Critical Review of SWATH Vessels

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Abstract

This paper describes the critical review of SWATH vessels based on experiments and operating experience, in order to demonstrate the SWATH that would probably result to fully explore its market potentials for a variety of maritime applications covering commercial, military and private sectors.

Therefore, the inherent design characteristic of the SWATH is appropriately explained, concerning several fundamental items in the content of this critical review, showing the superiority of performance of the SWATH.

1. Preface

As well known, a SWATH vessel* has an unique hull shape which consists of a pair of submerged main hulls or lowerhulls, an above-water platform structure, and streamlined surface-piercing struts which connect lowerhull with the platform structure. (See Figs. 12 and 13)

Noteworthy technical items of the SWATH verified so far are given as follows:

(1) small motion even in rough seas,

(2) capability of keeping small wavemaking resistance from the medium up to the high speed range,

(3) small speed drop in waves,

(4) wide working deck, and

(5) easy workability, etc.

Therefore, SWATH vessels will be further designed and constructed from now in more broad ways. In this paper described is the critical review of SWATH vessels based on experiments and operating experience.

Even though the design technology established for conventional ships is applicable to design of the SWATH, it is important to understand the inherent design characteristic of the SWATH. As all aspects of performance are sensitively affected by a small variation in the value of any one design parameter, the overall optimization process is always required for design of the SWATH. Accordingly, outlines of several fundamental items for design are explained below; such as propulsion, fin automatic control, maneuverability, stability, trim stability, strength, power transmission system, and so on.

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*SWATH Vessel: Small Waterplane Area Twin Hulled Vessel

Received September 25, 1986

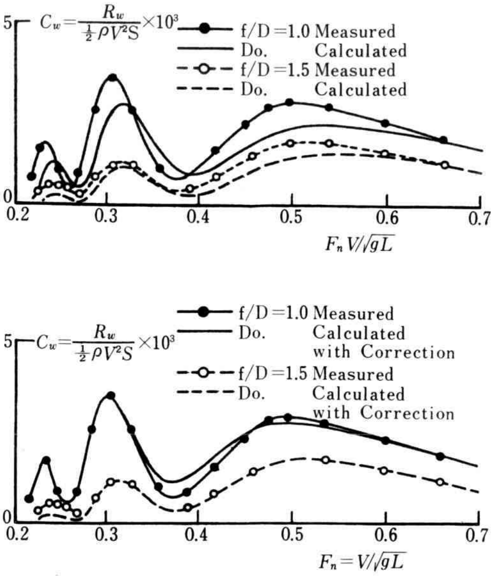

Fig. 1. Comparison of theoretical wave resistance of lowerhull with measured ones with and without correction

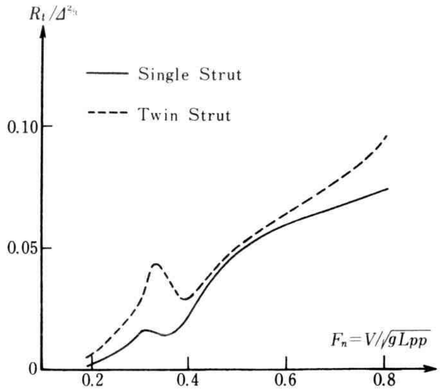

Fig. 2. Comparison of total resistance between twin strut and single strut SSC

2. Configuration

(1) Ideal lowerhull forms and struts

Using fundamental submerged models having several ratios of l/D (where l is length, D diameter), propulsion tests were carried out by changing ratio of f/D (where f is depth), in order to get the correction coefficients of phase rags and amplitudes, comparing with theoretical calculations for which Michell's approximation to struts and linear distribution Doublet's expression to submerged bodies are applied, respectively. These obtained correction coefficients are now introduced into theoretical equations to increase the accuracy due to theoretical predictions. In Fig. 1, a comparison of theoretical wave resistance of lowerhull with measured one with and without correction is given. This shows good coincidencies between measured data and calculated results with correction[1].

Propulsion tests of fundamental strut models were also carried out, and hence the difference of twin or single strut effect and interference between struts are obtained. A comparison of total resistance between twin strut and single strut SWATH is shown in Fig. 2[2].

As shown in Fig. 2, there exists a large hump around Froude number F_n 0.33 which was caused by wavemaking resistance of each strut in the case of twin strut. This phenomena are also examined theoretically. On the other hand, at the range beyond 0.6 of F_n the increase of resistance in the case of twin strut compared with single strut case was verified

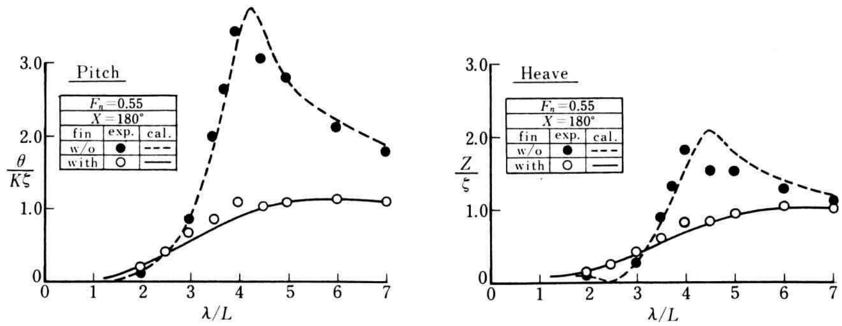

Fig. 3 Effect of fixed fins on pitch and heave at $F_n=0.55$

induced by added spray's resistance rather than the increase of wavemaking resistance[3].

In the case of a SWATH with the lower cruising speed ranging from 0.25 to 0.4 of F_n, optimal shape of the lowerhull changes drastically according to the design Froude number, and the end shape of the lowerhull tends to have larger bluntness and the midship section area of the lowerhull becomes smaller with the increase of design speed[4].

