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# HYDRONAUTICS, incorporated
## research in hydrodynamics

Research, consulting, and advanced engineering in the fields of NAVAL and INDUSTRIAL HYDRODYNAMICS. Offices and Laboratory in the Washington, D. C., area: Pindell School Road, Howard County, Laurel, Md.

HYDRONAUTICS, Incorporated TECHNICAL REPORT 762-1

# VOLUME II: ENGINEERING AND ECONOMIC FEASIBILITY STUDY OF ROLL STABILIZATION AND HEEL INDUCING SYSTEMS FOR NEW COAST GUARD POLAR ICEBREAKERS

February 1968

By

J. F. Dalzell
with Appendices by
Horst Nowacki

[redacted]

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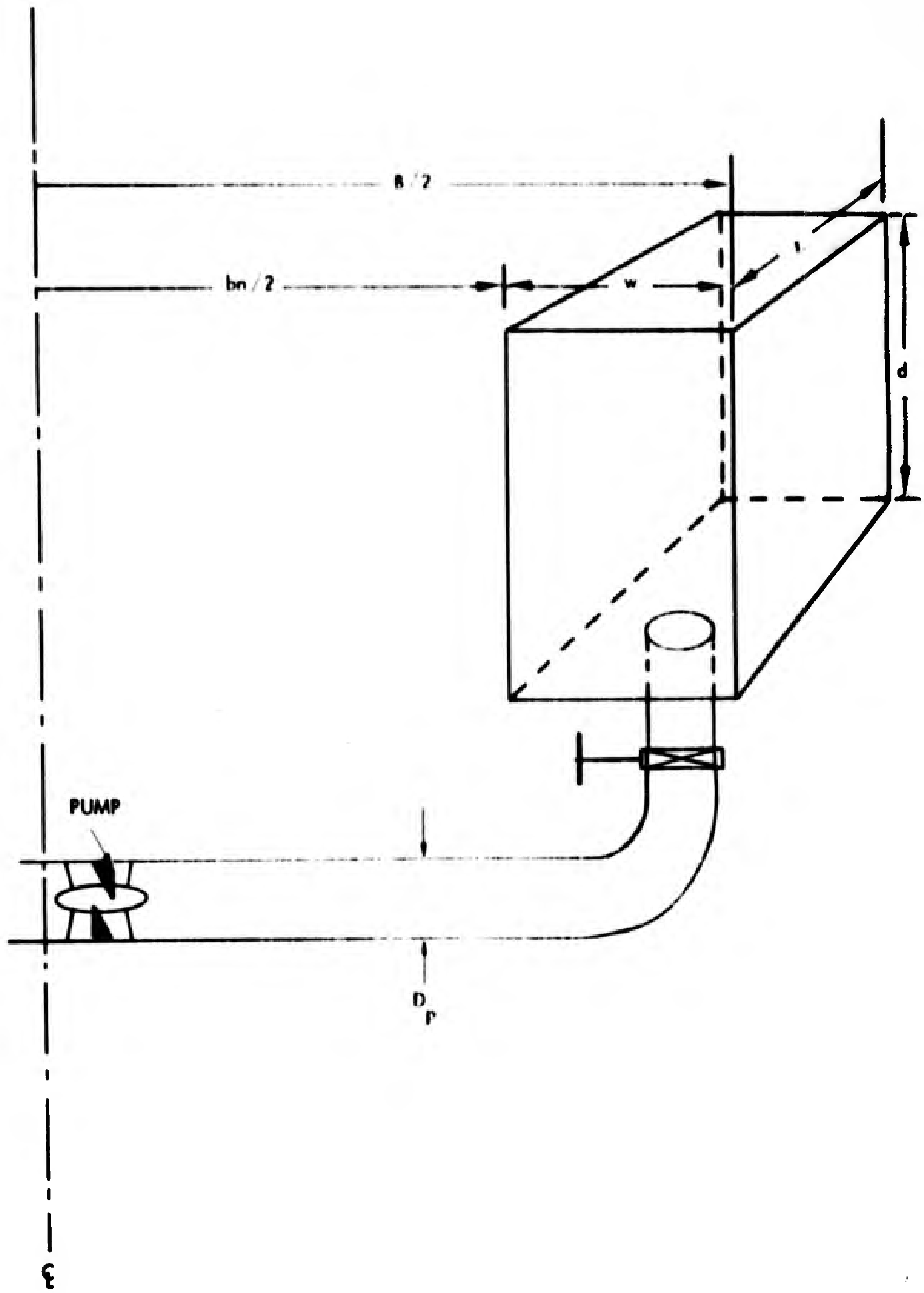



FIGURE 1 - NOMENCLATURE - SCHEMATIC HEELING TANK

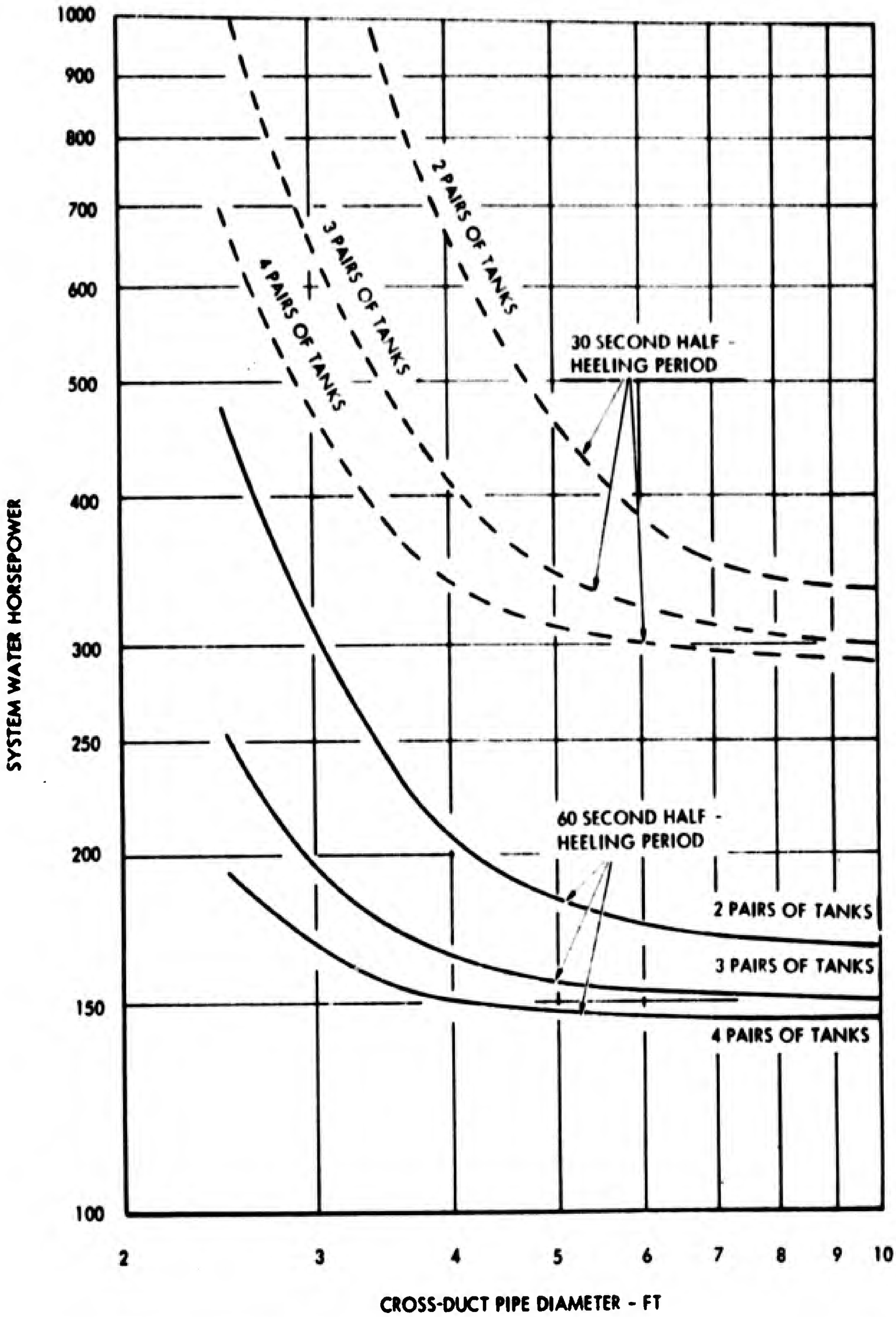



FIGURE 2 - VARIATION OF HEELING SYSTEM POWER WITH PIPE DIAMETER

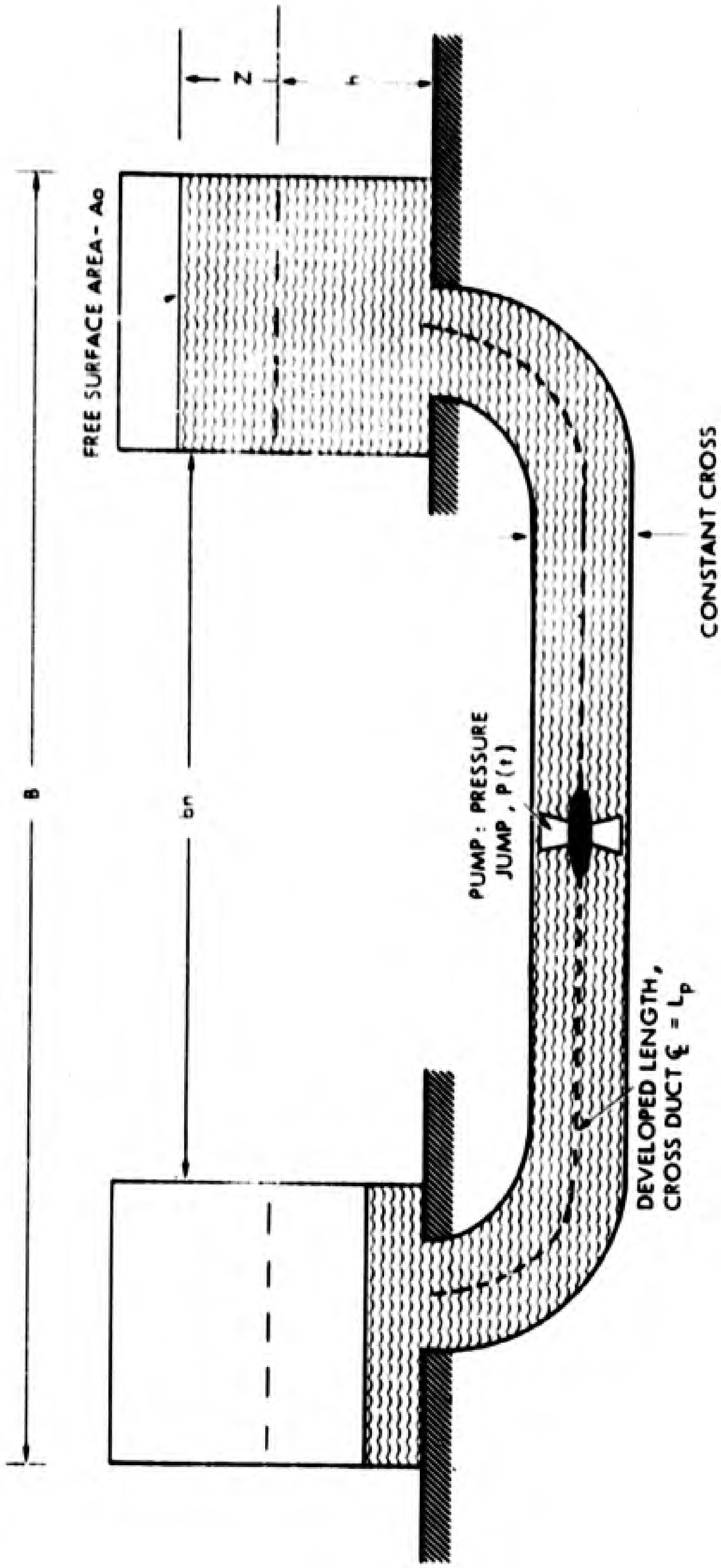



FIGURE 3 - NOTATION: HEELING TANK DYNAMICS

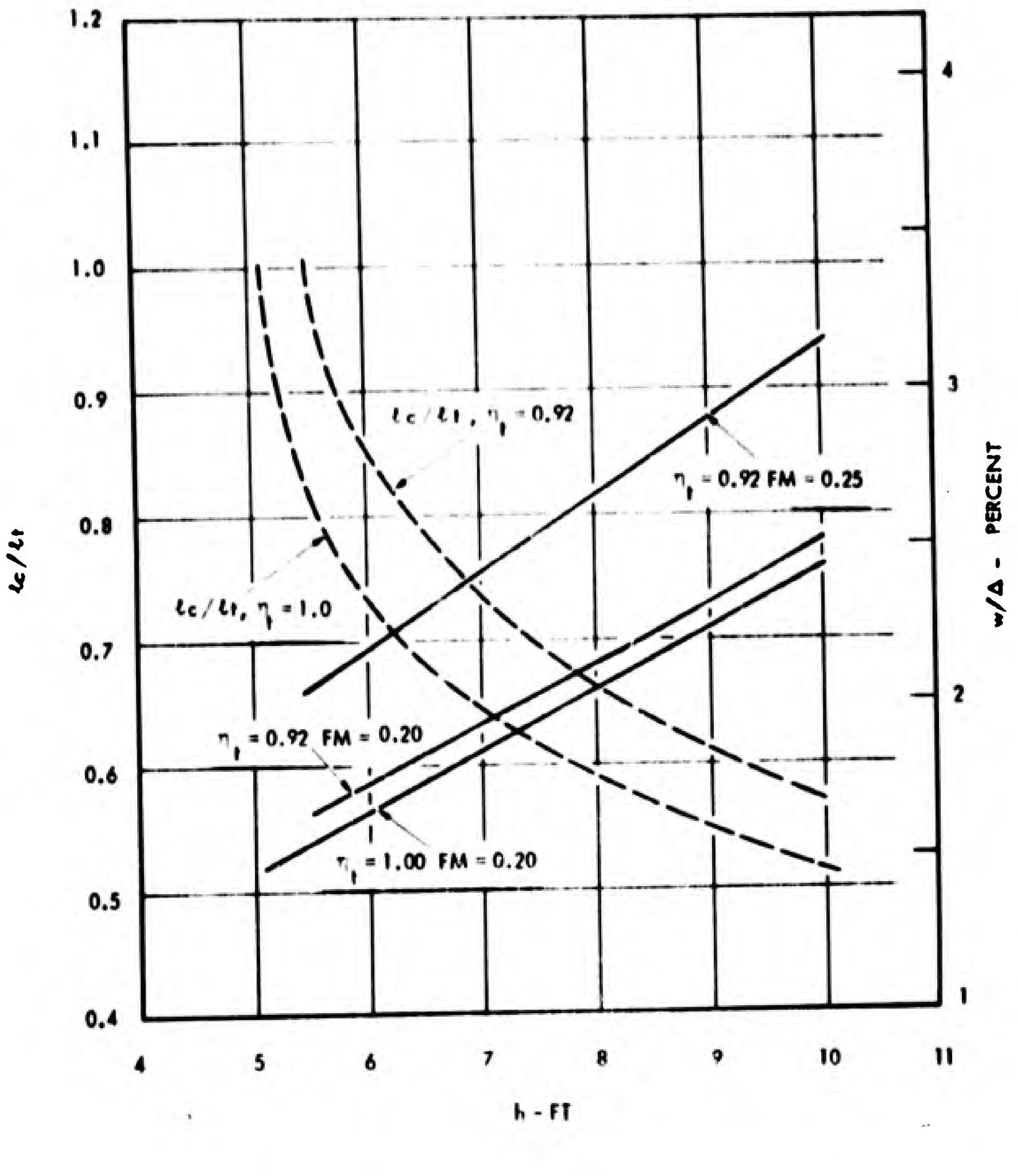



FIGURE 4 - SUMMARY OF "OPTIMUM" DESIGNS - TANK LENGTH 72 FEET, LOCATION 01 LEVEL, SHIP FREQUENCY - 0.533

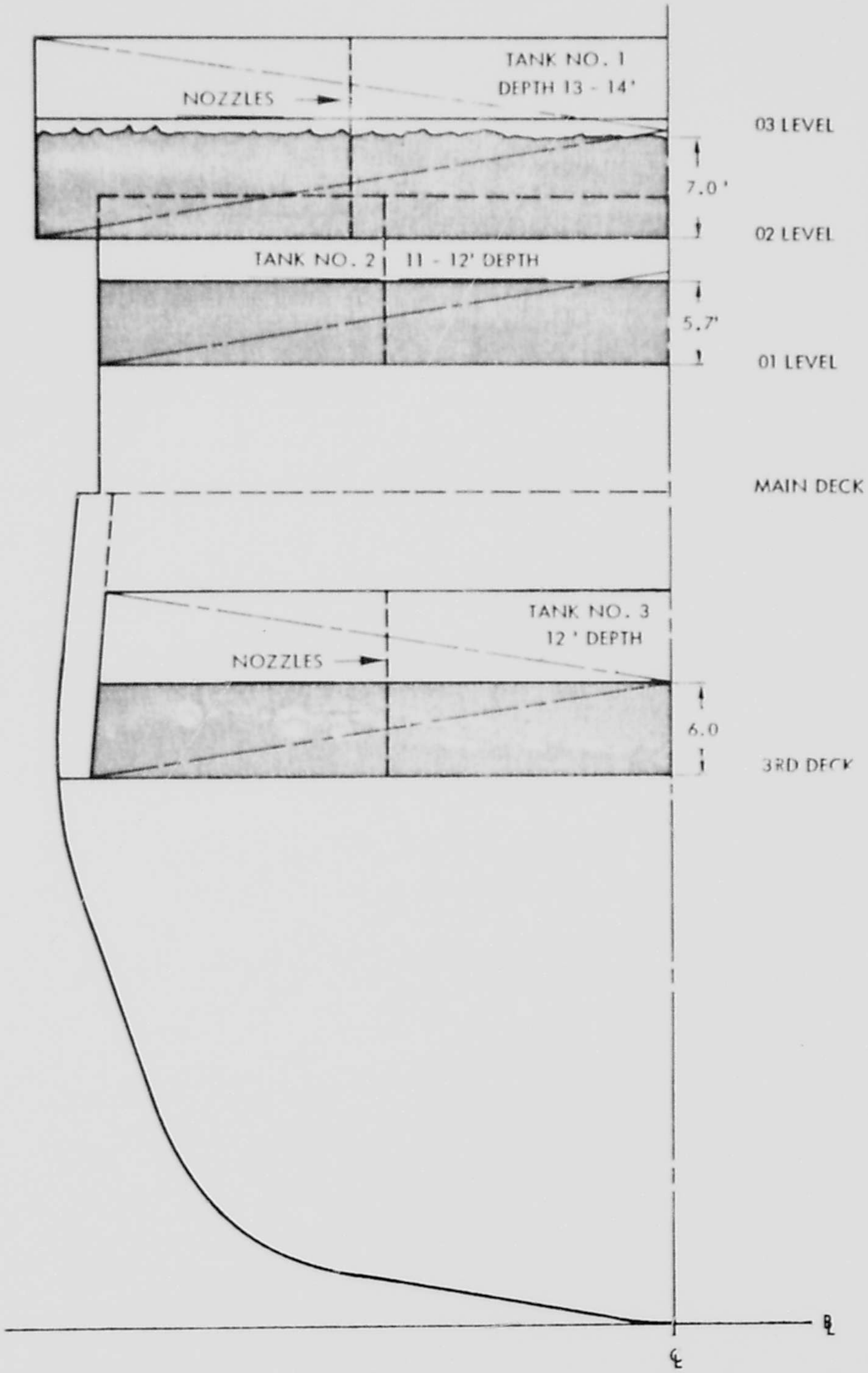



FIGURE 5 - FREE SURFACE TANK LOCATIONS

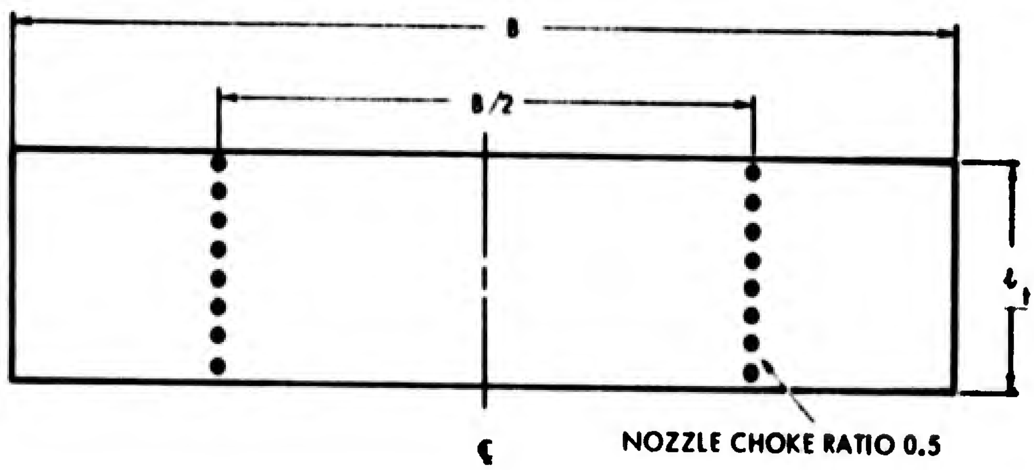
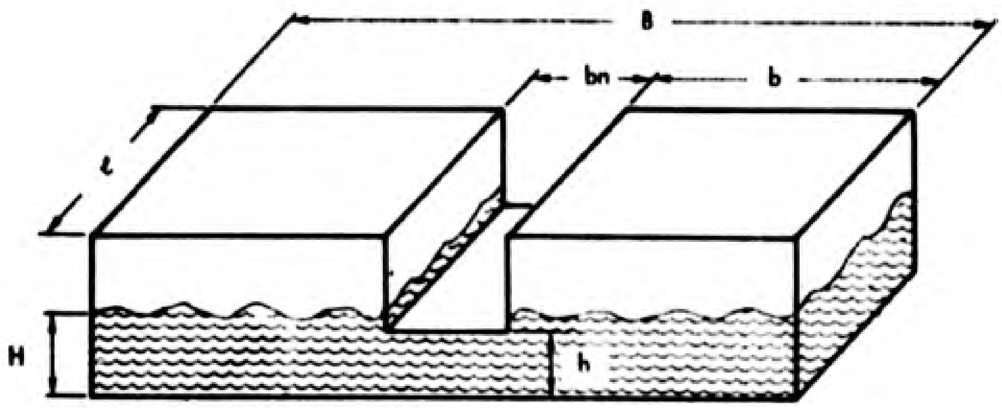



