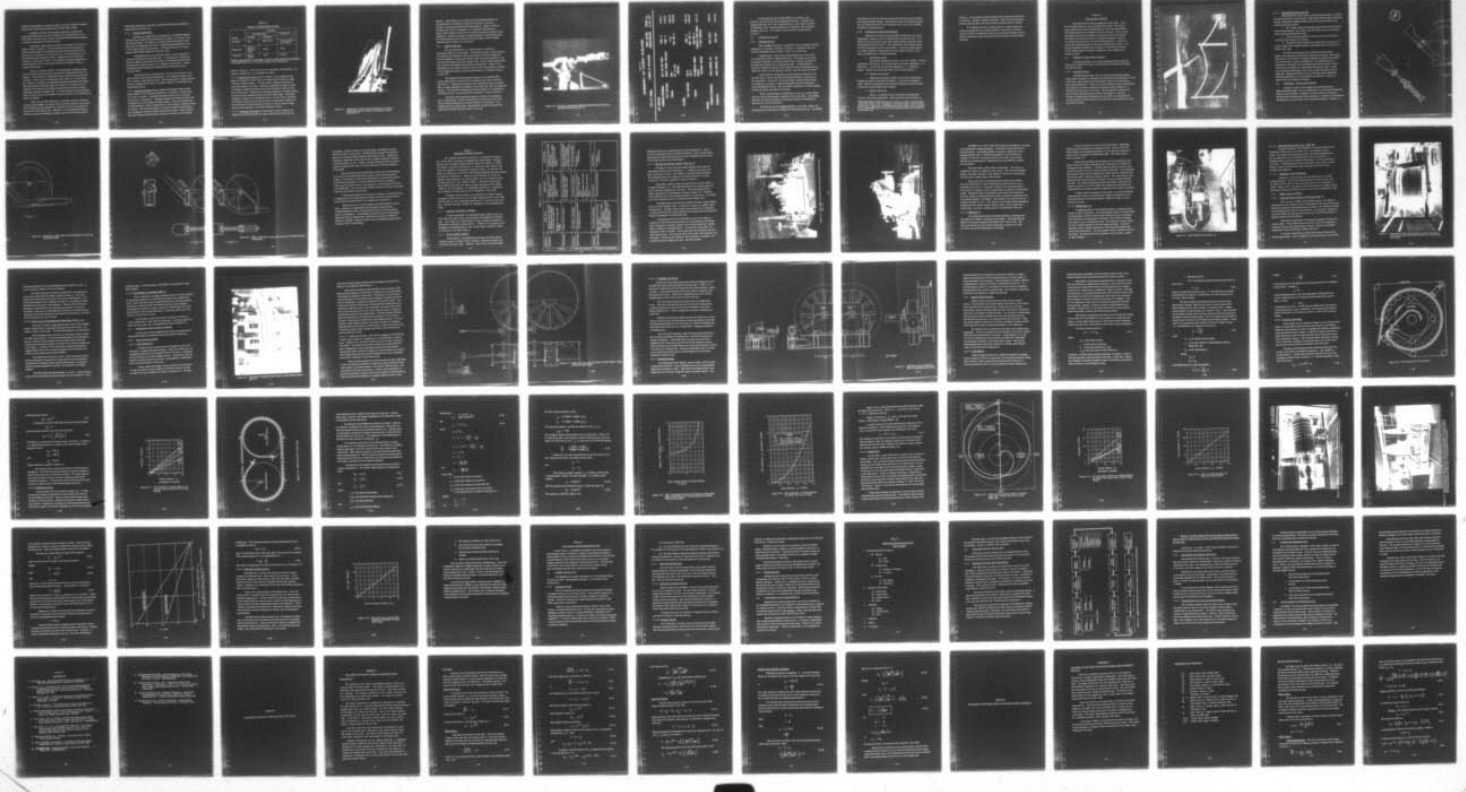
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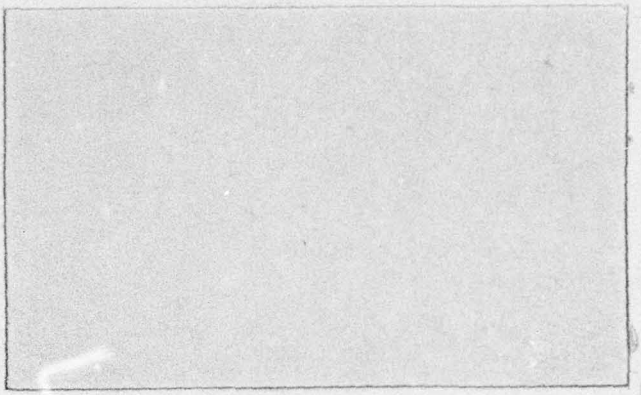
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# (6) FAIRED CABLE HANDLING SYSTEMS.

(11) 25 Nov 68

by

(10) D. B. Dillon

(9) Technical Report

(12) 117 p.

25 November 1968

(15) Contract N00600-68-D-0239, N00167-68-D-0007

Naval Ship Research and Development Center
Washington, D. C.

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## SUMMARY

This study combines a literature review of winching, fairleading and storage mechanisms for cables equipped with streamlined hydrodynamic fairing, as well as an evaluation of several new handling concepts. The handling systems are considered according to the suitability of their use with the various classes of cable fairings. Special emphasis is given to the problem of handling a faired cable aboard a submerged vehicle.

In this application, trailing fairings lack hydrodynamic efficiency and integrated fairings have not yet been developed to an acceptable level. Fully enclosing fairings are suitable, and it is believed that a reliable, functional submarine handling system can be developed to meet the requirements.

However, before a particular fairing/handling system combination can be chosen, considerable engineering data regarding the physical handling properties of the various fully enclosing fairings must be obtained. A five-year development program to measure these parameters, design, test, and produce an acceptable handling system is included.

Four towing systems constitute, in the opinion of HRC, the choices most likely to yield successful near-term designs for a submarine communications buoy handling system. After the necessary engineering data have been obtained, these systems should be compared in detail and the most suitable selected for further development. They are listed in descending order of preference.

1. Internal Winch/Guide Tube Storage/Segmented, Fully Enclosing Fairing
2. Conventional Drum Winch/Storage/Sectional Fairing
3. Conventional Drum Winch/Storage/Segmented, Fully Enclosing Fairing
4. Twin Drum Winch/Slack Drum Storage/Segmented, Fully Enclosing Fairing.

The study of functional requirements begins with a summary of the hydrodynamic function of cable fairing. The development of a classification scheme for cable fairings follows, excluding hair and ribbon vibration dampers. The primary handling characteristics of each type are identified. The discussion of the functional requirements continues with a brief analysis of the essential components of a faired-cable handling system(s): winching, fair leading, and storage. Finally, the constraints placed on any system operating on a submarine are briefly outlined.

The analysis of faired-cable handling systems is presented following an introduction in three major topics, and concluded evaluation and recommendations for the systems considered. The first topis is a discussion of the functional requirements of cable-fairings and their handling systems. The second topic deals with the problems of guiding the faired cable through the handling system. The final major topic is a study of existing faired cable winching and storage devices along with a detailed analysis of several devices proposed in order to alleviate the problems encountered in past systems.

Fairleading devices are classified according to the type of contact they make with the faired cable-rolling or sliding. Fairleading systems are classified according to the extent of control they exert on the faired cable-continuous or point-to-point. The fairleading systems that have been used on existing faired-cable handling systems are discussed, and their limitations noted. It is found that point-to-point fairleading has been used almost exclusively, and that with only moderate success. Only the experimental CHAN system has continuous fairleading; although it has not received extensive trials, the information available is entirely favorable to continuous fairleading.

None of the existing systems included a bellmouth at the overboarding sheave that was designed specifically for orienting faired cable as it enters the handling system. This has caused further problems with handling

faired cable to the extent that a practicable submarine handling system is impossible without this feature. Two concepts are proposed for further testing; one a rolling contact device, the other having sliding contact with the faired cable.

The winching/storage devices are the final main topic of the report, and nearly 40 percent of the total effort was expended in this phase. The devices that were studied were classified according to whether they applied tension to the cable directly or indirectly. Winching systems that have been used with faired cable were studied first to provide a historical background of successful concepts and problem areas. The existing systems ranged from experimental to fully operational devices. In accordance with the widely different data bases, the opinions expressed in the literature ranged from vague, but hoepful (experimental) to very specific condemnation of the inadequacies of some operational devices. Direct tension winches have been used exclusively except in trials of other devices. The handling systems studied include the SQA-series variable depth sonar; CHAN, the only faired-cable handling system designed for submarine use; the Bowing and North American Aviation hydrofoil-towed sonar; and the AQS-series air-towed sonar gear. The trials of indirect tension devices include the Pneumo Dynamics Corporation work with commercial cable haulers; the Pneumo Dynamics Twin Drum traction winch; and the slatted capstan designed by the Naval Research Laboratory. Inasmuch as all the existing systems except CHAN were designed for surface ship operation, two direct tension drum winches for submarine application were developed. One is intended to haul a relatively short length of fully fiared cable. The other was designed to handle a much greater length of cable, but only a fraction may be faired.

In addition, two indirect tension concepts were developed and analyzed in detail, seeking to provide the potential for slack storage that these devices afford, while avoiding the excessive intricacy and bulk of

existing indirect tension devices. Both systems are variations of the track-type cable hauler which grip the cable between two driven surfaces. In one, the hauler tracks follow a circular path with the storage drum mounted concentrically within the periphery. The gripping force is generated by tension in the cable and backs in conjunction with the curvature of the path.

The second concept uses a single track to clamp the cable against two thin, large diameter drums. This concept provides greater mechanical simplicity than the former, but at a penalty in compactness: the storage drum is not nested within the mechanism.

Finally, a hybrid system is described. The AIGS towing system handles a plastic-jacketed, unfaired cable from within the pressure hull of a submarine. This avoids the major drawback associated with submarine towing sysems: inaccessibility for maintenance and repair while submerged. Providing the towing requirements permit a faired-cable length no more than about half or three-fourths of the submarine length, the AIGS system could be adapted to handle faired-cable by storing it in a small guide tube laid along the submarine hull.

On the basis of this study, the various fairings and handling systems were evaluated, the four most promising combinations selected, as described in Section 5 and summarized in the opening paragraphs above.

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## Section 1

# INTRODUCTION

Perhaps the single most powerful way to improve the performance of a towed system is to fair the towcable. As greater demands are made of towed systems by the military and technical community, the use of faired towcables becomes increasingly necessary. Yet proposals to use faired cable are often met by opposition; it is claimed that faired cable is too difficult to handle and unreliable in operation. These claims are not unfounded, for many systems have been patched together from components designed to handle bare cable, with largely unsatisfactory results.

It is the purpose of this report to identify the essential functions a faired cable handling system must perform, to summarize the methods, or lack of methods, that have been used to perform these functions and to propose and evaluate new concepts for handling faired cables. Emphasis has been placed on handling faired cables from a submerged vehicle.

This report was prepared for the Naval Ship Research and Development Center, Carderock, Maryland, under Contracts N00600-68-D-0239 and N00167-68-D-0007.

## Section 2

## FUNCTIONAL REQUIREMENTS

In this section, the hydrodynamic function of cable fairing is summarized and the various types of fairing classified in order to identify the important factors that must be accounted for in a handling system design. The second subsection a discussion of the basic functional elements of any faired-cable-handling system, and a brief outline of the constraints placed on any system operating on a submarine.

### 2.1 CABLE FAIRING

Cable fairing is used for essentially two purposes: to decrease the hydrodynamic resistance of the typically round flexible strength member in a cable-tethered system, and to ameliorate the eddy-shedding characteristic of round members to reduce the tendency to vibrate laterally. Although simpler means have been found to alleviate the latter problem, * the only practical approach to drag reduction consists of streamlining (or fairing) the member in question.

#### 2.1.1 Hydrodynamic Function

In cross-section a typical fairing resembles a symmetrical hydrofoil section. No camber is used, as "lift" or side force is not desired. for the same reason, it is necessary that the fairing not assume an angle-of-attack. That is, it must be constructed such that it turns freely on the cable so that the axis of symmetry aligns with the stream. In this position, if the fairing is symmetric and undistorted, it will offer lower resistance to the stream than the enclosed member along and hence permit achievement

*Enclosing the cable with a fine wire braid from which tufts of plastic material are streamed is the most common method of reducing cable vibration without applying a fully streamlined trailing surface. Inasmuch as the problems of handling "haired fairing" are quite different from those of cable fairing, and the hydrodynamic efficiency of the tufts is much less than for fully faired cable, they are not considered further in this report.

of higher towing speeds, or greater depths, or the use of shorter lengths of cable at lower tensions than required if un-faired cable is used.

In addition to those characteristics necessary to achieve the primary goal that leads to the use of fairing, the fairing must be so contrived that it can be fair-led, winched, and stored without physical damage or serious compromise of the primary goal.

These basic requirements have led to the evolution of a series of distinct "types" of fairing. The factor that has exerted the greatest influence on their design is the requirement that the fairing align with the stream so as to produce no lift or side forces. It is quite feasible to produce fairings with acceptable tolerance regarding the cross-sectional symmetry. In general, however, the equilibrium towing configuration of a faired cable is a curve in the plane of maximum stiffness, which is structurally unstable.

Hence, to perform properly, the hydrodynamic stablizing torque per unit rotation must exceed the structural de-stabilizing torque per unit rotation. The structural de-stabilizing torque is proportional to the local curvature. Thus it may be avoided only if the cable is not curved (which is, in general, contrary to the physical laws governing cable-towed bodies). The structural de-stabilizing torque may, however, be reduced by minimization of the product of the elastic modulus and the moment of inertia taken about the axis of bending, (EI). It is also necessary that the effective center or line of action of the tensile forces lie upstream of the hydrodynamic center of pressure.

Thus, for any finite value of EI, a curvature exists for which the fairing will be unstable, in the sence that it will turn relative to the stream and, acting like a long, slender wing, will pull or divert the towed device to one side of the desired track. Moreover, if the unsupported length of fairing is too long, the build-up of tension may reach a level that results

in the same phenomenon, because of the stretch-induced accumulation of slack fairing at the lower end.

### 2.1.2 Fairing Classification

Fairings are classified on two parameters: the relative length of cable faired by a single fairing element and the degree of enclosure of the cable by the fairing. The length of a sectional fairing elements is of the same order of magnitude as its chord. Segmented fairing lengths are one or two orders greater than the chord. A continuous fairing is a fairing segment whose length equals the total cable scope.

Trailing fairings do not enclose the cable, but are attached to it at discrete intervals by clevis-like clips. A second group of fairings enclose the cable by a tube within the fairing or by nosepieces which attach the fairing to the cable along the entire length of each element. Finally, integrated fairings transmit the towing stresses through the fairing structure itself.

Any fairing may be classified into one of the nine fields in a 3 x 3 matrix formed from the above parameters. For pragmatic reasons, however, several of these fields are null. The fairings used in existing systems that were studied for this report are listed in Table 2-1 according to this classification matrix.

2.1.2.1 Sectional Fairings usually consist of essentially rigid elements, linked together to limit torsional deflections. In general, trailing fairing is not used in sections. Integrated sectional fairing is more commonly referred to as "articulated." Handling systems for sectional fairings must be designed to avoid excessive bending stresses. Drums and sheaves are of large diameter, may be resilient and, for long sections, polygonal instead of circular. The torsional rigidity of sectional fairings has been used by some designers to eliminate guide tubes and bell mouths. When, however, the inter-section torsion link fails, manual correction of the fairleading

Tabel 2-1

FAIRING CLASSIFICATION MATRIX

| Length | Enclosure | | |
|---|---|---|---|
| | Free Swivelling | | Integrated |
| | Trailing | Enclosing | |
| Sectional | none* | SQA series | North American (articulated) |
| Segmented | Pheumo-dynamics, CHAN | AQS-2 | none* |
| Continuous | SQA-10 "Mod 0" | none* | Boeing |

*"None" indicates that no reasonably complete handling system was developed, even though test pieces of the fairing may have been constructed.

system is required. This is not only time consuming, but hazardous. In a submarine application, it is impossible of course.

2.1.2.2 Segmented Fairings must be relatively limp to bending in the plane of the chord in order to pass over sheaves and to follow the cable "catenary" without twisting. For this reason, segmented fairings are often used trailing, with clips. Enclosing, segmented fairings are made by molding a flexible metal tube such as B-X electrical conduit into the nose of the fairing. Replacement of damaged dsegments is tedious because the cable must be threaded through the entire faired length. Segmented fairings as a group are torsionally quite limp, so that closely controlled fairleading is required for smooth operation as illustrated in Figure 2-1. The clips used for trailing fairing are prone to catch on nearby irregularities and protrusions, so the latter should be avoided.

2.1.2.3 Continuous Fairings have essentially the same properties as segmented fairings except that the entire cable length is faired by one

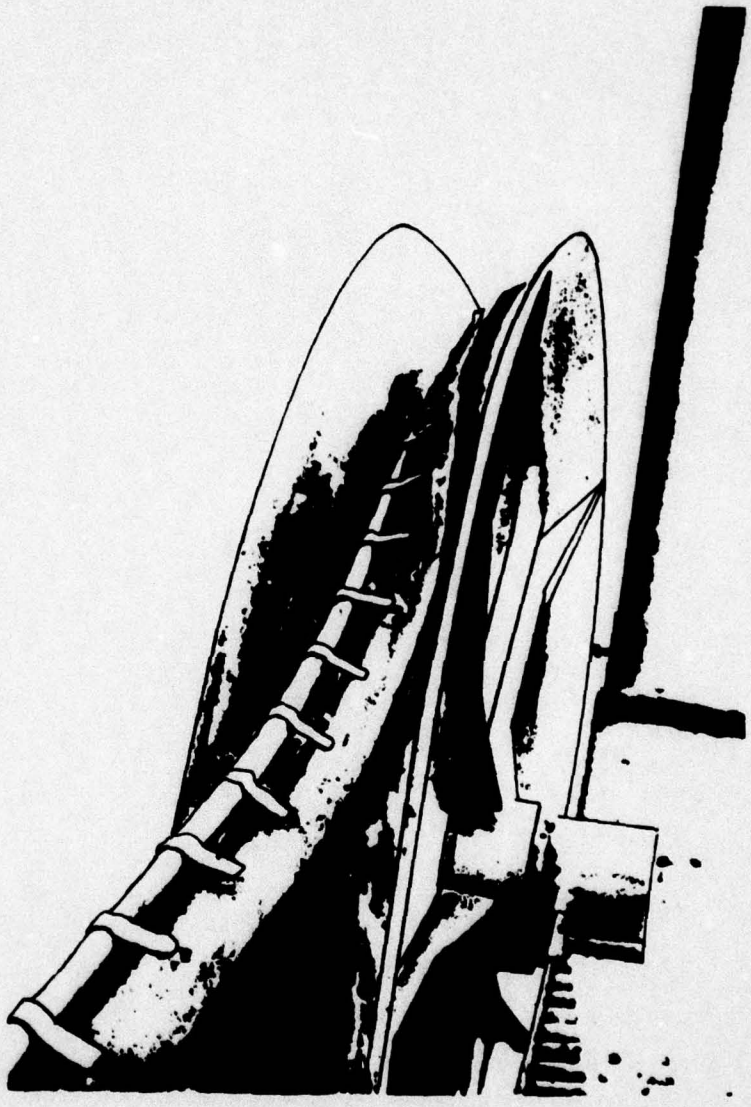


**Figure 2-1. Continuous, Trailing Fairing Handled Over a Sheave, Illustrating Need for Positive Fairleading Control. (from Ref. 7).**

segment. Long cables are not usually faired continuously because of the differential accumulation of length as the fairing is coiled for storage, and fairing stretching when fully deployed. Figure 2-2 shows the effects of stretch accumulation in a continuous trailing fairing. Figure-eight storage to avoid such accumuation has been developed to date. Integrated, continuous fairings are being developed for applications where the required performance envelop is very demanding, but the handling characteristics of these are not yet known.

### 2.1.3 Relative Efficiency

The principal function of a cable fairing is to reduce the hydronamic drag force on the cable. As discussed earlier, this is achieved by conceiling the cylindrical cable section within a symmetric hydrofoil. Various pragmatic compromises with this ideal have been introduced, leading to the many types of fairing available today. Each of these compromises has an effect on the ability of the fairing to reduce drag. Table 2-2 is a presentation of the relative efficiency of the various fairing types according to drag coefficient (based on faired cable thickness at a Reynold's number of about $10^5$).

The integrated fairings under development for hydrofoil VDS system application are clearly the most efficient, have drag coefficients less than 0.1. Segmented/continuous, fully enclosing fairings are second. When carefully designed with emphasis on low drag, they rival the integrated fairings. But when their handling and storage problems are considered in the design as well, the drag coefficient increases by a factor of about 2.

Sectional, fully enclosing fairings are the third-most efficient type of cable fairing, with drag coefficients that range from 0.2 to 0.35. All the trailing fairings may be lumped together in fourth place. The roughness of the mounting clips, and especially the junction between the cable and its fairing contribute heavily to the drag.



**Figure 2-2. Stretch Accumulation of Inadequately Supported Continuous, Trailing Fairing (from Ref. 4).**

Table 2-2

# SUMMARY OF FAIRED TOWLINES

| STYLE OF FAIRING | EXAMPLE OF APPLICATION | CROSS SECTION CHARACTERISTICS | TYPICAL $C_R$ @ $Rn \approx 10^5$ |
|---|---|---|---|
| **FREE SWIVELING** | | | |
| SEGMENTED/CONTINUOUS | AQS-2 AIR SHIP SONAR | DTMB #7 | 0.10-0.17 |
| FULLY ENCLOSING | PROJECT GENERAL PARAVANES | DTMB #7 | 0.10-0.25 |
| TRAILING | CHAN | TF-84 | 0.30-0.35 |
| | SQA -10 (VDS) (ORIGINAL) | MOD. TMB #7 | 0.25-0.30 |
| **SECTIONAL** | | | |
| FULLY ENCLOSING | SQA-10, | DTMB #7 | 0.20- 0.25 |
| | SQA-13 | MOD. TMB #7 | 0.30-0.35 |
| TRAILING | EXPERIMENTAL RUBBER MEANITE (GALLOPING TOWLINE) | MOD TMB #7 FABRICATED FROM RUBBER & STEEL COMPOSITE | 0.25-0.30 |
| **INTEGRAL** | | | |
| SEGMENTED/CONTINUOUS | NUWC/HYDROFOIL VDS | NACA 63A022 | 0.05-0.10 |
| ARTICULATED | NUWC/HYDROFIOL VDS | NACA 0020 | 0.07-0.10 |

In developing any faired-cable system it is natural to give priority in the same order as hydrodynamic efficiency. Consider first integrated fairing, but if cost, manufacturing, or other factors preclude its application, then consider segmented or continuous, fully enclosing fairings, and so on. The results of such a process are described in Section 5.1.

## 2.2 HANDLING SYSTEMS

### 2.2.1 Handling Function

Cable handling systems are comprised of three principle functional subsystems: fairleading, winching, and storing. In addition, there are auxiliary subsystems such as power, control, and information.

Fundamentally, faired cable is distinguished from bare in that it is bilaterally instead of radially symmetric. Fairleading a bare cable is therefore basically a one-dimensional problem: proper orientation of the cable axis through the various handing steps. The degradation of symmetry imposed by adding a fairing requires that the fairleading system control the cable in two dimensions, namely, axial and radial orientation.

Faired cable is distinguised from bane cable secondarily by the fact that most, if not all, fairings are fragile elements in comparison to the cable itself. This factor, as much as shape, restricts faired cables to a single lay on a drum or requires that hauler tracks be designed to avoid stressing the fairing. Yet a momentary malfunction of the fairleading through the tracks may destroy the fairing.

Third, cable fairings are useful only in so far as, by their shape, they improve the hydrodynamic properties of the bare cable. Any handling operation that alters the shape of the fairing elements is detrimental or destructive of the fairing function.

Of the three primary handling functions, most study, design, and development work has been concerned with winching and storage. Fairleading

mechanisims for faired cables have generally been patterned according to bare-cable fairleader design. This failure to obtain and maintain control over the radial orientation of the fairing has led to many of the handling problems in existing systems.

### 2.2.2 Constraints on Submarine Machinery

Submarine machinery may be broadly classified into two groups, depending on whether or not it is located within the pressure hull. It is unlikely that an acceptable faired cable handling system operating within the pressure hull can be found.* The constraints on the handling system are those that apply to any exo-hull machine. They may be summarized in a simple statement: the winch will be inaccessibly located in a sea-water environment on a military submarine.

A. Inacessibly located

Inaccessible machinery must be, in a word, reliable. Further distinctions can be made: the machine must be free from breakdowns, i.e., part failure. It must also be free from shutdowns, i.e., adjustment failure. Finally, the machine operation must be fully automatic.

B. Seawater environment

Machinery operating in seawater must not be susceptible to biological fouling or electrolytic corrosion. It must also be pressure proof. Electrical equipment must be heavily insulated and water proofed. Lubriaction must be highly water resistant.

C. Military submarine

Power consumption must be compatible with submarine sources, both in type and quantity. Hull penetrations must be kept to a

*The AIGS system is being developed to handle a circular, jacketed cable through the pressure hull. Adaptation of the AIGS system to store a faired portion of the cable withint the sheath outside the pressure hull is likely to be the closest approach to a faired-cable handling sytstem operating within the pressure hull.

minimum. The machinery profile and bulk must be kept small and faired to limit drag. Weight is critical to stability. Noise must be minimized; not only operating noise, but also cavitation and flow acoustic radiation.

It is readily seen that a suitable design for exo-hull submarine machinery usually will not be easily attained. Even when an excellent concept is found to accomplish the function, the details of design execution and the construction itself must also be followed through with the highest standards.