(2) Fin automatic control for maneuvering

The motion response of vessel affected with or without fins is discussed. As a SWATH has little wavemaking dumping force, there appears a large peak on response function at the resonance point. To decrease motion at the resonance point, it is effective to accommodate fins, especially when cruising, because of large dumping force according to the dynamic lift induced on fins. In Fig. 3 an effect of fixed fins on pitch and heave at 0.55 of F_n in head waves for twin strut is shown. A motion at the resonance point decreases well with fins in comparison with the case without fins. Therefore, the fin stabilizer system is quite useful for a SWATH to reduce pitching, rolling and heaving motions for the purpose of better comfort of the passengers[5~8].

The effect of fin control on motion decrement is now explained theoretically.

Linear equation of motion of a SWATH with six degree freedoms is given by[9].

$$(M+A)\ddot{X}+B\dot{X}+CX=F_e+F_l+F_f+F_c \tag{1}$$

where

M : mass or moment of inertia matrix

A : added mass or added moment of inertia matrix

B : dumping force coefficient matrix

C : stability force coefficient matrix

F_e : wave induced force vector

F_l : force vector by lifts

F_f : force vector by fixed fins

F_c : force vector by fin control

X : displacement vector of center of gravity of vessel, X_G, Y_G, Z_G, φ, ϑ, ψ

F_c of $Eq(1)$ is given as follows:
Controlled angle by fins, δ_c is given by

$$\delta_c = K_{zo}\ddot{Z}_G + K_{\varphi 0}\varphi + K_{\varphi 1}\dot{\varphi} + K_{\vartheta 0}\vartheta + K_{\vartheta 1}\dot{\vartheta} \tag{2}$$

where

$\ddot{Z}_G$: vertical acceleration at center of gravity

φ, $\dot{\varphi}$, ϑ, $\dot{\vartheta}$: angle and angle velocity of roll and pitch

K_{Z0}, $K_{\varphi 0}$, $K_{\varphi 1}$, $K_{\vartheta 0}$, $K_{\vartheta 1}$: controlled gain constant for feed back element due to be determined by simulation test or sea trial test

Then, F_c induced by fin control is given by

$$F_c = \frac{1}{2}\rho V^2 A_f C_{La}\,\delta_c \tag{3}$$

where

ρ : density of sea water

V : vessel's speed

A_f : projected area of fin

C_{La} : lift gradient of fin

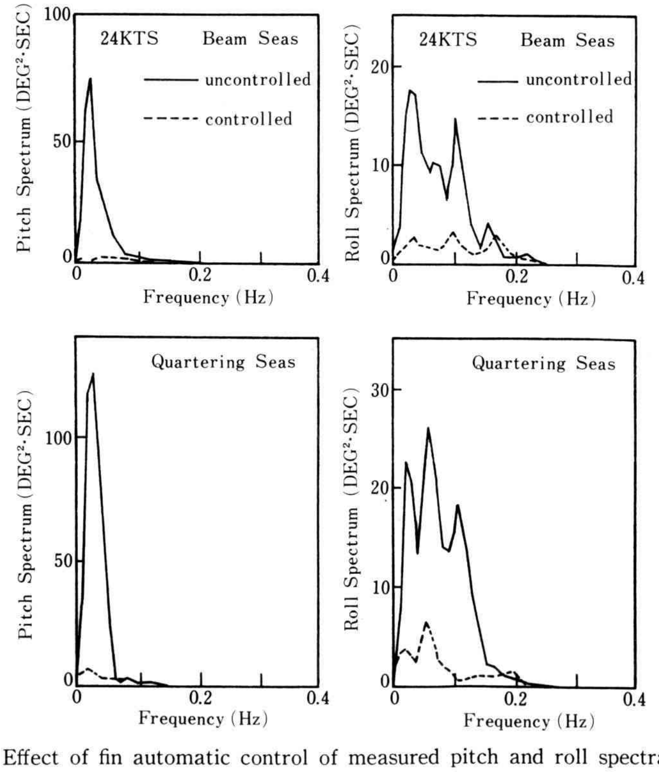

Fig. 4 Effect of fin automatic control of measured pitch and roll spectra at 24 knots for "SEAGULL"

If the acting point of fin lift exists at the distance, r_{xi}, r_{yi} and r_{zi} from the center of gravity of vessel, the force vector F_c of Eq (1) which indicates the force and moment around center of gravity of vessel, induced by fin control, is expressed by

$$F_c=\begin{bmatrix} 0 \\ 0 \\ \sum F_{Ci} \\ \sum r_{yi}\ F_{Ci} \\ -\sum r_{xi}\ F_{Ci} \end{bmatrix} = -A_C\ddot{X}-B_C\dot{X}-C_CX \tag{4}$$

F_C of Eq (4) is then introduced into Eq (1) directly, to predict easily the effect by fin control on motion decrement.

For theoretical prediction of motion used are irregular waves of the theoretical spectrum proposed at ISSC 1964, with the same frequencies of energy peak and the same effective wave heights as those of measured waves at sea trials. And $\cos^2 \chi$- distribution is also assumed as waves of the short wave length, based on calculated response functions in regular waves. This predicition was verified practical enough, indicating the good coincidencies between measured data at sea trials of both motion and accelerations. The measured motion spectrum at 24 knots for SEAGULL is shown in Fig. 4, putting encounter frequencies as abscissa for both uncontrolled and controlled cases, in which the effect by fin control on motion decrement significantly appears at the low frequencies range. The effect by fin control is also verified in comparison with the monohull vessel. For example, results of side-by-side running tests of SEAGULL and monohull in April 1980 demonstrated that the SWATH's superior seakeeping performance by only about 1/4 to 1/6 the roll angles and 1/3 to 1/7 the vertical accelerations at the bow, compared with those of the 35 meter high speed monohull. Hence, no one felt motion sickness on the SEAGULL.

Based on experiments the theoretical prediction of loads is then pursued.