FIGURE 6 - TANK PLAN, TYPICAL FREE SURFACE TYPE




$h/H = 0.5$

$\ell$ = FORE - AFT TANK LENGTH

b = ATHWARTSHIP DIMENSION OF TANK LENGTH

bn = LENGTH OF CROSSOVER

FIGURE 7 - NOTATION: U-TUBE TANK

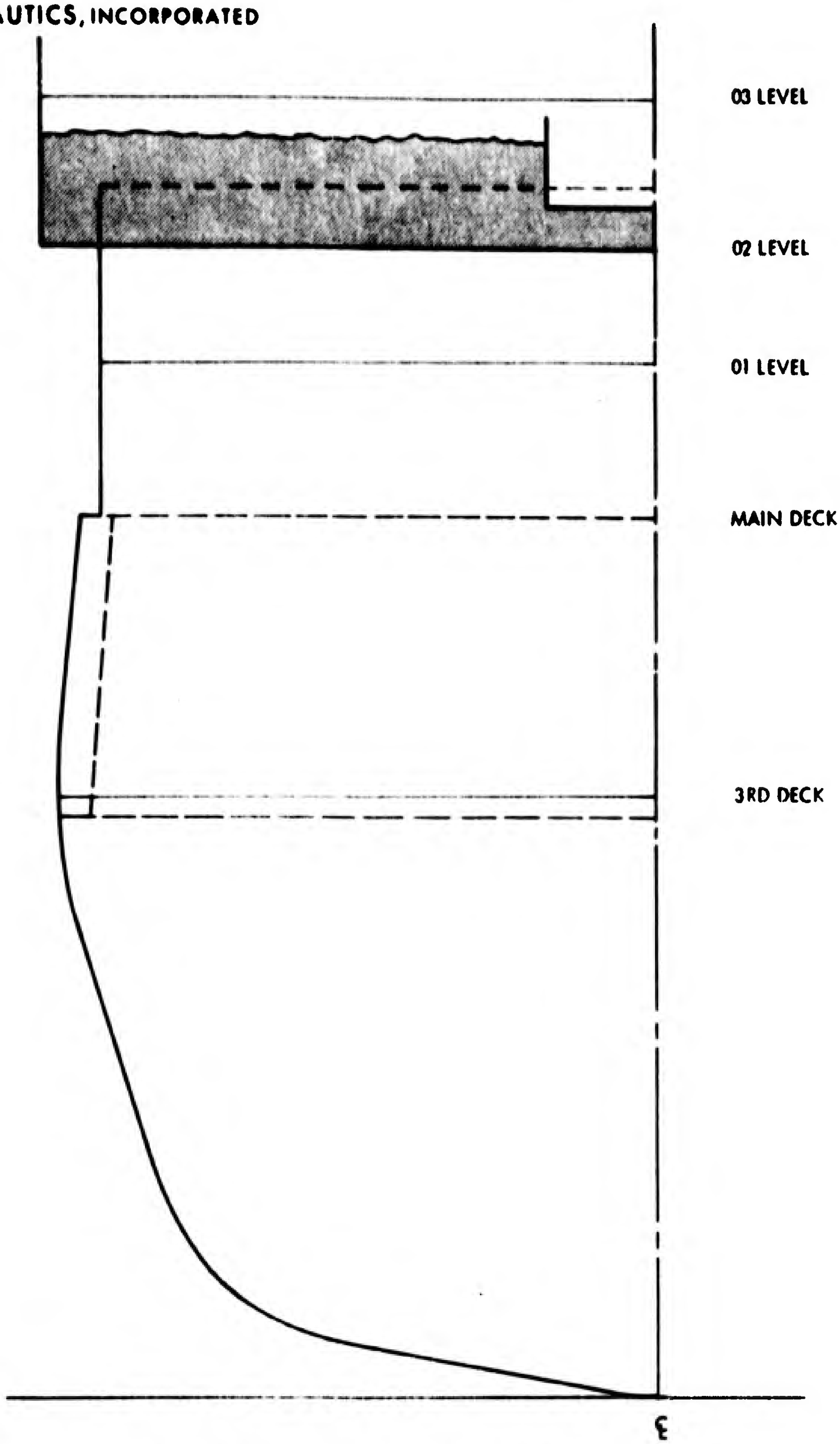



FIGURE 8 - TYPICAL U-TUBE TANK

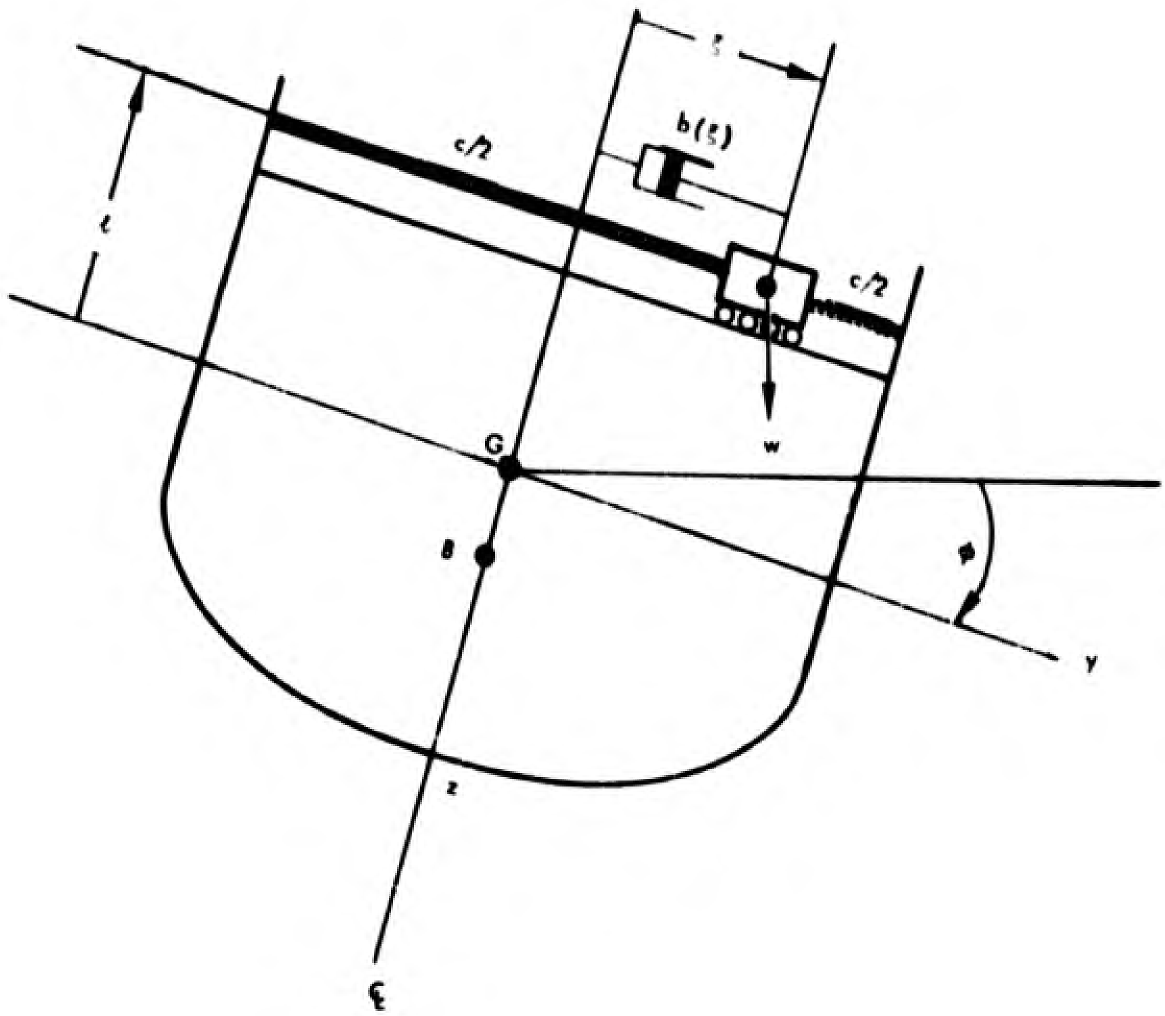



FIGURE 9 - PASSIVE MOVING WEIGHT STABILIZER SCHEMATIC

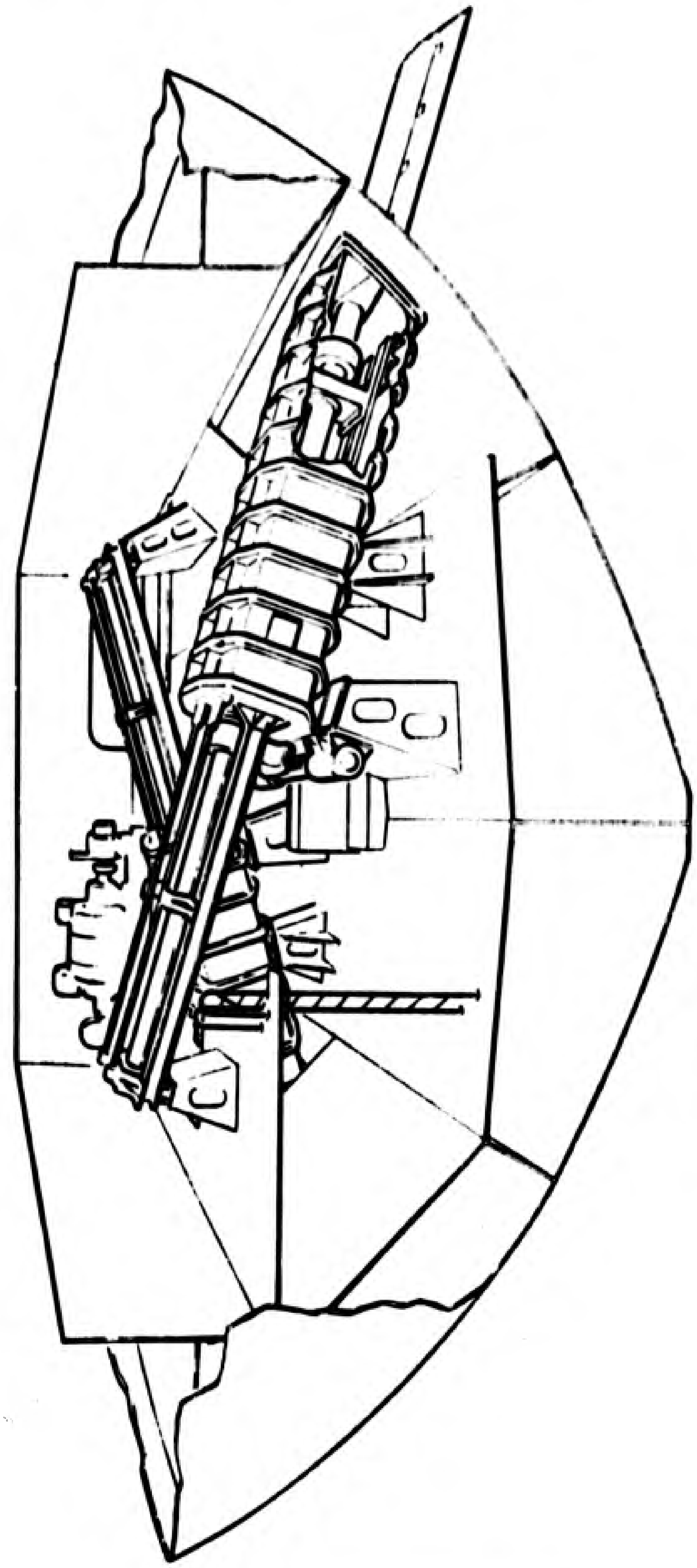


FIGURE 10 - ARRANGEMENT OF FIN STABILIZERS, LABRADOR (FROM REF. 7)

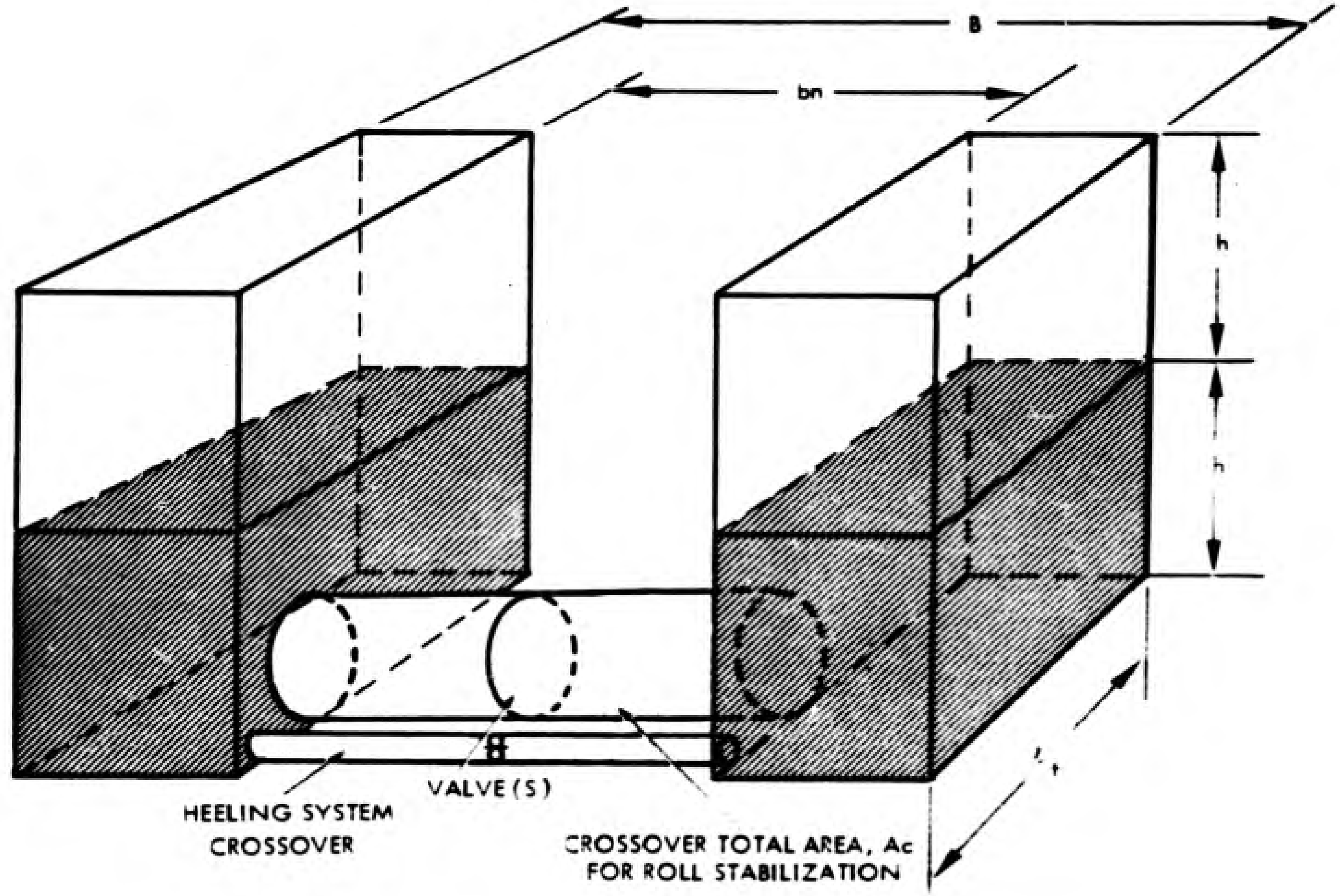



FIGURE 11 - CONCEPTUAL HEELING TANK, U-TUBE CONFIGURATION

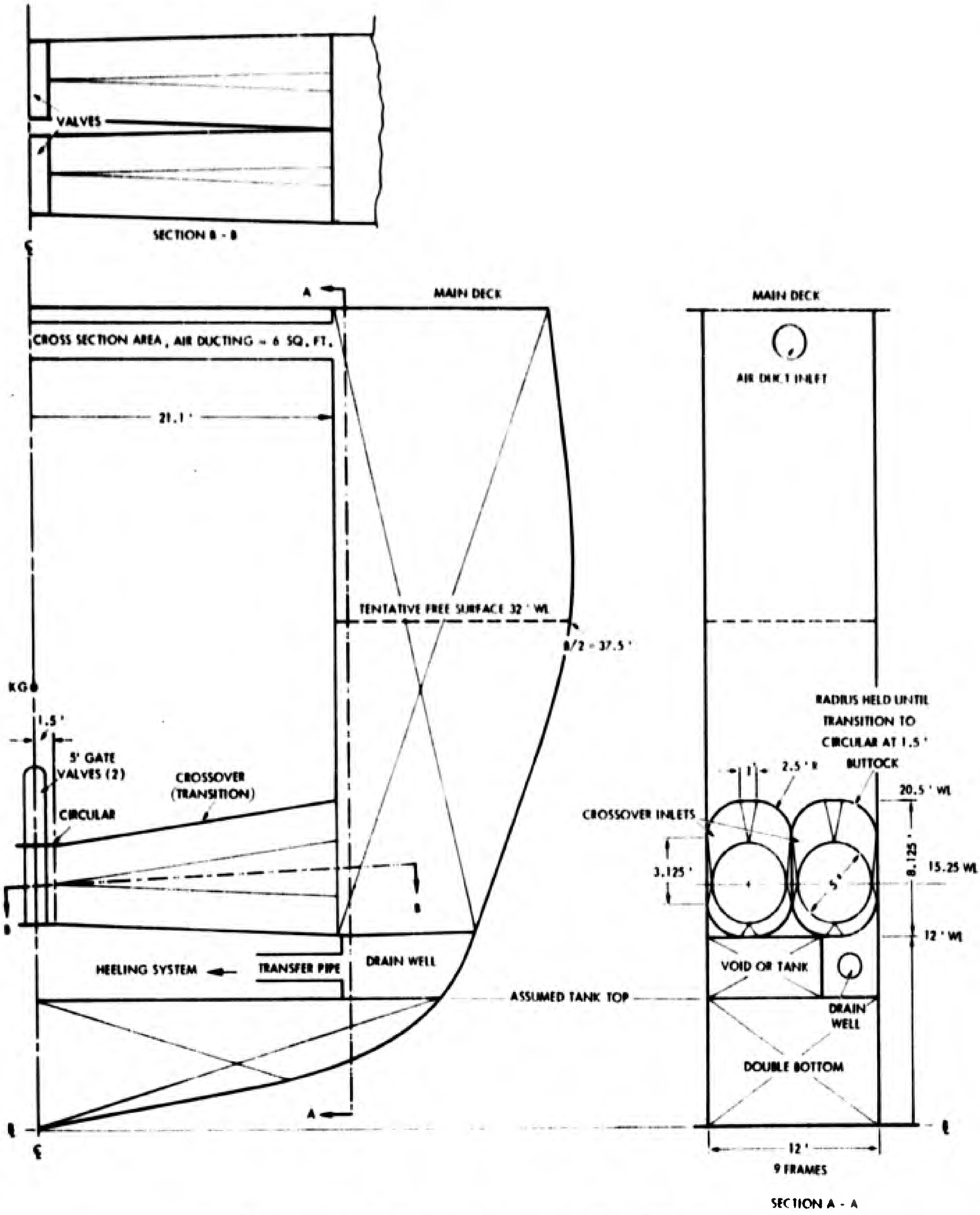



FIGURE 12 - COMBINATION HEELING U-TUBE STABILIZATION TANK (AFT TANK OF A SET OF TWO)

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June 27, 1967 S. B. FIELD 3,327,672

COMBINATION STABILIZATION AND HEELING SYSTEM FOR CARGO SHIPS, ICE BREAKERS, AND THE LIKE

Filed Sept. 23, 1965 3 Sheets-Sheet 1

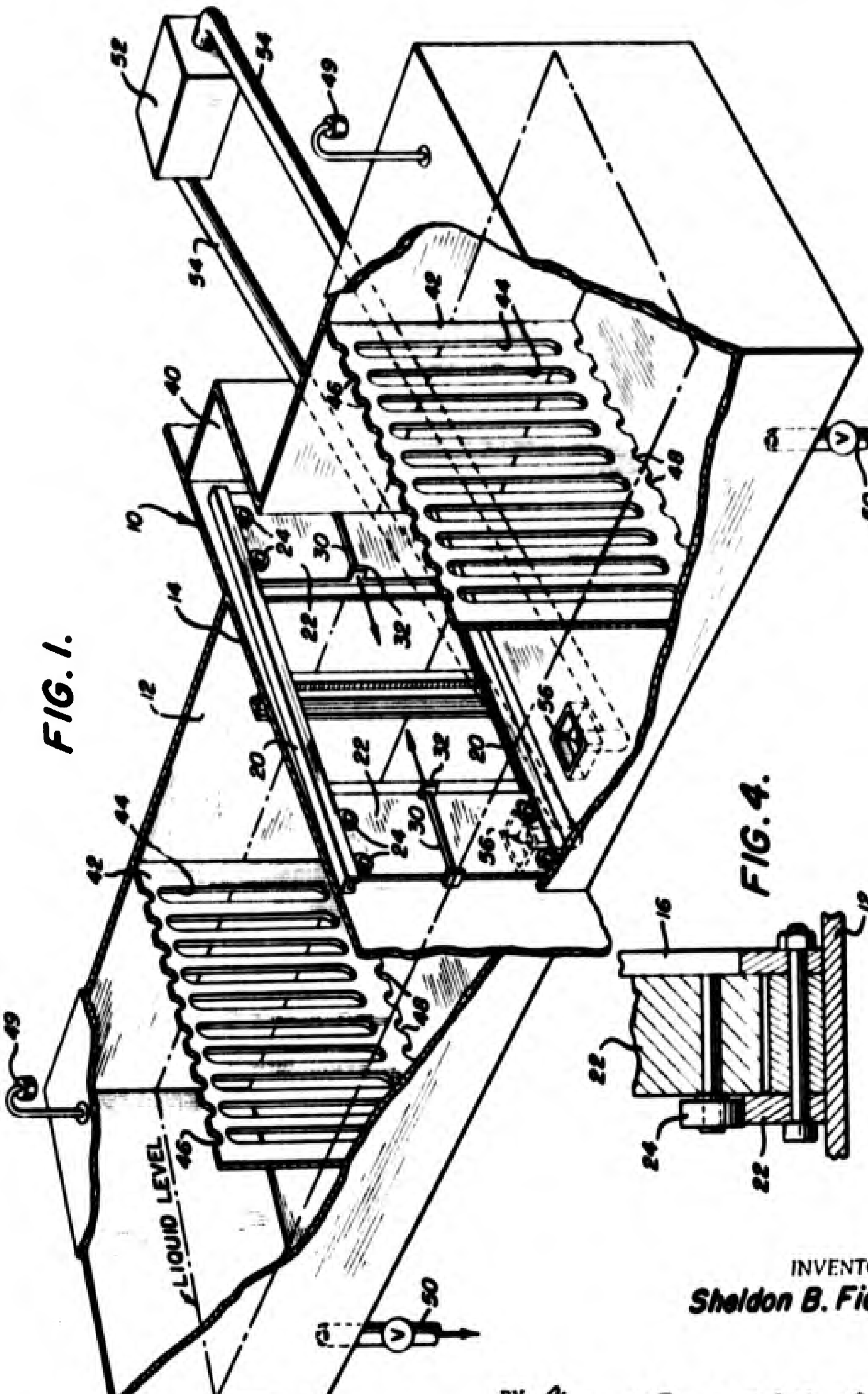


FIGURE 13 - COMBINED HEELING-FREE SURFACE TANK SYSTEM (FROM REF. 2)

INVENTOR
**Sheldon B. Field**

BY Stevens, Davis, Miller & Mosher
ATTORNEYS

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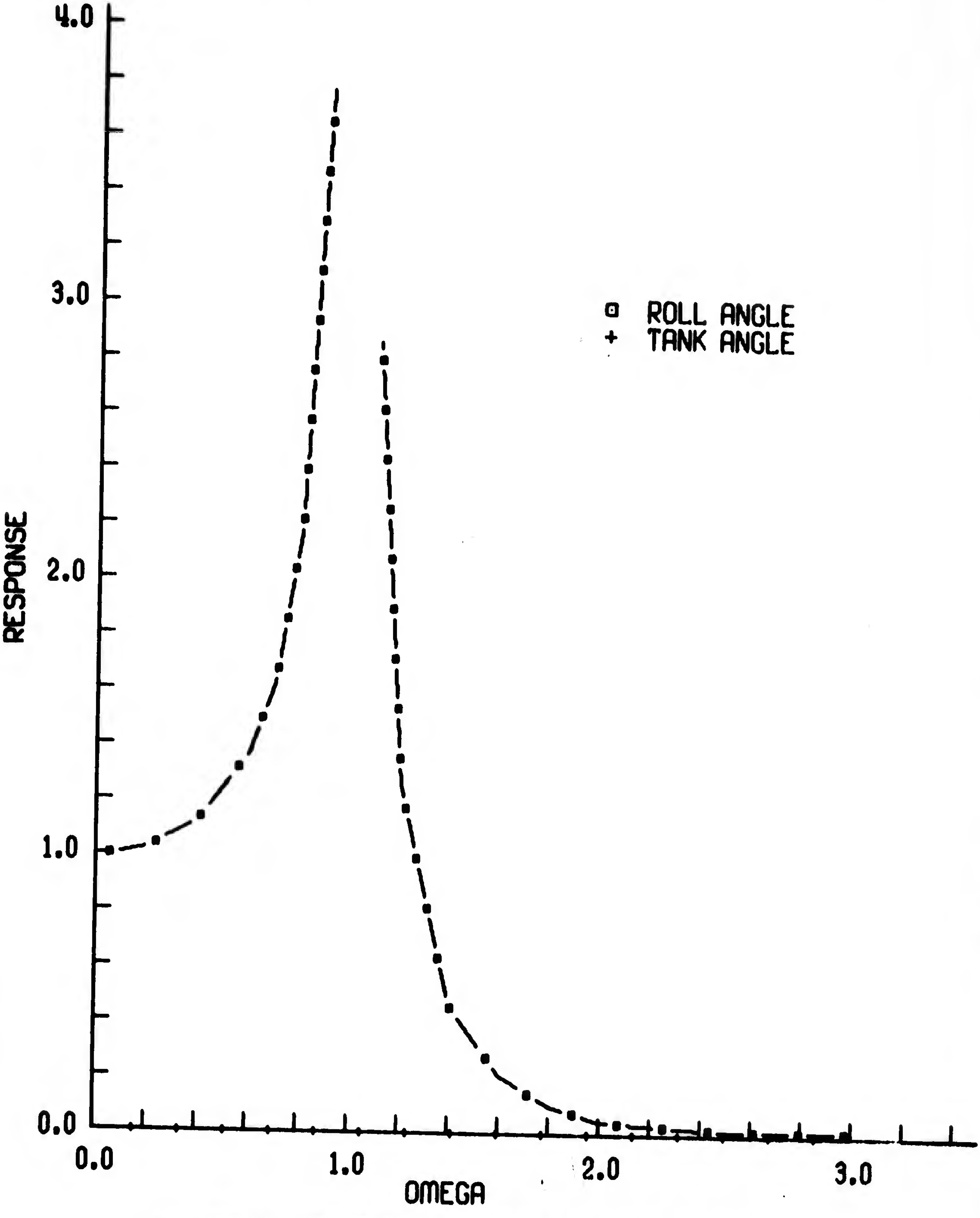


