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ANALOG SIMULATION OF A PASSIVE ANTI-ROLLING TANK SYSTEM FOR AN OCEANOGRAPHIC SURVEY VESSEL

by

Betty A. Oliver and James W. Church

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NOTATION

B_s'	Roll damping coefficient for unstabilized ship
B_s	Damping coefficient for stabilized ship (full tanks)
B_t	Coefficient of linear tank water damping term
B_{t_2}	Coefficient of nonlinear tank water damping term
g	Acceleration of gravity
J_s'	Roll moment of inertia of unstabilized ship
J_s	Moment of inertia of stabilized ship (full tanks, blocked ducts)
J_{st}	Mutual-coupling moment of inertia
J_t	Moment of inertia for tank water
K_s'	Static righting moment coefficient for unstabilized ship
K_s	Static righting moment coefficient for stabilized ship (full tanks, blocked duct)
K_{ss}	Coefficient of effective wave slope term
K_{st}	Mutual-coupling term coefficient
K_t	Static righting moment coefficient for tank water
Q_s'	Amplitude ratio of unstabilized ship
ω_s'	Resonant frequency of stabilized ship
ω_s	Undamped natural frequency of stabilized ship
ω_{st}	Decoupling frequency (frequency at which coupling terms cancel); secondary resonance frequency
ω_t	Resonant frequency of tank fluid
λ_t	Normalized coupling coefficient; equal to $\frac{K_t}{K_s}$
μ_{st}	Normalized "decoupling" frequency (frequency at which coupling terms cancel); equal to $\frac{\omega_{st}}{\omega_s}$

μ_t	Normalized frequency of tank fluid relative to natural frequency of ship; equal to $\frac{\omega_t}{\omega_s}$.
θ	Angle of roll of ship
Ω	Angle of tank fluid level with respect to ship
ψ	Effective wave slope for rolling

ABSTRACT

The roll stabilization of an Oceanographic Survey Vessel by passive anti-rolling tanks has been simulated on the Analog Computer Facility at the David Taylor Model Basin. A description of the simulation and the results for a satisfactory ship-tank system are presented.

INTRODUCTION

The Bureau of Ships considers that roll stabilization is an essential requirement for the satisfactory fulfillment of the mission of a 1200-ton Oceanographic Survey Vessel presently in design. The design requirements for stabilization with the ship dead in the water preclude the use of active fins. Weight, space, and power considerations make gyroscopic stabilization unattractive. Difficulty in the perfection of the control system introduces doubt that activated tanks will ever be really successful except in ships which have a large degree of natural damping. Hence, stabilization by a passive tank system has been proposed. This system should increase the natural damping of the ship to a point at which the activated tanks when reasonably well designed can be expected to be successful.[1]

The studies necessary for the design of the passive tank system for the Oceanographic Survey Vessel have been coordinated by Mr. K. C. Ripley, Code 442, Bureau of Ships. Assistance in some phases of this study has been provided by personnel of the David Taylor Model Basin's Stability and Control Division in the form of criticisms of design parameters, discussion of the possibilities of model test correlation, and computations on an analog computer. The results of the analog computations carried out in this connection are considered to be of sufficient interest to other investigators in the field to warrant their publication. This report presents the results of the analog computations, and a description and design procedure for the simulated ship-tank system. The equations of motion and the parameters used in the simulation of the system are presented in the Appendix. An evaluation of the hydromechanic or hydraulic aspect of the ship-tank system is considered to be beyond the scope of this report.

DESCRIPTION OF SHIP-TANK SYSTEM

The proposed Oceanographic Survey Vessel is a 196-foot research vessel which must be capable of satisfactory operation in a State 6 sea at zero speed. Pertinent characteristics of the unstabilized ship are presented in Table 1.

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[1] References are listed on page 14.

TABLE 1

Pertinent Characteristics of the Oceanographic Survey Vessel

Length, feet	196
Beam, maximum, feet	37
Displacement, tons	1200
Center of gravity above baseline, feet	14.1
Center of buoyancy above baseline, feet	12.05
Draft, feet	15
Metacentric height, feet	2.05
Natural period in roll, seconds	12.6

The motion of the unstabilized ship is usually described by a linear differential equation which contains terms reflecting the moment of inertia, the damping effect of the fluid, and the static righting moment. The inertia and righting moment terms are straightforward and are computed from ship geometry. The damping coefficient, however, cannot be determined with the same assurance. During the preliminary portions of this study, a damping coefficient was used which was derived from model tests conducted on a destroyer.[2] Further investigation of the serious scaling effects noticed in British experiments prompted Mr. C. G. Moody of the Stability and Control Division to recommend a larger value for the damping coefficient and this value is used in the studies given in this report.

