DEVELOPMENT OF A TECHNICAL PRACTICE FOR

ROLL STABILIZATION SYSTEM SELECTION

NAVSEC Report 6136-74-280
October 1974

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Prepared For
Hull Form and Fluid Dynamics Branch (SEC 6136)
Naval Ship Engineering Center
Hyattsville, Maryland 20782

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Under
Contract N00024-73-C-5572 new
Task 6120-141
(NAVSEC TPOC: R. A. Johnson)

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UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER Technical Report 7401.06-1	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) PHASE I REPORT ON THE DEVELOPMENT OF A TECHNICAL PRACTICE FOR ROLL STABILIZATION SYSTEM SELECTION. "Phase I."		5. TYPE OF REPORT & PERIOD COVERED Final Technical Report
		6. PERFORMING ORG. REPORT NUMBER TR-7401.06-1
7. AUTHOR(s) Eugene R. Miller, John J. Slager 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 Rd., Howard Co., Laurel, Maryland 20810		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS NAVSEC 6120-141.2
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Ship Engineering Center, Code 6136 Hyattsville, Maryland		12. REPORT DATE October 1974 Oct 74
		13. NUMBER OF PAGES 110 114p.
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)

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

NAVSEC 6136-74-280

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-Rolling Tanks
Anti-Rolling Fins
Bilge Keels

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

The determination of required roll motion performance or maximum permissible ship rolling, for a given application, is considered. The influence of human effectiveness and subsystem performance on this determination is discussed. Methods are given for the evaluation of and selection of the best type or types of roll stabilization systems for a given ship. Means are

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20. (Cont'd. fr p 1473 A)

provided for assessing the size and weight of various roll stabilization devices including active and passive tank roll stabilizers and active fin roll stabilizers. Means are given for estimating the damping coefficients of ships with no stabilization systems or with bilge keels only. Typical roll motions for various sea conditions, ship headings and ship speeds can be estimated from the extensive calculated results presented for destroyer and auxiliary type ships with and without stabilizers.

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INTRODUCTION

At the present time, there is considerable interest in improving the seakeeping performance of U. S. Naval vessels. In order to do this on a rational basis, it is necessary to identify and quantify the improvements in mission effectiveness and the changes in costs which will result from improved seakeeping. Of all of the ship motions, rolling is one of most objectionable and one of the most easily improved by means of a stabilization system. In order to aid in the evaluation of roll stabilization systems, the Naval Ship Engineering Center has undertaken the development of a Technical Practice for Roll Stabilization System Selection. The first phase of this effort was carried out by HYDRONAUTICS, Incorporated under Contract No. N00024-73-5572, Task No. 6120-141. The tasks specified for the first phase were as follows:

"Develop and describe a rational approach to the determination of ship roll stabilization requirements based upon ship mission and operational requirements. Present examples of properly stated, clearly defined ship roll stabilization requirements.

"For each currently feasible means of roll stabilization:

(1) Summarize key features from the standpoints of performance, ship impact (weight, space, cost, stability, ancillary equipment, etc.), and operational characteristics.

(2) Present relevant performance, ship impact, and operational characteristics data in a form which will enable the designer to quickly narrow the field of candidate systems and system sizes or capacities for a given ship roll performance requirement and hence reduce the scope, time and cost of subsequent trade-off studies."

In the final phase of the development of the Technical Practice for Roll Stabilization System Selection, it is planned that the following tasks will be carried out:

"Given that a properly defined ship roll stabilization requirement has been established, develop and describe a rational approach to the selection of stabilization system type, number, size or capacity, and location within the ship during the preliminary design phase. For each of the key steps of this approach:

(1) Define the current state-of-the-art in the technology associated with implementation based on a survey of the literature and the methodologies of the establishments currently active in the field. Review pertinent computer programs and their published documentation.

(2) Identify gaps in the current technology where additional R&D is required. Develop task statements for the required R&D efforts and recommend performing activities. Identify deficiencies in

the published documentation of pertinent computer programs and develop task statements to correct the deficiencies.

"Given that the stabilization system type, size or capacity, number of units, and location within the ship has been selected, develop and describe a rational approach to the further definition of the system during the contract design phase leading to the production of contract plans and specifications and final performance predictions. Discuss the proper role of model testing in the proposed approach for each type of stabilization system addressed. For each of the key steps of the proposed approach:

(1) Define the current state-of-the-art in the technology associated with implementation based on a survey of the literature and the methodologies of the establishments currently active in the field. Review pertinent computer programs and their published documentation.

(2) Identify gaps in the current technology where additional R&D is required. Develop task statements for the required R&D efforts and recommend performing activities. Identify deficiencies in the published documentation of pertinent computer programs and develop task statements to correct the deficiencies."

This report presents the complete results of the first phase of the development of the Technical Practice for Roll Stabilization System Selection. In particular, this report presents a rationale for the determination of requirements for ship roll performance, describes the basic types of stabilization systems, presents methods for the estimation of roll stabilization system performance during Concept Design and describes the impacts of roll stabilization systems on the ship design.

This report can be considered to be the first part of the Technical Practice for Selection of a Roll Stabilization System. However, it is believed that in its final format, the actual technical practice must be quite brief and contain references to the more complete technical practice descriptions. Since it was considered to be desirable to prepare the "abbreviated technical practice" in its entirety after the two phases of this effort are completed, the Technical Practice in its final format will be a deliverable at the end of the second phase.

DETERMINATION OF REQUIREMENTS FOR SHIP MOTIONS PERFORMANCE

General

The determination of whether or not a stabilization system should be installed on a projected new ship, and the selection of the most appropriate system, involves consideration of the following factors:

- The level of effectiveness (expressed in terms of the characteristics of the ship's motions) which the ship is required to attain,
- The comparative levels of effectiveness of the unstabilized ship and of the ship with each of several sizes and types of stabilization system installed, and
- The total impact of each stabilization system on the ship design.

The prediction of ship motions, with and without roll stabilization, and the estimation of the impact of roll stabilization on the ship design are discussed in later sections of this report. It is the purpose of this section to discuss the development of requirements for maximum acceptable ship motions. These requirements will be expressed in terms of the parameters used in ship motions analyses; thus they become, in essence, "ship motions performance requirements."

A methodology for derivation of ship motions performance requirements from the snip operational performance (or "effectiveness") requirements is presented below. First, the development of "ideal" requirements is discussed, i.e., ship motions performance requirements derived directly from the stated operational performance requirements, without regard to whether or not the ship impact (of the stabilization system which may be required) can be tolerated. Second, the development of "constrained" or "practical" ship motions performance requirements is discussed. This latter type of requirement would apply to the case in which improvement in ship performance (by means of stabilization) may be limited by ship impact limitations (such as cost and/or ship size and displacement constraints).

Determination of Critical Ship Performance Requirements

The required operational performance of a Naval ship can, to some extent, be stated in terms of its required capability to carry out each of the elements of a mission under specified environmental conditions. Those capabilities which are ship-motion-critical may be defined in the applicable Top Level Requirements (TLR) and/or the Plan for Use (PFU), and are normally stated in rather general terms. However, it is anticipated that the definition of these requirements will be more detailed in the future. These TLR or PFU requirements, plus any others which are defined, serve as the basic input to the determination of ship motions performance requirements. Examples of TLR or PFU requirements statements are given in Table 1.

Identification of Areas of Motion Sensitivity

In order to identify "areas" of motion sensitivity which may affect the ships' ability to meet its performance requirements,

TABLE 1

Sample of Ship Operational
Performance Requirements
Given By TLR or PFU

Performance Requirements	Environmental Conditions
Operation of embarked helicopters	Sea State 5 (significant wave height 12 ft, wind velocity 24 knots)
Replenishment and strikedown underway	Sea State 5 (significant wave height 12 ft, wind velocity 24 knots)
Required operational capability (other than replenishment and operation of embarked helicopters)	Sea State 6 (significant wave height 18 ft, wind velocity 28 knots
Limited operation and capability of continuing the mission without returning to port for repairs after sea subsides	Sea State 8 (significant wave height 50 ft, wind velocity 42 knots)
Survivability without serious damage to mission essential systems	Sea State 9 (hurricane conditions, when ship experiences maximum wave and wind conditions)

it is necessary to conduct a survey of the ship subsystems and operations. For studies of ship performance with and without roll stabilization, the effects of roll angle, roll rate, roll acceleration, and the combined effects of roll and sway, should all be considered; however, the state of the art is such that the information which can be developed will normally consist of data with limited factual back-up which is related to roll angle only, such as: "At roll angles greater than X degrees, the operation is difficult to carry out," or "At roll angles greater than X degrees, the subsystem will not function at 100% efficiency." In tabulating the results of the survey, the primary emphasis should be given to the definition of roll motion sensitivities which may affect the critical ship performance requirements (as specified in the TLR, PFU, etc.). A sample tabulation of the results of such a survey is given in Table 2.

Quantification of Motion Sensitivity Data

The next step in the development of ship motions performance requirements consists of an attempt to further quantify the motion sensitivity data. For the case in which roll motion sensitivity can be expressed in terms of roll angles (or if this is the only data available), the following steps are suggested:

a. Establish the limiting roll angles at which each roll-sensitive operation can be carried out at 100% effectiveness. If possible, also establish the roll angles at which the operations cannot be carried out (i.e., the angles at zero percent effectiveness) and the roll angles corresponding to intermediate levels of effectiveness.

TABLE 2

Sample Results of Roll Motion Sensitivity Survey

Subsystem or Operation	Effect of Roll Motion on Performance
Air Search Radar	Antenna stabilized to maintain coverage at 25° roll angles (vertical to out)
Bow Sonar	Initial indications are that there is no degradation until 20° or greater roll angles
RAS Conrep	System design based on satisfactory operation at 15° roll angles
Strikedown/Strike-up of Stores	No problem due to roll. Limited by human capability to lift loads when ship is rolling, but operation can be called off when rolling is severe.
Helo Handling System	Traversing system being designed to operate with ship rolling to 15°. Hauldown system not provided.
Helo Landing/Take off	Immediately after landing or before take-off, helo is vulnerable to sliding at roll angles greater than 6°.
Gun, Gun Ammo Handling	Gun system designed to operate at 15° roll angles. Gun reload requires men but ammo will be handled with positive control equipment (presumably also capable of operating at 15° roll).
Missile Launcher	Designed to operate at 15° roll angles.
Missile Handling (Magazine to Launcher)	Automatic; designed to operate at 15° roll angles and up to 30° roll at reduced capability.
Missile Strikedown	Handling of missile on dolly becomes very difficult at roll angles greater than 5°.
Torpedo Tubes	Designed to operate at 15° roll angles.
Loading of Torpedoes	Handling torpedo on dolly becomes very difficult at roll angles greater than 5°.
Personnel Effectiveness	Degrades at roll angles greater than about 6°.

b. Convert the roll angles discussed in a., above, to statistically definable quantities, as follows:

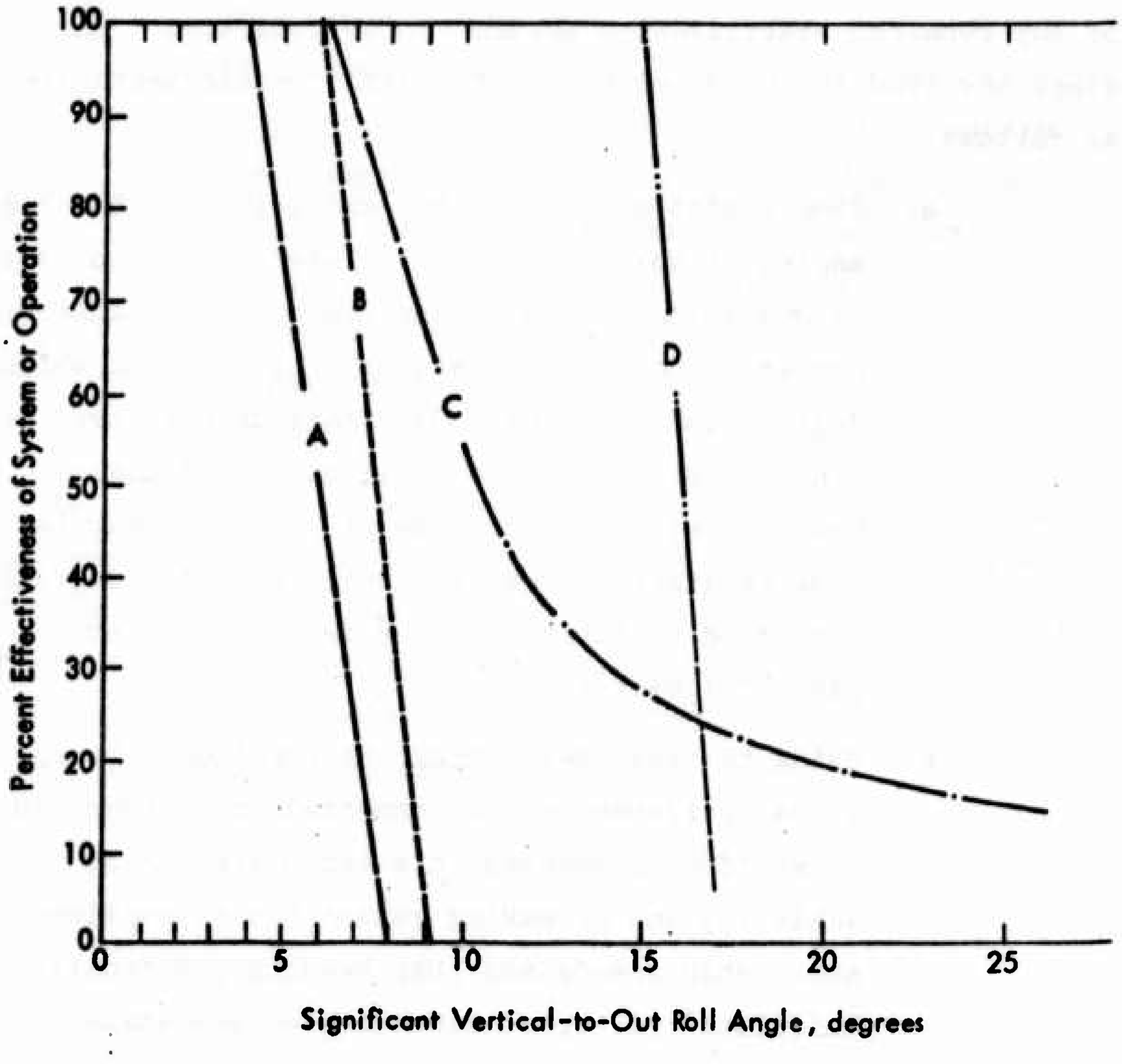
- o If, with occasional exceedance of a limiting roll angle the operation (to which the limiting roll angle applies) could still be carried out with about 100% effectiveness, then assume that this roll angle can be defined as "significant" (average of the one-third highest).

- o If, with even occasional exceedance of a limiting roll angle, the operation could not be carried out (or could not be carried out at about the 100% effectiveness level), then assume that this roll angle can be defined as "maximum" (a convenient definition for "maximum" is the average of the one-hundredth highest roll angles).

- o If possible, express the roll angles associated with intermediate levels of effectiveness, and at the zero percent level, in terms of the statistical quantities ("significant" or "maximum").

- o When the performance of the various roll-sensitive subsystems and operations have been expressed in terms of significant or maximum roll angles, convert the data, as necessary, to express it all in terms of either significant or maximum roll angle.

c. Plot the data developed in b., above, as percent effectiveness versus significant (or maximum) roll angle, as shown in the sketch below:



A - Missile and torpedo strike-down

B - Helicopter landing/takeoff

C - Human effectiveness (Averaged over many shipboard operations)

D - Refueling at sea

Development of "Ideal" Requirements

As noted previously, the ship motion performance requirements discussed immediately below may be called "ideal" since they are derived directly from stated operational performance requirements, without regard for whether or not the ship impact of any required stabilization system can be tolerated. The steps involved in the development of "ideal" requirements are as follows:

a. From a plot similar to the above sketch, the roll angles at which the most roll-sensitive subsystems or operations begin to experience performance degradation can be determined. The angle at which degradation from the 100% effectiveness level begins can be used as the "critical" roll angle; however, at least for some missions, it may be more realistic to use the angle at which degradation from an intermediate effectiveness level (say 80%) begins.

b. Using the sea states from the applicable operational performance/environmental conditions (such as would be presented in a table similar to Table 1), and by making reasonable assumptions about ship speeds and ship headings, definitive "missions" (i.e., combinations of sea states, ships speeds, and ship headings) can be formulated which appropriately describe the conditions under which each roll-sensitive operation must be carried out.

c. The data described in a. and b., above, can then be combined to formulate a ship motions performance requirement for each roll critical subsystem or operation in which the required ship performance is expressed as the required probability of exceedance of the specified critical significant (or maximum) roll angle. (Thus, required ship effectiveness could be expressed as the required percentage of mission time that the specified critical roll angle should not be exceeded.) The satisfaction of each ship motions performance requirement can be evaluated separately and a judgment made as to an overall effectiveness of the ship.

An example of the development of a ship motions performance requirement is given in Table 3. In this example, an attempt has been made to show the correlation between the operational performance requirement (Table 1), the subsystem or operation motion sensitivity (Table 2), and the finally developed ship motions performance requirement.

TABLE 3

Sample Development of Ship Motions Performance Requirement

Step 1 - Determine Critical Mission-Performance Requirement:

From the TLR, it was determined that the ship must replenish and strikedown ammunition in Sea State 5. (significant wave height = 12.0 feet)

Step 2 - Identify Motion-Sensitive Subsystem or Operation and Quantify the Data:

From a survey it was determined that handling missiles and torpedoes on a dolly, during strikedown, is hazardous at roll angles greater than about 5 degrees. It was assumed that missile and torpedo handling could be accomplished with 100% effectiveness at a 4 degree significant vertical-to-out roll angle, but would be impossible (0% effectiveness) at an 8 degree significant vertical-to-out roll angle. It was further assumed that a 5 degree significant vertical-to-out roll angle should not be exceeded if the requirement is to be satisfied.

Step 3 - Assume Additional Mission Characteristics:

The ship must strikedown missiles and torpedoes in Sea State 5. It was assumed that the operation must be possible at ship speeds of 15-25 knots; it was further assumed that the ship might not be free to choose an optimum heading during this operation.

Step 4 - Formulate Ship Motions Performance Requirement:

Based on steps 1, 2 and 3, above, the following requirement was formulated: "The ship shall have a zero percent probability of exceeding a 5 degree significant vertical-to-out roll angle while operating at any heading in Sea State 5 (significant wave height = 12.0 feet), at speeds of 15 to 25 knots.

Development of "Constrained" or "Practical" Requirements

As noted previously, the ship motions performance requirements discussed below may be called "constrained" or "practical" since they would apply when the improvement in ship performance required to satisfy the "ideal" ship motions performance requirements cannot be accomplished because of the dollar, ship size, and ship displacement constraints which have been placed on the design. Thus, in this discussion, it is assumed that "ideal" requirements have been developed, as outlined above, and that the preliminary ship roll performance and ship impact have been estimated using methods such as are described in the later sections of this report. It is further assumed that ship conceptual studies have shown that installation of the required stabilization system will result in a ship which exceeds one or more of the design constraints. The development of "constrained" or "practical" ship motions performance requirements then becomes an iterative process which has many similarities to the normal iterative design process used when trying to achieve an optimum weapon or electronics mix in a constrained ship design. This process is carried out as follows:

a. Using the "ideal" requirement as the upper limit, a less stringent requirement is assumed. Using the requirement formulated in Step 4 of Table 3 as an example, the "ideal" requirement could be relaxed in any one, or any combination, of the following ways:

- A small probability of exceeding the 5 degree roll angle could be allowed,
- The 5 degree roll angle could be relaxed to, say 6 degrees,
- It could be assumed that the operation (which is the primary basis for the requirement) would not be carried out at the ship headings giving the worst roll angles.

Conceivably, the sea state requirement could be relaxed, but this would appear to be a rather drastic change; also, restricting the ship to certain speeds which restrict rolling may not be realistic. It must be emphasized that any relaxed requirement which is finally recommended must be as reasonable as possible, with respect to ship operational performance. It is also obvious that the development of "constrained" ship motions performance requirements could involve a large number of iterative studies.

b. For each assumed "constrained" requirement, preliminary estimates of the ship motions performance must be made, to determine the approximate characteristics of the stabilization system needed to satisfy this requirement. The preliminary estimating method presented in this report can be used for this purpose. Likewise, the ship impact data presented herein can be used to prepare input for the ship conceptual studies necessary to determine if the "constrained" requirement allows a ship design which will satisfy the imposed constraints.

c. The iterative process outlined in a. and b., above, must be repeated until an acceptable balance is found between the ship design which satisfies the constraints and a "constrained" requirement which implies a minimum reduction in the desired ship operational performance.

The "constrained" ship motions performance requirement(s), developed by the methodology described above, can then serve as the basis for the detailed ship motions studies which are required for complete definition of the stabilization system chosen for the ship.

ROLL STABILIZATION SYSTEM TYPES

Bilge Keels

It has been recognized since the 19th century that the rolling motions of a ship are large because the hydrodynamic damping in this degree of freedom is small. W. Froude in 1865 recommended that bilge keels be fitted in order to increase the roll damping and thus reduce the roll motions. The damping moment generated by the bilge keels is due to a component supplied by the pressure resistance of the bilge keel itself and to a component created by the change in the pressure distribution on the hull. In a hull without bilge keels, the roll damping is provided by the dissipation of energy in surface waves, in viscous flow around the hull and by surface tension. The latter component is not important for the full scale case. The addition of bilge keels greatly increases the energy dissipation due to viscous flow.

Bilge keels have been shown to be effective even in the highest sea states and can be installed with little impact on a normal ship. The major impact is in the increase in the installed power and fuel capacity required to overcome the added resistance. This may be expected to be small except on high speed vessels. For these reasons almost all naval vessels are fitted with bilge keels.

Passive Roll Stabilization

The concept behind passive stabilization is a simple one. Most ships have very little roll damping. As a result, the energy which the waves impart to the ship is exhibited by large roll motions. The size of these motions must be sufficient so that the energy dissipated equals the energy imparted by the waves. When a passive stabilizer is placed on board a ship it is a dynamic system which has a resonance. This resonance is not so pronounced as the ship's resonance, since the stabilizer is much better damped. Now, the stabilizer is chosen so that its resonance is close to that of the ship. The tuning provides excellent dynamic coupling between the ship and stabilizer. This means that the roll energy of the ship can be efficiently transferred to the stabilizer, which, because of its high damping, converts this energy into heat. There are two important pointc. First, the stabilizer drains roll energy from the ship and thus greatly reduces the roll motion (particularly at the ship's roll resonance). Second, the stabilizer does not work unless the ship is already rolling. In other words, a passive stabilizer cannot eliminate roll motion entirely. Rather, it reduces the motion that is already there.

From the above discussion it is clear that a passive roll stabilizer can be any resonant system which couples well with roll. It must also be big enough so that it can absorb a significant amount of the ship's roll energy. Several general realizations of such passive systems have been invented, mostly during the latter part of the last century. These systems fall into two general categories: moving weight systems and tank systems.

Moving weight systems, as their name implies, involve a large weight which moves from side-to-side across the ship. The weight is converted to a resonant system typically by running it on a curved track or supporting it with springs, as shown in Figure 1. The damping can be provided mechanically or pneumatically. The moving weight system provides the smallest size passive stabilizer but has one significant problem. The weight must be large enough so that it can absorb a significant amount of roll energy. This generally means that the weight of the stabilizer must be of the order of 0.5% of the ship's displacement for ships of very small $\overline{GM}$ (compared to the ship's beam) to up to 2% of the displacement for large $\overline{GM}$ ships. For a 6,000 ton ship a moving weight of about 80 tons is required for a typical $\overline{GM}$. The mechanical problems of mounting and containing a 60 ton weight are enormous. For small boats, however, the moving weight may well be an ideal solution.

For the larger ships, tank systems have been used almost exclusively. Although these systems involve liquid weights somewhat larger than an equivalent moving weight, carrying, say 100 tons of water, in the above mentioned ship presents no particular difficulty. It is relatively easy to develop a resonant tank system. Consider the difficulty one has when walking with a full coffee cup. The coffee sloshes from side-to-side in a well defined resonant mode. Tanks of water which run the full beam of a ship, or nearly thereto, also have such a resonance. By proper installation of sufficient structure within this tank, an adequate amount of damping is obtained.

There are two general types of anti-roll tanks currently in usage: the free-surface tank and the U tube. The typical general arrangement of each is shown in Figure 2. Free surface tanks are often referred to by the trade name "Flume tank". There appears to be no substantial difference in the performance of the two systems.

Almost any fluid can be, and has been, used in tank systems. The only requirement is that it remains a fluid. Ordinary residual oils can become too viscous if not heated. However, in larger tanks Navy Special Fuel Oil or Bunker C has been used successfully. Since in tank systems the "weight" which moves back-and-forth across the ship is a fluid, it is easily disposed of. For instance, if the tank is installed high in the ship, the water (or fuel) can be readily dumped into the ocean. This gives the ship additional roll static stability for emergencies. An alternate arrangement is to provide a void tank low in the ship (usually directly below the stabilizer tank) into which the fluid can be dumped in an emergency.

In the U tube configuration, one must provide a path for the air above the fluid in one wing tank to move to the space above the fluid in the opposing wing tank. From continuity the air flow (volume rate) must be the same in this path as the water flow (volume rate) is in the lower crossover. If this air path is valved, then the amount of air flow (and thus the amount of water flow) can be controlled. Completely closing the valve virtually prevents motion of the tank water from side-to-side. In other words, closing of the valve "turns off" the tank. This might be a very important feature of any naval installation, since closing a valve can be accomplished more quickly than dumping a tank.

In free-surface type tanks, it is possible to change the resonant frequency of a tank by adding water (or draining away water). Thus, this type of tank is well suited to a situation in which the ship has a wide range of operating metacentric heights, leading to a wide range of roll resonant frequencies. It is not possible to change the tuning of a U tube so easily, although designs are in operation in which valves are placed in the water crossover duct of these tanks in order to change the resonant frequency. If the $\overline{GM}$ range varies by more than 2 to 1, it is usually necessary to install at least two stabilizer tanks, for either system, free-surface or U tube tanks. One tank would be designed to be optimum for the lower $\overline{GM}$'s and the other for higher $\overline{GM}$'s. Experience has shown that two such tanks operating in concert lead to good stabilization throughout the operating range.

One significant problem which occurs in all tank systems is saturation. The tank performs stabilization by sloshing of a fluid within it. The larger the ship roll motions, the larger are the slosh motions in the tank. When these motions become so severe that the fluid slams against the tank top, then the stabilization effectiveness decreases. Experience has shown that up to saturation, a passive tank reduces the roll motion by an almost constant percentage. Beyond saturation the roll reduction appears to be limited to a fixed number of degrees. A typical response, with and without stabilization, is shown in Figure 3. Almost all reasonably sized stabilizers will saturate in extremely high seas. If the wing tanks are restricted in height, saturation may begin in sea states as low as 5 or 6.

Unlike fin stabilization, passive stabilizers do not depend on the ship having a reasonable forward speed. The stabilizer performs well both at zero speed and at forward speeds. The major impacts of a passive anti-roll tank on a ship design are due to the required weight and volume and the effects of the reduction in stability due to the tank free surface. There are only a few passive anti-roll tank installations in U. S. Naval ships. There are, however, literally thousands of such installations in commercial ships.

Active Fin Roll Stabilization

In an active fin roll stabilization system, one or more sets of fins generate roll moments which oppose the wave excitation roll moments in response to the command of a control system. The roll motions are reduced by the resulting dissipation of energy.

Anti-roll fins are effective in all sea states at design ship speeds. Their effectiveness is reduced with reduced speed since the fin moment due to fin lift is a function of the ship speed. At zero speed the fins make only a small contribution to the passive damping of the ship.

The fin angle is controlled by a system which may sense the roll motions, velocity, acceleration and in some cases the lift on the fin. The fin is actuated by a hydraulic system which in most cases can change the fin angle from stop to stop in two seconds or less. The fin system may consist of a simple fin or a fin with a trailing edge flap.

The major impacts of an active fin roll stabilization system on the ship design are the added space and weight required for the fin control and activation system. Because of the large space required, fin installations on naval vessels are generally not retractable. There is also a small increase required in the installed power and endurance fuel to overcome the added resistance of the fins. Because of their proven effectiveness, there are about 68 U. S. Naval vessels with active fin systems. There are also a large number of installations on foreign naval vessels.

Active Tank Roll Stabilization

It is possible to use feedback control systems in tank systems in a fashion similar to the control systems used in active-fin stabilization. These systems are invariably the U tank configuration. The motion of the ship is sensed, this information is processed and some feature of the tank system is changed accordingly. Depending on what action is taken it is possible to define two different types of active tank systems. First, if the action is such to prevent (or permit) flow between the wing tanks by closing (or opening) valves in the air crossover, then the system is commonly called semi-active or controlled passive. Second, if the action is such that energy is put into (or extracted from) the tank fluid, then it is called a fully-active tank system, usually referred to simply as an active tank system. Figure 4 shows the typical arrangements of these two types of systems.

Either the semi-active or fully-active tank systems offer more stabilization than a passive tank system. The fully-active system offers the best performance. However, this system requires more complex components. For instance, a typical arrangement includes a variable pitch propeller pump connected to a motor. The pitch of the propeller is varied by hydraulic actuators commanded by the automatic control system. In a well designed system, during part of the roll cycle power is extracted from the tank and in other parts of the cycle power is supplied to the tank. In such a well designed system, the average power required is near zero and usually negative (meaning that a net amount of power must be extracted from the tank). However, the instantaneous power required (either into or out of the tank) is usually large. A typical 6,000 ton vessel may require a 2,500 HP pump for this purpose with an average net horsepower out of the tank of about 100 horsepower. It should not be surprising that the system extracts energy from the tank. It is this energy which the tank has extracted from the ship roll motion which must be dissipated. The pump system provides the means. Due to its complexity and cost, however, the active tank system does not seem very attractive. Its cost is equal to or greater than an active fin system but its performance is not as good, except at very slow ship forward speeds when the fins are ineffective.

The semi-active system does not require the mechanical complexity of the fully active system, but the performance is not as good. At a modest increase over the cost of a passive tank, one can construct a semi-active tank system which will offer a modest increase in the roll stabilization. Only a detailed systems analysis can reveal if this added cost and complexity is justified.

It should be pointed out that well designed active and semi-active tank systems have tanks whose slosh periods are considerably below that which would be desirable for a good passive tank. As a result, failure of any of the electronic or mechanical components will lead to a tank system which will not stabilize effectively or may even worsen the roll motion. This characteristic also needs to be considered carefully in performance of the overall systems analysis of stabilization. No installation of either active or semi-active systems in combatant ships is known.

In cases where it is necessary to minimize rolling at all ship speeds, or in cases where space and ship stability characteristics restrict the allowable size of both anti-rolling fins and tanks, both fins and tanks may be required. The fins are particularly effective at high speeds and tanks are particularly effective at low speeds.

At low speeds the roll damping of the fins is negligible, and all damping will be contributed by the hull (with or without bilge keels) and the tank or tanks. At high speeds the total roll damping will be approximately equal to the sum of fin, tank and hull damping, with the fin damping usually the most important. For combined fin and tank systems it will usually be desirable to make the fin area and tank size as large as possible, consistent with protecting the fins from damage and not decreasing GM or increasing ship displacement excessively.

ROLL STABILIZATION SYSTEM PERFORMANCE ESTIMATION FOR CONCEPT DESIGN

Objectives

In order to determine the potential of roll stabilization systems early in the concept design phase, it is necessary to have quantitative estimates of the potential of various types and sizes of roll stabilization systems. This quantitative information should be available rapidly and at low cost. To provide these data in the required form, a series of parametric roll motion calculations have been made and are presented in parametric plots. These data are applicable to conventional naval surface ships without roll stabilization or equipped with bilge keels, or bilge keels and passive tanks, active fins or active tanks. The results are available for short crested seas for a range of wave heights, ship speeds, heading angles and stabilizer sizes. It must be emphasized that these parametric calculations are intended for use only in the early stages of concept design to guide the formulation of definitive tradeoff studies. As soon as possible, specific calculations applicable to the particular ship design should be conducted. The parametric calculations should not be used for a definitive determination that a particular design will satisfy the roll motion requirements.

Parametric Data Development and Limitations

The parametric roll motion calculations were carried out using a computer program available to HYDRONAUTICS, Incorporated. In the form used, this program calculates a frequency domain solution to the linear equations of motion. The equations of motion include the roll, yaw and sway degrees of freedom of the ship and an equation of motion for the roll stabilizer. These equations are described in detail in Reference 1 . The seaway was described by the Pierson Moskowitz wave spectrum. The calculations were made for the short crested sea case by assuming that the wave energy was spread ±90 degrees with respect to the heading with maximum energy using a "cosine squared" relationship. This is the usual assumption made to approximate short crested seas in analytical ship motion studies.

There are two basic limitations which must be considered in the use of the parametric plots. The calculations were made for two specific naval vessels and the results were converted to a non-dimensional form. As a result the parametric data are only approximate for vessels which are "different" from the vessels used in the calculation. The definition of "different" is discussed in the following paragraphs which deal with the use of the parametric plots. The other limitation is that the calculations were made using a linear mathematical model. It is well known that roll motions are to some extent non-linear. As a result, the parametric data will overestimate the effectiveness of roll stabilization systems in high sea states unless sufficient margin is allowed in the design to prevent saturation. This is also discussed in the following paragraphs which deal with the use of the parametric plots.

Presentation of Systematic Performance Plots

Parametric plots for the roll motion of destroyer type ships are presented in Figures 5 to 34 and of auxiliary type ships in Figures 35 to 64 . For each ship type there are three types of plots. The first presents the significant out to out roll angle at the worst heading angle in short crested seas as a function of significant wave height/ship length. On each plot of this type there are data for a range of stabilizer sizes. Each plot is for a particular type of stabilizer and speed length ratio. These plots are Figures 5 to 19 for destroyer types and Figures 35 to 49 for auxilary types.

It is also necessary to have information on the variation of roll motion with heading angle. These data are presented in Figures 20 to 23 for the destroyer types and in Figures 50 to 53 for the auxiliary types. These figures apply to the ship with or without bilge keels. There are plots for each speed length ratio and curves on each plot for a range of wave heights.

When a stabilizer is added to the ship, heading angle for maximum roll, and the variation of roll with heading angle will change. In general the roll stabilizer will reduce the roll motions at the worst heading more than the motions at other headings. As a result, particularly for head seas, the roll motions for the stabilized case will be a higher percentage of the maximum roll at the worst heading. This effect is proportional to the size of the stabilizer installed. As a result, it is convenient to present the variation of roll motion with heading in terms of the unstabilized case with a correction for the effect of the stabilizer. These correction factors are presented in Figures 24 to 34 for destroyer types and Figures 54 to 64 for auxiliary types. These parametric curves may be used to estimate the variation of roll motion with heading angle in accordance with the following equation:

$$\varphi_{\frac{1}{3}\psi} = \varphi_{\frac{1}{3}max} \cdot C_{\psi} \left(1 + \frac{C_s \cdot (\text{size parameter})}{(\text{nominal size})} \right) \qquad [1]$$

where

$\varphi_{\frac{1}{3}\psi}$ = roll motion at heading angle ψ

$\varphi_{\frac{1}{3}max}$ = roll motion at worst heading angle (Figures 5 to 19 and 35 to 49)

C_{ψ} = fractional roll motion at heading ψ for case with or without bilge keels (Figures 20 to 23 and 50 to 53)

C_s = fractional change in roll motion at heading ψ for nominal stabilizer size (Figures 24 to 34 and 54 to 65)

For active or passive tank systems the size parameter is the fractional reduction in the ship GM due to the free surface of the tank. For active fins the size parameter is the static roll angle of the ship per degree of fin deflection at a speed length ratio of 1.2. The nominal stabilizer size parameter used in the calculation of C_s is 0.1 for the tanks and 0.1 for the active fins.

