AD-778 620

THE HYDRODYNAMIC DESIGN AND EVALUATION OF A TOWED INSTRUMENTATION VEHICLE

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April 1974

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Bethesda, Md. 20034

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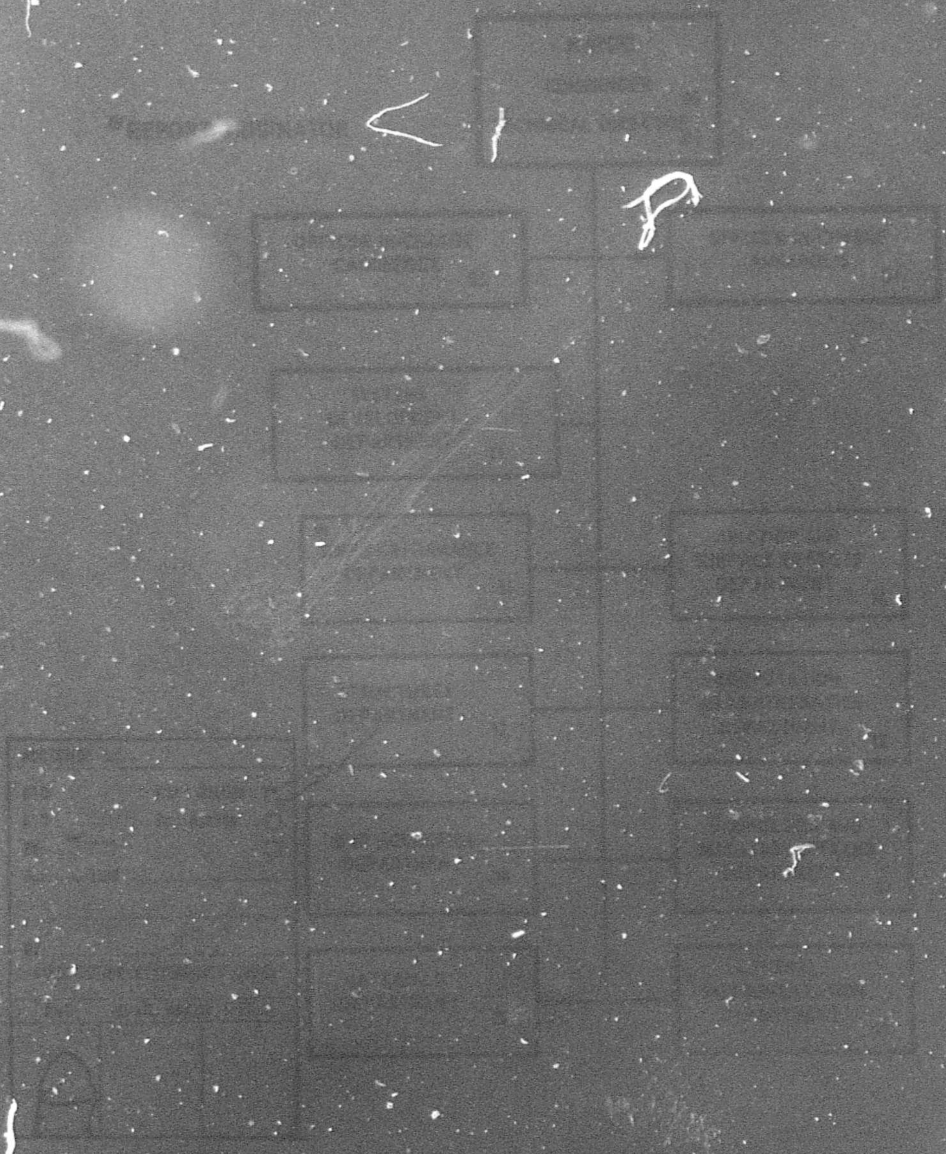

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NOTATION

A	Planform area
a	Aspect ratio b^2/A
B	Buoyancy
b	Wing or tail surface span
C_{D_C}	Wing crossflow drag coefficient
C_L	Wing lift coefficient L/qA
C_{L_α}	Slope of curve of lift coefficient versus angle of attack
C_R	Drag coefficient (based on diameter) of cable when normal to free stream
C_{ℓ_α}	Slope of curve of section lift coefficient versus angle of attack
D	Drag
d	Cable diameter
e	Wing offset distance from body centerline
F_V	Total downforce
K	Tail interference factor due to dihedral
$(k_2 - k_1)$	Virtual mass coefficient
L	Wing lift
ℓ	Characteristic length
ℓ_s	Distance from the sting crossbeam to the towpoint
ℓ_t	Distance from the center of pressure of the tail to the towpoint
M	Pitching moment
M'	Nondimensional pitching moment $M/q\ell^3$
N	Yawing moment
N'	Nondimensional yawing moment $N/q\ell^3$
q	Dynamic pressure $(1/2)\rho v^2$
R	Cable drag per unit length when normal to free stream
r	Body radius at point of attachment of wing
T	Cable tension
U	Free-stream velocity
V	Volume
W	Weight
W_f	Local body width
x_η	Length of nth body segment
α	Angle of attack relative to free stream, in degrees

β	Local flow angle, equal to free-stream angle plus angle of flow induced by wing upwash
Γ	Dihedral angle
ϵ	Wing upwash angle
η	Nondimensional dynamic pressure loss
θ	Pitch angle
Λ	Sweep of wing or tail surface quarter-chord
ρ	Water density
δ	Sidewash angle due to body and wing interference flow
ϕ	Cable angle relative to free stream
ϕ_C	Cable critical angle relative to free stream
ψ	Yaw angle

SUBSCRIPTS

b	Body
o	Cable values at body
s	Sting
t	Tail surfaces
wt	Wing-to-tail surfaces

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

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1. REPORT NUMBER 4143	2. GOVT ACCESSION NO.
3. RECIPIENT'S CATALOG NUMBER AD 778620	
4. TITLE (and Subtitle) THE HYDRODYNAMIC DESIGN AND EVALUATION OF A TOWED INSTRUMENTATION VEHICLE	5. TYPE OF REPORT & PERIOD COVERED
	6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) R.K. Knutson	8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Ship Research and Development Center Bethesda, Md. 20034	10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS NOL Project Order 9-0121 NSRDC Work Unit 1-1548-074
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Ordnance Laboratory Silver Spring, Md. 20910	12. REPORT DATE April 1974
	13. NUMBER OF PAGES 27
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)	15. SECURITY CLASS. (of this report) UNCLASSIFIED
	15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

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

18. SUPPLEMENTARY NOTES

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

Design and Evaluation
Towed Body

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

The hydrodynamic design approach and basin evaluation of a cable-towed vehicle are presented. The resulting body form reflects several features which were included to reduce cost of fabrication. The results of the basin evaluation indicate that the simplifying features did not significantly compromise the towing characteristics of the vehicle. The vehicle will perform satisfactorily at speeds substantially above the nominal design speed of 8 knots. Furthermore, cable predictions, also presented herein,

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ADMINISTRATIVE INFORMATION

The work reported herein was funded by Naval Ordnance Laboratory (NOL) Purchase Order 9-0121 of 4 March 1969, Naval Ship Research and Development Center (the Center) Work Unit 1-1548-074.