The values for all coefficients in the equations used to represent the ship-tank system are presented in the Appendix.

Stabilization of the ship is to be accomplished initially by passive anti-rolling tanks having the capacity to reduce the mean roll to 5 degrees or less in any sea of State 4 or less. Use of activated tanks is expected to bring the overall capacity up to that required for satisfactory operation in a State 6 sea. The tank system provided by Mr. Ripley for this study was designed to satisfy this requirement. Figure 1 shows the tank system and Table 2 presents the pertinent characteristics. The usual fluid for stabilizing tanks is sea water; however, a sodium chromate solution with a specific gravity of 1.35 was investigated in more detail in these studies. The tanks are assumed to have rectangular sections and the connecting ducts and transition pieces to have cylindrical sections.

The simulation of the ship-tank system is accomplished on the general purpose analog computer. A sine wave generator is used to provide an input representing the effective wave slope. The equations of motion are based on those presented by Chadwick[3] and include his equation for transverse motion of the roll axis. In addition, the velocity-squared tank damping term presented by Jacobson[4] is included. The authors are unaware of any previous treatment of roll stabilization that combines both the effects of sway and nonlinear damping although these effects contribute significantly to the computed response of the ship in a seaway.

TABLE 2

Characteristics of the Passive Tank System

Tank Size	
Width, feet	6.75
Length, feet	3.00
Weighted length of tank, feet	262.86
Area of one tank, square feet	20.25
Total Area, square feet	40.50
Duct Size	
Diameter, feet	1.73
Area, square feet	2.35
Tank and Duct Location	
Tank moment arm, feet	14.875
Weight of center of rotation above baseline, feet	13.0
Height of duct above baseline, feet	9.0

PROCEDURE OF THE STUDY

The procedure of the study may be delineated into two phases, the hand computation of the parameters representing the ship-tank system and the computer computation of the response of a particular design to various sine wave inputs. The activity in the first phase consists of the following steps:

(a) The inertial, damping, and righting moment coefficients of the unstabilized ship are determined from geometrical characteristics and empirical relations.

(b) The maximum wave slope is specified.

(c) Tanks are designed to have sufficient capacity for the specified wave slope with a natural frequency of the tank-water system about equal to that of the unstabilized ship.

(d) The parameters of the tank systems are determined on the basis of the design of step (c).

(e) The ship parameters of (a) are modified to include the effects of the tank on the ship characteristics.

(f) The parameters are reduced to forms suitable for computation.

The second phase of the study sees the equations representing the ship-tank system exposed to sine waves of constant amplitude and various frequencies. The results are presented as time histories of the roll angle and the input wave angle and are replotted as curves of magnification versus relative frequency for significant design changes.

Portions of these two phases are repeated for each design change investigated.

RESULTS

The roll stabilization effectiveness of the passive anti-rolling tank system for the Oceanographic Survey Vessel is presented in Figure 2 as a curve of roll magnification versus relative frequency for two effective wave slopes. The fluid in the tanks is the sodium chromate solution. The effects of the orbital motion of the axis of roll and of nonlinear tank damping are included. The data in tabular form are presented in Table 3.

Figure 3 compares the stabilization properties of the tanks for sea water and sodium chromate. The effective wave slope of 0.86 is used as a basis of comparison. It may be noted that as predicted by Chadwick in his study of the dynamics of anti-rolling tanks[3] the final system parameters are of approximately the same magnitude as those for full-scale ships having good stabilization properties. It should be noted, however, that his optimization equations (shown in the Appendix) fail to account for the density of the fluid used in the tanks. Consequently, they did not reflect the better of the two designs shown in Figure 3.

Figure 4 compares the effect on theoretical predictions of simplifying assumptions such as the use of a fixed axis of rotation or a linear tank damping term. The discrepancy between results based on linear damping and the more realistic presentation is most noticeable at the resonant frequencies, whereas the discrepancy between results based on the fixed axis of roll and the more realistic presentation is most apparent when the driving frequency approaches zero. The curve in Figure 4 which includes the effect of sway and velocity-squared damping are identical with the curve presented on Figure 3 for the ship-tank system with salt water.