The roll angle data are presented in terms of the significant or average of the ⅓ highest out to out roll motion. In many cases requirements will be stated in terms of other measures of the roll angle. These are all related to each other by constants which are defined in Table 4 .

TABLE 4

Relationship Between Significant Out to Out Roll Angle and Other Statistical Measures of Roll Angle

	Out to Out	Zero to Out
Significant (average of ⅓ highest)	1.0	0.5
Average of 1/10 highest	1.272	0.636
Average of 1/100 highest	1.667	0.834
Average value	0.626	0.313
RMS value (amplitude)		0.250

As noted, the parametric calculations were carried out for two specific ships and the results were plotted in a non-dimensional form. The characteristics of the two ships used are given in Table 5 . The roll motion data were plotted as a function of significant wave height to length ratio and speed length ratio. A number of calculations were carried out for other similar naval vessels and the results were found to be similar when presented in the non-dimensional manner described. In some cases a roll motion estimate may have to be made for a vessel which does not approximate the characteristics of the types calculated. In these cases, the motions can be interpolated or extrapolated from the available data on the basis of the ratios GM/B and L_{BP}/B.

TABLE 5

Characteristics of Ships Used for Parametric Calculations

	Destroyer Type	Auxiliary Type
Class	DLG 9	LKA 113
LBP	490 ft	550 ft
B_{max} @ W.L.	51.2 ft	82 ft
Draft	17.9 ft	25.5 ft
Displacement	5,876 ℓ. tons	18,690 ℓ. tons
$\Delta/(L/100)^3$	49	110
C_B	0.45	0.57
C_p	0.57	0.60
C_x	0.79	0.95
GM (uncorrected)	5.58 ft	6.49 ft
GM (corrected for F.S. of normal* tankage)	4.70 ft	5.27 ft
Vertical moment of F.S. of ship's actual anti-roll tank	-	20,237 ft tons
KG**	19.95 ft	30.31 ft***
KB	11.12 ft	14.30 ft
KM_r	25.53 ft	36.80 ft
LBP/B	9.59	6.71
B/Draft	2.86	3.22

*Excluding anti-roll tank.

**"Solid KG" (i.e., all liquids in tanks included and considered to be "frozen")

***Effect of weight and moment of ship's actual anti-roll tank is included.

Stabilization System Size Parameters

The roll motions are a function of the size of the stabilization system. For the purpose of the parametric plots a non-dimensional expression for the system size is used. There are different size parameters for each of the stabilization system types. These are described in the following paragraphs.

For the case of the ship alone or the ship with bilge keels, the critical parameter is the equivalent linear roll damping coefficient. This coefficient is the ratio of the actual damping to the critical damping of the linear system. The damping ratio, C/C_c, is a function of the ship characteristics and the size of the bilge keels. Methods of estimating the relationship between the ship characteristics, bilge keel size and damping ratio are given in the next section on ship impact data. The damping coefficient is also a function of the roll amplitude and tends to increase with amplitude. The relationships given in the next Section for roll damping versus ship characteristics are applicable to the heavy rolling (15 to 20 degrees zero to out) as well as the moderate rolling case.

For passive or active anti-roll tanks, the critical size parameter, K_{st}, is the fractional reduction in $\overline{GM}$ due to the free surface of the anti-roll tank. The parametric calculations were carried out for values of this parameter between 0.10 and 0.40. Other factors, such as tuning, were selected, based on experience, to produce optimum system performance. The parameters which define the performance and size of the tank are discussed in detail in the following sections on ship impact.

It should be noted, however, that the parametric roll motions data assume that the anti-roll tank does not saturate. For this to be valid, the saturation angle of the tank should be about twice the significant zero to out roll angle.

For active fin roll stabilization, the critical size parameter is the static angle ratio φ_s. This is the static roll angle of the ship per degree of deflection of the fins. For the parametric calculations, this angle was defined at a speed length ratio of 1.2. The range of the static angle ratio, φ_s, was from 0.1 to 0.3 for the parametric calculations. The parametric calculations were also based on available data for fin deflection rates and on a control system which senses roll angle and roll rate. Additional information is presented in the following section on ship impact.

Example of Roll Motion Estimate from Parametric Data

Given: Find the significant out to out roll motion of of a destroyer type vessel at a heading angle of 120 degrees for the following case:

$$V/\sqrt{L} = 0.8$$

Significant wave height/LBP = 0.025

Case 1 Ship with bilge keels $C/C_c = 0.085$

Case 2 Ship with passive tank, $K_{st} = 0.30$

Case 3 Ship with active fins $\varphi_s = 0.3$

Case 4 Ship with active tank, $K_{st} = 0.30$

Item	Case 1	Case 2	Case 3	Case 4
	Bilge Keels	Passive Tank	Active Fin	Active Tank
From Figure	7	11	14	18
Roll at worst heading $\varphi_{\frac{1}{3}max}$	15.6°	8.7°	5.7°	8.0°
From Figure	22	22	22	22
Heading angle correction C_ψ	0.97	0.97	0.97	0.97
From Figure		26	29	33
Stabilizer size correction C_s	None	0.0	0.0	-0.02
Size parameter	-	K_{st} = 0.3	φ_s = 0.3	K_{st} = 0.3
Nominal size (for Equation 1)	-	0.1	0.1	0.1
From Equation 1, calculate significant roll angle at ψ = 120°	15.2°	8.4°	5.5°	7.4°

ROLL STABILIZATION SYSTEM IMPACT ON SHIP DESIGN

Ship Alone and With Bilge Keels

As indicated in the section of this report dealing with the estimation of roll stabilization system performance, in order to determine the roll motions of a ship with or without bilge keels it is necessary to estimate the effective roll damping coefficient. The roll damping of a ship arises from the dissipation of energy in surface waves and viscous or form drag. As a result, the damping must depend on the ship's geometry, size of bilge keels, roll amplitude and frequency. Various methods have been proposed for the calculation of the effective roll damping as in References 2 and 3. However, there is very little systematic experimental data available to support these estimation procedures and the effort required is excessive for the concept design phase.

There are several empirical methods available for the estimation of the roll damping coefficient in the concept design phase. The first is the standard "rule of thumb" for zero ship speed i.e., roll damping coefficient of ship without bilge keels $\sim$ 0.01 to 0.03 and roll damping coefficient of ship with bilge keels $\sim$ 0.06 to 0.08. The roll damping coefficient can also be estimated from roll decay tests in calm water of similar ships (model or full scale). Figure 65 illustrates the calculation of the effective roll damping coefficient from the data of a roll decay test.

In order to provide more quantitative data on the relationship between roll damping and bilge keel size, the data available in the literature and at HYDRONAUTICS, Incorporated was analyzed. This results in the following empirical relationship between the ship characteristics, bilge keel size and roll damping coefficient (or damping ratio) at zero speed:

$$(C/C_c)_o = \frac{0.55 \left[A_{BK}\, w^{\frac{1}{2}} + .0024\ LBd^{\frac{1}{2}} \right] d^{5/2}\ \varphi^{\frac{1}{2}}}{\Delta B^2} \qquad [2]$$

where $(C/C_c)_o$ = damping coefficient at zero speed

A_{BK} = bilge keel area (total), ft^2

L = waterline length, ft

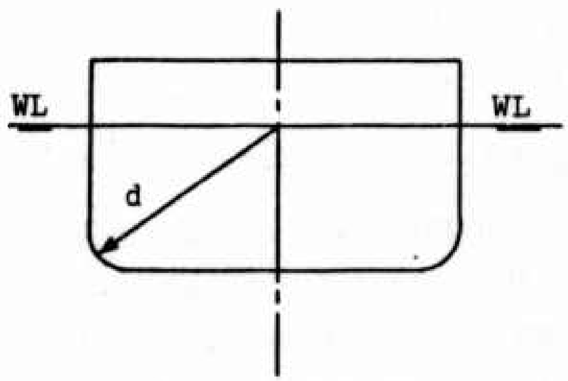
B = waterline beam, ft

w = bilge keel width, ft

Δ = displacement, long tons

φ = roll angle (zero to out) in radians

d = distance from centerline at load waterline to turn of bilge, ft (see sketch)



This equation appears to be valid for both moderate rolling and heavy rolling (i.e., d = 15 degrees or more zero to out). This

This equation is best suited to fine hull forms with relatively round bilges and to bilge keels which are relatively wide (bilge keel length to width of 40 to 50, which is typical of modern warships). The equation will tend to underpredict the damping coefficient of hulls with very sharp bilges or very narrow bilge keels. As equation [2] is empirical, it should not be used beyond the concept design stage. Figure 66, which compares calculated and measured damping coefficients for a number of ships, indicates the general accuracy of equation [2]

The change in roll damping coefficient with ship speed can be shown to be a function of Froude number and ship geometric characteristics;

$$\Delta(C/C_c) = (C/C_c)_u - (C/C_c)_o = 0.00085\,(L/B)(L/GM)^{\frac{1}{2}}\,(F/C_B) \times \left[1 + (F/C_B) + 2(F/C_B)^2\right] \qquad [3]$$

where $\Delta(C/C_c)$ = change in roll damping coefficient due to forward speed U

$(C/C_c)_u$ = damping coefficient at speed U

F = Froude number = $U/\sqrt{gL}$

C_B = block coefficient.

Equation [3] has been found to accurately predict the change in roll damping with ship speed for cases such as those given in References 2, 4 and 5. The change in damping coefficient can also be characterized, approximately, by a ratio of damping coefficients at finite speed and zero speed. Figure 67 shows a typical variation of this ratio with speed-length ratio, based on the data from References 2, 4 and 5.

Passive Roll Stabilization Tanks

There are a number of factors which must be addressed in the design of a passive tank system (either free-surface type or U tube type). They are:

a. Tuning. The tank should be tuned to a natural frequency near the ship's roll resonant frequency. Experience has shown that this frequency should be 6 to 10% higher than the ship's roll resonant frequency. The natural frequency of the ship can be estimated by the formula $\omega_n = 15.7\sqrt{\overline{GM}_{uncorrected}}/B$ radians per second, where B is the ship's beam. Thus the natural frequency of the tank, ω_{tn}, should be approximately $17\sqrt{\overline{GM}_{uncorrected}}/B$, radians per second.

b. $\overline{GM}$ Loss. The tank should have sufficient size so that it can absorb a significant amount of energy from the ship's roll motion. A theoretical analysis of this problem shows that the pertinent size parameter is the ratio of the free surface loss of the stabilizer to the metacentric height of the ship with the fluid in the stabilizer tank but frozen in position (no free surface loss). This ratio $\delta\overline{GM}/\overline{GM}_{uncorrected}$ should be in the range of 0.15 to 0.30.

c. Damping. The equivalent linear damping ratio for the tank sloshing should be in the range from 0.2 to 0.6. The damping of this slosh motion comes about from the drag of the fluid as it passes by the structure within the tank and when it enters and exits from the wing tanks. All of these losses are quadratic in nature and this causes the damping ratio of the tank to depend on the amplitude of roll motion as well as the actual structural configuration. Experience has shown that for typical tank configurations, the damping ratio for normal roll amplitudes is usually in the desirable range or somewhat below. The damping can easily be increased in these latter tanks by adding more-structural drag members within the water crossover. For instance "I" beam stanchions for damping are typical for both U tube and free-surface tanks.

For the purposes of feasibility design, it is usually safe to ignore the damping requirement. The tank is to be arranged so that all unnecessary structure is not inside the tank or (more important) inside the water crossover. Since it is not feasible to predict accurately the damping from theoretical considerations, a model test of the tank should be performed prior to finalization of design. During these tests various structural arrangements can be tried in order to achieve the proper damping.

d. Capacity. The volume of water in the tank must be sufficient so that the tank does not saturate in a low sea state. The requirement for free surface loss usually dictates the planform of the tank; the requirement for capacity then dictates the height of the tank. A good rule of thumb appears to be that the tank should have sufficient height so that the tank must be rolled 10°-15° before either the fluid hits the tank top on the low side or the bottom runs dry on the high side. If the tank is a given height, then the greatest tank capacity in this sense occurs when the tank is about one-half full.

e. Location. Experience has shown that a passive tank system can be located almost anywhere in the midlength of the ship, preferably no further forward than 0.25L forward of amidships or aft of 0.35L aft of amidships. For some ship types, the vertical location of the tanks can be important. Generally speaking, the higher the tank, the more effective it will be because the moment generated by the transverse acceleration of the fluid acts to reduce motions when the tank is located above the center of gravity. However, it is usually inconvenient to mount a tank high in a ship, since this is a most useful area. As a rule of thumb, if the ratio of $\overline{GM}$ to beam is 0.1 or less, it makes little difference where the tank is located vertically. For ratios of $\overline{GM}/B$ greater than 0.2, a tank located in the bilges may loose 50% of its effectiveness or more.

In summary, a passive stabilizing tank should have a frequency about 6 to 10% larger than the ship's frequency, a free surface loss 15-30% of the uncorrected $\overline{GM}$, a tank angle capacity of 10°-15°, and be located high in the ship (especially if the ship has a high $\overline{GM}/B$).

The material presented below is intended only for the purpose of rough sizing of the stabilizer tanks. For final design, model tests and numerical simulations must be performed if a good stabilizer is to result.

1. Free Surface Tanks. A design procedure suitable for preliminary design has been developed and exists in the BuShips Design Data Sheet #9290-4. The procedure for estimating the sizing is generally equivalent to that given in the DDS. A general configuration of a tank is given in Figure 68. In order to minimize the weight of the tank water it is very desirable that the tank run the full beam of the ship. For this configuration, the free surface loss is given by

$$\delta\overline{GM} = \frac{\rho_f}{\rho_s}\,[D(B^3-b^3) + db^3]/(420\ \Delta),$$

where Δ is the displacement of the ship, ρ_f is the density of the fluid in the tank and ρ_s is the density of the seawater. If b is small compared to B, then for a first approximation one can say

$$\delta\overline{GM} \approx \frac{\rho_f}{\rho_s} DB^3/(420\ \Delta) \qquad [4]$$

$\delta\overline{GM}$ is to be chosen on the basis of the required stabilization, the available $\overline{GM}$, the minimum allowable corrected $\overline{GM}$ and any other considerations. In typical systems $\delta\overline{GM}$ is 15% to 30% of the uncorrected $\overline{GM}$. Generally, the larger the $\delta\overline{GM}$, the greater the stabilization. With the desired $\delta\overline{GM}$, then, and the known B and Δ, Equation [4] can be used for a first approximation for the tank's fore- and-aft dimension D.

For the tank to have the maximum capacity it should be filled half-full. That is h should be H/2. In this case, the angle for the beginning of saturation is given by

$$\tau_{sat} = \tan^{-1}(H/B) \approx 57\ H/B \text{ degrees.} \qquad [5]$$

If h is not H/2, then

$$\tau_{sat} \approx 114\ h/B \text{ or } 114\ (H-h)/B \text{ degrees,} \qquad [6]$$

whichever is <u>smaller</u>. τ_{sat} should be chosen to be greater than 10°. For a half-full tank (h = H/2), a typical choice is H = 0.2B (corresponding to $\tau_{sat} = 11.4^{\circ}$).

It remains to choose b and d so that the tank has the proper tuning. If

$$B' = B + b\,(D-0.9d)/0.9d \qquad [7]$$

(The factor 0.9 is applied to account for damping structures)

then the tank natural frequency, ω_{nt}, is given approximately by:

$$\omega_{nt} = [(\pi g/B')\tanh(\pi h/B')]^{\frac{1}{2}} \qquad [8]$$

If h/B is less than 0.1, then we can approximate this result by

$$\omega_{nt} = \pi\sqrt{gh}/B' \qquad [9]$$

Equating this to the required tank frequency,

$$B' = 1.05B\sqrt{h/\overline{GM}_{uncorrected}} \qquad [10]$$

The quantities on the right-hand side are known and it remains to determine d and b in [7] so that B' is correct.

Combining all of these formulae, the volume of the tank fluid V_f can be estimated by

$$V_f = [D(B-b) + db]h \qquad [11]$$

and the weight of the tank fluid, W_f, by

$$W_f = \rho_f g\,V_f, \qquad [12]$$

where $(\rho_f g)$ is the density of the tank fluid. For this system, the total tank volume, V_t, is given by

$$\forall_t = \forall_f \, H/h \qquad [13]$$

Example: Suppose a ship with a beam of 80', an uncorrected $\overline{GM}$ of 6' and a displacement of 8,000 tons. Also suppose that longitudinal bulkheads exist 20' off the centerline so that it is convenient to choose b = 40', If it is determined that a tank free surface loss of 25% ($\delta\overline{GM}$ = 1.5') is required then from [4]

$$D \approx 420 \times 1.5 \times 8000/80^3 = 9.85'$$

Thus take D = 10'. If the tank is half full (h = H/2) and if the value of $\tau_{sat} = 11.4^o$,

$$h = 8' \quad \text{and} \quad H = 16'$$

From [10]

$$B' = 1.05 \times 80 \sqrt{8/6} = 96.9'$$

From [7]

$$B' = 80 + 40\,(10-0.9d)/0.9d = 96.9'$$

or

$$d = 7.78'$$

The exact formula is used to determine the exact free surface loss

$$\delta\overline{GM} = [10(80^3-40^3) + 7.78(40^3)]/(420\ \Delta), \text{ or}$$

$$\delta\overline{GM} = 1.482'$$

In order to achieve the required free surface loss of 1.5' we must increase all the longitudinal dimensions by the ratio of (1.5/1.482) = 1.0124. Note that this will not change the tuning.

In summary, the final configuration is

$$\left.\begin{array}{l} B = 80' \\ H = 16' \\ D = 10.12' \\ h = 8' \\ b = 40' \\ d = 7.87' \end{array}\right\}$$

the volume of tank fluid is, from [11]

$$\forall_f = (10.12 \times 40 + 40 \times 7.87) \times 8 = 5757 \text{ ft}^3$$

If sea water is used then the weight of tank water is

$$W_f = 5757/35 = 164.5 \text{ tons},$$

or approximately 2% of the ship's displacement.

A few notes should be made with regard to this calculation. First if B' turns out to be smaller than B then the tank will bulge out in the middle, rather than neck down as shown. If this is undesirable for some reason, then an alternative is to raise the liquid level in the tank, h. If the total tank height, H, is not raised correspondingly, then the tuning will be achieved at a sacrifice in τ_{sat}; that is, at a sacrifice in the high sea state capability. If B' = B, then the tank is simply a rectangular tank spanning the ship.

When the tank is necked down, as in the above example, then the tank can be configured either as shown in Figure 68I or 68C, without changing the performance. If the tank is to be used for a range of $\overline{GM}$'s for the ship, then optimum tuning will occur if the fluid level is approximately increased linearly with the change in $\overline{GM}$ over that for which the tank is designed. For the example ship the fluid level is 8' for a 6' $\overline{GM}_{uncorrected}$. If the ship is operating at an 8' $\overline{GM}$, the required water level is 8 x (8/6) = 10.7'.

2. U-Tube Tanks. A design procedure for U-tube tanks has been developed by Webster (7), and the procedure for sizing presented here follows the material in this report. It is assumed that the tank runs the full beam of the ship, as shown in Figure 69 . For this system, the free surface loss is given by

$$\delta\overline{GM} = \frac{\rho_f}{\rho_s} D(B^3-b^3)/(420\ \Delta), \qquad [14]$$

of if b is small compared to B,

$$\delta\overline{GM} \approx \frac{\rho_f}{\rho_s} D\ B^3/420\ \Delta \qquad [15]$$

As for the free surface tanks, this tank will have the maximum capacity if the wing tanks are approximately half full.

For this situation

$$\tau_{sat} = \tan^{-1} (H/B) \approx 57 \, H/B \text{ degrees} \qquad [16]$$

If h is not H/2

$$\tau_{sat} \approx 114 \, h/B \text{ or } 114 \, (H-h)/B \text{ degrees,}$$

whichever is <u>smaller</u>.

Again we desire τ_{sat} to be greater than 10°, yielding a usual choice of $H = 0.2B$.

In order to determine the tank tuning, we define an equivalent length of tank S', given by

$$S' = h + B-b + b[D(B-b)/2dp]. \qquad [17]$$

Then, the tank frequency is given by

$$\omega_{nt} = \sqrt{2g/S'}. \qquad [18]$$

Equating this to the desired tank frequency we get

$$S' = 0.223 \, B^2/\overline{GM}_{uncorrected} \qquad [19]$$

The quantities on the right hand side are known and thus determine S'. It remains to choose b, p and d so that S' is correct.

The volume of the tank fluid is then given by

$$\mathcal{V}_f = D(B-b)h + bdp, \qquad [20]$$

and the weight of the tank fluid by

$$W_f = \rho_f g\, \mathcal{V}_f. \qquad [21]$$

The total volume of the tank is

$$\mathcal{V}_t = D(B-b)H + bdp. \qquad [22]$$

Example: Choose the same ship as before (i.e., B = 80', $\overline{GM}_{uncorrected}$ = 6' and Δ = 8000 tons) and a free surface loss of 25% (δGM = 1.5'). Also choose b = 40', then from [15]

$$D = 420 \times 1.5 \times 8000/(80^3-40^3), \text{ or}$$

$$D = 11.25'$$

For τ_{sat} = 11.40, H = 16' and h = 8' for a half-full tank. From [19],

$$S' = 0.223 \times 80^2/6 = 237'$$

From [17], choosing D = d,

$$237 = 8 + 80-40 + 40[11.25 \times 40/(2 \times 11.25 \times p)],$$

or

$$p = 4.23'$$

In summary, the final configuration is

$$\left.\begin{aligned} B &= 80' \\ H &= 16' \\ D &= 11.25' \\ h &= 8' \\ d &= 11.25' \\ b &= 40' \\ p &= 4.23' \end{aligned}\right\}$$

The volume of the tank fluid is then

$$\forall_f = 11.25 \times 40 \times 80 + 40 \times 11.25 \times 4.23, \text{ or}$$

$$\forall_f = 5504 \text{ ft}^3$$

and, if sea water is used

$$W_f = 157 \text{ tons.}$$

With this U-tube tank, the tank frequency is virtually independent of the fluid level. That is, if this tank is used for a fuel tank, the tuning will not be much changed as the fuel is removed from the tank.

Active Fin Roll Stabilization

In the concept design phase the impact of an active fin system on the ship design requires an estimate of the fin size and the resulting weight, volume and control power. In the section of this report dealing with roll stabilization system performance, the fin size was defined in terms of the static roll angle which could be generated per degree of fin angle at a ship speed equal to a speed length ratio of 1.2. The fin size required to produce a given static roll angle, or the static roll angle developed by a given size fin can be determined as follows:

The static roll angle is given by:

$$\varphi_{static} = \arc\sin\left(\frac{K_{fin}}{\Delta\ \overline{GM}}\right) \qquad [23]$$

where φ_{static} = static roll angle

K_{fin} = total roll moment of fins

Δ = displacement

$\overline{GM}$ = metacentric height

The total roll moment generated by the fins is given by:

$$K_{fin} = (\tfrac{1}{2}\rho\, u^2\, A_{fin}\, C_{L_\alpha}\, \alpha_F) \cos\gamma \cdot d_{fin} \qquad [24]$$

where ρ = mass density of water — slug/ft³

u = speed at a speed length ratio of 1.2

$= 1.2 \times \sqrt{L_{WL}} \times 1.689$ — ft/sec

A_{fin} = total fin area (both sides of ship) — ft²

C_{L_α} = effective fin lift curve slope — /degrees

α_F = effective fin angle of attack (= 1.0° for this calc.)

γ = angle between fin and center of gravity, see Figure 70

d_{fin} = distance from CG to center of area of fin, ft

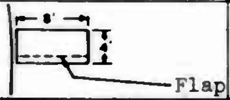
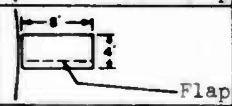
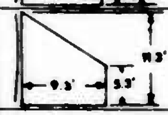
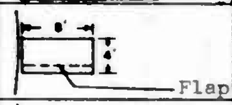
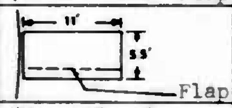
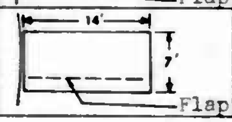
LWL = waterline length — ft

The effective fin lift curve slope will depend on the fin aspect ratio, planform and use of a trailing edge flap. The lift curve slope can be calculated for the purpose of concept design using the methods given in Chapter 8 of Reference 6. In the estimation of the lift curve slope it is recommended that the geometric aspect ratio be used with no account taken for the "image" effect in the hull. This will approximately account for the effects of the hull boundary layer, hull curvature near the fin and provide for interference from bilge keels. It is also recommended that if possible the fin span be limited so that it does not extend below the baseline or outboard of a line 5° to a line parallel to the centerline and tangent to the maximum beam at the station of the fin. These limits are shown in Figure 70.

There is not a sufficiently large body of data on existing systems to formulate parametric relationships between the fin area and weight, volume and control power. As a result, some available data from existing fin installations have been collected and are tabulated in Table 6. These data may be used during the concept design phase to estimate the required information to determine the impact on the ship design. Additional information and operating experience on U. S. Naval vessels can be obtained from Code 6165 of NAVSEC. The added resistance due to the fin may be estimated using the principles given in Chapter 7 of Reference 6.

TABLE 6

Ship Design Impact Data for Active Fin Systems

Installation	Fin Planform	Fin Area 1 Side	Total System Weight	Weight: Group 518	Weight: Group 113	Internal Volume 1 Side	Control Power	Maximum Angle/ Flap Angle	Time Stop to Stop	Maximum Design Lift	Comments
DE 1040 Class Sperry		32 ft^2	(Manual) 13.5 L.T.	-	-	134 ft^3	20 HP	$\pm 25^\circ$	~2 sec	23 L.T.	
DE 1040 Class Denny-Brown		32 ft^2	16.5 L.T.	~17.7 L.T.	4.4 L.T.	277 ft^3	20 HP	$\pm 23^\circ/33^\circ$	1.3 sec	23 L.T.	
DE 1052 Class Sperry		74 ft^2	20.5 L.T.	-	-	226 ft^3	50 HP	$\pm 30^\circ$	2 sec	38.4 L.T.	
DE 1052 Class Denny-Brown		77 ft^2	27.7 L.T.	25.4 L.T.	6.0 L.T.	308 ft^3	75 HP	$\pm 30^\circ$	1.6 sec	38.4 L.T.	
Sperry Gyro Fin		32 ft^2	34 L.T.				25 HP			16.5 L.T.	Retractable Lost Buoyancy = 14 L.T. Total
Sperry Gyro Fin		60.5 ft^2	54 L.T.				40 HP			31 L.T.	Retractable Lost Buoyancy = 26 L.T Total
Sperry Gyro Fin		98 ft^2	96 L.T.				75 HP			50 L.T.	Retractable Lost Buoyancy = 34 L.T. Total

Active Tank Roll Stabilization

As shown in Figure 4, an active tank system is basically the same configuration as a U-tube tank. Due to the complexity of fully-active systems, information about the pump and control system cannot be generalized. However, it is possible to define the required differences in tank geometry. The major differences between an active tank and those designed for pure passive stabilization is that the natural frequency of the tank with the air valve open is 30 to 40% greater than the ship's natural frequency and it is required that the tank have an equivalent linear damping ratio somewhat lower than a good passive tank. As a result, care must be taken in order to avoid any superfluous structure within the tank itself.

The design of the tank configuration, itself, follows along exactly as a U-tube. The only difference is that Equation [19] is to be replaced by

$$S' = .178\ B^2/\overline{GM}_{\text{uncorrected}}, \qquad [25]$$

in order to obtain the desired tuning.

Example: If we use the same ship as in the previous tank design examples, we have, from [25]

$$S' = .178 \times 80^2/6 = 190'$$

From [17], we get

$$190 = 8 + 80\text{-}40 + 40\,[11.25 \times 40/(2 \times 11.25 \times p)]$$

or

$$p = 5.64'$$

In summary, the final configuration is

$$\left.\begin{aligned} B &= 80' \\ H &= 16' \\ D &= 11.25' \\ h &= 8' \\ d &= 11.25' \\ b &= 40' \\ p &= 5.64' \end{aligned}\right\}$$

The volume of the tank fluid is then

$$\forall_f = 6138 \text{ ft}^3$$

and, if sea water is used

$$W_f = 170 \text{ tons.}$$

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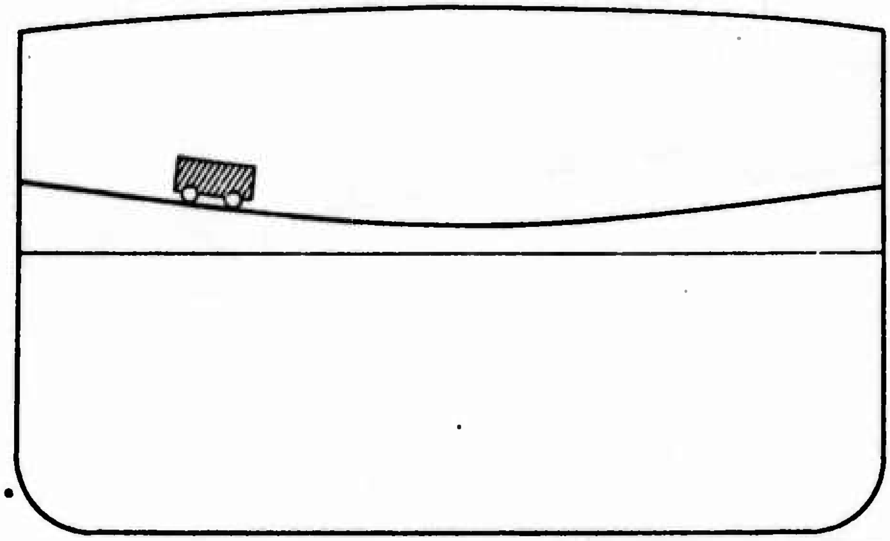
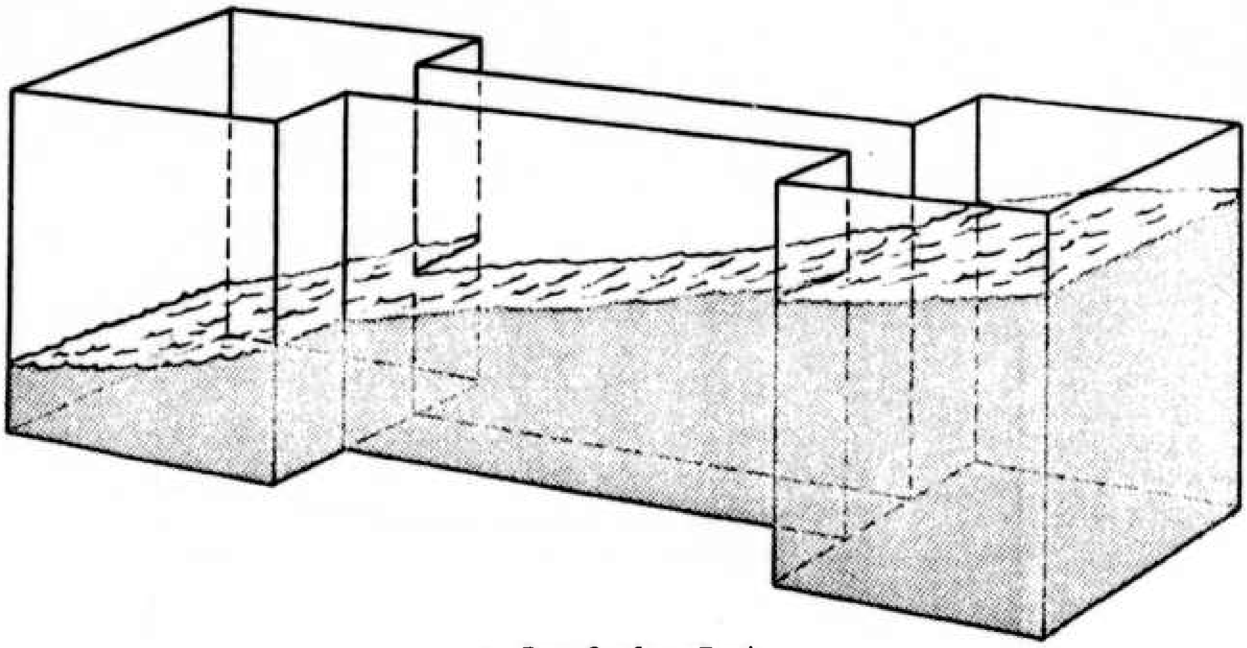
6. "Principles of Naval Architecture," Chapter VII, The Society of Naval Architects and Marine Engineers, 1967.

