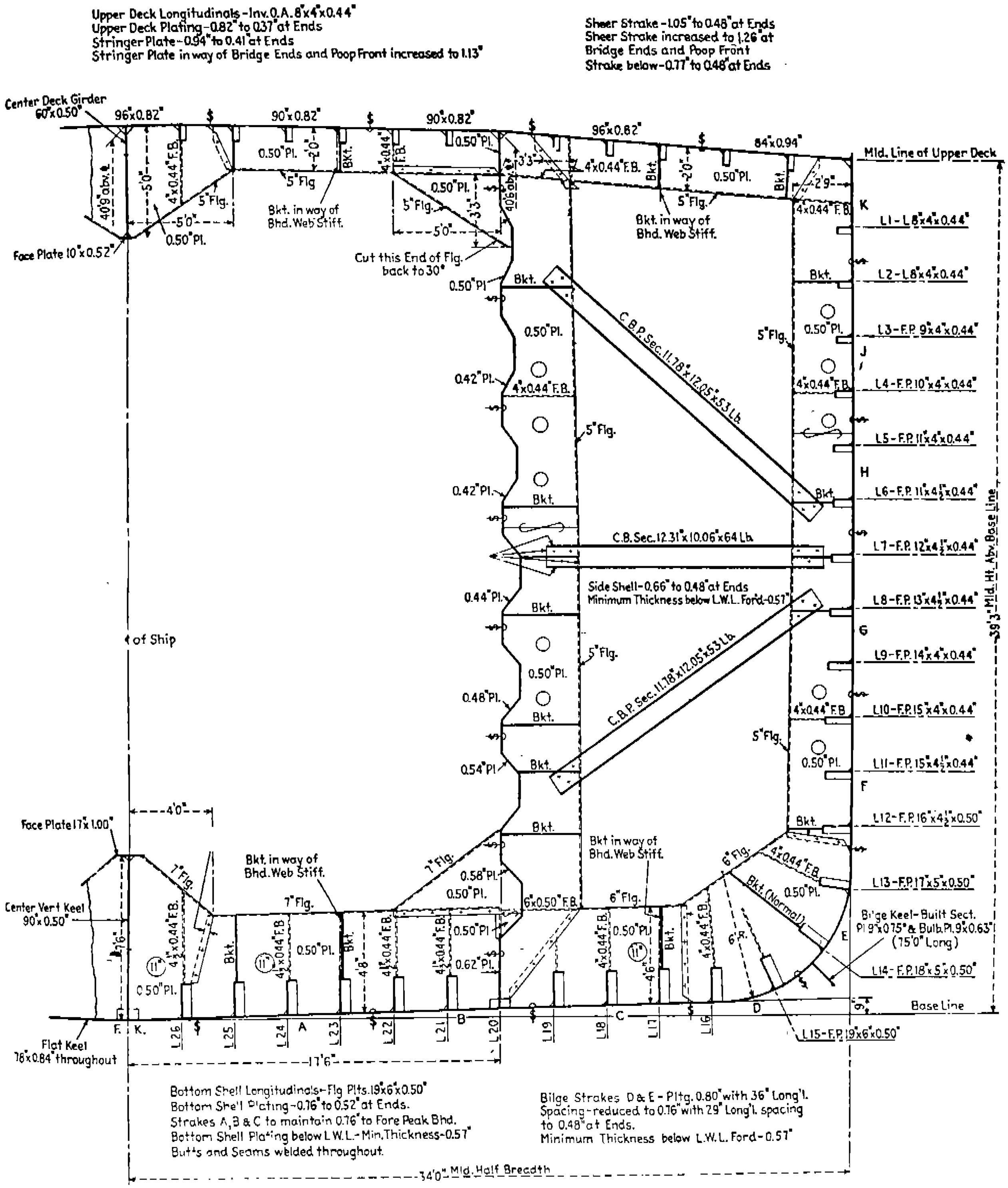
Midship Section
(Reference 15)

FIGURE 10



Longitudinally continuous material
(Reference 17)

