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ANALYTICAL DETERMINATION OF STRUCTURAL LOADING ON ASR CATAMARAN IN BEAM SEAS

by

R. M. Curphey and C. M. Lee

SHIP PERFORMANCE DEPARTMENT
RESEARCH AND DEVELOPMENT REPORT

April 1974

Report 4267

ANALYTICAL DETERMINATION OF STRUCTURAL LOADING ON ASR CATAMARAN IN BEAM SEAS

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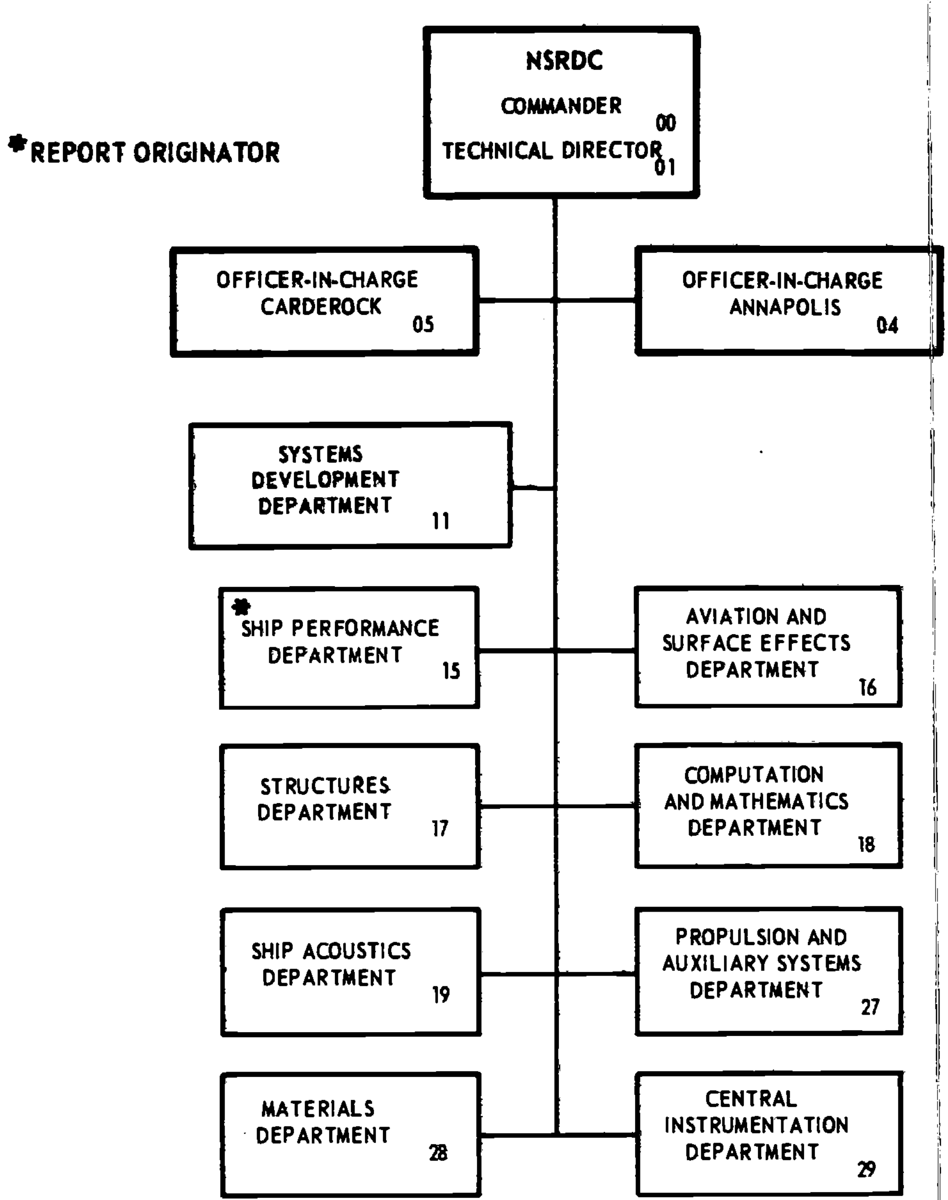

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REPORT DOCUMENTATION PAGE

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1. REPORT NUMBER 4267	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) ANALYTICAL DETERMINATION OF STRUCTURAL LOADING ON ASR CATAMARAN IN BEAM SEAS		5. TYPE OF REPORT & PERIOD COVERED 6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Richard M. Curphey and Choung M. Lee		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Ship Research and Development Center Bethesda, Maryland 20034		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS (See reverse side)
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Ship Systems Command Washington, D. C. 20360		12. REPORT DATE April 1974 13. NUMBER OF PAGES 25
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Loads
Catamaran
Wave Diffraction
Motion

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A two-dimensional model which incorporates the effects of wave diffraction and body motion has been developed to predict the dynamic structural loading on the crossbeam of a catamaran with zero forward speed in beam seas. Theoretical and model experimental results are compared for the amplitude of bending moment and vertical shear acting at the midpoint of the crossbeam of the ASR catamaran, a Navy submarine rescue ship. Correlation of theory with experiment is confirmed over the important frequency ranges.