TABLE 3

Results of Roll Stabilization Computations

Relative frequency ω/ω_s	Magnification of Roll ϕ/ψ					
	Unstabilized Ship	Stabilized Ship				
		Sea Water With Linear Damping ψ_0 = 0.86 degrees	Sea Water With Velocity-Squared Damping ψ_0 = 0.86 degrees	Sea Water With Velocity-Squared Damping, Including Effect of Sway ψ_0 = 0.86 degrees	Sodium Chromate With Velocity-Squared Damping, Including Effect of Sway ψ_0 = 0.86 degrees	Sodium Chromate With Velocity-Squared Damping, Including Effect of Sway ψ_0 = 2.47 degrees
.100		1.1				
.200	0.95	1.2	1.3	0.90	1.10	1.0
.401	1.10	1.55	1.6		1.20	1.35
.501	1.30			1.50	1.50	1.5
.601	1.50	2.3	2.6	1.90	2.15	2.0
.701	1.85	2.7	3.1	2.4	2.0	1.95
.802	2.70		2.75	2.0	1.2	1.90
.902	4.65	2.35	2.45	1.85	1.15	2.4
1.002	10.0	2.3	2.3	2.46	1.7	3.25
1.052		2.4				
1.102	4.25	2.5	2.55	3.15	2.6	3.3
1.152		2.45				
1.202	2.15	2.3	2.5	2.6	2.5	3.35
1.303	1.35	1.6	1.8	1.9	1.9	1.70
1.403	0.90	1.25	1.2	1.25	1.35	1.25
1.503					1.84	0.90
1.603	0.55	0.6		0.65	0.65	0.75
1.804		0.35	0.40	0.50		

The methods and procedures outlined in this report are considered to be valid and useful in the design process of a system of passive tanks for roll stabilization. For more accurate predictions of the effectiveness of such systems, it may be desirable to obtain precise coefficients for the ship and tank system from model tests.