FIGURE 14 – RESPONSE TO WAVE OF UNIT SLOPE
351FT. POLAR ICEBREAKER - 17KT.
NO STABILIZER

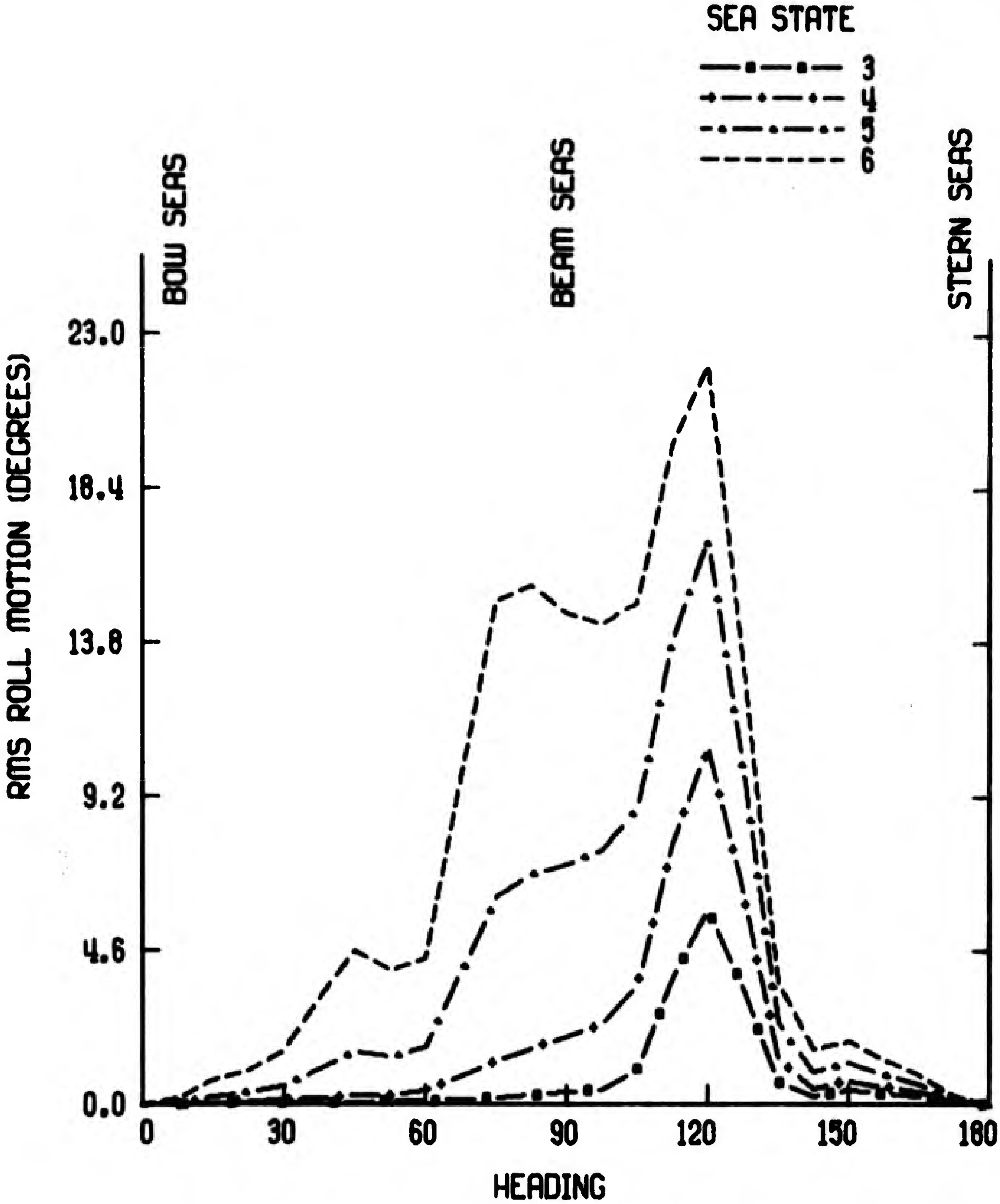


FIGURE 15 -RMS ROLL MOTION VS HEADING INTO LONG CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
NO STABILIZER

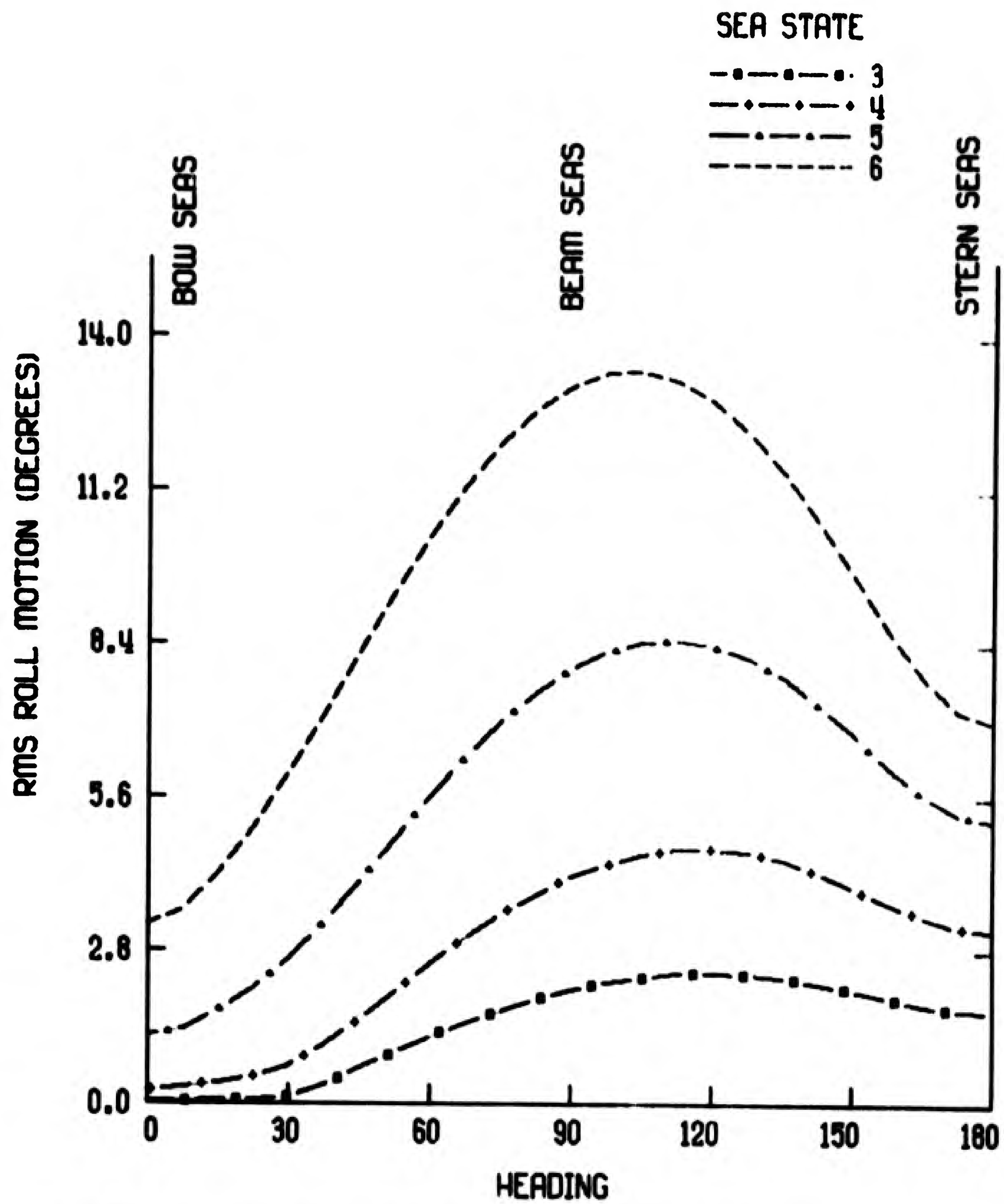


FIGURE 16 - RMS ROLL MOTION VS. HEADING INTO SHORT CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
NO STABILIZER

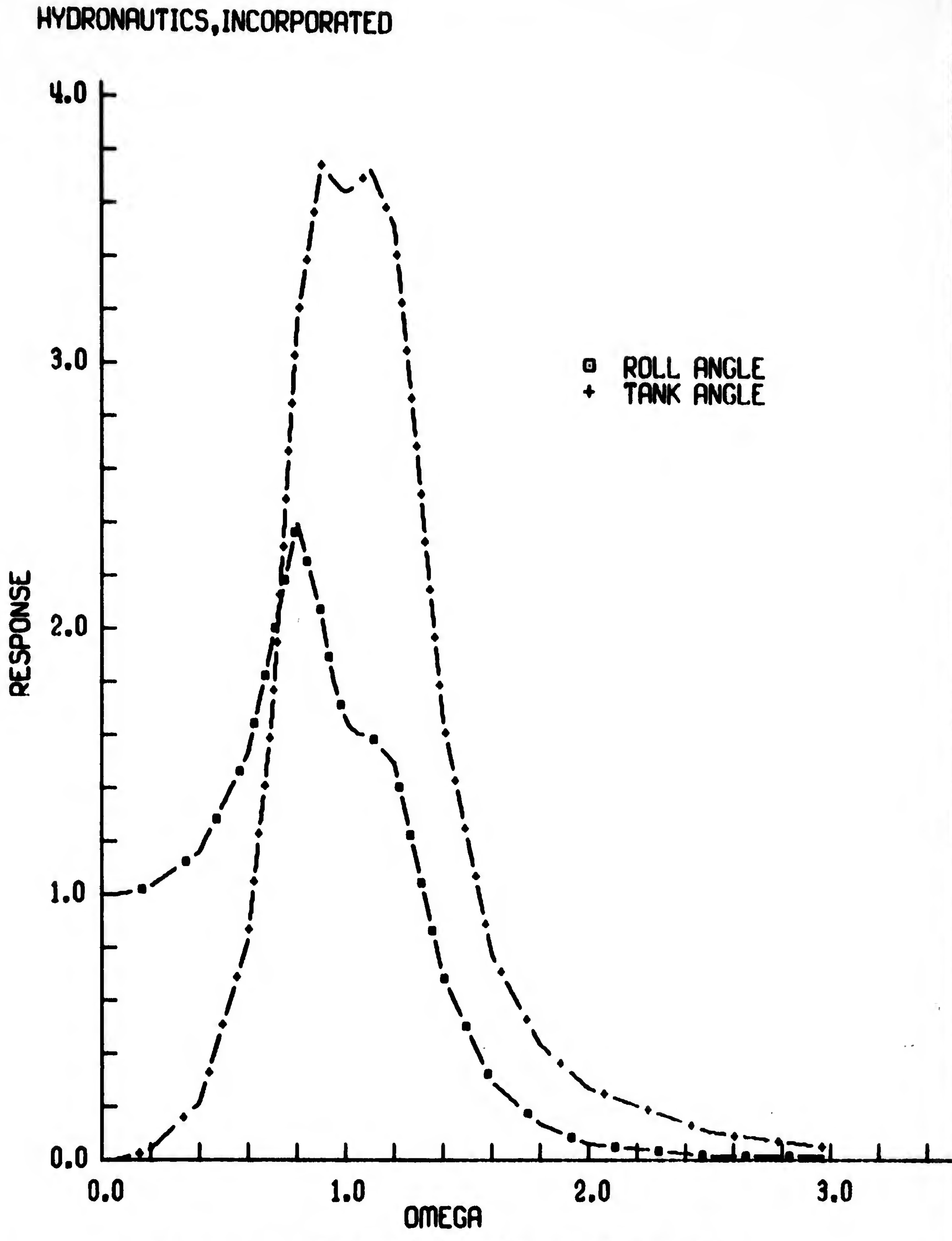


FIGURE 17- RESPONSE TO WAVE OF UNIT SLOPE
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK OR U-TUBE ON 02 LEVEL

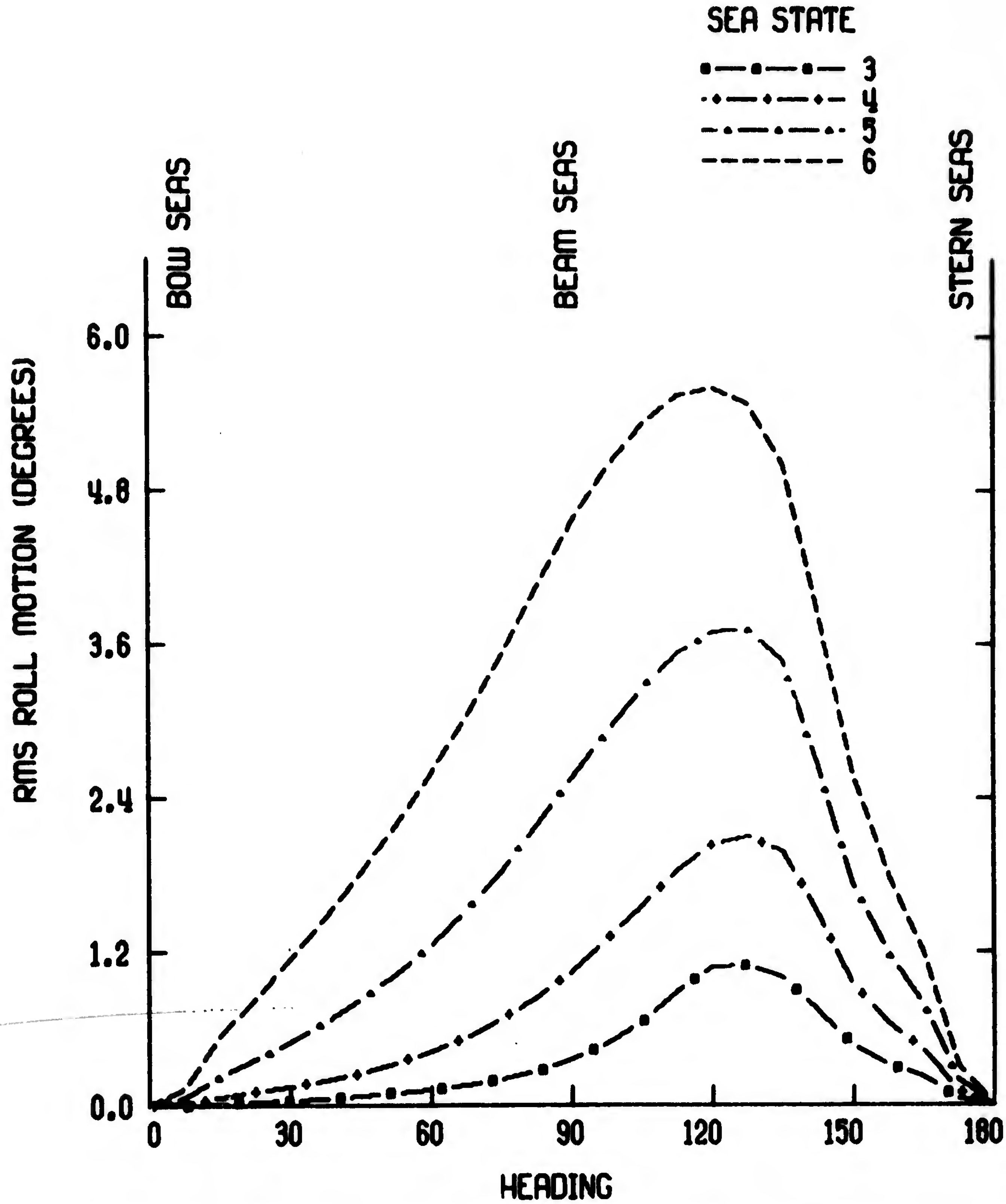


FIGURE 18 - RMS ROLL MOTION VS HEADING INTO LONG CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK OR U-TUBE ON 02 LEVEL

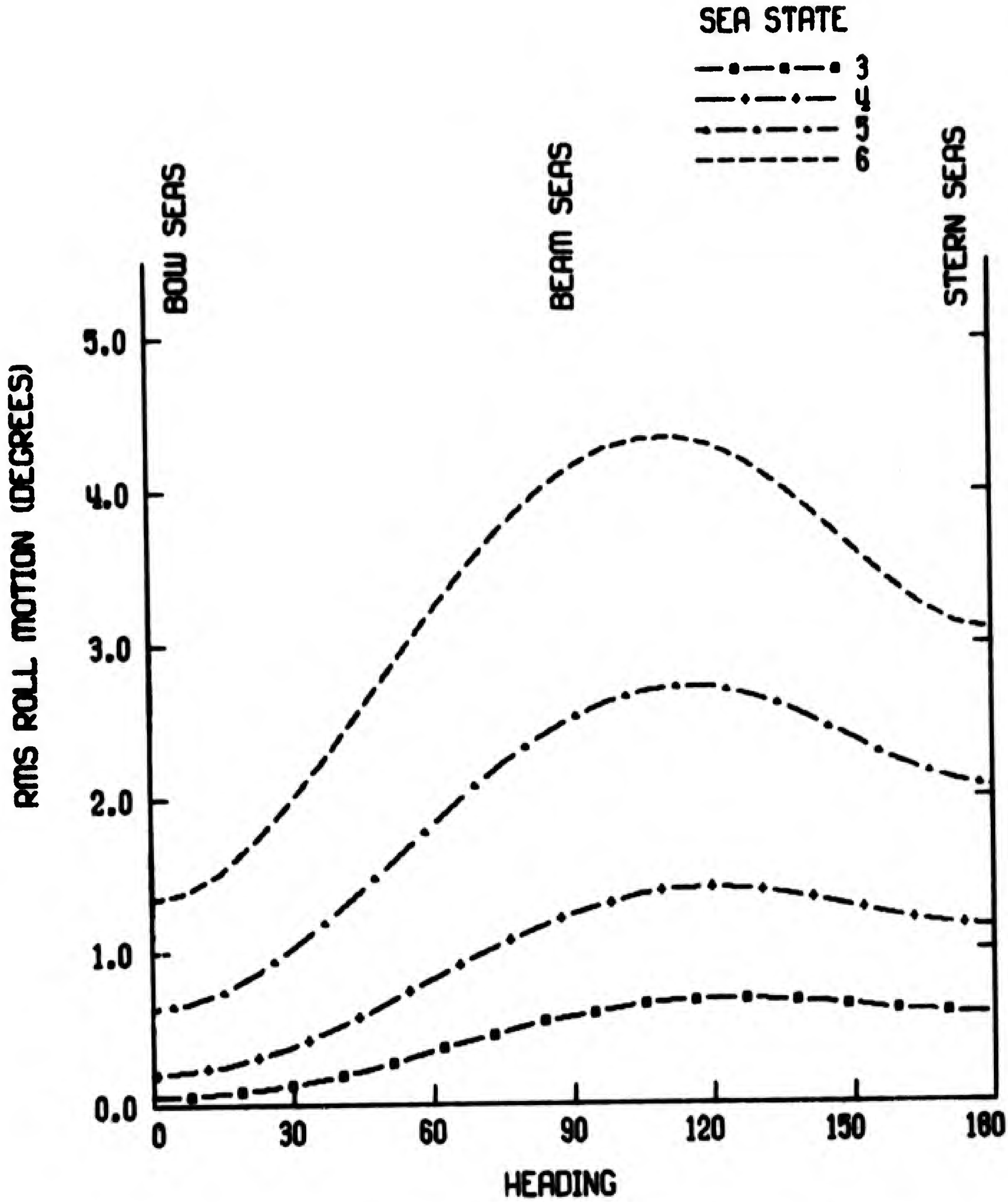


FIGURE 19 - RMS ROLL MOTION VS. HEADING INTO SHORT CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK OR U-TUBE ON 02 LEVEL

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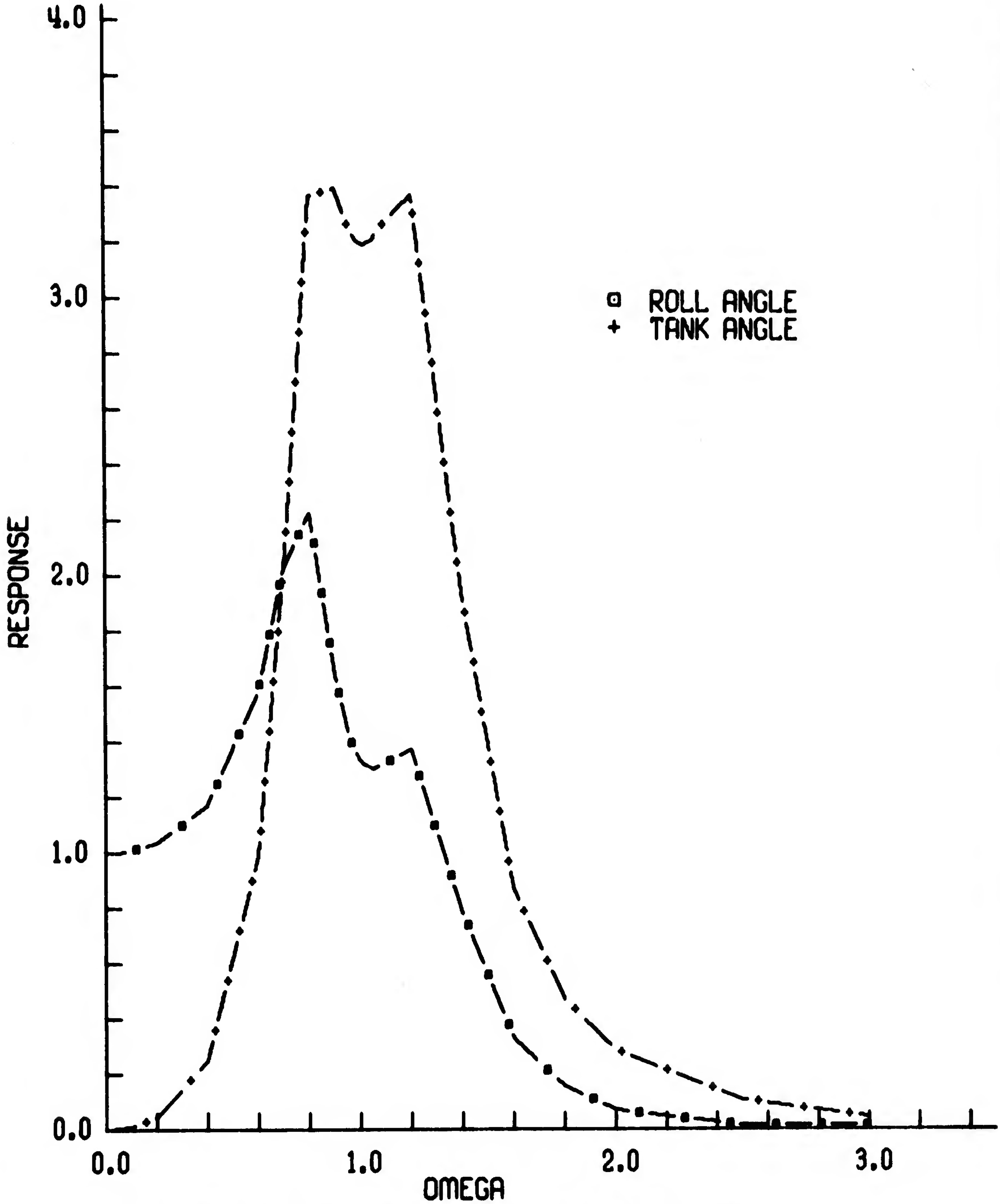