(Block 10)

Subproject SF43.422.411

Task 17204

Work Unit 4-1500-001

TABLE OF CONTENTS

LIST OF FIGURES

NOTATION

A	Wave amplitude
$A_{1/3}$	Significant amplitude of bending moment or shear forces
B	Beam of demihull
B_m	Maximum beam of catamaran cross section
b	Distance between catamaran centerline and demihull centerline
F_H	Horizontal force acting on demihulls
F_V	Vertical force acting on demihulls
g	Acceleration due to gravity
i	Denotes $\sqrt{-1}$
h_o	Distance from neutral axis of catamaran crossbeam to the mean free surface
K	Wave number (ω^2/g)
M	Bending moment on crossbeam
M_I	Bending moment on crossbeam contributed by mass acceleration effects
m	Mass of catamaran cross section (mass/unit length)
$\vec{n}$	Unit normal vector on hull surface, positive into hull
n_2, n_3	Components of $\vec{n}$ along the O_y and O_z axes, respectively
O_{yz}	Cartesian coordinates; O_y axis coincides with undisturbed free surface, O_z axis is positive upward along a line midway between the hulls

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P	Pressure in the fluid
$R(\omega)$	Response amplitude operator of bending moment or shear forces
$S(\omega)$	Prescribed wave energy spectrum
T	Draft of catamaran cross section
t	Time
V_2, V_3	Horizontal tension and vertical shear force, respectively, on crossbeam
V_{2I}, V_{3I}	Horizontal tension and vertical shear force, respectively, on crossbeam contributed by mass acceleration effects
y_o	Horizontal distance along crossbeam at which loading is evaluated
$\overline{y}$	Horizontal coordinate of center of mass of demihull
Δ_2	Ship displacement
λ	Wavelength of incident wave
ξ_i	Displacement of the catamaran section from its equilibrium position (i = 2, sway; i = 3, heave; i = 4, roll)
ξ_i^o	Complex amplitude of the displacement (i = 2, sway; i = 3, heave; i = 4, roll)
ρ	Mass density of fluid
Φ	Time-dependent velocity potential function
ϕ	Time-independent velocity potential function
ϕ_D	Diffraction potential
ϕ_I	Velocity potential of incident wave

ϕ_M	Velocity potential due to body motion
ϕ_D^O, ϕ_D^E	Superscript denotes even or odd function with respect to $y = 0$
ϕ_i	Velocity potential for forced oscillations ($i = 2$, sway; $i = 3$, heave; $i = 4$, roll)
ω	Angular frequency of incident wave

ABSTRACT

A two-dimensional model which incorporates the effects of wave diffraction and body motion has been developed to predict the dynamic structural loading on the crossbeam of a catamaran with zero forward speed in beam seas. Theoretical and model experimental results are compared for the amplitude of the bending moment and vertical shear acting at the midpoint of the crossbeam of the ASR catamaran, a Navy submarine rescue ship. Correlation of theory with experiment is confirmed over the important frequency ranges.

ADMINISTRATIVE INFORMATION

The work described herein was carried out during fiscal year 1973 as part of the Catamaran New Initiatives Program under the sponsorship of the Naval Ship Systems Command (NAVSHIPS 0342). Funding was provided under Subproject SF43.422.411, Task 17204, Work Unit 4-1500-001.

INTRODUCTION

A mathematical model has been developed to predict the dynamic structural loading on the crossbeam of a catamaran with zero forward speed in beam waves. The model is unique in the sense that it includes the effects not only of the incident beam wave but also of the scattered waves and body motion. To verify the theoretical model, a comparison was made with existing experimental data[1] for bending moment and vertical shear force acting at the midpoint of the crossbeam of an ASR catamaran model of a Navy submarine rescue ship.

The present report outlines the theoretical approach, defines problem geometry and sign conventions, describes the theoretical approximations, and compares predictions for the amplitude of the bending moment and vertical shear force with experimental data on the ASR catamaran model.[1] Important features of the loading responses are discussed including the effect of the incident and scattered waves and body motion.

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[1]Wahab, R. et al., "On the Behavior of the ASR Catamaran in Waves," Marine Technology, Vol. 8, No. 3, pp. 334-360 (1971).

THEORY

The mathematical model presented here applies either to conventional shaped catamarans or to small-waterplane-area twin-hull (SWATH) ships. It is assumed that the hulls are symmetric about the vertical center plane and possess sufficient longitudinal symmetry so that only the sway, heave, and roll modes of motion are excited by the incident beam waves. With no pitching or yawing motion, the three-dimensional loading problem has been simplified to that of finding the motion and loading on an equivalent two-dimensional body. The equivalent two-dimensional hull has the cross-sectional form of the midship section of the catamaran in question and is taken to be uniform over an equivalent length such that the actual displacement of the ship is obtained. This two-dimensionalization is a gross geometrical approximation especially for conventional catamarans whose hull forms deviate significantly from a true two-dimensional shape. Despite this approximation, the theory appears to provide satisfactory results.

Figure 1 shows the midship cross section of a conventional shaped catamaran. A coordinate system 0_{yz} is fixed at the vertical centerline of the section and the mean water surface. A plane sinusoidal wave with amplitude A is progressing in the positive y-direction. The beam B_m, draft T, and separation distance b of the hulls are shown in Figure 1. The height of the neutral axis of the crossbeam above the mean water surface is indicated by h_o. The vector $\vec{n}$ is the unit surface normal on the submerged portion of the hulls with components $+n_2$ and $+n_3$ along the +y and +z axes, respectively. Positive sway ξ_2 and heave ξ_3 are small displacements of the ship from the equilibrium position in the positive y- and z-directions, respectively, and positive roll ξ_4 is the angular displacement from the equilibrium in a counterclockwise direction.

The conventions for the bending moment, shear, and tension forces acting at the midpoint of the crossbeam are indicated in Figure 2.

The bending moment is the moment which tends to roll the hulls relative to each other or, equivalently, to sag or hog the crossbeam. Positive bending moment is defined as the moment which tends to roll the right hull in a counterclockwise direction or the left hull in a clockwise direction.

Vertical shear and horizontal tension are the forces which respectively tend to heave and sway the hulls relative to each other. Positive vertical shear is defined as the force which tends to heave the right hull upward or the left hull downward. Positive horizontal tension is defined as the force which tends to sway the right hull to the right or the left hull to the left.

As the incident beam wave propagates past the body, a pressure distribution is established over the hulls which tends to excite motion in sway, heave, and roll and to produce structural loading at points on the section. As motion is excited, additional loads are generated due to the motion itself. If it is assumed that the hydrodynamic pressure distribution, wave exciting forces, resulting motion, and loads are all linear in amplitude and frequency with respect to the incident sinusoidal wave, a linear analysis in the frequency domain can be pursued to determine the amplitude and phase of the motion and load quantities.