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

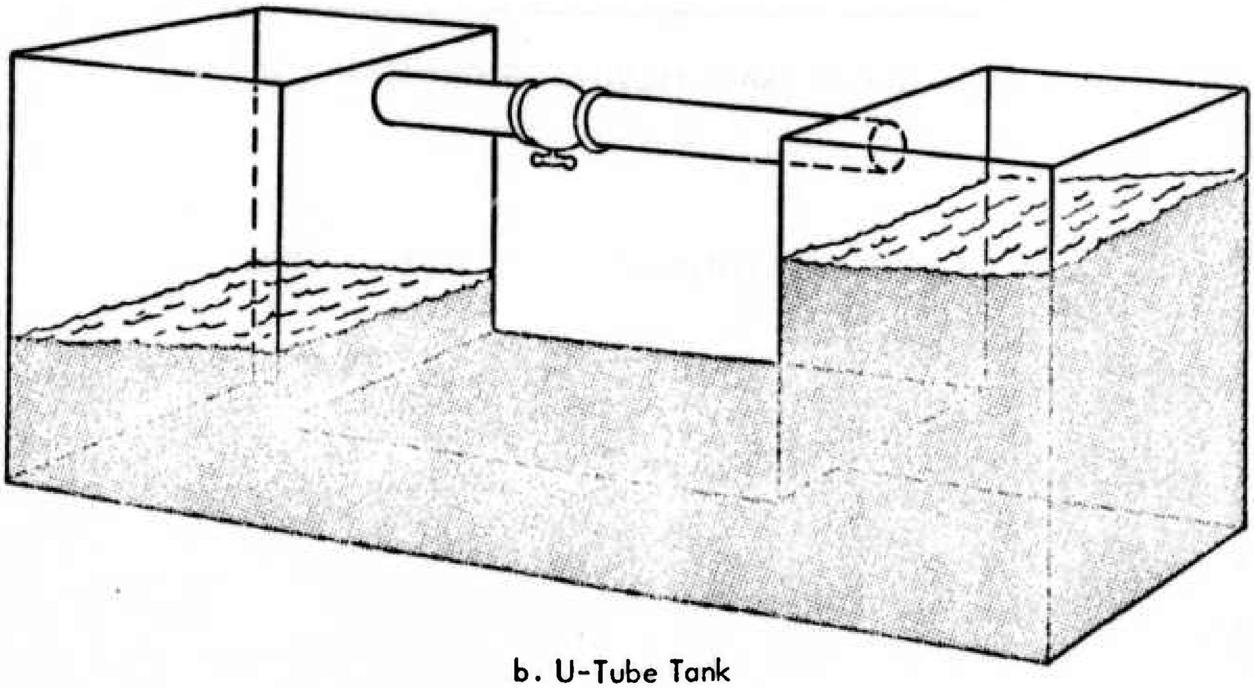


CURVED TRACK MOVING WEIGHT STABILIZER

FIGURE 1 - MOVING WEIGHT STABILIZER SYSTEM



a. Free Surface Tank



b. U-Tube Tank

FIGURE 2 - PASSIVE STABILIZER TANK TYPES

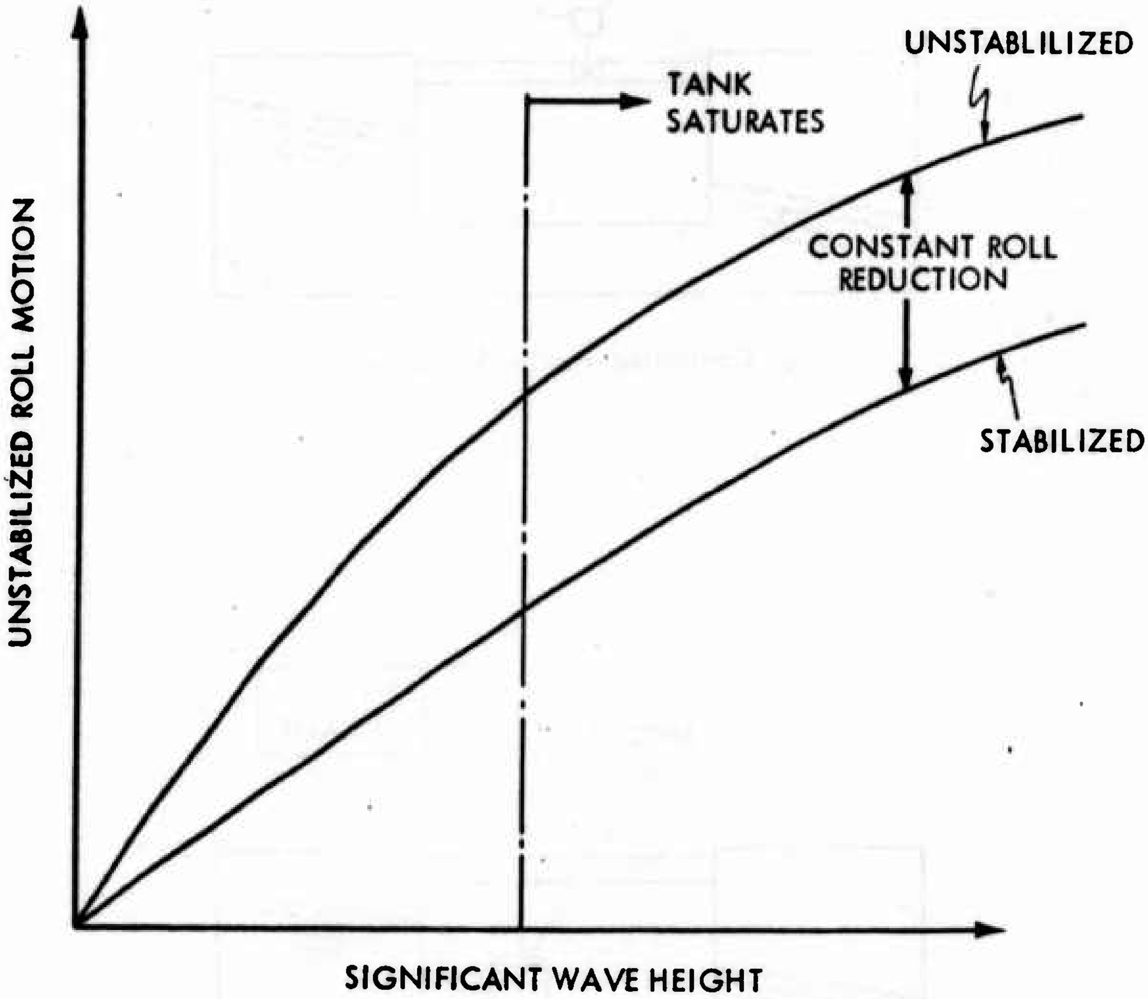
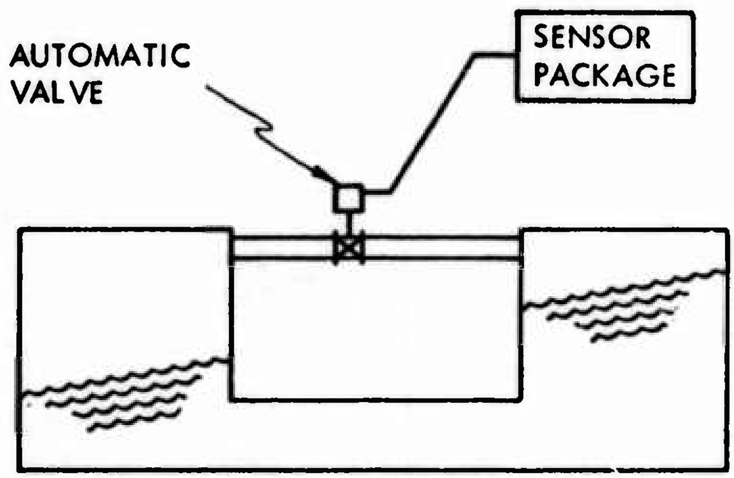
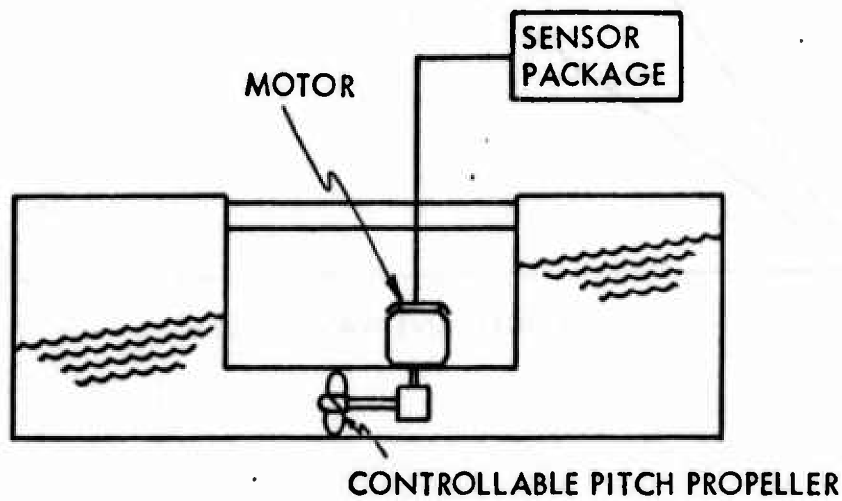


FIGURE 3 - THE EFFECT OF SATURATION



a. Controlled-Passive Stabilizer



b. Fully-Active Stabilizer

FIGURE 4 - ACTIVE STABILIZER TYPES

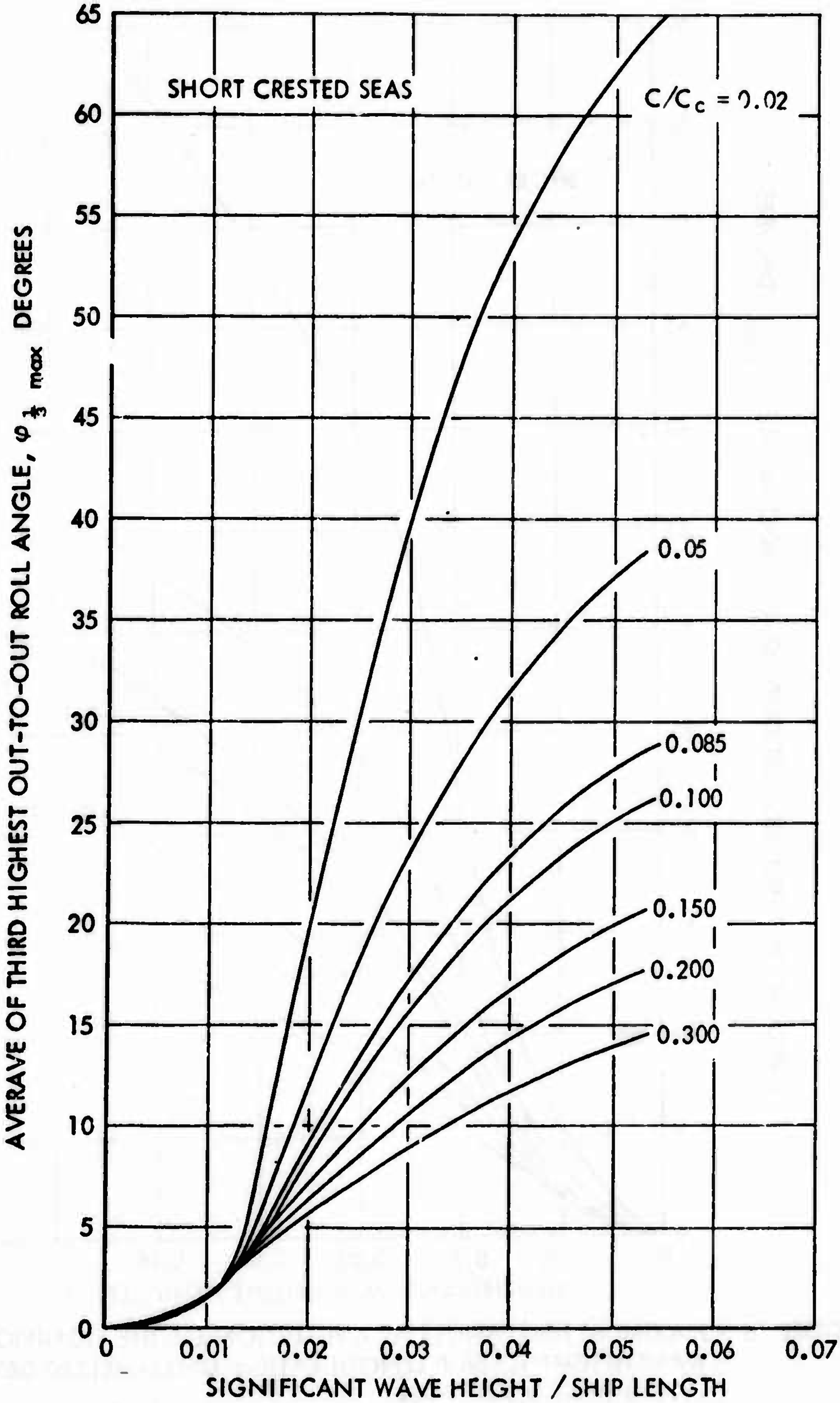


FIGURE 5 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO; UNSTABILIZED DESTROYER TYPE SHIP AT ZERO SPEED

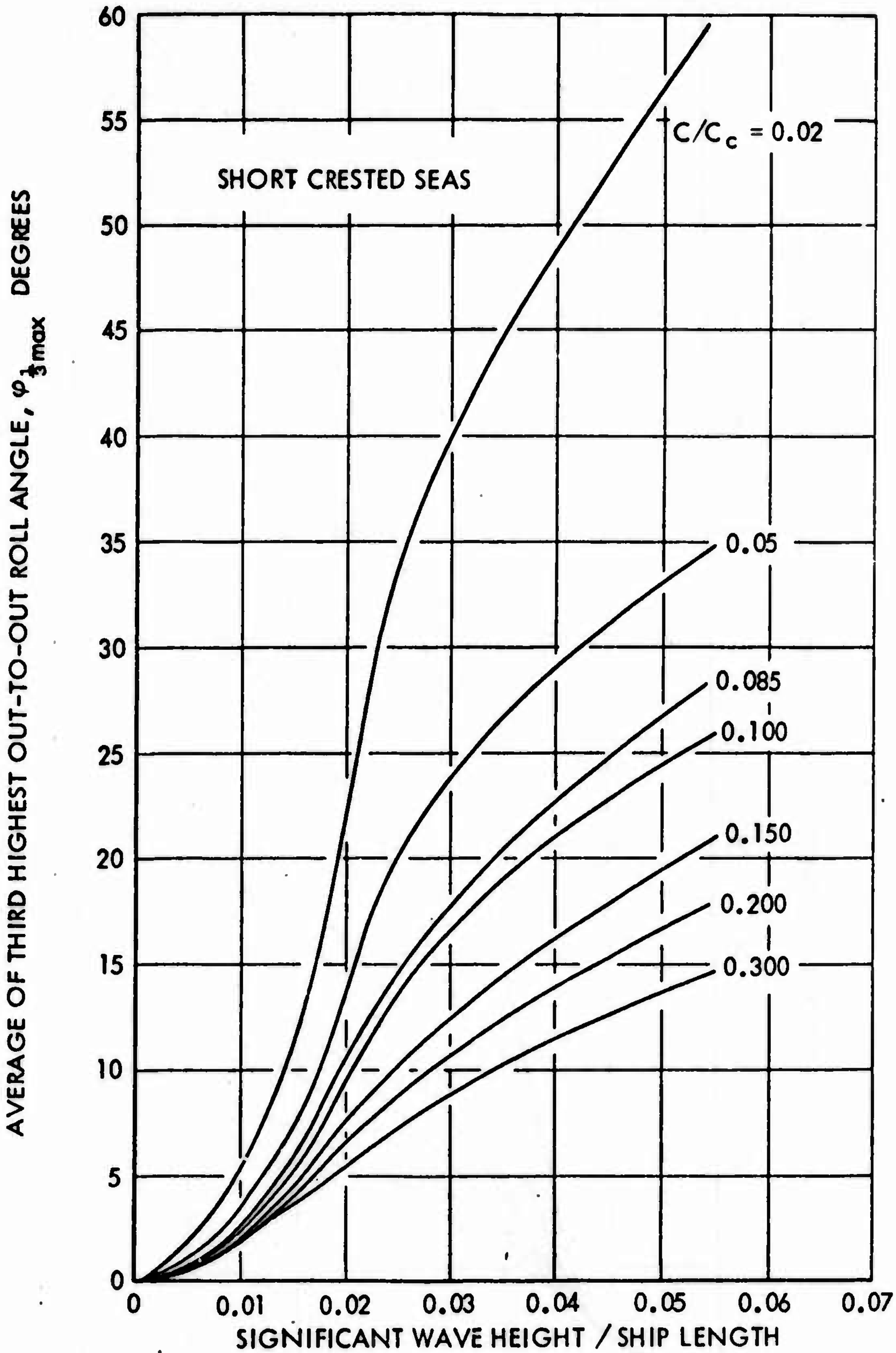


FIGURE 6 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO; UNSTABILIZED DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.4$

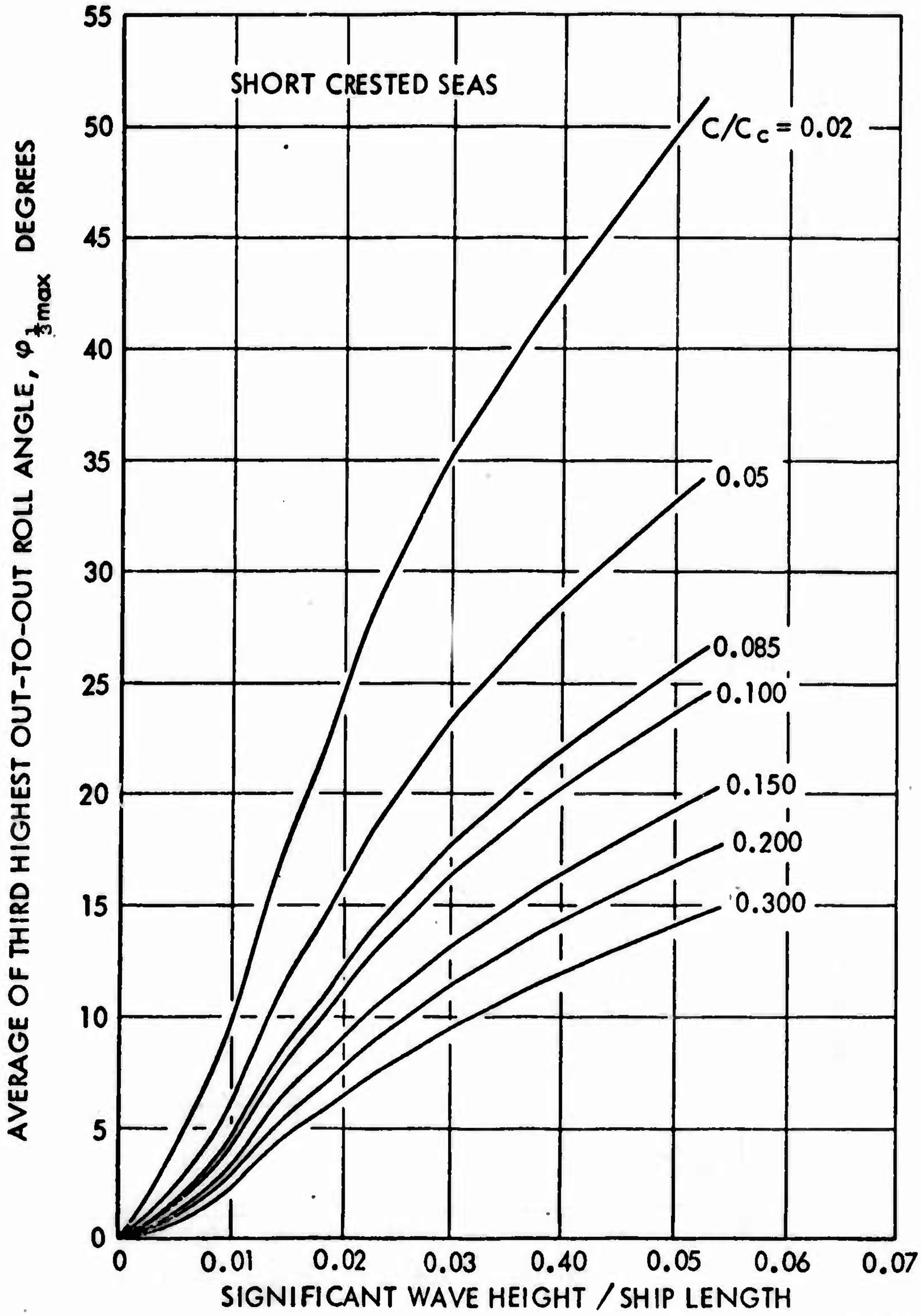


FIGURE 7 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO; UNSTABILIZED DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

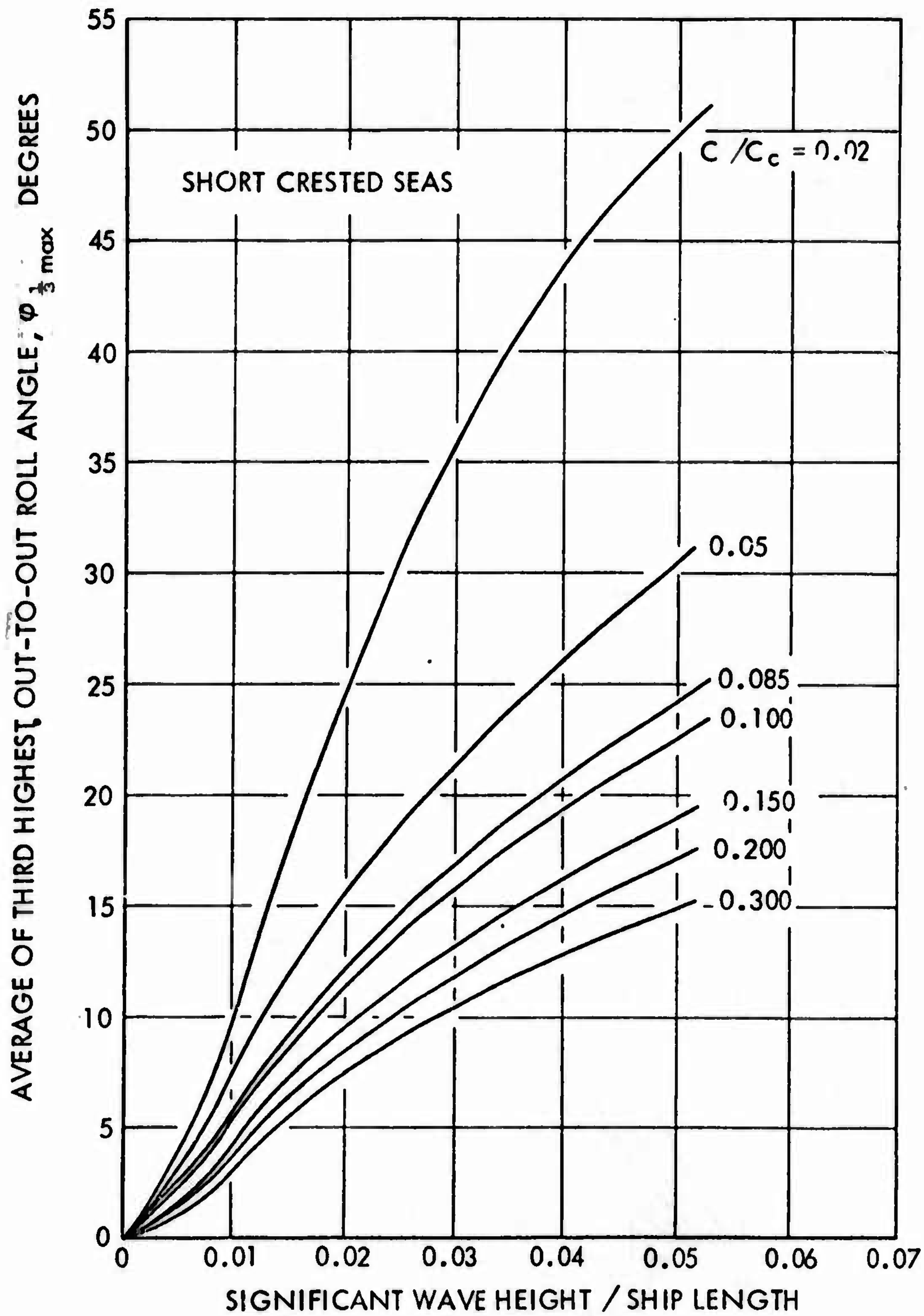


FIGURE 8 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO; UNSTABILIZED DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

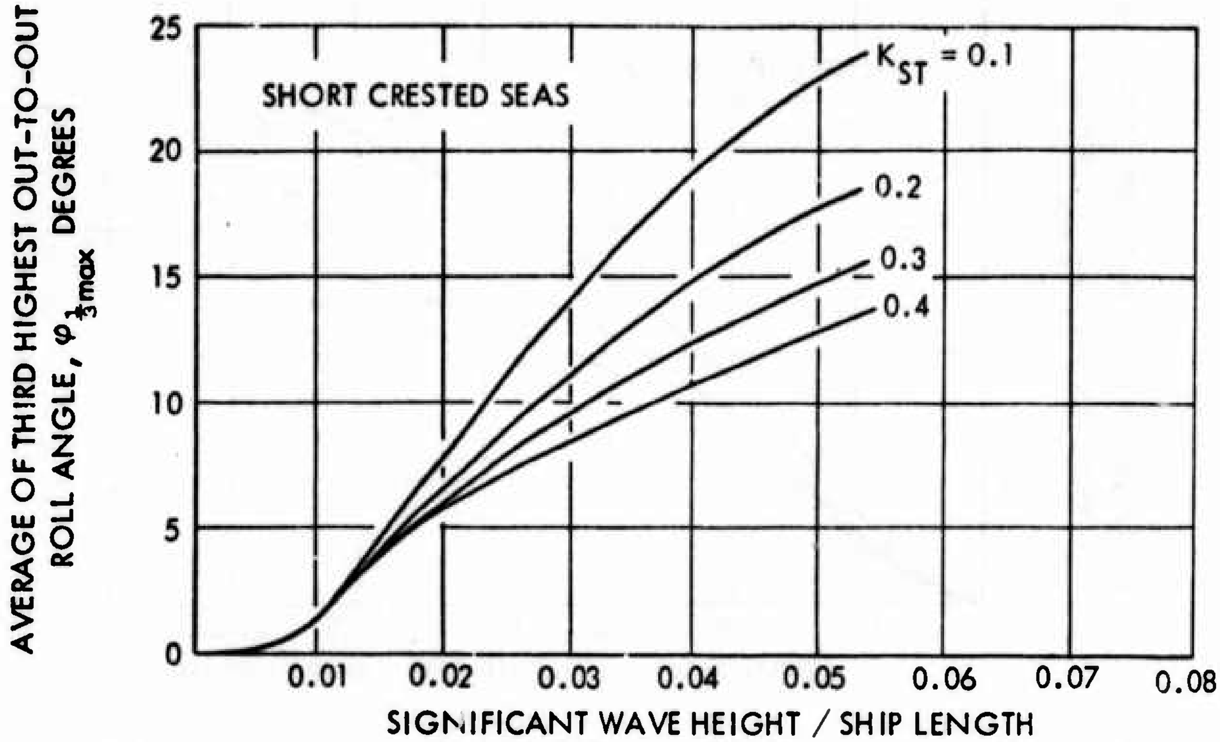


FIGURE 9 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT ZERO SPEED

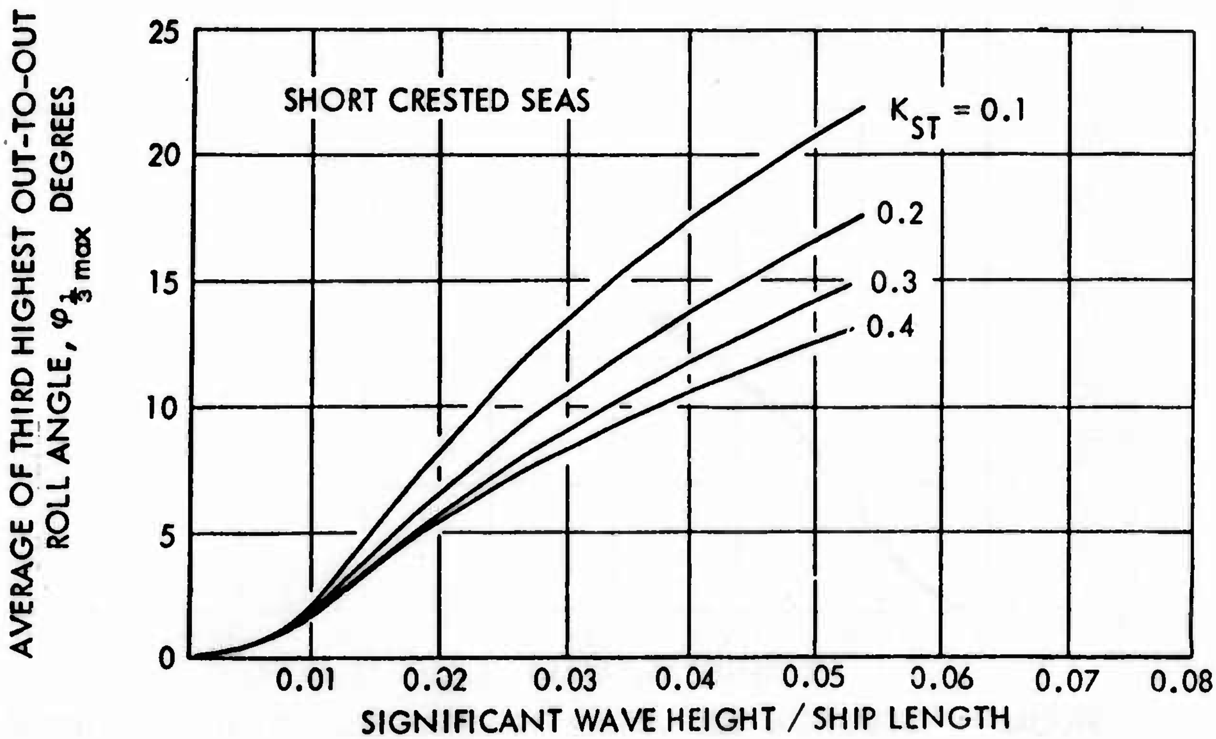


FIGURE 10 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L}$ = 0.4

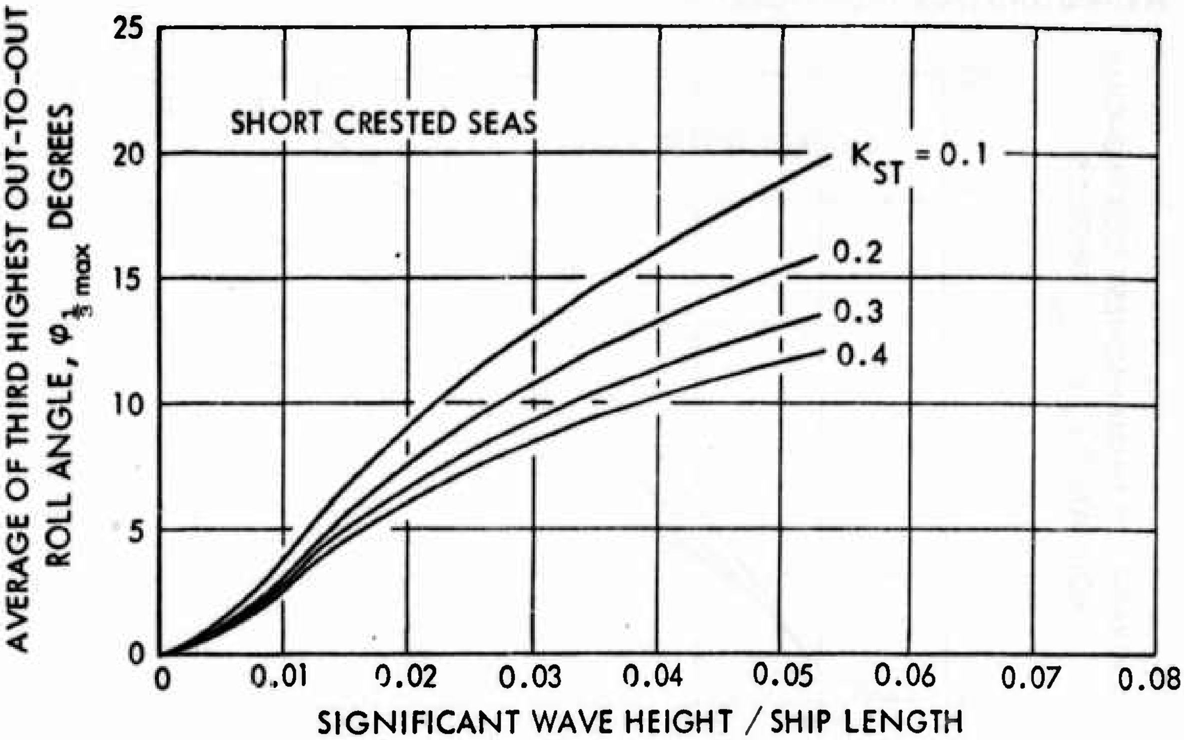


FIGURE 11 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

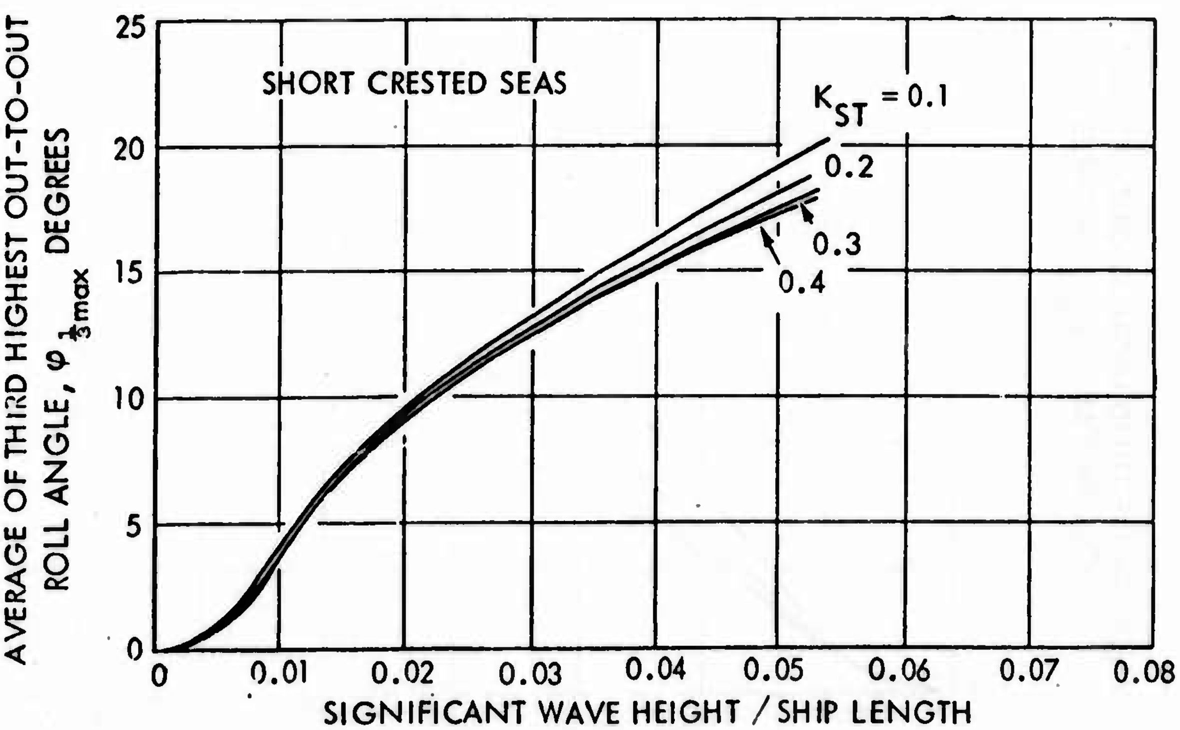


FIGURE 12 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

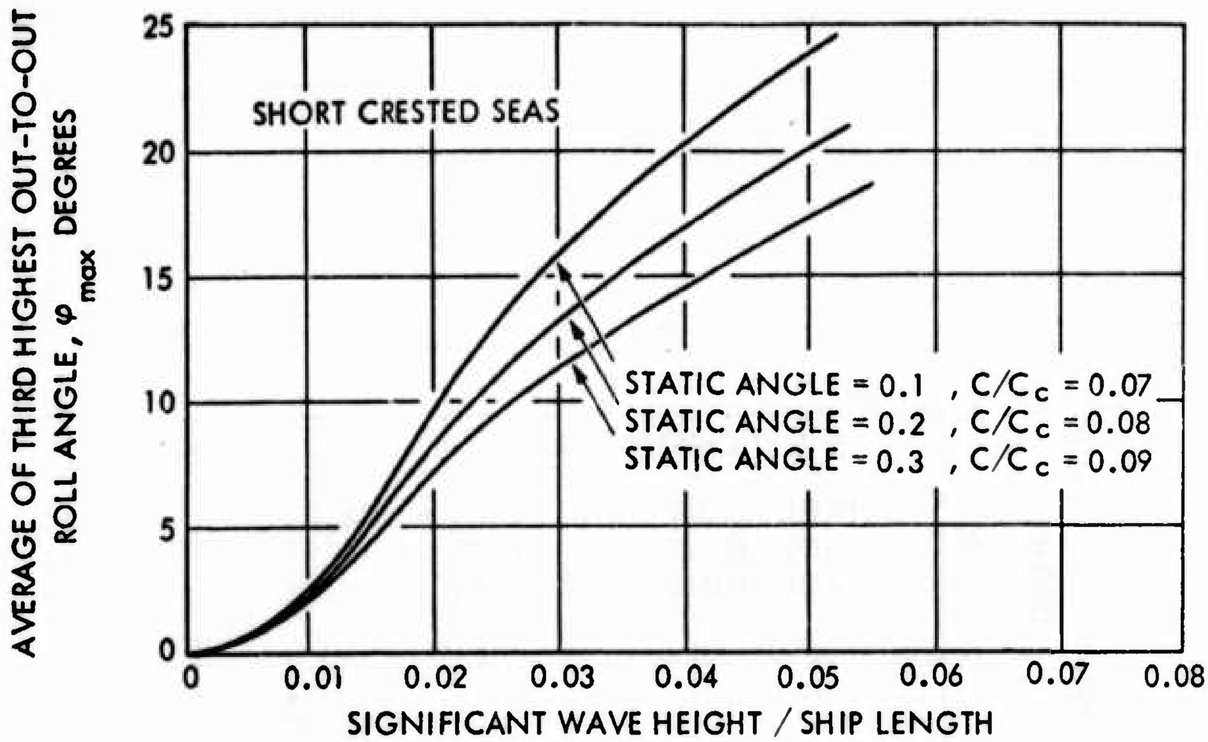


FIGURE 13 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE FIN CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.4$