FIGURE 11

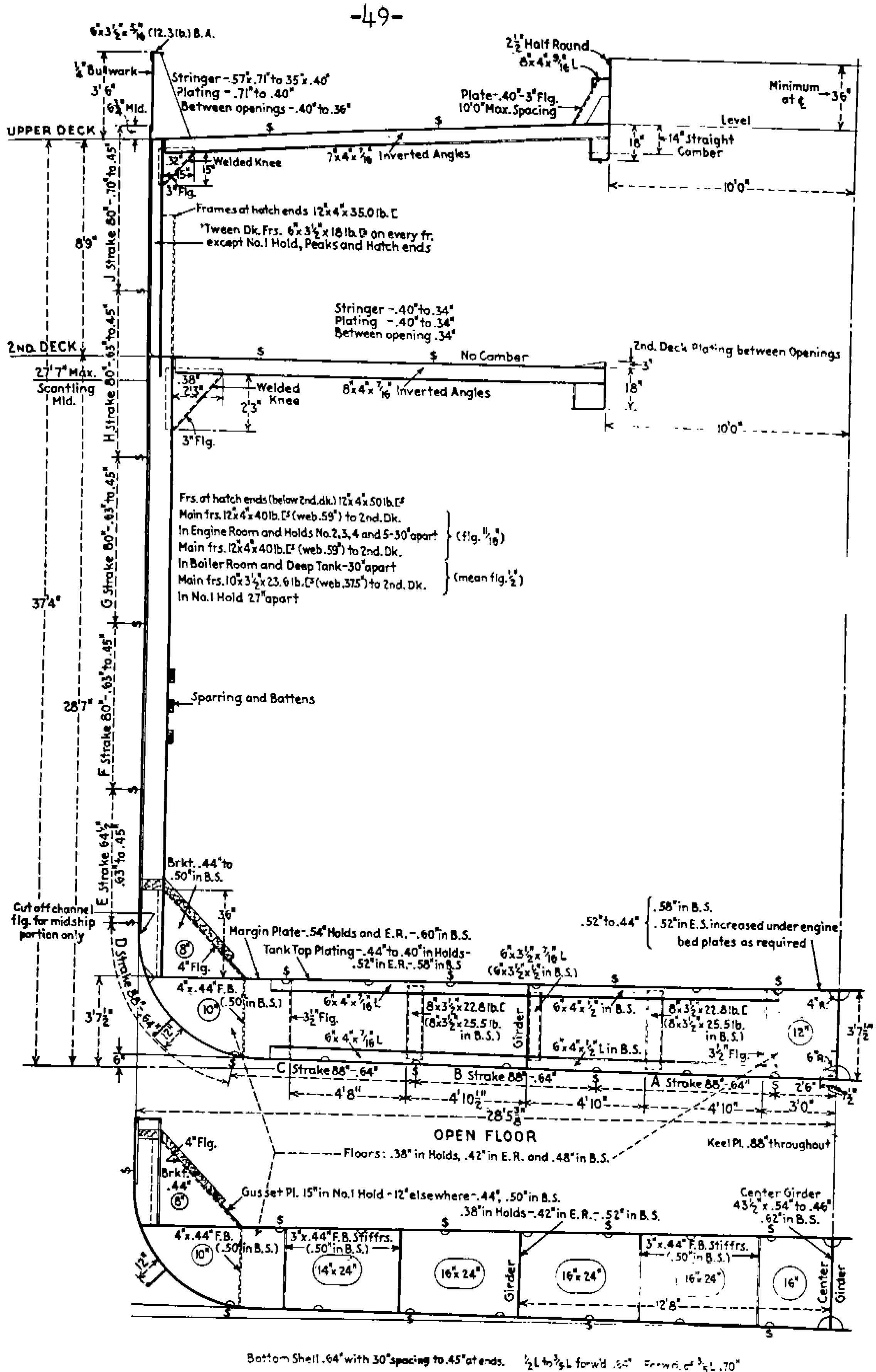


T2 tanker midship section similar to

SHILOH

VENTURA HILLS

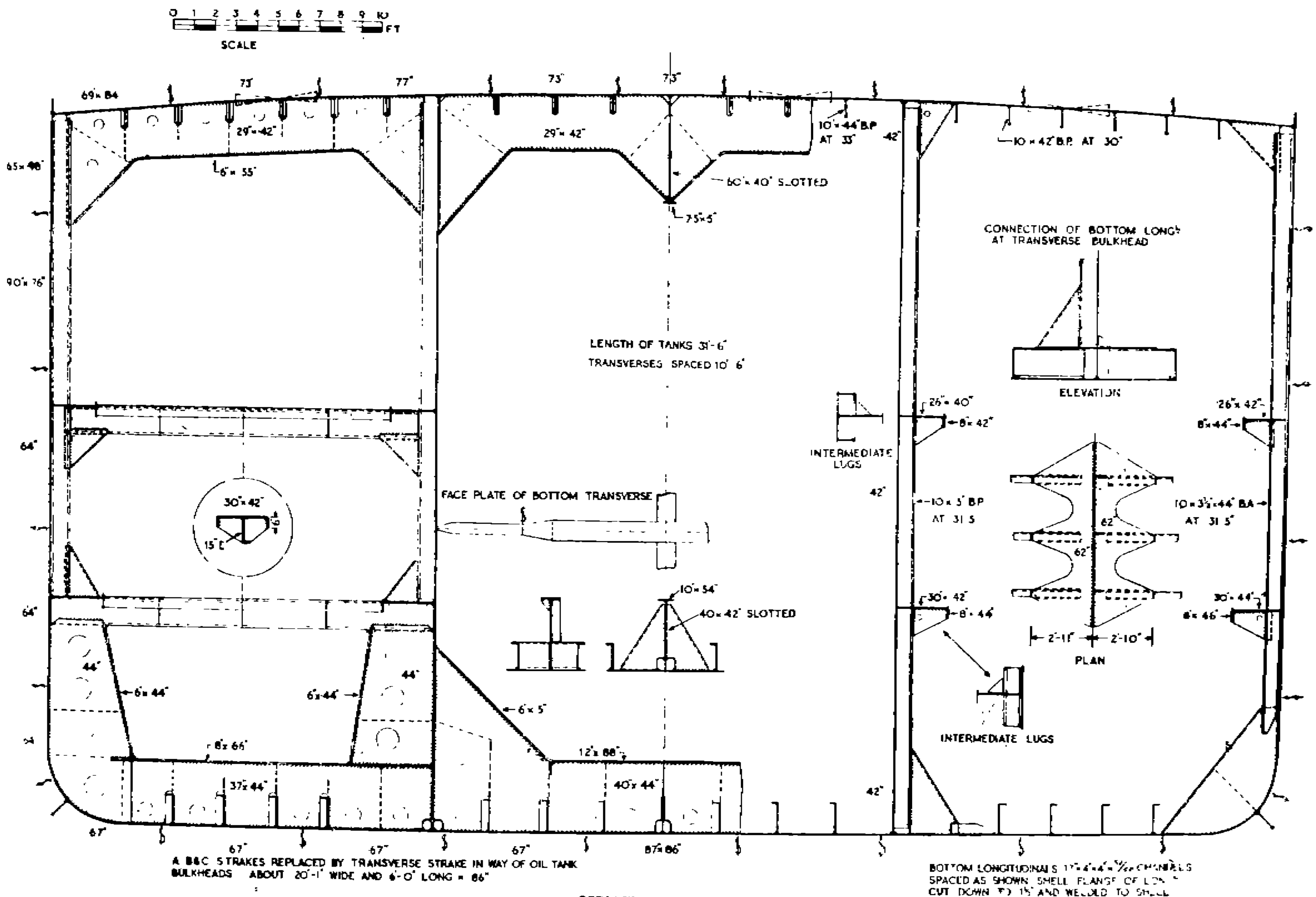
FORT MIFFLIN
("Marine Engineering"--1947)
FIGURE 12