FIGURE 20 – RESPONSE TO WAVE OF UNIT SLOPE
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK ON 01 LEVEL

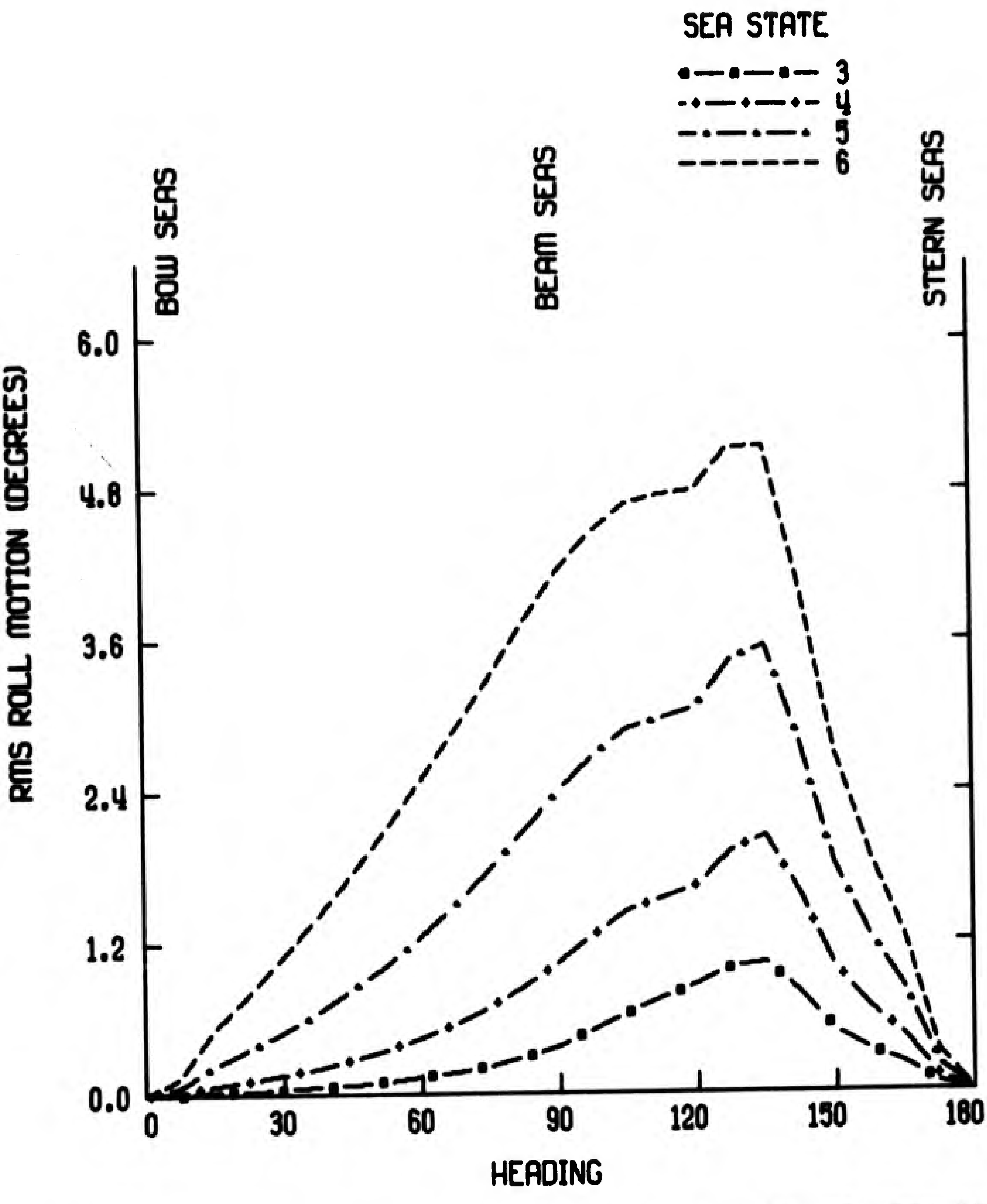


FIGURE 21 - RMS ROLL MOTION VS HEADING INTO LONG CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK ON 01 LEVEL

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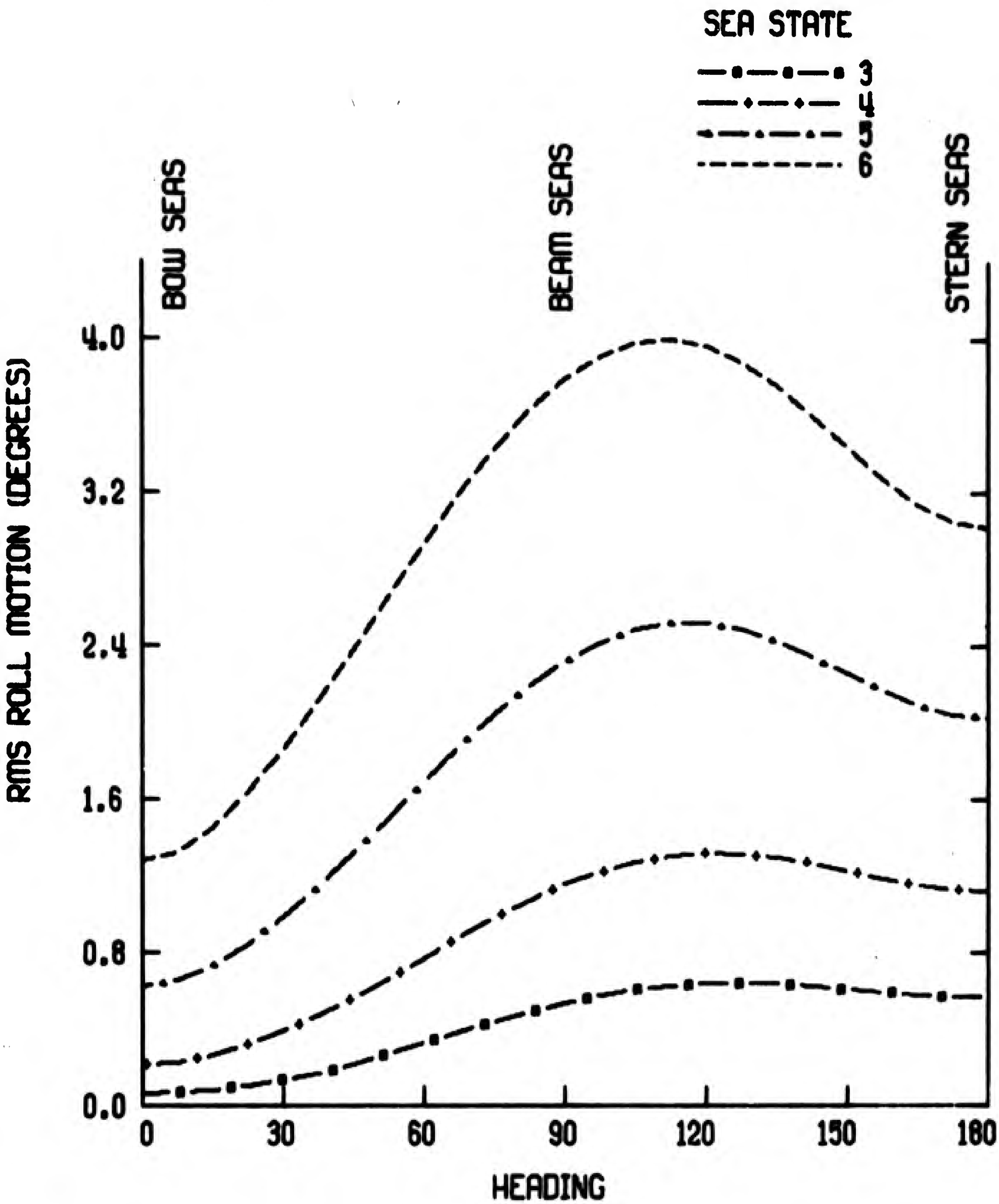


FIGURE 22-RMS ROLL MOTION VS. HEADING INTO SHORT CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK ON 01 LEVEL

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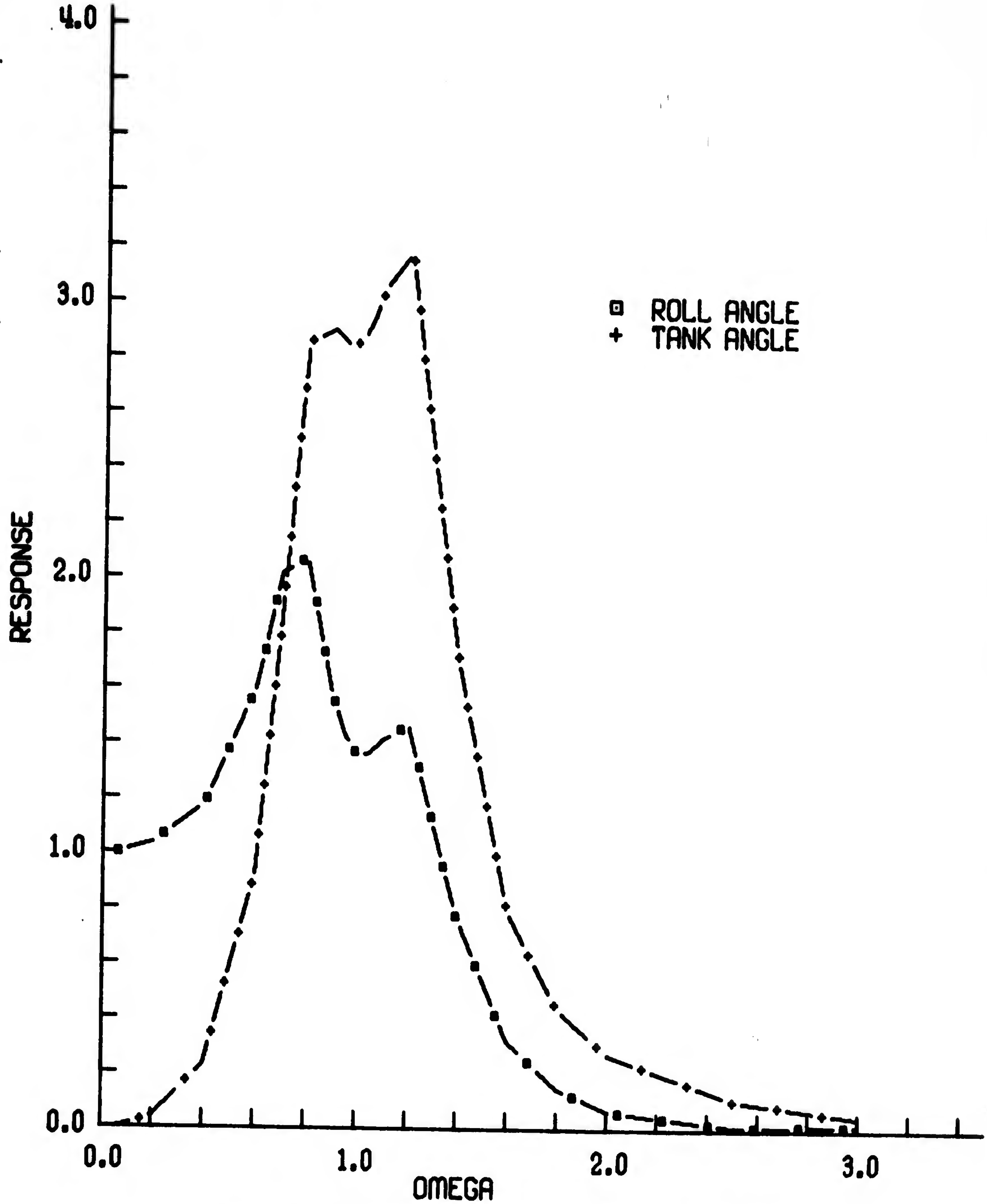


FIGURE 23 – RESPONSE TO WAVE OF UNIT SLOPE
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK ON THIRD DECK

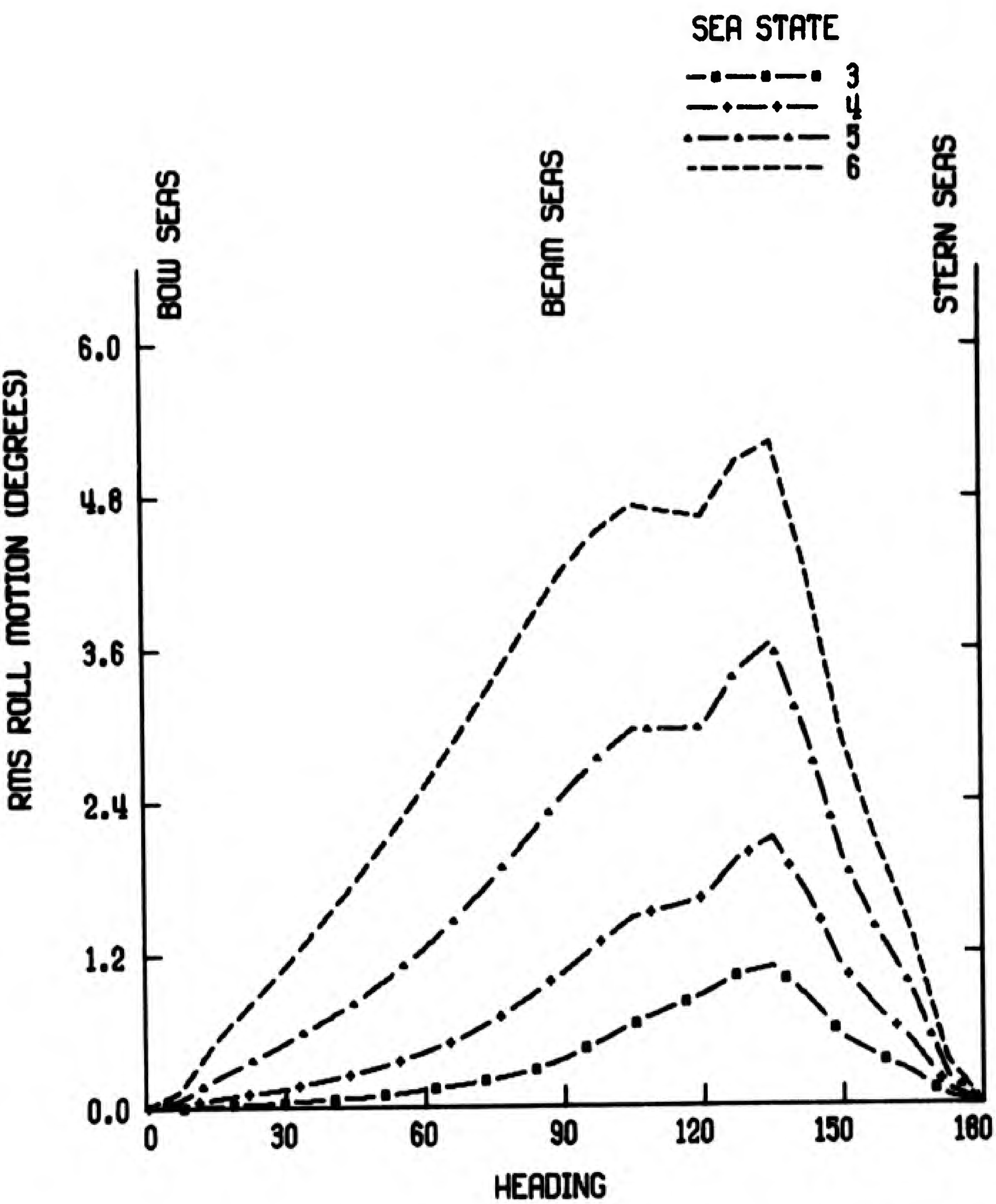


FIGURE 24 - RMS ROLL MOTION VS HEADING INTO LONG CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK ON THIRD DECK

HYDRONAUTICS, INCORPORATED

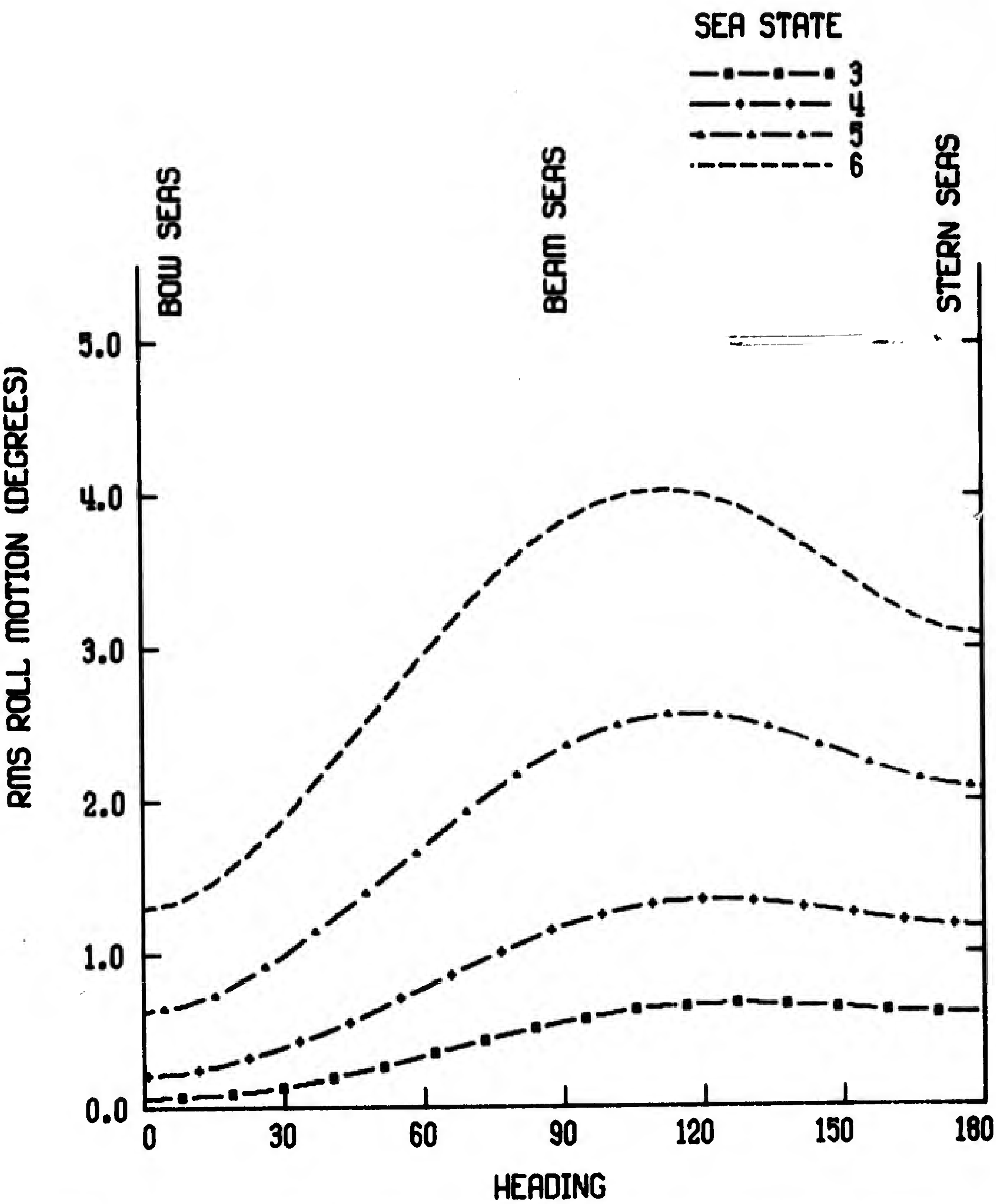


FIGURE 25 - RMS ROLL MOTION VS. HEADING INTO SHORT CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
FLUME TANK ON THIRD DECK

HYDRONAUTICS, INCORPORATED

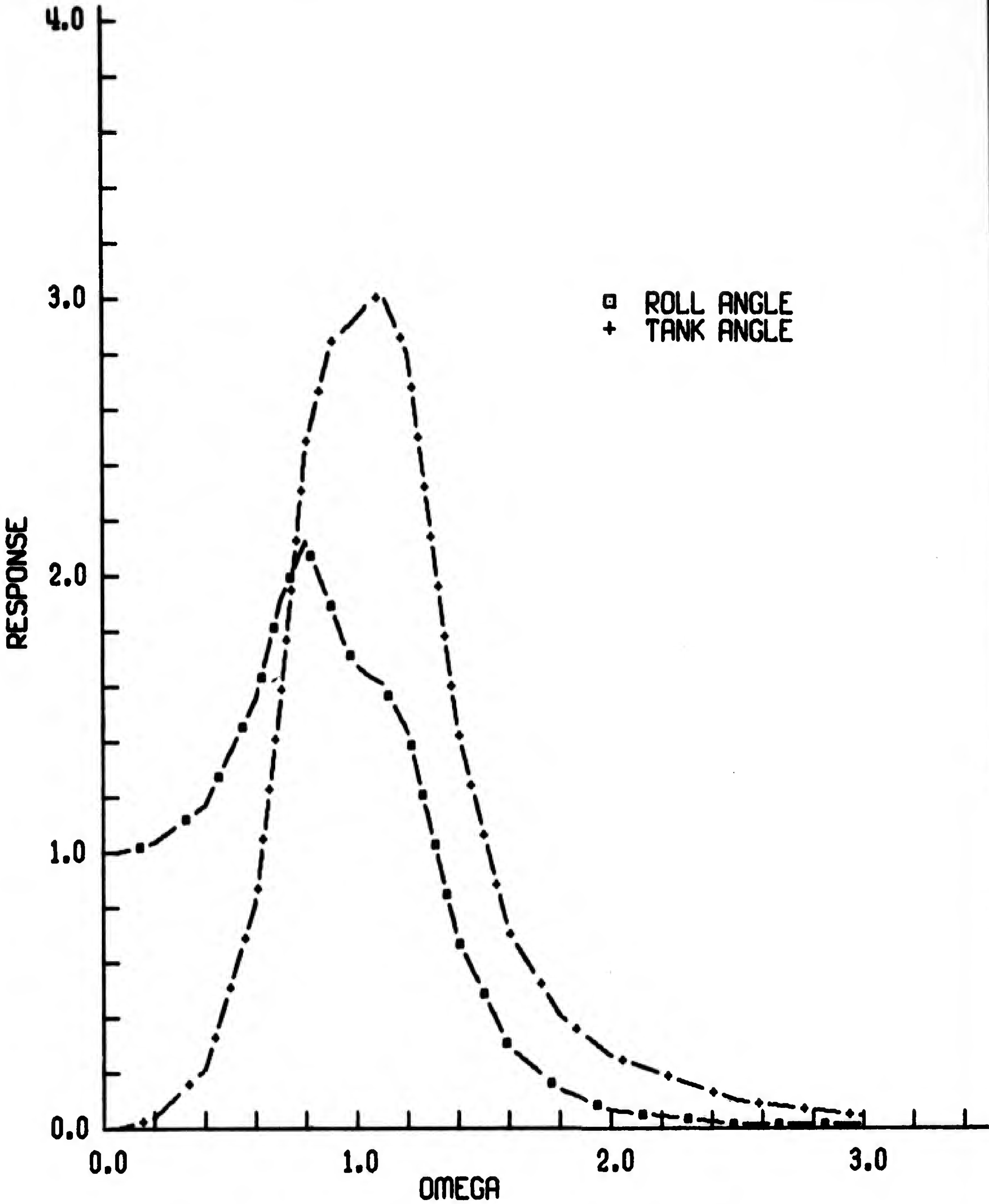


FIGURE 26 – RESPONSE TO WAVE OF UNIT SLOPE
351FT. POLAR ICEBREAKER - 17KT.
COMBINED HEELING-ANTI-ROLL TANKS

HYDRONAUTICS, INCORPORATED

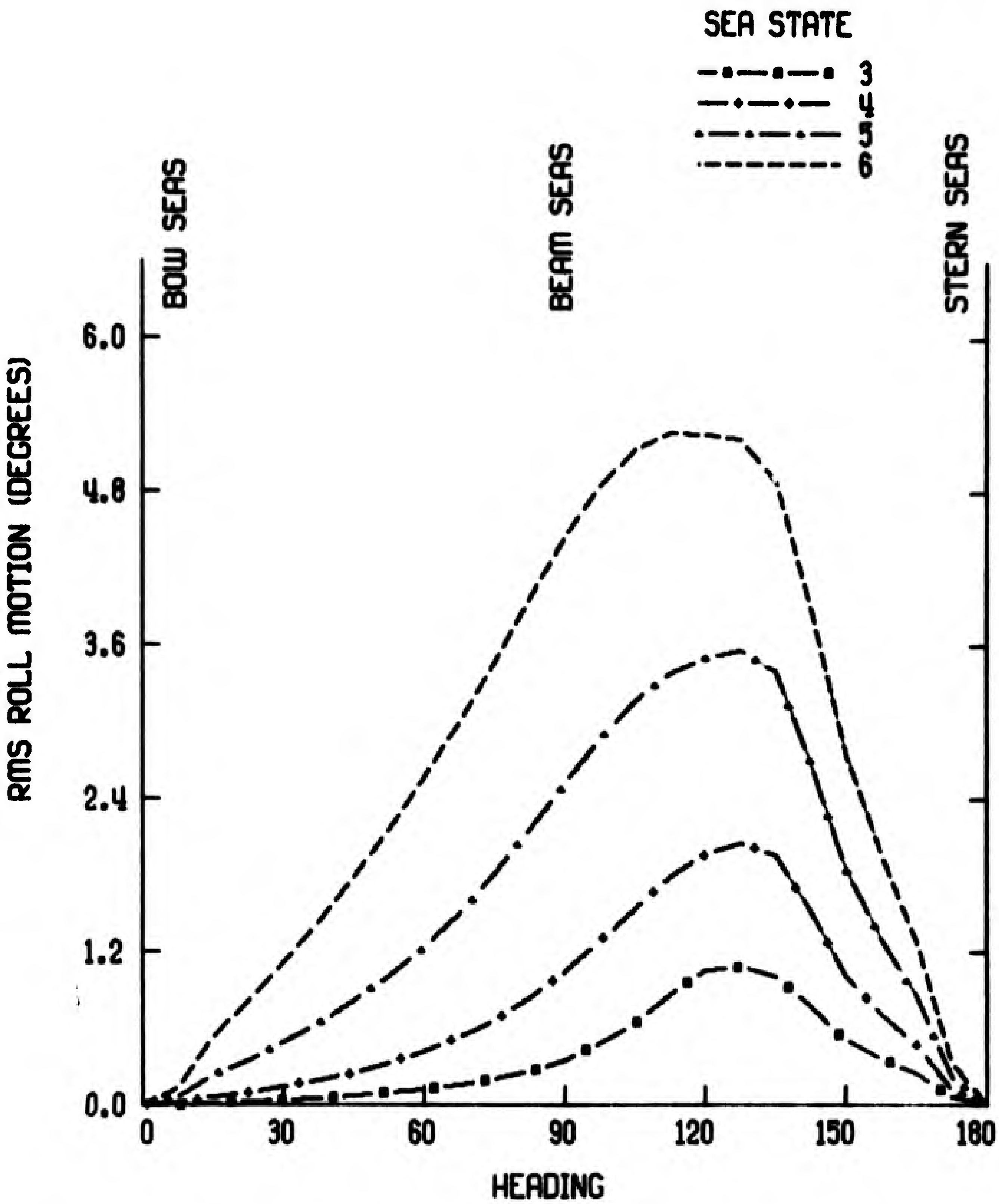


FIGURE 27 – RMS ROLL MOTION VS HEADING INTO LONG CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
COMBINED HEELING-ANTI-ROLL TANKS