INTRODUCTION

This Center was requested by NOL to design and construct a cable-towed vehicle to be towed from a surface ship. NOL specified that the main body of the vehicle was to be built from various sections of a discarded mobile mine casing, which was supplied to the Center for that purpose. NOL further indicated that the form of additional fixtures to the body would be suitable for inexpensive fabrication.

To accomplish the task, a cable study was undertaken to determine the hydrodynamic characteristics required in order for the vehicle to satisfy the operating criteria discussed in the following section. This study dictated the design of the depressor wings and stabilizing surfaces. These appendages were manufactured and installed, and experiments were conducted to confirm that design parameters had been fulfilled. System performance was predicted on the basis of the results of the basin experiments.

This report presents the design objectives, outlines the hydrodynamic design procedure, indicates the results of basin experiments that were conducted to ascertain the extent to which the objectives had been achieved, and presents graphically the predicted cable forces and configurations over the range of operating conditions of interest.

DESIGN OBJECTIVES AND CONSIDERATIONS

Specifications indicated that the vehicle would be required to operate most frequently at depths not greater than 250 feet and at speeds less than 8 knots. A further specification stipulated that a 0.528-inch-diameter, double-armored electromechanical cable measuring

500 feet in overall length would be used to tow the vehicle. This design information is summarized in Table 1.

To minimize the adverse effects on the towed vehicle of pitching and heaving motions of the towship, the cable angle at the water surface should be as small as possible. Therefore, the system should be designed in such a manner that all of the usable cable is required to obtain a towing depth of 250 feet at 8 knots.

TABLE 1 – DESIGN SPECIFICATIONS

Vehicle Housing: Section from Mobile Mine Casing	
Depth Capability, feet	250
Speed Requirement, knots	8
Overall Cable Length, feet	500
Cable Diameter, inches	0.528
Cable Weight in Seawater, pounds per foot	0.36

DESIGN PROCEDURE

The steady-state performance required of this vehicle was not unusual, and relatively straightforward design techniques were employed. The steps involved in the design of the system are outlined in the following subsections.

DEVELOPMENT OF GENERAL VEHICLE FORM

The vehicle housing was evolved by using selected sections from a mobile mine. The resultant form (Figure 1) consists of a cylindrical midbody, a disk-ogive nose (flat front and a curvilinear transition to the midbody), and a streamlined afterbody. The nose and tail sections are secured to the midbody by retainer rings to produce a watertight housing. The housing has an overall length of 7.55 feet and an external diameter of 1.73 feet. The fineness ratio is 4.35. A sting, constructed of 1.8-inch-diameter metal tubing and measuring approximately 23 inches in length, is secured to the nose of the housing. A short hydrodynamically faired crossbeam is located at the extreme forward end of the sting.

A depressor wing with a rectangular planform was selected to simplify construction. However, this shape will produce relatively high tip loadings and therefore will have a somewhat higher induced drag than one with a more elliptical spanwise pressure distribution. This selection seemed justified, nevertheless, since expense of fabrication was of primary concern. For further simplicity, the wing was designed to have zero dihedral angle. The selection of the wing aspect ratio was based on considerations of total wing span, lift-to-drag ratio, and structural stiffness. An aspect ratio of 3.0 was chosen to provide a satisfactory balance among these factors. A TMB-07507515 section shape[1] was selected for the wing section. This shape has a constant-radius leading edge with straight taper from the point of tangency to the trailing edge. The thickness-to-chord ratio is 0.15. The section has lift and stall properties comparable to that of an NACA 0015 section shape, but the maximum lift-to-drag ratio is somewhat less. The slight reduction in the lift-to-drag ratio which would result from using this section shape was accepted because of the reduced expense of fabrication.

The wing was designed to be mounted on top of the vehicle housing at approximately the longitudinal center of the housing parallel midbody. A mounting bracket was welded to the housing at this location. The towblock, to which the towcable is attached, is located on top of the wing and secured to the vehicle, along with the wing, by bolts which extend through the wing to the bracket. This method was used primarily for simplicity and ease of construction and offers a distinct advangage, i.e., with the wing and towblock located as a unit on top of the housing, the mounting bracket is not required to support the dynamic wing loads but only the static and dynamic forces produced by the housing. This is important since the skin of the casing is relatively thin, and forces applied to this surface must be kept to a minimum. One possible disadvantage of this location should be noted. A depressor wing located above the centerline of a body will produce some effective negative dihedral which could adversely affect the stability of the vehicle.

The original mobile mine had eight small stabilizing fins spaced at 45-degree intervals around the periphery of the tail section. Trim tabs were located on the horizontal and the vertical fins. Consequently, these same horizontal and vertical fins were retained for trimming the vehicle. The remaining four small fins were removed and replaced by larger fins in order to stabilize the body. The tail configuration thus developed is basically an X-form (interdigitated tail). For simplicity a rectangular planform was selected for the large fins. A fin section shape with a semicircular leading edge and a parallel after section with a squared off trailing edge was chosen for ease of construction.

CABLE FORCE

A schematic of the towing configuration is shown in Figure 2. The tension T_0 required to obtain the specified steady-state operating conditions were calculated from tables in earlier studies by Pode[2] and Pode and Rosenthal.[3] The velocity at an element of cable is

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[1] Whicker, L.F. and L.F. Fehlner, "Free-Stream Characteristics of a Family of Low-Aspect-Ratio, All-Movable Control Surfaces for Application to Ship Design," David Taylor Model Basin Report 933 (Dec 1958). A complete listing of references is given on page 19.

[2] Pode, L., "Tables for Computing the Equilibrium Configuration of a Flexible Cable in a Uniform Stream," David Taylor Model Basin Report 687 (Mar 1951).

[3] Pode, L. and L. Rosenthal, "Cable Function Tables for Small Critical Angles," Supplement to David Taylor Model Basin Report 687 (Sep 1955).

assumed to be constant and unaffected by the curvature of the cable, and the cable is assumed to be inelastic and completely flexible.

A first calculation was based on achieving a depth of 250 feet at 8 knots with 450 feet of wetted cable length. The specified and derived quantities necessary for making the computation are listed in Table 2 along with the resulting surface cable angle ϕ and required tension T_o.

The drag per unit length of cable when normal to the free stream R was computed from the equation:

$$R = C_R\ (\rho/2)\ d\ U^2 \tag{1}$$

A value of 1.5 was used for the normal drag coefficient C_R. This selection was based on the results presented by Gibbons and Walton.[4]

TABLE 2 – CABLE QUANTITIES FOR SPECIFIED PERFORMANCE

Input Quantities	
Velocity U, knots	8
Towing Depth, feet	250
Wetted Cable Length, feet	450
Drag per Unit Length of Cable When Normal to an 8-Knot Stream R, pounds per foot	11.98
Cable Critical Angle ϕ_c, degrees	10.0
Resulting Quantities	
Cable Tension at Body T_o, pounds	1702
Cable Angle at Body with Respect to Free-Stream ϕ_o, degrees	75.0
Cable Angle at Water Surface ϕ, degrees	19

The cable angle at the body ϕ_o is equal to the arctangent of the ratio of body downforce to body drag. A value of 75 degrees for this angle was arbitrarily selected as being a reasonable estimate and an easily attainable value for the vehicle.

