AD-A163 280 UNDERWATER WINCH STUDY FOR PROJECT LINEAR CHAIR(U) NAVAL FACILITIES ENGINEERING COMMAND WASHINGTON DC CHESAPEAKE DIV SEP 77 CHES/NAVFAC-FPO-1-77 (22)

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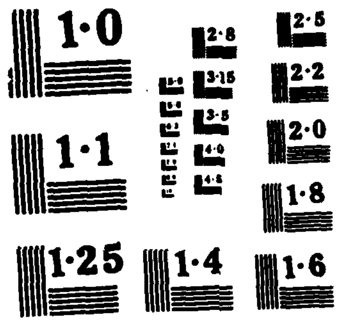

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UNDERWATER WINCH STUDY
FOR
PROJECT LINEAR CHAIR

FPO-1-77 (22)
AUGUST 1977

OCEAN ENGINEERING
AND CONSTRUCTION PROJECT OFFICE
CHESAPEAKE DIVISION
NAVAL FACILITIES ENGINEERING COMMAND
WASHINGTON, D.C. 20374

AD-A163280

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE

REPORT DOCUMENTATION PAGE

1a. REPORT SECURITY CLASSIFICATION
Unclassified

1b. RESTRICTIVE MARKINGS

2a. SECURITY CLASSIFICATION AUTHORITY

3. DISTRIBUTION AVAILABILITY OF REP.
Approved for public release; distribution is unlimited

2b. DECLASSIFICATION/DOWNGRADING SCHEDULE

4. PERFORMING ORGANIZATION REPORT NUMBER
FPO-1-77(22)

5. MONITORING ORGANIZATION REPORT #

6a. NAME OF PERFORM. ORG.
Ocean Engineering
& Construction
Project Office
CHESNAVFACENGCOM

6b. OFFICE SYM

7a. NAME OF MONITORING ORGANIZATION

6c. ADDRESS (City, State, and Zip Code)
BLDG. 212, Washington Navy Yard
Washington, D.C. 20374-2121

7b. ADDRESS (City, State, and Zip)

8a. NAME OF FUNDING ORG.

8b. OFFICE SYM

9. PROCUREMENT INSTRUMENT INDENT #

8c. ADDRESS (City, State & Zip)

10. SOURCE OF FUNDING NUMBERS

PROGRAM ELEMENT #	PROJECT #	TASK #	WORK UNIT ACCESS #

11. TITLE (Including Security Classification)
Underwater Winch Study for Project Linear Chair

12. PERSONAL AUTHOR(S)

13a. TYPE OF REPORT

13b. TIME COVERED FROM TO

14. DATE OF REP. (YYMMDD) 77-08

15. PAGES 84

16. SUPPLEMENTARY NOTATION

17. COSATI CODES

FIELD	GROUP	SUB-GROUP

18. SUBJECT TERMS (Continue on reverse if nec.)
Project Linear Chair, Linear Chair, Winch, Underwater ranges,

19. ABSTRACT (Continue on reverse if necessary & identify by block number)
A series of LINEAR CHAIR arrays are planned for installation on underwater tracking ranges. The purpose of this study is to identify potential underwater winch approaches for adjusting the length of several array configurations.

20. DISTRIBUTION/AVAILABILITY OF ABSTRACT
UNCLASS/UNLIMITED ✱ SAME AS RPT. DTIC

21. ABSTRACT SECURITY CLASSIFICATION

22a. NAME OF RESPONSIBLE INDIVIDUAL
Jacqueline B. Riley

22b. TELEPHONE
202-433-3881

22c. OFFICE SYMBOL

DD FORM 1473, 84MAR

SECURITY CLASSIFICATION OF THIS PAGE

UNDERWATER WINCH STUDY
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Three cases for use of a capstan were considered, depending on counter-weight size. A 500 lb counter-weight provides back tension to the capstan to prevent slippage. A 4000 lb counter-weight balances the buoyancy of the LINEAR CHAIR array so that the static torque on the capstan brake is essentially zero. Finally, a 4000 lb counter-weight, with the array buoyancy reduced to about 3000 lbs, uses the capstan power to raise the array and gravity for lowering in the emergency situation. Figure 3 shows a gravity powered counter-weighted capstan.

TRADEOFF ANALYSIS.

An analysis of selection criteria versus emergency energy sources was performed. The selection criteria included size, weight, power requirements, reliability, component availability, lead time, deployability, experience and cost.

Milestones for the development of the underwater winch were prepared. In addition, preliminary weight and cost estimates to design and fabricate the attractive winch approaches for the fixed LINEAR CHAIR array were developed.

CONCLUSIONS AND RECOMMENDATIONS.

None of the existing winches reviewed will meet the requirements of the LINEAR CHAIR Project Fixed Array. However, the-state-of-the-art has progressed to the point where a suitable winch could be designed and fabricated.

Several approaches appear promising for positioning the array. The prime candidates are the hydraulic accumulation winch for the single-leg configuration and the gravity capstan for the multi-leg configurations.

Refinement of the capstans and the hydraulic accumulation winch approaches, in conjunction with the array configuration development

process, is recommended. Also, the components (i.e., power cable, transformer, controller, etc.) required to support the winch must be identified, as well as the impact upon installation imposed by including the winch in the array.

EXECUTIVE SUMMARY

A series of LINEAR CHAIR arrays are planned for installation on underwater tracking ranges. The purpose of this study is to identify potential underwater winch approaches for adjusting the length of several array configurations. Potential array configurations are presented in Figure 1.

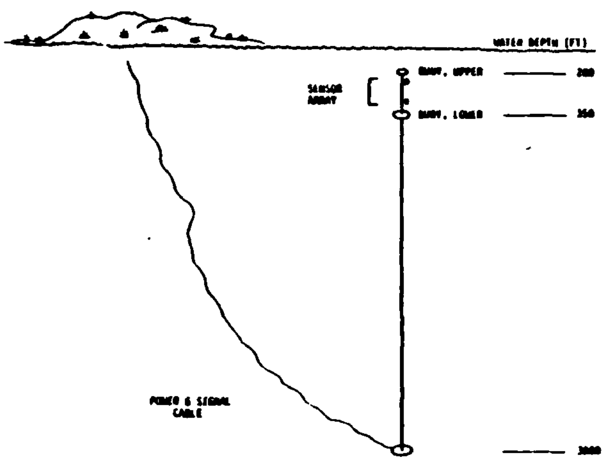
The underwater winch will provide a means for positioning the sensor array at specific depths between 200 and 1500 feet below the sea surface. The water depth will be 3000 feet. Normal positioning of the array will be at .5 foot/second. Emergency haul-down will be accomplished at 10 feet/second.

An underwater winch survey was conducted in 1972 that reported on ten underwater winches which represented the state-of-the-art design at that time. Since that time, five submersible winches have been developed or are under development. None of the existing winches reviewed will meet the requirements of the LINEAR CHAIR Project Fixed Array. → to p. vii

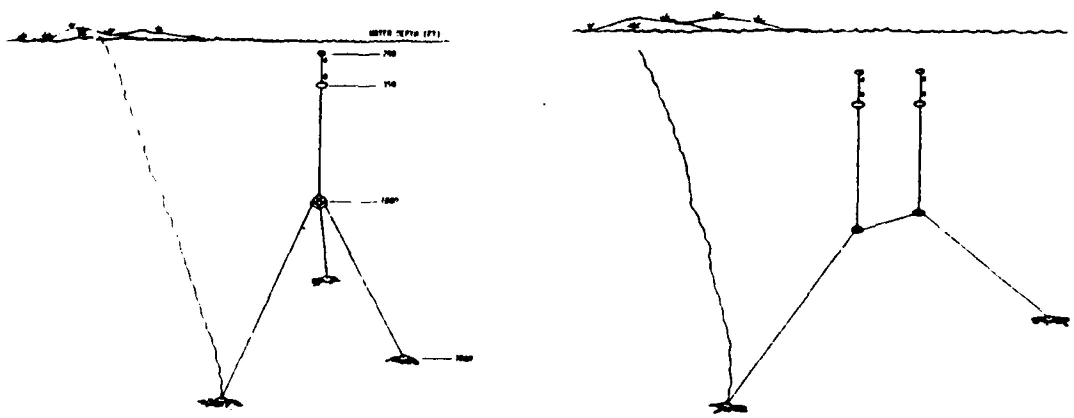
SINGLE-LEG ARRAY.

Nearly 120 horsepower are required to lower the LINEAR CHAIR array and buoys 120 feet in 12 seconds for the emergency situation. Since the 120 Hp rate need be sustained for only 12 seconds, it is feasible to store the energy at the site in a form readily converted to mechanical energy. Three energy storage principles were evaluated for the single-leg array.

- counter-weight using gravity
- hydraulic accumulators using compressed gas, and
- flywheels using rotational inertia.



Single-Leg Array Configuration



TRI-MOOR ARRAY

GOAL POST ARRAY

Multi-Leg Array Configurations

Figure 1. Array Configurations

The counter-weight is the most direct way to supply the energy for normal and emergency haul-down. However, the array would be very sensitive to ocean current variation due to the low cable tension below the weight. Since the LINEAR CHAIR array has a design goal of movements of less than a yard, the counter-weight for a single-leg installation is not suitable.

An electro-hydraulic winch consists of a hydraulic pump powered by an electric motor, a winch powered by a hydraulic motor, and an accumulator. The accumulator contains gas under pressure.

Flywheels store energy mechanically in their rotational inertia. Three factors dominate: rotor weight, windage, and gearing. Windage is the power required to maintain rotor speed against drag. Although the emergency haul-down can be accomplished by very small rotors, windage losses are prohibitive unless the rotor is in a near vacuum. Overall, the problems associated with flywheels for LINEAR CHAIR are judged to outweigh the benefits.

MULTI-LEG ARRAY.

Two multi-leg array configurations (tri-moor and goal post) were investigated. Since the two configurations differ only in the means of securing the apex buoy position, their winching techniques can be discussed together. The multi-leg configurations permit an efficient solution: mounting a depth adjustment winch on the apex buoy (Figure 2). If a drum winch is used at the apex buoy, then the discussion of the single-leg electro-hydraulic winch applies to the multi-leg cases as well. But, the major benefit, of the multi-leg moorings from an array handling standpoint, is that they provide a space beneath the cable handler for a counter-weight. The principal disadvantage, to mounting the winch at the apex buoy, is that the winch power cable must come up one of the mooring legs or be suspended from another buoy as shown in Figure 2.

