NAVAL SHIP ENGINEERING CENTER

DEVELOPMENT OF TECHNICAL PRACTICES FOR ROLL STABILIZATION TANKS DURING LATER SHIP DESIGN STAGES

NAVSEC Report 6136-75-12

15 April 1975

APPROVED FOR PUBLIC RELEASE - DISTRIBUTION UNLIMITED

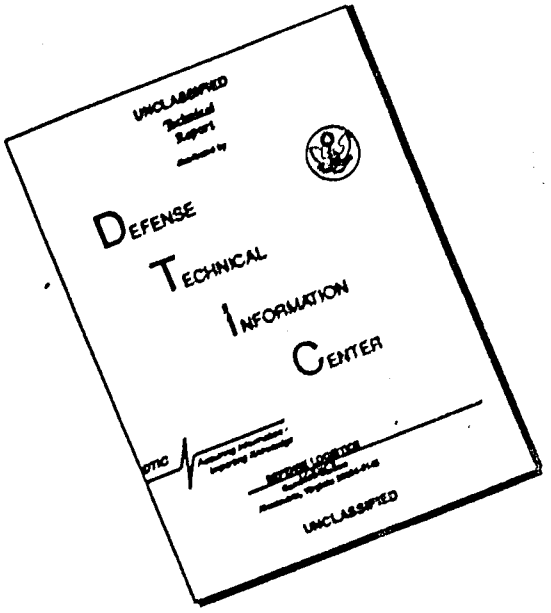
Prepared for

HULL FORM AND FLUID DYNAMICS BRANCH (SEC 6136)
NAVAL SHIP ENGINEERING CENTER
HYATTSVILLE, MARYLAND 20782

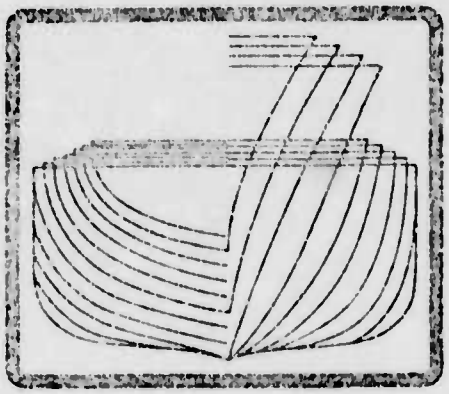
PEOPLE·PERFORMANCE·PRIDE·PROFESSIONALISM

HYATTSVILLE, MD.

DISCLAIMER NOTICE



THIS DOCUMENT IS BEST QUALITY AVAILABLE. THE COPY FURNISHED TO DTIC CONTAINED A SIGNIFICANT NUMBER OF PAGES WHICH DO NOT REPRODUCE LEGIBLY.



DEVELOPMENT OF TECHNICAL PRACTICES FOR ROLL STABILIZATION TANKS DURING LATER SHIP DESIGN STAGES

NAVSEC Report 6136-75-12

15 April 1975

APPROVED FOR PUBLIC RELEASE - DISTRIBUTION UNLIMITED

Prepared for

HULL FORM AND FLUID DYNAMICS BRANCH (SEC 6136)
NAVAL SHIP ENGINEERING CENTER
HYATTSVILLE, MARYLAND 20782

By

Roderick A. Barr and Vladimir Ankudinov
HYDRONAUTICS, Incorporated
and William C. Webster, Ship Research Incorporated
(HYDRONAUTICS Technical Report 7401.26-1)

Under

Contract N00024-C-5572
Task 6120-226
(NAVSEC TPOC: R. A. Johnson)

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE	READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER: Technical Report 7401.26-1	2. GOVT ACCESSION NO.
3. RECIPIENT'S CATALOG NUMBER	
4. TITLE (and Subtitle): DEVELOPMENT OF TECHNICAL PRACTICES FOR SELECTION OF ROLL STABILIZATION SYSTEMS - PHASE II - ROLL TANKS	5. TYPE OF REPORT & PERIOD COVERED: Technical Report
	6. PERFORMING ORG. REPORT NUMBER: T.R. 7401.26-1
7. AUTHOR(s): Roderick A. Barr, Vladimir Ankudinov, and William C. Webster	8. CONTRACT OR GRANT NUMBER(s): N00024-73-C-5572
9. PERFORMING ORGANIZATION NAME AND ADDRESS: HYDRONAUTICS, Incorporated, 7210 Pindell School Road, Howard County, Laurel, Maryland 20810	10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS: NAVSEC - 6120-226
11. CONTROLLING OFFICE NAME AND ADDRESS: Naval Ship Engineering Center, Hyattsville, Md.	12. REPORT DATE: January 1975
	13. NUMBER OF PAGES: 158
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)	15. SECURITY CLASS. (of this report): Unclassified
	15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for Public Release - Distribution Unlimited

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

18. SUPPLEMENTARY NOTES

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

Ship Roll Motions
Roll Stabilization
Anti-Roll Tanks

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

This report discusses the selection and design of anti-roll or roll tank stabilization systems for naval ships during ship contract design, and methods for validating the predicted tank performance. Drafts of Brief Technical Practices Sheets for roll stabilization system selection and design at the preliminary and contract design levels are included.

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

UNCLASSIFIED

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

Page

NOTATION AND DEFINITIONS

AIRSP	Airspring constant - controlled-passive tank
b	Tank crossover width
B	Tank maximum width
B'	Quadratic damping coefficient
$C_{\ell_{total}}$	Total head loss coefficient
d	Tank crossover length
D	Tank length (fore-and-aft)
g	Gravitational acceleration
g_n	Pump control gain
GM	Ship metacentric height
h	Fluid depth
h_t	Height of tank pivot
$h(R_\tau)$	Tank head loss coefficient
H	Pump head
HP	Pump horsepower
K_{st}	Fractional decrease in GM due to tank free surface loss
KR	Roll center height above keel
KG	Center of gravity height above keel
L	Ship length
L_g	Control gain - controlled-passive tank

M	Ship mass
M_{as}	Sway added mass
p	Tank crossover height
r	Scale ratio
R	Half width of U-tube tank
R_t	Tank fluid radius of gyration
R_T	Tank fluid velocity
s	Distance along tank fluid path
S'	Effective tank length
S''	Effective coupled tank length
t	Time interval
t_t	Effective height of tank fluid above roll center
T	Tank empty time
V	Ship speed
V_i	Local tank fluid velocity
w_{fs}	Tank free surface width
W_t	Weight of fluid in tank
α_p	Pump blade angle of attack
ζ_t	Tank linear damping ratio
η_p	Pump efficiency
η_{st}	Tank vertical position parameter

η_t	Ship-tank natural frequency ratio
ν	Kinematic viscosity
τ	Tank fluid angle
τ_s	Tank fluid angle to stop airflow
τ_{max}	Tank saturation angle
φ	Roll angle
ω	Frequency of oscillation
ω_s	Ship natural roll frequency
ω_{slosh}	Tank slosh frequency
ω_t	Tank natural roll frequency

Definitions

Heading Angle - The angle between the direction of ship motion and the wave direction (zero degree heading angle is stern seas, 180 degrees is head seas) where

1. bow seas are heading angles between 90 and 180 degrees
2. beam seas is a 90 degree heading angle
3. quartering seas are heading angles between 0 and 90 degrees.

Saturation Angle - The tank fluid angle (see) where fluid first touches the top of the tank.

Ship-Tank Coupling - Ratio of the ship natural frequency to the tank natural frequency or the tank tuning.

Significant Roll - The average of the one-third highest roll motions (out-to-out) in irregular seas.

Tank Fluid Angle - The angle the fluid surface makes with a horizontal fixed in the ship (such as the tank bottom or top).

Tank Damping - Ratio of the tank fluid flow damping to the critical damping for the fluid flow.

ABSTRACT

This report describes a detailed study to develop technical practices for the selection of roll stabilization systems for naval ships. Unlike an earlier study which considered all types of stabilization systems, this study is restricted to anti-roll or roll tank stabilization. Active, passive and controlled-passive roll tanks are considered. Methods for selecting the best type of roll tank for a given ship design and for the detailed design of the tank are presented. These methods are appropriate to the ship contract design phase. The role of model testing in tank design and methods for validating predicted tank performance are discussed. Drafts of Brief Technical Practices Sheets for roll stabilization system selection and design at the preliminary and contract design levels are given in Appendices.

INTRODUCTION

In an earlier report by Miller, et al. (1), the requirements for and selection and sizing of roll stabilization systems for a given naval ship are considered. The methods presented in Reference 1 for sizing the stabilization system and for estimating its performance (roll reduction) are suitable for those parts of the design process referred to as concept design and preliminary design. A more detailed treatment of roll stabilization system design and performance, including model tests and performance verification are required during the final or contract design of the ship.

This report addresses the detailed or contract design of passive and active anti-roll or roll tanks, one of the roll stabilization systems considered in Reference 1. A parallel effort will be completed in 1975 by NSRDC on bilge keel and active fin stabilization. The primary topics considered in this report are the selection of tank type, the detailed design of the tank, including the role of model testing, the validation of tank performance and an evaluation of available methods for designing and predicting the performance of roll tanks and for predicting ship roll motions.

Preliminary drafts of Brief Technical Practices Sheets to be used by navy personnel in selecting and designing tanks are given in Appendices A and B of this report. Appendix A, which considers all types of roll stabilization, is suitable for concept and preliminary design phases and is based on the material in Reference 1. Appendix B, which considers only roll tank stabilization, is suitable for the contract design phase and is

based on the material in this report. While these Technical Practices Sheets are designed to replace the existing Navy Design Data Sheet for roll tanks, it is not intended that these Sheets be totally self-contained or that they present formalized design procedures. It is intended that Appendix A and Reference 1 are together sufficient for concept and preliminary design and that Appendix B and this report are together sufficient for contract design.

It should be noted that the most important step in the design of a roll tank is the selection of tank geometry. This report does not consider the selection of tank geometry, although Section VI does present a critical review of several available methods for sizing. Tank sizing is considered in detail in the previous report on technical practices for roll stabilization, Reference 1.

II. SELECTION OF THE TYPE OF ROLL TANK

The decision on whether to use an active, passive or controlled-passive roll tank will often be made during contract design, since this decision may require extensive trade-off studies. The decision on whether to use a free surface or U-tube tank will usually be made during contract design since this decision is primarily influenced by the shape of available spaces, the type of tank and the range of ship operating metacentric heights or GM's.

Selection of Active, Passive or Controlled-Passive Tanks

The choice of type of tank will be made on the basis of tank performance, reliability, cost and on allowable tank GM reduction and available space for the tank. Active tanks can provide greater roll reduction for a given tank size, GM loss and weight, but are more costly to build and operate and are less reliable than passive tanks.

Passive Tanks - These tanks enjoy the advantages of low initial cost, known performance at all times and the absence of moving parts, with resulting high reliability and the need only for minimal routine maintenance. Free surface tanks can, by varying fluid depth, be tuned for good performance over a range of operating metacentric heights. Passive U-tube tanks can be used in tuned pairs designed to provide good performance over a range of GM's. Passive tanks are generally larger and require a larger free surface loss (reduction in GM) than active and controlled-passive tanks. The only moving parts required are the valves in fill, drain, vent and cross over lines.

Active Tanks - These are defined to be tanks in which water is moved primarily by action of a pump. These tanks can provide greater roll reduction than other type tanks, particularly at high speeds and in stern seas. Active tanks are, however, much more expensive to build and to operate, require significant routine and non-routine maintenance, and have relatively low reliability. The high cost of such tanks is due primarily to the large size of the required pump and prime mover, the large pump power consumption and the need for a variable pitch pump. Results presented by Webster for a Mariner (2) indicate a peak pump power of 4000 horsepower and an ideal mean power of several hundred horsepower. The mean power could easily exceed 1000 horsepower, however, if a large and expensive system is not provided for storing power during periods when the pump, acting as a turbine, extracts power from the fluid. The variable pitch pump is required to provide the required dynamic response and to minimize pump power requirements.

Figures 1-6, from the results of Reference 1, compare roll motions for destroyer and auxiliary type ships with active and passive roll tanks. The figures show that the active tanks are modestly better at $V/\sqrt{L} = 0.8$ and significantly better only at $V/\sqrt{L} = 1.2$. Active tanks are thus of no interest for applications in which roll motions are not important at speeds-length ratios of 1.0 or more.

The disadvantages of the active tank including high initial and operating cost, the need for significant maintenance, large size and weight of the pump drive system, relatively low reliability and poor tank performance when the pump is not fully

controlled (the tank will often increase roll motions in such cases) would seem to clearly outweigh the performance advantages of the active tank. The absence of good model or ship data verifying the performance of active tanks also makes the use of such tanks unattractive.

Controlled-Passive Tanks - These are defined to be tanks in which air flow in the tank is controlled dynamically by valves. The performance, cost and complexity of these tanks will be somewhere between those of active and passive tanks. These tanks will require only moderate maintenance but will be significantly less reliable than passive tanks. Preliminary cost estimates indicate that controlled-passive tanks should cost about twice as much as passive tanks. Controlled-passive tanks will have poor performance if control system or valve failure occurs; the tank damping is inadequate for good performance at roll resonance (3) and closing of the valves will make the tank inoperative.

Figures 7 and 8, from Reference 3, for a fast cargo ship model and a 75 foot experimental vessel with a 0.47 block coefficient indicate roll reductions with passive and controlled-passive tanks. Figure 7, for zero ship speed, indicates that the controlled-passive tank is most effective for very low ship GM. Figure 8, for $V/\sqrt{L} = 0.46$, also shows the controlled tank is most effective at low GM's. In both cases it is clear that the actual roll reduction in real seas will depend very much on sea state and wave frequencies.

Figures 9 and 10, from Reference 4, compare calculated roll motions of a fine oceanographic research ship at $V/\sqrt{L}$ = 0.81 in beam and oblique seas. Figure 9, for beam seas, indicates only slightly greater roll reduction with controlled-passive tanks. Figure 10, for quartering seas, indicates a great difference between controlled-passive and passive tanks, the latter causing an increase in rolling motions. Passive roll tanks are frequently ineffective, and in some cases are detrimental, at high speeds in quartering seas. In such cases the tank will often be made inoperative to avoid increased roll motions. Controlled-passive tanks may be attractive, despite their greater cost and maintenance and reduced reliability, when passive tanks are ineffective for important ship operating conditions, as in Figure 10.

Dalzell, et al. (5) present the additions and modifications to the methods of Reference 2 necessary to consider controlled-passive tanks. These modifications are described in Appendix C.

Selection of U-Tube or Free Surface Tanks

A number of factors influence the choice of a U-tube tank or a free surface tank. These include the geometry of available ship spaces, the required range of ship operating GM's and whether the tank is active, passive or controlled-passive. Neither type of tank is clearly superior for all applications. The sample calculations in Reference 1 indicate that for a typical tank application there will probably be little difference in required fluid weight. It also appears that there will

be little significant difference in cost. One advantage of U-tube tanks is that the tank can be made rapidly inoperative and the GM loss can be rapidly reduced by closing valves in the cross-over lines. Some factors affecting the choice are discussed below.

Shape of Available Spaces - The shape of available ship spaces which can be used for the roll tank can influence the choice of a U-tube or free surface tank. As examples, U-tube tanks will be favored when it is not possible to provide uniform depth across the ship beam, and free-surface tank will be favored when fore-and-aft length is severely restricted at midships.

Type of Tank - Either a free surface or a U-tube tank can be used for passive tanks. With active or controlled-passive tanks, however, it is desirable to use a U-tube tank. The active tank pump will have a diameter which is generally small compared with tank length and height. The tank must therefore be constricted or necked-down at the pump, restricting free surface action and making any tank act like a U-tube tank. The best tank performance will thus be obtained by using a true U-tube tank design. Controlled-passive tanks operate by controlling the air flow across the tank; this can be easily done only when a U-tube tank is used.

Operation Over a Range of Ship GM's - For some ships it will be necessary for the roll tank to provide significant roll reduction over a range of ship operating GM's. This requirement can influence the choice of a free surface or U-tube tank. Free surface tank natural frequency is proportional to the square

root of fluid depth. When the use of preservatives in the fluid does not prevent the varying of tank fluid depth, such tanks are well suited to operation over a moderate range of GM's. U-tube tank natural frequency can be varied only by changing tank geometry, which is not practical, but such tanks can be used in "tuned-pairs" (two tanks with different natural frequencies) to provide good roll reduction over a range of GM's which is probably larger than the range for which a single free surface tank is suitable. In some cases the insensitivity of U-tube tank natural frequency to depth may be an advantage. "Tuning" tanks for operation over a range of ship GM's is discussed in some detail in Section III.

A detailed trade-off of tank performance versus tank size(s) and weight(s) may be required to determine the type of tank best suited to operation over a wide range of GM's.

III. SELECTION OF PRIMARY ROLL TANK DESIGN PARAMETERS

Once the type of roll tank to be used has been selected, the detailed design of the tank during ship contract design can proceed. The selection of tank design involves selection of tank geometry and location, refinement of the tank design using computer trade-off studies and model tests and the design of the control system for active tanks. The final steps in the design, including details of scantlings, damping devices, piping, etc., are considered in Section IV.

Location of the Tank within the Ship

Ship arrangements will dictate, to a considerable extent, available locations for a roll tank. The longitudinal location of the tank has only a modest effect on roll motions, while the vertical location can have a significant influence on roll tank performance.

The tank location should be selected, whenever possible, to insure a tank large enough to provide the maximum allowable GM reduction and/or the desired roll reduction. It is also important to provide sufficient tank height to avoid or minimize tank saturation (water impact on the tank top or unwetting of the tank bottom) in all but the most severe conditions. The selection of tank angle capacity (tank fluid angle at which saturation occurs) is discussed in Reference 1. It will sometimes be necessary to carry out trade-off studies to determine the best combination of tank location and geometry.

Webster (2) has studied the influence of tank longitudinal location on roll motions and concluded that this influence is

not significant. Reference 1 indicates that the tank should lie between 0.25L (L is ship length) forward and 0.35L aft of the ship LCG. Forward tank locations tend to increase motions in quartering seas and decrease motions in bow seas; the opposite is true for aft tank locations. Since maximum rolling motions usually occur in quartering seas, it is probably better to use an aft rather than forward tank location when the tank cannot be located near midships.

It is usually desirable to locate the tank as high as possible in the ship – the ideal location from the standpoint of roll motions reduction would be in the superstructure. The effect of vertical location on roll motions is greatest for ships with large values of GM. A later section presents typical calculated results of the effect of vertical position on roll motions for both active and passive roll tanks. It probably will not be desirable to select a high tank location if this location results in a too small tank angle capacity.

Refinement of Tank Design Using Computer Design Studies

The preliminary tank performance estimates are based on a tank having near optimum natural frequency and damping and a given free surface loss. This preliminary design can be refined using either computer design studies or model tests. Model tests are much more costly and are best suited to design refinements or tuning, such as the selection of damping configuration. The effect on performance of systematic variations in all design variables can be made efficiently using theoretical methods and either digital or analog computers. It should be noted, however, that available computer programs are proprietary and not generally available.

The design parameters whose influence on roll motions can be readily studied using a computer are:

1. Tank free surface loss or GM loss - K_{st}
2. Tank natural frequency or ratio of tank to ship natural frequency (ω_t/ω_s) - $1/\eta_t$
3. Tank damping ratio - ζ_t
4. Tank location (vertical and horizontal).

The influence of tank GM loss will usually be considered in preliminary tank design. Roll motions decrease monotonically with increasing tank GM loss (see Reference 1). It is therefore desirable to select the largest tank and GM loss acceptable from ship stability and weight considerations. Typically a tank GM loss of 20 to 30 percent of unstabilized ship GM is selected.

To illustrate the refinement of tank design using computer calculations, the methods of Reference 2 have been used to calculate the effect of various tank design parameters on the roll motions of the ships considered in Reference 1. The results, presented as significant roll angles for the worst heading angle in short-crested, irregular seas and the corresponding tank fluid angles are discussed below.

Table 1 presents the variation of roll and fluid motions with tank frequency ratio (ship natural frequency divided by tank natural frequency) and sea state, for the auxiliary type ship at 1.2 speed length ratio. Table 2 shows the corresponding variation of tank weight, assuming a rectangular, constant beam, free surface tank. The variation of roll motion with frequency

TABLE 1

Effect of Passive Tank Frequency Ratio
(Ship Natural Frequency/Tank Natural Frequency)
on Significant Roll and Tank Angles of an
Auxiliary Type Ship in Short Crested Seas
at 1.2 Speed Length Ratio

Tank Frequency Ratio - η_t	Significant Roll at Sea State - Deg				Significant Tank Fluid Angle at Sea State - Deg			
	3	4	5	6	3	4	5	6
0.85	6.7	13.0	17.3	21.0	9.2	17.8	23.5	27 8
0.90	6.6	12.8	17.2	20.8	9.0	17.4	23.0	27.2
0.95	6.5	12.7	17.0	20.7	8.8	17.1	22.6	26.6

TABLE 2

Estimated Tank Weights for
Ship and Tanks
Considered in Table 1

Tank Frequency Ratio - η_t	$\frac{\text{Tank Weight}}{\text{Tank Weight at } \eta_t = 0.90}$
0.85	1.12
0.90	1.0
0.95	0.90

ratio is very small and hence it is likely that a low frequency ratio (say 0.85) and a small tank will be selected.

Tables 3 and 4 present the variation of roll and tank motions with tank damping ratio for active and passive tanks for the destroyer type ship at 1.2 speed length ratio. For active tanks, roll motions increase slightly and tank angles decrease slightly with increasing tank damping. As a result, a small damping ratio would be selected to minimize pump power consumption. For passive tanks, roll motions hardly change with tank damping, but tank angle decreases significantly with increasing damping. Smaller tank height and water weight can be used with smaller tank angles, and hence it is desirable to use relatively large damping (say $\zeta_t = 0.5$) for passive tanks.

TABLE 3

Effect of Active Tank Damping Ratio on
Significant Roll and Tank Angles for
Destroyer Type Ship
at 1.2 Speed-Length Ratio

Tank Damping ζ_t	Significant Roll at Sea State - Deg				Significant Tank Fluid Angle at Sea State - Deg			
	3	4	5	6	3	4	5	6
0.08	4.9	9.6	13.0	15.8	8.5	16.8	23.8	30.5
0.16	5.0	9.8	13.2	16.1	8.2	16.2	23.0	29.4
0.32	5.1	10.0	13.6	16.6	7.7	15.1	21.5	27.5

TABLE 4

Effect of Passive Tank Damping Ratio on
Significant Roll and Tank Angles for
Destroyer Type Ship
at 1.2 Speed-Length Ratio

Tank Damping ζ_t	Significant Roll at Sea State - Deg				Significant Tank Fluid Angle at Sea State - Deg			
	3	4	5	6	3	4	5	6
0.3	6.6	13.0	17.2	20.9	10.2	19.8	26.2	31.2
0.4	6.6	12.8	17.2	20.8	9.0	17.4	23.0	27.2
0.5	6.6	12.8	17.2	21.0	8.1	15.8	20.7	24.4

Tables 5 and 6 present the variation of roll and tank fluid angles with tank vertical location for active and passive tanks for the destroyer type ship at 1.2 speed length ratio. The tank vertical position parameter is defined by

$$\eta_{st}^2 = \frac{\omega_s^2}{2g/S''}$$

where ω_s is ship natural frequency

g is gravitational acceleration

S" is the effective coupled length,

$$S'' = \int t_t ds/R \text{ for U-tube tanks}$$

The dimensions used to compute S" are defined in Figure 11. For both active and passive tanks, roll motions in all sea states

TABLE 5

Effect of Active Tank Vertical Position on
Significant Roll and Tank Angles for
Destroyer Type Ship
at 1.2 Speed-Length Ratio

Vertical Position Parameter η_{st}^2	Significant Roll at Sea State - Deg				Significant Tank Fluid Angle at Sea State - Deg			
	3	4	5	6	3	4	5	6
-0.5	3.4	7.5	10.9	13.8	6.2	13.4	19.5	24.4
0	4.0	8.7	12.8	16.5	6.8	14.6	21.7	27.6
0.5	5.0	10.6	15.5	20.1	7.7	16.5	24.5	31.6
1.0	6.1	12.9	18.8	24.3	8.7	18.7	27.5	35.9

TABLE 6

Effect of Active Tank Vertical Position on
Significant Roll and Tank Angles for
Auxiliary Type Ship
at 1.2 Speed-Length Ratio

Vertical Position Parameter η_{st}^2	Significant Roll at Sea State - Deg				Significant Tank Fluid Angle at Sea State - Deg			
	3	4	5	6	3	4	5	6
-0.5	4.4	10.9	11.5	13.8	7.5	21.8	20.9	26.0
0	5.0	9.8	13.2	16.1	8.2	16.2	23.0	29.4
0.5	6.0	11.6	15.9	19.7	9.5	18.7	26.5	34.0
1.0	7.5	14.5	19.9	24.9	11.4	22.2	31.4	40.0

decrease significantly with increasing tank elevation above the ship center of gravity. Tank angle also increases with increasing elevation, but it is clear that the tank elevation should be maximized (minimum value of η_{st}^2). For typical applications the value of η_{st}^2 will be between about +0.2 and -0.4.

A time domain solution program, as developed at HYDRONAUTICS, may be used to study non-linear effects such as tank saturation. Webster (2) discusses the equations and numerical methods necessary for such solutions. Table 7 presents a comparison of roll motions calculated with and without non-linear effects. In all cases larger roll motions are calculated when non-linear effects are considered. The largest increase, about 15 percent, occurs with the smallest saturation angle and largest sea state. For some cases, particularly when tank angle capacity is severely limited, non-linear effects may be much larger.

Design of Tanks for Operation Over a Range of Ship's Metacentric Height

For ships which can operate with a wide range of GM's, it will be desirable, if not essential, for the roll tank to provide significant roll reduction over most or all of this GM range. Tanks "tuned" to a single GM or ship natural frequency will usually have poor performance at other GM's. A means of varying tank natural frequency or designing for operation over a range of GM's is required.

The natural frequency of free surface tanks varies as the root of fluid depth. Optimum tank tuning, in which the tank natural frequency is 6 to 10 percent greater than ship natural frequency

TABLE 7

Comparison of Calculated Linear and Non-Linear, Significant Roll Motions for Destroyer Type Ships with Passive Roll Tanks in Long-Crested Seas

Ship/Tank Coupling K_{st}	Tank Damping ζ_t	Saturation Angle τ_{max}	Sea State	Significant Roll Angle - Deg	
				Linear*	Non-Linear
0.2	0.3	10°	5	23°	26.0°
		10°	6	27°	31.5°
		15°	6	27°	29.3°
	0.5	10°	5	23°	24.4°
		10°	6	27°	30.8°
		15°	6	27°	30.5°
0.3	0.3	10°	5	19°	21.8°
		10°	6	23°	27.0°
		15°	6	23°	24.7°

*Interpolated for 70° heading angle.

(see Reference 1), can be achieved by changing fluid depth in proportion to ship GM. Two factors limit the range of GM's or ship natural frequencies over which the tank can produce significant roll reduction. If fluid depth is varied, tank depth must be increased to avoid tank saturation (water impact on tank top or tank bottom unwetting) in heavy seas at the maximum and minimum water depths; allowable tank depth will thus limit the effective range of operating depths and ship GM's. The tank GM loss must be small enough to provide adequate stability at the minimum ship GM. As the tank GM loss will vary little if any with depth, the tank GM loss may be insufficient to provide good tank performance at high ship GM's. A second tank, which is used only for high ship GM's, may be required.

The natural frequency of a U-tube tank, which is defined in Equation [18] of Reference 1 by

$$\omega_t = \sqrt{2gS'}$$

where ω_t is tank natural frequency

g is gravitational acceleration

S' is the effective tank length,

$$S' = h + B-b + b\,[D(B-b)/2dp]$$

and where the dimensions used to define S' are defined in Figures 11 and 13, can be varied significantly only by changing tank fluid depth, h, or beam, B, which is not possible. U-tube tanks can be used in "tuned-pairs," to achieve significant roll reduction over a range of ship GM's, if the natural frequencies of the tanks

are carefully selected. The tanks are designed to provide maximum roll reduction, if used alone, near the maximum and minimum ship natural frequencies or GM's, as shown in Figure 12. The resulting roll reduction will be fairly uniform over the frequency range. The optimum tank natural frequencies and allowable range of GM's for good tank performance must be determined by detailed design calculations or model tests. A relatively constant ratio of tank GM loss to ship GM can be obtained by using the tank with the higher natural frequency (ω_{2_t} in Figure 12) only at larger ship GM's. The use of two tanks has other potential advantages including greater flexibility in location.

Choice of Active Tank Control Parameters

For active tanks it is necessary to design the control system and to select the control system gains to provide maximum roll reduction and to avoid control system instabilities. A detailed discussion of control systems is given by Webster and Dogan (6).

References 2 and 6 consider a control system in which the pump angle of attack, α_p, is specified to be:

$$\alpha_p = g_3\ddot{\varphi} + g_2\dot{\varphi} + g_1\varphi$$

where φ is the roll angle

g_1, g_2 and g_3 are specified control gains

and the dots indicate time derivatives. The gain coefficients g_1, g_2 and g_3 are assumed to be independent and arbitrary. Their values are selected to provide the best tank performance while insuring that tank operation is always stable.

Webster (2) indicates that a detailed study of gains for an active tank for a Mariner resulted in the following set of values for optimum tank performance:

$$g_1 = 0.50$$

$$g_2 = 1.25$$

$$g_3 = 2.40$$

Other studies also indicate that these values lead to good tank performance. Table 8 presents results of a study of varying gains for the destroyer type ship of Reference 1. The values above were used in Reference 1. The other two sets of values (0.5, 1.25 and 3.0 and 0.0, 1.25 and 2.4) were selected from the results of Reference 6 as being nearly as good. Table 8 indicates that these other two sets of values are less desirable with K_{ST} of 0.20, due to the large tank angles, but that the set 0.5, 1.25 and 3.0 might offer a significant advantage with K_{ST} of 0.30. It is desirable, for a given design, to carry out at least a limited investigation of control gains, using the values above ($g_1 = 0.50$, $g_2 = 1.25$ and $g_3 = 2.40$) as a starting point.