## Section 3

## FAIRLEADING DEVICES

Fairleading devices may be classified in two groups. First are the rotating devices, such as sheaves and rollers. The second group includes bellmouths and guide-tubes in which the fairing slides. Fairleading systems are designed around two concepts. Point-to-point fairleaders control the orientation of the fairing at discrete locations and assume the fairing has sufficient rigidity, especailly in torsion, to maintain that orientation to the next fairleading device. Nearly all faired cable handling systems to date have employed this method. Continuous fairleaders establish the orientation of the fairing at one end of the handling system and guide it through the entire handling and storage process.

### 3.1 EXISTING FAIRLEADING SYSTEMS

In these paragraphs, the various fairleading systems that have been used to handle faired cable are discussed and classified according to the type of fairing.

#### 3.1.1 Sectional Fairing

The SQA systems (Ref.'s 10, 11) use point-to-point fairleading. Initial fairing orientation is assumed to be determined by hydrodynamic forces in the water, and maintained by the torsional rigidity of the fairing up to the overboarding sheave. Failure of a section requires manual reorientation at the sheave to prevent crushing the misaligned sections (Figure 3-1). Fairing is stored, nose-in, on a drum. This orientation is assured by a pair of rollers just off the drum surface on the SQA-10. The same mechanical design defects that plague the rest of the 10 mechanisms disrupt the functions of these rollers to the extent that some systems have had the rollers removed.

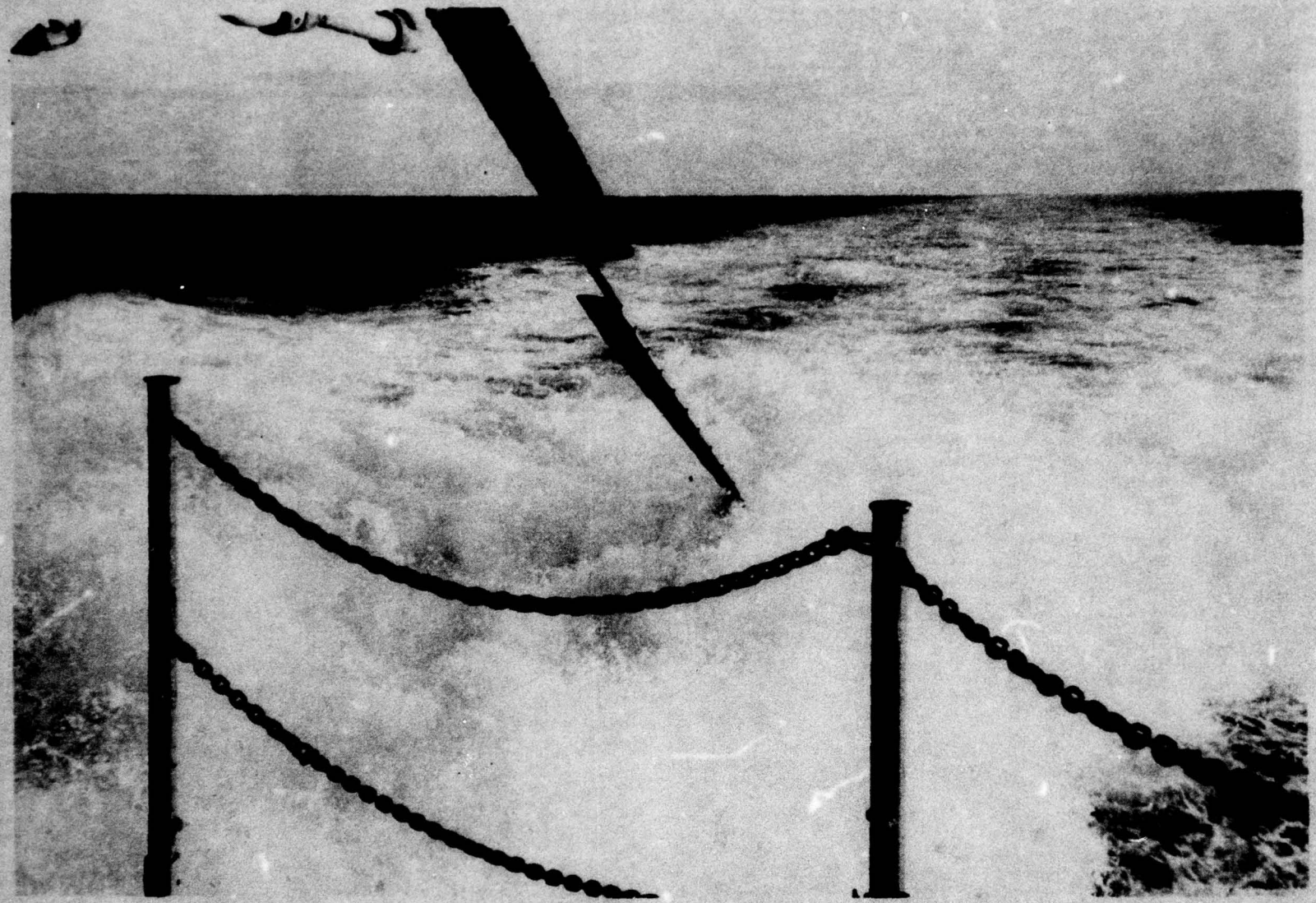


**Figure 3-1. Hauling Cable with Missing Fairing Section, Illustrating Need for Random Orientation Bellmouth (Ref. 11).**

### 3.1.2 Segmented/Continuous Fairing

Pneumodynamics used a point-to-point fairleading system of rollers and sheaves in their at-sea testing of a cable hauler with trailing, segmented fairing. Although the handling system was a purely experimental set-up, several points were noticed.

1. The trailing, segmented fairing showed little torsional rigidity, so that unguided spans of more than a few feet tended to lay over and hang with the fairing underneath the cable.

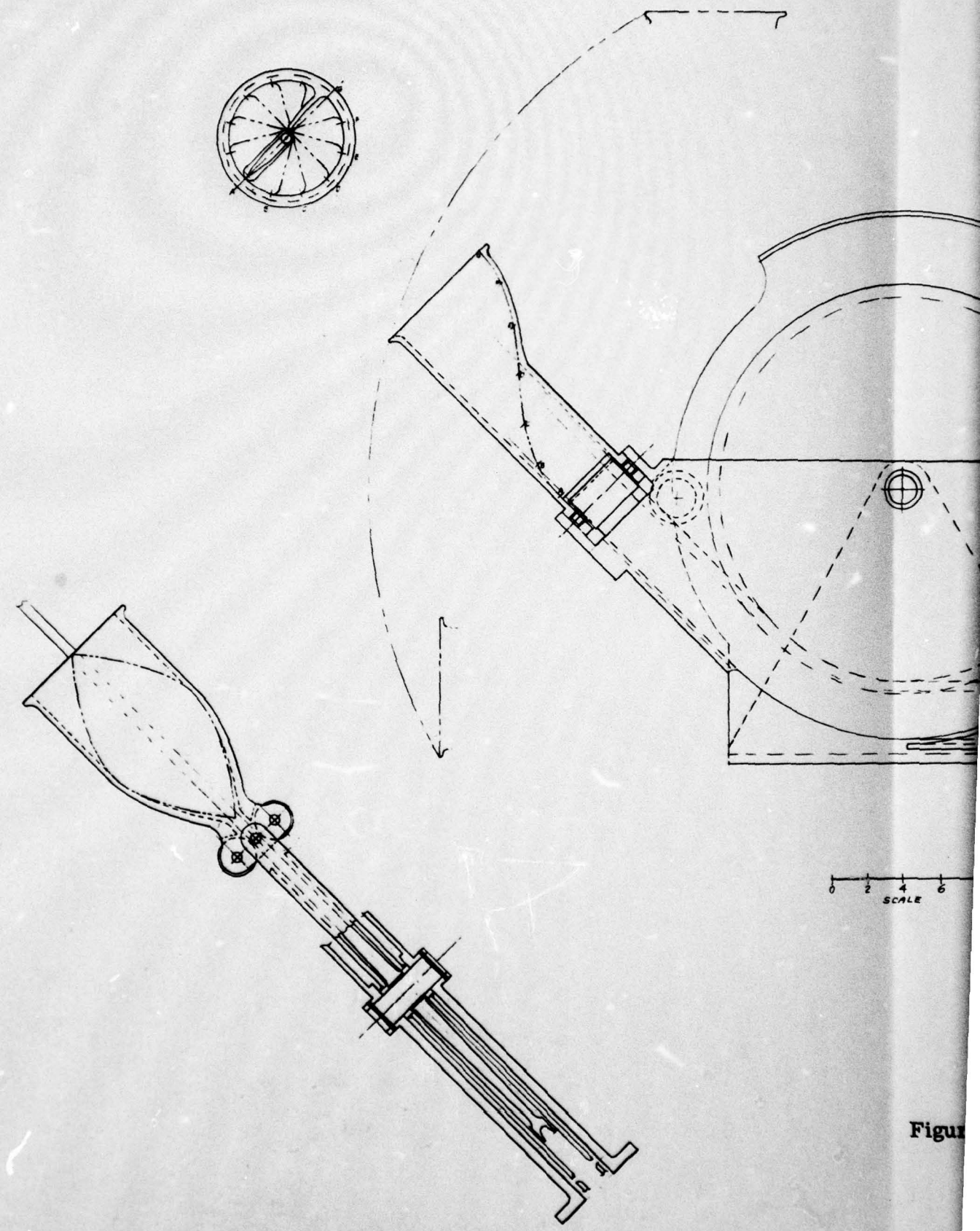
2. The fairing clips tended to catch on any nearby protrusions, resulting in bent clips and torn fairing.

3. The hauler tracks slipped on the cable before slipping on the fairing. Slight cable slippage produced large deformations of the fairing between the tracks.

The CHAN towing winch is the only faired cable handling system that uses what is essentially continuous farileading. Operational experience is too limited for conclusive judgments regarding this system, but it is expected to perform satisfactorily. One possible source of malfunction is the lack of any mechanisms that achieve positive control of the fairing orientation as it approaches the towpoint sheave. Under most conditions hydrodynamic forces will maintain the desired fairing alignment. Inasmuch as these forces are strongly velocity dependent and the possibilities for overcoming them so numerous, it seems unwise to rely on an assumed orientation.

## 3.2 PROPOSED FAIRLEADING SYSTEM

Figures 3-2 and 3-3 are sketches of two possible faired cable bellmouth designs. One is a modification of a conventional bellmouth, having a curved "splitter" to deflect a reversed fairing to one side or the other where the bellmouth walls constrain it to complete its reorientation. The splitter is curved so that if the fairing should happen to "hang up" on




Figur

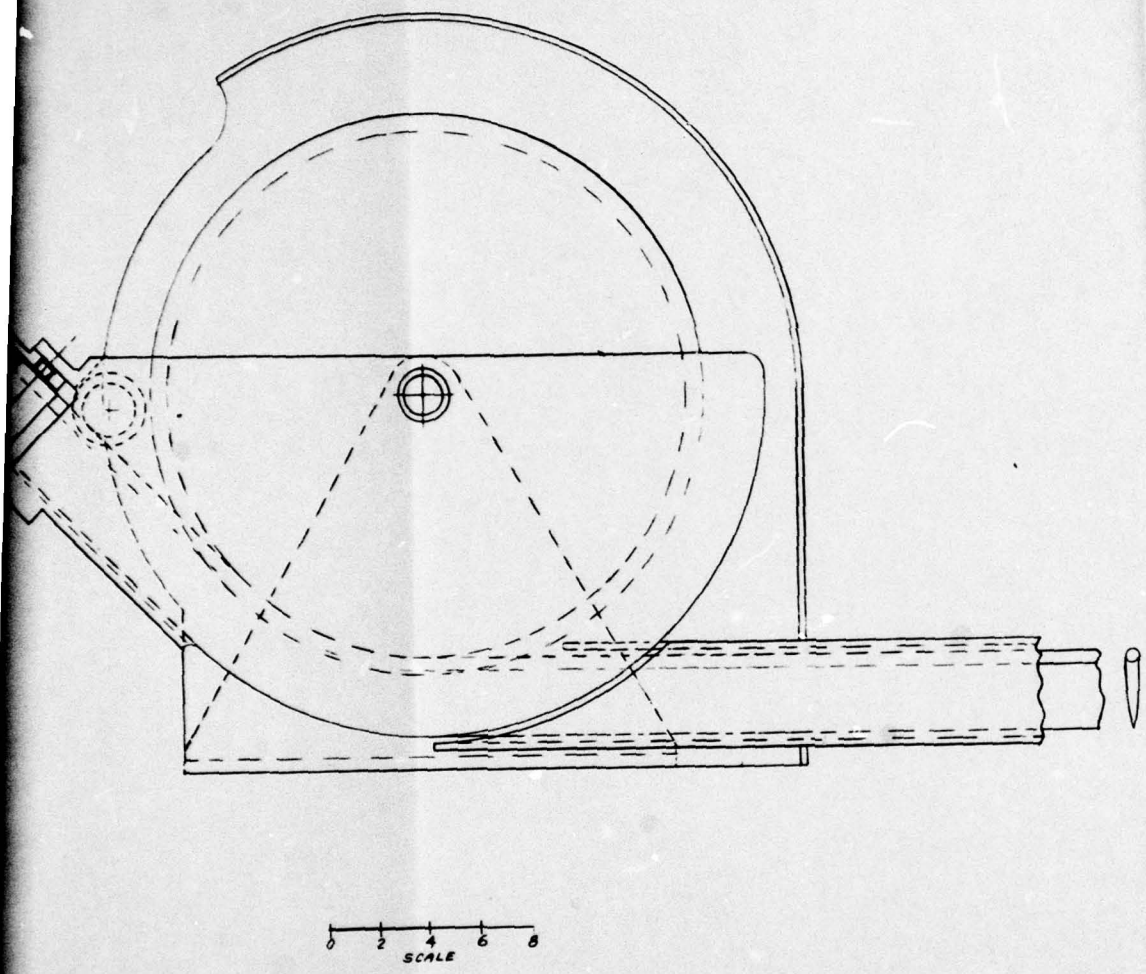
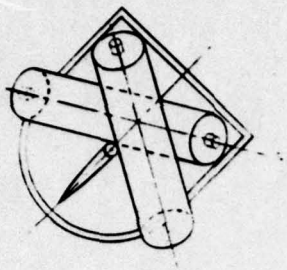
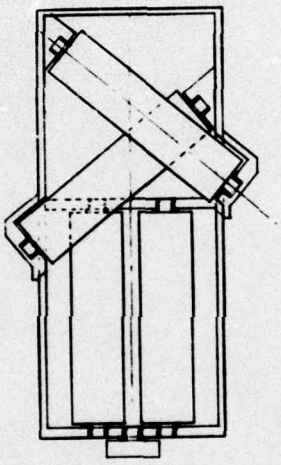
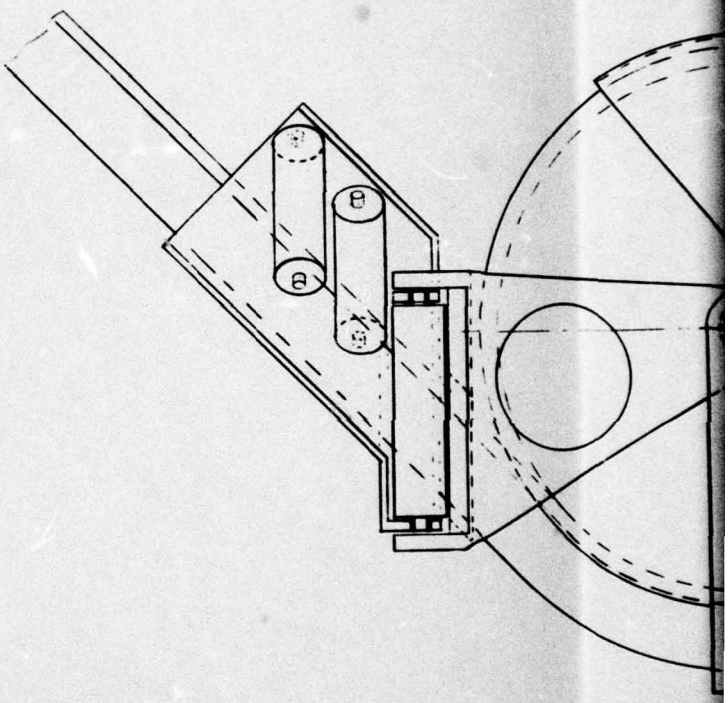
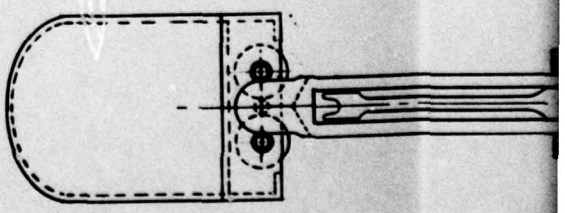


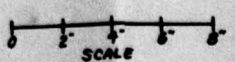

**Figure 3-2. Bellmouth to Align Cable Fairing with Sheave Groves and Fairleader Walls.**










0
1'
2'
3'
4'
SCALE

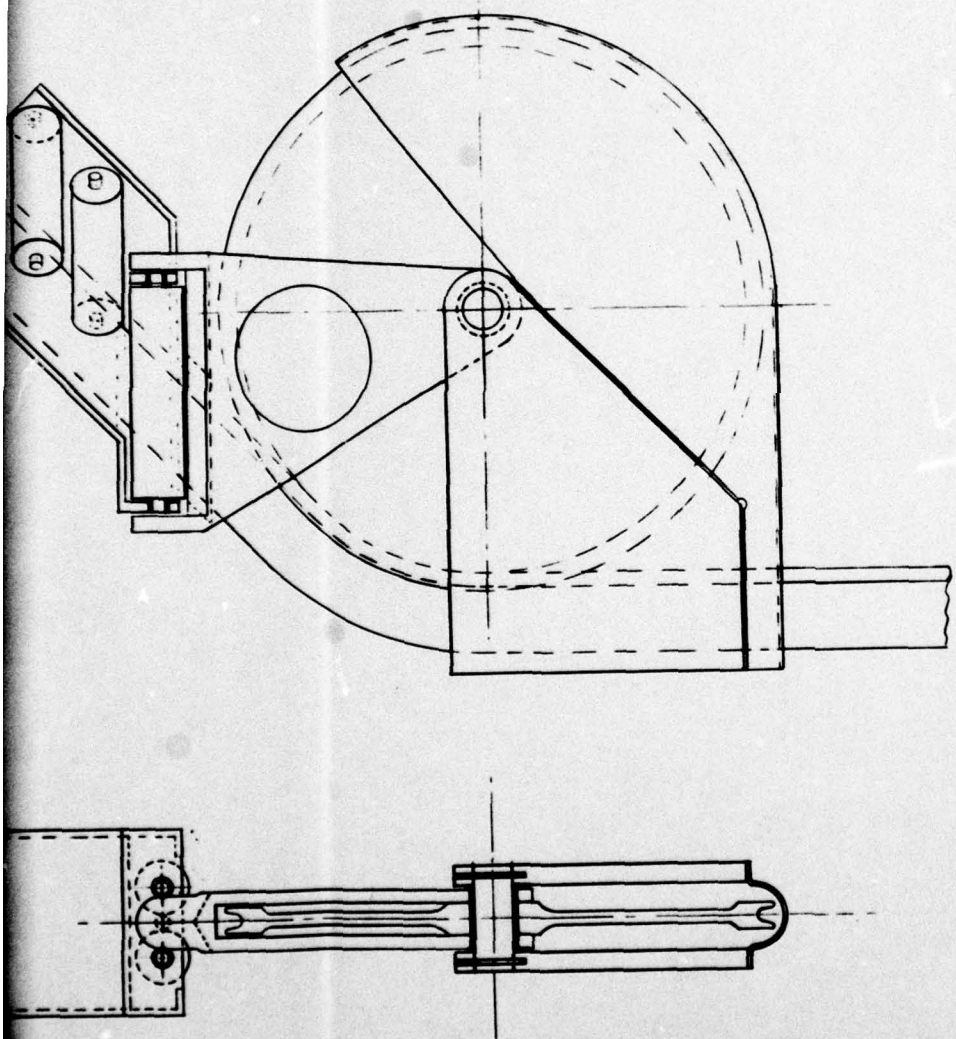


Figure 3-3. Roller-Bellmouth to Align Cable Fairing with Handling System Components.

the splitter, it will be carried to one side anyway, developing a moment which will cause it to snap free. The bellmouth is pivoted at the throat to permit it to track the lateral shifting of the towline in turns. Rollers are mounted on either side of the pivot in order to guide the fairing if the winch is operated while in a turn. Finally, the entire bellmouth frame is pivoted about the towpoint sheave axle so that the bellmouth tracks any change in towing angle.

The faired cable bellmouth sketched in Figure 3-3 avoids dragging the fairing over the bellmouth walls by replacing the walls with skewed rollers. If the fairing is within about 80 degrees of being exactly centered in the desired orientation, it passes without obstruction to the throat. There a pair of parallel rollers are mounted in a vertical plane but inclined to the cable axis. As the faired cable passes through, the fairing is smoothly rolled into the desired vertical plane.

If the fairing deflection exceeds about 80 degrees in either direction, it brushes one of two rollers skewed to either side and is rolled around to a point where the throat rollers can complete the reorientation.

These are but two of many possible bellmouth designs for faired cable. Inasmuch as experimental and operational experience are virtually nonexistent, HRC recommends that this be one of the specific areas for development. The necessity is underscored by the operational experience that is available when a bellmouth is not used: the SQA-10 system for example, requires a sailor to beat misaligned fairing sections into place with a 5 pound sledge.

## Section 4
## WINCHING/STORAGE DEVICES

The winching sub-system function is to control the tension in the towline while retrieving or deploying the towed system. Winching devices may be broadly divided between those that retrieve or deploy the towline in discrete increments in a "hand-over-hand" fashion, and those that maintain an essentially continuous motion. In general, the latter are preferred, the former devices being employed only when required by external circumstances. Hand-over-hand systems are used, for example, when portions of the towed system are to be stored linearly in racks because of their relative rigidity. Inasmuch as hand-over-hand systems presently used require manual resetting of the hauling linkage on the next segment, an impossible operation on a submarine, only continuous winching devices were considered in this study.

Continuous-motion devices can themselves be further divided according to those that do or do not apply tension directly to the cable. Of those that do, the conventional winch drum is by far the most common. Specific winching concepts are discussed in the paragraphs that follow.

### 4.1 EXISTING WINCHING SYSTEMS

In these paragraphs, the winching/storage systems that have been used with faired cable are discussed, and classified according to type. They range from experimental to fully operational (though not necessarily successful) towing systems. Table 4-1 is a summary of the advantages and disadvantages of the various types of winches, based on experience with these systems.

#### 4.1.1 Direct Tension Winches

Direct tension, conventional drum winches have been used exclusively except in trials of other concepts, apparently because they work well with bare cable. However, many of the so-called fairing

Table 4-1. Winch Systems

| Application | | Single Drum | Double Drum | Cable Hauler |
|---|---|---|---|---|
| General | Advantages | 1. Highly developed technology<br>2. Potentially most compact*<br>3. Direct tension application<br><br>*Except for very long cable lengths. | 1. Potentially compact faired cable storage.<br>2. Storage may be at another location and have high capacity. | 1. Potentially compact faired cable storage.<br>2. Storage may be at another location and have high capacity. |
| | Disadvantages | 1. Single lay drum capacity. | 1. Indirect tension application.<br>2. One side friction contact.<br>3. Potentially most bulky. | 1. Indirect tension application.<br>2. Fairing damage by track pressure or slippage<br>3. Potentially most complex. |
| Sectional Fairings | Advantages | See General Advantages. | See General Advantages. | See General Advantages. |
| | Disadvantages | 1. Nose pieces may be damaged by bending on drum. | 1. Rigid sections may bend or fracture on ungrooved drums.<br>2. Extra bending cycles under tension.<br>3. Extra sliding into grooved drums. | See General Disadvantages. |
| Segmented & Continuous | Advantages | See General Advantages. | 1. Slack Storage. | 1. Slack Storage. |
| | Disadvantages | Radius Differential and Storage Under Tension:<br>1. Damage to fairing at swaged rings or equivalent.<br>2. Trailing edge stretched: kiting<br>3. Increasing drum radius may negate compactness. | See General Disadvantages. | See General Disadvantages. |

handling problems are related directly to this traditionalism. This is not to say conventional drum winches are inferior to other concepts when hauling faired cable; only that the winch must be designed with the faired cable properties given full consideration.

4.1.1.1 SQA-Series VDS Sonar Hoists, (Ref's 10-11)

The SQA-10 is the only faired cable handling system that has approached operational fleet application. A critique of the system will differ from those of other, more nebulous, proposed handling systems. The advantages tend to be reduced to the simple statement: it works, sometimes. The list of problems is long, but specific.

Segmented, trailing fairing was used briefly on early models of the SQA-10 (Figure 4-1). The tension developed in the fairing under tow was not properly transferred to the cable. The fairing deformed and failed, destroying its hydrodynamic usefulness (Figure 2-2). Solutions to this problem were not successfully applied on a large scale. All models since have employed sectional, enclosing fairings (Figure 4-2).

The SQA systems are designed to handle sectional, fully enclosing fairing on large diameter cable. A single lay is taken on a padded, deep-grooved drum. The SQA-13 drum and driving machinery are mounted on a rotating turret with the towline and body led overboard and an articulated linkage is employed to overboard the body.

The principal problem with the SQA-10 hoist was a failure to account for marine environmental effects in certain phases of the design. Salt encrustation on hydraulic cylinder rams quickly abraded the cylinder seals, resulting in widespread leaks. The main level wind sheave was designed to slide on its shaft. Smooth operation was impossible after the shaft became rusty. The main drum drive used roller chain exposed to the salt spray. Corrosion and "setting up" were inevitable. Similarly, conventional truck air brakes were used. The drums rusted.

