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PRELIMINARY DEVELOPMENT AND TEST OF A CABLE-TOWED OCEANOGRAPHIC INSTRUMENTATION SYSTEM

Prepared under Contract Nonr 3201(00)
Sponsored by the
Office of Naval Research

Technical Report 6634-4

SYSTEMS ENGINEERING DIVISION
PNEUMODYNAMICS CORPORATION
BETHESDA, MARYLAND

Technical Report 6634-4

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December 1962

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ACKNOWLEDGMENTS

The contributions and services of a number of individuals and organizations made possible the successful completion of the project reported herein. The authors wish to express their appreciation, not only for the services provided, but for the personal interest and cooperation of the individuals involved.

In particular, the authors are indebted to Lt. Cdr. T. H. Sherman of the Office of Naval Research for his continued encouragement and support; the U. S. Naval Research Laboratory for the use of their pressure-test facilities; Mr. J. W. Danehower of the Bureau of Ships and Mr. Seymour Gross of the U.S. Navy Underwater Sound Laboratory for the use of the test-fairing and cable; Mr. H. Beck of the Hudson Laboratories of Columbia University for the loan of several depth sensors, as well as for discussions relative to instrumentation; Messrs. G. Springston, A. Brisbane, and T. Gibbons of the David Taylor Model Basin for consultations concerning cable fairing and the conduct of the depressor tow tests; the Portsmouth Naval Shipyard for the installation of the test gear; the officers and men of the <u>U.S.S. Yamacraw</u> for their cooperation, patience, and aid during the sea trials; and Mr. A. S. Burrows of the Bell Telephone Laboratories for his valuable assistance with instrumentation during sea trials.

FOREWORD

This report is the fourth in a series concerned with the development of a ship-towed instrument system capable of measuring selected oceanographic parameters in the vertical profile. This system differs from most other "profile" type developments in that measurements may be taken while underway, permitting determination of the "space-wise" variation of oceanic characteristics.

Reports previously issued under this contract are:

Wm. M. Ellsworth: "General Design Criteria for Cable-Towed Body Systems Using Faired and Unfaired Cable; Systems Engineering Division, Pneumo-Dynamics Corporation, Technical Report 6634-1, October 1960.

Wm. M. Ellsworth and S. M. Gay, Jr.: Preliminary Design of a Cable-Towed Oceanographic Instrumentation System; Systems Engineering Division, PneumoDynamics Corporation, Technical Report 6634-2, February 1961.

L. W. Bonde: Investigation of a Tractor-Type Winching Machine for Handling Faired Cables and In-Line Instrument Modules; Systems Engineering Division, PneumoDynamics Corporation, Technical Report 6634-3, July 1962.

In this report, the development of the submerged towed elements of the system is discussed. A preliminary investigation of a winching device is described.

SUMMARY

The components of a cable-towed oceanographic instrumentation system have been developed and tested to demonstrate hydromechanical feasibility. As part of the project, a depressor, instrument module, and devices for attaching trailing-type fairing to the tow cable were developed and tested. Designs for these elements are presented. The investigations also included a basic study of specialized winching equipment capable of handling rigid "lumps" spaced intermittently along the cable. Representative elements of the system, including the winching equipment, were tested at sea.

The results of the developmental and sea tests verify the hydromechanic feasibility of towing the system to a depth of 5,000 feet at speeds of seven or eight knots. The capability of the winch in handling and storing cables with intermittent, rigid "lumps" was also proven.

With feasibility established, development should be continued to finalize the design of the winching and handling equipment. Design studies on the data-transmission system should be initiated.

INTRODUCTION

In May 1960, under Contract Nonr-3201(00), the Systems Engineering Division initiated a study of the requirements for a cable-towed oceanographic measurement system which would permit simultaneous measurements of selected oceanic characteristics at a number of depths to 5000 feet. The results of this study were presented in two technical reports (1)* and (2), the latter of which was concerned with the requirements for the towed elements and shipboard handling and storage of such systems. Preliminary requirements for the instrumentation, including a method for sequential interrogation of as many as 128 sensors using only a limited number of electrical conductors, were also presented to illustrate the feasibility of transmitting such an amount of data through a cable small enough to permit the use of acceptably sized handling and storage equipment. A recommendation was made that further developmental work be directed toward demonstration of feasibility of hydromechanical specifications, with minor emphasis on instrumentation problems.

Hydromechanical feasibility depends on the satisfaction of certain conditions:

1. The submerged elements of the system must be towable at the design speed range;

---

* Numbers in parentheses refer to references given in Appendix V.

a. When towed with design values of cable scope and speed, the *in situ* value of the depth must not be substantially less than 5,000 feet and the *in situ* value of the cable tension must not be substantially greater than 15,000 pounds.

b. The components must be structurally adequate; and

c. The tow must be stable.

2. Winching equipment must demonstrate satisfactory:

a. Inhaul and payout of the faired cable without damage to the fairing or cable; and

b. Englutment and passage of the instrument modules without slippage or damage.

It was considered that hydromechanical feasibility could be most effectively demonstrated with an engineering model constructed of full-scale elements of the prototype system. Development and test of representative quantities of each major element of the system was accordingly undertaken. Arrangements were made to test the resulting model at sea with representative winching equipment.

The results of this program constitute the subject matter of this report. Since this document is essentially an extension of (2), liberal reference has been made to that report for explanation of the underlying philosophy and design approach. This report presents only component design considerations in those few areas where significant changes have been made in philosophy or approach.

GENERAL SYSTEM DESCRIPTION

The system, shown diagrammatically in Figure 1, consists of a number of lengths of three-quarter-inch-diameter cable coupled by pipe-like housings. Provision is made for instrumentation in the central sections of the housings, with cable terminals and appropriate electrical fittings in the end pieces.

The cable is covered with a free-swiveling, hydrodynamic fairing to reduce drag and vibration; the instrument modules are also faired. A core-space, to accommodate any suitable electrical cable, is available within the load-carrying cable armor.

The cable-fairing-module assembly is retained at proper depth by a depressor, which develops the requisite depressing force by a combination of weight and hydrodynamic reaction.

Shipboard handling is accomplished by an endless-track-type winch for systems with rigid instrument modules distributed along the faired cable, or by a twin-load-drum winch for systems lacking modules.

The faired cable is delivered to the stowage area under a low tension, permitting storage in multiple layers on a drum or in a well.

For convenience, the elements of the system have been grouped as follows:

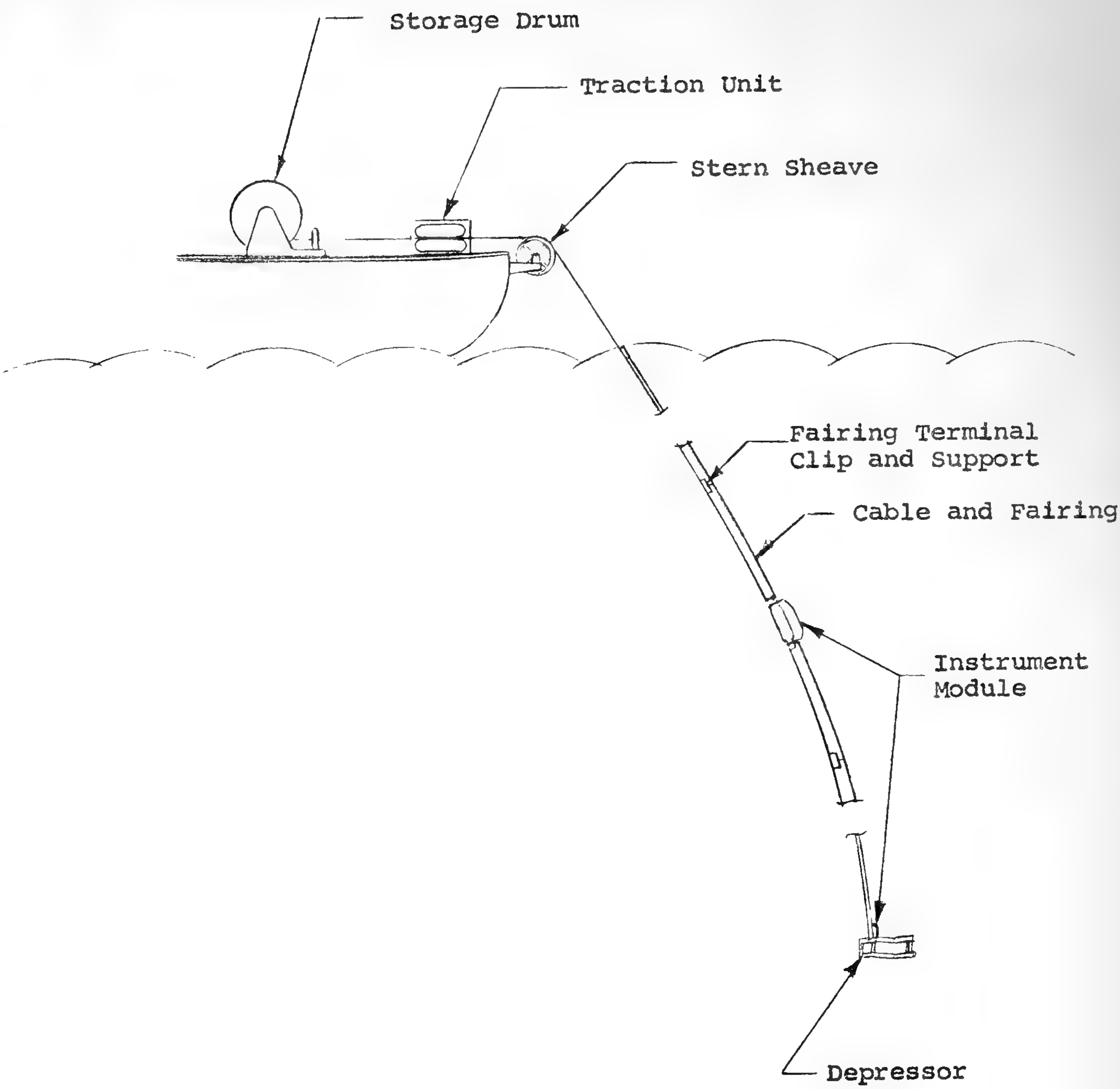


Figure 1 - Generalized Arrangement for a Cable-Towed Oceanographic Instrumentation System

1. Towed system: all mechanical elements that are placed overboard;

2. Shipboard-handling equipment: all items necessary for launching, retrieval, and storage of the towed elements; and

3. Instrumentation system: all equipment relevant to the sensing, transmission, storage, and display of oceanic data.

SYNOPSIS OF THE INVESTIGATION

The elements of the towed system were developed along the lines indicated in (2), and subjected to tests at sea. On the basis of the tests, minor modifications and improvements were effected, and the design of the towed elements finalized to the extent possible at this time; the resulting design is presented in the following section. The details of component development are discussed in Appendix I, and the description of the sea trial is given in Appendix II.

During the development, a major change in the length of faired cable was found necessary to attain the desired performance; this is discussed in Appendix III.

The feasibility of using an endless-track-type winching machine for handling the faired cable and instrument modules was investigated. Comprehensive tests, to determine the effect of cable fairing on the machine's tractive effort, were conducted with a commercially available unit; a sea test was carried out with a similar, but larger unit. The results of the first tests are reported in (3); the sea test is described in Appendix II of this report.

A very limited investigation was made of the problems of passing the towed system over the stern of a vessel and the final storage of the system aboard ship. The results of this investigation and a discussion of the winching problem are given in the section headed "Results of Handling System Investigation."

DESIGN DISCUSSION

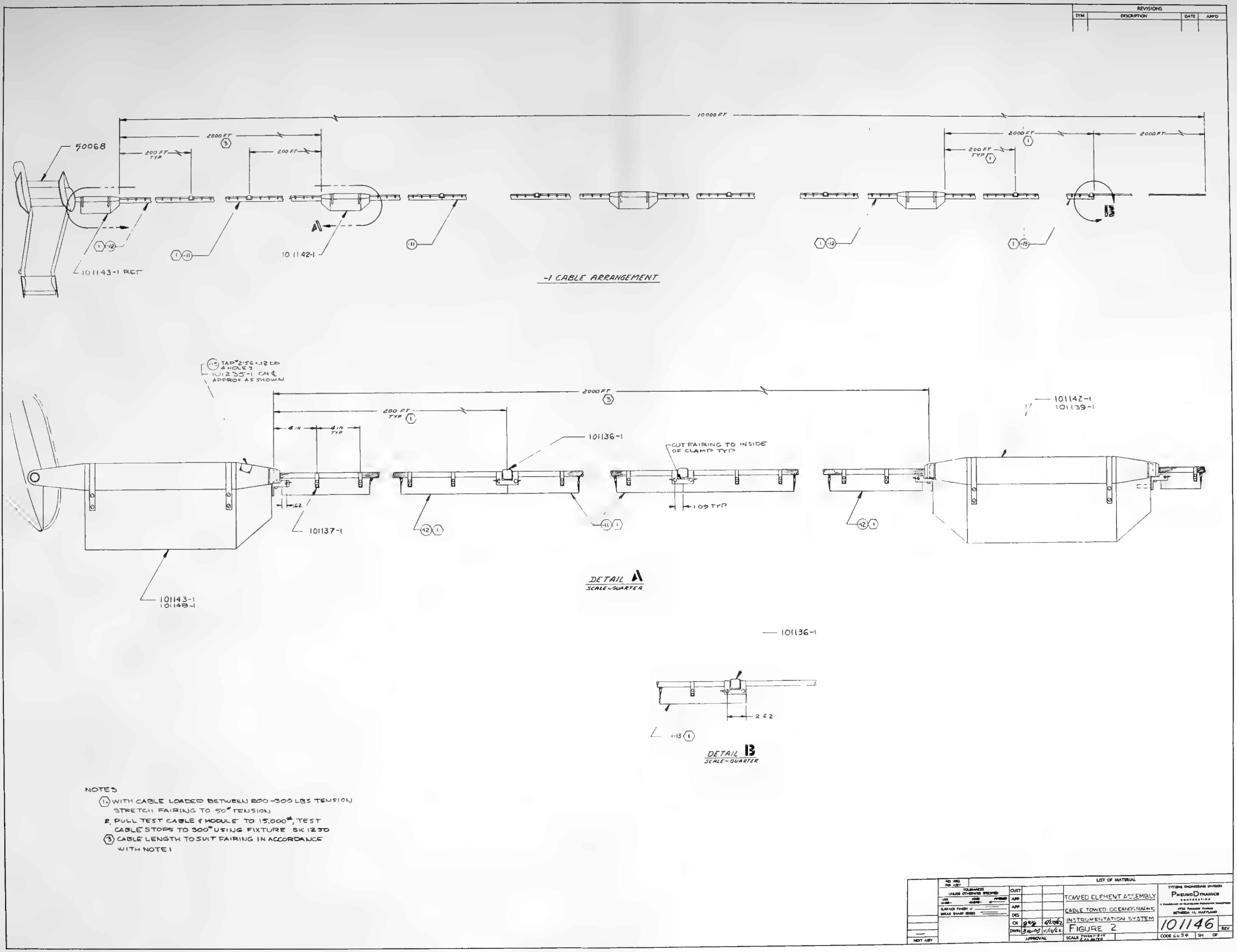
GENERAL

As illustrated on Figure 2, the towed-system design resulting from the development and tests to date consists of 8,000 feet of cable covered with a freely-swiveling rubber fairing and additional unfaired cable to permit rigging. The instrument modules are placed "in-line" and serve as electrical and structural links between the segments of cable. The depressor is attached to the bottommost module. In the discussion of the components that follows, the relevant SED Assembly Drawing number for each component is listed to the right of each heading.

DEPRESSOR — SED Drawing No. 50068

The depressor, shown on Figure 3, has circular-arc-airfoil wing sections and flat-plate stabilizers. The structural members are fabricated of plate and bar stock, the edges of which are simply rounded where streamlining is required.

The structure consists of welded subassemblies bolted together to facilitate proper alignment. The two ballasting weights are cast in halves which are bolted together; each weight is then bolted to the forepart of the structure. If desired, the unit can be disassembled to facilitate shipping and storage. All nuts, bolts, and movable hardware are made of stainless steel.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SYM
DESCRIPTION
DATE
APP'D
10,000 FT
2000 FT
200 FT TYP
200 FT
50068
101143-1 REF
101142-1
A
B
-1 CABLE ARRANGEMENT
TAP #2-56 x .12 DP 4 HOLES
101235-1 ON ℄ APPROX AS SHOWN
101136-1
CUT FAIRING TO INSIDE OF CLAMP TYP
101142-1
101139-1
4 IN
4 IN TYP
.62
101137-1
1.09 TYP
101143-1
101148-1
DETAIL A
SCALE ~ QUARTER
101136-1
2.62
DETAIL B
SCALE ~ QUARTER
NOTES
1. WITH CABLE LOADED BETWEEN 200-300 LBS TENSION STRETCH FAIRING TO 50# TENSION
2. PULL TEST CABLE & MODULE TO 15,000#, TEST CABLE STOPS TO 300# USING FIXTURE SK 1230
3. CABLE LENGTH TO SUIT FAIRING IN ACCORDANCE WITH NOTE 1
LIST OF MATERIAL
NO. REQ PER ASSY
TOLERANCES UNLESS OTHERWISE SPECIFIED
SURFACE FINISH
BREAK SHARP EDGES
CUST
APP
APP
DES
CK
DWN
APPROVAL
NEXT ASSY
TOWED ELEMENT ASSEMBLY
CABLE TOWED OCEANOGRAPHIC
INSTRUMENTATION SYSTEM
FIGURE 2
SCALE
SYSTEMS ENGINEERING DIVISION
PNEUMODYNAMICS CORPORATION
BETHESDA 14, MARYLAND
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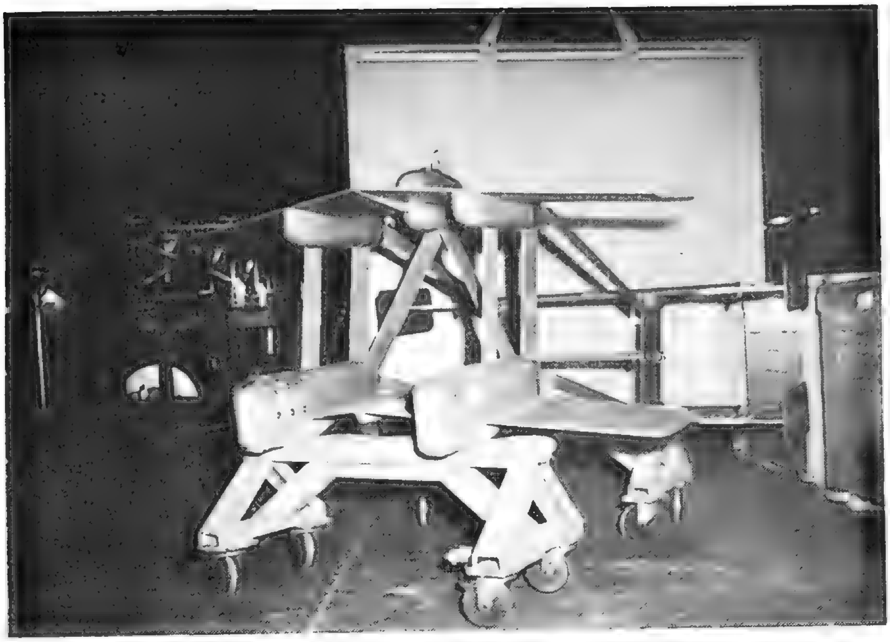
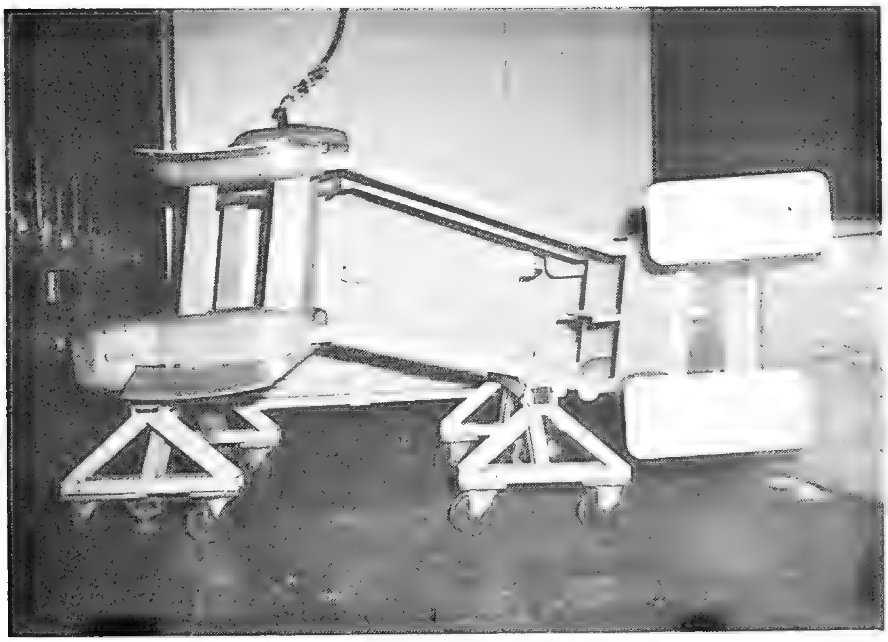




Depressor Developed for the Cable-Towed
Underwater Instrumentation System

Figure 3

High-drag areas, such as occur at the junction of lift-producing surfaces and other structural members, are enclosed in Fiberglas-reinforced plastic fairings. The horizontal and vertical stabilizers incorporate adjustable trim tabs so that the depressor's trim can be maintained or changed. The assembly is finished with International Orange enamel.

