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SWATH VESSELS FOR NAVAL APPLICATIONS DESIGN AND ANALYSIS

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ABSTRACT

The Small Waterplane Area Twin Hull (SWATH) vessel design concept is currently being developed for a range of naval vessel applications which, by virtue of their mission requirements, call for a platform with unusual stability attributes by comparison to conventional hull forms. The history of SWATH design development evolves from catamaran and semi-submersible hull forms with refinement of the variables of lower hull geometry and size, distance between and connection to hull/strut members, strut area and shape, distance from underdeck structure to operating draft, methods of propulsion, methods of steering and weight distribution. When properly combined, these variables provide for a platform design which achieves desirable ship motion characteristics for the intended operating range. A most critical component in developing an effective SWATH design is the determination and application of reasonable hydrodynamic loading criteria based upon anticipated combinations of sea state, heading and speed. The utilization of both analytically derived hydrostatic loads as well as those obtained from model testing will be discussed. The current methods of sophisticated structural analysis applicable to an unusual hull form of the SWATH type will be discussed in detail. A review of current SWATH vessel design methods and applications will be provided. Conclusions are reached as to design considerations best suited for SWATH vessels.

INTRODUCTION

At the present time, the application of the concept of Small-Waterplane-Area-Twin-Hull (SWATH) is of increasing interest to the naval design community. The SWATH ship of comparable size to a conventional catamaran or monohull displacement ship has more favorable characteristics for certain applications, namely: large transverse stability; low vertical motion and acceleration amplitudes; high sustained speed at most headings to ocean waves; and large deck area for relative displacement. These attributes make the SWATH design very desirable as a naval platform in cases where primary mission requirements necessitate the use of a vessel with the ability to operate with unique seakeeping characteristics at a variety of speeds and headings throughout a broad spectrum of sea conditions. As a result of its small waterplane area, the SWATH ship is characterized as having relatively long motion natural periods, particularly roll motion. The long natural periods enable the vessel to avoid resonant excitation in most sea states. The SWATH ship resonance motions may however be excited by long swells or very high sea states of long period. In addition to

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the attributes noted, the SWATH design offers a number of secondary benefits which are desirable for naval vessel applications, namely: generally good damaged stability characteristics; unique layout flexibility due to rectangular platform geometry; relatively high propeller efficiency; preordained opportunity for physical system redundancy; relatively low underwater noise signature; and outstanding maneuverability characteristics at low speeds. The SWATH design does have some disadvantages which must be considered when designing for certain naval vessel roles namely: extremely weight sensitivity; unusual draft and freeboard considerations; potential for upper hull underdeck slamming; unusually large beam ratio; relatively slow speed in calm water. Naval designers may attempt to mitigate the effects of some of these disadvantages by a variety of means including sophisticated ballasting systems, variable draft capability and active control surfaces.

The SWATH design therefore may be considered for a variety of naval vessel roles including: strategic positioning; surveillance; oceanographic and hydrographic survey; personnel transport; offshore patrol; offshore range support; mine warfare; towing; and vertically deployed aircraft platform.

Historically, the SWATH design evolved from conventional catamaran and semi-submersible platform technology. Much is available in the literature concerning the chronology of this evolution. Some of the difference between the SWATH design and those from which it evolved are very important in concept, as well as application if an effective SWATH design is to be generated. The size and shape of the lower hulls must provide effective buoyancy, as well as acceptable hydrodynamic properties without sacrificing strength requirements. The methods of attachment of the lower hulls to the struts, and the struts in turn to the upper hull are very important given the levels of stress transition at those areas. The number, size and geometry of the struts must be determined to provide the optimum water plane area and air draft while maintaining required strength. Propulsion configuration must consider unique hull geometry as well as acceptable weight distribution in layout. Control surfaces and steering interface must effectively provide both the desired seakeeping and maneuvering characteristics while handling the unusual loads encountered as a result of hydrodynamic considerations. Both structural and mechanical weight requirements and distribution must be precisely considered due to the weight sensitivity of the hull form.

The determination and application of motion and wave loads must be of primary concern during design. The structural response and fatigue life associated with the design must be carefully ascertained. Given the unique stress paths endemic in a SWATH design, certain structural details require analysis. Examples are openings in bulkheads and brackets.

LOADS

Like all marine structures, SWATH vessels are subject to seaway loads which are generally classified as Primary and Secondary Loads as follows:

- Primary Loads are those loads which affect the structure as a whole and therefore generally govern the general configuration and scantlings of the vessel.
- Secondary Loads are more localized by nature and therefore generally govern local scantlings and structural details.

The determination of both types of loads for SWATH vessels is both different and more complex than required for monohulls.

PRIMARY LOADS

Primary Loads for monohulls are those acting along the length of the vessel which produce shear and bending forces along the longitudinal axis of the hull girder. Transverse loads have been shown to be of minor consequence in the vast majority of cases. Conventional practice for monohulls is therefore based in the two-dimensional analysis of the vessel at various load conditions upon waves of various heights and shapes as well as in still water. In general, the design wave height or shape does not equate to specific sea conditions or extreme cases, but represent a base line, or reference condition, from which the structural adequacy of the hull can be judged in comparison with previous, successful designs. This practice is not suitable or adequate for SWATH vessels.

The separate hulls and greater beam of SWATH vessels result in a predominance of forces in the transverse direction. Differential hydrostatic forces and dynamic effects alternatively spread the hulls apart and squeeze them together. Further, these transverse forces generally vary along the length of the vessel due to the configuration of the struts and lower hulls resulting in torsional effects. Thus for a SWATH vessel it is necessary to consider three-dimensional, dynamic (or at least semi-static) loading conditions. The alternating nature of the loads and typical scantlings of SWATH vessels raise concern for fatigue strength which require consideration of long term and extreme case loading consideration. The current state of SWATH design development is further compounded by the lack of extensive full scale experience. These factors combine to make the determination of loads for SWATH vessels more complex and more critical than for monohulls, and thus justify the expenditure of much more time and effort during design.

SECONDARY LOADS

The nature of secondary seaway loads such as localized wave impacts are the same for monohulls and SWATHs. SWATH type vessels, however, generally have extensive flat areas on the underside of the cross structure which vastly increase the likelihood of wave impact at very high pressure. Recent model tests even suggest that the outboard surfaces of the struts (which are very similar to the side shell of a monohull) may be subjected to high, localized wave impact pressure. It is, therefore, necessary to include consideration of these secondary loads throughout the design development of a SWATH vessel.

PRELIMINARY LOAD ESTIMATION

During early stages of design (prior to development of refined hull form, reliable weight estimates and final bulkhead locations) it is necessary to rely on load estimation methods for initial sizing of scantlings and evaluation of proposed structural configurations. For this purpose, the Primary Load estimating procedures developed by the U.S. Navy at the David Taylor Research Center (DTRC) have proven useful.

Full descriptions of the procedures and their development are given in references [1] and [2].

As noted previously, the most significant wave induced loads on a SWATH type vessel are the side forces which act on the side struts and lower hulls. Studies have indicated these forces are greatest in the beam sea case and decrease to insignificance in the head and following sea cases. The quartering sea cases are important since these appear to produce the greatest torsional effects.