A detailed discussion of control system stability, based on the use of open loop response or a Bode plot, is given by Webster and Dogan (6). While there appears to be little or no danger of instability when the control gains listed above (0.50, 1.25 and 2.40) are used, it is always wise to check stability for a given design.

TABLE 8

Calculated Effect of Active Tank Control Gains
on Significant Roll of Destroyer Type Ship
at 1.2 Speed-Length Ratio in Short-Crested Seas

Ship/Tank Coupling - K_{ST} $\Delta GM/GM$	Control Gain Values			Significant Roll Angle in Sea State				Significant Tank Angle in Sea State			
	G_1	G_2	G_3	3	4	5	6	3	4	5	6
NO TANK		0		6.8	14.5	21.2	27.5	NO TANK			
0.2	0.5	1.25	2.4	4.0	8.7	12.8	16.5	6.8	8.7	12.8	16.5
	0.5	1.25	3.0	3.7	8.1	11.9	15.2	7.5	16.3	24.1	30.6
	0	1.25	2.4	3.8	8.2	12.1	15.4	7.4	16.2	23.9	30.2
0.3	0.5	1.25	2.4	3.5	7.7	11.2	14.3	5.9	13.0	19.3	24.4
	0.5	1.25	3.0	3.2	7.1	10.4	13.0	6.6	14.6	21.4	26.9
	0	1.25	2.4	3.3	7.3	10.6	13.4	6.5	14.3	25.7	26.2

Selection of Damping Structure and Calculation of Tank Damping

The tank should be designed to provide optimum damping (typically a damping ratio of 0.2 for active tanks and 0.4 to 0.5 for passive tanks). Tank damping arises from three basic sources; friction on tank walls, form losses due to changes in tank cross-section or due to bends (for U-tube tanks) and losses due to internal damping structure, stanchions, etc.

Methods for calculating the damping due to the basic tank geometry are given for U-tube tanks, by Webster (7). These methods can also be adapted to free surface tanks. Methods for estimating the effect of damping devices, such as "nozzles" or stanchions are given in Reference 8. Appendix G summarizes some useful results from References 7 and 8. The actual magnitude of tank damping is best determined using tank bench tests, and damping devices are usually designed or positioned on the basis of such tests.

Model Testing and Its Role in Tank Design

Model testing will usually play two roles in the design of roll tanks. It can be used to: 1) refine design characteristics such as tank natural frequency and damping and to 2) verify or validate the tank performance. Experimental verification of tank performance through model tests is perhaps the most important phase of the tank design.

In this section bench tests of a tank model and tests of a ship model with tank model installed are discussed in some detail. Suitable test techniques and methods for interpreting the results are described. The role of bench tests in refining

the details of the tank design are considered. In Section V the role of model tests in validating tank performance is considered.

Bench Tests - The purpose of the bench tests of a model of a roll tank is to ascertain and verify the following basic dynamic characteristics of the tank:

- o the tank tuning (the tank's natural frequency)
- o the tank damping.

In addition, there are several other aspects of the tank's performance which are, or can be, observed during a normal bench test. These are:

- o the effect of configuration (such as water level or valve position) on tuning and damping
- o the effect of non-linearities
- o the inception of saturation.

Scaling of Bench Tests - It is desirable to test a small model of the tank in order to determine the above characteristics. As a result, it is necessary to have a geometrically similar model and to preserve certain dynamic laws if the model is to perform exactly as the full-scale tank. For precise modeling, it would be necessary to preserve the Froude number, Reynolds number, Weber number and cavitation number. Because of the limited number of fluids available, it is not possible to preserve all of these ratios. The situation is analogous to ship model testing. Weber number (relating to the effect of surface tension) and cavitation number govern phenomena which are not important for most roll tanks and, as a result, lack of preservation of these quantities is not critical.

Froude scaling can be accomplished by scaling time in the following way:

$$t_m = t_p \cdot \sqrt{r} \quad ,$$

where t_m is the interval of time in the model scale which corresponds to the interval of time, t_p, for the prototype.

r is the scale ratio, the ratio of a linear dimension of the model to a corresponding dimension on the prototype.

Since we are dealing with a small scale tank model, the model has a shorter natural period than the full scale tank.

Reynolds scaling requires that the viscous properties of model and prototype are preserved, and thus for a Froude-scaled model we require that

$$\nu_m = \nu_p \cdot r^{3/2}$$

where ν_m is the required kinematic viscosity of the fluid used in the model corresponding to the kinematic viscosity, ν_p, of the fluid used in the prototype.

This above relation indicates that we require a very much smaller kinematic viscosity in the model than in the prototype.

If a heavy oil such as Navy Special is used in the ship tank, it can turn out that water in a small model satisfies the Reynolds scaling law. Typically, the prototype fluid is fresh water or light fuel oils (such as kerosene) and it is impossible to

find a fluid with a low enough kinematic viscosity to preserve Reynolds scaling. In this situation water is usually used in the model and the model is made as large as practical so that the discrepancy in Reynolds scaling is minimized.

Two facets of roll tank testing ameliorate the Reynolds number discrepancy. First, as long as the flow, both full scale and model scale, is turbulent, then little difference occurs in such quantities as frictional drag. If the tank models are constructed with internal structure, then these structures assure constant "tripping" of the low Reynolds number flow in the model, assuring turbulent flow. Further, since the fluid is trapped in the model, the vorticity generated by the flow is retained in the fluid. This results in a high level of background turbulence in the tank fluid, also assuring "tripping" of the flow. Second, most of the losses in the tank occur due to sudden constrictions in the flow. The entrance and exit losses resulting from these constrictions are known to be practically independent of Reynolds number.

In conclusion, it can be stated that lack of Reynolds scaling will probably not produce large errors in measurement of the tank model properties, as long as the model is reasonably large. Practical experience indicates that models larger than about 30" in beam generally lead to reasonable measurements. Models smaller than about 20" in beam have been known to exhibit difficulties. In this latter case, the damping measured in the tank model can be significantly greater than that measured in a larger model. In other words, tank models this small can lead to erroneous conclusions, particularly with regard to the

effect of internal structure. It should also be pointed out that in quite small free surface tanks (Flume-type) the effect of surface tension becomes large enough to prevent the waves created by the tank motion from breaking. It is not known what effect this has on interpretation of the test results.

Test Procedures for Bench Tests - Appropriate test procedures and equipment for bench tests are discussed in some detail in Appendix D.

Other Types of Bench Tests

The two standard types of bench tests described in Appendix D have limitations. It is noted there that the oscillating table tests apply only to the situation of rolling in still water. However, if one is careful with the tares, then one can measure moments which can be interpreted directly in terms of stabilization effectiveness for this still water situation. The impulse test can only be used for U-tube type tanks and yields only a dynamic characterization of the tank. This characterization must be interpreted through the use of a computerized dynamic simulation of the ship and tank.

Recently the Naval Ship Research and Development Center has developed a servo-controlled oscillating table test facility in which the table can roll and sway simultaneously under computer control. The oscillating or roll table is used in conjunction with an analog computer in a combined, real time dynamic simulation; in other words in a hybrid computation. In principle, the process is straightforward. The table is oscillated so that the roll angle and sway displacement match

the instantaneous values of same quantities generated in real time using an analog computer. The force and moment generated by the tank are measured, corrected for tares and "fed-back" into the computer simulation. In this way, the tank model has the same swaying and rolling motions as it would on a ship in a seaway (unlike the normal bench test). This method is currently limited to regular and long-crested irregular seas, although it is proposed to extend it to short-crested seas.

This simulation facility is described by Zarnick et al. (9). An evaluation of this facility is given in Appendix E. It appears that this facility is more suitable for tank performance validation than for selection of overall tank geometry.

Ship Model Testing

The purpose of ship model tests is generally to verify the following characteristics of the stabilization system:

- o the roll reduction at resonance
- o the roll response at other frequencies.

In addition, several other aspects are usually noted during these tests. These are:

- o the effect of forward speed and hull configuration (such as bilge keels) on the stabilization
- o increases in rolling in stern seas due to the tank.

Scaling of Ship Model Tests - The scaling relations for the ship model are virtually the same for the ship model as for the tank model. Geometric scaling must be preserved as well as Froude scaling. As a result, Reynolds scaling cannot be preserved

Although this is not very important for the bench test of the tank model, it is quite significant for the ship model. The fluid motion in the tank is quite well damped; the roll motion of the ship is not. Therefore, care must be taken to select a ship model large enough so that a mismatch of the viscous fluid characteristics in the flow about the hull will not cause a large discrepancy in the roll response. This usually means that the ship model must be quire large, generally 15 feet or more. Swaan (10) indicates that a length of 10 feet may be satisfactory for ships with reasonable size bilge keels. Greater lengths will be required for hulls without bilge keels. Martin (11) indicates that the bilge keel width should be at least 0.5 inches to avoid scale effects — this dimension may set the minimum hull length.

Even if a ship model length of 15 to 20 feet is used, the on-board model of the roll tank may be significantly smaller than the tank bench test model. In this case it is wise to perform at least a rudimentary bench test of the roll tank to be installed on the ship model to verify that it has the same characterisitcs as determined in the complete bench test. Any discrepancy can likely be attributed to scale effects. If the ship model tank is significantly smaller than the bench test model, then it may be necessary to omit some or all of the structural detail on the smaller model to obtain the same damping characteristics.

It is also necessary to note that in the ship model tests the fluid used in the tank should have a specific gravity

(compared to the test basin water) identical to the specific gravity of the tank fluid (relative to sea water) of the prototype.

Test Procedures for Ship/Tank Model Tests - Procedures for conducting tests of ship models equipped with roll tanks are discussed in some detail in Appendix F.

Refining of the Design

As was pointed out in Reference 1, it is desired for the tank to have particular values of certain characteristics such as natural frequency and critical damping ratio. It is probable that when the detailed tank design is completed and a tank model is first tested it will not have exactly the desired characteristics. At this point, a model of the tank can be used to refine the design. If the tank does not have the correct natural frequency, then it is usually necessary to change some overall dimension of the tank. For instance, too low a natural frequency means that the flow area from one side of the tank to the other is too small. If the tank does not have the correct damping, then it is usually necessary to change some of the small details of the tank. For instance, too low a critical damping ratio means that more structure with a high fluid drag should be placed in the tank, ideally in an area of high flow, such as the crossover duct in a U-tube. This iterative process of experimental design development is continued until a satisfactory design is achieved.

IV. DEVELOPMENT OF CONTRACT DESIGN AND DRAWINGS

In the previous section means for selecting the type of tank and the design parameters and geometry of that tank are considered. To complete the tank design at the contract design level, it is necessary to select tank scantlings, required damping structures and tank piping and valving. It is also necessary, as part of the contract design, to prepare tank drawings and an operations manual for the tank.

Integration of Tank Design with Ship Structure

The selection of tank location to make the best use of available ship spaces has been discussed earlier. The space selected will dictate the tank planform shape and height and the structural members and penetrations within the tank space.

A wide range of tank planform shapes, including rectangular, I-shape and C-shape, as shown in Figure 13, can be used. The shape, per se, will have little effect on tank performance unless excessive damping occurs due to flow constrictions. These planform shapes can be used for both free surface and U-tube tanks. For active tanks either an I-shape or C-shape tank will usually be required. Rectangular spaces can always be converted to either I- or C-shape spaces.

Care must be taken to insure that the bulkhead stiffeners, stanchions or other structure within the tank space do not result in excessive tank damping, with a resulting increase in roll motions and, in the case of active tanks, increase in required power. It will usually be desirable to place most or all bulkhead stiffeners outside the tank. Structural stanchions are

usually widely spaced and hence do not result in excessive damping. Other penetrations of tank space, such as cableways or pipeways will not be a problem unless their location results in severe restriction of tank flow and hence excessive damping. Figure 14 shows the integration of a typical free surface tank into the ship structure.

The influence on tank damping of tank shape and structures within the tank is discussed in Section III.

Selection of Tank Scantlings

Tank scantlings must be adequate to meet both the hydrostatic and dynamic pressures which occur during tank operation. Dynamic pressures occur because of fluid acceleration in the tank, saturation and perhaps fluid sloshing. Much if not all of the main tank structure will be existing ship structure (decks, bulkheads, etc.). Required scantlings for these parts must be adequate to meet both ship and tank structural requirements.

Required tank bottom and side scantlings can be determined using the Navy Design Data Sheet for deep tanks (12). The tank should be assumed to be filled to the top for estimating hydrostatic pressure.

If tank saturation occurs, the tank top can be subjected to large dynamic pressures. The use of very heavy tank top scantlings, to resist these pressures, can be avoided if the tank is fitted with several sets of damping devices just below the tank top. These damping devices prevent large dynamic

pressures on the tank top during saturation conditions, but have little or no effect on tank damping during normal operating conditions. When the tank has a large angle capacity and saturation is anticipated only rarely, such damping devices are not needed.

Large tank loads can occur if the natural or sloshing frequency of the tank fluid and the exciting frequency (frequency of encounter) are equal. Sloshing can occur in either free surface or U-tube tanks, although it will be most serious in free surface tanks, due to the much larger free surface. The lowest sloshing frequency, which is usually the most important, is given in Reference (7) as:

$$\omega_{slosh} = \sqrt{\frac{g\pi}{w_{fs}} \tanh \frac{\pi h}{w_{fs}}} \cong \frac{\pi}{w_{fs}} \sqrt{gh}$$

where ω_{slosh} is the sloshing frequency

h is the fluid depth

w_{fs} is the width of free surface (in either transverse or longitudinal directions)

A formula for higher harmonics of sloshing is given by Webster (7). If ω_{slosh} corresponds to a frequency of encounter with significant wave energy, sloshing may cause significant dynamic pressures on tank sides or ends and it may be necessary to increase tank scantlings. If sloshing appears to be a problem, it is generally desirable to increase sloshing frequency by increasing water depth or reducing free surface width, either by longitudinal subdivisions or by using a U-tube rather than a free surface tank.

Provision of Necessary Piping and Valving

The roll tank should be fitted with fill, drain and vent lines, with appropriate valving, and sounding tubes. The fill and drain lines may be the same or separate lines depending on the procedure used for rapid tank draining. Vent lines are required at the top of the tank to prevent over-pressurization and possible tank damage during filling. Some means for rapid tank draining must be provided unless means are provided for subdivision of the tank in case of ship damage.

If the tank is located above the source of fluid (fuel oil tanks, fresh water tank, etc.) a pump will be required in the fill line. Vent lines are required at the tank tops to permit tank filling and prevent damage due to overpressurizing the tank during filling. These lines should be fitted with stop check valves to prevent flow into the tank. If possible, these vent lines should have a significant height and open on clear deck; when tank saturation occurs during heavy rolling, fluid can be forced well up these vent lines. For U-tube tanks one vent line should be provided at the top of each vertical leg.

Some means must be provided for rapid tank draining or reduction of tank free surface (GM) loss, in cases of lost intact stability due to ship damage. If damaged stability conditions require dumping the tank overboard, it will be necessary to use a pump to drain, if the tank bottom is at or below the damaged waterline; it is not adequate to provide for gravity drainage only in undamaged conditions. It is currently considered

necessary to affect this draining or tank subdivision in three to five minutes. Current preferred Navy practice is to drain the tank to the ship double-bottom tanks. When this is not possible, as for the Sea Control Ship, the tank must be drained overboard or the tank subdivided. With U-tube tanks, free surface loss may be sufficiently reduced by closing the vent line and crossover line valves.

It should not be too difficult to subdivide a tank with moving gates or bulkheads under normal conditions, including ship rolling, but it may be extremely difficult to fully subdivide the tank when serious damage has occurred. If the subdividing member does not fully seal, fluid can flow past and no reduction in free surface loss will occur.

Large drain lines will be required to drain a tank in three to five minutes. For a tank with 800 tons of fluid (as the Sea Control Ship design) 3 to 4 square feet of drain cross-sectional area will be required to drain in three minutes at a drain velocity of 50 feet per second. Much higher drain velocities are probably not practical unless the drain lines are free of bends and obstructions. Port and starboard drain lines are required. If a pump is required for draining, large powers can be required. The required power to empty a tank in a given period, pumping against a given head is:

$$HP = \frac{W_t H}{14.7\ T\ \eta_p}$$

where HP is the pump horsepower

W_t is the weight of tank fluid in long tons

H is the required pump head rise

T is the empty time in minutes

η_p is the pump horsepower.

As an example 114 horsepower will be required, with a pump efficiency of 0.80, to empty 800 tons of fluid in three minutes with a five foot pump head. The pump head is the sum of the static head (elevation of the discharge above the tank bottom) and the piping losses. The latter can be quite large.

Tank Operations Manual

It is necessary when a committment has been made to build a ship and, as part of the tank contract design, to prepare a tank operations or instruction manual. While the primary purpose of this manual is to describe tank operation and operating procedures, some discussion of routine maintenance should also be included for passive tanks. For active tanks and controlled-passive tanks it will be necessary to prepare a separate and more detailed maintenance manual.

A tank operation manual should include the following items:

1. A brief description of ship rolling and the tank design to reduce rolling.
2. A description of the tank installation including location, piping, valving, gaging, etc.
3. Instructions for operation under normal conditions including:

a. Water level
b. Valve positions

4. Instructions for operation under special conditions including:

   a. Low initial ship stability or GM
   b. Very heavy weather
   c. Resonant rolling
   d. Damaged ship conditions

5. Instructions for operation during tank filling and draining.

6. Discussion of simple, routine maintenance.

7. Curves of predicted tank performance (roll reduction as a function of ship speed, heading angle to waves, wave height and GM.

The manual should be kept as simple as possible, to facilitate its use by ships' personnel.

V. VALIDATION OF STABILIZER PERFORMANCE

The problem of validating the performance of a roll stabilization system is an extremely difficult one. This problem is compounded because even when the ship and roll tank have been built and the ship puts to sea, validation is very difficult, and thus at sea trials of roll tanks are very limited. For these reasons it has not been possible to establish meaningful levels of confidence for theoretical methods and experimental methods for predicting roll tank performance. Since it is at-sea performance of the roll tank which is of primary interest, it is appropriate to consider measurement of this performance before considering available methods for validating the predicted roll tank performance.

At-Sea Measurements of Roll Motions

The worst roll motions typically are experienced in stern quartering seas. In a real seaway, the waves are random in length, height and direction. This means then that any measure of the roll motion of the ship must be statistical in nature. The determination of meaningful statistics (that is, values with a high level of confidence) requires that a large number of roll cycles be examined. In stern quartering seas, the encounter frequency is low and, as a result, it takes a long time to encounter a large number of cycles. For instance, if the ship's roll resonant period is 12 seconds, then the worst speed and heading combination will lead to an average encounter frequency of 12 seconds. This means that in a half-hour, the ship will be affected by about 150 waves. This is approximately the

number of encounters required to determine a reasonably confident value for the root-mean-square (rms) roll angle.

Since measurement of the waves themselves is extremely difficult, a determination of the effectiveness of the stabilization system requires that one set of tests be performed with stabilization and another without. The mechanics of performing the tests with and without stabilization results usually in the two tests being performed over a span of about 2 hours. One must then hope that the seaway has not changed much in this span of time. In conclusion, then, one can say that full-scale tests are quite difficult to execute and interpret.

Validation Using Ship Model Tests

Difficulties similar to those encountered in ship trials will be encountered in ship model tests carried out in a random seaway. If a random seaway is produced in a model basin it may require several runs of the model through this seaway to determine meaningful motions statistics. If this process were to be repeated for various scale sea states and headings the cost could be prohibitive. This problem becomes more acute as speed-length ratio increases and/or model length increases. A minimum model length of at least 10-15 feet and a minimum roll tank beam of at least 20 inches are required to minimize scale effects, as described in Section III.

Ship model tests in regular waves will permit a shorter test run, but roll motions are not linear, particularly with a roll tank and in large waves, and it may not be possible to accurately predict motions in irregular waves from those in

regular waves. This difficulty is illustrated by Figure 35, which is discussed in a later section. Considerably different RAO's are obtained for a ship with no roll tank from tests in irregular seas corresponding to sea states 4, 5 and 6. The situation is almost certain to be worse with a roll tank. It therefore seems essential to conduct any validation tests in irregular waves corresponding to the sea states of interest for the ship.

Validation Using Bench Tests and Simulation

Probably the most direct and economical way to validate the stabilization system performance is to combine the methods of computerized dynamic simulation with model tests. The general approach here is to validate the computer simulation by means of a few selected model tests. The computer simulation can then be used to inexpensively predict the long-term motions of the ship in various conditions and these results can be used to validate the design.

In this hybrid approach, it is imperative to have a good characterization of the tank and this can be obtained by any of the bench test methods described previously. The modeling of ship motions dynamics is a rapidly maturing field, in which most of the basic concepts appear to be fairly well understood. As a result, it is possible to obtain fairly good models of the motions of unstabilized ships from the literature, although one important parameter which is generally not known is the ship's unstabilized roll damping, both at zero speed and at forward speed. Combining the tank and ship dynamics is a generally straightforward

undertaking. The impulse ship model test, if performed at zero speed and at forward speed, can be used to determine this damping parameter. Methods, based on available model data, are given in Reference 1 for estimating ship damping. The results of the computer simulation can then be compared with the regular wave, beam-seas tests and refined, as necessary. This "calibrated" dynamic simulation then can be used to compute the roll response in a variety of situations of interest.

Although there is no guarantee that the above method is foolproof, it appears to be the method which is, at present, most attractive, from the standpoints of accuracy and expense, for validating predicted roll motions and roll tank performance.

The NSRDC oscillation table simulation facility is similar in concept but considers both roll and sway motions in the time domain. It is thus potentially more accurate than other hybrid methods which consider only roll motions and typically use a frequency-domain solution. Initial validations of this facility indicate that it can predict roll motions with and without roll tank in regular and irregular waves with reasonably good accuracy. Further validation of the facility is needed, however, to insure that this facility is suitable for validating predicted roll motions for any ship, and to establish a sufficient level of confidence to permit dispensing with all ship model tests. It seems likely that validation of performance with this facility will be considerably more expensive than with the normal bench table test - computer simulation. The elimination of all ship model testing might make this facility economically more attractive, however.

VI. REVIEW OF METHODS FOR SIZING ROLL TANKS

One of the most important steps in the design of a roll tank is the determination of the tank size required to produce the desired roll reductions. In this section the adequacy of two available methods for tank sizing are considered, using data for two roll tanks designed for the Sea Control Ship as a standard.

Table 9 compares the dimensions and natural frequency of the PABL and J. J. McMullen tank designs for the Sea Control Ship with the dimensions and natural frequencies for these tanks calculated using the U. S. Navy Design Data Sheet, Reference 8 and the Phase I Report, Reference 1. The PABL tank was designed using Reference 8 , while the McMullen tank was designed using methods which are unpublished.

Tank design 3, in Table 9, is an independent check of the PABL tank, design 1, assuming a tank beam of 76.4 feet. The dimensions and fluid weight of design 3 are in very close agreement with design 1, verifying the PABL tank design calculations. The calculated tank natural frequency of 0.619 is 11 percent more than the measured value reported by Zarnick, et al. (14). Design 5 is a check of the McMullen tank natural frequency using Reference 8 , and the dimensions of the McMullen tank, but neglecting tank vertical taper. The calculated natural frequency of 0.71 is 14 percent more than the measured value given in Reference 15. Reference 13 notes that the PABL tank performance is improved by decreasing tank beam and thus increasing tank natural frequency. The methods of Reference 8 thus appear questionable for sizing even a simple rectangular tank.

TABLE 9

Comparison of Calculated and Actual Characteristics
of Roll Tanks for the Sea Control Ship

Tank Design	Tank Beam	Tank Length	Fluid Depth	Fluid* Weight	Tank Natural Frequency-1/sec	$\frac{\omega_{ns}}{\omega_{nt}}$
1. NAVSEC-PABL (Rectangular)	76.4	40	5	372	0.555	1.06**
2. J. J. McMullen (C-Shape)	78/60	40/24	16	820	0.625	0.94**
3. Reference 12 (Rectangular)	76.4	40.5	5	376	0.619	0.95
4. Reference 1 (Rectangular)	76.4	40	5	372	0.525	1.12
5. Reference 12 (C-Shape)	78	40/24	16	845	~0.71***	0.83
6. Reference 1 (C-Shape)	78	40/24	16	845	0.627	0.94
7. Reference 1 (Rectangular)	78	40	7.72	587	0.625	0.94

*Tons of Navy Distillate
**Model test data from References 14 and 15
***Calculated using estimated damping structure

Tank design 4 is a check of the PABL tank natural frequency using the dimensions of that tank and the method of Reference 1. The calculated natural frequency of 0.525 is about six percent less than the measured value. Tank design 6 is a check, using Reference 1, of the McMullen tank design dimensions, but neglecting tank vertical taper. The calculated natural frequency of 0.627 is in excellent agreement with the measured value. Comparing designs 6 and 7 illustrates the weight reduction that can be achieved, for a given tank GM or free surface loss and tank natural frequency, by using a rectangular rather than a C-shaped tank. The method of Reference 1 thus appears much better for estimating tank natural frequency and tank sizing than does the method of Reference 12.

VII. REVIEW OF STATE-OF-THE-ART IN PREDICTING ROLL MOTIONS WITH AND WITHOUT ROLL TANKS

A number of methods are available for predicting ship roll motions with and without roll tanks. These include theoretical, experimental and hybrid methods which combine bench test data with computer simulation. Experimental and hybrid methods have been considered earlier in this report. In this section available theoretical methods and computer programs for predicting roll motions with and without tanks are considered, and several methods are evaluated on the basis of comparisons of predicted and measured roll motions for several ships. Methods and computer programs which do not include roll tanks are of interest since these can usually be readily modified to include a roll tank.

Theoretical Methods for Predicting Roll Motions

A number of theoretical methods are available for calculating ship roll motions. Some of these include a roll tank. Most methods are based on linear equations of motion, some incorporating an equivalent linearized damping term. Other methods include non-linear ship or roll tank terms.

Available theoretical methods can be conveniently classified according to the degrees-of-freedom of ship lateral motion considered. The one-degree-of-freedom (1 DOF) approach considers only roll, the two-degree-of-freedom (2 DOF) approach roll and sway and the three-degree-of-freedom (3 DOF) approach roll, sway and yaw. Recent five-degree-of-freedom (5 DOF) methods, such as that described by Salvesen, et al. (16),

consider three-degrees-of-lateral-freedom which are uncoupled from the two-degrees-of-vertical-freedom (pitch and heave), and are thus really 3 DOF methods for calculating lateral motions. In each case the tank can be treated as an additional degree of freedom, as done by Webster (2) or as applied moments and forces, as done by Conolly (17).

The comparisons of calculated and measured roll motions for the Sea Control Ship discussed later in this section indicate that 1 DOF methods are generally not adequate, and that there is probably little to choose between 2 DOF and 3 DOF approaches. These conclusions are not surprising since roll-sway coupling is known to be important while roll-yaw coupling is generally not important. For ships which are highly asymmetrical, such as ships with large sonar domes, roll-yaw coupling could be important in oblique seas. Additional comparisons are required to insure that roll-yaw coupling can be safely neglected and a 2 DOF approach used. It should also be noted that sway damping, as well as roll damping, can have a significant effect on roll motions and must therefore be estimated with care.

The method described by Conolly (17) is an example of a 1 DOF approach. This method was originally applied to ships with active fin stabilization but can be readily extended to ships with roll tanks, as done at NSRDC. Zarnick, et al. (9) describes a 2 DOF approach which includes an applied force and moment due to a roll tank, although no means is provided for calculating this force and moment. Salvesen, et al. (16) and Raff (18) describe 3 DOF approaches for calculating roll motions

without roll tanks. Webster (2) describes a 3 DOF approach for calculating roll motions for ships with roll tanks, using either linear or non-linear equations of motion. The other methods all use only linear equations of motion. These methods can all be considered state-of-the-art methods, although all are not equally sophisticated.

Available Computer Programs for Predicting Roll Motions

A number of theoretical methods are available for predicting roll motions with and without roll tanks, but only two computer programs for predicting roll motions, both for ships without roll tanks, are generally available. These programs are the 5 DOF program SCORES described by Raff (18) and the MIT 5 DOF seakeeping program described by Steen (19). The treatment of roll in both programs is not wholly state-of-the-art, one particular deficiency being the absence of viscous effects in sway damping. The treatment of roll appears to be somewhat better in the MIT program than in SCORES.

Computer programs based on the 1 DOF method of Conolly (17) for ships with roll stabilizers, and the method of Salvesen, Tuck and Faltinsen (16), for ships with no stabilizers, have been developed at NSRDC. While these programs are not generally available, they are probably available to NAVSEC. The method of Reference 16 appears somewhat better for predicting roll than that used in the other two 5 DOF programs.

The methods of References (16-19) are based on linear equations of motion and frequency-domain solutions. Motions in irregular seas are calculated using the techniques of linear

superposition. These programs can be readily modified to include a roll tank if a linear tank equation of motion and linear tank-ship coupling terms are used. Tank damping can be treated by equivalent linearization, as described by Webster (2). Important non-linearities such as tank saturation cannot be considered in such frequency-domain solutions, however. Existing programs which are modified to include a linear treatment of the tank will thus be useful only for cases of moderate roll motions, where little tank saturation occurs.

A computer program which solves the non-linear equations of motion in the time-domain is required for proper analysis of passive roll tank performance and is absolutely essential for active and controlled-passive tanks. A methodology for solving the coupled 3 DOF equations of lateral motion in the time domain is given in Reference 2.

Comparison of Calculated and Measured Roll Motions of the Sea Control Ship with and without Roll Tank

A detailed comparison of measured and predicted motions has been made for the Sea Control Ship for several reasons. These include the size of the ship model (17 foot length) and tank model (2.3 foot beam) which should insure reasonable accuracy of the data, the extensive scope of the tests in regular waves and the availability of predicted performance based on two methods developed at NSRDC. Comparable data and calculations are not available for any other ship.