CABLE

The cable is of double-armoured construction with the electrical leads contained in an insulated core. Approximately one-half inch of the overall 3/4-inch-(nominal)-diameter is reserved for the electrical core. The overall breaking strength is approximately 45,000 pounds. These cables were originally developed for the petroleum industry for use in logging deep wells; they may be obtained from several major American cable manufacturers.

Details of the core construction have not yet been specified, pending final selection of the data-transmission scheme.

CABLE FAIRING AND ATTACHMENTS

SED Purchase Specification M.S. - 101

The fairing is David Taylor Model Basin type TF-84. The letters "TF" stand for "trailing fairing," the first numeral designates the ratio of the maximum thickness of the fairing to the diameter of the cable, in tenths, and

the last digit gives the overall fineness ratio, i.e., the ratio of the overall length of a right-section of the fairing plus cable, to the diameter of the cable. A trailing-type fairing with a thickness ratio of 8/10 and a fineness ratio of 4 was selected.

The fairing is constructed of a single-durometer rubber; the leading edge is reinforced with a five- or six-ply Dacron tire-cording to provide the necessary structural strength. The ends of the Dacron cording are turned back to form a loop, as shown in Figure 4.

The fairing is to be procured in maximum lengths of 200 feet; shorter lengths may be specified, as required, for lesser module spacings. It may be produced from Bureau of Ships Mold G60818.

CLIPS SED Drawing No. 101137

The fairing is secured to the cable, at intervals of four inches, by clips which encompass the cable and are free to swivel thereon. It is attached to the ears of the clips by rivets passed through the rubber body aft of the Dacron-cord reinforcement.

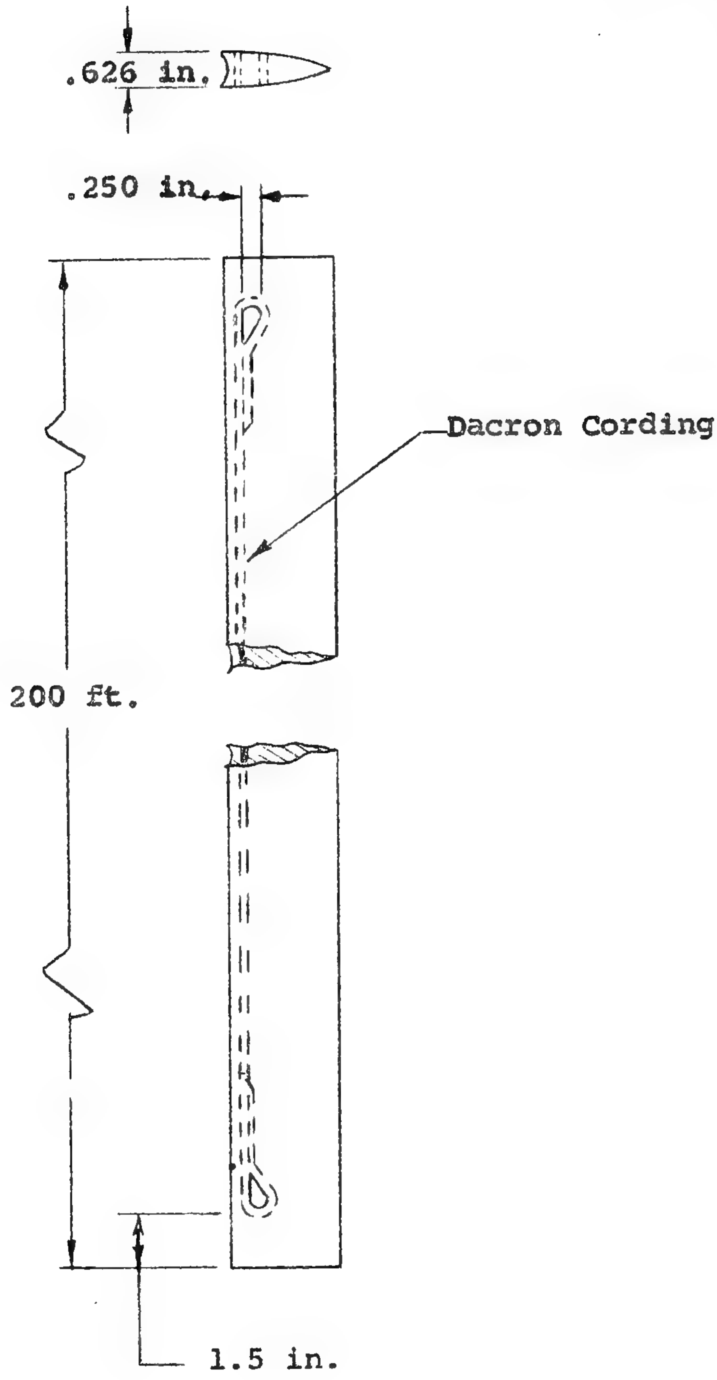


Figure 4 - Cable-Fairing, Trailing Type, General Arrangement

Each section of fairing is terminated at a hanger, to which it is secured by means of a rivet passed through the eye of the loop described above. An alternate hanger, used during sea trials, permits the Dacron cording to be terminated without the loop, thus affording a technique for effecting repairs in the event the fairing is ruptured or an eye pulled out.

The general arrangement of the developed hanger is shown in Figure 2. The principle of operation is similar to that of the alternate hanger shown on Figure 5. The hanger spans a stop which serves to transmit the hydrodynamic load of the fairing into the cable for those lengths of cable fitted with modules spaced more than 200 feet apart, For module spacings of 200 feet or less, the hanger is attached to the module and is free to swivel with respect thereto.

As may be seen on Figure 5, the stop is simply a split, friction-type clamp with sufficient clearance between the halves to permit them to bear on the cable when bolted together. A resilient elastomer pad is placed between the stop and cable to aid in maintaining the clamping pressure when the cable diameter decreases under load. A sand-impregnated polyurethane liner, developed by the David Taylor Model Basin, is used for this purpose. Self-locking bolts are used to secure the two halves.

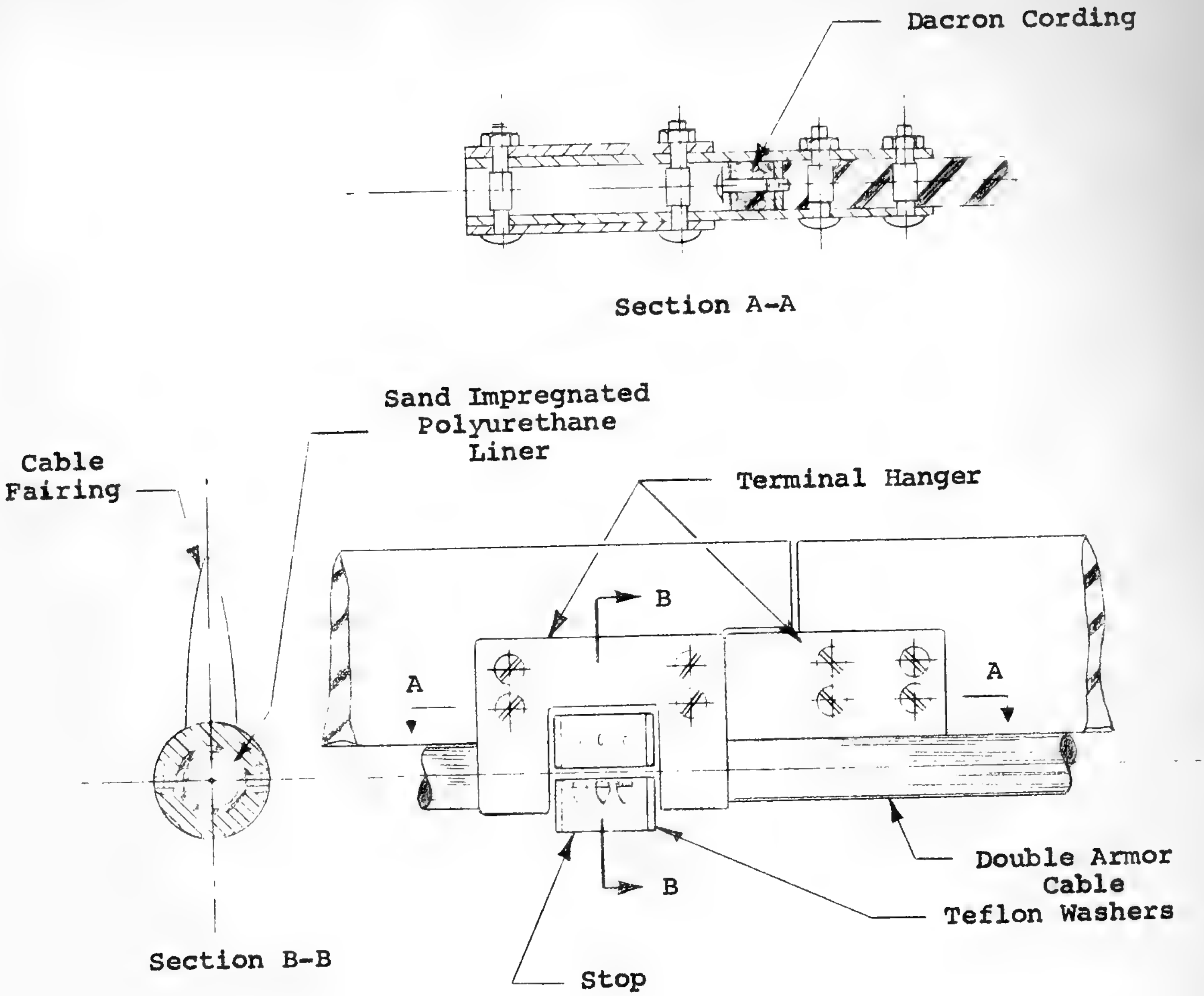
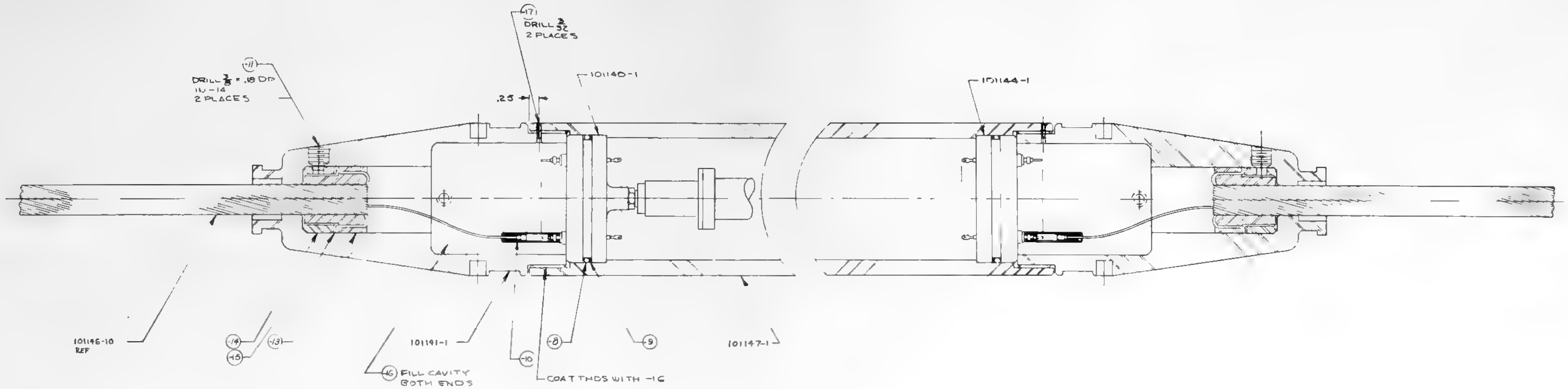


Figure 5 - Stop and Fairing Hanger

As previously stated, the instrument module is a pipe-like housing coupling the various lengths of 3/4-inch-diameter cable. In the present design, shown on Figure 6, the body is closed at each end by plug-type bulkheads with O-ring seals. The module shown has an end plug suitable for mounting a pressure sensor. Commercially available, bulk-head-type, single-pin electrical connectors provide the means for transmitting electrical signals. A commercially marketed cable fitting is used to terminate the individual load-carrying wires and transfer the loads to the cable-fitting housing, which may be rotated relative to the cable fitting, permitting attachment to the instrument case without rotation of the case relative to the cable.

The cable fairing terminates at each module, which is, itself, faired to reduce drag and vibration.

The module at the bottom of the line also serves as the depressor towstaff (Figure 2).



NOTES

1. -13, -14, & -15 PURCHASE FROM MECCA CABLE & SERVICE INC FOR CABLE .755 O.D., 24 WIRES INNER .062 DIA 24 WIRES OUTER .079 DIA. O.D. 1.625

NO. REQ PER ASSY	PART NO	DASH NO	DESCRIPTION	MATERIAL	MATERIAL SPEC
-	101146-10		CABLE REF		
2		-17	SPRING PIN	3/32 DIA × 3/8 LG, CRES	
AR		-16	GREASE	DOW CORNING #3	
2	COM'L	-15	WASHER	1	
2		-14	LOCK NUT	1	
2		-13	THIMBLE	1	
2		-11	SET SCREW SEL LOC	HALF DOG POINT, CRES TYPE 18-8 1/4-20 UNF-3A × 1/4	
32		-10	FEMALE BOOT	M-25-FB #8417	MECCA CABLE & SERVICE INC
2		-9	BACK UP RING	PARKER SEAL CO. 8-233-N300-9	
2	COM'L	-8	"O" RING	47-741 BUNA N	
1	101146-1		PLUG ASSY		
2	101141-1		CABLE FITTING		
1	101140-1		PLUG ASSY.		
1	101147-1		BODY		
		-1	MODULE ASSY		

LIST OF MATERIAL

TOLERANCES UNLESS OTHERWISE SPECIFIED

SURFACE FINISH

BREAK SHARP EDGES

NEXT ASSY: 101146

CUST | APP | APP | DES | CK | DWN

APPROVAL

INSTRUMENT MODULE ASSY

INTERMEDIATE

FIGURE 6

SCALE: FULL

SYSTEMS ENGINEERING DIVISION

PNEUMODYNAMICS CORPORATION

BETHESDA 14, MARYLAND

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RESULTS OF HANDLING-SYSTEM INVESTIGATION

GENERAL

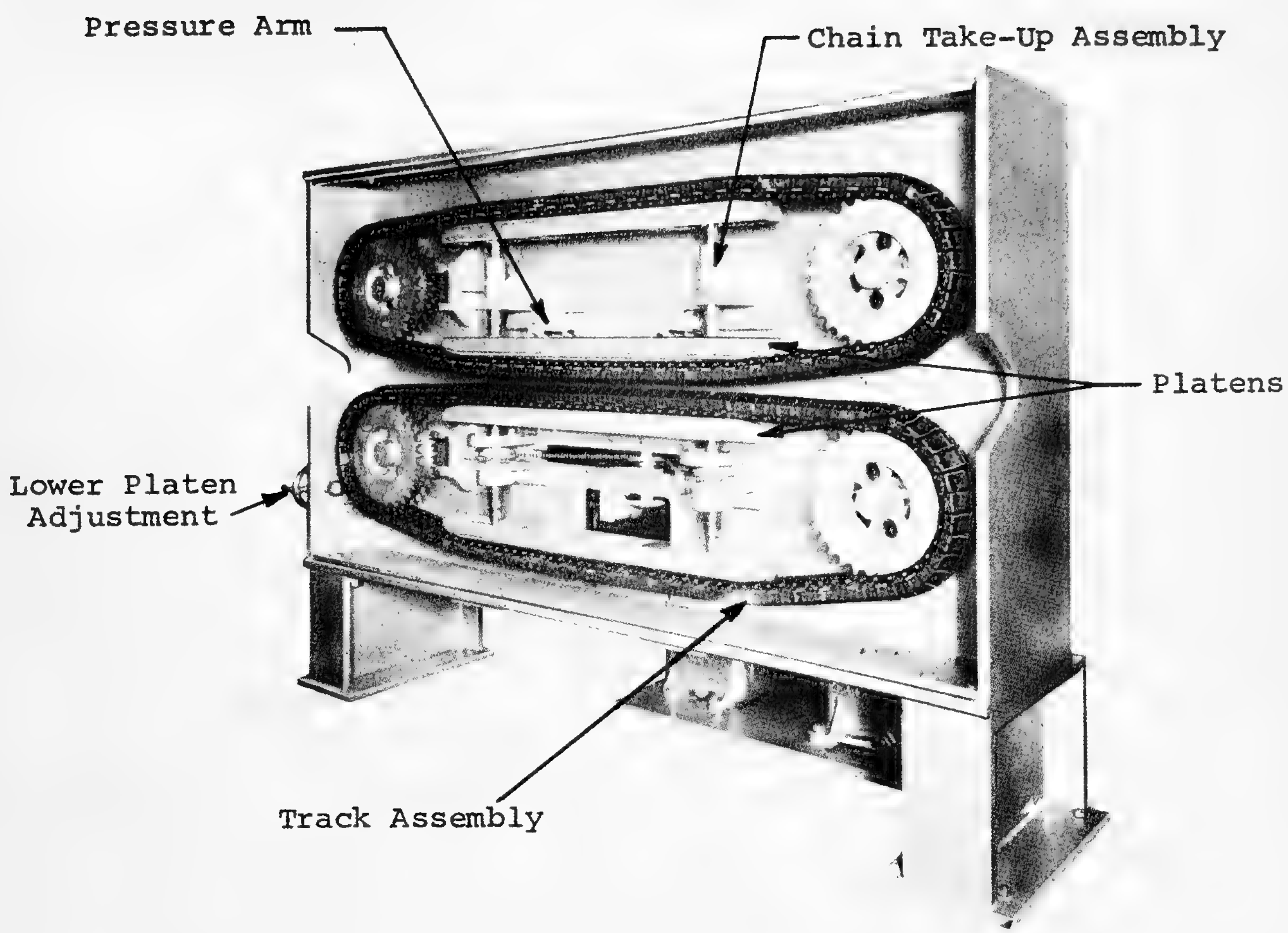
Although the handling system investigation has been primarily concerned with determination of the feasibility of adapting the endless-track winching concept to the task at hand, some preliminary thinking has been devoted to the problems of passing the modules over a stern sheave and ship-board storage of the system.

ENDLESS-TRACK WINCH

Two commercially available models of endless-track-type winching machines were tested to determine the feasibility of utilizing machines of this type for handling faired cable and in-line modules.

These machines consist simply of two tractor-like treads between which the cable is clamped. Tractive effort on the cable is developed by friction between the surface of the cable and the tread. The clamping force is normally developed by air-loaded pistons, arranged to permit a limited degree of motion of one or both tracks. The space between the tracks will automatically increase or decrease to accommodate various-sized objects.

One model, marketed by the Entwistle Manufacturing Company under the trade name "Caterpuller" (Figure 7), was used to investigate the general capability in receiving and passing



CLASS D
TYPE D-VA-72
CATERPULLER

Track Design: Floating and Both Driven
Track Arrangement: Vertical
Loading: Single Track - Multiple Pneumatic
Effective Track Length: 45 Inches
Maximum Pull: 4000 Lbs. at 100 FPM
Maximum Power Input: 20 HP
Tread Design: Flat (Material - Black Neoprene Jacket Compound, 60 ± 5 Durometer)

Figure 7

the fairing and modules. In addition, an attempt was made to establish quantitative values of the tractive force with faired and unfaired cable for various conditions of service and for various values of the ratio of the maximum thickness of the fairing to the cable diameter. For convenience, wire rope was used for these tests. The results, presented in (3), are summarized in Table 1, below.