Sikora and Dinsenbacher propose [3] an algorithm to estimate maximum side load based on analysis and model tests of over 15 SWATH design studies which included single and tandem struts per side. The algorithm applies to vessels in the 3,000 ton to 30,000 ton range operating in the North Atlantic at random headings for a 20 year period at 50 percent operability (e.g. 3,600 at sea days), The maximum lifetime beam sea side load is estimated as:

$$F = \Delta DTL$$

where

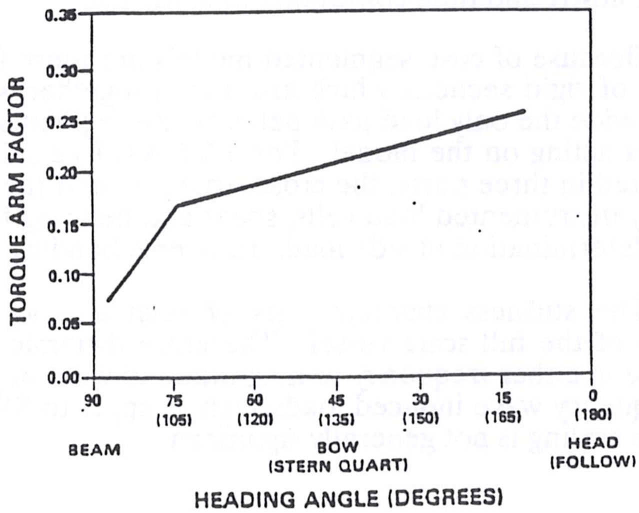
F = maximum lifetime load (tons)
Δ = displacement (tons)
$D = 1.55 - 0.75 \tanh (\Delta/11{,}000)$
$T = 0.532 \times \text{draft (ft)} / \Delta^{1/3}$
$L = 0.75 + 0.35 \tanh (0.5L_s - 6.0)$
$L_s = \text{strut length (ft)} / \Delta^{1/3}$

Studies indicated that this force appears to act at mid-draft for the beam sea case and approximately so for the quartering sea cases.

The longitudinal center of this force is more difficult to estimate, and is critical in distributing the side load along the length of the vessel. For this purpose Sikora and Dinsenbacher propose a torque arm factor, figure 1, which is the percent of the strut length from the mid length point of the strut at which the side force is centered. For design they recommend distribution of the side force along the length of the strut (ignoring lower hull overhang) as a trapezoidal function with the centroid of the trapezoid located as noted above.

FIGURE 1

TORQUE ARM FACTOR



The DTRC studies indicate correlation of algorithm estimates to model test results to be slightly greater than 10 percent, which is adequate for parametric studies and preliminary designs.

Studies in the United Kingdom for the Royal Navy [4] resulted in endorsement of the Sikora - Dinsenbacher algorithm for preliminary load estimates. Estimates of potential error were developed which allow designers to make corrections based on engineering judgement.

ESTIMATES OF SECONDARY LOADS

No "rule-of-thumb" has yet been proposed for secondary or wave impact loads although extensive work is being carried out. These loads are too dependent upon vessel configuration (haunch shape, vessel length, wet deck height above water line, etc.) and operating conditions (sea state, vessel speed, etc.) to allow reliance on simplistic estimates. It is clear, however, that these loads are substantial. The T-AGOS 19 wet deck was designed for approximately 50 psi forward, 63 psi at midship and 20 psi in the stern [3], where as the KAIYO wet deck was designed for approximately 100 psi at the bow, 25 psi at midship and 50 psi at the stern [5]. The large discrepancy in these values and their distribution indicate the unsettled nature of the load determination.

LOAD DETERMINATION BY MODEL TESTING

The testing of SWATH models for load data is complex and requires extensive facilities. The models must be hydrodynamically accurate and of adequate scale to allow instrumentation for either direct load measurement or stress response. The testing tank must be equipped with wave makers programmed to reliably generate a variety of wave conditions. The facility must be large enough to allow testing of beam and quartering sea cases without interference by side wall effects.

Load models are of two types; flexible and segmented, rigid body type. A flexible model duplicates the relative stiffness of the full scale structure and provides similar stress response. Rigid vinyl plastic such as PVC (Polyvinal Chloride) is frequently used for such models. They normally provide highly accurate results, but are very costly and time consuming to construct.

Because of cost, segmented models are more frequently used. This type model consists of rigid sections which are joined together with flexible load cells. The load cells provide the only load path between the segments and thereby allow determination of forces acting on the model. For a SWATH vessel such a model would typically be segmented in three parts; the cross structure, and the two struts and lower hulls. With properly instrumented load cells, shear and bending force data may be obtained which allows determination of side load, transverse bending and longitudinal tension.

The stiffness characteristics of such a model do not generally scale to the stiffness of the full scale vessel. Therefore dynamic responses from such a model do not scale in either frequency or magnitude to what might be expected at full scale. For low frequency wave induced loads such as apply to SWATH primary loads, this lack of dynamic scaling is not generally significant.

The use of models to determine wave impact loads requires pressure sensors or load diaphragms located at the areas of interest. Both systems have limitations and add significantly to the complexity of the process.

Pressure sensors are relatively easy to install on a model, but since they measure only point, peak pressure, their meaning is often difficult to assess. Peak pressures, while of interest, are not really significant in the design since they occur only over a very small area. Pressure sensor data must, therefore, be adjusted to apply to larger areas such as plate panels or stiffener grillages.

Pressure diaphragms are more difficult to install but yield more useful results. Pressure diaphragms are large and are designed to simulate the relative strength properties of the actual structure. The diaphragm may represent plate panels or entire grillage systems. They are instrumented to provide strain response data from which significant load data may be obtained.

In either case, statistical processing of the wave impact data is necessary to determine design load values.

Extensive wave impact model experiments on a SWATH model have been conducted in Canada by Defense Research Establishment Atlantic (DREA) as reported in reference [6]. These experiments involved a large radio-controlled model operating in Bedford Basin near Halifax where sea conditions approximate North Atlantic conditions. The intent of this experiment was to develop a simple empirical method to predict peak impact pressures. The results, however, exhibited excessive scatter in the data (in excess of 20%) which precluded achievement of the goal.

Previous model experiments by Zarnick at DTRC are also reported in reference [6] to have had the same problem. More recent tests may have resolved there difficulties but these have not been reported on in public literature.

ANALYTICAL DETERMINATION OF PRIMARY LOADS

The availability of powerful computer facilities have made possible the practical development of highly accurate load prediction programs for complex vessels such as SWATH hull forms. Work is progressing in the U.S., U.K. [4] and Japan [5] with varying degrees of success. The following is a review of analytical investigations conducted by the American Bureau of Shipping for several SWATH type vessels with excellent results as reported in references [6] and [7].

The analytical approach followed by ABS for determining wave loads is based on the linear response of the vessel to harmonic wave excitation. Strip theory and a two-dimensional source sink distribution method are used to obtain the hydrodynamic coefficients associated with motion equations which take into account the hydrodynamic interaction of the two hulls.

The hydrodynamic pressure distribution over the surface of the wetted hull section consists of four components:

1. the incident wave pressure due to plane pressure waves,

2. the diffracted wave pressure due to the presence of the body held in the wave train,
3. the radiated wave pressure due to the body oscillating in otherwise calm water,
4. the quasi-hydrostatic pressure due to the change in vertical displacement of the body. The velocity potential of the incident wave (with time factor $e^{-i\omega t}$ omitted) is given by:

$$\Phi_I = \frac{ga}{\omega_o} e^{k_o y} e^{i(k_o x \cos \mu - k_o z \sin \mu)}$$

where g, a, k_o, μ are gravitational acceleration, wave amplitude, wave number, and wave heading angle respectively.