The roll motions of the Sea Control Ship have been predicted by NSRDC using the oscillating table simulation facility

described by Zarnick, et al. (13) and the approach of Conolly (17). These predictions have been previously compared with the model data in Reference 14. For this study motions have been calculated using the method described by Webster (2) and program SCORES (18). A comparison of all of these results is discussed in this section.

Ship Without Roll Tank - Figures 15-20 compare the measured roll motions in regular waves with no roll tank with the various predictions of roll motions for the same ship speeds and headings. Figures 15-17 are for a ship speed of five knots while Figures 18-20 are for a ship speed of 20 knots. Figures 23-26 compare predicted roll motions in irregular waves representing sea states 5 and 6 for ship speeds of five and 20 knots.

From Figures 15-17 it can be concluded that the predictions made using the NSRDC analog simulation are slightly better than the predictions made using the method of Reference 2 (labeled HYDRONAUTICS) and are significantly better than the predictions made using the methods of References 17 (labeled Conolly) and 18 (labeled SCORES) for a ship speed of five knots. Conolly's method is particularly bad for the 60 degree (quartering) heading angle while SCORES badly overestimates roll resonance for all cases. From Figures 18-20 it can be concluded that the predictions made using the method of Reference 2 and those made using the NSRDC analog method are in equally good agreement with the data. The predictions made using SCORES are too large near resonance. There is clearly no real difference in the predictional capabilities of the NSRDC analog (2 DOF) and the HYDRONAUTICS digital (3 DOF) methods for the Sea Control Ship

in regular waves. Both of these methods are, however, clearly superior to the other two methods considered.

The comparisons of Figures 23-26, for which there are no model data, indicate general agreement between the calculations made using the NSRDC analog method and the HYDRONAUTICS digital method. The predictions made using program SCORES are also generally in agreement with the other predictions, although the maximum RMS roll angles are (except at five knots and sea state 5) significantly larger than those predicted using the other methods. In the absence of model data the only conclusion that can be reached from these comparisons is that program SCORES is probably less suitable than the other methods.

Ships With Roll Tanks - The comparisons of predicted and measured roll motions in regular waves with PABL roll tank indicate that the predictions made using the HYDRONAUTICS method are in somewhat better agreement with the model data than are the predictions made using the NSRDC roll table or oscillator simulation facility. The differences in the predictions are most significant near the roll resonance and for long wave lengths. The HYDRONAUTICS predictions were made using the linear, frequency domain method of Reference 2. This approach is suitable because of the small roll response at these conditions and the resulting absence of tank saturation. Unfortunately no data are available for heading angles of 60-70 degrees, where the maximum roll response usually occurs with a roll tank.

The comparisons of Figures 27-30, for which there are again no model data, indicate considerable differences in the predictions for heading angles between 50 and 90 degrees. Agreement between the NSRDC simulation facility predictions and the HYDRONAUTICS predictions is generally good for other heading angles. The predictions made using Conolly's method are in agreement with the other predictions only for heading angles of 60 degrees or less.

The maximum response at heading angles of 60 to 80 degrees, predicted by the NSRDC simulation, is consistent with most experience with passive roll tanks, and is probably due in large part to the low tank angle capacity or saturation angle (approximately seven degrees). The HYDRONAUTICS calculations were made using a frequency domain solution which does not consider tank saturation. Additional calculations were carried out for the case of Figure 30 using the non-linear, time-domain method of Reference 2. These calculations resulted in a significant increase in predicted roll motions at heading angles between 50 and 90 degrees, although the resulting predictions are still significantly less than those obtained from the NSRDC simulation. It is clear that non-linear, time-domain methods of Reference 2 must be used for predicting roll motions in real seas.

Comparison of Calculated and Measured Roll Response for the SL-7

The SL-7 Containership is of interest because it is a high speed, very fine hull form. Model tests of this design have been carried out at Davidson Laboratory and NSMB. Only the results of the Davidson Laboratory tests, as reported by Dalzell

and Chiocco (20) are readily available. A comparison of these data and predictions made using programs SCORES has been reported by Kaplan, et al. (21). Calculations have been made for this study using a frequency domain, 3 DOF computer program based on the methods of Reference 2.

Model tests have been carried out and roll response reported for 25 knot ship speed, for heavy (47,686 tons) and light (41,367 tons) load conditions, and for 30 and 60 degree heading angles. The measured and calculated roll responses for these conditions are compared in Figures 31-34. No calculations are reported in Reference 21 for the case of Figure 34. The agreement between the data and the SCORES calculations ranges from good (Figure 33) to poor (Figure 31). The agreement between the data and the HYDRONAUTICS calculations range from good (Figure 32) to fair (other cases). On average, the HYDRONAUTICS calculations are in better agreement with the data than are the SCORES calculations. For the case of Figure 32, the choice of damping ratio is very important, while for the other cases it is not important. The best agreement is obtained with the lower damping, which is considered to be the more realistic.

The large discrepancies between predictions and measurements are probably due, in part, to the difficulties encountered in conducting the model tests (20). Significant scale effects are likely for the five foot long model used in these tests. The small size of the model bilge keels probably makes them somewhat ineffective, with a resulting increase in roll motion; this might explain why measured roll motions are larger than predicted motions.

Illustrative Comparisons for Fine Ships

Two comparisons of predicted and measured roll motions of fine hulls at high speed and in oblique seas are presented to illustrate potential shortcomings of both theoretical methods and ship model tests for predicting roll motions. Both comparisons are for ship without roll tanks.

Figure 35 compares predicted and measured roll motions for a recent destroyer type ship with a 0.49 block coefficient, operating at a speed-length ratio of 1.1, at a 70° heading angle and in irregular waves. The model length of 21 feet should insure against any scale effects. The predictions were made using the methods described in Reference 2. The calculated peak angles agree with the data, but the measured response is much more broad-banded.

The model data in Figure 35 indicate significant roll motions up to a frequency of encounter of 0.74. Figure 36 shows that this frequency of encounter occurs, at the stated ship speed and heading angle, only for a wave length of 24 feet for the actual ship. Since little or no rolling will occur at such wave lengths, it seems likely that the actual heading angle was at least 75 rather than 70 degrees. Heading angles for tests in irregular, oblique waves may therefore have to be treated as nominal values.

The width of the response amplitude operation (RAO) curve may reflect nonlinear damping or the behavior of frequency of encounter shown in Figure 36. It is concluded that care should be taken in interpreting roll motions in regular, oblique waves deduced from roll motions in irregular, oblique waves.

Figure 37 from a report by Baitis and Wermter (22) indicates how theoretical methods such as those of Salvesen, et al. (16) tend to over-predict peak roll response at high speeds, probably due to the underestimation of roll or sway damping. These results are for a fine hull form (0.485 block coefficient) at a speed length ratio of 1.55. The agreement between model data and predictions are significantly better at a speed-length ratio of 0.50, but are significantly worse for some other GM's and bilge keel sizes considered. These comparisons illustrate that state-of-the-art seakeeping theories can be inadequate for predicting ship roll motions at high speeds.

VIII. AREAS NEEDING ADDITIONAL WORK

While it might appear from reading the preceding section that means for designing roll tanks and predicting the roll motions of ships with and without roll tanks are fairly well in hand, additional work is needed, if not required, in a number of areas. These include roll motions predictions in quartering seas, minimum acceptable model size and correlation of ship data, model data and theoretical predictions. In this section these areas are discussed.

Existing theoretical and hybrid methods are capable, for most ship operating conditions, of reproducing model roll data for ships with and without roll tanks with reasonable accuracy. The area where agreement appears least satisfactory is high speed operation in oblique stern seas. This is significant because maximum rolling motions in irregular seas often occur at these conditions, particularly when passive roll tanks are used.

Observed discrepancies between model data and predictions may be due to a number of causes including:

1. Inadequate model size
2. Unrealistic model constraint during tests
3. Failure to hold model on desired oblique heading or unrealistic model rudder action.
4. Failure to properly account for non-linear ship roll damping or viscous contributions to sway damping in calculations.
5. Failure to account for rudder action and heading changes in calculations.

Items 1, 2 and 4 are applicable to tests at all heading angles while Items 3 and 5 are applicable only to tests in oblique seas. From the discussions in this report it would appear that Items 1, 3 and 4 are likely to be the most significant.

It is clear that systematic model experiments are needed to determine minimum acceptable model size and minimum test time required to obtain adequate model data in both regular and irregular waves. Tests of models having lengths of 15 to 20 feet would be desirable.

It would be highly desirable to carry out much more detailed comparisons of measured and predicted roll motions than presented in this report, particularly for ships with roll tanks. These comparisons should include predictions made using a number of theoretical methods and should include an investigation of the effect of the rudder and viscous contributions to sway damping.

Comparisons of full scale data with model data or theoretical predictions are very limited. One great difficulty in any such comparison is the accurate determination of the sea conditions in the full scale tests. Conolly (17) indicates generally good agreement between trial data and prediction, using a one-degree-of-freedom method, for a small, fine ship (Ship A) operating with and without fin stabilizers. Brunsell, et al. (23) present comparisons showing significant differences between full scale and model measurements of roll motions of a weathership with and without operating roll tanks. The largest discrepancy is in the roll band-width rather than in the peak

roll motions. Agreement improves with increasing ship speed. Agreement is somewhat better with the roll tank in operation. A long-term project to correlate ship motions, model motions and theoretical motions predictions for the SL-7, which has no roll tanks, is currently being carried out for the Ship Structure Committee of the National Science Foundation. It is likely that analysis of the SL-7 ship data will not be completed for two or more years.

Additional comparisons of full scale data with model data and predictions for modern naval ships and ships with roll tanks are clearly needed to verify the adequacy of current model test procedures and theoretical methods. The difficulties in obtaining the full scale data should not be underemphasized, however.

IX. CONCLUSIONS

While this report is concerned primarily with defining methods for designing roll tanks, at the contract design level, certain tentative conclusions stated in the report bear repeating. These include:

1. The design methods given in the Navy Design Data Sheet DDS 9290-4 for roll tanks do not appear to be adequate.

2. The design methods given in the Level I Report, Reference 1, lead to tank designs which are similar to current commercial design practice.

3. One-degree-of-freedom methods, such as that proposed by Conolly, do not appear wholly adequate for predicting roll motions with or without roll tanks, particularly in oblique, stern seas.

4. The roll table-simulation method developed at NSRDC and the three-degree-of-freedom method described by Webster (2) appear to predict roll motions, with and without roll tank, about equally well. For motions in irregular waves, it is necessary to use the non-linear, time-domain solution of Reference 2.

5. Program SCORES does not appear to predict roll motions as well as the methods described in Item 4 above.

It should be noted that all of these conclusions are based on limited data and results, and must therefore be considered as tentative.

X. REFERENCES

1. Miller, Eugene R., John J. Slager and William C. Webster, "Phase I Report on the Development of a Technical Practice for Roll Stabilization System Selection," NAVSEC Report 6136-74-280(HYDRONAUTICS Report 7401.06-1) October 1974.

2. Webster, William C., "Analysis of the Control of Activated Antiroll Tanks," Transactions of the SNAME, Volume 75, 1967.

3. Bell, John and William P. Walker, "Activated and Passive Controlled Fluid Tank System for Ship Stabilization," Transactions of the SNAME, Volume 74, 1966.

4. Chopra, H., and J. Dalzell, "Controlled Passive Tank Analysis for the U.S.C.G. Oceanographic Research Ship (WHEO)," HYDRONAUTICS, Incorporated Technical Report 766-2, February 1968.

5. Dalzell, J. F., W. C. Webster and H. S. Chopra, "Analysis of Controlled Passive Roll Stabilization System for Glomar XI," HYDRONAUTICS, Incorporated Technical Report 832-1, February 1968.

6. Webster, William C., and P. Dogan, "An Analysis of the Control of Activated Anti-Roll Tanks," HYDRONAUTICS, Incorporated Technical Report 490-2, December 1966.

7. Webster, William C., "The Design of Tanks for Use in an Activated Tank System," HYDRONAUTICS, Incorporated Technical Report 490-3, July 1967.

8. Department of the Navy (U. S.), "Design Data Sheet DDS 9290-4, Passive Anti-Roll Tanks," September 1962.

9. Zarnick, Ernest A. and Jack A. Diskin, "Modelling Techniques for the Evaluation of Anti-Roll Tank Devices," Paper VIIC-1 presented at Third Ship Control Systems Symposium, September 1972.

10. Swaan, W. A., "The Effect of Model Size on Rolling in Waves," Contribution to the Tenth ITTC Conference, July 1963.

11. Martin, Milton, "Roll Damping Due to Bilge Keels," Report of the Iowa Irstitute of Hydraulic Research, November 1958.

12. Department of the Navy (U.S.), "Design Data Sheet DDS 9110-3, Structural Design of Flat Plating and Stiffeners Subject to Water Pressure," November 1955.

13. Zarnick, Ernest A., W. G. Meyers and N. K. Bales, "Validation of Technique for Simulating Anti-Roll Tank Behavior in Waves," Paper presented at 1974 Summer Computer Simulation Conference, Houston, Texas.

14. Zarnick, Ernest E., et al., "Sea Control Ship Anti-Roll Tank (PABL) Evaluation," NSRDC Ship Performance Department Evaluation Report 496-H-05," January 1974.

15. John J. McMullen Associates, Inc., "Final Technical Report Oscillating Table Model Tests Stabilizer Tank Sea Control Ship," Report 1221M, December 1973.

16. Salvesen, Nils, E. O. Tuck and Odd Faltinsen, "Ship Motions and Sea Loads," Transactions of the SNAME, Volume 78, 1970.

17. Conolly, J. E., "Rolling and its Stabilization," Quarterly Transactions of the RINA, Volume III, No. 1, January 1969.

18. Raff, Alfred I., "Program SCORES - Ship Structural Response in Waves," Ship Structure Committee Report SSC-230, 1972.

19. Steen, Atle, "5-D Seakeeping Program User's Manual," MIT, Department of Ocean Engineering.

20. Dalzell, J. F., and M. J. Chiocco, "Wave Loads in a Model of the SL-7 Containership Running at Oblique Headings in Regular Waves," Ship Structure Committee Report SSC-239, 1973.

21. Kaplan, Paul, Theodore P. Sargent and John Cilmi, "Theoretical Estimates of Wave Loads on the SL-7 Container Ship in Regular and Irregular Seas," Oceanics, Incorporated Technical Report 73-96, April 1973.

22. Baitis, A. E. and R. Wermter, "A Summary of Oblique Sea Experiments Conducted at the Naval Ship Research and Development Center," Appendix VIII to the Report of the Seakeeping Committee, Thirteenth ITTC Conference, 1972.

23. Brunsell, P.A., et al., "Correlation of Full Scale and Model Basin Stabilization Test Results for the Canadian Weather Ships," Quarterly Transactions of the RINA, Volume III, No. 1, January 1969.

APPENDIX A

DRAFT OF BRIEF TECHNICAL PRACTICES SHEET FOR ROLL STABILIZATION SYSTEM SELECTION - CONCEPTUAL AND PRELIMINARY DESIGN PHASES

Introduction

In order to incorporate satisfactory roll performance in the design of a new ship, it is first necessary to determine what the roll motion performance requirements are for this ship; the next step is to determine what, if any, roll stabilization system must be installed in the ship if those performance requirements are to be met. In some cases it may be concluded that a combination of roll stabilization systems are required or that there is no way in which to meet the specified requirements.

The major steps in the selection of a roll stabilization system up to the preliminary design level can be summarized as follows:

1. Definition of roll motion performance requirements
2. Definition of potentially suitable roll stabilization systems and estimation of roll performance with each
3. Review of performance requirements and revisions if necessary
4. Selection of the most suitable roll stabilization system
5. Documentation of all work for reference during contract design.

This document, which is based primarily on Reference 1, outlines methods for carrying out these steps and for estimating the gross size (area, volume or weight) of roll stabilization systems. These methods are suitable for the phases of design usually referred to as concept design and preliminary design.

Roll Motion Performance Requirements

A process for determining suitable roll motion performance requirements for a given ship design is discussed in some detail in Reference 1. This process is based on defining the required level of mission effectiveness of the ship and its component systems as a function of ship roll motions. The steps required to define this effectiveness are:

1. Identify any operational requirements, as defined in the Top Level Requirements (TLR) or Plan For Use (PFU), which the ship might not be able to meet because of ship roll motions.

2. Identify all areas of motion sensitivity (weapons systems, etc.) which affect the ability of the ship to meet these operational requirements and quantify these sensitivities, as far as possible, in terms of statistical roll motions quantities such as significant (one-third highest) or maximum (or one-hundredth highest) roll angle versus ship or subsystem performance degradation.

3. Develop a set of "ideal" roll motion performance requirements based on probabilities of exceedance of appropriate roll characteristics in the specified or assumed operating conditions (sea state, ship speed, heading) where "ideal" requirements are those which imply little or no degradation of ship operational performance (i.e., nearly 100% mission effectiveness).

4. Develop a set of "constrained" or "practical" roll motion performance requirements based on allowing greater probabilities of exceedance or assuming less stringent operating conditions than in the case of the "ideal" requirements.

The constrained or practical performance requirements are used either when the ship cannot meet its "ideal" requirements with available roll stabilization systems or to carry out trade-off studies of effectiveness versus stabilizer system cost, size, etc. The development of constrained or practical requirements is an iterative process in which alternate roll stabilization systems are posited and one "works backwards" to determine roll motion performance and the corresponding ship mission effectiveness values. This process must be repeated until an acceptable trade-off between ship effectiveness and ship and stabilizer cost, size, etc. is achieved.

Available Roll Stabilization Systems

A number of roll stabilization systems, including bilge keels, passive and active roll tanks and active fin stabilizers, which are attractive for naval applications, are described in Reference 1. The performance and design of each of these types at a level suitable for concept and preliminary ship design, is considered in the following sections.

Performance and Geometry of Stabilization Systems

Reference 1 presents methods for estimating the performance and required dimensions and/or weight of each stabilization

system as a function of key design parameters. Systematic calculations of rolling motions with and without stabilization systems are given in Figures 5-64 of Reference 1 for a range of significant wave heights (or sea states), ship heading angles (head seas to stern seas) and ship speeds (speed length ratios of zero to 1.2) for destroyer and auxiliary type ships. These results can be used to estimate roll stabilization system performance for any ship having a hull form similar to one of these two types. For significantly different hull forms, similar calculations must be made using a method such as that given in Reference 2. From these results, the key design parameters required to meet the roll motion performance requirements can be determined, as described in the following paragraphs.

Bilge Keels - These keels increase hull roll damping and hence reduce roll motions, and are particularly effective at low speeds, where bare hull damping is very small. Equations [2] and [3] of Reference 1 are used to estimate roll damping coefficient with and without bilge keels. The bilge keel contribution to damping is proportional to bilge keel area (Equation [2]). Roll motions with and without bilge keels can be estimated from Figures 5-9, 20-23, 35-38 and 50-53 of Reference 1 using appropriate damping ratios. Bilge keel size will usually be limited by considerations of vulnerability and added drag.

Roll Tanks - These may be of either free surface or U-Tube type (see Reference 1). For active tanks the U-tube type is clearly advantageous. For ships which are required to operate over a range of GM's, free surface tanks may be advantageous.

Tank performance, size and weight are determined by tank free surface loss, tank natural frequency, tank angle capacity and, to a lesser extent, tank damping. For preliminary design purposes it is usually reasonable to assume the tank has satisfactory damping and adequate angle capacity. Equations [4] and [14] can be used to calculate required tank dimensions for a given tank GM loss. Equations [6] and [16] can be used to calculate tank height required to give the desired tank angle capacity. Equations [8], [17] and [18] of Reference 1 gives tank natural frequencies as a function of tank dimensions. Equations [12] and [21] give corresponding fluid weights. Tank dimensions are usually selected to give ratios of tank to ship (without tank) natural frequencies for optimum or near optimum tank performance (1.06-1.10 for passive tanks and 1.30-1.40 for active tanks). Figures 9-12 and 24-27 (passive tanks) and 16-19 and 31-34 (active tanks) of Reference 1 can be used to estimate roll motions as a function of K_{st} (ratio of tank GM reduction divided by ship GM without tank). All of these results are for optimum tank frequency and damping. The desirability of using a large K_{st} or tank free surface loss, particularly for lower speeds, is obvious from these figures. Tanks are most effective at low speeds but are effective at almost all conditions. Increasing K_{st} results, however, in larger tank dimensions and fluid weight. It is typical to use a K_{st} of 0.2 to 0.3 for passive tanks and a somewhat smaller value for active tanks. A tank angle capacity of 12 to 15 degrees should be used to avoid tank saturation.

Active Fins - The performance, size and power of fin stabilization systems are determined primarily by fin static

angle which is proportional to fin area and lift coefficient. Figures 13-15, 28-30, 43-45 and 58-60 of Reference 1 can be used to estimate roll angles as a function of fin static angle. From these results it is clear that fins are effective only at higher speeds. Fin static angle can be estimated using Equations [23] and [24] of Reference 1. Lift and drag can be obtained using References 3 or 4 of Reference 1. Fin span and area, and hence roll reduction due to fins, will usually be limited by vulnerability, required storage space (for retractable fins), weight and/or power. Typical powers and weights can be estimated as a function of fin area using Table 6 of Reference 1.

Selection of Roll Stabilization System

The selection of the type of roll stabilization system will usually be a trade-off between performance (ship roll motions or roll reduction), reliability, cost, weight, required space and added drag. For ships with low initial GM, roll tanks will not be considered. Reliability and associated performance degradation, due to system malfunction, is probably the most important item, besides performance, to be considered in trade-off studies. In some cases severe restrictions on available space will limit the number of feasible systems. When roll performance is important at both low speeds and high speeds, it may be desirable to use combinations of systems such as bilge keels and active fins or passive tanks and active fins. It will often be appropriate, during preliminary design, to use computer methods to make parametric studies of stabilizer performance. The results presented in Reference 1 are illustrative of such parametric studies.

Documentation

It is essential that the work on roll motions performance requirements and on selection of the roll stabilization system carried out during concept and preliminary design be properly documented for use during contract design.

APPENDIX B

DRAFT OF BRIEF TECHNICAL PRACTICES SHEET FOR ROLL STABILIZATION SYSTEM (ROLL TANK) SELECTION AND DESIGN - CONTRACT DESIGN PHASE

Introduction

The Brief Technical Practices Sheet for Contract and Preliminary Design Phases (Appendix A of this report) describes the selection of the appropriate type of roll stabilization for a given ship design. This Brief Technical Practices Sheet described the selection and design of an anti-rolling or roll tank at the contract design level.

The major steps in the selection and design of a roll tank during ship contract design can be summarized as follows:

1. Selection of an active, passive or controlled-passive type tank.
2. Selection of a free surface or U-tube tank.
3. Selection of tank location in the ship and resulting constraints of tank geometry.
4. Detailed design of the tank using computer calculations and model tests.
5. Preparation of design drawings and operations manual.
6. Validation of predicted tank performance.

This document, which covers all of these areas, is based primarily on the material contained in the present report.

Selection of Active, Passive or Controlled-Passive Roll Tank

For most applications a passive roll tank should be used because of its low cost, high reliability, need for almost no maintenance and known performance at all times. Passive tanks have been widely used and available methods for designing and

predicting the performance of such tanks have been validated. Passive tanks are usually larger (with greater fluid weight and GM loss) than active and controlled passive tanks, and sometimes produce no roll reduction or even a roll increase in stern quartering seas.

Controlled-passive tanks are more costly and less reliable than passive tanks, but can often be made somewhat smaller for equal roll reduction, and generally have good performance at all ship operating conditions. Since the control and mechanical systems are relatively simple and operating powers are small, these tanks may be attractive for cases where passive tanks have poor performance at some ship operating conditions or where allowable GM loss is severely restricted. Controlled-passive tanks will have poor performance if the control system fails. A number of applications of controlled-passive tanks exist, and the feasibility of such tanks has been demonstrated.

Active tanks are much more costly to build and to operate than other types of tanks. This increased cost, coupled with the increased maintenance requirements and poor performance of the tank if the control system or pump fails, makes such tanks generally unattractive for naval applications. The absence of shipboard applications of active tanks and validation of predicted performance for such tanks are further deterrents to the use of active tanks.

Selection of Free-Surface or U-Tube Roll Tank

The selection of the type of tank depends on several considerations including:

1. Whether the tank is passive, active or controlled-passive.
2. Requirements for operation over a range of ship GM's or with varying fluid level in the tank.
3. The size and shape of available spaces for the roll tank.

Either type of tank can be used for passive tanks while U-tube tanks are required for active and controlled-passive tanks. When operation over a moderate range of GM's (say with a ratio of maximum to minimum GM of two or less) is required, free surface tanks are attractive because the tank natural frequency can be "tuned" to obtain optimum tank performance at all GM's by varying tank fluid depth. This method results, however, in great tank depths and fluid weights when the range of GM's becomes too large. For large ranges of GM's, the use of a "tuned-pair" of U-tube tanks, is probably more attractive. When tank fluid level must be changed, as in cases where the fluid is to be fuel or fresh water, U-tube tanks should be used as their natural frequency is not significantly affected by fluid level. Sometimes the shape of available spaces will make one type of tank more attractive.

Selection of Tank Location

Ideally the roll tank should be located near midships and as high in the ship as possible, and should have a sufficient planform size and depth to achieve the desired free surface or GM loss and tank angle capacity. For most designs, however, tank location and size will be dictated, at least to some extent, by

ship arrangements and available ship spaces. While it is desirable to locate the tank between 0.35L aft and 0.25L forward of midships, it is usually more important to pick a location which gives the desired or best possible tank size rather than one which is at a certain longitudinal or vertical position. Tank location may be selected using computer trade-off studies, as described in the next section.

Detailed Tank Design Using Computer Calculations and Model Tests

At the end of preliminary design only the overall design characteristics of a roll tank, such as GM loss, overall dimensions and fluid weight will have been determined. During the contract design phase it may be necessary to modify these characteristics due to decreases in allowable tank GM loss, restrictions on tank dimensions or locations, etc. An important part of the roll tank contract design is to refine and optimize tank design, taking into account necessary trade-offs between cost, weight, location and tank effectiveness. This refinement leads directly to the final tank design. Both computer calculations and model tests are generally used for design refinement and detailed tank design, as described in detail in Section III of this report.

Computer calculations, based on a method such as that of Reference 2, can be used to study the effect on roll tank performance and size of all important tank design parameters, including GM loss, tank natural frequency, tank damping ratio, and tank location. Tables 1-6 show the effect of such parameters. Table 7 illustrates how the methods of Reference 2 can be used to evaluate non-linear effects and tank angle capacity.

During the design process, bench tests, as described in detail in Appendix D, are used to determine necessary modifications to tank dimensions, damping devices, etc., required to obtain the desired tank natural frequency and damping ratio. Tests of a ship/tank model are usually not used during the design process but only during validation of tank performance.

Preparation of Design Drawings and Operations Manual

Once the tank design has been completed it is necessary to prepare detailed drawings of the tank. These drawings should include:

1. All tank scantlings and structural members on tank walls.
2. All piping and valving associated with the tank.
3. Internal damping devices.

Section IV of this report discusses means for estimating scantlings, and tank damping and necessary tank piping and valving. For controlled-passive or active tanks, drawings showing details of valve controls, pumps, etc. must also be prepared.

When a decision to build the ship has been made, an Operations Manual must be prepared. The information that should be included in this Manual is discussed in Section IV of this report. For active and controlled-passive tanks it will also be necessary to prepare a Maintenance Manual.

Validation of Tank Performance

Once the design of the tank is completed, it is essential that tank performance be validated for all ship operating conditions

of interest. This can be done using tests of a ship model with a model tank, using bench tests with ship motion simulation or using a combination of the two methods.

The use of ship/tank model tests, as described in Appendix F, while inherently attractive, is often not attractive for validation because:

1. A large ship model (length of 15 feet or more) and tank model (tank beam of 2 feet or more) is required.
2. The model must be tested in irregular, oblique seas, necessitating a large number of tests and a considerable expense.

If a large facility is available (such as the MASK at NSRDC) and cost is not a primary consideration ship/tank model tests will probably be used for validation.

A more probable method of validating roll tank performance will be to use bench tests of the tank, together with a computer simulation of ship motions. The bench tests may incorporate only roll, as described in Appendix D, or both roll and sway as in the NSRDC simulation facility described in Appendix E. With normal roll only bench tests it is essential to conduct ship/tank model tests at a few important operating conditions to verify the predicted tank performance or to suitably adjust the predictional techniques. Work currently in progress on the NSRDC simulation facility could result in this facility being used in the future in lieu of all model tests.

APPENDIX C

PERFORMANCE CALCULATIONS FOR CONTROLLED-PASSIVE TANKS

The methods for calculating the performance of passive and active roll tanks given by Webster (2), can be readily extended to controlled passive tanks. Dalzell, et al. (5) presents the necessary modifications for a controlled-passive tank in which the air flow in two crossover pipes is controlled by valve action. The following excerpts from Reference 4 summarize these modifications and the solution of the modified equations of motion.

Mathematically, the control and the valve operating apparatus can be simulated as follows:

$$L_g = \ddot{\varphi} + a\omega_s\dot{\varphi}$$

where L_g = net control signal

ω_s = ship roll frequency

$\dot{\varphi}$ = roll velocity

$\ddot{\varphi}$ = roll acceleration

a = a constant

The sense of roll and motion of tank fluid is defined so that positive roll is starboard deck edge down, and positive tank motion implies decrease in fluid in the starboard reservoir:

When L > 0 (positive)

Air is permitted to flow from port to starboard.

When L < 0 (negative)

Air is permitted to flow from starboard to port.

When L switches from positive to negative, the valves change instantaneously.