WING DESIGN

The overall span of the wing (and thus the wing size) should be minimized to facilitate vehicle handling, and the mass and moments of inertia of a towed vehicle should be kept small for good dynamic performance. Therefore, both the mass and the mass distribution are of concern. Consequently, in order to concentrate the mass of the vehicle toward the longitudinal center, the nose and tail sections of the housing should be kept as light as possible. Initial

[4] Gibbons, T. and C.O. Walton, "Evaluation of Two Methods for Predicting Towline Tensions and Configurations of a Towed Body System Using Bare Cable," David Taylor Model Basin Report 2313 (Dec 1966).

investigations of the volume inside the middle section indicated that enough space would be available for lead ballast to make the vehicle weigh approximately 400 pounds in seawater.

For a cable tension T_o of 1702 pounds acting at an angle ϕ_o of 75 degrees, the required total downforce is:

$$F_V = T_o \sin \phi_o = 1644 \text{ pounds} \tag{2}$$

If the body weight in water (W–B) is 400 pounds, then the wing must produce a downward force of:

$$L = F_V - (W-B) = 1244 \text{ pounds} \tag{3}$$

The lift developed by a wing which intersects a cylindrical body may be estimated by[5]

$$L = 1/2 \rho U^2 C_L A \left[1 - \frac{4r^2}{b^2 \left(1 + \frac{4e^2}{b^2}\right)} \right] \tag{4}$$

The quantity shown in brackets in Equation (4) is an approximate factor which accounts for the wing-body interference. For most applications, $b \gg e$ and the ratio $4e^2/b^2$ may be neglected. Then with the substitution $b^2 = aA$, where a is the effective aspect ratio of the wing, Equation (4) becomes:

$$L = 1/2 \rho U^2 C_L A \left(1 - \frac{4r^2}{aA}\right) \tag{5}$$

Equation (5) may be rearranged to solve for the wing planform area A. The result is:

$$A = \frac{L}{1/2 \rho U^2 C_L} + \frac{4r^2}{a} \tag{6}$$

The lift coefficient C_L may be calculated from the relation[1]

$$C_L = \left[\frac{C_{\ell\alpha} a}{\cos \Lambda \sqrt{\frac{a^2}{\cos^4 \Lambda} + 4 + \frac{57.3 C_{\ell\alpha}}{\pi}}} \right] \alpha + \frac{C_{D_C}}{a} \left(\frac{\alpha}{57.3}\right)^2 \tag{7}$$

For a wing with a rectangular planform, $\Lambda = 0$ and $C_{D_C} = 1.7$.[1] If these values are substituted into Equation (5) along with a value of a = 3.0 for the wing aspect ratio, then:

$$C_L = 0.055\alpha + 0.00017\alpha^2 \tag{8}$$

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[5] Gay, S.M., Jr., "The Hydrodynamic Design of a Cable-Towed Body Suitable for Economical Production," David Taylor Model Basin Report 1389 (Dec 1959).

A relatively high effective angle of attack α of 11 degrees was chosen to minimize the required wing area. However, this angle is substantially below the predicted stall angle for a wing with this section shape and aspect ratio. Then using Equations (6) and (8), the wing area A required to achieve a wing downforce L of 1244 at 8 knots is 11.9 square feet.

The wing was mounted to the vehicle housing as shown in Figure 3. Note that a "wedge" is sandwiched in between the towblock and the top surface of the wing. This wedge is necessary if the bolts which secure the wing and towblock to the mounting bracket are to seat properly, i.e., the upper surface of the towblock base and the lower surface of the mounting bracket should be parallel.

STABILITY ANALYSIS

The dynamic performance of a towed vehicle is influenced by many interacting body and cable hydrodynamic and inertial parameters. At present, however, neither the necessary magnitudes nor even the relative significance of these factors is sufficiently well known to design a dynamically optimum cable-towed system. Consequently, tail surfaces are generally designed for static stability only with some static margin that past experience has indicated will provide satisfactory dynamic behavior.

Eames[6] indicates that only a small static pitch margin should be provided in the vertical plane, so that the vehicle will respond primarily in heave to surges in the cable tension. He recommends that for a body with stabilizer surfaces, the slope of the pitching moment curve versus angle of pitch should be approximately one-half the magnitude (but of opposite sign) of that of the bare body. In the horizontal plane, however, the slope of the yawing moment curve versus yaw angle for a body with stabilizer surfaces should have a magnitude equal to or greater (but of opposite sign) than that of the bare body so that significant body yaw angles will not occur.

Stability in Pitch

The slope of the nondimensional pitching moment curve versus the angle of pitch for a winged body without stabilizers may be expressed by the following equation as given by Multhropp[7] and by Perkins and Hage[8]

$$\frac{\partial M'_b}{\partial \theta} = \frac{1}{q\ell_b^{\,3}} \frac{\partial M_b}{\partial \theta} = \frac{\pi}{2\ell_b^{\,3}} \Sigma W_f^{\,2} \frac{\partial \beta}{\partial \theta} \Delta x_\eta \tag{9}$$

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[6]Eames, M.C., "Experimental Bodies for High-Speed Underwater Towing Research," Naval Research Establishment (Canada) Report 66/7 (May 1967).

[7]Multhropp, V.H., "Aerodynamics of the Fuselage," NACA TM 1036 (Dec 1942).

[8]Perkins, C.D. and R.E. Hage, "Airplane Performance Stability and Control," John Wiley and Sons, Inc., New York (1967).

To evaluate the quantities in Equation (9), the vehicle length is divided into segments, not necessarily of the same length. Then for segments behind the wing[8]

$$\frac{\partial \beta}{\partial \theta} = \frac{x_\eta}{\ell_{wt}} \left(1 - \frac{d\epsilon}{d\alpha}\right) \tag{10}$$

For segments in front of the wing, the value of $\partial\beta/\partial\theta$ may be evaluated from curves presented in Perkins and Hage.[8] In the region between the wing leading and trailing edges, $\partial\beta/\partial\theta$ is considered to be equal to zero.

