SSC-222

CATAMARANS–TECHNOLOGICAL LIMITS TO SIZE AND APPRAISAL OF STRUCTURAL DESIGN INFORMATION AND PROCEDURES

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1971

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The Ship Structure Committee has completed a project that assesses the present state of the art for designing Catamarans, large platform, twin hulled ships. The purpose of the project was to collect and analyze design techniques and data presently available and assess their usefulness for catamarans approaching 1000 feet in length.

This report contains procedure for the initial design of a large catamaran and indicates where additional tests should be made before the final design stage is completed.

W. F. REA III
Rear Admiral U.S. Coast Guard
Chairman, Ship Structure Committee

SSC-222

Final Report

on

Project SR-192, "Catamaran Designs"

to the

Ship Structure Committee

CATAMARANS - TECHNOLOGICAL LIMITS TO SIZE AND APPRAISAL OF STRUCTURAL DESIGN INFORMATION AND PROCEDURES

by

Naresh M. Maniar and Wei P. Chiang
M. Rosenblatt & Son, Inc.

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ABSTRACT

Existing United States shipbuilding facilities can handle 1000-foot catamarans with up to 140-foot individual hull beams on the premise that the hulls would be joined afloat. Major harbors and channels of the world suggest an overall beam limit of 400 feet and 35-foot draft. Drydocking for catamarans over 140-foot in breadth will require new facilities or extensive modification to existing facilities. Scantlings of a 1000-foot catamaran cargo liner can be expected to be within current shipbuilding capabilities. The uniqueness of the catamaran design lies in the cross-structure and the important facets of the cross-structure design are the prediction of the wave-induced loads and the method of structural analysis. The primary loads are the transverse vertical bending moments, axial force, shear, and torsion moments. Designers have relied heavily on model tests to obtain design loads and have used general structures principles and individual ingenuity to perform the structural analysis in the absence of established guidelines. Simple semi-empirical equations are proposed for predicting maximum primary loads. A structural analysis method such as the one proposed by Lankford may be employed for conceptual design purposes. The Lankford method assumes the hulls to be rigid and the cross-structure loads to be absorbed by a group of transverse bulkheads and associated effective deck plating. This procedure in general should provide an overall conservative design and not necessarily an economic or optimized design. Additional research and development work including systematic model test programs are necessary for accumulating additional knowledge in areas of uncertainty and for the establishment of reliable design methods for catamaran structure.

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LIST OF SYMBOLS

Where equations are reproduced from references, definitions of their symbols are also provided. Each appendix has its own list of symbols.

Symbol	Definition
a_h	Aggregate horizontal acceleration
B	Beam of each hull
b	Hull centerline spacing
C_b	Block coefficient
C_{LA}	Centerplane area coefficient
C_w	Waterplane coefficient
C	Oblique wave coefficient
	$= \dfrac{S + B}{\sqrt{(S + B)^2 + (\frac{L}{2})^2}}$
C	Midship coefficient
D_o	Draft
d	$d_1 - 0.65\ D_o$
d_1	Distance of cross-structure neutral axis above base line
F_{sc}	Vertical shear at juncture of cross-structure and hull due to total cross-structure weight
F_{s1}	Maximum shear at juncture of cross-structure and hull
F_{so}	Maximum wave-induced shear at juncture of cross-structure and hull, weightless cross-structure
g	Gravitational acceleration
H	Wave height
$\overline{H}_{1/3}$	Significant wave height
H_L	Side hydrostatic force on outboard shell
H_R	Side hydrostatic force inboard shell
h	Horizontal shift of center of buoyancy of one hull
L	Length between perpendicular
M_1	Maximum vertical bending moment at juncture of cross-structure and hull
M_c	Moment at juncture of cross-structure and hull due to weight of cross-structure
M_o	Maximum wave-induced bending moment on cross-structure, weightless cross-structure
P	Maximum axial force
S	Clear hull spacing

Symbol	Definition
T_1	$\rho C_b g \, 0.6 \sqrt{\lambda_T} \; L^2/2\pi$
T_c	Maximum torque on cross-structure about its twist center, $t \neq o$
T_o	Maximum torque on cross-structure about its twist center, $t = o$
t	Longitudinal distance between ship LCG and cross-structure twist center
V_L	Centroid of H_L below neutral axis of cross-structure
V_R	Centroid of H_R below neutral axis of cross structure
W	Total width of catamaran
Y_L	Wave surface above still waterline at outboard shell
Y_R	Wave surface below still waterline at inboard shell
Δ	Total (both hulls) displacement
Δ_1	g x added mass in sway of both hulls
λ	Wave length
λ_T	LC_λ
ρ	Mass density of water
ω	Circular wave frequency

1. INTRODUCTION

The history of catamarans is old, references (1) and (2). However, in this century, it is only in the last decade that there has been a revival of serious interest in catamarans resulting in the construction of some sixteen vessels.

Except for one cargo vessel for use on the Volga, all these vessels are special purpose vessels, such as ferries, oceanographic research ships, fishing boats, drilling rigs and pipe-laying barges. Also, it is pertinent to note that these ships are under 315 feet in length, except for two, the 400-foot Duplus (Dutch) and the 425-foot Kyor Ogly (Russian). It may be recognized that for the special purposes in question, catamarans were selected over monohulls mainly to take advantage of the large deck area, high transverse stability,and good maneuverability at low speeds offered by the catamaran configuration.

The question has been raised, "why not large catamarans?" - both in the commercial sector and the Navy. In both groups, the interest is related to high-speed vessels for low density pay load. To answer this question, the Maritime Administration began with the Catamaran Study (1), performed by General Dynamics, and the Navy has undertaken a comprehensive assessment of catamaran technology (2), (3) and (4). Litton Industries claim an actual design of a semi-submerged catamaran container ship (5) and (6), and Fisher, et al, have prepared a preliminary design of a catamaran container ship for the Trans-Atlantic trade (7).

A salient obstacle in assessing the desirability of large catamarans has been the lack of technical information to establish the structural requirements. The purpose of the project reported here was to investigate into the technological limits to size and proportions of catamarans, appraise existing design procedures, and determine the additional structural knowledge required to insure their structural adequacy.

The features examined that could impose size limits were powering and propulsion, cross structure scantlings, construction problems, repair facilities, and harbor and pier limitations.

In order to estimate the cross-structure scantlings it was necessary to accomplish at least the first cycle of the preliminary design of a large catamaran of a size indicated by considerations other than cross-structure scantlings.

The major effort of the project was centered around the procedure for the structural design of the cross-structure. The task was divided into three parts, viz: (a) Assembly and comparison of all available model test data on the loads on the cross structure; (b) Evaluation of the analytical methods for estimate of cross-structure load and (c) Structure analysis methods.

Numbers in parentheses refer to references listed.

New equations are proposed for the estimate of wave-induced vertical bending moment, axial force and shear force. Modifications are proposed to an existing equation for torsion.

The project scope was limited to conventional surface catamarans as opposed to semi-submersible catamarans (column-stabilized or strut-stabilized). No attempt was made to analyze the influence of symmetrical hulls or non-symmetrical hulls on the size limit or the cross-structure of catamarans.

Of all the aspects of catamaran design, resistance has received the most attention in the past. Considerable work has been done in the areas of theoretical prediction and model test measurements, as well as their correlation. A brief statement on the most important aspects of catamaran resistance as gathered from the literature is provided in Appendix 1.

Recommendations are made for the future research and development program for large catamarans.

2. ANALYSIS OF FEATURES THAT MAY IMPOSE SIZE LIMITS

It appears, in principle, that there are no insoluble technical considerations which would preclude the design and construction of a 1000-foot catamaran in the United States. This does not imply that the facilities exist to build many ships immediately, that there will not be special problems to overcome, or that there is no need for future research and development effort necessary to build an efficient vessel. What is meant is that if economics strongly favor a large catamaran, the venture to design and build one may be undertaken without a strong reservation that some unknown technological problem would force the premature termination of the venture.

The features considered in reaching the foregoing conclusion are as follows:

a. Resistance-Powering-Propulsion:

Main machinery and propulsion system for a large catamaran does not present a situation not found in large monohull designs. Depending on speed and draft, very large catamarans may require more than one propeller per hull. However, this need not set an upper limit to the catamaran size, assuming that hull beam is sufficient, and form can be designed to accommodate more than one propeller. Machinery weight and volume should be acceptable.

b. Wave Loads, Cross-Structure Scantling and Structural Material:

The hydrodynamic effect unique to catamarans and of prime consideration is, of course, the differential wave loading on the hulls to be absorbed by the cross-structure. Design checks for up to approximately 1000-foot catamaran with 100-foot clear hull spacing show that cross-structure with practical scantlings can be designed to absorb the wave loads. With full transverse bulkheads at approximately 50-foot spacing and making the conservative estimate of effective flange, the maximum steel (100,000 psi yield) plate thickness is 1-1/4 inches. There is no doubt that the cross-structure material would have to be steel.

c. Drafts:

Water depths at existing cargo piers around the world suggest draft limitation of approximately 35 feet.

d. Construction:

Existing United States drydock facilities can build up to approximately 1050' x 140' monohulls. Bethlehem Steel Company's new drydock at Sparrows Point, Maryland will measure 1200' x 200'. One million ton drydocks under construction in Japan and Northern Ireland will be approximately 1965' x 329'. Catamarans with overall beam larger than the width of the available dock would have to have the hulls and the center-body assembled with hulls afloat. The latter technique was used in the E.W. Thornton construction. Twin docks with equal depth, just the correct depth and just the correct width, may be an answer, if available.

e. Drydocking:

Drydocking poses a problem if the desired catamarans are too large for the dry-dock sizes mentioned in the previous paragraph. Modification of existing facilities or construction of new facilities will be required. From a technical viewpoint, use of two floating docks may be feasible.

One must not underestimate the ingenuity of shipyards to solve the drydocking problem. Evidently no serious reservation was held regarding drydocking when the construction of the 250-ft wide Mohole Platform was initiated.

The Levingston Shipbuilding Company has drydocked the 105-ft wide E.W. Thornton on a single floating drydock split into two longitudinal halves held together by spacer beams.

It is believed that the Russians have a scheme for dismantling their relatively small catamarans for maintenance and repairs.

f. Cargo Handling and Piers:

The problems of cargo handling and piers are economic problems. They can be solved, at a price, if the economics of catamarans were so attractive. Use of twin piers or discharge of cargo offshore have possibilities.

g. Channels and Harbors:

Certain unpublished studies claim that the majority of major harbors around the world can accept 1000' x 400' catamarans.

h. Economics:

The General Dynamics study (1) and certain unpublished studies claim that the economics of catamarans as compared to economics of monohulls are unfavorable or at the most marginal. Captain M. Eckhart, Jr. reporting on the Navy's findings to date (3) states "No compelling reason is yet in sight for a general shift from the monohull to the multihull or catamaran configuration."

3. EXISTING STRUCTURAL DESIGN METHODS

3.1 General

The coverage of existing design procedures is limited to the cross-structure since without exception individual hulls have been treated as monohulls.

Neither the classification societies nor the governmental agencies have established design criteria or guidelines for cross-structure design and designers must follow general structural engineering techniques. In the case of the T-AGOR 16 Catamaran Research Ship design the Navy did suggest the use of the paper "The Structural Design of the ASR Catamaran Construction" by Lankford (8) as guidance.

3.2 Cross-Structure Loads

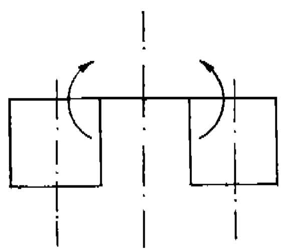
As for any structure, there are two phases to the cross-structure design, namely, the determination of the loads and the design of the structure to absorb the loads.

The loads experienced by the cross-structure are:

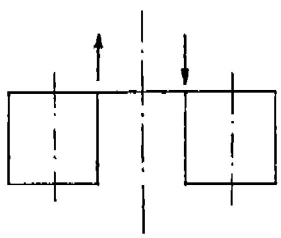
A. Calm water load due to the weight (lightship weight and deadweight) of the cross-structure.

B. Wave-induced loads due to differential wave loads on the individual hulls.

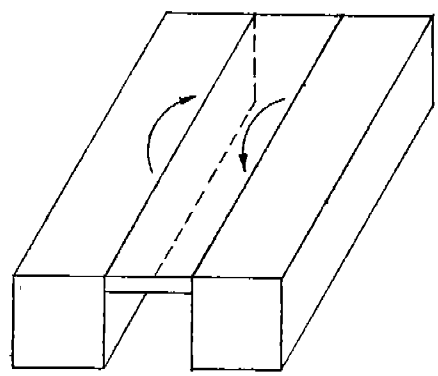
i. Transverse vertical Bending Moment, usually referred to as just the Bending Moment or sometimes even as the Roll Moment.



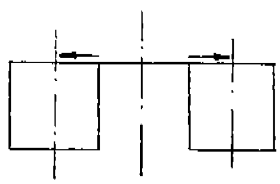
ii. Vertical Shear Force, usually referred to as just the Shear Force.



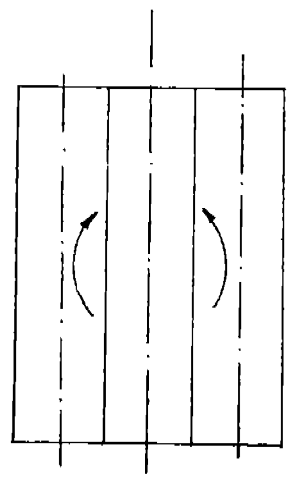
iii. Torsion Moment, sometimes referred to as the Pitch Moment.



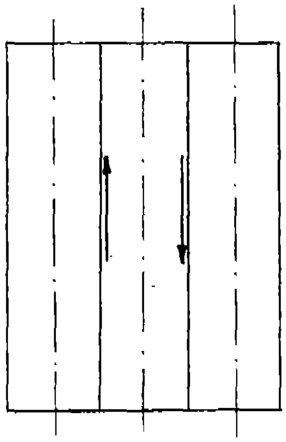
iv. Transverse in-plane Horizontal Force or Side Force.



v. Horizontal in-plane Moments or Yaw Moment.



vi. Longitudinal in-plane Force.



vii. Water impact loads.

C. Grounding and Docking Loads

The controlling loads in the cross-structure design are the wave-induced loads numbered i, ii, and iii, grounding and docking loads (if grounding and docking is considered a design criteria) and the calm water loads. Impact loads are treated as local loads and require reinforcement of the cross-structure bottom and inboard shell of the individual hulls.

Side forces which appear to be instrumental in causing the maximum vertical bending moments are of sufficient magnitude to be included in the direct stress calculation. Earlier designers tended to neglect them and only in one conventional catamaran model test (9) (report unpublished) were the side forces measured. Loads (v) and (vi) cause negligible stresses.

The rest of this section is devoted to the survey of the existing structural design methods. However, at this point it may be desirable to point out that the project investigators' conclusions as to the vessel positions with respect to the waves that are likely to give rise to the maximum response and the recommended method for design load estimate appear in Section 5.

Table 1 - Catamaran Load and Structure Analysis

		Areas of Contribution							
		C-S		Wave Loads					
	Ref.	Steel Wt Est.	Grounding Loads	Bending Mom.	Shear Force	Torsion Moment	Structure Analysis Bending	Structure Analysis Shear	Structure Analysis Torsion
R. Scott	10			+		+	+		+
B.W. Lankford, Jr.	8		+				+	+	+
H.A. Schade	12 & 13			+	+	+			
A.L. Dinsenbacher	13			+	+	+			
G.O. Thomas	4	+					+	+	+
J.L. Glaeser	14			+	+				
C.W. Levingston and W.H. Michel	15	Description of E.W. Thornton Structure							
W.H. Michel	16	Description of Univ. of Miami Catamaran Design Structure							

3.3 Survey of Existing Design Methods

Table 1 lists load and structure analysts and their published contributions. It is emphasized that designers of catamarans actually built have relied heavily on model tests to provide the numbers for wave loads. Model test data analysis is covered in Section 4. Brief description and discussion on the work of each structure analyst listed in Table 1 follow. However, any calculations performed to assess their methods are included in tables of Section 5. These tables compare model test predictions, calculations by existing methods and calculations by new equations presented in this report.

3.3.1 R. SCOTT

While still a Naval Architectural student at the University of Michigan, Scott proposed expressions for the stresses due to torque and transverse bending of a catamaran cross-structure (10). They are as follows:

Torsion:

To obtain the torsional bending moment, a fine-lined 300-foot long vessel was poised obliquely on a trochoidal wave, 170' x 10'. The crest coincided with the forward quarter point of one hull and the aft quarter point of the other hull, with the trough at the extremities. (Scott has not provided additional information on the vessel or the basis for selecting a 10-foot high wave.) Under this attitude of the vessel, the center of buoyancy of the hulls moved toward the crest by an amount equal to 4 percent of the length. Thus, each hull had a torque of 0.04L times the displacement per hull and the total torque on the cross-structure was given by $T = 0.04L\Delta$

Where Δ = Total displacement of catamaran

Assuming the wing structure as a thin walled rectangular tube in torsion, the stress, S, was given by

$$S = \frac{T}{2\,At}$$

where A = Area of the tube and

t = Tube thickness

The approach to obtain the total torque moment, as simple as it may be, has merit for application in early stages of the design. Torque as given by $0.04L\Delta$ have been compared with model test results in Table 9. Except in the case of one vessel where the test value is 16% higher, in all other cases, $0.04L\Delta$ would provide conservative estimates.

Little application can be found for the stress expression as all known catamarans have longitudinally discontinuous cross-structure which can not be idealized as a single tube.

Transverse Bending:

It was assumed that during severe rolling in beam seas one of the hulls can become partially emerged where one-half of the entire displacement of one hull is cantilevered from the end of the cross-structure. Under this assumption the stress on the cross-structure is expressed as

$$\text{Stress} = \frac{\text{Hull separation} \times 1/2 \text{ displacement of one hull}}{\text{Section modulus of cross-structure on centerline}}$$

$$= \frac{(W-2B)\Delta/4}{\text{Section modulus}} = \frac{S\,\Delta/4}{\text{Section modulus}}$$

A portion of Table 7 is a comparison of bending moments given by $S\Delta/4$ with available model test results. It shows that the test value for ASR is higher than $S\Delta/4$ while for other vessels $S\Delta/4$ is higher than the test values.

(Note: Here S = clear hull spacing)

Even though Scott's assumption provides bending moment values higher than the model tests it is questionable whether the particular assumption of the ship-wave relationship generates the maximum bending moment. A more detailed discussion on the condition for maximum bending moment appears in Section 5.