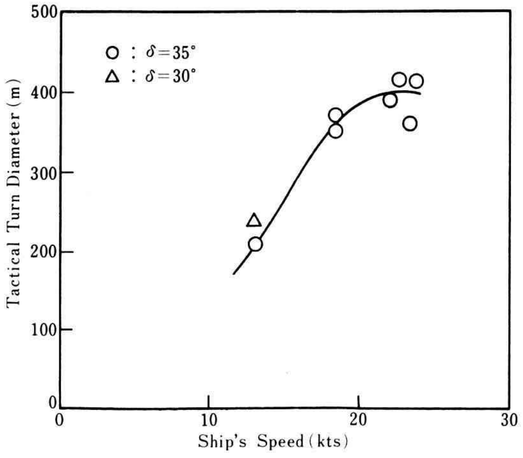

Fig. 5. Tactical turn diameter vs approach speed for "SEAGULL"

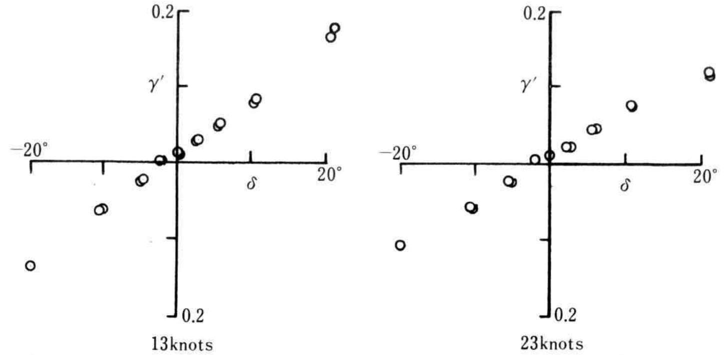

Fig. 6. δ-γ' Curves at low and high speed for "SEAGULL"

By simulation of loads which are measured, the effect of vessel's form, i.e., twin or single strut, and speed, draft, trim, heel etc. on tactical turn diameter is tested. In the case of SEAGULL, a tectical turn diameter changes largely according to approach speed as shown in Fig. 5, i.e., for example, that at 23 knots increases about twice as large as that at 13 knots.

At the range beyond 0.7 of F_n, sea trial tests are needed for maneuverability performance, because technically difficult conditions exist in modeling tests.

Seakeeping performance of a SWATH is superior at the low and high speed range as shown in Fig. 6, that is the relation between δ and r' obtained from spiral tests of SEAGULL. Having quick rudder responses, the spot turning is capable either by changing each thrust force of port or starboard propeller, or by rotating each propeller reversely[8,9].

(3) Stability

The flair shape at top of struts and the bulkhead arrangement are determined by the

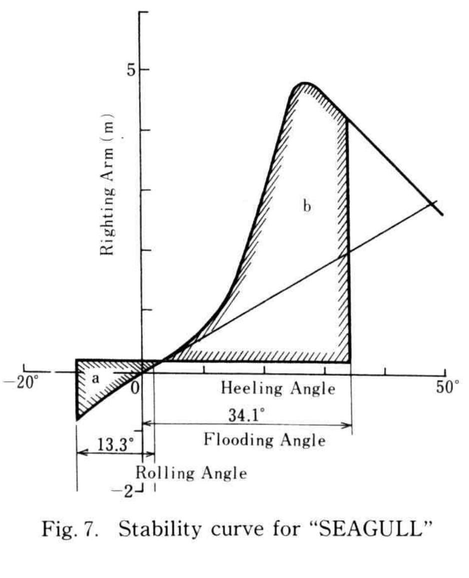

Fig. 7. Stability curve for "SEAGULL"

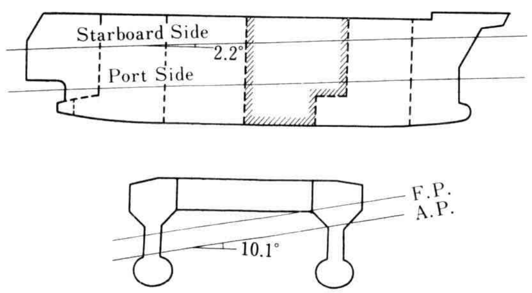

Fig. 8. Damaged stability of "SEAGULL" after one compartment flooding

stability under damaged or undamaged conditions, using stability calculation or undamaged conditions, using stability calculation program of ocean structures. Under undamaged condition, the initial slope of hull is large because of small stability force effect at even keel condition, but the stability force due to upper structure is extremely large compared with the monohull, and hence a SWATH is called a safety vessel with no capsize[2].

But in order to prevent abruptly induced large slope of hull by transferring heavy weights on the deck, hull structures should keep sufficient magnitudes of GM. Fig. 7 shows the righting arm of SEAGULL from rolling angle 13.3 degrees up to flooding angle 34.1 degrees at wind speed 55 knots. As shown in Fig. 7 at the small initial slope, the increment of righting arm is also small, but it increases rapidly from heeling angle 14.2 degrees where the upper structure touches on the water surface, and reaches the maximum value at about 27 degrees, even further at 34.1 degrees flooding moment, it still keeps enough large magnitude of righting arm. The ratio of area b/a is much larger than required 1.0, by which large enough stability is verified.

Under damaged such as at the collision, the inclination of vessel is examined. Damaged stability of SEAGULL is calculated when one compartment flooded without counter flooding, the result of which is shown in Fig. 8. Fig. 8 shows that after flooding the wateline between port and starboard sides is small enough as 10.1 degrees inclination and it doesn't reach at the deck. After the counter flooding, the inclination generally becomes small, hence, dangerous flooding of sea water onto the deck never occur[10].

3. Current SWATH Vessels and their Operating Experience

The U.S. navy began a SWATH program in the late 1960 and constructed the 190 ton SSP KAIMALINO as a work boat[11].

The SSP KAIMALINO has been used for demonstrations to show its potential applications for various military and Coast Guard missions and has indicated real capabilities for future applications, such as V/STOL air carriers and coastal and offshore patrol cutters[12,13].