("Marine Engineering")
FIGURE 13

Liberty ship midship section similar to

PHILIP SCHUYLER

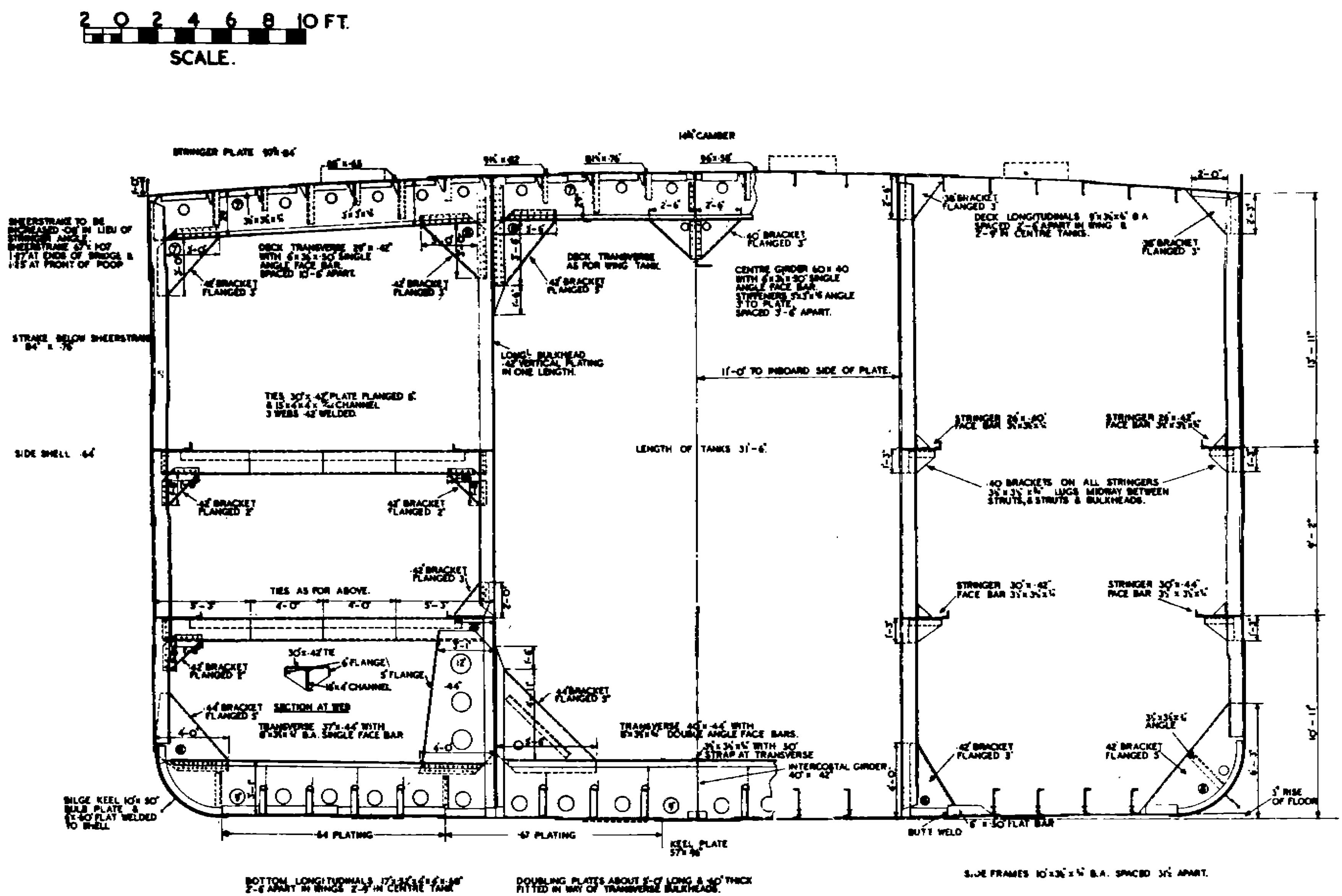


NEVERITA

Midship Section

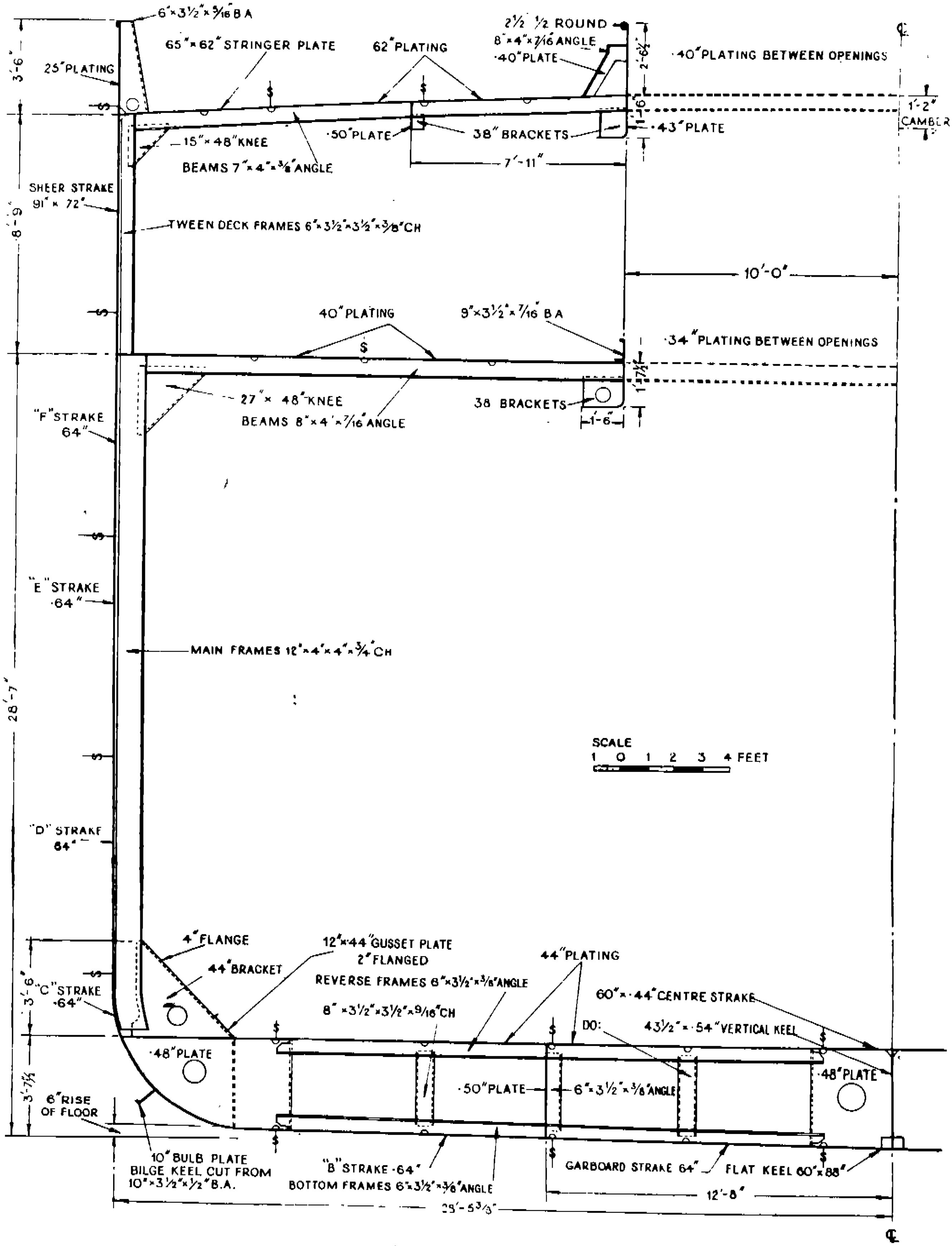
(Reference 37)

FIGURE 14



NEWCOMBIA

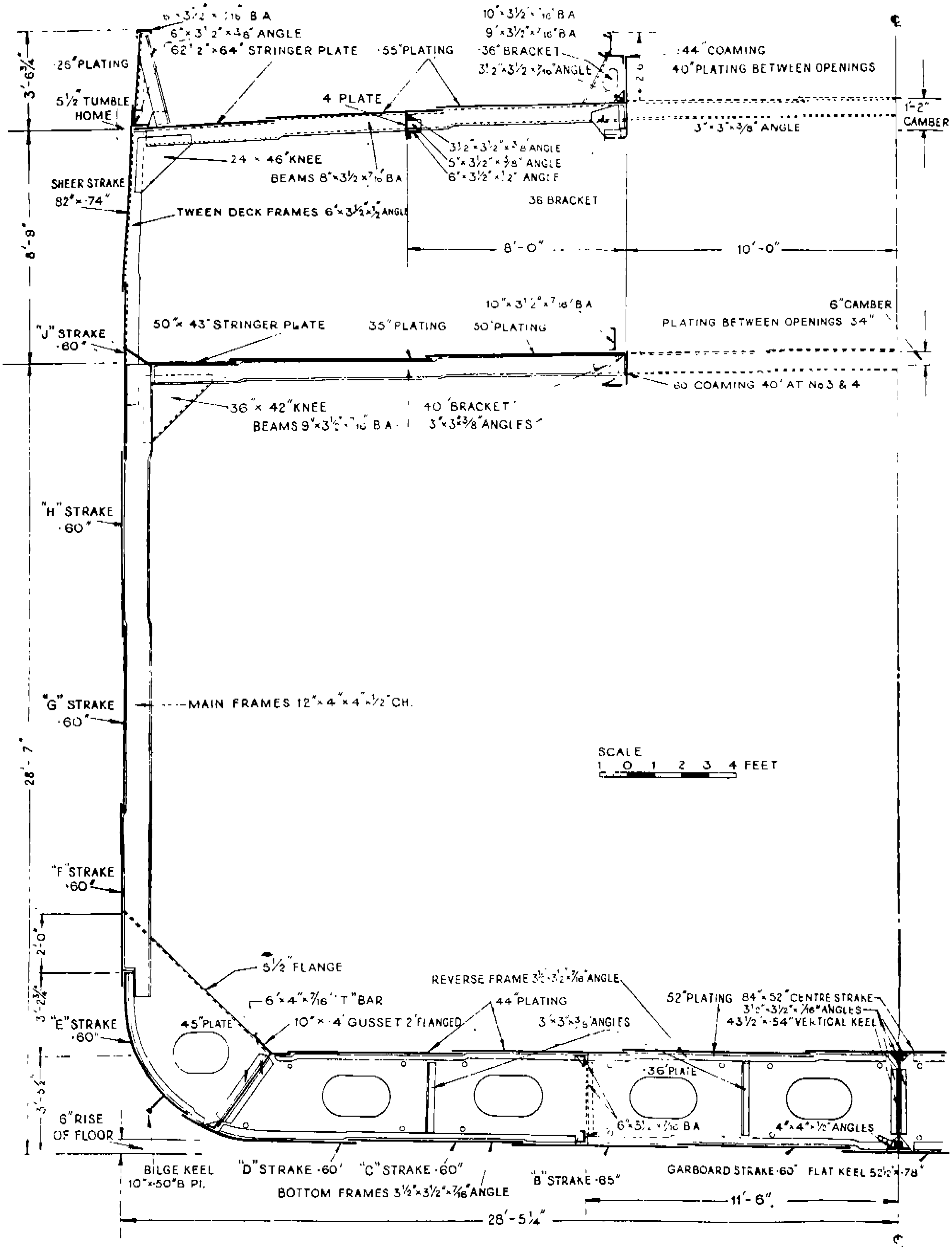
Midship Section
(Reference 39)
FIGURE 15