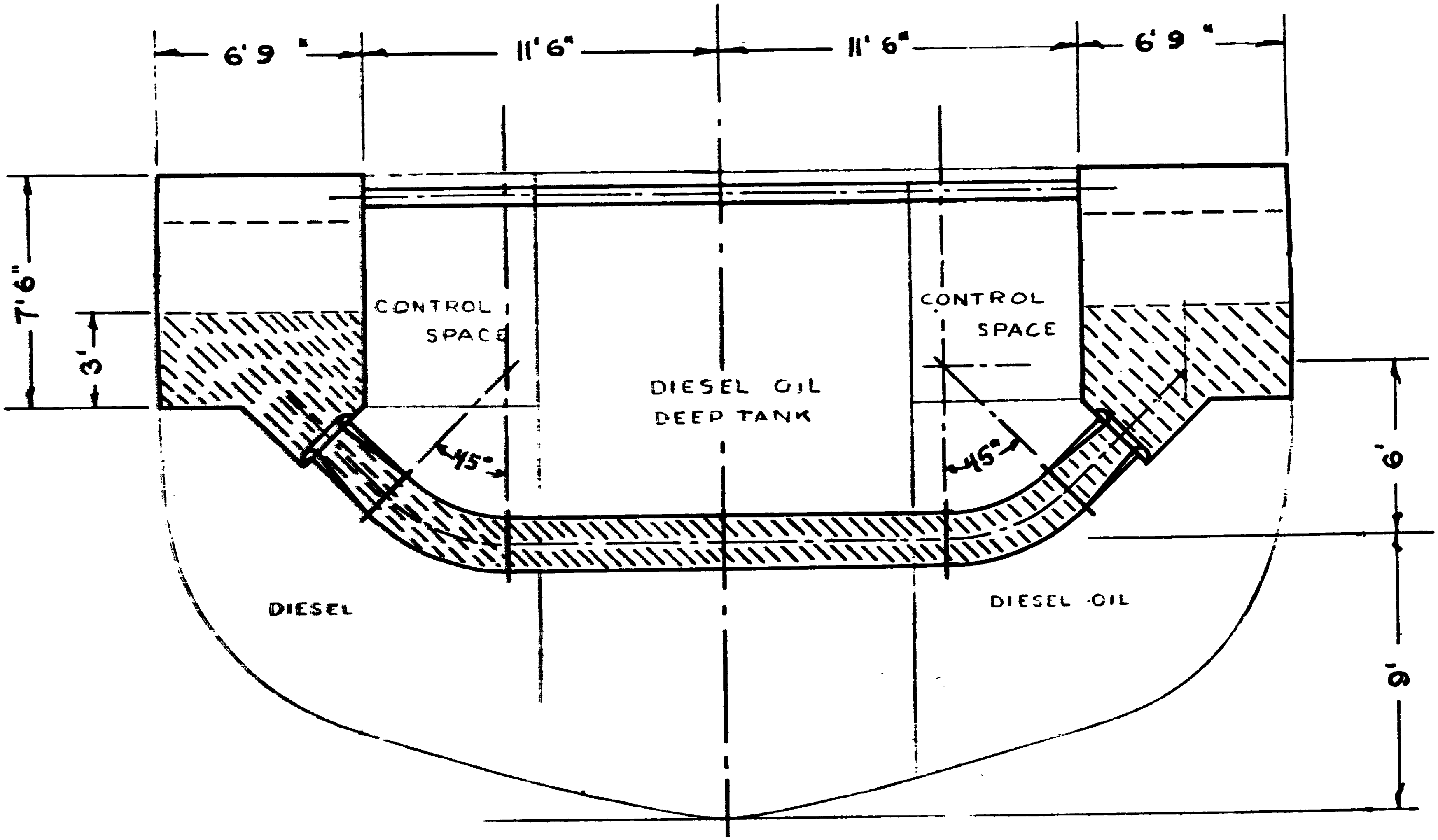

Figure 1 - Cross-Section of Oceanographic Survey Vessel Showing Passive Anti-Rolling Tanks