Mitsui Shipbuilding Co. has also constructed various types of the SWATH, i.e., the experimental craft MARINE ACE, the high speed passenger ferry SEAGULL, the hydrographic survey vessel KOTOZAKI and other SWATH type pleasure boats. And now the support vessel for underwater work and experiments KAIYO was constructed. But the

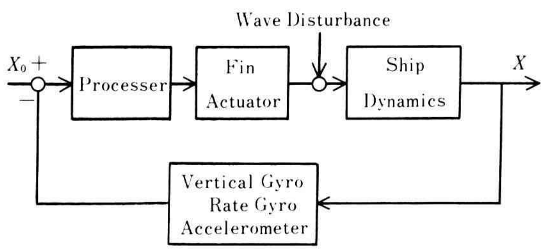

Fig. 9. Block diagram for fin control

Table 1. Main particulars of SWATH vessels

Ship name	Kaimalino	Marine Ace	Seagull	Kotozaki	Ohtori	Suave Lino	Kaiyo
Kind of ship	Work& Demonstration	Experimental Craft	Passenger Ferry	Hydrographic Survey	Hydrographic Survey	Fishing & Demonstration	Support Ship
Completion	1973	Oct. 1977	Sep. 1979	Dec. 1980	Dec. 1980	1973	Oct. 1984
Length O.A. (m)	26.8	12.35	35.9	27.0	27.0	19.2	60.0
Length P.P. (m)	23.5	11.0	31.5	25.0	24.0	16.8	53.0
Breadth O.A. (m)	13.7	6.5	17.1	12.5	12.5	9.1	28.0
Depth (m)		2.7	5.85	4.6	5.1		10.6
Draft (m)	4.66	1.55	3.15	3.2	3.4	2.13	6.3
Displacement (MT)	193/224	18.4/22.2	343	236	239	53	
Payload (complement)		(20 personnel)	402-466 Pass.	abt. 36 tons	(20 personnel)	abt. 14 tons	abt. 860 tons
Strut type	Twin	Twin/Single	Single	Single	Single	Single	Single
Hull material	Hybrid	All AL	All AL	Hybrid	Steel (Hull) & AL (Deck House)	All AL	All Steel
Propulsion system	Gas turbine & Chain drive 2250 PS×2	Gasoline & Bevel gear 200 PS×2	Diesel & Bevel gear 4050PS×2	Diesel & Bevel gear 1900 PS×2	Diesel & Bevel gear 1900 PS×2	Diesel & Bevel gear 425 PS×2	Diesel/Electric D/G 1250 kw×4 Motor 850 kw×4
Propeller	C.P.P.	F.P.P.	F.P.P.	C.P.P.	C.P.P.	F.P.P.	C.P.P.
Fin control	Automatic	Automatic	Automatic	Manual	Manual	Automatic	Manual

number of SWATH vessels is still few excluding some small prototypes of less than 10 meters long. The main particulars of several constructed SWATHs are tabulated in Table 1.

Fig. 9 shows a block diagram of the fin control system installed on MARINE ACE and SEAGULL. Ship motion and accelerations sensed by means of gyros and accelerometers on board are fed into a computer, then the fin actuators are independently driven in accordance with each command signals from the computer. This feed back system is also useful in keeping the ship's trim and keel favorable when turning and changing speed drastically.

By fin stabilizer system, the significant pitch and roll motions measured in Sea State 4 with a 2.4 meter significant wave height at 24 knots were less than 1.5 degrees for all headings. Measured vertical accelerations were less than 0.1 g in the same conditions as above.

For the development of the SWATH, the completion of the 2,800 gross-ton SWATH KAIYO is really an epoch-making event. It means not only enlarging the size from several hundred to several thousand tons but also entering a new area of medium or low speed type SWATH design from the design of high speed type SWATH vessels. Sea trials have been conducted with respect to the items of ship performance tests and equipment tests. Ship performance tests to investigate powering, seakeeping, maneuvering and fin control response are described below.

The maximum speed recorded was about 14 knots and sufficient power margin at a cruising speed of 12 knots was comfirmed. KAIYO showed excellent maneuverability at the full range of speed up to the maximum speed with tactical diameter of approximately 5 times of the ship length as well as good course keeping ability. As for seakeeping, it was reported that the ship motions were extremely small where she was running in Sea State 3 to 4 of significant wave heights 1.25 to 2.5 meters. She was easy to maneuver by using quick and smooth response of the controllable pitch propellers and differential thrusts between port and starboard propellers. Trim and heel adjustment by manually operated fore fins was reported very effective and useful during the operation.

The excellent performance of KAIYO was therefore verified by sea trials, as being a stable offshore base handling various computer aided equipments in order to get accurate data on board[14,15].

For the exploitation of sea bed oil, an offshore crane vessel MICOPERI 7000 having extraordinary lifting capacity by two 7,000 ton rotating cranes installed aboard is under construction at the Trieste's Fincantieri Shipyards. After completed in early 1987, the vessel will be the largest of its kind in the world. The main characteristics of this vessel are 175 m $\times$ 87 m platform, height 43.5 m, trial speed 9.5 knots, maximum lifting capacity of two 7,000 ton totating cranes and automatically controlled dynamic positioning system[16].

4. Loads

Before discussing strength analyses of the SWATH structure, we shall explain how to

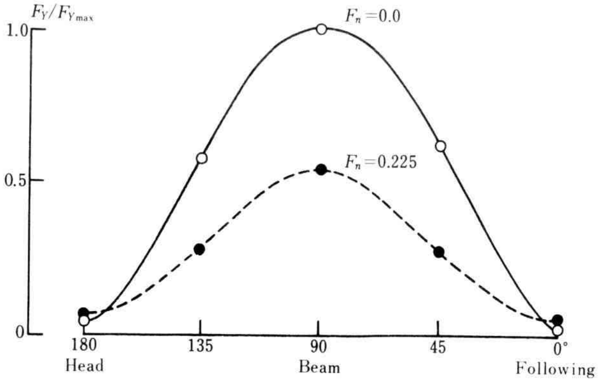

Fig. 10. Side force in regular waves

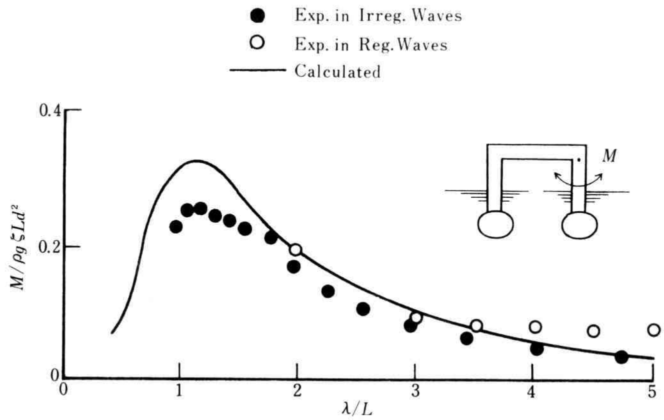

Fig. 11. Bending moment in beam waves

predict the wave loads first[2].