3.3.2 B.W. LANKFORD, JR.

Lankford's well-known and valuable paper, "The Structural Design of ASR Catamaran Cross-Structure" (8) includes the following:

i. Analytical approach to sea load prediction

ii. Distribution of the design sea loads

iii. Drydocking and grounding loads

iv. Structural configuration of the ASR

v. The design procedure

The design wave-induced vertical bending moments were obtained by making a long term prediction. The prediction calculations used response amplitude operators provided by model tests (11), ocean wave spectrum derived from data on 12 most severe storms at the National Institute of Oceanography (Great Britain), and wave frequency occurrence in the North Atlantic.

The part of the paper which covers points (ii) through (v) mentioned above, together with the references, is reproduced in Appendix 2 of this report.

Lankford uses drydocking and grounding loads as design criteria. Based on the assumption that the vessel is docked or grounded with maximum weight in such a manner that one hull is supported forward at station 4 and the other is supported aft at station 18, the design torque is given by $\Delta d/4 = 0.175\,L\Delta$. This criteria is considered overly conservative and it gives torque values which are much higher than wave induced torque as can be seen in Table 9. The assumed loading condition where no buoyancy support is available can occur during docking only. Further, one must assume that the hull flexibility is not such that the vessel weight can force the keel down to the blocks.

The Lankford method of cross-structure analysis is likely to attract designers for two reasons, viz:

i. It is neatly stated and simple and quick to apply.

ii. It is the only available method which has been applied to vessels actually built, namely the ASR and the T-AGOR 16.

However, the readers must be cautioned against the unreserved acceptance of this method as it appears to oversimplify the structure and make some questionable assumptions. Further, the method does not assure an economic nor a conservative structure. The primary oversimplification is that the hulls are rigid. The primary questionable assumption is that there is no relative rotation between the hulls and the cross-structure at the junction of the hulls and the cross-structure.

3.3.3 H.A. SCHADE and A.L. DINSENBACHER

Schade's and Dinsenbacher's works (12) and (13) are considered together since the methods employed by Dinsenbacher to develop equations for axial forces, vertical moment, shear and torsion moment are refinements of methods developed by Schade. The Ship Structure Committee project reported here benefited from the information and style of presentation in these two references. The following paragraphs are taken directly from the Introduction and Analysis section of Dinsenbacher's paper and they state the refinements made to Schade's methods, and the assumption of the methods. The Summary and Discussion (from the same paper) which include the equations developed are reproduced in Appendix 3 of this report. (The reference numbers in the quotation refer to the references in the paper which are also included in Appendix 3.)

> "In 1965 Professor H.A. Schade made a feasibility study of an ocean-going catamaran in which equations were developed for estimating the cross-structure loads (1). The author assumed the hulls to be prismatic forms acted upon by vertically fronted waves. It was decided to compare the loads resulting from this method to results from a model test of an ASR catamaran (2). The comparison showed Schade's loads to be somewhat higher than those found from the model test. Also, Schade's method relates wave height only to ship dimensions, and not to wave length. It was thus decided to employ many of the general aspects of Schade's method but to modify the waves used in his study. Sinusoidal waves are substituted for the vertically fronted waves. The wave lengths are related to the ship dimensions in an effort to optimize loads. Also, the wave amplitudes are related to the current design wave height-length relationship and to the loads measured on the ASR catamaran model."

> "The resulting empirical equations devised herein are simple and quick to employ. They are founded on a combination of a more realistic wave shape, the current design wave height-length relationship used for longitudinal strength, model and full-scale evaluations of current surface-ship hull girder design loads, and loads measured on a catamaran model in waves. A procedure for estimating primary stresses resulting from the gross loads is also included."

Assumptions: (Quotation Continued)

> "For this study, in a manner similar to that of Schade, the ship is idealized as two rectangular prisms (representing the hulls) connected by a rectangular box (the cross-structure). The longitudinal and transverse dis-

tributions of weight are taken as uniform in the hulls and in the cross-structure. The length, beam, draft, and weight of the prismatic representation of the hulls are taken as those of the actual hulls. The interconnecting box has the same length (span between hulls), width, depth, weight, clearance above still water, and vertical location of neutral axis as does the actual cross-structure. The fluid density used for the computation of vertical forces is modified here to compensate for the difference in displaced fluid between the rectangular blocks and the actual hull forms as was done by Schade; however, the fluid density is not modified in the computation of transverse loads. Also, the drafts are found for the prismatic forms which produce vertical accelerations of $\pm 0.4g$, and these accelerations and drafts are used in computing the loads on the prismatic idealization. These heave acceleration amplutudes of $\pm 0.4g$ are not unrealistic maxima to expect for the ship's service life (3). Sinusoidal waves rather than vertically fronted waves are used. Pressures are assumed hydrostatic. Inertia forces on the ship mass are included in calculating loads. Slamming and whipping are ignored. It is further assumed herein that relative positions of wave and ship similar to those which produced the highest cross-structure loads in Schade's work will result in the worst conditions. Therefore, only the loading conditions shown in Figures 1 and 2 will be considered."

Figures 1 and 2 are included in Appendix 3.

Comments on the equations developed and their associated assumptions follow:

Axial Force:

The equation for axial force in beam seas does not account for the possible force contribution due to the horizontal acceleration, which can be substantial.

Detailed discussion on the probable conditions for maximum loads as concluded from some independent analysis and available test data is covered in Section 5.

Bending Moment and Shear:

i. Although not stated specifically the bending moment equations development assume that maximum side hydrostatic force and maximum vertical acceleration occur at the same time for the relative wave and ship position of Loading Condition 1 (see Appendix 3, Figure 1). Also that the sense of the acceleration on both hulls is the same.

ii. The second term on the right side of equation (75), Appendix 3, for maximum shear is obtained by relating the shear and bending moment RMS values in 40-knot wind beam seas for the ASR catamaran. Intrinsic to this operation is the assumption that shear and moment are in phase or that the particular shear is the result of the particular moment.

The validity of the foregoing two assumptions is doubted. The two assumptions do play a very important part in the resulting equations for maximum bending moments and shear. The reasoning behind the objections will be found in Section 5.

Torque:

Equation (79) Appendix 3, developed for maximum torque (which occurs in oblique seas) about the twist center of the cross-structure is

$$T_o = \left| \rho C_{bg} BAL^2/2\pi \right| + \left| 0.14\, M_Q\, t/S \right|$$

The first term on the right represents the torsion about the center of gravity of the ship, while the second term represents the torsion due to shear acting through the ship's center of gravity, which tends to differentially heave the hulls. The latter term is obtained by relating the maximum shear to the maximum bending moment (for a catamaran with weightless cross-structure) in the same oblique wave which causes the maximum torsion. This assumption is the same as the second assumption listed under bending moment and shear and its validity is doubted also. Attention is drawn to the fact that the term in question is not likely to be large unless t, the distance from center of center of gravity of the ship to the center of twist of the cross-structure, is large.

The development of the first term in the torsion equation is found to be logical and preferred over Scott's expression for torque. It seems to take in as many details as possible without beginning with the fundamental equations of motions. The first term is employed to nondimensionalize the test data (Section 4).

3.3.4 G.O. THOMAS

G.O. Thomas delivered a lecture (4) entitled "Structural Analysis of Catamarans" as one part of a short course on "Modern Techniques of Ship Structural Analysis and Design" at the University of California in September 1970. It was a generalized lecture based on the conceptual design of a naval strike platform for which considerable design information was collected and design criteria developed.

The material on design load derivation was as presented by Dinsenbacher and discussed earlier in this report.

In developing the design criteria for aircraft carriers, Thomas was able to refer to some very recent work performed at the British National Physical Laboratories (unpublished) and at the Naval Ship Research and Development Center. The section on structural design criteria selection contained formulas for cross-structure clearance and slamming loads which are applicable to catamarans in general.

Thomas' formula for cross-structure clearance above load waterline is

$$C = 3 + 1.1 \sqrt{2(S + B)} \quad \text{but } C \not< 20$$

The clearance as calculated by this formula compared quite closely to the actual clearance for the E.W. Thornton and the ASR but it gave much higher values than actual for the University of Miami design and the Ridgely Warfield. In this respect it is pertinent to note that the forward end of the Ridgely Warfield's cross-structure is bow shaped and designed for low clearance. It is suspected that for very large catamarans the cross-structure clearance may be controlled by the minimum depth and freeboard requirements for the individual hulls. Also, the designer is likely to pay some penalty in terms of additional clearance if the ends of the cross-structure are within approximately 0.15L of the ends of the hulls.

Thomas provides a fairly lengthy discussion on the design criteria for cross-structure slamming. He elects to treat the relatively small forward-and-aft areas as local areas since they are of minor importance to the overall cross-structure weight. The following discussion on the slamming loads on the large middle areas (referred to as Region 2) is quoted directly from Thomas' lecture notes (4).

"In Region 2, slamming of the largest area of cross-structure bottom plating was assumed to be caused by the descent of the cross-structure right on top of a wave passing through the catamaran flume. This may not be strictly the case, but lacking specific information, it was taken to be so. Wave buildup within the tunnel was neglected since it primarily effects slamming aft. A second unpublished report by the National Physical Laboratory shows that high-impact pressures aft for a catamaran with water pile-up and without anti-pitching fins were a little less than at the forward quarter point."

"Loads from slamming on the cross-structure bottom in Region 2 can be divided into two kinds: (a) short-term high-impact pressures acting locally in the lateral direction for panels and on the edges of floors and (b) longer duration for lower pressures used for cross-structure bottom bent and overall cross-structure bottom grillage design."

"The highest pressures for short-term slamming can be taken as for flat bottom impact. This can be justified by considering that welding distortion can cause a slightly concave appearance to the cross-structure bottom plating which could then slam on wave crests as a flat bottom. The equation used for flat-bottom slamming is from Chuang*

$$p = 4.5V \quad 64/62.4$$

where p is the flat bottom slamming pressure in pounds per square inch, V is the relative motion between ship and fluid in feet per second, and the value 64/62.4 converts pressures from those for fresh water to those for sea water. The slamming station for relative motion was taken at 0.46L forward of amidships and, since impact pressures are assumed to occur when the ship descends on top of the wave, impact velocities were based on design maximum pitch motion. Pressures greater than those from flat-bottom slamming can be experienced as transients for relatively shallow deadrise angles of hull to fluid. However, these pressures are usually very localized to the water-structure interface and were assumed to carry insufficient momentum to affect the design of the plating."

*Chuang, S.L., "Experiments on Flat-Bottom Slamming," Journal of Ship Research (March 1966)

"The first mentioned unpublished NPL report showed that raising the cross-structure on a catamaran model reduced the frequency of slams of a given severity but did not reduce the intensity when they did occur. Pritchett** has confirmed this conclusion in more recent testing at NSRDC. The general concensus so far is that for the higher most probable sea conditions (Beaufort 6 in one case and State 7 sea in another), short-term, high-impact slamming pressures can be assumed to be between 80 and 120 psi, regardless of the size of the ship or height of the cross-structure (within reason). Slamming pressures from the Chuang equation fell within this range for all catamarans of the series."

"High impact flat-bottom slamming pressures were applied over single panels of bottom plating which were then designed as for boundaries of tanks, and to floors and double bottom longitudinal girders to design against local collapse."

"Following the initial slam on the bottom plating of the cross-structure, the pressure can be assumed to drop very rapidly to that given by $1/2\, \rho V^2$ where ρ is the mass density of sea water, i.e.,

$$P = 0.994\, V^2$$

where P is the flat bottom pressure in pounds per square foot and V is the relative motion between ship and fluid in feet per second."

"For this relationship the relative velocity between ship and fluid can be taken to include the orbital velocity of particles in the wave since the cross-structure bottom might now be well below the crest of the wave. Pressures from this equation ranged from 600 to 900 pounds per square foot for the catamaran series studied. These pressures were then applied to the overall cross-structure bottom grillage design."

Thomas has also developed a weight equation for the cross-structure of a catamaran but its application is extremely restricted. Actually, it was developed for the conceptual design on aircraft carriers. The equation is not presented here due to its acknowledged limitations and high probability of involving large errors when applying it to nonaircraft carrier type structure.

3.3.5 JOHN L. GLAESER

While at the Webb Institute of Naval Architecture, Glaeser prepared an undergraduate thesis entitled "A Theoretical Investigation Into the Motions of a Catamaran and the Shear and Bending Moments on its Cross-Structure" (14). The responses considered were heave, roll, shear and vertical bending moment. As a check on his theory, Glaeser calculated the responses for the ASR and compared them with the model test results (11). Figure 1 (taken from the summary of the thesis) shows the comparison.

**Pritchett, C., "Model Studies of ASR-Catamaran Impact Pressures on Between Hull Structure," Naval Ship Research and Development Center T & E Report 340-H-01 (January 1970).

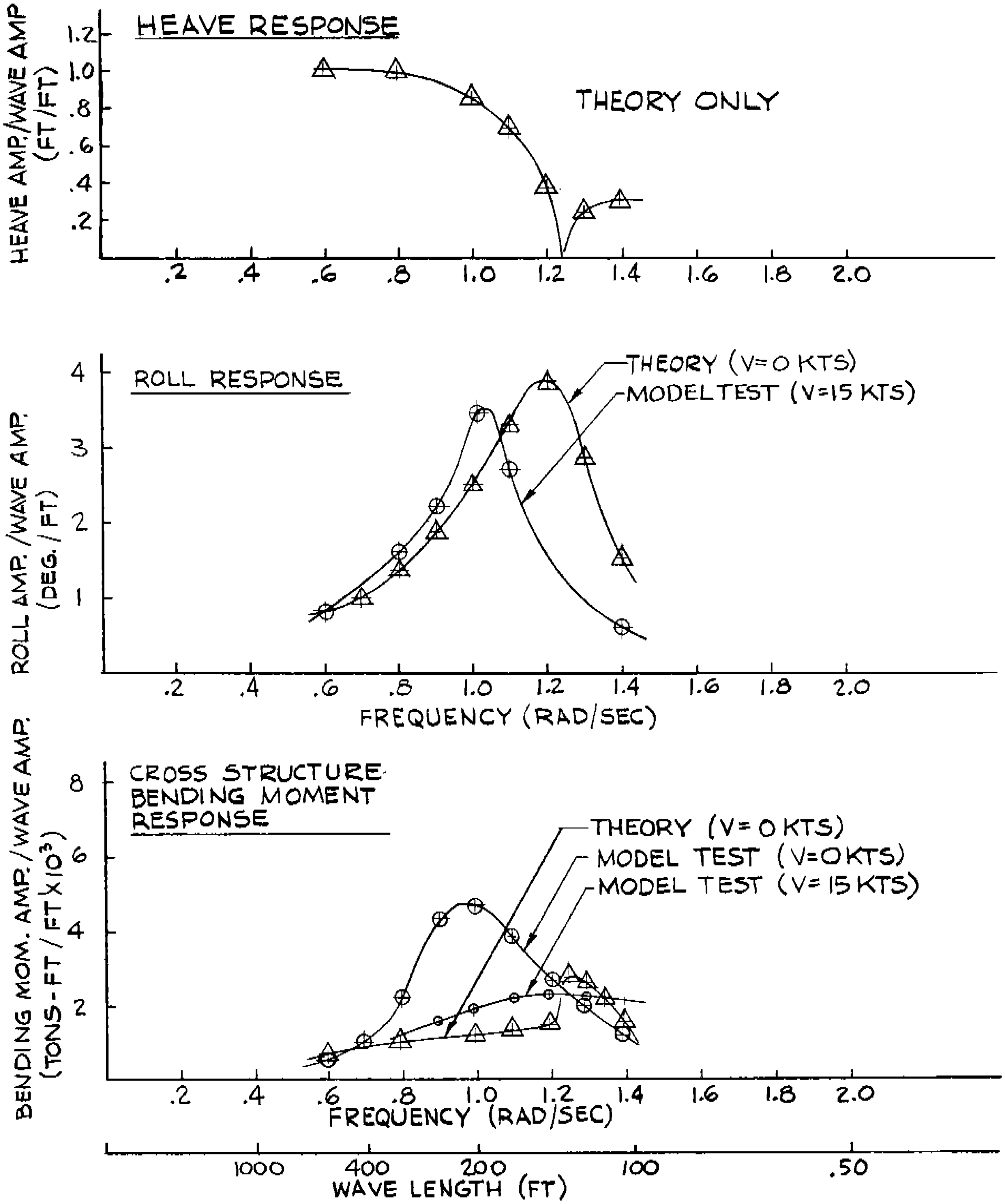


Fig. 1 - Catamaran Response in a Regular Beam Sea
(Reproduced from Ref. 14)

To permit the most basic analysis the problem was simplified by taking a catamaran at zero speed in a two-dimensional cosine wave. This is reasonable as maximum roll and vertical moments appear to occur in beam seas. Other primary assumptions of the theory are as follows:

Motion Calculations:

1. The hulls are thin enough, and the roll small enough so that the wave height at the center of a hull is the same as at the sides. The catamaran is wall sided.

2. All the hydrodynamic, hydrostatic and inertial forces act through a point on the centerline of each hull.

3. The catamaran is not pitching and there is no cross coupling effect between heave and roll.

Shear and Bending Moment Calculations:

1. All hydrodynamic and hydrostatic forces act through the center of buoyancy of each hull as it moves.

2. The cross-structure is weightless (in accordance with the model test).

3. The catamaran rolls about its center of gravity and is wall sided.

First, Glaeser wrote and solved the differential equations for heave and roll. Then knowing the motions of the vessel, the forces on each individual hull were calculated, the forces being those which made up the original differential equations. The constants of proportionality, added mass and damping were calculated using Grim's coefficients.

Comments on the Comparison of Theoretical Calculation and Model Test Results for the ASR:

See Figure 1. Although the shear response comparison is not included in the summary, it is included in the principle thesis. However, the shear comparison is nearly identical to the roll motion comparison.

The roll and shear correlations are very respectable except that the theoretical maximum occurs at $\omega \approx 1.2$ ($\lambda \approx 2$ hull centerline spacing) while the experimental maximum occurs at $\omega \approx 1$ ($\lambda \approx 2$ overall width). It is suspected that this is due to the simplification that the hulls are thin and that the vertical force acts through a single point. The agreement in magnitude leads one to conclude that the theory has succeeded in identifying, at least, the principle parameters which influence roll motion and shear force.