The horizontal, faired crossbeam that is attached to the forward end of the sting will produce a destabilizing moment when the body pitches. The slope of the nondimensional pitching moment curve versus angle of attack of the sting which is assumed to be in undisturbed flow is[8]

$$\frac{\partial M'_s}{\partial \theta} = (C_{L_\alpha})_s \frac{\ell_s A_s}{\ell_b{}^3} \tag{11}$$

The slope of the nondimensional pitching moment curve versus angle of pitch for stabilizing fins may be expressed by the equation[5,8]

$$\frac{\partial M'_t}{\partial \theta} = -(C_{L_\alpha})_t \frac{\ell_t A_t}{\ell_b{}^3} (1 - \dot{\eta}_t) \left(1 - \frac{d\epsilon_t}{d\alpha}\right) \tag{12}$$

For an X-tail configuration,[9]

$$(C_{L_\alpha})_t = (C_{L_\alpha})_{\Gamma=0} K \cos^2 \Gamma \tag{13}$$

The tail planform area required to make the slope of the total pitching moment curve versus pitch angle equal to −0.5 times that of the bare body is

$$\frac{\partial M'_b}{\partial \theta} + \frac{\partial M'_s}{\partial \theta} + \frac{\partial M'_t}{\partial \theta} = -0.5\left(\frac{\partial M'_b}{\partial \theta} + \frac{\partial M'_s}{\partial \theta}\right) \tag{14}$$

Equations (9), (11), (12), and (13) are substituted into Equation (14) to obtain the required tail area

$$A_t = 1.5 \frac{\pi/2 \left[\Sigma W_f{}^2 \frac{\partial \beta}{\partial \theta} \Delta x_\eta\right] + \left[\frac{2\pi}{(1 + 3/a_s)}\right] \ell_s A_s}{\left[\frac{2\pi}{(1 + 3/a_t)}\right] K \ell_t (\cos^2 \Gamma)(1 - \eta_t)\left(1 - \frac{d\epsilon_t}{d\alpha}\right)} \tag{15}$$

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[9]Purser, P.E. and J.P. Campbell, "Experimental Verification of a Simplified Vee-Tail Theory and Analysis of Available Data on Complete Models with the Vee Tails," NACA Report 823 (1945).

The upwash angle at the tail ϵ_t due to the wing was estimated by using the method of Silverstein and Katzoff.[10] For a wing with an elliptical spanwise load distribution and an aspect ratio of 3.0, the upwash angle at the tails for this vehicle is:

$$\epsilon_t = 5.4\, C_L \text{ degrees} \tag{16}$$

If Equation (8) is substituted into Equation (16), then after differentiating with respect to the wing angle of attack α, the result is

$$\frac{d\epsilon_t}{d\alpha} = 0.0018\alpha + 0.297 \tag{17}$$

For an angle of attack of 11 degrees, $d\epsilon_t/d\alpha = 0.32$.

The dynamic pressure loss at the tail η_t due to the wing wake is zero since the tail is located outside of the wing wake.[10] The value of the tail interference factor K was determined from curves in Purser and Campbell.[9] For a tail with a rectangular planform and an aspect ratio of 3.8, K = 0.70. The tail surface area equation values were calculated as follows:

ℓ_t = 4.75 feet (from body geometry)
Γ = 45 degrees (selected tail orientation)
a_t = 3.8 (selected geometry)

$\Sigma W_f^2 \dfrac{\partial\beta}{\partial\theta} \Delta x_\eta$ = 11.09 cubic feet (from Equation (10), from information in Perkins and Hage,[8] and from body geometry)

a_s = 2.37 (sting geometry)
ℓ_s = 3.9 feet (sting length)
A_s = 0.24 square feet (sting geometry)

Then from Equation (15), the required tail area A_t is 7.6 square feet.

Stability in Yaw

In the horizontal plane, wing effects on local sideslip angles at the body are assumed to be negligible and, therefore, the slope of the nondimensional yawing moment curve versus angle of yaw for the housing may be expressed by[8,11]

$$\frac{\partial N'_b}{\partial\psi} = \frac{2V\,(k_2 - k_1)}{\ell_b^{\,3}} \tag{18}$$

For an X-tail configuration, the yawing moment curve for the stabilizer is identical to the pitching moment curve, except that the upwash factor is replaced by a sidewash factor. Thus

[10] Silverstein, A. and S. Katzoff, "Design Charts for Predicting Downwash Angles and Wake Characteristics behind Plain and Flapped Wings," NACA Report 648 (1939).

[11] Munk, M.M., "The Aerodynamic Forces on Air Ship Hulls," NACA Report 184 (1924).

$$\frac{\partial N'_t}{\partial \psi} = -(C_{L_\alpha})_t \left(\frac{\ell_t A_t}{\ell_b^{\,3}}\right)(1 - \eta_t)\left(1 - \frac{d\sigma_t}{d\psi}\right) \qquad (19)$$

where σ_t is the sidewash angle at the tail due to body and wing interference flow. The sidewash is primarily dependent on the vertical location of the wing on the body; for a depressor wing mounted above the body centerline, the sidewash should be slightly stabilizing.[8] In any event, the sidewash angle is likely to be small and was therefore neglected in this analysis.

The resulting total yawing moment equation for the body with wing and tail is:

$$\left(\frac{\partial N'}{\partial \psi}\right)_{total} = \frac{2V(k_2 - k_1)}{\ell_b^{\,3}} - (C_{L_\alpha})_t \left(\frac{\ell_t A_t}{\ell_b^{\,3}}\right)(1 - \eta_t) \qquad (20)$$

The factor $(k_2 - k_1)$ may be evaluated by using data from Munk.[11] For a vehicle with a fineness ratio of 4.35 (see DEVELOPMENT OF GENERAL VEHICLE FORM), the value of $(k_2 - k_1)$ is 0.80. The volume V of the housing was estimated to be 13.04 cubic feet. Then from Equation (18):

$$\frac{\partial N'_b}{\partial \psi} = 0.048 \qquad (21)$$

Also, since the factors (C_{L_α}), ℓ_t, A_t, and η_t in Equation (20) have the same values as those in Equation (12),

$$\left(\frac{\partial N'}{\partial \psi}\right)_{total} = -0.055 \qquad (22)$$

By comparing Equations (21) and (22) and using the criterion that the magnitude of $(\partial N'/\partial \psi)_{total}$ should be greater than $-1.0\ \partial N'_b/\partial \psi$, the previously developed stabilizer geometry should provide satisfactory stability in yaw as well as in pitch.

SUMMARY OF FINAL DESIGN

The general physical characteristics of the towed vehicle are listed in Table 3. The weights given for the vehicle are without the instrumentation, which is expected to weigh approximately 70 pounds.

EXPERIMENTAL APPARATUS AND PROCEDURE

For the basin experiments, the vehicle was suspended on a 9-foot length of 0.528-inch-diameter, double-armored electromechanical cable. The measurement apparatus consisted of a tension dynamometer and potentiometer pendulum mounted on a towing beam of the carriage in the arrangement shown in Figure 4. An additional pendulum potentiometer inside the vehicle housing was used to monitor the pitch angle of the body during the

TABLE 3 – PHYSICAL CHARACTERISTICS OF THE VEHICLE

Complete Configuration	
Overall Length, feet	9.5
Weight in Air without Instrumentation, pounds	1367
Weight in Seawater without Instrumentation, pounds	322
Estimated Weight of Instrumentation, pounds	70
Housing	
Length, feet	7.55
Maximum Diameter, feet	1.73
Fineness Ratio	4.35
Wing	
Section Shape: TMB-07507515	
Incidence Angle (leading edge down), degrees	11.0
Sweep Angle of Quarter-Chord, degrees	0
Dihedral Angle, degrees	0
Span, feet	6.0
Root Chord, feet	2.0
Tip Chord, feet	2.0
Planform Area, square feet	11.9
Taper Ratio	1.0
Aspect Ratio	3.0
Tails	
Section Shape: Rectangular with Semicircular Leading Edge	
Incidence Angle, degrees	0
Sweep Angle of Quarter-Chord, degrees	0
Span (measured along surface), feet	3.8
Chord, feet	1.0
Planform Area, square feet	7.6
Taper Ratio	1.0
Aspect Ratio	3.8

experiments. Output from each of the measurement transducers was transmitted to a digital recorder on the carriage. The tension measurement system used in these experiments is accurate to ±50 pounds. The pendulums have accuracies of ±0.5 degree.