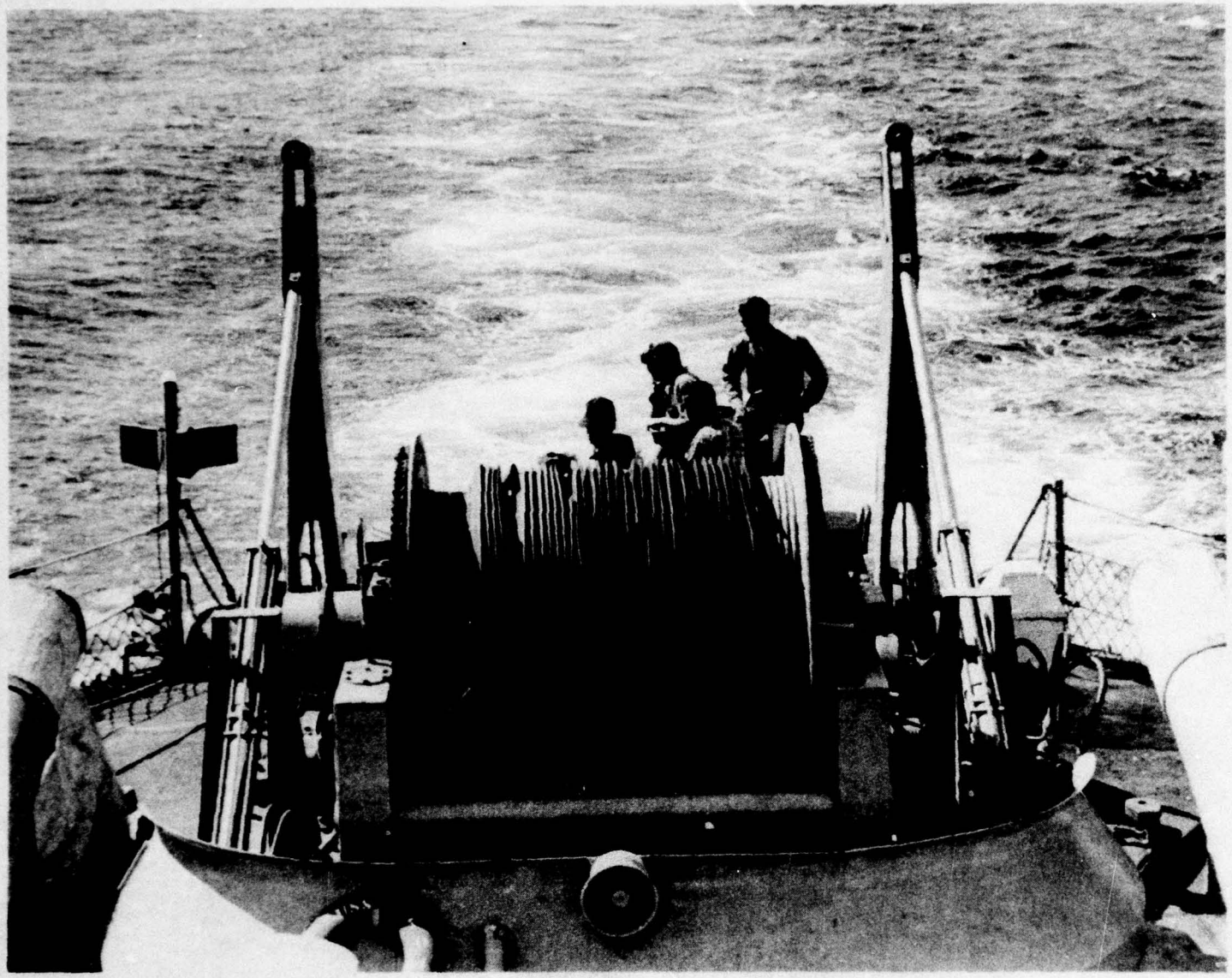


**Figure 4-1. Early SQA-10 System Using A Sequencted Trailing Fairing (Ref. 11)**

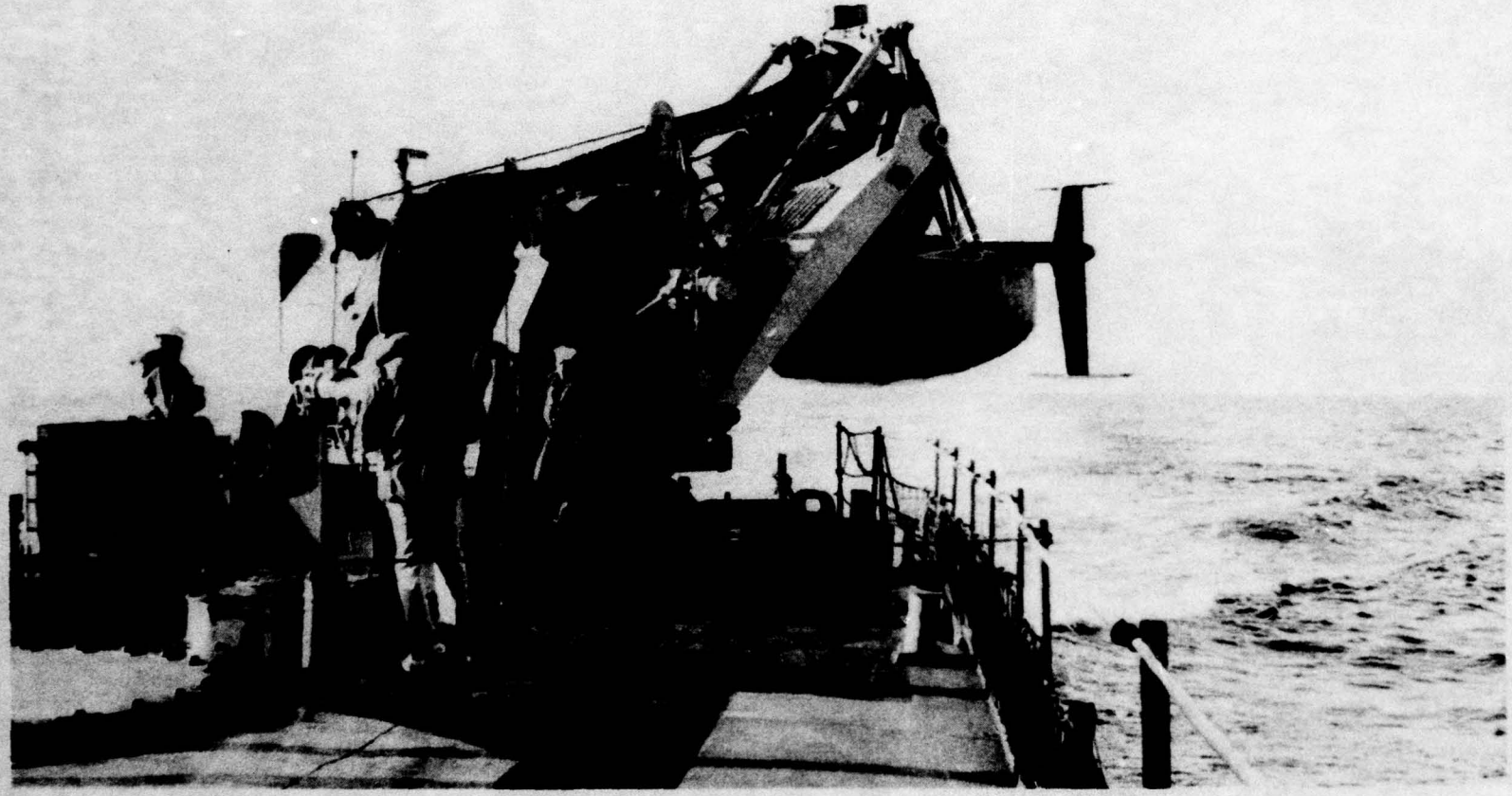


**Figure 4-2. Sectional Enclosing Fairing Handled on SQA-11 VDS Sonar Drum Winch (Ref's. 10, 11)**

The SQA-13 is a more recent VDS system than the SQA-10, and uses more sophisticated design throughout. The towline is fully enclosed by a sectional fairing. The handling system is similar in appearance to the grading machines used in roadside landscaping: The winch system is mounted in a turret on a king post pivot set in the deck, and the cable and body are overboarded by a telescoping boom hydraulically extended from the turret.

The winch system itself is quite conventional. The towline is wrapped on a drum in a single lay, nose inward. The rest of the handling system is devoted, basically, to the problem of placing the body beneath the water from a ship moving on the interface.

Only a few more or less experimental models of this system have been constructed, so that its operating characteristics are not as well known as for the 10. In general, however, the concensus seems to be that the 13 is superior to the 10, particularly in regard to environmental design. Because the SQA-13 has different electronics and transducers, there are related mechanical differences that complicate comparisons of the two handling systems' s suitability. Most notable is the smaller size and weight of the SQA-13 "fish". The influence of this single feature propagates through the entire system design: smaller cable, more compact drum, lighter boom, less power, smaller turret, less deck load, and so on.

4.1.1.2 CHAN (ref. 6)

The CHAN towing system has been assembled in prototype form and tested in the NSRDC circulating water channel. The winch has been taken to sea aboard a destroyer and tested with a "heavy" body. The winch was mounted on deck, and the cable given a half turn as it was fair-led to the overboarding sheave at the fantail.

A deep grooved drum is used to store the towline. Segmented, trailing fairing is attached to the towline by clips. It is one of the few winch systems ever designed specifically to be mounted on a submarine, and the only one designed to handle faired cable. The CHAN winch is shown on Figure 4-3.

Owing to the somewhat unusual history of the development of the CHAN system, the CHAN winch has become obsolescent even before its development was completed. A weight reduction analysis would be necessary if the system were to be used on modern submarines. The mechanical slip ring-less contact system designed for CHAN could be replaced by one of the newer types, which use mercury contacts. The winch motor control panel has been at least partially converted to silicon controlled rectifier circuitry. The rapid development of small submersible craft in the five years since active work on the CHAN winch stopped has provided a technological base from which a significant upgrading of the system is possible.

In spite of the obsolescense of the system, one cannot escape the feeling that this winch is one likely place to begin development of a new submersible faired cable handling system, within the limits of its performance envelope.

#### 4.1.1.3 BOEING (Ref. 3)

A handling system for the Boeing integrated, continuous faired towline is under construction. Our understanding, based on a telecon with the project office at NEL is that no major departures from conventional drum winching are incorporated in the design. The towline is wrapped, nose inward, on a 6-foot diameter drum. The diameter is large in order to reduce residual stresses in the fairing during storage. An helical aluminum strip is welded to the drum to hold the fairing upright and to control the lay placement. The strip height is about 15 percent of the towline chord. The drum and helix are greased, in lieu of teflon, rubber, or other coatings.

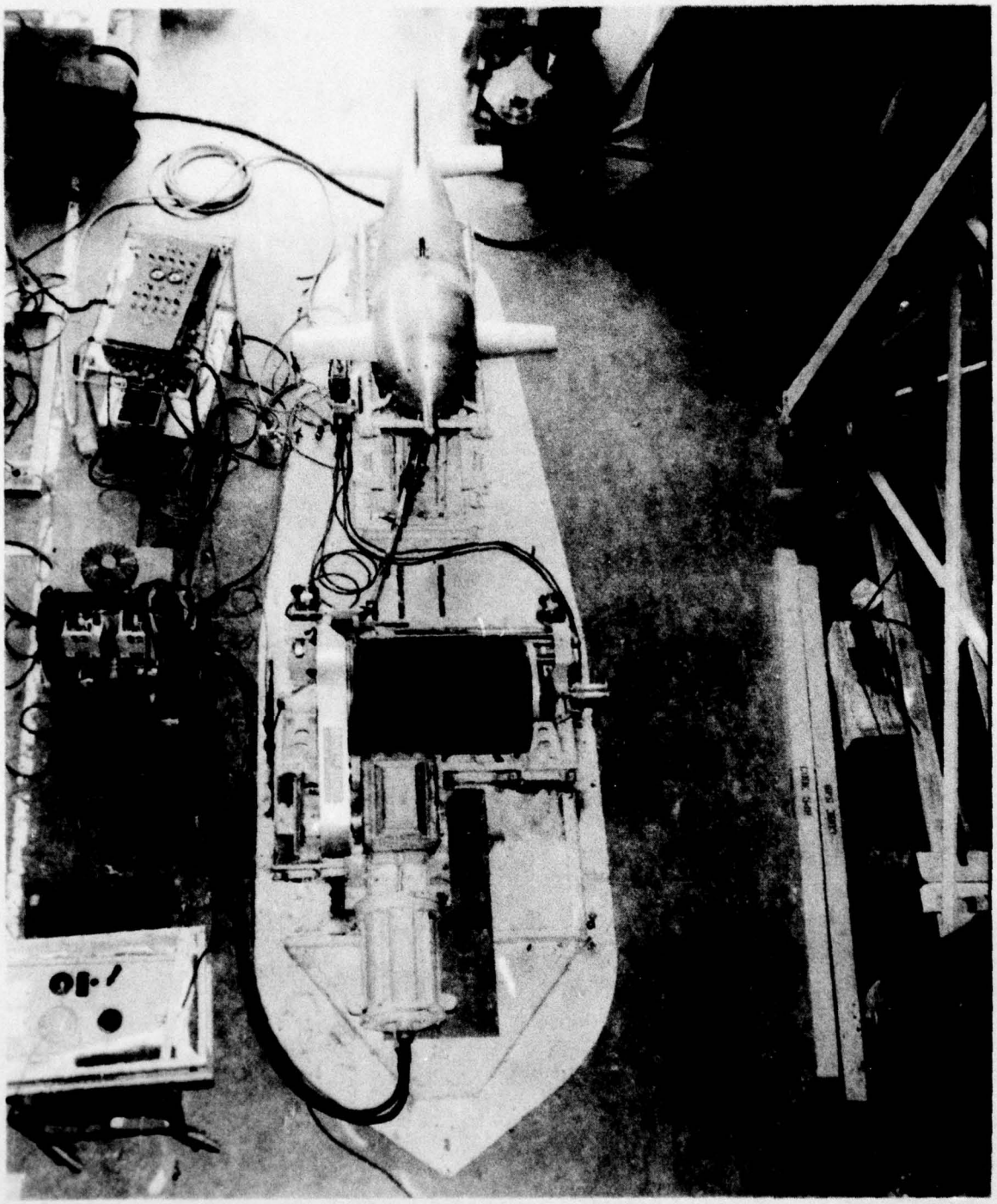


**Figure 4-3. The CHAN Winch (from Ref. 6).**

#### 4.1.1.4 North American Aviation, Inc., (Ref. 12)

A single test length of the integrated, articulated faired towline developed by North American Aviation, Inc., was produced and tow-tested. No handling system per se was used, the towline being deployed by jury-rigged gear and removed similarly at the end of the test. A hexagonal drum winch was intended for the system, had it been developed more fully. Fairleading and level wind mechanism would require careful design in order to keep fleeting angles small and protect the towline knuckles and stabilizers.

#### 4.1.1.5 AQS-Series, Air-Towed Sonar

Segmented, trailing fairing has been handled from air-ships towing sonar gear. Most of the double-armoured towcable is bare; only the lower portion, which is under water when the system is deployed, is faired. Figure 4-4 is a photograph of the storage drum of an AQS-2 system, showing the faired portion freely wound over the evenly wrapped bare cable.

### 4.1.2 Indirect Tension Winches

#### 4.1.2.1 Entwistle/Pneumodynamics Cable Hauler (Ref. 2)

This was not a complete cable handling system, but rather an experimental rig to evaluate the use of cable haulers with faired cable having instrument cannisters interspersed along its length. The cannister diameter was somewhat greater than the cable diameter. A segmented, trailing fairing of soft rubber was used with wire rope of 7/16 to 3/4-inch diameter at tensions between 1000 and 3200 lbs.

Difficulty was encountered in passing the cannisters through the traction heads. This problem was overcome in a later trial by applying pressure to the traction chain through a series of rollers on articulated suspensions rather than through a single track beam.

It was also noted that the traction head pressure deformed the fairing clips slightly. This is probably not a serious problem if the treads

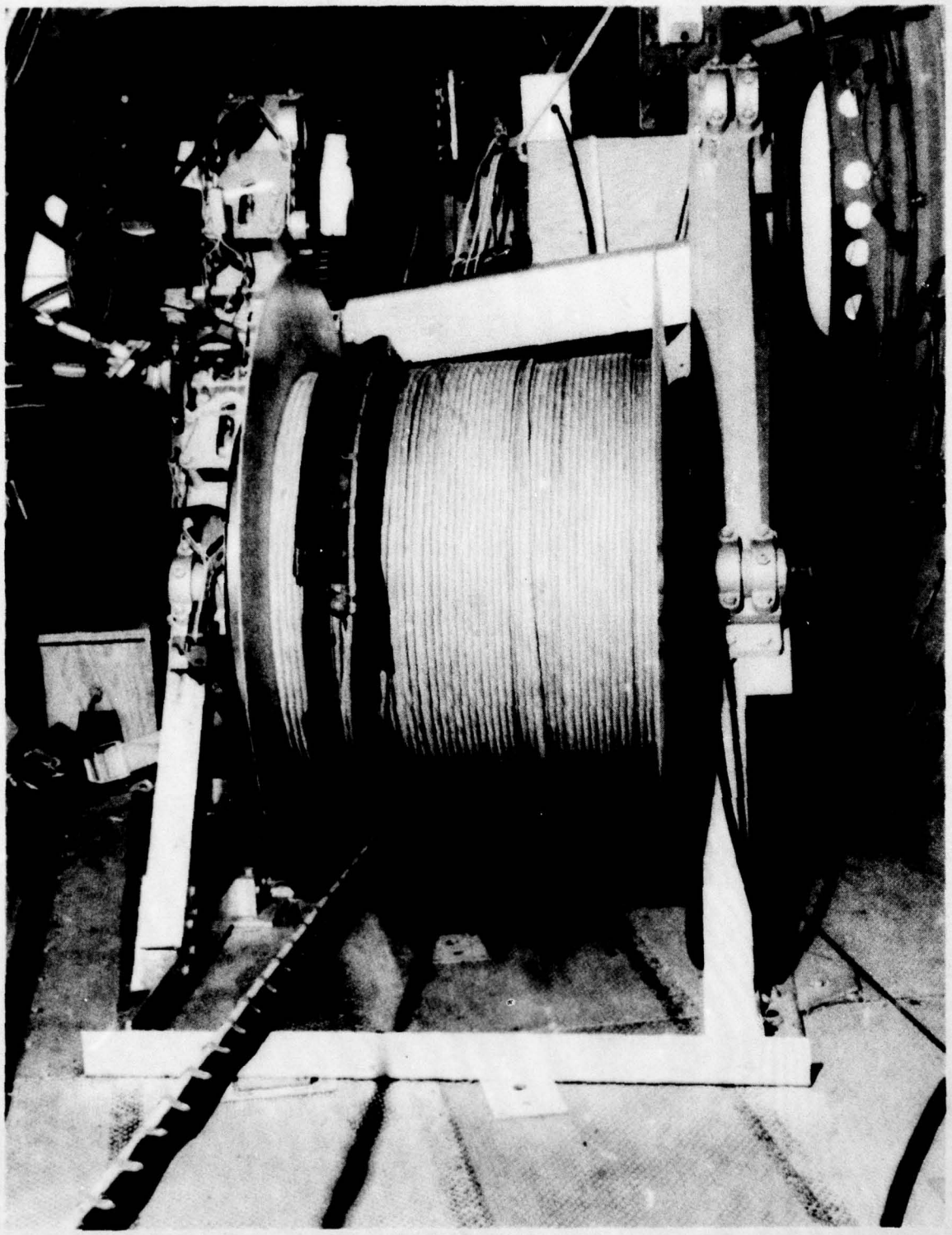


Figure 4-4. Segmented Trailing Fairing Stored by Free-Laying Over Level-Wound Bare Cable (AQS-2 Helicopter-Towed Sonar) PSD 313035.

are properly shaped and the clips designed with the problem in mind. It is not a problem, however, that can be ignored.

The static coefficient of friction calculated from the experiment for dry, bare cable pulled by rubber treads was about 0.4. Wet cable reduced the coefficient by about 25 percent; greasy cable halved it. Dry, faired cable showed a static coefficient of about 0.2, providing the fairing thickness was less than the cable diameter. When the fairing thickness equalled the cable diameter the coefficient was reduced by a factor of two thirds. Inasmuch as the hydrodynamic efficiency of trailing fairing is impaired if the fairing thickness is less than the cable diameter, the tread must be shaped to grip only the cable.

4.1.2.2 Western Gear/Pneumodynamics Cable Hauler (Ref's 7, 14)

This was a complete, though experimental, faired cable handling system in that all three components were present and used at sea: fairleading, winding, and storage. The cable had segmented, trailing fairing.

The fairleading system was point-to-point using rollers to guide the cable around corners. It performed satisfacotrily for the duration of the test, except for the fairleading within the hauler itself, where the fairing clips occasionally caught on the hauler frame and were yanked free by the treads. A tendency for the limp fairing to sag between supports suggests that for long term success the supports must be spaced more closely or fully enclosing guide tubes be added.

The hauler did not perform as well. Fairing clips deformed so that the fairing was no longer free to align with the flow when deployed. Occasional track slippage, albeit experimentally induced, revealed the extent of fairing damage that can be done very quickly by even slight slipping.

The faired cable was stored loosely on a drum. A level wind was used, but strict control of the lay in the drum was not maintained. In this

particular case, a very large drum, with respect to the amount of cable, was available.

#### 4.1.2.3 Naval Research Laboratory (Ref. 9)

N. R. L. has built a winching mechanism (Figure 4-5) that combines the functional operation of a canted twin drum winch into a capstan-like drum. Each of the twin drums is axially slotted (as if removing alternate staves from a barrel) and the two slotted drums are meshed together, maintaining the original cant angle, to provide a lateral shift of the cable with each turn, and a small displacement, so that the cable bears on the slats of only one drum at a time.

In tests using a small scale model a coefficient of friction of 0.15 for steel cable on steel slats was measured. A full-size model was tested with bare cable, and trailing fairing as shown in Figure 4-5.

## 4.2 PROPOSED WINCH/STORAGE SYSTEMS

In this section several possible winch system concepts are presented and evaluated for handling faired cable aboard a submarine.

### 4.2.1 Direct Tension Winches

#### 4.2.1.1 Conventional Drum

Conventional drum winches offer the intangible but very real advantage of a highly developed technology. They are, furthermore, by integrating the storage and winching functions on the drum , potentially the most compact of all cable handling systems. Because the end of the towline is secured directly to the drum, the possibility of catastrophic loss of the towed system is minimized.