Figure 1 does not show the model test heave. In this respect it is valuable to note that the theoretical heave curve is very much like the Thornton model test curve in which heave/wave height is approximately zero at that wave frequency when roll, shear and bending moment are maximum and approximately unity at low frequencies.

The bending moment correlation is poor casting a doubt on the theory. As Glaeser himself suspected it is most likely due to neglecting both the hydrostatic and hydrodynamic side forces. It will be observed that locations of maximums are the same as for roll motion.

4. MODEL TEST DATA ANALYSIS

As mentioned earlier model tests have assisted greatly in the estimation of wave-induced loads on the cross-structure of catamarans. What is more important to recognize is that they will continue to do so until theoretical and semi-empirical methods have been proven to a high degree of confidence (which takes time).

This section consolidates and compares the available model test data on the loads imposed by sea waves on the catamaran cross-structure. Limitations of the various test programs and the consequent limitations of the data comparison are enumerated. The purpose of the comparison was to determine the gross relationship between the loads and the major parameters of the catamaran design and waves.

4.1 Test Background

4.1.1 Test Vessels

The prototype characteristics of the vessels whose model test data were available to this project, are provided in Table 2. It will be observed that within the data plots appearing in the report are data points marked "Undisclosed Series." These are from an unpublished test report of a conventional catamaran.

The bulk of the analysis has been centered around the "Thornton" and the "ASR" whose test programs included a large range of sea conditions and the data, as reported, are amenable to extrapolation and comparison. The amenability to extrapolation was most valuable as it was helpful in estimating loads on large catamarans.

The portion of the Mohole and the Levingston 6-column semi-submersible platforms test data which were useable were the data for the ocean tow condition. In this condition the water lines are below the top of the lower longitudinal hulls and the vessels are essentially surface catamarans. Test program for the University of Miami Research Vessel Design was quite limited.

4.1.2 Loads Compared

The loads compared were the two moment and one force measured in each test with model at zero speed, viz:

- Vertical Bending Moment in Beam Seas
- Vertical Shear Force in Beam Seas
- Torsion Moment in Oblique Seas

The crucial side forces which are the major cause of the maximum vertical moment were measured in the Levingston test only. The reported acceleration data for the various tests are inadequate to attempt a meaningful comparison.

4.1.3 Pertinent Notes on the Tests

a. All the test models simulated the total weight, centers and gy-radii of the catamaran as a rigid body. None of the models simulated the structural rigidities of the centerbody or the cross members.

b. The ASR report (11) provides random wave test results (only) in terms of response amplitude operators and response spectral energy.

The other tests which were all performed at the Davidson Laboratory reports both regular wave and random wave test results. However, the random wave test results are in terms of averages only.

c. The all important information on phase relationship between the various loads and the wave are available for the Mohole and the Levingston tests only.

d. Each test was performed for a specific configuration and one loading condition only.

e. Load measurement system: The ASR test used four strain gages mounted on two rigid aluminum bars, one forward and one aft to measure loads.

The Davidson Laboratory used Schaevitz force measurement dynamometers which are linear variable differential transformers to measure loads. (The dynamometers have a core mounted between two springs and the voltage output is proportional to the displacement of the core.) Although the actual instrumentation arrangement was not the same for every Davidson Laboratory test, the following paragraph from the "Thornton" Report (17) is informative of the principle of the system.

"The hulls were connected by a rigid bridge structure which was a part of the force and moments measuring system. The bridge was fixed to the port hull and was connected to linear force measurement-dynamometers in the starboard hull. The bridge was made up of three frames which spanned the hulls at the L.C.G. and at two points 12 inches forward and aft of the L.C.G. The frame at the L.C.G. was

Table 2 - Prototype Characteristics of Model Test Vessels

	E.W. Thornton Drilling Ship	ASR Submarine Rescue Ship	Univ. of Miami Research Vessel	Mohole - Semi-Submersible Platf	Levingston Semi-Submersible Platform
Reference Number	15, 17	11, 18	16, 19	20, 21	22
Test Facility	Davidson	NSRDC	Davidson	Davidson	Davidson
Hull Symmetry	Unsym	Unsym	Unsym	Sym	Sym
Length Overall	-	-	146'-8"	390'-0"	260'-0"
Length Bet. Perp, L	255'-0"	210'-0"	136'-6"	* 355'-0"	-
Beam Overall, W	105'-0"	86'-0"	50'-5"	250'-0"	200'-0"
Beam Each Hull, B	37'-0"	24'-0"	16'-10"	35'-0"	36'-0"
Hull ℄, Spacing, b	68'-0"	62'-0"	33'-7"	215'-0"	164'-0"
Clear Hull Spacing, S	31'-0"	38'-0"	16'-10"	180'-0"	128'-0"
Test Draft, D_o	17'-0"	18'-0"	9'-5"	28'-7"	16'-0"
Total Displacement, Δ	6700 T	2797 T	695 T	16,800 T	7700 T
Block Coef, C_b	0.73	0.54	0.56	0.75	0.90
Waterplane Coef, C_w	0.84	0.737	-	1.0	1.0
Centerplane Coef, C_{LA}	0.92	0.92	-	1.0	1.0
L/b	3.75	3.387	4.063	1.163	1.220
L/D_o	15.00	11.67	14.44	13.64	16.25
B/D_o	2.18	1.33	1.78	1.24	2.25
L/B	6.89	8.75	8.13	11.14	7.22
b/W	0.648	0.721	0.667	0.860	0.820
Oblique Wave Coef, C_λ	0.47	0.51	0.46	0.77	0.78

* Assumed value

Table 3 - Particulars of "E. W. Thornton" Series Ships

TABLE 3 - PARTICULARS OF "E.W. THORNTON" SERIES SHIPS

	Ship A	Ship B	Ship C	E.W. Thornton
Scale	1:2.383	1:2	1:1.278	1:1
LBP, L	607.67'	510.0'	325.89'	255.0'
Beam Overall, W	250.22'	210.0'	134.19'	105.0'
Beam Each Hull, B	88.17'	74.0'	47.29'	37.0'
Hull Spacing, S	73.87'	62.0'	39.62'	31.0'
Hull ℄ Spacing, b	162.05'	136.0'	86.9'	68.0'
Draft, D_o	40.51'	34.0'	21.73'	17'- 0"
Displacement, Δ	90,800 T	53,600 T	14,000 T	6,700 T
d	52.43'	44.0'	28.12'	22.0'
2 (W-B) = 2b	324.10	272.0	173.8	136.0

Table 4 - Particulars of "ASR" Series Ships

TABLE 4 - PARTICULARS OF "ASR" SERIES SHIPS

	Ship A	Ship B	Ship C	ASR
Scale	1:3.19	1:2.675	1:1.71	1:1
LBP, L	669.90'	561.75'	359.1'	210.0'
Beam Overall, W	274.34'	230.05'	147.06'	86.0'
Beam Each Hull, B	76.56'	64.20'	41.04'	24.0'
Hull Spacing, S	121.22'	110.65'	64.98'	38.0'
Hull ℄ Spacing, b	197.78'	165.85'	106.02'	62.0'
Draft, D_o	57.42'	48.15'	30.78'	18.0'
Displacement, Δ	90,800 T	53,600 T	14,000 T	2,797 T
d	93.79'	78.65'	50.27'	29.4'
2 (W-B) = 2b	395.56	331.70	212.04	124.0

Table 5 - Particulars of the University of Miami Series Ships

TABLE 5 - PARTICULARS OF THE UNIVERSITY OF MIAMI SERIES SHIPS

	Ship A	Ship B	Ship C	Univ. of Miami Ships
Scale	1:5.080	1:3.339	1:1.143	1:1
LBP, L	693.4'	455.8'	156.0'	136.5'
Beam Overall, W	256.0'	168.3'	57.6'	50.4'
Beam Each Hull, B	85.3'	56.1'	19.2'	16.8'
Hull Spacing, S	85.3	56.1'	19.2'	16.8'
Hull ℄ Spacing, b	170.7'	112.2'	38.4'	33.6'
Draft, D_o	48.0'	31.5'	10.8'	9.45'
Displacement, Δ	91,108 T	25,827 T	1,042 T	695 T
d	106.2'	69.8'	23.9'	20.9'
2 (W-B) = 2b	341.4	224.4	76.8	67.2
Significant Wave Ht, $\overline{H}_{\frac{1}{3}}$	45.7	30.0	10.3	9.0

attached to two dynamometers spaced three inches apart while the two other frames were attached to single dynamometers located on the centerline plane of the starboard hull. All dynamometers gave the relative shear force and the relative pitch moment, while the outputs of the two dynamometers at the L.C.G. registered relative roll moments."

It should be clarified that the ASR System measured the total vertical bending moments, i.e., primary moments and secondary moments due to shear, whereas the Davidson Laboratory System measured primary bending moments only.

4.2 Data Consolidation and Comparison

As mentioned previously the data analysis is centered around the "Thornton" and the ASR tests. To accomplish data extrapolation, the "Thornton" and the ASR prototypes were expanded into a series of geometric ships up to 90,800-ton displacement. Tables 3 and 4 provide the particulars of the series. The wave loads response amplitude operators were expanded by Froude scaling. The ASR test report provided the R.A.O.s* while the Thornton R.A.O.s were based on the regular wave data. It should be clarified that the ASR R.A.O.s picked from the report were the mean values of two runs for each condition. The response of each ship in the series was obtained in sea state 5 ($\bar{H}_{1/3}$ = 10'), sea state 7 ($\bar{H}_{1/3}$ = 30') and sea state 8 ($\bar{H}_{1/3}$ = 50') using the Pierson-Moskowitz spectrum represented as follows:

$$S(\omega),\ \text{ft}^2\ \text{sec}^2 = \frac{16.78}{\omega^5}\ e^{-\dfrac{33.56}{\bar{H}_{1/3}\ \omega^4}}$$

Area under curve of S (ω) vs ω equals $\bar{H}_{1/3}/2.832$

The University of Miami Research Vessel design test data was too limited to deduce response amplitude operators. For the one random wave test, the wave and response information is reported in terms of averages only. To make the most of the data, it was expanded to three prototype ships which had test significant wave height equivalent to 10.3 feet (sea state 5), 30.0 feet (sea state 7) and 45.7 feet (sea state 8). Particulars of these ships appear in Table 5. The Undisclosed Series was developed in the same manner.

The semi-submersible platform data was used "as is."

All the test data assembled are for zero speed. In case of the ASR model tests (11), the load measurements were made in forward speeds up to 20 knots and it was found that the maximum loads occurred at zero speed. This finding need not be applicable to all craft, particularly very high speed craft.

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*Response Amplitude Operators

There is general agreement among the different test data that maximum wave-induced bending moments and shear force occur in beam seas while the maximum torsion moment occur in oblique seas (45° to 60° off 0° or 180° heading). A significant correlation between the "Thornton" and the ASR tests, the two tests for which R.A.O.s are available, is that the maximum bending moment and shear occur in waves with length equal to approximately 1.8 to 2.0 times the overall beam.

Non-dimensionalized data is presented in the following plots:

Figure 2: $\dfrac{\text{Max. vert. bend. mom.}}{d\,(\Delta + \Delta_1)/2}$ Versus Δ, Beam Seas

Figure 3: $\dfrac{\text{Max. vert. bend. mom.}}{d\,(\Delta + \Delta_1)/2}$ Versus L, Beam Seas

Figure 4: $\dfrac{\text{Max. vert. bend. mom.}}{d\,(\Delta + \Delta_1)/2}$ Versus b, Beam Seas

Figure 5: $\dfrac{\text{Max. shear force}}{\Delta/2}$ Versus Δ, Beam Seas

Figure 6: $\dfrac{\text{Max. shear force}}{\frac{\Delta}{2}\ \frac{b}{W}\ C_w}$ Versus Δ, Beam Seas

Figure 7: $\dfrac{\text{Max. torsion mom.}}{T_1}$ Versus Δ, Oblique Seas

Where $T_1 = \rho C_b g\ B \times 0.6\sqrt{\lambda_T}\ \ L^2/2\pi$

Figure 8: $\dfrac{\text{Max. torsion mom.}}{T_1}$ Versus ΔL, Oblique Seas

Each figure includes data from all the tests in three sea states. The symbols used in the plots for the various tests are as follows:

—+—+—+—	Thornton Series
—⊖—⊖—⊖—	ASR Series
×	Univ. of Miami Catamaran Series
⊡	Undisclosed Series
⊗	Mohole Platform
•	Levingston Platform

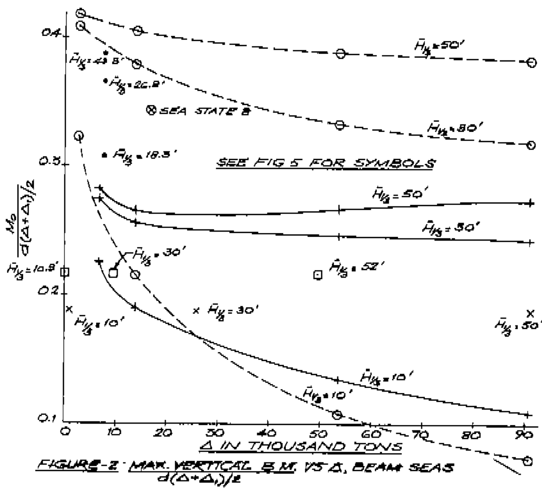


Fig. 2 - $\frac{\text{Max. Vertical B.M.}}{d(\Delta+\Delta_1)/2}$ vs. Δ, Beam Seas

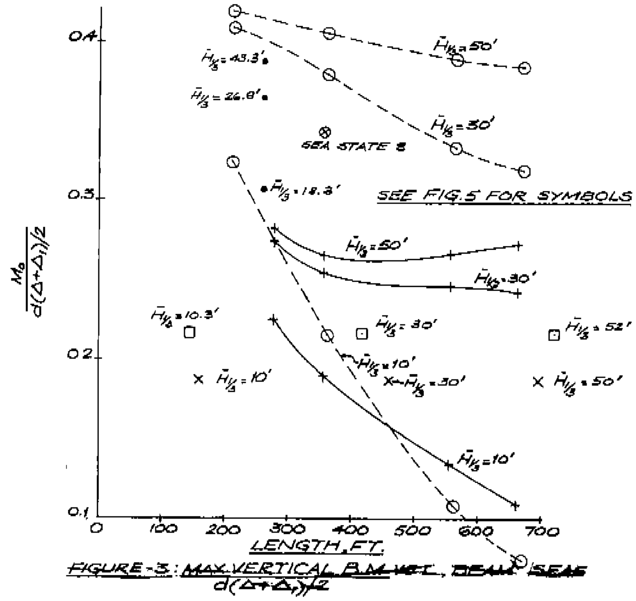


Fig. 3 - $\frac{\text{Max. Vertical B.M.}}{d(\Delta+\Delta_1)/2}$ vs. L, Beam Seas

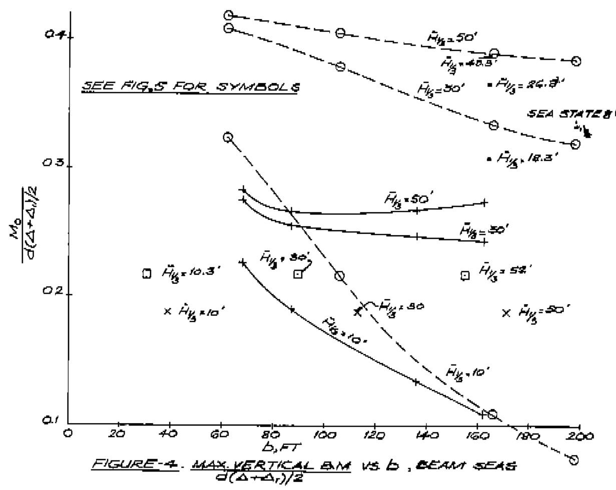


Fig. 4 - $\frac{\text{Max. Vertical B.M.}}{d(\Delta+\Delta_1)/2}$ vs. b, Beam Seas

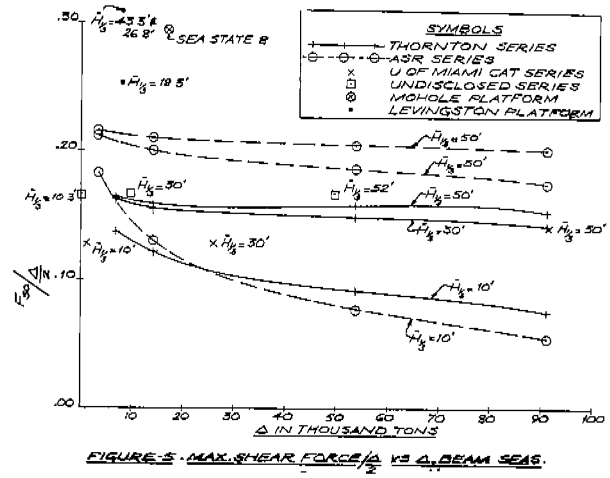


Fig. 5 - Max Shear Force/$\frac{\Delta}{2}$ vs. Δ, Beam Seas

The plots are for loads in terms of maximum single amplitudes where maximum is taken as follows:

Thornton and ASR:	Average of the 1/1000 highest calculated for the Pierson Moskowitz spectrum.
All Other Tests:	Maximum measured or average 1/1000 highest (obtained from significant or 1/10 highest average values), whichever is greater.