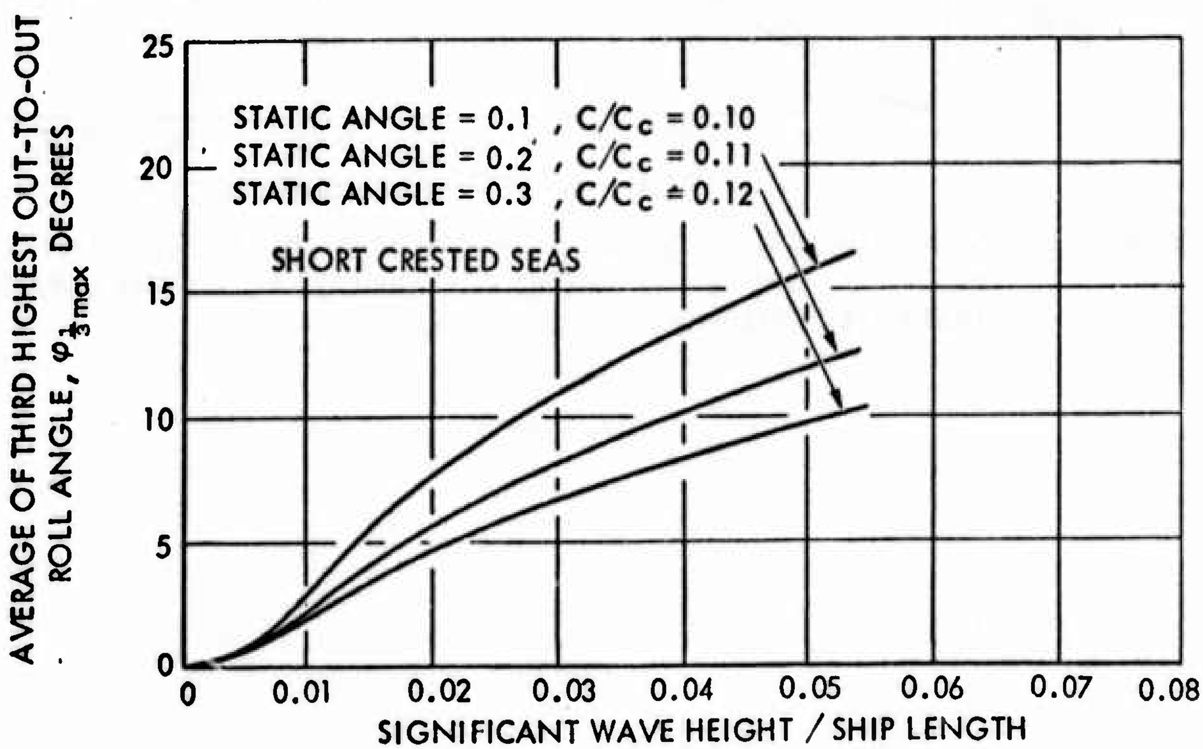


FIGURE 14 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE FIN CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

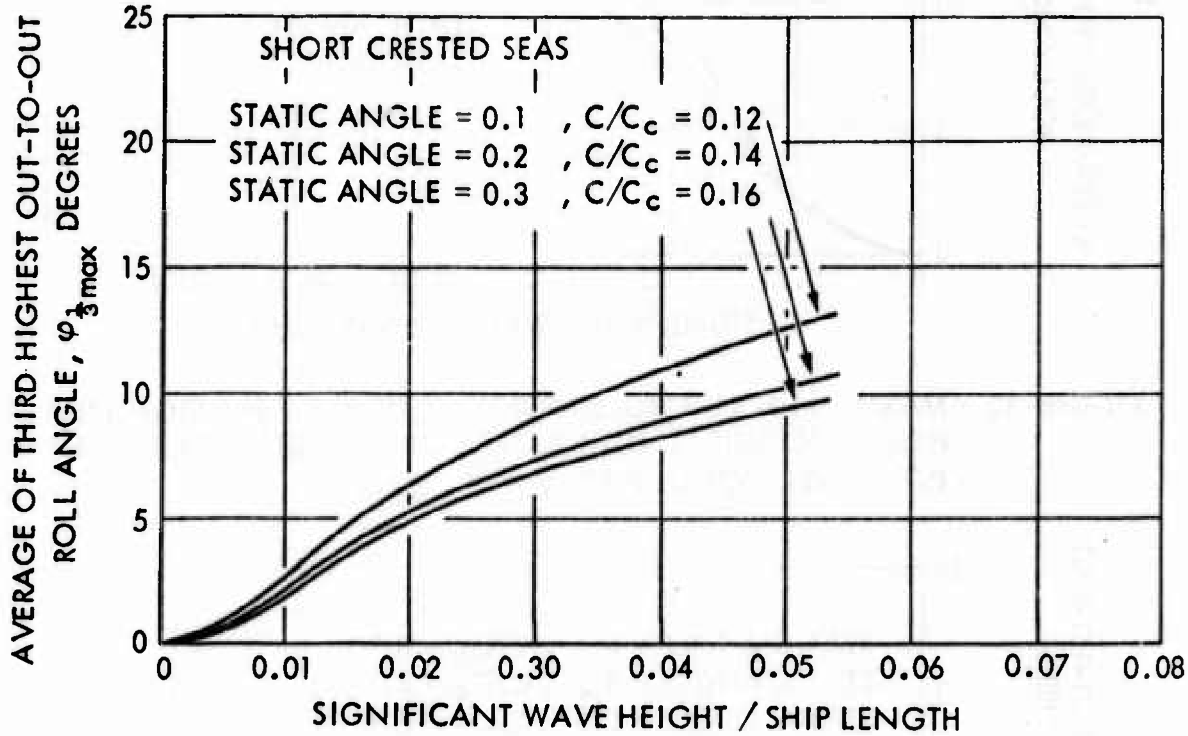


FIGURE 15 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE FIN CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

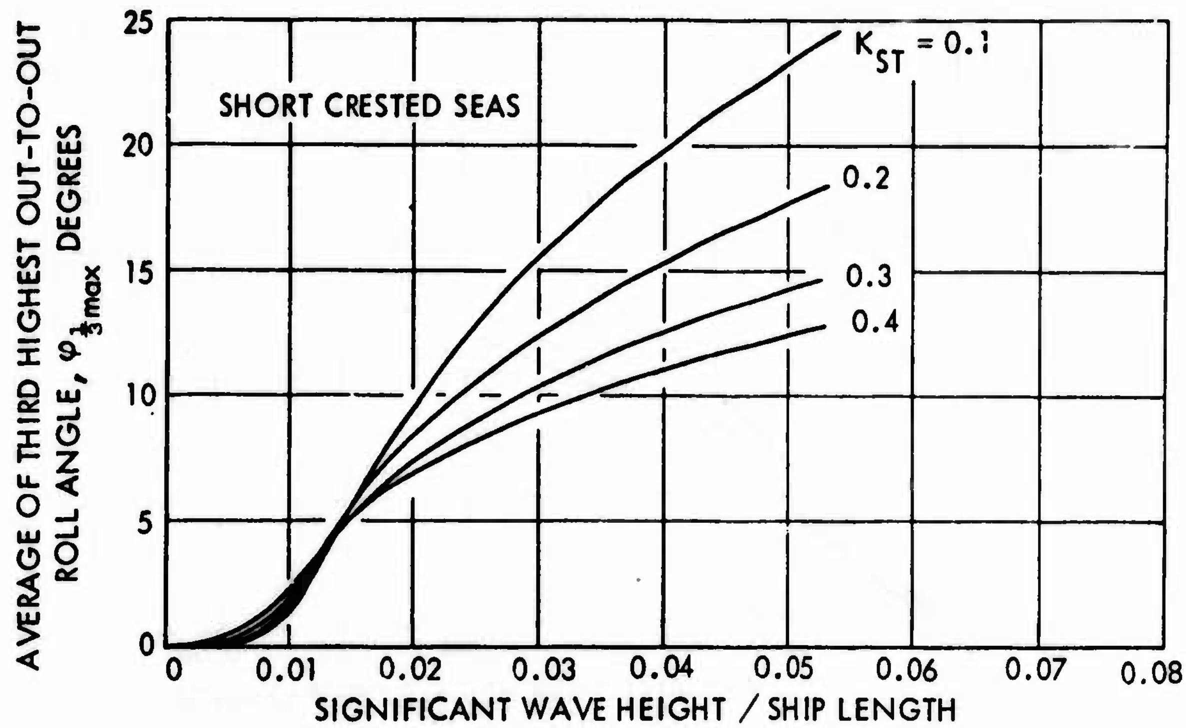


FIGURE 16 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT ZERO SPEED

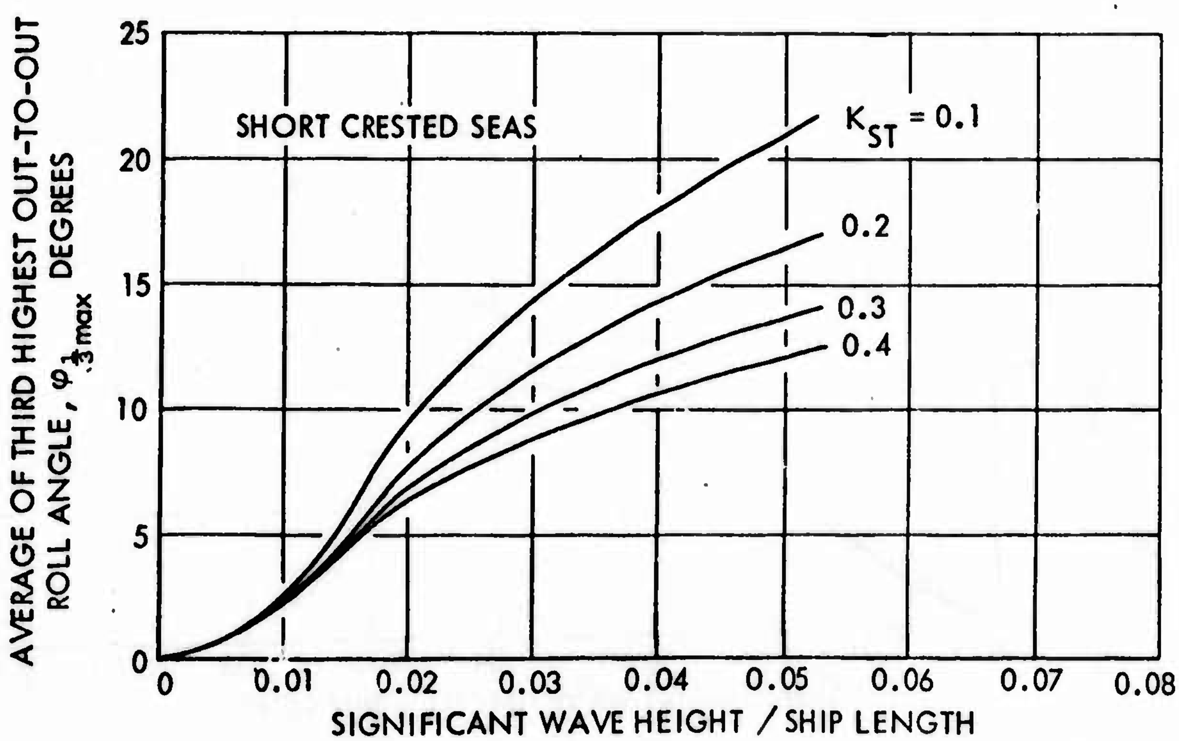


FIGURE 17 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.4$

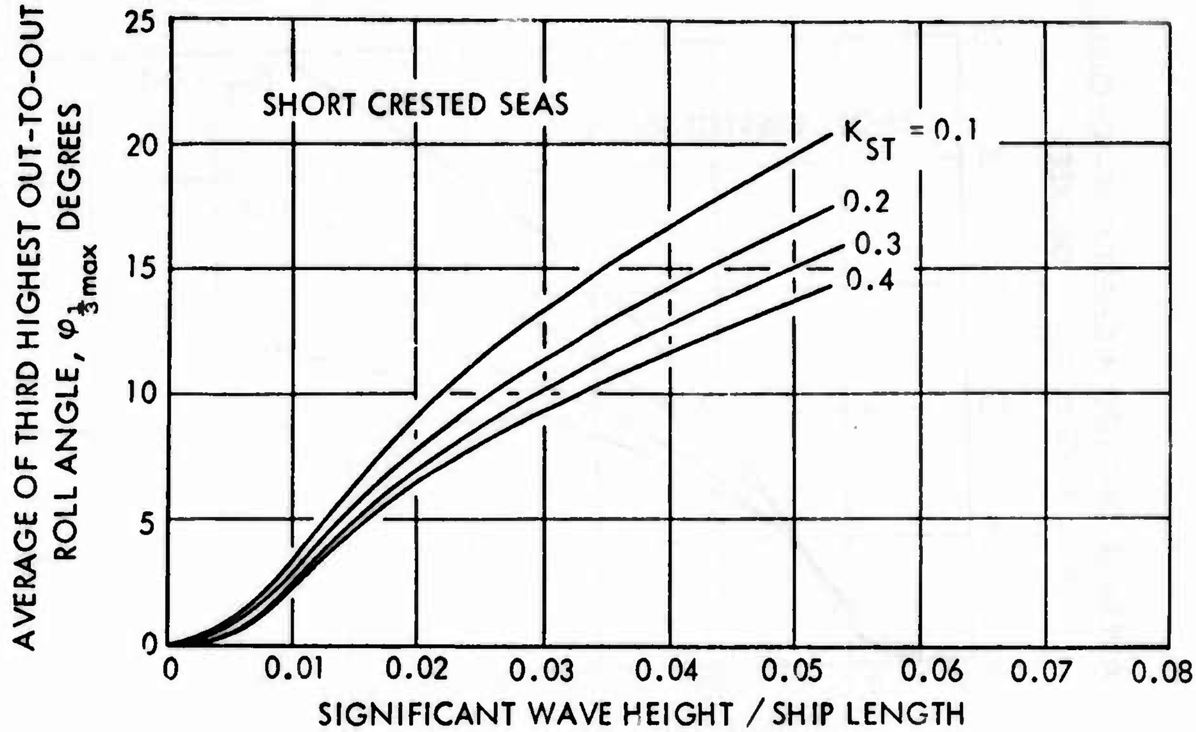


FIGURE 18 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

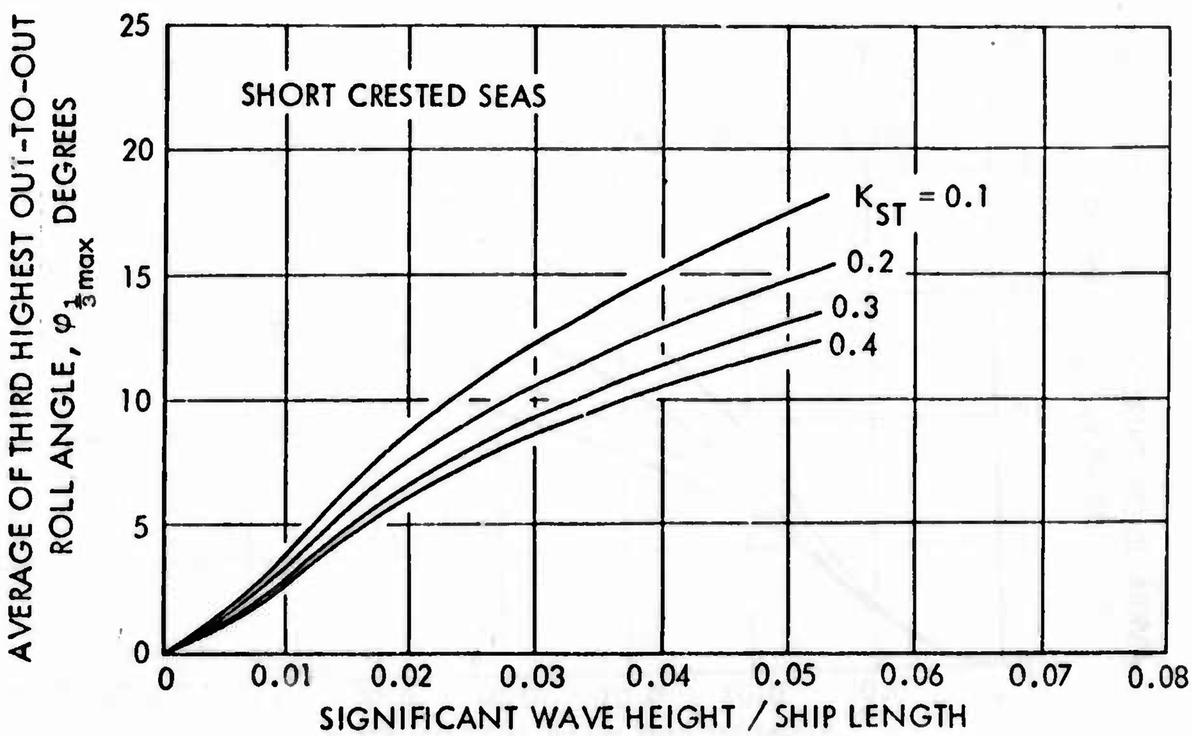


FIGURE 19 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

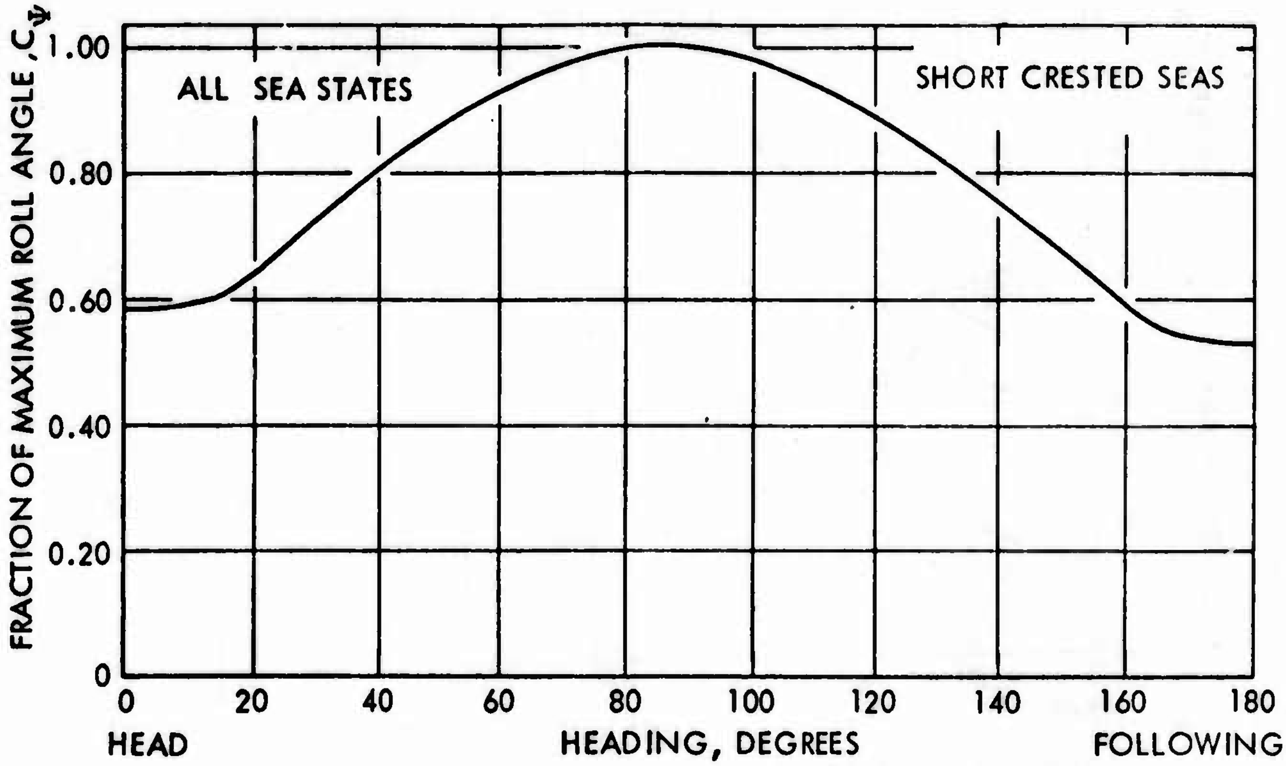


FIGURE 20 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR ALL SEA STATES; UNSTABILIZED DESTROYER TYPE SHIP AT ZERO SPEED

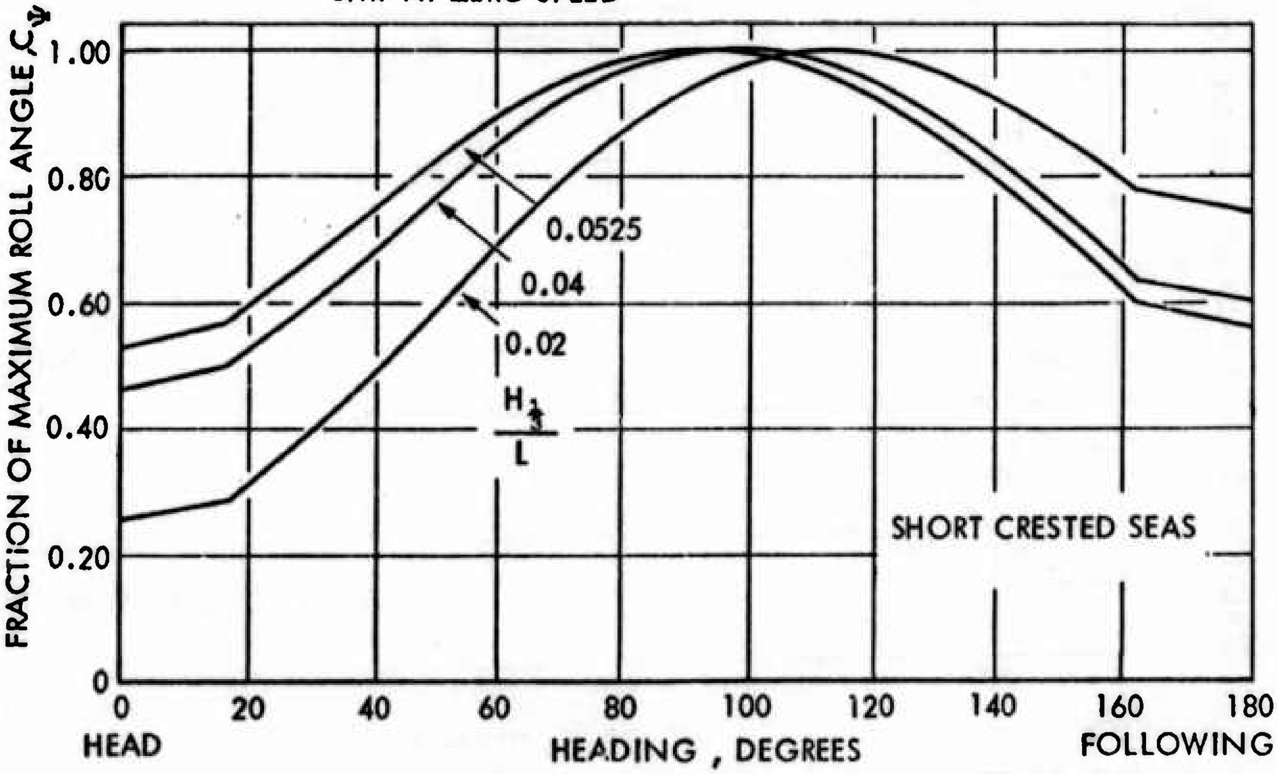


FIGURE 21 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS; UNSTABILIZED DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.4$

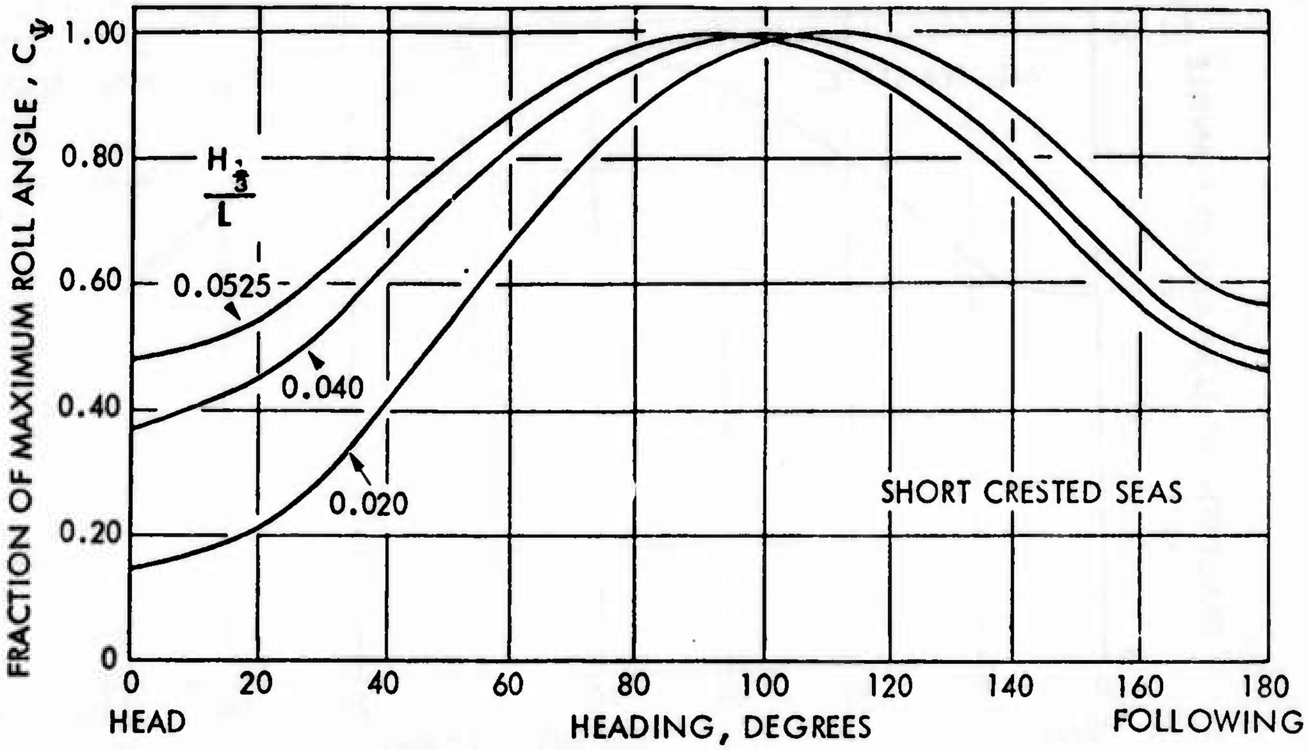


FIGURE 22 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS; UNSTABILIZED DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

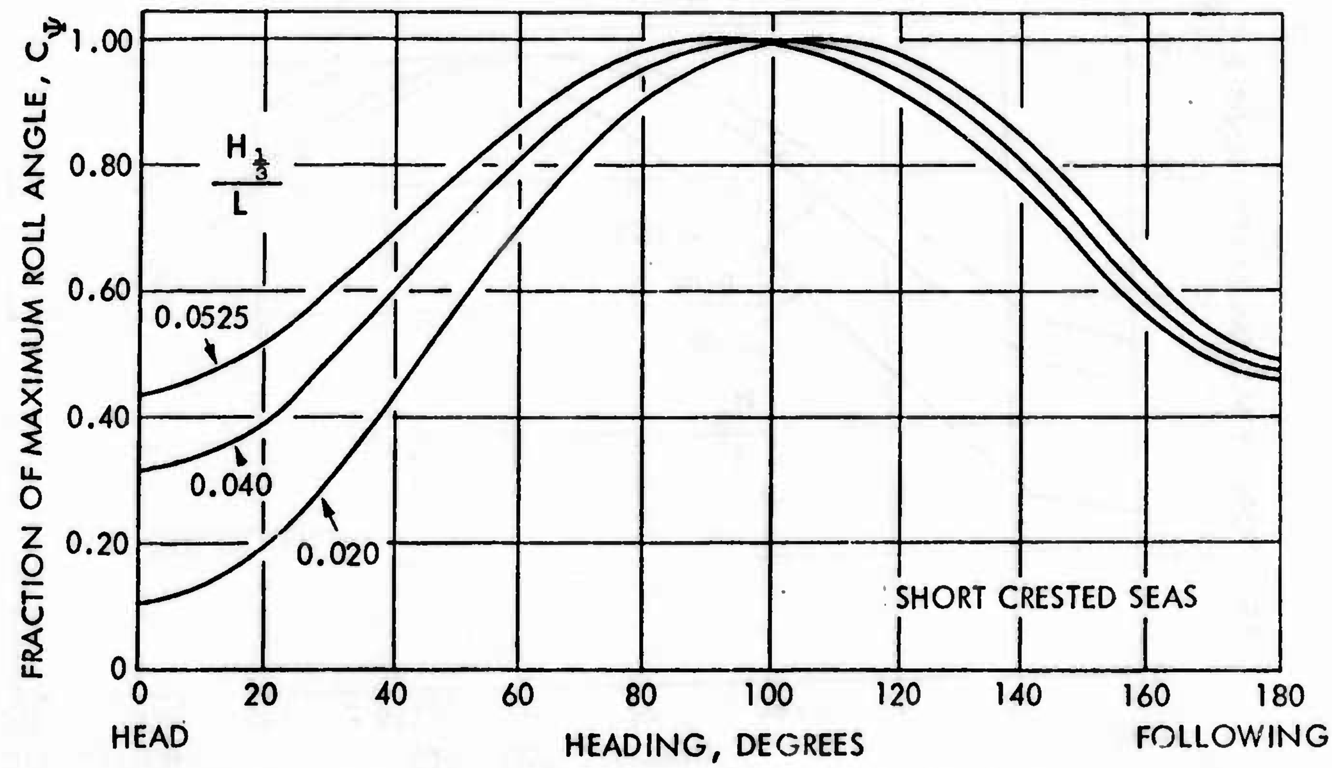


FIGURE 23 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS; UNSTABILIZED DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

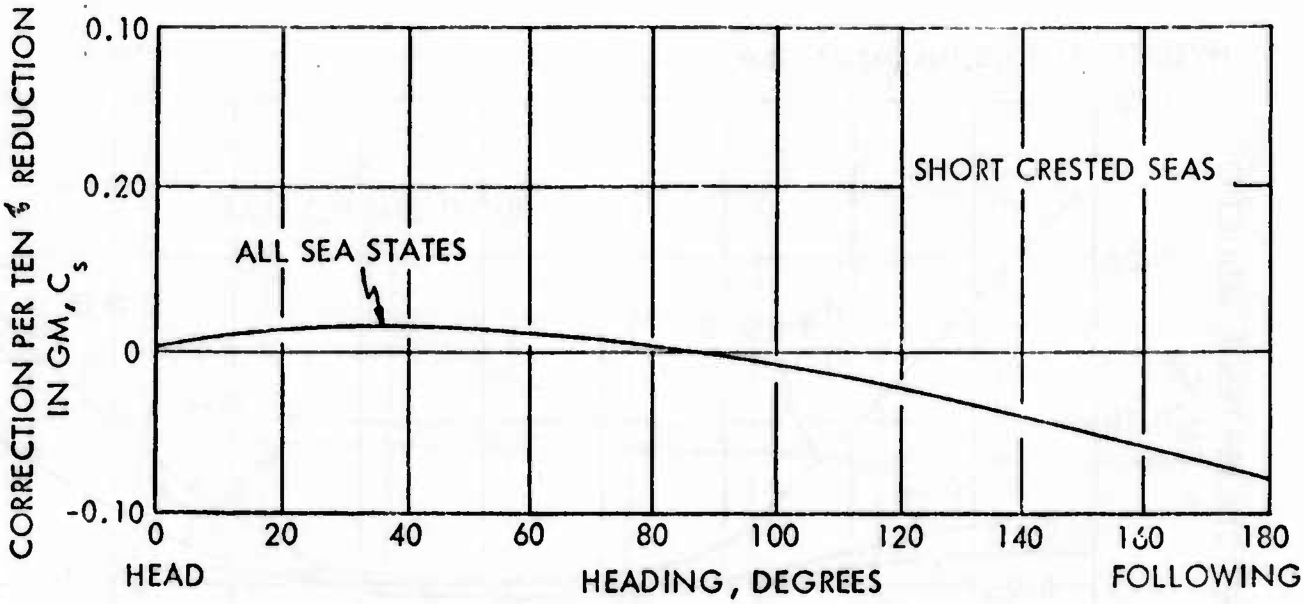


FIGURE 24 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE WITH PASSIVE STABILIZER TANKS INSTALLED; DESTROYER TYPE SHIP AT ZERO SPEED

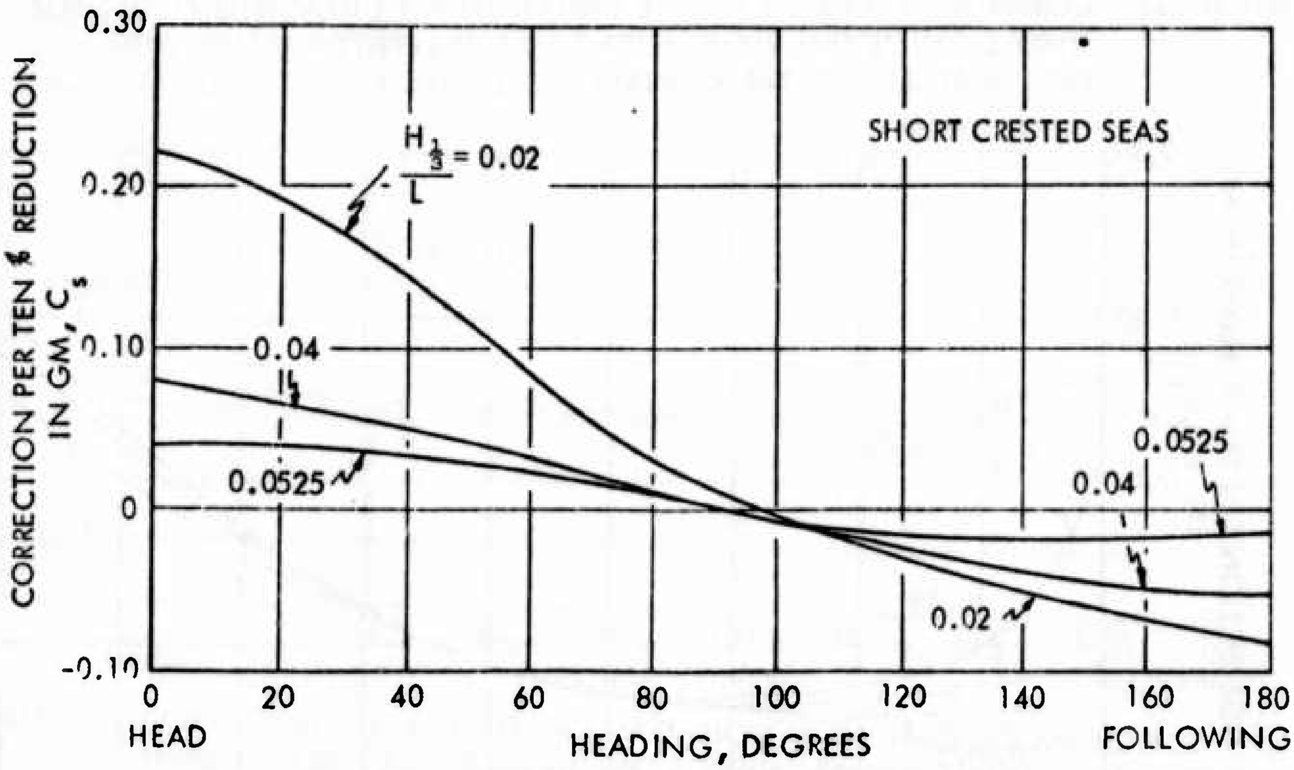


FIGURE 25 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE WITH PASSIVE STABILIZER TANKS INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.4$