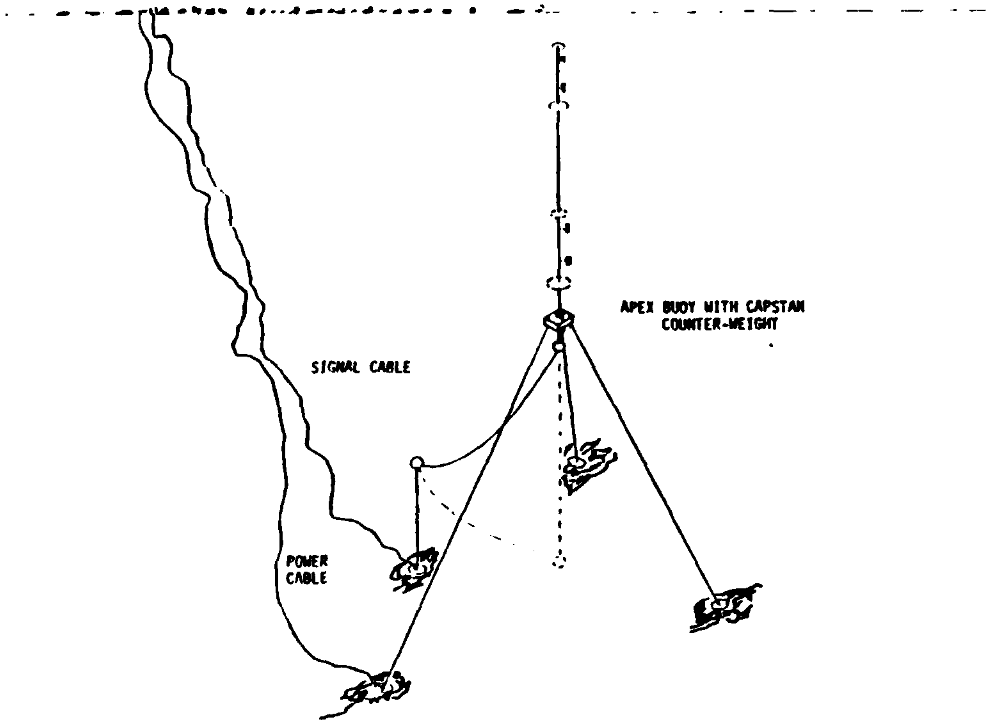


Figure 2. Tri-Moor with Counter-Weighted Capstan

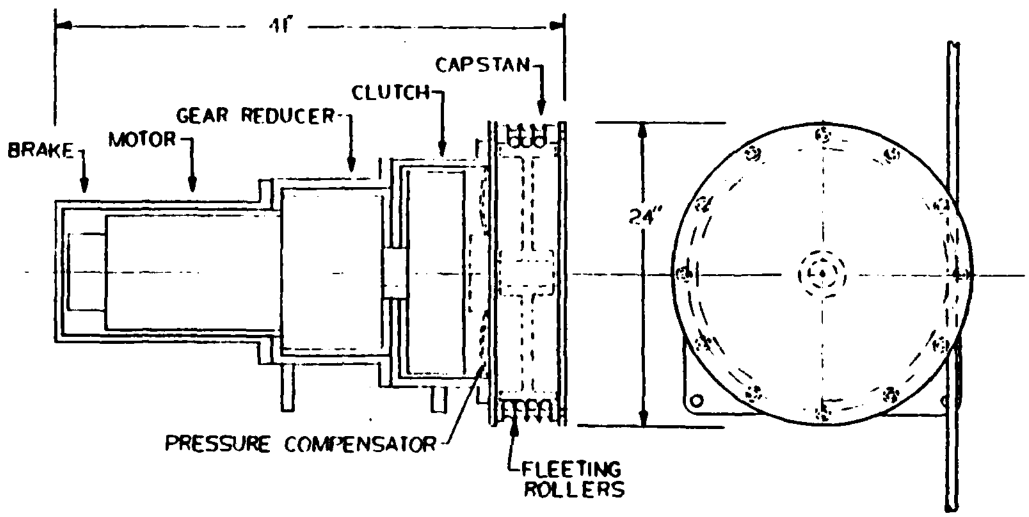


Figure 3. Counter-Weighted Capstan

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SECTION I

INTRODUCTION

A series of LINEAR CHAIR arrays are planned for installation on the Atlantic Fleet Range Support Facility Underwater Tracking Range, St. Croix, U. S. Virgin Islands and other selected locations. The depth of the array sensors must be adjustable. The approach under consideration is to adjust the length of the array mooring line using a shore powered remote-controlled underwater winch.

At the request of the Ocean Engineering and Construction Project Office of the Chesapeake Division of the Naval Facilities Engineering Command, EG&G Washington Analytical Services Center, Inc., Rockville, Maryland, undertook a study to identify potential underwater winch approaches for adjusting the length of several array configurations. This report documents the study.

SECTION II

REQUIREMENTS

2.1 GENERAL.

The underwater winch shall provide a means for positioning the sensor array at specified depths between 200 and 1500 feet below the sea surface. The array is composed of two magnetometer and E-field sensors 150 feet apart. Near each sensor, a pinger will be mounted for underwater tracking of the array. Figure 2-1 shows the sensor array with its upper and lower support buoys in a single-leg configuration. Two other multi-leg array configurations under consideration are shown in Figure 2-2.

The signals from the sensors will be multiplexed in the lower buoy and transmitted through a coaxial cable to shore. For the purpose of this study, the cable shown in Figure 2-3 will be used from the lower buoy to the winch.

2.2 ENVIRONMENT.

Since the array winch is intended for ocean installation periods of at least 1 year in up to 3000 feet of water, specific allowance in its design must be made for the high ambient pressure, corrosion in low velocity flow, fouling, foreign particle intrusion, and general salt incrustation. Therefore, proper material selection and pressure compensation are required in order to maximize reliability.

2.3 PERFORMANCE.

At this stage of the program , size restrictions have not been imposed. However, it must be kept in mind that since this equipment must be installed at-sea, size and weight must be minimized based upon good design practices.

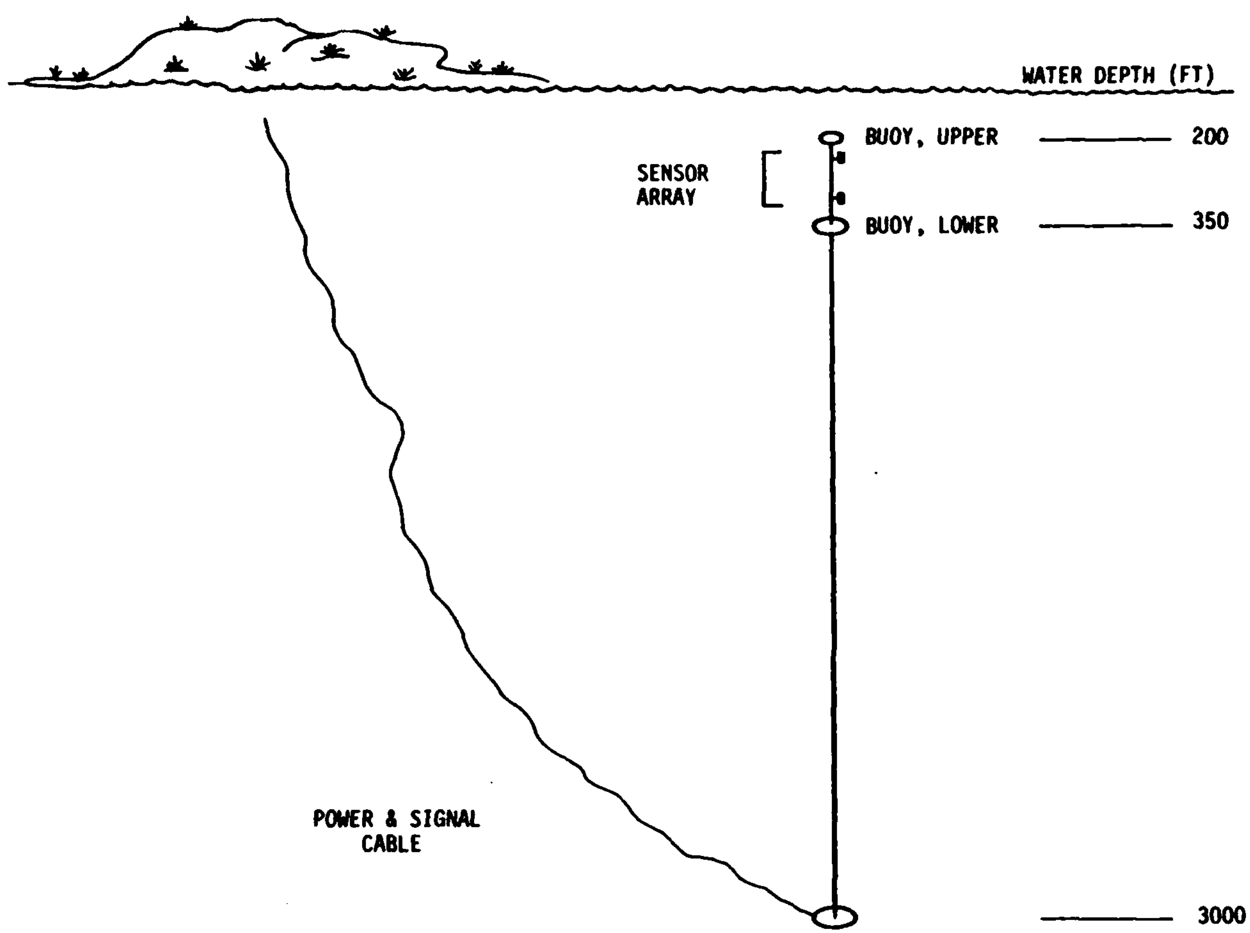

Figure 2-1. Single-Leg Array Configuration

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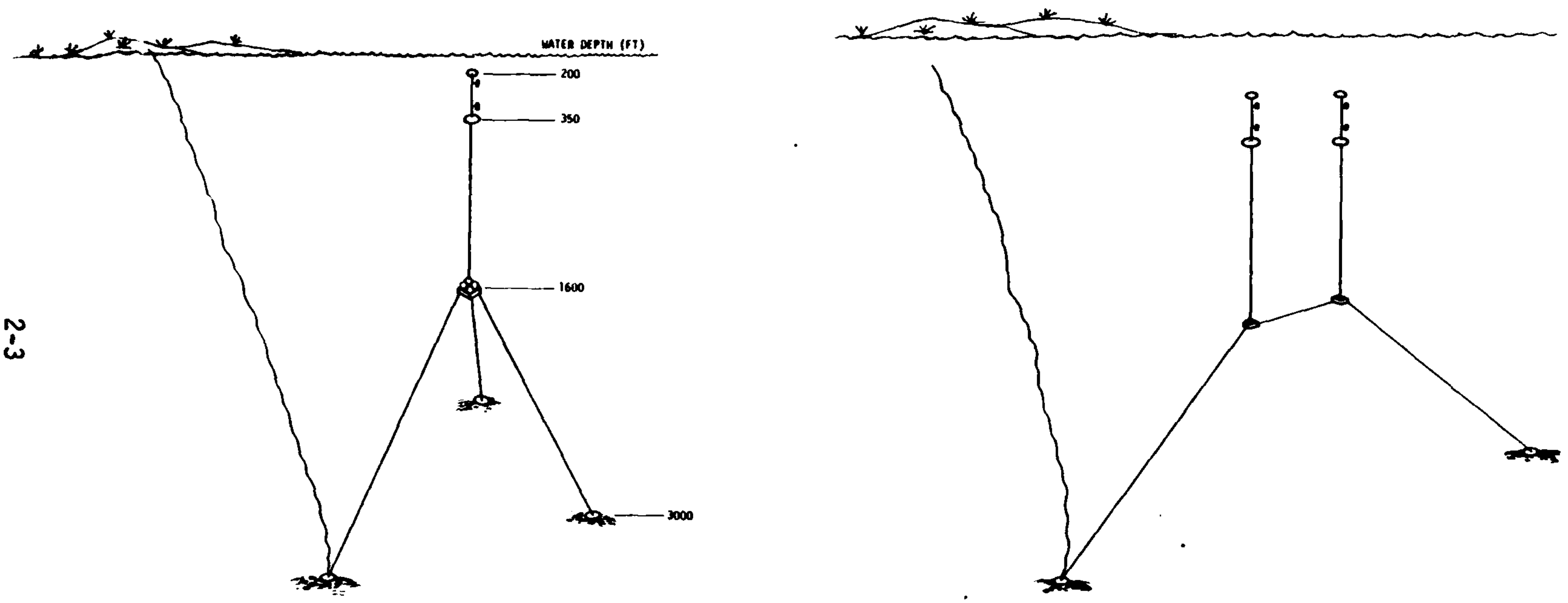

TRI-MOOR ARRAY

GOAL POST ARRAY

Figure 2-2. Multi-Leg Array Configurations

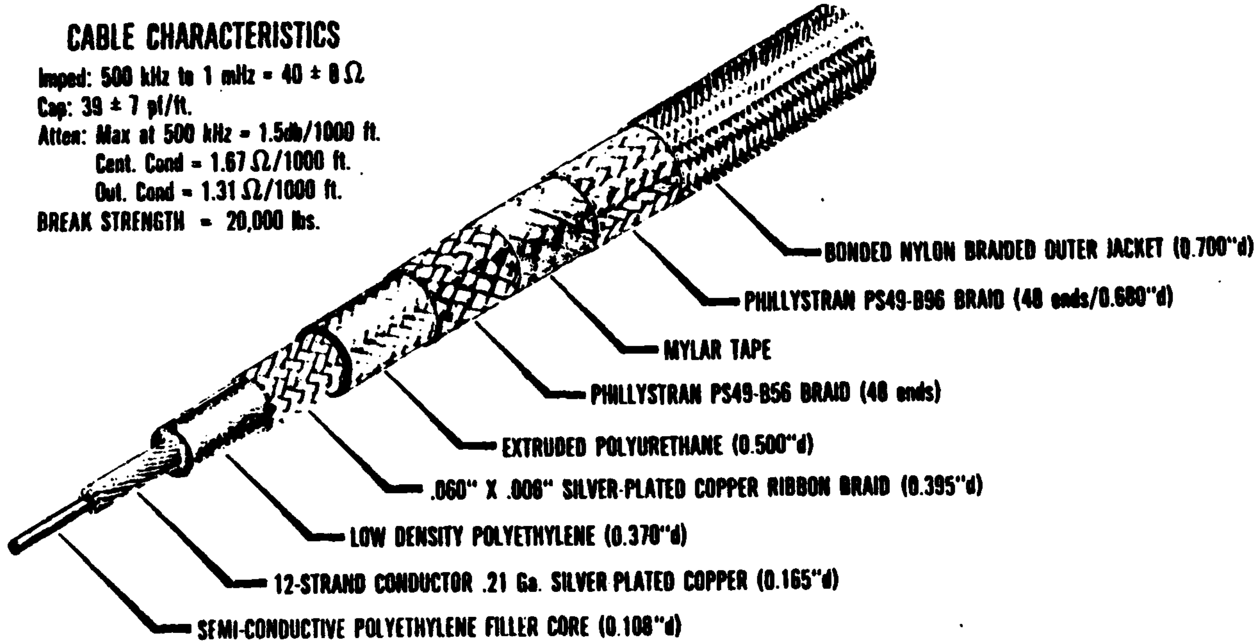

Figure 2-3. Sensor Array Cable

Tentative performance characteristics have been identified in order to provide a base for the development of possible approaches. Table 2-1 summarizes these characteristics. In addition to positioning, the winch must be capable of lowering the sensor array at high speed under emergency situations. Horizontal movement of less than a yard, due to currents, is a design goal.