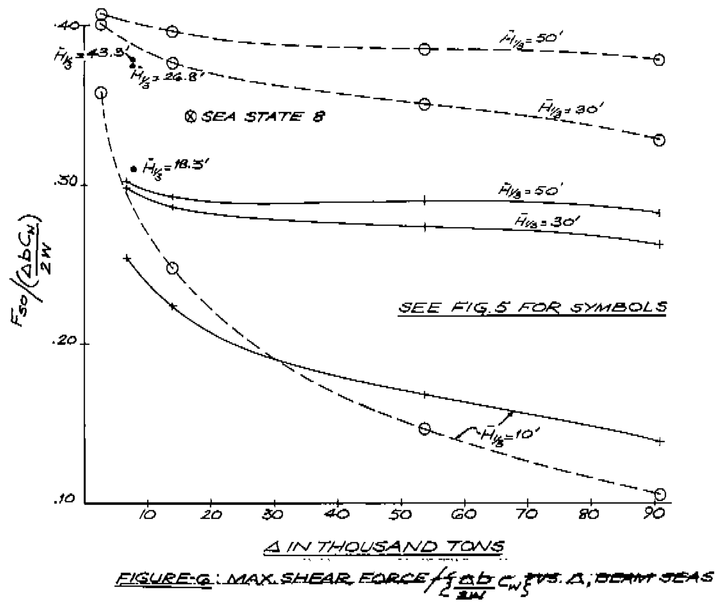


Fig. 6 - $\frac{\text{Max. Shear Force}}{\{\frac{\Delta b}{2W} C_W\}}$ vs. Δ, Beam Seas

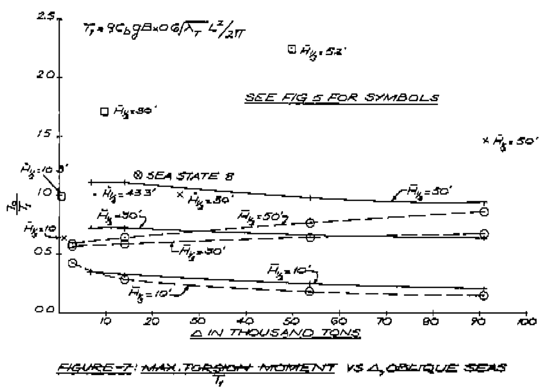


Fig. 7 - $\frac{\text{Max. Torsion Moment}}{T_1}$ vs. Δ, Oblique Seas

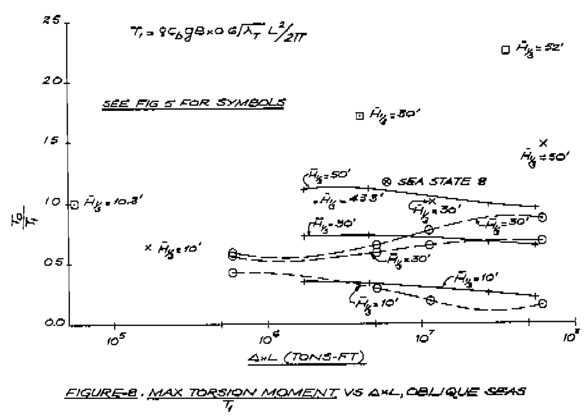


Fig. 8 - $\frac{\text{Max. Torsion Moment}}{T_1}$ vs. ΔxL, Oblique Seas

The reported phase relationships between cross-structure loads for Levingston 6-column platform (in towing condition) in both beam seas as well as oblique seas are as follows:

- Maximum shear 90° out of phase with bending moment
- Maximum side force in phase with bending moment
- Maximum yaw moment 90° out of phase with bending moment
- Maximum torsion moment 180° out of phase with bending moment

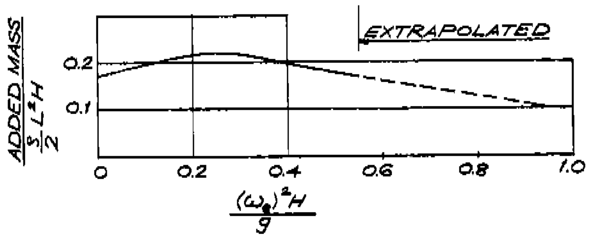


Fig. 9 - Added Mass For Sway Direction, Series 60 (Ref. 23)

Table 6 gives the ratios of maximum magnitude of each load in beam seas and oblique seas for the Thornton, ASR and the Levingston Platform. They were valuable in deducing the load schedule, Table 10.

Table 6 - Ratios of Maximum Loads in Beam Seas and Oblique Seas

	Thornton	ASR	Levingston	Mean
Bending Moment, $\frac{\text{Oblique Seas}}{\text{Beam Seas}}$	0.54	0.36	0.55	0.48
Shear, $\frac{\text{Oblique Seas}}{\text{Beam Seas}}$	0.52	0.55	0.53	0.53
Torsion Moment, $\frac{\text{Beam Seas}}{\text{Oblique Seas}}$	0.55	0.36	0.55	0.49

It was realized that added mass (Δ 1/g) was an important term contributing to the vertical bending moments. However, the scope of the project would not permit detailed added mass calculation for each test vessel. Also, a literature search for reference material on the added mass in sway of unsymmetrical vessel was futile. In view of this it was decided to calculate the added mass based on Series 60 coefficients provided by Eda and Crane (23) and reproduced in Figure 9 here.

4.3 Discussion of the Plots

4.3.1 Vertical Bending Moments

a. The first observation that can be made of the plots is that the ASR series coefficients are consistently higher than the Thornton series, and that the differences are large enough not to be attributed to experimental inaccuracies alone.

b. The plots of coefficient against length and centerline hull spacing in addition to displacement do not help to explain the data distribution.

c. It is recognized that in plotting ASR test data together with the other test data, it is assumed that shear is 90° out of phase with the bending moment implying that the contribution of secondary moments to the total maximum bending moment is zero. This assumption is in accordance with the Levingston Platform tests results. It should be pointed out that the Levingston Platform hulls are much more widely spaced than the ASR hulls (see Table 2), and that this assumption may be inaccurate for the ASR. Further, that the inaccuracy of this assumption may be one of the reasons why the ASR bending moment coefficient is much higher than for other ships.

d. The data is too insufficient to deduce the influence of form coefficients on the difference noted above or the general trends. For the same reason it is not possible to develop a better representation of size than just displacement to the first power.

e. The bending moments are non-linear with respect to significant wave height. Also that the non-linearity increases with decrease in vessel size. There is a plausible explanation for this trend. Maximum bending moments are experienced in waves with $\lambda = 1.8$ to $2.0W$ (2 W range from 100 feet to 548 feet for vessels represented on the plot). Now, as the sea state rises, the maximum spectral energy shifts to longer waves and wave height for waves with $\lambda \approx 1.8$ to $2.0W$ does not increase proportionally resulting in the non-linear load response.

4.3.2 Shear Force

Discussions of items (a), (d), and (e) under Vertical Bending Moments apply to shear force also.

The purpose of using both $\frac{\Delta}{2}$ and $\frac{\Delta}{2}\,\frac{bC_W}{W}$ to nondimensionalize force was in the hope of explaining the reason for the high values of MAX Fso/ $\Delta/2$ for the semisubmersible platforms. The apparent differences between the platform and the other vessels which could particularly influence the shear force are their very wide hull centerline spacing, b, and high waterplane coefficients, C_W. It is realized that the introduction of C_W tends to increase the differences in the ASR and Thornton coefficients in the higher sea states.

4.3.3 Torsion Moment

a. The Thornton series torsion moment coefficients are higher than the ASR series, whereas, in the case of the vertical bending moment and shear force, the opposite is true.

b. Just as the vertical bending moment and shear, the torsion moments are nonlinear with respect to significant wave height, but not to the same degree.

c. No apparent explanation is available as to why the data point representing the University of Miami design and the Undisclosed design are much higher than the other ships, although they are both conventional catamarans similar to the Thornton and ASR.

d. At the upper end of the Δ and ΔL scale the correlation between the ASR and Thornton series is good. Further, in sea state 8 the torsion moment coefficient approaches unity implying that the expression used to nondimensionalize the moment is most promising to estimate maximum torsion moments.

5. CONDITION FOR MAXIMUM RESPONSE AND RECOMMENDED METHOD FOR DESIGN LOADS ESTIMATE

The purpose of this section is (i), to determine the probable wave and ship position in which the maximum catamaran motions and cross-structure loads are caused, (ii) develop simple load equations and (iii) suggest a design load schedule. It is intended to concentrate on the beam sea condition in items (i) and (ii) since it is proposed to use the torsion equation in nearly the same form as developed by Dinsenbacher (Appendix 3).

5.1 Condition for Maximum Response in Beam Seas

Figure 10 depicts a catamaran poised in several locations in three different waves. In Figure 10-I, the wave length equals b, the centerline hull spacing; in Figure 10-II, the wave length equals 2b, and in Figure 10-III, the wave length is supposed to be several times bigger than b.

By inspection it can be seen that when $\lambda \approx b$, the wave-induced forces (hydrostatic, inertial and damping) on both the hulls have the same direction and magnitude. Since the loads on the cross-structure are due to the differential loading on the two hulls (besides the loads due to the mass of the cross-structure), in this particular condition the cross-structure loading should be small. Intuitively, the heave magnification should be high and roll magnification small.

When wave length is much bigger than the catamaran width, as in Figure 10-III, the differential loading on the hulls should be small and consequently the cross-structure loading should be small. Also, the roll and heave magnification should be roughly unity.

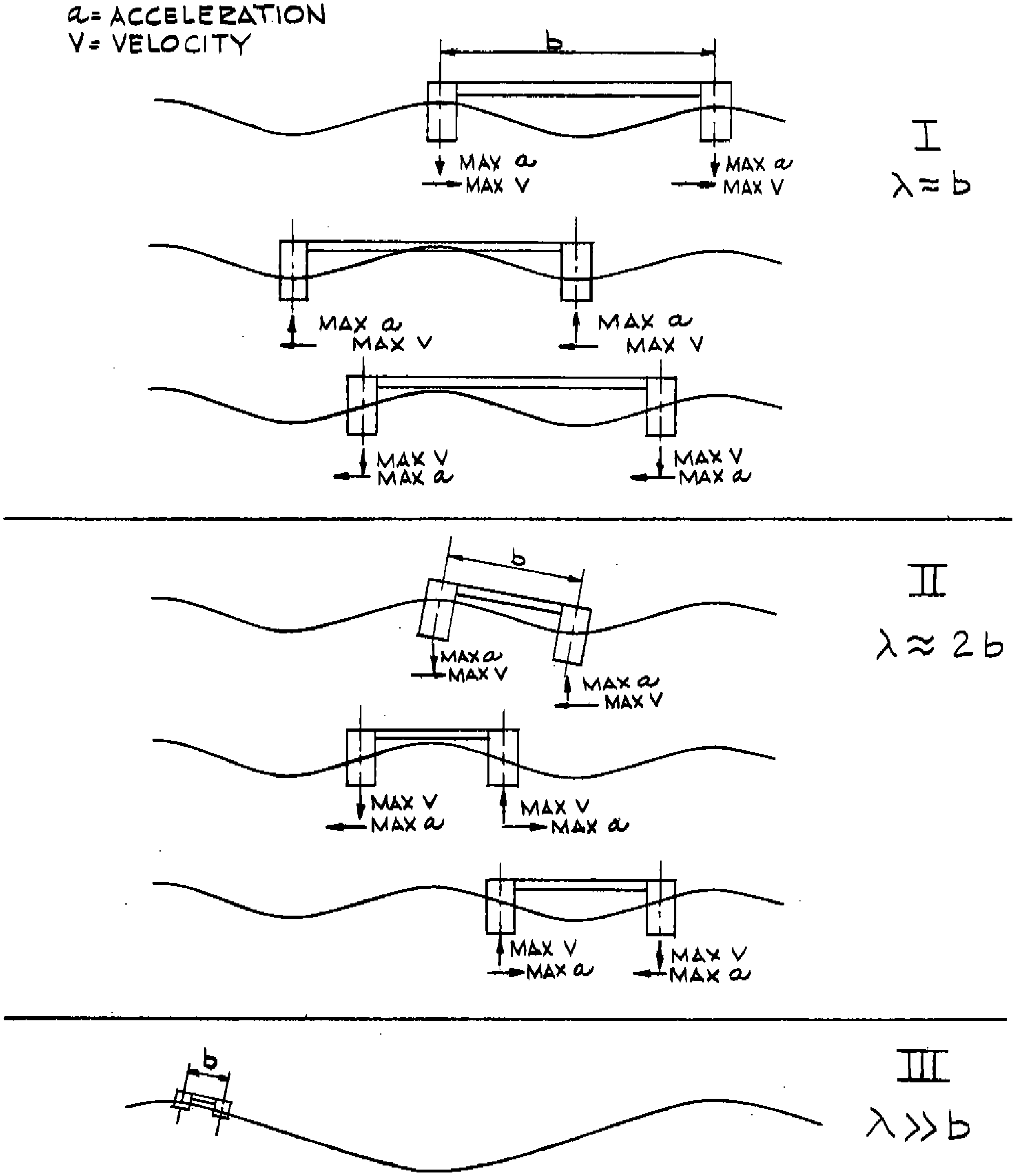


Fig. 10 - Catamaran in Beam Waves of Different Length

Waves of $\lambda \approx 2b$, Figure 10-11, have the potential for generating conditions for high differential loading on the hulls. When one hull is on the crest and the other in the trough they experience maximum vertical acceleration of opposite sense generating high shear force on the cross-structure and at the same time inducing large catamaran roll. The velocity dependent (damping) force would induce bending moment, however, it is believed to be small. If the wave is considered to be of highest steepness possible, then the roll and shear should be maximum. When the hulls are at the nodes (with crest or trough on the catamaran centerline), they experience maximum equal and opposite side forces, both hydrostatic and hydrodynamic, which result in maximum moment on the cross-structure. Further, when the crest is on the centerline the moment at the juncture of the cross-structure and the hulls due to side forces have the same sense as the moment due to the weight of the cross-structure, whereas when the trough is on the centerline the particular two moments have opposite sense. Figure 10-11 makes another valuable suggestion; that a catamaran heave should be small when $\lambda \approx 2b$ because the vertical wave force on the hulls cancel one another.

In the foregoing paragraphs, tentative conclusions were reached as to the wave and ship locations in which maximum response are caused. Now the model test results will be inspected for the same purpose.

The principal clues from the model test results regarding the conditions for maximum response in beam seas are as follows:

i. There is general agreement among the different test results that maximum roll, shear force and vertical bending moment occur with vessel at zero forward speed in beam wave with $\lambda \approx 1.8$ width to 2.0 width.

ii. In both the Thornton and the Levingston Platform test, where heave was measured as well as other responses in a wide range of regular waves, it was found that heave approached zero in waves when shear, roll and bending moment were maximum.

iii. Phase data from the Levingston Platform test in beam seas is as follows:

Maximum shear 90° out of phase with bending moment
Maximum side force in phase with bending moment
Maximum yaw moment 90° out of phase with bending moment
Maximum torsion moment 180° out of phase with bending moment

This implies that maximum bending moments are caused by side forces and not by vertical forces since heave is minimum or zero in waves which cause maximum bending moment, and shear is 90° out of phase with maximum bending moment.

It can be stated that there is good agreement between the conclusions reached on the basis of the model test results and the visual inspection. This agreement provided the encouragement to set up simple equations for maximum vertical bending moment, axial force and shear force, whose presentation follow. Indeed, it is admitted that the test data available to reach the conclusions is limited.

5.2 Development of Design Load Equations

5.2.1 Equation for Estimating Maximum Transverse Vertical Moments and Axial Force (See Figure 11)

Assumptions:

- Wave is sinusoidal
- Wave length = twice hull centerline space, $\lambda = 2b$
- Wave height = $\lambda/10$

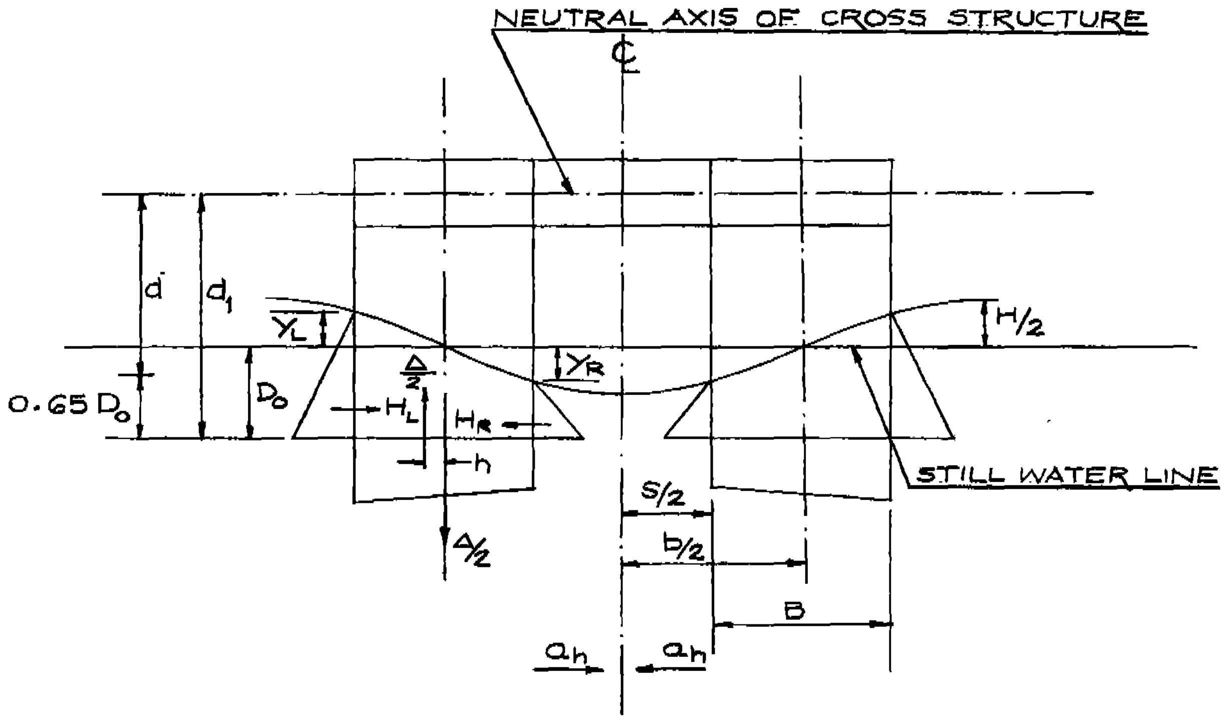


Fig. 11 - Loading Condition for Maximum Vertical Bending Moment in Beam Seas

- Trough at centerline of catamaran
- Vertical acceleration is 1g (displacement of one hull equals half weight of catamaran)
- Magnitude and distribution of side hydrostatic force per foot of length remain constant as at transverse section with maximum beam
- The aggregate magnitude of the horizontal acceleration causing the dynamic side force equals the intact wave acceleration at a point 1/4 beam off the centerline of each hull and 0.65 draft above keel
- Cross-structure weight is evenly distributed
- Cross-structure extends between inboard shell of hulls and the ends are built in.
- Velocity dependent forces and impact of water particles on the hulls are negligible.