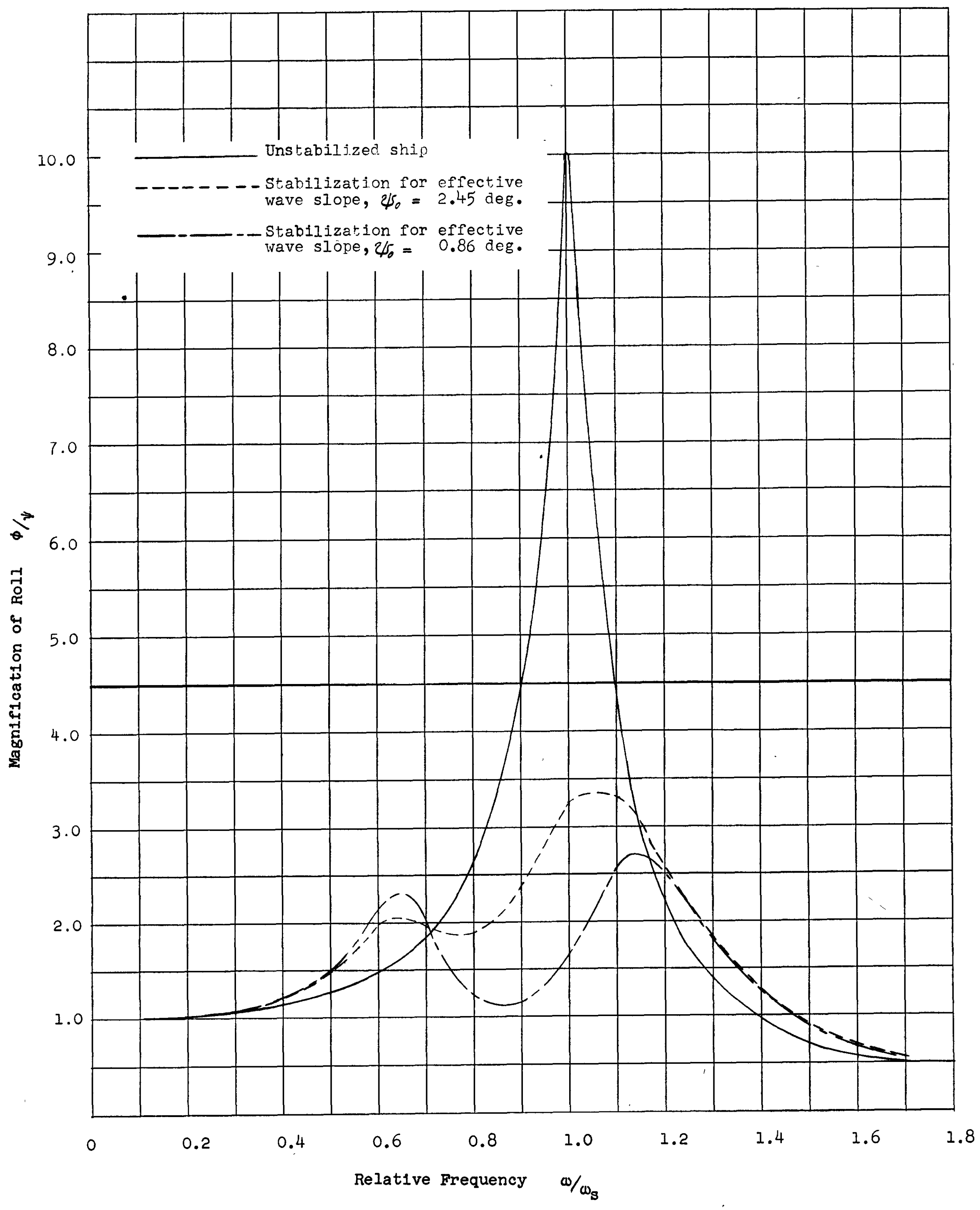

Figure 2 - Roll Stabilization of Oceanographic Survey Vessel

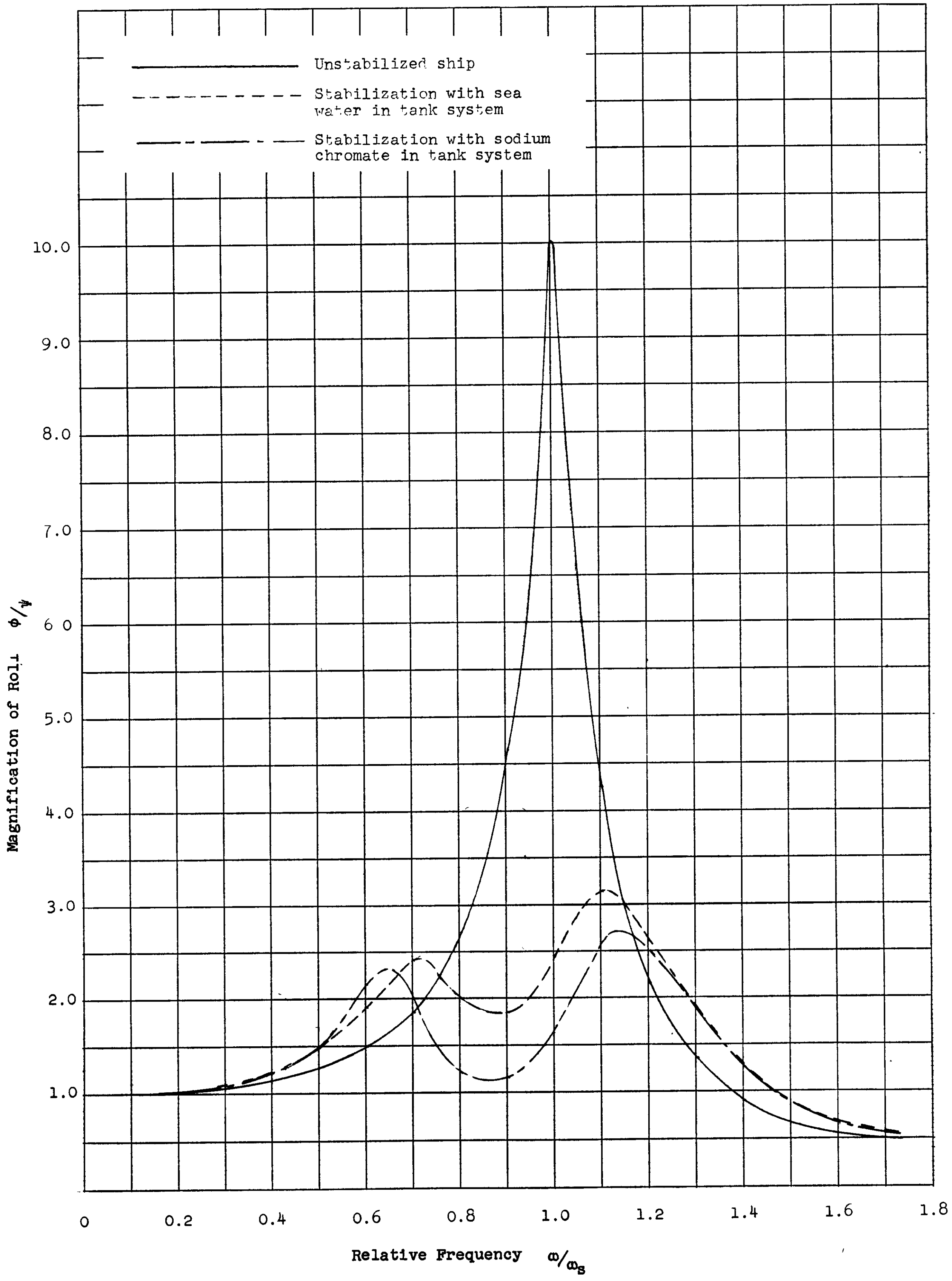