The experiments were conducted in the high-speed basin at speeds ranging between 3 and 12 knots. During each experimental run, cable tension, cable angle, and body pitch angle were measured, and the vehicle was observed for steadiness and alinement. The trim tabs on the vehicle were set at zero angle throughout.

REDUCTION AND PRESENTATION OF DATA

The vehicle towed in a steady and stable manner at all experimental speeds. Some very slight horizontal perturbations of the vehicle were noted occasionally, but these damped out rapidly. The body maintained pitch angles of 0 ± 0.5 degree over the experimental speed range.

Since the towcable tensions and angles were measured at the juncture of the towcable with the carriage, the recorded values did not directly represent the vehicle hydrodynamic characteristics alone, i.e., they also contained the effects of the cable forces. Therefore, these combined characteristics were used as input to a computer program developed by Cuthill[12] for solving cable problems to resolve the cable tensions and angles at the body. The results were corrected for viscous effects of seawater by the method presented in Gay.[5]

The corrected towcable tension and angle data at the body are given graphically in Figure 5. At the design speed of 8 knots, the resolved body tension is 1675 pounds and the resolved cable angle is 74.3 degrees. This tension is equal to the design tension within the accuracy of the measurement system.

PREDICTION OF CABLE CONFIGURATIONS

The information presented in Figure 5 and the cable computer program described by Cuthill[12] were used to predict the equilibrium cable configurations, tensions, and angles over the speed range.

The drag coefficient (based on frontal areas) for the cable when normal to the stream and the ratio of tangential drag to normal drag were assumed to be 1.5 and 0.02, respectively. These values were taken from Gibbons and Walton.[4] Also, the cable diameter and weight in seawater were assumed to be 0.528 inch and 0.36 pound per foot, respectively. The predictions are presented graphically in Figures 6–8.

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[12]Cuthill, E., "A FORTRAN IV Program for the Calculation of the Equilibrium Configuration of r Flexible Cable in a Uniform Stream," NSRDC Report 2531 (Feb 1968).

CONCLUSIONS

Based on the results of the experiments and the predictions of cable configurations and forces, the following conclusions are drawn:

1. The vehicle adapted from a mobile mine case will exhibit adequate steady and stable towing behavior.

2. The desired depth of 250 feet at 8 knots will be obtained by using approximately 460 feet of wetted cable length.

3. With 460 feet of cable payed out while the vehicle tows at 8 knots, the cable angle at the ship relative to the direction of tow will be slightly less than 20 degrees at 8 knots (Figure 7). This should provide considerable decoupling of the towing ship and the towed vehicle, and this decoupling should tend to minimize motions of the vehicle due to the operation of the ship in waves.

4. Maximum tension under calm-water towing conditions will not exceed 3100 pounds at 11 knots (Figure 8). With a towcable breaking strength of approximately 20,000 pounds, an adequate margin of safety should be available for towing in a seaway.