Once air is prevented from flowing from port to starboard (say), further motion of water from the starboard to port reservoirs is impeded by the compression of air in the port reservoir and expansion of air in the starboard reservoir. Defining the terms in Figure 38:

τ = "Tank Angle" = 0 (when water in each reservoir is at same level) and $\approx$ (amount water level falls in starboard reservoir)/R

τ_{max} = Maximum tank angle or saturation angle

τ_s = Tank angle at which valves stop flow of air

R = Distance from ship $\mathbb{C}$ to $\mathbb{C}$ of reservoir

Assuming adiabatic compression of air when tank fluid moves from $\tau = \tau_s$ to $\tau = \tau$: The air pressure difference between reservoirs becomes:

$$\Delta p = 2p_o K (\tau - \tau_s) \left[\frac{\tau_{max}}{\tau_{max}^2 - \tau_s^2} \right]$$

where p_o = Atmospheric pressure

K = 1.4

After conversion of the above pressure difference to head and non-dimensionalizing to conform to the development by Webster (2), there result the following terms to be added to the left-hand side of the roll and tank equations, Equations 24 and 25, of Reference 2:

Terms due to air compression to be added to left-hand side of:

$$\text{Roll, Eq. 24:} \quad -k_{st}(\text{AIRSP})\,[D(L_g,\dot{\tau})](\tau-\tau_s)$$

$$\text{Tank, Eq. 25:} \quad (\text{AIRSP})\,[D(L_g,\dot{\tau})](\tau-\tau_s)$$

where:

$$\text{AIRSP} = \text{airspring} = \frac{p_o K}{\rho g R \tau_{max}}$$

$$D(L_g,\dot{\tau}) = \begin{cases} \left[\dfrac{1}{1-\left(\dfrac{\tau_s}{\tau_{max}}\right)^2}\right] & \text{If the pressure difference between reservoirs is such that air could flow only in a direction opposite to that commanded by the control.} \\ (0) & \text{If the air in the crossover can flow or is flowing in the direction commanded by the control.} \end{cases}$$

Because of the nonlinearities in both the control and the additional terms added to the roll and tank equations, only the Nonlinear, Time Domain Computer Simulation described in Reference 2 could be used in the present case. In this method, a 4th order Runge-Kutta integration is performed on the equations of motion, including nonlinearities in tank damping, and saturation (water at the top of one reservoir). The terms outlined above were inserted in the computer programming and the logic of the control was incorporated in the middle of each 0.6 sec Runge-Kutta time step.

In the controlled-passive tank, all other mathematical parameters defining the tank are of the same type as those

defining a pure passive tank; and suitable estimates of loss in $\overline{GM}$, tank natural frequency and quadratic tank damping are made from the geometry of the tank.

APPENDIX D
BENCH TEST PROCEDURES

There are two popular methods available for performing bench tests: oscillating table tests and impulse tests. The oscillating table tests are required for free-surface tanks (Flume-type). The simpler impulse type test is more appropriate for U-tube tanks. The procedure for each method is described below.

Oscillating Table Tests

In this test procedure, the tank model is installed on an oscillating table as shown in Figure 39. The pivot for the table is usually selected as the scaled location of the so-called roll center. That is, the pivot is located at a distance above or below the bottom of the tank model which corresponds (in the scale of the tank model) to the vertical location of the roll center. The vertical position, $\overline{KR}$, of the roll center above the keel, shown in Figure 11, is given by:

$$\overline{KR} = (\overline{KG} \cdot M + \overline{KA} \cdot M_{as})/(M + M_{as})$$

where $\overline{KG}$ is the height of the ships' center of gravity above the keel

$\overline{KA}$ is the vertical location of the line of action of the lateral added mass forces due to sway. A usual estimate is that $\overline{KA} \approx \overline{KB}$, the vertical location of the center of buoyancy.

M is the mass of the ship

M_{as} is the lateral added mass due to sway.

In practice, the roll center is located about half-way between the center of gravity and the center of buoyancy. It is the point about which the ship rolls in the absence of any sway forces. For instance, if a ship model were held at given angle of heel in a calm tank of water and then released (without imparting any sway forces) the model would roll about the roll center. Since this is not an untypical experiment for determining the magnitude of the roll damping, it is of interest to choose this point as a pivot for the oscillating table tests. In this way, the tank will undergo the same motions as it would if it were installed in a ship model undergoing a still-water, roll damping test (as described above). It should be noted that in actual rolling in waves, the ship does not roll about the roll center, or for that matter about any one point. Therefore, care must be exercised in interpreting oscillating table tests.

The tests themselves are performed in two steps. In each of these steps the table is oscillated sinusoidally and the tests are performed until a steady state has been reached. The table is assumed to be oscillating at an angle, $\alpha(t)$, given by

$$\alpha(t) = \alpha_o \cos \omega t \ .$$

The tests are performed for a range of frequencies, ω, above, below and near the tank's resonant frequency.

In the first step, the tank model is installed on the oscillating table and the pivot point is adjusted to the proper height (as discussed above). The empty tank and table are oscillated at several different frequencies and amplitudes. A

time history of the moment required (formed by multiplying the force measured at the load cell by the transverse distance between the load cell and pivot) is recorded as a tare. That is, this is the moment required to move the table, bearings and tank model structure under these conditions and this extraneous moment must be subtracted from any measurements taken with the tank operating. For a given moment, the tare moment, $M_1(\omega)_1$ is:

$$M_1(\omega) = a_1(\omega) \cos \omega t + b_1(\omega) \sin \omega t$$

In the second step, the tank is filled to the desired level and the same tests are performed. For a given frequency, the moment measured in these tests is given by

$$M_2(\omega) = a_2(\omega) \cos \omega t + b_2(\omega) \sin \omega t$$

From the results of these two tests a corrected moment can be computed, given by:

$$M_c(\omega) = a_c(\omega) \cos \omega t + b_c(\omega) \sin \omega t$$

where $$a_c(\omega) = a_2(\omega) - a_1(\omega) - \alpha_o W_t h_t \left(1 - R_t^2 \omega^2 / g h_t\right)$$

$$b_c(\omega) = b_2(\omega) - b_1(\omega)$$

W_t is the weight of fluid in the tank

h_t is the height of the pivot above the centroid of the volume of the tank fluid

g is the acceleration of gravity

R_t is the radius of gyration of the fluid in the tank, measured about the pivot axis.

The term $w_t h_t \alpha_o (1 - R_t^2 \omega^2 / g h_t)$ in the expression for a_c represents the moment (in-phase to, and proportional to the table angle) which would arise if the tank fluid were frozen in position and not allowed to move. As a result, the corrected moment, $M_c(\omega)$, is the moment which the fluid in the tank exerts due to its motion.

As a check on the results, at very low frequencies, the value of $a_c(\omega)$ should approach $\alpha_o \rho_t g I_t$ and the value of $b_c(\omega)$ should approach zero. Here, ρ_t is the density of the fluid in the tank; g is, as before, the acceleration of gravity and I_t is the moment of inertia of the free surface of the tank. The combination $(\rho_t g I_t)$ is just the free surface loss of the tank.

For subsequent calculations, it is somewhat more convenient to consider the non-dimensional form of the corrected moment given by

$$\overline{M}_c(\omega) = M_c(\omega)/(\alpha_o \rho_t g I_t) = \overline{a}_c(\omega) \cos \omega t + \overline{b}_c(\omega) \sin \omega t$$

where $$\overline{a}_c(\omega) = a_c(\omega)/\alpha_o \rho_t g I_t$$

$$\overline{b}_c(\omega) = b_c(\omega)/\alpha_o \rho_t g I_t$$

Therefore, $\overline{a}_c(\omega)$ approaches unity as ω approaches zero.

Several analyses of the functions $a_c(\omega)$ and $b_c(\omega)$ can be made. A few of the simpler approaches are outlined below.

Tuning - It is usually assumed that the dynamics of the tank fluid are equivalent to a second-order oscillator. If this is true, then at the natural frequency of the system, the

in-phase moment is zero and the out-of-phase moment is close to its maximum. These two conditions then can be used to determine the natural frequency of the tank. It is typical that the frequency for which $\bar{a}_c(\omega) = 0$ is not quite the same as that which yields a maximum value of $b_c(\omega)$. Since the determination of the maximum is somewhat more difficult than a zero crossing, usually the condition that $\bar{a}_c(\omega) = 0$ is used to determine the tank resonant frequency, ω_t.

Damping - As with the tuning, the damping can be determined using the second-order system analogy. The non-dimensional damping ratio of the tank, ζ_t, is then given by

$$\zeta_t = -1/[2\bar{b}_c(\omega_t)]$$

This damping is positive since it is typical that $\bar{b}_c$ is negative.

Non-linear Effects - If the motions in the tank are linearly dependent on the motions of the ship (or in this case, the motions of the table) then $\bar{a}_c(\omega)$ and $\bar{b}_c(\omega)$ should be constant for all values of α_o. Whereas it is usually true that the value of $\bar{a}_c(\omega)$ is independent of α_o, it is typical that value of $\bar{b}_c(\omega)$ becomes smaller when α_o becomes larger. Since this is the term which reflects the damping in the tank, we see from above that this behavior corresponds to an increase in tank damping with an increase in roll angle. This is due principally to the fact that most of the damping in the tank is quadratic in nature.

Still-Water Ship Roll Performance - Perhaps the most meaningful interpretation of the results can be obtained by a

direct prediction of the stabilized performance of the ship itself. If it is assumed that the ship can be described by a second-order system for still water rolling, then one can show that the apparent stabilized ship non-dimensional damping at resonance, $\hat{\zeta}_s$, is

$$\hat{\zeta}_s = \zeta_s - \bar{b}_c(\omega_s) \cdot \bar{k}_{st}/2$$

where ζ_s is the non-dimensional damping coefficient of the unstabilized ship

$\bar{k}_{st} = \delta\overline{GM}/\overline{GM}$, the ratio of the free surface loss of the tank to the uncorrected metacentric height.

The magnification factor at resonance (the roll angle divided by the wave slope) is given by $(1/2)\hat{\zeta}_s$. Thus the roll reduction at resonance, P, afforded by the tank is given by

$$P = \zeta_s/\hat{\zeta}_s = 1/[1 - \bar{b}_c(\omega_s) \cdot \bar{k}_{st}/2]$$

It should be remembered that $\bar{b}_c$ is negative and therefore P is less than 1. The value of $\bar{b}_c(\omega_s)$ which should be used corresponds to an α_o equal to the resulting roll angle.

It should be noted in comparing the formula for the tank damping and that for the ship damping that small values of ζ_t lead to large values of ζ_s. In other words, a small tank damping leads to a large apparent ship damping at resonance. Of course, small values of tank damping will lead to roll resonances at other frequencies. This undesirable situation will occur only if $\bar{b}_c(\omega)$ has a very sharp peak at the ship 's

resonant frequency and therefore can be detected by inspection of the behavior of $\overline{b}_c(\omega)$ with frequency.

Impulse Tests

A simple and reliable alternative to oscillating table tests exists for the case of the U-tube stabilizing tank. The typical test set-up is shown in Figure 40. The model of the tank is filled with fluid to the desired level and the tank is tilted at an angle. A height gage is installed near the center of one of the vertical legs of the tank. At a given time, the tank angle is reversed impulsively (or as close thereto as possible) and the motion of the fluid in the one leg of the tank is recorded as a function of time. From continuity, knowledge of the flow in one part of a U-tube tank allows one to determine the flow in all of the other parts. The equation of motion of the tank fluid immediately after the impulse, expressed in terms of the tank angle, τ (and defined in Figure 11) is given by

$$\ddot{\tau} + 2\zeta_t \omega_{nt} \dot{\tau} + B\omega_{nt}^2 \dot{\tau}|\dot{\tau}| + \omega_{nt}^2 \tau = \omega_{nt}^2 \tau_o$$

where ω_{nt} is the natural frequency of the tank

ζ_t is the non-dimensional tank linear damping ratio

τ_o is the angle of the tank after the impulse

B' is the quadratic damping coefficient.

If the impulse tests are performed for various different initial and final tank angles, one can determine the three values ω_{nt}, ζ_t and B which describe the tank dynamics by direct

computer simulation. Values of these three parameters are selected and modified until the solution of the tank equation above best matches the measured impulse test result; typically, the method of least squares is used. These three values now completely characterize the tank. If the value of ζ_t turns out to be substantially more than that for similar tanks then this is, in general, a strong indication that the tank model is too small and substantial laminar flow may be occurring.

The impulse test of a tank model has advantages and disadvantages over the oscillating table tests. An advantage is that since only the water itself is measured, there is no need for any tares. A disadvantage is that since no moments are measured the results cannot be used directly to estimate the roll reduction afforded by the tank, but must be used with a detailed simulation model of the dynamics of the ship and tank, as discussed under methods for prediction of roll motion.

APPENDIX E

NSRDC ANTI-ROLL TANK SIMULATION AND EVALUATION FACILITY

The simulation facility developed at NSRDC, as described by Zarnick and Diskin (9), for evaluating the performance of ship roll tanks is described briefly and compared with other methods for computing ship rolling motions in the text. In this Appendix, the facility, which consists of a roll-sway oscillator table coupled in real time to an analog computer simulation of ship motions is examined critically as a potential tool for validating predicted roll tank performance and ship roll motions in regular and long-crested, irregular waves.

Existing Validation of NSRDC Simulation Facility

At present the NSRDC simulation facility has been validated only for one ship, the recent U. S. Navy Sea Control Ship design. This validation, which is based on comparison of roll motions with and without roll tanks, is discussed in References 13 and 14. When no roll tank is installed, only the analog computer part of the simulation is used.

In References 13 and 14 the roll motions with and without roll tank and in regular and irregular waves determined from tests of a 17 foot long model and from predictions made with the NSRDC simulation facility are compared. In general the agreement between model data and predictions is good, although the following should be noted:

1. Comparisons with roll tanks are limited to four cases (two in regular waves, one in irregular waves and one in transient waves). In the case of greatest interest (ship speed of 20 knots and 45 degree heading angle) the comparison is incomplete and agreement is not so good.

2. Comparisons in irregular waves are limited to two cases. Excepting the poor agreement near the end of the tests, agreement for these cases is generally good, although peak roll angles predicted by the NSRDC facility are 10 to 30 percent less than the measured peak roll angles for many cycles. For a very few cycles the opposite is true.

3. Agreement between predictions and ship model data is somewhat less satisfactory for the transient wave cases than for the irregular wave cases.

If this facility is to be used for validating predicted roll tank performance, a more thorough validation of the facility for cases with roll tanks and in irregular waves is required.

Comparison of NSRDC Simulation Facility with Other Predictional Methods

From the comparisons of various predictional methods presented in Section VI several conclusions can be drawn about the relative accuracy of the NSRDC simulation facility and other methods, and in particular the methods described by Webster (2). These conclusions, are based on comparisons for one ship, however, and must therefore be considered provisional.

The theoretical method of Reference 2 predicts roll motions in regular waves which are in as good agreement with the model data as are the predictions made using the NSRDC simulation facility. For the two cases with roll tanks, the method of

Reference 2 gives slightly better agreement, even though the time-domain solution was not used. No comparison has been made for the irregular wave cases since wave histories were not available. A comparison of the significant roll angles in sea states 5 and 6 without roll tanks indicates the two sets of predictions are in reasonable agreement. A comparison of the significant roll angles with roll tanks indicates that the predictions made using the NSRDC facility are probably better, although this cannot be confirmed from the model tests. The few non-linear time-domain calculations made using the methods of Reference 2, are in considerably better agreement with the NSRDC facility predictions, as seen in Figure 30, indicating that it is probably essential to use such a time-domain solution to predict motions in irregular waves. Comparisons for a number of irregular wave cases are needed to make a valid comparison of these methods.

It should be noted that more accurate predictions can no doubt be made if the methods of Reference 2 are used in combination with bench test data for the roll tank.

Potential Limitations

Integration of the ship equations of motion in real time is straightforward, but realization of the hydraulics necessary to move the model correctly and the sensing equipment to measure the loads is not simple. Tank inertial effects are important and it is thus necessary to have correct instantaneous roll and sway acceleration as well as roll and sway displacements. This places a heavy burden on the hydraulic system, particularly servo-valves. Any "hunting" will cause large

spurious accelerations and loads, which can limit the accuracy. Since loads are supposed to be fed back into the analog simulation simultaneously, little or no smoothing to eliminate extraneous noise can be used.

If any phase lags exist in the facility, part of the gross inertial loads in the table will appear as damping loads in the analog simulation. Since tare inertias are large, a phase lag of even a few degrees between roll angle or sway signal and measured roll moment or sway force would make the computer results suspect. Zarnick and Diskin (9) discuss the feed-ahead technique which is used to minimize phase lags. A thorough evaluation of the complete facility would be required to determine the effectiveness of this feed-ahead procedure, particularly for irregular waves.

Conclusions

The NSRDC anti-roll tank simulation facility appears capable of predicting roll motions in regular and long-crested, irregular seas with acceptable accuracy. In view of the results for the Sea Control Ship, and the absence of validation or comparisons for other ships, however, it is not clear that this facility is superior to other predictional methods, such as bench test-digital computer simulations and tests of ship models of adequate size, for validating predicted roll tank performance. A method such as that presented by Webster (2), used together with bench test data for the tank, appears, at least at present, potentially as good as the NSRDC facility and is likely to be less costly to use. Sufficient results are not available to

insure that system phase lags might not be a problem, particularly in irregular waves, although there is no evidence of this in the results for the Sea Control Ship. Further validation of this facility, using reliable model data for other ship designs and for irregular waves is needed.

APPENDIX F

SHIP/TANK MODEL TEST PROCEDURES

There are a variety of tests usually performed on models of stabilized ships. A few are discussed below.

Inpulse Test

A quick indicator of the success of the stabilization system can be obtained by an impulse test of the ship model in still water. In this test, the model is given an initial roll angle by means of an external moment (often times in practice applied by a stick). The moment is released and the subsequent roll motion recorded. For this purpose, a roll gyro is usually installed in the model.

The recorded roll motion is a damped, oscillatory curve from which one can extract an average logarithmic decrement as a measure of the effective roll damping at resonance. Performing this impulse test with and without stabilization gives a good overall view of the effectiveness of the stabilizer. For instance, it was mentioned that very low internal damping in the tank leads to a low ship response at its unstabilized resonance. However, such a tank also introduces two resonances one at a frequency above, the other at a frequency below the original resonance. In an impulse test of a system in which the tank damping is too low, the recorded roll angle exhibits large, slowly decaying motions at these two "side" resonances. In conclusion, then, the impulse test is a good, although primarily qualitative, measure of the effectiveness of the system.

Beam Seas Tests

In these tests the model is oriented beam to a set of regular waves, and the resulting roll motions are measured.

Usually these tests are performed for a range of different wave lengths (or periods) and for each test the ratio of the steady-state roll angle amplitude to the wave slope amplitude is computed. This response amplitude operator (RAO) is plotted as a function of wave length or period or frequency.

Determination of the RAO with and without the stabilizer operating gives a good quantitative measure of the roll reduction over the range of tested wave lengths. It should be noted that it is always prudent to perform the roll impulse tests first in order to determine if side resonances occur so that these frequencies can be examined closely.

One difficulty in performing beam seas tests is in maintaining the orientation of the model. An unrestrained model often yaws so as to place itself bow to the waves. However, any restraints on the model to prevent this yawing can, if not placed correctly, affect the roll motions. Experience indicates that highly elastic restraints placed near the waterline, bow and stern, appear to have the least influence.

Forward Speed Tests

The worst rolling motion of larger ships occurs in quartering (50-70 Degree) seas when the ship is underway. Performance of tests in these conditions requires a very large seakeeping basin, of which only a few exist in the world. Restraint of the model during this kind of testing is even more challenging than for the zero speed situation above. Ideally, a good, radio-controlled model would be best. As a result, forward speed tests in quartering seas are rarely performed.

APPENDIX G
ESTIMATION OF TANK DAMPING

It is useful to be able to estimate tank damping and to select any damping structures required to achieve the desired damping before any tests of the tank are made. This will help to insure that major modifications to the tank design are not required. This is especially important for active and controlled-passive tanks where relatively small tank damping is desired.

Webster (7) has considered the calculation of damping of U-tube tanks in detail. Those methods and similar methods can be used to calculate the damping of free surface tanks. This Appendix summarizes some of the material presented in Reference 7. For a detailed treatment, the reader must refer to Reference 7.

Tank Damping Coefficient

The roll tank equation of motion can be expressed, in non-dimensional form, after applying the process of equivalent linearized to the damping term, as:

$$\ddot{\tau} + 2\,\bar{\zeta}_t\,\omega_t\,\dot{\tau} + \omega_t^2\,\tau = 0$$

where τ is the tank fluid angle

$\bar{\zeta}_t$ is the equivalent linear damping ratio

ω_t is the tank natural frequency

The damping ratio can be expressed as

$$\bar{\zeta}_t = \frac{\omega_t}{4} \left\langle \left| \frac{h(R\dot{\tau})}{(R\dot{\tau})} \right| \right\rangle_{\text{average}}$$

where $R\dot{\tau}$ is the fluid velocity in the tank

$h(R\dot{\tau})$ is the total head loss function

It can be seen that $\bar{\zeta}_t$ is proportional to the mean of the absolute value of the ratio $h(R\dot{\tau})$ to $R\dot{\tau}$. For typical cases, $h(R\dot{\tau})$ is proportional to $(R\dot{\tau})^2$ and this average value of the bracketed quantity varies linearly with the average value of $\dot{\tau}$. Thus for each amplitude of motion we can associate an equivalent linear damping ratio.

The computation of the damping can be carried out in a direct manner. First, the U-tube is decomposed into its hydraulic elements: the vertical legs, bends, entrances, exits, valves, transitions, etc. The loss coefficient for each is estimated by means of the information set forth below and these coefficients are summed to form $h(R\dot{\tau})$ and then to calculate the equivalent linear damping ratio

$$\bar{\zeta}_t = \frac{\omega_t R}{8g} C_{\ell_{total}} |\dot{\tau}|_{ave}$$

where $C_{\ell_{total}}$ is the total head loss coefficient

R is the tank dimension defined in Figure 11

The calculation of the loss coefficients for each tank hydraulic element from loss coefficients for pipes, bends, contractions and expansions is discussed in the following sections.

The Damping in U-Tube Tanks

There are several causes of the damping in a conduit such as a U-tube. All of the causes are due to the viscosity of

the fluid. The direct effect of the viscosity is one of the friction drag on the wetted surfaces of the conduit. There are several other losses in the conduit which are caused in a less direct way by viscosity. These effects are due to restrictions in the flow and may exhibit themselves as losses due to a contraction, an enlargement, an entrance, an exit, an obstruction, a bend, etc. Most of the material concerning these phenomena are empirical in nature and a large body of engineering data exists in a form which is convenient for the computation of damping.

In a hydraulic system consisting of many individual devices or distinguishable flow sections, the total loss is normally taken as the sum of the losses of each element, i.e.,

$$h_\ell = \sum_{i=1}^{n} \text{loss}_i = \sum_{i=1}^{n} C_{\ell_i} \frac{V_i^2}{2g}$$

However, the velocity V_i in this equation is the local velocity associated with the element in question. It is more convenient to work simultaneously with the local velocities at many individual elements by referring them all to a single velocity at some reference section of the flow path. For roll tanks the reference section is taken as the rectangular side tank area A_o. The principle of continuity for incompressible flows states

$$A_o V_o = A(s_i)\, V_i$$

where V_i, V_o are the velocities at points i and o respectively, and

$A(s_i)$, A_o are the cross-section areas at i and o.

The total head loss can then be expressed as

$$h_\ell = \sum_{i=1}^{n} C_{\ell_i} \frac{V_i^2}{2g} = \sum_{i=1}^{n} C_{\ell_i} \left(\frac{A_o}{A(s_i)} \right)^2 \frac{V_o^2}{2g} = \sum_{i=1}^{n} C_{\ell_i}' \frac{V_o^2}{2g} \quad .$$

Pipes - The loss due to turbulent flow in a pipe is usually expressed in the following form.

$$H_{\ell_p} = f \frac{L}{D} \frac{V^2_{pipe}}{2g}$$

where f is the friction coefficient,

L is the pipe length, ft,

D is the pipe diameter, ft, and

V_{pipe} is the mean velocity in the pipe.

For preliminary estimates of the damping, all the quantities on the right side of this equation are known except the factor f. This factor is a function of both the roughness of the pipe and the Reynolds number, and plots of pipe flows can be found in numerous handbooks and textbooks. One should bear in mind that the flow in a stabilization tank is not steady. Fortunately, for rough pipes at high Reynolds numbers, the curves of f are very flat. During a large portion of an oscillation cycle, the

Reynolds number is probably high enough that f can be assumed constant.

As an estimate, the Reynolds number and f can be determined using the mean velocity over one half a cycle of the water pendulation. During this interval the water flows in one direction. The effective loss coefficient C_{ℓ_i}' referred to the side tank velocity is

$$C'_{\ell_{pipe}} = f \frac{L}{D} \left(\frac{A_o}{A_{pipe}} \right)^2$$

Conduits - Frictional losses in uniform conduits with non-circular cross-sections can be treated in a similar fashion to pipes. The head loss for a constant cross-section conduit is expressed

$$H_\ell = f \frac{L}{4r} \frac{V_c^2}{2g}$$

where f is the friction factor as before,

L is the length of section,

r is the hydraulic radius = cross-section area/wetted perimeter, and

V_c is the average velocity through conduit.

Values for the friction factor f can be selected in the same manner as with round pipes by replacing the diameter D by four times the hydraulic radius, 4r.

Bends - For smoothly turning duct bends without guide vanes, the loss coefficient depends on the bending angle, the bend radius and the duct dimensions. The loss coefficient C_ℓ for right angle bends in rectangular ducts is given in Table G1.

TABLE G1

Loss Coefficients for Rectangular Ducts

w/d \ r/d	2/3	1	5/3
3	0.55	0.22	0.15
6	0.38	0.16	0.09

where r is the centerline radius, w is the bend width and d is the bend depth in the plane of curvature.

Expansions and Contractions - In a stabilization tank, the geometry is almost always symmetrical. A contraction on one side will be accompanied by an expansion on the other side. Also a contraction in one part of a cycle of oscillation will become an expansion in another part of a cycle when the flow reverses. Hence, transition sections can be treated interchangeably. The design of a transition influences the damping of the system. If large damping is desirable a transition should be made abrupt so as to increase the loss.

The design of a gradual transition is usually governed by its behavior during the part of the cycle which causes expansion. Because of the presence of a positive pressure gradient

in an expansion, the flow will have a tendency to separate. When separation occurs there is a marked increase in the loss, along with irregular velocity distributions.

The phenomenon of separation in a diffuser is quite complicated and is governed by a number of factors: angle of divergence, length of transition, upstream velocity distributions, and the entrance conditions. Because of the number of factors involved, there is no really comprehensive information available. There is general agreement that C_ℓ depends a great deal on the divergence angle and the diameter ratios. More recent experiments show that the upstream velocity distributions affect the loss, although not in a drastic way.

When the angle of divergence is small and separation is not present, the flow is not very different from that through a uniform pipe, and the friction loss can be estimated in a similar fashion. The velocity variation along the length must be taken into account. For rectangular sections the loss coefficient can be approximated by:

$$C_{\ell_t} = f \frac{1}{4r_2} \left[\sqrt{\frac{A_2}{A_1}} + \frac{A_2}{A_1} + \left(\sqrt{\frac{A_2}{A_1}} \right)^3 + \left(\frac{A_2}{A_1} \right)^2 \right]$$

where A_1, A_2 are the cross-section areas at the two ends, and
r_2 is the hydraulic radius = A_2/wetted perimeter at section 2.

For abrupt expansions from A_1 to A_2 the Table G2 gives the loss coefficients C_ℓ for the expression (based on V_1):

$$H_\ell = C_{\ell_e} \frac{V_1^2}{2g} \quad .$$

TABLE G2

Loss Coefficients for Sudden Expansions

Area Ratio $\frac{A_1}{A_2}$	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C_{ℓ_e} (expansion)	1.00	0.81	0.64	0.49	0.36	0.25	0.16	0.09	0.04	0.01	0

In a like fashion, abrupt contraction losses are computed from the formula (based on V_2):

$$H_\ell = C_{\ell_c} \frac{V_2^2}{2g}$$

for which the coefficients are given in Table G3

TABLE G3

Loss Coefficients for Sudden Contractions

$\frac{A_2}{A_1}$	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C_{ℓ_c}	0.50	0.48	0.45	0.41	0.36	0.29	0.21	0.13	0.07	0.01	0

Nozzles and Stanchions

The head loss coefficient for nozzles of the type used in roll tanks, stanchions and other similar vertical elements in the tank is given by

$$C_\ell = C_d \frac{aN}{A_{flow}}$$

where C_d is the element 2.d. drag coefficient based on frontal area a of the element

A_{flow} is the flow area at the cross-section where the N elements are located

N is the number of elements.

For free surface tanks the average frontal area and flow area, corresponding to the undisturbed tank fluid level, can be used. Reference 12 gives drag coefficients for various nozzle shapes.