TABLE 1

COMPUTED VALUES OF STATIC COEFFICIENT OF FRICTION, μ

Rope Diameter in inches	μ Unfaired Dry	Unfaired Wet	Unfaired Greased	Faired $\left(\frac{\text{Fairing Thickness}}{\text{Rope Diameter}}\right)$	Faired Dry	Faired Wet	Faired Greased
7/16	0.38	--	--	1.0	0.07	--	--
9/16	0.38	0.27	0.16	0.8	0.20	0.21*	--

In the table, μ is defined as the ratio of the tension in the rope to the normal force exerted on the rope by the track treads. As is more fully explained in (3), the normal force was deduced from measurement of the air pressure in the

---

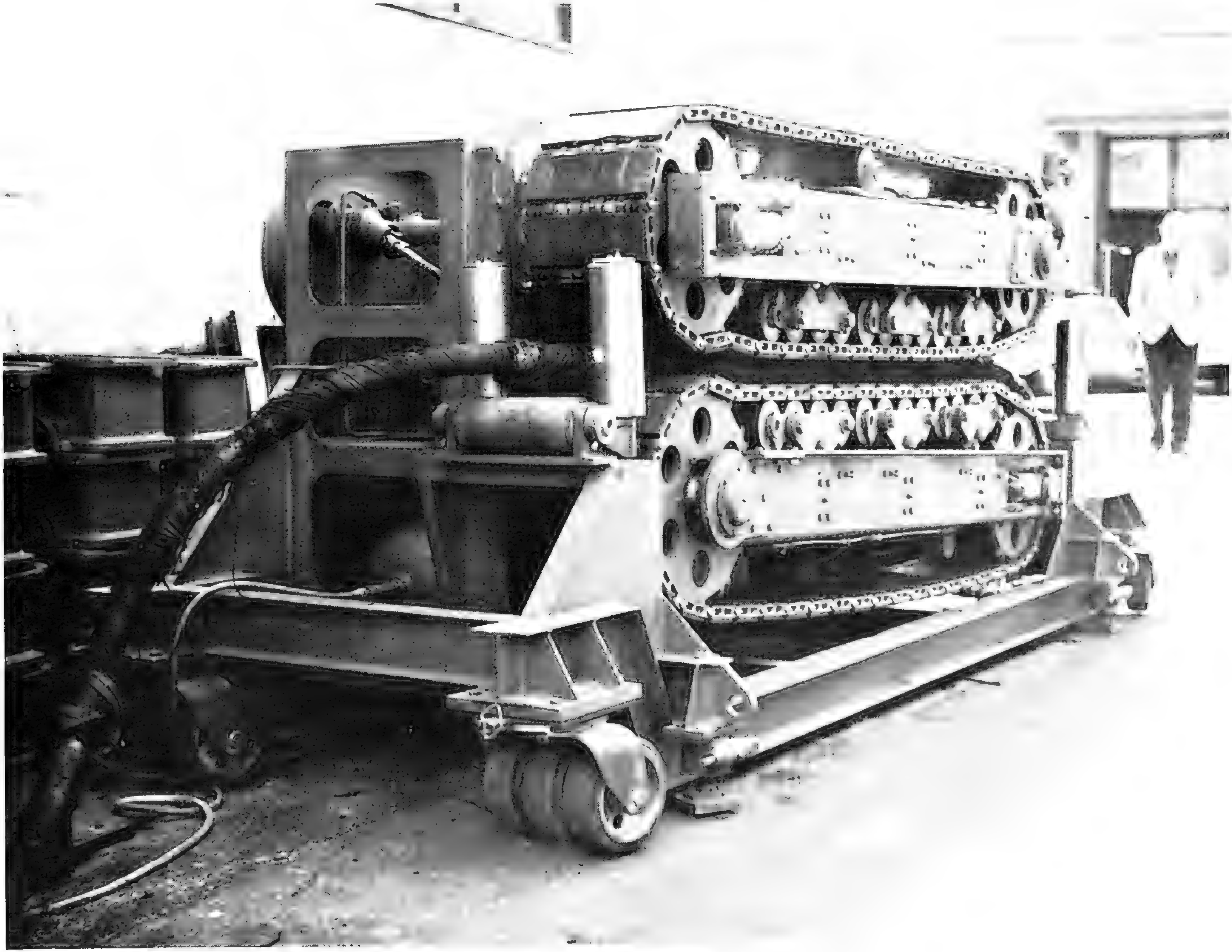
* This seemingly anomalous result would indicate that wetting increases the tractive coefficient of friction; the true explanation would appear to be that at the higher tread pressures the water has a negligible effect, the discrepancy in the results being attributable to inexactitude in determination of the inception of slipping or lifting.

loading cylinders and the geometry of the linkages connecting the pistons and track.

This machine did not satisfactorily englut the modules tested although the faired wire rope assemblies did track through the machine without damage to the rope or fairing. Some deformation of the fairing clips was observed, and it was noted that the rope tended toward the edges of the treads when slack was allowed to accumulate at the point of discharge and when proper alignment of the rope was not maintained at the point of entry.

The second winch, manufactured by the Western Gear Corporation and marketed under the trade name "Cable Hauler," was tested at sea with full-scale faired cable and a dummy instrument module. This winch, shown on Figure 8, and its installation for test are described in Appendix II. The purpose of the test was to evaluate the proficiency of the machine in handling the faired cable and module, to evaluate a roller-and-guide technique for orientation and direction of the fairing, and to determine the tractive effort attainable with faired cable. The details of the tests are given in Appendix II.

The winch satisfactorily handled the faired cable assembly, and englutted and passed the module, as shown on Figure II-2. The roller and guide assembly satisfactorily performed its intended function. The maximum measured line pull was just over 10,000 pounds, with pressure of 110 psig



Western Gear Corporation "Cable Hauler"

Figure 8

applied to the track-loading air cylinders. Slippage at this load occurred when the air pressure was reduced to 90 psig. From these data, and the geometry of the machine, a value for μ of 0.29 was computed.*

STERN SHEAVE

Three methods of passing the cable, with in-line modules, over the stern have been considered:

1. Single, large-diameter sheave

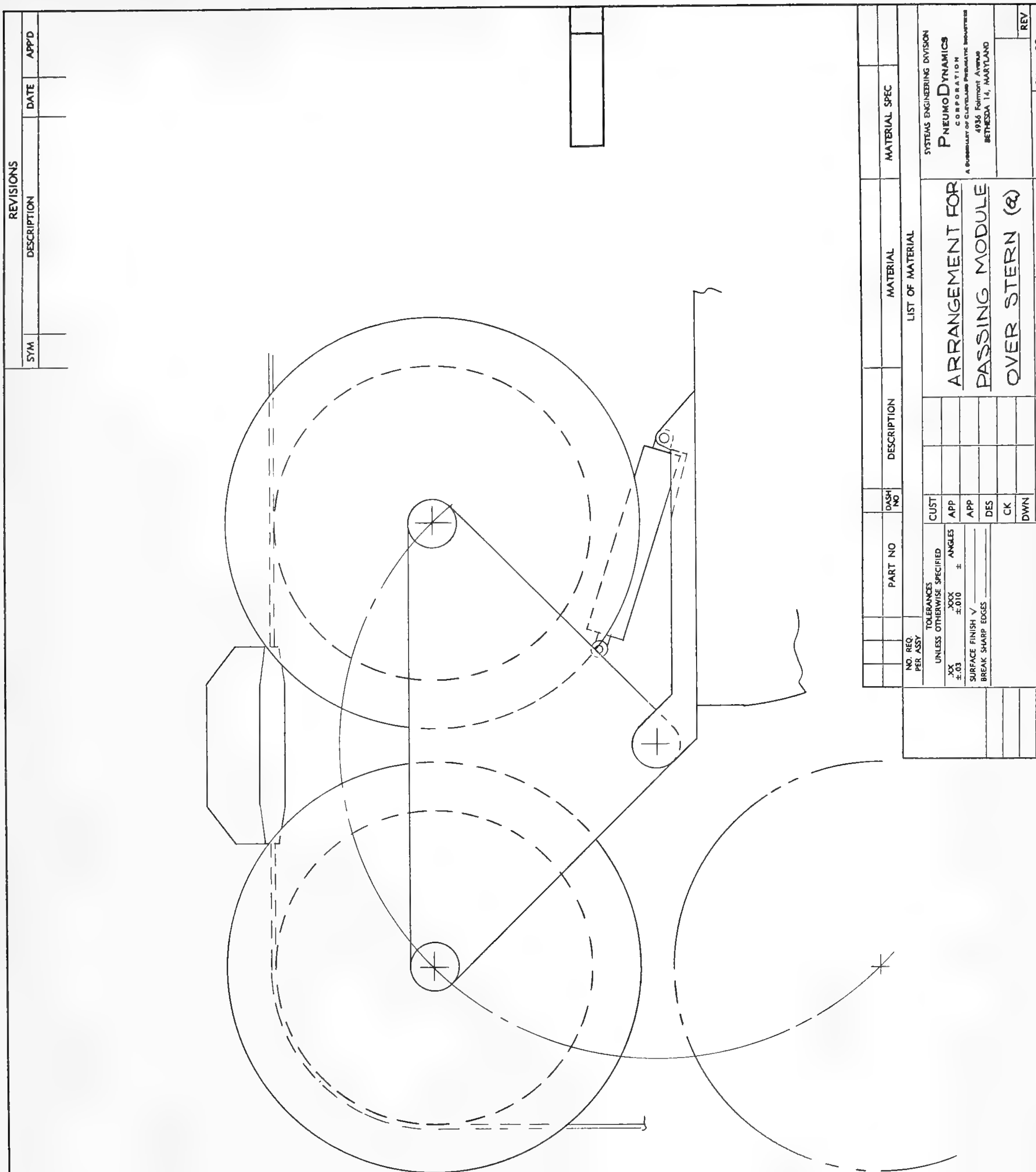
   Use of a single, conventional sheave would necessitate a sheave about twenty feet in diameter to preclude excessive local stress in the individual cable-armour wires near the fitting which secures the cable to the module.

2. Tandem-sheave arrangement

   Excessive sheave diameter could be avoided by transferring the cable between two tandem-mounted sheaves, avoiding contact between modules and sheaves as shown in Figure 9a. The diameters of such sheaves need not exceed the minimum requirement for reasonable cable life (3 feet for $\frac{3}{4}$-inch-diameter cable).

---

* The lack of agreement with the value of μ given in Table 1 for equivalent conditions may be attributed to inexactitude in the pressure measurement, the use of double armoured cable rather than wire rope, and friction in the mechanical elements of the track loading system. The use of double armoured cable should yield a real increase in the value of μ_0 due to the lay of the outer wires; whereas the increase due to friction in the mechanical linkages is present only because the measurement was taken with decreasing pressures.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MATERIAL
MATERIAL SPEC
LIST OF MATERIAL
TOLERANCES
UNLESS OTHERWISE SPECIFIED
XX ±.03
XXX ±.010
ANGLES ±
SURFACE FINISH
BREAK SHARP EDGES
NEXT ASSY
CUST
APP
APP
DES
CK
DWN
APPROVAL
ARRANGEMENT FOR
PASSING MODULE
OVER STERN (a)
SCALE
SYSTEMS ENGINEERING DIVISION
PNEUMODYNAMICS
CORPORATION
A SUBSIDIARY OF CLEVELAND PNEUMATIC INDUSTRIES
4936 Fairmont Avenue
BETHESDA 14, MARYLAND
CODE
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FIG. 9a

As shown in Figure 9a, an actuation system would be required to transfer the load from one sheave to the other.

3. Single sheave with lift-over device

The cable could be passed by a single sheave, about seven feet in diameter, straddled by a segment of a larger sheave functioning as a cam (Figure 9b). On the periphery of the cam would be a receptacle for the module, at each end of which a groove would support the cable in such a way that the axis of the cable near the module would coincide with the axis of the module itself. The cable would be led about a segment of decreasing radius so that it would exit from the cam on a line tangent to the periphery of the sheave. The sequence of action required to lift a module over the sheave is shown in Figure 9b. An actuator would be required to engage the cam.

STORAGE

As discussed in (2), storage could be accomplished in several ways. Cable sections could be decoupled at each module and the cable layed out in deck wells or reeled onto a multiplicity of small drums. Alternatively, it might be stored as a unit in a well or on a single drum where the problem of fleeting the module-encumbered cable onto the storage reel could be avoided by traversing the reel relative to the winch.

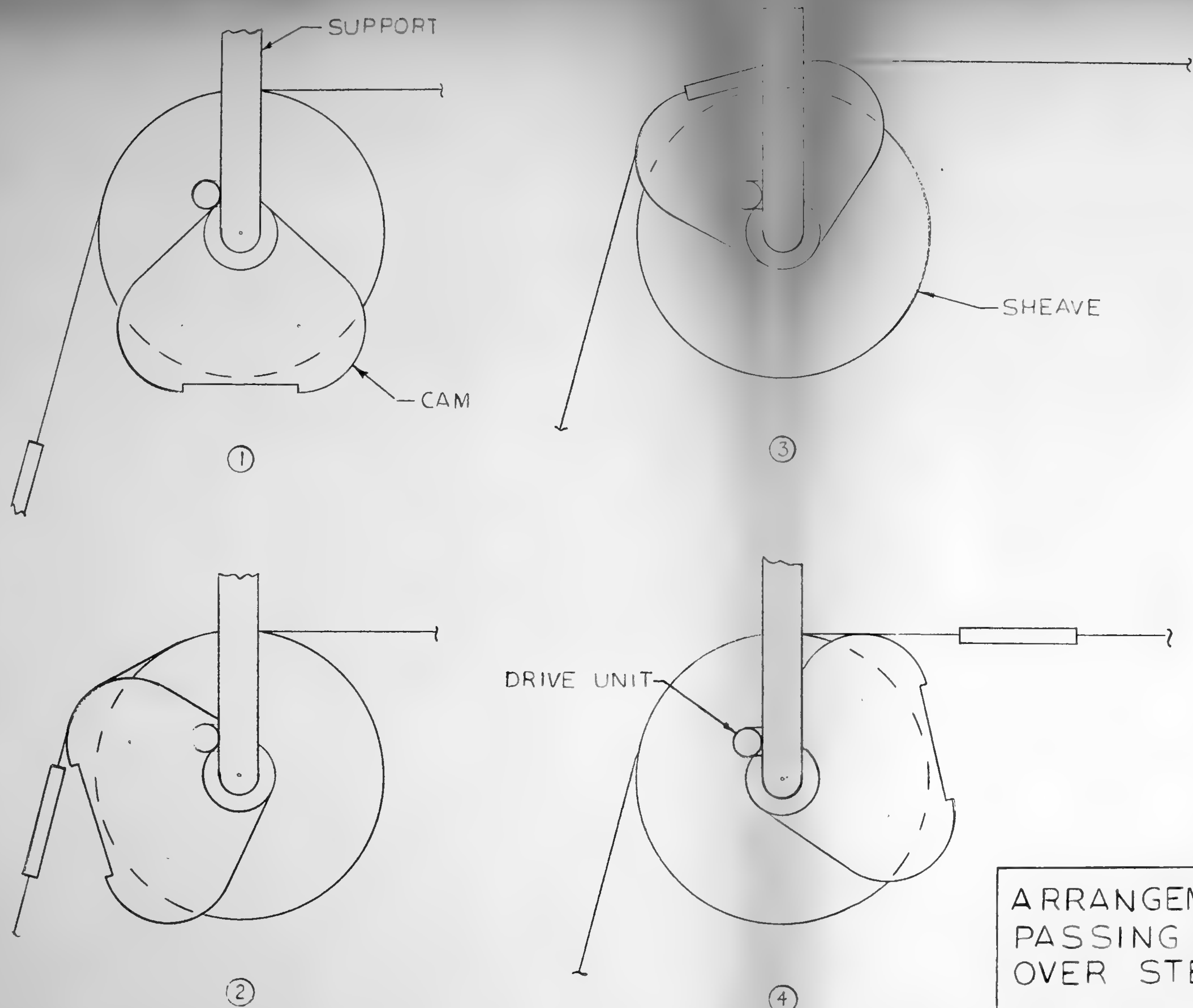

FIG. 9b

ARRANGEMENT FOR PASSING MODULE OVER STERN	SYSTEMS ENGINEERING DIVISION PNEUMODYNAMICS CORPORATION 4936 Fairmont Avenue BETHESDA 14, MARYLAND
	SK-1321

DISCUSSION OF RESULTS

GENERAL

The objectives of the present project could best be accomplished by investigation of only those factors representing potential problem areas in the realization of the design goals. In each such instance, analysis or appropriate testing of the specific elements was performed. These individual results must be appropriately weighted to determine overall system feasibility.

PERFORMANCE

During the course of these studies, it was determined that the resistance of the faired cable might be as much as fifty percent greater than that assumed for the preliminary design. The effects of such an increase in the resistance are discussed in Appendix III. It is pointed out therein that the depth objective can be attained, in spite of the higher resistance, by increasing the scope of faired cable.

Effecting such an increase without exceeding the limitation of one-third the cable breaking strength depends on the validity of the assumed hydrodynamic-loading functions. Two sets of computation, one by Whicker (4) and the other by Eames (5), are available for estimation of the hydrodynamic forces acting on faired cable, and for prediction of resulting geometries. Whicker's formulation is the more general; Eames' results represent a special case. If the loading

assumed by Eames is correct, the maximum predicted tension in the system at design speed is about sixty percent of the cable breaking strength, and it is obvious that either the speed or depth must be reduced to ensure a reasonable life for the system. If, on the other hand, the loading assumed by Whicker is correct, the depth objective can be attained with no sacrifice in speed. Since Eames' formulation is based on the assumption that the resistance is due only to viscous drag, some error will attend application of his theory to the present fairing. This fairing has a fineness ratio of four. A substantial portion of the total resistance must, consequently, be attributed to the pressure distribution rather than the viscous drag. Whicker's formulation accounts for the existence of this pressure component, but the accuracy of his assumptions is unknown. The most important aspect in this development program, therefore, was the demonstration of technical feasibility in achieving the depth objective.

Since full scope of cable was not available for the trials, an attempt was made to compare predicted with experimentally attained values of depth, cable tension, and cable angle, for given scope. Because the test cable was of slightly larger diameter than that selected for the system, and the depressor was found to have a slightly lower downforce-to-drag ratio than originally predicted, new performance predictions, on the basis of Eames' theory, were made; results are given in Appendix III.

The comparison (Appendix III) of cable configurations resulting from application of the two theories, show that the geometries of the cables differ but little, whereas the tensions show significant variation. The surprising agreement in geometries, as indicated by Figure III-1,* indicates that Eames' theory may be used to predict geometries with fair accuracy, particularly for those situations like the presently described sea trials in which the cable does not depart significantly from the vertical. In Appendix III, it is shown that either theory predicts the same radius of curvature of the cable near the depressor. If the system, then, performs in accordance with Whicker's theory, the geometry will be in substantial agreement with predictions based on Eames' theory,but the measured tensions will be noticeably lower.

The limited results attained from the sea trials substantiate this hypotheses. As shown on Figure III-2, the measured depth and cable angle correspond very well with the predictions (based on Eames), and the measured tensions, shown on Figure III-3, are substantially lower than those predicted.

Unfortunately, due to an intermittent open in an electrical lead in the cable, only one reliable measurement of depth was attained on a cable scope of sufficient length to yield

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* Page III-4, the significant curves are those which illustrate variation of scope with towing speed.

significant results. The high seas in which the tests were conducted caused the tensions to fluctuate continuously, so that only a range of tensions could be observed. Ideally, such a test should be conducted on a smooth sea, The rough seas did, however, remove any doubts concerning the dynamic stability of the system.

COMPONENT PERFORMANCE

Towed Elements

General

The sea trial served to demonstrate the structural and hydrodynamic adequacy of all elements of the system except for the instrument modules (which were not subjected to maximum hydrostatic or tensile loads), the stops and the cable. The adequacy of the first two components was demonstrated by other tests, reported in Appendix I, however. The cable is certainly adequate for the static loads if the limit placed on the maximum allowable static tension is not exceeded; the overall adequacy depends on the maximum dynamic loads expected and will be discussed later. Some minor improvements to various components were indicated as a result of the various developmental tests and the sea trial.

Depressor

Static instability in pitch was observed during tow tests with the original set of horizontal stabilizers. As is explained in Appendix I, this condition was corrected

by the addition of horizontal and vertical tip-plates, as shown in Figure II-1. This configuration towed satisfactorily in basin tests and at sea. These extensions are vulnerable during handling, however; Figure II-1(b) shows how the starboard end plate was bent during a launching mishap. The extensions also interfere with storage on deck as they protrude below the rear supports.

Observation of the towing characteristics, made during the basin towing tests, give indication that the deficiency can be remedied by effecting minor changes in the location of the stabilizing surfaces.