OCEAN VULCAN

Midship Section (Reference 46)

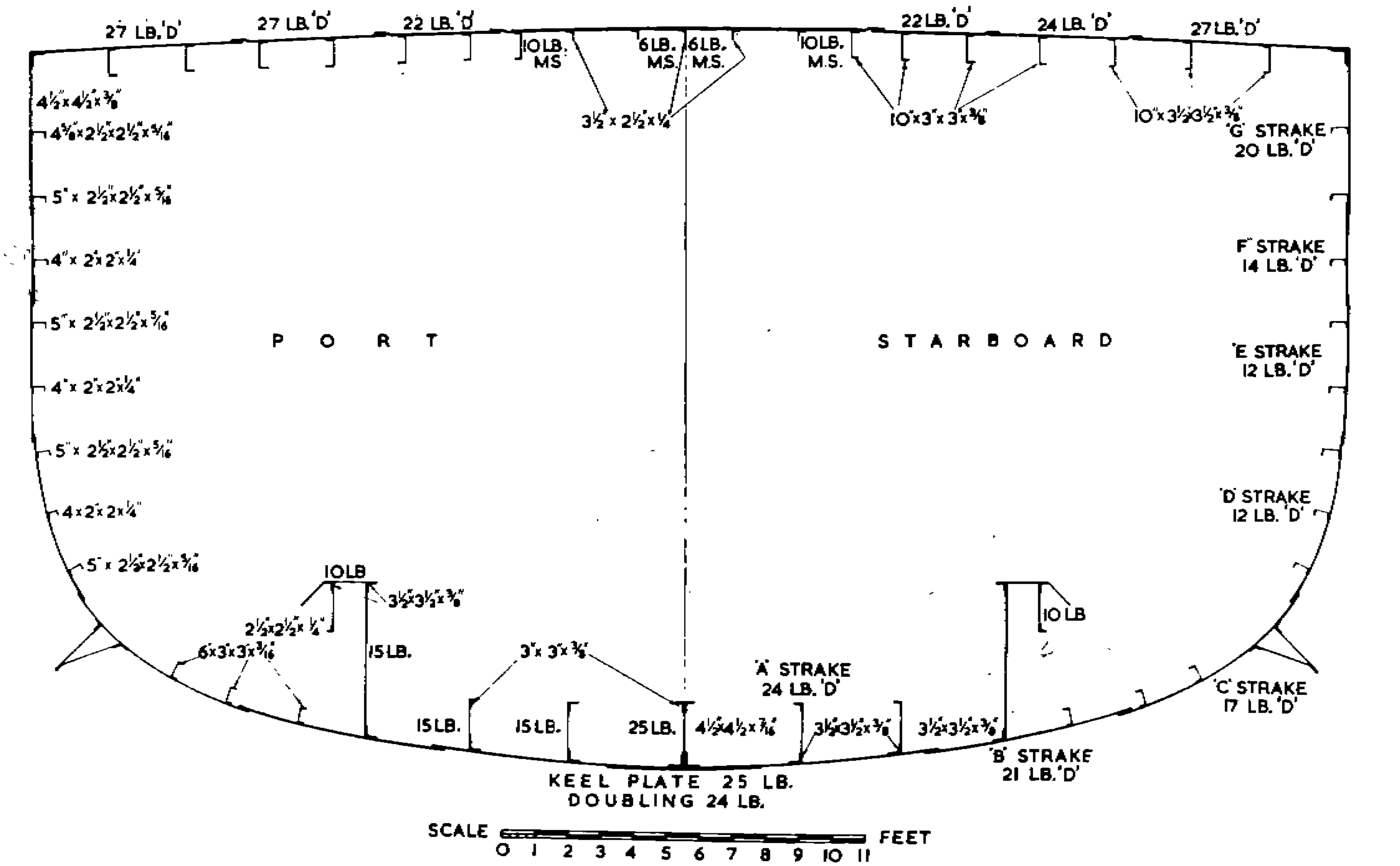
FIGURE 16



CLAN ALPINE

Midship Section (Reference 46)