The incident wave pressure is then represented by:

$$P_I = -i \rho g a\, e^{(k_o y)}\, e^{-ik_o z \sin \mu}\, e^{ik_o x \cos \mu}$$

The diffraction pressure and radiation pressure are as follows:

$$P_D = \rho(i\omega + U\partial/\partial x)\ \phi_D\ (x, y, z; k, \mu)\, e^{ik_o x \cos \mu}$$

$$P_R = \rho(i\omega + U\partial/\partial x) \sum_{m=2,3,4} S_m(x)\ \phi_R^{(m)}(x, y, z)$$

with

$$S_m(x) = \begin{cases} \zeta_2 + x\zeta_6 - iU/\omega\, \zeta_6 \\ \zeta_3 - x\zeta_5 + iU/\omega\zeta_5 - OG\,\zeta_4 \\ \zeta_4 \end{cases}$$

where U is the ship speed, ϕ_D is the diffraction potential; $\phi_R^{(m)}$ is the radiation potential in mode m per unit motion displacement. Modes 2, 3, 4, 5 and 6 denote heave, sway, roll, yaw and pitch respectively. Surge motion, although it is calculated, is not included in the pressure calculation. Both ϕ_D and ϕ_R are obtained as solutions to the two-dimensional fluid boundary value problem by the method of source-sink distribution. These hydrodynamic pressures are used for the evaluation of wave exciting force and added mass and damping coefficients. OG is positive when the center of gravity is above the coordinate origin.

The quasi-hydrostatic pressure due to body motions is:

$$P_s = -\rho g\, (\zeta_2 + x\zeta_6 - Z\zeta_4)$$

Hence, the resultant hydrodynamic pressure is given by:

$$P = P_I + P_D + P_R + P_S$$

An example of these pressure calculations is shown in Figure 2.

Figure 2 - Sample Pressure Load Distribution

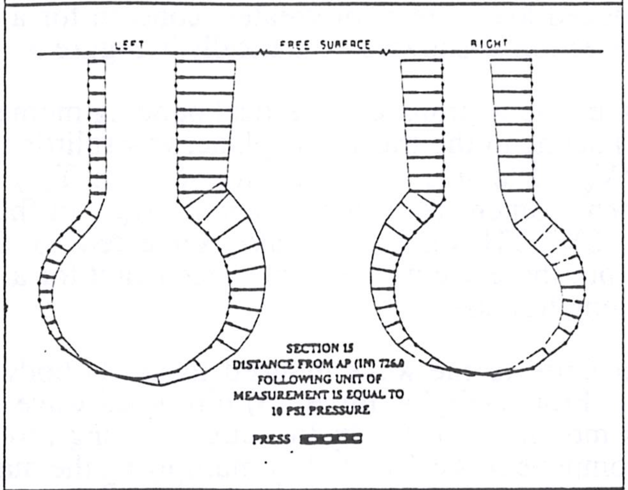


Figure 3 - Shear forces and moments at the midpoint of the cross structure in the oyz and oxz planes

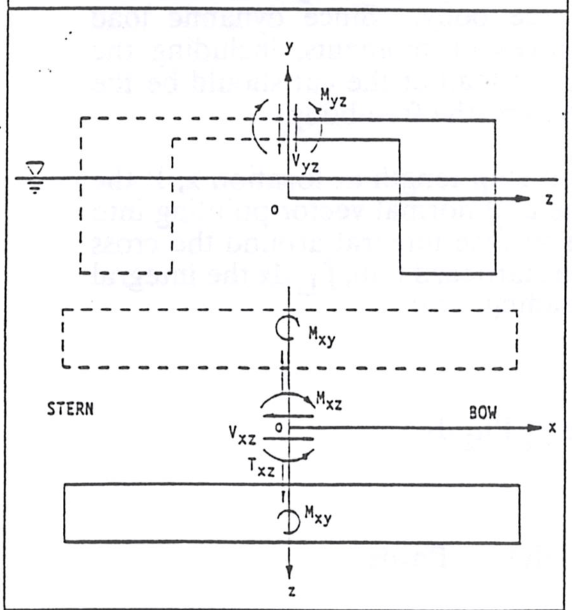
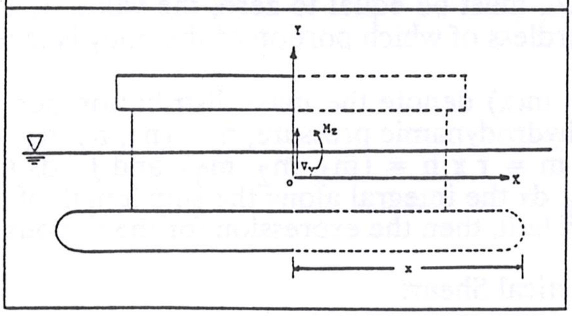


Figure 4 - Shear forces and bending moments at station x



These hydrodynamic forces must be calculated for a SWATH vessel in beam and quartering seas for which the maximum critical loads can be expected according to model test results. The wave induced load effects of greatest concern for a SWATH type vessel are in the transverse direction as shown schematically in Figure 3.

These loads are the prying moment (transverse vertical bending moment, M_{zy}) and the vertical shear force (V_{yz}) acting in the transverse plane, yaw splitting moment (M_{xz}), longitudinal shear force (V_{xz}) and side force (transverse force T_{xz}) acting in the horizontal plane, and the pitch moment (torsional moment, M_{xy}) in the vertical plane. As shown in Figure 4, SWATH vessels are also subjected to the same longitudinal forces as monohulls, but these are not generally significant for a SWATH as currently configured (beam to length ratio).

The following factors contribute to the wave induced loads: 1) body mass or inertia force; (2) incident wave or Froude-Krylov force; (3) diffracted wave force (4) hydrodynamic force due to body motions; and (5) hydrostatic restoring force due to vertical displacement. The first component is calculated by multiplying the mass by the acceleration. The components (2) through (5) can be represented by a resultant pressure force which is obtained by integrating the hydrodynamic pressure over the wetted hull surface. These force components must be superimposed, with the proper phase angles, to give the wave induced loads.

For computations, the structure is assumed cut along the centerline to calculate the effects of the wave loads at the midpoint of the cross-structure. The removed portion of the structure is replaced by three forces and three moments.

These reaction forces and moments are calculated by summing the force and moment components noted above acting on the free body. Since dynamic load equilibrium requires that the summation of the forces or moments, including the inertia load, must be equal to zero, the value of a given load at the cut should be the same regardless of which portion of the body is taken to be the free-body.

Let m(x) denote the mass distribution per unit ship length at location x, P the resultant hydrodynamic pressure, $n = (n_1, n_2. n_3)$ the unit normal vector pointing into the body, $m = r \times n = (m_1, m_2, m_3)$, and $\int_c ds$ the surface integral around the cross section, $\int_R dx$ the integral along the ship length of the starboard hull, $\int_L dx$ the integral of the port hull, then the expression for the various loadings are:

Vertical Shear:

$$V_{yz} = 1/2 \int_R \omega^2 m(x) \left(\zeta_2 + x\zeta_6 - z\zeta_4\right) - \int_R dx \int_c P n_2 \, ds$$

or

$$V_{yz} = 1/2 \int_R \omega^2 m(x) z\zeta_4 \, dx - 1/2\left(\int_R dx - \int_L dx\right) \int_C P n_2 ds$$

Transverse Shear or Side Force:

$$T_{xz} = - 1/2 \int_R \omega^2 m(x) \left[\zeta_3 - x\zeta_5 - (y - OG)\zeta_4\right] dx - \int_R dx \int_C P n_3 \, ds$$

or

$$T_{xz} = -1/2\,(\int_R dx - \int_L dx) \int_C Pn_3 ds$$

Prying moment:

$$M_o = -1/2\int_R \omega^2\, m(x)\,\zeta_2\, Z\, dx - 1/2\,(\int_R dx - \int_L dx) \int_C Pm_1\, ds$$

where Z is the distance of the center of gravity of the starboard hull from the oy-axis.