As the wave loads are there horizontal side force F_Y, shearing force F_V, lengthwise force F_X, vertical bending moment M_B, torsional moment M_T and horizontal bending moment M_H, but others except F_Y and M_B are compared as small from the results of tank tests.

Measurement of F_Y and M_B in regular waves, therefore, were carried out under the condition of different speed, wave length and wave direction. As shown in Fig. 10, horizontal side force F_Y reaches the maximum at $F_n=0$, decreasing with increase of Froude number in beam wave. It rapidly decreases as the wave direction changes from heading to following.

Bending moment in beam waves, M at the top flair connection of struts with upperhulls were compared between calculated results and model experiments in regular and irregular waves as shown in Fig. 11.

Although calculated results in irregular waves in Fig. 11 are the response obtained from the analyses of measured bending moment spectrum and wave spectrum, Fig. 11 shows good coincidencies between the response obtained by experiment in regular waves.

From this phenomena, we can obtain the response in any irregular wave by linear superimposition method as the case of monohull, and hence theoretical prediction of wave loads is also verified.

5. Structural Design and Analysis

(1) Design considerations

Fatigue

As explained in chapter 4 Loads, loading from beam seas of little concern for monohull designs, however, becomes of significant impact for a SWATH concept. The twin hulls would be flexed athwartship with the passage of waves. The likelihood of fatigue failure must be anticipated at lowerhull-strut, strut-sponson and sponson-cross structure welded joints as shown in Fig. 12.

Luedeke et al.[17] say that a SWATH craft of the HALCYON's size, for example, could expect to endure a maximum lifetime side force equivalent to 0.95 to 1.0 times its displacement value. For the HALCYON design, a conservative approach was taken, and a side load of 1.2 times its displacement was selected and applied at the centerline of the lowerhull.

Sikora et al.[18] present a method for predicting lifetime fatigue load spectra for monohull and SWATH. It investigates how conventional monohull ship designs would fare against the maximum lifetime expected vertical bending moments. Similarly, the maximum lifetime side force was predicted for single strut and tandem strut SWATHs. A design side force can be determined from the lifetime maximum loads by applying the ratio of design to predicted moment as determined from the monohulls, namely 0.73. For a mild steel SWATH to have comparable strength to the average monohull of the past thirty years, it should be designed so that 0.73 times the lifetime side force produces a single amplitude stress 8.5 tsi (130 MPa).

Unlike the monohulls which are affected disproportionately by whipping, it is assumed that the tensile and compressive stresses are equal for SWATHs.

Stress concentrations, etc.

Sikora et al.[18] say that as the proposed stresses for the design of a SWATH are for the nominal structure, at the areas of stress concentration, such as the strut-box intersection or changes in transverse bulkheads, the consideration should be given. It is possible that local loads due to slamming and machinery, etc., and minimum plating and scantling requirements

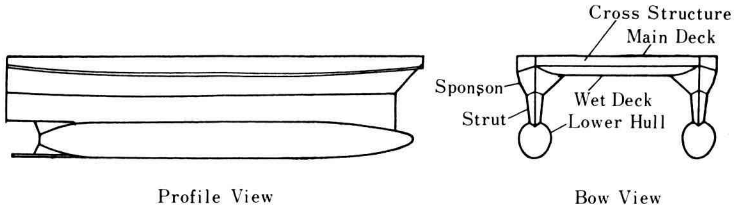

Fig. 12. HALCYON hull form

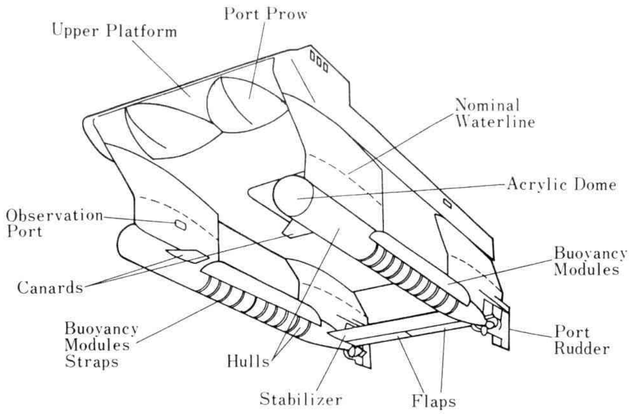

Fig. 13. SSP KAIMALINO hull form

for buckling may govern the design of the structure to the extent that the nominal stress does not even approach the recommended limits.

Damage

Hightower et al.[19] say that the periodic structural inspection was planned to evaluate the durability of the SSP design, of which hullform was shown in Fig. 13. The first was carried out in 1973, shortly after construction. This inspection formed the baseline for subsequent studies conducted in 1979 and 1983. At the time of the last inspection, the SSP had operated more than 5,000 hours at sea, with most of that experience accumulated in rough Hawaiian waters. The last inspection did not find any significant defects in the primary load-bearing structure.

A number of local discrepancies were defected, most of which were associated with defective welds. One recurrent fatigue crack was found at the intersection of the aft strut trailing edge and the upperhull. This was most probably caused by a focusing of propeller-induced vibrations. The addition of two small chocks on each strut has eliminated this problem.