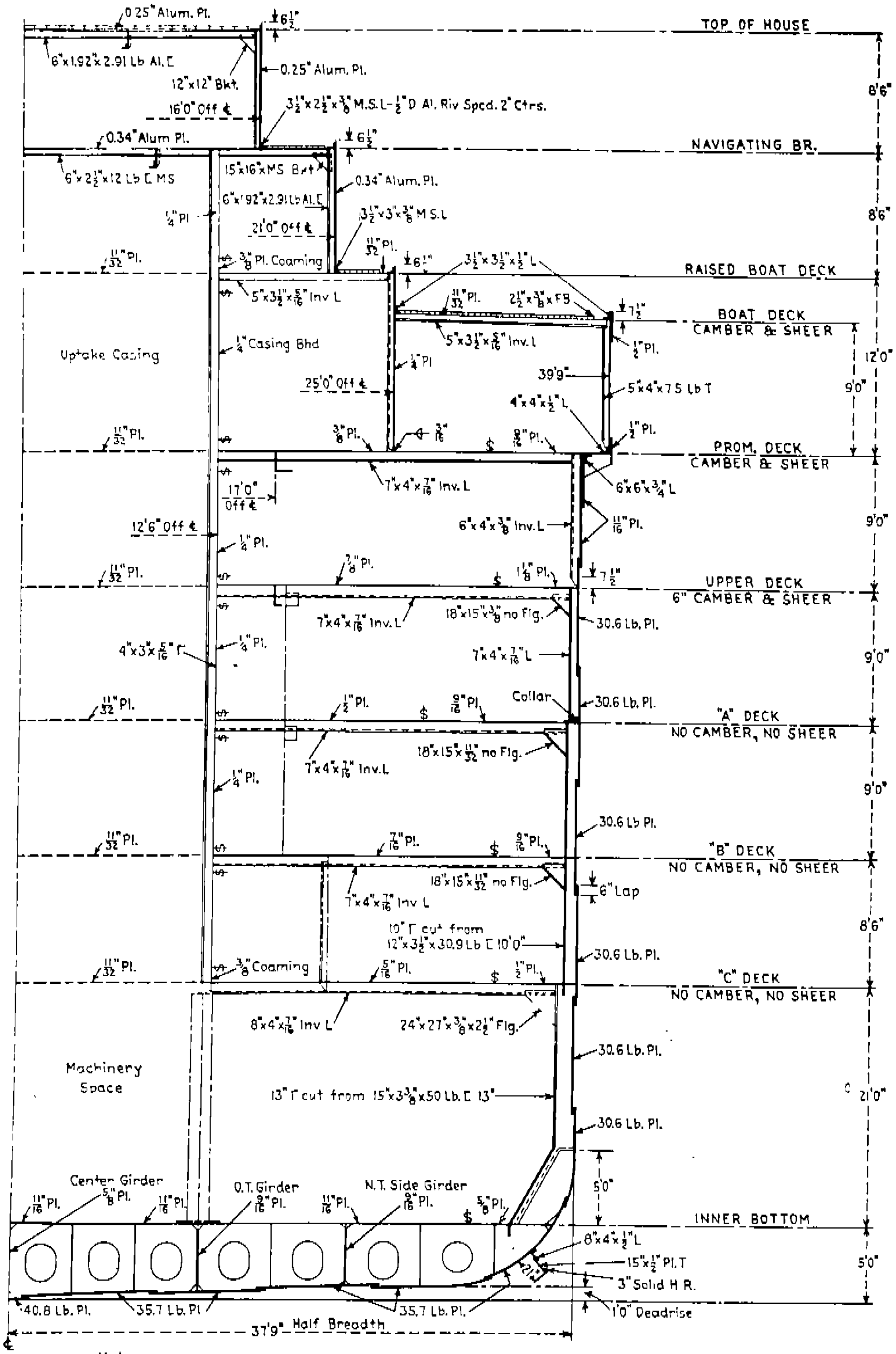
FIGURE 17



ALBUERA

Longitudinally continuous material
(Reference 48)

FIGURE 18



Note:-
Structure above raised boat deck of aluminum.
Below all steel.

Scale
0 2 4 8

PRESIDENT WILSON

Midship Section
(Reference 47)

FIGURE 19

APPENDIX A

"Waviness in the Bottom Shell Plating of Ships with All-Welded or Partially Welded Bottoms"

by

H. E. Jaeger
H. A. Verbeek

University of Delft

APPENDIX A

Waviness in the Bottom Shell Plating of Ships with All-Welded or Partially Welded Bottoms

Preliminary Report

Summary

This preliminary report discusses the first results of an investigation into the permanent corrugations which originated in the bottom shell plating of American-built all-welded vessels and in the plating of the bottoms of partially riveted vessels recently built on the Continent of Europe.

It is shown that the scantlings of the Liberty- and Victory-ships do comply with the requirements of the classification societies holding for riveted vessels. It is also shown that the mechanical properties of the employed steel and the loading of the ships in a seaway which gives rise to longitudinal bending of the hull remain within normal proportions.

The deformation of plating between stiffeners resulting from the riveting and welding processes are considered, and it is shown that the shrinkage of the welds connecting the floors to the bottom plating is the prime cause which creates the permanent corrugations. A theory interpreting this phenomenon is developed in the report and is here reproduced in simplified form.

A longitudinal strip of bottomplating extending between two adjacent floors is considered. Fig. 1 shows the shrinkage and the distortion of the fillet welds connecting the floors to the bottomplating, Fig. 2 illustrates the initial deflection of the plating between floors and the bending moments due to the distortion.

When the ship is in hogging condition, the strip of plating will be subjected to an axially directed compressive stress P/h, the maximal stress σ_{max} working in the extreme fibre of the plating then is given by the following expression holding for a plate simply supported along two opposite sides.

$$\sigma_{max} = \frac{P}{h}(1 + \frac{6f}{h}).$$

in which:

h = thickness of plating

f = momentary maximal deflection of the strip.

With a small initial deflection of the plating f_o = 1/6 h the extreme fibre stress is more than twice the axial stress. It is shown that in welded panels initial deflections often reach values far in excess of 1/6 h; in these cases the yieldpoint is exceeded and permanent corrugations will result. This phenomenon of plastic setting-in is still facilitated by the bending moments attendant upon the weld distortion and in the bottomplating once more by water pressure.

As far as known no permanent corrugations have been observed in the weather decks of the Liberty-and Victory-ships.

In this relatively heavy deckplating the initial deflection generally is small and the ratio 6f/h is still smaller than for bottom panels, so it can be explained that the yield-stress is not exceeded in this case. In ships with light welded weatherdecks permanent waviness sometimes has been observed.

The report discusses the permanent set which sometimes has taken place in the bottomplating of recently-built ships having welded double bottoms and floors riveted to the welded bottom shell plating.

In all these cases the deformations have not been brought about by a shortage of longitudinal strength of ships, but they must be attributed to initial deflections caused by distortions arising from welding and riveting.

Butt-welds connecting prefabricated bottom sections usually show deep indents and bulges. It is shown that these deformations result from ill-alignment of the plating, before welding of the butts is started. Practically no plastic set is observed here during the ship's life.

Measurements of permanent deformations in bottom plating and tanktops have been performed, the outcomes of which are discussed in detail and are explained with the aid of the theory developed before.

Precautions to be taken to prevent permanent deformation in the bottom panels of ships under construction are discussed and measures to improve the condition of existing vessels are reviewed.

It is thought, that not much danger is to be expected in navigating with ships having bottom-waviness.

On the other hand, the report thinks it advantageous to come to a construction of longitudinal frames in the bottom and under the deck, maintaining transverse frames in the ships' sides.

Finally some remarks about the continuation of the present investigation are made.

APPENDIX B

"Recent Developments in the Study of Longitudinal Strength"

by

James Turnbull

Lloyd's Register of Shipping

Recent Developments in the Study of Longitudinal Strength*

By James Turnbull, O.B.E.

THE longitudinal strength of ships has been constantly under review since the days when it was first appreciated by naval architects that a ship's hull was simply a large girder subject to variations in loading and having, when among waves, a violent and varying form of support, difficult if not impossible to assess. Theories were therefore devised for estimating the longitudinal bending moments to which ships are subjected in a seaway, and these have been applied and interpreted by experienced naval architects. Notwithstanding these reasonably satisfactory theories and the greater knowledge of ocean waves acquired within recent years, the longitudinal scantlings of ships are still based mainly on the records of service behaviour of earlier similar ships.

The principal theory has been the standard graphical longitudinal bending moment calculation, in which the ship is assumed to be poised momentarily on a trochoidal wave having a length from crest to crest equal to the length of the ship, and having a height equal to 1/20 of its length. The "Smith correction," although theoretically acceptable, is not often applied, presumably because designers of hull structures consider it a refinement of a calculation that is used only for purposes of comparison and which, in any event, will not give the actual bending moments a ship is likely to experience in service. For the same reason the "Read correction" has not been generally included.

Although the methods hitherto adopted have proved satisfactory, naval architects are not entirely satisfied in assuming values of the support given to a ship by the sea, and therefore for decades they have been striving to obtain reliable information on ocean waves and the forces they exert on ships. Furthermore, knowledge of the detailed behaviour, in regard to stresses and strains, of hull structures in a seaway is by no means complete, and this also impels them to seek information on this important subject.