After transforming M_o to the midpoint of the deck cross structure, the prying moment is:

$$M_{yz} = M_o - Y\, T_{xz}$$

where Y is the vertical distance of the cross deck neutral axis from the center of gravity of the ship.

The yaw splitting moment and pitch torsional moment can be derived as:

$$M_{xz} = M_y|_{x-l} - l \cdot V_z|_{x-l}$$

$$M_{xy} = M_z|_{x-l} - l \cdot V_y|_{x-l} + Y \cdot V_{xz}$$

where My, Mz and Vy, Vz are lateral and vertical bending moments and shear forces, respectively, evaluated at the end of the ship where $x - l$ for one side of the ship.

The results of their wave induced load analysis may be applied directly to a Finite Element Model (FEM) as part of the strength assessment to be discussed later. In addition, the hydrodynamic forces, P, are used for the analysis of lower hull shell plating and framing scantlings as secondary loads. Finally, the analysis of specific sea conditions (wave height, period, etc.) combined with specific vessel operating conditions (speed, heading, etc.) are used to quantify RAO's used in the fatigue analysis.

Considering the very limited full scale data available for SWATH type vessels, the question is raised as to the accuracy and adequacy of analytical techniques to predict loads. This question is commonly answered by comparison of the analytical results with model test results. However, caution must be taken when such correlation is attempted to ensure that conditions are identical in both the analysis and the model test. If not properly taken into account, even slight differences may lead to erroneous conclusions and misleading correlation.

The correlation between the analytic results and the model test results for side force at the intersection of the deck cross structure and strut in beam seas is shown in Figure 5. The side force is non-dimensionalized by $\rho g A_p$ where A_p is the projected area of the ship's profile below the mean waterline. Figure 5 shows good correlation between the analytical and model test results for side force although the peak value is somewhat higher than the model test results.

Figure 5 - Side force at the cross structure strut in beam seas

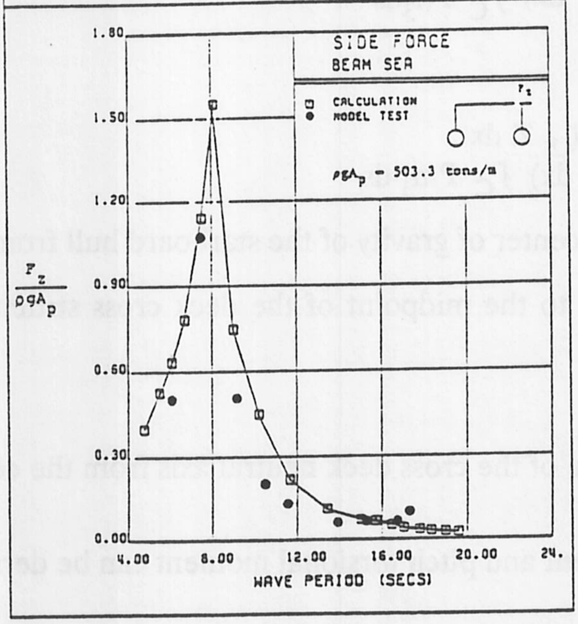


Figure 6 - Prying moment at the cross structure midpoint in beam seas

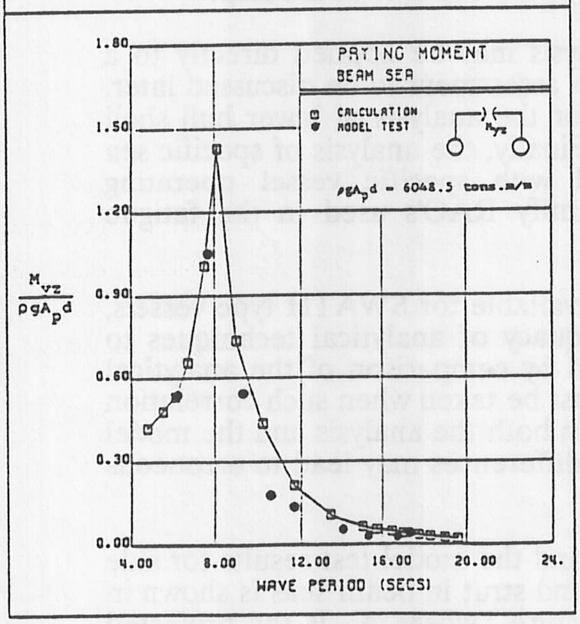
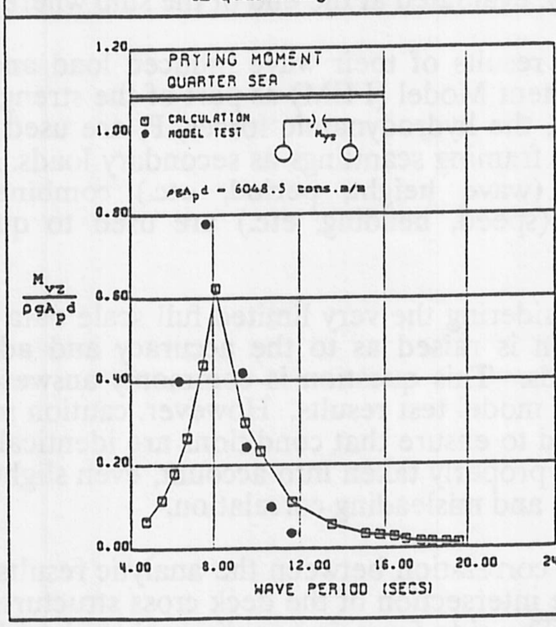


Figure 7 - Prying moment at the cross structure midpoint in quartering seas



The correlation results for the prying moment at the center of the deck cross structure in beam seas and bow quartering seas are given in Figures 6 and 7, respectively.

The prying moment is non-dimensionalized by $\rho g A_p d$ where d is the distance from the neutral axis of the deck cross structure to half the mean waterline. The prying moment correlation is also good, which is expected because the prying moment is primarily a function of the side force acting at the mid-draft with an equivalent arm from the deck cross structure.

ANALYTICAL DETERMINATION OF SECONDARY LOADS

The major Secondary Loads of concern for a SWATH vessel are the wave impact loads on the underside of the cross structure, the haunch and the strut sides.. Various attempts have been made to analytically predict these loads as discussed in several technical papers references [5], [8] and [9]. The confidence which can be placed on the results is open to question because of the almost total lack of full-scale test data and the wide scatter found in reported model test results.

The following method of analysis is a review of the ABS procedures which are based on existing, proven theory. The analysis has been specifically adapted for SWATH type hulls using the following assumptions:

1. the wave impact pressure is a result of linear superposition of the short-duration (approximately 0.05 second) impulse pressure that occurs when water impacts on the SWATH hull surface, and the slowly varying pressure that is caused by the post impact quasi-static head of water and the fluid particle velocity (see Figure 8);

2. the steep wave can be described by a Stoke's second-order wave which is not deformed due to the presence of the vessel;

3. random sea conditions can be represented by uni-directional wave spectrum;

4. linear seakeeping theory applies for SWATH vessels in a seaway;

5. no air is entrapped between the wave surface and the impact surface and no spraying occurs;

6. the impact surface is considered rigid, smooth and flat.

Given the above assumptions, equations may be formulated to assess the short-duration impulse component and the slowly varying pressure component.