2.4 CONSIDERATIONS.

The winch(s) for positioning the sensor array could be located at the anchor(s) or at the cable junction buoys in the multiple leg configurations. For simplicity, one winch to perform both the positioning and emergency lowering will be investigated.

Due to the severe array excursion requirement, fairing of the array cable with either fringe or an airfoil section is under investigation. This study will focus on the unfaired or bare cable. Fringe fairing can be handled similarly to bare cable, making allowance for the inefficiencies in storage volume required for direct tension winching. Indirect handling would require an increase in capstan length due to the potential reduction in friction coefficient. Airfoil fairing has been handled on ships at sea by direct tension devices but with some difficulty. It should be noted that the type of fairing greatly affects its handling approach; no experience has been gained underwater. However, submerged systems are in the early stages of development for submarine communication buoy systems. If airfoil fairing is required to meet the excursion requirement, the winch will require development using the experience gained by the submarine communication buoy systems, as a starting point.

Electrical power for the winch will be supplied from shore over a bottom mounted commercially available cable. Alternating current will be supplied, since submersible AC motors are inherently simpler in design

TABLE 2-1. WINCH PERFORMANCE CHARACTERISTICS

Parameter	Positioning	Emergency
Distance (ft)	1500	120
Speed (ft/sec)	.5	10
Duty Cycle	20 up & downs of 300 ft per day	2-3 downs per day*
Water Depth (ft)	3000	
Power Transmission Cable Length (miles)	3	

*Must be available for operation in 30 minutes.

and construction than DC motors, require less maintenance, and have attained a high state of development.

In bare cable winch design, the drum diameter is minimized, not only to reduce torque required to maintain line pull, but also to reduce the ratio of wasted core volume to storage area. The drum must only be large enough to prevent excessive stresses in the cable as it is wrapped upon the drum. Drum diameter of 15 to 25 times cable diameter have yielded 59×10^3 and 943×10^3 bend cycles to failure for Kevlar ropes, when loaded at 20% rated actual breaking strength.[*1] Reference 2 indicated 183×10^3 cycles to failure for a braided jacket 19 x 7 impregnated Kevlar 29 rope loaded at 20% of ultimated strength over pulleys of 24 times diameter. Over a period of 1 year, assuming 200 days operation, the sensor array cable would be bent around the winch drum 8000 times, if it moved up and down twenty times a day. Based upon the test results enumerated above and the number of cycles expected, a minimum drum diameter of 20 inches (i.e., 29.6 times cable diameter) and a working load range of 20 to 30 percent, the sensor array cable braking strength (4000 to 5000 lbs) will be utilized in this study for developing approaches. This should yield cable life in excess of a year.

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* References are listed in Section VIII

SECTION III

UNDERWATER WINCH SURVEY

3.1 GENERAL.

A survey[3] in 1972 reported on ten underwater winches that represented the state-of-the-art design at that time. Of the ten winches, eight were direct tension, conventional drum winches with cable stored either in single or multiple layers. Two were indirect tension winches. All the direct tension winches have given satisfactory results. Only the double-drum capstan type indirect tension winch with fixed basket storage gave satisfactory service. The single-drum capstan-type winch, utilizing a pressure roller and rotating cable storage basket, proved unsatisfactory due to the lack of back tension and had to be redesigned along the double-drum capstan-type indirect tension winch.

3.2 LEVEL WIND DEVICES.

Level wind devices for those winches requiring such a device, whether single layer or multi-layer cable storage, functioned without causing any problem. The CHAN winch with a level wind design for spooling a cable with fairing never operated as a functional system long enough to evaluate its ability to satisfactory level wind a faired cable for an extended period of operation. The VERSUS winch with a level wind that will accomodate a cable of two different diameters through an automatic sensing system functioned for an extended period of time.

3.3 POWER DRIVES.

Power drives for three winches were AC motors and gear reducers in oil filled pressure compensated housings. Two winches used DC motor power drives in air filled housings. One winch used an open-winding water

filled AC motor direct drive hydraulic pump/motor power drive. One winch used a battery powered electro-hydraulic power drive, where the hydraulic oil reservoir serves as the pressure compensation housing for the electric drive motor. One winch used a DC motor sealed in an air filled housing powered by a seawater battery. Another winch was driven by a battery powered DC encapsulated motor. The encapsulation technique proved to be a problem. One winch has a hydraulic power drive located in the submarine.

3.4 MATERIALS.

Materials used for winch mounting frames were steel, corrosion resistant steel (both 304 and 316), and anodized aluminum. Properly treated, the steel has held up well. The aluminum frame was used on an expendable system. The 304 stainless steel is unacceptable due to crevice corrosion and marine fouling.

3.5 PERFORMANCE CHARACTERISTICS.

Table 3-1 gives the performance characteristics of the ten underwater winches reviewed in the 1972 survey, as well as the performance characteristics of five additional underwater winches developed or under development since the 1972 survey. Table 3-2 lists the winch developers, and the user for whom the winch was developed.

The 1972 survey also presents a brief summary of lessons learned through those programs for design of a deep-water, long life winch. Table 3-3 provides a brief comparison of winch design characteristics of the functional components of the ten winches reviewed in 1972 along with the five winches developed since 1972.

3.6 NEW WINCH DEVELOPMENTS.

Since the 1972 survey, five submersible winches[4,5,6,7] have been developed or are under development. Four are state-of-the-art direct tension, conventional drum winches with the cable stored on the drum in either

TABLE 3-1. UNDERWATER WINCH PERFORMANCE CHARACTERISTICS

Name	CABLE Diameter (in.)	CABLE Length (ft)	RATED Depth (ft)	RATED Pull (lbs)	RATED Speed (ft/min)
CHAN	0.30	240	1000	3500 1000	75 150
AN/BRA-27	0.219	1000	1000	600	100
AN/BRA-8	0.35	classified	classified	2500	150
AN/BSQ-5 (BIAS)	0.35	1500(1)	1300	5500	60/180
VERSUS	0.95 2.25 x 32 long(2)	2000 1000	classified	2000	200
Makai	0.25	800	600	500	30
MARK II DDS	0.375	1200	1000	2500	15/40
NEMO	0.25	750	unlimited	1000	30/60
LOLITA	0.096	1200	1200	750	20
WEBS	0.50	1000	1000	6500	15
BOMIS I	0.50	3000	1300	3300(3)	50
BOMIS II	0.50	3000	600	3300(3)	50
OCEAN MASTER					
Mod. 0124-1	5/16	300	600	2000	50
0124-2	7/16	170	600	5000	26
SCAT	.35	1026(4)	1300	5600	75/150

(1) 300 ft is ribbon faired
(2) modules on 32-foot centers
(3) calculated maximum
(4) fully fiared with enclosed section fairing

TABLE 3-2. WINCH DEVELOPERS

WINCH	DEVELOPER	CUSTOMER
CHAN	NSRDC	
BRA-27	Union Carbide	NRL
BRA-8	ITT	NAVSEC
BIAS	Union Carbide	NRL
VERSUS	Western Gear	Westinghouse
Makai	ACCO	
Mark II	DORTEC	Navy
NEMO	Southwest Research	NCEL
LOLITA	Ocean Science Eng.	NAVAIR
WEBS	Ocean Science Eng.	
BOMIS	Naval Torpedo Station	
SCAT	EG&G, WASC, INC./ Western Gear	DTNSRDC
Ocean Master	ACCO	

TABLE [illegible] CHARACTERISTICS

[illegible]	Drum	Power Drive	Level Wind	Mounting Frame Material	Bearings
[illegible]	[illegible] grooved [illegible] cast Alum[illegible]	AC motor [illegible] Final reduction [illegible] Nickel Copper	Acme Screw [illegible] Monel	Medium - Steel	Impregnated, Bronze Sleeve
[illegible]-2	Multi lay CRES Drum with Alum[illegible]	[illegible] Motor in air [illegible], Gear & [illegible] reduction	Diamond Screw Hastalloy-C	Hot Rolled Steel	Fabriod Sleeve
SP[illegible]	Grooved Single La[illegible] (304) CRES	[illegible]raulic Motor [illegible] submarine	[illegible] Helix Follower	Not applicable	UHMW Polyethylene Sleeve
B[illegible]C-E [illegible]	Multi Lay [illegible] (6061-T6) [illegible], Hot Rolled steel	AC Motor [illegible] with planetary [illegible] and electro-disk brake [illegible]	Diamond Screw Hastall[illegible]-C	Hot Rolled Steel	Fabriod & UHMW poly. Sleeve
VER[illegible]	Multi lay, (316)	DC Motor conventional [illegible] in air filled housing, cone drive reducer	Automatic Two Speed	CRES	Conventional ball reverse seal oil-filled
MA[illegible]	Multi lay, steel (Epoxy paint finish)	AC Motor, gear reducer, mechanical brake	None	Nodular cast iron (Epoxy paint)	Delrin Sleeve
M[illegible] II	Multi lay, CRES	Electro hydraulic [illegible], planetary reducer [illegible]	None	CRES	Not available.
NEMO	Lebus grooved, multi lay, (316L) CRES	Battery, electro hydraulic, exposed 17-4-pH [illegible]	Lebus	(316L) CRES	Glass filled Teflon sleeve.
LOLITA	Double, pressure roller back tension [illegible] Aluminum.	Battery, electro mechanical in air, non compensated	None	Anodized Al.	Conventional sealed, steel ball
W[illegible]	Single, fleeting [illegible] pressure roller back tension [illegible]	Battery, electro mechanical in air, non compensated	None	CRES	Conventional sealed. steel ball
B[illegible]S, I, II	Multi Lay, steel (Zinc-rich paint finish)	AC motor and worm gear driven one step reduction oil filled pressure compensated	None	Mild steel (zinc rich paint finish	
Ocean Master	Multi Lay, Steel	AC motor, work gear reduction oil fill pressure compensated	Self-reversing Tight Lay Level wind	Steel	Ball or roller bearing sealed oil filled. Water lubricated bushings outside pressure housing
SCAT	Multi Drum Concentric Axially Transfersing grooved for fully enclosed sectional fairing (CRES.)	AC motor three stage ([illegible] spur and a planetary) reduction oil filled pressure compensated	Traversing drum	Steel	CRES Ball

single or multiple layers. The fifth winch is a direction tension winch having multiple storage drums on a common axis, thus being able to store multiple layers of faired cable. Each of the new winches will be discussed below.

3.7 BOMIS I WINCH.

The Bottom Mounted Instrumentation System (BOMIS)[4] winch was designed to raise a buoyant instrument package to the desired operating depth from the ocean floor at a depth of 1300 feet. The instrument package can be brought to the surface for calibration and refitted with new equipment. The BOMIS concept was conceived to avoid lowering the instrument string through the water surface.

BOMIS consists of a winch and instrument package installed on the ocean floor and electronics equipment located ashore; these components are connected by underwater cables. The coaxial instrument cable and the three-phase power cable from shore are separate. The winch and cables were designed for a maintenance free life of 5 years.

The winch is a conventional single drum design. For maximum reliability, the number of moving parts was minimized. The drum is horizontally mounted and 8 inches wide. The cable guide point is 10 feet above the drum which makes the lead angle sufficiently small that a level wind mechanism is unnecessary.

The winch is powered by a 1200 rpm, five horsepower, three-phase, 60 hertz, 440-volt AC motor. A 200:1 worm-gear reducer drives the winch drum. Power is transmitted at 4,160 volts to a step down transformer in the winch housing. The gears, motor, and transformer are all in a temperature-pressure compensated oil filled housing. A coaxial slip ring is used to transfer electrical signals to conductors in the cable on the winch drum.

All construction is mild steel. Corrosion protection is accomplished with zinc-rich paint and zinc anodes. A second winch, BOMIS II, located at a depth of 600 feet, is of the same design. The BOMIS winch configuration is shown in Figure 3-1.

3.8 OCEAN MASTER.

The Ocean Master[5] underwater winch model 0124-1 is built from standard components, which include worm gears; spline-connected shafts and couplings; self-reversing, tight-lay level wind; and a temperature and pressure compensator.

The winch is a conventional single drum design. The drum is horizontally mounted, fabricated from mild steel, and finished with epoxy paint.

The input is 5 horsepower with a single gear reduction. Shafts in the gear housing are supported on both ends by precision ball or roller bearings. Water lubricated bushings are used outside the housing. The winch can be used for applications requiring deep submergence.