Cable

Surges in tension of approximately double steady state values were observed in the system during the sea trial due to high seas. Since the system was at short scope (700 feet), for which condition the cable angle at the towpoint was quite steep (61 degrees), any heaving motion of the towing vessel must result in a corresponding heave of nearly equal magnitude at the depressor. The small margin of static stability in pitch of the depressor, and the location of the towpoint well aft of the main-wing quarter chord line, result in characteristics which tend to minimize response in pitch for disturbance frequencies of the order encountered during the trials. The depressor on short scope is thus literally pulled vertically through the water when the towing vessel heaves. Making allowance for human error in judging

the motion of the vessel, the maximum total vertical motion of the fantail of the test vessel was about six feet. Since the period of the vertical motion was about six seconds, the maximum vertical velocity of the depressor was, say, $\frac{2\pi}{6 \text{ sec}}$ x 3 feet, or about 3 feet per second. With a forward velocity, at 7 knots, of roughly 12 feet per second, the maximum increase in the angle of attack at the depressor would be about 15 degrees. Since the effective angle of attack of the depressor is approximately 14.8 degrees, this would correspond to doubling the lift coefficient and hence the tension, which is about what was observed. Conversely, on the downstroke, the tension should have dropped and it did indeed fall to extremely low values following the appearance of a high surge load.

The foregoing discussion is overly simplified, but does serve to illustrate the problem that attends large motions of the towing vessel. This situation might be alleviated by decreasing the impedance offered to the cable at the tow point. It is possible, of course, that pendulous motion of the system might thus be excited, with equally undesirable results, but the fact that the cable experienced no pronounced fore and aft motions during the periods of peak surging tends to discount this possibility.

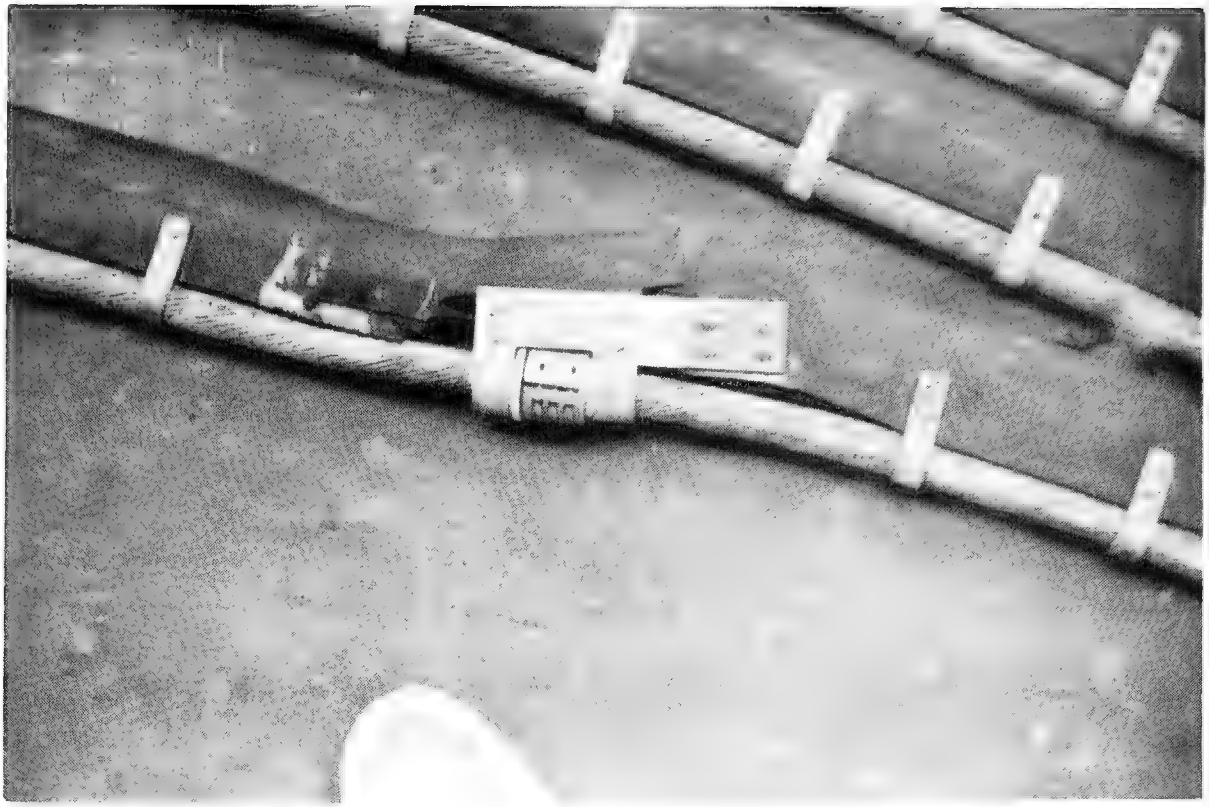
If the mechanism discussed above is responsible for the observed surges in tension, increased scope will undoubtedly

improve the situation materially, as the shallower angle at the tow point would reduce the influence of the disturbance. For example, at full scope, the cable angle at the vessel in the prototype system will be about 25 degrees. For this case, vertical motion of the tow point of as much as 20 feet will result in negligible deformation of the cable catenary unless the disturbance is amplified in passing down the cable. The latter possibility must be investigated. Reduction in speed, when on short scope in heavy seas, may also help in keeping the peak loads within the limits of the strength of the cable.

Fairing

The possibility of effecting some reduction in the cost of the cable fairing by using a single-compound construction is discussed in Appendices I and IV. Results concerning the effect on stability of the stiffness in bending seem plausible, but were not tested, since an available length of conventional, double-durometer construction was used for the sea trial.

During inhaul of the system at sea, the fairing crept slightly upward relative to the cable, so that some strain was exerted on the upper side of the hangers, elongating the holes in the fairing on that side of the hangers (Figure 10). To alleviate this condition, the fairing terminations were subsequently designed to hold the fairing with equal strength in both directions.



Condition of Hanger and Stop
After Completion of Inhaul Operation

The "ears" on the hanger point toward
the depressor when the system is streamed

Figure 10

The method of terminating the Dacron-cord reinforcement in a loop or eye, developed by the David Taylor Model Basin and the U. S. Rubber Company, was adopted as a less expensive method of attachment to the hangers; the Dacron loop is able to sustain the maximum load that can be carried in the fairing.

Fairing Attachments

The apparent lack of resistance to deformation exhibited by the mild steel fairing clips used for the sea trials, and the tediousness of assembling them to the fairing with the threaded bushing and "Nylock" bolts, led to the specification of a "springier" material for the clips and assembly by means of rivets. This technique was used with success in an assembly of 3/8-inch-diameter cable and TF-84 fairing, and resulted in a significant decrease in the time required for assembly.

It was found that no intermediate bearing surface was required between the stop and cable hanger, so these were eliminated in the final design. Teflon washers used for such bearing surfaces during the sea trials were found to be badly chewed, and several were lost.

The adequacy of the stop used during the sea trial was not completely determined since the steepness of the cable in the test installation precluded development of the maximum tangential load in the fairing. However, no failure due to vibration or neck-down of the cable was observed. Since the

design load is conservatively estimated and the load-holding ability of the stop can be tested independently, failure to test under worst conditions need be of no concern.

Modules

The present module is designed as a basic instrumentation-carrying vehicle. Only minor modification should be required to adapt it for temperature measurements. More extensive modifications will be needed to adapt it for measurements (sound velocity, salinity, conductivity) requiring a flow of water across the sensor. It is believed, however, that the present design can be modified to provide the necessary ports and vents.

Handling and Storage Elements

The feasibility of using the endless-track-type winching machine for handling the modules and faired cable depends to a large extent on the compactness of a unit which will develop the necessary pull. The tests with the Entwistle unit indicate that a coefficient of friction of about 0.2 between faired cable and tread may be expected.

For effective performance, the winch need be capable of inhauling the system with the vessel traveling at a speed of only three knots. For this condition, the initial line tension will be about 10,000 pounds, progressively lessening with inhaul of the system. If we assume a margin fifty percent greater than the anticipated load, the design

load becomes 15,000 pounds. The normal force required to develop this line pull would be about 37,500 pounds since the frictional forces act on both sides of the cable. If we assume the maximum permissible loading over the total length of track, L, to be p pounds per inch of contact between track and cable, the length of track is given by

$$L = \frac{(37,500 \text{ lbs})}{p}.$$

If a value of 500 pounds per inch is taken for p, in accordance with one manufacturer's recommendations, the length of track becomes 75 inches, which is about that of the Western Gear unit tested at sea. It would not be desirable, in any event, to reduce the track length used in that unit, since it very nicely envelops the modules while maintaining pressure on the remaining cable, as shown in Figure II-2.

The total tractive capacity can be increased by increasing the allowable value of p. Discussions with cable manufacturers indicate that it should be possible to load the cable transversely by several thousands of pounds per inch. Hence, if the track and fairing materials permit, or if other suitable materials can be found, it appears reasonable to retain about a six-foot effective track length and increase the tractive capacity by increasing the normal loading. If p could be increased to 1500 pounds per inch, for example, the winch could sustain the maximum pull imposed by the cable.

One caution must be observed if very high track loadings are used, however, since a module in entering the tracks is subjected to a force tending to expel it due to the geometry of the entrance, although this force may be minimized by tapering the end of the module as was done with the dummy test modules in the present design.

This factor does suggest that by designing the lower ends of the module to maximize this force, any module could be used to sustain the load when under tow. For example, if the winch were capable of sustaining 15,000 pounds through the tracks and an additional 10,000 pounds could be developed by the aforementioned technique, a 25,000-pound loading could be sustained with only a 10,000-pound load on the cable fitting in the module.

The track-loading technique used in the Western Gear unit is nicely adapted to englutment of the module and appears to be superior to other methods.

The problem of passing the module over the stern with equipment of reasonable size is still in the conceptual stage. The cam technique described in the preceding section seems to be the most promising approach since it appears that this technique will result in minimum size equipment.

The most promising approach to the storage problems seems to be to reel the fairing loosely on a large drum, letting it overlap the modules as required, and to level wind by moving the storage drum transversely relative to the winch.

CONCLUSIONS

The hydromechanical feasibility of this cable-towed oceanographic instrumentation system has been demonstrated. Analysis indicates feasibility of attainment of the depth objective at design speed within the limitation on maximum line tensions. The limited evidence attained is basically favorable and lends assurance that the predicted performance can be achieved. The hydromechanical performance of the towed elements, the results of the tests with the endless-track-type winches, and the conceptual solutions to the handling and storage problems were satisfactory. The feasibility of the "in-line" module concept, providing a flexible arrangement for varied distribution of instrument containers in the vertical profile, has been established.

RECOMMENDATIONS

With the hydromechanical feasibility of the system established, and much of the design of the towed elements well in hand, it is recommended that development of a fully operational system be undertaken. It is suggested that this be accomplished in two steps.

The first step should consist of design, procurement, and evaluation of a prototype system which would include representative oceanographic instrumentation, a full-depth towing system, and handling and storage systems. Selection should be made of a data-transmission technique, with a preliminary choice of sensors. Electronic components should be limited to those sufficient to provide a comprehensive test of the electronic system. Emphasis should be placed on the measurement of sound-velocity, temperature, and pressure at this stage, since these appear to comprise the areas of greatest interest.

The second step should consist of procurement of a fully operational system tailored to the needs of a particular project. Practically all the material procured for the prototype system should be usable in the operational system.

Since the cable fairing represents a major cost of the system, it is recommended that a 20-foot length of the

single-compound fairing be thoroughly tested to establish its adequacy for application to the prototype system.

The problem of surge loading should be studied further, and techniques for effecting reduction established.

APPENDIX I

TOWED SYSTEM COMPONENT DEVELOPMENT

APPENDIX I

TOWED SYSTEM COMPONENT DEVELOPMENT

GENERAL

Details relevant to the major components of the towed system, including requirements, design approach, development, and resulting characteristics are discussed in this Appendix.

DEPRESSOR

Requirements

Based on the analysis presented in (2), the depressor was required to provide a total downforce of 4,450 pounds at a towing speed of 7¼ knots. Since its only function was the provision of downforce, it seemed reasonable to require that the depressor be inexpensive.

Design Approach

The total downforce was developed by a combination of weight and hydrodynamic depression for the reasons discussed in (2). The following specific objectives were established:

Weight in water...............1,000 pounds

Hydrodynamic downforce........3,450 pounds

Arctan (downforce/drag)....... ≥ 84 degrees

Development

The detailed design considerations involved in attainment of the objectives stated above are given in Appendix I of (2). On the basis of the work reported therein, a prototype design was executed and a full-scale depressor fabricated and tested (Figure 3).

The depressor was test-towed from the No. 2 carriage at the David Taylor Model Basin. Towline tension and the cable angle at the depressor, relative to the horizontal, were measured and recorded as a function of tow speed. The first test demonstrated the depressor to be hydrodynamically unstable in pitch, thus making it impossible to control the dynamic downforce. For the second test, the effectiveness of the horizontal stabilizers was increased by adding stabilizer area and tip plates. In this condition, the depressor was towed at speeds to seven knots and found to be stable and responsive to trim adjustments. The results (6) were corrected to standard sea conditions for design trim and are given in Figures I-1, I-2, and I-3 which show, respectively, cable tension at the depressor as a function of tow speed, towstaff angle as a function of tow speed, and cable tension at 7¼ knots as a function of horizontal trim tab setting.

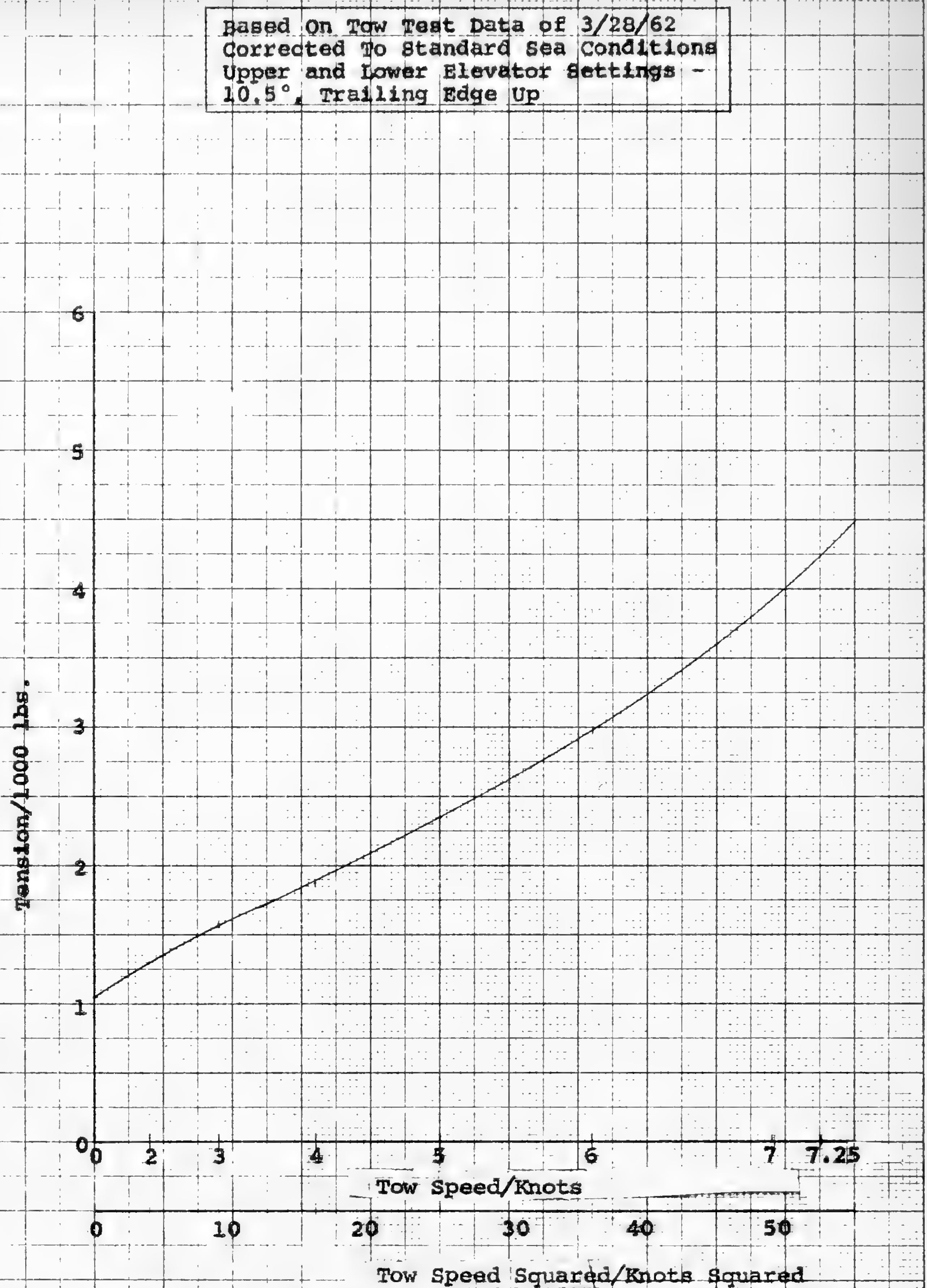


FIGURE I-1 - Effect Of Towing Speed On The Cable Tension Developed By The Depressor

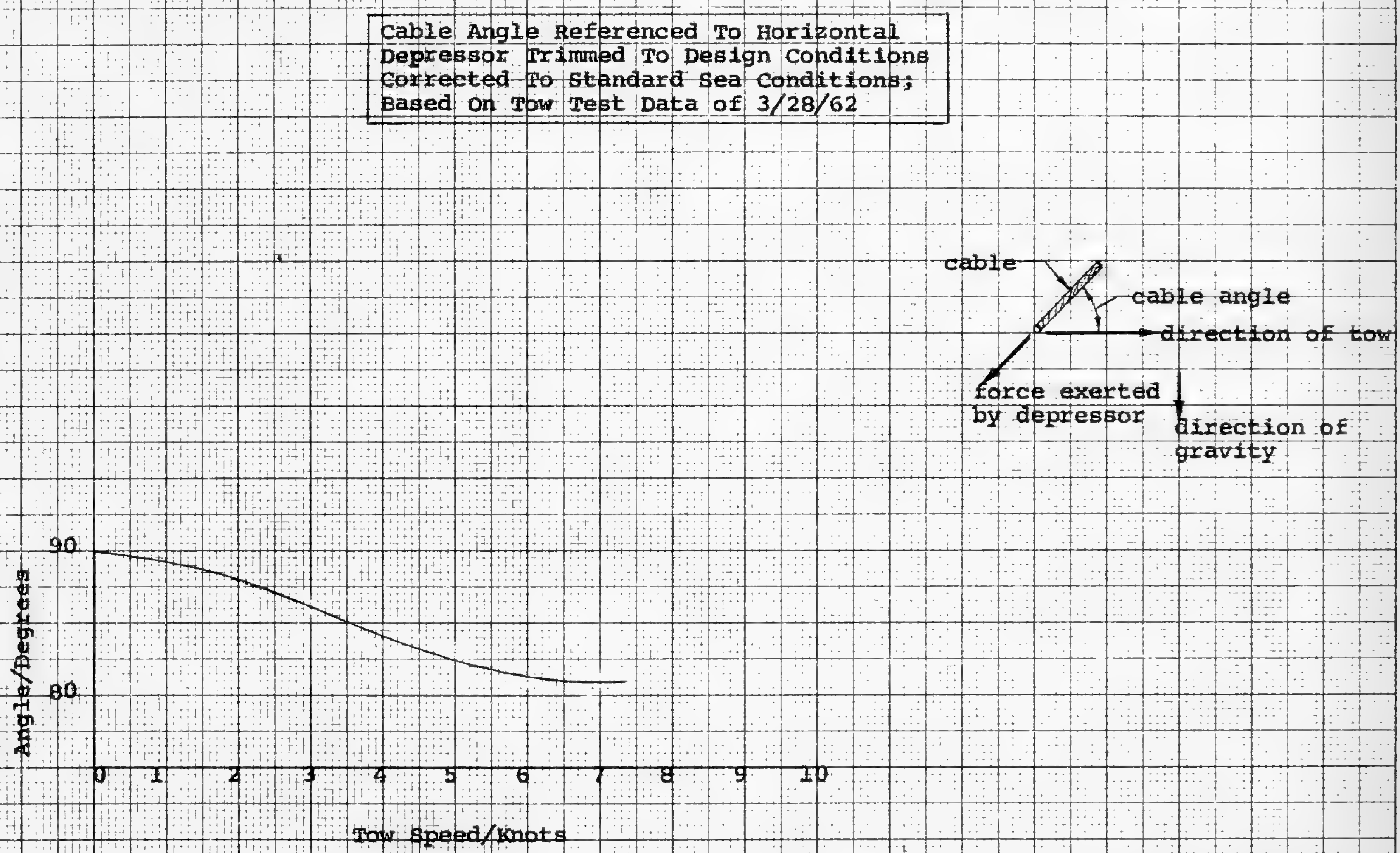


FIGURE I-2 - Cable Angle Developed By The Depressor As A Function Of Towing Speed