Impact pressure is a time dependent function of the mass and the effective velocity of the water normal to a defined impact area. For computation, the mass of water is assumed to be contained within a finite volume related to the wetted surface of contact, and the relative impact velocity of the water surface to the impact surface. These relationships may be written as,

$$P_i/\gamma = \frac{\pi}{2g} V_n \frac{ds}{dt}$$

From this general equation, separate equations may be developed for specific locations as follows:

(a) underdeck centerline (head sea)

$$P_i/\gamma = \frac{\pi}{2g} V_n \left[\frac{C\text{-}V\cos\mu}{\cos\phi} + \frac{V_L}{\cos\phi} + V_n \cot\theta \right], \quad 0^o < |\phi| < 15^o$$

from horizontal

$$V_n = V_v \cos\phi \qquad ,\theta > 0^o$$

(b) midship outboard strut side (beam sea)

$$P_i/\gamma = \frac{\pi}{2g} V_n \left[\frac{V_v}{\cos\phi} + V_n \cot\theta \right], \quad 0^o < |\phi| < 15^o$$

from vertical

$$V_n = (C\text{-}V\cos\mu + v_t)\cos\phi \qquad ,\theta > 0^o$$

(c) midship underdeck centerline (beam sea)

$$P_i/\gamma = \frac{\pi}{2g} V_n \left[\frac{C\text{-}V\cos\mu}{\cos\phi} + \frac{V_t}{\cos\phi} + V_n \cot\theta \right], \quad 0^o < |\phi| < 15^o$$

from horizontal

$$V_n = V_v \cos\phi \qquad ,\theta > 0^o$$

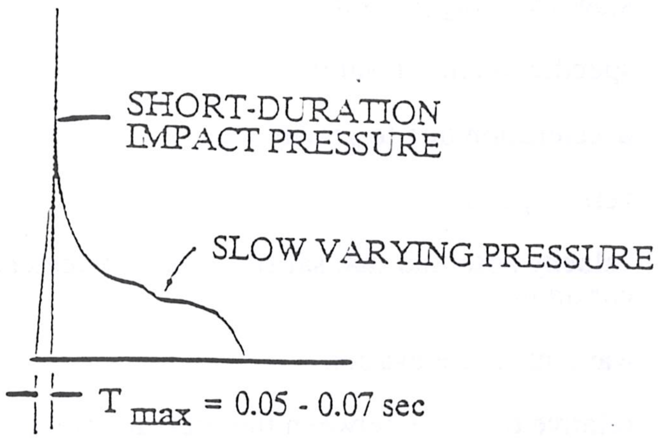
Where Pi is the impact pressure γ, the specific weight of waters, ϕ and θ are illustrated in Figure 9, C is wave celerity; V is the vessel speed, V_v is the relative vertical velocity; V_L and V_t are longitudinal and transverse velocities, respectively; μ is the wave heading with 180^d being the head sea.

In order to determine the slowly varying pressure, the solution of wave characteristics of the Stoke's second-order wave are substituted into the dynamic equation in its integrated form (Bernoulli's equation). That is

$$\frac{P_s}{\gamma} = -y - \frac{1}{g}\left[\left(\frac{\partial\phi}{\partial t} - V\frac{\partial\phi}{\partial x}\right) \right] - \frac{1}{2g}\left[\left(\frac{\partial\phi}{\partial x}\right)^2 + \left(\frac{\partial\phi}{\partial y}\right)^2 \right]$$

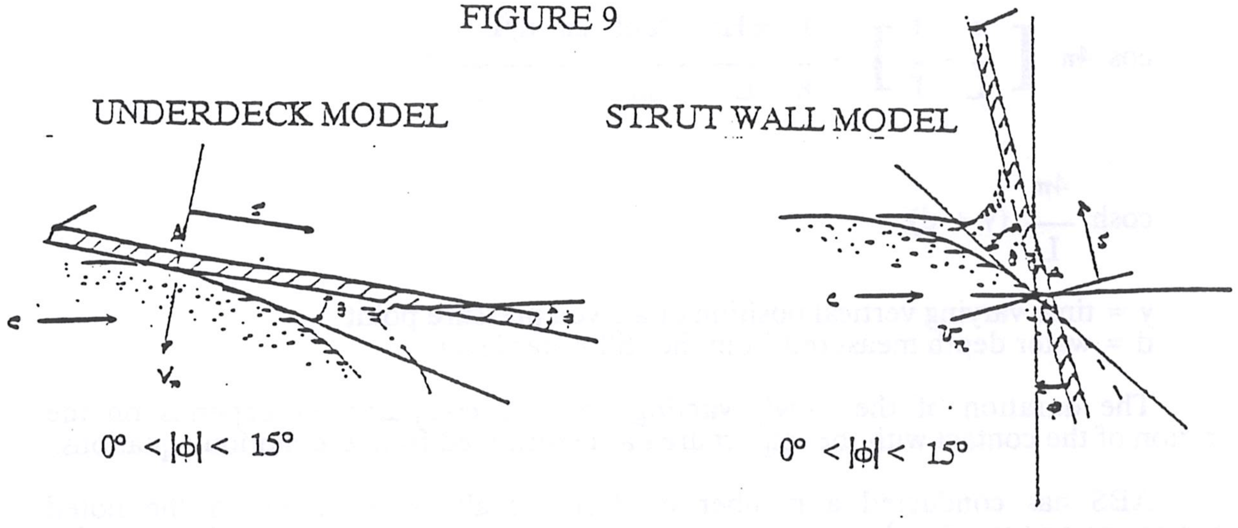
for $\eta > d$

FIGURE 8



TYPICAL WAVE IMPACT PRESSURE TIME HISTORY

FIGURE 9



PROGRESSIVE WAVE INCIDENT TO AN INCLINED FLAT SURFACE where,

P_s = slowly varying pressure

τ = specific weight of water

g = acceleration due to gravity

V = vessel speed

ϕ = velocity potential that satisfies the Laplace's equation of continuity

γ = wave surface elevation

d' = relative distance between the pressure point position on the hull and the mean water surface.

Thus it may be determined, as shown in reference (Wiegel), that

$$\frac{P_s}{\gamma} + y = \frac{H}{2}\,\frac{\cosh\ 2\pi\ (y + d)/L}{\cosh\ 2\pi\ d/L}\cos 2\pi \left[\frac{x}{L} - \frac{t}{T}\right] +$$

$$\frac{3}{8}\,\frac{\pi H^2}{L}\,\frac{\tanh\ 2\pi\ d/L}{\sinh^2\ 2\pi\ d/L}\left[\frac{\cosh\ 4\pi\ (y + d)/L}{\sinh^2\ 2\pi\ d/L} - \frac{1}{3}\right]\cdot$$

$$\cos\ 4\pi \left[\frac{x}{L} - \frac{t}{T}\right] - \frac{1}{8}\,\frac{\pi H^2}{L}\,\frac{\tanh\ 2\pi\ d/L}{\sinh^2\ 2\pi\ d/L}\cdot$$

$$\cosh \frac{4\pi}{L}(y + d)$$

y = time varying vertical position of a given pressure point
d = water depth measured from the still water level

The duration of the slowly varying pressure compartment depends on the duration of the contact with the impact area as determined from the motion equations.

ABS has conducted a number of design evaluations based on the noted assumptions and the developed equations. In order to deal with the complexity of this nonlinear problem, a design wave approach has been adapted. The following calculation procedure has been adapted for assessing wave impact leads on SWATH vessels:

1. calculate the maximum design wave for a given seastate based on a short-term extreme value approach.

$$H_{max} = H_s \left[(0.5 \, \text{Log}_e \, N)^{0.8} + 0.2886 \, (2 \, \text{Log}_e \, N)^{-0.8} \right]$$

where

H_{max} = maximum wave height in N waves
H_s = significant wave height
N = number of waves

2. calculate the associated wave period based on the steepness ratio (wave height/wave length) of 1/8, and the associated wave profile based on the Stoke's second-order wave.