In general the structural loading may be resolved into the following contributing effects:

1. Incident wave - When the body is restrained from moving, this component of the structural loads arises from the pressure distribution of the undisturbed incident wave over the submerged portion of the body surface. The assumption that the presence of the body does not distort the incident wave is commonly called the Froude-Krylov hypothesis.

2. Diffraction - This component accounts for the scattering of the incident wave by the presence of the body. When summed with the incident wave effect, the two contributions provide the loading on a body section which is restrained from moving.

3. Motion - As mentioned previously, when the body executes motion, additional loads are generated due to the motion itself. These are a result of mass acceleration, buoyant restoring, and hydrodynamic (added mass and wavemaking damping) effects.

LOADING FORMULATION

A standard approach to determine the structural loading on the crossbeam of the catamaran would be to cut the structure at the point where the loads are to be determined and to consider all of the forces and moments (both inertial and hydrodynamic) acting on the free end as in Figure 2.[2] The values obtained for the loads must be identical regardless of whether the portion of the body to the right or the left of the cut is taken to be the free end. Hence another approach for evaluating the loading is applicable[3] in which the loads contributed on both portions of the body are added with a sign consistent with the conventions defined in Figure 2 and the result is then divided by two.

This approach allows mass acceleration and pressure quantities to be evaluated for the whole body section. If the loads are evaluated at the midpoint of the crossbeam, the computation can be simplified by utilizing the symmetric and antisymmetric nature of the mass acceleration effects and pressure distribution with respect to $y = 0$.

The dynamic loading at the midpoint of the crossbeam ($y = 0$, $z = h_o$) is given by the mass acceleration effects minus the appropriate integral of the hydrodynamic pressure over the submerged body surface.

Bending Moment:

$$M = \frac{1}{2} M_I - \frac{1}{2} \int_{R+L} p[n_3 y + n_2(h_o - z)] \operatorname{sgn}(y) \, dl \tag{1a}$$

Horizontal Tension:

$$V_2 = \frac{1}{2} V_{2I} - \frac{1}{2} \int_{R+L} p n_2 \operatorname{sgn}(y) \, dl \tag{1b}$$

Vertical Shear:

$$V_3 = \frac{1}{2} V_{3I} - \frac{1}{2} \int_{R+L} p n_3 \operatorname{sgn}(y) \, dl \tag{1c}$$

[2] Pien, P. C. and C. M. Lee, "Motion and Resistance of a Low-Waterplane-Area Catamaran," 9th Symposium on Naval Hydromechanics, Paris, France (1962).

[3] Ogilvie, T. F., "On the Computation of Wave-Induced Bending and Torsion Moment," Journal of Ship Research, Vol. 15, No. 3, pp. 217-220 (1971).

Here M_I, V_{2I}, and V_{3I} are the mass acceleration effects of the full body section, R+L denotes integration over the submerged portion of the right and left hulls, sgn(y) denotes the sign of y which is positive on the right hull and negative on the left hull, and p is the hydrodynamic pressure. In the above equations, pressure is multiplied by appropriate surface normals--or surface normal moment arm in the case of bending moment--to provide loads consistent with the definitions of Figure 2 for the right and left hulls.

Since the catamaran is symmetric with respect to $y = 0$, n_2 and n_3 are respectively symmetric and antisymmetric with respect to $y = 0$, and it is clear that if the pressure is some arbitrary distribution over the right and left hull surfaces, only the symmetric part of the pressure distribution with respect to $y = 0$ can contribute to the bending moment and horizontal tension and only the antisymmetric part can contribute to the vertical shear.

Although the two approaches are completely equivalent, when mass acceleration effects are considered on the half body, all modes of motion must appear formally in the load equations. However, when summed over both halves of the body section, it is clear that some modes of motion cannot contribute to bending or shear, and these may be immediately neglected. For example, heaving of the ship section results in a vertical inertial force on each hull which is symmetric with respect to $y = 0$; as indicated by Figure 2, this force configuration can contribute only to the bending moment. Roll motion generates a vertical inertial force on each hull which is antisymmetric about $y = 0$ and can contribute only to vertical shear. Sway motion generates a symmetric horizontal force which cannot contribute to either bending or tension.

In order to evaluate the loading from Equations (1), it remains to determine the hydrodynamic pressure acting on the body hulls and the resulting motion. As mentioned previously, the pressure has components due to the incident and diffracted waves and motion; these include added mass, wavemaking damping, and buoyant restoring effects. The pressure is determined from potential-flow theory,[2,4] and the sway, heave, and roll motion may be obtained from the solution of the coupled equations of motions,[2] where pitch and yaw motion are taken to be zero and all hydrodynamic, mass, and restoring coefficients are evaluated in a two-dimensional sense.

The hydrodynamic pressure is given in terms of the velocity potential by the linearized form of the Bernoulli equation plus buoyancy terms:

$$p = -\rho \frac{\partial \Phi}{\partial t} - \rho g(\xi_3 + y\xi_4)$$

Since a time harmonic disturbance has been assumed, the velocity potential can be written as

$$\Phi(y,z,t) = \mathrm{Re}\ [\phi(y,z)e^{-i\omega t}]$$

where ϕ is a time-independent potential, which generally has real and imaginary parts. The pressure is then written as

$$p = \mathrm{Re}\ [i\rho\omega\phi - \rho g(\xi_3{}^{o} + y\xi_4{}^{o})]e^{-i\omega t} \tag{2}$$

The time-independent potential ϕ can be further resolved into the following components:

$$\phi = \phi_I + \phi_D + \phi_M$$

which respectively represent the fluid disturbance due to the incident waves, the diffracted waves, and the motions of the body. The potentials ϕ_I and ϕ_D generally have even and odd components with respect to $y = 0$ which may be denoted by

$$\phi_I = \phi_I^0 + \phi_I^E \text{ and } \phi_D = \phi_D^0 + \phi_D^E$$

[4] Lee, C. M. et al., "Added Mass and Damping Coefficients of Heaving Twin Cylinders in a Free Surface," NSRDC Report 3695 (1971).