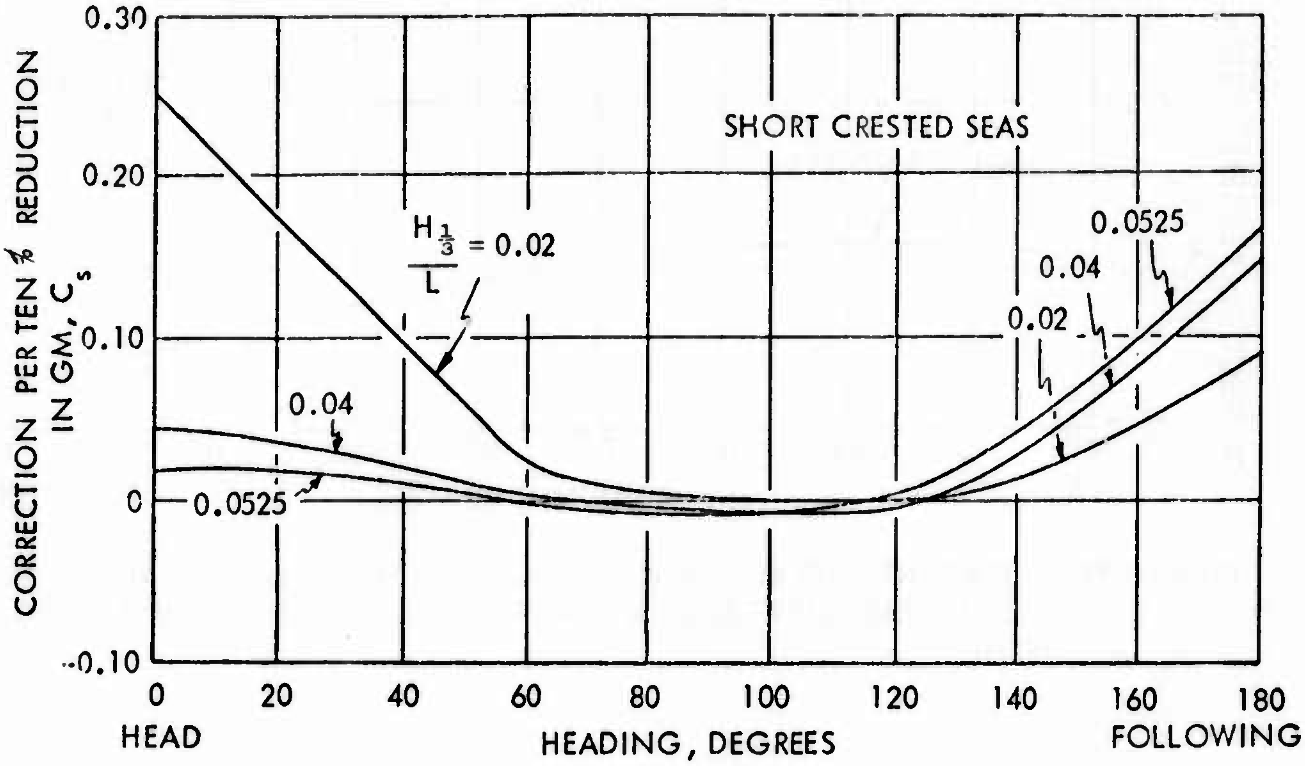


FIGURE 26 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH PASSIVE STABILIZER TANKS INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

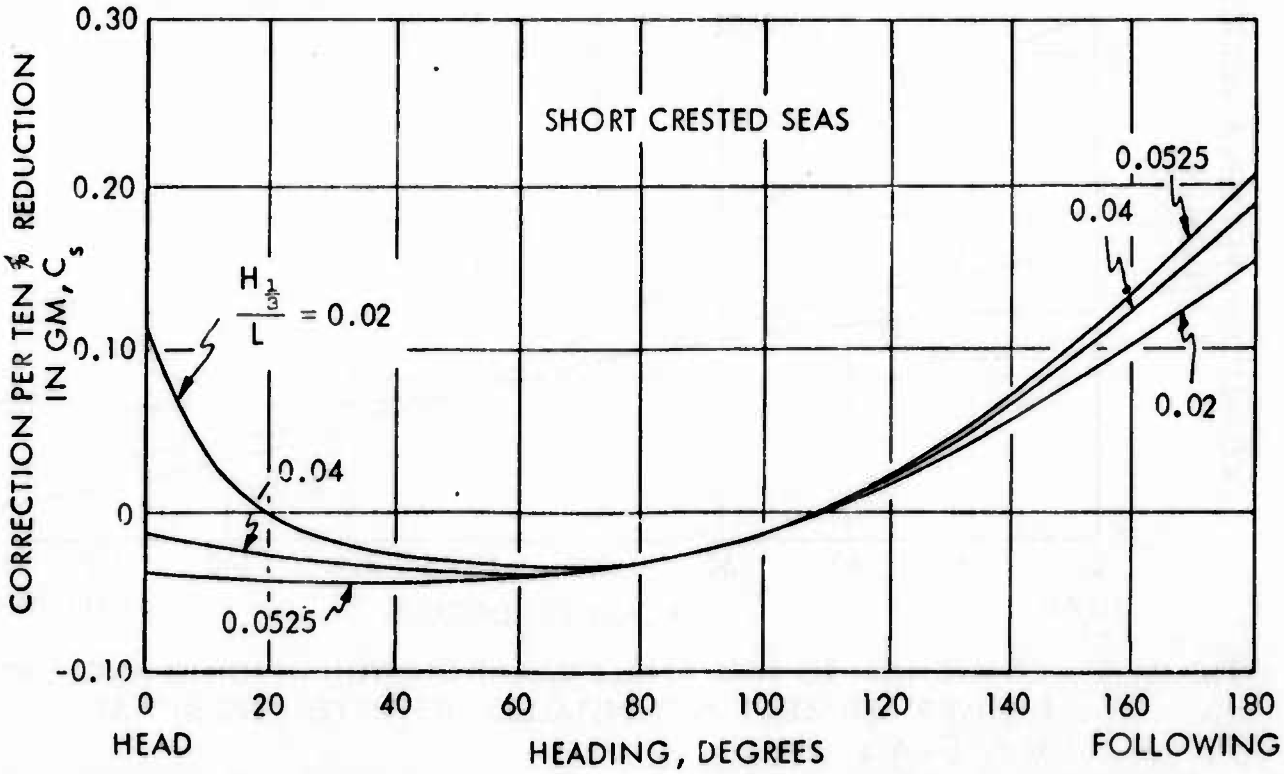


FIGURE 27 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH PASSIVE STABILIZER TANKS INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

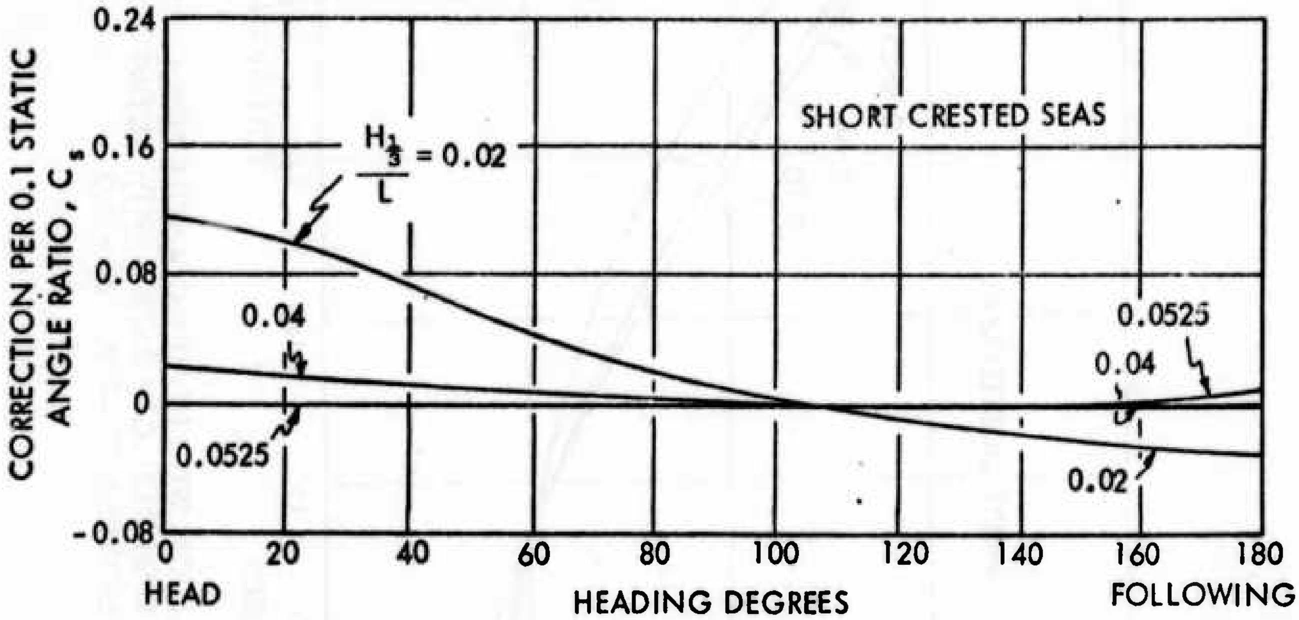


FIGURE 28 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE FIN STABILIZER INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.4$

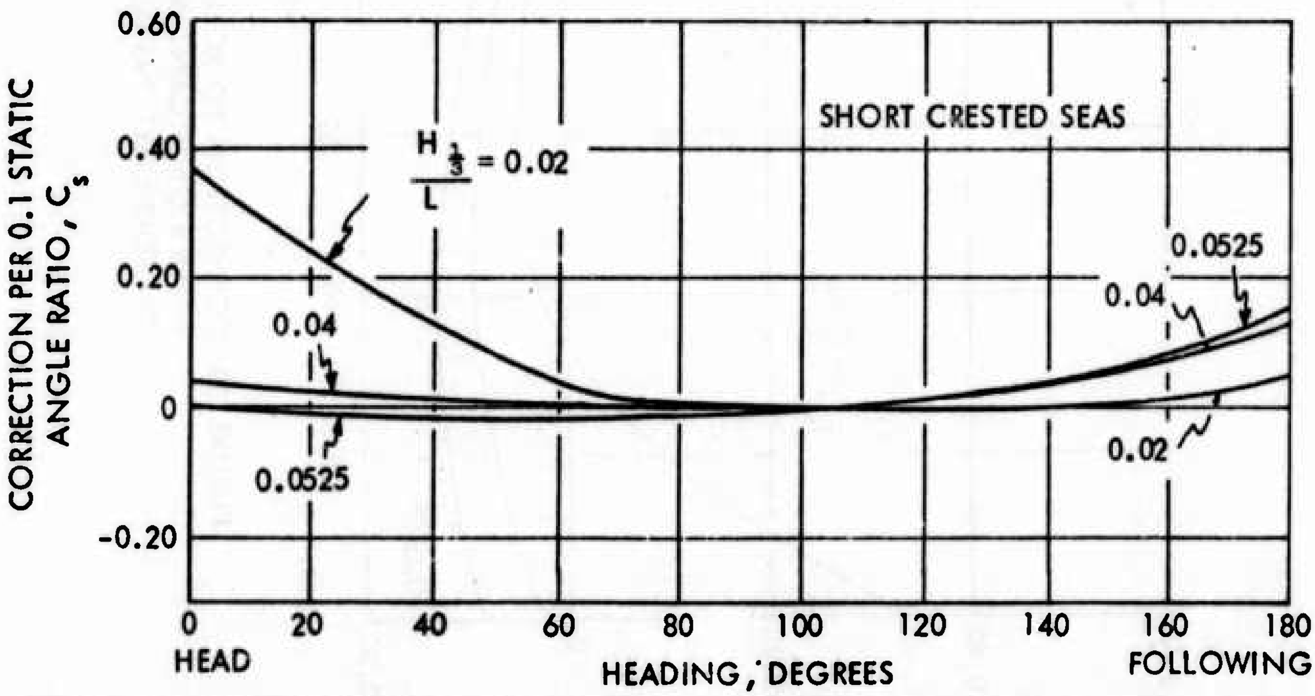


FIGURE 29 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE FIN STABILIZER INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

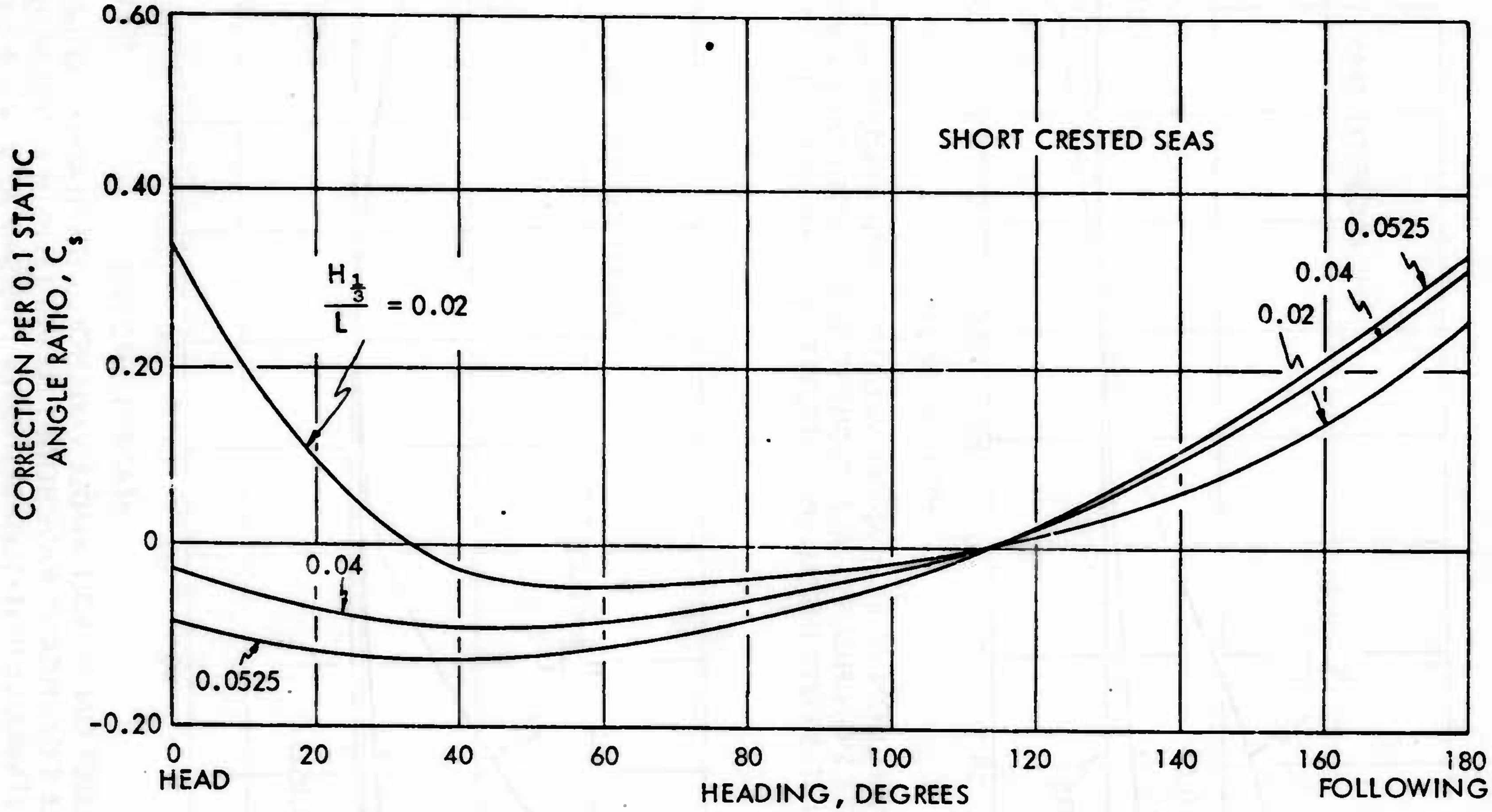


FIGURE 30- CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE FIN STABILIZER INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

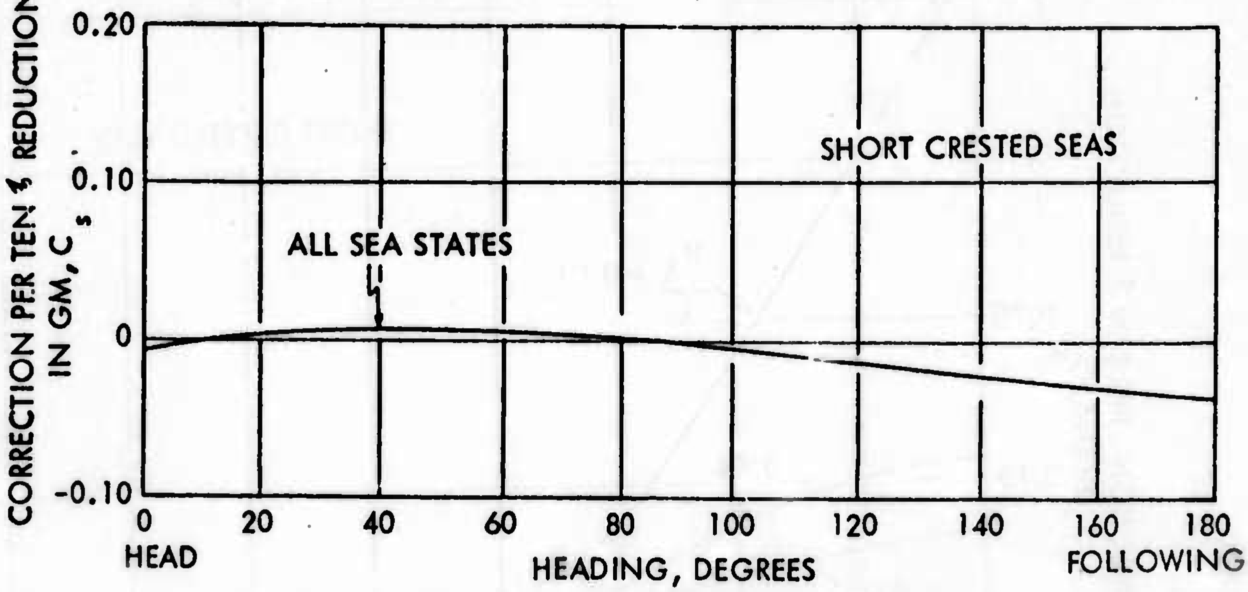


FIGURE 31 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE WITH ACTIVE STABILIZER TANKS INSTALLED; DESTROYER TYPE SHIP AT ZERO SPEED

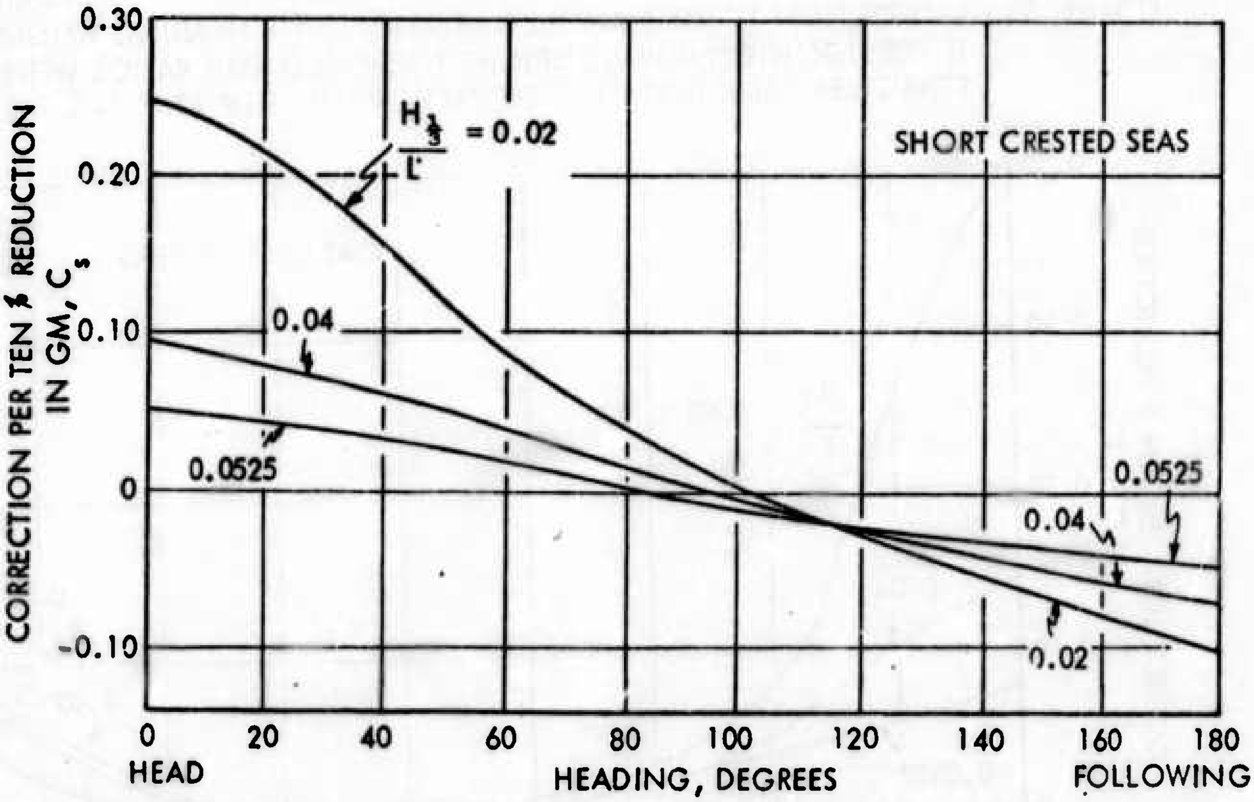


FIGURE 32 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE WITH ACTIVE STABILIZER TANKS INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.4$

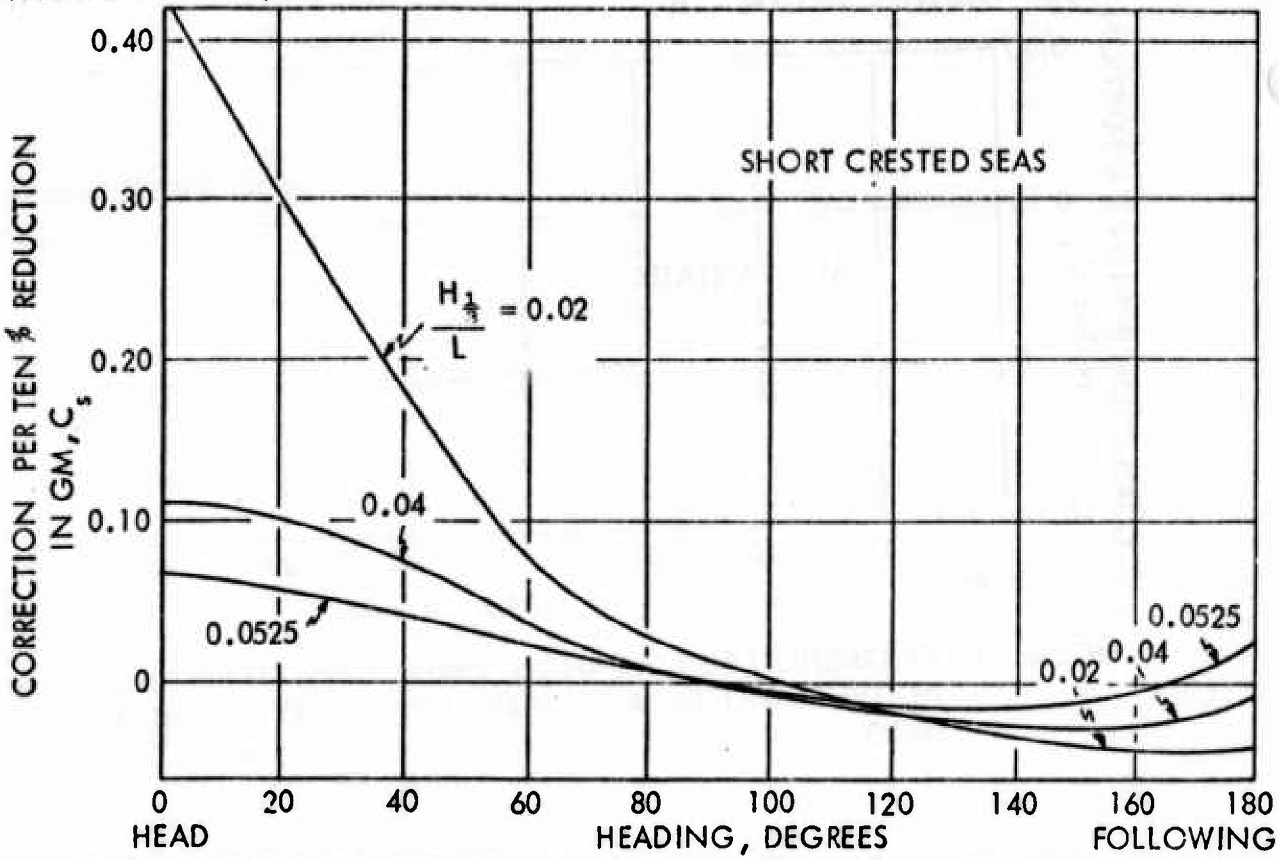


FIGURE 33 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE STABILIZER TANK INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 0.8$

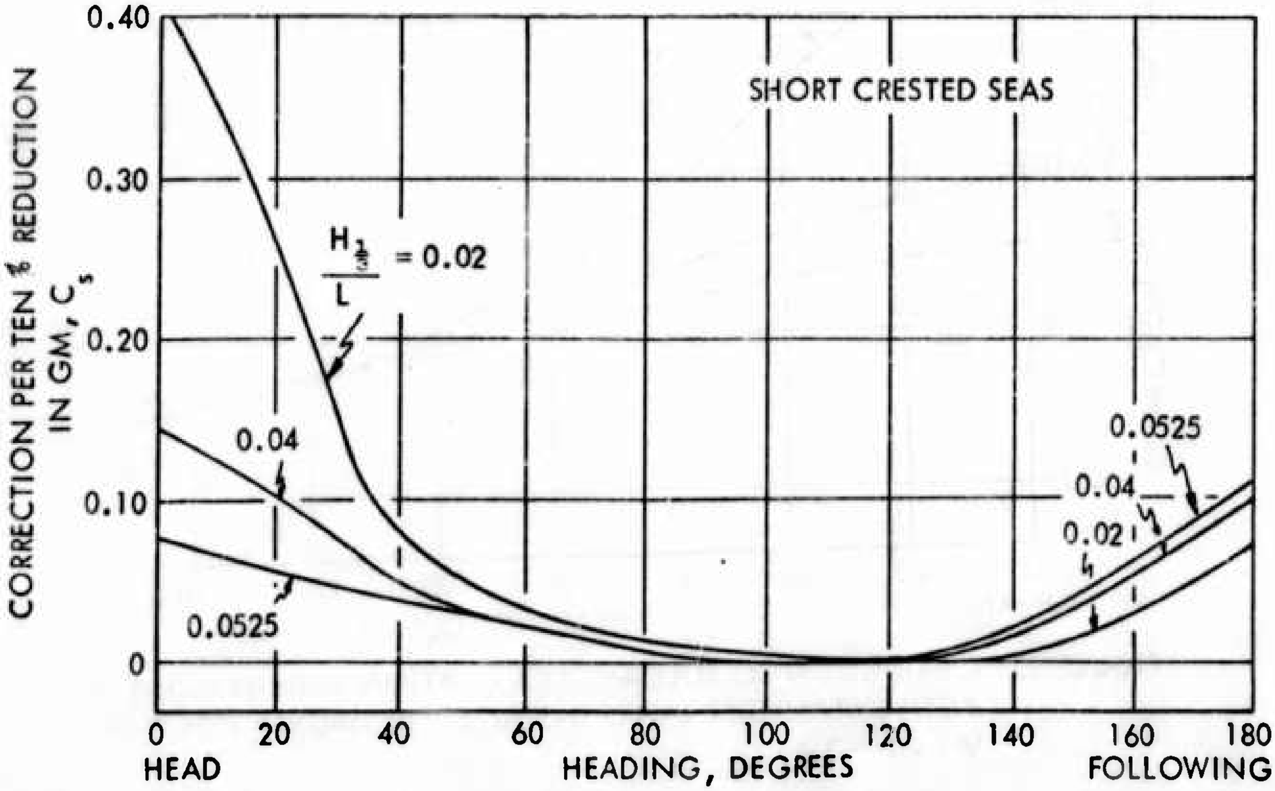


FIGURE 34 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE STABILIZER TANK INSTALLED; DESTROYER TYPE SHIP AT $V/\sqrt{L} = 1.2$

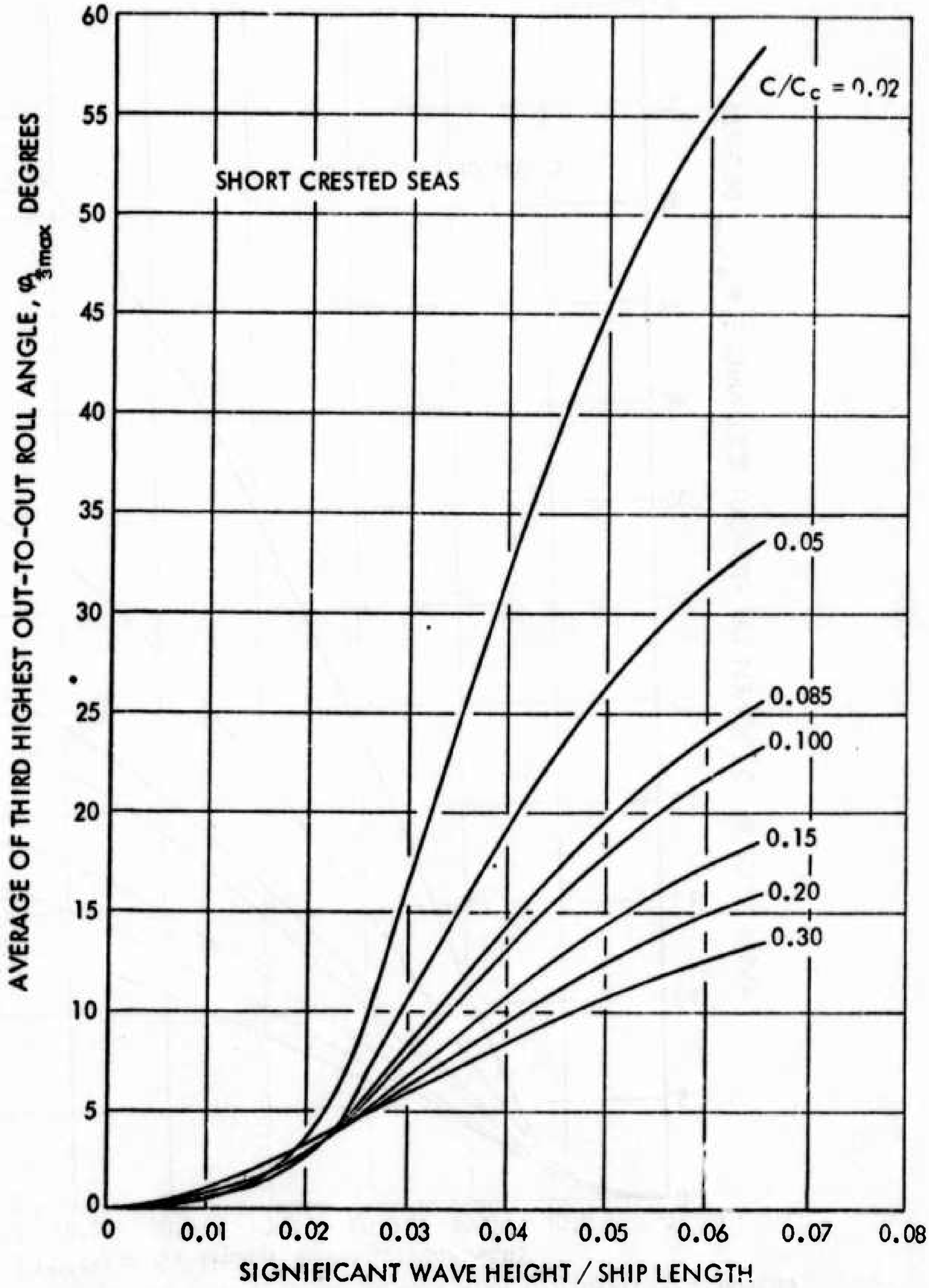


FIGURE 35 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS SHIP DAMPING RATIOS; UNSTABILIZED AUXILIARY TYPE SHIP AT ZERO SPEED