The Ocean Master underwater winch model 0124-2 is the same winch as Ocean Master model 0124-1 except that it has a double gear reduction giving a 5000 pound line pull at 26 feet per minute.

3.9 SCAT WINCH.

The Submarine, Concentric, Axially Traversing (SCAT) winch[6,7] is a new and unique design for high density storage of a faired towline to control a submarine towed communication buoy. The SCAT winch meets the requirements for handling a cable faired with fully enclosed sectional fairing, is submersible, operates by remote control, and is highly compact. The concept is based on multiple drums, concentrically nested on a common shaft, and traversing axially along the shaft for level winding. (Figures 3-2 and 3-3).

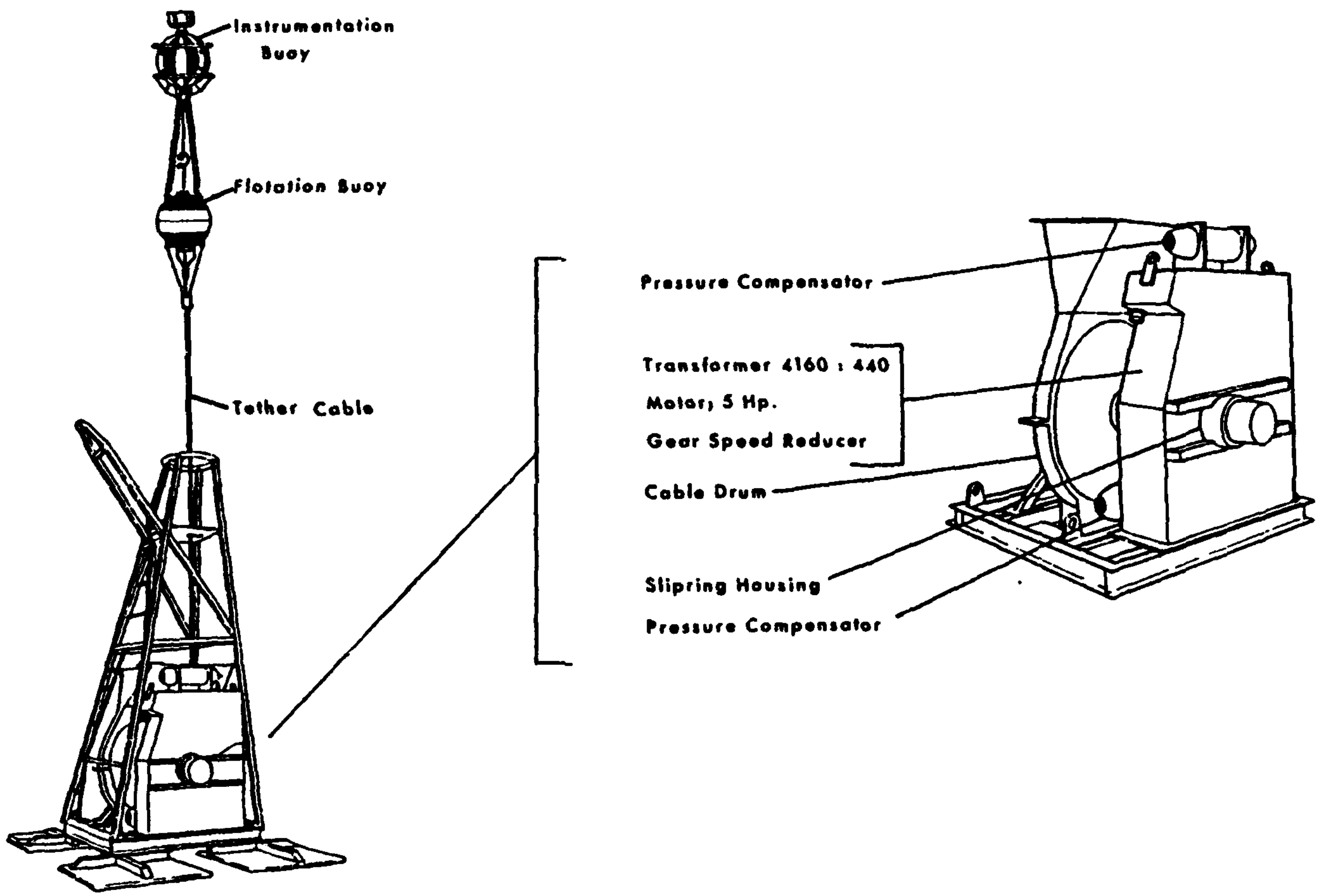

Figure 3-1. BOMIS Winch Configuration

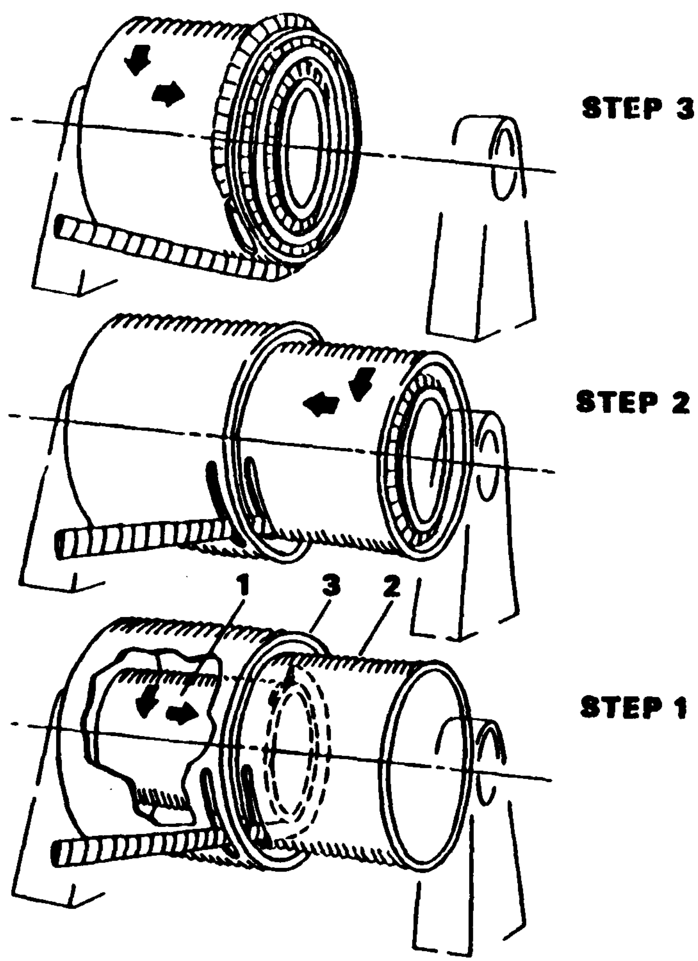

Figure 3-2. SCAT Winch

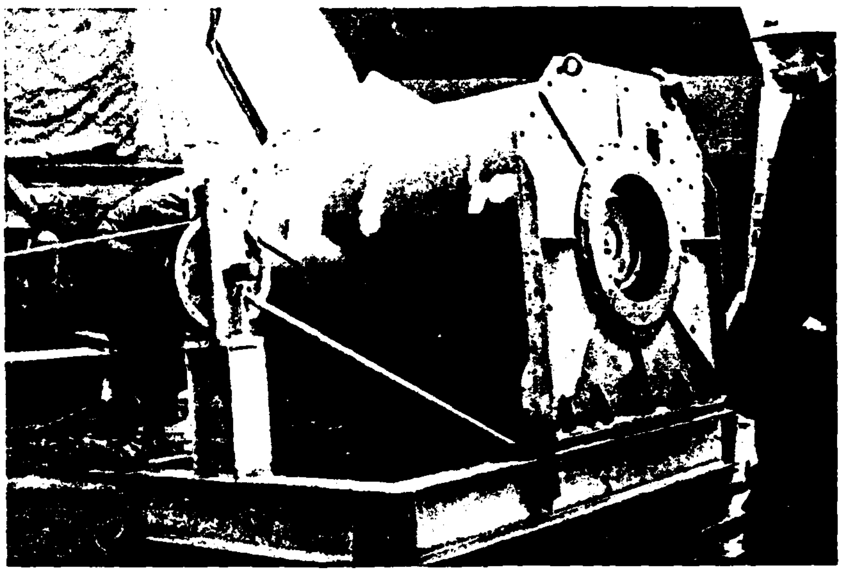

Figure 3-3. SCAT Winch Under Load Test

The drums are grooved internally and externally, direct tension, horizontal, and nested on a common shaft. The drums store electromechanical cable faired with a fairing whose nose width is .5-inch and cord length is 2.50 inches. Drum flange inserts with threads matching the internal drum threads provide the drum fleeting mechanism.

Mounted within the central shaft tube is a tubular housing enclosing the drive system, consisting of a 20/40 horsepower 440 volt, three phase, 60 hertz, 900/1800 rpm, submersible electric motor; planetary gear reducer; and spring set, multiple disc magnetic brake. All these components are oil filled and provided with a spring loaded pressure compensation system. The tubular drive housing is mounted rigidly to the right side winch frame. The reducer output is coupled to the main drive tube. The entire drum system rotates about a fixed drive housing within a fixed outer shell.

The winch has successfully undergone both land based and shallow underwater tests. These tests have proven the functional feasibility of the winch.

SECTION IV

APPROACH

4.1 GENERAL.

Cable hauling devices fall into one of two classes. Direct tension winches apply tension to the cable by pulling directly on the end of the cable, which is secured to the winch drum. Slip rings are used if access to conductors in the cable is required. Direct tension haulers merge the storage and hauling functions into a single part: the drum.

Indirect tension devices grip the sides of the cable by friction. They do not store the cable, but merely haul cable in at one point and expel it at another. Indirect tension haulers cannot use slip rings. If storage is required, a separate storage device must be supplied. Some indirect tension devices grip the cable by various clamping mechanisms; others require tension in the expelled cable to produce friction. The simplest indirect tension device, the capstan, is in the latter category.

The direct tension drum winch and the indirect tension capstan are discussed in this section, based on the performance characteristics detailed in Section II. Details of the analyses are presented in appendices as referenced.

4.2 POWER TRANSMISSION.

Power to operate the cable hauler might be obtained by conversion of on site energy sources, such as solar, wind, wave, tide, current, or nuclear energy. Since the site is only three miles offshore, none of these "exotic" techniques can compete economically with energy transmitted from shore.

Of the two most common power transmission methods, fuel pipelines and electrical conductors, the latter is simpler to install and the

energy transmitted is easier to convert and control. Transmitting electrical power three miles offshore does not press the state-of-the-art nor seriously limit the power available for the winch. A few hundred horsepower could be supplied through a cable* having three #8 AWG conductors carrying 5000 VAC from shore to a 10:1 transformer supplying 500 VAC to the hauler motor. By way of comparison, submersible motors are commercially available up to 25 Hp, and special motors have been built as large as 40 Hp.

4.3 <u>ENERGY STORAGE.</u>

It is shown in Appendix A that nearly 120 horsepower are required to lower the LINEAR CHAIR array and buoys 120 feet in 12 seconds. This short duration peak can be transmitted through the power cable, but would require a four-motor drive train: one 5 to 10 Hp motor clutched and geared for normal depth adjustment at 0.5 feet/second, and three 40 Hp motors clutched and geared together for emergency draw down. Separate motors for normal and emergency operation are required because of the large ratio (20:1) of emergency to normal speed.

The limiting factor is not the rate of transmission of electrical energy to the site; it is the rate at which the transmitted energy can be converted into mechanical energy. Since the 120 Hp rate need be sustained for only 12 seconds, it is feasible to store the energy at the site in a form readily converted to mechanical energy. Three energy storage concepts are addressed in the paragraphs that follow. A fourth, using batteries to store energy electrochemically, is very commonly used, but does not escape the problem of converting electrical energy at a 120 Hp rate. The three energy storage principles to be evaluated are:

- counter-weights using gravity

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* as used for the BOMIS System

- hydraulic accumulators using compressed gas, and
- flywheels using rotational inertia.

4.4 SINGLE-LEG ARRAY.

Figure 4-1 shows the LINEAR CHAIR array stretched vertically between an upper and a lower subsurface buoy. The mooring cable passes through an optional in-line counter-weight to the adjusting winch attached to the anchor on the sea floor.

4.4.1 COUNTER-WEIGHT ENERGY STORAGE.

It is shown in Appendix A that about 89% of the energy required for emergency haul-down and 99.95 percent of the normal haul-down energy is absorbed as potential energy in the two buoys. The counter-weight is, therefore, the most direct way to supply that energy. However, the counter-weight must be attached to the mooring line high enough to allow the array to be pulled to the maximum depth. Since the portion of the mooring below the weight is nearly slack, the array depth would be very sensitive to ocean current variation. Since the LINEAR CHAIR array has a design goal of movements of less than a yard, the counter-weight for a single-leg installation is not suitable.