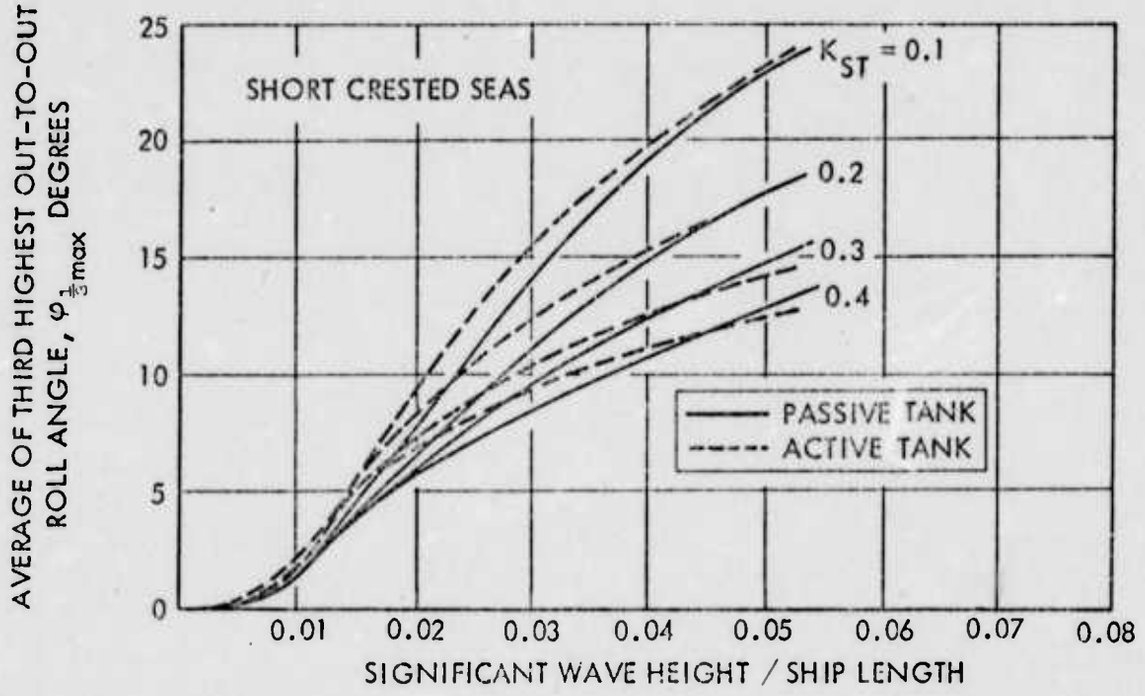


FIGURE 1 - COMPARISON OF SIGNIFICANT ROLL ANGLES WITH ACTIVE AND PASSIVE ROLL TANKS FOR A DESTROYER AT ZERO SPEED

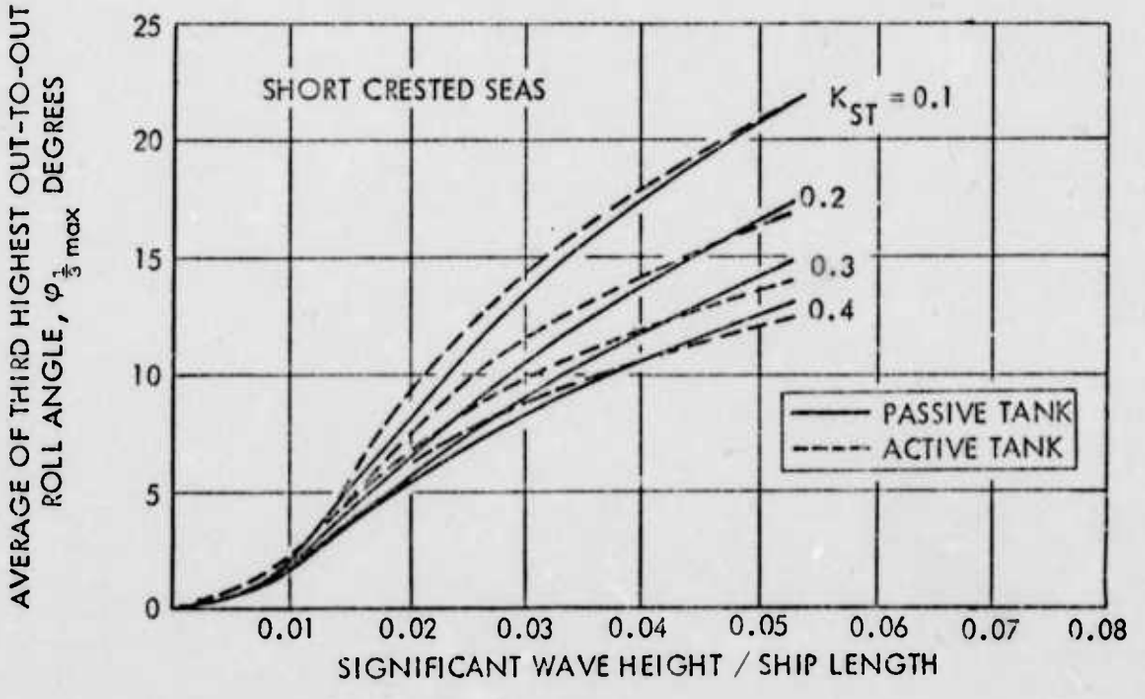


FIGURE 2 - COMPARISON OF SIGNIFICANT ROLL ANGLES WITH ACTIVE AND PASSIVE ROLL TANKS FOR A DESTROYER AT $V/\sqrt{L} = 0.4$

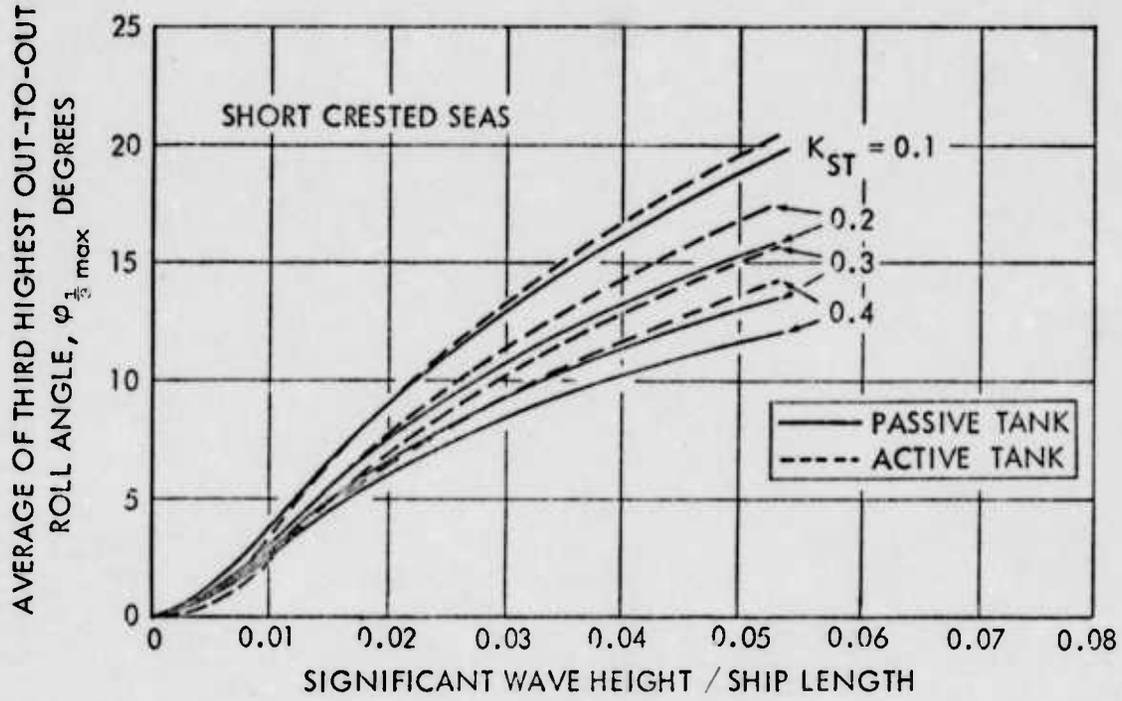


FIGURE 3 - COMPARISON OF SIGNIFICANT ROLL ANGLES WITH ACTIVE AND PASSIVE ROLL TANKS FOR A DESTROYER AT $V/\sqrt{L} = 0.8$

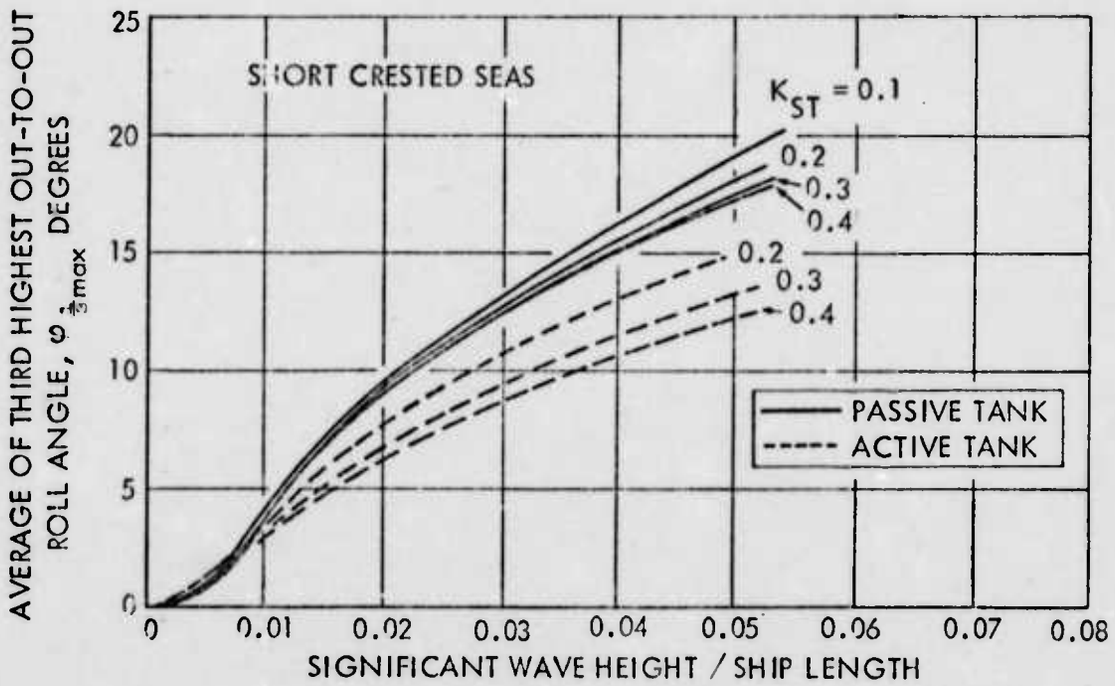


FIGURE 4 - COMPASISON OF SIGNIFICANT ROLL ANGLES WITH ACTIVE AND PASSIVE ROLL TANKS FOR A DESTROYER AT $V/\sqrt{L} = 1.2$

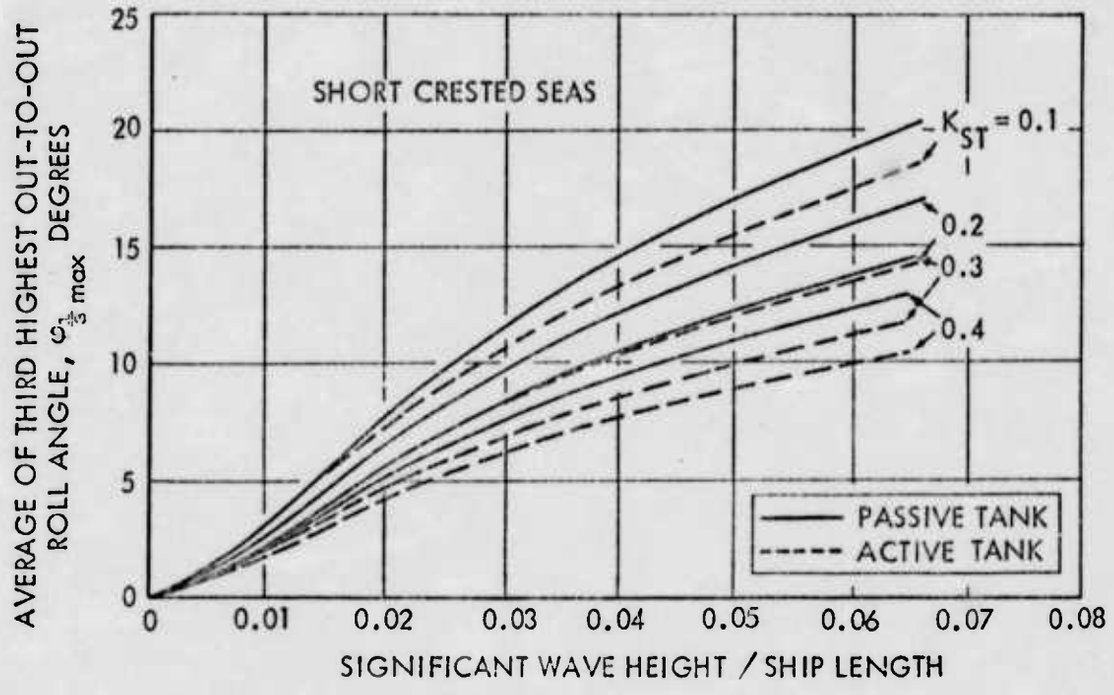


FIGURE 5 - COMPARISON OF SIGNIFICANT ROLL ANGLES WITH ACTIVE AND PASSIVE ROLL TANKS FOR AN AUXILIARY AT $V/\sqrt{L} = 0.8$

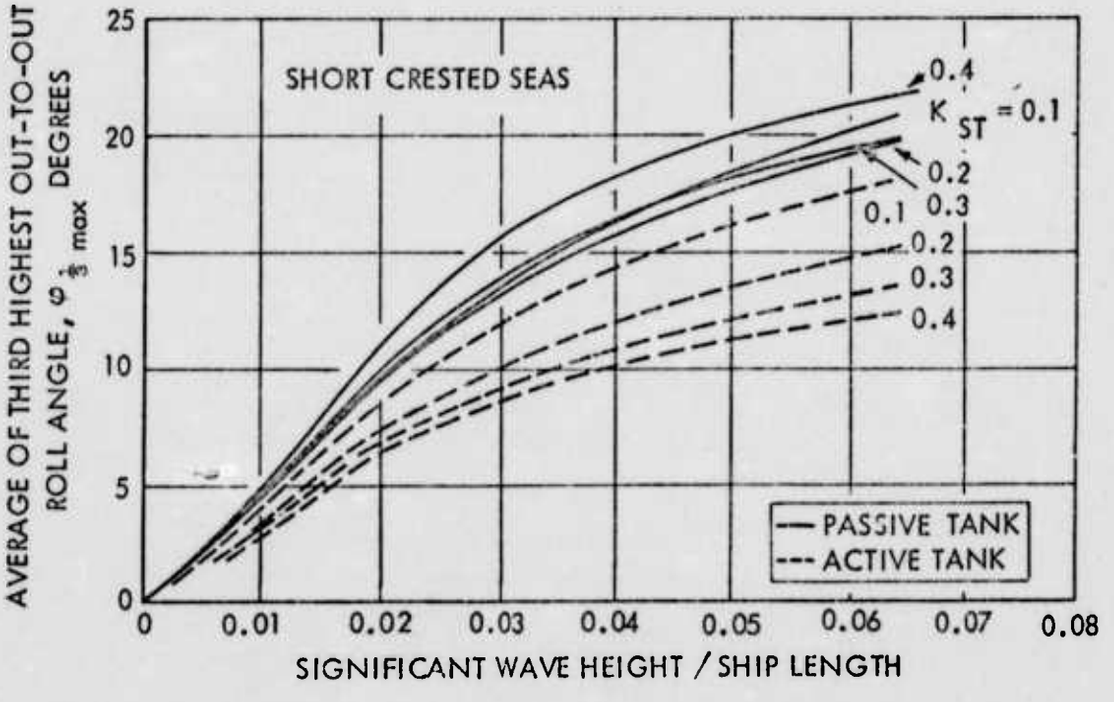


FIGURE 6 - COMPARISON OF SIGNIFICANT ROLL ANGLES WITH ACTIVE AND PASSIVE ROLL TANKS FOR AN AUXILIARY AT $V/\sqrt{L} = 1.2$

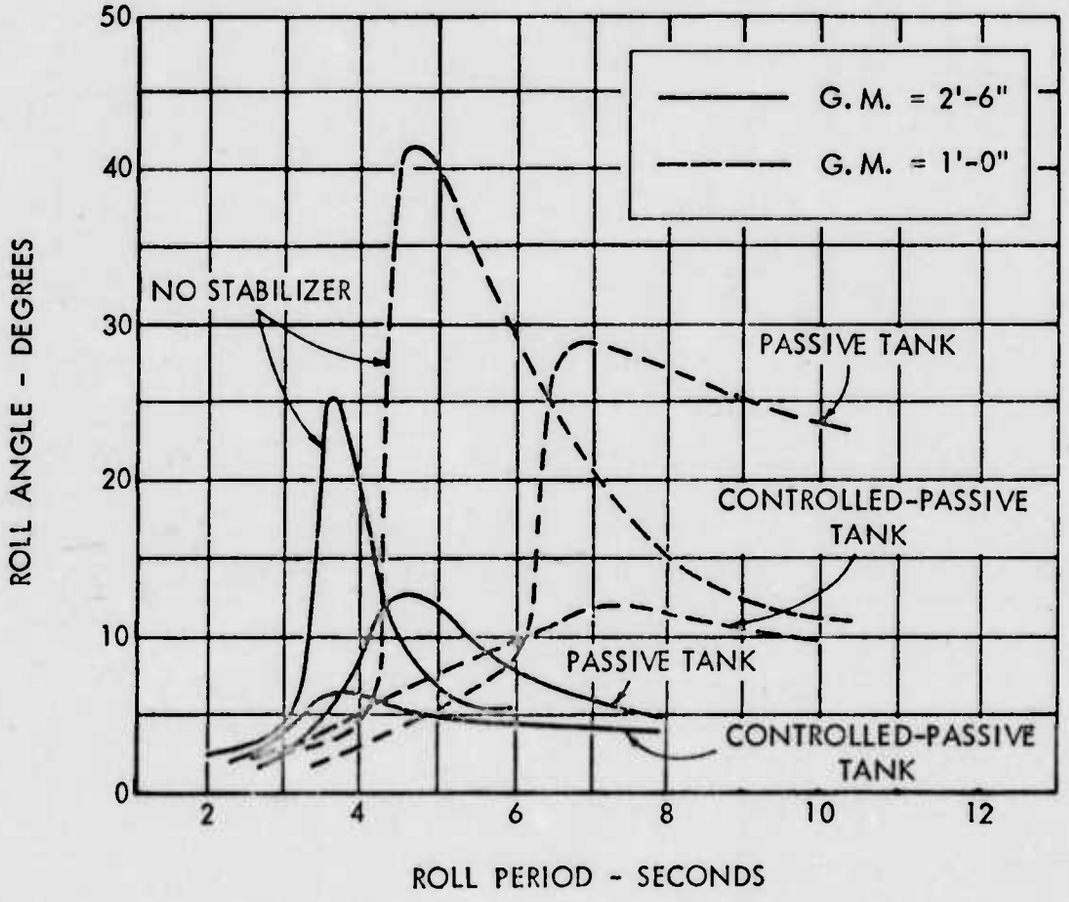


FIGURE 7 - MODEL TEST PERFORMANCE OF PASSIVE AND CONTROLLED-PASSIVE ROLL TANKS ON A FAST CARGO SHIP (From Ref. 3)

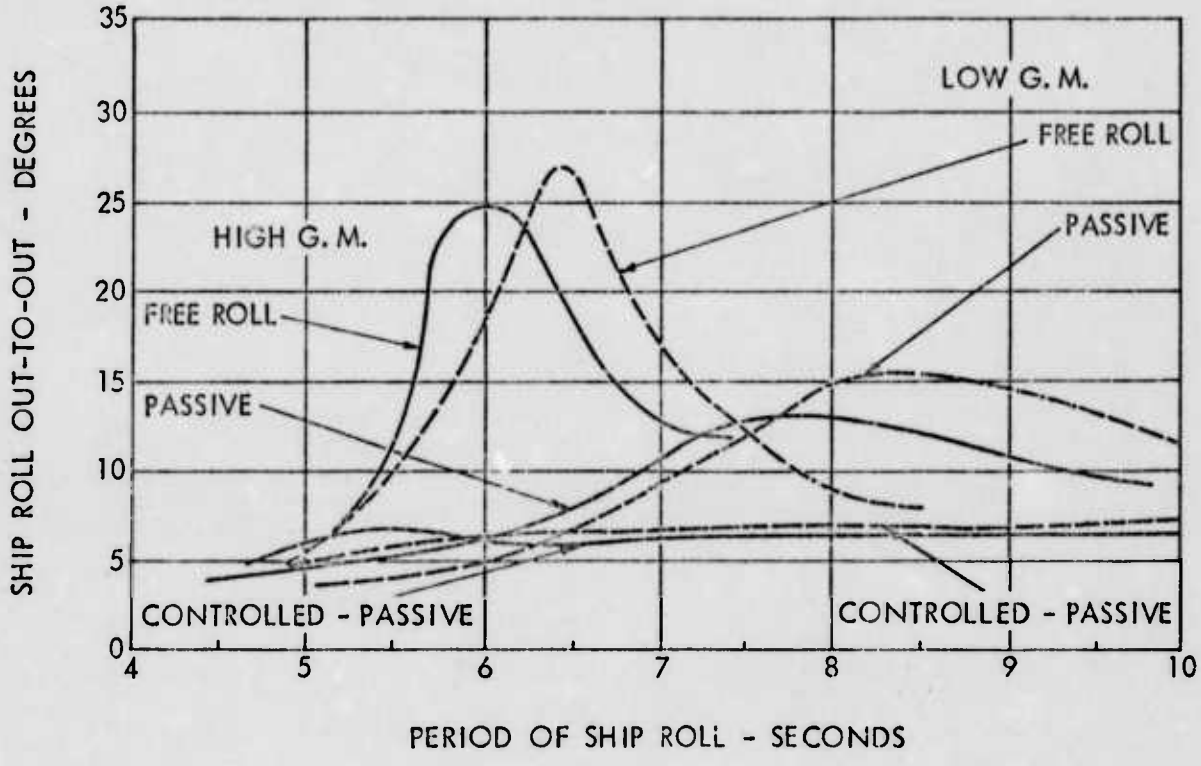


FIGURE 8 - MEASURED TRIAL PERFORMANCE OF PASSIVE AND CONTROLLED-PASSIVE ROLL TANKS ON THE "SECOND SNARK" (From Ref. 3)

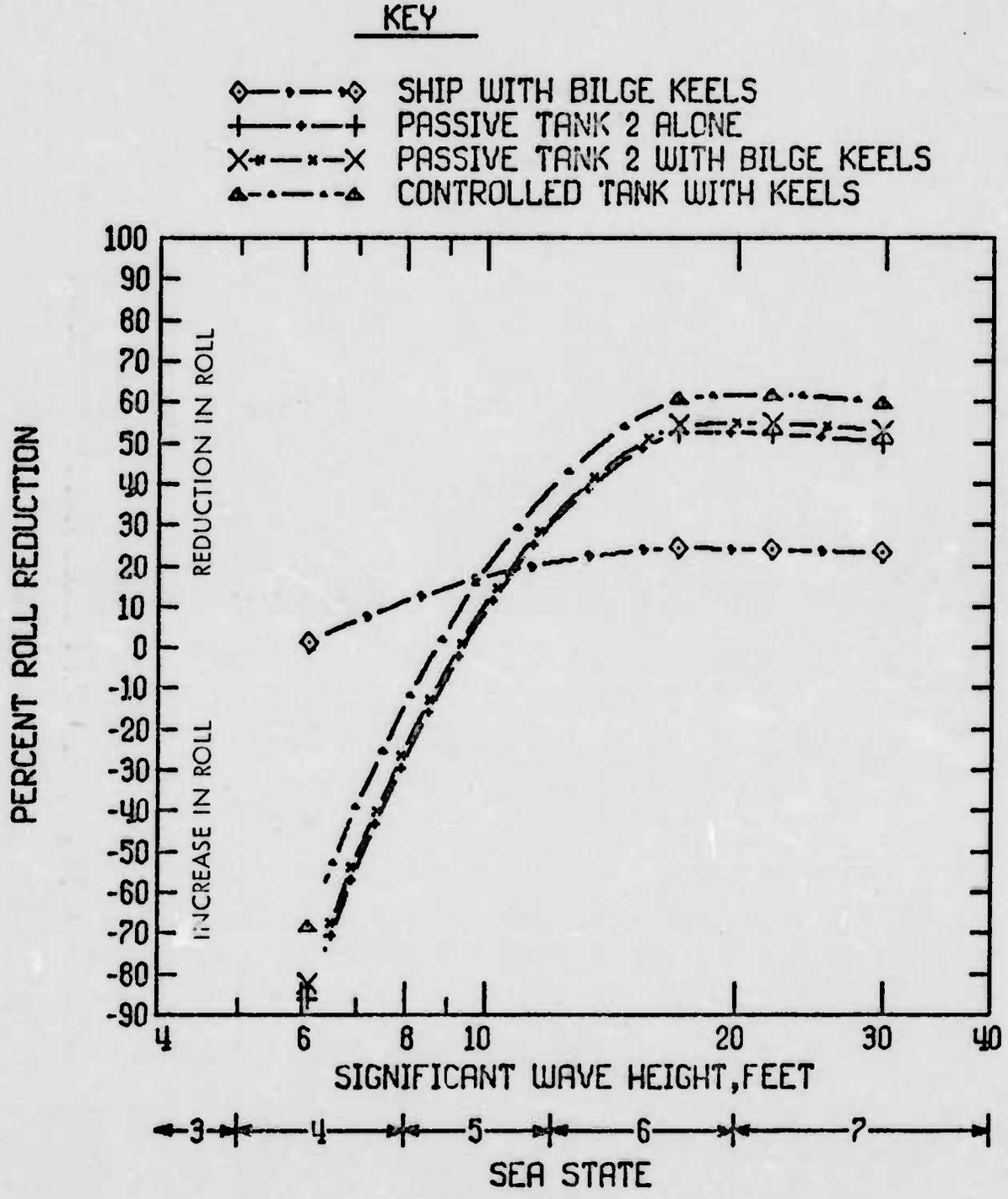


FIGURE 9 - ROLL REDUCTION AS FUNCTION OF WAVE HEIGHT FOR OCEANOGRAPHIC RESEARCH SHIP - AT 15 KNOTS IN LONG-CRESTED BEAM SEAS

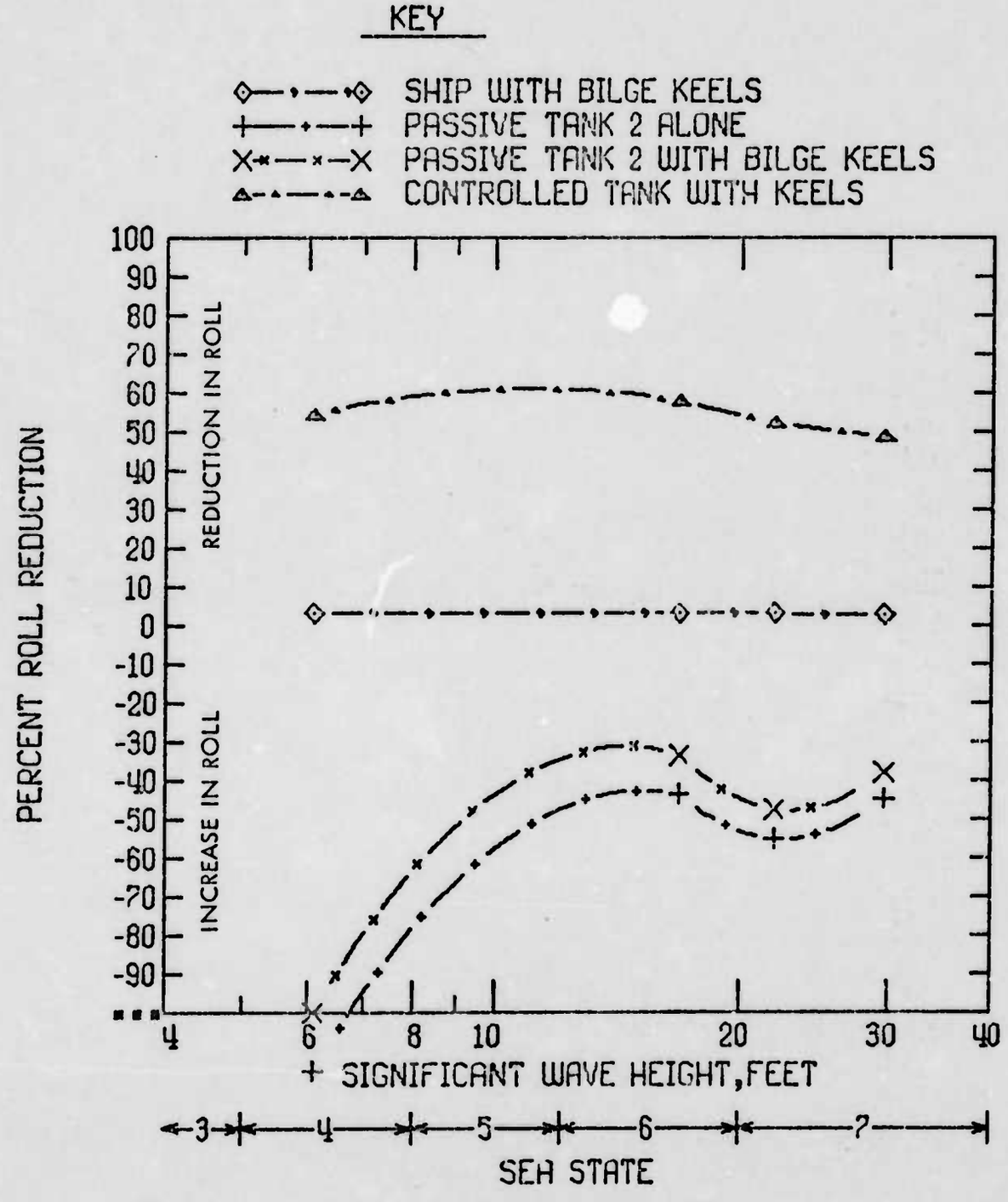


FIGURE 10 - ROLL REDUCTION AS FUNCTION OF WAVE HEIGHT FOR OCEANOGRAPHIC RESEARCH SHIP - AT 15 KNOTS IN LONG-CRESTED BEAM SEAS

$$S'' = \int \frac{t_t \, d_s}{R}$$

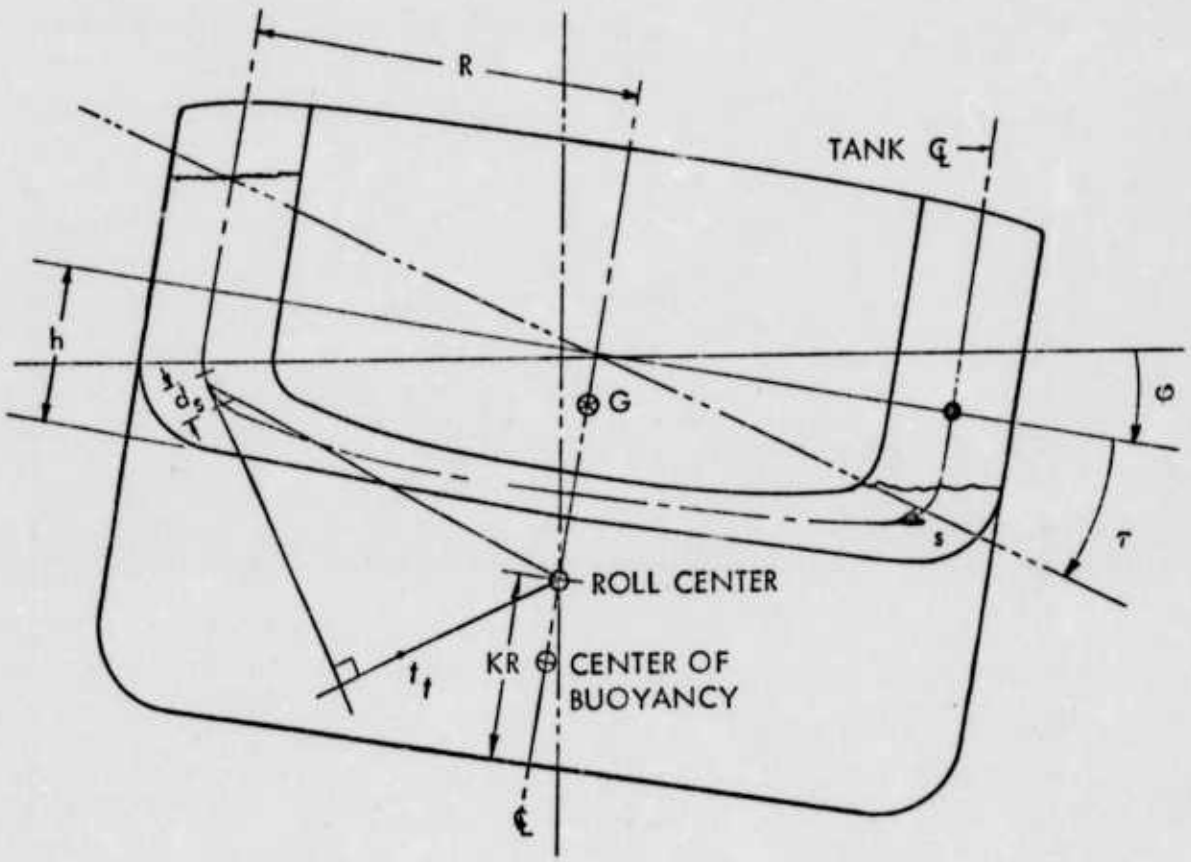


FIGURE 11 - DEFINITION SKETCH FOR LENGTH S"