3. calculate the corresponding design value of the absolute vertical and transverse vessel velocities by multiplying their RAO values at the calculated wave period and the design wave amplitude. However, the upper limit of these design values of absolute velocities are bounded by their short-term extreme values, which are calculated for the same sea condition by using the same approach as the maximum design wave formulation given above.

4. calculate the corresponding relative velocity by considering the proper phase angle between absolute body velocity and wave velocity. This phase angle is calculated by the linear seakeeping theory.

5. the time varying wave impact pressures are calculated for specific locations as shown previously, where a small time step is to be used. Finally, the average wave impact pressures are computed for the given impact area. They are obtained by averaging the wave impact pressures over the wetted area.

This computational procedure is applicable for the underdeck centerline region of the SWATH ship in head sea and for the midship outboard strut side of the SWATH ship in beam sea. However, the starboard beam sea case, for example, both short-duration impulsive and slowly varying pressure components are calculated on the outboard strut side, while only the slowly varying pressure component is calculated on the lower hulls, inboard strut sides and the port outboard strut side.

In summary, given the above assumptions and computational procedure, the time varying wave impact loads (average wave impact pressures) can be calculated across the underdeck of a fore end frame station, using the approximated transverse distribution of wave impact pressures. These wave impact loads can be applied in a structural analysis and subsequently, an assessment of structural adequacy of the underdeck cross structures for a critical head sea condition.

Similarly, for a critical beam sea condition, the wave impact loads can be calculated directly on the outboard strut side that faces the incoming wave, while the rest of the submerged hulls will take on a pressure due only to the slowly varying component. These wave impact loads can be used in a structural analysis and subsequently, as assessment of the structural adequacy of the strut structural arrangement at midship.

STRUCTURAL ANALYSIS OF SWATH VESSELS

At the present time, the structural analysis of SWATH vessels depends primarily on finite element method (FEM) analysis since there are not satisfactory data available to develop scantling formulas for a simplified analysis. Initial designs may be developed by scaling up from previous design or applying ship structural design methods or computer programs specifically developed for SWATH vessels based on model testing and previous FEM analyses. The American Bureau of Shipping (ABS) has experience in reviewing several independent SWATH vessel designs and has performed FEM analyses in order to gain experience. The ABS "Preliminary Guide for Building and Classing SWATH Vessels" has included appendix B "Guide for Hull Girder Strength for Concept Level Design" which provides a simplified method for determining the transverse bending moment, longitudinal bending moment, vertical shear force and wave impact. This concept level design method is derived from U. S. Navy model test data developed by the David Taylor Research Center (DTRC) [4].

From a review of the present SWATH designs, the transverse direction is considered the primary structural element. As SWATH vessels become larger, it may be shown that the longitudinal direction could share the primary loading with the transverse direction due to the effects of wave loading. The FEM analysis is based on three-dimensional (3D) coarse mesh analysis and a two-dimensional (2D) fine mesh analysis. The primary purpose of the three dimensional (3D) FEM analysis is to calculate the overall structural response and develop the loading distribution and displacement input for the two dimensional (2D) FEM fine mesh analysis of the transverse bulkheads and frames. A general analysis procedure of a SWATH vessel is as follows:

1. Determination of hydrodynamic loadings including wet deck slam due to wave spectra, heading, and vessel speed.

2. Determination of all estimated live loads, including deck loading, tank loading and any cargo loading or mission related loading conditions and inertial loads due to acceleration.

3. Determination of all dead loads and lightship weight distribution.

4. Develop coarse mesh 3D FEM model of entire vessel for initial analysis to determine overall response to applied loadings.

5. Develop fine mesh 2D FEM model of transverse frames and bulkheads to determine stress distribution and stress levels.

6. Review stress results and redesign structural elements as necessary and modify FEM model and rerun stress analysis.

7. Perform structural analysis with the wave slam loads and green sea loading on exposed decks as estimated from model tests or hydrodynamic analysis.

8. Perform fatigue analysis using S - N for actual life cycle.

9. Perform vibration analysis on final design.

10. Perform local analysis of structural components for strength capability and buckling analysis to Classification Society Rules.

The hydrodynamic loading analysis, has been previously discussed. The live loading would be given by an owner's design requirements or by the following recommended minimums:

Deck Loadings -	460 Kg/m^2	Crew Spaces
	920 Kg/m^2	Work Areas
	1,325 Kg/m^2	Storage Areas

Tank Loadings - Total weight of liquid in the tank with pressures distributed over area of tank.

Machinery and Equipments Loadings - Concentrated weight provided by the manufacturer.

The coarse mesh 3D FEM model is developed from the basic arrangements of decks, bulkheads and shell structure. The actual stiffness of the model is calculated from the individual membrane plate and beam elements. A typical model would consist of a lower hull section, strut section and half of the cross deck structure modeled about the centerline for symmetry considerations. The lower hull would have stiffened bending plates representing the outer hull plating and longitudinal stiffeners. The lower hull ring frames and transverse bulkheads that support the shell plating and stiffeners are represented by beam elements using an effective plate with the same structural properties of inertia and shear area as the actual structural element. The longitudinal and transverse plate structure above the lower hull are modeled using quadrilateral and triangle membrane plate elements. The transverse frames of the hull structure are lumped as beam elements with frame spacing as the effective plate width. Longitudinal girders and transverse bulkhead stiffeners are idealized as beam elements. The doors and archways in the transverse and longitudinal bulkheads are accounted for by reducing the plate element thickness to a value that would approximate the effective shear area across the section.

Since SWATH vessels are symmetric about the longitudinal centerline plane, only one side of the ship is modeled to reduce the number of node points, elements and solution time for the finite element analysis. Based on the principle of superposition, the symmetric and anti-symmetric loadings are computed to solve for the deflections and stresses on the port and starboard sides of the ship.

The FEM analysis is normally performed for the following loading conditions: (1) still water condition; (2) beam sea maximum prying side force; (3) beam sea maximum squeezing side force; (4) quartering sea maximum splitting moment. For load conditions (2) through (4), the dynamic pressures together with the appropriate vertical and lateral inertia forces are superimposed on the static still water loads to achieve total load equilibrium for the static finite element analysis. Figure 10 shows a isometric view of a finite element model of the SWATH ship.

It should be noted that the stress results obtained by the 3-D coarse mesh model should not be viewed as actual stresses expected in the structure due to the coarseness of the model in which averaging of plate thickness and use of equivalent thickness to account for the openings were made. The 3-D analysis provides only appropriate boundary conditions which are needed for the subsequent 2-D fine mesh analyses.

After reviewing all of the load case results, it is generally found that the critical load case is the maximum side force in a beam sea. The results show that for almost all transverse bulkheads, the high stress occurs in the transverse plating, between the main deck and second deck, outboard of a longitudinal bulkhead. An "arch way" opening or passage way may be present in the structure and the plate thickness in the 3-D model is reduced to account for the loss in shear area. It should be noted that the stress value obtained from 3-D coarse mesh analysis may not represent the actual level of stresses in the structure; however, this local area may warrant further investigation using 2-D fine mesh FEM. Typical results of the 3-D structural response in terms of deflections are presented in Figures 12 through 15 for the beam sea condition of a high sea state loading. On each section, the shaded area indicates the location where high stress levels occur.

In order to determine the actual local stresses in critical locations, a 2-D fine mesh FEM is performed for a typical transverse bulkhead and web frame. The models are extended from the lower hull into the struts and cross structure. Membrane plate elements are used to model the plate panels. The local stiffeners and effective width of the side shell are modeled as equivalent bar elements with axial stiffness.