Figure 3 - Effect on Roll Stabilization of Tank Fluid Density

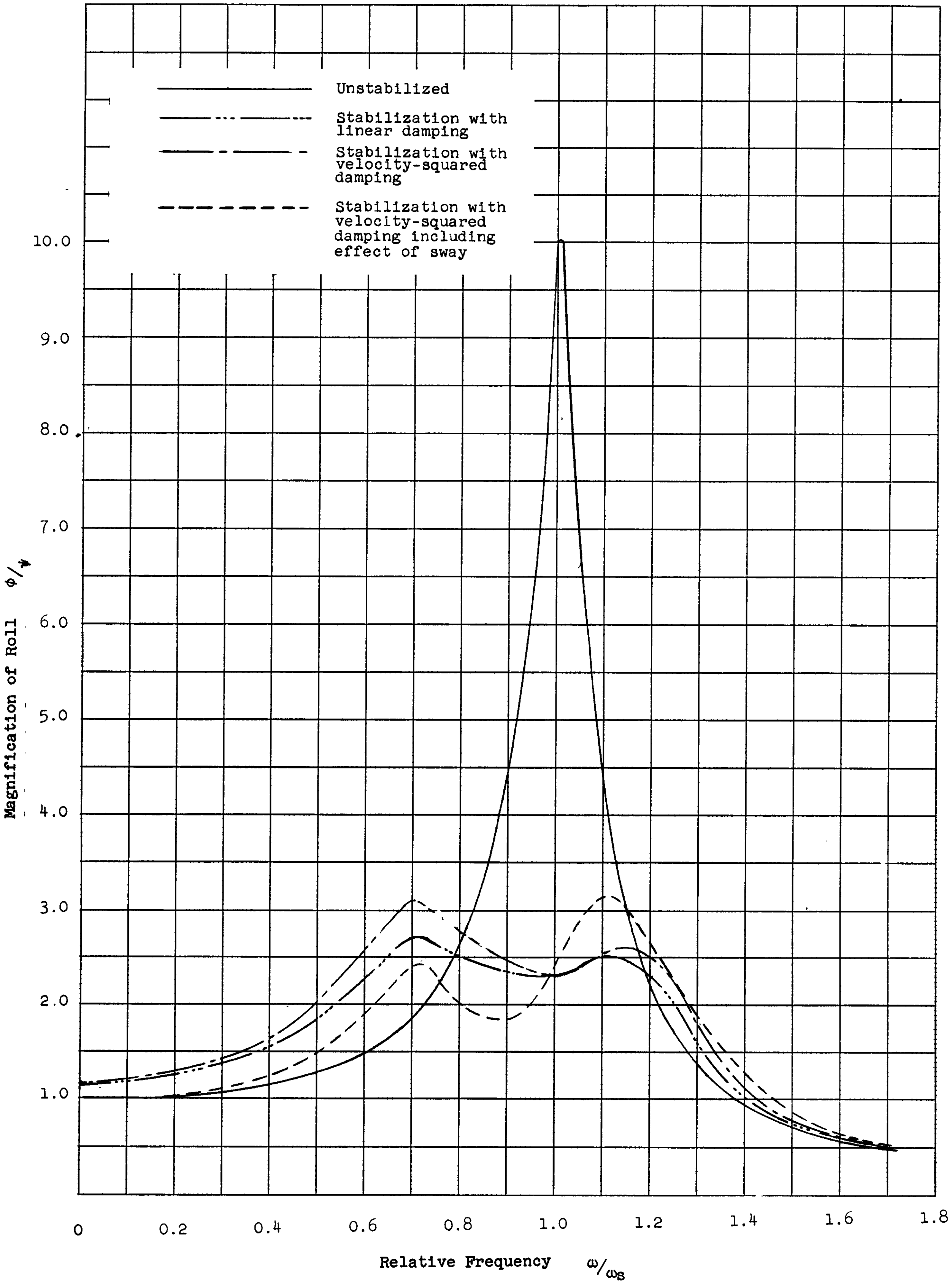

Figure 4 - Effect of Simplifications in the Equations of Motion on Computed Stabilization