In 1942-43 several welded ships developed serious fractures, and at that time it was thought that there must be some fundamental difference in structural behaviour between welded and riveted ships, since the latter had not suffered to nearly the same extent. It was obvious that new problems had arisen and in consequence it was decided, almost simultaneously in the U.S.A. and in the United Kingdom, to set up research committees to investigate why welded ships were behaving differently from riveted ships. This presented a good opportunity to obtain additional knowledge of ocean waves as well as of the detailed behaviour of some typical ships' structures. The work which these committees instigated has undoubtedly resulted in a clearer understanding of many of the factors involved in the longitudinal strength of ships.

The first full-scale experiment carried out under the direction of the Admiralty Ship Welding Committee was a comparison between the behaviour of the welded tanker *Neverita* and the riveted sister ship *Newcombia* under hogging and sagging bending moments applied in still water, but no important overall difference was revealed. These ships were 460 ft. (140 m.) in length, 59 ft. (18 m.) in breadth, and 34 ft. (10·38 m.) in depth. It should be noted that they were framed longitudinally, and that the maximum mean stress induced was of the order of 5 tons per sq. in. (790 kilo. per sq. cm.) in the deck and bottom plating.

The next ships compared were the welded *Ocean Vulcan* and the riveted *Clan Alpine*, sister dry cargo ships of 416 ft. (126·9 m.) in length, 56 ft. 10½ in. (17·33 m.) in breadth, and 37 ft. 4 in. (11·38 m.) in depth to the strength deck. They were of standard design and were transversely framed, and slight differences in behaviour were observed during the still water tests. Some references to these will be made later.

The decision was made to fit the *Ocean Vulcan* with instruments capable of recording at sea the wave pressures, the wave profiles on the ship's sides, the wind velocities, angles of roll, pitch and yaw and the forces imposed on the ship and cargo by accelerations. The observations to be made at sea were to include the determination of ocean waves by stereophotographic survey and other methods.

With these instruments the *Ocean Vulcan* made eight double crossings of the Atlantic over a period of seventeen months and rough sea conditions were experienced on several occasions. Much valuable information was obtained from these sea trials, and in addition to the measurement of waves and wave pressures on the ship, investigations were carried out on the effects on the hull structure of vertical bending, horizontal bending, torsion, heaving and pitching, axial compression and slamming. This was a new approach, some of these factors not having been considered in previous investigations.

It was evident from the observations made that waves of earlier storms were almost always superimposed on the existing wave system, with the result that the seas were seldom regular. However, the important conclusion was come to that under the most severe storm conditions there is a tendency for the waves produced by that storm to dominate all earlier disturbances and therefore to closely resemble trochoidal form.

Since the waves recorded on this trial were evidently not the most severe that could be met, a study was made of the greatest waves reported by earlier investigators, many of which were estimated by visual observations of ships' officers. It is usual, in such circumstances, for wave heights to be over-

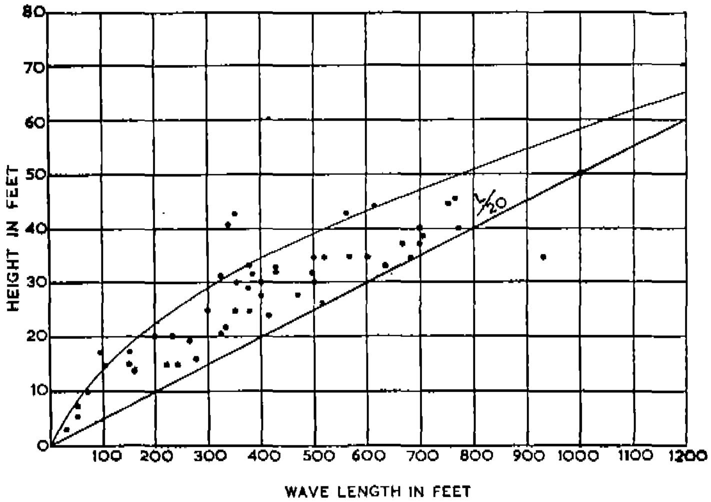


Fig. 1. Maximum wave heights for various wave lengths as observed and estimated by various investigators

* Abstract of a paper *Longitudinal Strength—A Review of Some Recent Developments* presented at the Autumn Meeting of the Institution of Naval Architects in Genoa on September 26

estimated and lengths to be underestimated. These observations, together with those obtained on the *Ocean Vulcan* sea trials, are plotted in Fig. 1.

It will be seen that the highest waves having the length of the *Ocean Vulcan* are probably 35 ft. (10·7 m.) in height, *i.e.*, a height to length ratio of 1 to 12, which is much steeper than the value L/20 assumed in the standard longitudinal bending moment calculation. According to Fig. 1, the greatest waves for ships of 300 ft. (90 m.) in length have a height to length ratio of approximately 1 to 10 and for 600 ft. (180 m.) ships approximately 1 to 14.

It is noteworthy that Schnadel reported that the 430-ft. (131 m.) *San Francisco* experienced waves of a height of L/13·5.

In many instances waves of exceptional height have been reported, but the length from crest to crest has not been mentioned. For instance, the highest wave ever reported was 112 ft. in height. Its length from crest to crest is not known but was probably about 3,000 ft. (915 m.).

It is possible that ocean-going ships seldom experience these maximum wave conditions during their lifetime, and when they do they are most probably proceeding on a course inclined to the direction of the waves, which would have the effect of reducing the relative steepness of the wave traversing the ship's sides. The wave pressure records showed reasonable agreement with those derived from the "Smith correction," and this, too, has the effect of reducing the effective steepness of the waves.

From the foregoing it may be deduced that in using the L/20 wave without the "Smith correction" the theoretical longitudinal bending moment amidships would approximate the actual bending moment experienced in severe storm conditions by ships of about 400 ft. (120 m.) in length. In longer ships the stresses derived from the classical theoretical calculation would, of course, be higher and in shorter ships lower than those actually experienced.

Actions at sea affecting strengths

In the *Ocean Vulcan* sea trials the horizontal bending moments were found, as would be expected, to be greatest when the seas were advancing at an angle of between 30 and 45° on either the bow or the stern and appreciably high stresses resulted. Horizontal bending was frequently in phase with the vertical bending so that at one sheerstrake (or bilge), the stresses due to the horizontal and vertical bending moments became additive, while on the other side of the ship they tended to cancel each other. However, when the vertical longitudinal bending moments were at their highest values the horizontal longitudinal bending moments were at their minimum.

The greatest range of vertical bending moment derived from these observations at sea was 190,000 tons feet (58,800 tonnes metres), corresponding to a range of stress of 8 tons per sq. in. (1,260 kilo. per sq. cm.) at the top of the sheerstrake amidships. There is no experimental evidence to show the actual separation of this range into hogging and sagging, although the investigators are inclined to the view that the sagging moment was slightly greater than the hogging. This range was associated with waves 35 ft. (10·7 m.) in height and between 600 ft. (180 m.) and 700 ft. (210 m.) in length. More severe conditions than these could, no doubt, be encountered with a correspondingly larger range of stress. The highest range of stress recorded on the 430-ft. (131 m.) *San Francisco*, with waves L/13·5, was 9·6 tons per sq. in. (1,510 kilo. per sq. cm.) [8·2 tons per sq. in. (1,290 kilo. per sq. cm.) without the addition for slamming which occurred at the time], which bears a reasonable relationship to the highest range recorded on the *Ocean Vulcan*.