The incident wave potential is given by

$$\phi_I = -\frac{igA}{\omega} e^{Kz + iKy}$$

where g = the acceleration due to gravity

A = wave amplitude

ω and K = the angular frequency and wave number

The motion potential ϕ_M is given by

$$\phi_M = \phi_2\xi_2^o + \phi_3\xi_3^o + \phi_4\xi_4^o$$

Here ϕ_2 is the potential for forced oscillations in sway; for a body symmetric about the centerline, it is an odd function with respect to $y = 0$. ϕ_3 is the potential associated with heave and is even, and ϕ_4 is the potential associated with roll motion and is odd. ξ_2^o, ξ_3^o, and ξ_4^o are the complex amplitudes of motion in sway, heave, and roll. The potentials ϕ_D and ϕ_i (i = 2, 3, 4) are calculated by the method of source distribution.[2,4]

Substitution of the above velocity potentials into Equation (2) and subsequent substitution of Equation (2) into (1) provide expressions for the loading at the midpoint of the crossbeam.

Bending Moment:

$$M = \mathrm{Re}\left\{\left[-\frac{1}{2}\omega^2 m\bar{y}\xi_3^o - i\omega\rho\int_R (\phi_I^E + \phi_D^E + \phi_3\xi_3^o)[yn_3 + (h_o - z)n_2]dl + \rho gBb\xi_3^o\right] e^{-i\omega t}\right\} \tag{3a}$$

Horizontal Tension:

$$V_2 = \mathrm{Re}\left\{\left[-i\omega\rho\int_R (\phi_I^E + \phi_D^E + \phi_3\xi_3^o)n_2 dl\right] e^{-i\omega t}\right\} \tag{3b}$$

Vertical Shear:

$$V_3 = \mathrm{Re}\left\{\left[-\frac{1}{2}\omega^2 m\bar{y}\xi_4^o - i\omega\rho\int_R (\phi_I^0 + \phi_D^0 + \phi_2\xi_2^o + \phi_4\xi_4^o)n_3 dl\right.\right.$$

$$\left.\left. + \rho g B b \xi_4^o\right] e^{-i\omega t}\right\} \tag{3c}$$

where m = mass of the full body cross section

$\bar{y}$ = y-coordinate of the center of mass of the right demihull

B = demihull beam

b = distance between the centerline of the two hulls and the demihull centerline

In summary, Equations (3) for the loading at the *midpoint* of the cross-deck show that the incident and diffracted waves contribute to all load quantities. However, heave motion affects only the bending moment and horizontal tension, and sway and roll motion affect only the vertical shear.

Once the loading at the midpoint of the crossbeam is known, these results may be used to determine the loads at any other point y_o along the neutral axis of the crossbeam. In particular, if it is assumed that the crossbeam is massless, the vertical shear and horizontal tension forces remain unchanged along the crossbeam, and the bending moment is given by

$$M(y_o) = M(0) - y_o V_3(0) \tag{4}$$

where $M(0)$ and $V_3(0)$ are the bending moment and vertical shear at the midpoint. If the crossbeam is not taken to be massless, the appropriate mass inertia effect of the beam section between the midspan and y_o must be subtracted from the results of Equations (3a), (3c), and (4).

RESULTS AND DISCUSSION

Regular wave results were computed for bending moment and vertical shear force based on the theoretical model just described. They were then compared to corresponding experimental data for the ASR catamaran model 5061. In addition the statistical bending moment and vertical shear were computed as a function of significant wave height by using the regular wave loading transfer functions together with a specified wave energy spectrum.

Pertinent geometric information for the ASR catamaran is given in Table 1. Experimental results were obtained from ASR model tests performed by Wahab et al.[1] for a hull separation distance of 1.41.* (1.58 ?)

Figure 3 indicates the predicted and experimental amplitudes of the bending moment and vertical shear at the midpoint of the crossbeam together with heave motion and roll motion for the ASR as a function of the ratio of the wavelength to overall beam (λ/B_m). The amplitude of the bending moment has been nondimensionalized by the total ship displacement times the wave amplitude ($\Delta_2 A$), the vertical shear force by the total ship displacement times the wave amplitude divided by the ship length ($\Delta_2 A/L$), the heave motion by the wave amplitude, and roll motion by the wave slope ($KA = 2\pi A/\lambda$). As mentioned previously, three-dimensional theoretical results were obtained by multiplying the two-dimensional results for the midship section by an equivalent ship length. The use of ship displacement and length in the nondimensionalization of the bending moment and vertical shear force is not intended to represent any particularly predominant functional relationship of the loading quantities. Two ships with the same displacement and length but different geometrical shapes could have significantly different loading amplitudes.

It is seen from Figures 3a and 3b that theoretical and experimental loading results were in relatively good agreement for both shape and magnitude. It is known that the apparent frequency shift of the bending and shear response peaks is due to the two-dimensional approximation. There was some discrepancy in the motion results at the longer wavelengths (Figures 3c and 3d). This effect and the sharply peaked nature of the theoretical heave and roll responses are also attributed to the two-dimensional approximation.

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*The ratio of the distance between the inner hull faces at the waterline to the demihull beam.

It is of some interest to examine the separate effects of the incident and diffracted wave and body motion on the loading quantities. The bending moment and vertical shear are plotted in Figure 4 to show the effects of the various components. The broken line curves represent the loading due to the undisturbed incident wave (Froude-Krylov effect). The dotted curves present the sum of the effects contributed by the undisturbed incident wave and the diffracted wave and represent the restrained body loading. The solid line curves indicate the addition of motion effects to the restrained body case and are simply replots of Figures 3a and 3b.