4.4.2 HYDRAULIC ACCUMULATION.

Figure 4-2 is a schematic diagram of an electro-hydraulic winch. It consists of a hydraulic pump powered by an electric motor, a winch powered by a hydraulic motor, and an accumulator. The accumulator contains gas under pressure. As hydraulic fluid is forced into the accumulator tank, the gas is compressed further, increasing the pressure. For normal positioning of the array, the electric motor would drive the pump which would drive the hydraulic motor directly through the control valve. Emergency haul-down is accomplished by discharging the fluid in the accumulator through the control valve to the hydraulic motor. After the haul-down is complete, the electric motor and pump are coupled through the control valve to the accumulator to recharge it. Figure 4-3 is a sketch of a potential hardware layout.

5960-2

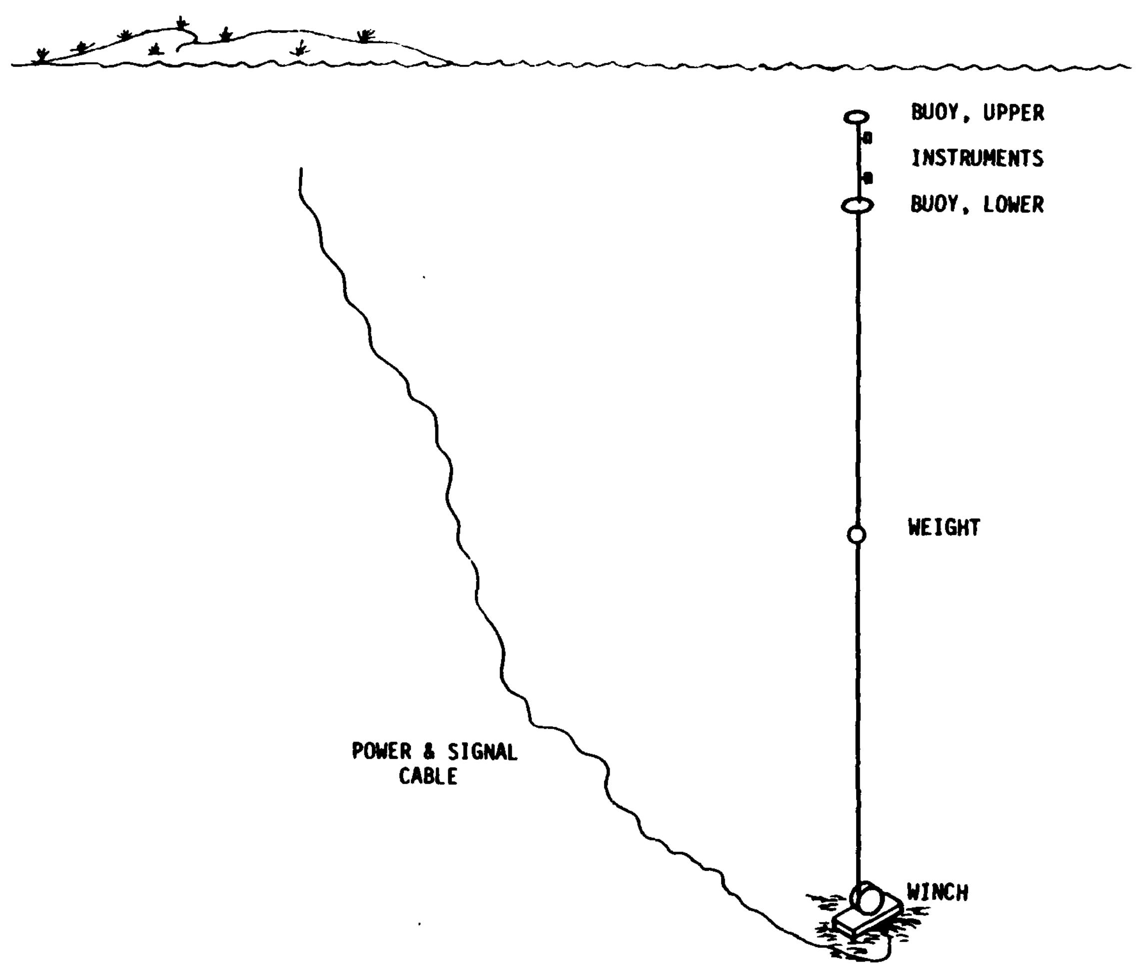

Figure 4-1. Single-Leg Array With In-Line Weight

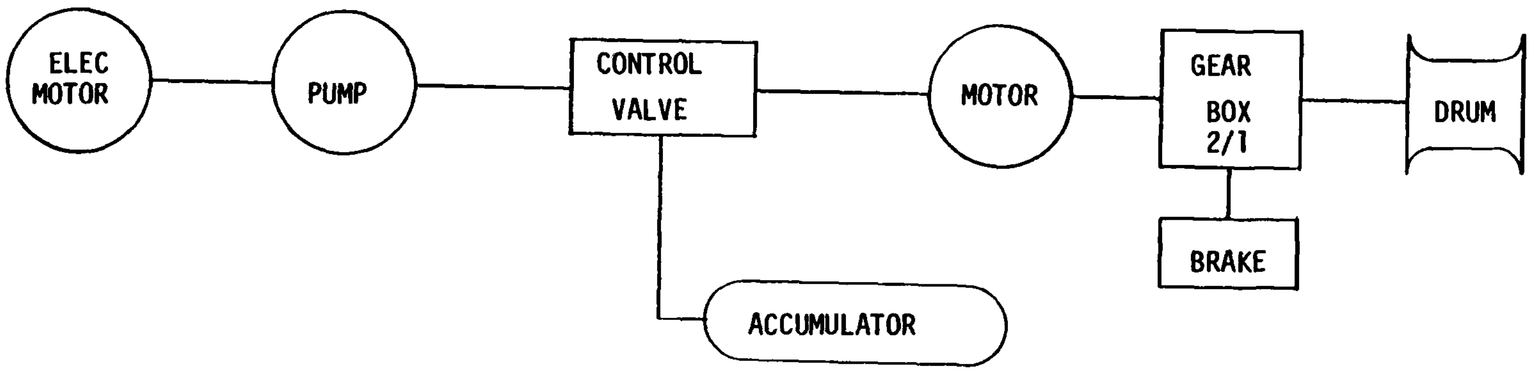

Figure 4-2. Electro Hydraulic Winch Schematic

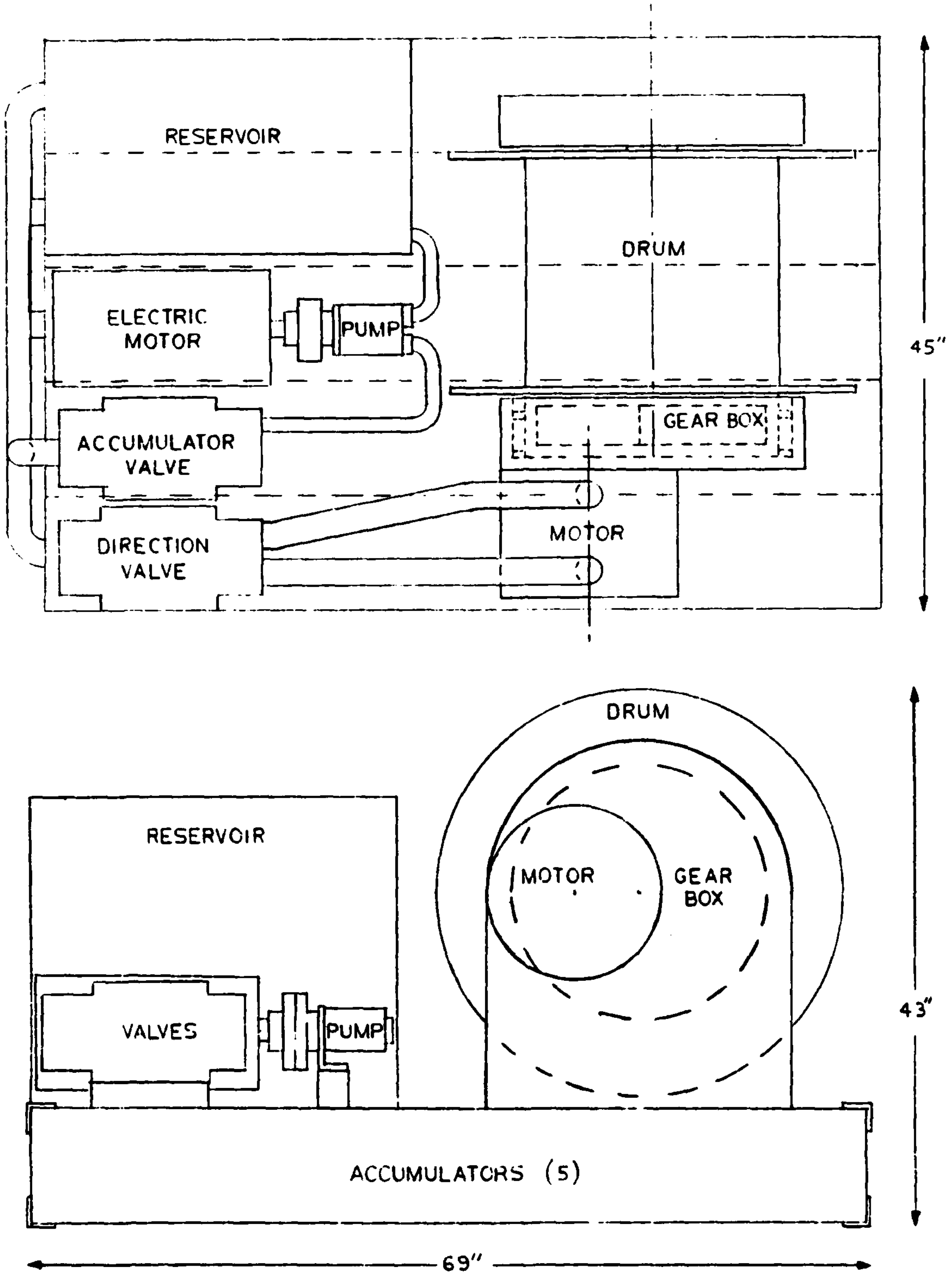

Figure 4-3. Electro-Hydraulic Winch Arrangement

It is shown in Appendix B that adding a second hydraulic motor to the gear box allows the accumulator volume to be substantially reduced; from 378 gallons to 105 gallons. This is because the second motor allows the accumulator to discharge to a lower pressure while maintaining the same line tension. The emergency flow velocity in the motor supply pipes is also reduced by a factor of 7 using two motors.

The apparently simple expedient of doubling the gear ratio would reduce the pressure needed during emergency haul-down too, but would make the drum turn too slowly for normal positioning. A two-speed electric motor would then be needed; slow speed, high torque for charging the accumulator, and high speed, less torque for normal positioning.

4.4.3 FLYWHEEL ROTATIONAL INERTIA. Flywheels store energy mechanically in their rotational inertia. Three factors are shown in Appendix A to dominate the design tradeoff for the LINEAR CHAIR mooring: rotor weight, windage, and gearing. Windage is the power required to maintain rotor speed against drag. Although the emergency haul-down can be accomplished by very small rotors - 100 pounds or less using high strength steel, windage losses are prohibitive unless the rotor is in a near vacuum. Furthermore, bearings, clutches, and gears for spin up and haul-down become major design problems. When the rotor speed is reduced to one or two times the normal electric motor speed, and windage in air kept within a few horsepower, the rotor weight required is measured in thousands of pounds. Windage losses for a rotor exposed to the sea are totally prohibitive - thousands of horsepower. Even in air at 1 atmosphere, it is necessary to have a housing that closely conforms to the rotor. A rotor in a large housing loses nearly twice the energy as a rotor in a conformal housing because it pumps air along the axle and expels it radially.

Special gearing and clutches are required for a flywheel to allow for smooth engagement of the emergency drawdown load without dissipating excessive flywheel inertia in slippage. The spin up motor must be of special high slip design or be fitted with a torque converter.

Special care will be required during installation to avoid bumps and shocks, lest the massive rotor bend its shaft or damage its bearing. Deployment impact with the sea bottom is a particular instance.

Overall, the problems associated with flywheels for LINEAR CHAIR are judged to outweigh the benefits.

4.5 MULTI-LEG ARRAY.

Figures 4-4 and 4-5 show two multi-leg array configurations - tri-moor and goal post.

4.5.1 WINCH. Depth adjustment could be effected by winches mounted on the anchors as discussed for the single-leg array. But in the multi-leg array configurations, these winches would be inefficient, since they would have to draw down not only the LINEAR CHAIR array, but also a large apex buoy. Furthermore, the line of action of the winch tension would operate at an angle to the vertical buoy force reducing its effectiveness. This would mean larger winch forces and greater lengths of cable to be stored on the drum.

The multi-leg configurations permit a much more efficient solution: mounting the depth adjustment winch on the apex buoy structure. In this case, the winch load does not include the apex buoy.