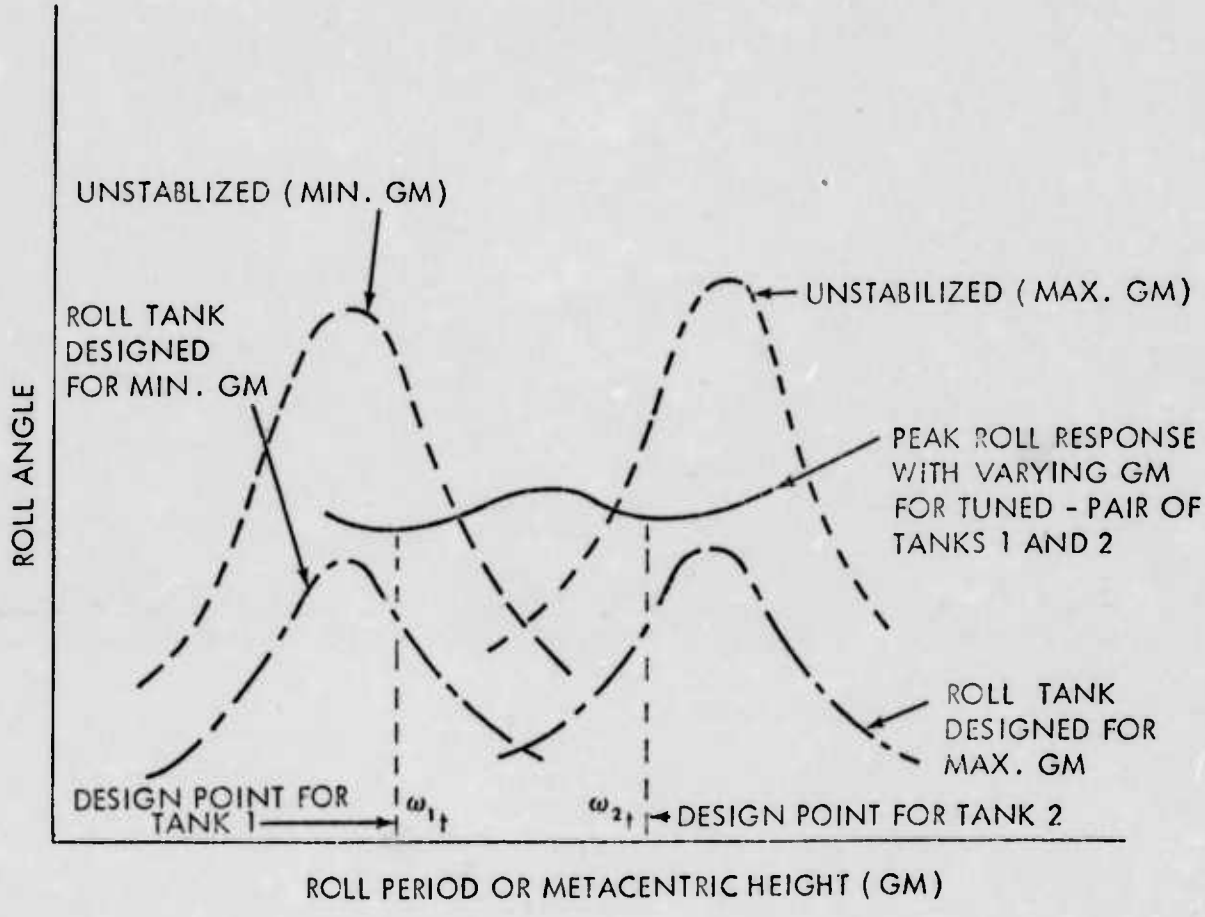
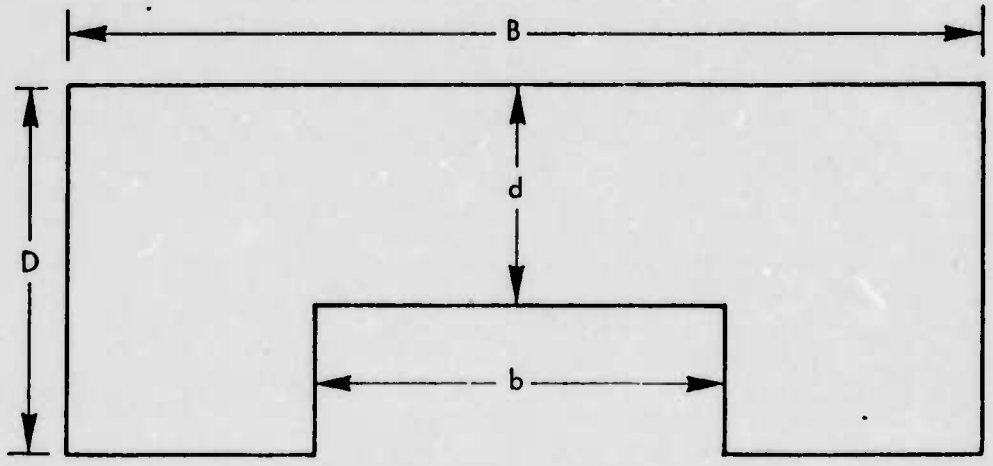
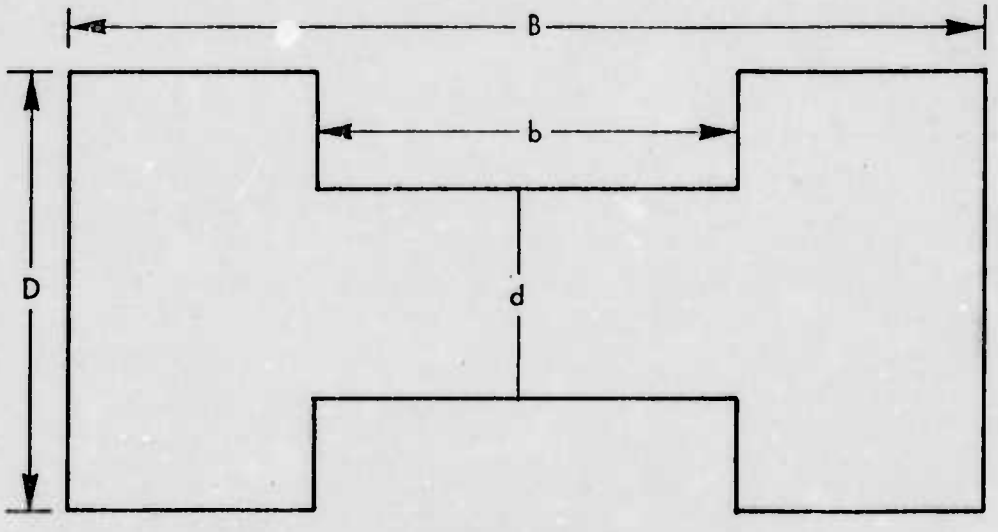


FIGURE 12 - APPLICATION OF U-TUBE TANKS IN "TUNED-PAIRS"



A. C-SHAPE PLANFORM



B. I-SHAPE PLANFORM

FIGURE 13 - TANK PLANFORM SHAPES

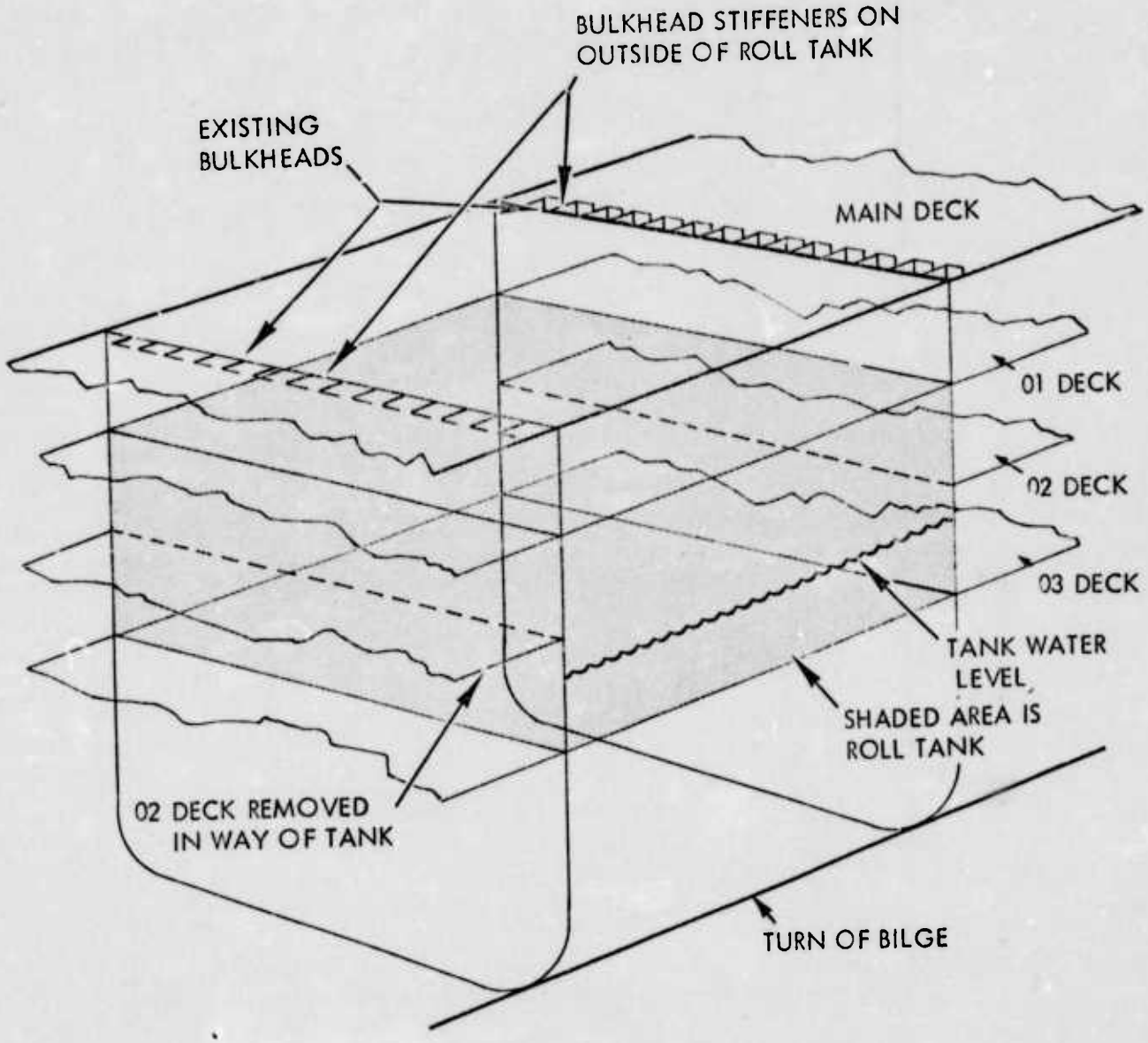


FIGURE 14 - SKETCH SHOWING INTEGRATION OF ROLL TANK WITH SHIP STRUCTURE

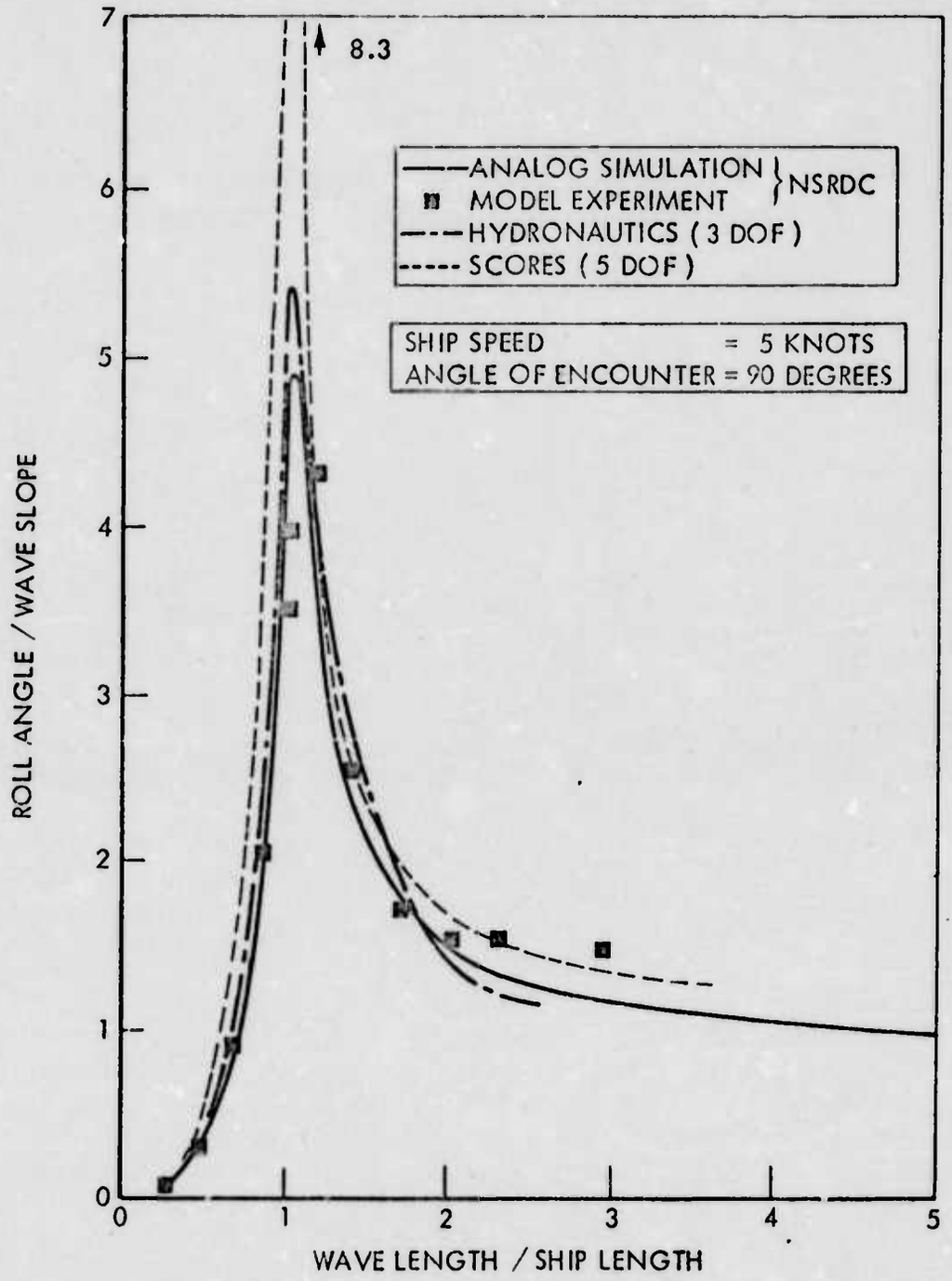


FIGURE 15 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK AT 5 KNOTS IN BEAM SEAS

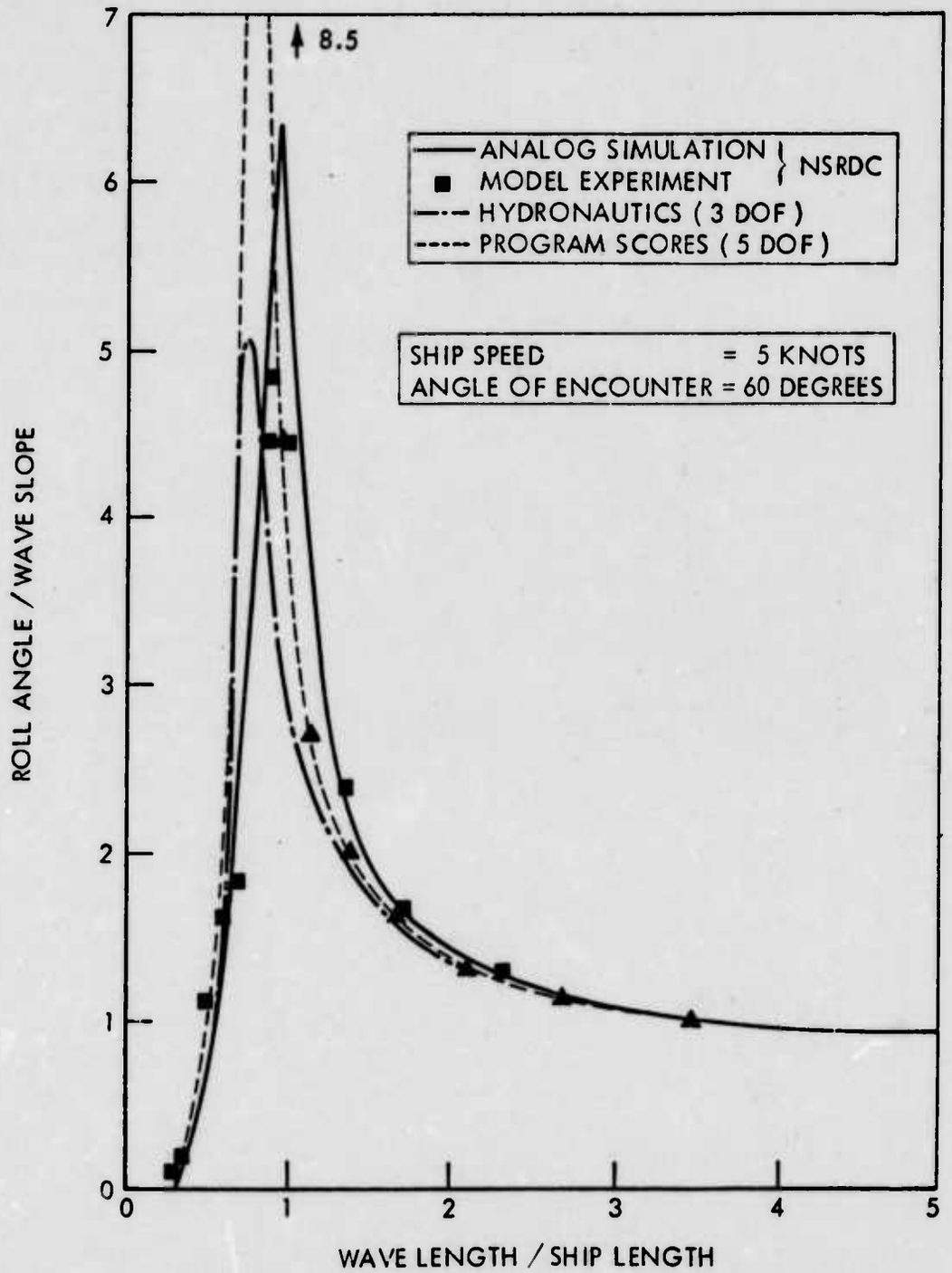


FIGURE 16 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK AT 5 KNOTS IN FOLLOWING (60°) SEAS

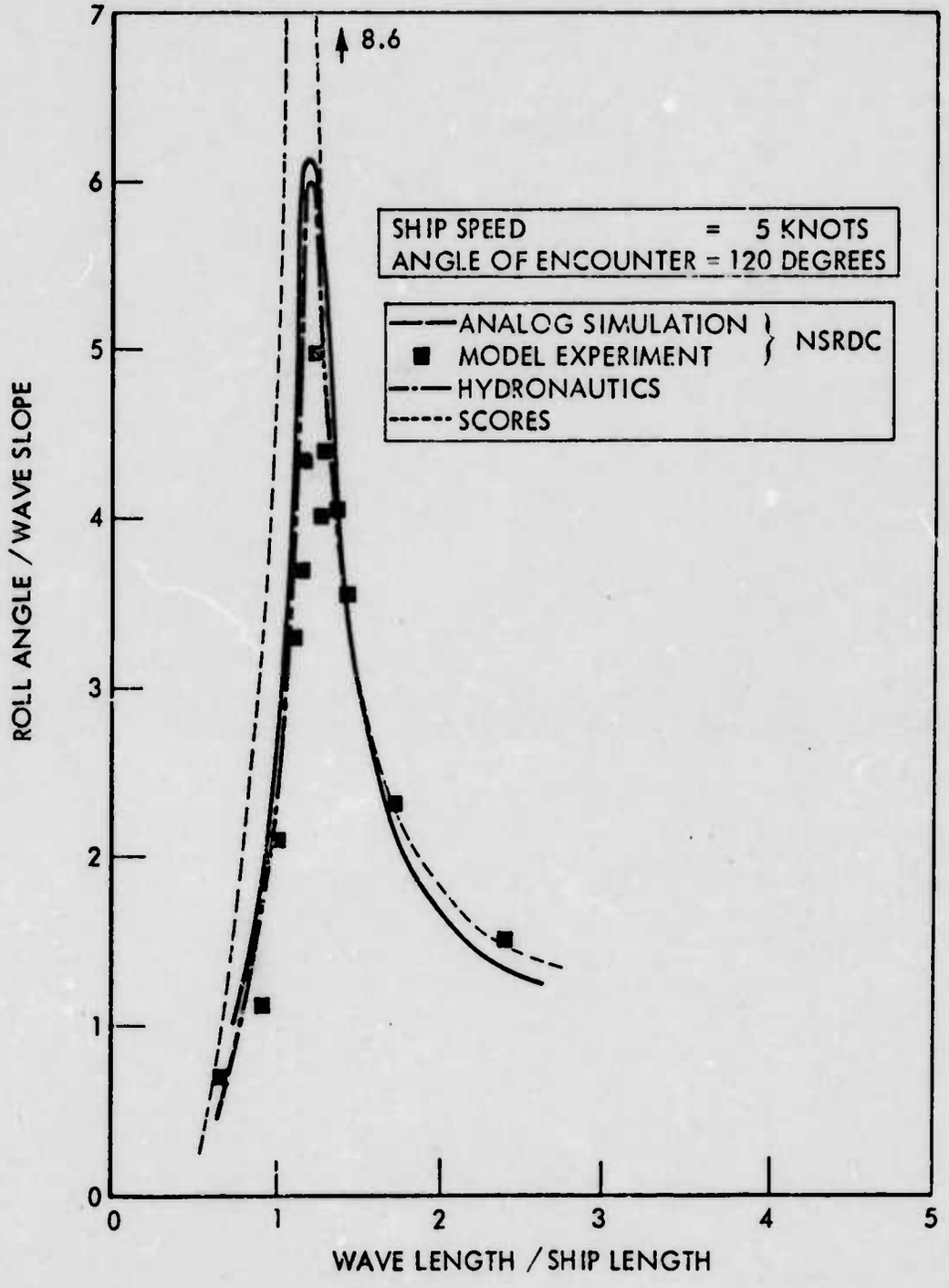


FIGURE 17 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK AT 5 KNOTS IN HEAD (120°) SEAS

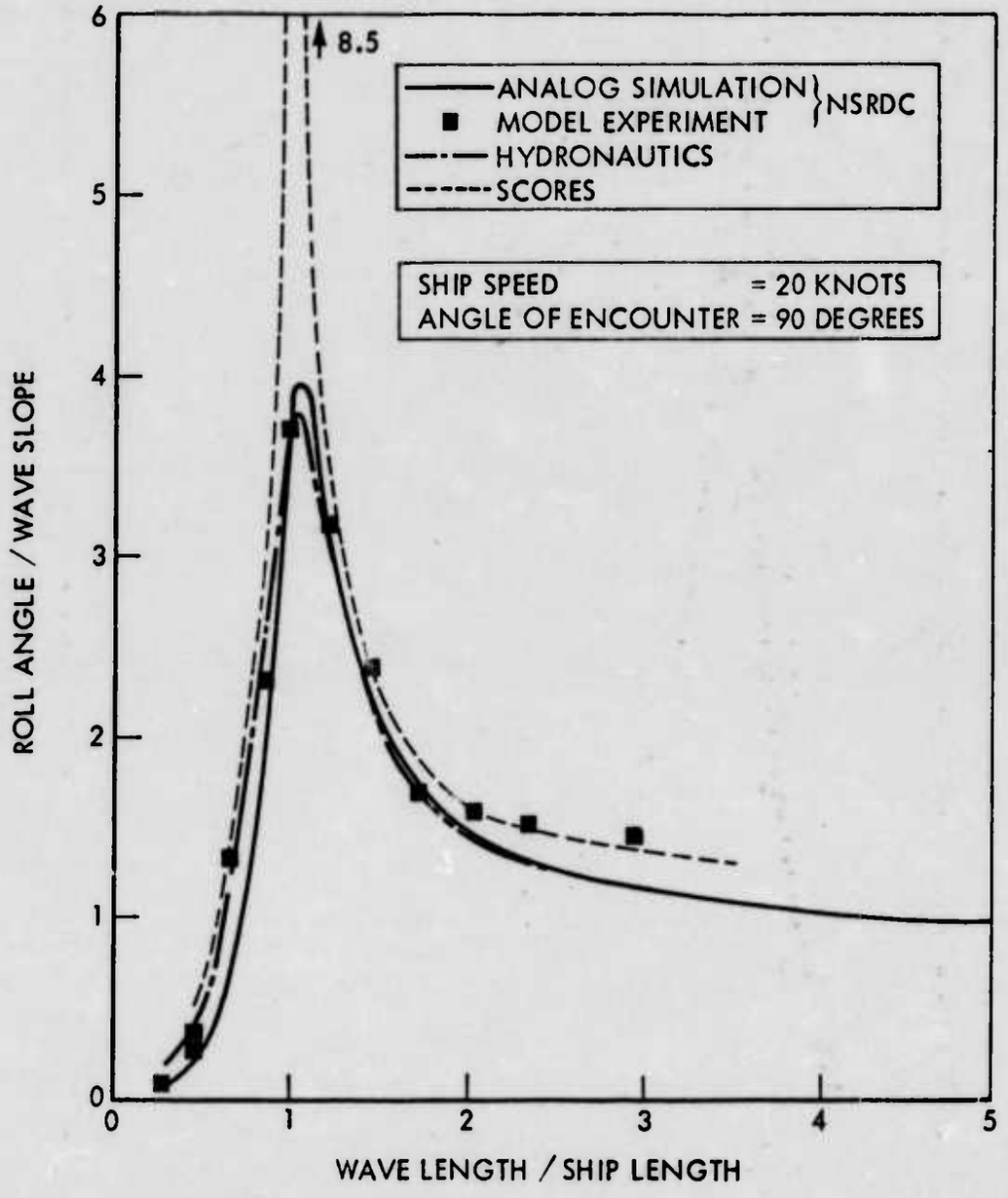


FIGURE 18 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK AT 20 KNOTS IN BEAM SEAS

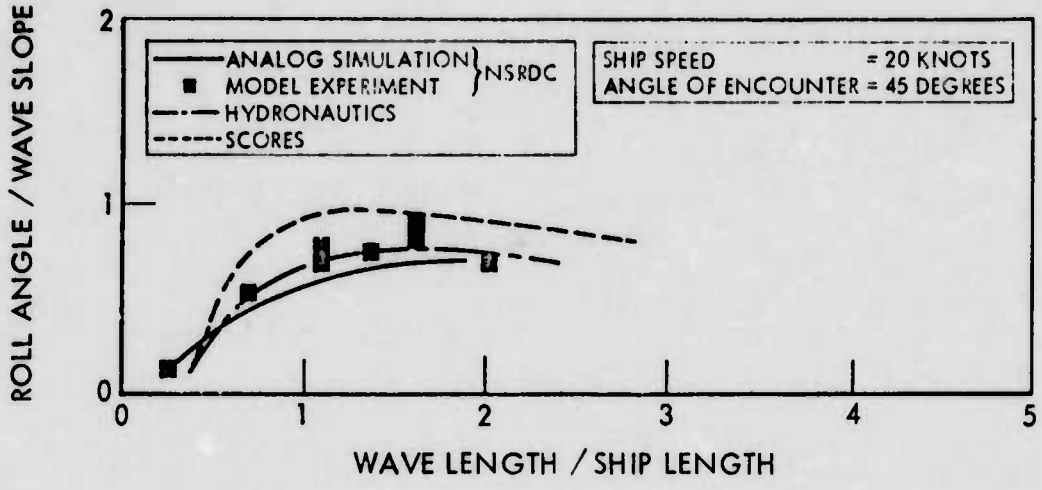


FIGURE 19 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK AT 20 KNOTS IN QUARTERING (45°) SEAS