In bare cable winch design, the drum diameter is minimized, not only to reduce the torque required to maintain line pull, but also to reduce the ratio of wasted core volume to storage area. The drum

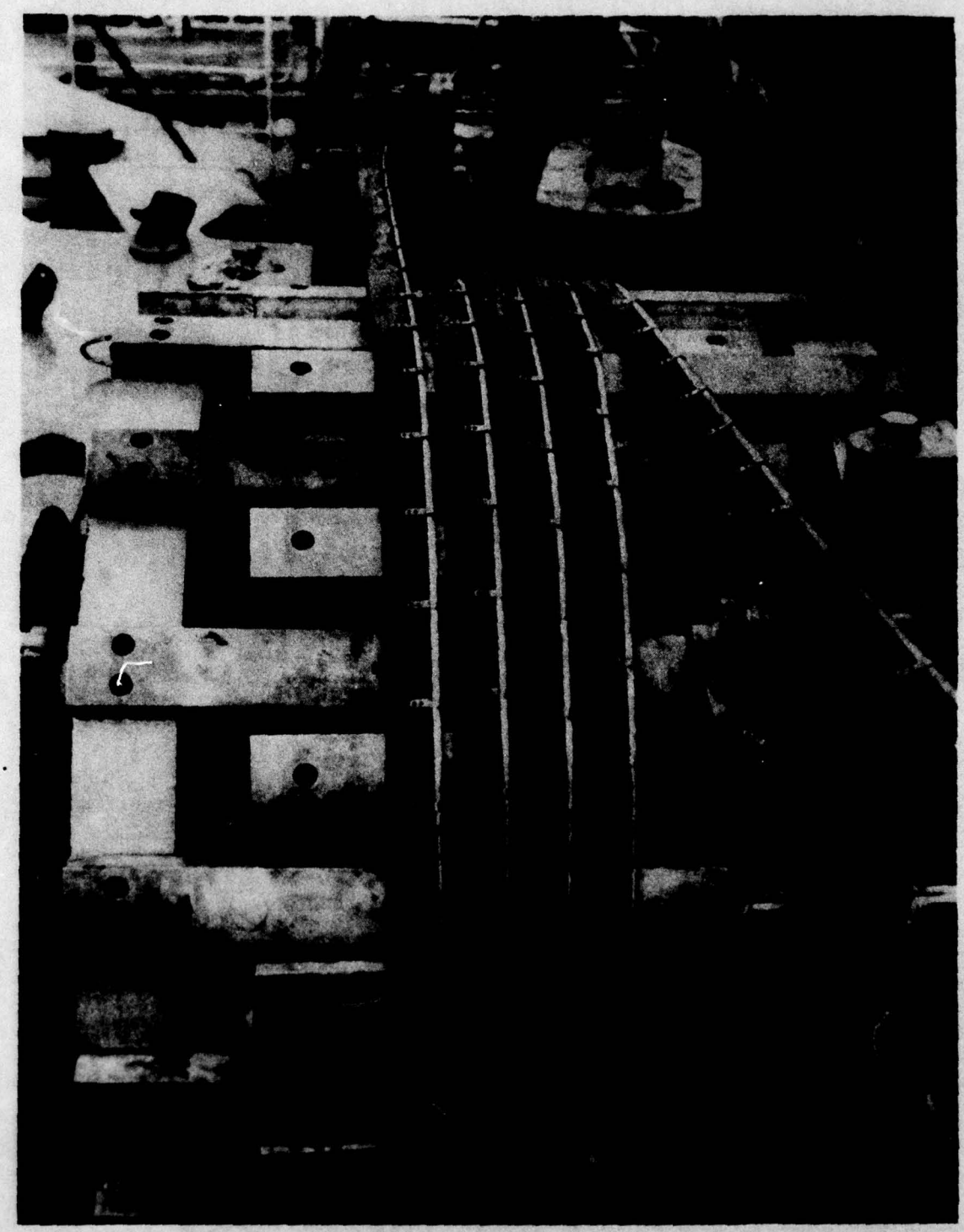


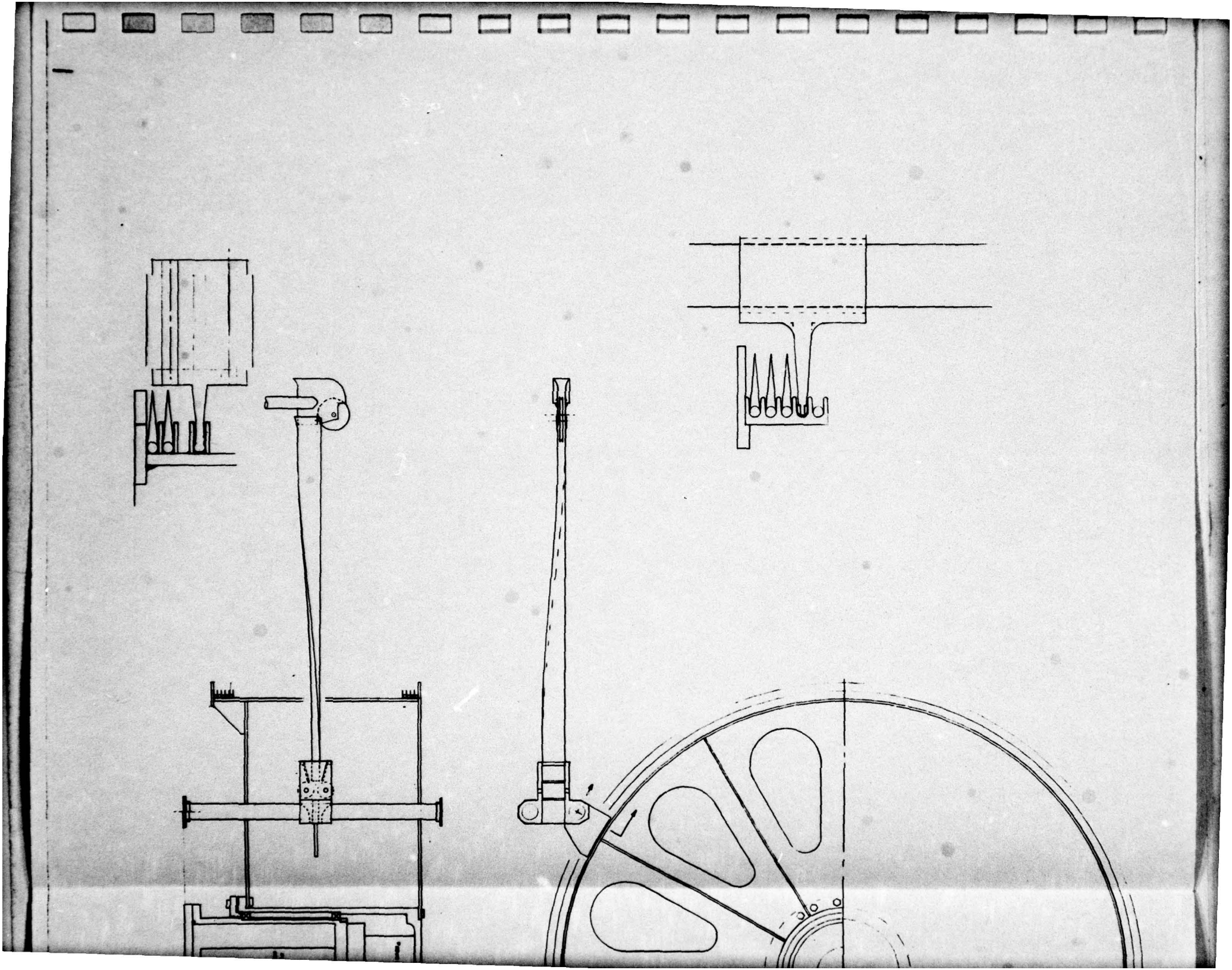
**Figure 4-5. Continuous Trailing Fairing Winched on NRL Slatted Capstan (Ref. 9).**

must only be large enough to prevent excessive fatigue stresses in the cable as it is wrapped around the drum.

When these winches are used with faired cable, however, their fairing storage capacity is restricted to a single lay on the drum because of the shape and fragility of the fairing. Furthermore, the nose pieces of fully enclosing, section fairing may be damaged by bending around the drum if its diameter is improperly matched to the section length. Segmented and continuous fairings may be damaged by the differential accumulation of cable and fairing. The concomitant stretching of the trailing edge of these fairings may not fully relax when the system is deployed, degrading the function of the cable fairing. Although all these disadvantages are relieved by increasing the drum diameter, this is done at a sacrifice of the potential compactness of the handling system. Two preliminary design layouts involving conventional winches for submarine application have been prepared. One assumes a single lay of cable, fare or faired; the other accepts multiple lays of bare cable with the outer most lay faired if desired.

4.2.1.1.1 *Single Lay Concept.* This winch is designed to hold 1000 feet of cable, bare or faired, in a single lay on a large diameter, grooved drum. (Figure 4-6.) The drum is laid on its side in order to present a low profile. The electric drive motor and gear are located within the hub, to regain some of the compactness lost by the large diameter. Because the cable is stored in a single lay, the level wind screw need be only a single helix; the drum grooves themselves are used, eliminating the need for a separate screw and drive line.

The faired cable passes from the drum directly into a guide tube on the level wind which positively controls the fairing orientation as the cable moves to the towpoint sheave in the bottom of the buoy nest. The nest itself is not shown on the figure. A faired cable bellmouth is required at the sheave to achieve control of an arbitrarily oriented fairing; this is not shown here because it is required whatever winch is used.



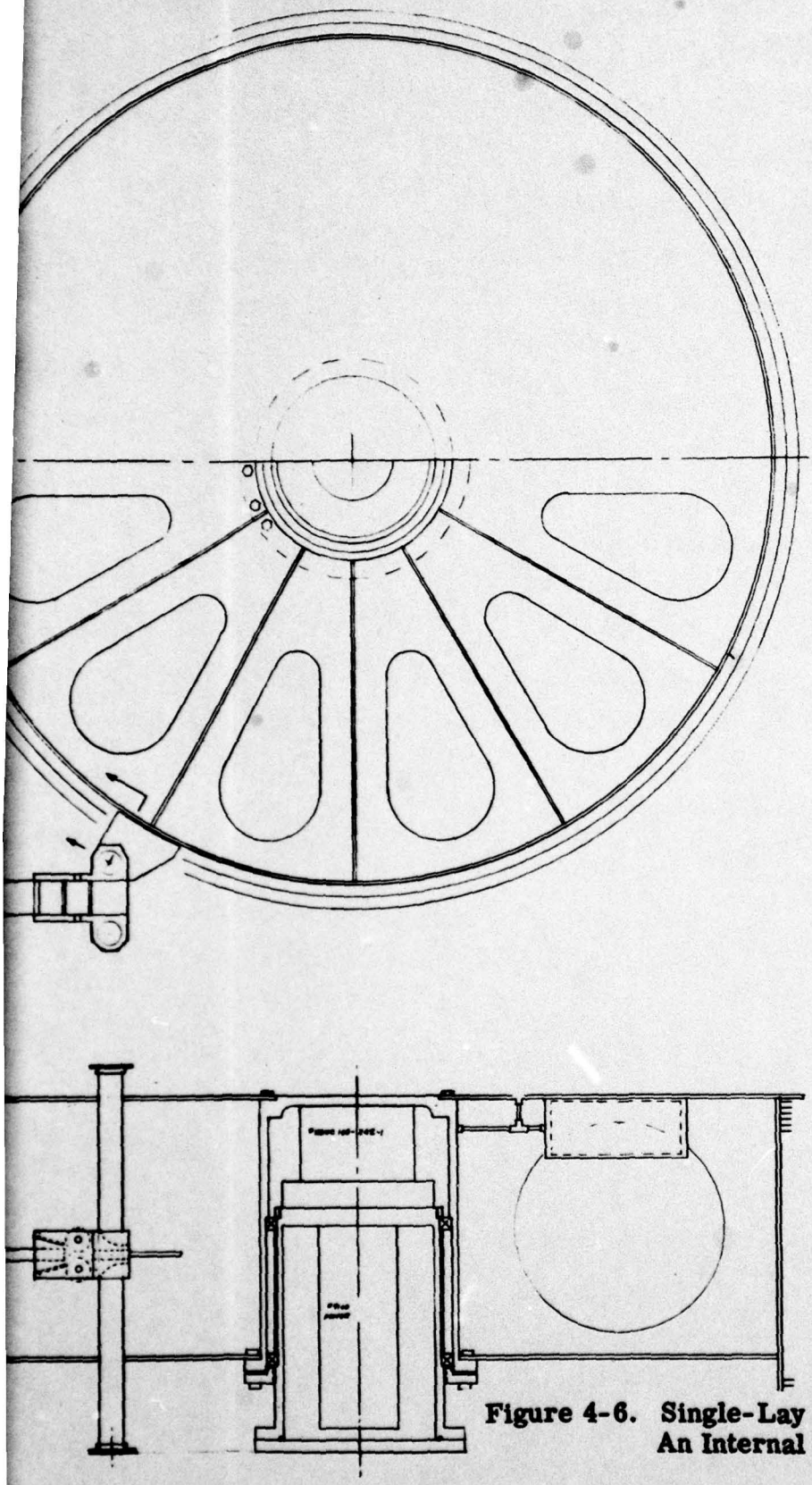


**Figure 4-6. Single-Lay Drum Winch for Bare or Faired Cable, Using An Internal Drive Train.**

#### 4.2.1.1.2 Multiple Lay Concept

The second conventional drum winch layout is designed for holding 1200 feet of bare cable and 300 feet of faired cable. (Figure 4-7.) The drum is electrically driven through two worm speed reducers coupled in series. In this layout the drum axis is horizontal, presenting a tall, thick profile to the flow. With minor modifications the winch could be designed with a vertical drum axis to be flat on the hull.

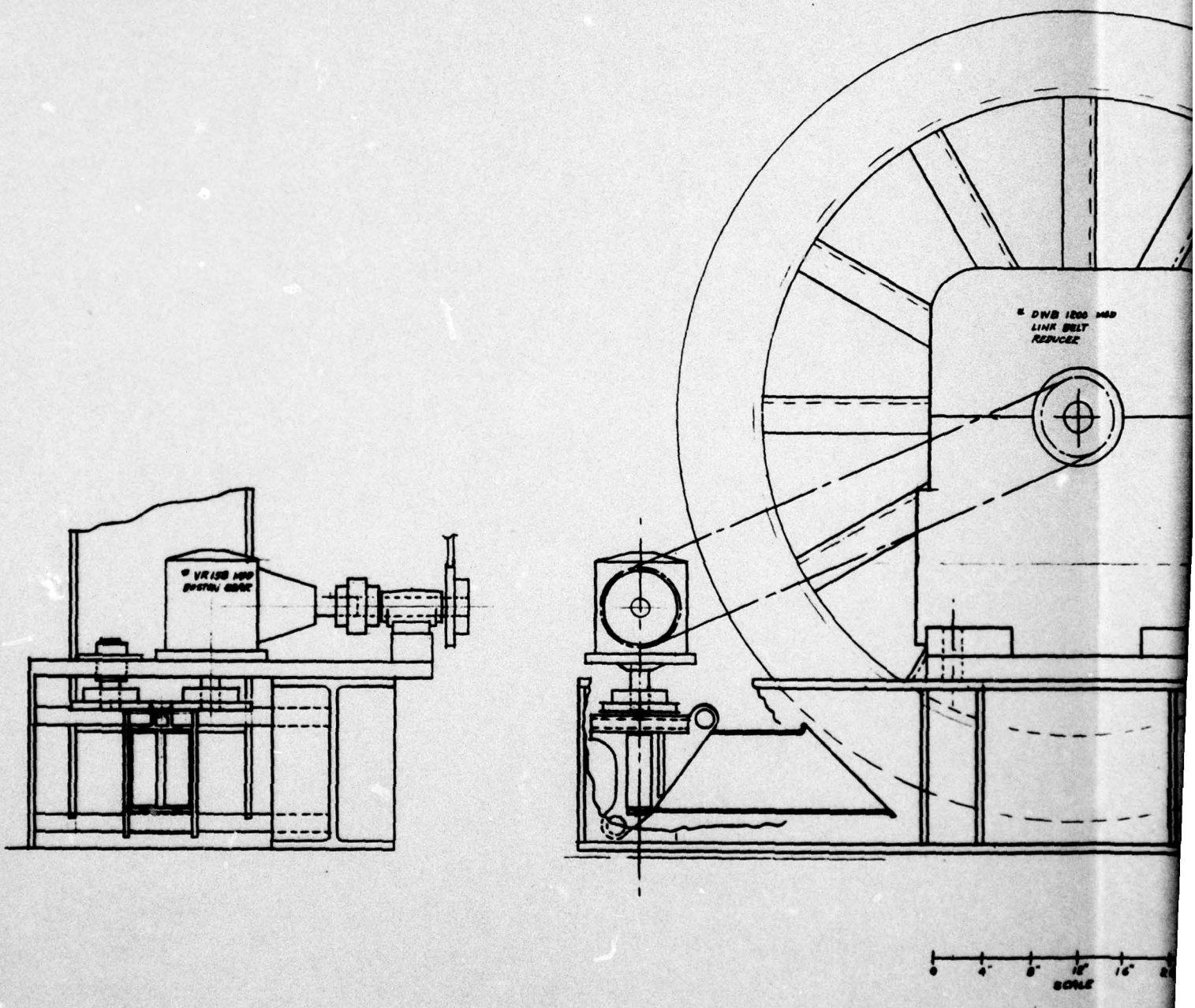
The level wind is chain driven rather than by a double helix screw. The error introduced by harmonic motion as the chain passes around the sprockets at each end of its travel amounts to less than 1 degree tracking error. This is insufficient to produce a winding error in the lays.

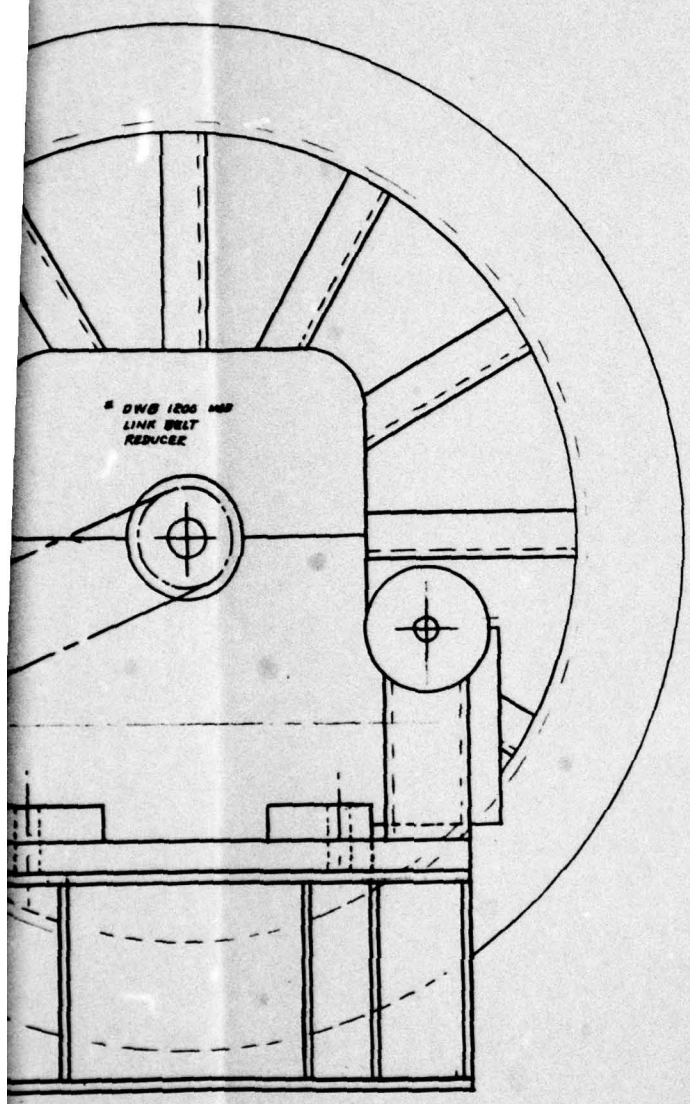
Inasmuch as the fairing winds over the bare cable lays, grooves cannot be used to insure that the fairing orients consistently on the drum. By laying the fairing partly over on its side, its inherent instability when bent under tension will cause it to remain there, wrapping in partially overlapping turns.

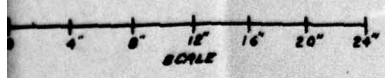
The level wind mechanism, after the faired cable bellmouth, is the second area that HRC recommends for a detailed development and testing investigation. Both double helix and chain-carrier traverse methods involve moving parts that will be directly exposed to the sea, where lubricants can be washed away and corrosion and fouling are always a problem to be reckoned with. Numerous possibilities exist to provide a more effectively sealed system; careful design now will prevent a multitude of problems later.

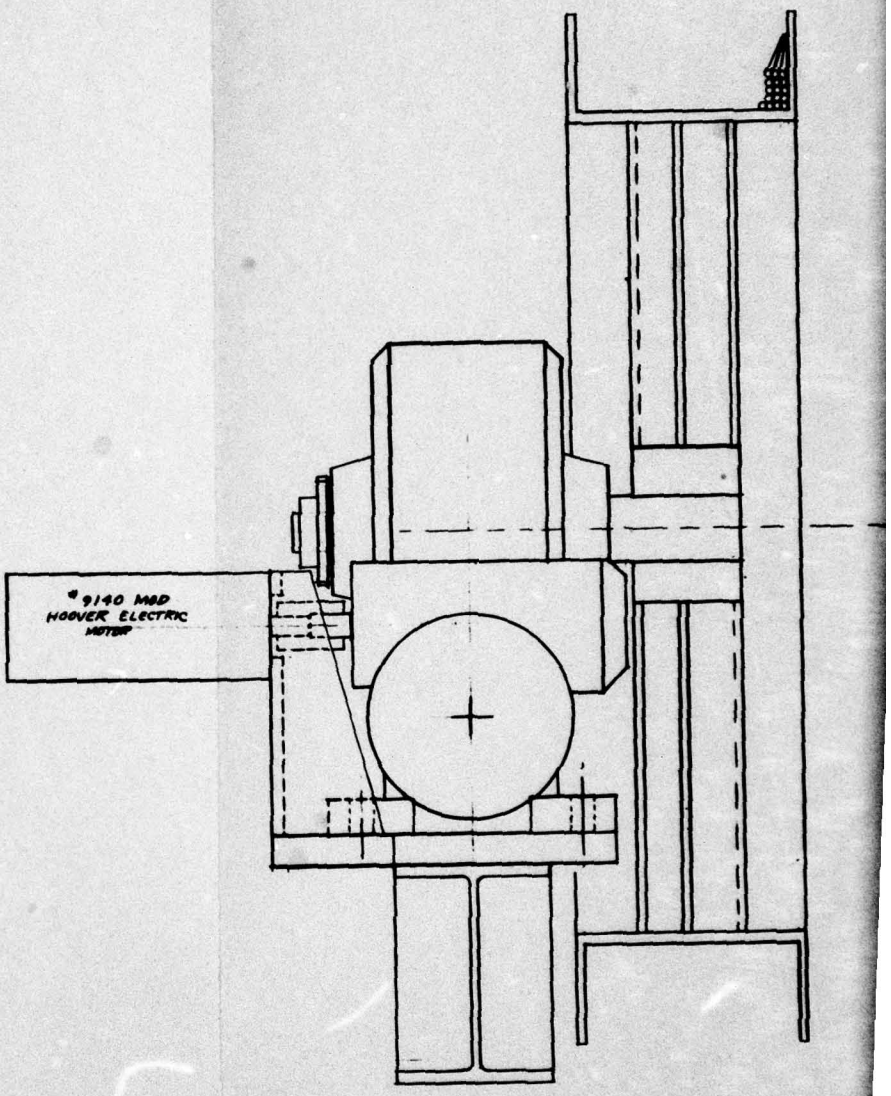
### 4.2.1.2 Sprocket Winches

By swaging appropriate fittings, such as rings, to the tow cable at appropriate intervals, it may be hauled by means of an appropriately shaped sprocket wheel or chain. This allows the storage function to be separated from the winch, so that the cable is not under tension. The


DWB 1200 HSD
LINK BELT
REDUCER
SCALE




300 FT FAIRED CABLE
1200 FT TOW CABLE

**Figure 4-7. Multiple-Lay Drum Winch for Bare or Partially Faired Cable**

concept is limited by the relatively low axial load capacity of swaged rings and the impairment of the fairing function by the large number of rings anticipated to be necessary. These limitations are judged sufficient to disqualify the concept from further consideration at this time. The concept might be useful if it were desired to winch sectional, integrated (articulated) fairing, yet allow slack storage. The knuckles of the fairing would have to be designed to engage the sprockets.

### 4.2.2 Indirect Tension Devices

Indirect tension winches drive the cable solely by contact friction. Conventional drum winches usually transmit most of the tension to the cable by friction with the drum, but they have the "fail-safe" feature that a properly designed cable terminal on the drum can transmit the tension in the absence of drum friction. At a sacrifice of compactness, indirect tension winches separate the storing and winching functions, allowing the storing device to be maximized towards its function. When very great lengths of cable are to be stored, the indirect tension winches become the most compact; perhaps least bulky is more expressive.

Five indirect winch concepts are discussed in the following subparagraphs. The first represents the present state-of-the-art for winches which grip the cable between two surfaces. Subparagraph 4.2.2.2 and 4.2.2.3 are discussions of two attempts to increase the compactness and mechanical simplicity of the cable hauler concept. The final two subparagraphs present the state-of-the-art in winches using multiple turns of cable about the driving drum(s).

#### 4.2.2.1 Cable Haulers

These winching devices were originally developed for handling extreme lengths of relatively rigid cable, such as laying submarine cables. They consist of a pair of linked belts carrying treads looped over a driven sprocket and roller or track assembly (not unlike military tank

tracks) the loops are parallel, tread-to-tread so that the cable can be squeezed between them and drawn along as the tracks are driven.

Cable haulers with properly shaped tracks are expected to work well with continuous, integrated faired towlines, where the track friction on the fairing is transmitted directly to the integrated tension member, and there are no metal clips or nosepieces to catch or be bent. Sectional and segmented, non-integrated faired cables offer moderate success with cable haulers in surface ship applications where they can be monitored, but are subject to damage by excess track pressure or slippage. They are not considered a viable option for submarine applications because fairing damage, slipping, or hang ups cannot be readily detected and remedied. Furthermore, the intricate track roller mechanism poses a serious lubrication, sealing, and fouling design problem.

When design requirements preclude the conventional drum design, such as when very long lengths are to be winched, as in undersea cable laying, or if the cable fairing cannot be stored under tension without an excessively large drum, then traction cable haulers must be considered.

The basic traction equation is

$$T_T = \mu N + T_B \ , \qquad (4\text{-}1)$$

where

$T_T$ is the towline tension,

$\mu$ is the friction coefficient,

$N$ is the total normal force, and

$T_B$ is the back tension to the storage unit.

In general, a traction winch should be designed, if possible, so that it will just slip at the breaking tension of the cable. In this way, unnecessary slippage by moderate impulsive surges is eleiminated, and yet the winch can slip to avoid breaking the cable.

1. Rectilinear Track

For a conventional rectilinear traction winch with one track driven

$$N = n_s L, \tag{4-2}$$

where $n_s$ is the unit normal force and $L$ is the effective track length. If both tracks are powered in synchronism, the total normal force is, of course, twice as large.

The above equation pin-points the third area where design information is lacking: what are the limits on the unit normal force before double armored cable collapses or otherwise fails? Another related question is tensile failure on a sheave or drum. Cable strengths are set based on linear tension testing, and drum and sheave diameters calculated to prevent fatigue failure. But can a double armored cable be wound on such a drum at near breaking tension without collapse? In this report it is assumed the cable would be as likely to collapse as part; i.e., the allowable unit normal force is given by

$$n_s = \frac{2\,T_u}{\lambda_f\, d_c} \tag{4-3}$$

where

$T_u$ is the ultimate cable strength,

$\lambda_f$ is the ratio of drum to cable diameter based on fatigue life, and

$d_c$ is the cable diameter.

Setting

$$T_B = 0,$$

$$T_T = T_u,$$

and substituting for N, gives the equation

$$T_u = 2\mu\, \frac{T_u}{\lambda_f\, d_c}\, L, \tag{4-4}$$

so that

$$L = \frac{\lambda_f d_c}{2\mu} \qquad (4\text{-}5)$$

is the expression for the effective track length required for a rectilinear traction winch. In general,

$$\lambda_f \sim 40,$$

and the PneumoDynamics work with cable haulers indicates that the friction coefficient cannot be reliably assumed greater than about 0.1. Hence, we find

$$L \geq 100\, d_c,$$

if both chains are driven. For the winch to handle cable in the quarter to half inch diameter range, the effective track length must be between about 4 and 8 feet if only one track is driven, or 2 to 4 feet if both are driven.

4.2.2.2 Circular Track Winch

In this design concept, sketched in Figure 4-8, the hauler tracks follow a circular path so that the slack-storage drum can be mounted within the track periphery in order to maintain the maximum compactness. A second benefit accrues, however, in that the curvature of the track allows the tension in the tow cable and outer belt to provide the normal force required for traction. The rectilinear puller must have a system of air or hydraulic cylinders to provide this force; the circular hauler uses only one ram to generate a pretension in the outer belt.