Since the two configurations differ only in the means of securing the apex buoy position, their winching techniques can be discussed together. In the goal post configuration, two winches would be required, since it is equivalent to two tri-moorings.

If a drum winch is used at the apex buoy, then the discussion of the single-leg mooring winch applies to the multi-leg cases as well. But, the major benefit, of the multi-leg moorings from an array handling standpoint, is that they provide a space beneath the cable handler for a counter-weight. The principal disadvantage, to mounting the winch at the apex buoy, is that the winch power cable must come up one of the mooring legs.

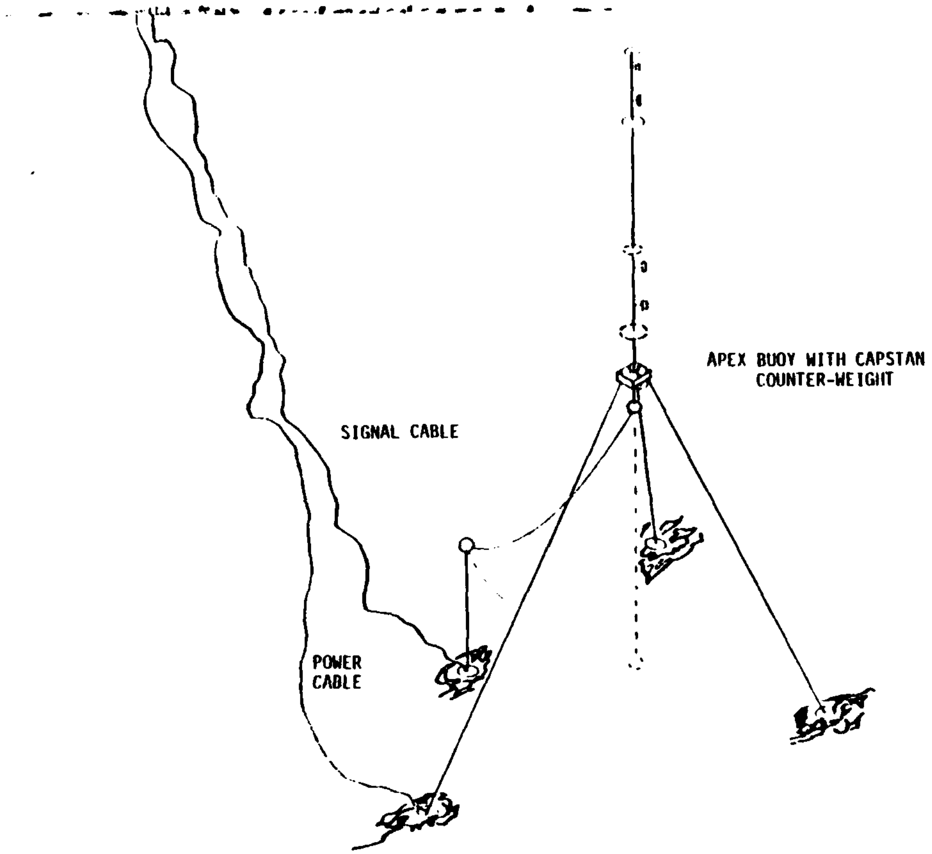

Figure 4-4. Tri-Moor with Counter-Weighted Capstan

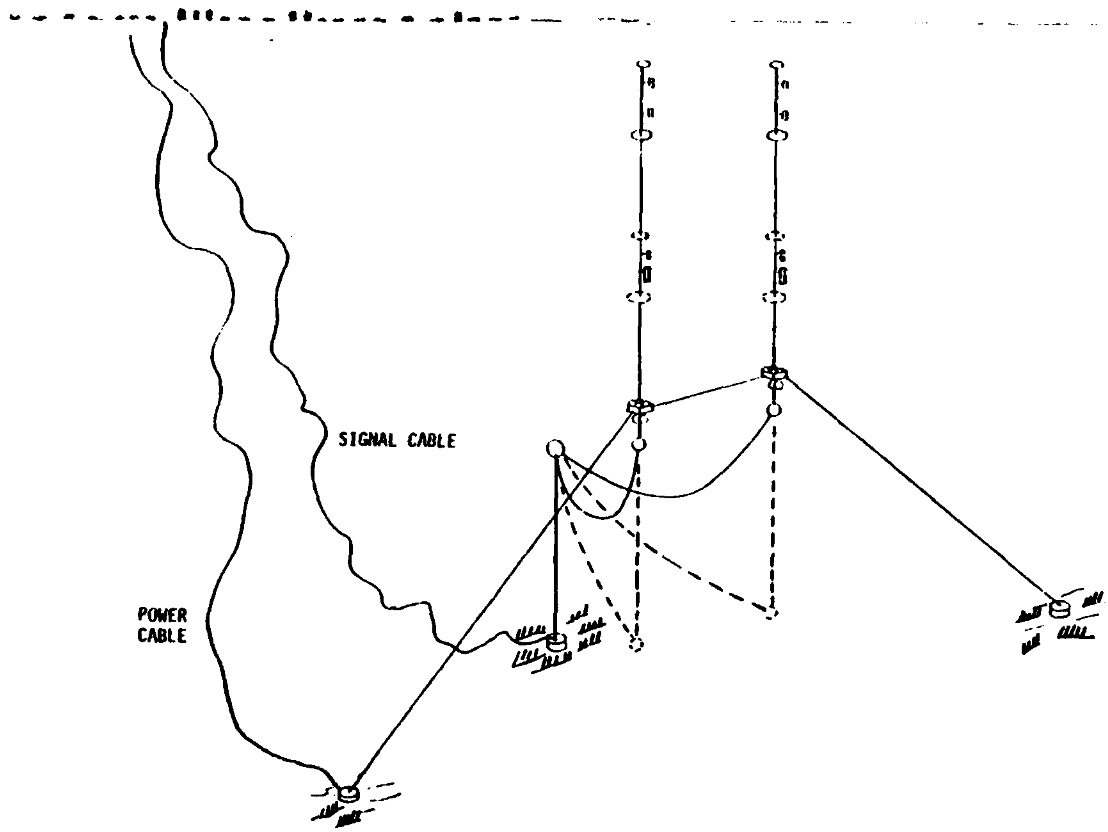

Figure 4-5. Goal Post with Counter-Weighted Capstan

4.5.2 CAPSTAN. There is no need for slip rings to maintain conductor continuity, since the cable does not end at the capstan as it must with a winch. Provision must be made, however, to keep the cable centered on the capstan. This is done by mounting grooved rollers around the periphery of the capstan. The axes of the rollers are essentially parallel to the axis of the capstan. A slight angle may be imparted to help the rollers fleet the cable along the capstan. If this axial slippage does not occur, the cable could creep off the end of the wheel like a nut spinning from a bolt. The grooved rollers also prevent a slack cable from slipping off the end of the capstan as shown on Figure 4-6.

Three cases are considered, depending on counter-weight size. A 500 lb counter-weight provides back tension to the capstan to prevent slippage. A 4000 lb counter-weight balances the buoyancy of the LINEAR CHAIR array so that the static torque on the capstan brake is essentially zero. Finally, a 4000 lb counter-weight with the array buoyancy reduced to about 3000 lbs uses the capstan power to raise the array instead of lowering it. These three cases are discussed below.

4.5.3 500 LB. COUNTER-WEIGHT. With a 500 lb counter-weight, the power needed for normal haul-down is reduced from about 5 to 4 horsepower, as shown in Appendix B. The counter-weight is too small to eliminate the need for an energy discharge device for emergency drawdown (Figure 4-6).

Using the hydraulic accumulator required for the single-leg winch as a basis for comparison, it is shown in Appendix B that the 500 lb counter-weight significantly reduces the required accumulator volume. If the capstan is driven by a single hydraulic motor, the required accumulator volume is reduced from 378 gallons to 86 gallons by the counter-weight. If a second hydraulic motor is engaged during emergency haul-down, the total accumulator volume with a counter-weight is only 64 gallons.

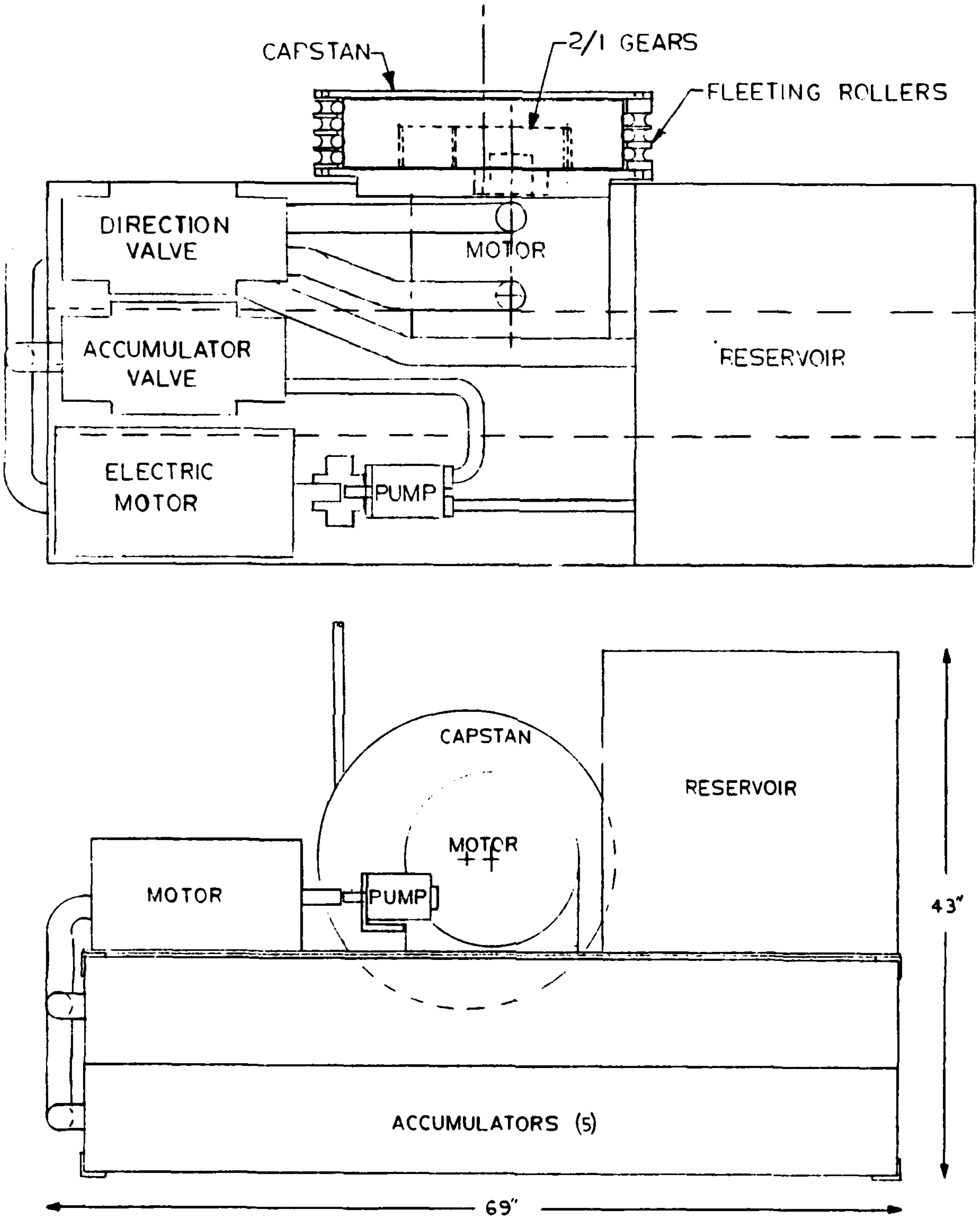

Figure 4-6. Electro Hydraulic Capstan

4.5.4 <u>4000 LB. COUNTER-WEIGHT.</u> A 4000 lb counter-weight balances the array buoyancy at a cable safety factor of 5. In this case, the power required for normal positioning is very small, so that a fractional horsepower electric motor is adequate. A 30 horsepower motor, separately geared for a line speed of about 13 ft/sec, can perform the emergency haul-down in 12 seconds. The 4000 lb counter-weight makes an electro-mechanical drive possible.

It is shown in Appendix B that a 32 gallon accumulator, using a single hydraulic motor, will perform the emergency drawdown. Since the motor selected for analysis in Appendix B is substantially oversized for the 4000 lb counter-weight loads, a smaller accumulator can probably be used.

4.5.5 <u>4000 LB. COUNTER-WEIGHT WITH FREE FALL.</u> If the LINEAR CHAIR array buoys are reduced from 2000 lb net buoyancy to about 1500 lbs each, the combined array plus counter-weight will be about 1000 lbs heavy in sea-water. Normal positioning of the array requires about 1 horsepower. It is shown in Appendix B that by releasing the brake and de-clutching the capstan from its drive gears, the array will free-fall 120 feet in 12 seconds. Figure 4-7 shows an arrangement of an electrically powered capstan for this case.