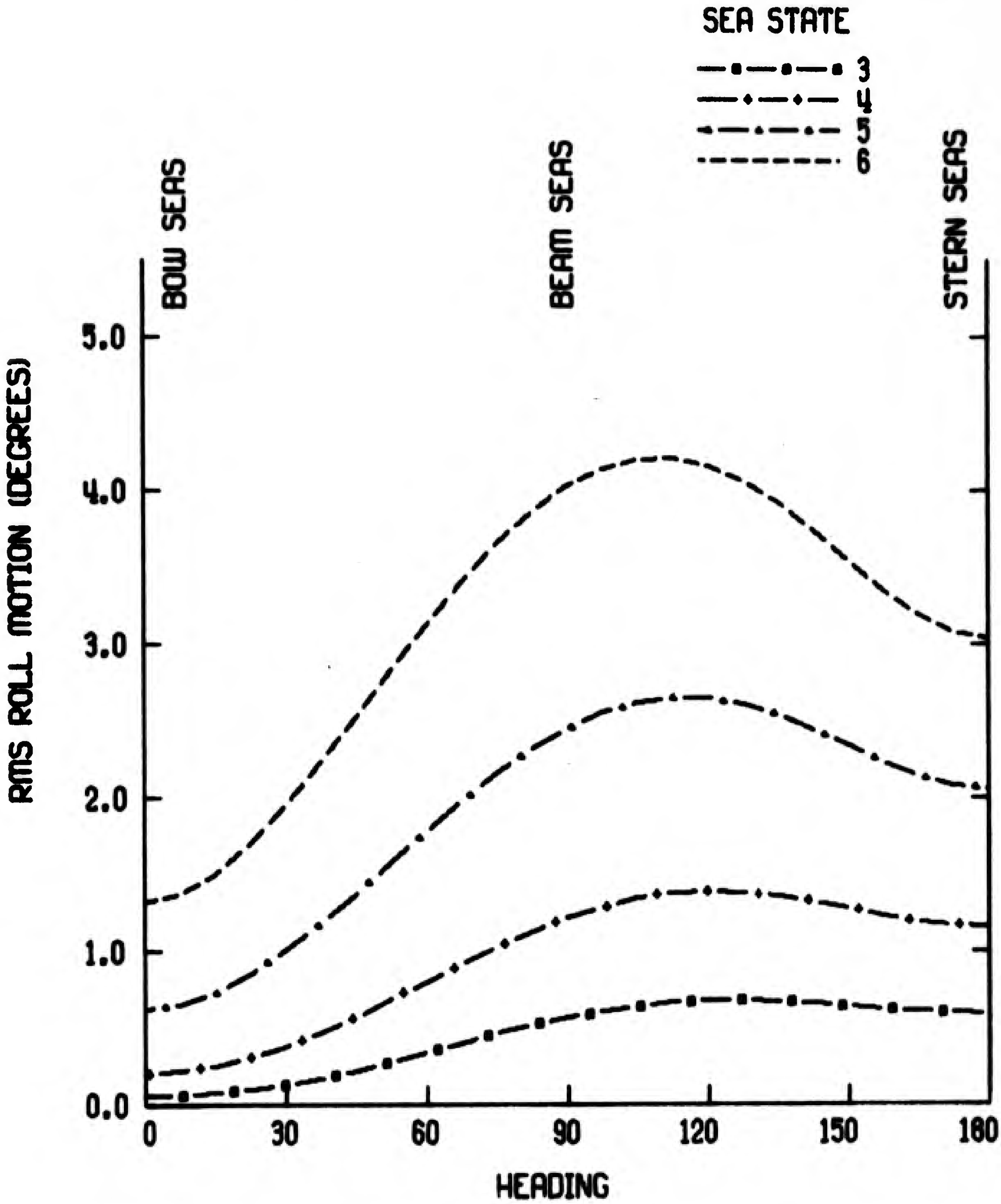


FIGURE 28 – RMS ROLL MOTION VS. HEADING INTO SHORT CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
COMBINED HEELING-ANTI-ROLL TANKS

HYDRONAUTICS, INCORPORATED

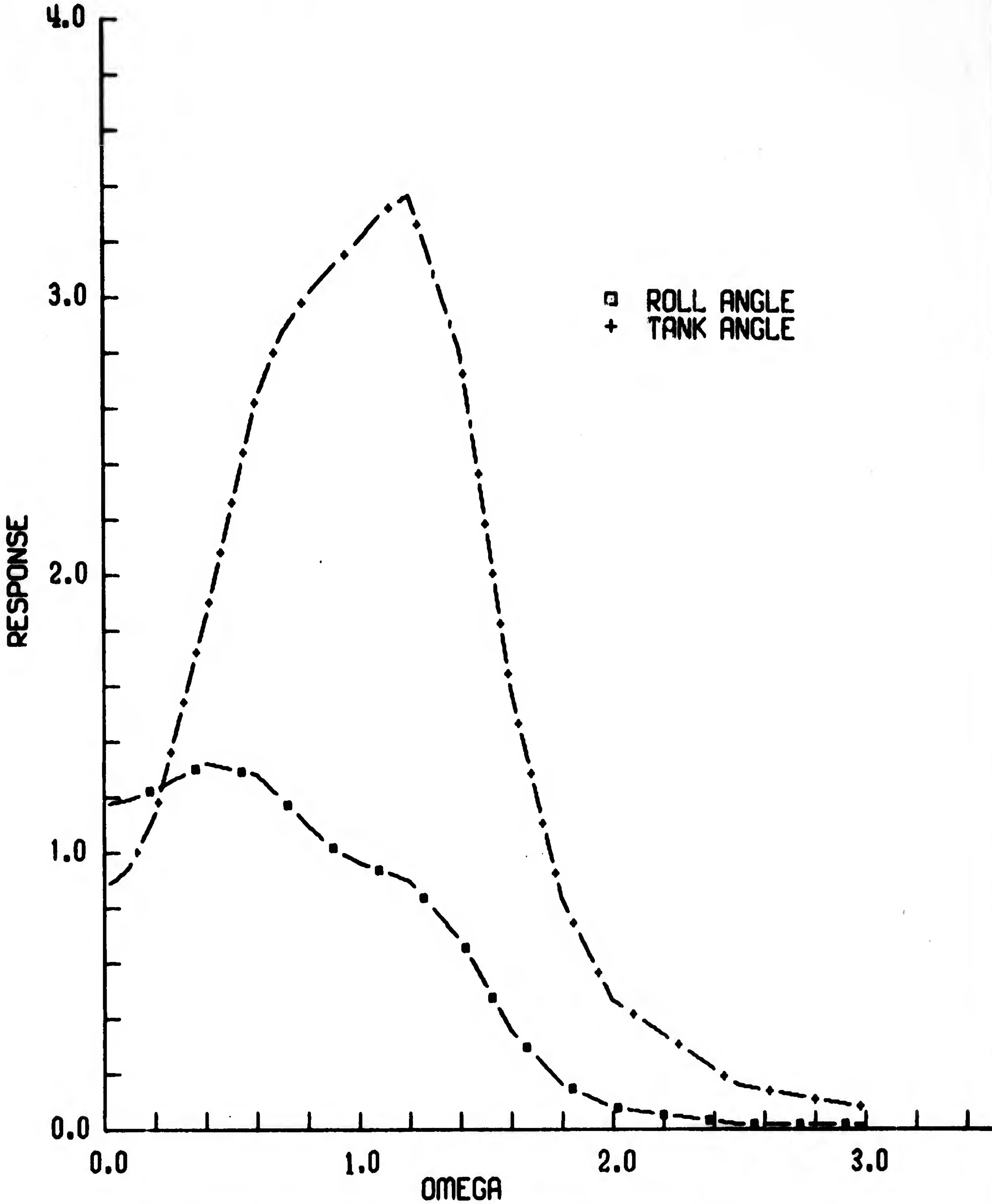


FIGURE 29 - RESPONSE TO WAVE OF UNIT SLOPE
351FT. POLAR ICEBREAKER - 17KT.
ACTIVE HEELING-ANTI-ROLL TANKS
(G1 = -0.60 G2 = 0.30 G3 = 3.00)

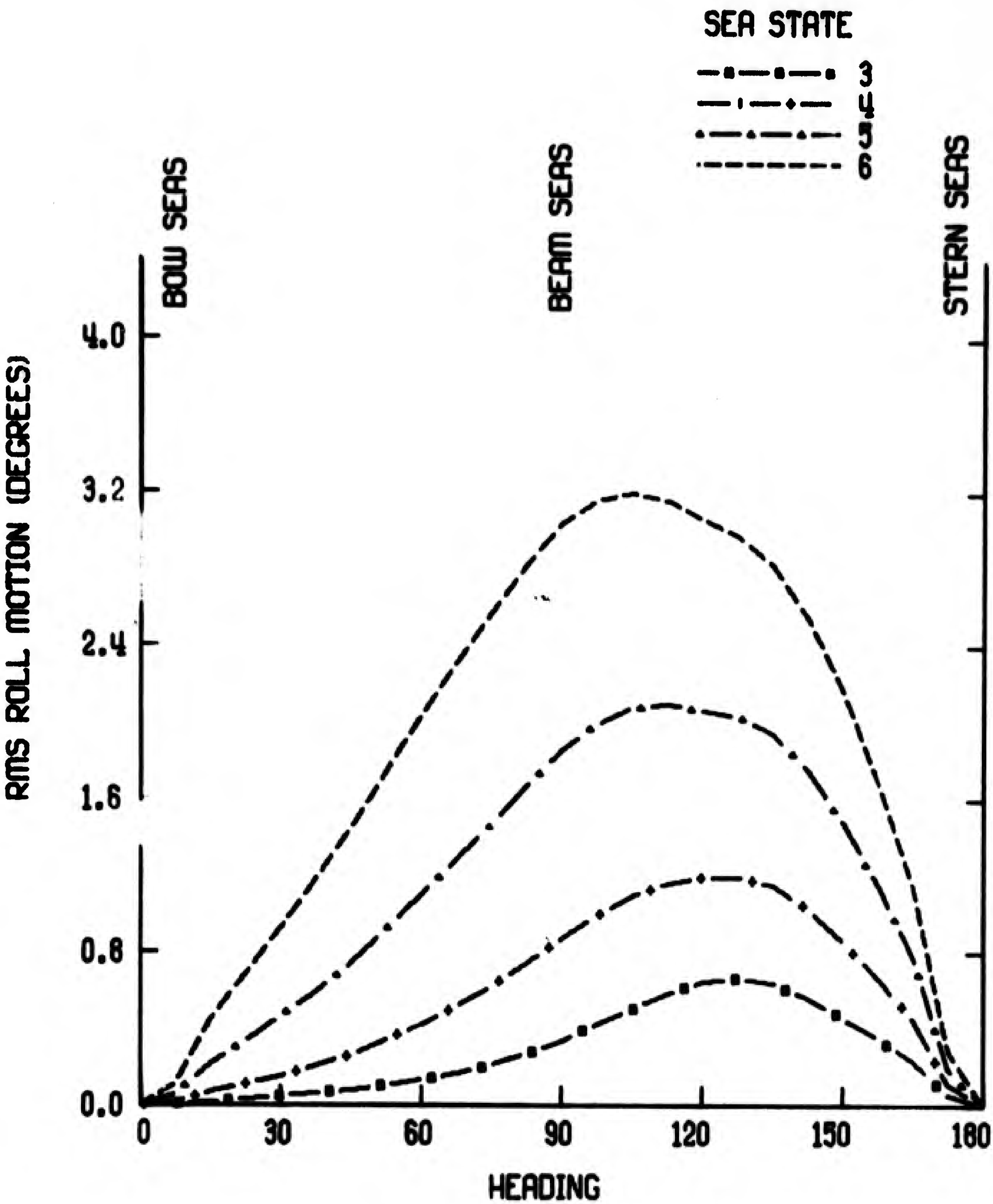


FIGURE 30 – RMS ROLL MOTION VS HEADING INTO LONG CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
ACTIVE HEELING-ANTI-ROLL TANKS
(G1 = -0.60 G2 = 0.30 G3 = 3.00)

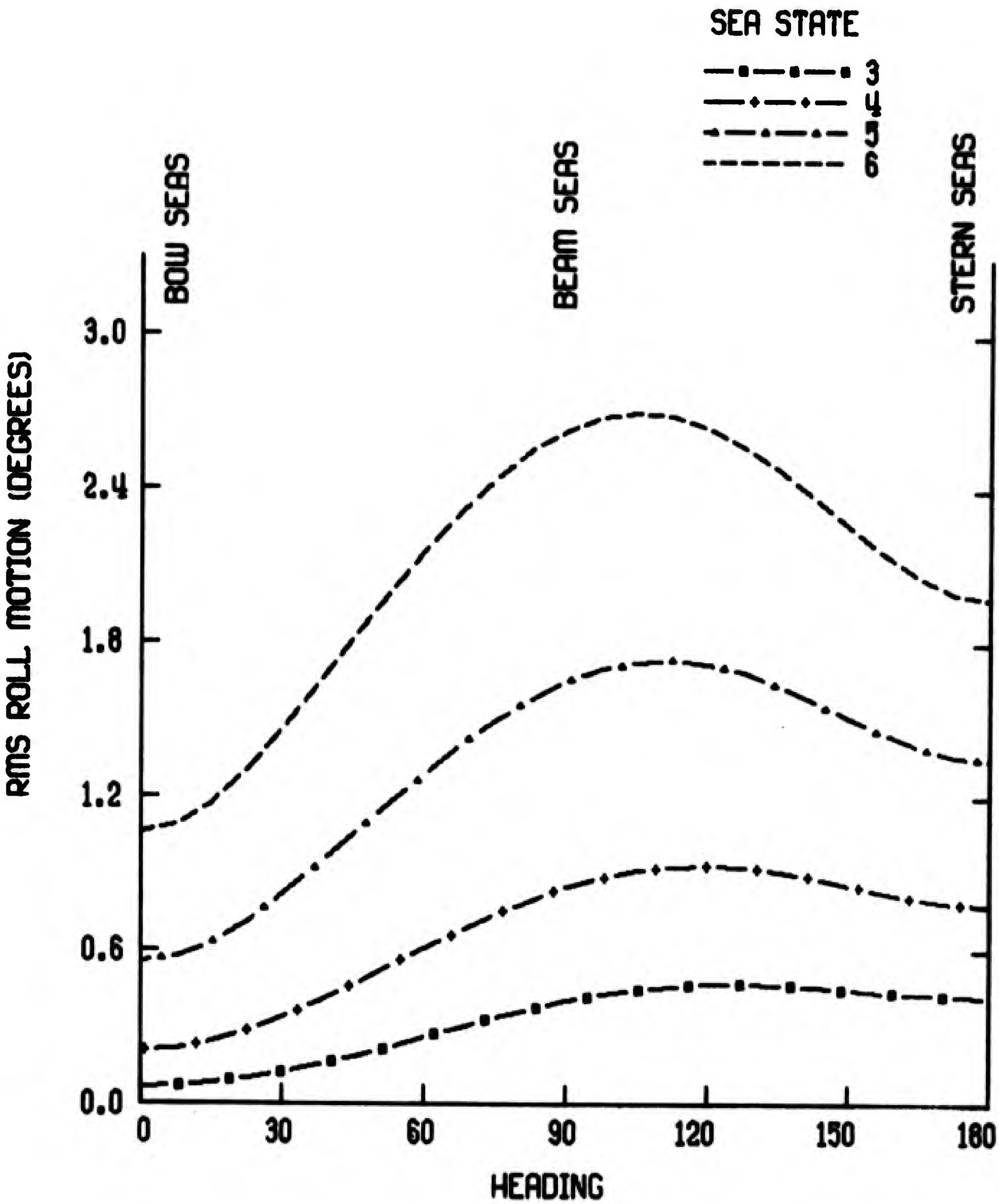


FIGURE 31 – RMS ROLL MOTION VS. HEADING INTO SHORT CRESTED SEAS
351FT. POLAR ICEBREAKER - 17KT.
ACTIVE HEELING-ANTI-ROLL TANKS
(G1 = -0.60 G2 = 0.30 G3 = 3.00)

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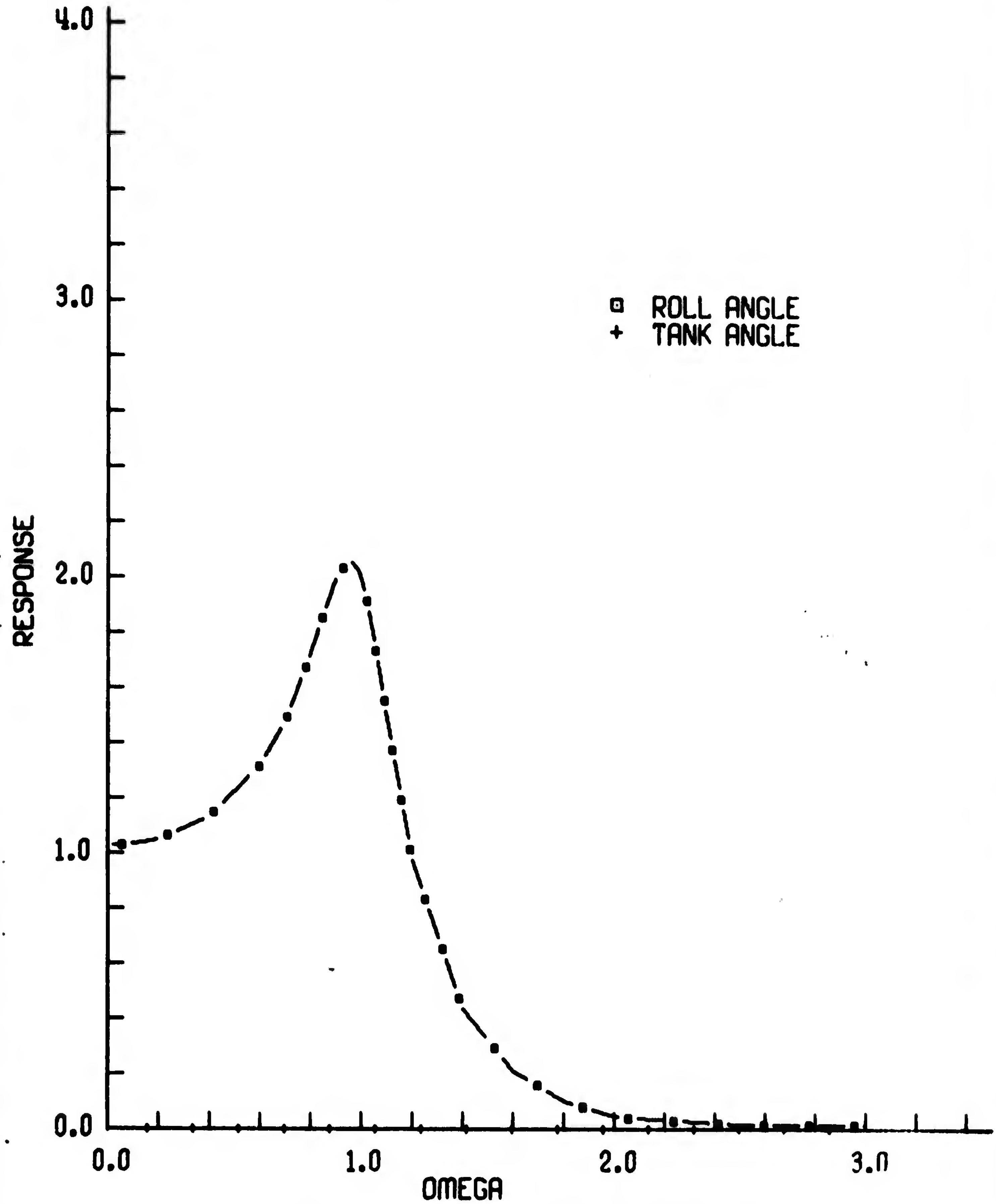



FIGURE 32 - RESPONSE TO WAVE OF UNIT SLOPE
351 FT. POLAR ICEBREAKER
1 DEGREE CAPACITY ACTIVE FINS
SHIP SPEED 17.0 KNOTS

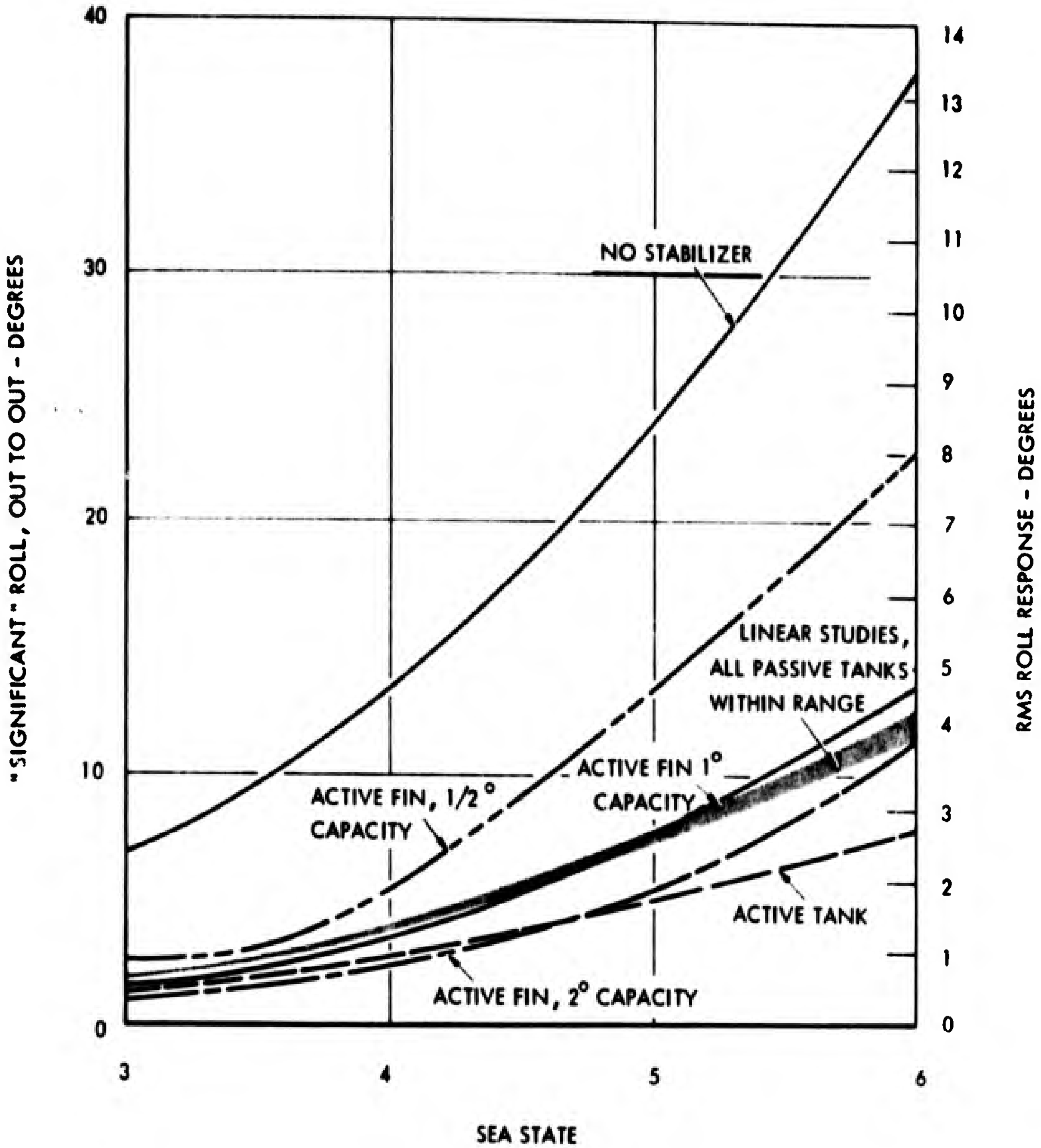



FIGURE 33 - ESTIMATED ROLL RESPONSE: FULLY DEVELOPED SHORT CRESTED SEAS FOR 351 FOOT POLAR ICEBREAKER STABILIZED AND UNSTABILIZED LINEAR STUDIES