Maximum Vertical Bending Moment:

M_o = Wave-induced bending moment for a weightless cross-structure, constant over the breadth of cross-structure

M_o = Side hydrostatic force moment - couple due to the horizontal shift in center of buoyancy + side inertia force moment

$$M_o = (H_L V_L - H_R V_R) - \frac{\Delta}{2} h + \left(\frac{\Delta + \Delta_L}{2g}\right) a_h d \quad \ldots\ldots\ldots \quad (E\,1)$$

$H_L = \rho g L \dfrac{(D_o + Y_L)^2}{2}$ = Side hydrostatic force on outboard shell

$V_L = d_1 - \dfrac{1}{3}(D_o + Y_L)$ = Centroid of H_L below neutral axis of cross-structure

$Y_L = \dfrac{H}{2}\cos\left(\pi \dfrac{S}{\lambda}\right)$ = Wave surface above still waterline at outboard shell

$H_R = \rho g L \dfrac{(D_o - Y_R)^2}{2}$ = Side hydrostatic force on inboard shell

$V_R = d_1 - \dfrac{1}{3}(D_o - Y_R)$ = Centroid of H_R below neutral axis of cross-structure

$$Y_R = \frac{H}{2} \cos\left(\pi \frac{S}{\lambda}\right) \quad = \text{Wave surface below still waterline at in-board shell}$$

$$h = \frac{B}{2} - \frac{B}{3} \left[\frac{2\,(D_o + Y_L) + (D_o - Y_R)}{(D_o + Y_L) + (D_o - Y_R)} \right]$$

$$h = \frac{B}{2} - \frac{B}{3} \left[\frac{3\,D_o + Y_L}{2\,D_o} \right] = \text{Horizontal shift in center of buoyancy}$$

$\frac{\Delta_1}{2g}$ = Added mass of one hull in horizontal direction

a_h = Aggregate horizontal acceleration

$d = d_1 - 0.65\,D_o$ = lever arm for inertia force

M_c = Moment at ends due to weight of cross-structure

$$= -\frac{W_c\,S}{12} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(E 2)}$$

M_1 = Maximum vertical bending moment at juncture of cross-structure and hull

$$M_1 = M_o + M_c \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(E 3)}$$

Maximum Side Force

P = Maximum axial compression

$$P = H_L - H_R + \frac{\Delta + \Delta_1}{2g} \quad a_h \quad \ldots\ldots\ldots \quad \text{(E 4)}$$

Due to the symmetry of the assumed wave and vessel, it is possible, by intuition, to set down the equations of moment and axial force for the condition of wave crest at centerline.

M_1 = Maximum vertical bending moment at the juncture of cross-structure and hull

$$M_1 = \left|(M_o)\right| + \left|(M_c)\right|$$

$$M_1 = \left|(H_L V_L - H_R V_R) - \frac{\Delta}{2}\,h + \left(\frac{\Delta}{2g} + \frac{\Delta_1}{2g}\right) \quad a_h d\right| + \frac{W_c\,S}{12}$$

$$P = \text{Maximum axial tension} = \left| H_L - H_R + \left(\frac{\Delta + \Delta_1}{2g}\right) \quad a_h \right|$$

It is important to note that absolute values are signified since symbols refer to figure II for trough at centerline.

It should be recognized that whether crest at centerline or trough at centerline will result in the higher direct stress will depend on the relative size of stress due to M_c and P. However, by rough checks, it was found that for existing catamarans stress due to axial force was greater than stress due to cross-structure weight (or local loads).

5.2.2 Equation for Estimating Maximum Shear Force

According to the analysis at the beginning of this section, maximum shear occurs, probably, when one hull is on the crest and the other in the trough. In this position the hulls experience maximum vertical acceleration in opposite direction to one another. Again, according to the analysis, maximum roll should occur at the same time as maximum shear.

Combination of vertical acceleration and roll will not permit an immediate writing of a shear force equation as it was possible in the case of vertical bending moment and axial force. It is proposed to resort to the model test data to obtain an expression for maximum shear. This is done simply by picking the highest nondimensionalized shear coefficient for a weightless cross-structure from Figure 6. Since the vertical wave-induced acceleration on the hulls are of opposite sense, the cross-structure can be assumed to have 1g acceleration only. Then,

$$F_{so} = 0.41 \frac{\Delta}{2} \frac{b}{w} C_w \quad \text{.....................} \quad (E5)$$

= Wave induced shear at ends, weightless cross structure

$$F_{sc} = \frac{W_c}{2} \quad \text{.....................} \quad (E6)$$

F_{sc} = Shear at ends due to cross-structure weight

F_{sl} = Maximum shear at juncture of cross-structure and hull

$$F_{sl} = F_{so} + F_{sc} \quad \text{.....................} \quad (E7)$$

5.2.3 Equation for Estimating Maximum Torsion Moment

Dinsenbacher's torsion moment equation which is also reproduced in Appendix 3 is

T_c = Torque about center of twist of cross-structure

= Torque about center of gravity of ship + torque due to shear acting through the ship's center of gravity

$$T_c = \left| \rho C_b\, g\, B\, 0.6 \sqrt{L_w}\, L^2 \;/2\pi \right| + \left| 0.14\, M q\, t/S \right|$$

Torsion values as provided by the first item, T_1, can be compared with the model test results, as was done in Figure 7, since t for model was zero except for the ASR model. Even for the ASR model, t was relatively small making the second term of secondary importance.

It can be seen from Figure 7 that if the constant 0.6 in T_1 was replaced by 0.7 then T_1 would provide torsion values at least as large as any test value in an irregular sea with 50-foot significant wave height if the data scatter due to the University of Miami model test and the undisclosed test is neglected. A 50-foot significant wave height represents sea state 8 and it is considered sufficiently severe for design purposes. It is pertinent to point out at this time that Dinsenbacher selected 0.6 to suit the ASR long term prediction of torsion moments. Even though the use of 0.7 may overestimate torsion, conservativeness is justified in light of the limited test data and the many simplifications that had to be made to derive the equation.

It is proposed to replace the second term in light of the objection raised to it in Section 3 of this report. According to the model test results, maximum shear and torsion are out of phase, and maximum shear in oblique seas is approximately 53 percent of maximum shear in beam seas. (This applies to a weightless cross-structure.) It is conjectured that it would be conservative to assume that shear in phase with torsion is half of maximum shear. Then, using the symbols of this report, the torsion equation would be

$$T_c = \left| \rho C_b g \; 0.7 \sqrt{\lambda_T} \; L^2/2\pi \right| + \left| (t)\; (0.53 \times 0.5 \times \text{max shear in beam seas}) \right|$$

$$T_c = \left| \rho C_b g \; 0.7 \sqrt{\lambda_T} \; L^2/2\pi \right| + \left| (t)\; 0.11 \frac{\Delta}{2} \frac{b}{w} C_w \right| \quad \text{......(E8)}$$

If t = Longitudinal distance from ship LCG to cross-structure twist center = 0

then

$$T = T_o = \left| \rho C_b g \quad 0.7 \sqrt{\lambda_T} \quad L^2/2\pi \right|$$

5.2.4 Comments on the Proposed Equations

- The equations are quasi-dynamic and semi-empirical in nature. They neglect velocity dependent forces as well as the impact of water particles on the hulls.

- Although any other assumption than that wave form remains intact as it passes the catamaran would be difficult to handle, in reality, it is seen that wave form does deform between the hulls. It is conjectured that the deformed wave would not cause higher acceleration dependent forces or larger hydrostatic loadings than a wave which remains intact.

- The new equations presented do not have any back-up derivation associated with them.

- The procedure for calculating side hydrostatic force is the same as used by Schade and Dinsenbacher (12) and (13).

- The use of $\lambda = 2b$ in beam sea condition is not quite in accordance with the model test results which suggest $\lambda \approx 1.8W$ to $2.0W$. The possible refinement is sacrificed to sustain symmetry and simplicity.

- The method does not account for unsymmetrical hulls and form of hulls.

- As far as it can be determined, there is no published information on the added mass in the horizontal direction for catamarans. Whether it is satisfactory to consider the added mass of each hull as if they were independent hulls is quite questionable since they can constrain one another's sway motion. This should be particularly true in waves with $\lambda \approx 2b$ where the horizontal acceleration of the two hulls have opposite sense. Unfortunately, model test results gathered do not have sway results to evaluate this. Until new information on added mass in sway at low frequencies (wave encounter frequencies) and for unsymmetrical hulls is forthcoming, estimates using Series 60 data, Figure 9, will have to suffice.

- It is suspected that for small catamarans the proposed method could very much overestimate the bending moment. The reason being that frequency of occurrence of the critical wave with $\lambda \approx 2W$ and $H \approx 2W/10$ is likely to be slim.

5.3 Comparison of Loads Calculated by Proposed Equations and by Other Method

Tables 7, 8 and 9 provide for the catamarans listed in Table 2, the vertical bending moment, shear, and torsion moment respectively, as calculated by the proposed equation and other methods. Other methods include model tests, Scott's method for bending moment and torsion, and Lankford's method for torsion moment due to grounding. All calculations are for catamarans with weightless cross-structure since model tests results are for weightless cross-structure.

As a matter of interest, shear and bending moment for the Thornton, ASR, and Levingston Platform were also calculated for a wave with $\lambda = 2b$ and $H = \lambda/10$ assuming load/wave height remains constant. The values of maximum load/wave height were obtained from the test reports.

5.4 Method for Design Loads Estimate

Table 10 presents a recommended design load schedule which is based on the equations developed in Section 5.2 and the ratios of the maximum load in the beam seas and the oblique seas as given in Table 6.

Table 7 - Wave-Induced Transverse Vertical Bending Moment in Beam Seas

Note: All values are single amplitudes in foot tons and for weight less cross-structure

	E.W. Thornton	E.W. Thornton Ship A	ASR	ASR Ship A	U. of Miami Ship A	Mohole Platform	Levingston Platform
(1) * Model Test Max. in Sea State 8	-	-	-	-	1,820,000	729,000	199,286
(2) ** Calc. 1/1000 Highest in Sea State 8	33,240	1,045,344	32,547	3,091,982	-	-	-
(3) SR 192 Method	50,438	1,626,323	40,518	4,195,760	4,325,545	-	220,764
(4) $S\Delta/4$ (Scott's Method)	51,925	-	26,572	-	1,942,878	756,000	246,400
(5) $\sqrt{RAO}\,(2b/10)$ 1/2	54,040	-	55,051	-	-	-	200,607
(6) (1)/(3) or (2)/(3)	0.659	0.643	0.803	0.737	0.421	-	0.903
(7) (1)/(4) or (2)/(4)	0.640	-	1.255	-	0.936	0.964	0.806
(8) (3)/(5)	0.933	-	0.736	-	-	-	1.100
(9) *** Long Term Prediction of Maximum	-	-	63,300	-	-	-	-

* Max. or 1/1000 highest, whichever is greater
** RAO from model tests and sea state described by Pierson-Moskowitz Spectrum
*** From Reference (8)

Note: All values are single amplitudes in foot tons and for weightless cross-structure

Table 8 - Wave-Induced Shear in Beam Seas

Note: All values are single amplitude in foot tons and for weight less cross-structure

	E.W. Thornton	E.W. Thornton Ship A	ASR	ASR Ship A	U. of Miami Ship A	Mohole Platform	Levingston Platform
(1) * Model Test Max. in Sea State 8	-	-			6,450	2,480	1,190
(2) ** Calc. 1/1000 Highest in Sea State 8	551	6,964	302	9,134	-	-	-
(3) $0.41(\Delta/2)(b/W)(C_w)$ SR 192 Method	749	10,110	304	9,880	-	2,960	1,400
(4) $\sqrt{RAO}$ (2b/10) 1/2	605	-	349	-	-	-	890
(5) (1)/(3) or (2)/(3)	0.726	0.686	1.0	0.923	-	0.837	0.850

Note: All values are single amplitude in foot tons and for weightless cross-structure

* Max. or 1/1000 highest, whichever is greater
** RAO from model tests and sea state described by Pierson-Moskowitz Spectrum

Table 9 - Wave-Induced Torsion Moment in Oblique Seas

Note: All values are single amplitudes in foot tons and for weightless cross-structure

	E.W. Thornton	Thornton Ship A	ASR	ASR Ship A	U. of Miami Ship A	**** Mohole Platform	Levingston Platform
(1) * Model Test Max. in Sea State 8	-	-	-	-	1,625,000	193,452	93,304
(2) ** Calc. 1/1000 Highest in Sea State 8	58,545	1,044,577	9,536	809,401	-	-	-
(3) *** SR 192 Method	61,400	1,280,000	18,810	1,090,000	1,285,000	192,000	99,500
(4) $0.04\,L\Delta$ (Scott's Method)	68,340	2,206,878	23,495	2,433,077	2,526,971	238,560	80,080
(5) $0.175\,L\Delta$ (Grounding)	298,988	9,655,082	102,790	10,644,711	11,055,500	1,043,700	350,350
(6) (1)/(3) or (2)/(3)	0.95	0.82	0.51	0.74	1.26	1.01	0.94
(7) (1)/(4) or (2)/(4)	0.86	0.47	0.64	0.33	0.64	0.81	1.16

Note: All values are single amplitude in foot tons and for weightless cross-structure

* Max. or 1/1000 highest, whichever is greater
** RAO from model tests and sea state described by Pierson-Moskowitz Spectrum
*** $\rho C_b\, g\, B \times 0.7\sqrt{\lambda_w}\; L^2/2\pi$
**** Assumed L = LBP = 355 Ft

Table 10 - Design Load Schedule

Load	Beam Waves	Oblique Seas
Loading for Direct Stress at Midspan of Cross-Structure		
Axial Force	P from (E4)	0.48 of P from (E4)
Moment, Weightless Cross-Structure	M_o from (E 1)	0.48 of M_o from (E 1)
Local Load (Cross-Structure Weight)	W_c	W_c
Loading for Direct Stress at Juncture of Cross-Structure and Hull		
Axial Force	P from (E4)	0.48 of P from (E4)
Moment, Weightless Cross-Structure	M_o from (E 1)	0.48 of M_o from (E 1)
Local Load (Cross-Structure Weight)	W_c	W_c
Torsion	0.49 of T_c from (E8)	T_c from (E8)
Loading for Shear at Juncture of Cross-Structure and Hull, Acting Concurrently with Moment		
Torsion	0.49 of T_c from (E8)	T_c from (E8)
Local Load	W_c	W_c
Loading for Shear at Juncture of Cross-Structure and Hull, Acting Out of Phase with Moment		
Shear	F_{so} from (E5)	0.53 of F_{so} from (E5)
Local Load	W_c	W_c

The method is considered satisfactory for conceptual designs.

It will be noted that the grounding and docking loads are not included in the schedule. In the opinion of the authors, grounding torsion loads are nearly impossible to estimate as they are so subjective to vessel speed, shape, size and strength of striking objects and water depth. As far as torsion loads due to docking are concerned it is suggested that individual designer consider oblique docking with most likely docking weight and realistic support points appropriate to his vessel.

6. HULL FLEXIBILITY AND CROSS-STRUCTURE STRESSES

It was apparent at the beginning of the project that in order to attempt the establishment of catamaran size limits it was necessary to select a suitable method for the preliminary structural analysis of the cross-structure of a large catamaran, once the critical loads were estimated.

Lankford's method, discussed in Section 3 and detailed in Appendix 2, was readily available. However, as mentioned previously, it appeared to have two major weaknesses. It assumes the hulls to be rigid and there is no relative rotation between the hulls and the cross-structure at their junction. Hence, it was deemed desirable to find a method which did not have these weaknesses and to try it out on a vessel for which structural calculations using Lankford's method were available.

The method of space frame analysis had an immediate attraction and it was decided to try it out on the T-AGOR 16 Oceanographic Research Catamaran for which structural calculations based on Lankford's method were available in-house. It must be mentioned at once that only the hull bending flexibility and shear deformation in the longitudinal direction were simulated in the mathematical model. The space frame analysis had the following advantages:

- Representation of structure partially by its flexibility is inherent to the method. It should provide, at least, indicative numerical values on the influence of hull flexibility and the relative rotation between the hulls and the cross-structure on the cross-structure, and the influence of the cross-structure on the individual hull structure in the transition area.

- The method is computerized which could be a great asset later in the project if structural analysis was necessary for several ships.

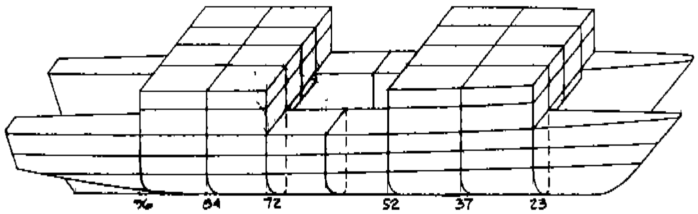
- It can assume several different types of loading at once and permits quick changes in the structural configuration.

- It can include maximum amount of structure effective in taking primary and secondary loads by employing progressively more detailed mathematical model.

- It can conveniently handle structure with more than one material, say steel and aluminum.

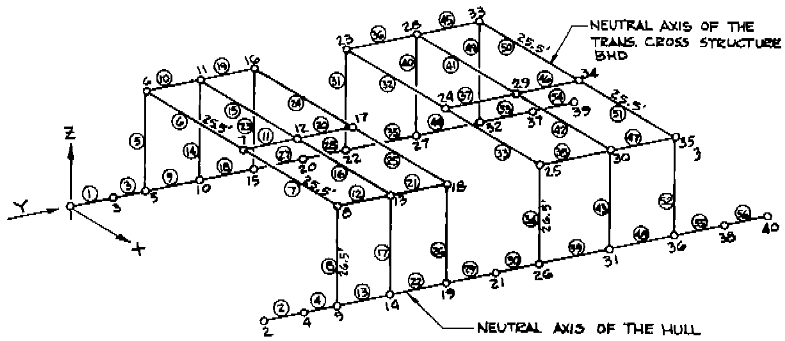
Figure 12 shows the bare outline of the T-AGOR structure and Figure 13 delineates its mathematical model incorporated in the space frame analysis which employed the IBM-1130 "Stress" program.

The analysis used the original T-AGOR 16 design loads. The loadings which controlled the primary members of the cross-structure were the grounding loads and the transverse vertical bending moments in beam seas. The former were obtained as suggested by Lankford and the latter were obtained from the ASR load estimates (with necessary modification to reflect different principal characteristics).



LENGTH BET. PERP.	22'-0"	HULL CLEAR SPACING	27'-0"
BEAM OVERALL	75'-0"	DEPTH TO MN. DK. AT SIDE	34'-0"
BEAM EA. HULL	24'-0"	DEPTH OF CROSS STRUCT. AT ℄	21'-6"
HULL ℄ SPACING	51'-0"	DRAFT	18'-6"

Fig. 12 - T-AGOR16 Structural Configuration



NOTE: CIRCLED FIGURES ARE MEMBER NUMBERS.
UNCIRCLED FIGURES ARE JOINT NUMBERS.