Derivation of the equations which model the performance of a circular track winch is presented in Appendix A. The results are summarized here. Let the cable tension be reduced from $T_T$ to zero in passing around the track of total arc, $\theta$. Then the pretension in the outer belt must be

$$T_{po} = T_T / (e^{2\mu\theta} - 1), \qquad (4\text{-}6)$$

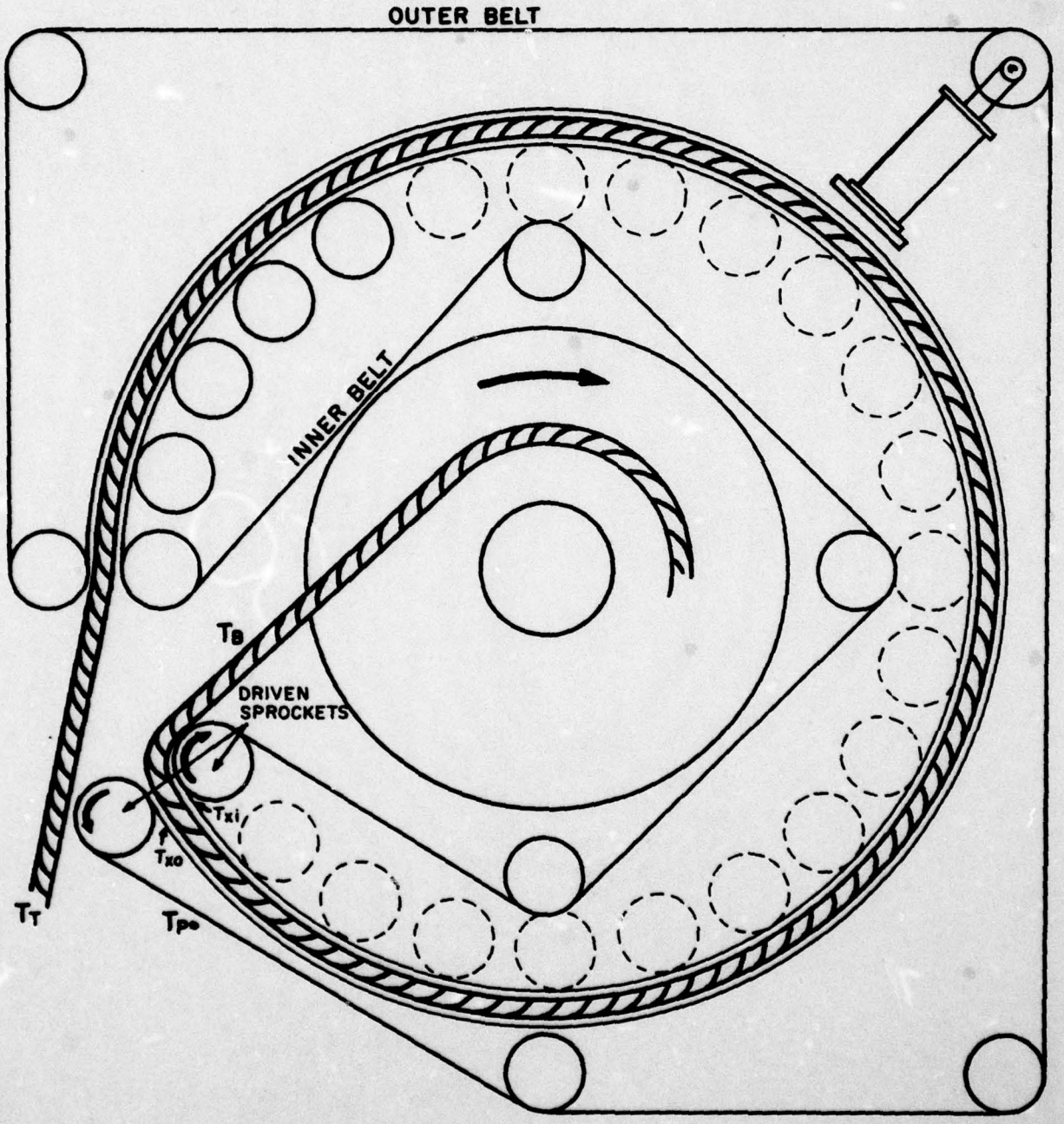



**Figure 4-8. The Circular-Track Cable Hauler**

and the maximum tension

$$T_{xo} = T_{po}\, e^{\mu\theta}. \qquad (4\text{-}7)$$

Pretension in the inner belt does not increase tractive ability:

$$T_{pi} = 0,$$

but the maximum tension in the inner belt is given by

$$T_{xi} = T_T \left( \frac{e^{\mu\theta} - 1}{e^{\mu\theta} - e^{-\mu\theta}} \right) . \qquad (4\text{-}8)$$

If 30 degrees are allowed for towcable access to the track, $\mu$ is taken as 0.1, and the towing capability is equated to the cable strength, these equations become

$$T_{po} = .462\, T_u,$$

$$T_{xo} = .822\, T_u,$$

and

$$T_{xi} = .640\, T_u.$$

These relations are plotted as Figure 4-9.

The analysis of the circular track winch is carried further in Appendix A. It is shown that the track cannot be formed of uniformly spaced rollers if the inner tread belt is flexible. The roller diameter is excessively large in order to prevent crushing the cable by the localized normal force. The inner belt must be formed of roller chain moving on a rigid circular track with sprockets at each end.

4.2.2.3 Belted Drum Winch

A third traction winch is sketched in Figure 4-10. A single traction belt clamps the towcable to two drums which also serve as sprocket wheels for the belt. Thus the point-to-point contact of the circular arc traction winch is replaced by continuous contact with the drum surface. Furthermore, the heavy sprocket loads imposed at the ends of the circular arc track are eliminated, as well as the complexity

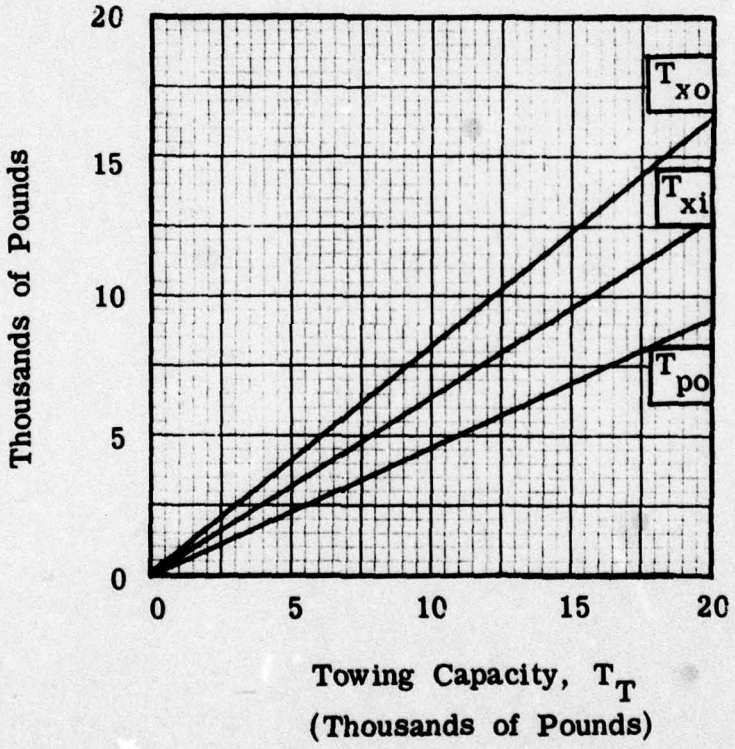



Figure 4-9. Belt Tensions vs Towing Capacity for Circular Traction Winch with no Back Tension

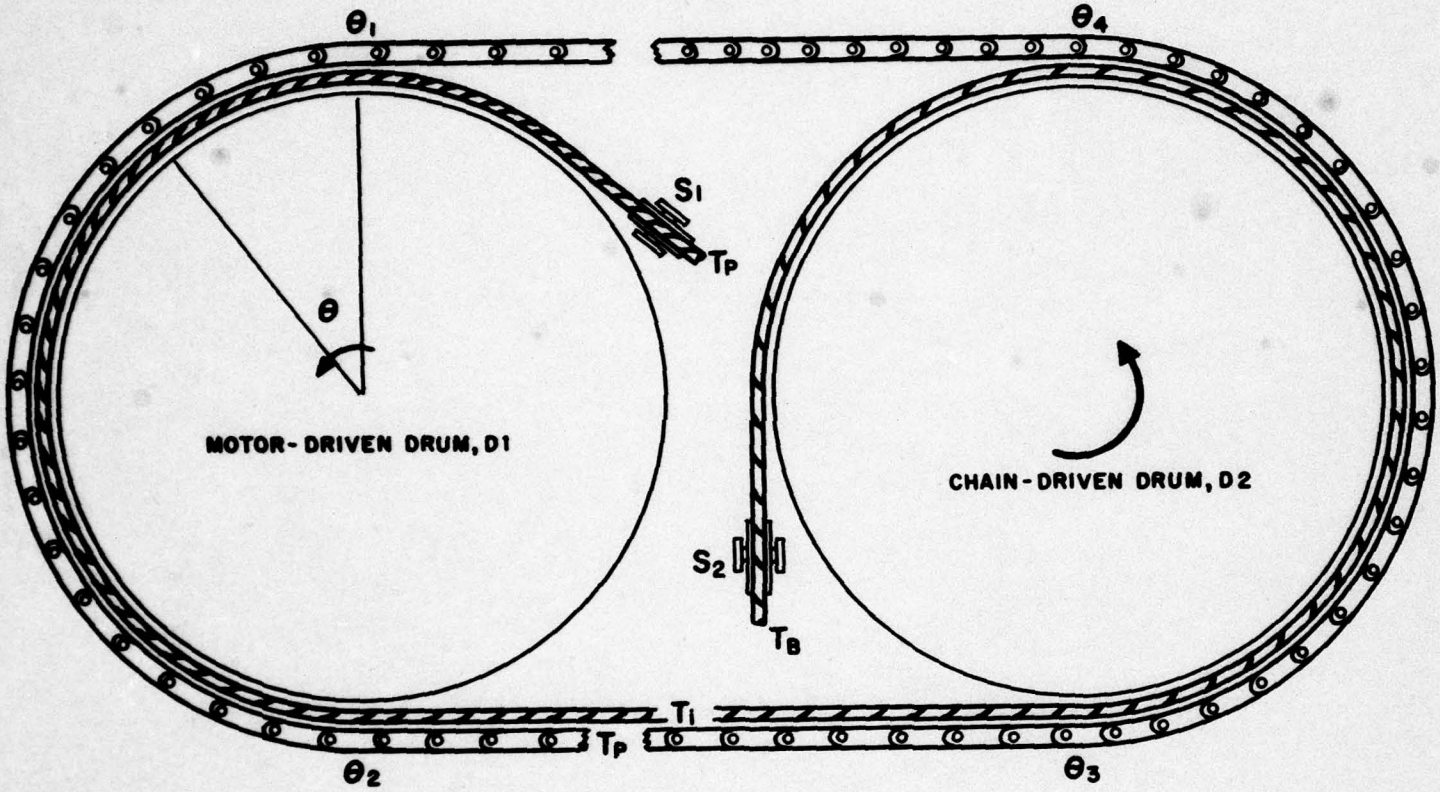



**Figure 4-10. Belted Twin Drum Cable Hauler**

of the slack-side return rollers for the inner and outer belt. Like the linear puller, however, the storage mechanism for the belted drum winch is separated from the winch itself.

The operation of the belted drum winch is as follows. Drum D1 is rotated by a suitable prime mover and transmission. Drum D2 is an idler whose shaft position may be adjusted relative to D1. Sprocket teeth on the periphery of each drum engage the traction belt, so that D1 drives the belt which drives D2 in turn. The adjustment of the shaft position of D2 allows the belt to be given a pretension, $T_p$.

The towcable is fair-led from the towpoing sheave, around a second sheave S1, into the plane of the drums. From there it is clamped between tread pads on the face of D1 and on the inner surface of the traction belt. After passing around D1, the cable follows the belt and is again clamped against D2. When the cable has passed around D2 and its tension has been reduced to a small value, it is passed around sheave S2, out of plane, and fair-led to the storage mechanism.

A derivation for the equations describing the tensile variation is given in Appendix B. The results of that analysis are summarized below:

$$T_B = \lambda_B T_T, \tag{4-9}$$

$$T_p = \lambda_p T_T, \tag{4-10}$$

and

$$T_x = \lambda_x T_T; \tag{4-11}$$

where

$T_T$ is the cable towing tension,

$T_B$ is the cable back tension to the storage unit,

$T_p$ is the belt pretension,

and

$T_x$ is the belt maximum tension.

Furthermore,

$$\lambda_p = \frac{A - F(1-C) - \lambda_B}{G(1 - C) - B - 1} , \qquad (4\text{-}12)$$

and

$$\lambda_x = F + G\,\lambda_p ; \qquad (4\text{-}13)$$

where

$$A = e^{-\mu\phi_1} ,$$

$$B = -2 \quad 1 + \frac{e^{-\mu\phi_1}}{\mu\phi_1} - \frac{1}{\mu\phi_1} \quad ,$$

$$C = 2 \quad e^{-\mu\phi_1} + \frac{e^{-\mu\phi_1}}{\mu\phi_1} - \frac{1}{\mu\phi_1} \quad ,$$

$$D = e^{-\mu\phi_2} ,$$

$$E = D - 1 ,$$

$$F = \frac{\lambda_B - AD}{CD + E} ,$$

and

$$G = - \frac{BD + E}{CD + E} .$$

In the above,

$\mu$ is the tread friction coefficient,

$\phi_1$ is the tread contact arc around D1,

$\phi_2$ is the tread contact arc around D2, and

$\lambda_B$ is the ratio of the back tension in the cable to the towing tension as defined by Equation 4-9.

Setting

$\mu = 0.1$,

$\phi_1 = \pi$,

and $\phi_2 = \pi$,

as in the previous examples, gives

$$T_p = (0.36369 - 0.86367\,\lambda_B)\,T_T \quad ,$$

$$T_x = (0.78926 - 1.28925\,\lambda_B)\,T_T \quad .$$

From the first equation, note that the maximum value of $\lambda_B$ is

$$\lambda_{BX} = 0.421,$$

since negative values of the pretension have no application. Figure 4-11 is a plot of the ratio of the maximum tension to the pretension in the belt against the cable tension ratio, $\lambda_B$, as expressed by the following equation:

$$\frac{T_x}{T_p} = \frac{0.78926 - 1.28925\,\lambda_B}{0.36369 - 0.86367\,\lambda_B} \quad . \tag{4-14}$$

Inasmuch as the winch should be able to part the towline even if the storage unit has failed, the design should assume

$$\lambda_B = 0 \quad ,$$

and

$$T_T = T_u .$$

The ultimate strength in pounds, $T_u$, of double armored cable is approximately related to the cable diameter, d, in inches, by the equation

$$T_u = 78{,}000\,d^2 \quad . \tag{4-15}$$

Then the required belt pretension is given, in the same units, by

$$T_p = 28{,}350\,d^2 . \tag{4-16}$$

This equation is plotted as Figure 4-12.

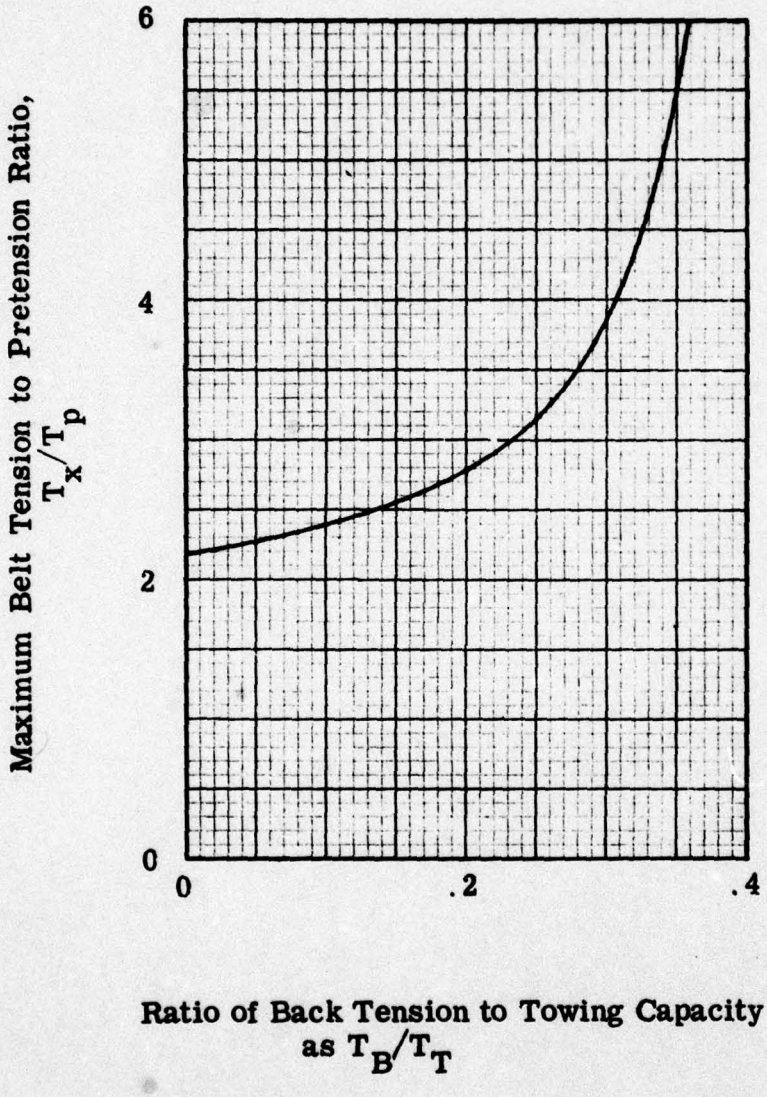



**Figure 4-11. Ratio of Maximum Tension to Pretension in Belt versus Ratio of Back Tension to Towing Capacity for Belted Twin Drum Winches**

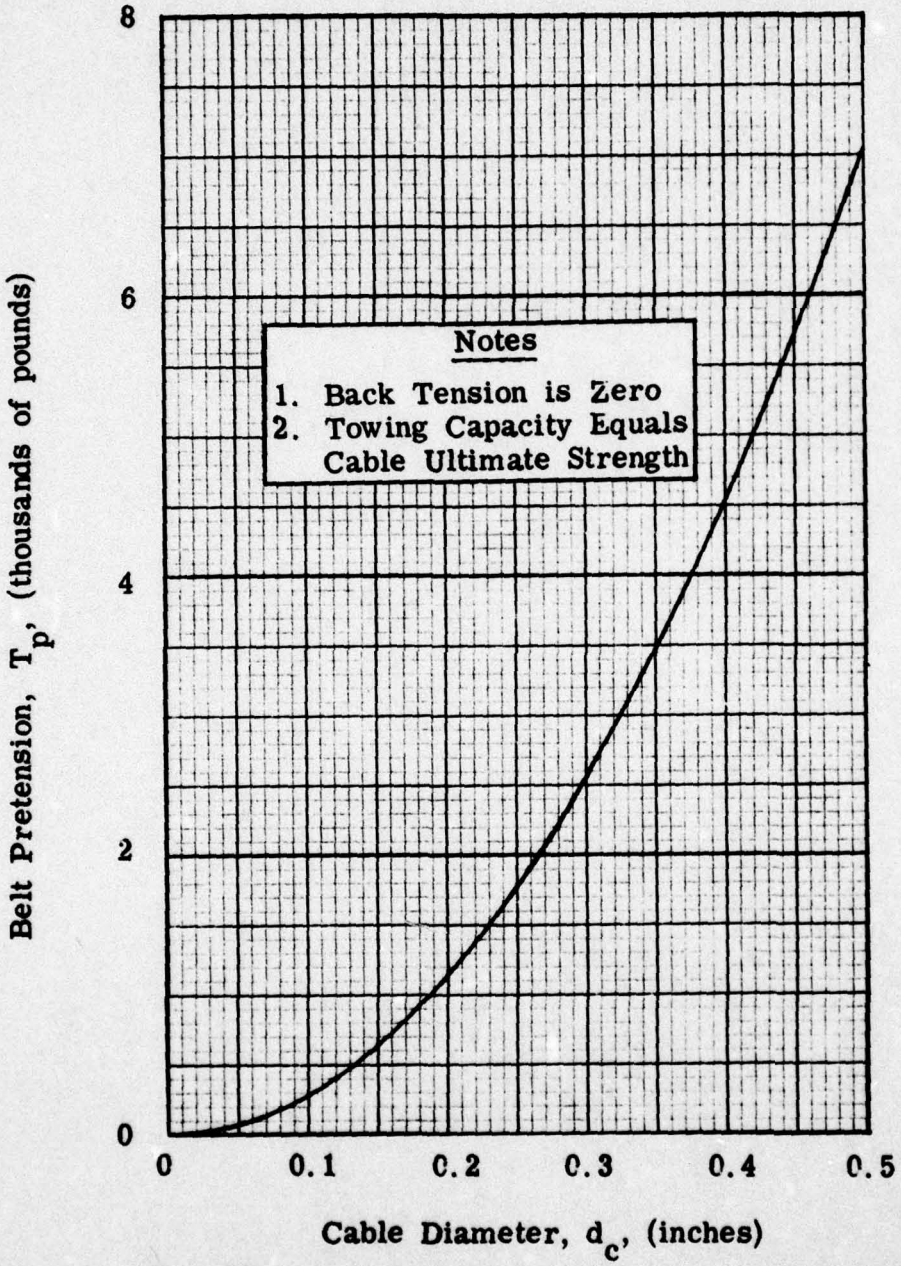



**Figure 4-12. Belt Pretension vs Cable Diameter for Belted Twin Drum Winch**

Figure 4-13 is a polar plot of the cable and belt tensions, where the angle is the contact arc. Thus $0 \leq \theta \leq \pi$ represents drum D1 and $\pi \leq \theta \leq 2\pi$ represents drum D2.

Figure 4-14 shows $T_x$, $T_p$, and $T_i$, the inter-drum cable tension, plotted against towing capacity, $T_T$.

A relation between drum and cable diameter (Equation 4-11) is derived in Appendix B, which limits the normal force on the cable in order to prevent the cable strands from crushing the core wires.

$$D = 40 (1 + \lambda_x) d_c. \qquad (4\text{-}17)$$

The maximum normal force is set equal to the normal force generated when the cable is under its maximum tension over a flat drum whose diameter has been selected to eliminate the effects of fatigue stresses. The relation, given is plotted as Figure 4-15.

4.2.2.4 Double Drum

On this winch, a pair of grooved or canted drums are driven in tandem, side by side, with several loops of the faired cable passing around them. This concept eliminates the potential mechanical problems inherent in the complex roller-and-track of the cable hauler, but it is potentially the most bulky of all the winching systems studies, except in the case of a very long cable where the storage bulk far exceeds that of the winch drums. Figure 4-16 shows continuous, trailing fairing being hauled on a twin-drum winch built by PneumoDynamics Corporation. The drum in the foreground is plainly canted upward to the right, displacing the faired cable turns without a tendency to "crawl" off the end of the drums. Figure 4-17 shows the back tensioner and storage drum used with this winch.