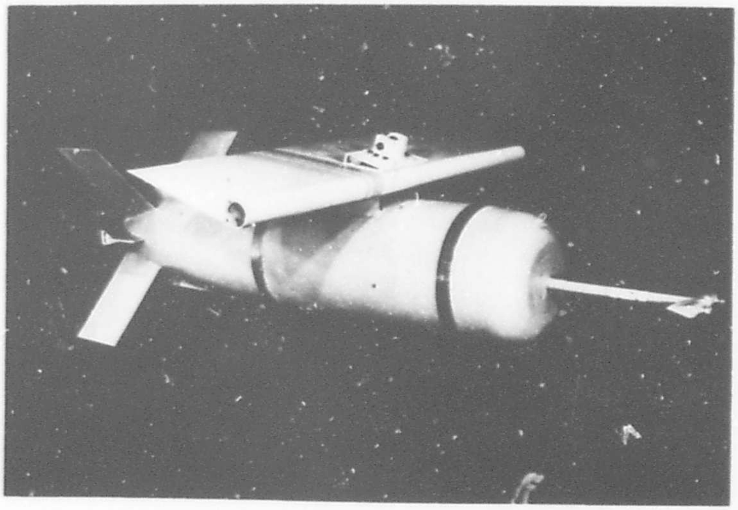

Figure 1a – Starboard Bow View

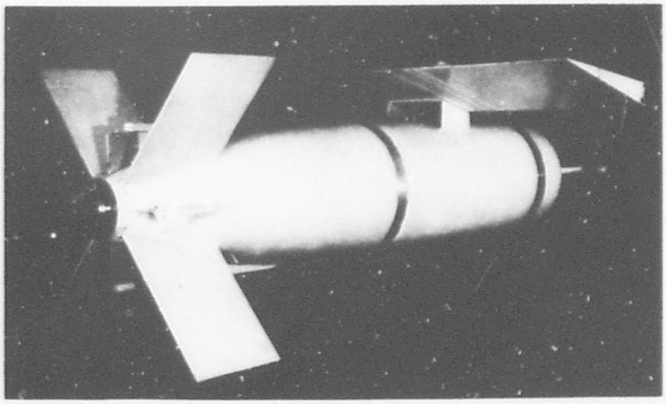

Figure 1b – Starboard Quarter View

Figure 1 – Views of the Towed Vehicle

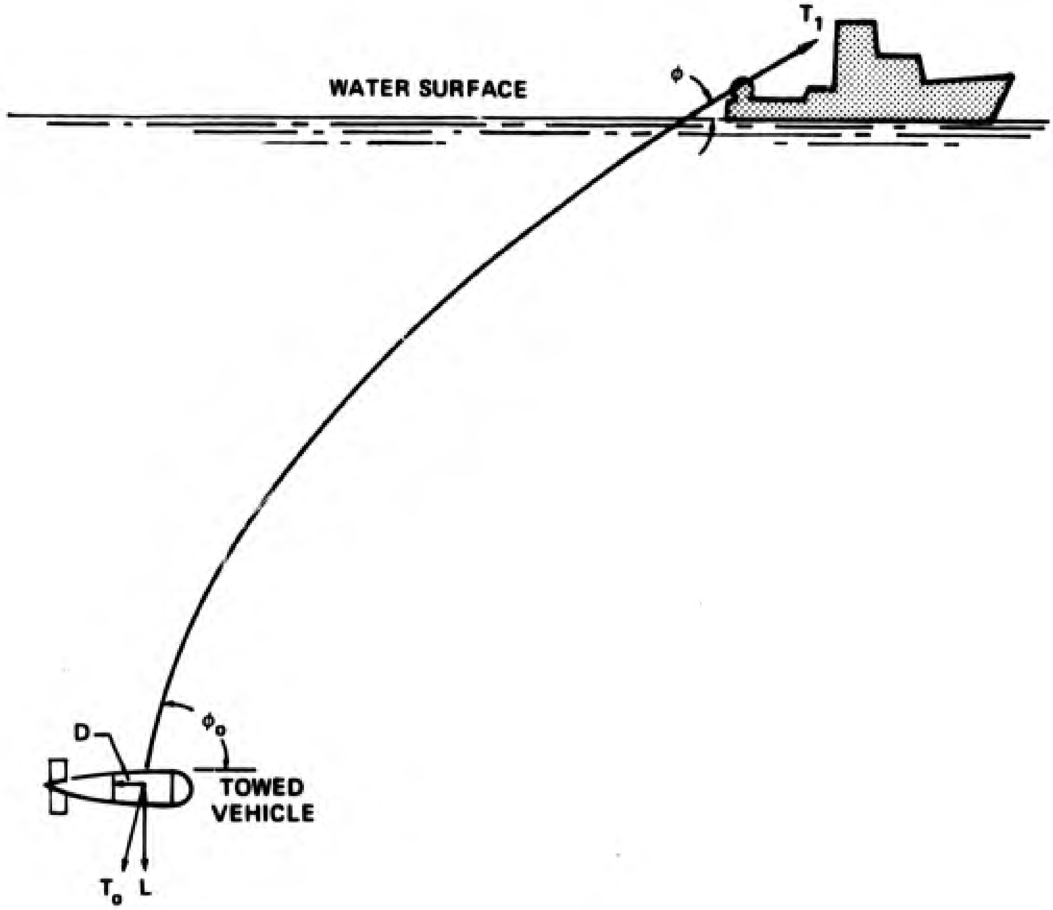

Figure 2 – Towing Configuration

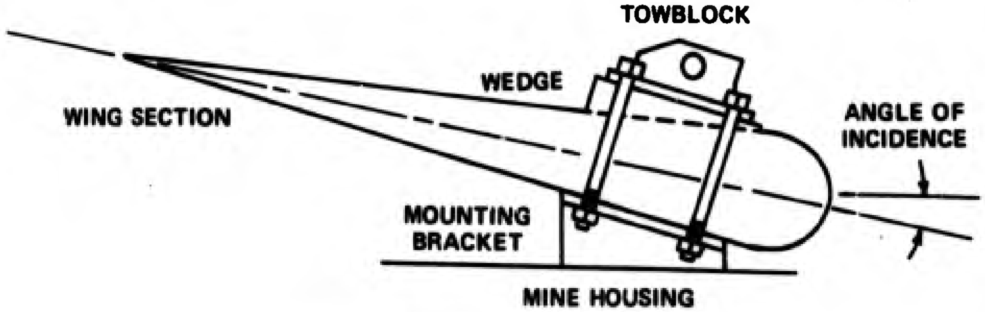

Figure 3 – Wing Mounting Arrangement

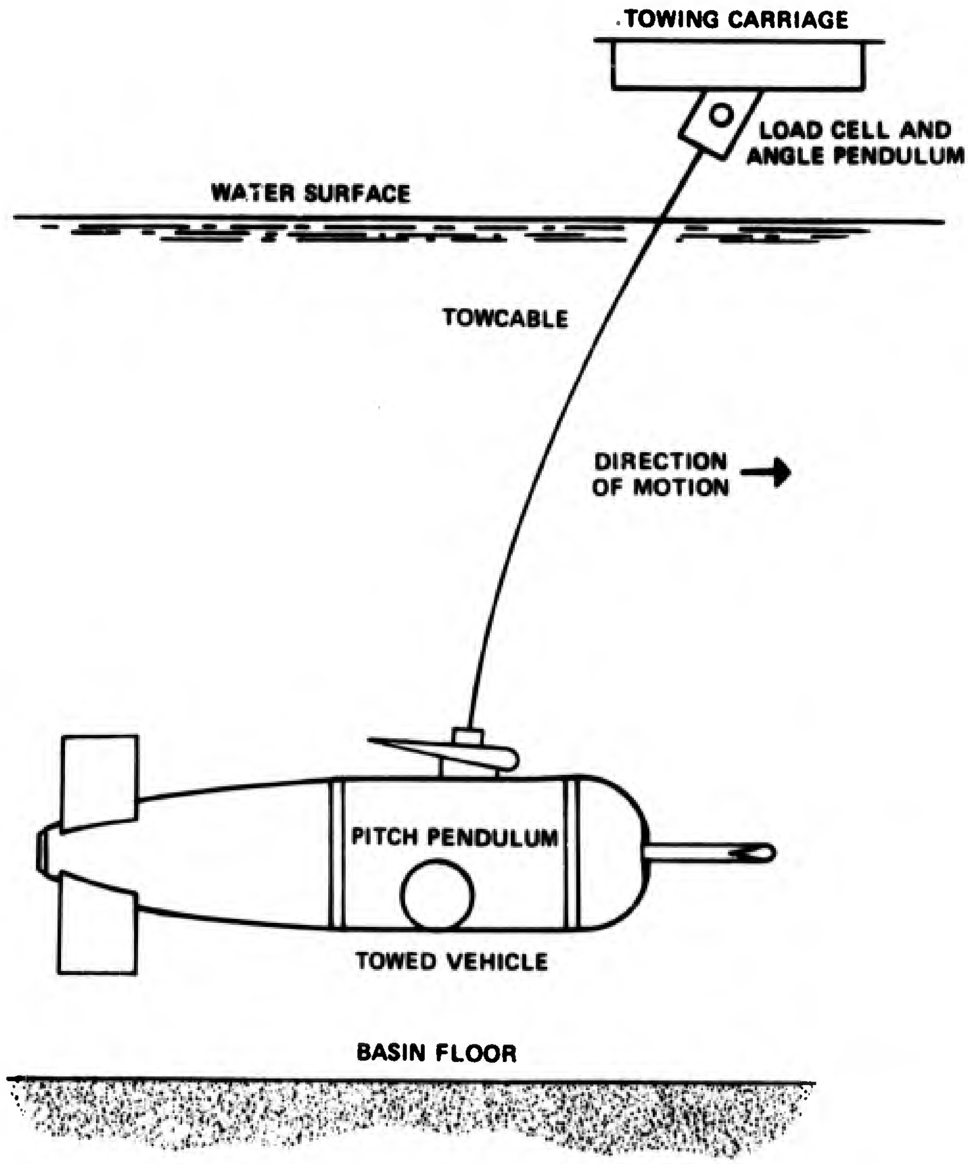

Figure 4 – Basin Towing Arrangement

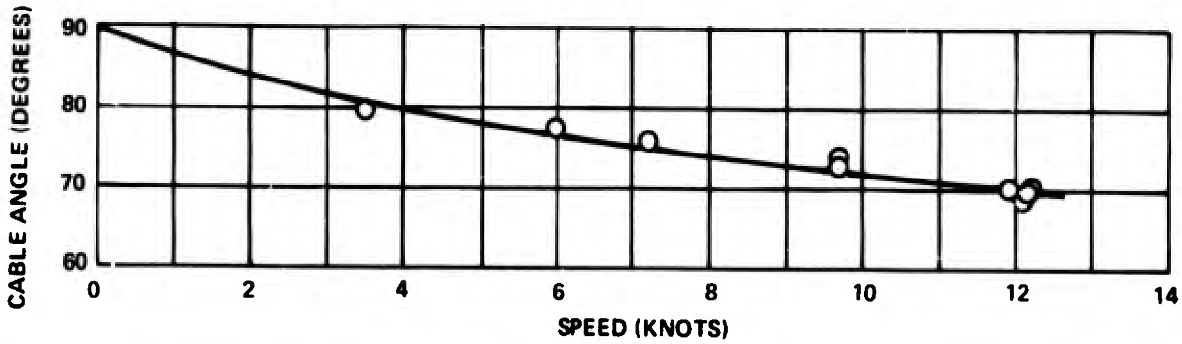

Figure 5a – Cable Angle

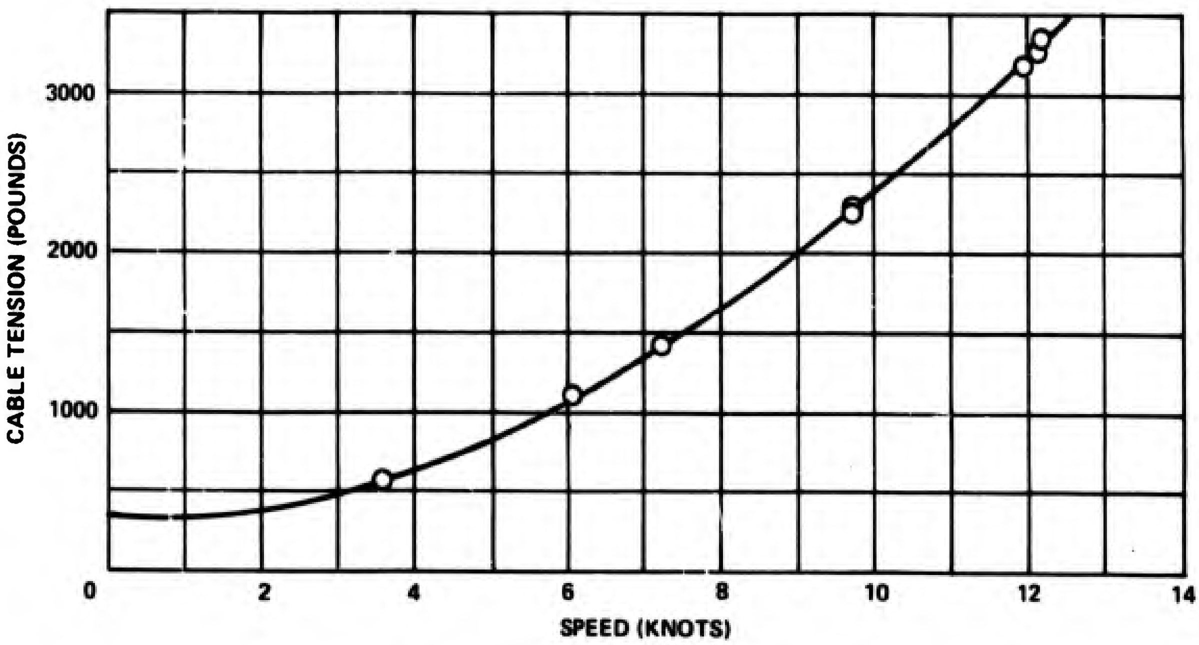

Figure 5b – Cable Tension

Figure 5 – Measured Cable Angle and Tension at the Body as Functions of Speed

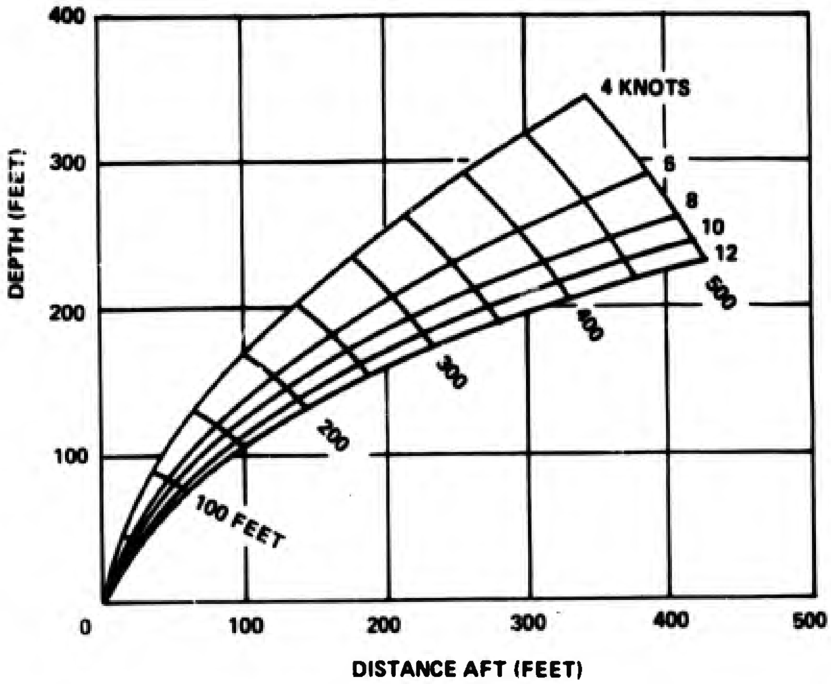

Figure 6 – Predicted Towcable Configurations for Various Speeds and Cable Scopes

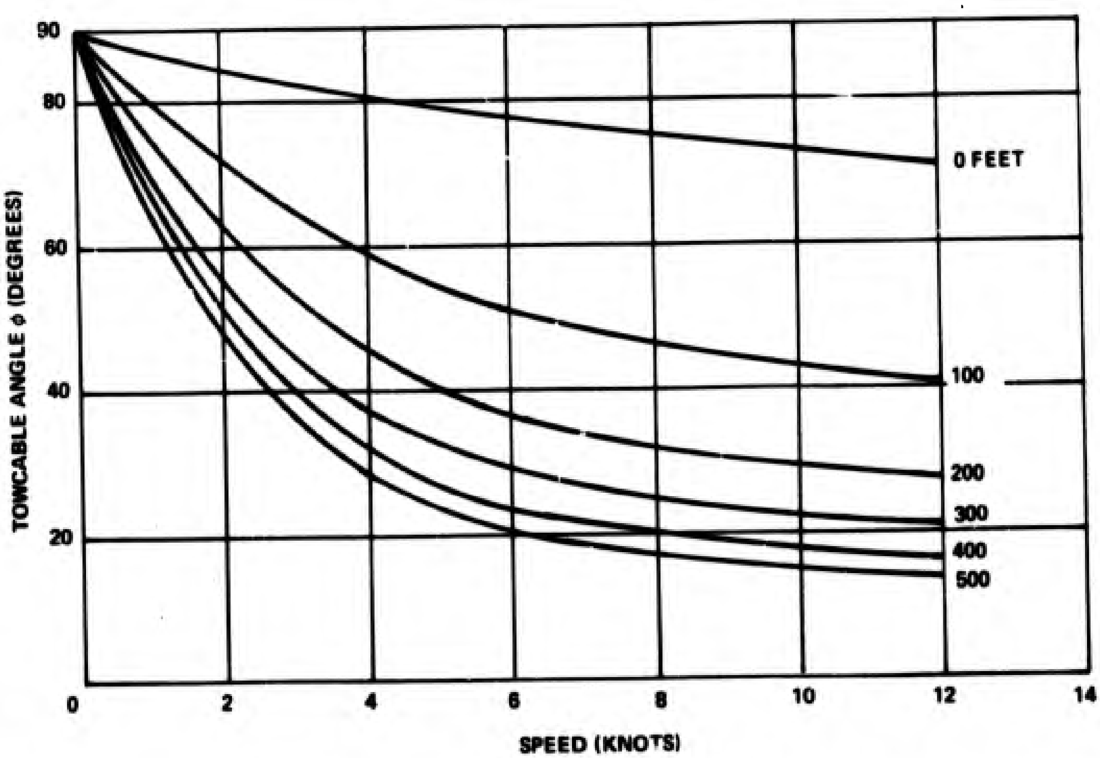

Figure 7 – Predicted Towcable Angle as a Function of Speed for Various Cable Scopes

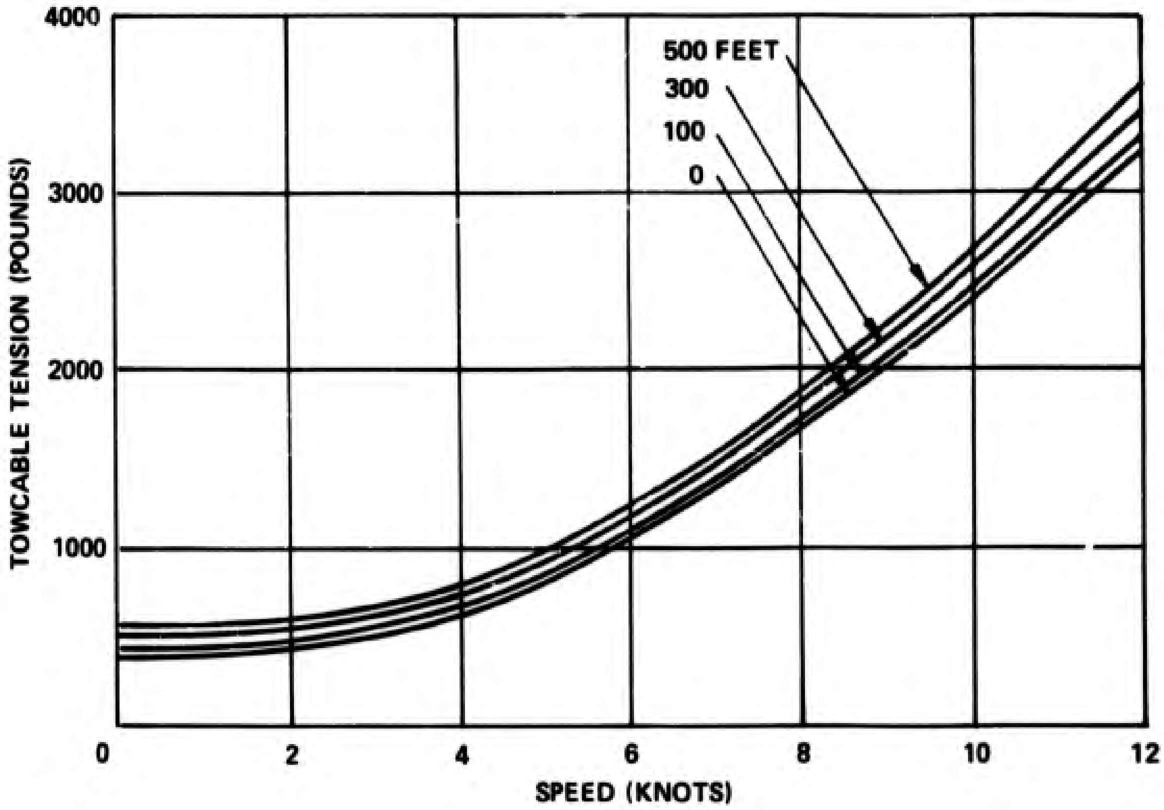