HULL LOADS INTRODUCED AT JOINTS ALONG THE NEUTRAL AXIS OF THE HULLS

Fig. 13 - Structural Model of T-AGOR16 For IBM - 1130 "Stress" Program

The resulting moments and shear forces in the beam sea condition and grounding condition for the cross-structure from the "Stress" program output are provided in Table 11. Other less critical conditions are omitted. The flexural stresses and shear stresses in the six cross-structure members based on the stress program output and those as calculated in the T-AGOR 16 Structural Design are also tabulated for comparison. Stresses in the structures other than the cross-structure are not tabulated, since the structural design for those members were based on American Bureau Rule and their stresses can not be calculated readily.

From the tabulation, the following conclusions can be drawn with respect to hull flexibility and cross-structure stresses:

- The flexural stresses calculated based on the structural model are in good agreement with those taken from the T-AGOR 16 structural analysis using Lankford's method.

- The shear stresses for grounding condition are in fair agreement. Those for beam condition show less agreement. Since the shear stresses are less critical than flexural stresses in beam sea condition, the discrepancy in shear stresses is not considered important.

- It appears, admittedly based on this limited check only, that the introduction of the longitudinal flexibility of the hulls has small influence on the stress in cross-structure, i.e., the simplification which assumes the hulls to be rigid would not affect the scantlings selection.

- Since the hulls can be assumed rigid the mathematical model can be greatly simplified. For a preliminary study, all the transverse cross-structure bulkheads can be assumed structurally similar, i.e., they all have the same section modulus, moment of inertia, shear area, etc.

In light of the last two conclusions, it may be stated that the preliminary analysis of a catamaran cross-structure can be conveniently handled with a conventional method, such as Lankford's method, with about the same accuracy in results, and about the same time requirement as the space frame analysis. Detail design analysis should consider, in addition to hull longitudinal flexibility, such structure response as the hull transverse and torsional deformation, cross-structure deformation in various directions and component structure (decks, bulkheads, etc.) deformation.

Table 11 - T-AGOR16 Catamaran Stress Summary

Bhd	Member	Section Modulus In^2 Ft	Shear Area In^2	Bend. Mom. Ft Kips	Shear Kips	Stress, Kips/In^2 Stress Program Flexural	Stress Program Shear	Design Calcs Flexural	Design Calcs Shear
				Beam Sea Condition					
96	6 & 7	650.0	105.0	16,900	74	24.6	0.7	23.6	3.3
84	15 & 16	833.3	102.0	19,244	67	23.1	0.7	23.9	3.1
72	24 & 25	632.9	82.5	19,749	68	31.2	0.8	24.1	2.8
52	32 & 33	784.6	94.9	24,951	55	31.8	0.5	26.5	2.8
37	41 & 42	833.0	102.0	21,901	74	26.3	0.7	23.9	3.1
23	50 & 51	853.0	120.0	20,471	70	24.0	0.7	26.5	3.2
				Grounding Condition					
96	6 & 7	650.0	105.0	13,619	952	21.0	9.1	21.4	10.5
84	15 & 16	833.3	102.0	9,216	506	11.0	5.0	11.5	7.7
72	24 & 25	632.9	82.5	4,187	118	6.6	1.4	8.1	4.6
52	32 & 33	784.6	94.9	2,535	24	3.2	0.3	4.2	3.6
37	41 & 42	833.0	102.0	10,999	554	13.2	5.4	10.8	7.6
23	50 & 51	853.0	120.0	15,780	1,114	18.5	9.3	8.9	10.8

7. DESIGN SHIP

7.1 Purpose

The analysis of the features that may impose catamaran size limits, Section 2, indicated that existing U.S. shipbuilding facilities could handle approximately 1000-foot catamarans on the premise that individual hulls would be built in a drydock and joined together afloat. Whether 1000-foot length should be proposed as a present probable upper limit was dependent on whether the necessary scantling size and the weight of the cross-structure were practical. Hence, once the available methods for cross-structure loads prediction and structural analysis were evaluated, the logical next step was to make a preliminary design of an approximately 1000-foot catamaran. Also, it is believed that in the course of the design, the inadequacies, if there be any, of the available structural design information would become apparent.

Table 12 - Design Ship Particulars

Hull Symmetry		Symmetrical
Length Bet. Perp., L		942'- 0"
Beam Overall, W		300'- 0"
Beam Each Hull, B		100'- 0"
Hull ℄ Space, b (corresponding to $\frac{b}{L} = 0.21$)		200'- 0"
Clear Hull Spacing, S		100'- 0"
Depth to Upper Deck at Side		106'- 0"
Depth of Cross-Structure		45' - 0"
Length of Cross-Structure		800'- 0"
Draft		31' - 0"
Cross-Structure Clearance from Waterline		30' - 0"
Displacement		90,800 Tons
Block Coefficient, C_b		0.54
Midship Coefficient, $C_{⊠}$		0.952
Prismatic Coefficient, C_p		0.572
Waterplane Coefficient, C_w		0.701
Service Speed (corresponding to $\frac{V}{\sqrt{gL}} = 0.24$)		25 Knots
Install Shaft Horsepower		150,000
Lightship Weight		52,687 Tons
Hull Structure	28,439	
Cross-Structure	5,598	
Electric Plant	1,150	
Propulsion	2,680	
Communication & Controls	280	
Auxiliary Systems	5,950	
Outfit & Furnishings	3,800	
Margin, 10%	4,790	
Deadweight		38,113 Tons
Container Capacity @ 11 Tons/Container		3,101 Containers
Container Capacity @ 15 Tons/Container		2,247 Containers
Container Capacity on Upper Deck, 8' x 8' x 20'		3,136 Containers

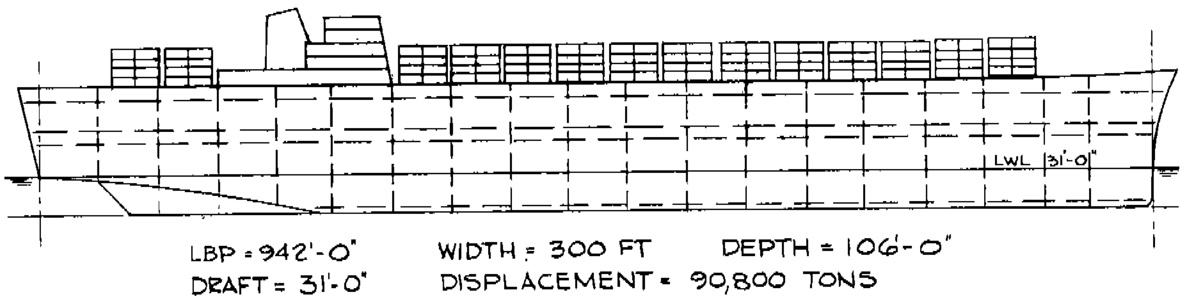


Fig. 14 - Design Ship Profile and Plan

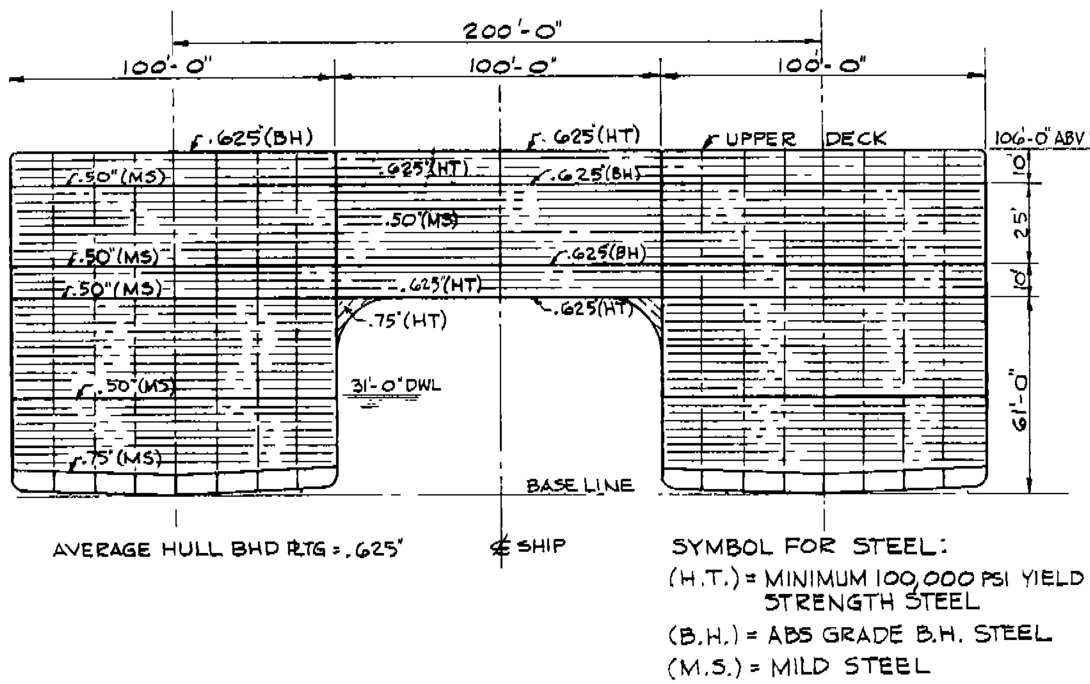


Fig. 15 - Design Ship - Typical Bulkhead Structure

7.2 Design Description

The preliminary design presented here is not optimized (or recycled) by far. The readers can expect no more defense from the authors for the design other than for its suitability to provide the limited information desired. The selected shape coefficient, bulkhead and deck arrangements, assumed framing system, etc., can all absorb considerable improvement.

It is assumed that the vessel would be a container ship since it is well accepted that if large catamarans are at all found superior to monohulls it would be as high-speed, payload carriers. Table 12 lists the design particulars and Figure 14 shows the profile and plan views. A rough set of lines were made to obtain hydrostatic properties and various plating areas.

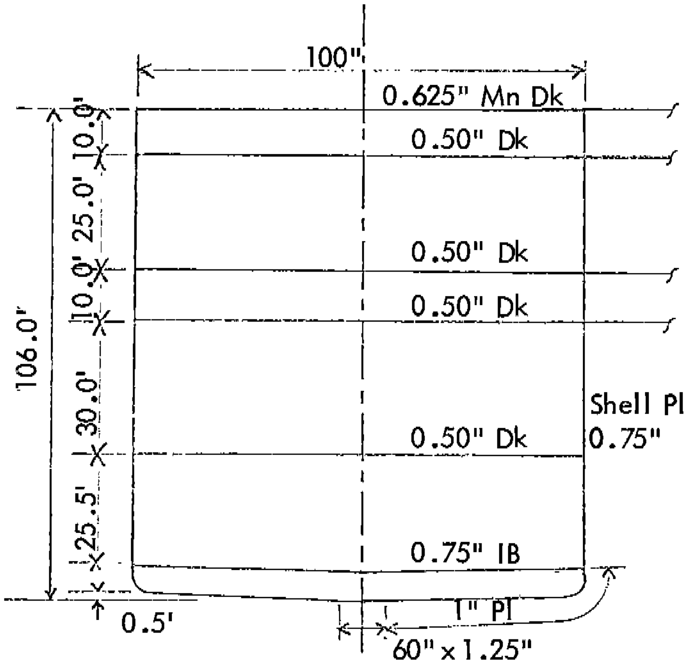
It will be observed that the design's Froude number of 0.24 and the hull centerline spacing to ship length ratio of 0.21 do not correspond to the values of 0.3 to 0.4 and 0.3 respectively, suggested for good resistance characteristics (see Appendix 1). To design for Froude number of 0.35 would require a speed of 36 knots. It was felt that a 36-knot speed would render the design uneconomical. To design for hull centerline spacing to ship length ratio of 0.3 would require hull centerline spacing of 314 feet which was considered impractical.

The 100' x 800' cross-structure is composed of four structural decks, including the upper deck and the bottom, and seventeen identical full structural transverse bulkheads spaced at 50 feet. The cross-structure is assumed to be fixed at the inboard shell of the hulls. In order to validate this assumption, four of the decks and seventeen of the full transverse bulkheads in the hulls are aligned with decks and bulkheads of the cross-structure. Figure 15 depicts the catamaran structure at a bulkhead.

Figure 16 provides the information on the section moduli of the individual hulls and the cross-structure. It includes sketches of the assumed effective structures, calculated section moduli and the required section modulus for the hulls based on the American Bureau of Shipping rules. It will be noted that the minimum permissible scantlings result in a section modulus considerably in excess of that required. This is due to the increased depth as compared to a monohull to have sufficient cross-structure clearance above the waterline.

7.3 Explanation for Effective Structure

Explanation is warranted for the structure assumed effective in cross-structure bending. On the face of it an immediate question may come to the mind of the reader; why should all the deck plating be considered effective in bending just as in the conventional longitudinal strength calculation, rather than just 24-foot breadth with each bulkhead. The structural analysis (as distinct from the design load estimate) was performed following Lankford's method, Appendix 2. In Lankford's method, all the principle loads, vertical bending moment, axial force, shear as well as the torsion moment, are absorbed by the bulkheads together with effective deck plating acting as fixed-end beams between the two hulls. The effective deck plating breadth of 24 feet was calculated by reference



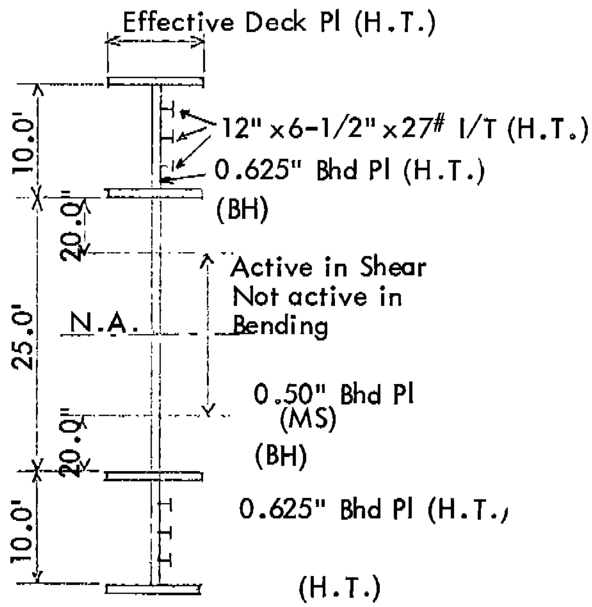
Main Hulls

Main Hull

Section Modulus Required by ABS
= 198,500 In^2 Ft per Hull

Section Modulus, Including 10% for Longitudinals:

Deck: 269,000 In^2 Ft
Bottom: 266,090 In^2 Ft



Cross-Structure

Cross Structure

	Effective Plating Four Decks	
	5/8" x 24'	1-1/4" x 10'
Section Modulus In^3	148,800	130,056
Shear Area, In^2	300	300
Axial Load Area In^2	923	803

Symbol for Steel:
(H.T.) = Minimum 100,000 psi yield Strength Steel
(B.H.) = ABS Grade B.H. Steel
(M.S.) = Mild Steel

Fig. 16 - Design Ship Section Moduli

to the well-known paper (24) on the subject by Professor Schade. The bulkheads were considered as multiple webs for each deck with length of 100 feet and plating width of 50 feet between webs. Among the various combinations of load and end fixity considered by Professor Schade, two were applicable to the structure in question, viz: - equal moment at both ends or uniform load and fixed ends. Even though the structure's major loading is due to equal moment at both ends, the latter combination was used as it gave the smaller effective breadth. Using the same reference, Professor Adams proved (see discussion to Professor Schade's paper) that for a monohull with no centerline bulkhead and with side shell as double webs, the effective breadth of the deck plating is 97%.

The 20-inch effective web plating at top and bottom of the center web, Figure 16, was reached by taking one-sixth of the length of the outer webs. The reasoning for this is provided by Lankford, Appendix 2.

Figure 16 also shows that if arbitrarily a 10-foot deck plating width were to be considered effective, 1-1/4 inch plate thickness would be necessary to provide approximately the same section modulus as available with 5/8-inch x 24-foot effective plate; 10 feet should be quite conservative.

Although 24-foot effective breadth was arrived at with probably adequate interpretation of the structure, it is acknowledged that the structure in question is really integrated box structures. Further, that there is insufficient test data on box girders to derive effective structure directly in lieu of the method employed.

7.4 Cross-Structure Loads and Stresses

The wave-induced design loads as deduced from the method (labeled SR-192) proposed in Section 5 of this report, by Dinsenbacher's (13) method and from the Thornton and ASR Series, Section 4, are summarized in Table 13. The stresses which are summarized in Table 14 were calculated by using maximum loadings predicted by SR-192 equations. The stresses are within the allowable stresses for 100,000 psi yield strength steel.

Although grounding is not considered a design criteria, stresses were also calculated for the grounding condition and are included in Table 14. If grounding was to be considered as a design criteria the selected scantling would be quite inadequate as the shear stress is 47,280 lb/in^2.

7.5 Design Conclusions

a. Direct stresses are higher in beam seas than in oblique seas. Shear stress is higher in oblique seas than in beam seas.

   Required largest deck plating thickness is very much dependent on the assumptions related to value of effective plating.

b. The required largest scantling of approximately 1-1/4 inch, based on a hopefully conservative assumption and steel yield strength of 100,000 psi, are common to shipbuilding today. Of course, this is true only for the particular structural configuration employed.

c. If grounding was to be considered a design criteria the assumed structure would be quite inadequate.

d. The imperative need to sustain the continuity of structural members (17 bulkheads and four decks) of the cross-structure into the main hulls causes the main hulls structural configuration to be uneconomical, e.g., unlikely that a 1000-foot monohull would require 50-foot main bulkhead spacing.