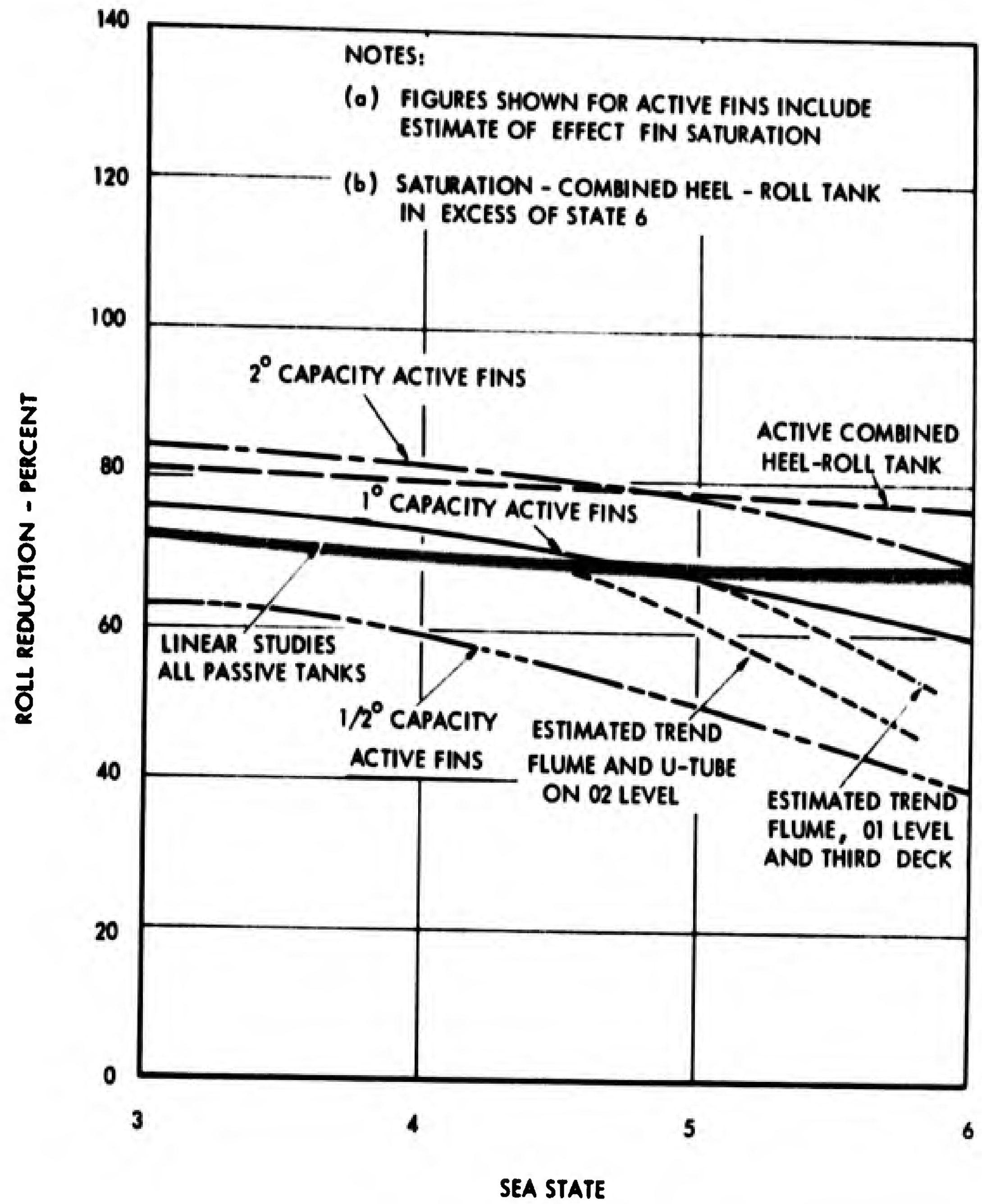



FIGURE 34 - ESTIMATED ROLL REDUCTION FOR VARIOUS STABILIZERS, 351 FOOT ICEBREAKER, 17 KNOTS SHORT CRESTED SEAS

# APPENDIX A

By

Horst Nowacki

## CONSIDERATIONS ON THE HEELING MOMENT REQUIRED TO FREE THE ICEBREAKER

Existing specifications on heeling tanks such as the Norske Veritas rules are limited to some recommendations for the tank dimensions. It remains uncertain, however, what physical effects can be brought about by tanks of given size, and how these effects are related to the required freeing capability in various iced-in conditions.

In order to develop a feeling for the magnitude of the physical requirements a simplified analysis of the critical situation with the icebreaker bow on the ice was carried out, which will at least establish a lower bound for the required freeing capability. Ship characteristics correspond to the "nominal" design point (displacement 11060 tons).

1. Assumptions

(a) The icebreaker bow has moved upon the ice up to the sharp dent formed at the lower end of the forefoot where the stem contour meets the keel. The force of support at this point equals 20 percent of the displacement, or P = 2210 L.T.

(b) Freeing is attempted by a combination of heeling by means of the tanks, and backing with all propellers. No "jack-knifing," i.e. yawing moments are applied simultaneously, e.g. by operating the propellers in opposite directions.

(c) In actuality, in order to free the forefoot by rotation about a longitudinal axis, the static friction must be overcome, but it is also necessary to exert normal pressures upon the ice obstructing the rotation until this ice is crushed. As a third influence, adhesive forces are acting between the bow and the ice, in particular if the bow has been at rest for some time.

In the analysis, the aspect of normal pressures is eliminated for simplification by assuming that the shape of the iced-in bow sections normal to the axis of rotation is circular. Only the tangential forces of friction and adhesion are thus considered. It is for this reason that only a lower bound of the required freeing moments is obtained.

(d) In conformity with the previous assumption, the bow sections in the iced-in domain are converted into circular sections of the same girth. For the iced-in area it is assumed as an extreme case that the whole forefoot is uniformly covered with ice. This region extends from the design waterline at 28 feet down to the 4 foot waterline, and from station 0 back to station 1.4 (bow dent). From this geometry the half-girths can be measured and the equivalent radii be determined:

| Station | Half-girth (ft.) | Equiv. radius (ft.) |
|---|---|---|
| 0 | 0 | 0 |
| 0.5 | 12 | 7.65 |
| 1 | 25 | 15.9 |
| 1.4 | 37.25 | 23.7 |

It can be seen that the slope of the substitution bow is nearly constant so that the shape is approximately conical. The average slope corresponds to $\alpha$ = 25.6 degrees.

The surface of the "semi-cone" equals:

$$S = \int_{0}^{1.4} \pi \cdot r(x) \cdot dx = \frac{\pi}{2} R_{max} \cdot \ell$$

$$= \frac{\pi}{2} \; 23.7 \cdot 49.25 = 1840 \text{ sq. ft.}$$

(e) The thrust axis is assumed to be colinear with the axis of bow rotation, i.e. the axis of the conical substitution bow.

(f) The fluid is located in wing tanks at 0.4 B from the centerline.

(g) The coefficient of static friction between steel and ice is assumed to be:

$$\mu_S = 0.6$$

(h) Adhesion is usually measured by the shear stress $\tau$ required to overcome the locking forces. If S is the area subject to adhesion, the shear force required to disrupt the connection is

$$P_A = \tau \cdot S$$

To treat adhesion in analogy to friction let us suppose the same restrictive effect is to be obtained by a friction force. This would require a normal force

$$N = \frac{P_A}{\mu_A} = \frac{\tau \cdot S}{\mu_A}$$

The analogous adhesion coefficient, which is applicable to the ship geometry and type of grounding described in the foregoing, thus becomes:

$$\mu_A = \frac{S}{N} \cdot \tau$$

where

$N = 0.2\ \Delta = 2210$ L.T. $= 4{,}950{,}000$ pounds

$S = 1840$ sq. ft. $= 265{,}000$ sq. in. for whole forefoot

$S/N = 0.0535$ sq. in./lb.

Values of adhesion between ice and steel vary drastically depending upon the type of paint, its condition, and its content of ice-phobic additives. A correspondence from the New York Naval Shipyard, Reference A.1, quotes the following test results obtained by their Material Laboratory:

| | Average Adhesion at 10°F, $\tau$, lb. per sq. in. |
|---|---|
| Ice on Mild Steel | 120 |
| Ice on Paint on Mild Steel | 80 |
| Ice on Paint with $Ni(CN)_2$ Additive on Mild Steel | 16 |

For the purpose of this study, two variations in adhesion were investigated; low adhesion of $\tau$ = 20 psi, and high adhesion of $\tau$ = 100 psi.

The extent of icing-in was also varied. The cases were with the whole forefoot iced in, and with 50 percent of the corresponding area iced in. This resulted in the following combinations:

(1) Static friction only:

$$\mu = \mu_S = 0.6$$

(2) Static friction and low adhesion, half forefoot in ice:

$$\mu = \mu_S + \mu_A = 0.6 + 0.535 = 1.135$$

(3) Static friction and low adhesion, whole forefoot in ice:

$$\mu = \mu_S + \mu_A = 0.6 + 1.07 = 1.67$$

(4) Static friction and high adhesion, half forefoot in ice:

$$\mu = \mu_S + \mu_A = 0.6 + 2.68 = 3.28$$

(5) Static friction and high adhesion, whole forefoot in ice:

$$\mu = \mu_S + \mu_A = 0.6 + 5.35 = 5.95$$

2. <u>Derivation of Required Heeling Moment and Backing Thrust to Free Icebreaker Bow</u>

During the freeing attempt the forces acting on the conical bow beside the heeling moment are thrust, T, and force of support P, see Figure A.1. These forces have the following normal and tangential components:

$$N = P \cdot \cos\alpha - T\sin\alpha$$

$$F_T = P \cdot \sin\alpha + T\cos\alpha$$

The tangential force produces an average freeing stress

$$p_T = \frac{F_T}{S}$$

The heeling moment causes a circumferential freeing stress, $p_h$, which is assumed to be uniformly distributed. The moment of these stresses developed in the whole area of contact must be in equilibrium with the heeling moment:

$$M_h = \int_0^{\ell} p_h \cdot \pi \cdot r(x) \cdot r(x)dx = p_h \cdot \pi \int_0^{\ell} r^2(x)\, dx$$

$$p_h = \frac{M_h}{V}\text{ , where } V = \pi \int_0^{\ell} r^2(x)\, dx = \frac{\pi \cdot \ell}{3} R^2_{max}$$

$$= \text{volume of semi-cone}$$

$$(V = 29{,}000 \text{ cu. ft.})$$

The resulting freeing stress, which is obtained by vectorial addition, must exceed the friction and adhesion stress to release the bow.

$$p_{res} = \sqrt{\left(\frac{F_T}{S}\right)^2 + \left(\frac{M_h}{V}\right)^2} \geq p_\mu \ , \quad p_\mu = \mu \cdot \frac{N}{S}$$

$$p_{res} = \frac{1}{S}\sqrt{F_T^2 + \left(\frac{M_h}{V/S}\right)^2} \geq \mu \cdot \frac{N}{S}$$

$$\frac{M_h}{V/S} \geq \sqrt{(\mu N)^2 - F_T^2}$$

Substituting for N and $F_T$:

$$\frac{M_h}{V/S} \geq \sqrt{C_{PP} \quad P^2 + C_{PT} \quad PT + C_{TT}T^2}$$

where

$$C_{PP} = \mu^2 \cdot \cos^2\alpha - \sin\alpha$$

$$C_{PT} = -2 \quad (\mu^2 + 1) \cdot \sin\alpha \cos\alpha$$

$$C_{TT} = \mu^2 \cdot \sin^2\alpha - \cos^2\alpha$$

In particular, for T = 0:

$$\frac{M_h}{V/S} = P \cdot \sqrt{C_{PP}}$$

and for $M_h = 0$:

$$F_T \geq \mu N \text{ or}$$

$$T \geq \frac{\mu \cdot \cos \alpha - \sin \alpha}{\mu \cdot \sin \alpha + \cos \alpha} P$$

Finally, in non-dimensional form:

$$\frac{M_h}{V/S \cdot P} \geq \sqrt{C_{PP} + C_{PT} \frac{T}{P} + C_{TT} \left(\frac{T}{P}\right)^2}$$

For $T = 0$:

$$\frac{M_h}{V/S \cdot P} \geq \sqrt{C_{PP}}$$

For $M_h = 0$:

$$\frac{T}{P} \geq \frac{\mu \cos \alpha - \sin \alpha}{\mu \sin \alpha + \cos \alpha}$$

3. Results and Conclusions

The results of the heeling moment requirement calculations for the icebreaker design are shown in Figure A.2. When half the forefoot is iced in, the assumption is that the area S is reduced by 50 percent; this is associated with a 75 percent reduction in iced-in volume V for a conical bow, so that V/S is reduced by 50 percent. In other words, when the bow is covered with ice to a smaller extent, the lever arms of the frictional stresses, which are proportional to V/S, are reduced in the same measure. This explains the large differences in required heeling moment for zero thrust with the extent of ice coverage, whereas this fact plays no part in backing the bow down at zero heeling moment.

Figure A.3 shows the heeling moment and thrust requirements in nondimensional form, which is applicable to any icebreaker bow of conical shape.

The following conclusions can be derived from the analysis:

(a) The full power backing thrust alone is insufficient to overcome even static friction alone. It would take about 25 percent increase in backing thrust to make this possible.

(b) A heeling moment of 5,500 L.T. feet, or less than 50 percent of the Norske Veritas requirement, would be enough to release the bow from static friction in combination with the full power backing thrust. This does, however, not allow for any additional bow restraints.

(c) Tanks dimensioned to Norske Veritas rules could free the conical bow from static friction even without propeller support.

(d) Tanks designed for the equivalent of a $5^{\circ}$ static heel would be capable of freeing the bow from friction and a small amount of adhesion or other restrictive forces.

(e) Considering the case of friction alone, the installation of a 5,500 L.T. foot heeling tank is equivalent in effectiveness to a 25 percent SHP-increase.

With more severe restraints, the trade-offs are even more striking. For $\mu$ = 1.135 e.g., it appears that a tank of about 15,000 L.T. feet can have an effect that would otherwise require about 770 L.T. additional thrust, or more than five times the design - SHP.

REFERENCE

A.1 Letter from Commander, New York Naval Shipyard to Chief of the Bureau of Ships (Code 324) dated 1 February 1952 on "De-Icing Materials and Methods Program — Progress on."

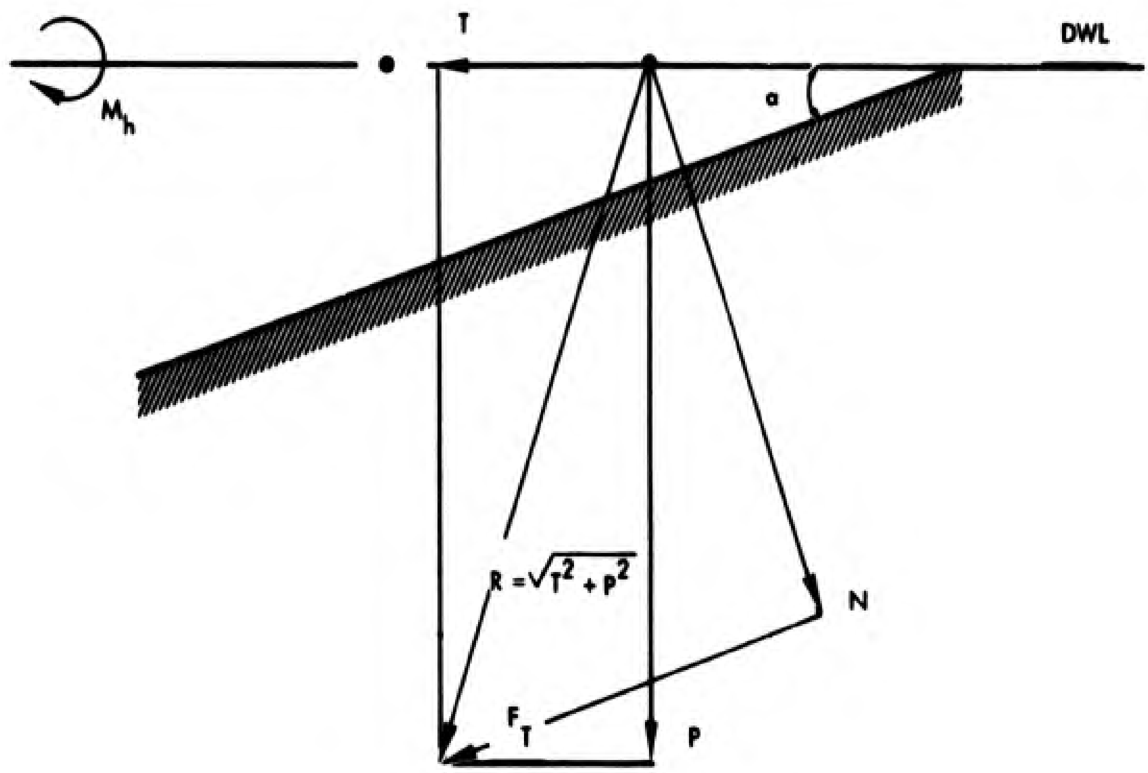



FIGURE A-1 - FORCE SITUATION AT CONICAL BOW

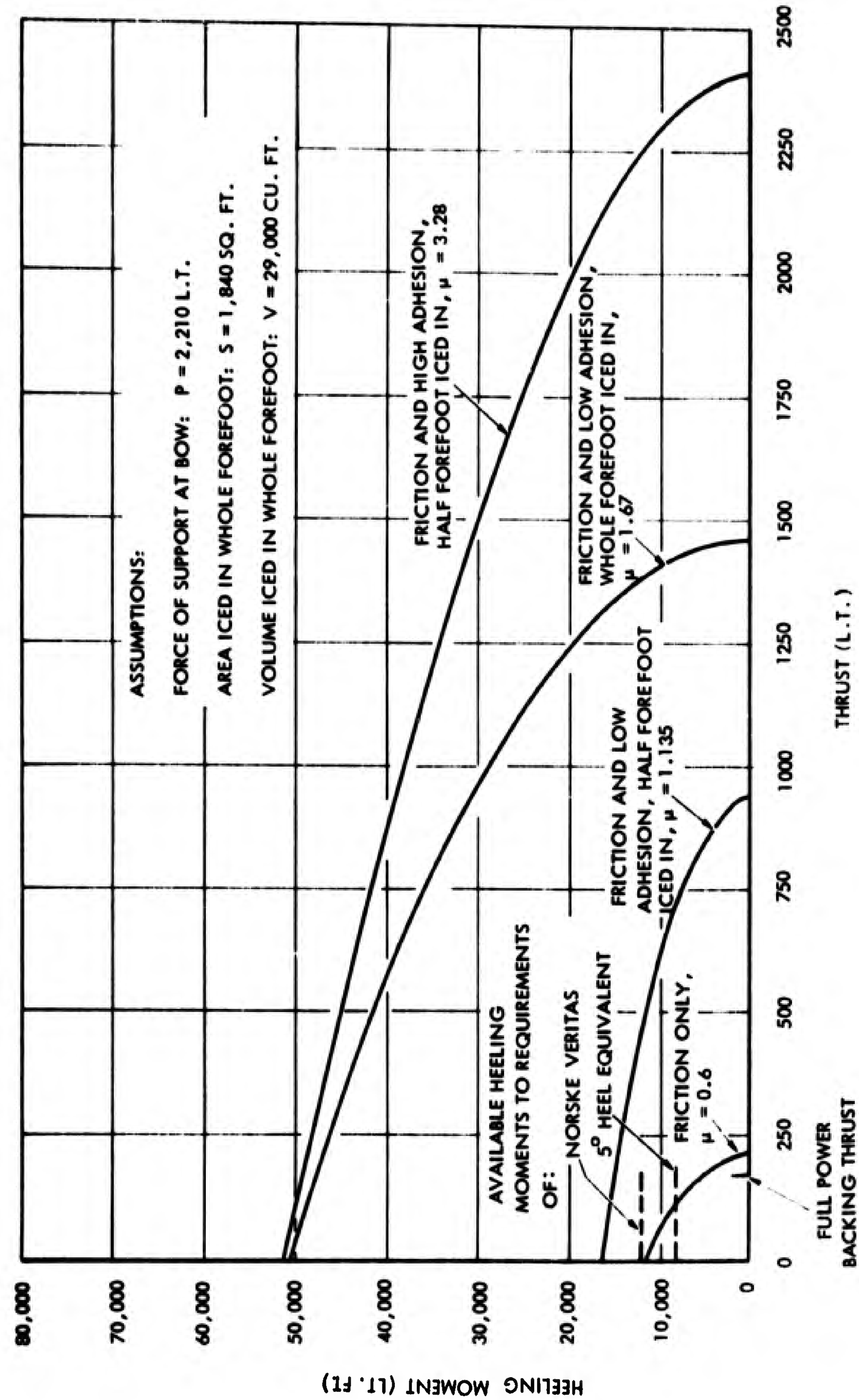



FIGURE A – 2 – REQUIRED AND AVAILABLE HEELING MOMENTS AND BACKING THRUST TO FREE CONICAL 11060 TON ICE BREAKER BOW

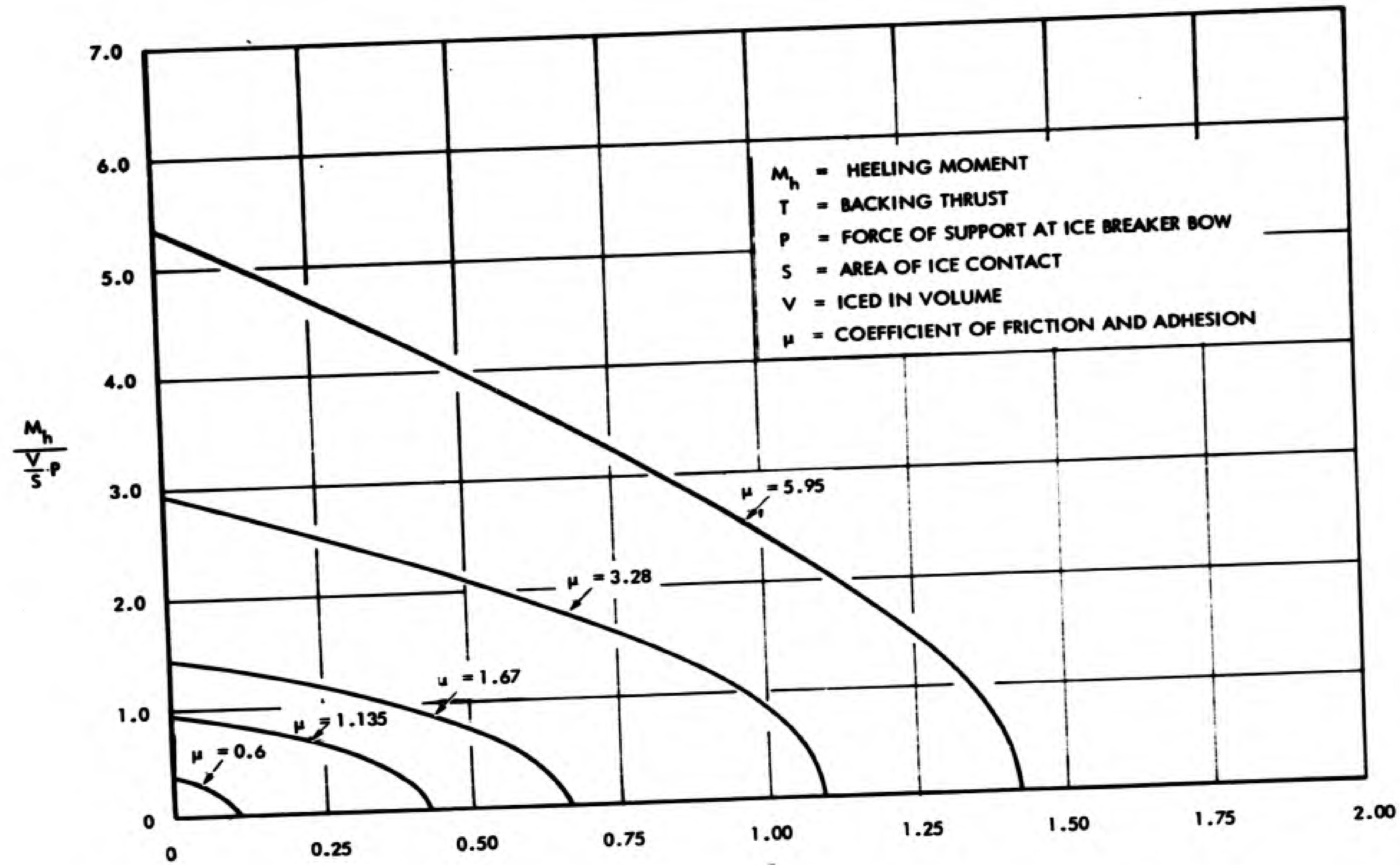



FIGURE A-3 - MONDIMENSIONAL HEELING MOMENT AND BACKING THRUST REQUIREMENTS TO FREE CONICAL ICE BREAKER BOW

## APPENDIX B

## HYDROSTATIC RIGHTING MOMENTS: FLARED PRISMATIC BODY

A flared prismatic body is assumed as in Figure B-1. The body floats at waterline $WL_1$ with no heel and at waterline $WL_2$ with a heel angle of $\theta$. Flare is defined by the angle $\alpha$. When the body heels about the centerline of the original waterplane ($WL_1$), more volume is immersed than emerges because of the flare. As a consequence, the body must rise to a new waterline $WL_2$ to maintain constant displacement. The new waterplane intersects the old a distance ($\delta$) from the centerline. The body length is assumed equal to dx.