Torsion moments in the case of the *Ocean Vulcan* were estimated to cause longitudinal stresses not exceeding ½ ton per sq. in. (80 kilo. per sq. cm.). However, under the wave conditions which cause the greatest vertical bending moments the torsion moments were small. It would appear, therefore, that no special allowance may be necessary for horizontal bending or for torsion when computing the probable greatest longitudinal stress.

Heaving and pitching

The effects of heaving and pitching were investigated, but were considered to be relatively unimportant, even under the most severe wave conditions, so long as slamming did not occur. Although the instrumentation on the *Ocean Vulcan* was not suitable for recording shock loading, some general observations were made. It was estimated that normal slams resulted in stresses at the strength deck amidships of the order of ± 1½ tons per sq. in. (240 kilo. per sq. cm.). The highest stress due to slamming recorded by Schnadel on the *San Francisco* was 1·4 tons per sq. in. (220 kilo. per sq. cm.) under very severe weather conditions when winds had reached force 12 on the Beaufort scale and the ship was pitching as much as ± 12°. Slamming stresses recorded by other earlier investigators were less than ½ ton per sq. in. (80 kilo. per sq. cm.).

Slamming was only noted when the ship was in the light condition with a draught forward of from 8 ft. 2 in. (2·5 m.) to 10 ft. 1 in. (3·1 m.), *i.e.*, from about 0·02 L to 0·024 L. Slamming was experienced on one day in every three while the ship was in the open ocean on a ballast voyage, and while no exact count was kept it is estimated that between 2,000 and 4,000 slams were experienced during the 17-month period of the sea trials. From the evidence of two slams identifiable from the records it was deduced that slamming had resulted in a greater increase in the sagging stresses amidships than in the hogging stresses. It was observed that it was not necessary for the ship's bottom forward to leave the water for slamming to occur.

The greatest axial thrust was estimated to result in a compressive stress of less than ½ ton per sq. in. (80 kilo. per sq. cm.) over the section amidships. In high-speed ships this factor may, however, be more important.

It would appear from the foregoing that of these various actions only vertical bending, axial compression and slamming require special consideration.

It is not suggested that the foregoing views should be accepted without reserve, as the investigations have been carried out on one type of ship only.

Deflections of main girders

As it was generally believed that the failures in welded ships were, to some extent, due to such ships being more rigid than riveted ships and since such a conception could not be proved or disproved without actual tests, the Admiralty Ship Welding Committee arranged for a comparison to be made between the deflections of the welded tanker *Neverita* and the riveted tanker *Newcombia* and between the welded cargo ship *Ocean Vulcan* and the riveted cargo ship *Clan Alpine*. The results show that, contrary to general expectations, there was no significant difference in deflection between the riveted and welded ships. A slight difference was detected in the case of the dry cargo ships, the welded ship being slightly the more flexible. However, this slight difference might be accounted for by the normal inaccuracies in recording.

It should be noted that the ships were not subjected to very high stresses during these tests.

Another general belief was that while riveted structures, because of rivet slip, could automatically adjust themselves in such a way that each part of the structure took its fair share of the load, welded ships did not possess this desirable property. Because of this belief, special efforts were made to detect rivet slip. The accuracy of the instruments was such that any slip large enough to have an appreciable effect on the behaviour of the structure would have been revealed, but no rivet slip was noted. In the paper describing the British Admiralty's bending tests on the riveted destroyer *Albuera*, which was tested to destruction, it is specially mentioned that no rivet slip was observed.

The foregoing does not prove that rivet slip never occurs in ships' structures in service. Taking it by and large, the evidence supports the view that riveted construction, under high stresses, is capable of an adjustment the exact nature of which, however, is not yet fully understood.

Many ships have been subjected to longitudinal bending tests in still water, and it has been found from the tests that, in general, the resulting stresses agreed with those arrived at by the clasicals beam theory. In the Admiralty Ship Welding Committee's investigations a further step was taken in comparing the behaviour of certain welded ships with the behaviour of sister ships of riveted construction.

Fig. 2*a* shows the distribution of stress across the bottom near amidships for a ship such as the riveted *Clan Alpine* in the hogging condition compared with that for a welded sister ship under the same conditions. Fig. 2*b* shows the comparison for the sagging condition.

It will be seen that these stresses, which are heart of plate stresses, have a distribution that follows the general trend of the distributions given by the simple beam theory but that there are several notable departures. The most outstanding of these occurs in the vicinity of longitudinal stiffening members, where the observed stresses are higher than the theoretical values, while clear of such stiffening the observed values are smaller than the theoretical values. These departures are much more prominent in the welded than in the riveted ship.

The unfairness of the bottom plating between frames clear of longitudinal stiffening in the welded *Ocean Vulcan* was in general about double that of the riveted sister ship. This unfairness is the main explanation for the greatly reduced heart of plate stress in the bottom plating away from the longitudinal stiffening members. The corrugations of the bottom plating of the *Ocean Vulcan* were kept under observation at each dry docking following the bending tests. They were found to have increased on each occasion and ultimately fairing and the fitting of additional longitudinal stiffening became necessary in order to prevent a recurrence.

In thin and abnormally unfair plating in a transversely framed welded ship, the surface stress may reach three times the heart of plate stress.

The foregoing observations, combined with the results from the longitudinally framed *Neverita* and *Newcombia* which showed only small departures from the theoretical stress distribution, show conclusively that welded ships of appreciable size should preferably be stiffened longitudinally on the bottom and the strength deck over the midship portion at least.

As several wartime built welded ships sustained fractures at hatchway corners, a special study was made by the Admiralty Ship Welding Committee of the stress at such discontinuities. Concentrations of stress of the order of two and a half times the nominal stress were found at certain discontinuities near amidships, and there was a tendency for the concentrations to be greater in the welded than in the riveted ships. These concentration factors are approximate.

The subject of fatigue in ships' structures has not received much attention by naval architects, due, no doubt, to the absence of reliable information regarding the actual stresses and the number of times the various stress ranges are experienced in service.