Double drum winches are expected to function quite well with segmented and continuous faired cables. The typically rigid sectional fairing would require the grooved drum design or other means to hold

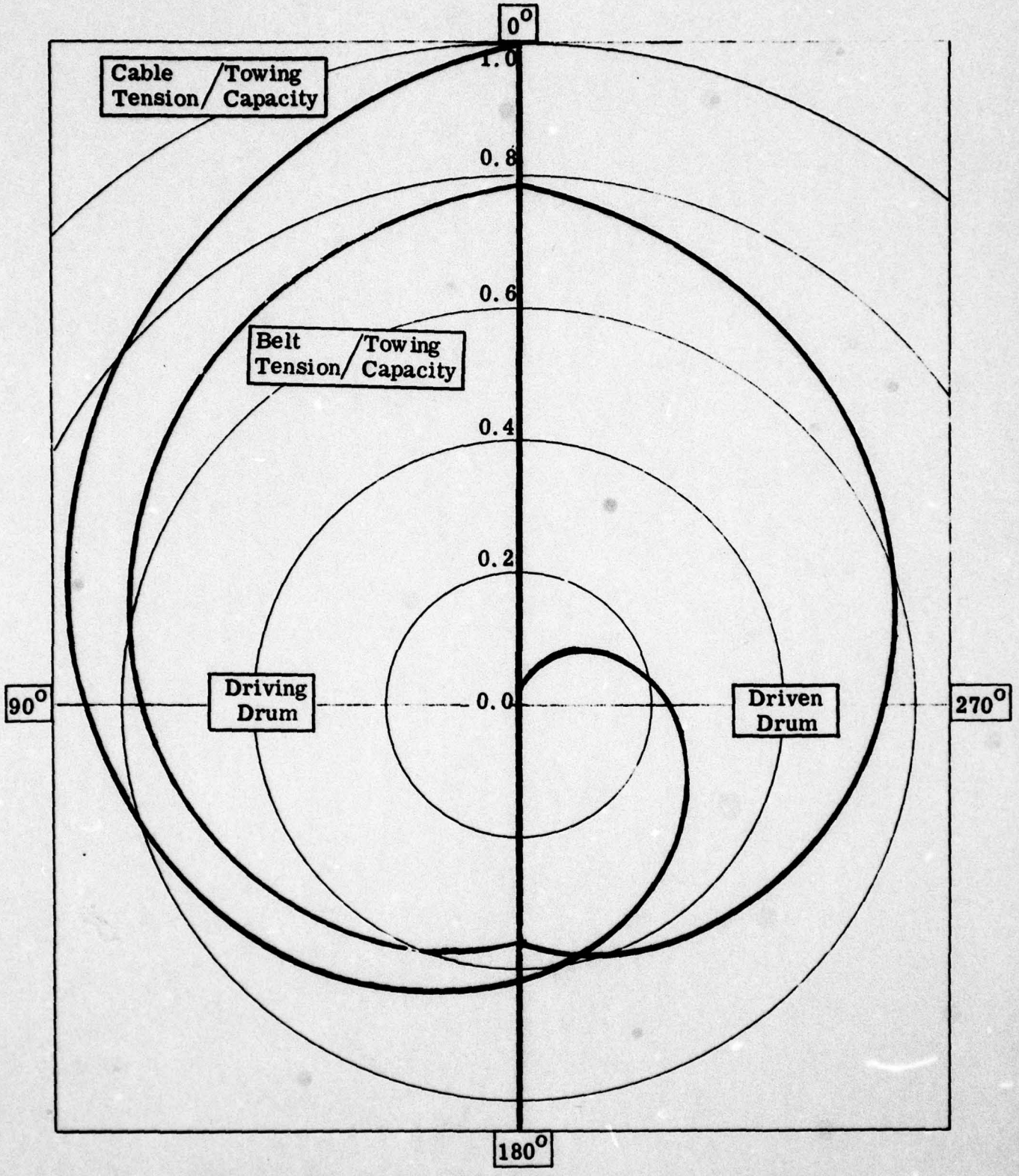



**Figure 4-13.** Cable and Belt Tension Ratios vs Contact Angle with a Belted Twin Drum Winch, Polar Plot

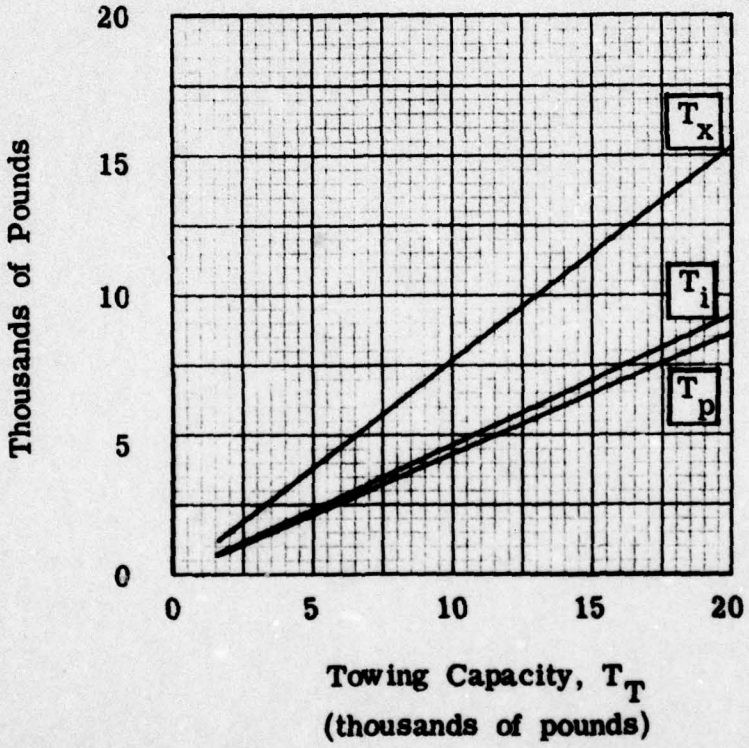



**Figure 4-14. Belt and Cable Tensions vs Towing Capacity for Belted Twin Drum Winch with No Back Tension**

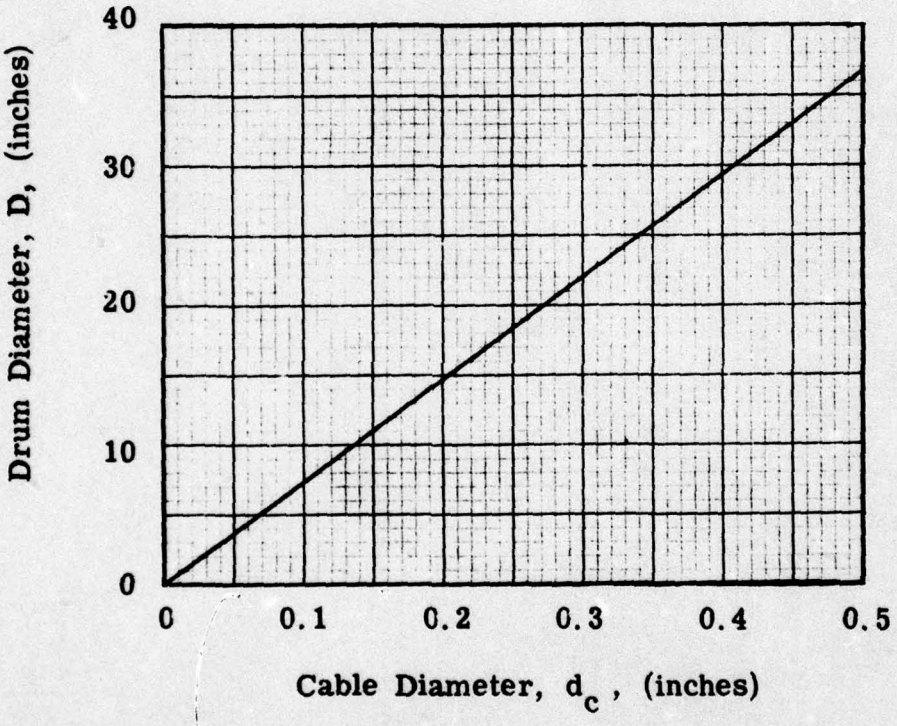



**Figure 4-15. Drum vs Cable Diameter for Belted Twin Drum Winch**

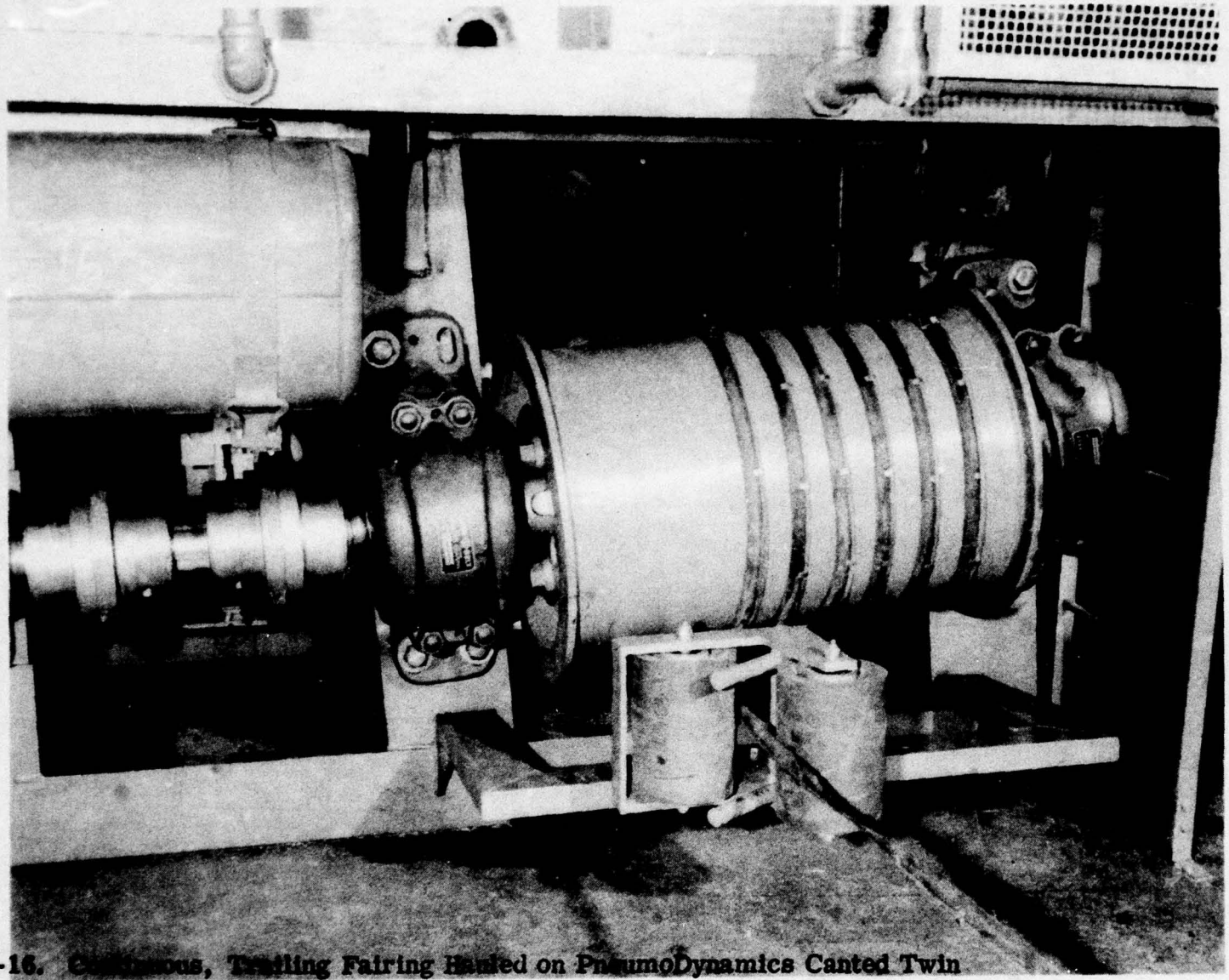


Figure 4-16. Continuous, Trailing Fairing Hauled on PneumoDynamics Canted Twin Drum Winch (from unpublished work performed for the Lamont Geophysical Observatory by PneumoDynamics Corporation.)

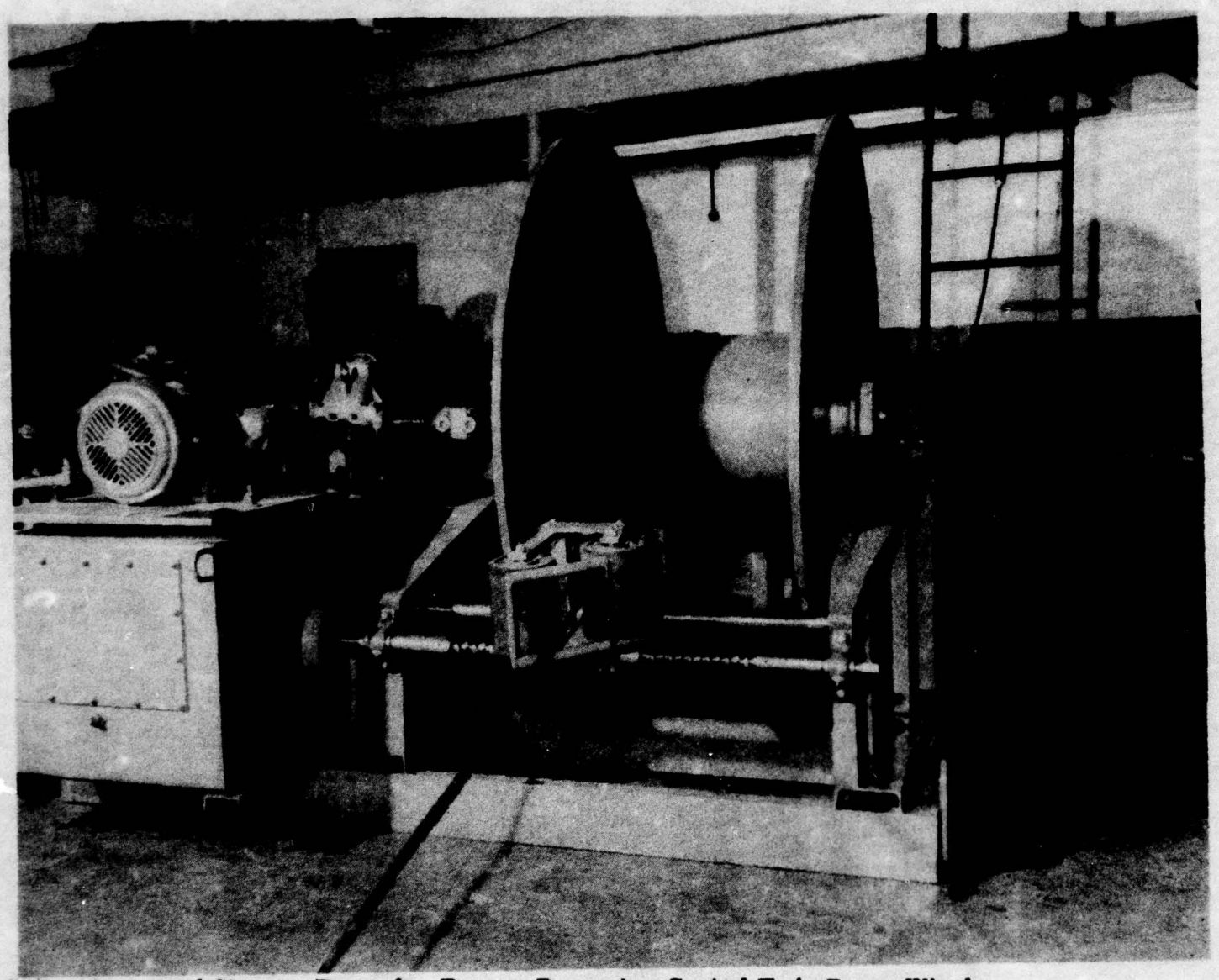


Figure 4-17. Back Tensioner and Storage Drum for PneumoDynamics Canted Twin Drum Winch, Reeving Continuous, Trailing Faired Cable (from unpublished work performed for the Lamont Geophysical Observatory by PneumoDynamics Corporation.)

it nose inward, lest the sections be warped or broken. Fully enclosing nosepieces are, however, subject to deformation and high wear by sliding into the grooves. Failure by bending fatigue must also be considered.

The twin drum winch follows the classic belt equations

$$T_T = T_B \, e^{-\mu\theta}, \tag{4-18}$$

where $\theta$ is the total arc of contact with a powered surface.

Setting

$$T_B = \lambda_B \, T_T, \tag{4-19}$$

gives

$$\theta = \frac{1 \, n \, \lambda_B}{\mu} \, . \tag{4-20}$$

Now

$$\theta = k \pi N,$$

where k=1, 2 is the number of powered drums, and N is the number of turns of cable taken about the drums. From the above, we arrive at

$$N = \frac{1 \, n \, \lambda_B}{k \pi \mu} \, . \tag{4-21}$$

Figure 4-18 is a plot of N versus $\lambda_B$, assuming $\mu$ is 0.1 as before. Unlike the three preceding winch systems which have external means of generating traction forces, the twin drum winch cannot operate at zero back tension. However, a few turns are sufficient to reduce the back tension to a few percent of the towing tension.

Inasmuch as no external normal forces are generated to improve traction on a twin drum winch, the drum diameter may be the same as the rest of the handling system sheaves:

$$D \sim 40 \, d_c.$$

Sectional fairing, which is usually made of rigid parts, should be pulled using a grooved drum to hold the section's nose inward. Flexible fairings can use either grooved or flat, canted drums. The latter, of course, would allow the fairing to lay over, and avoid stretching the

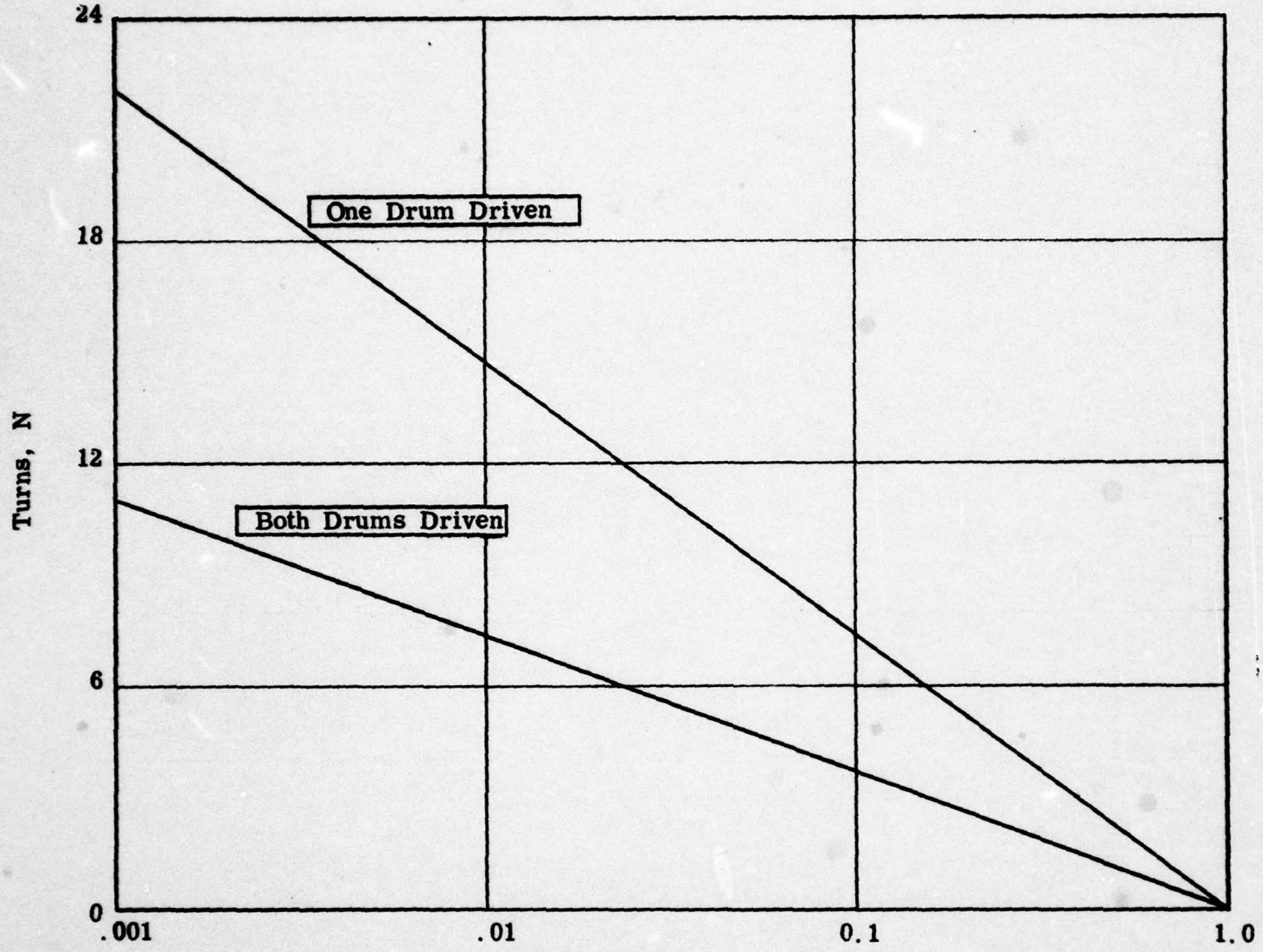



Figure 4-18. Turns About Twin Drums vs Back Tension to Towing Capacity Ratio with One or Both Drums Driven

trailing edge. The cant angle required to prevent adjacent turns from overlapping is given by

$$\sin \theta = C/D \tag{4-22}$$

where C is the total chord of the faired cable, and D is the drum diameter. Thus, approximating sin θ, and substituting for D,

$$\theta = \frac{1}{40} \quad \frac{C}{d_c} \tag{4-23}$$

Cant angle is plotted against the faired cable fineness ratio in Figure 4-19.

4.2.2.5 AIGS Cable Handling System

The AIGS (Ref. 1) system was developed to handle a plastic-jacketed, but otherwise bare, cable towing an acoustic array. If the array were replaced by cable fairing, and a buoy nest added, the AIGS system meets the requirements of a partially faired cable handling system. Appendix D is a description of the AIGS cable handling equipment, taken from Reference 8.

There are 3 notable features of the AIGS system. First, the winch proper is contained within the submarine hull, the cable passing through a packing gland. Second, the winch is placed far forward whereas the towpoint sheave is near the stern, so that a long fairleading tube is used to protect the cable in passing the length of the submarine. Third, seawater is pumped through this tube to lubricate the sliding of the cable and to provide a drag force sufficient to pull the cable out through the packing gland.

By using a faired cable bellmount and a rectangular oval guide tube, the fairing could be winched into the guide tube up to the packing gland and stored there. As long as the faired length is compatible with the potential length of the guide tube - that is, less than the submarine length - this concept offers several performance gains:

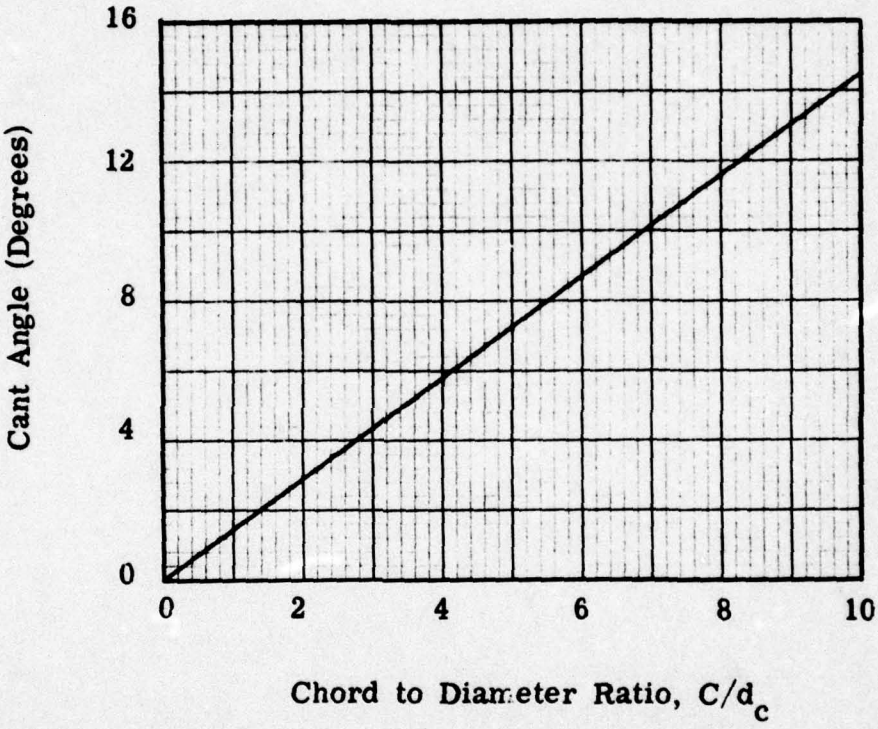



**Figure 4-19. Cant Angle versus Faired-Cable Fineness-Winch-Ratio for Twin Drum Winch**

1. The winch is accessible for repair and service.
2. The winch noise may be decoupled from radiating into the water through the hull.
3. The fairing is stored essentially straight and relaxed.
4. There is no winch housing drag or flow noise.

The concept is limited by practical considerations. One is the availability of space far enough forward for the winch. Another is the control of the buoy as it approaches the nest on the end of a spring several hundred feet long. Finally, only two to three hundred feet of fairing are possible with this system, on present submarines, even if the winch is located in the bow.

In at sea tests aboard HRC's research vessel, HARRIS, and aboard the submarine, TINOSA, it was noted that extracting the initial length from the guide-tube was difficult, but possible, using the hydrodynamic flushing system. This problem will be largely ameliorated in the submarine communications application because the buoy maintains a large tension in the cable.

## Section 5

## EVALUATION AND RECOMMENDATIONS

In this section, a comparative evaluation of the several types of cable fairing is given, followed by a listing of the quantitative measurements needed in order to specify the handling characteristics of the fairings. A third subsection a discussion of the relative merits of the potential handling systems for the submarine communications buoy. The final topic a description of the towing systems recommended for further study after the handling characteristic data have been obtained.

### 5.1 FAIRING TRADE-OFFS

The suggestion was made in paragraph 2.1.3 that fairings should be chosen in order of hydrodynamic efficiency. This format will be followed here.

#### 5.1.1 Integrated Fairing

The two integrated fairings proposed for the hydrofoil VDS offer the highest efficiency of any fairing to date. The monolithic character of the Boeing design should also have very high resistance to vibration damage. The North American articulated design will probably be much more susceptible to such failure.

There are many unanswered questions, however, about these fairings. Practical questions such as, "Can they be manufactured to high standards of quality control on a production basis at reasonable cost?" Operational questions such as, "Will they tow stably in the desired operating conditions?"; "Will the trailing edge relax if stored over a drum?" Some of these questions have been partially answered by the development program currently in progress.

It is the opinion of HRC that

(1) too much development work remains for integrated fairings to be considered in the submarine communications buoy system at the present time;

(2) the drag reduction offered by integrated fairings justifies their continued development, and that on a high level of priority, so that they will be available for the demands of the next generation of submarines.

### 5.1.2 Fully Enclosing Fairings

Free-swivelling, fully enclosing fairings have higher drag than integrated fairing, but have considerably more development background. The properties and construction of sectional, fully enclosing fairing are much different from those of the segmented/continuous versions. Therefore, they are discussed separately.

#### 5.1.2.1 Segmented/Continuous Fairing

These fairings can be designed to achieve drag coefficients nearly as low as for integrated fairings. The operational problems of creep in storage and bunching in tow have restricted their application. In the opinion of HRC, both of these problems can be eliminated by proper handling system selection and design. The solution is predicated, however, on quantitative engineering data about the physical properties of the fairings. Such data are also required with respect to the flex and wear resistance of the cable sheath within these fairings.

Continuous fairing is not expected to respond to these corrective measures as readily as segmented fairing.

#### 5.1.2.2 Sectional Fairing

Free-swivelling, sectional, fully enclosing fairing has higher drag than segmented fairing, and will in fact, barely provide a system that meets the submarine communications buoy performance requirements.

However, it offers the largest base of operational experience of any fairing, from its use on VDS sonar cables.