It was mentioned in the previous section that at the midpoint of the crossbeam, the only motion contribution to the bending moment would arise from heave and that both sway and roll should contribute to the vertical shear force. This trend is not particularly apparent from the experimental data for the ASR since the heave and roll resonances occurred at approximately the same frequency (Figures 3c and 3d). This point, however, has been verified for a MODCAT hull form, where the roll and heave resonances are widely separated in frequency. The experimental data indicated that a large roll resonance at low frequency had absolutely no effect on the bending moment at the midpoint of the crossbeam.

Under the assumption that the loading quantities are linearly superposable, the significant amplitudes of the bending moment and vertical shear may be obtained by

$$A_{1/3} = 2 \left\{ \int_0^\infty [r(\omega)]^2 S(\omega)\, d\omega \right\}^{1/2}$$

where $R(\omega)$ = response amplitude operator of either bending moment or vertical shear as shown in nondimensional form in Figures 3a and 3b

$S(\omega)$ = appropriate wave energy spectrum

ω = angular frequency of the incident beam waves

Figures 5 and 6 respectively present the significant amplitudes of the bending moment and vertical shear force for the ASR as a function of significant wave height. The solid curves indicate the dimensional values of the significant bending moment and vertical shear obtained when a

Pierson-Moskowitz sea spectrum is used. The cross marks show the significant amplitudes obtained for 307 Station India sea spectra.[5]

CONCLUSIONS

1. The two-dimensional theoretical model developed to predict the dynamic structural loading on the crossbeam of a catamaran with zero forward speed in beam waves provides results which are in good agreement with experiment for the amplitude of the bending moment and vertical shear force at the midpoint of the crossbeam.

2. Resolution of the theoretical results into components due to the incident wave, diffracted wave, and motion effects shows that all three have a very significant effect on the loading responses. Inclusion of wave diffraction and motion effects in this analysis is necessary to obtain good correlation with experimental results.

ACKNOWLEDGMENTS

The authors are grateful to Dr. J. P. Feldman and Mr. J. B. Hadler for their suggestions and careful review of the report and to Nadine Hubble for providing the results of Figures 5 and 6.

[5] Miles, M., "Wave Spectra Estimated from a Stratified Sample of 323 North Atlantic Wave Spectra," National Research Council Canada, Division of Mechanical Engineering Report LTR SH-128 (May 1972).