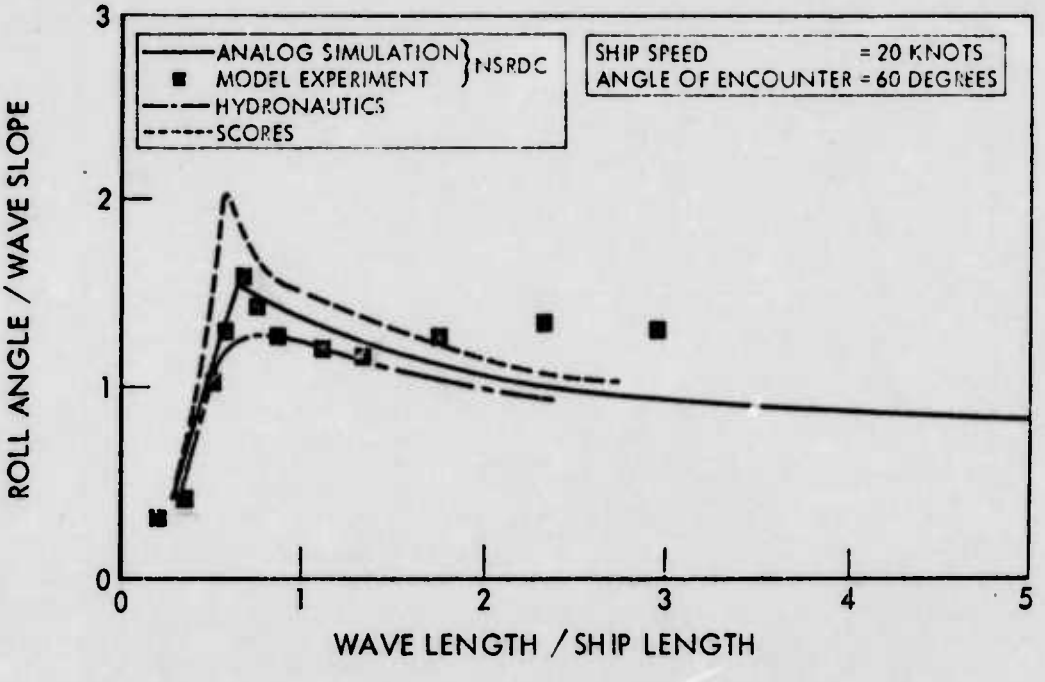


FIGURE 20 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK AT 20 KNOTS IN QUARTERING (60°) SEAS

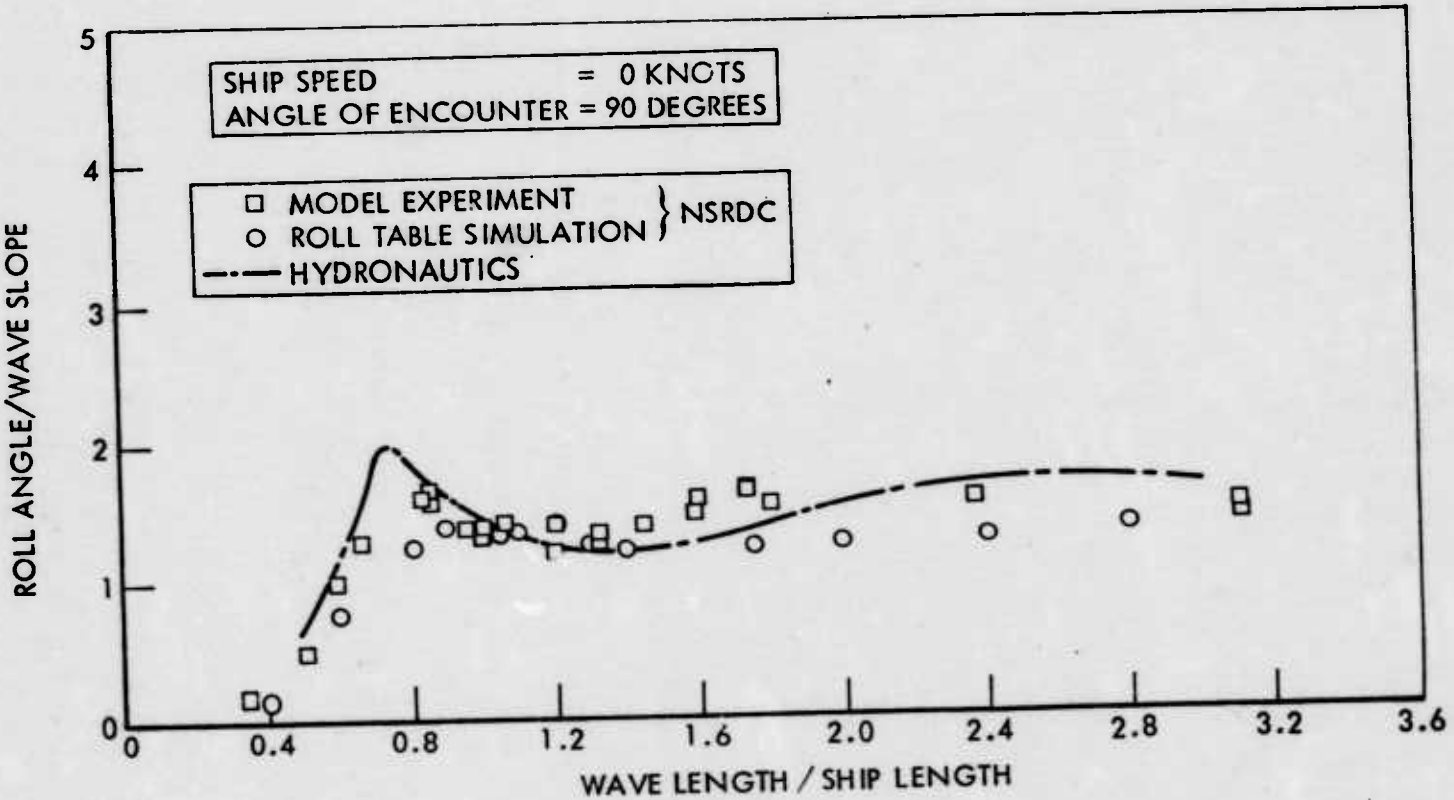


FIGURE 21 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITH PABL TANK AT ZERO SPEED IN BEAM SEAS

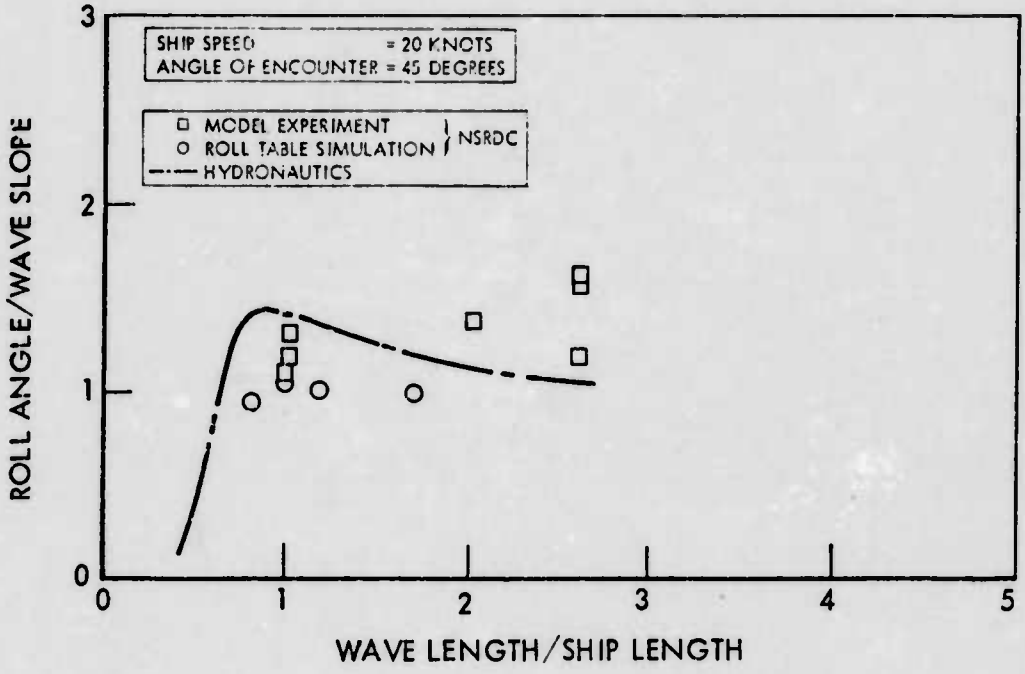


FIGURE 22 - MEASURED AND PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITH PABL TANK AT 20 KNOTS IN QUARTERING (45°) SEAS

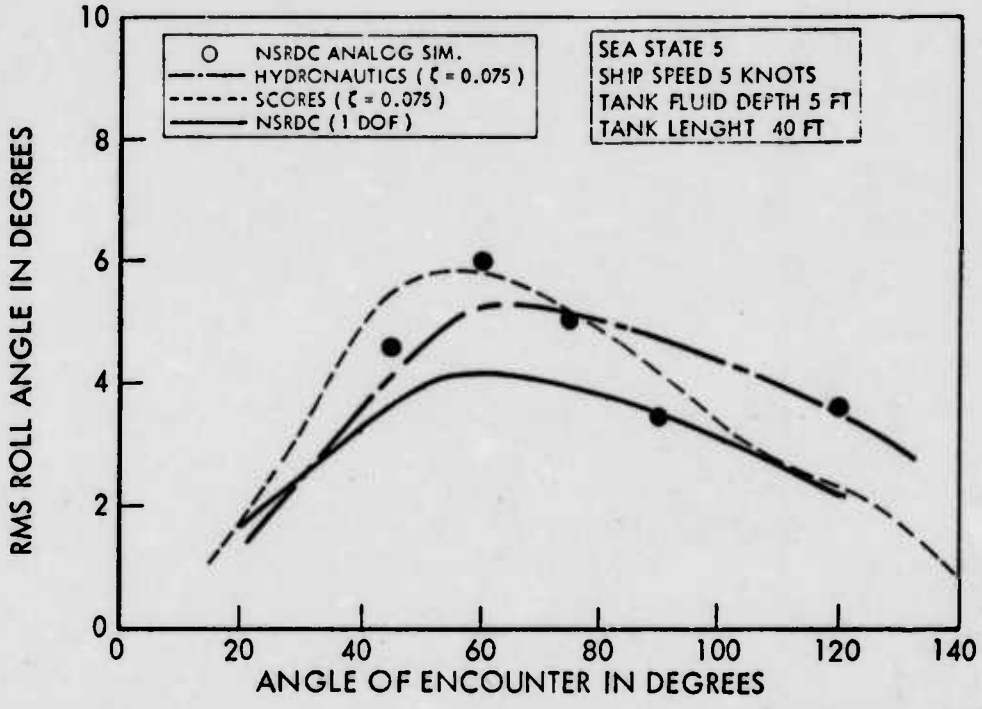


FIGURE 23 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK IN SEA STATE 5 AT 5 KNOTS

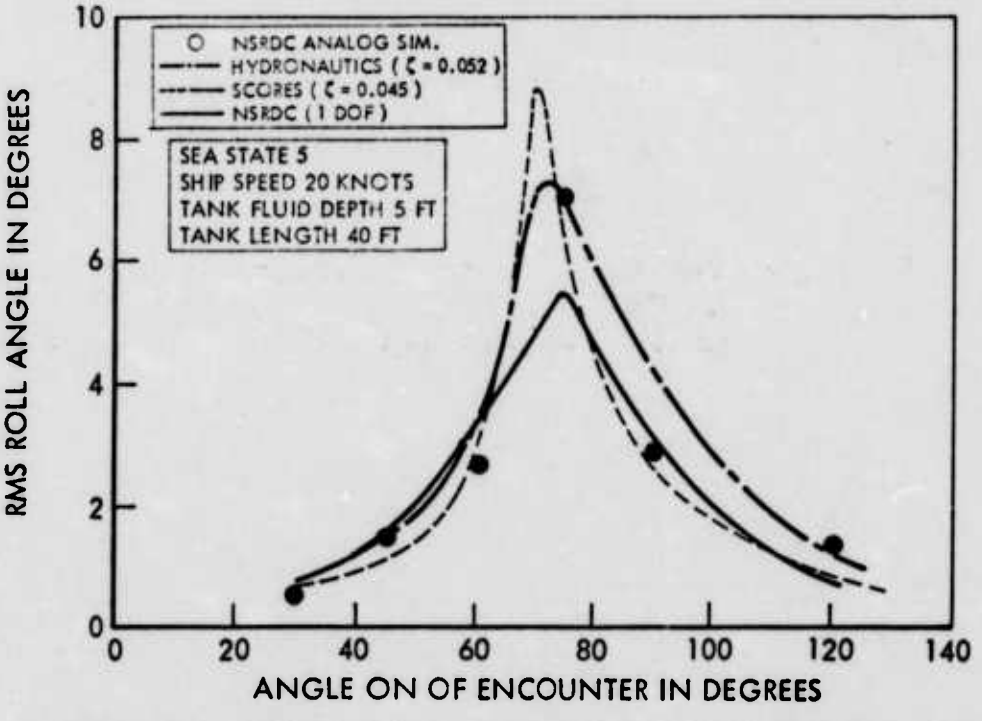


FIGURE 24 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK IN SEA STATE 5 AT 20 KNOTS

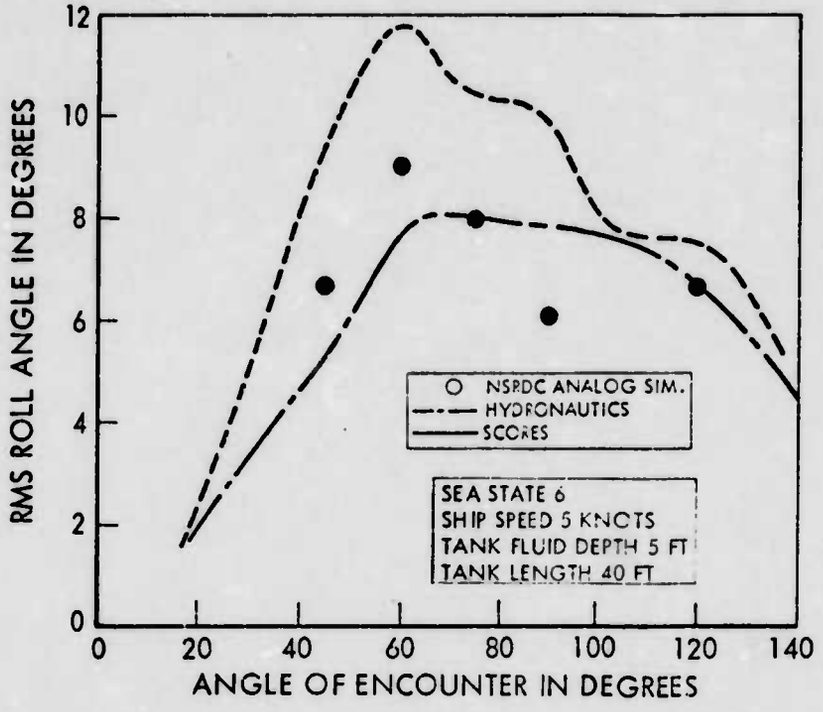


FIGURE 25 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK IN SEA STATE 6 AT 5 KNOTS

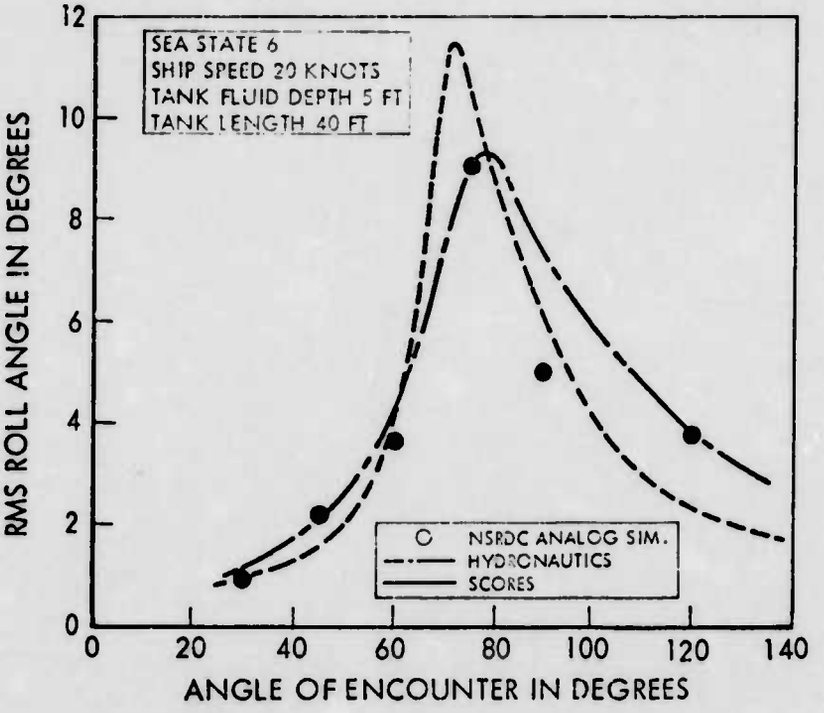


FIGURE 26 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITHOUT TANK IN SEA STATE 6 AT 20 KNOTS

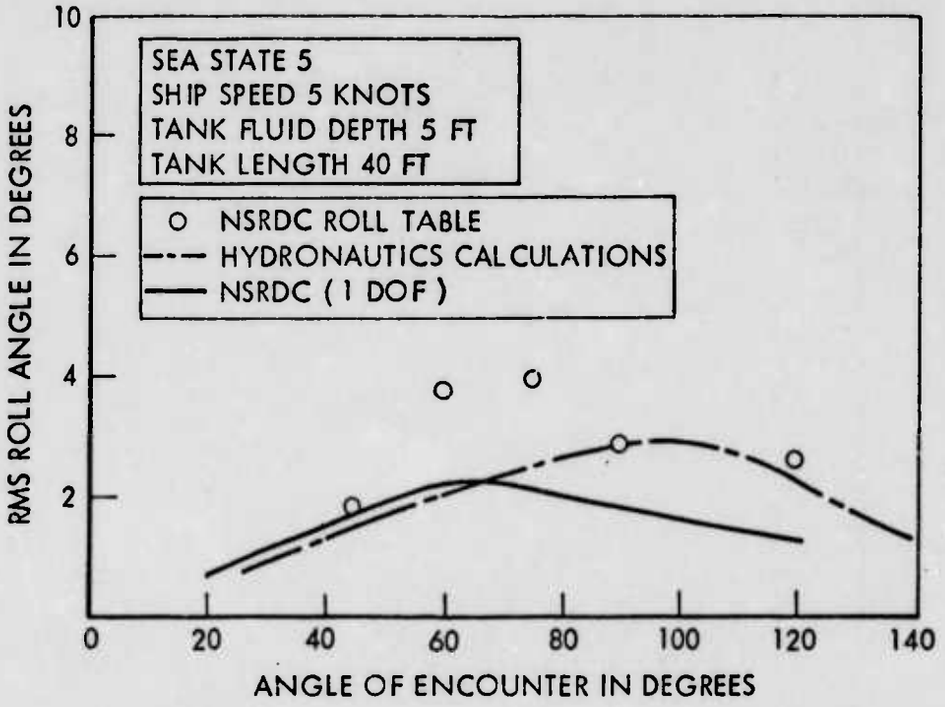


FIGURE 27 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITH PABL TANK IN SEA STATE 5 AT 5 KNOTS

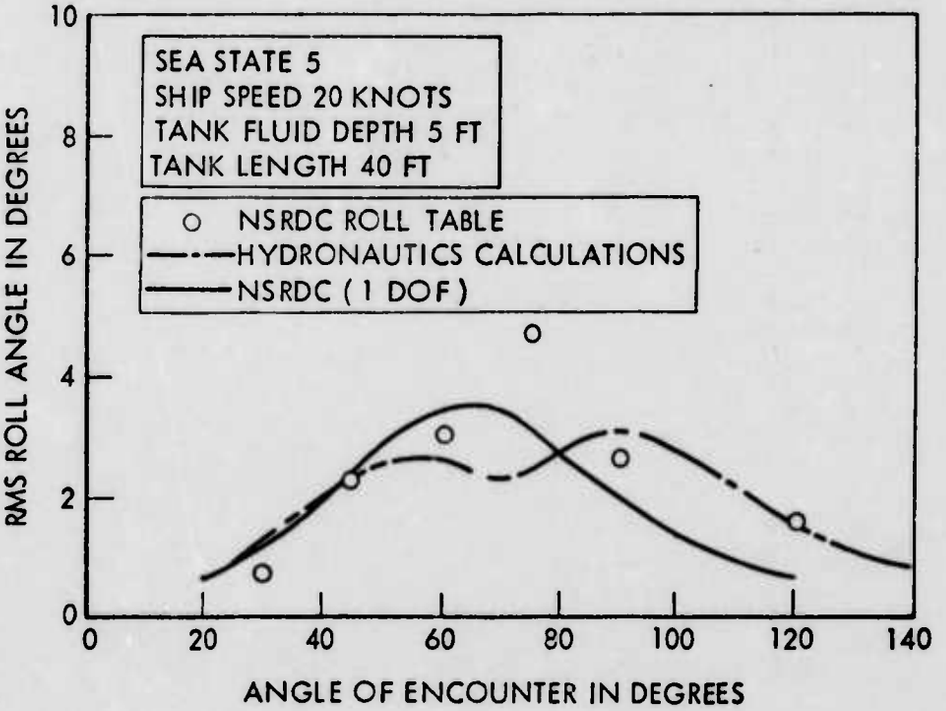


FIGURE 28 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITH PABL TANK IN SEA STATE 5 AT 20 KNOTS

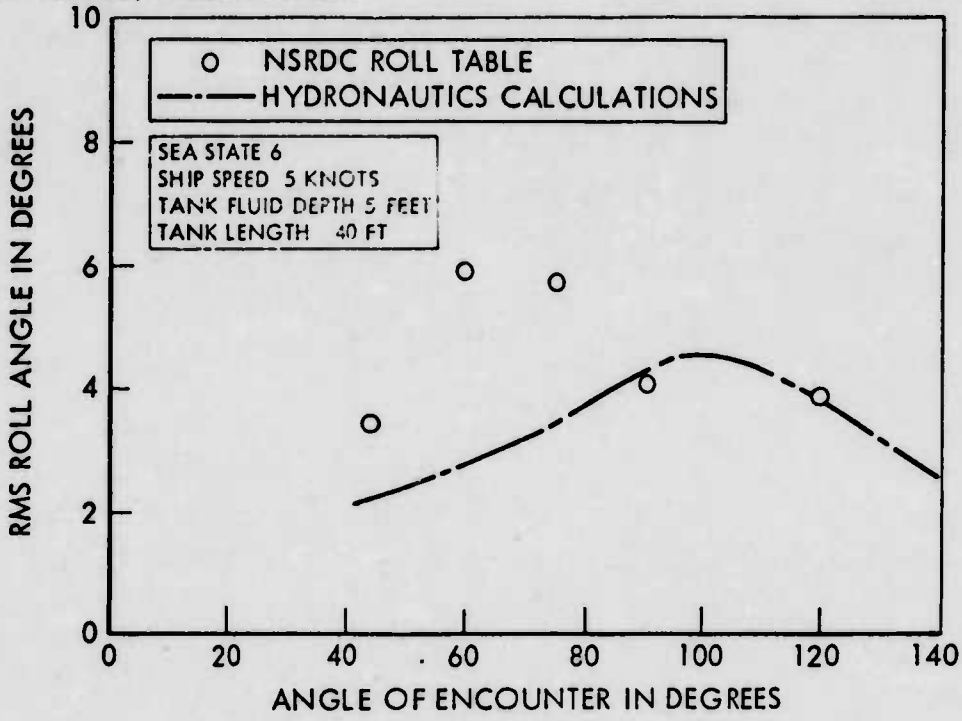


FIGURE 29 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITH PABL TANK IN SEA STATE 6 AT FIVE KNOTS

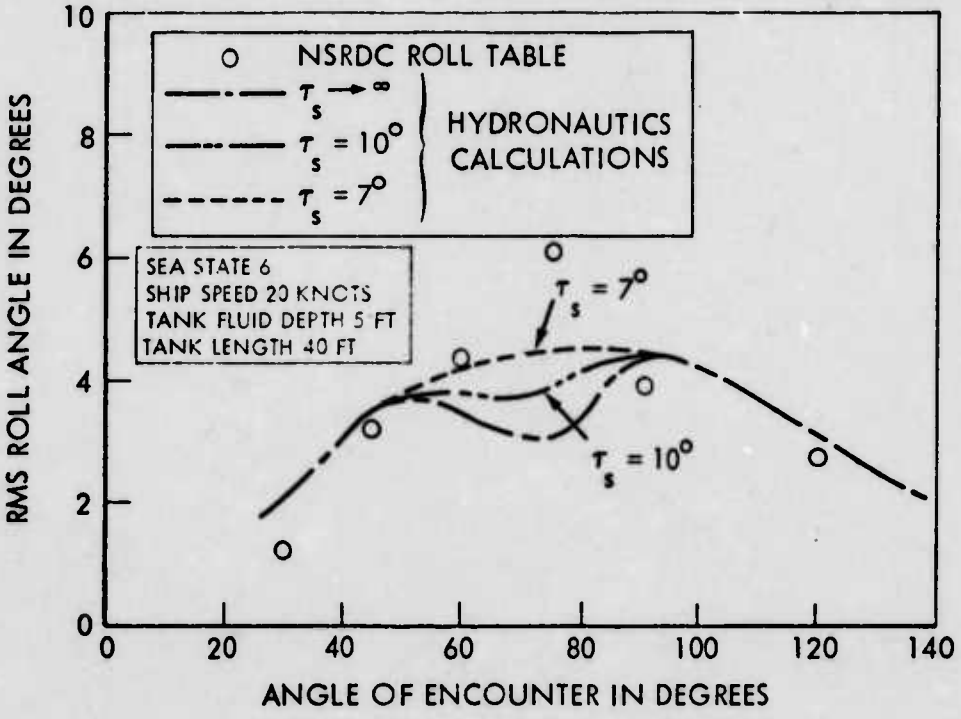


FIGURE 30 - PREDICTED ROLL MOTIONS OF SEA CONTROL SHIP WITH PABL TANK IN SEA STATE 6 AT 20 KNOTS

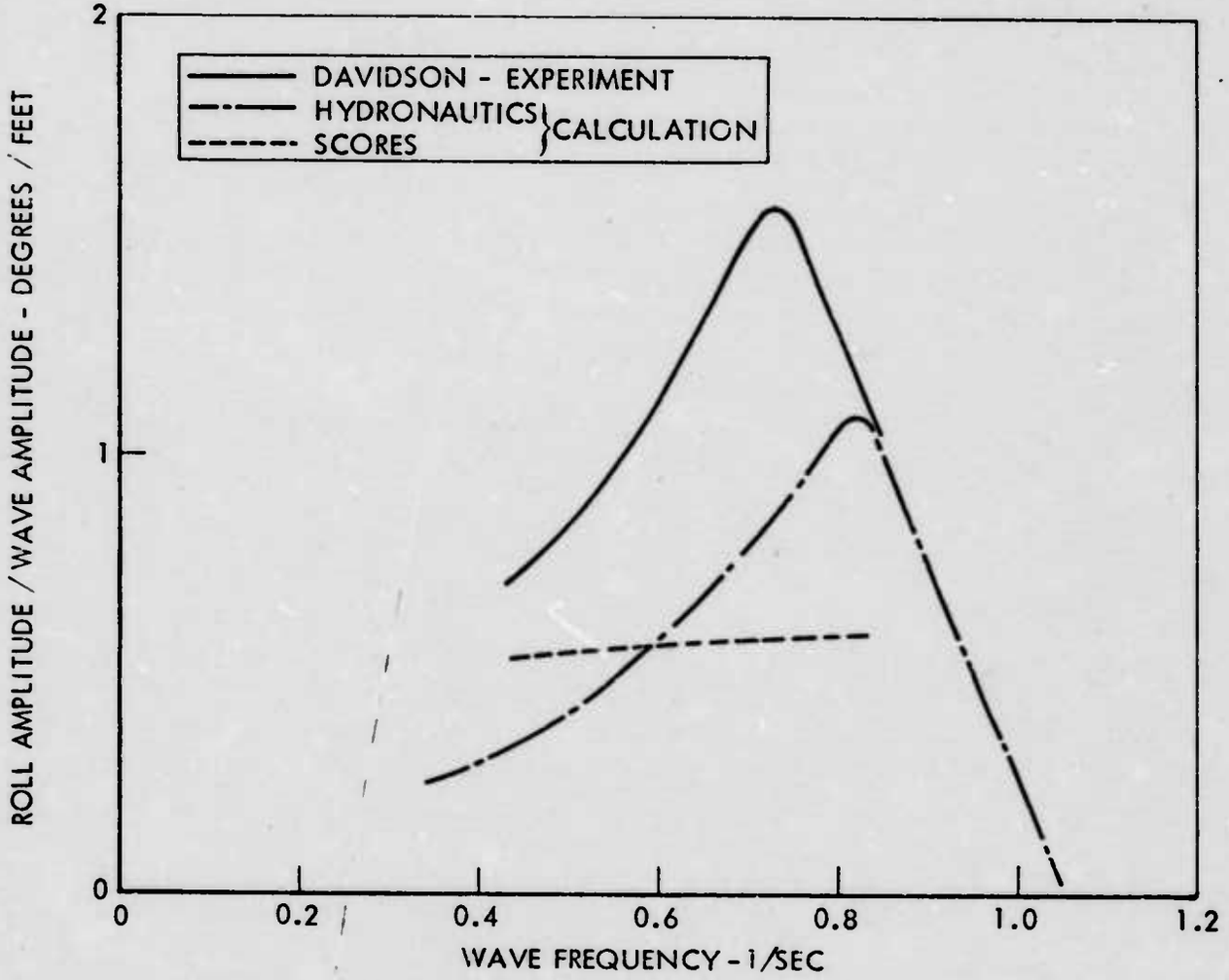


FIGURE 31- MEASURED AND CALCULATED ROLL MOTIONS OF SL-7 CONTAINERSHIP AT 25 KNOTS AND QUARTERING (60°) SEAS - HEAVY DISPLACEMENT