During the 1983 survey, personnel of NSRDC discovered substantial damage to the framing of the port prow. This damage was caused by slamming and is thought to have occurred while the ship was stationkeeping in rough waters with an exceptionally heavy load aboard. Cross-structure clearance at the time was less than 4 ft. This area of the SSP bow was designed to yield under a gross loading of about 14 psi.

Based on experience gained through SSP operations and discussions with the crew, it appears that the design would have been adequate if sufficient cross-structure clearance had been maintained. Seakeeping studies conducted in 6.5- to 10.5-ft significant seas with 5-ft 11-inch clearance support this conclusion.

A second area which was found to have possible damage due to slamming was located in the wet deck under the diesel generator compartment. The deformations seen were small, not exceeding 1/4 inch, and therefore they may be associated with other causes such as welding distortion that occurred during construction.

The SSP wet deck is designed to yield under a uniform load of 2.8 psi and the crew reports that slams have occurred in this area when the SSP is heavily loaded. Again the design is considered adequate provided sufficient clearance is maintained.

Hightower et al.[11] say that the underside of the upperhull has a nominal clearance of 6 ft above the waterline which ultimately determine wave conditions that would provide unacceptable operating characteristics from the standpoint of slamming. Smooth transits have been made in swells of approximately 15 ft with no impacts at all ; yet in short steep waves we have, on occasion, experienced hard slams. In all cases to date we have been able to alleviate the problem by proper platform trimming. Altering course can also eliminate or reduce slamming ; however, to date this has not been required.

Following operations in heavy seas when slamming was encountered we have always made inspections of the upperhull structure and plating but have not detected evidence of structural damage.

Hightower et al.[19] say that shortly after the trials began, the SSP suffered a broken shaft in the port chain drive system. The problem was traced to the initiation of fatigue cracks at the sharp edges of key slots. The design was revised to minimize the effects of stress concentrations, and a modified chain drive was installed.

(2) Hull structure (upperhull, lowerhull, appendages)

Luedeke et al.[17] say that for the HALCYON a plating pressure of 20 psi was used to design specific plate and longitudinal stringer combination in the wet deck area forward of the knuckle in anticipation of possible slamming contacts.

Sidehull pressures were estimated at 7.8 psi based on a combination of side loads and V-line loads. Pressures on the wet deck aft of the knuckle were estimated at less than 5 psi for normal operations.

Primary hull bending, shear, and torsion loads for dry docking and hoisting of the craft were also determined. A load factor of 4 was applied to the lightship weight and was considered to be appropriate for both dry docking and hoisting operations. The HALCYON can be supported on keel blocks in dry dock and can be hoisted from four lifting points designed into its main deck structure.

Five interior transverse watertight bulkheads were provided that continued into the cylindrical lower sidehulls. Circular tee frames located at 36 inch on center stiffened the lowerhulls and were joined to the nontight frames of the struts. Inboard of the shell were three longitudinal watertight bulkheads located at the hull centerline and at 7.5 ft off centerline (port and starboard) at the inboard joint of the sponson and wet deck. Longitudinal and transverse framing were of increased thickness in the areas of machinery openings and at hull lift points.

Many of the design features are introduced in the HALCYON structure, in particular, to provide continuity at the joint interface such as the cross structure-sponson and sponson-strut joints.

Continuous, double sided filet welds were employed throughout to mitigate against fatigue failure and stress concentrations at weld bead terminations. Only the HALCYON's lightly loaded superstructure and pilothouse used conventional ship welding structure.

A major portion of the hullform was constructed using marine grade 5456-H-116 aluminum alloy plating and flatbars. This alloy provides the highest welded strength of any of the marine grade aluminum alloys, while at the same time retaining good weldability and corrosion resistance characteristics. Extrusions were of 5083-H-111 aluminum alloy. In a few areas where lower loads were anticipated, 5086-H-111 alumimum alloy was used. The HALCYON uses an "ELECTRO-GUARD" corrosion protection system. This is an automatically controlled cathodic protection system utilizing high purity aluminum alloy sacrificial anodes.

(3) Rigid vinyl modeling and other experiments

Lang and Sloggett[20] say in a case of the SSP design, one decision was to maximize the stability of the platform both at rest and underway. The multi-strut-per-side configuration shown in Fig. 3 with a full-span stabilizer was chosen for this reason.

Hay[21] says that to determine load criteria the SSP prototype craft and a scaled vinyl model were instrumented with strain, pressure and accelerometer transducers. Static load evaluations were conducted on both to obtain a basis for comparison of prototype and rigid vinyl model responses and to obtain load per unit strain relationships. Prototype and model static strain data indicate that for the lowerhulls and vertical struts agreement is good, for the upperhull, however, agreement is inconsistent and at best fair. It is believed that simplifications made in scaling the model structure were responsible for these inconsistencies. Load per unit strain sensitivities developed from the static load evaluations are applied to strain data collected during sea trials to establish a trials derived loads data base.

Estimates are then made of wave induced primary side loads for various sea states and headings. For a 20 year operational lifetime the maximum side loads are predicted to be approximately one-half the craft displacement. Agreement of the model and prototype based loads data is excellent and further validates rigid vinyl modeling as a design tool for estimating full-scale loads.

Luedeke et al.[17] say that in order to more fully explore the effects of transverse loading of the HALCYON's hull structure, a NASTRAN finite element model was developed. The model was representative of the hull structure between bulkheads at Stations 37 ft and 49 ft. This segment of the structure was selected for analysis because of the large opening in the

Table 2. HALCYON structural design safety factors

Condition	Operational	Emergency	V-Line
Yield Strength	1.15	1.00*	No Req't.
Ultimate Strength	1.50	1.20	1.20

* Local yielding is allowed.