This arrangement would seem to provide the most reliable emergency operation. However, the penalty is that the array has less vertical stability under the effects of current variation because of the reduced buoyancy. Whether this is a serious penalty is beyond the scope of this report. A possible solution is to retain the original buoy size, but use a larger counter-weight. In this case, the cable safety factor is reduced to about 3.5.

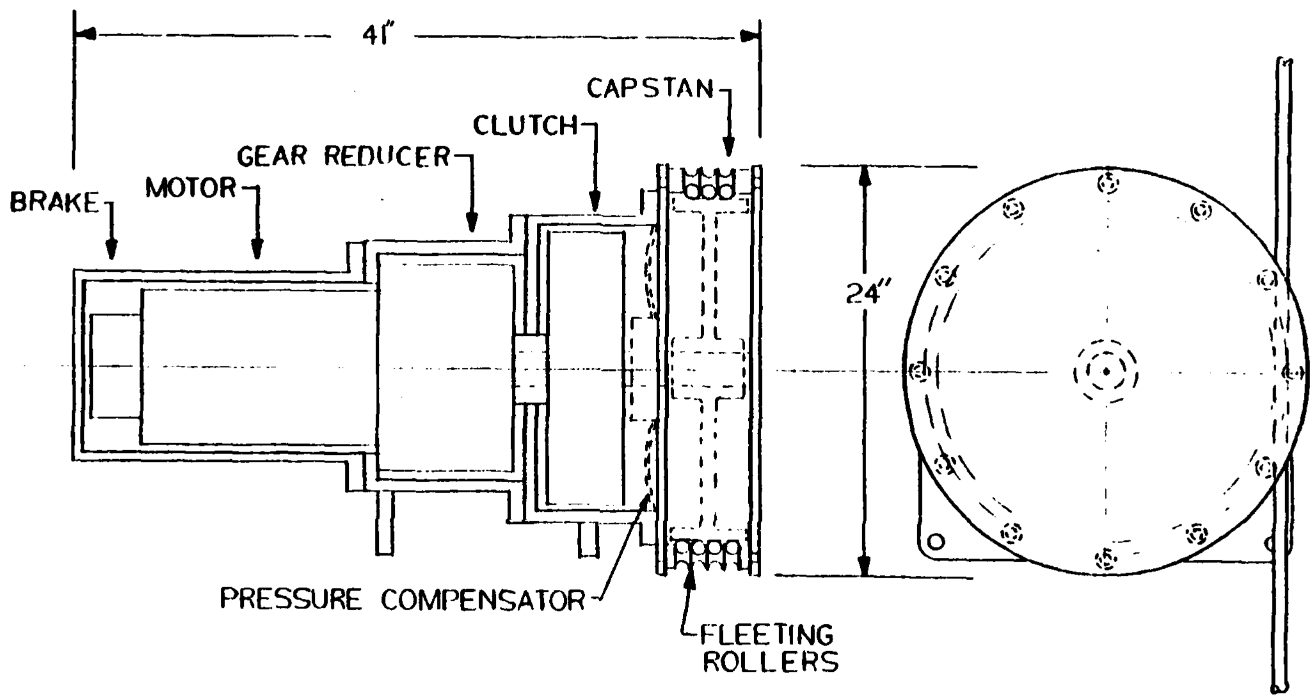

Figure 4-7. Counter-Weighted Capstan

SECTION V

TRADEOFFS

Figure 5-1 presents a matrix of the selection criteria versus emergency energy sources. Definitions of the terms used in the tradeoff matrix, for each of the criteria, are presented below:

PHYSICAL CHARACTERISTICS (SIZE)

Small - individual components displace less than 10 cubic feet

Medium - individual components displace between 10-35 cubic feet

Big - individual components displace more than 35 cubic feet

WEIGHT

Heavy - individual components weigh more than 1 ton

Light - individual components weigh less than 1 ton

POWER REQUIREMENTS

High - no energy stored on site. Buoyant forces not counterbalanced.

Medium - buoyant forces not counterbalanced

Low - buoyant forces counterbalanced

CRITICAL / EMERGENCY ENERGY SOURCE	WINCH			CAPSTAN				
CRITERIA	ELECTRIC	HYDRAULIC	FLYWHEEL	HYDRAULIC	GRAVITY	FLYWHEEL AND GRAVITY	HYDRAULIC AND GRAVITY	COMMENTS
PHYSICAL CHARACTERISTICS (SIZE)	B	B	B	B	S	M	M	S = Small M = Medium B = Big
WEIGHT	L	H	H	H	H	H	H	H = Heavy L = Light
POWER REQUIREMENTS	H	M	M	M	L	M	L	H = High M = Medium L = Low
RELIABILITY	H	M	L	M	H	L	M	H = Highest M = Moderate L = Lowest
COMPONENT AVAILABILITY	D	S	D	S	O	D	O	O = Off-the-shelf S = State-of-the-Art D = Developmental
LEAD TIME	L	M	L	M	S	L	M	S = <6 months M = 6 months - 1 year L = >1 year
DEPLOYABILITY	D	M	D	M	E	D	E	D = Difficult M = Moderate E = Easy
EXPERIENCE	N	S	N	S	S	N	S	N = None S = Some E = Extensive
COST	H	M	H	M	L	H	M	H = High M = Moderate L = Low

Figure 5-1. Tradeoff Matrix - Selection Criteria Versus Emergency Energy Sources

RELIABILITY

Highest - all electric or simple gravity system
Moderate - hydraulic and electric system
Lowest - flywheel always spinning, even in a static environment. The flywheel must be in a vacuum or air chamber.

COMPONENT AVAILABILITY

Off-the-Shelf - made from parts that are commercially available - can be ordered from a catalog
State-of-the-Art - parts would have to be specially ordered and designed
Developmental - would require development before manufacture

LEAD TIME

S (6 months) - Off-the-shelf items
M (6 mo. - 1 yr.) - State-of-the-Art items
L (>1 yr.) - Developmental items

DEPLOYABILITY

Difficult - large masses on bearings
Moderate - rotating items of moderate size
Easy - heavy items with no moving parts

EXPERIENCE

None - reflects developmental items
Some - been accomplished before
Extensive - (See Section II)

COST

High - developmental costs
Moderate - requires both electric and hydraulic
Low - mechanically simple

Based on the tradeoff analysis, the following appear the most promising candidates for further investigation:

a. The hydraulic winch for the single-leg configuration

b. For the multi-leg configuration, listed in order of preference:

(1) Gravity Capstan
(2) Hydraulic and Gravity Capstan
(3) Hydraulic Capstan

SECTION VI

DEVELOPMENT PLAN

6.1 MILESTONES.

Figure 6-1 presents the milestones for the development of the underwater winch system.

6.2 COST AND WEIGHT ESTIMATES.

Preliminary weight and cost estimates to design and fabricate the attractive winch approaches for the fixed LINEAR CHAIR array are presented below. Costs are predicated upon concept definition being completed and a detailed specification being provided. No estimates are made for critical component investigations or testing.

APPROACH	WEIGHT (POUNDS)	COST (1000 dollars)	
		Design	Fabrication
Winch - Hydraulic	17,000	30	65
Capstan			
Hydraulic	12,000	30	65
Gravity	5,000	15	35
Hydraulic & Gravity	8,000	30	50

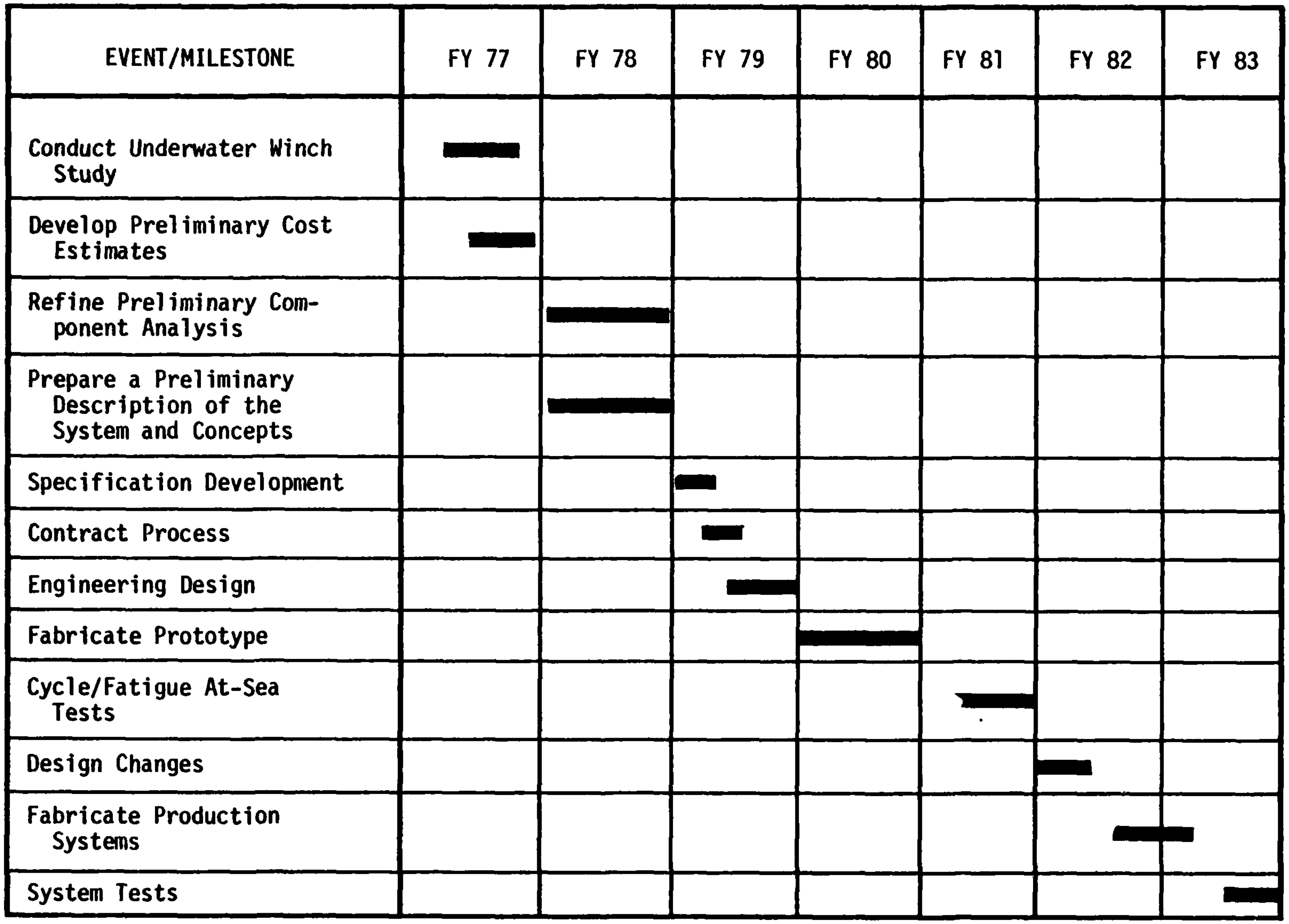

Figure 6-1. Underwater Winch System Development Milestones

In addition to the cost for the winch, the cost of equipment (i.e., power cables, transformer, controller, etc.) for transmitting the power to the winch three miles offshore must be added. It is estimated that this equipment will cost \$90,000 of which the cable accounts for \$80,000. An additional \$10,000 should be included for spare parts in order to minimize equipment down time. Due to the constraints of this study, the additional cost of array installation, caused by the inclusion of the winch, was not developed. This should be determined in the succeeding phase of the LINEAR CHAIR Project as the concepts are identified.

SECTION VII

CONCLUSIONS AND RECOMMENDATIONS

None of the existing winches reviewed will meet the requirements of the LINEAR CHAIR Project Fixed Array. However, the state-of-the-art has progressed to the point where a suitable winch could be designed and fabricated.

Several approaches analyzed in this study appear promising for positioning the array. The approaches are dependent upon array configuration, thus a single approach cannot be recommended until an array configuration has been established. However, the prime candidates are the hydraulic accumulation winch for the single-leg configuration and the gravity capstan for the multi-leg configurations.

Refinement of the capstans and the hydraulic accumulation winch approaches in conjunction with the array configuration development process is recommended. Also, the components (i.e., power cable, transformer, controller, etc.) required to support the winch must be identified, as well as the impact upon installation imposed by including the winch in the array.