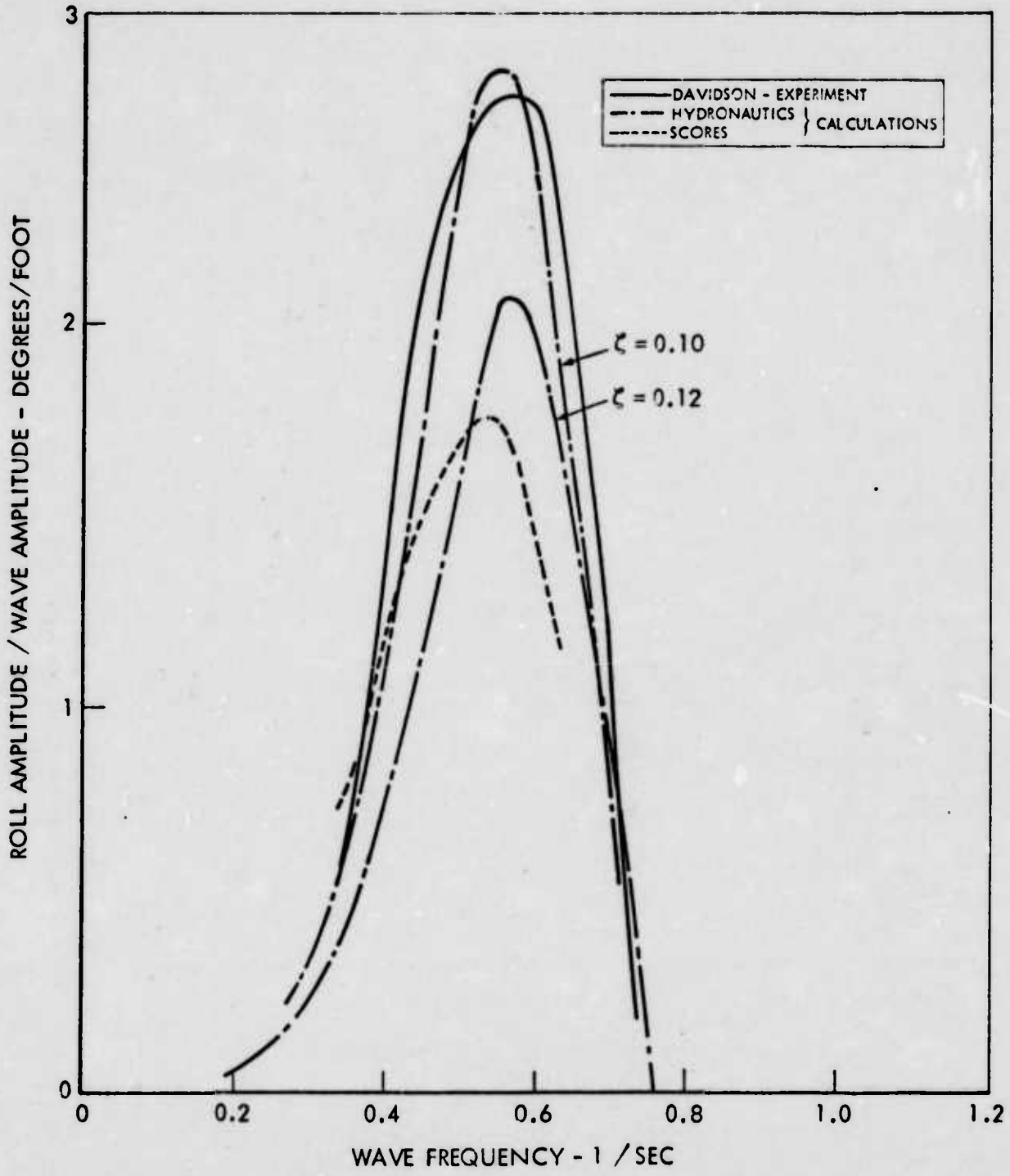


FIGURE 32 - MEASURED AND CALCULATED ROLL MOTIONS OF SL-7 CONTAINERSHIP AT 25 KNOTS AND QUARTERING (30°) SEAS - HEAVY CONDITION

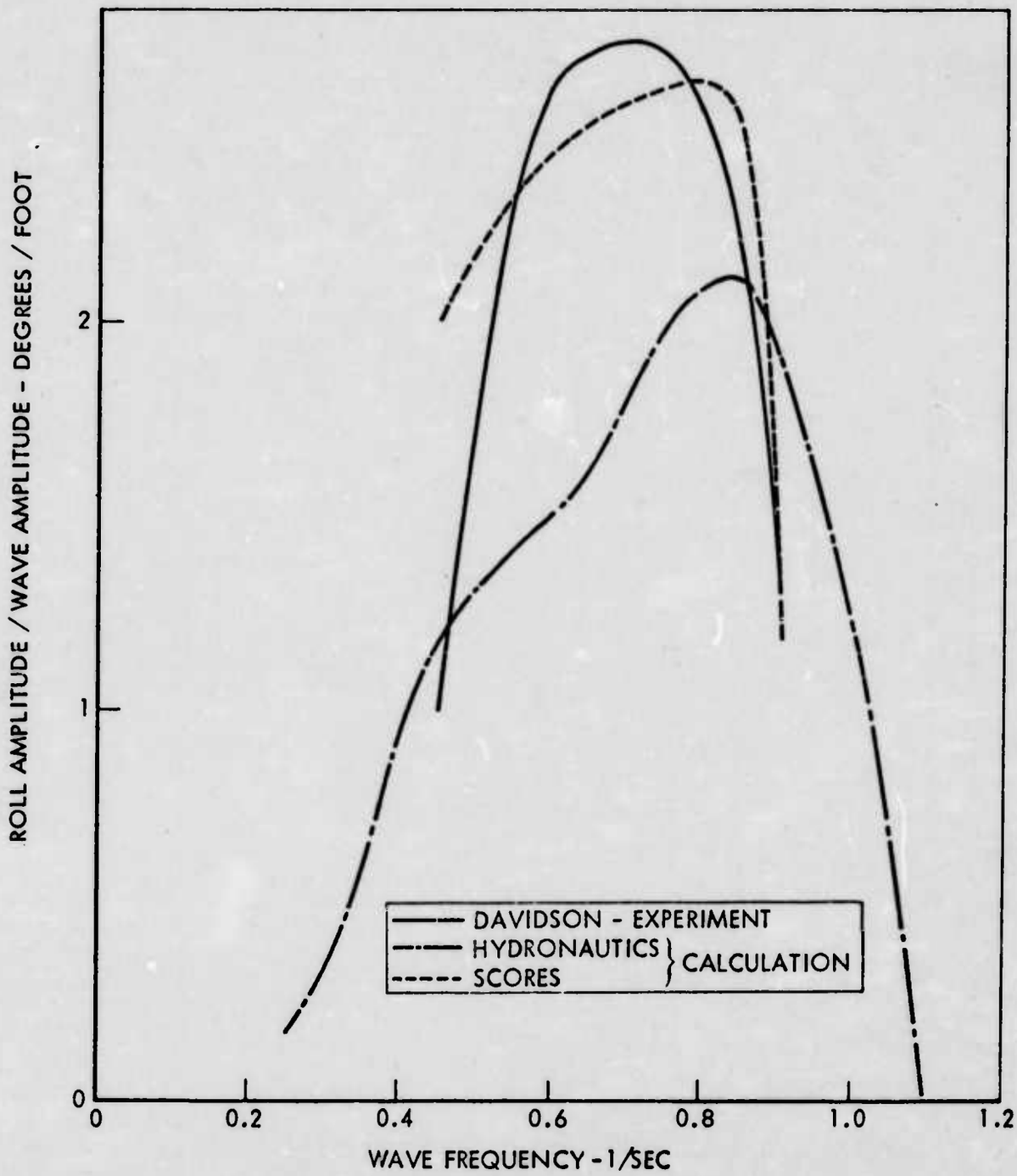


FIGURE 33 - MEASURED AND CALCULATED ROLL MOTIONS OF SL-7 CONTAINERSHIP AT 25 KNOTS AND QUARTERING (60°) SEAS - LIGHT DISPLACEMENT

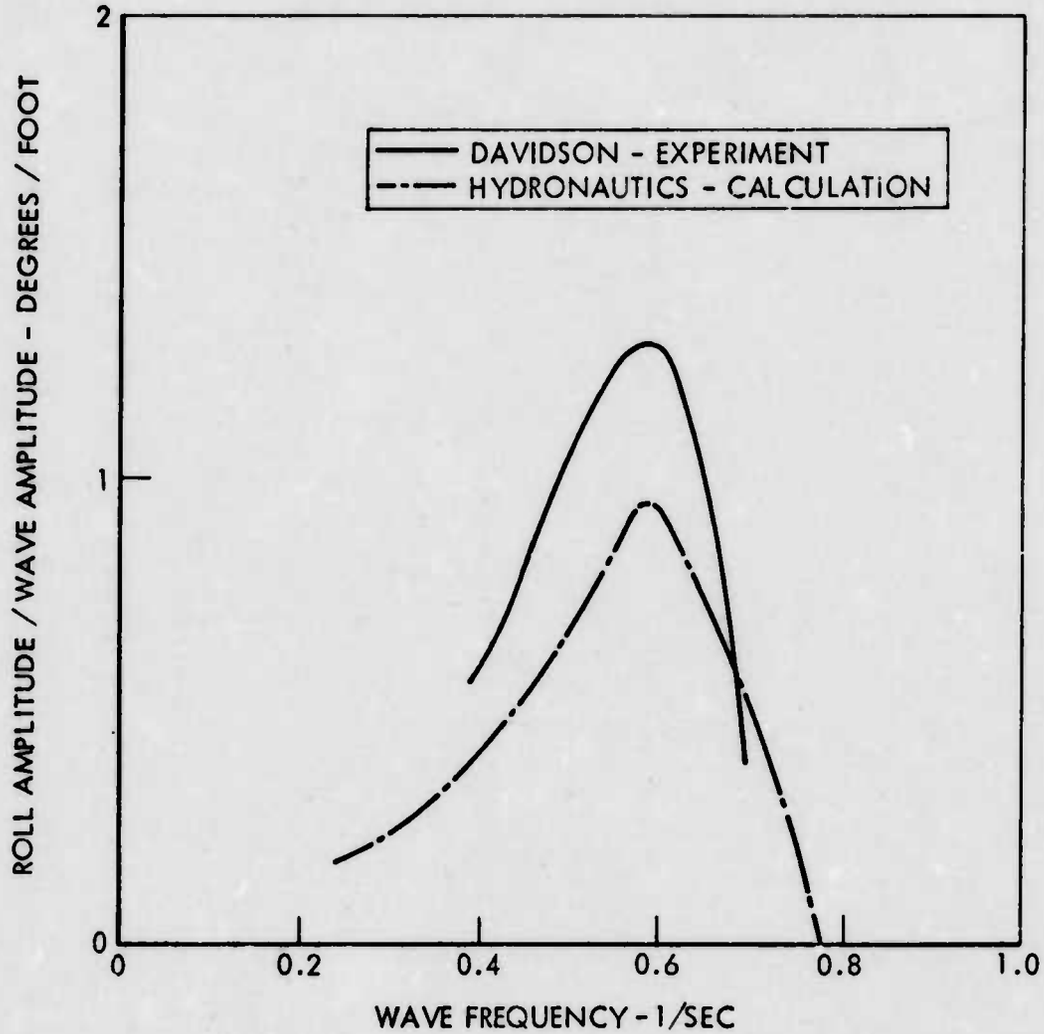


FIGURE 34 - MEASURED AND CALCULATED ROLL MOTIONS OF SL-7 CONTAINERSHIP AT 25 KNOTS AND QUARTERING (30°) SEAS - LIGHT DISPLACEMENT

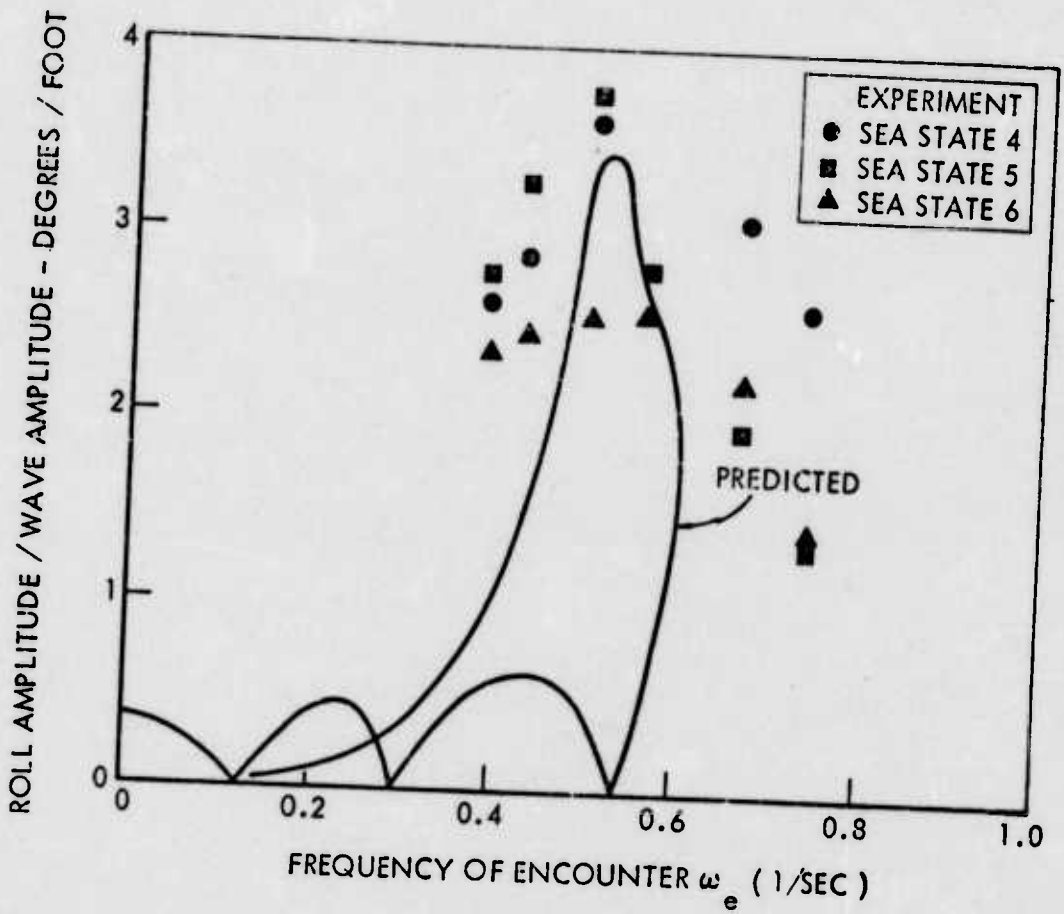


FIGURE 35 - MEASURED AND PREDICTED ROLL MOTIONS OF DESTROYER TYPE SHIP IN QUARTERING (70°) SEAS AT 24 KNOTS

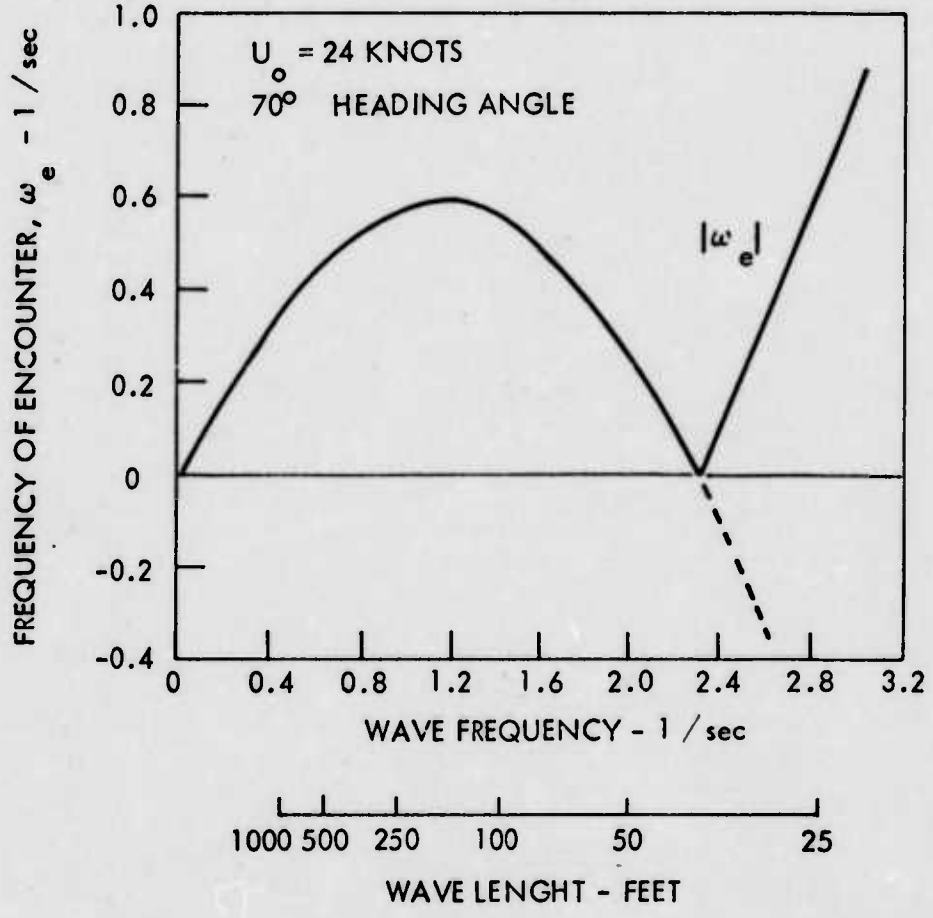


FIGURE 36 - VARIATION OF FREQUENCY OF ENCOUNTER WITH WAVE FREQUENCY AND WAVE LENGTH FOR 24 KNOT SHIP SPEED, 70° HEADING ANGLE

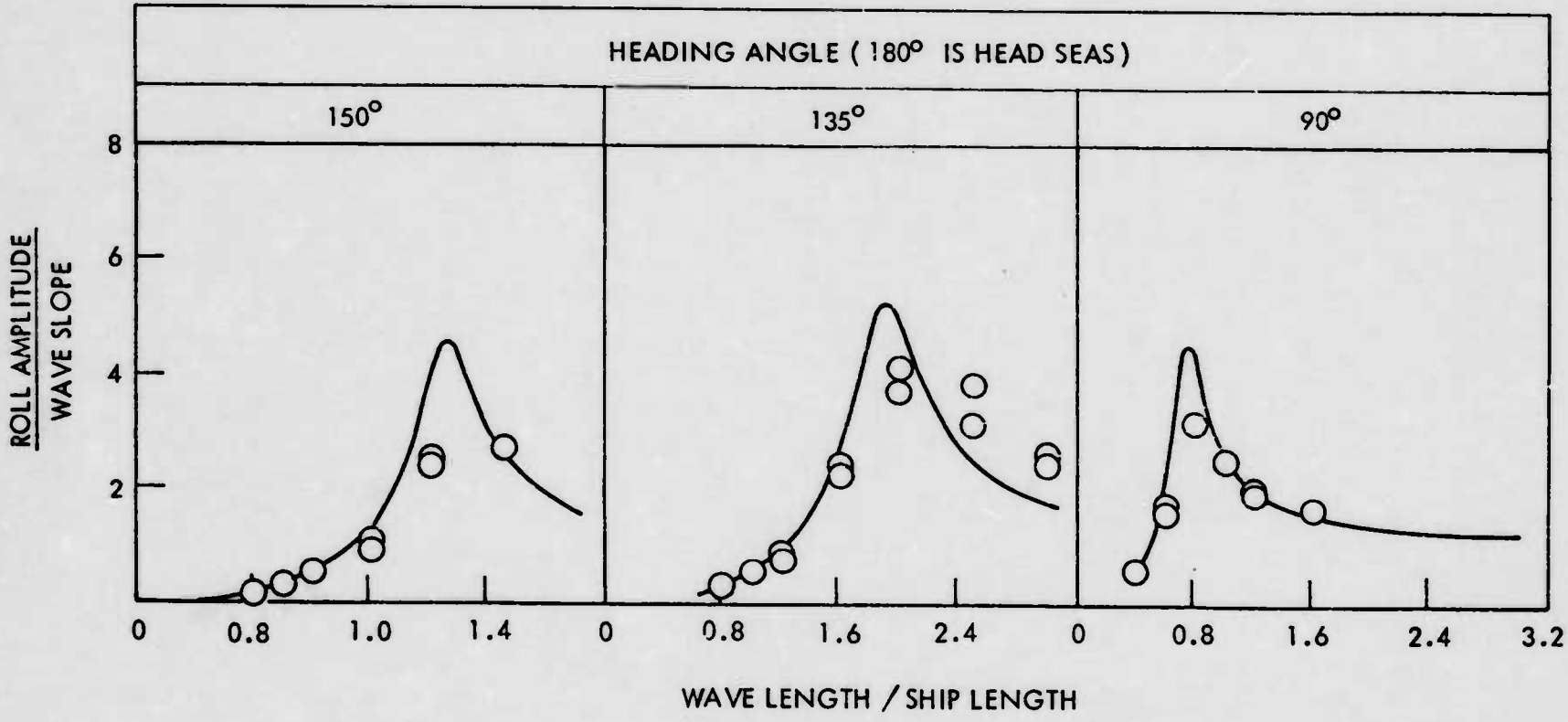


FIGURE 37 - COMPARISON OF MEASURED AND PREDICTED ROLL MOTIONS OF A DESTROYER TYPE SHIP AT 1.55 SPEED LENGTH RATIO

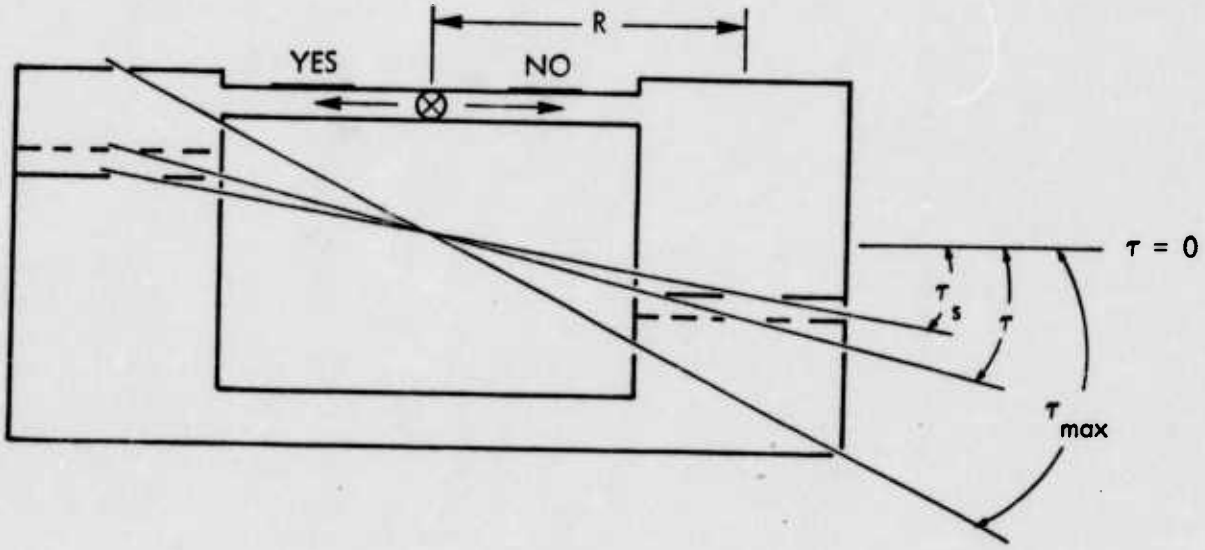


FIGURE 38 - DEFINITION SKETCH FOR CONTROLLED-PASSIVE TANKS

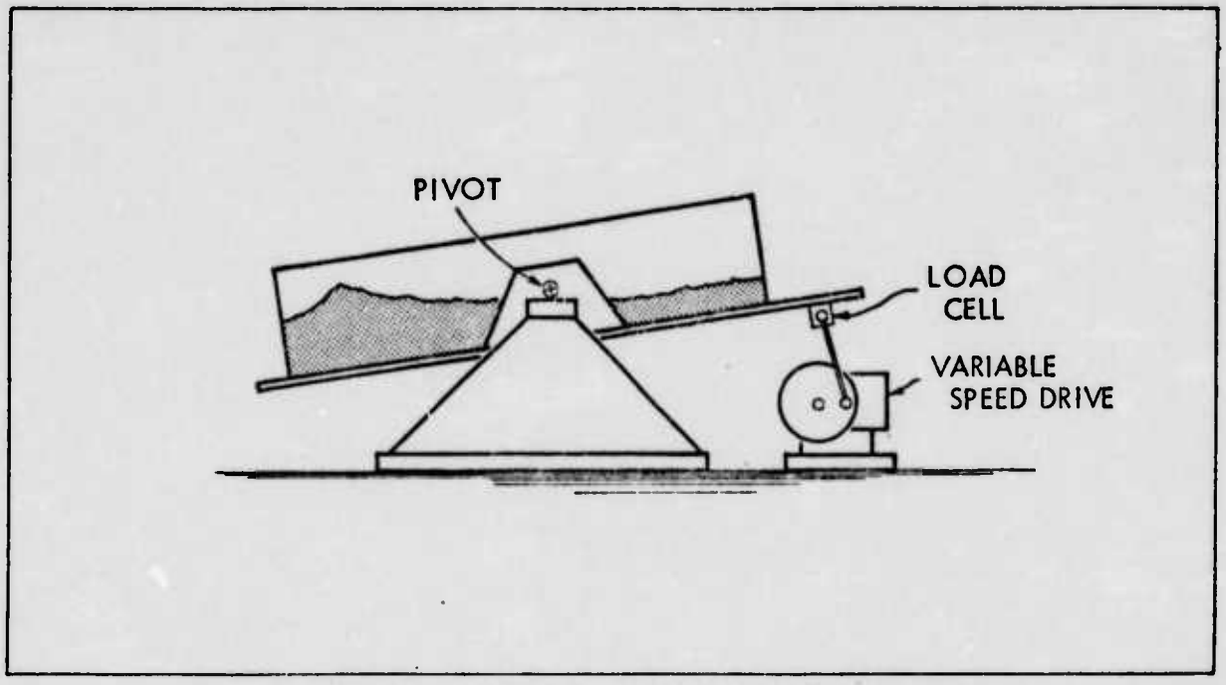


FIGURE 39 - OSCILLATING TABLE TEST FACILITY

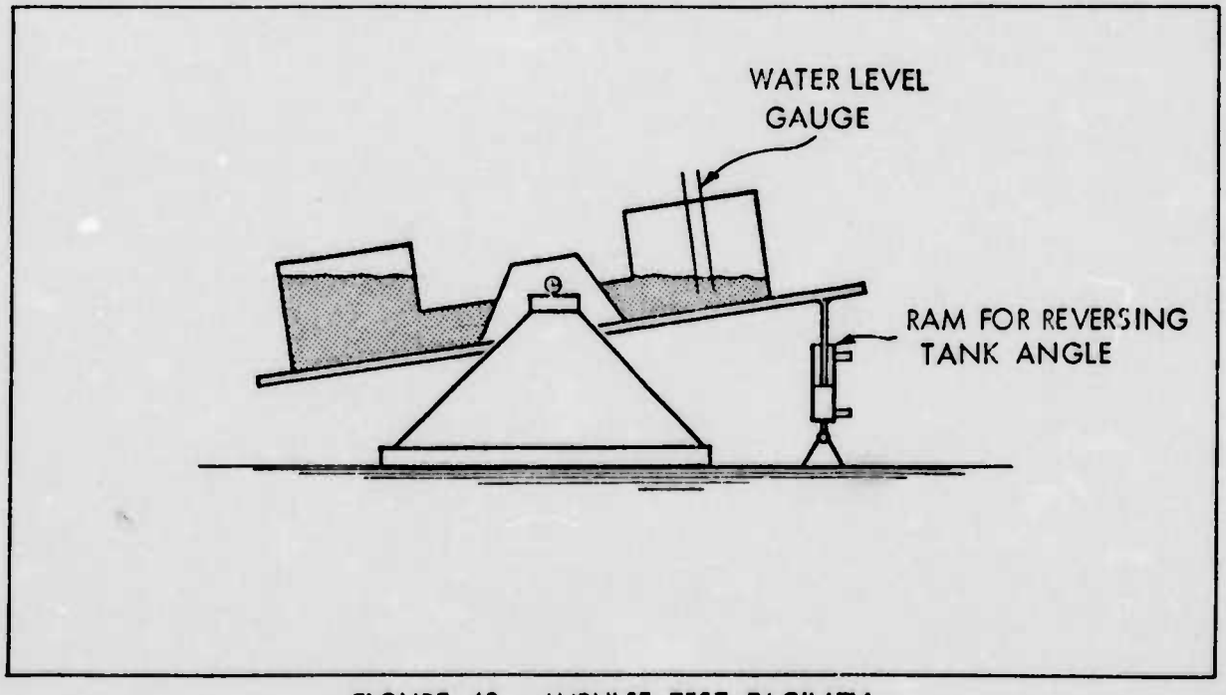


FIGURE 40 - IMPULSE TEST FACILITY