The steel weight, equipment weight, etc., are distributed at various levels of the model. Boundary conditions in the form of deflections from the 3-D FEM analysis are applied at the appropriate locations.

Typical 2-D FEM analyses are performed in way of openings in transverse bulkheads, connecting areas between the strut and tapered hatch section and between the strut and lower hull to determine the actual stress levels for the maximum loading condition determined from the 3-D FEM analysis. Figure 11 shows typical 2-D FEM models.

The results of the FEM analysis are generally given in principle stresses as well as the combined stresses, known as Henky-Von Mises stress. Typical permissible Henky-Von Mises stresses for FEM analysis including all loadings may be as high as 85 percent of the yield stress (Fy) of the material. Additional permissible bending stress, axial stress and shear stress are provided by ABS in the "Guide for Building and Classing SWATH Vessels" and are as follows:

Static Loading Axial or Bending Stress =	Fy/1.67
Static Loading Shear Stress =	Fy/2.50
Dynamic Loading Axial or Bending Stress =	Fy/1.25
Dynamic Loading Shear Stress =	Fy/1.88

It is interesting to note that the local load strength analysis such as tank bulkhead stresses and deck loading stresses may govern over the finite element stress results. Therefore, all structural elements are to be analyzed for local loading as well as FEM analysis and checked for permissible stress levels.