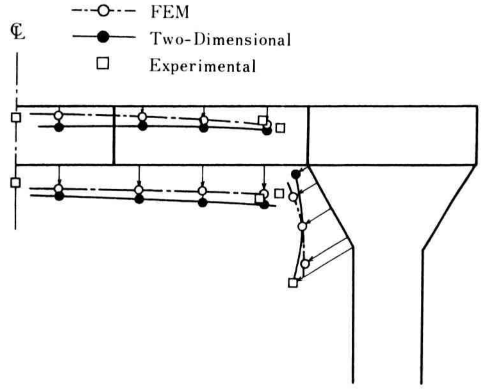

Fig. 14. Comparison of stress between measured data and theory for "MARINE ACE"

main deck for machinery. It was considered an area likely to cause the most severe internal stress distributions and concentrations. Given the HALCYON's beam sea loading assumptions, the average stress for the NASTRAN section was calculated to be $\pm 3{,}905$ psi which was well within the material allowables for the alloys selected.

This calculated stress provided a good basis for computing the total deck loads. Analysis indicated that relatively high stress peaking would exist at the transverse bulkheads adjacent to the deck cut out. It also showed that transverse loading resulted in compression in the wet deck that would result in higher stresses at frames and bulkheads since they would be carrying higher loads to take up the slack. Table 2 lists the safety factors used for the HALCYON structural analysis.

Oshima et al.[2] say that as preparation to discuss the strength, both analyses by FEM or others and load tests on land were carried out to get the strength response under given external load condition at the time of construction of MARINE ACE.

Under the horizontal force given by oil jackscrew, load tests between bending moment and stress response of the upper strut corner were conducted to compare with theoretical results by FEM and two-dimensional analyses. As the results shown in Fig. 14, FEM results have good coincidencies with experimental results. Afterwards, sea trials were carried out to get the data concerning strength response of structure for many different speeds, wave directions, turning circles by using the SEAGULL. Bending stress distribution in the vertical direction of struts by wave loads and pressure distribution on the surface of the submarged bodies are also measured.

Nagai[22] says that to evaluate the strength of wet deck or underside of the upperhull against hard slams, stiffened 5083 aluminum alloy plates were used for experiments with the help of FEM. Analyses of accumulated data indicate that the deck and underside structures stiffened with appropriate rigidities absorb the slam energy and their responses are well within elastic region.

Table 3. HALCYON Weight

Weight Group	Weight (LT)
100-Hull Structure	24.9
200-Propulsion Plant	10.0
300-Electric Plant	3.7
400-Command & Surveillance	0.4
500-Auxiliary Systems	3.8
600-Outfit & Furnishings	4.7
Lightship Weight :	46.5
F00-Variable Load	
Operating Fluids (lubricants, hydraulic fluid, coolant)	1.0
Payload (fuel, crew, provisions, fresh water, cargo/passengers)	9.5
Full Load Displacement :	* 57.0

* Full load displacement at 7 ft-0 Draft.

Table 4. Comparison of weight and manufacturing costs

Hull Material	Hull Weight Index	Cost Index
All High Tensile Steel	1.0	1.0
All Aluminum Alloy	0.55	3.1
H.T. Steel and Al-Alloy	0.70	2.1

6. Materials and Lightweight Structural Concepts

To increase the payload capability of SWATH, it is necessary to lighten the hull structural weight by selecting a suitable hull material and a proper structural arrangement based on overall strength analyses. Hull materials have to be selected from the viewpoint of cost, strength, weight and manufacturing capability to satisfy basic design reqirements such as payload and speed.

Hightower et al.[19] say that in the SSP KAIMALINO we selected a hybrid materials approach : aluminum upperhull, steel struts and steel lowerhulls. The primary structural joint between the steel and aluminum was made using Dataclad, Dupoint's explosion-bonded steel-and-aluminum plate, with no signs of structural failure.

On the other hand, Luedeke et al.[17] say that the all-welded, all-aluminum hull structure of the HALCYON is unique in terms of its design and construction and reprensents a departure from conventional boat building practice. Table 3 shows the weight breakdown of the HALCYON.

Mabuchi et al.[14] show in Table 4 a comparison of weight and manufacturing costs for a 400 ton SWATH design using three different hull materials, high tensile steel, aluminum alloy, and a combination of high tensile steel for the lowerhulls and struts and aluminum alloy for the deck structure.

In the case of MARINE ACE and SEAGULL, anti-corrosive aluminum alloy was used as the hull materials to reduce the structural weight and therefore use less power to attain the design speed.

According to the design experience so far, the anticorrosive aluminum alloy or much lighter structural materials is usually adopted for a high speed, small size SWATH to reduce hull weight in spite of its high cost.

As the hull material of KOTOZAKI, three alternatives were considered, i.e., Al-alloy, high tensile steel and a hybrid structure. The adoption of aluminum alloy for the entire hull showed an unacceptably high building cost. Also for required payload and design speed of 20 knots a high tensile steel structure needs much bigger engines. Therefore, a hybrid structure with the main hull made of high tensile steel and the deck parts of anti-corrosive aluminum alloy were chosen to allow a reasonable design including the size of engine and a good performance cost figure.

Narita et al.[14,15] say that on the other hand, in the case of KAIYO designed as an ocean going vessel, steel was selected to satisfy the requirements of low cost and good hull maintenance. High tensile steel was used for bottom and side shell plates as well as the longitudinal members of lowerhull and the transverse webs below the upper deck, while the deck house was made of mild steel.

7. Fabrication

Fabrication methods of two different kinds of hull materials such as a case of SSP KAIMALINO[19], say, a hybrid material approach and an another case of HALCYON[17] constructed by only aluminum are explained.

(1) Fabrication method of SSP KAIMALINO[19]

The process of construction was highly modularized, with various assemblies constructed in many separate areas of the yard, utilizing conventional shipyard fabrication technology. The lower hulls were fabricated as modules from high-tensile-strength steel (50,000 psi yield) in 8-ft sections on buoy assembly machines. These sections were aligned and joined along with the tailcones on the floor of the metal fabrication shop.

The struts as modules from high-tensile-strength steel (50,000 psi yield) were built in halves in female molds, with the two halves of each of the four struts mated and transported to an outside assembly site.

Here the lower hulls and struts were fitted together and welded. When these sub-units were completed, the strut/lower hull assemblies and the stabilizer were transported to the yard's large floating drydock.