Table 13 - Design Ship, Wave-Induced Cross-Structure Loads

	Method			
	SR-192	Dinsenbacher	Thornton Series	ASR Series
BEAM SEAS: MAXIMUM TRANSVERSE VERTICAL BENDING MOMENTS Single Amplitude in Foot Tons				
Weightless Cross-Structure, Constant	*3,061,106	1,658,464	2,048,785	2,869,346
**With Cross-Structure Weight,	*3,427,439 At ends of Cross-Structure	3,966,364 At Midspan	-	-
BEAM SEAS: MAXIMUM SHEAR AT ENDS Single Amplitude in Tons				
Weightless Cross-Structure	8,666	5,636	5,978	8,032
With Cross-Structure Weight	*30,646	36,411	-	-
BEAM SEAS: MAXIMUM AXIAL FORCE Single Amplitude in Tons				
	*52,367	33,074	-	-
OBLIQUE SEAS: MAXIMUM TORSION MOMENTS Single Amplitude in Foot Tons				
	*2,948,449	***2,527,242	2,403,794	2,188,600

* Used for Structural Analysis
** Assumed Cross-Structure Weight = Steel + Ship's Deadweight
*** Assumed LCG of Ship Coincides with Longitudinal Location of Cross-Structure Twist Center

8. TOPICS FOR FUTURE RESEARCH AND DEVELOPMENT PROGRAM

Researchers (1), (2), (4), (8) and (13), who have appraised catamaran technology have generally reached very similar conclusions as to the deficiencies in the technology and the topics for the desirable future research and development program. A significant conclusion of this project is that a safe large catamaran structure can be designed now by conducting model tests, using existing design information and generally adopting a conservative approach. However, by nature of the design method the resulting structure would be unduly heavy. Also, such an approach would be unacceptable if a large num-

Table 14 - Design Ship, Cross-Structure Stress Summary

Section Modulus	=	148,800 In^3	(See Figure 16)
Shear Area	=	300 In^2	
Axial Load Area	=	923 In^2	

BEAM SEA CONDITION (Trough at Centerline)

Total Loading on Cross Structure:	
Vertical Bending Moment Without Cross-Structure Weight	3,061,106 Ft Tons
Torsion Moment = 0.53 x Max. in Oblique Seas	1,562,000 Tons
Local Load (Cross-Structure Weight)	43,960 Tons
Axial Load	52,367 Tons
Stress on End Bulkheads:	
Primary Bending	32,527 Lb/In^2
Bending due to Shear due to Torsion	5,535 Lb/In^2
Bending due to Local Load	- 3,893 Lb/In^2
Subtotal	34,169 Lb/In^2
Axial Compression	7,473 Lb/In^2
Total Stress	41,642 Lb/In^2
Shear Acting Concurrently with Bending and Torsion:	
Shear due to Torsion	1,371 Kips
Shear due to Local Load	2,897 Kips
Total Shear	4,268 Kips
Shear Stress	14,230 Lb/In^2
Shear Out of Phase with Bending and Torsion	
Shear	4,039 Kips
Shear Stress	13,463 Lb/In^2

Table 14 - Design Ship, Cross-Structure Stress Summary, (Cont'd)

OBLIQUE SEA CONDITION

Total Loading on Cross-Structure:	
Vertical Bending Moment Without Cross-Structure Weight 0.48 x Max. in Beam Seas	1,469,331 Ft Tons
Torsion Moment	2,948,449 Ft Tons
Local Load (Cross-Structure Weight)	43,960 Tons
Axial Load, 0.48 x Max. in Beam Seas	25,136 Tons
Stress on End Bulkheads:	
Primary Bending	15,613 Lb/In^2
Bending due to Shear due to Torsion	10,430 Lb/In^2
Bending due to Local Loads	3,893 Lb/In^2
Subtotal	29,936 Lb/In^2
Axial Tension	- 3,587 Lb/In^2
Total Stress	26,349 Lb/In^2
Shear Acting Concurrently with Bending and Torsion:	
Shear due to Torsion	2,587 Kips
Shear due to Local Load	2,897 Kips
Total Shear	5,484 Kips
Shear Stress	18,280 Kips

GROUNDING CONDITIONS

For Reference Only - Not Used as a Design Criteria

Total Loading on Cross Structure:	
Torsion Moment	12,850,000 Ft Tons
Local Load	43,970 Tons
Stress on End Bulkheads:	
Bending due to Shear due to Torsion	45,500 Lb/In^2
Bending due to Local Load	3,893 Lb/In^2
Total Stress	49,393 Lb/In^2
Shear due to Torsion	11,288 Kips
Shear due to Local Load	2,897 Kips
Total Shear	14,185 Kips
Shear Stress	47,280 Lb/In^2

ber of vessels were contemplated. In view of this conclusion, a following comprehensive list of study topics is prepared to close the major gaps in catamaran technology, and ensure the availability systematic design information to develop catamaran structure which would tend towards the optimum.

a. The nature, magnitude, location and frequency of hydrodynamic loads on the hulls; the distribution of loads in the cross-structure or the centerbody; magnitude and location of local wave impacts on the centerbody and the hulls. These will require theoretical and experimental programs.

   Model Test Program:

   - Series tests which would include symmetrical and unsymmetrical hull forms; range of hull spacing; variations in vertical location; longitudinal extent and longitudinal locating of centerbody.

   - Series suitable for ships from 100 feet to 1000 feet.

   - Model test methods which can simulate the centerbody, at least its weight and weight distribution.

b. Full scale centerbody load measurement program. (Necessary to develop acceptable measurement technique and data analysis once the data is gathered.) It would be prudent to select a catamaran whose cross-structure is relatively simple and amenable to clean analysis.

c. Dynamics of structural response in various vibratory modes.

d. Hull form and spacing for minimum resistance and ship motions in a seaway. Hull form, particularly unsymmetrical, for multi-screw installation.

e. Added mass and mass moment of inertia for the horizontal motion of symmetrical and unsymmetrical bodies at wave encounter frequencies (individually and as catamarans). Added mass and mass moment of inertia for the vertical motion of unsymmetrical bodies.

f. Damaged stability and compartmentation requirements.

g. Construction techniques to minimize need for new facilities (shipyard responsibilities). Drydocking facilities.

h. Contribution by cross-structure to the longitudinal strength of the vessel.

i. Behavior of box girders under combined bending, torsion and shear loads.

j. Stress concentration at the hull and cross-structure juncture. Nature and extent of necessary reinforcement and structural details.

9. CONCLUSIONS

1. The major constraints to catamaran size will be imposed by economics, individual shipyard construction capabilities, drydock facilities and pier facilities.

   Existing United States yard facilities can handle individual hulls of approximately 1050 ft x 140 ft. The hulls and the centerbody would have to be joined with hulls afloat; 35-foot draft is acceptable in most major harbors. New drydocking facilities and modified or new pier facilities will be essential. Discharge of cargo in the streams could remove the pier problem.

2. Existing design information for the estimation of loads on the cross-structure is just adequate to provide guidance to make preliminary prediction of loads on large catamarans.

   With respect to scantlings, a 1000-foot long catamaran with 100-foot beam hulls, 100-foot hull spacing and 31-foot draft is feasible. This does not imply that the structural configuration will necessarily be attractive.

3. The available model test data for predicting cross-structure loads are not sufficient and the existing analytical methods are not adequately developed to provide great confidence in either.

4. Model tests to date have been performed for specific designs only and have had the drawback of not simulating the centerbody.

5. Additional research and development work including systematic model test programs are necessary for the establishment of reliable design methods for optimum catamaran structure.

ACKNOWLEDGEMENTS

To an appreciable degree, the accomplishment of the project is due to the availability of unpublished model test data belonging to private companies. In this respect, special acknowledgement is due to the Reading & Bates Offshore Drilling Company for their permission to use the complete model test data on the E.W. THORNTON, and to Friede and Goldman Inc. through whom the tests were contracted. Friede and Goldman Inc. are also to be thanked for permission to use the model test data on their catamaran design for the University of Miami. Thanks are due to the Levingston Shipbuilding Company for providing the model test data for their drilling platform design. Mr. John L. Glaeser supplied a copy of his senior thesis which was appreciated.

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Mr. Walter H. Michel and Dr. Haruzo Eda willingly contributed with informal discussions of some aspects of the project for which the authors are grateful.

Appendix 2 is quoted from "The Structural Design of the ASR Catamaran Cross-Structure" by Benjamin W. Lankford, Jr., published in August 1967, Naval Engineers Journal, pages 625-635, by permission of the American Society of Naval Engineers.

Appendix 3 is quoted from "A Method for Estimating Loads on Catamaran Cross Structure" by Alfred L. Dinsenbacher published in October 1970 Marine Technology, Vol. 7, No. 4, pages 477-489, by permission of The Society of Naval Architects and Marine Engineers.

Last, but not least, acknowledgements are due to all the members of the Advisory Group II, Ship Structural Design, Ship Research Committee who provided enthusiastic and practical guidance to the project.

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APPENDIX 1

CATAMARAN RESISTANCE

Of all the aspects of catamaran design, resistance has received the most attention. Considerable work has been done, both in the areas of theoretical prediction and model test measurements, as well as their correlation. References listed at the end of this appendix represent valuable published information on the subject.

The main reason for the interest in resistance is because it has been shown that under certain conditions, net resistance of catamarans can be made smaller than the total resistance of the two hulls considered singly.

According to general practice, it is assumed here that resistance can be separated into two independent components, namely, frictional or viscous and wave-making. Contributions to resistance by other phenomena, including the influence of wave-making on viscous resistance, are relatively small and are omitted in this discussion.

Frictional resistance is a function of the wetted surface, degree of surface roughness and speed, and, for the catamarans, it is equal to the sum of the frictional resistance of the individual hulls.

Catamaran calm water wave-making resistance is a function of the Froude number ($V/\sqrt{gL}$), hull form and hull spacing. Eggers (references 1 and 2) has demonstrated theoretically that the wave-making effects between the hulls can interfere favorably to reduce the catamaran wave drag to below the level appropriate to the two hulls running in isolation. This is possible where given frequency components of the combined wave pattern are out of phase by approximately 180°.

There is general agreement between theory and model test data that the beneficial interference can occur in the Froude number range of approximately $0.3 < V/\sqrt{gL} < 0.4$, irrespective of hull separation. Beneficial hull separation in terms of the center to center spacing as a ratio of the ship's length appears to be in the order of 0.3. Further, optimum spacing varies with speed. Of course, what may be beneficial for resistance may not be compatible with the rest of the design.

The purpose of the foregoing discussion is to show that beneficial wave-pattern interference effects are obtained in a narrow range of Froude number and hull separation. From a practical design viewpoint the net resistance benefit has to be appreciable to constrain the design within the above narrow range of Froude number and hull separation.

The conclusion reached on the basis of data available to date is that no more than 15% net reduction in total resistance of large catamarans should be expected in ideal conditions when compared to the total resistance of two hulls running independently. At the same time, the increase is not expected to be more than 15%.

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APPENDIX 2

This is a reproduction of reference (8), "The Structural Design of the ASR Catamaran Cross-Structure" by Benjamin W. Lankford, Jr.," excluding the first part which is devoted to the description of the statistical methods used to predict the response of this ship's hull to sea condition beyond the capabilities of model tests.

SHEAR FORCES

The foregoing discussion has only described the bending moment resulting from a beam sea condition. There is, however, a slight shear force in the beam sea condition. The shear force is of a higher value in the quarter sea heading, but the associated moment results in a negligible design value. The shear force used is approximately 600 tons. Shear becomes more of a design problem from loads of other sources which will be described in another paragraph.

Shear or any other design response can be determined in the same way as the moment in the foregoing discussion. All the designer needs to do is to use the proper response amplitude operators from the model test.

DISTRIBUTION OF THE DESIGN SEA LOADS

The final bending moment predicted of 63,300 foot tons represents the total moment on one side of the ship. Since this moment is independent of any bending caused by the weight of the structure, a dead load bending moment must be added to this moment. The dead load moment for the ASR was calculated assuming this ship in still water since the effect of any sea waves has already been determined. The total maximum moment including the dead load effect is 72,000 foot tons (nearest 1000 foot tons). Since dead load opposes the sea forces in the upward direction the design load is less, or about 55,000 foot tons. The distribution of this moment to each major cross-structure member was based on a ratio of the assumed moments of inertia of each member. The shear loads were distributed as a ratio of assumed web areas. A summary of the sea loads for each bulkhead will be given in a later paragraph.

OTHER LOADS CONSIDERED

As a separate condition, the cross-structure was designed for what was considered the maximum possible torsional load on the cross-structure. To determine these loads, the following conditions were considered:

a. The ship could be drydocked with the port hull blocks and starboard hull blocks out of plane or the keel could be out of plane.

b. The ship may possibly run aground.

For these conditions the ship was assumed to be supported on one hull forward (Station 4) and on the other hull aft (Station 18). The load distribution of the applied torsional moment to each bulkhead in the cross-structure is assumed to be a function of the linear distance from the center of torsion and the vertical deflection in each loaded bulkhead.

STRUCTURAL CONFIGURATION OF THE ASR

The ASR is a 251 foot LOA catamaran with an 86 foot maximum breadth (26 foot wide hulls) and full load displacement of 3600 tons. Figure 9 shows a typical cross-section of one of the transverse support bulkheads for the ASR. The transverse bulkheads between the two hulls along with an effective breadth of plating as the upper and lower flanges is considered the primary supporting cross-structural member. There are six of these bulkheads similar to Figure 9 carrying the loads. Locations of these bulkheads are shown on Figure 8. The use of six bulkheads has no special significance other than that it provides a satisfactory arrangement for structure commensurate with compartment and access requirements and distributes the loads into the hull girders through scantlings of normal dimension. The three bulkheads forward and the three bulkheads aft form two separate deckhouses with an open well between for rescue operations.

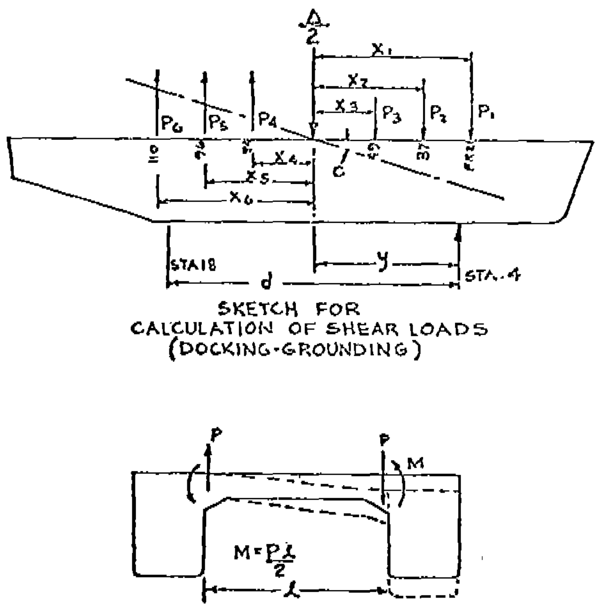

Fig. 8 - Forces Caused by the Settlement of Supports

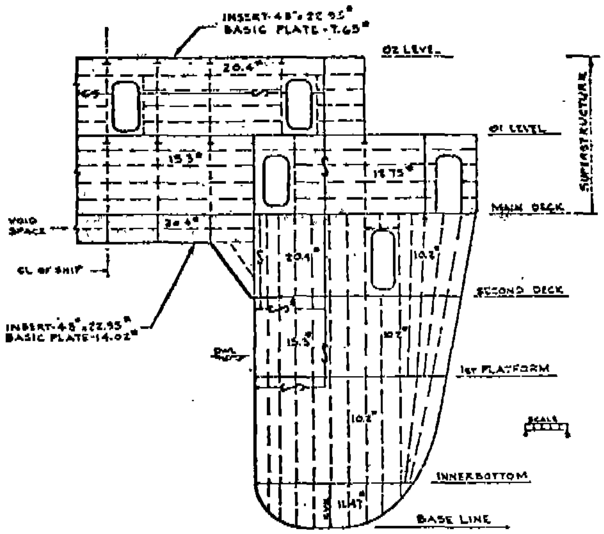

Fig. 9 - Typical Transverse Bulkhead for ASR

THE DESIGN PROCEDURE

The calculation of the load distribution to each bulkhead from the grounding-dock condition is as follows (See Figure 8):

Assumptions

1. The algebraic sum of moments about the center of torsion = 0, where center of torsion is assumed to be the centroidal axis of the assumed bulkhead areas.

$$\text{Torque} = \frac{\Delta d}{4} \qquad \text{(EQUATION 1)}$$

Where:

Δ = Total ship displacement (both hulls)
d = Distance Station 4 to Station 18
P_n = Shear Load on Bulkhead n.
X_n = Distance to Bulkhead n.

2. Deflection in each bulkhead is directly proportional to the linear distance (X_n) from center of torsion.

$$C = \frac{\delta_n}{X_n}$$

Where:

δ_n = Deflection of Bulkhead n.
C = Tangent of the Angle (for small angles)

In order to evaluate forces in the bulkheads it is convenient to compute spring constants, K_n, due to support settlement. Bending and shear strains were included in the spring constant. It was assumed that the torsional strength of the hull is large compared to the spring constants.

$$\delta_n = \frac{P_n}{K_n}, \text{ hence: } C = \frac{P_n}{K_n X_n}$$

or $$\frac{P_1}{K_1 X_1} = \frac{P_2}{K_2 X_2} + \ldots \frac{P_n}{K_n X_n} \qquad \text{(EQUATION 2)}$$

3. The cross-structure bulkheads are assumed to be fixed ended beams undergoing a settlement of the support. (See Figure 8)

Calculations

Equating the externally applied forces and internal resisting forces -

(Equation 1) $$\text{Torque} = \frac{\Delta d}{4} = \frac{\Delta y}{2}$$

Where:

$$y = \frac{1}{2} d$$

$$\left[\frac{\Delta}{2}\right] y = P_1 X_1 + P_2 X_2 + \ldots\ldots P_n X_n$$

From Equation 2:

$$P_2 = \frac{P_1 X_2 K_2}{K_1 X_1}, \quad P_3 = \frac{P_1 X_3 K_3}{K_1 X_1}, \ldots\ldots P_n = \frac{K_n X_n P_1}{K_1 X_1} \qquad \text{(EQUATION 3)}$$

$$\left[\frac{\Delta}{2}\right] y = P_1 \left[X_1 + \frac{K_2 X_2{}^2}{K_1 X_1} + \ldots\ldots \frac{K_n X_n{}^2}{K_1 X_1}\right] \qquad \text{(EQUATION 4)}$$

The equation is solved for P_1. All other P_n values can now be obtained from Equation 3. From the shear loads (P_n), bending moments in the bulkheads are attained from:

$$\text{Moment} = \frac{P_n L_n}{2}$$

Where:

L_n = Span of cross-structure between the hulls (Figure 8).

The final moments and shear values for each bulkhead in the cross-structure calculated by the foregoing condition and the predicted sea forces are recorded below and represent the actual values used for the ASR design. It must be noted however that after an initial scantling selection has been made the design loads were re-cycled and checked for the new inertias and web areas obtained as opposed to those assumed originally.