Inasmuch as each section is an assembly of about 10 discrete pieces, sectional fairing is judged the most susceptible to damage (i.e., section loss) from vibration. Engineering data are required on this and other questions before a reasonable system can be designed to handle sectional fairing aboard a submarine.

With such information available, HRC believes that a system using sectional fairing would be the most straightforward to design, and would require the least development effort of all the fairing types.

### 5.1.3 Trailing Fairings

Although trailing fairings offer the least drag reduction of the conventional cable fairings, they are still superior to hair or ribbon vibration dampers. Furthermore, a significant amount of engineering data already has been compiled, chiefly by the Naval Ship Research and Development Center at Carderock, Maryland. Should serious problems prohibit the use of the more sophisticated fairings, trailing fairings should be considered, in conjunction with relaxed towing requirements.

## 5.2 ENGINEERING DATA REQUIREMENTS

Integrated and trailing fairings have been shown unsuitable for the operational development of a submarine communications buoy, although for opposite reasons. The choice between segmented and sectional fully enclosing fairings cannot be made a priori. Rather, these fairings must stand or fall as parts of an entire buoy-towing system.

But before handling systems can be designed for these fairings, their handling characteristics must be known. A program of engineering data collection must be devised and implemented. Table 5-1 is a list of the pertinent parameters. Tests to measure them in a meaningful way must first be devised.

Table 5-1

Engineering Data Requirements for Cable Fairings

1. Strength and Elastic Modulus
   - (a) Tension
     - (1) Tear
     - (2) Creep
   - (b) Compression
     - (1) Buckling or stacking
     - (2) Crushing
   - (c) Torsion
     - (1) Fairleading
     - (2) Bellmouth
2. Minimum Bend Radius
   - (a) Chord plane
   - (b) Lateral plane
   - (c) Fatigue effects
   - (d) Wear
3. Vibration
   - (a) Fatigue
   - (b) Disintegration
   - (c) Wear
4. Abrasion
5. Impact
6. Corrosion

With these data, the fairings and handling systems can be compared and optimized as specific designs, from which an operational system may be developed. Figure 5-1 outlines such a program.

## 5.3 HANDLING SYSTEM TRADE-OFFS

Following the same method used in evaluating the fairing types, the handling system that intrudes least upon the function of the submarine will be given initial priority in the selection process.

### 5.3.1 Internal Winch With Guide-Tube Storage

The AIGS array-towing winch concept offers several very great advantages. It has the least amount of machinery outside the pressure-hull, which minimizes the drag and noise output of the system. Conversely, it has the maximum proportion of machinery within the hull, accessible for inspection, maintenance and repair. Furthermore, the fairing is stored linearly in the guide tube, eliminating the problems of relaxation and creep in storage. The concept and major components have been successfully tested aboard a submarine at sea.

The major limitation to the concept, and it is admittedly serious, is that the faired portion of the cable is restricted to the length of the guide tube: probably not much more than three quarters of the submarine length, even if the nest and winch are afforded maximum separation.

The operating characteristics desired for the submarine communications buoy could be satisfied by the AIGS system in conjunction with the Boeing integrated fairing. Segmented, fully enclosing fairing does not quite meet the most stringent towing specifications when the faired cable length is restricted to potential guide tube lengths.

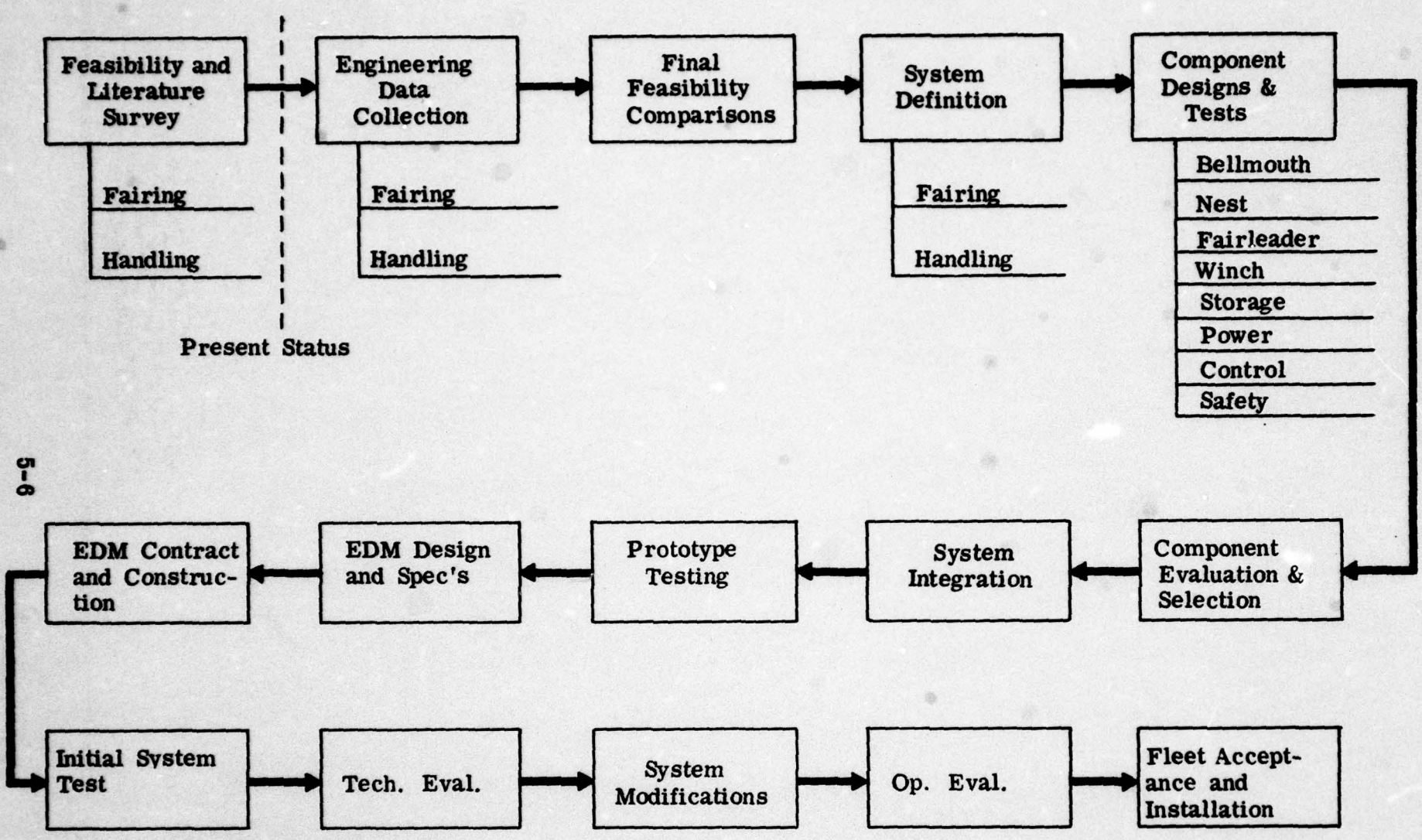



Figure 5-1. Suggested Development Program for Submarine Faired Cable Handling System

However, the Navy might rather accept reduced towing performance in order to retain the knot or two of top speed that bulky deck machinery may cost.

Furthermore, the system could be easily upgraded if and when an integrated fairing gets operational approval.

### 5.3.2 Conventional Deck Mounted Drum Winch

Conventional drum winches offer, of course, the largest technical base from which a design can be developed. They have a positive mechanical attachment to the cable. By combining the winching and storage function on the drum, the most compact of the on-deck systems is achieved.

Although the total cable length may be large, the faired length is restricted to one lay on the drum. Nonetheless, the faired length can exceed that allowable with the AIGS concept. The exo-hull machinery must, of course, be fully automatic, and is inaccessible for maintenance or repair.

A conventional drum winch handling a cable partially faired with fully-enclosing sectional fairing will meet the towing performance specifications for the communications buoy, but the sacrifice in topside weight and top speed will probably not be negligible.

### 5.3.3 Deck-Mounted Twin-Drum and Traction Winches

The principal advantage offered by these winch concepts is that they allow slack storage of much greater lengths of faired cable in multiple lays on the storage drum. The slack storage eliminates the question of segmented fairing relation. However, neither concept has ever been tested with fully enclosing fairing. Whether sufficient tractive effort can be applied to the cable through the fairing without damage depends on the outcome of the test program described before.

Traction winches in particular are not well-developed, especially for submarine applications. They have no positive attachment to the cable, and even a momentary slip can destroy the fairing.

## 5.4 SYSTEM RECOMMENDATIONS

Four towing systems constitute, in the opinion of HRC, the choices most likely to yield successful near-term designs for a submarine communications buoy handling system. After the necessary engineering data have been obtained, these systems should be compared in detail and the most suitable selected for further development. They are listed in descending order of preference, according to the criteria established above. That is, the winch that the intrudes least upon the submarine's operation, together with the hydrodynamically most efficient fairing, is given the highest priority.

1. Internal Winch/Guide Tube Storage/Segmented, Fully Enclosing Fairing
2. Conventional Drum Winch/Storage/Sectional Fairing
3. Conventional Drum Winch/Storage/Segmented, Fully Enclosing Fairing
4. Twin Drum Winch/Slack Drum Storage/Segmented Fully Enclosing Fairing

## 5.5 DEVELOPMENT RECOMMENDATIONS

In addition to the fairing engineering data program recommended in Section 5.2, HRC recommends that specific study be given to faired cable bellmouth and fairleader design. This program is necessary regardless of the handling system/cable fairing combination selected. The bellmouth must orient the fairing without damage for passing over a sheave or into a fairleader; the latter must maintain control of the orientation as the fairing passes from handling element to element. Both

the bellmouth and the fairleaders must function even when the fairing is initially misaligned or has gaps owing to missing sections or segments.

The bellmouth must also align itself with vertical and lateral cable deflections in order to accomodate changes in towing speed and direction. The concepts illustrated in Figures 3-2 and 3-3 represent, in the opinion of HRC, the best starting point for a bellmouth development, but it must be stressed that neither of these designs has been constructed or tested.

Furthermore, HRC recommends evaluation of existing level-wind mechanisms for submersible drum winches. The level-wind is in an exposed location and subject to fouling, corrosion, and lubrication failure. Therefore, there is a significant potential for failure of diamond-screw or chain driven level-wind mechanisms. In the event that conventional mechanisms are found inadequate for submarine applications, then alternate concepts must be investigated, developed, and tested.

## Section 6

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# Appendix A

# ANALYSIS OF TWIN BELT CIRCULAR TRACTION WINCH

## Appendix A

## ANALYSIS OF TWIN BELT CIRCULAR TRACTION WINCH

### DESCRIPTION

The operation of this winch is essentially similar to that of conventional linear traction winches. The cable is pinched between two belt-loops and pulled along with them by friction. Each belt is driven by a sprocket. The original motivation for the circular traction winch was compactness with slack cable storage - the tracks are formed around the central storage drum.

The winch construction is perhaps best understood by examining the sketch (Figure 4-8). A ring of closely-spaced rollers is formed around a storage drum with only enough clearance for three rollers between the periphery of the drum and the ring. One of the traction belts is looped about the ring using the three inner rollers to return the slack-side. If the belt is constructed of roller chain, the ring of rollers may be replaced by a circular track, except at the discharge end where a driving sprocket must be used.

The outer chain also tracks around the ring against the inner chain, but its slack side is returned by four rollers spaced around the outside of the winch. The cable is fed between the inner and outer belts, around the ring, and inward to the storage drum.

Although compactness was the original motivation, an additional benefit accrues in that the normal force between cable and belts is produced by the cable tension and a pretension in the outer belt. No pretension is used in the inner belt, because this only tends to increase the bearing loads on the ring. The outer belt pretension is produced by a hydraulic ram or screw jack acting on one of the outer, slack-side return rollers.

## ANALYSIS

Inasmuch as all the pulling is done in the increments of arc where the belts pass around the rollers, and none occurs in the straight segments between rollers, it is plain that the multiplicity of rollers only serves to expand the track circumference. The pulling capacity is the same as if the cable and belts were wrapped around a single roller.

### Outer Belt Tension

The tension in the outer belt increases from the pretension, $T_p$, to the maximum tension, $T_x$, as the belt proceeds around the rollers. When the outer belt reaches its driving sprocket at the storage end of the track, its tension is reduced to $T_p$ again. The outer belt obeys the classic belt equation,

$$\frac{dT_o}{d\theta} = \mu T_o \ , \tag{A-1}$$

which has the solution:

$$T_o = T_p e^{\mu\theta} \ . \tag{A-2}$$

If there are M rollers, each having a contact arc, $\alpha$,

$$T_x = T_p e^{\mu M\alpha} \ . \tag{A-3}$$

### Cable Tension

The cable loses tension to both belts. The rate of tension transfer to the outer belt was written in part A. The rate of tension transfer to the inner belt may be written by considering the cable and outer belt to be a single unit bearing against the inner belt:

$$\left(\frac{d\,T_c}{d\theta}\right)_i = -\,\mu T' \tag{A-4}$$

where T' is the combined tension, and the negative sign indicates tension loss. Thus

$$\left(\frac{dT_c}{d\theta}\right)_i = -\mu (T_o + T_c) \quad . \tag{A-5}$$

The total tension loss in the cable is, therefore,

$$\frac{dT_c}{d\theta} = -\mu (2\, T_o + T_c) \tag{A-6}$$

or

$$T'_c + \mu T_c = -2\mu T_o, \tag{A-7}$$

but substituting for $T_o$ the function found in part A gives

$$T_c' + \mu T_c = -2\mu T_p\, e^{\mu\theta} \quad . \tag{A-8}$$

the general solution of the reduced equation is

$$T_{cr} = C\, e^{-\mu\theta} \tag{A-9}$$

and a particular solution is

$$T_{cp} = -T_p\, e^{\mu\theta} \, . \tag{A-10}$$

The complete solution is therefore

$$T_c = C\, e^{-\mu\theta} - T_p\, e^{\mu\theta}. \tag{A-11}$$

The boundary condition is that when the angle is zero, $T_c$ equals the towing tension, $T_T$. Thus

$$C = T_T + T_p, \tag{A-12}$$

and

$$T_c = T_T\, e^{-\mu\theta} + T_p\, (e^{-\mu\theta} - e^{\mu\theta}) . \tag{A-13}$$

In addition, when the angle is $M\alpha$, $T_c$ equals the back tension on the storage drum, i. e.,

$$T_B = T_T\, e^{-\mu M\alpha} + T_p\, (e^{-\mu M\alpha} - e^{\mu M\alpha}) \, , \tag{A-14}$$

from which are find

$$T_p = \frac{T_T\, e^{-\mu M\alpha} - T_B}{e^{\mu M\alpha} - e^{-\mu M\alpha}} \cdot \qquad \text{(A-15)}$$

Substituting for $T_p$ in the cable tension equation gives

$$T_c = T_T \left( \frac{e^{\mu M\alpha}\, e^{-\mu\theta} - e^{-\mu M\alpha}\, e^{\mu\theta}}{e^{\mu M\alpha} - e^{-\mu M\alpha}} \right) + T_B \frac{(e^{\mu\theta} - e^{-\mu\theta})}{e^{\mu M\alpha} - e^{-\mu M\alpha}} \cdot \qquad \text{(A-16)}$$

Inner Belt Tension

The sum of the tension increase in the inner and outer belts equals the loss in tension in the cable:

$$(T_i - T_{p_i}) + (T_o - T_{p_o}) = T_T - T_C \cdot \qquad \text{(A-17)}$$

Inasmuch as the inner belt tension only loads the rollers without increasing the normal force between cable and belt, no pretension is applied to the inner belt.

$$T_i = T_T - T_C - T_o + T_p \qquad \text{(A-18)}$$

When the appropriate substitutions are made from Equations A-2, -15, and -16 for $T_c$, $T_o$ and $T_p$, the result is

$$T_i = \left( T_T\, e^{\mu M\alpha} - T_B \right) \left( \frac{1 - e^{-\mu\theta}}{e^{\mu M\alpha} - e^{-\mu M\alpha}} \right) . \qquad \text{(A-19)}$$

The maximum tension in the inner belt occurs when $\theta = M\alpha$:

$$T_{i_x} = (T_T\, e^{\mu M\alpha} - T_B) \left( \frac{e^{\mu M\alpha} - 1}{e^{2\,\mu M\alpha} - 1} \right) . \qquad \text{(A-20)}$$

Normal Force and Roller Diameter

Following the analysis in Appendix B the roller diameter affects the conversion from normal force per radian to force per inch:

$$n_s = N/(d_r/2)$$

or

$$d_r = \frac{2N}{n_s} \qquad \text{(A-21)}$$

The roller diameter is fixed by the ratio of the maximum normal force per radian occurring around the track, to the maximum normal force per inch that the cable can withstand without failure.

It is assumed that the maximum normal force the cable can withstand occurs when the cable is stretched to its breaking tension around a flat-faced drum whose diameter has been selected to mimimize fatigue damage.

$$d_f = d_c$$

where

$$\lambda \sim 40$$

Thus

$$n_s = T_u / (d_f/2) \qquad \text{(A-22)}$$

$$= \frac{2Tu}{d_c}$$

The normal force exerted in the track is largest between the cable and the inner belt, where

$$N = T_o + T_c$$

$$= \left( \frac{T_T e^{\mu M \alpha} - T_B}{e^{\mu M \alpha} - e^{-\mu M \alpha}} \right) e^{-\mu \theta} \qquad \text{(A-23)}$$

Thus, N is a maximum when $\theta = 0$.

$$N_x = \left( \frac{T_T e^{\mu M \alpha} - T_B}{e^{\mu M \alpha} - e^{-\mu M \alpha}} \right) \qquad \text{(A-24)}$$

Setting

$$T_B = 0$$

$$T_T = T_u = \tau_u d_c^2$$

gives

$$d_r = \left( \frac{2\tau_u d_c^2 e^{\mu M \alpha}}{e^{\mu M \alpha} - e^{-\mu M \alpha}} \right) \frac{\lambda d_c}{2\tau_u d_c^2} \qquad \text{(A-25)}$$

$$\boxed{d_r = \frac{\lambda d_c}{1 - e^{-2\mu M \alpha}}} \qquad \text{(A-26)}$$

For

$\lambda = 40$

$M\alpha = 2\pi$

$\mu = 0.1$,

$$d_r \left( \frac{40}{1 - e^{-1.257}} \right) d_c ,$$

or

$$d_r = 56\, d_c$$

For half-inch cable, the required roller diameter is 28 inches.

Inasmuch as a satisfactory winch must be able to pull the cable to parting when the brake is set, one must conclude that the track length compression that occurs in the discrete bends around the rollers produces either unacceptably large normal forces or unacceptable bulk.

## Appendix B

# ANALYSIS OF TWIN DRUM WINCH WITH DRIVEN CHAIN OVERWRAP

## APPENDIX B

## ANALYSIS OF TWIN DRUM WINCH WITH DRIVEN CHAIN OVERWRAP CONCEPT

Drum D1 is rotated by the prime mover, at a rim speed of 100 feet/min, through an appropriate transmission. The drum surface carries a tread of appropriate shape and resiliency to grip one side of the cable. The edges of the drum rim carry sprocket teeth, which have a pitch diameter appropriate for a special roller chain. The roller chain has tread pads on each link to match the drum tread. The motor driven drum drives the chain through the sprocket teeth. The tension in the chain causes it to bear against the cable, and the cable is then passed between the drum and chain treads.

Drum D2 is similar to D1 except that it is driven by the roller chain from D1. The pivot shaft of D2 is not fixed, so that the center distance to the shaft of D1 can be forcefully varied. In this way, the chain may be placed under pretension to increase its gripping force on the cable. The cable, having passed around the motor-driven drum, D1, is passed around D2, under the pads of the chain.

In operation, tension is removed from the cable by friction, with the drums on one side and friction with the chain on the other. The normal forces necessary to produce friction are generated by the tension in the cable and in the chain.

## VARIABLES AND CONSTANTS

| | |
|---|---|
| D1 | motor-driven drum (fixed shaft) |
| D2 | chain-driven drum (movable shaft) |
| $T_T$ | towing tension in cable at winch input |
| $T_B$ | back tension in cable at winch output |
| $T_i$ | intermediate tension in cable between drums |
| $T_p$ | preload tension in chain |
| $T_x$ | maximum tension in chain |
| | coefficient of friction |
| $\theta_1 = 0$ | tangent point of chain and D1 on input (hauling) side |
| $\theta_2$ | angle from $\theta_1$ to tangent point of chain and D1 on intermediate side in rotation direction of D1 |
| $\theta_3$ | angle from $\theta_1$ to tangent point of chain and D2 on intermediate side |
| $\theta_4$ | angle from $\theta_1$ to tangent point of chain and D2 on output (storage) side |
| $T_o(\theta)$ | running chain tension variable |
| $T_c(\theta)$ | running cable tension variable |

## MOTOR DRIVEN DRUM, $D_1$

The chain enters $D_1$ under the maximum tension, $T_x$, and leaves at the preload tension, $T_p$. The difference between $T_x$ and $T_p$ represents the tension removed from the chain in passing around $D_2$. Tension is transferred from the cable directly to $D_1$ under the combined pressure of the cable and chain tension on the drum traction pads. In addition, tension is transferred through the chain and sprocket teech to $D_1$ the pressure of the chain tension on its traction pads. It is assumed that the latter tension transfer occurs through the sprocket tooth immediately following each pad.

### Chain Tension

The reduction in chain tension from $T_x$ to $T_p$ is assumed to occur in a linear manner around $D_1$. A roller chain manufacturer's representative, who was consulted about this question, said that function was non-linear, with most of the tension being relieved in the first few teeth. The increased complexity required to model this non-linearity was judged unwarranted for this report.

Using the linear assumption, the chain tension around the motor-driven drum is directly proportional to:

$$T_o(\theta) = \frac{T_p\theta + T_x(\theta_2 - \theta)}{\theta_2} \quad . \tag{B-1}$$

$$\theta_1 \equiv 0 \leq \theta \leq \theta_2$$

### Cable Tension

Differential Equation. The rate of increase of cable tension around the driving drum (a negative quantity) is defined by the coefficient of friction, $\mu$.

$$\frac{dT_c}{d\theta} \equiv -\mu \lim_{\Delta\theta \to 0} \left| \frac{N}{\Delta\theta} \right| \quad . \tag{B-2}$$

Some of the cable tension is transmitted to the chain by the normal force, $N_o$, and some is transmitted to the drum surface by the combined normal forces, $N_o + N_c$.

$$N = 2 N_o + N_c \quad .$$

$$\frac{dT_c}{d\theta} = - \mu \lim_{\Delta\theta \to 0} \left\{ \frac{\left[ 2 \left( T_o \left| \theta + \frac{\Delta\theta}{2} \right| + T_o \left| \theta - \frac{\Delta\theta}{2} \right| \right) + T_c \left| \theta + \frac{\Delta\theta}{2} \right| + T_c \left| \theta - \frac{\Delta\theta}{2} \right| \right] \sin \frac{\Delta\theta}{2}}{\Delta\theta} \right\} ,$$

(B-3)

$$= - \mu \left| 2 T_o (\theta) + T_c (\theta) \right| \quad .$$

Applying equation B-1 gives the differential equation,

$$T_c' + \mu T_c = - \frac{2\mu}{\theta_2} \left[ T_x (\theta_2 - \theta) + T_p \theta \right] , \qquad \text{(B-4)}$$

for which the reduced equation is

$$T_c' + \mu T_c = 0 . \qquad \text{(B-5)}$$

Solution. The reduced equation has the common solution

$$T_{cr} = C_1 e^{-\mu\theta} . \qquad \text{(B-6)}$$

The particular solution is

$$T_{cp} = \frac{2}{\theta_2} \left[ \left( T_x - T_p \right) \theta - T_x \theta_2 - \frac{T_x - T_p}{\mu} \right] . \qquad \text{(B-7)}$$

Combining these solutions and using the boundary condition

$$T_c(0) = T_T$$

to find the expression for $C_1$, gives the specific solution

$$T_c(\theta) = T_T e^{-\mu\theta} + 2 \left\{ \left( T_x - T_p \right) \frac{\theta}{\theta_2} + \left( T_x + \frac{T_x - T_p}{\mu \theta_2} \right) \left( e^{-\mu\theta} - 1 \right) \right\} .$$

(B-8)

$$\theta_1 = 0 \leq \theta \leq \theta_2$$

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In addition, $T_c(\theta_2) = T_i$

$$T_i = T_T\, e^{-\mu\theta_2} + 2\,T_x\left(e^{-\mu\theta_2} + \frac{e^{-\mu\theta_2}}{\mu\theta_2} - \frac{1}{\mu\theta_2}\right) - 2\,T_p\left(1 + \frac{e^{-\mu\theta_2}}{\mu\theta_2} - \frac{1}{\mu\theta_2}\right) . \qquad \text{(B-9)}$$

CHAIN DRIVEN DRUM, $D_2$

In passing around $D_2$, the chain tension increases from $T_p$ to $T_x$. This increase comes in part through (a) the effort needed to rotate $D_2$, which pulls on one side of the cable, and (b) the effort needed to pull directly on the other side of the cable. This can be seen to be significantly different from the case of the motor driven drum, where the tension taken off the inner surface of the cable has no influence on the tension removed from the outer face.