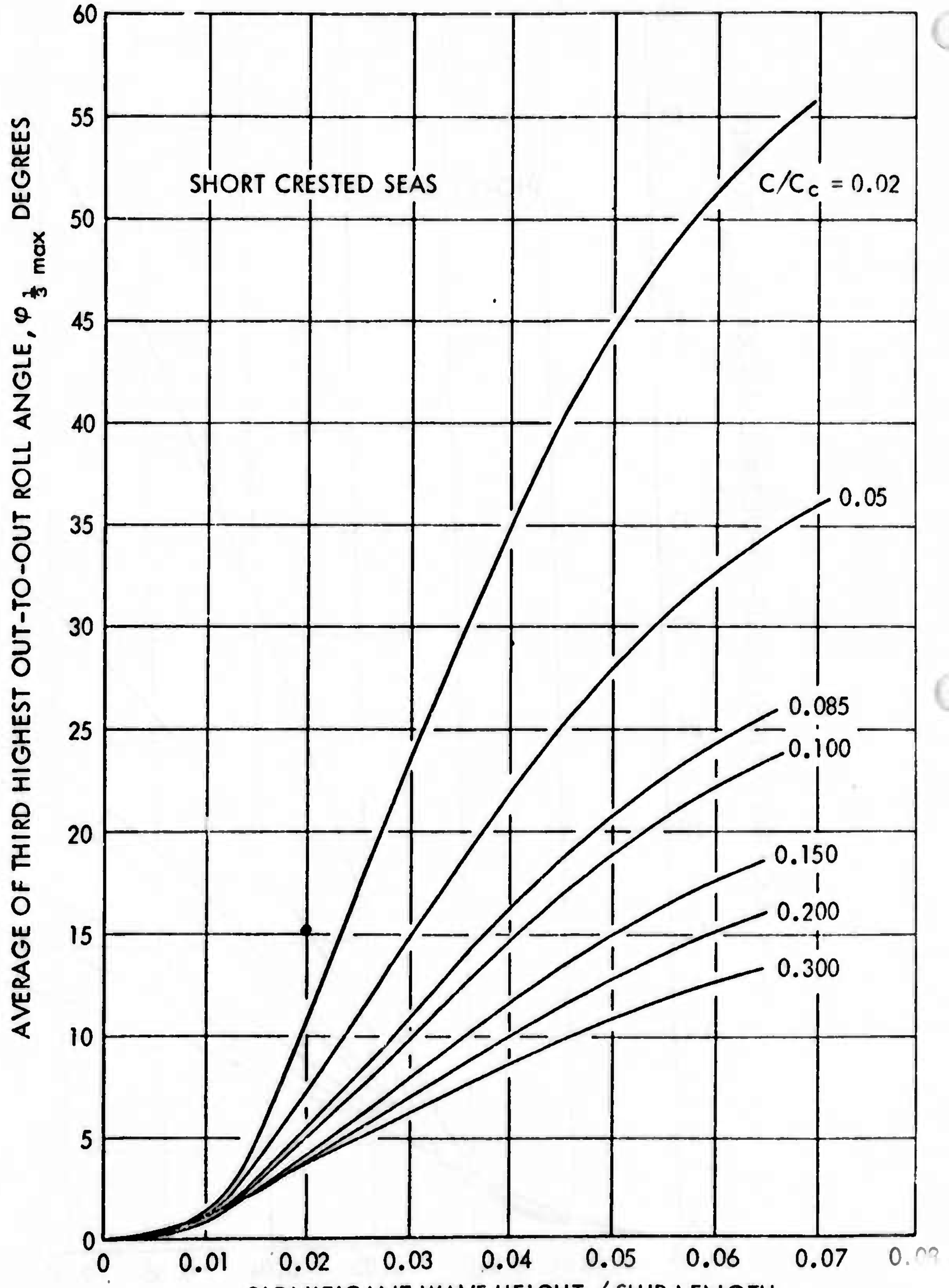


FIGURE 36 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS SHIP DAMPING RATIOS; UNSTABILIZED AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.4$

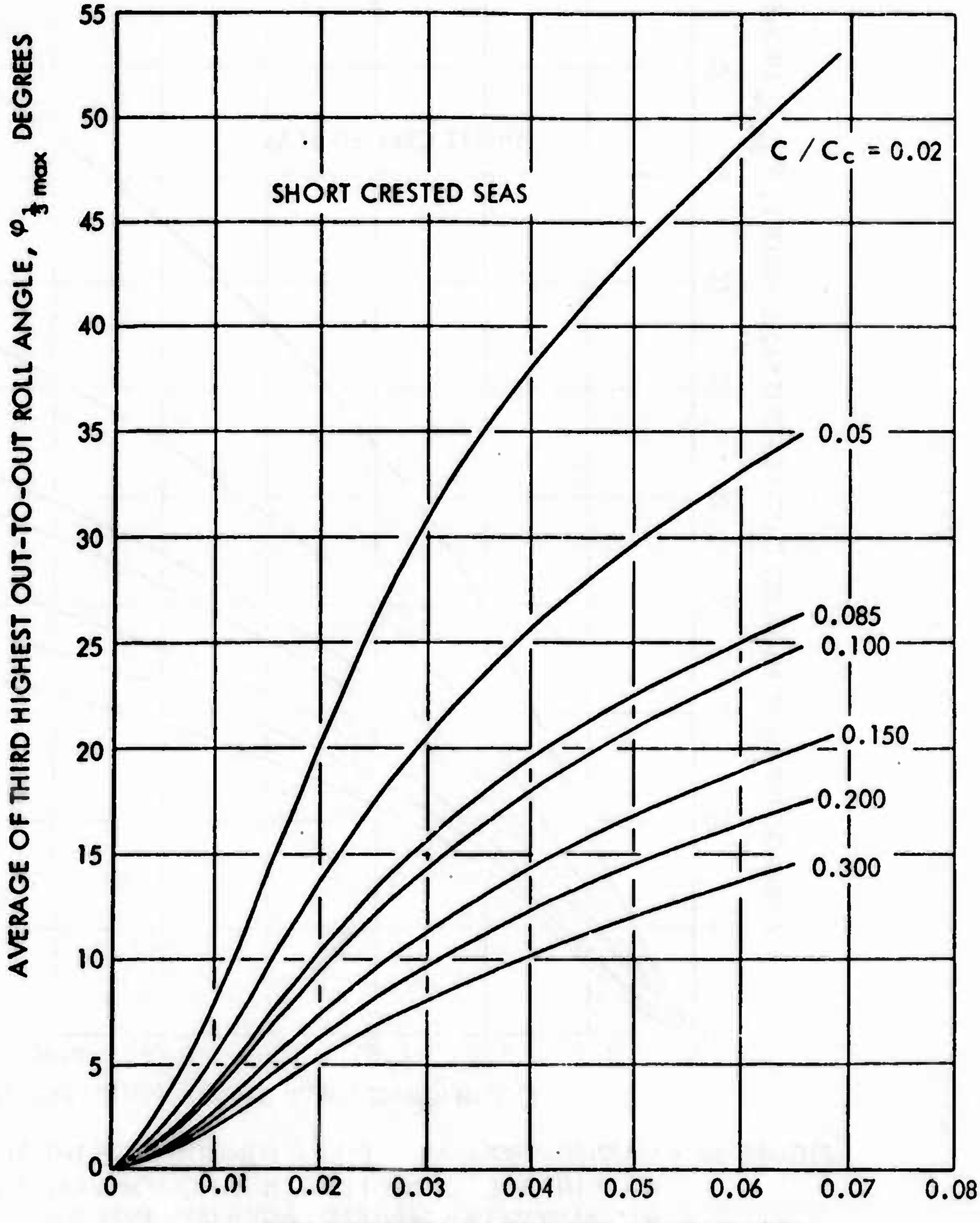


FIGURE 37 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS SHIP DAMPING RATIOS; UNSTABILIZED AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.8$

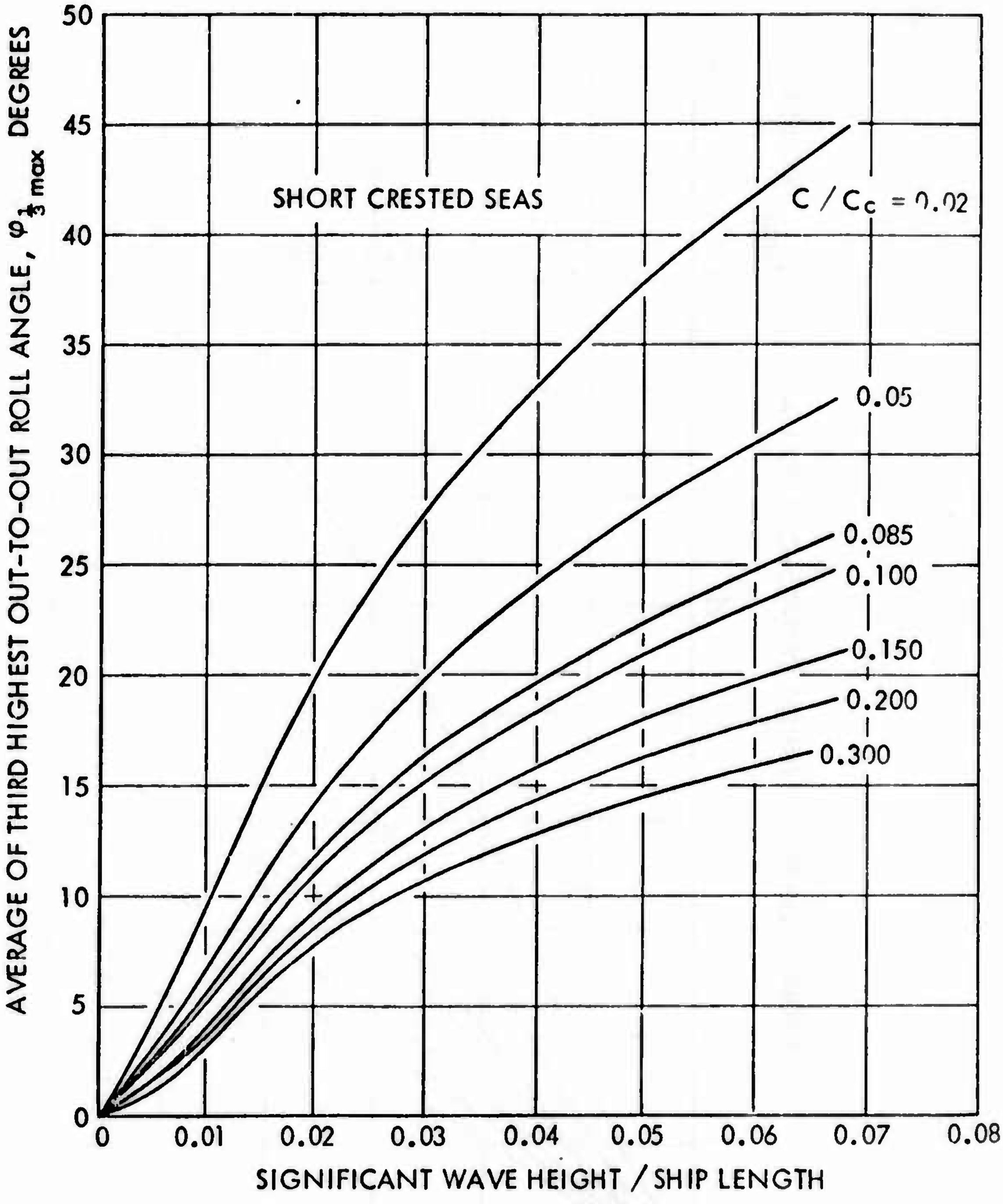


FIGURE 38 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS SHIP DAMPING RATIOS; UNSTABILIZED AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 1.2$

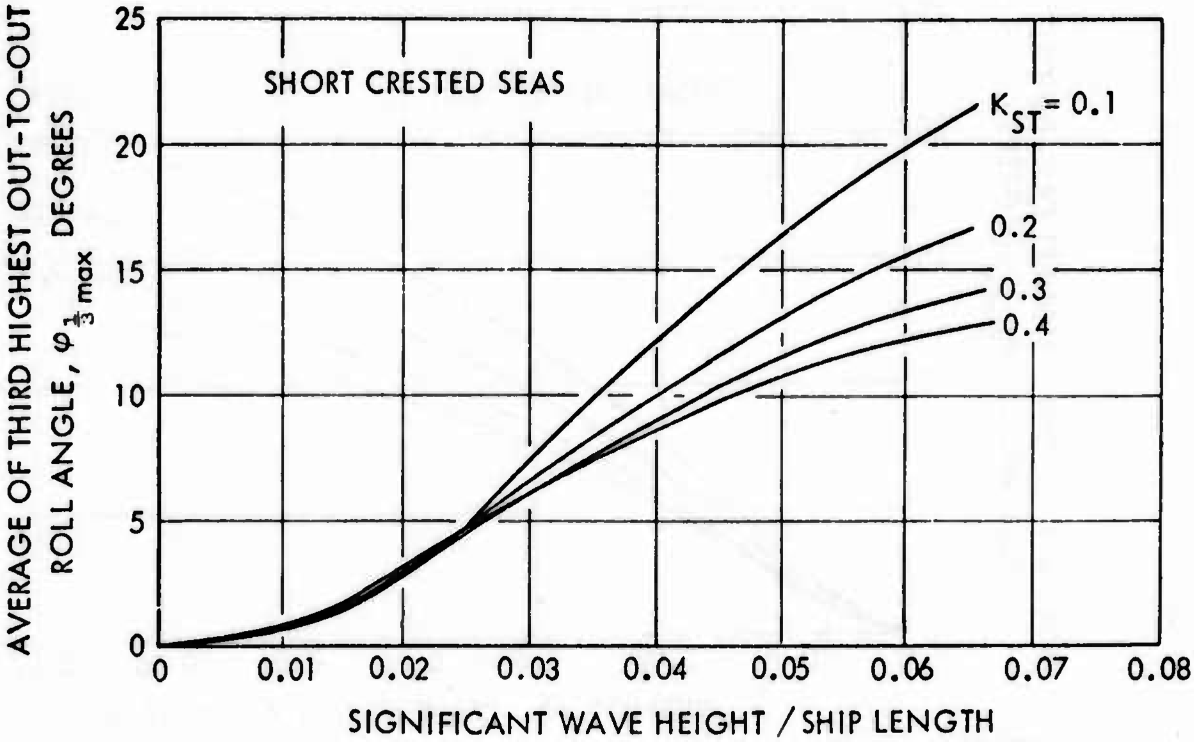


FIGURE 39 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT ZERO SPEED

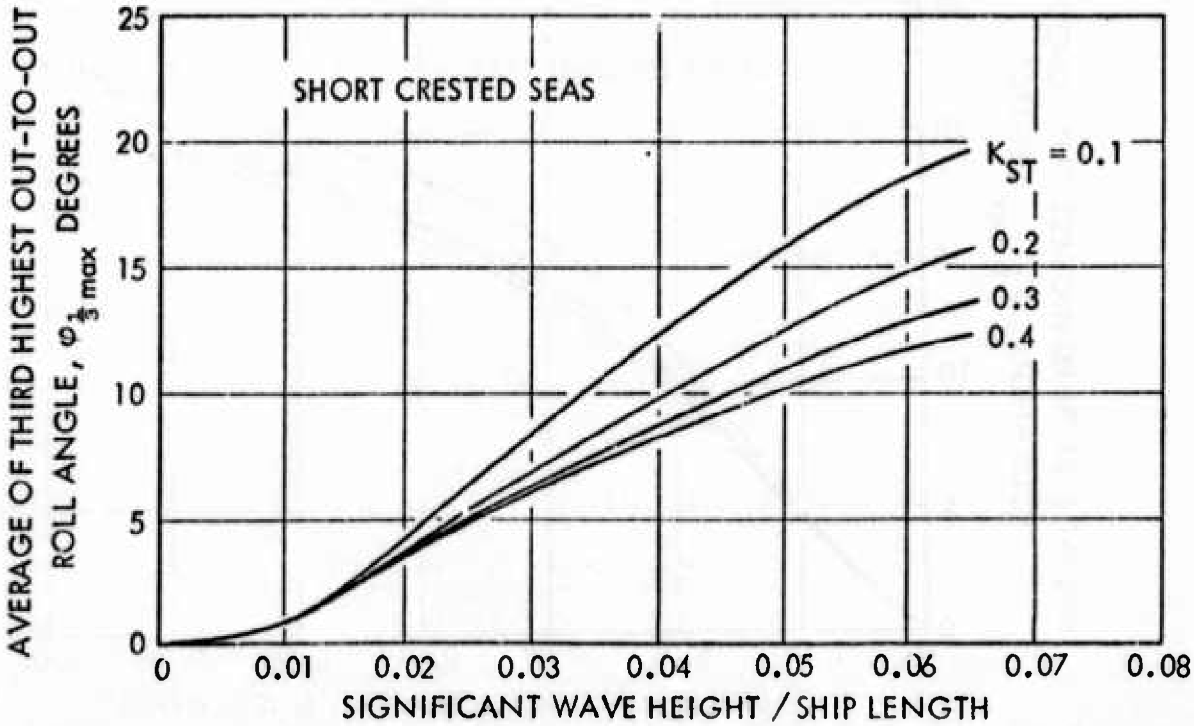


FIGURE 40 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.4$

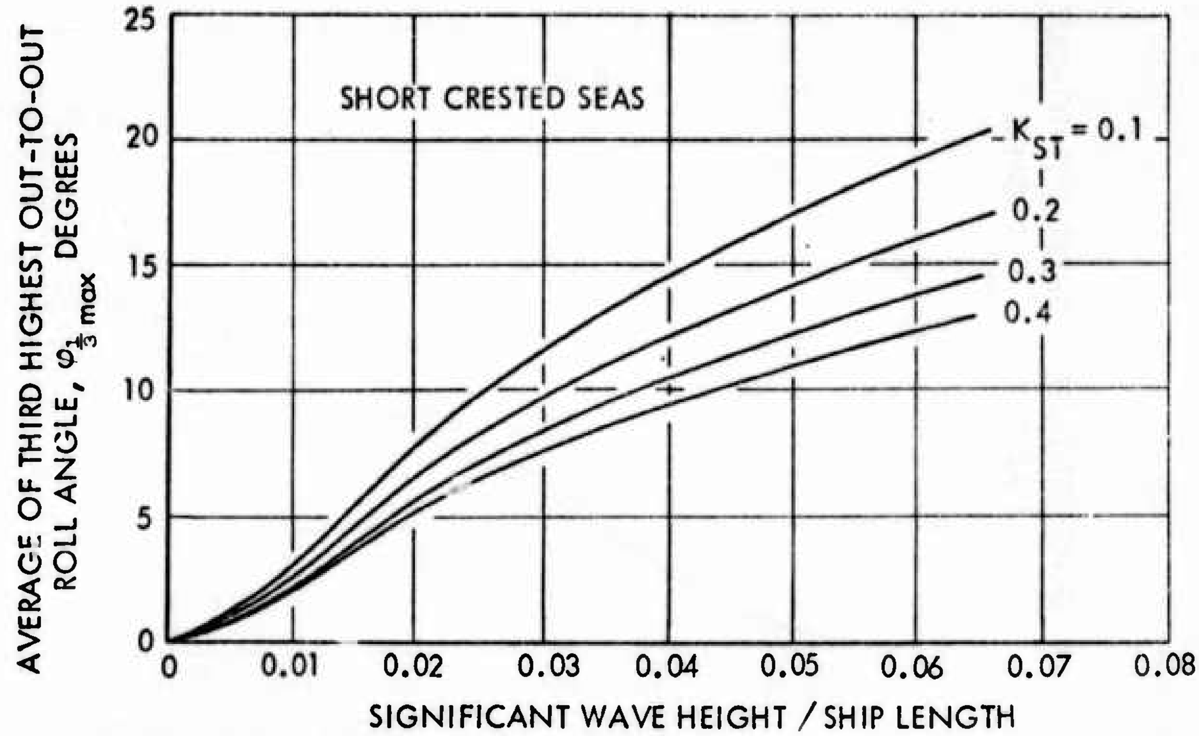


FIGURE 41 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L}$ = 0.8

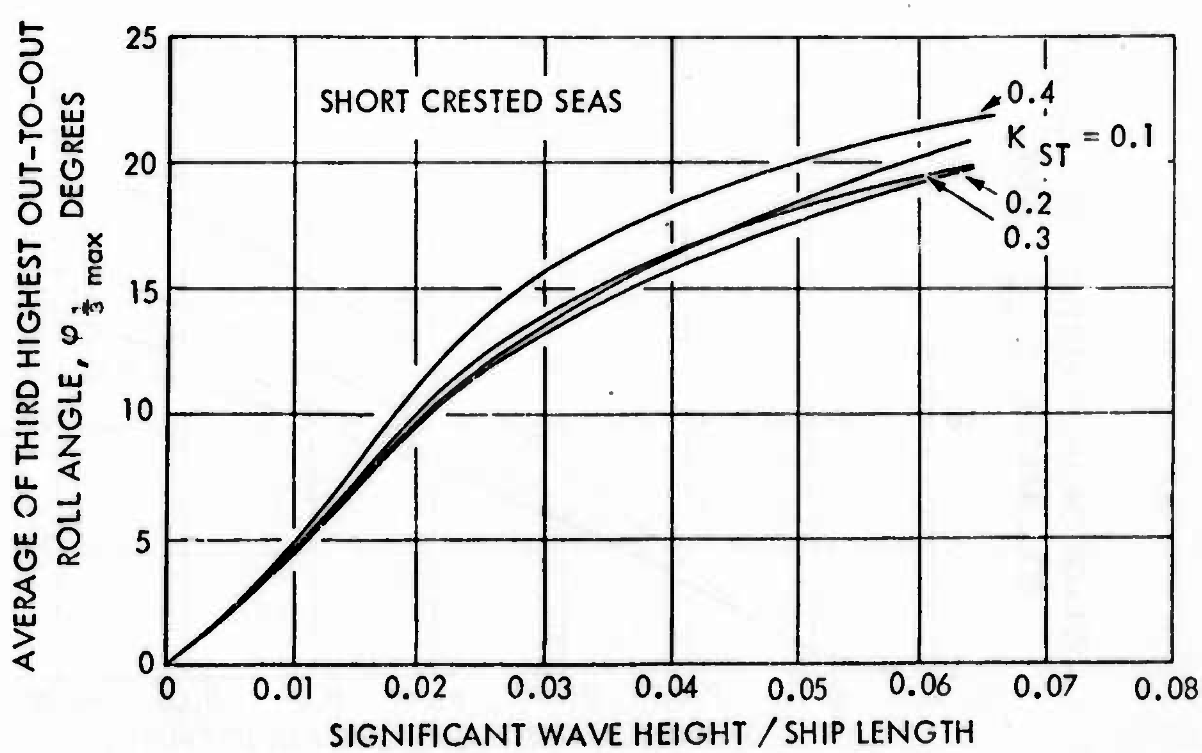


FIGURE 42 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS PASSIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L}$ = 1.2

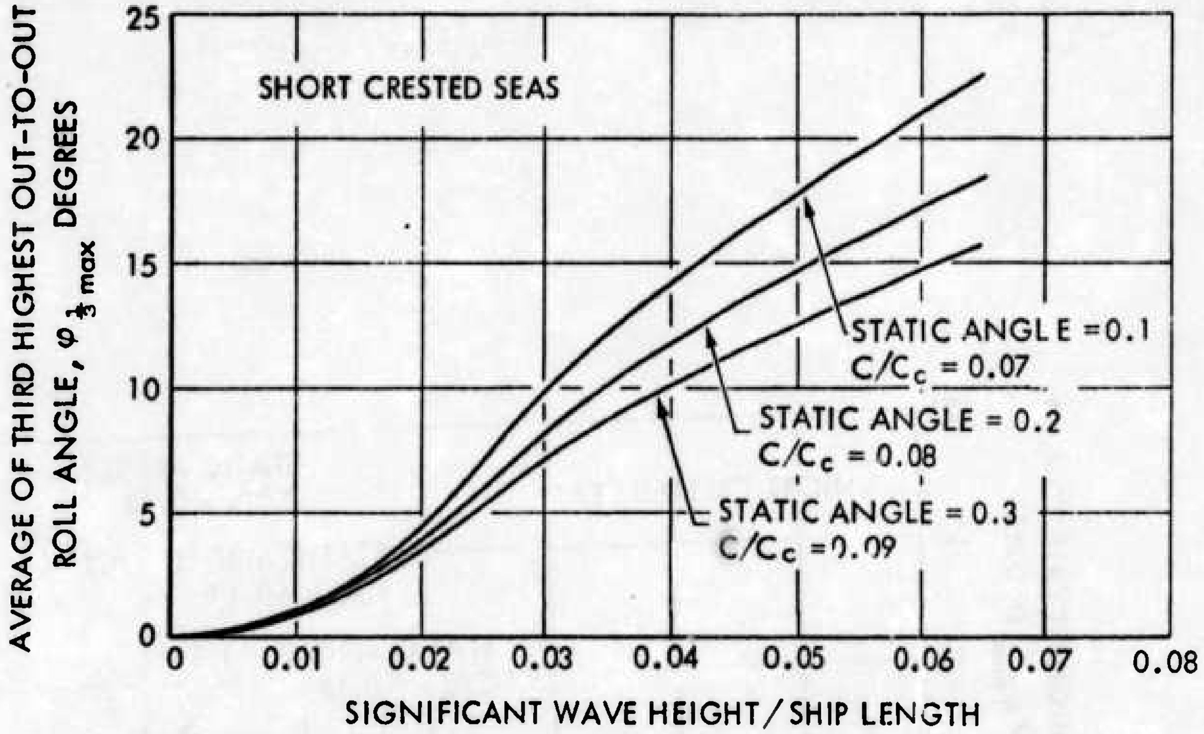


FIGURE 43 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE FIN CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.4$

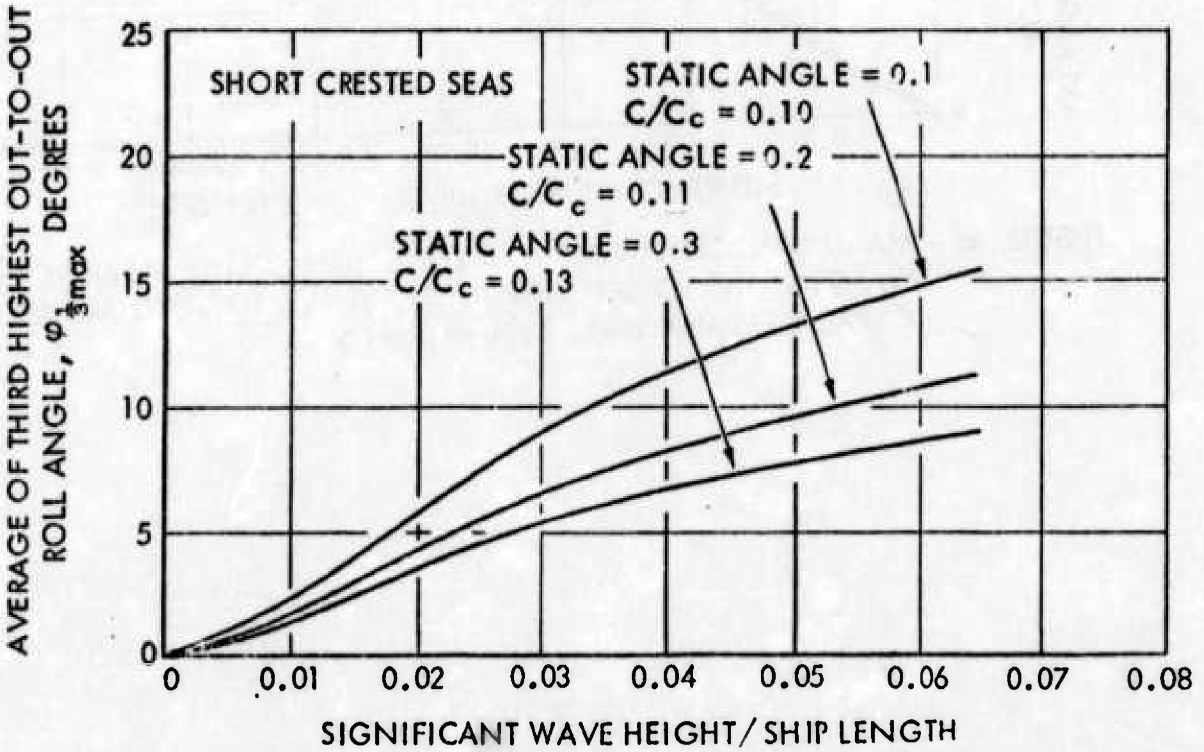


FIGURE 44 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE FIN CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.8$

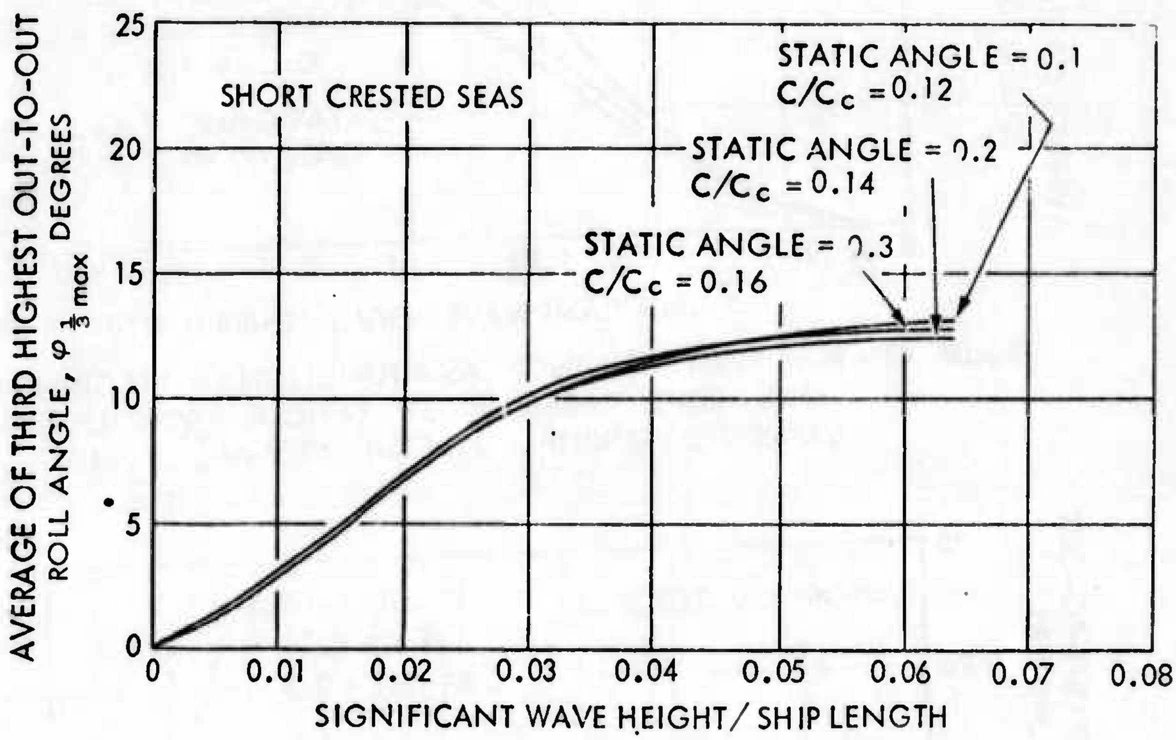


FIGURE 45 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE FIN CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 1.2$

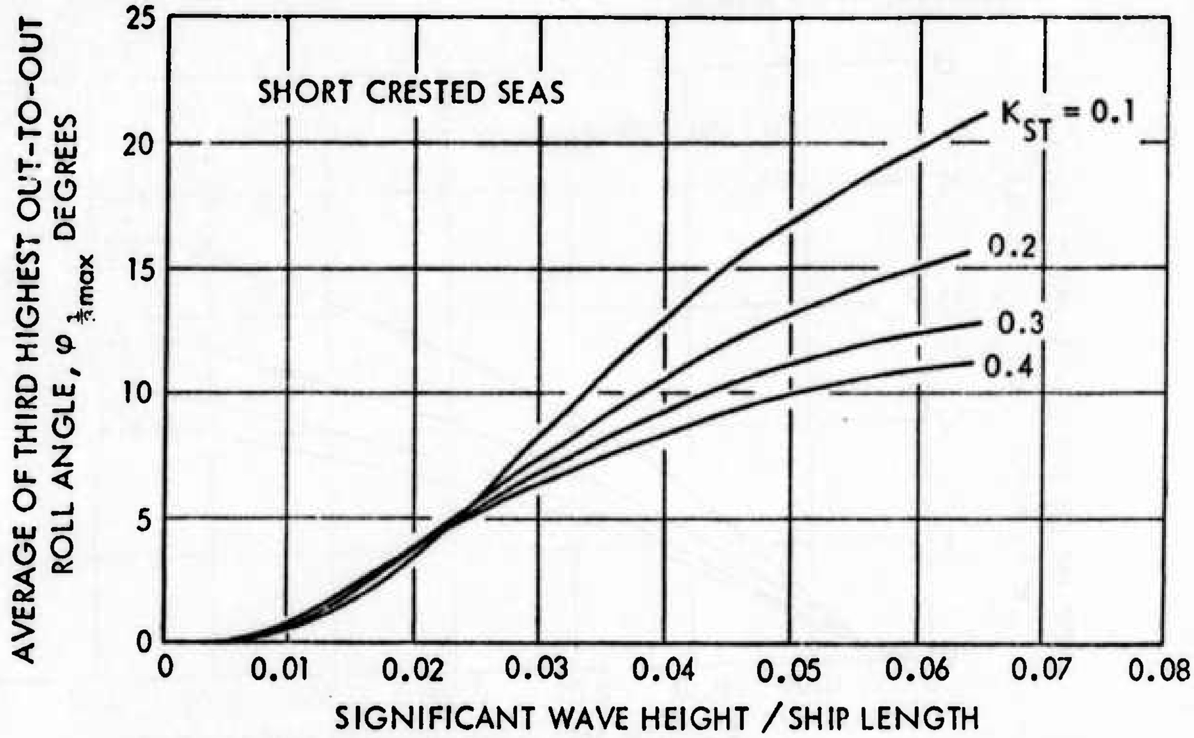


FIGURE 46 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT ZERO SPEED

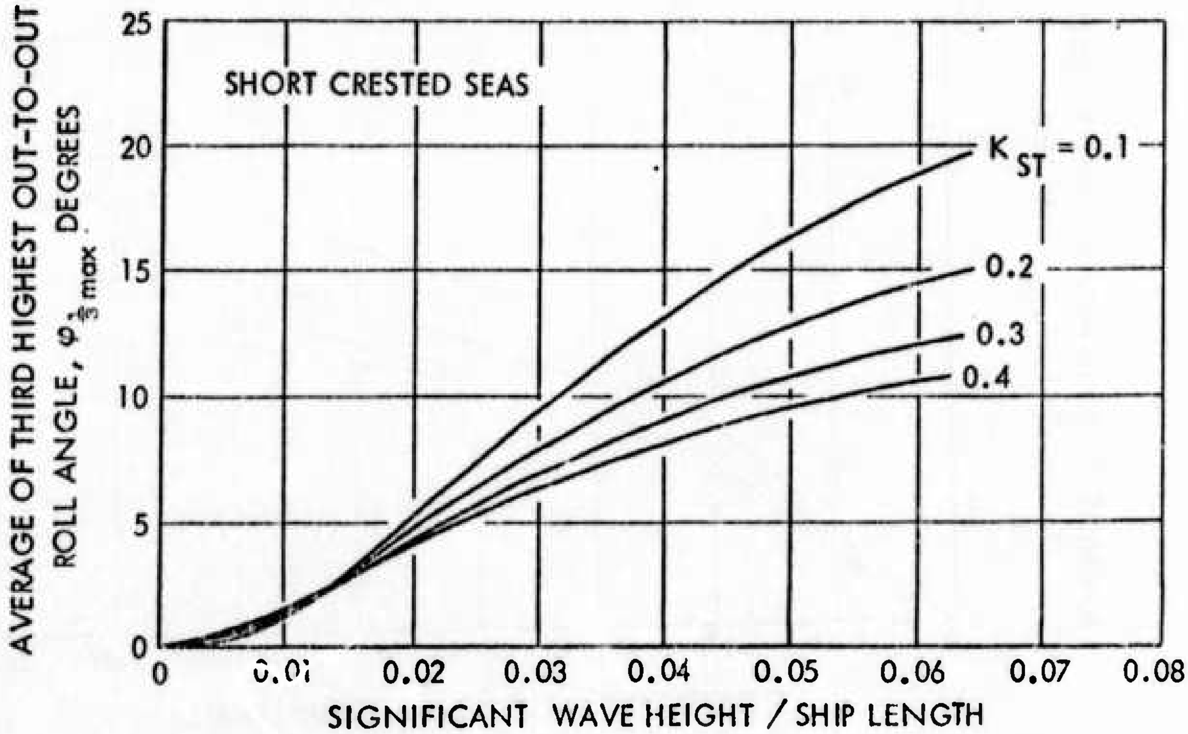


FIGURE 47 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.4$

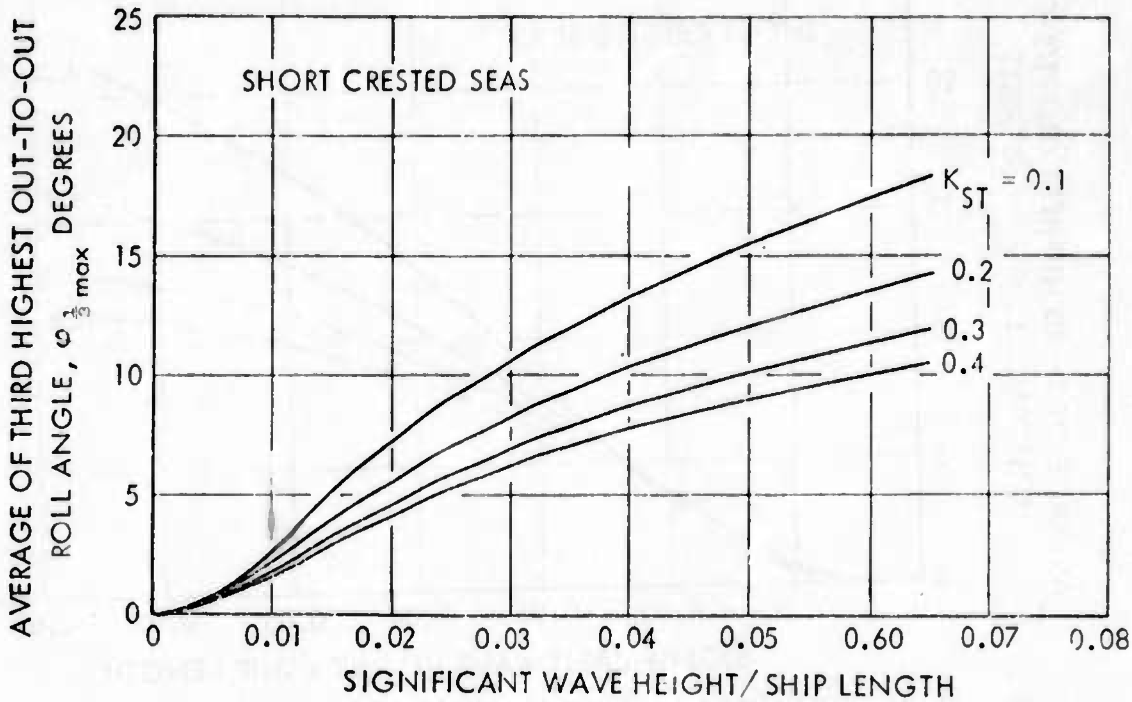


FIGURE 48 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.8$

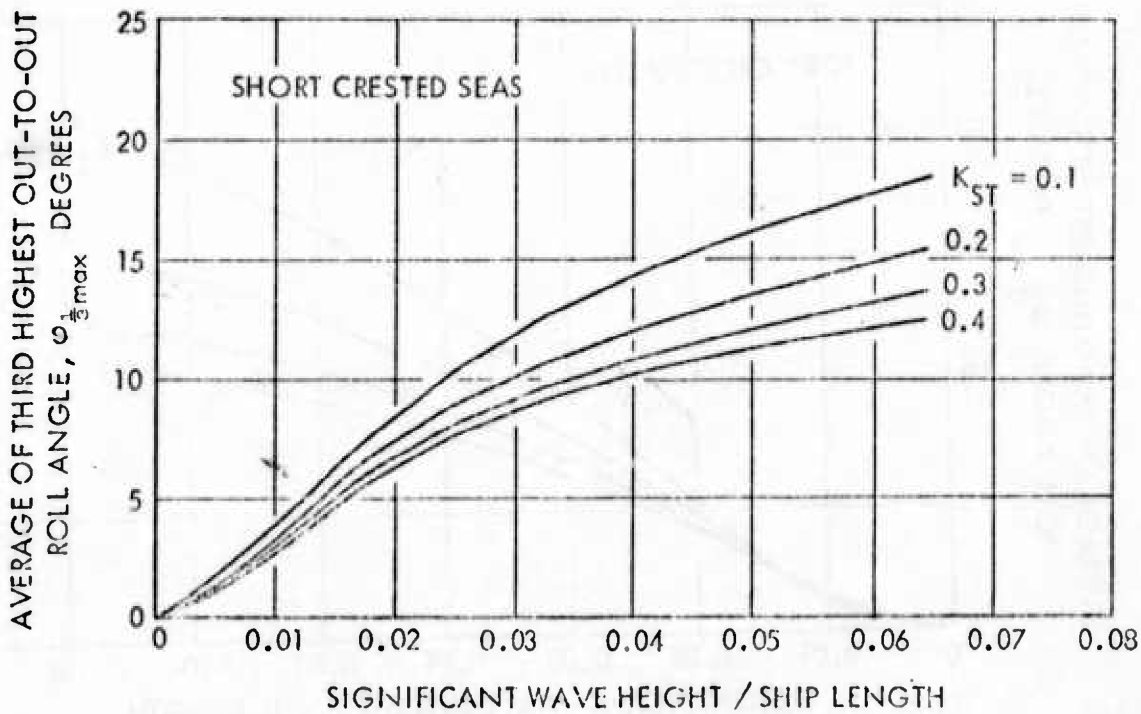


FIGURE 49 - MAXIMUM ROLL ANGLE AS A FUNCTION OF THE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIO FOR VARIOUS ACTIVE TANK CAPACITIES; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 1.2$