The areas of the emerged and immersed triangles are equal by the definition of the problem:

$$\frac{\ell}{2}(b-\delta)\sin\theta = \frac{(b+\delta)}{2} m \sin\theta \qquad \text{(B-1)}$$

From geometrical considerations:

$$\frac{\ell}{b-\delta} = \frac{\sin\alpha}{\sin(\alpha-\theta)} \qquad \text{(B-2)}$$

$$\frac{m}{b+\delta} = \frac{\sin\alpha}{\sin(\alpha+\theta)} \qquad \text{(B-3)}$$

Combining Equations B-1, B-2, B-3:

$$\left(\frac{\delta}{b}\right)^2 \frac{1}{2}\frac{\tan\theta}{\tan\alpha} - \left(\frac{\delta}{b}\right) + \frac{1}{2}\frac{\tan\theta}{\tan\alpha} = 0 \qquad \text{(B-4)}$$

Appendix B (continued)

Equation (B-4) yields solutions for $(\frac{\delta}{B})$.

The restoring couple on the body is the product of the buoyancy represented by the immersed or emerged triangles and the projection on $WL_2$ of the total distance between centroids $g_1$ and $g_2$.

From geometrical considerations, the horizontal shift in centroid is:

$$\frac{1}{3}(\ell + m + 2b \cos \theta) \tag{B-5}$$

The restoring couple is then:

$$M = \frac{\gamma b^3 dx}{3} (\frac{\ell}{b})(1-\frac{\delta}{b}) \sin \theta \left[\frac{\ell+m}{2b} + \cos \theta\right] \tag{B-6}$$

After some manipulation:

$$M = \frac{\gamma 2b^3 dx}{3} \sin \theta \left[\cos \theta (1-\frac{\delta}{b})^2 \frac{\sin \alpha}{\sin(\alpha-\theta)} \left[\frac{1}{2} + \frac{\sin^2 \alpha \sqrt{1-\tan^2 \theta/\tan^2 \alpha}}{2 \sin(\alpha-\theta) \sin(\alpha+\theta)}\right]\right] \tag{B-7}$$

When $\alpha$ $90^{\circ}$ Equation (B-7) becomes:

$$M_{\alpha=90} = \frac{\gamma 2b^3 dx}{3} \sin \theta \left[\frac{1}{2} + \frac{1}{2 \cos^2 \theta}\right]$$

$$= \frac{\gamma 2b^3 dx}{3} \sin \theta \left[1 + \frac{\tan^2 \theta}{2}\right] \tag{B-8}$$

The last is the expression arrived at for the righting couple on a wall sided ship.

Appendix B (continued)

The question at hand is how much the righting moments for a prismatic body with a $70^{\circ}$ flare angle ($\alpha$) differ from those for a wall sided prismatic body. Evaluation of the terms in the large brackets in Equations B-7 and B-8 was made for $\alpha = 70^{\circ}$. The results indicated that at most there is only a 1% difference between restoring moments for the wall sided body, and the flared body ($\alpha = 70^{\circ}$) up to heel angles of $15^{\circ}$. For heel angles up to $10^{\circ}$, there appears to be a maximum error (of -2%) in the expression:

$$M = \frac{\gamma 2b^3 dx}{3} \sin \theta \qquad \text{(B-9)}$$

$$\text{For } 90^{\circ} \geq \alpha \geq 70^{\circ}$$

The integral of (B-9) over ship length is:

$$\int_0^L M = \Delta\, \overline{BM} \sin \theta \qquad \text{(B-10)}$$

The moment required to heel the ship to $\theta$ is:

$$\Delta\, \overline{BM} \sin \theta - \Delta\, BG \sin \theta = \Delta\, GM \sin \theta \qquad \text{(B-11)}$$

(Per standard development). Assuming $BG \approx BM/2$, the error involved in using the small angle-wall sided approximation for ships with flare angles not less than $70^{\circ}$ is likely to be no more than a 5% underestimate of moment for $10^{\circ}$ heel and about a 2% underestimate for $5^{\circ}$ heel.

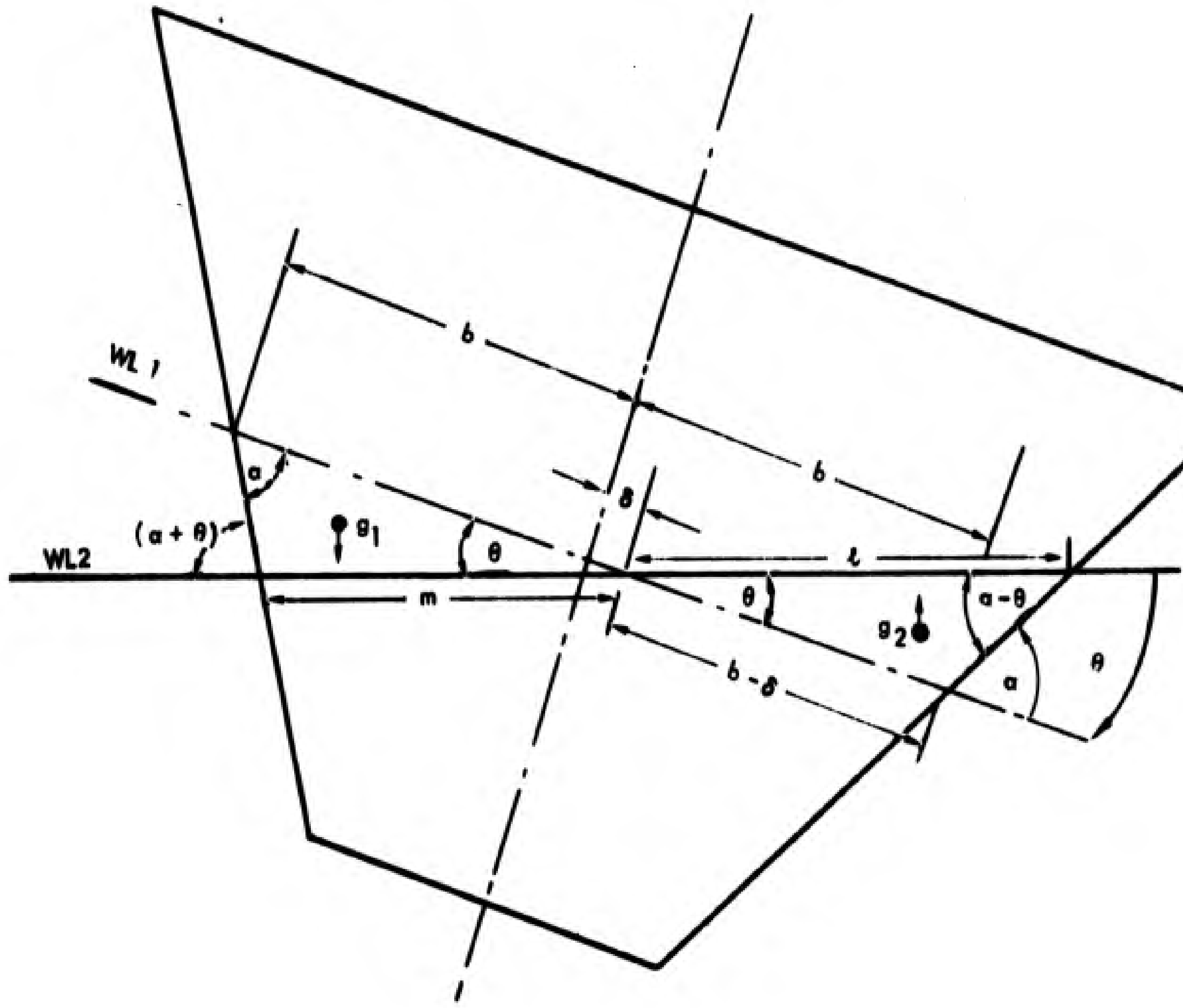



FIGURE B-1

# APPENDIX C

## PRELIMINARY ACTIVE FIN DESIGN

by Horst Nowacki

### A. Derivation of Fin Design Requirements from the Given Roll Reduction Percentage

The fin design requirements are given only indirectly in terms of the desired roll reduction so that conversion is necessary. The fins act essentially by dissipating a certain amount of energy each cycle and can be treated in analogy to other damping devices. Fin effects upon the system inertia are minor, and therefore neglected in this context. These assumptions allow an estimate to be made of the increase in damping provided by the fins. The derivation is based on the following assumptions and steps:

1. The rolling motion of the ship is considered to be governed by the linear equation of motion:

$$I_x' \ddot{\phi} + 2w \dot{\phi} + \Delta \cdot \overline{GM} \cdot \phi = K_o(t)$$

where

$I_x'$ = the mass moment of inertia of the ship in rolling about the axis of roll, including hydrodynamic inertia.

$2w$ = damping coefficient

$\Delta$ = displacement

$\overline{GM}$ = metacentric height

$K_o(t)$ = excitation moment due to waves

Or in normalized form:

$$\ddot{\phi} + 2\zeta\,\omega_n\,\dot{\phi} + \omega_n^2\,\phi = \hat{K}_o(t)$$

where

$$2\zeta = \frac{2w\cdot\omega_n}{\Delta\cdot\overline{GM}} = \frac{2w}{\omega_n\,I_x'} = \text{nondimensional damping coefficient}$$

$$\omega_n = \sqrt{\frac{\Delta\cdot\overline{GM}}{I_x'}} = \text{natural frequency of roll}$$

$$\hat{K}_o(t) = \frac{K_o(t)}{I_x'}$$

The above equations of motion imply the assumptions that coupling influences are negligible, and that the nonlinear influence upon the restoring moment is insignificant.

It is further assumed that the response remains essentially harmonic, $\phi = \phi_o \cos \omega t$, even though the installation of fins introduces a somewhat nonlinear contribution to the damping moment.

2. It is assumed that ship speed by far exceeds the circumferential velocity component induced by the rolling motion at the fin. In this event, the inflow velocity into the fin is nearly constant, when the fin is at angle of attack, and so are fin lift and moment.

To support this assumption, compare the icebreaker ship speed of V = 17 knots = 28.7 ft/sec to the circumferential velocity $U_c$ at resonance:

$$U_c = b \cdot |\dot{\phi}|_{max} = b \cdot \omega\, \bar{\phi} = 0.55 \cdot 78' \cdot 0.53 \cdot 0.1 = 2.3 \text{ ft/sec} < 10\%\ V_s$$

where

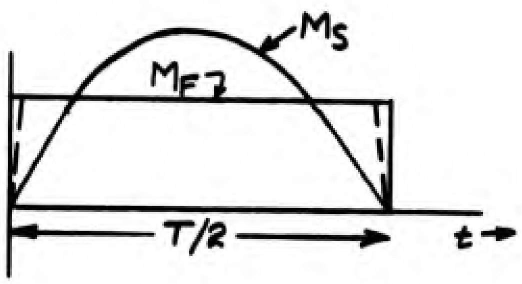
$b = 0.55B$ = distance from fin to axis of roll

$\omega = 2\pi/T$ = resonant frequency

$\bar{\phi} = 5.7° = 0.1_{rad}$ = estimated resonance amplitude with fins in operation

The variation in inflow velocity during one cycle would be less than 5% in this case.

3. The fin moment, which is considered to be constant during each half-cycle, is now replaced by a sinusoidally varying moment $M_s$ that dissipates the same amount of energy per half-cycle.




<u>The work done by $M_F$:</u>

$$W_F = \int M_F d\phi = M_F \int_0^{T/2} \frac{d\phi}{dt}\, dt = M_F \int_0^{T/2} \bar{\phi} \cdot \omega \sin \omega t\, dt$$

$$W_F = 2\bar{\phi}\, M_F$$

The work done by $M_S$:

$$W_S = \int M_s d\phi = \int_0^{\frac{T}{2}} 2w\,\dot{\phi}\,\frac{d\phi}{dt}\,dt = 2w\int_0^{\frac{T}{2}} (\bar{\phi}\omega)^2 \cdot \sin \omega t\,dt$$

$$W_S = \frac{2w\cdot\pi\cdot\omega^{o}\cdot\bar{\phi}^2}{2} = (2w\omega\bar{\phi})\cdot\frac{\pi^{o}\bar{\phi}}{2} = M_{S_{max}} \cdot \frac{\pi\bar{\phi}}{2}$$

Since $W_F = W_S$, the maximum equivalent fin moment becomes:

$$M_{S_{max}} = \frac{4}{\pi} M_F = 1.27\ M_F$$

If this figure is corrected for the time required for fin reversal, during which less than the full fin moment is produced, one obtains a slightly different result. E.g. with 1.3 sec. reversal time (as on the "Labrador" Denny-Brown system) and 5.9 sec. half-cycle duration (resonant condition), $M_F$ has to be about 11% higher or:

$$M_{S_{max}} = 1.14 \cdot M_F = k_1 \cdot M_F$$

4. The effect of the fin has now been replaced by an equivalent sinusoidal damping term, and one can solve for the corresponding damping coefficient.

$$M_{S_{max}} = 2w \cdot \dot{\phi}_{max} = 2w\,\bar{\phi} \cdot \omega$$

$$2w = \frac{M_{S_{max}}}{\bar{\phi} \cdot \omega}$$

$$2\zeta_F = \frac{2W \cdot \omega_n}{\Delta \cdot GM} = \frac{M_{S_{max}}}{\Delta \cdot GM \cdot \phi} \cdot \frac{\omega_n}{\omega}$$

$$2\zeta_F = \frac{k_1 M_F}{\Delta \cdot GM \cdot \phi} \cdot \frac{\omega_n}{\omega}$$

This represents the contribution of the fin to the damping coefficient.

5. The next question is what fin damping coefficient is required for a certain roll reduction percentage $\alpha$.

At resonance, the motion amplitude $\phi$ for any given wave slope $\bar{v}$ is inversely proportional to the resulting damping coefficient:

$$\phi = \frac{1}{2\zeta} \bar{v}$$

The damping coefficient consists of contributions from the ship $(2\zeta_S)$ and the fins $(2\zeta_F)$:

$$2\zeta = 2\zeta_S + 2\zeta_F$$

In order to reduce $\phi$ by the percentage $\alpha$ by means of the fins:

$$\frac{1}{2\zeta_S + 2\zeta_F} \bar{v} = \frac{1-\alpha}{2\zeta_S} \bar{v}$$

or:

$$2\zeta_F = \frac{\alpha}{1-\alpha} \, 2\zeta_S$$

The required fin damping coefficient thus depends on the damping already present in the ship. It is thus easier to meet the roll reduction requirement for a ship that has only little damping itself.

Figure C-1 shows the required fin damping coefficient as a function of $\alpha$ for various estimated ship damping coefficients:

$2\zeta_S = 0.04$, zero speed damping

$2\zeta_S = 0.07$, realistic estimate of finite speed damping of icebreaker

$2\zeta_S = 0.08$, high, "pessimistic" estimate of finite speed damping of icebreaker

6. It is also customary to measure the effectiveness of a stabilizing system by its capability to compensate a wave slope if its maximum stabilizing moment is applied statically.

The righting moment of the ship can be approximated for small angles of heel by

$$M_R = \Delta \cdot \overline{GM} \cdot \sin \theta$$

The heeling moment produced by static application of the fins designed for the damping coefficient as required at resonance ($\omega/\omega_n = 1$) equals

$$M_F = \frac{2\zeta_F \cdot \Delta \cdot \overline{GM} \cdot \bar{\phi}_{res}}{k_1}$$

where $\bar{\phi}_{res}$ = the resonance amplitude with fins operating.

The relations between wave slope capability, design damping, and roll reduction rate thus become:

$$\sin \theta = \frac{2\zeta_F}{k_1} \cdot \bar{\phi}_{res} =$$

$$= 2\zeta_S \cdot \frac{\alpha}{1-\alpha} \bar{\phi}_{res}$$

The wave slope capability based on this approximate relation is plotted in Figure C-2 as a function of roll reduction rate, with resonant roll amplitude of the stabilized motion as parameter.

7. Conclusions:

It appears from this result that a wave slope capability of 1 degree is sufficient to meet the 60% roll reduction requirement for the icebreaker.

This figure is much lower than the 5 to 6 degrees usually designed for on passenger vessels. But, it must be borne in mind that there the same reduction rate may be aimed at, but the ships are fitted with bilge keels and have usually more than twice as much damping without fins ($2\zeta_S$). It is, therefore, necessary to install fins of more than double capacity for the same rate of improvement.

Another comparison may be drawn with the Canadian icebreaker "Labrador" and its Denny - Brown stabilizers, Reference 7. Its wave slope capability can be estimated as follows:

$\Delta = 6400$ L.T.

Estimated $\overline{GM} = 6.75'$, so that $\overline{GM}/B = .08$

Fin area: $A = 10.67' \cdot 5.67' = 60.0$ sq. ft.

Lever from axis of roll $b = 21'$

Speed: $V = 16$ knots $= 27$ ft/sec

$V^2 = 730$ ft$^2$/sec$^2$

Lift coefficient $C_L = 1.6$ (conservative)

Lift: $$L = \frac{C_L \cdot 0.5\ \rho V^2 \cdot A}{2240} = 31.2 \text{ L.T.}$$

Lift of two fins $2L = 62.4$ L.T.

$$\sin\theta = \frac{(2L)\cdot b}{\Delta \cdot \overline{GM}} = 0.0303$$

Wave slope capability: $\theta = 1.75$ deg

This value is only little higher than that derived above. The system may have been designed with some allowance for the inevitable insufficiencies of control in an irregular seaway.

In conclusion, a variation of the wave slope capability between 0.5 and 2 degrees is considered as sufficient to investigate in the preliminary design studies. At $\phi_{res} = 5.7$ deg. $= 0.1$ rad, this corresponds to a range of $2\zeta_F$ between 0.1 and 0.4.

## B. Preliminary Fin Design Procedure

The design of the fin system is based on the following basic relationships. The notation is explained below.

### Fin Dimensions:

Fin moment for given wave slope requirement:

$$M_F = \Delta \cdot GM \cdot \sin\theta$$

Lift per fin:

$$L = \frac{M_F}{n \cdot b}$$

Required area per fin:

$$A = \frac{L}{0.5\ \rho \cdot V^2 \cdot C_L}$$

Cavitation number:

$$\sigma = \frac{P_o - e}{0.5\ \rho\ V^2} = \frac{\gamma(ho - he)}{0.5\rho\ V^2}$$

Chord length and span:

$$c = \sqrt{A/a}$$

$$s = A/c$$

### Strength:

Lift per fin:

$$L = C_L \cdot 0.5\rho\ V^2 \cdot A$$

Drag per fin:

$$D = C_D \cdot 0.5\rho\ V^2 \cdot A$$

Resultant force per fin:

$$R = \sqrt{C_L^2 + C_D^2} \cdot 0.5\rho\ V^2 \cdot A$$

Bending moment and torque in fin shaft at bearing:

$$M_B = (0.5\ s + \bar{s}) \cdot R$$

$$M_T = \bar{c} \cdot R$$

Bending and torsional stresses:

$$\sigma_B = \frac{M_B}{\pi/32 \cdot d^3}$$

$$\tau = \frac{M_T}{\pi/16 \cdot d^3}$$

Maximum principal stress, allowable stress:

$$\sigma_1 = \frac{\sigma_B}{2} + \sqrt{\left(\frac{\sigma_B}{2}\right)^2 + \tau^2} = \sigma_A$$

$$= \frac{M_B}{\pi/16 \cdot d^3} \left[1 + \sqrt{1 + \left(\frac{M_T}{M_B}\right)^2}\ \right]$$

Minimum diameter of fin shaft:

$$d = \sqrt[3]{\frac{M_B}{\pi/16 \cdot \sigma_A} \left(1 + \sqrt{1 + \left(\frac{M_T}{M_B}\right)^2}\ \right)}$$

Power:

Power to overcome appendage drag of two fins:

$$P = \frac{0.5\rho(2A)V^3 \cdot C_D}{550}, \text{ in HP}$$

Tilting moment and power per fin to overcome steady flow force:

$$M_1 = C_N \cdot 0.5\rho V^2 \cdot A \cdot \bar{c}$$

$$P_1 = \frac{M_1 \cdot \frac{dx}{dt}}{550}, \text{ in HP}$$

Tilting moment and power per fin due to acceleration:

$$M_2 = D_H \cdot \frac{d^2\alpha}{dt^2} = \frac{\pi}{8}\rho \cdot c^4 \cdot s \frac{d^2\alpha}{dt^2}$$

$$P_2 = \frac{M_2 \frac{d\alpha}{dt}}{550}, \text{ in HP}$$

Total tilting power:

$$P_T = \frac{1}{550}\left(C_N \cdot 0.5\rho V^2 \cdot \bar{c} \cdot A + 0.125 \cdot \pi \cdot \rho \cdot c^4 \cdot s \frac{d^2\alpha}{dt^2}\right)\frac{d\alpha}{dt}$$

Notation:

| | |
|---|---|
| $\Delta$ | displacement |
| $\overline{GM}$ | metacentric height |
| $\theta$ | wave slope |
| $M_F$ | moment of all fins |
| $n$ | number of fins |

| | |
|---|---|
| $L$ | lift of one fin |
| $C_L$ | $= L/(0.5\rho V^2 \cdot A)$ = fin lift coefficient |
| $b$ | fin lever arm, distance from center of roll to midspace of fin |
| $A$ | area of one fin |
| $V$ | fin advance speed (ft/sec), taken as ship speed (=17 knots = 28.7 ft/sec) |
| $\rho$ | mass density of water |
| $\gamma$ | specific weight of water |
| $p_o, h_o$ | static pressure and corresponding head at fin, consisting of atmospheric pressure and static water pressure |
| $e, h_e$ | vapor pressure and corresponding head, $h_e = 0.41$ ft. at 50 deg. F |
| $c$ | chord length of fin |
| $a$ | aspect ratio |
| $s$ | span of fin |
| $D$ | fin drag |
| $C_D$ | $D/(0.5\rho V^2 \cdot A)$ = fin drag coefficient |
| $M_B$ | bending moment in fin shaft at outer bearing |
| $M_T$ | torque in fin shaft |
| $\overline{s}$ | distance from inner end of fin to center of adjacent bearing |
| $\overline{c}$ | distance from center of pressure to shaft axis |
| $\sigma_1$ | principal stress |
| $\sigma_A$ | allowable stress |
| $d$ | required diameter of fin shaft |

$P$ additional propulsive power to overcome fin drag

$C_N$ $= C_L \cdot \cos\alpha + C_D \sin\alpha$ = fin normal force coefficient

$D_H$ hydrodynamic inertia coefficient for rotation of a rectangular plate about its spanwise axis through centroid, Ref. C-5, page 112.

$dx/dt, d^2\alpha/dt^2$ = angular velocity and acceleration of fin while tilting

$P_T$ total tilting power, peak power

Design Assumptions:

Number of fins:

Two fins are sufficient fo all wave slope requirements of interest.

Type of fin:

The fins are assumed to consist of movable main fins fitted with movable flaps as high-lift devices. These arrangements have proven successful in raising the maximum obtainable lift coefficient, in the same time delaying separation to higher angles of attack. They thus reduce the required area.

It is also assumed that end plates are mounted to the fins to minimize the induced drag.

Cavitation:

The lift that the fins can produce when operating in water may be limited by cavitation in particular at higher speeds.

In the present case, with a 20 ft. fin submergence:

$$V = 17 \text{ knots} = 28.7 \text{ ft/sec}$$

$$= 64 \text{ lbs/ft}^3$$

$$h_o = 53 \text{ ft.}$$

$$\sigma = 4.1$$

It can be seen from Reference C-2, Figures 5 and 6, and from Reference C-1, Figure 31, that for this cavitation number, the lift coefficients are not yet seriously affected.

Lift coefficient:

In view of the preceding conclusions on cavitation, it appears realistic to expect a maximum lift coefficient around $C_L = 1.6$ at about 20 degrees main fin angle, see Reference C-2. Reference C-1, Figure 31, recommends $C_L = 1.55$, which is in close agreement.