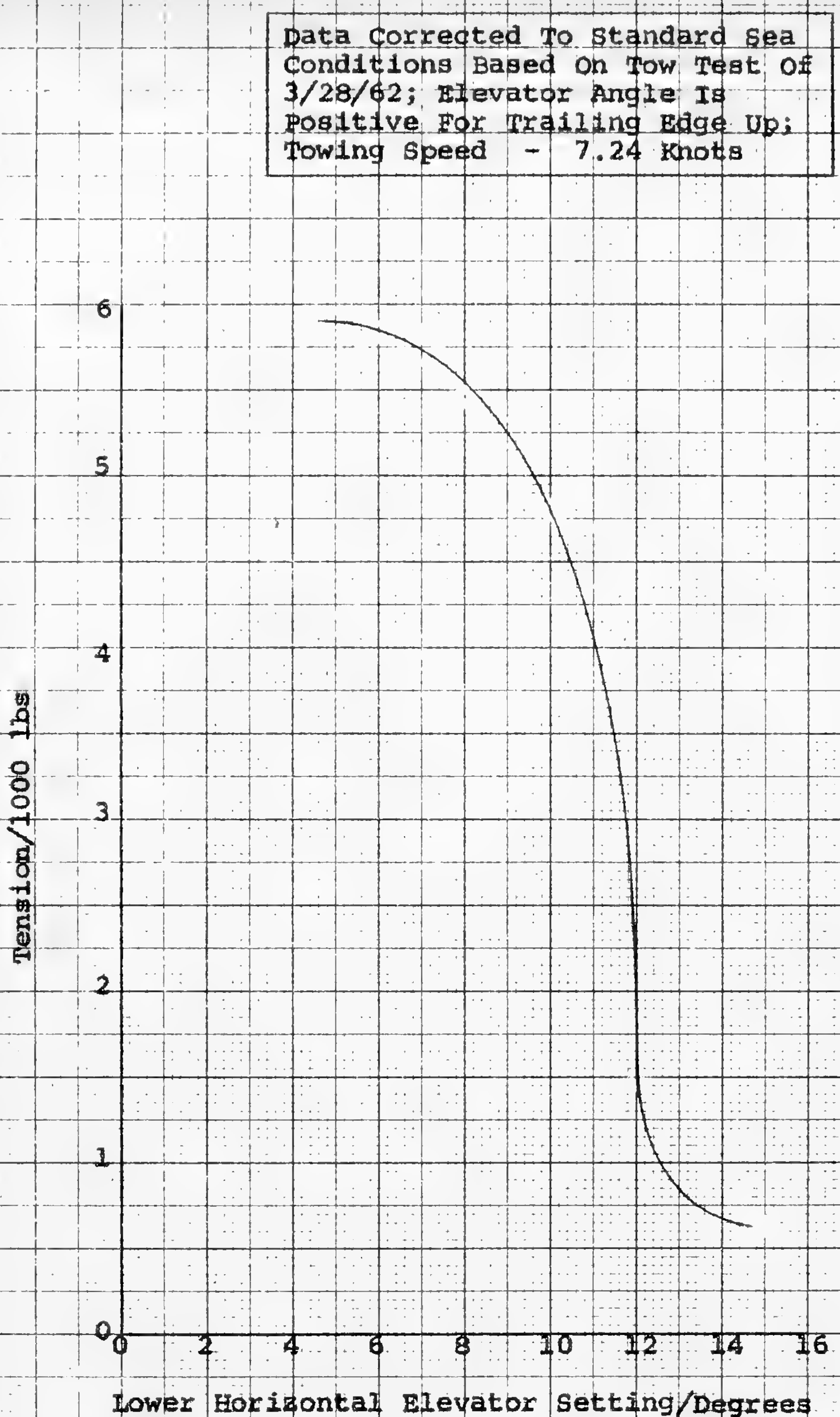


FIGURE I-3 - Cable Tension Developed By The Depressor As A Function Of Lower Elevator Setting

Requirements

The purpose of the cable fairing is to reduce the hydrodynamic load on the towline. Quantitative evaluations of the requirements are difficult to formulate because of the effect of fairing shape and materials on towing performance, handling difficulties, cost and practicability of fabrication. Qualitative conclusions, derived from consideration of these factors, will thus have to suffice.

The fairing must possess the following characteristics:

1. Stability under tow;
2. Structural strength adequate for withstanding hydrodynamic and handling loads;
3. Rigidity, to resist excessive stretching or bunching under hydrodynamic loading;
4. Flexibility, to permit grasping of the cable for inhauling and paying out;
5. Resistance to abrasion, crushing, sunlight, oil, ozone, etc.;
6. Resistance against permanent set or damage when stored in multiple layers;
7. Low cost in fabrication; and
8. Simplicity in assembly to the cable.

Design Approach

Most of the requirements are satisfied by a previously developed fairing, the DTMB TF-84; to minimize developmental costs, it was decided to adapt that fairing to the present application. Items 3, 6, 7, and 8 of the preceding list of requirements constituted the only problem areas in this adaptation. Solutions to these problems are discussed below:

1. Resistance against permanent set or damage when stored in multiple layers

   Damage due to storage in multiple layers was avoided by storing under lightly loaded conditions.

2. Low cost in fabrication

   Cost of fabrication could be reduced by the use of a single-rubber compound in the fairing in place of the two-compound construction usually used; the results of an investigation of this possibility are discussed later.

3. Simplicity in assembly of the fairing to the cable

   Since the expense of assembling the fairing to the cable is primarily due to the labor involved, the clips and hangers used for this purpose were designed

for assembly-line techniques. In particular, riveted rather than screw-type fasteners were selected for the attachment of the clips to the fairing, to speed the operation. A looped-end-type construction was selected for termination of the Dacron-cord reinforcement to simplify construction of the hangers and reduce the number of fasteners required to secure the fairing to the hangers.

4. Resistance to excessive stretching or bunching under hydrodynamic loading

The problem of stretching and excessive accumulation of fairing at the lower end of the cable was solved by suspending the fairing in short lengths between hangers attached to the cable.

Development

Estimate of Tangential Load

In order to select the optimum length of fairing between cable stops, and to determine the loading that the cable stops and hangers must sustain, an estimate of the tangential component of the hydrodynamic drag on the cable fairing had to be made. In estimating this load,

it was necessary to assume that the tangential component of the hydrodynamic load was sustained solely by the fairing and that the cable was towed at the critical angle* and at design speed. Based on Eames' and Whicker's loading functions, estimates of 2.55 lb/ft and 0.67 lb/ft, respectively, were determined for the tangential loading. Both estimates are conservative since the cable must sustain some part of the tangential component of the hydrodynamic drag. Also, the angle an element of fairing makes with the horizontal is not constant, but increases from the critical angle of approximately 24 degrees at the upper end, to approximately 84 degrees at the depressor end of the catenary.

Whicker's loading functions, which are the more realistic of the two formulations, were used in determining the length of fairing between cable stops. Eames' loading functions were used to estimate the load that the cable stops and hangers must sustain. The use of the very conservative load estimate in the design of the cable stops and hangers provides a safety factor in the design.

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* The critical angle is the acute angle subtended by a long towline and the horizon, when the towline is towed from the upper end along a path parallel to the water's surface with no load attached to the lower end.

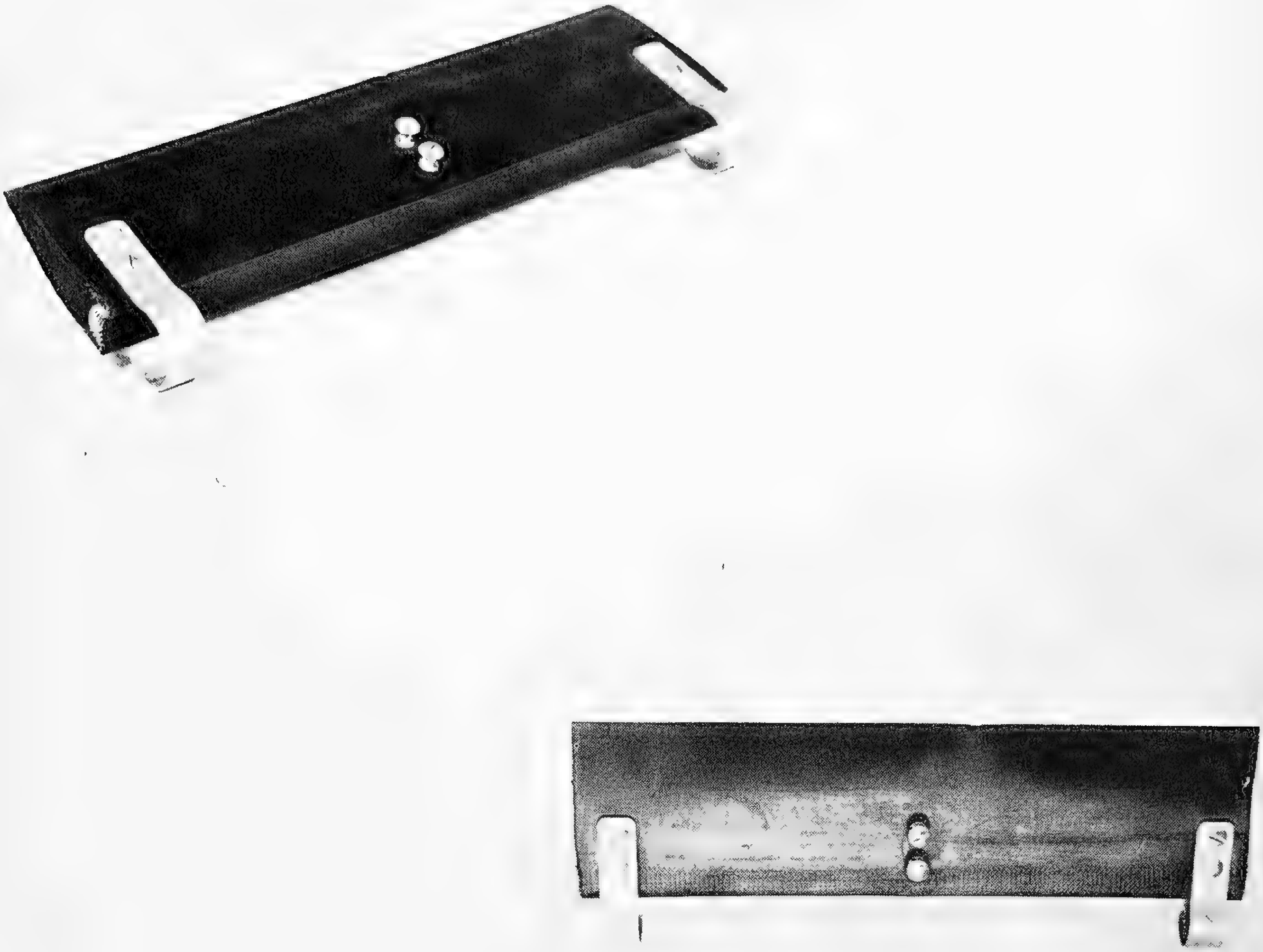
Construction

Cable fairing must conform to the curvature of the cable catenary; to reduce stress buildup, the trailing edge is generally constructed of a more elastic compound than is the fore part of the fairing. The line of demarcation between the harder and softer rubber compounds in a typical construction may be seen in Figure I-4 as the change in shading near the trailing edge of the fairing, just aft of the screws used for attaching the clips.

To reduce costs, without impairing performance, the possibility of using a single-durometer compound was considered. The substitution of a single-durometer compound for the two-durometer construction can be justified only if a fairing so constructed retains its hydrodynamic stability.

The destabilizing moments acting on the fairing are those due to bending as a result of cable curvature, and those due to axial loads resulting from accumulation of the tangential component of the external loading. The result, derived in Appendix IV, is a moment per unit length,

$$L_B + L_T = (IEC_o^2 + TC_o\bar{y})\ \psi \qquad [1]$$



Short Length of the Clip-Type Fairing
Used for Sea Trials

Figure I-4

where

L_B = the moment per unit length due to bending

L_T = the moment per unit length due to axial loading

T = the total tension due to external loads

C_o = the local curvature of the cable

$\overline{y}$ = the distance from the center of the cable to the centroid of the force T

E = the effective modulus of elasticity of the fairing material

I = the second moment of the cross-sectional area of the fairing referenced to the center of the cable

ψ = the local angle of inclination of the center-line of the fairing relative to the plane containing the cable.

To determine the net moment acting on the fairing, there must be added

L_H, a hydrodynamic moment, and

L_B, a weight moment.

Although existing information relative to L_H is inconsistent, the effectiveness of the single-compound construction can be settled at least qualitatively.

The maximum cable curvature occurs at the depressor and is given very nearly by

$$C_o = R/T_o \quad ,$$

where T_o is the force imposed on the cable by the depressor and R, the resistance experienced by a unit length of the towline when caused to move at V, some steady velocity relative to a fluid, along a path normal to the axis of rotational symmetry of the cable. The value of C_o at a speed of 7¼ knots is about 1/1680 feet. Since the approximate value of E is 72,000 lb/ft^2, and of I, 10^{-4}ft^4, the contribution due to bending is, at most

$$L_B = 10^{-4}\ \text{ft}^4 \times 72 \times 10^3\ \text{lb-ft}^{-2} \times (1680\ \text{ft})^{-2}$$

$$= 2.55 \times 10^{-7}\ \text{ft-lb/ft}$$

The contribution due to the externally imposed loads is, at most, that corresponding to the maximum load (500 pounds) which can be carried by the stops. Since this load is carried almost exclusively in the Dacron cording, we may estimate $\bar{y}$ to be nearly equal to the radius of the cable, hence:

$$L_T = TC_o\bar{y} \cong 500\ \text{lb} \times (1,680\ \text{ft})^{-1} \times \frac{0.375}{12}\ \text{ft}$$

$$= 9.3 \times 10^{-3}\ \text{ft-lb/ft} \ ,$$

which is approximately $3 \times 10^4 L_B$.

We may, therefore, conclude that the contribution due to bending is negligible in comparison with that due to axial loading. Substitution of a homogeneous compound for the present two-compound construction should have virtually no effect on the location of the centroid of the externally imposed forces, since practically all the load is carried in the Dacron-cord reinforcement. Long sections of the two-compound-construction fairing have evidenced no instability while under tow when excessive stretching was avoided. Substitution of the single-durometer construction, then, should result in no adverse effects on stability.

Selection of Length

It was desired to minimize the number and size of stops to avoid problems in reeving over drums and sheaves, and yet eliminate excessive stretch and bunching of the fairing near the lower stops. The selected length represents a compromise between these two goals.

The elastic modulus of the fairing was estimated on the basis of the experimentally determined characteristics of similar fairings, which indicate that the product of A, the area in cross section, and E, the modulus of elasticity, is about 12,000 pounds. ϵ, the strain under the load T, at a point,

s, from the bottom of the segment of fairing, is given by

$$\epsilon = \frac{T}{EA} \quad ,$$

where

$$T = \int_0^L \frac{dT}{ds}\, ds \quad ,$$

and L is the total length of fairing involved.

From Whicker's work

$$\frac{dT}{ds} \cong 0.67 \text{ lb/ft} \quad ,$$

when the cable is towed at the critical angle corresponding to design speed.

Hence, ξ , the total extension in the length, L, not greater than

$$\xi = \int_0^L \frac{0.67 \text{ lb/ft}}{EA}\, s\, ds = \frac{1}{2} \; \frac{0.67 \text{ lb/ft}}{EA} L^2 \quad .$$

with an assumed fairing length of 200 feet, the load at the upper end would be about 133 pounds, and

$$\xi = \frac{1}{2} \left(\frac{133 \text{ lbs} \times 200 \text{ ft}}{12{,}000 \text{ lbs}} \right) = 1.11 \text{ ft}$$

or about 0.5 percent of the length. In actuality, the fairing elongation will be less than one-half of one percent of its length because of the assumptions previously mentioned, and so it was concluded that 200-foot sections of fairing should tow satisfactorily.

Characteristics

The fairing will have the following characteristics:

Resistance coefficient	$\cong$ 0.3
EA	$\cong$ 12,000 pounds
Breaking strength	$\cong$ 1,200 pounds
Maximum length	= 200 feet
Construction	40-to 60-durometer neoprene rubber with 6-ply Dacron-cord reinforcement
Weight in air	$\cong$ 0.67 pounds per foot
Weight in water	$\cong$ 0.17 pounds per foot

INSTRUMENT HOUSING

Requirements

The general requirements for the instrument housing are given in (2). Since the tension in the cable is maximum at the top, decreasing with depth, and the hydrostatic pressure is at a minimum at the surface, increasing with depth, the forces acting on the instrument case vary according to the position of the housing along the cable. The end caps of the housing, being free-flooding, are not affected by the hydrostatic pressure and are subject only to the tensile load imposed by the cable. This is not true however, of the central, watertight, cylindrical compartment which will house

the sensors and associated electronics. This section of the housing is subjected to both an external hydrostatic pressure, acting on it radially, and an axial force. This axial force may be either tensile or compressive depending on the magnitude of the cable tension at the housing and the hydrostatic pressure acting on the sealed bulkhead located in the end of the cylindrical instrument compartment. The housing nearest the surface, then, must sustain the highest tensile loading, and the housing nearest the depressor, the greatest compressive loading. However, to facilitate interchangeability, it was decided that the housings should be designed to sustain the maximum load, either tensile or compressive.

Since each housing is essentially a connecting link in the cable, it was decided to design each to withstand a load equal to the rated breaking strength (45,000 pounds) of the cable. The maximum depth attainable by the housing nearest the depressor would be, at zero ship speed, just the length of the cable. With the original estimated cable length of 6,200 feet, the maximum pressure on the bottom housing would have been 2,795 psi; with 8,000 feet of cable, the latest estimate for the system, the maximum pressure would be 3,556 psi.

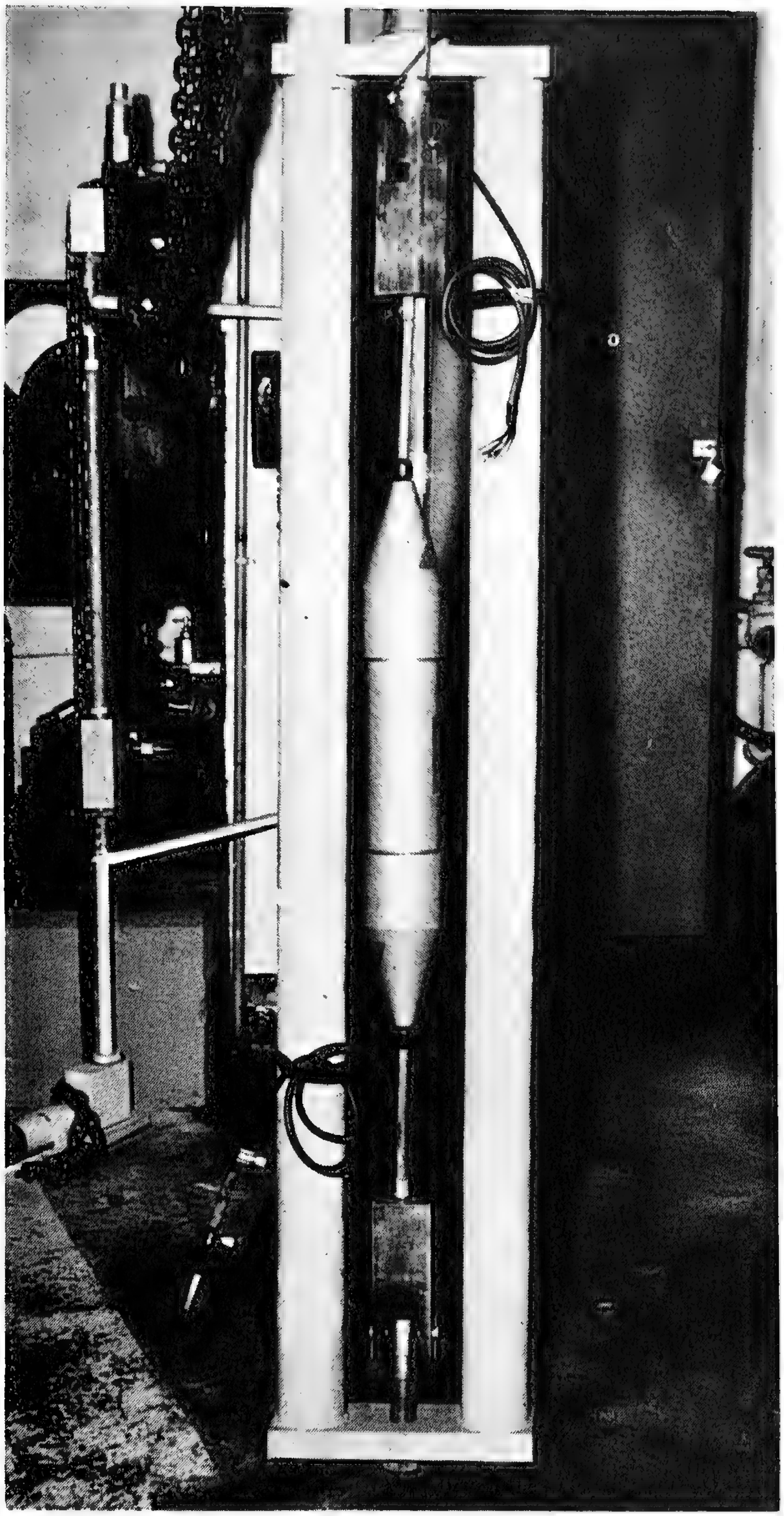
Design Approach

The first instrument housing designed to meet the above specifications was essentially the same as the housing shown on Page 43 of (2). It consisted of two tapered end caps, each housing a mechanical cable lock, threaded onto a central instrument section. The electrical leads from the cable were sealed by a potting compound, which served as an electrical insulator as well as a watertight seal. The threaded section was sealed by neoprene O-rings. On each end of the central section, a phenolic plate, containing the female receptacles for the electric feed-throughs, was held in place by snap rings. The center section had an inside diameter of approximately three inches, a wall thickness of 0.345 inch, and a length of 12 inches. The overall length of the assembled housing was 28 inches.