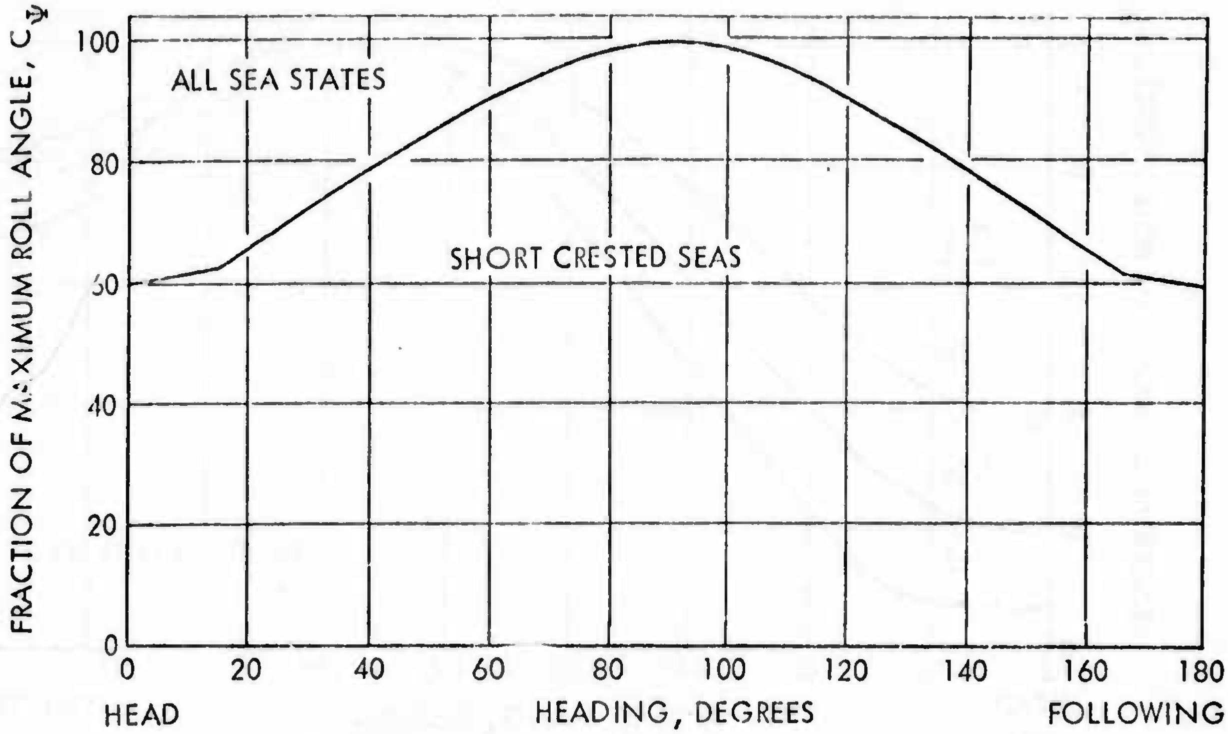


FIGURE 50 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR ALL SEA STATES; UNSTABILIZED AUXILIARY TYPE SHIP AT ZERO SPEED

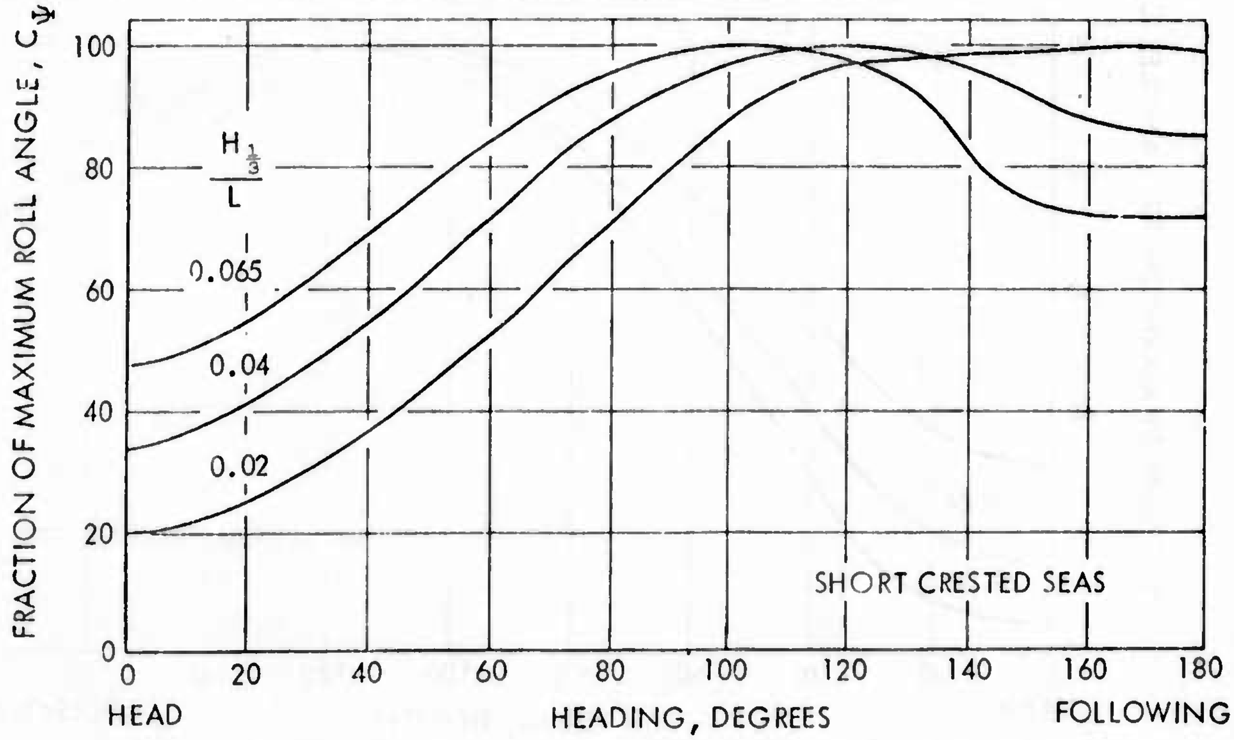


FIGURE 51 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS; UNSTABILIZED AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.4$

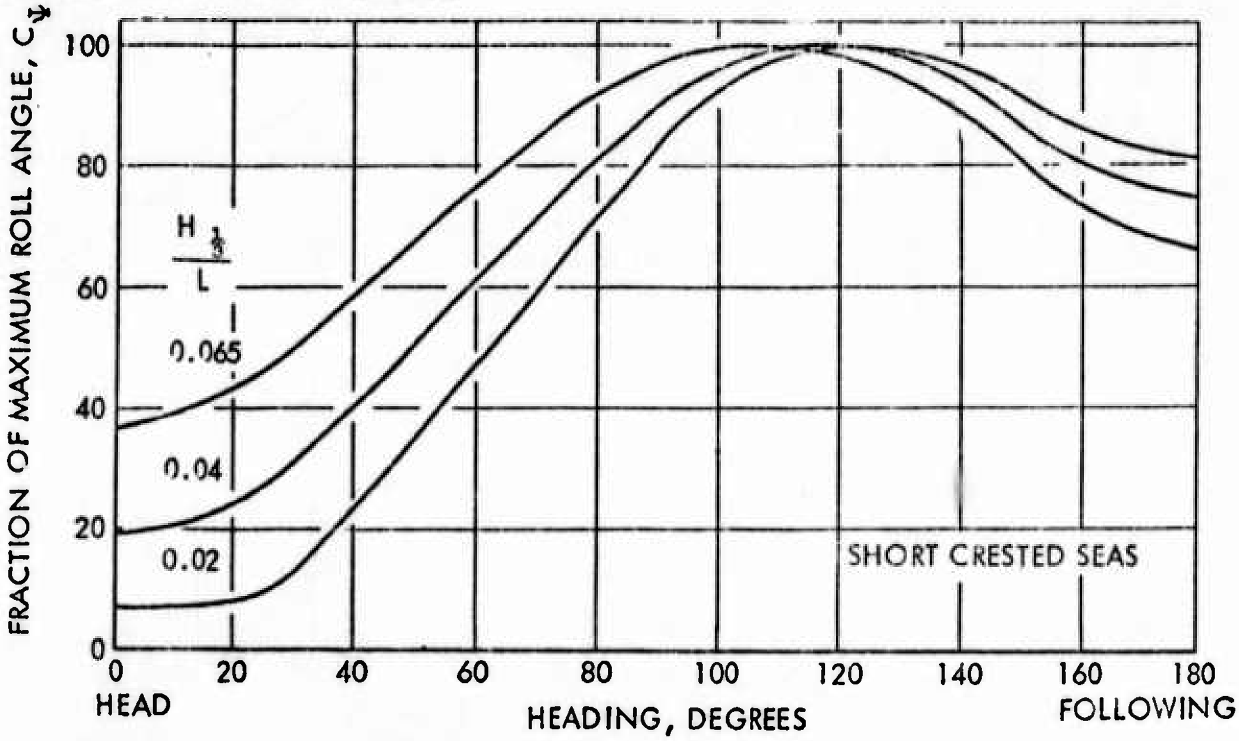


FIGURE 52 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS; UNSTABILIZED AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.8$

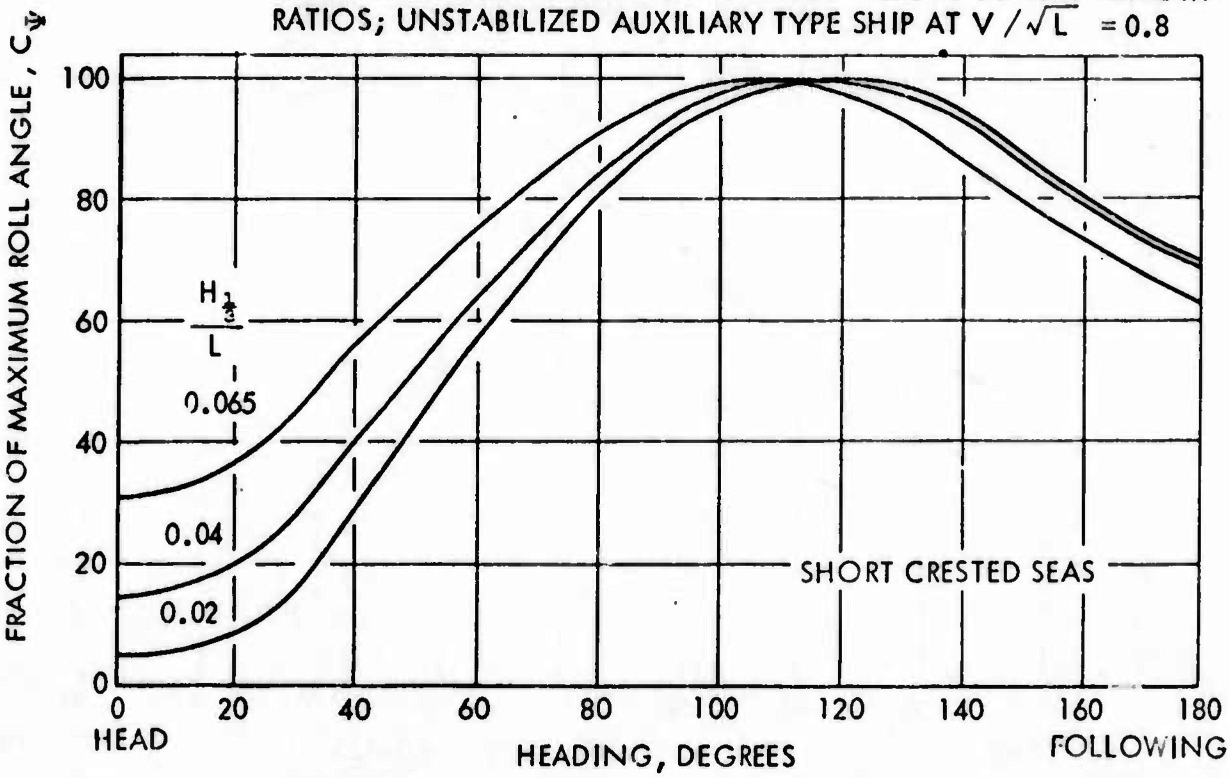


FIGURE 53 - CHANGE IN ROLL ANGLE AS A FUNCTION OF THE HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS; UNSTABILIZED AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 1.2$

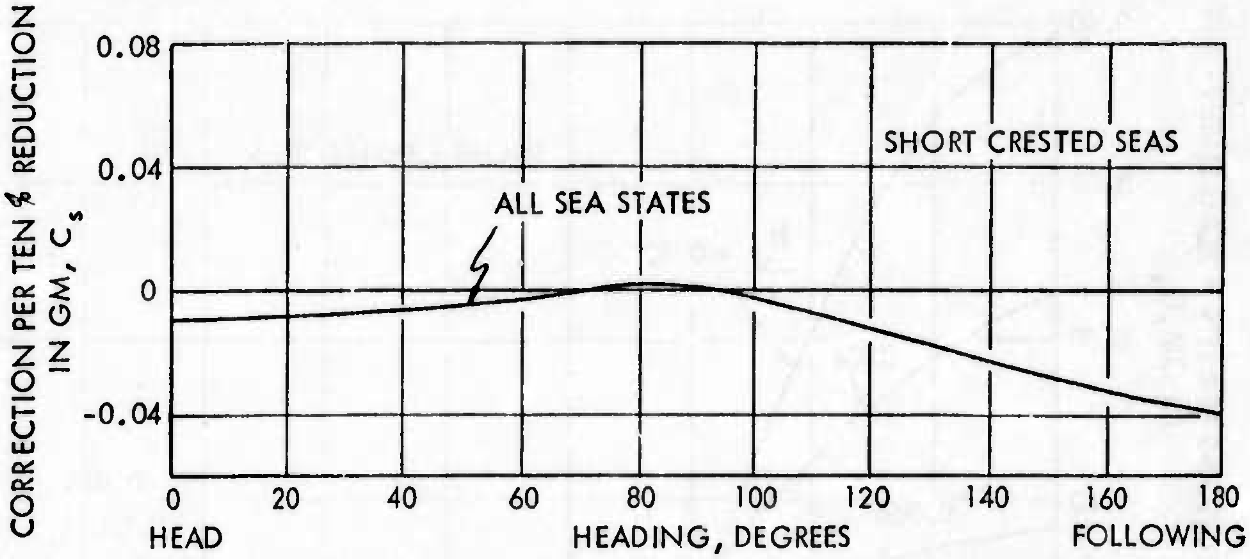


FIGURE 54 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE WITH PASSIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT ZERO SPEED

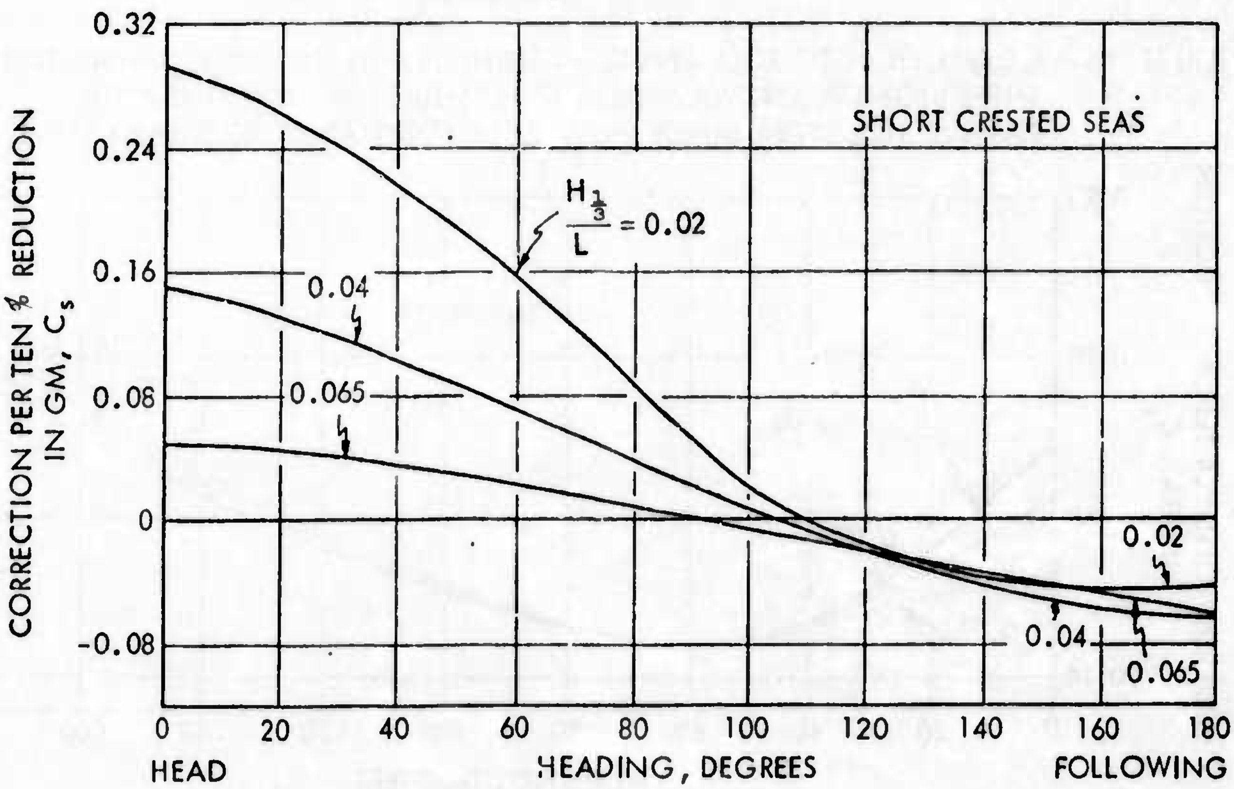


FIGURE 55 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH PASSIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.04$

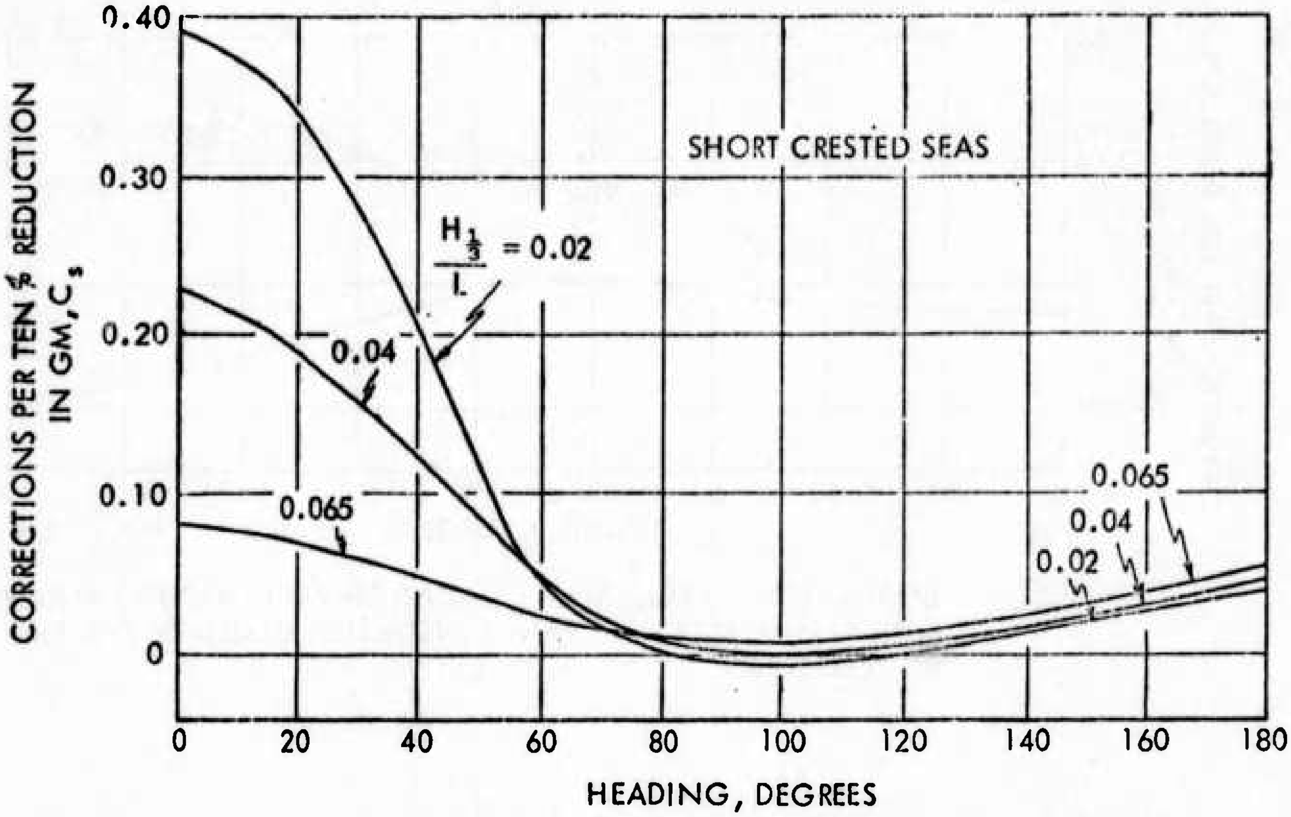


FIGURE 56 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH PASSIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.8$

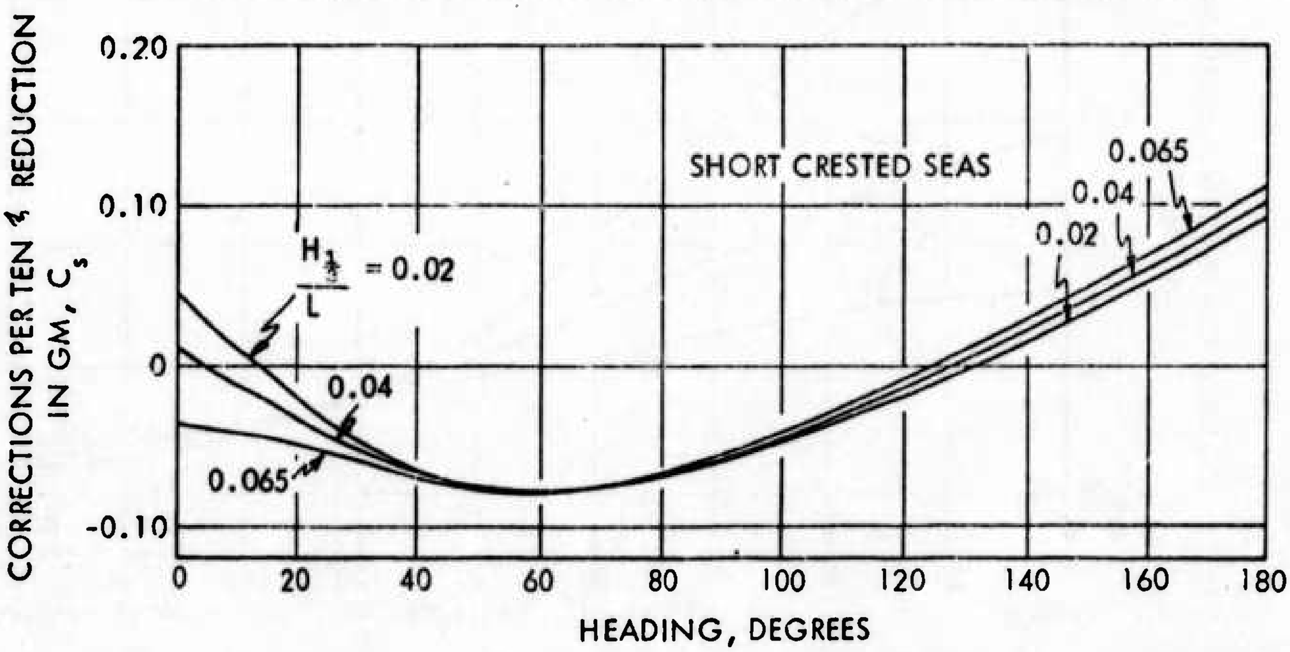


FIGURE 57 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH PASSIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 1.2$

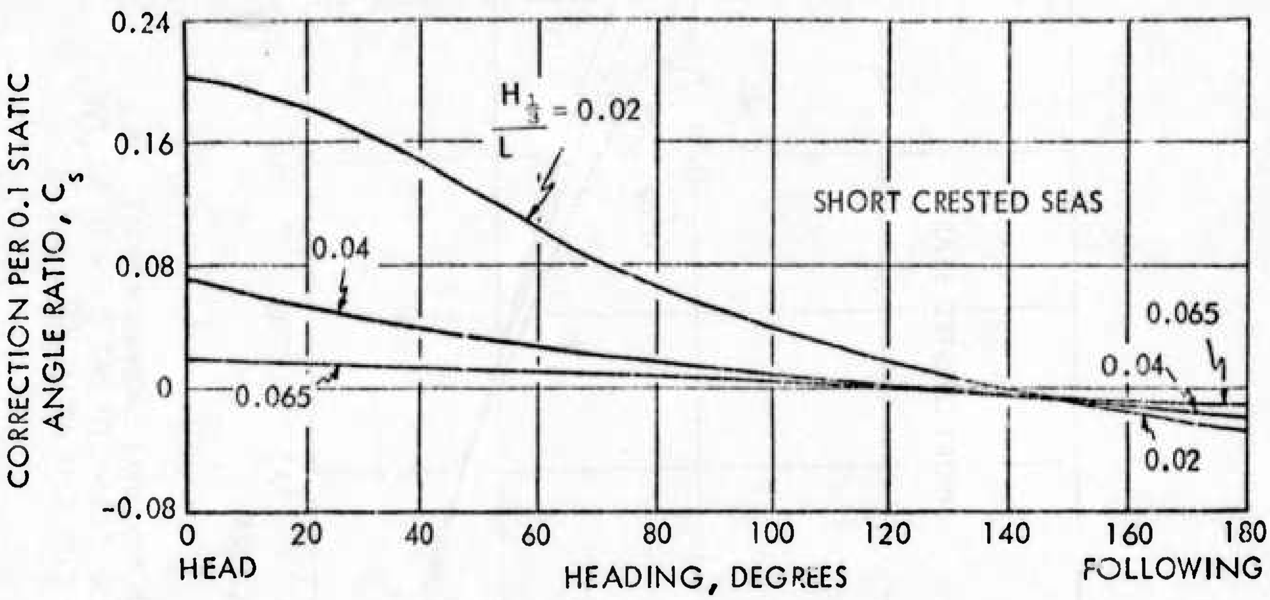


FIGURE 58 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE FIN STABILIZER INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.4$

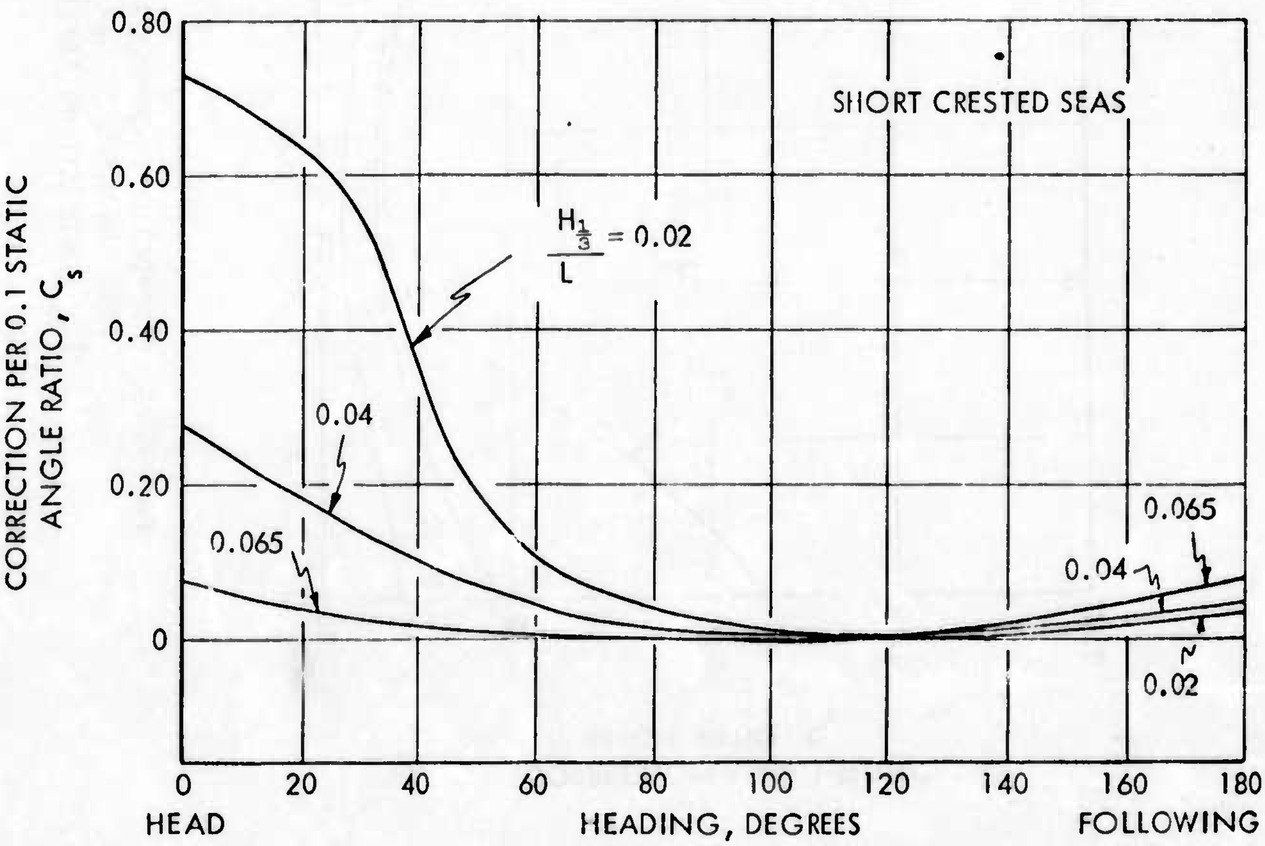


FIGURE 59 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE FIN STABILIZER INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.8$