A statistical strain gauge was fitted to

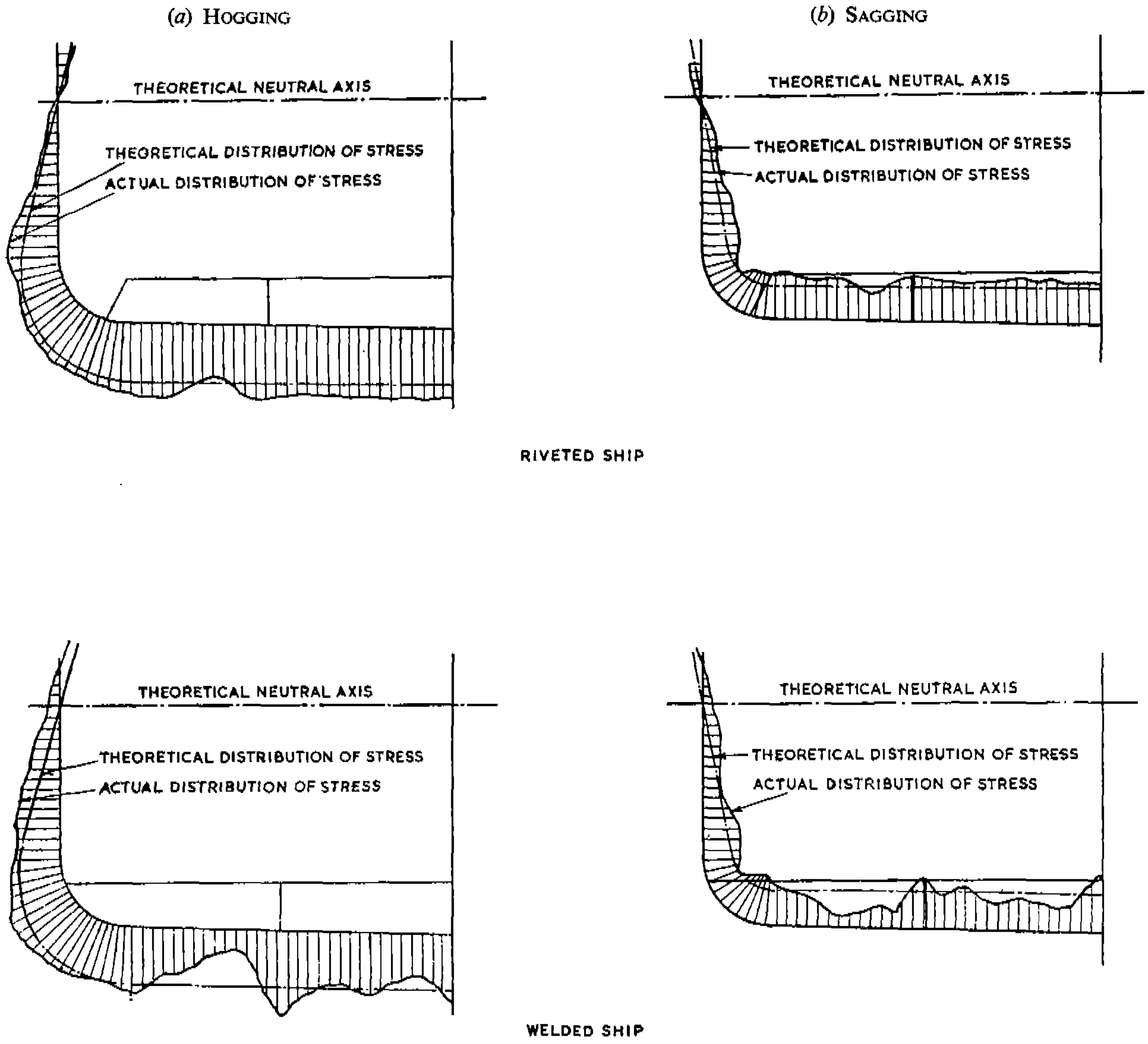


Fig. 2 (a & b). Distribution of longitudinal stresses for bottom shell plating near amidships the *Ocean Vulcan* and it has been in satisfactory operation for over a year. Some interesting data have now been collated.

Typical results taken over a period of one year are as follows:—

Range of stress	Number of times experienced
1 ton per sq. in. (158 kilo. per sq. cm.)	266,884
2 tons per sq. in. (315 kilo. per sq. cm.)	7,105
3 tons per sq. in. (473 kilo. per sq. cm.)	1,329
4 tons per sq. in. (630 kilo. per sq. cm.)	102
5 tons per sq. in. (788 kilo. per sq. cm.)	5
6 tons per sq. in. (945 kilo. per sq. cm.)	2

It will be noted that the maximum range was only 6 tons per sq. in. (945 kilo. per sq. cm.), and that range on only two occasions during one year of service. It is obvious that much more severe conditions could be experienced, and these may be recorded during this investigation, which is being continued.

Although ships are subject to fatigue loading, it is by no means clear that fatigue is an important factor in the longitudinal strength of ships. It may, however, be important in regard to regions of stress concentrations when a ship has been consistently subjected to an injudicious longitudinal distribution of cargo.

Distribution of cargo

Unlike the conditions which ships meet at sea, the distribution of cargo can be controlled. It is perhaps true to say that many structural fractures experienced at sea are due mainly to injudicious loading.

In tankers, unless loading distribution is properly controlled, there is a danger of excessive sagging stresses, particularly where the total length of the cargo spaces extend over a short length of the midship portion of a ship. Even when the cargo tanks are well spread out over the length of the ship, if the end tanks are left empty when high-density cargo is being carried, high sagging stresses may result. In these circumstances it is advisable to reduce the cargo loads in the midship half length. Fore and aft distribution of ballast must also be carefully arranged if dangerously high stresses are to be avoided.

Twenty years or so ago, when large deep tanks were incorporated amidships in most dry cargo ships, the ballast sagging condition was generally more severe than the load hogging condition. While the modern cargo ship has a better distribution of ballast she has finer lines, with large flare at the ends and cruiser stern, often with a forecastle and poop for cargo, with the result that the loaded hogging condition is now generally the more severe. This condition is worsened when, as sometimes happens, No. 3 is the only hold left empty.

There is such a wide variety of types of ships and possible loading arrangements that it is not possible in this paper to discuss the subject fully. The matter is, however, of paramount importance.

Residual stresses were considered at one time to be one of the primary causes of the structural failure of welded ships. Research and experience have shown that, if they do in fact exist in an appreciable magnitude, they need not concern the designer of hull structures provided good notch tough steel is used in the construction.

There is no doubt that the research on ships' structures carried out under the direction of the Admiralty Ship Welding Committee constitutes one of the most important contributions to our knowledge of the strength of ships. The full effect of that research will not be felt until all the reports have been published and studied. While ships' scantlings will continue to be based mainly on the service behaviour of earlier similar ships, it should now be possible to make a closer estimate of the actual stresses imposed by the forces of the sea. It is likely that the standard longitudinal bending moment calculation will be superseded by a simplified still-water bending moment calculation, to which will be added an estimate of the effects due to dynamic action.

Most research on actual ships' structures has been carried out on ships between 400 ft. (120 m.) and 500 ft. (150 m.) in length. It would add much to our knowledge and assist in arriving at reliable values for the dynamic factor if an investigation of the behaviour at sea of ships of other lengths, depths, forms, speed and draught-length ratios could be carried out.

With the tremendous advances being made in the development of scientific recording equipment, it is conceivable that such additional information may be obtainable without great expense in the near future. An example of this type of equipment is the statistical strain gauge fitted to the *Ocean Vulcan*. The chief officer takes the records, and the gauge, which does not interfere with the operating of the ship, is serviced only when the ship visits the United Kingdom.

Welded longitudinal framing

Although knowledge of all the factors affecting the longitudinal strength of ships is still incomplete, the items requiring special attention are clearly shown by a study of structural failures and of the results of research.

Perhaps the most outstanding deduction is the superiority in welded construction of longitudinal framing over transverse framing for the bottom and strength deck amidships.

The high concentration factors show how important it is for the design of welded structural details at discontinuities to receive most careful consideration. These high factors and the probable existence of residual stresses show the desirability of using, in welded construction, good ductile notch tough steel, especially for thick plating.

It is obvious that the danger of excessively high stresses being experienced at sea can be greatly reduced by arranging the fore and aft distribution of the cargo loading in such a way that the bending moment in still water is as near as practicable to the neutral condition, and in order to minimise the effects of slamming, the draught forward in the ballast condition should be kept as great as possible consistent with other features, such as immersion of the propeller.