Development

To test the housing, a fixture (Figure I-5) was designed which, when enclosed in a pressure tank, would subject the housing to tensile, as well as hydrostatic load, thus paralleling operational conditions.

The first instrument housing (2) tested in this manner failed, due to the porosity in the compound potted around



Test Fixture with First Instrument Housing
Rigged for Test

Figure I-5

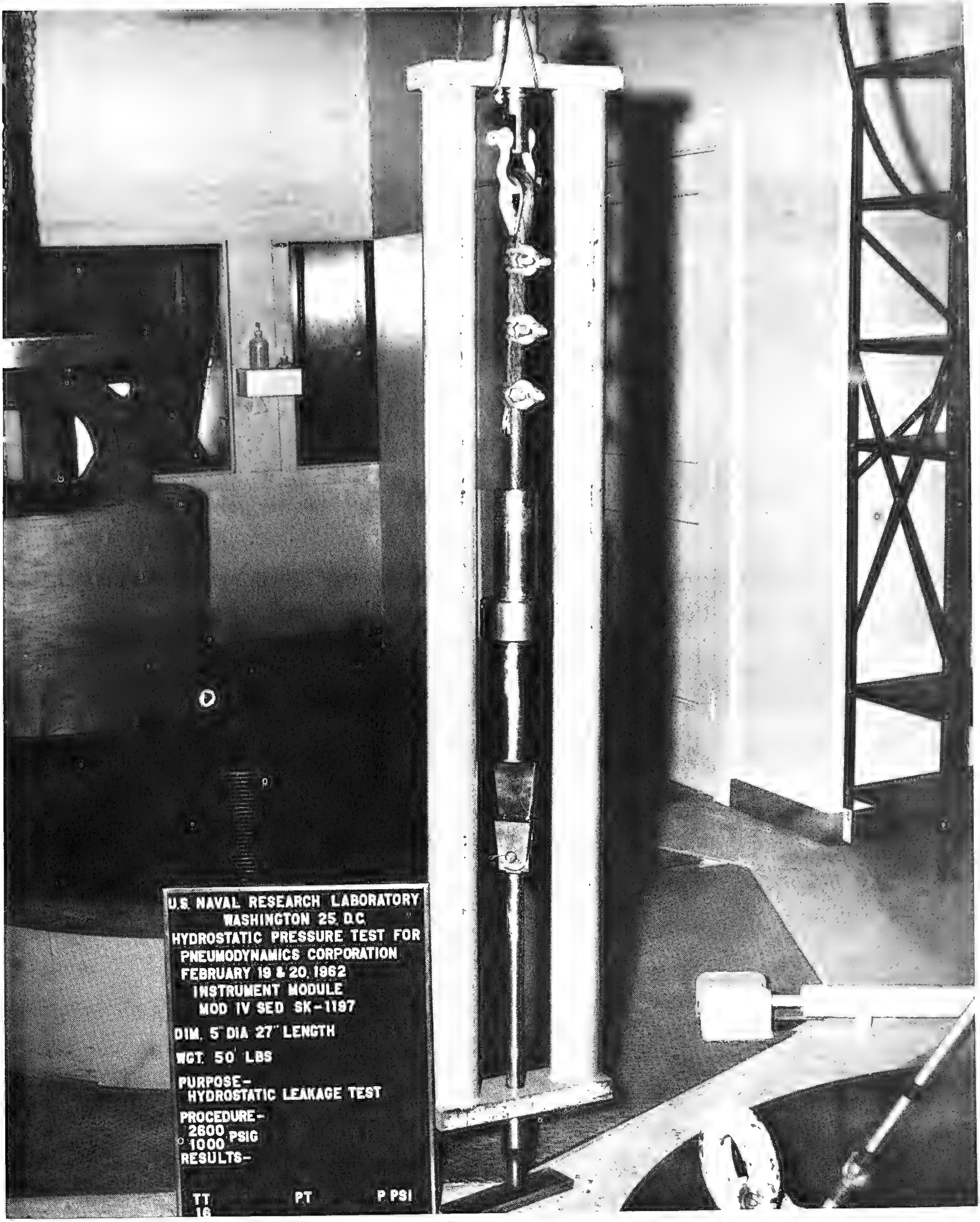
the electrical terminal leads and the insufficent bonding of the compound to the adjoining wall of the end cap cavity. Water under high pressure (2,600 psig) entered the housing through the cable armor wires, ruptured the bond between the sealing compound and the end-cap cavity wall, and extruded the compound into and around the phenolic plate mentioned earlier.

Since no evidence of failure of the mechanical cable lock in the housing was observed (it had been subjected to a tensile load, under water, of approximately 7,000 pounds), it was decided that a test should be run to compare it with a poured epoxy fitting of the type used successfully by the David Taylor Model Basin. A poured fitting of this type was designed in the form of a clevis for attachment in a tensile testing machine, and a comparative test to destruction was run. Failure occurred at a tensile load of 17,300 pounds. The rated breaking strength of the cable was 18,000 pounds. The failure occurred just inside the mechanical lock, approximately 5/8 inch before the first bend of the wires into the spool. Three wires were broken; examination revealed a "necking down" indicative of tensile failure in the wires.

After this housing failed to maintain a watertight instrument compartment, a "test" housing was designed specifically to test the sealing characteristics of commercially available watertight electrical connectors.

One method evaluated was a packing-gland-type fitting which seals by squeezing the insulation of each of the individual conductor wires. Although a pressure test indicated that this fitting successfully sealed around the conductor-wire insulation, it was discarded as its large size would have required lengthening the housing thus increasing the handling problems. It also provided no method for disconnecting the cable from the central instrument section.

A more compact design (Figure 6) was selected in which small pressure-terminal seals, screw-mounted to the housing bulk-head, may be utilized. Also, a more compact clamping arrangement for securing the cable armor wires to the housing was evaluated in the test fixture, as shown in Figure I-6. A hydrostatic pressure test (2600 psig) of the assembly revealed no leakage around the terminal seals, or any slippage or failure of the armor wires. This housing was used during sea trials.



Test Fixture with Test Housing Rigged for Pressure Tests

Figure I-6

CABLE STOP

Requirements

The towed cable fairing is subjected to a force (the tangential component of the hydrodynamic drag) acting along its length. To prevent the fairing from sliding down the cable under the action of this force, the fairing must be attached to the cable by a free-swivelling attachment. The attachment must grip the cable with sufficient force to sustain the load even under the condition in which the cable, under load, may reduce slightly in diameter.

For design purposes, the maximum load requirement was based on Eames' loading functions. The maximum load required to be carried by the stop was thus taken to be 510 pounds for a 200-foot length of fairing.

Design Approach

For ease of assembly to the cable, a split-ring-type stop that grips the cable by means of friction, was developed. To allow for possible reduction in cable diameter, a resilient liner was selected which, under pre-compression, retains enough normal force to develop the frictional forces required to sustain the load. Because of encouraging results of DTMB tests of cable stops lined with sand-impregnated polyurethane, this material was selected for this application.

A test stop, utilizing a split ring which grips a 3/8-inch diameter cable through the liner by means of three socket-head cap-screws, was accordingly designed and manufactured. For tests, operating conditions were simulated by loading the fairing and simultaneously exerting a load on the cable. The stop was tested to a load of only 400 pounds, however, due to the limitations of the test fixture.

Requirements

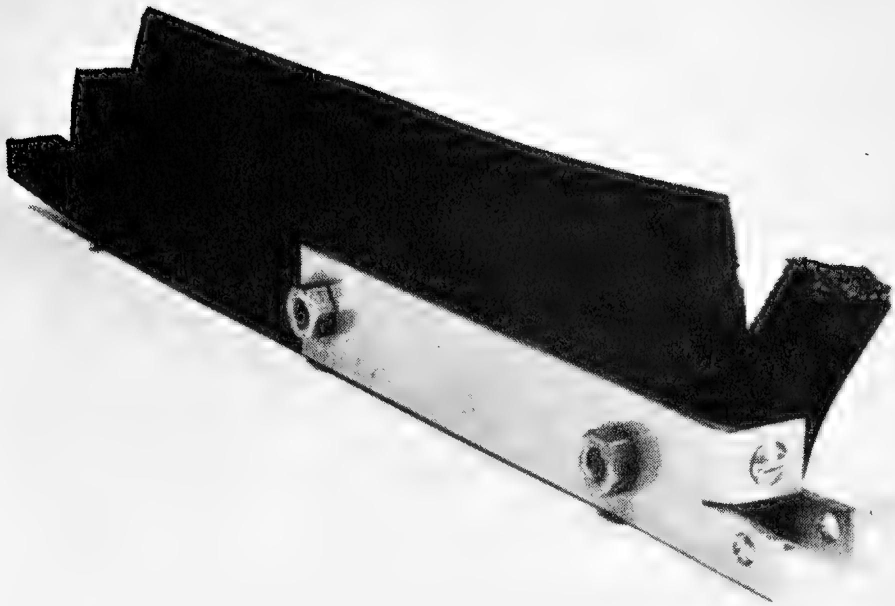
A hanger, mounted at the top of each 200-foot length of fairing, is required to transmit the hydrodynamic drag on the fairing to the cable stop. This hanger must grip the fairing firmly so as to sustain the maximum load without damage to the fairing. For design purposes, the maximum load was taken to be 510 pounds, the same as that required of the stop.

Design Approach

It was decided that the terminal clip must attach to the load-carrying member of the fairing (which is the Dacron cording) in order to prevent tear-out or rupture of the fairing. A design was conceived (Figure 5) in which this was accomplished by means of bending a small length of the Dacron cording, from which the faired section had been removed, and gripping this cording in a clamp. Additional clamping effort was achieved by side plates which clamped a length of fairing approximately six inches long. The developmental clamp is shown, attached to a short length of the fairing, in Figure I-7.

Under test, this device sustained a load of 900 pounds.

During assembly of the cable and fairing used for the sea trials, all terminal clips and stops were subjected to a 300-pound proof load.



Developmental Fairing-Hanger
Attached to a Short Length of Fairing

Figure I-7

Although the hanger described above performed satisfactorily, it was obvious that reduction in fabrication costs and assembly time could be effected if the reinforcing cord in the fairing could be terminated as an eyelet. As mentioned previously, this proved feasible, hence the hanger shown in Figure I-7 was designed.

APPENDIX II

SEA TRIALS

APPENDIX II

SEA TRIALS

INTRODUCTION

A model of the towed system was tested at sea aboard the U.S.S. YAMACRAW (ARC-5) during the period 13 to 17 April 1962. The tests were conducted to establish the hydromechanical feasibility of the system. For this purpose, it was considered necessary to test only selected full-scale elements of the system, and to provide only enough instrumentation to define the operating characteristics; a single pressure sensor located at the depressor seemed sufficient, provided measurements of cable tension and tow-cable angle were taken simultaneously.

DESCRIPTION OF EQUIPMENT

General

The towed system consisted of 1,200 feet of cable (the lower 685 feet of which was faired), a prototype depressor, and one test module. The handling and storage equipment consisted of an endless-track-type winch, an existing storage reel, an existing stern sheave, and miscellaneous fairleads and rollers peculiar to the specific installation. A tensiometer, depth indicator, and simulated instrument module were also utilized.

Cable

Cable of the desired type was not immediately available so two existing 600-foot lengths of 0.782-inch diameter, double-armor, 35-conductor cable were spliced together. One of the cable lengths had been previously rejected due to cracks in its insulation. After a hydrostatic pressure test, eight good conductors were selected for splicing to the other cable. The electrical continuity of each selected conductor was checked after the splice was completed.

Fairing and Clips

The fairing was a TF-84, sized for 0.782-inch diameter cable, produced from Bureau of Ships mold No. G-60818. The after portion was constructed of 30- to 40-durometer rubber, and the leading edge of 50- to 60-durometer rubber. The line of demarcation between the two compounds is visible in Figure I-4. A six-ply Dacron cord, capable of sustaining a maximum load of about 1,200 pounds with a 10 percent elongation, was molded into the leading edge.

The fairing was attached to the cable with clips spaced on four-inch centers. The clips were made of mild galvanized steel and were secured to the fairing with "Nylock" bolts screwed into stud-type inserts passed through the body of the fairing aft of the Dacron cord,

as illustrated in Figure I-4. The general configuration of the assembly may be seen in Figures II-3(b) and II-5 .

Hanger and Stop

The hanger and stop assembly is shown in Figure 10. The clamp permitted termination of the tangential load in the fairing in the absence of a loop in the Dacron cord. The split-type stop, described in Appendix I, was used to transfer the load from the fairing into the cable. In this installation, the Dacron cord in the lower section of the fairing was clamped within the downward extending ears of the clamp. The upper section of fairing was secured to the clamp only by bolts passed through the rubber after body as shown in Figure 5.

Instrument Module

A development module, described in Appendix I, was modified to function as a towstaff by the addition of a clevis, a sheet-metal fairing, and a bail for use in launching and retrieving. The module is shown in Figure II-1 installed on the depressor. The handling bail may be clearly seen in Figure II-1(a) and the fairing in Figure II-1(b).

The depth sensor of the module was mounted on the end plug as shown on Figure 6.

Depressor

The depressor is described in Appendix I and shown on Figure 3.

Instrumentation

The instrumentation consisted of:

1. A Bourns Model 304, miniature gage-pressure transducer which converts pressure into an electrical output by moving a wiper contact linearly across a precision wire-wound potentiometer;

2. A bridge circuit for the Bourns gage, designed to permit reading of gross pressure and fluctuations about a selected gross pressure;

3. A two-channel Brush recorder, Model Mark II; and

4. A 20,000-pound-capacity direct-reading Dillon dynamometer.

Handling Equipment

The handling equipment consisted of five major components:

1. A conventional stern sheave, approximately five feet in diameter;

2. A Western Gear "Cable Hauler," Model No. 1142, as shown in Figure 8. The clamping force in this unit is developed by means of three pairs of opposed air cylinders and is transmitted to the tracks by means of three pairs of opposed bogies, as shown on Figure II-2. Since each pair of bogies moves independently, englutment of lumps by one pair does not affect the clamping action of the others. The entire upper track assembly rotates about the forward drive sprocket. This arrangement, coupled with the independently suspended bogies, provides a means for accommodating large discontinuities in cable diameter.

   The unit is hydraulically powered, and is rated at 10,000-pounds line pull at 50 ft/min on jute-covered cable. It was modified by the addition of entrance and exit guide-roller assemblies to orient the cable fairing properly, as shown on Figure II-3;

3. A fairleading system consisting of roller assemblies and a trough which guided the fairing to the cable well;

4. A powered storage reel mounted in the forward cable well, equipped with a variable-speed drive system and band-type brake. An associated fleeter was mounted just above the reel, as shown in Figure II-4; and

5. A davit for lowering the depressor over the side.

TEST INSTALLATION

The installation was tailored to the accommodations available on the test vessel. The winch was mounted near the stern sheave (Figure II-5), and the cable led from the winch by means of a roller and trough system to a large powered storage reel in the forward cable well.

The winch was mounted with the open side of the tracks facing the starboard side. The centerline of the tracks was canted relative to the centerline of the stern sheave by that amount required to make the extension of the centerlines intersect at a small angle, in azimuth, at the location of the vertical guide roller mounted on the after end of the winch. The winch was thus placed in a bight of the cable, so that the load on the cable tended to force it into the tracks.

The after guide-roll assembly is shown in Figure II-3. The two horizontal rollers serve to orient the fairing for entry into the tracks; the upper roller is hinged to permit passage of the stops or modules. In operation, the trailing edge of the fairing is turned away from the vertically oriented guide roller (Figure II-3(a)). The cable is shown, in Figure II-6, passing through the forward fairing guide rollers and about the forward transfer roller; from the forward transfer roller to the cable trough in Figure II-7; forward to a second set of transfer rollers, Figure II-8; and thence into the cable well, Figure II-9.

The electrical conductors were connected to an existing slip-ring assembly on the storage-reel shaft. The slip-ring leads were led to a nearby laboratory in which the recorder was installed.

The depressor was placed under a davit on the port side of the winch.

The dynamometer was shackled to a pad eye on the after end of the winch frame, and thence to a wire rope shackled to a cable clamp.

PROCEDURES

For launching, the depressor was lifted over the side by means of the davit and lowered until the load was transferred to the main towline. An alternate procedure consisted of using an available ship's boom to place the depressor over the side, transferring the load to the main towline which was kept on short scope so that the depressor was suspended in the space between the bottom of the stern sheave and the waterline. Anti-sway lines were attached and the depressor thus carried to the launching area where launching was accomplished by merely paying out the main towline.

For payout, cable was pulled by the winch off the storage reel, which was braked to prevent slack.

For in-haul, the storage-reel operator attempted to synchronize the storage reel with the winch to maintain

tension in the towline between the winch and reel.

The "freeness" of the fairing on the cable was checked at the stern sheave during payout, with manual alignment of the sections that did not swivel freely.

DESCRIPTION OF TESTS

The winch was specifically tested to determine its capability for englutting the simulated instrument module, in-hauling and paying out the faired cable under operational conditions, and developing the predicted loading on the faired cable.

To determine the first item, a length of 3/4-inch-diameter wire rope, with an attached 1,000-pound weight, was rigged over the stern sheave and led forward between the winch tracks. The simulated module was then bolted to the cable between the stern sheave and winch. Tests were conducted by causing the winch to in-haul and payout the module several times. These tests were conducted in fair weather with dry equipment.

The second item was determined by observing the action of the winch during tests of the towed system. These tests were conducted in a sea state between three and four, with almost continuous heavy rainfall.

To determine the third item, a length of the faired cable was attached to the dynomometer, which was shackled, in turn, to the stern sheave, and the winch was engaged so

as to place a pull on the cable. The air pressure in the loading cylinders was maintained at 110 psig for this operation. The tension was noted and the air pressure in the loading cylinders then slowly reduced until slippage relative to the tracks was evidenced by a decrease in the cable tension. As the winch was hydraulically powered, it was possible to maintain a steady pull under fully stalled conditions for a short period of time. These tests were conducted in fair weather with dry equipment. Tests of the type described in (3) were not run, due to lack of sufficiently sensitive pressure gauges.

For the tests of the towed system, the depressor was placed overboard and launched on a short scope of cable at a ship's speed of three knots. Approximately 100 feet of cable was payed out and the towing characteristics observed at speeds of three, six, and eight knots. The cable scope was then increased, while at a speed of eight knots, and the depth of the depressor monitored continuously during the lowering. The pressure signal was lost, however, at an indicated depth of 420 feet. The lowering was continued until 700 feet of cable was in the water. The ship's speed was then decreased to six knots and the tensiometer attached to the towline. When the speed was decreased, the pressure signal was regained.

The tow was continued at six knots. Depth and tension indications were recorded and the angle of the tow cable at

the surface was measured. Speed was then increased to eight knots and the pressure signal again lost. Tension indications were recorded and 90-degree port and starboard turns made with a standard rudder.

Speed was then decreased to three knots and the tensiometer removed from the line. The pressure signal was regained when the speed was reduced. The system was in-hauled and, due to a difficulty with the storage reel, the cable was flaked-out on the deck ahead of the winch. It was noted that the depth signal was lost when the spliced section of the cable traversed the stern sheave and was regained when the traverse was complete.

The in-haul was stopped at a cable scope of approximately 50 feet, and the system towed at a speed of six knots toward sheltered water. While towing at six knots, the cable slipped relative to the winch. Speed was reduced, the tensiometer attachments (without tensiometer) attached to the towline, and the tow toward sheltered water continued at a speed of six knots. Retrieval was completed after reaching sheltered waters.