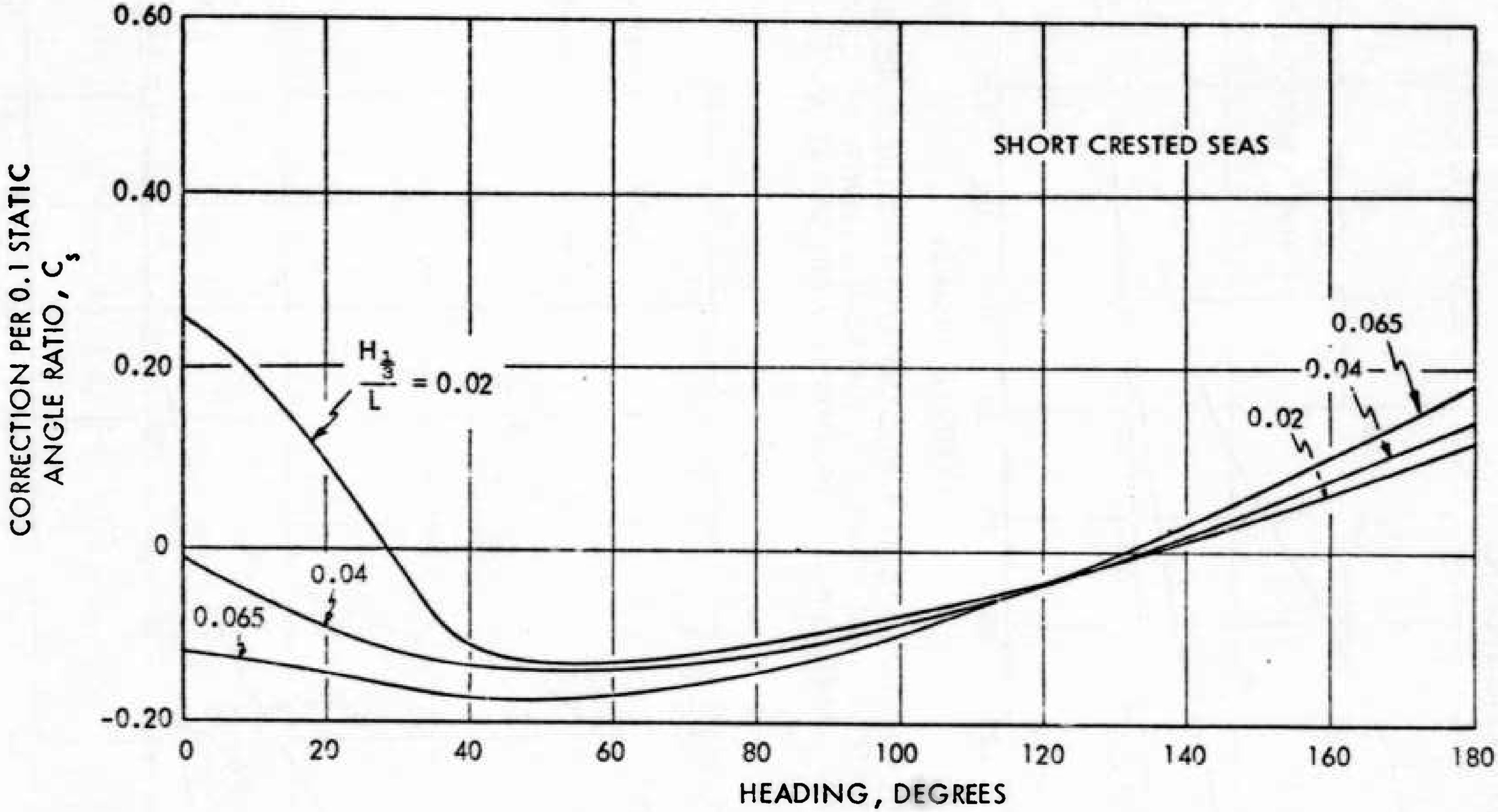


FIGURE 60 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE FIN STABILIZER INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 1.2$

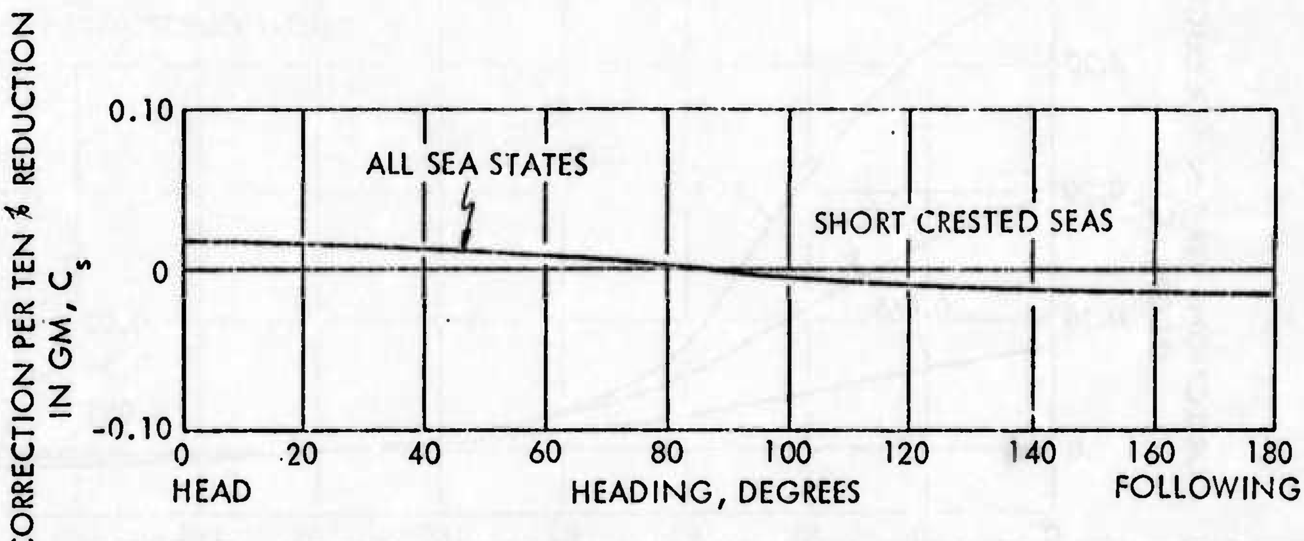


FIGURE 61 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE WITH ACTIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT ZERO SPEED

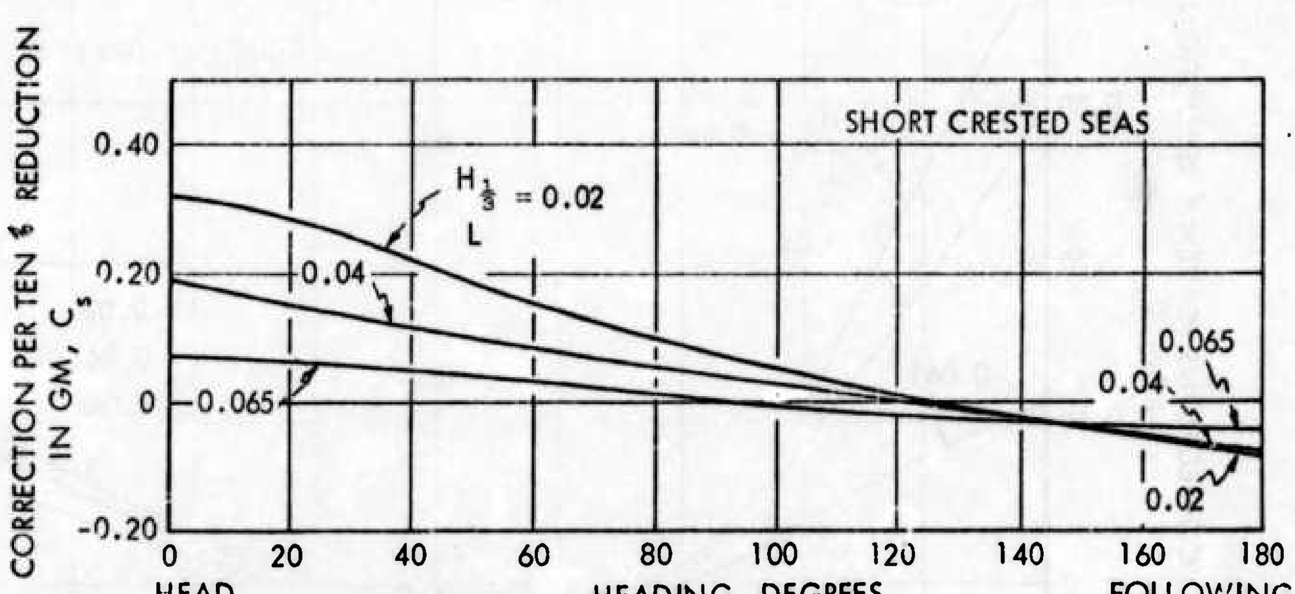


FIGURE 62 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.04$

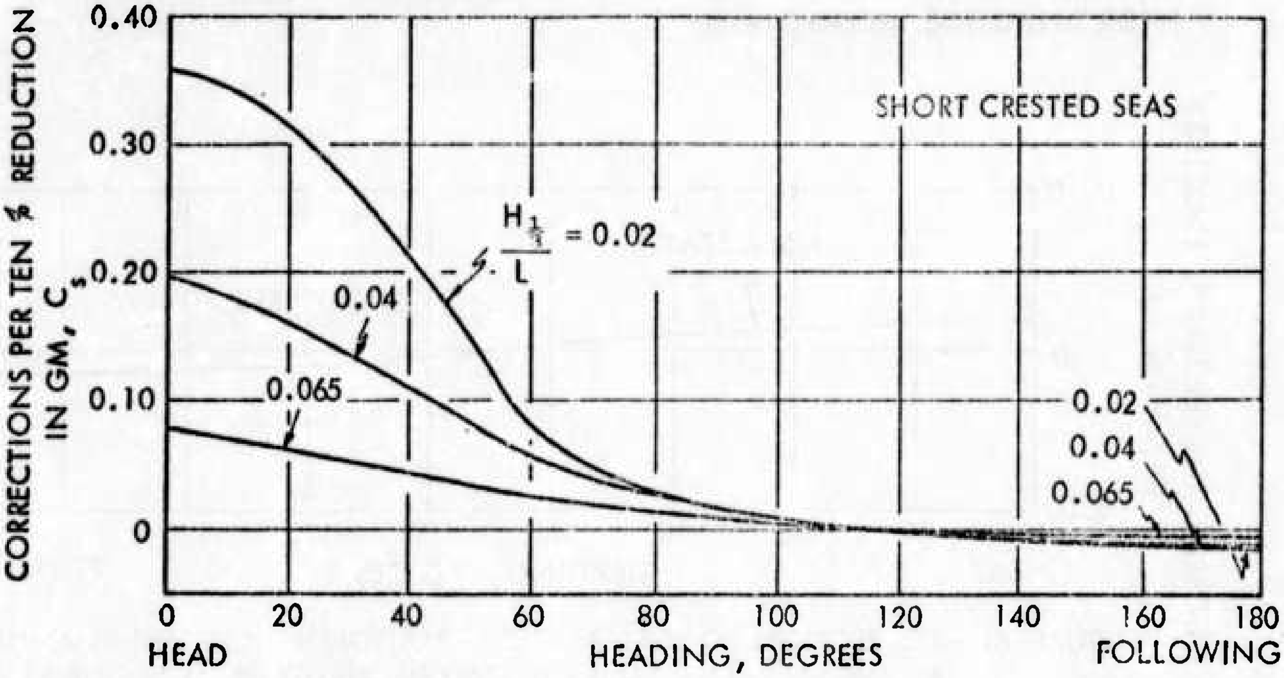


FIGURE 63 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 0.8$

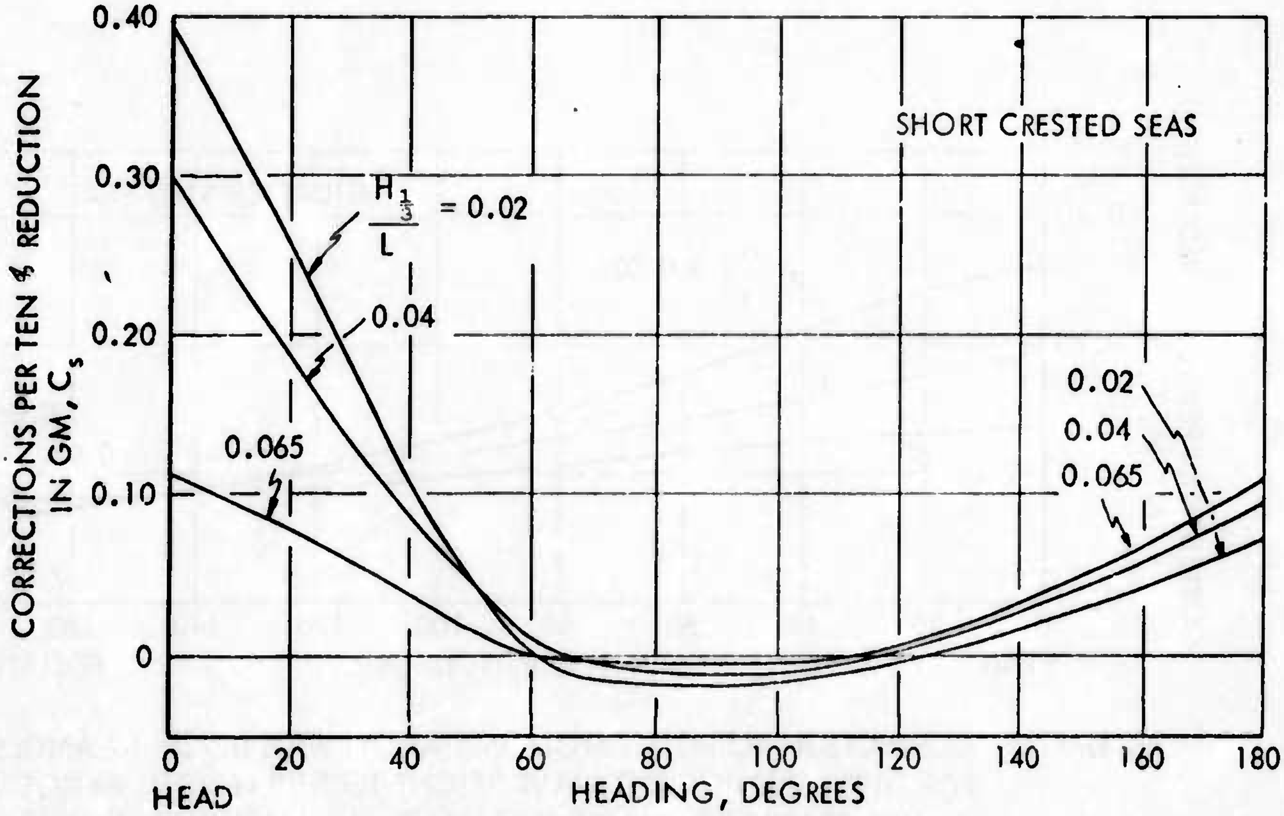
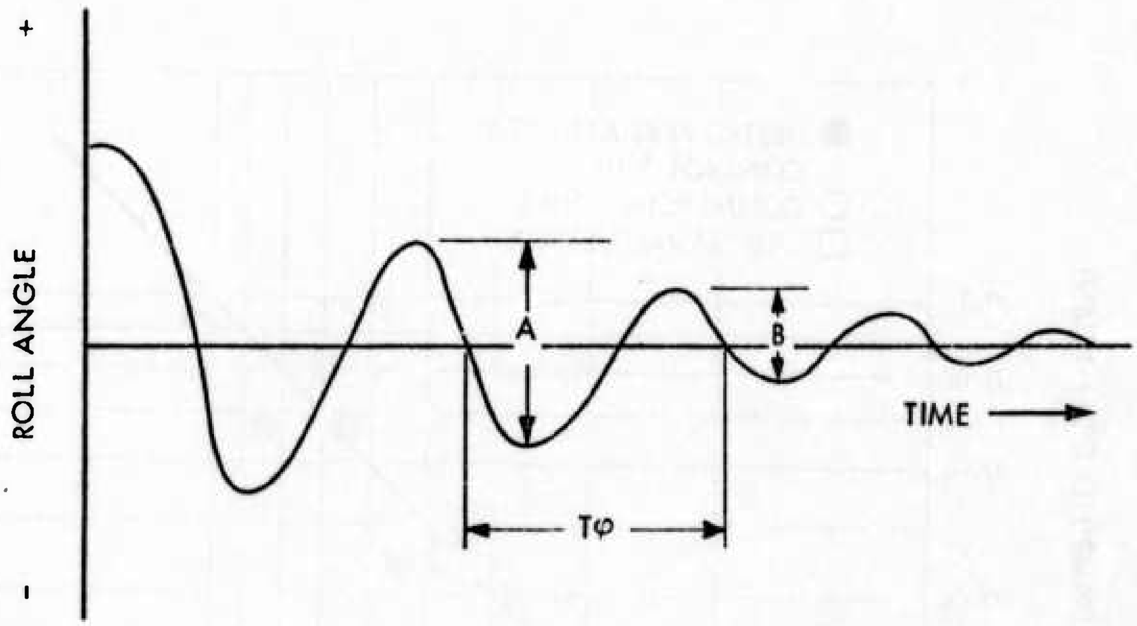


FIGURE 64 - CORRECTION TO ROLL ANGLE VARIATION WITH HEADING ANGLE FOR THREE SIGNIFICANT WAVE HEIGHT TO SHIP LENGTH RATIOS WITH ACTIVE STABILIZER TANKS INSTALLED; AUXILIARY TYPE SHIP AT $V/\sqrt{L} = 1.2$



$$\ln \frac{A}{B} = 2\pi (C/C_c)$$

C/C_c = EFFECTIVE ROLL DAMPING COEFFICIENT

$T\varphi$ = ROLL PERIOD IN CALM WATER

FIGURE 65 - CALCULATIONS OF EFFECTIVE ROLL DAMPING COEFFICIENT FROM A ROLL DECAY TEST

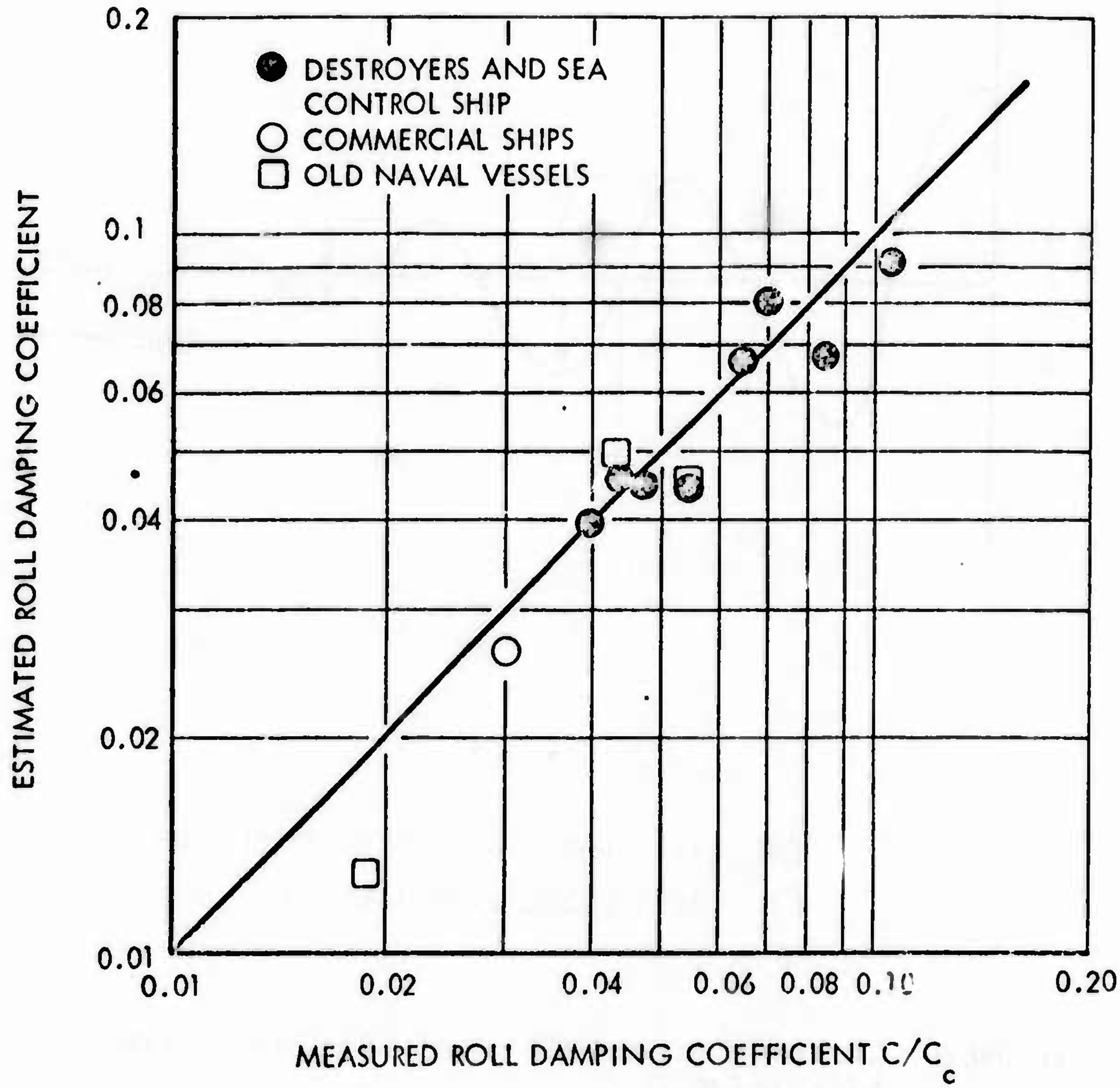


FIGURE 66 - COMPARISON OF ESTIMATED AND MEASURED ROLL DAMPING COEFFICIENTS

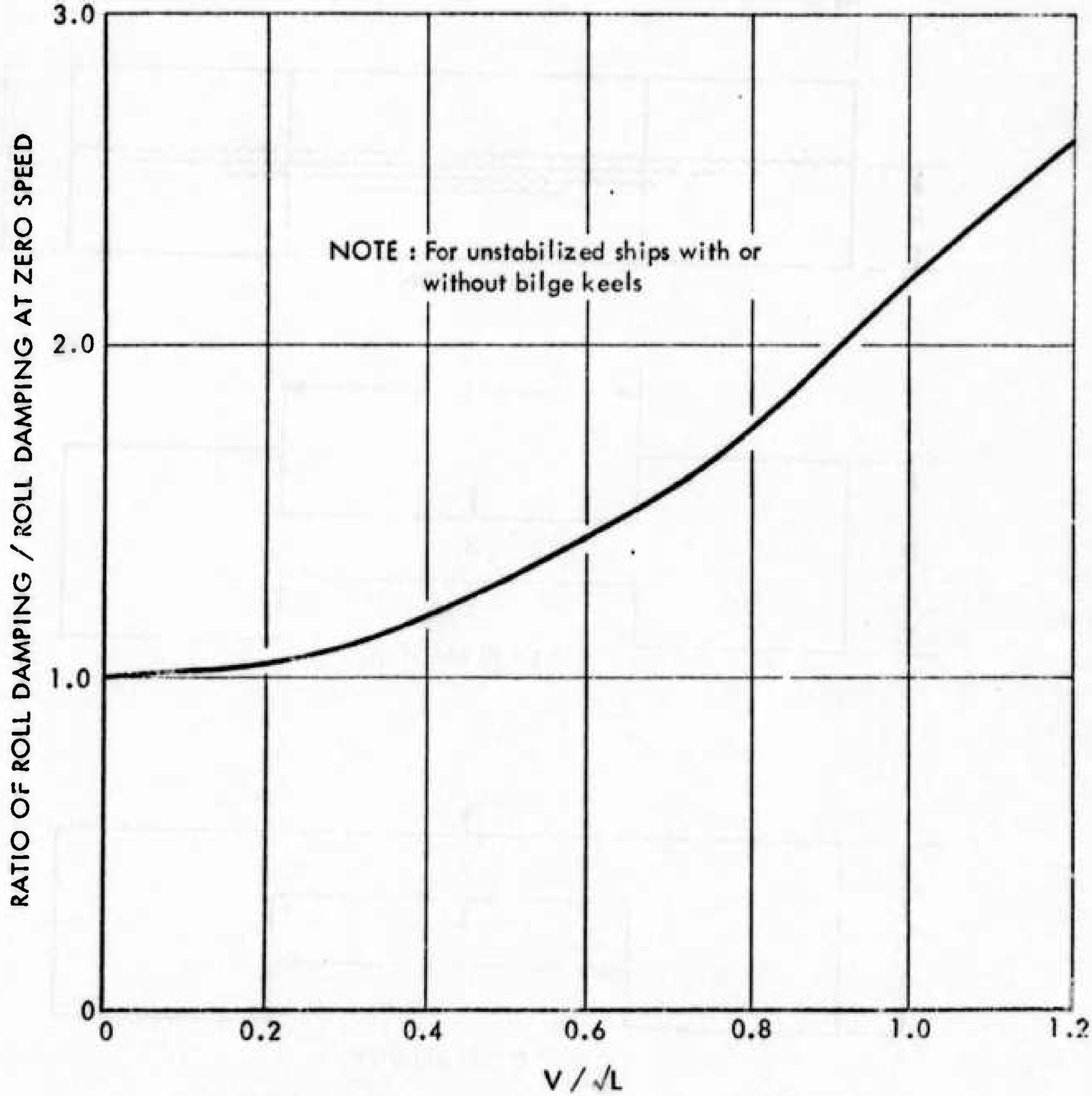


FIGURE 67 - TYPICAL VARIATION OF ROLL DAMPING COEFFICIENT WITH SPEED LENGTH RATIO

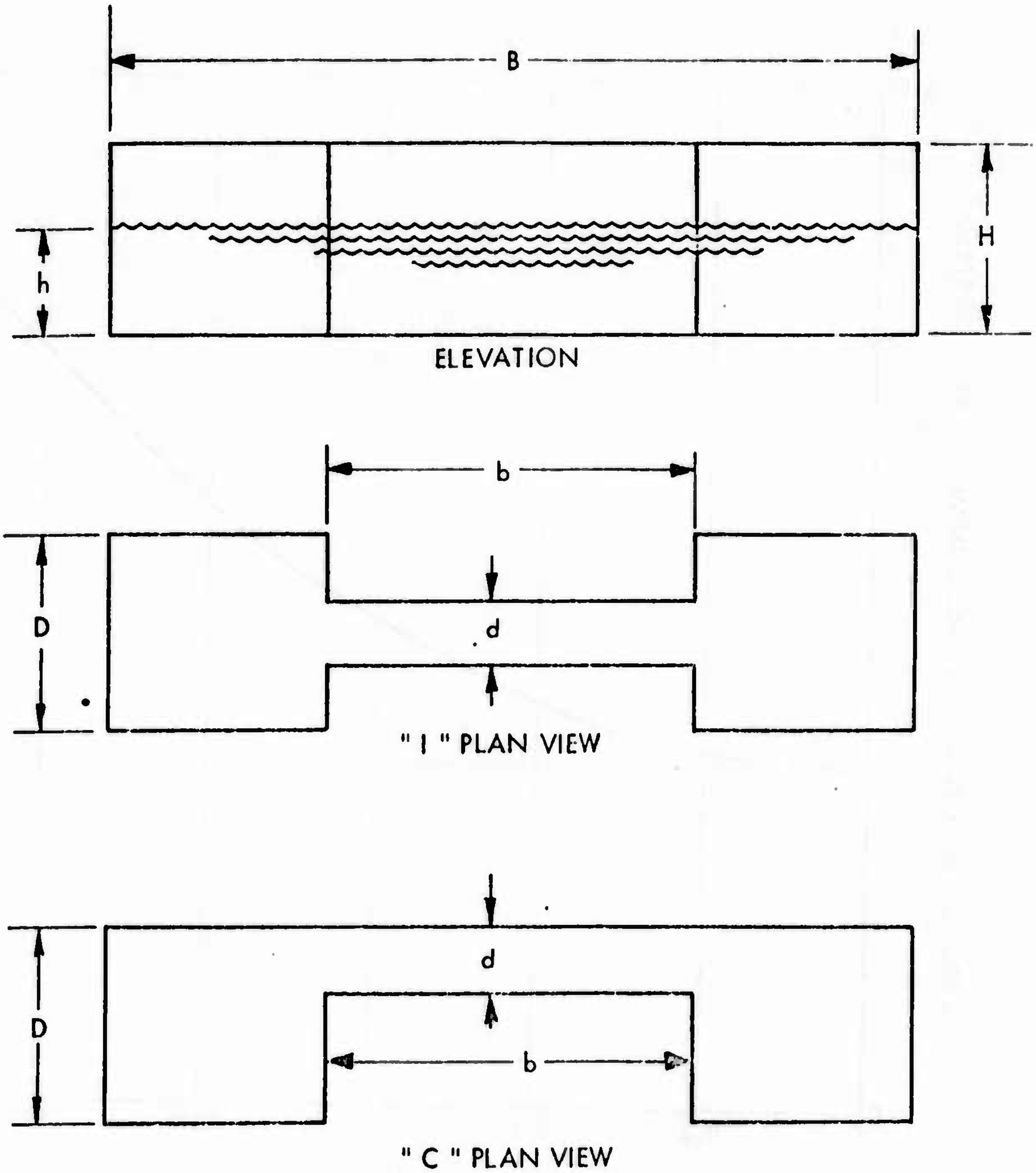


FIGURE 68 - DIMENSIONS FOR FREE SURFACE TANKS

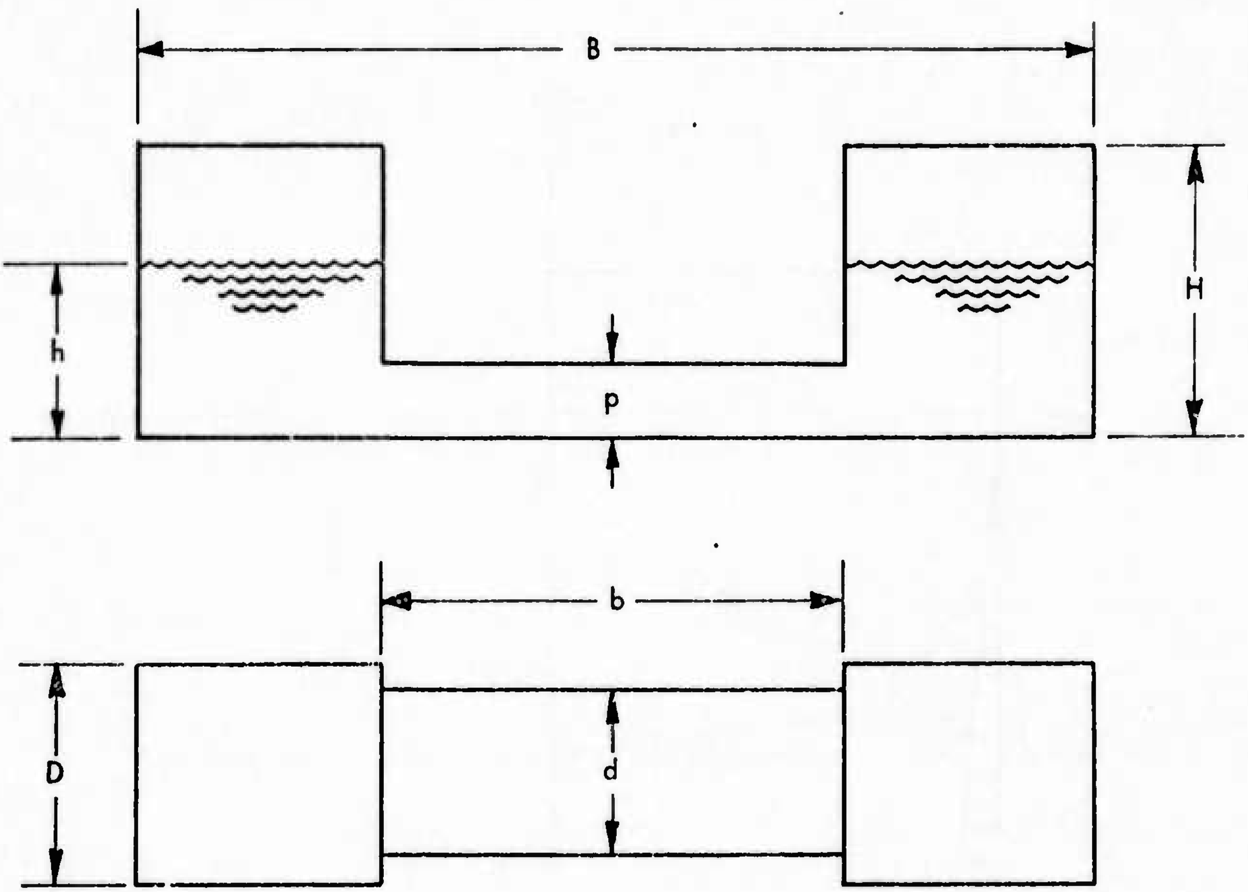


FIGURE 69 - DIMENSIONS OF U-TUBE TANK

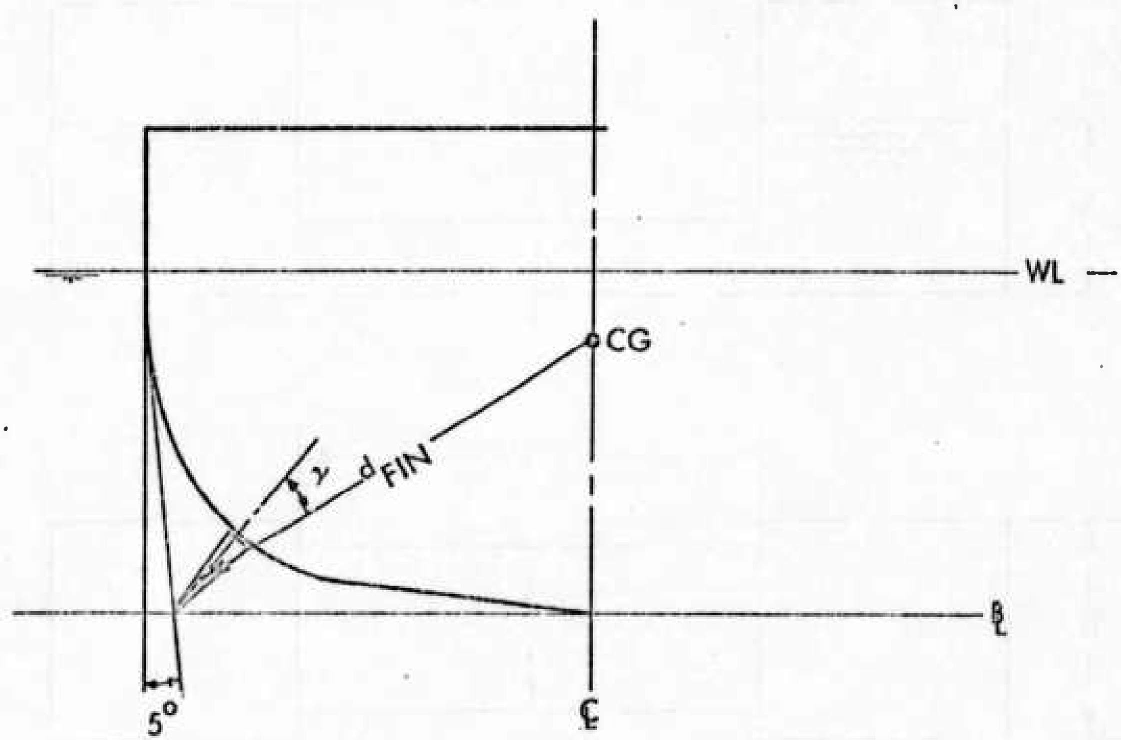


FIGURE 70 - ACTIVE FIN LOCATION