The upper hull fabricated as a single module from 5086 aluminum was under construction in four sections in another area of the fabrication shop.

As each hull section was completed, it was moved to a second outside assembly area, where these sections were joined. The completed upper hull was then transported to the floating drydock for joining with the two strut/lower hull assemblies. The upper hull was positioned by crane over the struts in order to allow the tops of each strut to be scribed and cut to the contour of the Dataclad, which had been previously attached to the upper hull. With the struts prepared, the upper hull was positioned and welded with explosively-bonded dataclad strips of steel and aluminum alloy in place.

Hull penetrations for the various control surfaces and propeller shafts were plugged to allow the hull to be floated out of drydock to a pierside area, where outfitting could continue without occupying valuable dry-dock space.

Several months later, after the SSP had been partially outfitted at pierside, the vessel reentered the drydock for installation of propellers, control surfaces, fuel bladders, and verious hull-mounted sensors. In March 1973 the SSP was christened and launched. Outfitting and installation of main engines continued at pierside, with construction completed in the fall of 1973.

No unusual problems were encountered during the construction phase, although the hullform was unique.

(2) Fabrication method of HALCYON[17)]

Careful planning went into material selection, joint fit-up, weld sequencing, module sizing and tooling needs of the HALCYON. The HALCYON's hull was built in seven sections. These were two sidehull modules (port and starboard) consisting of a lower hull, strut and sponson, and five transverse modules making up its cross-structure.

The sidehulls were first erected (in a laid down position) on dual platen tables. They were then raised, aligned, and brased to receive the cross structure modules. Three cross-structure modules were also fabricated on a platen table. Each was partially erected. Only the wet deck, transverse and longitudinal bulkheads and framing were installed. As each was completed, they were raised, positioned, and joined to the sidehulls. Main deckplating was added once they had been welded in place. The remaining two cross-structure sections (bow and stern) were built in place spanning the sidehulls. The pilothouse and deckhouse structure were then constructed as a complete module and installed as a unit, to complete the HALCYON hull. Piping systems, for example, were installed early in the erection process before modules were closed out to minimize the number of connections needed. Although the HALCYON's hull is designed to be accessible, the installatjon of long pipe runs would not be possible with the deck and shell in place.

The hull design required that all welds be double sided and continuous, a close tolerance joint fit-up was found to be important. Gaps in the fit-up were held within 0.05 In. thus ensuring minimum distortion. Proper weld sequencing played an important role. Plate was allowed to move by controlling the weld direction, as well as, starts and stops, thereby reducing to a minimum, the residual stress buildup in completed sections.

Tooling aids, such as templates and fixtures, played an important role during the construction of the HALCYON. They proved valuable for assuring repeatability and close tolerance fit-up. Form core, for example, was used to template directly off the structure where contours had to be matched.

During the construction phase, the structural details were considered to be designed to ease fabrication and to eliminate potential crack problems.

The only remaining items yet to be fabricated are the rigid glass reinforced plastic sandwich foam covers for the propulsion engines and diesel generator sets, and the boat's accommodation ladder. Each engine cover will contain panels for routine maintenance access, observation windows to monitor local instrument panels, and forced ventilation systems.

8. Propulsion Systems

Propulsion system for the SWATH is required to be carefully selected from various aspects such as weight, size and space, cost, reliability and so on[14].

In order to achieve a commercially acceptable economical SWATH, the high speed diesel engines were preferred to the gas turbines in spite of the heavier weight and larger size because of the lower fuel cost and the lower maintenance cost.

In the case of a relatively small SWATH, the main engines are usually located on the upper deck due to the lack of sufficient space in the lowerhull. Therefore, reliable power transmission system with high efficiency is required in order to drive the propellers placed in the aft end of the lowerhulls.

Three mechanical transmission systems such as a bevel gear system, a silent chain system and a hydraulic system were evaluated for use in a 400 ton SWATH at Mitsui as summarized in Table 5.

Then a Z-drive system with two bevel gears was adopted because of higher efficiency, lighter weight and lower cost than other systems. The bevel gear type power transmission system with twin vertical shafts was installed on SEAGULL.

In future, it will be recommended to use smaller reduction gear system rather than the solar gear system.

For a relatively large SWATH, the electric propulsion system is recommended from the

Table 5. Comparison of typical mechanical transmission system

System	Bevel Gear System	Silent Chain System	Hydraulic System
Efficiency (%)	97-98	97-98	75-80
Max delivered power (PS)	10,000	3,700	2,000
Dia of casing (m/m) (in case of 2,000 PS)	800	1,200	1,500
Reverse	N.P.	N.P.	Possible
Noise	Low	Medium	High
Vibration	Low	Medium	High

viewpoint of reasonable arrangements for transmission between engine and propeller, or of less noise and vibration.

For example, for the support of underwater work and experiments in the sea and on seabeds, it is needed to minimize underwater noise because of efficient work of underwater transmission and audio measurements. The arrangements of engines on the deck are recommended, and skewed blades are used both for the main propellers and the impellers of side-thrusters.

In future, the large diesel-electric propulsion system being used so far will be replaced by the superconduction system from the viewpoint of lighter weight and smaller space and lower cost than now, for a relatively small SWATH an application of much effective electric-driven system will be taken into consideration.

9. Closing Remark

As the SWATH has been developed recently, the number of constructed SWATH vessels around the world is few. To date, however, SWATH has been operating successfully with no further structural problems event, and considered as completed a feasibility concept design package of its own for a next generation SWATH vessel. That next generation vessel has been developed such as an underwater working support vessel KAIYO or an offshore crane vessel MICOPERI 7000.

The intent of this review was to demonstrate the SWATH that would result to fully explore its market potentials for a variety of maritime applications covering commercial, military and private sectors.

Noteworthy technical items, therefore, were critically reviewed from both viewpoints of experiments and of operating experience by current SWATH vessels.
(The writer is a member of Sigma Xi, and also of International Ships and Offshore Structures Congress)

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