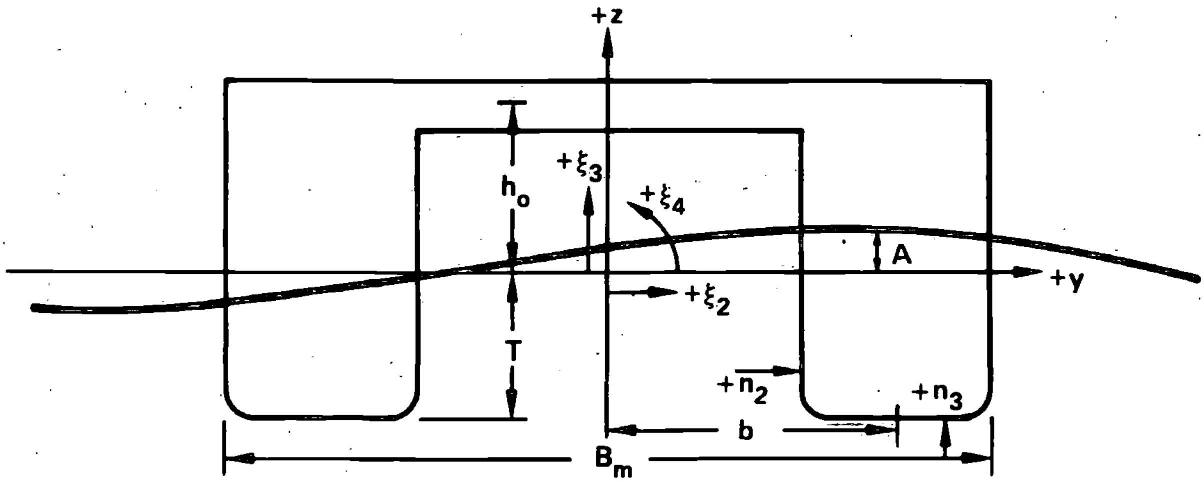

Figure 1 - Catamaran Geometry and Sign Conventions

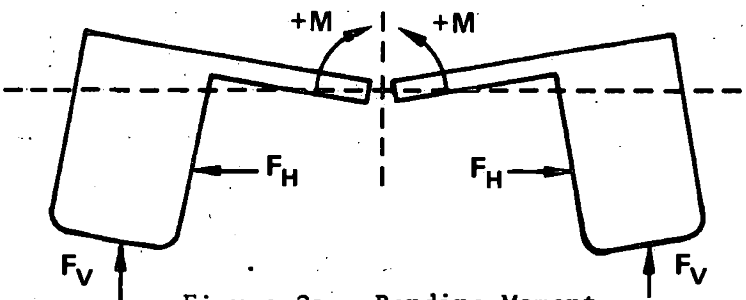

Figure 2a - Bending Moment

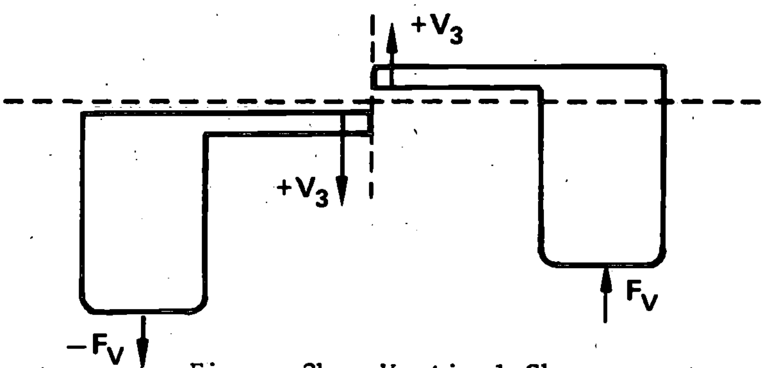

Figure 2b - Vertical Shear

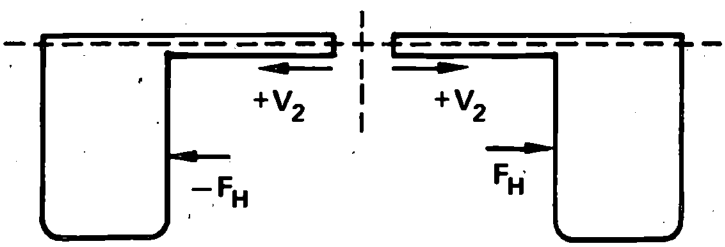

Figure 2c - Horizontal Tension

Figure 2 - Catamaran Loading Conventions

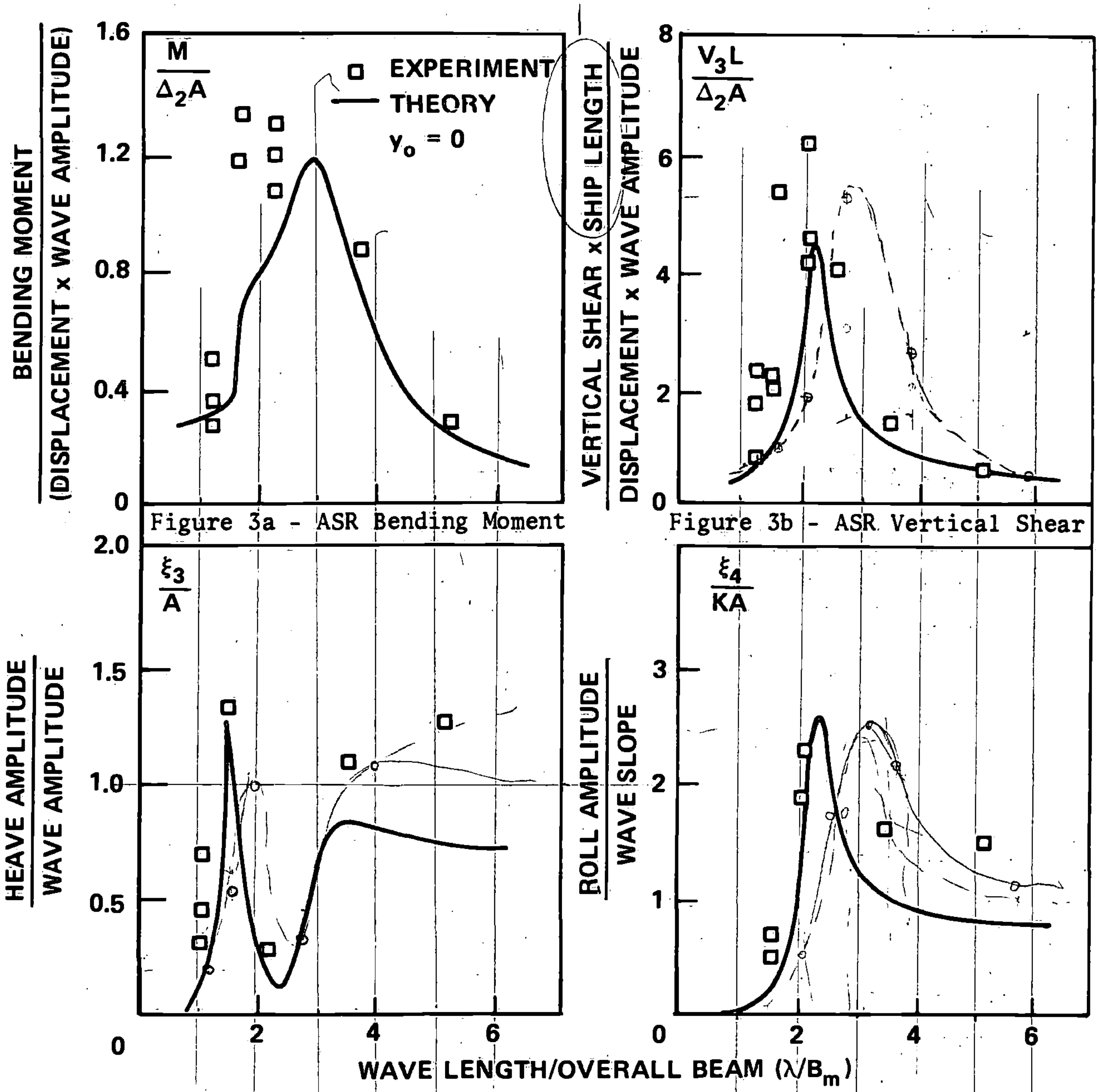