After retrieval, the cable was flexed, under no load, in the region of the splice; it was observed that the pressure signal fluctuated and was occasionally lost during the flexing. The signal was regained and became steady when the spliced region was straightened.

RESULTS

The winch englutted the module without difficulty or adverse mechanical effects. As shown in Figure II-2, the flexible tracks conformed to the module and maintained contact with the cable over nearly the full remaining length of track.

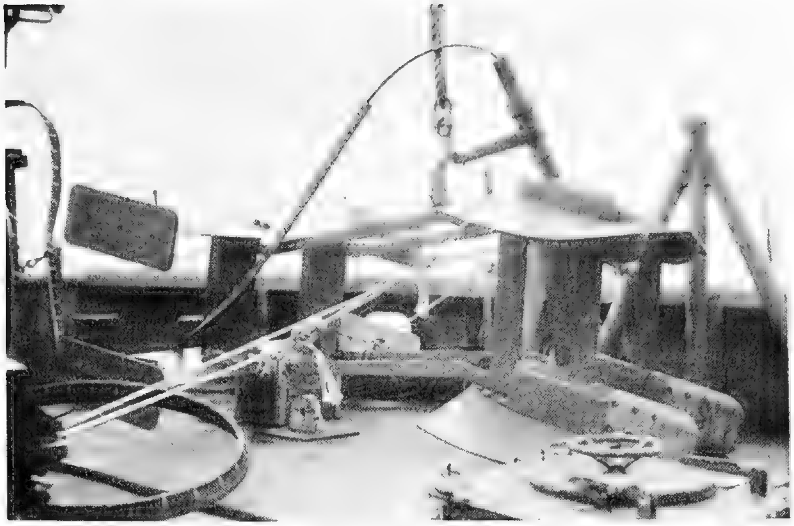
The faired cable assembly was held, in-hauled, and payed out by the winch with no evidence of adverse mechanical effects on the winch or the cable, fairing, and clips, while cable tension was within winch capacity; when excessive tension was placed on the winch, slippage of the cable relative to the track and fairing occurred (Figure II-10).

The winch exerted a maximum pull of 10,000 pounds under stalled conditions, and sustained that load for all loading-cylinder air pressures down to 90 psig. As the winch was stalled, slippage was evidenced by creeping of the tracks, which carried along the unanchored fairing.

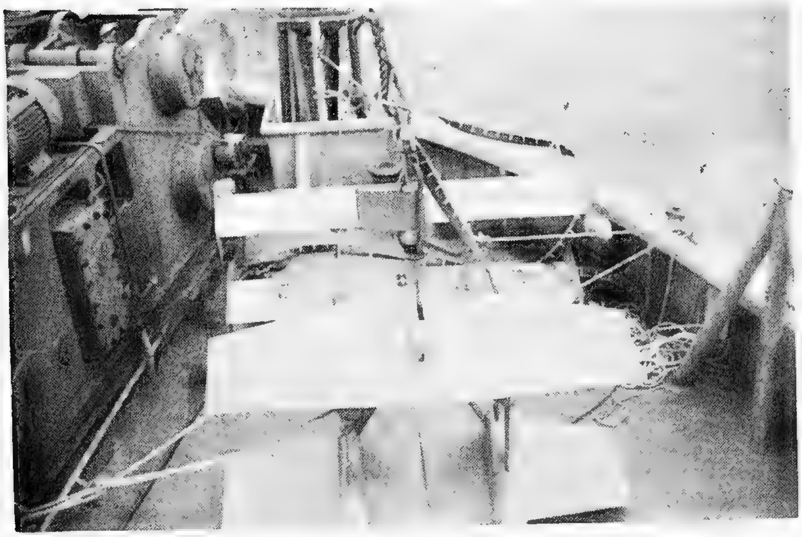
The towed-system elements exhibited stability for all conditions of the test. The system towed with a very slight "kite" to starboard. It maintained depth during turns; when entering a turn, the tow cable assumed a slight angle to the inside of the turn, returned to a straight course during the steady part of the turn, and veered slightly to the opposite side when the turn was checked, thereafter returning to its straight-course position.

With a cable scope of 700 feet and a speed of 6 knots, the indicated depth was 660 to 665 feet; the line tensions averaged 3500 to 4000 pounds, with occasional surges to 9000 pounds. The angle between the axis of the cable and the water surface was approximately 61 degrees. At a speed of 8 knots, the observed tensions were between 5500 and 6000 pounds, with occasional surges to as much as 9000, 10,000, and 11,000 pounds. One surge to 13,000 pounds was observed.

The depressor was damaged when launched in rough weather. The cable fairing, clips, and clamps suffered no damage during towing or passage through the winch, although a number of clips were torn loose and deformed by "hang-ups" in passage through the roller and trough system. As a result, some clips failed to swivel freely on the cable and had to be manually aligned.



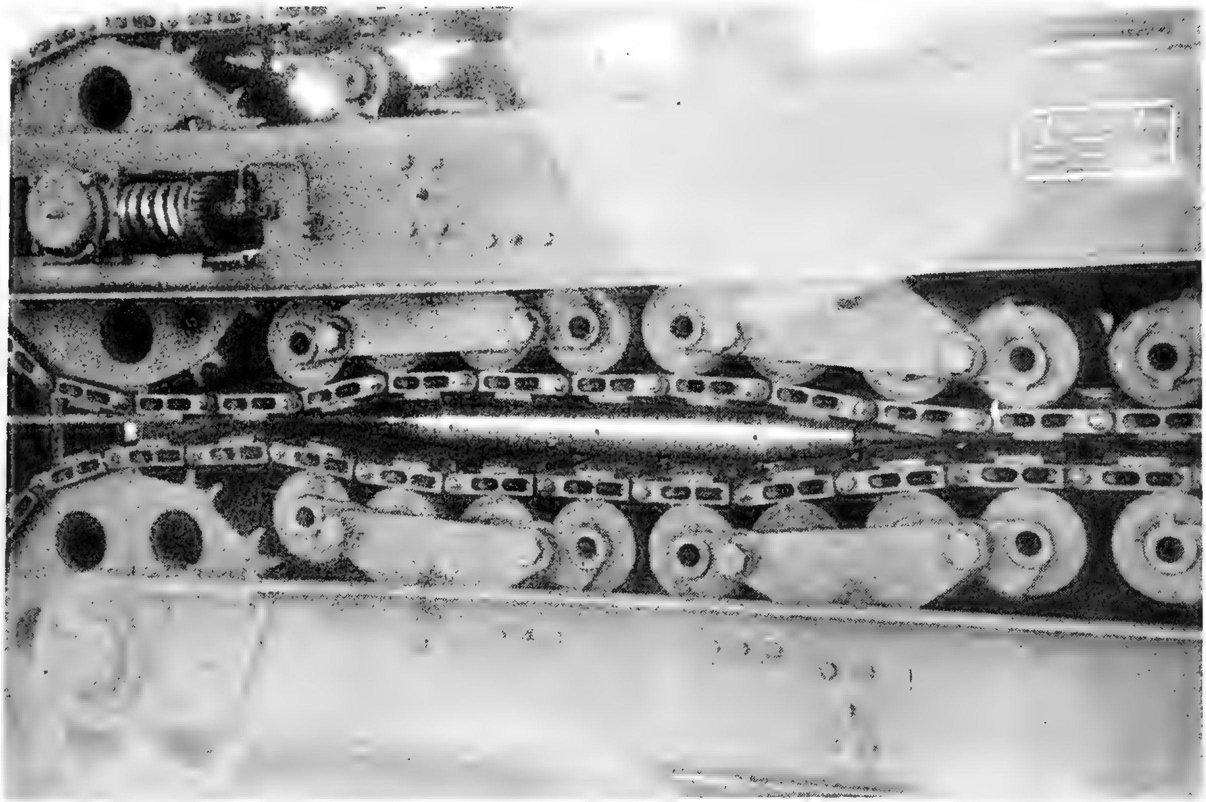
a. View Showing Module and Handling Bale



b. View Showing Module Fairing, Bent Starboard Vertical-Tip Plate, and Additions to Stabilizers

Depressor as Tested at Sea

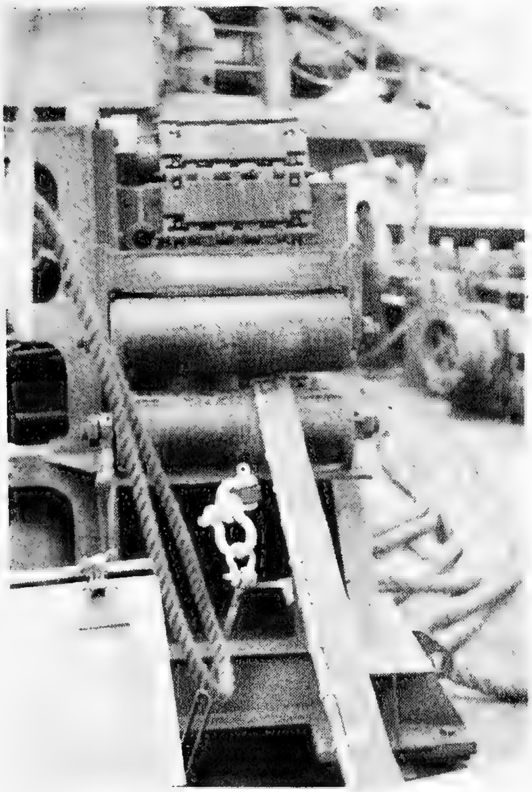
Figure II-1



View of "Cable Hauler" Showing Dummy Module
Gripped Between Treads and
Method of Loading the Tracks

Figure II-2

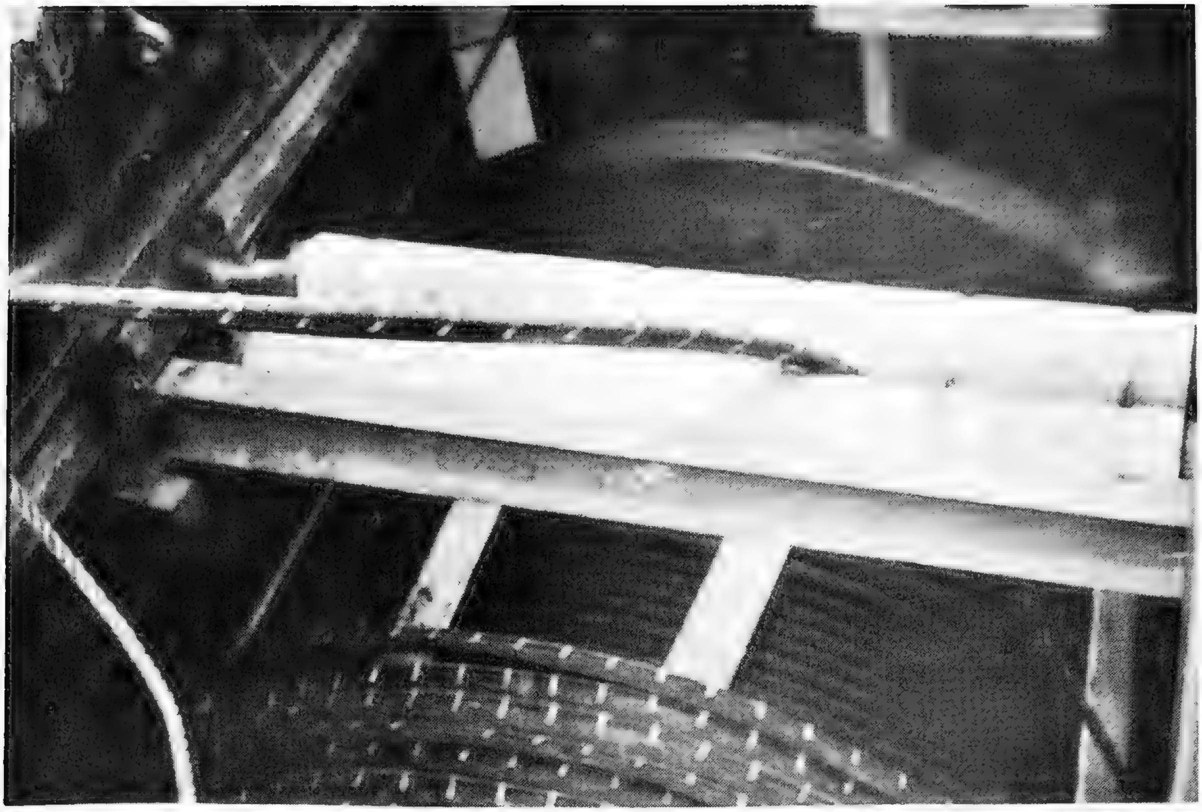
a. View showing vertical and horizontal rollers. The fairing is normally positioned with the trailing edge facing outward.



b. View showing normal position of the fairing in relation to the guide rollers.

Fairing Guide Roller Installation

Figure II-3



Cable Reel Used for Storage of Faired Cable
During Sea Trials

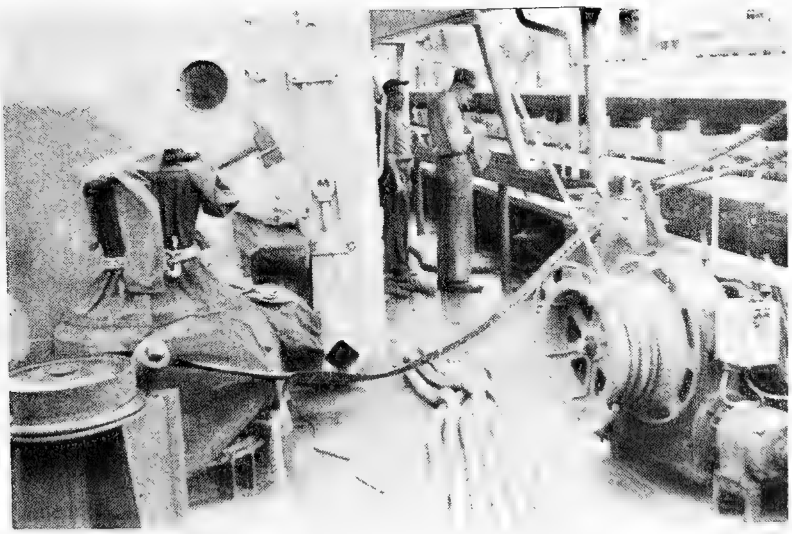
Figure II-4

Sheave Used for Handling Faired Cable
Over the Stern

Figure II-5

Forward Guide Rollers and Transfer Sheave as Seen from Port

Figure II-6



Arrangement for Transferring Fairing from Forward Guide Rollers to Trough as Seen Looking Forward

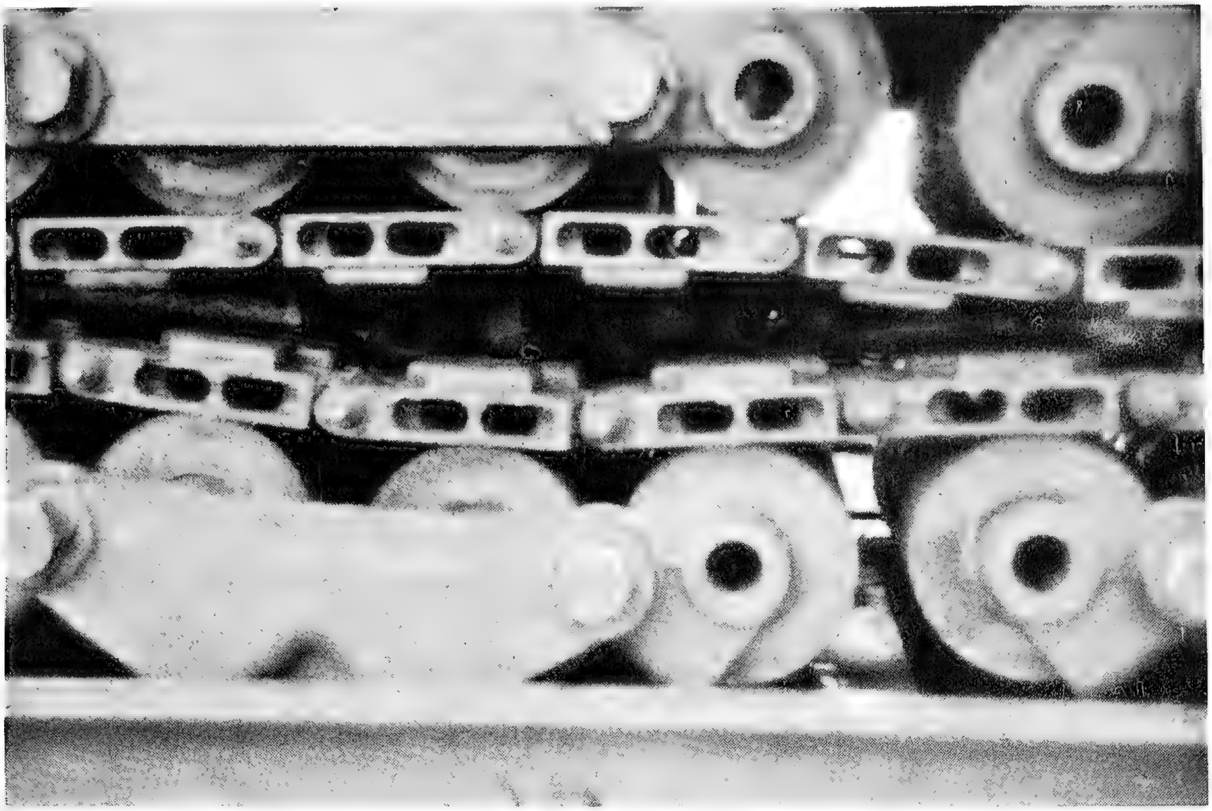
Figure II-7

View Looking Aft Showing Forward Transfer Roller

Figure II-8

"...and Thence into the Cable Well"

Figure II-9



Results of Cable Slip

Figure II-10

APPENDIX III

ADDITIONAL PERFORMANCE CONSIDERATIONS

APPENDIX III

ADDITIONAL PERFORMANCE CONSIDERATIONS

PREVIOUS ANALYSES

In order to perform the analysis prerequisite to the design of the cable-towed oceanographic instrumentation system, it was necessary to make specific assumptions regarding the nature of the hydrodynamic loads imposed on the faired cable and to estimate the value of the resistance coefficient. In the analysis reported in (1), it was recognized that the expression for the dependence of the hydrodynamic loads on the cable angle, as given by Whicker (4), is probably more nearly correct than that given by Eames (5). Eames' formulation was selected for this application, however, since its use results in a conservative estimate of the geometry of the cable catenary and overstatement of the tension. A value of 0.2 was selected for the resistance coefficient on the basis of the best experimental data available at that time.

On the basis of the foregoing, and the simplifying assumption that the cable angle at the depressor, with respect to the horizontal, would be nearly 90 degrees, the results of the preliminary design study indicated that with a depressor downforce of 4,450 pounds and a length of 6,200 feet of faired cable, a maximum depth of 5,000 feet could be obtained at a towing speed of 7¼ knots.

ADDITIONAL CONSIDERATIONS

Since completion of the preliminary design, further information concerning the hydrodynamic resistance of faired cables has come to hand. Results of recent tests on faired cable (7) show that the minimum value of the resistance coefficient to tbe expected for TF-84 fairing is closer to 0.3 than 0.2 in the Reynolds' number range characteristic of the present system.

In light of this information, a study was made of the effects of resistance coefficient and cable-loading assumptions on the performance of the system. A digital computer solution based on Whicker's loading assumptions, with a resistance coefficient of 0.3 and using computed values for the cable angle at the depressor (84 degrees at the 7¼-knot speed) was provided by the David Taylor Model Basin. Calculations based on Eames' assumptions, using computed values for the cable angles and resistance coefficients of 0.2 and 0.3, were also made. Results are tabulated below for a towing speed of 7¼ knots, and are shown as a function of speed on Figure III-1. The results at 7¼ knots given in (2); based on Eames' method, a resistance coefficient of 0.2, and a cable angle at the depressor of 90 degrees, are included in the table for comparison.