Aspect ratio:

A high aspect ratio promises a high lift per unit fin area, but practical considerations about space requirements for retractable fins, bearing design, and fin shaft strength limit the aspect ratio to the neighborhood of 2.

In some cases in the following, values a little below 2 had to be assumed to ensure acceptable fin shaft dimensions. This means that for higher aspect ratios, the chord length and consequently the root profile thickness would become too small to be compatible with the shaft dimensions.

Drag coefficient:

The drag coefficient pertains to the fin and flap at their design angles of attack (20 and 30 degrees respectively). It must include the induced drag caused by finite span, although this effect is greatly reduced by end plates.

The drag coefficient is taken as $C_D = 0.35$ from the similar case of Reference C-3, Figure 20, which applies to fins with end plates of 1.72 aspect ratio. No other applicable design data are available to check this assumption, but in comparison with airfoil data for infinite span, it appears to be in the right order of magnitude, Reference C-4.

Lever arm of fin bending moment:

The distance from the inner end of the fin to the center of the adjacent bearing is estimated as $\overline{s} = 2$ ft. for all designs in the following. This is added to the half-span to obtain the lever arm.

Lever arm of hydrodynamic fin torque:

This is the distance from the shaft axis to the hydrodynamic center of pressure. The location of the latter varies with cavitation number and, hence to some extent with angle of attack and speed. But, this variation is weak in the neighborhood of the design condition, Reference C-2, Figure 9.

It is, therefore, considered conservative to assume that the shaft can be located so that its distance from the center of pressure at no time exceeds $\overline{c} = 0.12c$.

Allowable stress:

The allowable stress is taken as $\sigma_A$ = 40,000 psi according to naval specifications for rudder stocks.

Angular velocity and acceleration in tilting fin:

In order to reverse the fin from +20 degrees to -20 degrees within 1 second, it takes an average angular velocity of $d\alpha/dt = 0.7$ rad/sec.

For the acceleration, it is assumed that the speed of 0.7 rad/sec must be reached from rest within 0.1 sec., so that $d^2\alpha/dt^2 = 7$ rad/sec$^2$.

Rated power of tilting motor:

It has to be noted that the acceleration phase lasts only a fraction of a second during a cycle of several seconds. The maximum tilting power $P_T$ is a pronounced peak power, therefore, and it is common practice in such cases to give the motor a reduced power rating taking advantage of its overload capacity. We assume a motor rated at 0.5 $P_T$ would be acceptable.

However, a 25% correction for mechanical losses in power transmission should be added. The power rating will thus equal 0.625 $P_T$.

Weights:

Weights have to be estimated according to the scarce data available from other designs, which are listed in the table below. All weights are total weights including lost buoyancy.

TABLE C-1

| Ship | Fin Area, Ft. | Design Speed | Weight Long Tons | Source(Ref.) |
|---|---|---|---|---|
| Cross-Channel Ferry | 18 | 20 | 20 | C-3 |
| Icebreaker "Labrador" | 60 | 16 | 77 | 7 |
| Passenger Ship | 70 | 14 | 58 | C-3 |
| DD "Timmerman" | 90 | 15 | 70 | C-2 |

These figures do not give unique trends, but it appears that the icebreaker installation is heavier than average, while the "Timmerman" data seem to be projected low. No meaningful relation can be derived for the weights, because too little is known about the influence of parametrical variations. It has to suffice instead to conclude that the weight of the smaller installations in L.T. roughly equals the area of one fin in sq. ft. (for two fins in total) while larger installations tend to be relatively lighter. But, in view of the variations, all weight estimates here were made conservatively high, i.e., somewhat higher than corresponding to the fin area.

Space requirements:

The space required inside a vessel for retractable fins and the associated gear may vary with fin aspect ratio, inclination of fin axis and other arrangement peculiarities.

But it is thought that in most cases, the net result can be made fairly competitive to the following dimensions:

Beamwise dimension = 2.2s

Longitudinal dimension = 2.0c

Vertical dimension = 3.0c

Volume required per fin = $13.2 \cdot s \cdot c^2$

Further space may sometimes be lost because of difficulties to fit the compartmentation closely and effectively around the stabilizer volume.

Results:

The results are presented in Table C-2 on fin design data and in summary in Figure C-3. A sketch of possible arrangements for the medium size design is given in Figure C-4.

Because of the moderate wave slope capability, the design specifications are generally favorably low in comparison to passenger vessels of comparable size and speed. The systems proposed here seem also more favorable than those of the ice-breaker "Labrador." There, some performance was sacrificed by installing the fins in the forebody at 1/3 length from the forward end where their lever arms are shorter than amidships.

## REFERENCES - APPENDIX C

C-1 Flipse, John E., "Stabilizer Performance on SS MARIPOSA and SS MONTEREY," SNAME 65, 1957.

C-2 Hagen, G. R., "Feasibility Studies of the Roll Stabilization of the USS BOSTON (CAG-1)," DTMB Report 950, 1955.

C-3 Allen, J. F., "The Stabilization of Ships by Activated Fins," RINA 87, 1945.

C-4 Abbot and von Doenhoff, "Theory of Wing Sections," Dover Publications, New York, 1959.

C-5 Korvin-Kroukovsky, B. V., "Theory of Seakeeping," SNAME, New York, 1961.

TABLE C-2

FIN DESIGN DATA

| Item | Dimension | Des. 1 | Des. 2 | Des. 3 |
|---|---|---|---|---|
| Wave slope capacity $\theta$ | Degrees | 0.5 | 1.0 | 2.0 |
| Required fin moment $M_F$ | LT. ft. | 820 | 1640 | 3280 |
| Span s | ft. | 6 | 8 | 12 |
| Lever b = 32'+0.5 s | ft. | 35 | 36 | 38 |
| Required lift per fin L | LT. | 11.7 | 22.8 | 43.1 |
| Required area per fin A | sq.ft. | 18.85 | 36.7 | 69.5 |
| Aspect ratio a | -- | 1.91 | 1.75 | 2.07 |
| Chord length c | ft. | 3.14 | 4.59 | 5.8 |
| $\sqrt{C_L^2 + C_D^2}$ | -- | 1.635 | 1.635 | 1.635 |
| Resultant force R | lbs. | 25,400 | 49,500 | 93,800 |
| Bending lever = $0.5s+\bar{s}$ | ft. | 5 | 6 | 8 |
| Bending moment $M_B$ | lb.in. | 1,525,000 | 3,565,000 | 9,000,000 |
| Torque $M_T = 0.12 \cdot C \cdot R$ | lb.in. | 115,000 | 327,000 | 783,000 |
| $M_T/M_B$ | -- | 0.0755 | 0.092 | 0.087 |
| $M_B/(\frac{\pi}{16} \cdot \sigma_A)$ | $in.^3$ | 194 | 454 | 1142 |
| Required diameter of fin shaft d | in. | 7.3 | 9.7 | 13.2 |
| Shaft diameter - chord length ratio | -- | 0.193 | 0.176 | 0.19 |
| Propulsive power for fin drag P | HP | 56.5 | 110 | 208 |
| Tilting moment $M_1$ | lb.in. | 9530 | 27,100 | 64,900 |
| Tilting moment $M_2$ | lb.in. | 3210 | 19,350 | 74,300 |
| Total tilting moment | lb.in. | 12,740 | 46,450 | 139,200 |

TABLE C-2 (Continued)

| Item | Dimension | Des. 1 | Des. 2 | Des. 3 |
|---|---|---|---|---|
| Wave slope capacity $\theta$ | Degrees | 0.5 | 1.0 | 2.0 |
| Peak tilting power $P_T$ | HP | 16.2 | 59 | 177 |
| Rated power of each tilting motor | HP | 10 | 37 | 111 |
| Total weight of installation, estimated | LT. | 21 | 37 | 70 |
| Total weight/displacement | -- | 0.2 | 0.33 | 0.63 |
| Space requirement, estimated both fins | $ft^3$ | 1570 | 4440 | 10,600 |

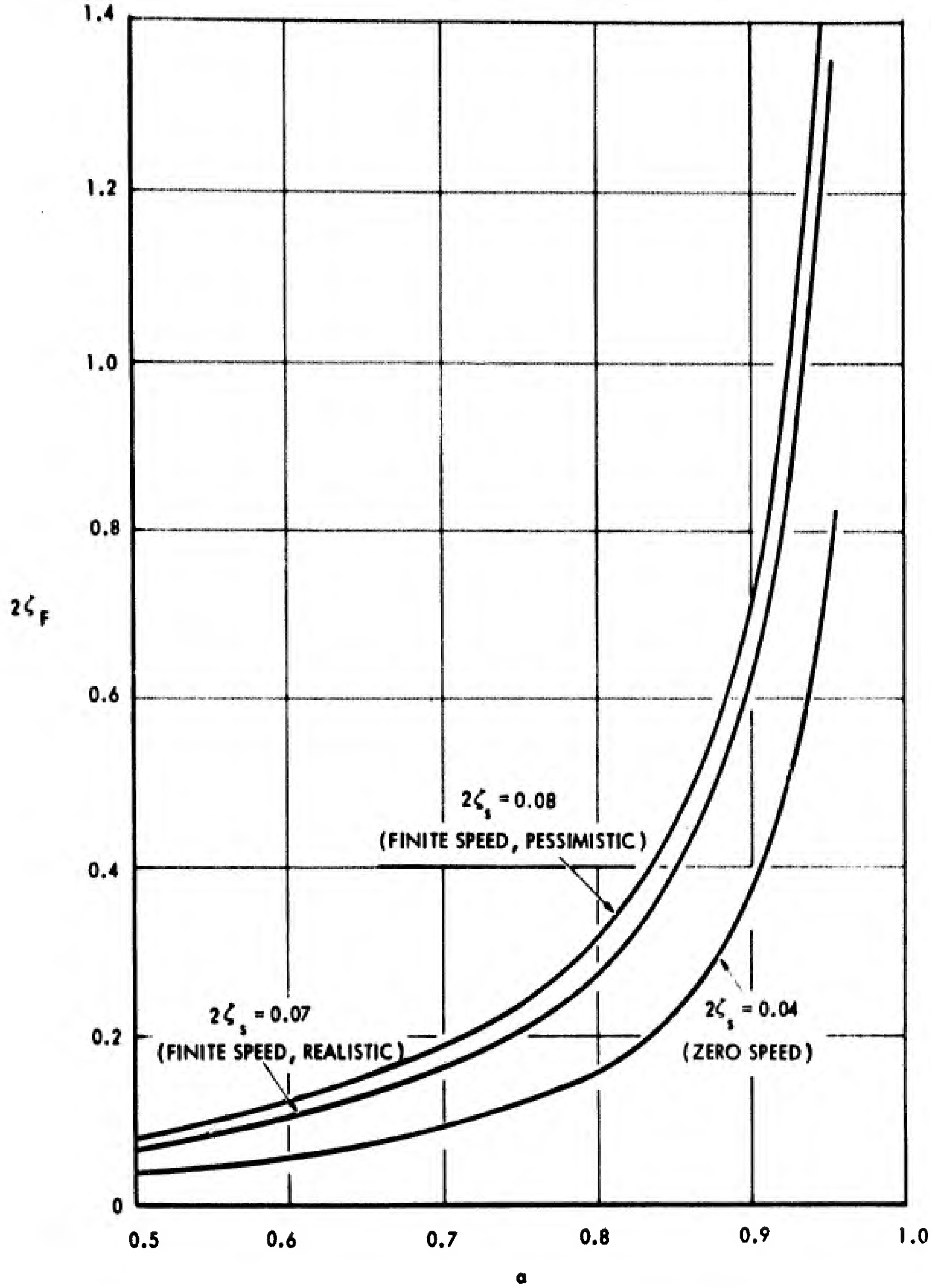



FIGURE C-1 - REQUIRED FIN DAMPING COEFFICIENT AGAINST ROLL REDUCTION PERCENTAGE

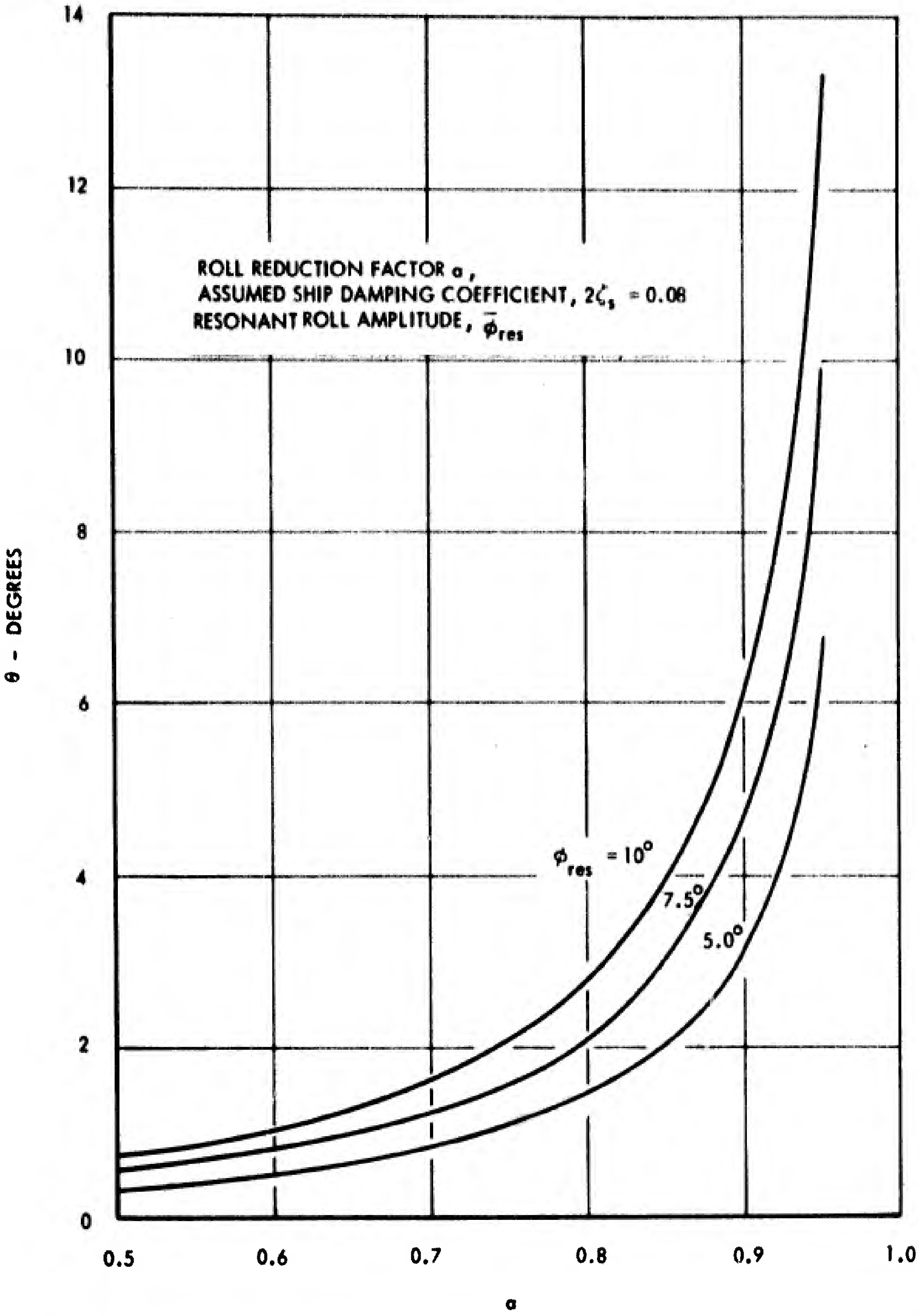



FIGURE C - 2 - APPROXIMATE EQUIVALENT WAVE SLOPE CAPABILITY

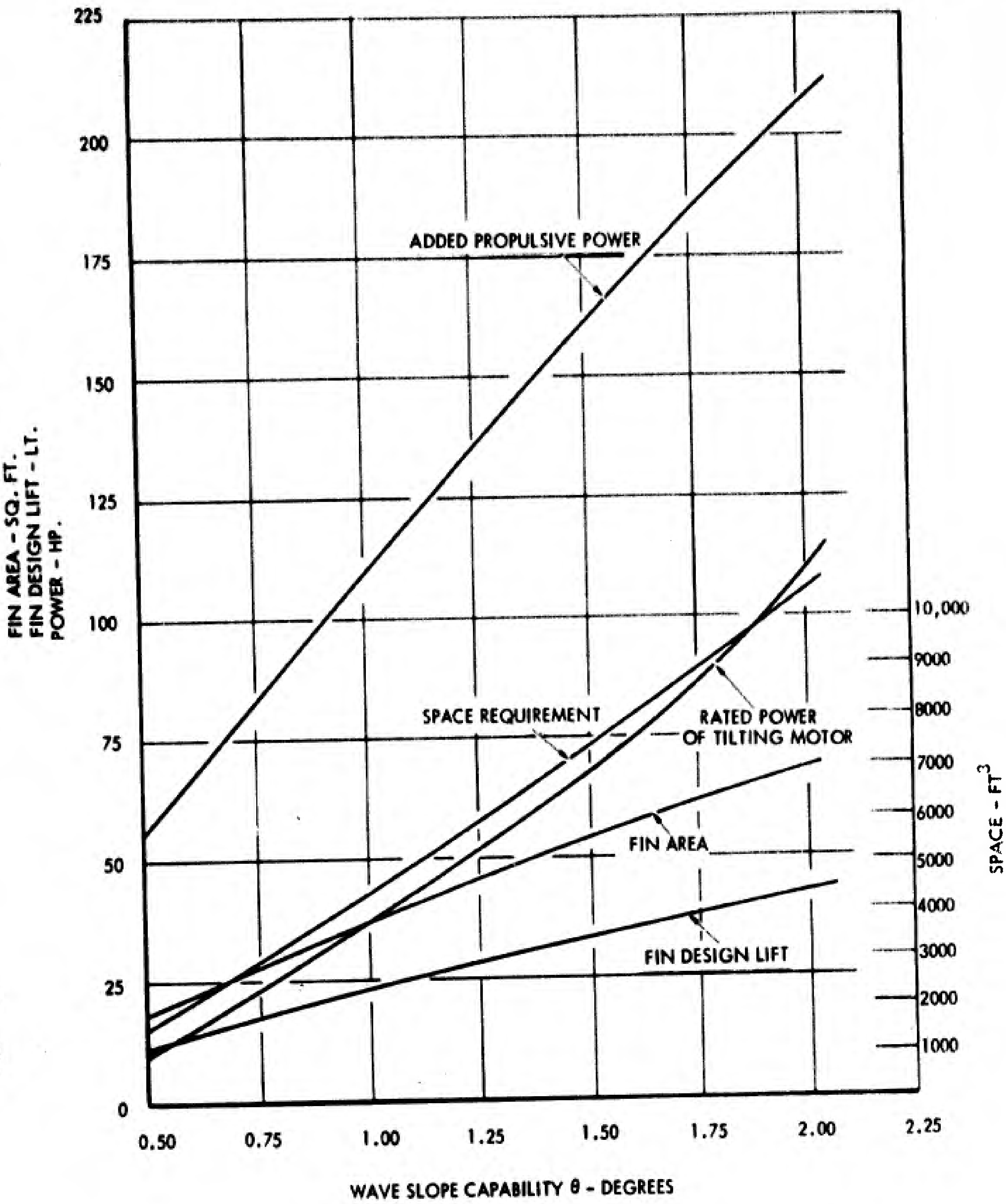



FIGURE C-3 - ESTIMATED SPECIFICATIONS OF FIN STABILIZING SYSTEM

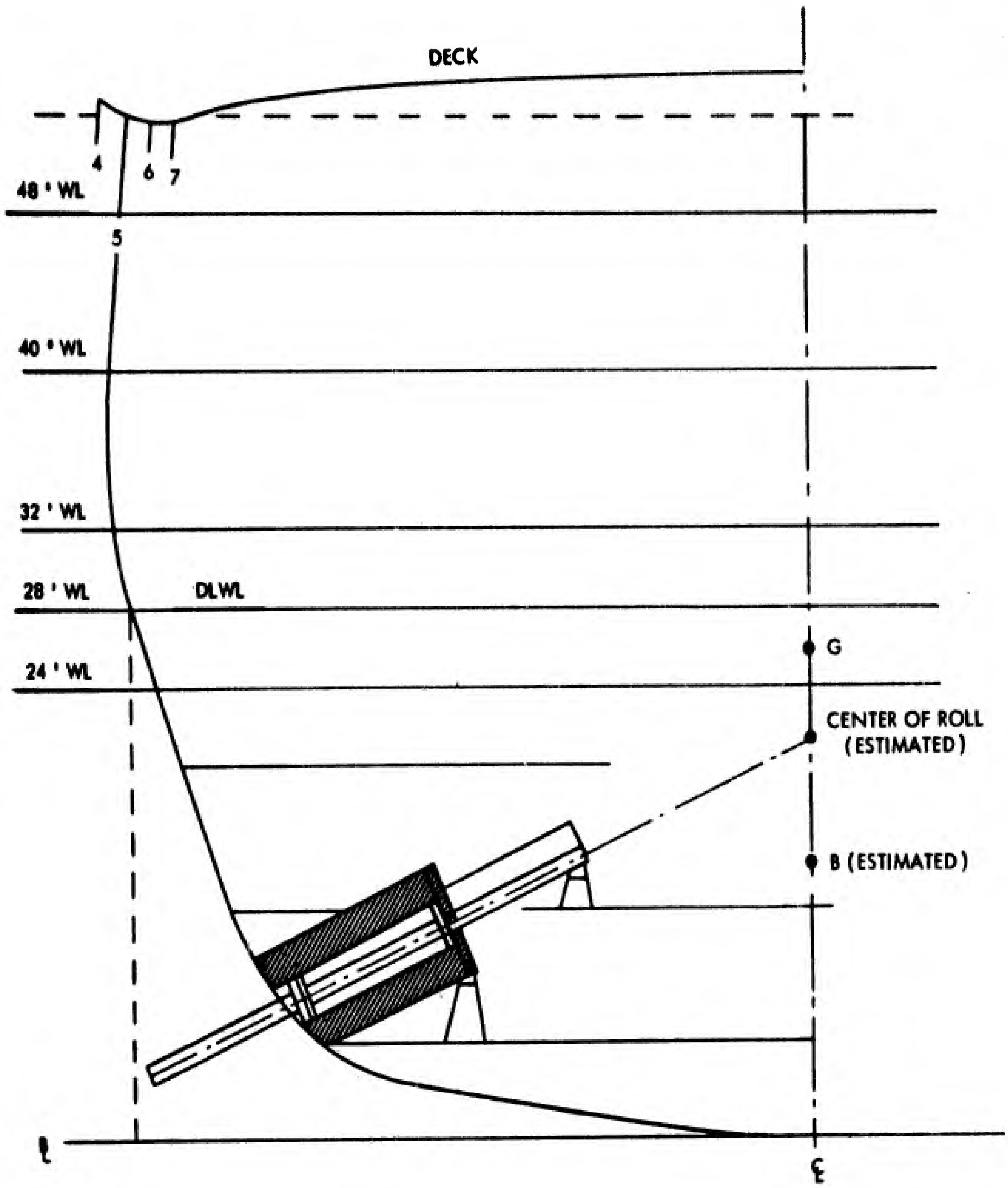



FIGURE C-4 - ACTIVE FIN: SCHEMATIC ARRANGEMENT

## APPENDIX D

## SPERRY GYROFIN DATA

Data on three standard sizes of Sperry Gyrofins were furnished by Sperry Piedmont and are summarized below:

| Fin Type Size | "A" | "B" | "C" |
|---|---|---|---|
| Fin Span | 8 ft. | 11 ft. | 14 ft. |
| Fin Chord | 4 ft. | 5.5 ft. | 7 ft. |
| Fin Area | 32 $ft.^2$ | 60.5 $ft.^2$ | 98 $ft.^2$ |
| Fin Lift (at 17 kts.) | 16.5 tons | 31 tons | 50 tons |
| Fin Box Length | 15.5 ft. | 23 ft. | 25 ft. |
| Fin Box Width | 6 ft. | 8 ft. | 9 ft. |
| Fin Box Height | 4.5 ft. | 5.5 ft. | 6.5 ft. |
| Lost Buoyancy (per fin with fin extended) | 7 tons | 13 tons | 17 tons |
| Estimated Weight (per fin assembly) | 24 tons | 40 tons | 65 tons |
| Hydraulic Drive Motor (continuous duty rating) | 25 hp | 40 hp | 75 hp |
| Budgetary Cost (2 fins/ship) | $155,000 | $250,000 | $350,000 |

All fins are fully flapped with geometric aspect ratio of 2. Control systems include lift feedback. All systems above are of the type in which the fin is housed by rotating $90^{\circ}$.

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| FEASIBILITY STUDY | | | | | | |
| ROLL | | | | | | |
| STABILIZATION | | | | | | |
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