Figure 3a - ASR Bending Moment

Figure 3b - ASR Vertical Shear

Figure 3c - ASR Heave

Figure 3d - ASR Roll

Figure 3 - Crossbeam Loading and Motion of ASR Catamaran in Regular Beam Waves

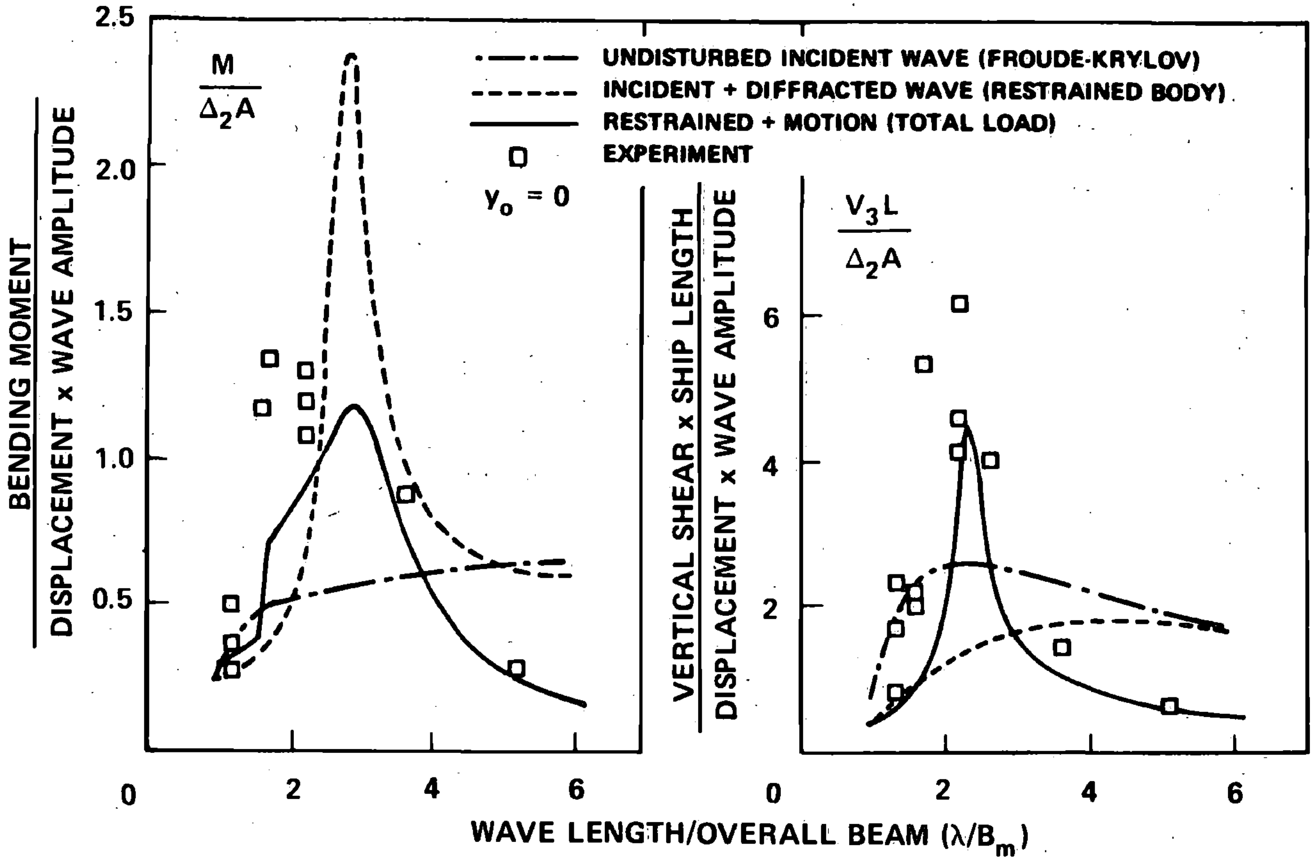

Figure 4a - ASR Bending Moment

Figure 4b - ASR Vertical Shear

Figure 4 - Decomposition of Loading Effects for ASR Catamaran in Regular Beam Waves