APPENDIX A

Equations of Motion

The following equations of motion are similar to those presented by Chadwick[3] except for the tank damping term and the change in nomenclature to conform with SNAME Bulletin 1-5:[5]

$$J_s\ddot{\phi} + B_s\dot{\phi} + K_s\phi + J_{st}\ddot{\Omega} + K_t\Omega = K_{ss}\Psi \qquad [1]$$

$$J_{st}\ddot{\phi} + K_t\phi + J_t\ddot{\Omega} + f(\dot{\Omega}) + K_t\Omega - \frac{K_t}{g}\ddot{x} = 0 \qquad [2]$$

$$\frac{K_t\ddot{\phi}}{g} + m_x\ddot{x} = m_x g\Psi \qquad [3]$$

Where $f(\dot{\Omega}) = B_t\dot{\Omega}$ or $B_{t2}\dot{\Omega}^2$.

For analog computer solution these equations are rewritten:

$$\ddot{\phi} = -\frac{B_s}{J_s}\dot{\phi} - \frac{K_s}{J_s}\phi - \frac{J_{st}}{J_s}\ddot{\Omega} - \frac{K_t}{J_s}\Omega + \frac{K_{ss}}{J_s}\Psi \qquad [4]$$

$$\ddot{\Omega} = -\frac{f(\dot{\Omega})}{\Sigma(J_t,K_t)} - \frac{K_t}{\Sigma(J_t,K_t)}\Omega - \frac{J_{st}}{\Sigma(J_t,K_t)}\ddot{\phi}$$

$$- \frac{K_t}{\Sigma(J_t,K_t)}\phi + \frac{K_t}{\Sigma(J_t,K_t)}\Psi \qquad [5]$$

Where $\Sigma(J_t,K_t) = J_t + \frac{K_t^2}{m_x g}$.

$$\Psi = \Psi_0 \sin \omega t \qquad [6]$$

The parameters used in the study are listed in Table 4.

The optimization equations from Reference 2 are restated for convenience as follows:

Optimization with respect to Roll Angle

$$\mu_t^2 = \frac{1}{1 - \lambda_t \left(\frac{1}{\mu_{st}^2} - \frac{1}{\mu_{st}^4}\right)} \qquad [7]$$

Optimization with respect to Roll Velocity

$$\mu_t^2 = \frac{1}{1 - \frac{\lambda_t}{2}\left(1 - \frac{1}{\mu_{st}^4}\right)} \qquad [8]$$

Optimization with respect to Roll Acceleration

$$\mu_t^2 = \frac{1}{1 - \lambda_t \left(1 - \frac{1}{\mu_{st}^2}\right)} \qquad [9]$$

TABLE 4

Parameter Values Used in Study

(a) Unstabilized Ship

K_s', lb-ft/rad	2460
J_s', lb-ft-sec^2/rad	9880
B_s', lb-ft-sec/rad	493
ω_s', rad/sec	0.499
Q_s', -	10.0
ψ_{max}, deg	0.856
	2.47

(b) Stabilized Ship

Ship Terms	With Sea Water in Tanks	With Sodium Chromate in Tanks
K_s, lb-ft/rad	2343	2343
J_s, lb-ft-sec^2/rad	9830	9830
B_s, lb-ft-sec/rad	493	493
K_{ss}, lb-ft/rad	2460	2460
ω_s, rad/sec	0.495	0.495
Tank Terms		
K_t, lb-ft/rad	512	691
J_t, lb-ft-sec^2/rad	2076	2776
B_t, lb-ft-sec/rad	565	755
ω_t, rad/sec	0.495	0.495
Q_t, -	1.834	1.834
	0.820	0.820
B_{t_2}, lb-ft-sec/rad	410	548
ω_{st}, rad/sec	1.790	1.790
J_{st}, lb-ft-sec^2/rad	161	216
$\Sigma^{2}(J_t, K_t)$, lb-ft-sec^2/rad	-	2840

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