Chain Tension

It is assumed that the sprocket teeth of $D_2$ are uniformly loaded, so that the increase in chain tension by driving $D_2$ is linear. Let K be the constant of proportionality. Then the differential equation for the outer chain tension is

$$\frac{dT_o}{d\theta} = \mu T_o + K, \quad \theta_3 \le \theta \le \theta_4$$

or

$$T_o' - \mu T_o = K . \qquad \text{(B-10)}$$

The general solution is readily obtained

$$T_o = C_2\, e^{\mu\theta} - K/\mu . \qquad \text{(B-11)}$$

From the boundary condition

$$T_o(\theta_3) = T_p,$$

it is found that

$$C_2 = \left( T_p + \frac{K}{\mu} \right) e^{-\mu\theta_3} \quad . \tag{B-12}$$

The solution for $T_o(\theta)$ is, therefore

$$T_o(\theta) = T_p \, e^{\mu(\theta - \theta_3)} + \frac{K}{\mu} \left( e^{\mu(\theta - \theta_3)} - 1 \right) . \tag{B-13}$$

At the other extreme,

$$T_o(\theta_4) = T_x,$$

from which K is found to be

$$K = \mu \left( \frac{T_x - T_p \, e^{\mu(\theta_4 - \theta_3)}}{e^{\mu(\theta_4 - \theta_5)} - 1} \right) . \tag{B-14}$$

Thus,

$$T_o(\theta) = \frac{T_p \left( e^{\mu(\theta_4 - \theta_3)} - e^{-\mu(\theta - \theta_3)} \right) + T_x \left( e^{\mu(\theta - \theta_3)} - 1 \right)}{\left( e^{\mu(\theta_4 - \theta_3)} - 1 \right)} . \tag{B-15}$$

$$\theta_3 \le \theta \le \theta_4$$

Cable Tension

The differential equation for the cable tension around $D_2$ is the same as for $D_1$:

$$T'_c + \mu T_c = -2 \mu T_o \quad . \tag{B-16}$$

But the expression for $T_o$ is different, as developed in the preceding paragraphs.

$$T'_c + \mu T_c = \frac{-2\mu (T_x - T_p)}{e^{\mu\theta_3} \left( e^{\mu(\theta_4 - \theta_3)} - 1 \right)} e^{\mu\theta} + \frac{2\mu \left( T_x - T_p \, e^{\mu(\theta_4 - \theta_3)} \right)}{\left( e^{\mu(\theta_4 - \theta_3)} - 1 \right)} . \tag{B-17}$$

When this equation is integrated, the result is

$$T_c(\theta) = C_3 e^{-\mu\theta} - \frac{(T_x - T_p) e^{\mu(\theta - \theta_3)}}{(e^{\mu(\theta_4 - \theta_3)} - 1)} + \frac{2(T_x - T_p e^{\mu(\theta_4 - \theta_3)})}{(e^{\mu(\theta_4 - \theta_3)} - 1)} . \quad \text{(B-18)}$$

From the boundary condition

$$T_c(\theta_3) = T_i$$

it is found that

$$C_3 = \left[ \frac{T_i + T_p(2 e^{\mu(\theta_4 - \theta_3)} - 1) - T_x}{(e^{\mu(\theta_4 - \theta_3)} - 1)} \right] e^{\mu\theta_3} . \quad \text{(B-19)}$$

Thus the solution for $T_c(\theta)$ is

$$T_c(\theta) = T_i e^{-\mu(\theta - \theta_3)} + T_p \left[ \frac{e^{\mu(\theta-\theta_3)} + 2(e^{\mu(\theta_4-\theta)} - e^{\mu(\theta_4-\theta_3)}) - e^{-\mu(\theta-\theta_3)}}{(e^{\mu(\theta_4 - \theta_3)} - 1)} \right] + T_x \left[ \frac{2 - e^{\mu(\theta - \theta_3)} - e^{-\mu(\theta - \theta_3)}}{(e^{\mu(\theta_4 - \theta_3)} - 1)} \right] . \quad \text{(B-20)}$$

At the other extreme

$$T_c(\theta_4) = T_B$$

which results in

$$T_B = T_i e^{-\mu(\theta_4 - \theta_3)} + (T_p + T_x) \left( \frac{2 - e^{\mu(\theta_4 - \theta_3)} - e^{-\mu(\theta_4-\theta_3)}}{e^{\mu(\theta_4 - \theta_3)} - 1} \right) . \quad \text{(B-21)}$$

An expression for $T_i$ was developed in Section III. This may be substituted in the above equation to give $T_B$ as a function of $T_T$, $T_p$, and $T_x$ as follows:

$$T_B = T_T\ e^{-\mu(\theta_4 - \theta_3 + \theta_2)}$$

$$+ T_x\left[\frac{2-e^{\mu(\theta_4-\theta_3)} - e^{-\mu(\theta_4-\theta_3)}}{\left(e^{+\mu(\theta_4-\theta_3)} - 1\right)} + 2e^{-\mu(\theta_4-\theta_3)}\left(e^{-\mu\theta_2} + \frac{e^{-\mu\theta_2}}{\mu\theta_2} - \frac{1}{\mu\theta_2}\right)\right]$$

$$+ T_p\left[\frac{2-e^{\mu(\theta_4-\theta_3)} - e^{-\mu(\theta_4-\theta_3)}}{e^{\mu(\theta_4-\theta_3)} - 1} - 2e^{-\mu(\theta_4-\theta_3)}\left(1 + \frac{e^{-\mu\theta_2}}{\mu\theta_2} - \frac{1}{\mu\theta_2}\right)\right]. \qquad \text{(B-22)}$$

Neglecting inertia and friction (except between the traction pads and the cable), then the increase in belt tension equals the decrease in cable tension around $D_2$. That is,

$$T_x - T_p = T_i - T_B.$$

Substituting for $T_B$ and solving for $T_x$ gives after extensive simplification:

$$T_x = T_T\left[\frac{\mu\theta_2\left(e^{\mu(\theta_4 - \theta_3)} - 1\right)}{2\left(e^{\mu\theta_2} - 1 - \mu\theta_2\right)\left(e^{\mu(\theta_4 - \theta_3)} - 1\right) + \mu\theta_2\ e^{\mu\theta_2}}\right]$$

$$+ T_p\left[\frac{2\left(e^{\mu\theta_2} - 1\right)\left(e^{\mu(\theta_4 - \theta_3)} - 1\right) + \mu\theta_2\ e^{\mu\theta_2}}{2\left(e^{\mu\theta_2} - 1 - \mu\theta_2\right)\left(e^{\mu(\theta_4 - \theta_3)} - 1\right) + \mu\theta_2\ e^{\mu\theta_2}}\right]. \qquad \text{(B-23)}$$

## SUMMARY OF TENSION EQUATIONS

In previous sections the equations have been developed relating $T_x$ $T_B$ to $T_T$ and $T_p$.

$$T_x = A_x T_T + B_x T_p.$$

$$A_x = \left[ \frac{\mu\theta_2 \left( e^{\mu(\theta_4 - \theta_3)} \right)}{2 \left( e^{\mu\theta_2} - 1 - \mu\theta_2 \right) \left( e^{\mu(\theta_4 - \theta_3)} - 1 \right) + \mu\theta_2 \, e^{\mu\theta_2}} \right] .$$

$$B_x = \left[ \frac{2 \left( e^{\mu\theta_2} - 1 \right) \left( e^{\mu(\theta_4 - \theta_3)} - 1 \right) + \mu\theta_2 \, e^{\mu\theta_2}}{2 \left( e^{\mu\theta_2} - 1 - \mu\theta_2 \right) \left( e^{\mu(\theta_4 - \theta_3)} - 1 \right) + \mu\theta_2 \, e^{\mu\theta_2}} \right] . \quad \text{(B-24)}$$

$$T_B = A_B T_T + B_B T_p + C_B T_x .$$

$$A_B = e^{-\mu(\theta_4 - \theta_3 + \theta_2)} .$$

$$B_B = \left[ 2 \, e^{-\mu(\theta_4 - \theta_3)} \left( \frac{1}{2} + e^{-\mu\theta_2} + \frac{e^{-\mu\theta_2}}{\mu\theta_2} - \frac{1}{\mu\theta_2} \right) - 1 \right] .$$

$$C_B = \left[ -2 \, e^{\mu(\theta_4 - \theta_3)} \left( \frac{1}{2} + \frac{e^{-\mu\theta_2}}{\mu\theta_2} - \frac{1}{\mu\theta_2} \right) - 1 \right] . \quad \text{(B-25)}$$

Unlike the conventional twin-drum winch, the belted drum winch may operate with zero back tension, for which the following relations are easily obtained.

$$T_p = \lambda_p T_T ,$$

$$T_x = \lambda_x T_T$$

where

$$\lambda_p = - \left( \frac{A_B + A_x C_B}{B_B + B_x C_B} \right),$$

$$\lambda_x = A_x + B_x \lambda_p .$$

## DRUM DIAMETER

The tension equations describing the winching performance of this system do not include the drum diameter as a variable. The winching friction is developed in terms of the normal force, $n_\theta$, between the cable and traction pads per radian of arc. But the crushing strength of the cable limits the normal force per inch of arc, $n_s$, that can be tolerated in the winch. The relation between radians and inches is

$$n_s = 2\, n_\theta / D \ ,$$

where D is the drum diameter in inches.

The limit for computing $n_\theta$ was taken before in establishing the rate of change of tension in the cable. The drum diameter must be computed for the extreme value of $n_\theta$, which occurs between the cable and the drum

$$n_\theta = \lim_{\Delta\theta \to 0} \frac{N}{\Delta\theta} = T_o(\theta) + T_c(\theta) \text{ lbs/radians} \ . \qquad \text{(B-26)}$$

The maximum value of $n_\theta$ occurs at $\theta = 0$ where both $T_o$ and $T_c$ are largest.

$$n_{\theta x} = T_x + T_T \ .$$

$$D = \frac{2\,(T_x + T_T)}{n_{s_x}} = \frac{2T_T}{n_{s_x}} \quad (1 + \lambda_x) \ . \qquad \text{(B-27)}$$

The remaining problem is to establish the crushing strength of the cable. For want of empirical data, the following theory is proposed.

In designing systems with cables running over sheaves, it is customary to specify the sheave diameter in terms of the cable diameter,

$$D_s = \lambda_f\, d_c \ ,$$

where the proportionality constant, $\lambda$, is chosen large enough so that fatigue stresses are insignificant.

$$\lambda_f \sim 40 \quad .$$

It is assumed that when a cable is stretched to its breaking tension over a flat drum whose diameter is fixed by the radio, $\lambda_f$, the cable is also on the verge of collapsing. That is,

$$n_{s_x} \approx \frac{2\,T_u}{\lambda_f\, d_c} \quad ,$$

where $T_u$ is the breaking tension of the cable.

For the belted winch, then

$$D = \lambda_f\, d_c \, \frac{T_T}{T_u} \, (1 + \lambda_x) \quad . \tag{B-28}$$

The winch design should be such that

$$T_T = T_u,$$

so that an impulse that would part a cable on a locked drum winch will produce slippage in the belted drum winch, perhaps saving the cable and body from loss. Thus,

$$D = \lambda_f \, (1 + \lambda_x)\, d_c \quad . \tag{B-29}$$

## Appendix C

## SUMMARY OF FAIRED CABLE HANDLING SYSTEM DATA

### Handling System: AIGS II

| | | |
|---|---|---|
| Winch | Type: | Capstan |
| | Power: | Submarine hydraulic service line |
| | Tension: | 1300 lbs @ 200 feet/min. |
| | Dimensions: | Capstan @ 1 x 2 x 3 feet |
| | | Storage @ 2 x 2 x 2 feet |
| | Turns: | 6 |
| Cable | Type: | Unfaired, plastic jacketted |
| | Diameter: | 0.35 in. |
| | Length: | 2500 feet |
| Reference No. | | 1, 7 |

### Handling System

### Entwhistle Manufacturing Co., BM 878 Caterpiller

| | | |
|---|---|---|
| Winch | Type: | Linear Traction |
| | Power: | 20 Hp |
| | Tension: | 4000 lb. |
| | Speed: | 100 ft/min. |
| | Track: | 45 in. contact length |
| | Pad: | 60 durometer flat neoprene |
| Fairing | Type: | Segmented, Trailing |
| | Material: | Fabric reinforced soft rubber |
| | Thickness: | 0.44 in. |
| | Chord: | 2.00 in. |
| | Length: | 25 feet |
| | Identification: | DTMB TF-84 No. 7 |
| | Cable: | 0.75 in. diameter |
| Reference No. | | 2 |

## Handling System

### Boeing High Speed

| | | |
|---|---|---|
| Winch | Type: | Single Lay, Grooved Drum |
| Fairing | Type: | Continuous, Integrated |
| | Material: | Fiberglass/polyethylene/dacron |
| | Thickness: | 0.58 in. |
| | Chord: | 2.64 in. |
| | Length: | 600 feet |
| | Identification: | NACA 63 A 022 |
| Reference No. | | 3 |

## Handling System

### CHAN

| | | |
|---|---|---|
| Winch | Type: | Single Lay, grooved drum |
| | Power: | 20 Hp |
| | Tension: | 3500 lb at 75 ft/min. |
| | | 1000 lb at 150 ft/min. |
| | Length: | 7 feet |
| | Width: | 6 feet |
| | Height: | 27 in. |
| | Drum: | 21 in. diameter x 36 in. length |
| Fairing | Type: | Segmented, Trailing |
| | Material: | Fabric reinforced rubber |
| | Thickness: | 0.24 in. |
| | Chord: | 0.88 in. |
| | Length: | 16-15 foot segments |
| | Identification: | DTMB B 5 |
| Reference No. | | 5 |

## Handling System

## Western Gear Hauler

| | | |
|---|---|---|
| Winch | Type: | Linear Traction |
| | Tension: | 10,000 lb. |
| | Speed: | 50 ft/min. |
| | Pads: | Flat rubber |
| Fairing | Type: | Segmented, Trailing |
| | Material: | Dacron reinforced dual hardness rubber |
| | Thickness: | 0.63 in. |
| | Chord: | 3.13 in. |
| | Length: | 200 feet |
| | Identification: | DTMB TF-84 |
| | Cable: | 0.782 in., diameter |
| Reference No. | | 6, 12 |

## Handling System

## SQA-Series VDS Hoists

| | | |
|---|---|---|
| Winch | Type: | Single lay, Padded, grooved drum |
| | Material: | Polyethylene/Polypropylene |
| | Length: | 4 inches |
| | Cable: | SQA-10: 1.34 in. diameter |
| | | SQA-13: 0.78 in. diameter |
| Reference No. | | 8, 9 |

## Handling System

### North American High Speed

| | | |
|---|---|---|
| Winch | Type: | Hexagonal Drum, 8 ft. major diameter |
| Fairing | Type: | Segmented, Integrated (articulated) |
| | Material: | Stainless steel/polypropylene |
| | Thickness: | 0.36 in. |
| | Chord: | 1.80 in. |
| | Length: | 4 feet |
| Reference No. | | 10 |

## Handling System

### Pneumodynamics Twin Drum

| | | |
|---|---|---|
| Winch | Type: | Twin Drum |
| | Diameter: | 20 in. |
| | Width: | 30 in. |
| | Cant. Angle: | Variable, 0 to 5 degrees |
| | Power: | Diesel-Hydraulic prime |
| | | 10 hp Electro-Hydraulic back tensioner |
| | Tension: | 14,500 lbs, stalled |
| | | 6,600 lbs at 720 feet/min. |
| | Length: | 116 in. |
| | Width: | 104 in. |
| | Height: | 138 in. |
| | Weight: | 20,000 lb. |
| Reference No. | | 11 |

## APPENDIX D

## AIGS Handling System

### D.1 INTRODUCTION

The purpose of the Towed Array Handling and Stowage System is to deploy, retrieve, and stow a Sonar Array. Basically the array is deployed by allowing the hydrodynamic force generated by the ship's forward motion to drag it away from the ship. The force necessary to move the array from its showed position into the open seawater is provided by a flushing unit. A cable is attached to the array. The cable passes through a hull penetration joint and fastens to a cable handling package. Deployment speed is controlled by applying a variable braking force with a capstan-like traction unit. Retrieval is accomplished by utilizing the traction unit to winch the cable and array back into its showed position against the hydronamic forces. Stowage space is provided by a tube at least as long as the array which is attached to the ship.

Positive control is provided during deployment and retrieval. The cable handling package provides this fun ction . Extreme positions and top speed are automatically limited. Operating parameters of the system are sensed and displayed at the operating station. In addition, array position and cable tension signals are fed to the Sonar Array Complex for remote display and recording.

### D.2 DESCRIPTION OF CABLE HANDLING PACKAGE

The cable handling package contains three main units.

Cable transfer unit - consisting of shearing, sealing, traction, tension, length sensing and guiding devices.

Cable storage unit - consisting of power rewind, rotary joints, end position control devices.

Hydraulic control unit - consisting of manual valving for control of speed, direction, adjustments for torque, rewind tension, overload projection, etc.

These three units are made up of hardware devices each of which has a specific function to perform. The following section describes each device relative to system operating and orientation.

## D.3 CABLE TRANSFER UNIT

### D.3.1 Shear Gate Valve

The shear gate valve is a cut-off device used to jettison paid out cable in an emergency and seal the opening. It is primarily hydraulically actuated with capability for manual override. It contains the necessary subsafe requirements.

### D.3.2 Static Seal

This manually adjustable elastomer seal is normally operated to provide a static seal around the cable. It can also be adjusted to act as a "line-wiper" as the cable is paid out or retrieved. With the cable completely retracted from the transfer unit, the shut-off seal in an emergency can be completely closed, providing shut-off redundancy with the shear valve.

### D.3.3 Ball Check

This ball check will seal against sea pressure and will seat, whenever the cable is not present in the cable passage. It contains the necessary subsafe requirements. It is the automatic shut-off for the condition when the cable breaks and unthreads the system.

### D.3.4 Dynamic Seal

The dynamic seal consists of an elastomer sealing cartridge mounted at the inboard flange of the static seal. Pressure is admitted to the outer surface of the cartridge which tends to balance the sea pressure

on both sides of the seal. When assembled, the cartridge is placed in tension. Elongation of the element changes its internal diameter. The tension is adjusted to achieve optimum sealing fit and pressure. Axial tension also results in hoop tension in the circular section of the seal in contact with the cable. As cable diameter decreases due to repeated payout and retrieval, seal adjustment and efficiency are maintained.

### D.3.5 Traction Drive

The traction drive is a mechanism consisting of a capstan-like drum surrounded by grooved cable guide rollers. These are mounted in a ring-shaped housing. Two hydraulic motors are mounted in the open center of the ring. They drive the drum through shielded live bearings. The guide roller grooves are offset progressively from roller to roller to establish a helical path for the cable as it passes around the drum. They act to make the drum self-threading. Provision is also made for emergency hand cranking of the unit.

### D.3.6 Lock and Tension Gage

A mechanical locking device is provided for use during streaming. This device is capable of holding the cable at any load up to the breaking strength of the cable. Incorporated in this device is a means for sensing cable tension during streaming. Output from this sensor is an electrical signal proportional to cable tension.

### D.3.7 Tracking Device

The tracking device consists of a wide friction roller which rides on four turns of cable. It is mounted in the traction unit in place of one of the regular guide rollers. Sufficient back-up pressure is provided to reduce slippage to a minimum.

### D.3.8 Length and Speed Sensor

The sensor components are contained in an enclosure which is mounted to the traction unit. The tracking device drives a tachometer and a digital encoder through a right angle gear set. The ratio of the gear set has been chosen to produce one foot of cable per rotation of the encoder. The output signal is used to drive receivers located in the local operating station and in the Sonar Console. These operate electronic counters for visual indication. In addition, a signal representing array position is fed from the Sonar Console to the recorder. Zeroing of the counters is accomplished simultaneously by electronic means.

### D.3.9 Cable Guide

A pivotable pulley arrangement is provided at the end of the housing to level win on to the stowage drum.

## D.4 CABLE STORAGE UNIT

### D.4.1 Reel

Storage capacity of the reel is 2500 feet of 0.350 inch diameter cable. A mechanical lock is incorporated to prevent slippage during streaming. The lock contains a fail safe ratchet pawl.

### D.4.2 Power Rewind

The cable storage reel is driven by a combination hydraulic motor and pump to provide a minimum of 50 pounds of inhaul tension on the cable during both payout and retrieval. This drive is interlocked through the hydraulic control unit to operate whenever the handling system is turned on. A pressure regulator is mounted on the pressure port of the motor to adjust the reel-back torque. This is permanently adjusted during the first calibration runs, and then locked in position.

D.4.3 Rotary Joint

A rotary joint provides the interface between the deployed cable and fixed cable on board ship. The connection between the cable end and the rotary joint is such that parting occurs should the array hang up and pull the cable after it. Parting occurs at a load which will not damage equipment.

D.4.4 Automatic Shut-off Actuator

A mechanism is incorporated to shut-off the system when the last 75 feet of cable are still wrapped on the reel and in retrieval when 300 feet of cable are out. The mechanism accomplishes this function by triggering two separate limit switches at the storage reel which actuates the automatic shut-off valve at the manifold.

HYDRAULIC CONTROL UNIT

D.5.1 *Manifold*

All hydraulic controls are mounted on the manifold. This in turn is mounted on a panel in the operating equipment area.

D.5.2 (On-Off-Shear) Control Valve

This control is a hand operated 3 position, 4-way valve. When in the "OFF" position, the entire Cable Handling System is shut down. When in the "ON" position, the Cable Storage Unit receives hydraulic pressure and produces back tension. Hydraulic pressure is simultaneously delivered to the traction unit direction control. When in the "shear" position, the hydraulic piston in the shear and gate valve is pressurized while the rest of the system is vented to the return line. A safety latch is provided to prevent inadvertent operation of the shear mechanism.

### D.5.3 System Pressure Regulator

The system pressure regulator serves the function of setting maximum Hydraulic Control Unit pressure at some level below ship's system pressure. This is tentatively set at 1500 psi, but can be adjusted at installation, if necessary. Once set, the regulator is secured and does not become a part of the operating procedure.

### D.5.4 (In-Stop-Out) Control Valve

Direction is controlled by a hand operated 3 position, 4-way valve. It controls flow to the traction unit to provide three modes of operation:

1. Cable "IN",
2. Cable "STOP"
3. Cable "OUT"

### D.5.5 Speed Control

This is a hand operated precision flow control valve with a variable range adjustment. It is capable of controlling the speed of the traction unit to produce cable speeds in the range of 0 to 200 fpm. Speed is unaffected by temperature and pressure changes. Four ball check valves provide the logic to permit the control to meter into the traction unit during retrieval, and to meter out during payout.

### D.5.6 Overload Protection

Overload protection is obtained from a pressure regulating valve. Should the array hang-up external to the submarine, it will attempt to drive the traction unit motor as a pump. If the mechanical lock is not in place when this happens, excessive pressures may develop in the hydraulic lines. Under this condition, this valve provides a hydraulic flow path for the purpose of relieving the overpressure.

### D.5.7 Replenishing Circuit

This consists of a pressure regulating valve and a ball check valve. The ball check is in parallel with the regulator. They are oriented to permit free flow from the traction unit motor to the reservoir during retrieval of the array, and low pressure flow to the motor during payout. During the first part of the payout process, this circuit provides limited power to the traction unit to overcome seal forces and traction unit internal friction, the motor begins to act as a pump. This circuit provides hydraulic fluid as needed to prevent the motor from drawing a vacuum and cavitating.

### D.5.8 Dynamic Cable-Tension Indicator

Indication is provided under operating conditions by a pressure gage which reads pressure delivered to the traction unit. The scale reads "pounds force" pull in the cable.

### D.5.9 Automatic Shut-Off Valve

This is in a 2 position, 4-way solenoid operated valve. It is spring leaded in the open position, and electrically actuated to obtain the closed position. It is controlled by switches located at the storage reel to provide automatic shutdown when the extreme positions of deployment and stowage are reached.

UNCLASSIFIED

Security Classification

**DOCUMENT CONTROL DATA - R & D**

*(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)*

1. ORIGINATING ACTIVITY *(Corporate author)*

Hydrospace Research Corporation
5541 Nicholson Lane
Rockville, Maryland 20852

2a. REPORT SECURITY CLASSIFICATION

UNCLASSIFIED

2b. GROUP

Does not apply.

3. REPORT TITLE

FAIRED CABLE HANDLING SYSTEMS

4. DESCRIPTIVE NOTES *(Type of report and inclusive dates)*

Final Report

5. AUTHOR(S) *(First name, middle initial, last name)*

David B. Dillon

| 6. REPORT DATE | 7a. TOTAL NO. OF PAGES | 7b. NO. OF REFS |
|---|---|---|
| 25 November 1968 | 107 | 14 |

| 8a. CONTRACT OR GRANT NO. | 9a. ORIGINATOR'S REPORT NUMBER(S) |
|---|---|
| N00600-68-D-0239 - N00167-68-D-0007 | Technical Report No. 210 |
| b. PROJECT NO. | |
| c. | 9b. OTHER REPORT NO(S) *(Any other numbers that may be assigned this report)* |
| d. | None |

10. DISTRIBUTION STATEMENT

[redacted]

DISTRIBUTION STATEMENT A

Approved for public release;
Distribution Unlimited

11. SUPPLEMENTARY NOTES

12. SPONSORING MILITARY ACTIVITY

Naval Ship Research & Development Center
Washington, D. C.

13. ABSTRACT

This study combines a literature review of winching, fairleading and storage mechanisms for cables equipped with streamlined hydrodynamic fairing, as well as an evaluation of several new handling concepts. The handling systems are considered according to the suitability of their use with the various classes of cable fairings. Special emphasis is given to the problem of handling a faired cable aboard a submerged vehicle.

In this application, trailing fairings lack hydrodynamic efficiency and integrated fairings have not yet been developed to an acceptable level. Fully enclosing fairings are suitable, and it is believed that a reliable, functional submarine handling system can be developed to meet the requirements.

DD FORM 1473 REPLACES DD FORM 1473, 1 JAN 64, WHICH IS OBSOLETE FOR ARMY USE.

UNCLASSIFIED

Security Classification

| 14. KEY WORDS | LINK A | | LINK B | | LINK C | |
|---|---|---|---|---|---|---|
| | ROLE | WT | ROLE | WT | ROLE | WT |
| Cable Fairings | | | | | | |
| Winching | | | | | | |
| Cable Handling | | | | | | |
| Submarine Towing | | | | | | |