As can be seen from the resulting loads above, the docking-grounding condition gives the highest combination of shear and moment which governs the forward and aft bulkhead especially the web plating to resist buckling. The large variance of loads on Bulkhead 21 resulted from the fact that there was a deck height difference in depth from the other bulkheads.

	Predicted Sea Loads		Calculated Docking-Grounding Loads	
	Moment Ft. Tons	Shear Tons	Moment Ft. Tons	Shear Tons
Frame 21	4,600	80	5,180	305
Frame 37	11,600	104	6,650	342
Frame 49	13,200	104	3,900	230
Frame 84	13,200	104	4,150	244
Frame 86	14,700	104	6,900	406
Frame 110	14,700	104	10,200	596
Total	72,000	600	36,980	2,173

The resisting cross-structure bulkheads (21, 37, 49, 84, 86 and 110) have been designed similar to transverse bents on aircraft carriers. Of the total plate girder, the outer flange plus about 1/6 of the depth of the web assumed to entirely resist the bending moment. The total shearing force was assumed to be equally distributed to the entire web. For stability, the outer 1/6 portion of the web was sized to develop the the necessary shear or compressing buckling strength, whichever was worse, while the remaining 2/3 portion of the web was designed to develop a shear buckling strength equal to the shearing yield stress of the material. The above plate girder theory is based on actual tests indicating that the moment in a plate girder is concentrated in the flanges but drops off rapidly toward the neutral axis, unlike the straight line distribution used generally. As a result, the center portion of the web cannot be assumed to contribute. The 1/6 web depth used is an approximate value covering the actual moment. Reference (8) describes this method and test results.

As mentioned above, the upper and lower levels form a flange for the major cross-structure bulkheads. The actual width of plating, or "effective breadth" considered in the design is probably somewhat conservative.

Approximately four feet of normal deck plating is considered to act with the bulkheads as an effective breadth. The cross-structure, however, is more of a box girder with stiffened plating. Although under torsional and bending loads it is assumed that much more plating is effective, test data of large box girders is limited. It was, therefore, decided to consider the plating between bulkheads to provide an additional factor of safety rather than including it in the design at this time. It is hoped that structural model tests can be conducted on box structures in the near future to determine a more precise effective breadth. With the assumption that plating between bulkheads is ineffective, the design is then reduced from a torsional box girder problem to one of bulkheads resisting the loads imposed through pure bending and shear. With only four feet considered for the effective breadth, it was necessary to use inserts to provide the necessary section modulus for the structure shown on Figure 9.

The joint between the cross-structure and main hull was considered a most critical area. Additional web plating was added in the cross-structure and main hull bulkheads to reduce the nominal stress resulting from stress concentrations. There was also a problem of plate delamination. If the cross-structure was made intercostal to the hull, the shell plate could delaminate, and vice versa. To solve this problem, the insert plate acting as the lower flange of the cross-structure was carried continuously through the inboard shell into the second deck and the transverse bulkhead plating was carried continuously through the inboard shell and second deck. This provides an interlacing of the highly stressed structure so plate delamination would not lead to a major failure.

Since the cross-structure presented most of the problems, and is the basis for this paper, little has been said about the main hull and local loads. The hulls are designed using standard longitudinal strength calculations. Design of structure for local hydrostatic loads is similar to that found on conventional ships except for the shell plating inboard and bottom of the cross structure. During a visit to the catamaran drilling rig, E.W. Thornton, in the Gulf of Mexico, it was discovered that shell stiffening was badly damaged as a result of the pocketing effect of seas between the two hulls while the ship was moored. These forces somewhat resembled the effect found by the model test for the ASR. However, no effect on local stiffening could be predicted by the model test. As a result of the local damage found on the drilling rig, the shell inboard on the ASR was designed for 1500 pounds per square foot. The drilling rig had framing members intermittently welded. These welds suffered cracking throughout the length of the ship. For the ASR, continuous welding is specified.

CONCLUSION

The primary purpose of this paper has been to provide the ship design engineer with some basic knowledge of the problems encountered with catamaran hull structure and a simple approach to the solution of these problems. It will be necessary to conduct more tests on various hull forms and spacings before a completely analytical solution can be developed. It will then become important to instrument these hulls once they are built and attempt to correlate analytical predictions with full scale ship tests. The ASR and the commercial ship E.W. Thornton, along with a new oceanographic research catamaran now being designed, will provide valuable information for future designs.

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APPENDIX 3

This is a reproduction of the "Summary and Discussion" section with the associated nomenclature for reference (13), "A Method for Estimating Loads on Catamaran Cross-Structure" by A.L. Dinsenbacher.

NOMENCLATURE

(For Appendix 3 Only)

A	=	wave amplitude
A_b	=	wave amplitude for computing bending load
A_t	=	wave amplitude for computing torque load
B	=	beam of one hull
b	=	half beam of entire ship
C	=	center of twist of cross-structure
D	=	instantaneous mean draft
D_o	=	stillwater draft
d	=	distance from top of cross-structure down to neutral axis of cross-structure
G	=	center of gravity of ship
g	=	gravitational acceleration
H_1	=	depth (keel to top of cross-structure)
H_L	=	horizontal hydrostatic force on outboard side of a hull
H_R	=	horizontal hydrostatic force on inboard side of a hull
L	=	ship length (L_{BP})
L_w	=	wave length
$\overline{M}(y)$	=	cross structure bending moment at transverse coordinate y
$M(0)$	=	bending moment at midspan of cross-structure
M_Q	=	moment at junction of cross-structure and hull, not including effects of weight and mass of cross-structure
P	=	transverse axial load on cross-structure
p	=	horizontal hydrostatic pressure
Q	=	vertical shear force
S	=	clear span between hulls
T_c	=	torque on cross-structure about its twist center
t	=	horizontal distance (positive forward) from cross-structure twist center to ship's CG
v_L	=	distance from keel to center of pressure of horizontal hydrostatic force on outboard side of a hull
v_R	=	distance from keel to center of pressure of horizontal hydrostatic force on inboard side of a hull
W	=	ship weight
W_c	=	weight of cross-structure
x,y,z	=	coordinate system fixed on ship representation with origin at center of gravity
β	=	half the clear span between hulls
Δ	=	instantaneous displacement or buoyancy
ρ	=	density of water
ξ	=	distance, positive downward, from mean surface elevation to wave surface

SUMMARY AND DISCUSSION

The equations obtained thus far for estimating cross-structure loads will now be summarized. The symbols have been defined in the Nomenclature, and some are illustrated in Figs. 1 and 2. In several of the equations given in the following, the term (1 ± 0.4) appears; the positive sign indicates the ship accelerating upward, the negative sign corresponds to downward acceleration. Also, we have now substituted S/2 for β and (S/2) + B for b in the equations developed previously in the text.

For loading condition 1, the waves are approaching from the beam and are assumed to produce the greatest axial, vertical bending and shear loads. The directions of positive loads are shown in Fig. 3.

The wave length and amplitude, A (wave height = $2\,|A|$), for this loading case are taken to be

$$\underline{Lw} = 2(S + B) \tag{70}$$

$$A = A_b = \pm 1.0 \sqrt{\underline{Lw}} \tag{71}$$

The axial load on the cross-structure is

$$P = -2\rho g L A\,(1 \pm 0.4)\,D_o \sin \frac{\pi B}{2(S + B)} \tag{72}$$

The moment at the transverse mid-span of the cross-structure is

$$M(0) = -\rho g L A \sin \frac{\pi B}{2(S+B)} \left[2 D_o (1 \pm 0.4)(H_1 - d) - D_o^2 (1 \pm 0.4)^2 - \frac{A^2}{3} \sin^2 \frac{\pi B}{2(S+B)} \right] + \frac{W(S+B)A}{2\pi D_o} \left[\frac{2(S+B)}{\pi B} \sin \frac{\pi B}{2(S+B)} \cos \frac{\pi B}{2(S+B)} \right] + (1 \pm 0.4) \frac{W_c (S + 2B)}{8} \tag{73}$$

where A is given in (71). The sign of A is positive for a wave trough between the hulls, and negative for a crest.

The moment at the junction of cross-structure and hull is

$$M\left(\pm \frac{S}{2}\right) = M(0) - (1 \pm 0.4)\, W_c S/8 \tag{74}$$

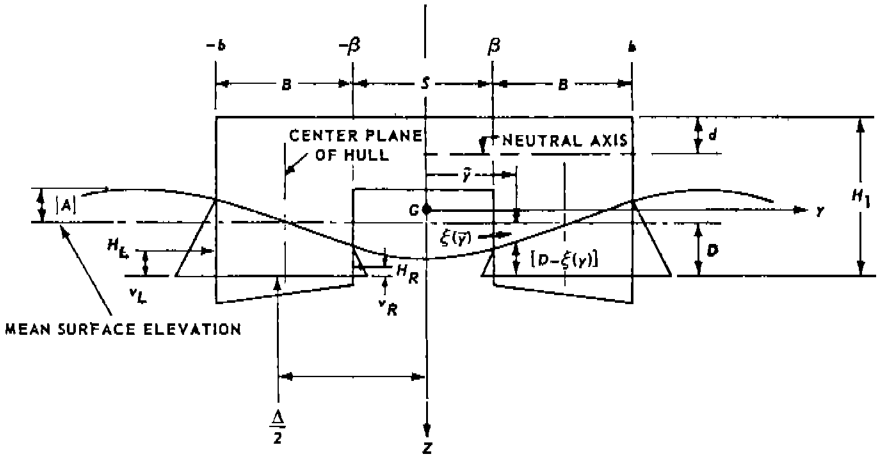
where M(0) is obtained from (73). Whichever sign is chosen in the term of (1 ± 0.4) in (72), the same sign must be employed in that same term in (73) and (74).

NOTES: View looking forward from stern

Coordinate system fixed on ship with origin at center of gravity (G) of ship

Wave length, $L_w = 2(b+\beta) = 2(S+B)$

A is wave amplitude (if A negative, crest between hulls)



D is height of wave surface above keel at center plane of hull (also distance from keel to mean surface elevation)

$\xi(\bar{y})$ is vertical location of wave surface from mean elevation, at transverse coordinate $\bar{y}$ (ξ positive in trough)

For wave surface: let $\xi(\bar{y}) = A \cos \frac{\pi \bar{y}}{(b+\beta)}$

Immersion of hull at $\bar{y} = D - \xi(\bar{y}) = D - A \cos \frac{\pi \bar{y}}{(b+\beta)}$

Fig. 1 - Loading Condition 1

The estimate for maximum shear, in beam waves, at the junction of cross-structure and hull, is

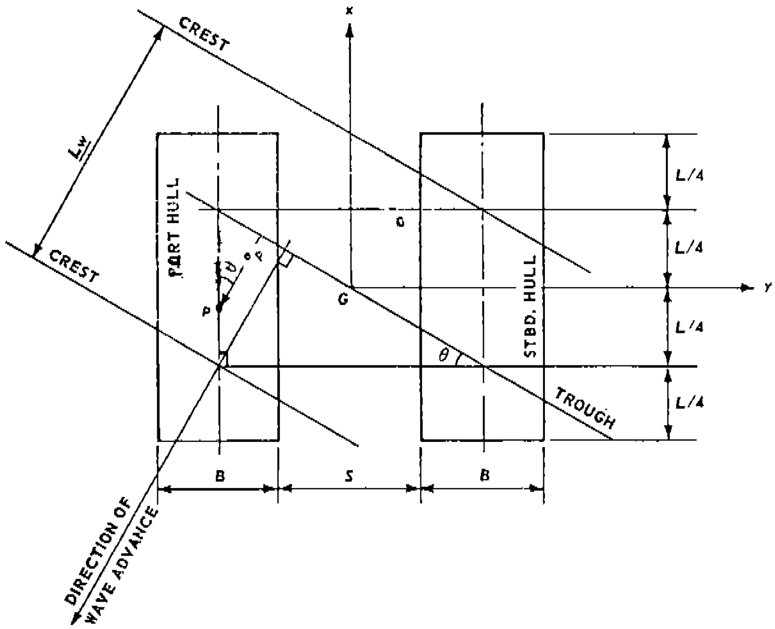
$$Q = (1 \pm 0.4) \frac{W_c}{2} + 0.34 \frac{M_Q}{S} \qquad (75)$$

in which

$$M_Q = M(0) - (1 \pm 0.4) W_c (S + 2B)/8 \qquad (76)$$

M(0) in (76) is obtained from (73). Again, the choice of sign in (1 ± 0.4) must be kept consistent throughout (73-76).

It is important, in estimating the shear and moment acting on the cross-structure, that the various combinations resulting from the choices of $\pm$ A and (1 ± 0.4) be computed. This is to insure that the maximum loads are found.



$$\theta = \tan^{-1}\frac{L/2}{S+B}$$

$$\underline{Lw} = 2(S+B)\sin\theta = \frac{L(S+B)}{\sqrt{(S+B)^2+(L/2)^2}}$$

At point p (on port hull centerplane), distance from mean surface elevation to wave surface is

$$\xi_p = -A\cos\left(\pi + \frac{2\pi u_p}{\underline{Lw}}\right) = A\cos\frac{2\pi u_p}{\underline{Lw}}$$

$$\text{But, } u_p = \left(\frac{L}{4} - x_p\right)\cos\theta$$

$$\text{so } \xi_p = A\sin\frac{2\pi x_p}{L}$$

$$\text{and immersion} = D - \xi_p = D - A\sin\frac{2\pi x_p}{L}$$

Fig. 2 - Loading Condition 2

For loading condition 2 (for maximum torsion), the wave advances obliquely as as shown in Fig. 2. The wave length and amplitude are

$$\underline{Lw} = L(S+B)/\sqrt{(S+B)^2 + (L/2)^2} \tag{77}$$

$$A = A_t = 0.6\sqrt{\underline{Lw}} \tag{78}$$

The reader is reminded that the wave height is twice the magnitude of the amplitude.

The estimate for maximum torsional load in waves is

$$T_c = \left|\rho C_b g\, BAL^2/2\pi\right| + \left|0.14 M_q t/S\right| \tag{79}$$

Table 1 - Loading Schedule A - For Direct Stress at Mid-Span of Cross-Structure

Load	Beam Waves	Quartering Waves
Axial Force	P from (72)	0.4 of P from (72)
Moment	M (0) from (73)	0.4 of M (0) from (73)
Shear	Not Applicable (N.A.)	N.A.
Torsion	N.A.	N.A.

Table 2 - Loading Schedule B - For Direct Stress at Junction of Cross-Structure and Hull

Load	Beam Waves	Quartering Waves
Axial Force	P from (72)	0.4 of P from (72)
Moment	$M\left(\pm\frac{S}{2}\right)$ from (74)	0.4 of $M\left(\pm\frac{S}{2}\right)$ from (74)
Shear	N.A.	N.A.
Torsion	0.4 of T_c from (79)	T_c from (79)

Table 3 - Loading Schedule C - For Shear Stress at Junction of Cross-Structure and Hull

Load	Beam Waves	Quartering Waves
Axial Force	N.A.	N.A.
Moment	N.A.	N.A.
Shear	Q from (75)	0.4 of Q from (75)
Torsion	0.4 of T_c from (79)	T_c from (79)

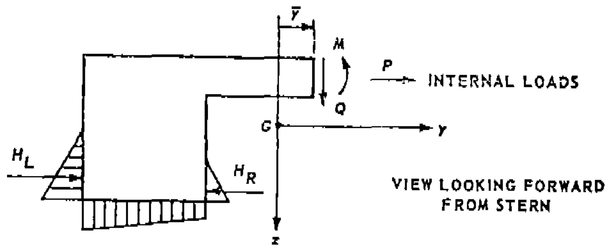


Fig. 3 - Positive Internal Loads

in which A is defined by (78), M_Q is computed from (76), and T_c is the magnitude of the torque about the center of twist of the cross-structure.

To estimate the maximum stresses on the cross-structure it is necessary to apply simultaneously, in certain proportions, the loads found in the foregoing. As has been stated in the text, the model test results showed moments and shears in quartering seas to be about 0.4 of their magnitude in beam waves. Also the torque in beam seas was found to be about 0.4 of its value in quartering waves. Therefore, to obtain estimates of maximum stresses, it is suggested that the loads should be applied in accordance with the loading schedules in Tables 1, 2 and 3. Each loading schedule is for a specific stress, and for the ship operating in both beam and oblique waves. For each case the stresses of interest, which are produced by the loads in the "Load" column, are calculated and summed. The equation to use to obtain a particular load is indicated in the row in which that load is designated.

It may be observed in Loading Schedule B, Table 2, that torsion loads are used in computing the direct stresses at the junction of the cross-structure and hulls. The reason for this is that the torsional load can produce bending moments on the ends of the transverse bulkheads spanning the hulls (2, 9).

CONCLUDING REMARKS

An attempt has been made herein to develop simple expressions for estimating gross loads on the structure linking the hulls of a catamaran. Although several gross assumptions and approximations have been made, some compensation for these has been introduced by relating, albeit empirically and/or heuristically, to model test results and current design practices for longitudinal strength.

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13. ABSTRACT

Existing United States shipbuilding facilities can handle 1000-foot catamarans with up to 140-foot individual hull beams on the premise that the hulls would be joined afloat. Major harbors and channels of the world suggest an overall beam limit of 400-feet and 35-foot draft. Drydocking for catamarans over 140-foot in breadth will require new facilities or extensive modification to existing facilities. Scantlings of a 1000-foot catamaran cargo liner can be expected to be within current shipbuilding capabilities. The uniqueness of the catamaran design lies in the cross-structure and the important facets of the cross-structure design are the prediction of the wave-induced loads and the method of structural analysis. The primary loads are the transverse vertical bending moments, axial force, shear, and torsion moments. Designers have relied heavily on model tests to obtain design loads and have used general structures principles and individual ingenuity to perform the structural analysis in the absence of established guidelines. Simple semi-empirical equations are proposed for predicting maximum primary loads. A structural analysis method such as the one proposed by Lankford may be employed for conceptual design purposes. The Lankford method assumes the hulls to be rigid and the cross-structure loads to be absorbed by a group of transverse bulkheads and associated effective deck plating. This procedure in general should provide an overall conservative design and not necessarily an economic or optimized design. Additional research and development work including systematic model test programs are necessary for accumulating additional knowledge in areas of uncertainty and for the establishment of reliable design methods for catamaran structure.

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Advisory Group II, "Ship Structural Design" prepared the project prospectus and evaluated the proposals for this project:

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MR. C. M. COX, *Asst. Naval Arch., Hull Des. Div., Newport News Shipbuilding & Dry Dock Co.*

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