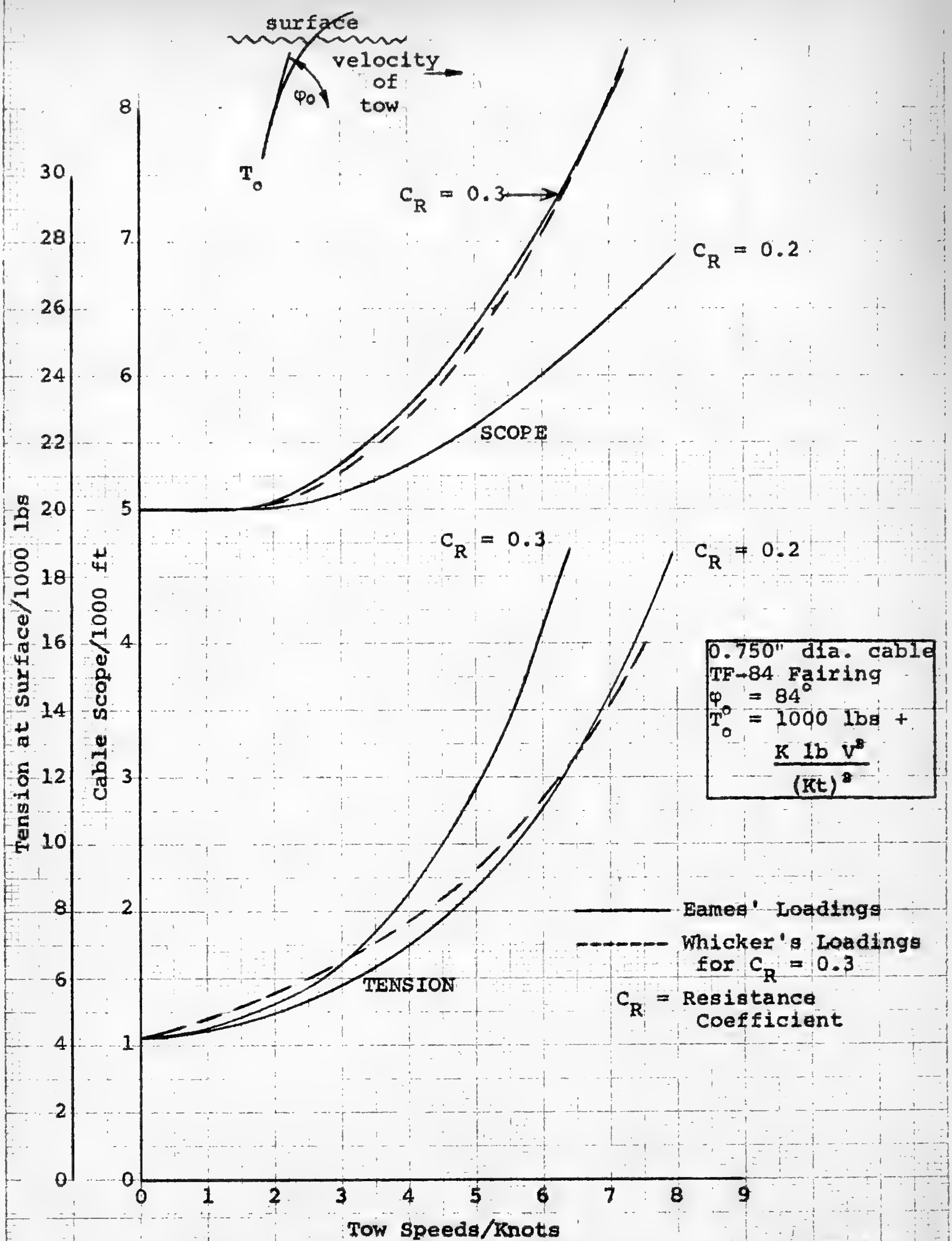


Figure III-1 - Comparison of Cable Lengths and Maximum Tensions Required to Attain a Depth of 5000 Feet Computed According to Whicker and Eames

Cable Loading Assumption	Eames	Eames	Eames	Whicker
Resistance Coefficient	0.2	0.2	0.3	0.3
Cable Angle at Depressor at 7¼-knot tow speed	90°	84°	84°	84°
Scope of Cable Required to Attain a Depth of 5000 ft	6,200'	6,530'	8,350'	8,350'
Tension at the Surface	15,000 lbs	15,400 lbs	26,000 lbs	15,000 lbs

The results indicate that if Whicker's loading assumptions are correct, the desired performance can be attained, even with a resistance coefficient as high as 0.3, without exceeding the arbitrarily imposed limitation of 15,000 pounds on cable tension.

Computations for Sea Trials

It was determined from tow tests that at the higher values of towing speed the depressor imposes an angle of 81 degrees on the lower end of the cable. Since this value is slightly less than the 84 degrees predicted in (2), and since an 0.782-inch-diameter cable was to be used for the sea trials in place of the 0.75-inch-diameter cable on which previous performance predictions were based, estimates of the cable geometry for these new conditions were required for comparison with sea-trial data. These computations were made on the basis of Eames' formulation, using a resistance coefficient of 0.3; the results are shown in Figures III-2 and III-3.

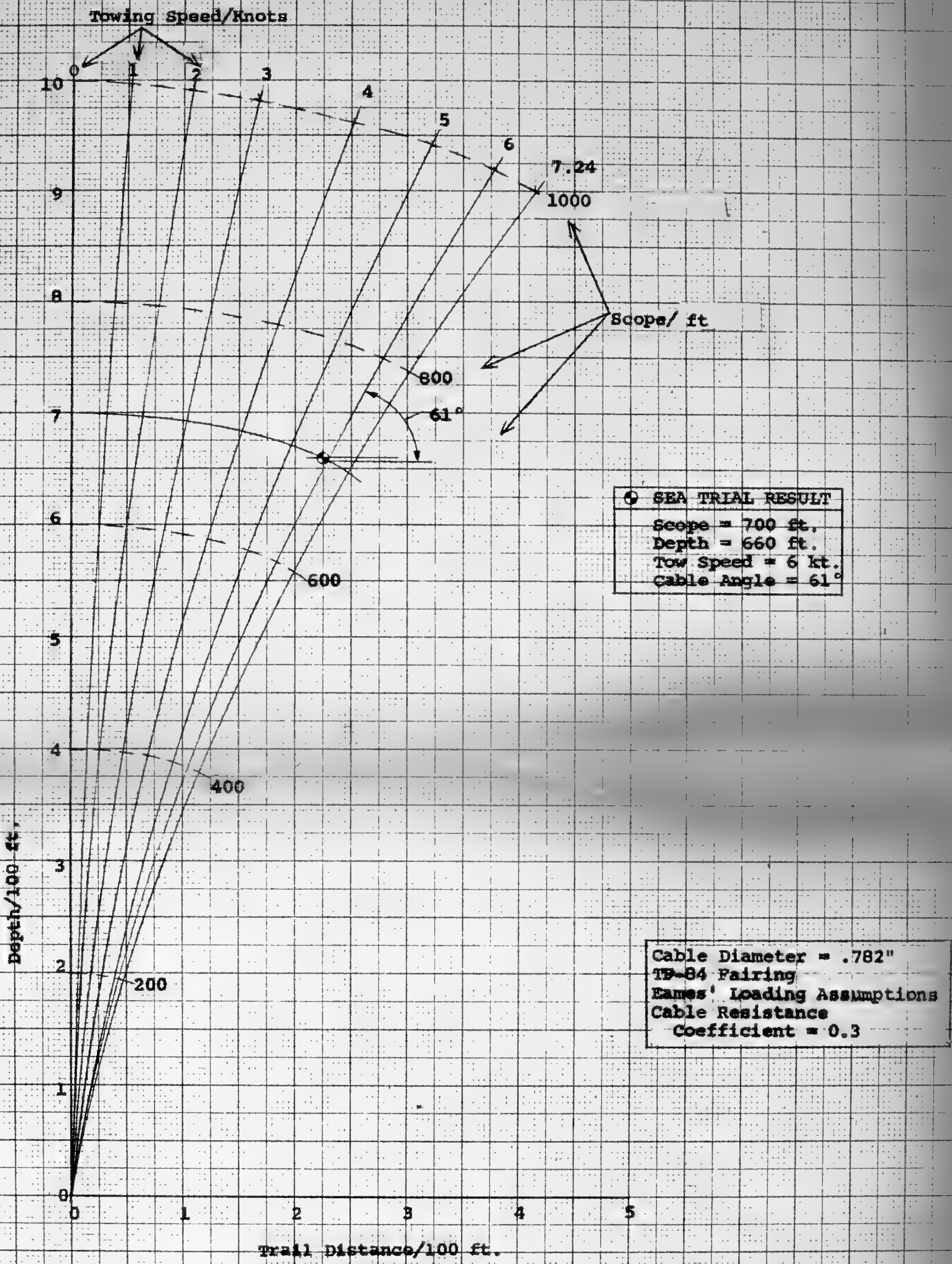


Figure III-2 - Predicted Depth as a Function of Trail Distance for Various Values of Cable Scope and Towing Speed

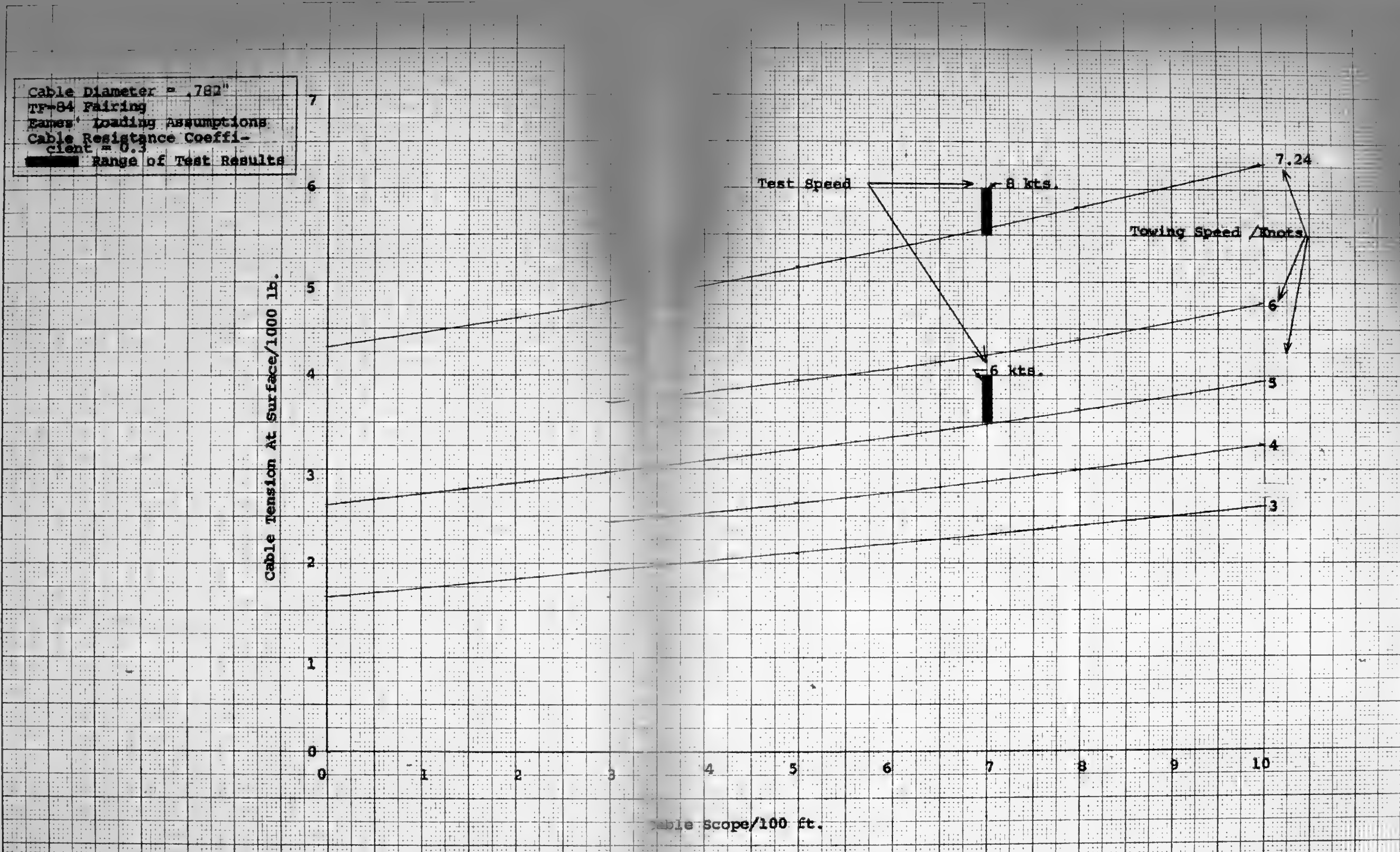


FIGURE III-3 - Predicted Cable Tensions As A Function Of Scope For Various Towing Speeds

Eames' method was selected for these computations as an expediency, since solutions for the cable equations based on Whicker's loading functions were not readily available. It is believed, however, that the differences in catenary estimates made by the two methods is small in the present case. This may be shown by considering the differences in the loading functions basic to the two assumptions.

First, for large values of the ratio of the weight per unit length of cable to the resistance per unit length, i.e., for low towing speeds, the ratio of the functions approaches unity. Second, for large values of the cable angle, the ratio of the functions approaches unity. Third, for the initial cable angle pertaining to the present case, $\varphi = 81^{\circ}$, the radius of curvature of the cable in the neighborhood of the depressor is found to be 1,675 feet on the basis of Eames' assumptions, and 1,680 feet on the basis of Whicker's assumptions, i.e., they may be considered equal.

APPENDIX IV

THE EFFECTS OF BENDING AND TENSION ON THE STABILITY OF CABLE FAIRING

APPENDIX IV

THE EFFECTS OF BENDING AND TENSION ON THE STABILITY OF CABLE FAIRING

INTRODUCTION

A trailing type fairing, attached to a strut of circular cross section and moved through a fluid, trails directly behind the strut if hydrodynamically stable. Only the hydrodynamic forces are of interest in determining the static or "zero-frequency" stability. If, however, the strut is permitted to curve in the direction of motion, forces are induced in the fairing which tend to rotate it relative to the strut. The relevant forces are those due to the bending of the fairing and those due to externally imposed axial loads. The nature of these effects is discussed below.

ASSUMPTIONS

For the purpose of this discussion, the following assumptions are made:

1. The axial load carried by the cable, relative to that carried by the fairing, is so great that the cable can be considered a rigid member;

2. The fairing is constructed of a homogeneous material, the elastic modulus of which is quite small compared to that of the cable;

3. The fairing is free to swivel about an axis normal to the center of a right section of the cable;

4. The end points of the fairing are constrained from up or down motion relative to the cable; and

5. The fairing is of neutral buoyancy and is attached to the unbent cable under zero strain.

BENDING

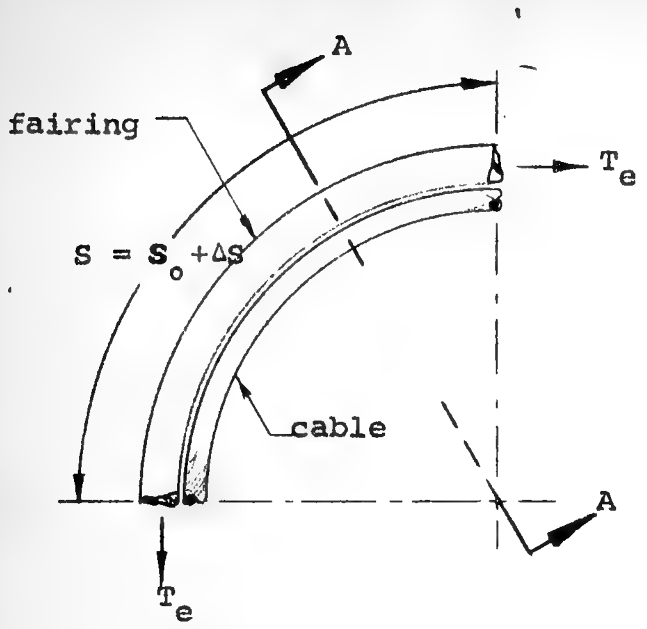
Assume a segment of fairing of initial length S_o, attached to a segment of cable with curvature C_o, as illustrated in Figure IV-1. Consider first the case for a small elastic cord of area ΔA, as shown on Figure IV-1(c). When the cord is positioned such that the angle ψ obtains between the line a-b and the plane of curvature of the cable, the moment per unit length, ΔL, due to the elemental axial force, ΔT, acting on the cord, is clearly

$$\Delta L = yC \sin \psi \, \Delta T \quad , \qquad [1]$$

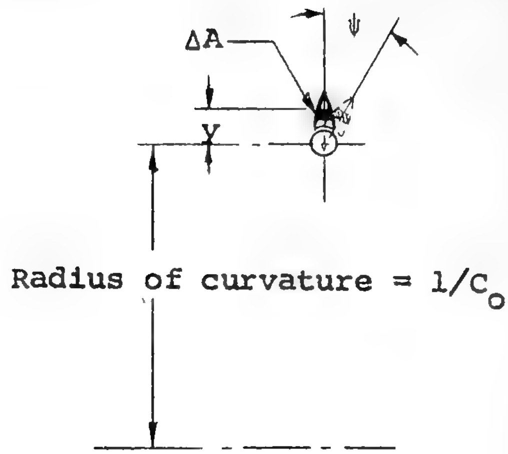
where C is the curvature of the elemental cord positioned at distance y from the center of the cable.

The assumptions previously stated require that the elongation of the cord per unit length, ϵ, be given by yC_o, if ψ is small. Then, since

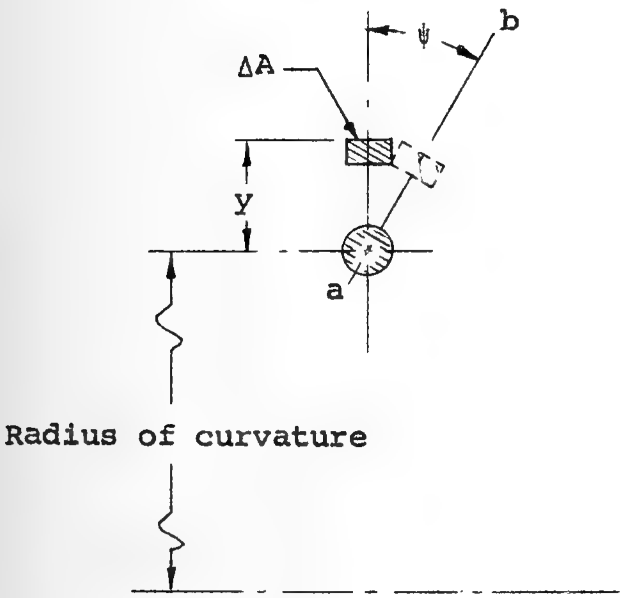
$$\Delta T = \epsilon E \Delta A \quad ,$$



a. General Configuration



b. Section A-A



c. Section A-A Idealized

FIGURE IV-1 - Definition Sketches For Derivation Of Structural Effects On Fairing Stability

where E is the elastic modulus of the cord, the moment per unit length of cord necessary to maintain the cord in equilibrium in the displaced position, is

$$\Delta L = Ey^2 C_o C \psi \Delta A .$$

But

$$C = \frac{C_o}{1+yC_o} ;$$

hence

$$\Delta L = EC_o^2 \psi \frac{y^2}{1+yC_o} \Delta A .$$

L_B, the total moment per unit length due to bending, is then simply the summation of the moments due to all such elementary cords, and is given by

$$L_B = EC_o^2 \psi \int \frac{y^2}{1+yC_o} dA .$$

Since $yC_o << 1$ for most practical problems, we may write

$$L_B = IEC_o^2 \psi , \qquad [2]$$

where I is simply the second moment of the cross-sectional area of the fairing, referenced to the axis normal to line a-b which intersects the center of the cable.

TENSION

L_T, the moment per unit length due to an externally imposed load, T_e, may be found by integrating [1] over the total area of the fairing, assuming ψ to be a small angle and $yC_o \ll 1$. y, in this case, is just the distance from the center of the cable to the line of action of the external load, and is written $\bar{y}$. Dropping the subscript e, we obtain

$$L_T = TC_o\bar{y}\psi \quad . \qquad [3]$$

APPENDIX V

REFERENCES

REFERENCES

(1) Ellsworth, Wm. M.: General Design Criteria for Cable-Towed Body Systems Using Faired and Unfaired Cable; Systems Engineering Division, PneumoDynamics Corporation, Report No. TN-SEDU-6634-1, October 1960

(2) Ellsworth, Wm. M. and Gay, S.M.: Preliminary Design of a Cable-Towed Oceanographic Instrumentation System; Systems Engineering Division, PneumoDynamics Corporation, Report No. TN-SEDU-6634-2, February 1961

(3) Bonde, L.W.: Investigation of a Tractor-Type Winching Machine for Handling Faired Cables and In-Line Instrument Modules; Systems Engineering Division, PneumoDynamics Corporation, Report No. TN-SEDU-6634-3, July 1962

(4) Whicker, L.F.: The Oscillatory Motion of Cable Towed Bodies; Doctoral Dissertation; University of California, 1957.

(5) Eames, M.C.: The Configuration of a Cable Towing a Heavy Submerged Body from a Surface Vessel; Naval Research Establishment (Canada) Report PHx-103, November 1956.

(6) David Taylor Model Basin letter 9250/7100/(549:JT:lm) to PneumoDynamics Corporation, dated 21 May, 1962

(7) Gibbons, Thomas: Hydrodynamic Characteristics of a Systematic Series of Trailing-Type Cable Fairings; David Taylor Model Basin unpublished manuscript.

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