FIGURE 10
ISOMETRIC AND FRONT VIEW OF 3-D FEM MODEL

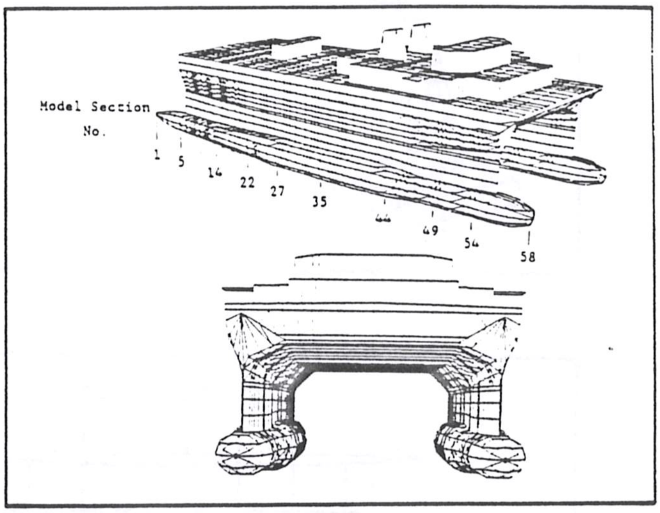


FIGURE 11
2-D FEM TRANSVERSE FRAME MODEL WITH FINE MESH

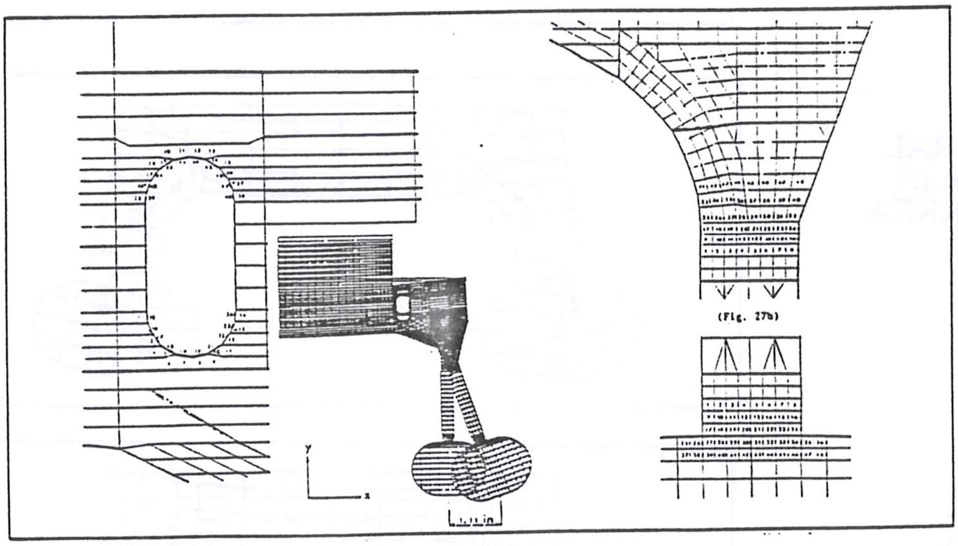


FIGURE 12
STRUCTURAL
RESPONSE AT
FORWARD END

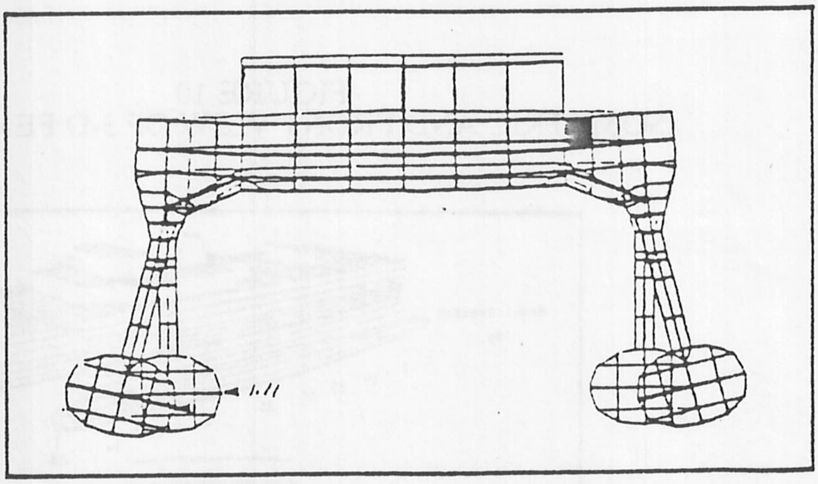


FIGURE 13
STRUCTURE
RESPONSE IN
WAY OF FORWARD
DECK HOUSE

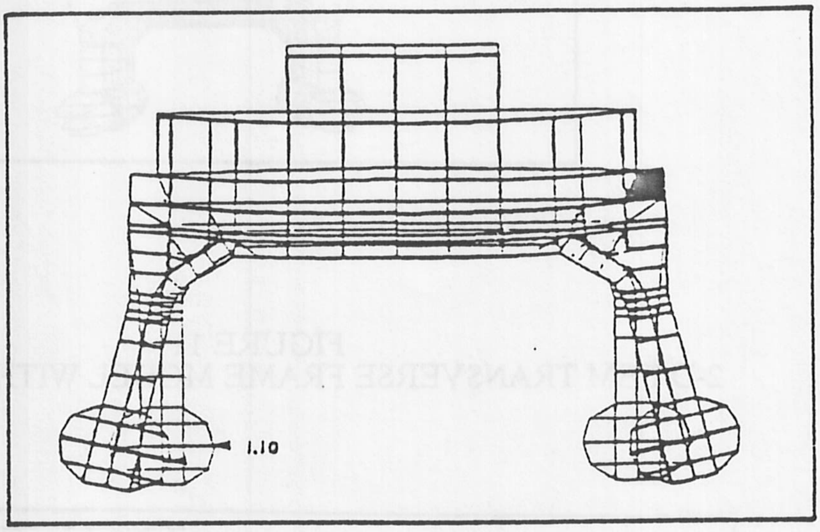


FIGURE 14
STRUCTURAL
RESPONSE IN
MIDSHIP AREA

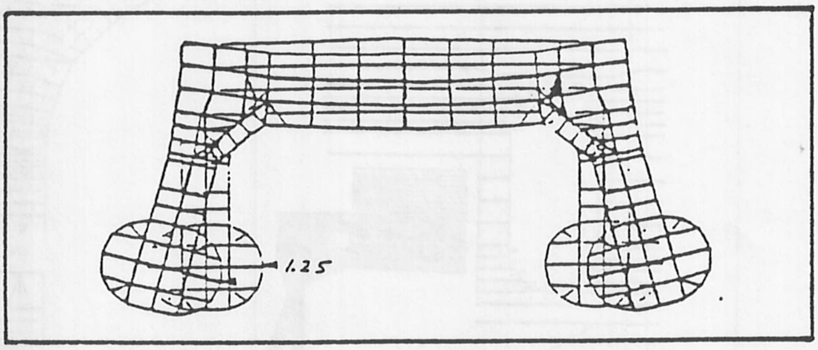
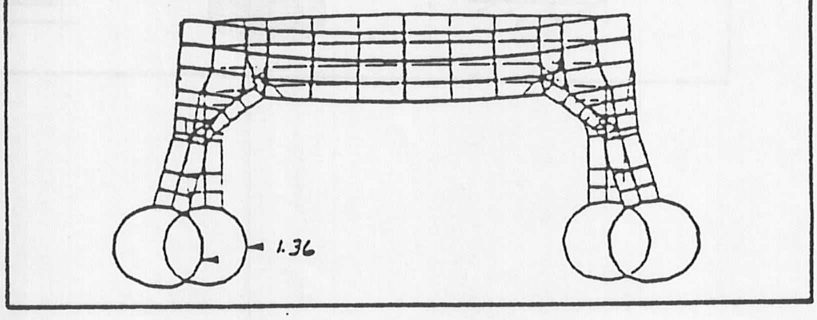


FIGURE 15
STRUCTURAL
RESPONSE AT
AFT END



Since high tensile steels are often used for SWATH vessels, it is important to assess buckling strength and fatigue strength. Buckling strength may be determined from local loads or from the results of the FEM analysis. The "ABS Guide for SWATH Vessels" has incorporated the plate buckling requirements for compression as well as shear. The fatigue analysis is performed to determine whether the structure has adequate capability to resist the effects of cyclic loadings that are likely to be present during its lifetime. The spectral fatigue assessment includes the following procedure:

a. Describe the long term environment,
b. Calculate motions, sealoads and local hydrodynamic pressures,
c. Calculate structural response and stress transfer functions,
d. Select appropriate S-N curves and stress concentration factors,
e. Estimate fatigue life and compare with acceptance criteria.

The wave environment to be used in the fatigue assessment depends on the anticipated areas of operation with the North Atlantic wave environment used for unrestricted service. The finite element analysis method of fatigue analysis is generally used for SWATH vessels due to the complexity of the hull form. The FEM fatigue analysis is performed in a two step approach (similar to structural analysis) consisting of a coarse mesh analysis followed by a finer mesh analysis of local structural areas of interest. The transfer function stresses causing fatigue need to be determined over a range of relevant frequencies and headings. The selection of the Stress-Life or S-N curves is the next step in the fatigue analysis. The S-N curves represent the fatigue performance of structural details under constant amplitude loadings. There are many compilations of S-N curves such as the U.K. Department of Energy, The Welding Institute, the University of Illinois Fatigue Bank, the U.S. Navy. The S-N curves should be selected based on the survival probability of that curve. The estimation of the fatigue life is determined using the stress transfer function and the wave energy spectra. The mean square value of the fatigue stress for each seastate is determined. ABS uses a computer program called ABS/SWIFT (for SWATH Integrated Fatigue) system which calculates the entire range of S-N curves for any given finite element. The final assessment of the fatigue life is based on an appropriate preselected valves such as 20-year target life. If is important to note that the actual workmanship of the shipyard in fabricating the structural detail should be considered in the selection of the weld joint for fatigue purposes.

Similarly, the FEM model may also be used to determine the structural vibration from a particular known vibration source. ABS has performed vibration analysis on many ship structures to determine the structural reinforcements necessary to dampen the structure from harmful structural vibrations. The structural vibrations of a SWATH vessel may be due to the excitations from the main engines as well as from the vibration of the hulls in a particular operating condition. Hull vibration due to wave action may be determined during model testing.

The complete structural analysis of a SWATH vessel will ensure the designer and owner that the vessel will be able to perform the mission requirements of the vessel.

SWATH APPLICATIONS

The SWATH vessel's superior seakeeping capability and operational flexibility is ideally suited to many commercial and military applications. The virtues of the SWATH can be summarized as follows:

Superior seakeeping performance in waves
Less speed loss in waves
Greater deck space
Greater cargo handling and working efficiency
Excellent low speed maneuvering and course keeping

The limitations of the SWATH are related to the weight sensitivity and slower speed in calm seas due to the greater wetted surface area when compared with a similar displacement monohull. It is interesting to note that the twin strut design SWATH has a smaller turning circle than that of the single strut. ABS has recently been involved with a USCG-design SWATH patrol vessel, U.S. Navy T-AGOS 19 Oceanographic research vessel, Pacific Marines twin-strut SWATH Passenger vessel, and the U.S. Navy T-AGOS 23 Oceanographic research vessel design. Three of the above mentioned SWATH designs are being built to ABS classification.

In the commercial vessel fleets, SWATH designs have been considered and/or developed for offshore crew and supply vessels, ferry boats, fishing boats, rescue-patrol vessels, passenger vessels, offshore drill ship and scientific oceanographic research vessels.

The large stable platforms which are capable of operation in relatively high sea states permit the owner to operate without reducing the vessel's speed. The cruise trade is seen as an area of major interest since the passenger will be less likely to experience sea sickness that would be typical on a monohull vessel. The large open deck space and reduced ship motions will be a major asset to scientific vessels, as well as fishing vessels.

In the government and military vessel fleets, SWATH designs have been considered for aircraft carriers, frigates, patrol vessels and Oceanographic Research vessels. The SWATH air craft carriers envisioned would be designed for helicopter and vertical and short take off and landing aircraft. The limited ship motions would permit aircraft operations in relatively high sea states. The SWATH frigate design offers a vessel that has a large stable platform for weapons and combat systems that will have no problem in keeping the speed necessary to stay with the fleet. Additionally, the SWATH frigates, patrol vessels and oceanographic vessels will have the capability to perform anti-submarine operations in relatively high sea states. The location of power plants, generators and major machinery components on the cross deck structures help to isolate the noise signature of the vessel from sonar measurements. Oceanographic side scan survey operations performed on a SWATH vessel will be more accurate than from a similar monohull due to ship motions. Government agencies are presently considering SWATH vessels for dredging work, customs patrols, buoy tenders, Coast Guard operations, border patrols and coast and geodetic survey. The large deck area and stable platform attributes of a SWATH make it an idea platform for helicopter operations which provide increased effectiveness for patrol-type vessel.

Further applications of the SWATH vessel will become inevitable after more experience has been gained with actual operating conditions. Several small SWATH pleasure vessels have been designed for the yachting market.

CONCLUSION

SWATH type vessels have been shown to be advantageous for operations in extreme sea conditions where a stable platform with large deck area is needed.

Hydrodynamic performance, sea way load estimation and strength assessment are the critical factors in the design of these unique vessels. Design processes and analysis tools have been developed which permits the confident development of proposed designs.

Based on the evaluation of four classes of SWATH type vessels and various design studies, the American Bureau of Shipping has developed a "Preliminary Guide for Building and Classing SWATH Vessels". This document provides designers with the necessary guidance for determination of loads and scantlings for a SWATH vessel design.

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