Figure 8 – Predicted Towcable Tension as a Function of Speed for Various Cable Scopes

REFERENCES

1. Whicker, L. F. and L. F. Fehlner, "Free-Stream Characteristics of a Family of Low-Aspect Ratio, All-Movable Control Surfaces for Application to Ship Design," David Taylor Model Basin Report 933 (Dec 1958).

2. Pode, L. "Tables for Computing the Equilibrium Configuration of a Flexible Cable in a Uniform Stream," David Taylor Model Basin Report 687 (Mar 1951).

3. Pode, L. and L. Rosenthal, "Cable Function Tables for Small Critical Angles," Supplement to David Taylor Model Basin Report 687 (Sep 1955).

4. Gibbons, T. and C.O. Walton, "Evaluation of Two Methods for Predicting Towline Tensions and Configurations of a Towed Body System Using Bare Cable," David Taylor Model Basin Report 2313 (Dec 1966).

5. Gay, S.M., Jr., "The Hydrodynamic Design of a Cable-Towed Body Suitable for Economical Production," David Taylor Model Basin Report 1389 (Dec 1959).

6. Eames, M.C., "Experimental Bodies for High-Speed Underwater Towing Research," Naval Research Establishment (Canada) Report 66/7 (May 1967).

7. Multhropp, V.H., "Aerodynamics of the Fuselage," NACA TM 1036 (Dec 1942).

8. Perkins, C.D. and R.E. Hage, "Airplane Performance Stability and Control," John Wiley and Sons, Inc., New York (1967).

9. Purser, P.E. and J.P. Campbell, "Experimental Verification of a Simplified Vee-Tail Theory and Analysis of Available Data on Complete Models with the Vee Tails," NACA Report 823 (1945).

10. Silverstein, A. and S. Katzoff, "Design Charts for Predicting Downwash Angles and Wake Characteristics behind Plain and Flapped Wings," NACA Report 648 (1939).

11. Munk, M.M., "The Aerodynamic Forces on Air Ship Hulls," NACA Report 184 (1924).

12. Cuthill, E., "A FORTRAN IV Program for the Calculation of the Equilibrium Configuration of a Flexible Cable in a Uniform Stream," NSRDC Report 2531 (Feb 1968).