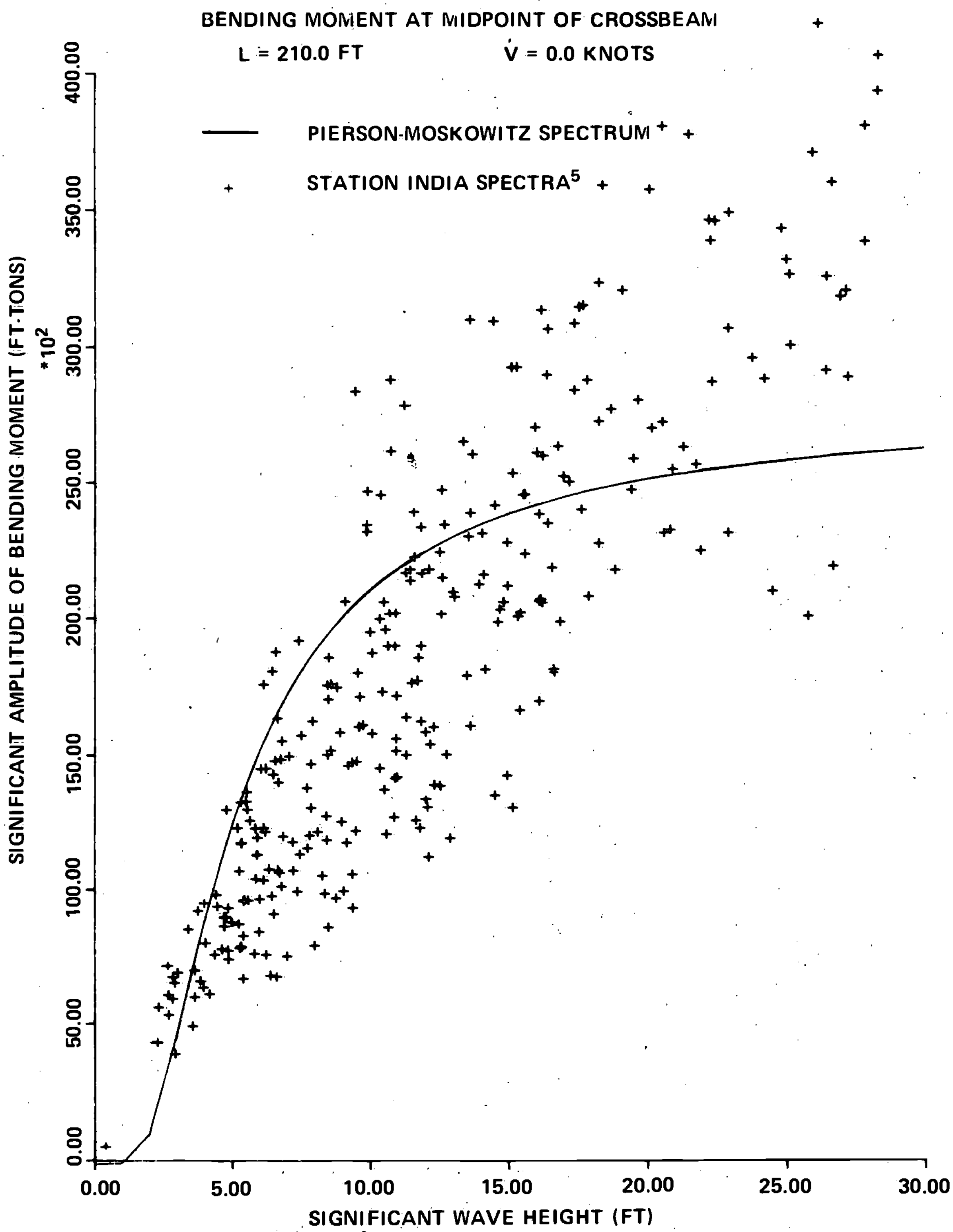

Figure 5 - Crossbeam Bending Moment on ASR Catamaran in Irregular Beam Seas

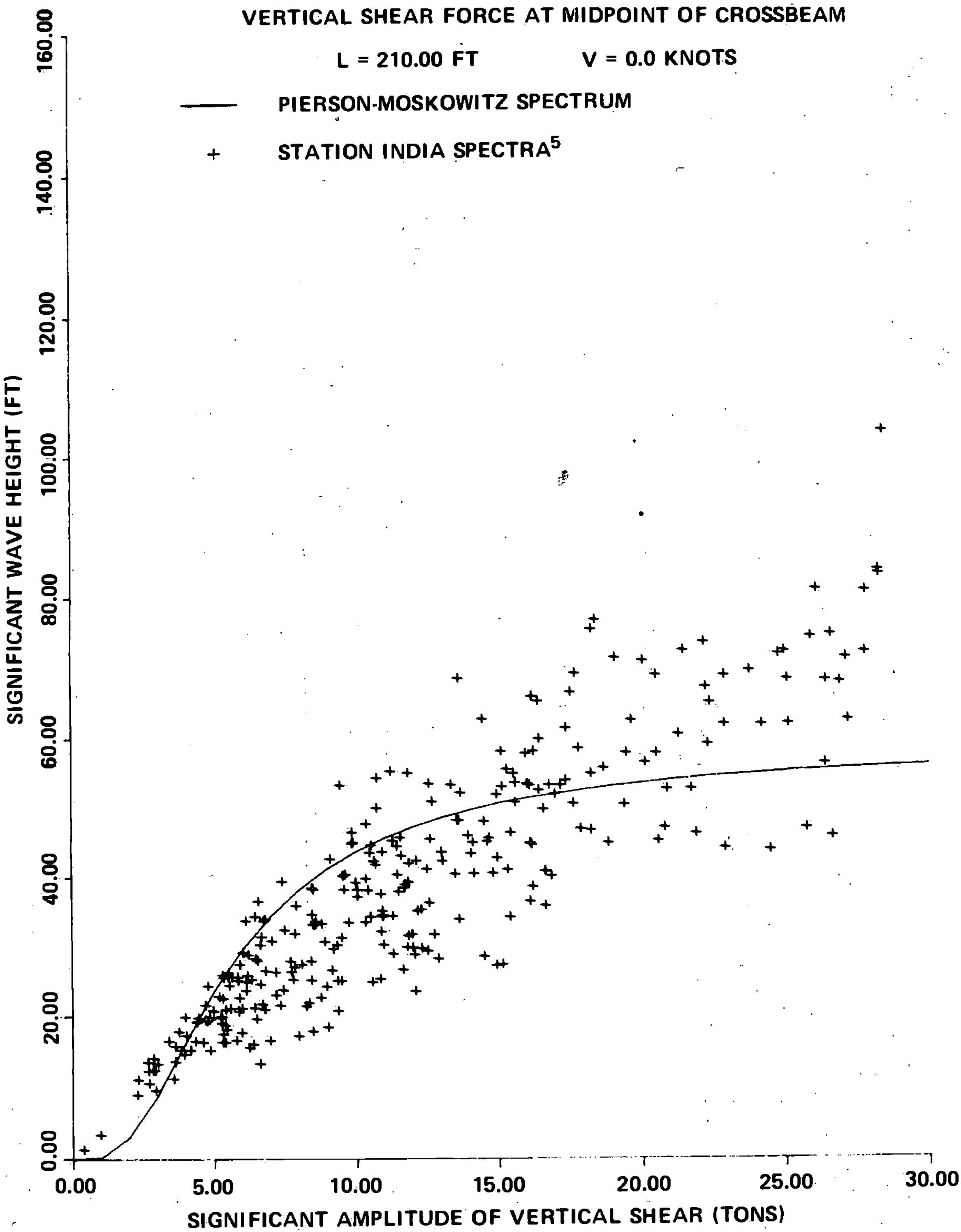

Figure 6 - Crossbeam Vertical Shear on ASR Catamaran in Irregular Beam Seas

TABLE 1 - ASR DIMENSIONS

NSRDC Model Number	5061
Beam (Each Hull) at the Waterline, feet	24.0
Draft (Station 10), feet	18.0
Length at the Waterline, feet	210.0
Displacement of Each Hull, long tons	1386 (S.W.)
Hull Spacing, feet	38.0
Longitudinal Center of Gravity Aft of FP, feet	105.6
Longitudinal Radius of Gyration, feet	0.233L
Block Coefficient	0.55
Scale Ratio	16.89
Diameter, feet	–
Vertical Height of Neutral Axis from Mean Waterline, feet	23.0

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