SECTION VIII

REFERENCES

1. Gibson, Philip T. and Jacobs, Ralph P. "Bending Fatigue Evaluation of Kevlar Rope to RCA Services Company", Battelle Columbus Laboratories, 14 February 1975.
2. "Design Guide for Selection and Specification of Kevlar Rope for Ocean Engineering and Construction", July 1976, Project Order N-62477-75-P05-0021. Developed for Chesapeake Division, Naval Facilities Engineering Command, Ocean Engineering and Construction Project Office.
3. Bonde, Leslie W. and Shelly, Philip E., "A Survey of Underwater Winches", Hydrospace Research Corporation Technical Report No. 348, 31 March 1972.
4. "Bottom Mounted Instrumentation System", Naval Torpedo Station, Keyport, Washington.
5. Information Sheet for Underwater Winch Model 0124, ACCO Equipment Division, York, Pennsylvania.
6. Bonde, Leslie W., "A Compact Submersible Winch for a Faired Towline", Development Report Phase I and II, Hydrospace Challenger, Inc., TR-43710001, June 1973.
7. Bonde, Leslie W., "A Compact Submersible Winch for a Faired Towline", Development Report Phase III and IV, TR 43710002, January 1975.

APPENDIX A

SINGLE-LEG CONCEPT

APPENDIX A

SINGLE-LEG CONCEPT

This concept, Figure 4-1, represents the most direct approach. The LINEAR CHAIR array is suspended vertically between upper and lower buoys which are moored by a single leg to a winch fixed to an anchor. The depth of the array is controlled by the winch. In an emergency, the depth must be increased by 120 feet within 12 seconds. Pertinent fixed parameters are summarized in Table A-1.

A.1 BUOYANCY.

For extended deployment, the cable should not be highly stressed; the combined buoyancy of the suspension buoys should not exceed 20 percent of the breaking strength of the cable. If the buoys are of equal size, then

$$B = T_B/10 = 2,000 \text{ lbs.}$$

Spherical syntactic foam buoys are simple and easy to build. The buoys are moored in moderate depths, so 30 lb/cu ft foam will provide crush depth in excess of the water depth. The volume of foam required for each buoy is:

$$V = B/(\gamma_w - \gamma_f) = 2000/(64 - 30) = 58.8 \text{ cubic feet.}$$

TABLE A-1. LINEAR CHAIR PARAMETERS

Given Values

Seawater density	1.99 slugs/cuft. @ 59F
Kinematic viscosity	1.28×10^{-5} ft^2/sec @ 59F
Cable diameter	0.7 inches
length	1250 feet
strength	20,000 pounds
material	Kevlar, smooth jacket
Adjusting speed	0.5 feet/second
Emergency adjustment	120 feet
Emergency time	12 seconds

Assumed Values	Value	Justification
Cable safety factor	5	accepted value for extended use
Upper/lower buoy size ratio	1	interchangeable
Buoy shape	sphere	simple mold, moderate drag
Buoy construction	syntactic foam	simple manufacture
Foam density	30 lb/cu ft	pressure
Buoy drag coefficient	0.11	turbulent flow Reynolds No.
Winch efficiency	0.7	

For a spherical buoy, the radius is

$$r = \left(\frac{3V}{4\pi}\right)^{1/3} = \sqrt[3]{14.04} = 2.41 \text{ feet;}$$

the diameter is

$$d = 2r = 4.83 \text{ feet.}$$

In air, the buoys weigh $W = \gamma \cdot V = 30 \cdot 58.8 = 1765$ lbs.

A.2 HYDRODYNAMIC DRAG.

As the winch hauls the array down, it is resisted not only by the buoyancy of the spheres, but also by their drag combined with the drag of the cable being pulled through the water. The drag depends on the Reynolds Number, UL/ν. For the buoys under normal winching,

$$R_{NB} = \frac{.5 \times 4.83}{1.28 \times 10^{-5}} = 1.89 \times 10^5,$$

and in a "crash dive" $R_{NE} = 20 \cdot R_{NB} = 3.77 \times 10^6$. These values are above the turbulent transition for spheres, so that $C_D = .11$ may be used.[1] The drag of each buoy is thus

$$D_B = \frac{1}{2} PC_D (\pi r^2) \ U^2$$

$$= \frac{1}{2} \times 1.99 \times .11 \times (\pi \cdot 2.41^2) \ U^2 = 2.00 \ U^2.$$

The Reynolds Number of the cable, referenced to its diameter is, for normal winching,

$$R_{NC} = \frac{.5 \times .7/12}{1.28 \times 10^{-5}} = 2.28 \times 10^3,$$

[1] Hoerner, Fluid Dynamic Drag, published by the author, p. 3-8.

and at 10 ft/sec,

$$R_{NC} = 4.56 \times 10^4.$$

The drag coefficient of a smooth, jacketed cable is[2]

$$C_t = 0.006,$$

based on wetted circumference. Thus, the cable drag is

$$\begin{aligned} Dc &= \frac{1}{2} PC_t (\pi dL) U^2 \\ &= .5 \times 1.99 \times .006 (\pi \times .7/12 \times 1250) U^2 \\ &= 1.37 U^2. \end{aligned}$$

A.3 WINCH POWER.

The tractive force required of the haul-down winch is thus,

$$\begin{aligned} Fw &= 2 \times B + 2 \times 2.00 U^2 + 1.37 U^2, \\ &= 4000 + 5.37 U^2. \end{aligned}$$

The horsepower required is

$$Hp = \frac{F \times U}{550 \times \zeta} = \frac{(4000 + 5.37 U^2)U}{550 \times \zeta}.$$

Assuming the winch mechanism is 70 percent efficient, the normal power required is

$$Hp = \frac{(4000 + 5.37 (.5)^2) \times .5}{385} = 5.2,$$

but under the emergency haul-down,

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[2] Pattison, J.H., et al, "Handbook on Hydrodynamic Characteristics of Moored Array Components," DTNSRDC Report SPD-745-01, p. 3-40.

$$Hp = \frac{(4000 + 537) \times 10}{385} = 118.$$

The 5.2 horsepower required for normal winching may be conducted by underwater cable. The emergency haul-down power must be accumulated during quiescent periods for abrupt release upon demand. Furthermore, although 2-speed electric motors are available, the speed ratio required between emergency and normal conditions (20:1) is beyond the capacity of commercial submersible motors. The horsepower ratio, nearly 23:1, aggravates the issue. Emergency haul-down requires a special device for releasing stored energy through the winch.

A.4 ENERGY ACCUMULATION AND RELEASE.

Three methods for accumulating energy and releasing it to drive the winch for emergency hauldown are described below.

A.4.1 BATTERIES.

Batteries protected in an oil-filled chamber and charged through the undersea cable supplying the normal-speed motor power, might be discharged through a separate, high speed, high power motor for emergency hauldown.

The brushes of DC motors for direct coupling to the batteries must be protected from seawater, as by operating in oil. This has not been very reliable in past practice. The alternative is to invert the battery DC to AC electronically, so that a brushless AC motor can be used.

In either case the winch requires two complete transmission systems with an either/or clutch between them in order to allow the motors to operate at reasonable speeds.

In summary, battery accumulation is not judged practical for emergency hauldown.

A.4.2 FLYWHEEL INERTIA.

Energy may be stored in the angular momentum of a flywheel and released through appropriate gearing for emergency hauldown. A relatively low power electric motor, operating through a torque converter or variable-slip drive, is used to bring the flywheel up to speed and maintain it against windage.

The energy required for emergency hauldown is:

$$Er = F \times D = 4537 \times 120 = 544{,}440 \text{ Ft-lbs.}$$

The energy available from a flywheel is:

$$Ea = \frac{1}{2} I \omega^2 (1 - \lambda^2) \text{ , where}$$

I is the moment of inertia of the wheel,

ω is its initial rate of rotation, and

λ is the ratio of the final rate of rotation to ω.

For simplicity, consider a simple steel disk:

$$I = mr^2 = \rho V r^2 = \rho \pi r^4 t = \mu \rho \pi r^5,$$

where m is the flywheel mass,

r is its radius,

ρ is the mass density of the material,

V is the volume of material,

t is the disk thickness, and

μ is t/r.

Equating energy available and energy required, assuming a 70% transmission efficiency, η, gives:

$$r = \left\{ \frac{2 \times F \times D}{\eta \times \mu \times \pi \times \omega^2 \ -(1 - \lambda^2)} \right\}^{1/5}, \text{ or}$$

in terms of RPM, N,

$$r = \left\{ \frac{1800 \times F \times D}{\eta \times \pi \times N^2 \times \mu \times (1 - \lambda^2)} \right\}^{1/5}$$

Assuming the thickness is half the diameter gives

$$\mu = 1,$$

and assuming the flywheel slows by 50 percent during hauldown gives

$$\lambda = 0.5.$$

Further assuming that the flywheel speed is maintained by a direct drive AC motor at 1725 RPM, gives

$$r = \left\{ \frac{1800 \times 4537 \times 120 \qquad 1/5}{.7 \times \pi^3 \times 1725^2 \times 1 \times (1 - .5^2)} \right\}^{1/5}$$

$$= (20.2)^{1/5}.$$

Thus,

$$T = r = 1.82 \text{ ft},$$

and the flywheel weight is

$$W_f = g \cdot \rho \cdot \pi \, r^2 \, t = 480 \times \pi \times 1.82^3 = 9090 \text{ lb}.$$

The simple disk is a relatively inefficient shape for a flywheel. Advanced flywheel design using anisotropic fiber composites is not suitable for this application. An isotropic (i.e., solid steel) disk tapered such that the rim thickness is 2-percent of the axial thickness on a shaft whose diameter is 10 percent of the rotor diameter is shown (page[1] 5-20) to be within 4 percent of the optimum stress design.

Using a transmission efficiency of 70 percent, the 544,440 ft-lb energy required for emergency haul down becomes 777,770 ft-lb, or 293 watt-hours, of energy in the flywheel. Substituting this value in the formulas cited gives:

$$E_1 = \left(\frac{C_E}{C_W}\right)\left(\frac{W}{\rho_2}\right) \sigma_W, \text{ at the top rotation speed, and}$$

and

$$E_2 = \left(\frac{C_E}{C_W}\right)\left(\frac{W}{\rho}\right) \sigma, \text{ at the end of the haul down, where:}$$

where:

$$C_E = 9.9755 \times 10^{-6},$$

$$C_W = 0.38260,$$

---

1 "Economic and Technical Feasibility Study for Energy Storage Flywheels" ERDA report 76-65 UC 94B, prepared by Rockwell International, Space Division, Dec. 1975, section 5.2, p. 5-11 ff.

E is the kinetic energy in the rotor, watt-hr,

W is the rotor weight, lb,

ρ is the rotor density, lb/in^3,

σ_W is the working stress, psi and

σ_2 is the rotor stress at the end of the hauldown.

The rotor radius and spin rate are related by

$$\rho r^2 N_1^2 = C_s \quad \sigma_W \text{ , and}$$

$$\rho r^2 N_2^2 = C_s \quad \sigma_2 \text{ ,}$$

where:

r is the rotor radius (inches)

N_1 is the rotor RPM before hauldown,

N_2 is the rotor RPM after hauldown, and

$C_s = 1.8831 \times 10^5$.

Thus $\sigma_2 = \sigma_W \left(\frac{N_2}{N_1}\right)^2$, and the change in energy of the rotor during hauldown is:

$$E_1 - E_2 \doteq \Delta E = \left(\frac{C_E}{C_W}\right)\left(\frac{W}{\rho}\right) \sigma_W \left(1 - \left(\frac{N_2}{N_1}\right)^2\right) .$$

Solving for the weight gives:

$$W = \left(\frac{C_W}{C_E}\right)\left(\frac{\rho}{\sigma_W}\right)\Delta E \left\{ \frac{\left(\frac{N_1}{N_2}\right)^2}{\left(\frac{N_1}{N_2}\right)^2 - 1} \right\}$$

If $N_1 = 2N_2$, as before, then

$$W = \frac{\frac{.38260}{9.9755\times10^{6}} \times \frac{480}{1728} \times 293 \times \frac{4}{3}}{\sigma_W},$$

$$= \frac{4.162 \times 10^{6}}{\sigma_W}$$

which is plotted on Figure A- 1 (a). The rotor radius in feet is given by:

$$R(ft) = \frac{1}{N_1}\sqrt{\frac{12.C_s \cdot \sigma_W}{\rho}}$$

$$= \frac{68.6}{N_1}\sqrt{\sigma_W}$$

This is plotted on Figure A-1 (b) for several values of σ_W.

Comparing parts (a) and (b) of the figure shows that the haul down can be accomplished with rotors much lighter than the simple disk analysis showed. For example, at a working stress of 40,000 psi, the rotor weighs only 105 lbs (A-1 (a)). But if the rotor has 2-ft radius, it must spin nearly 7000 rpm (A-1 (b)). In order to moderate both the rotor diameter and spin rate, the rotor weight must be relaxed. Table A-2 shows the parameters of a flywheel operating at 3450 rpm.

TABLE A-2. EMERGENCY HAUL DOWN FLYWHEEL PARAMETERS

RPM, before hauldown	3450.
RPM, after hauldown	1725.
Diameter, feet	4.0
Weight, lbs	416.
Length, hub, inches	2.16

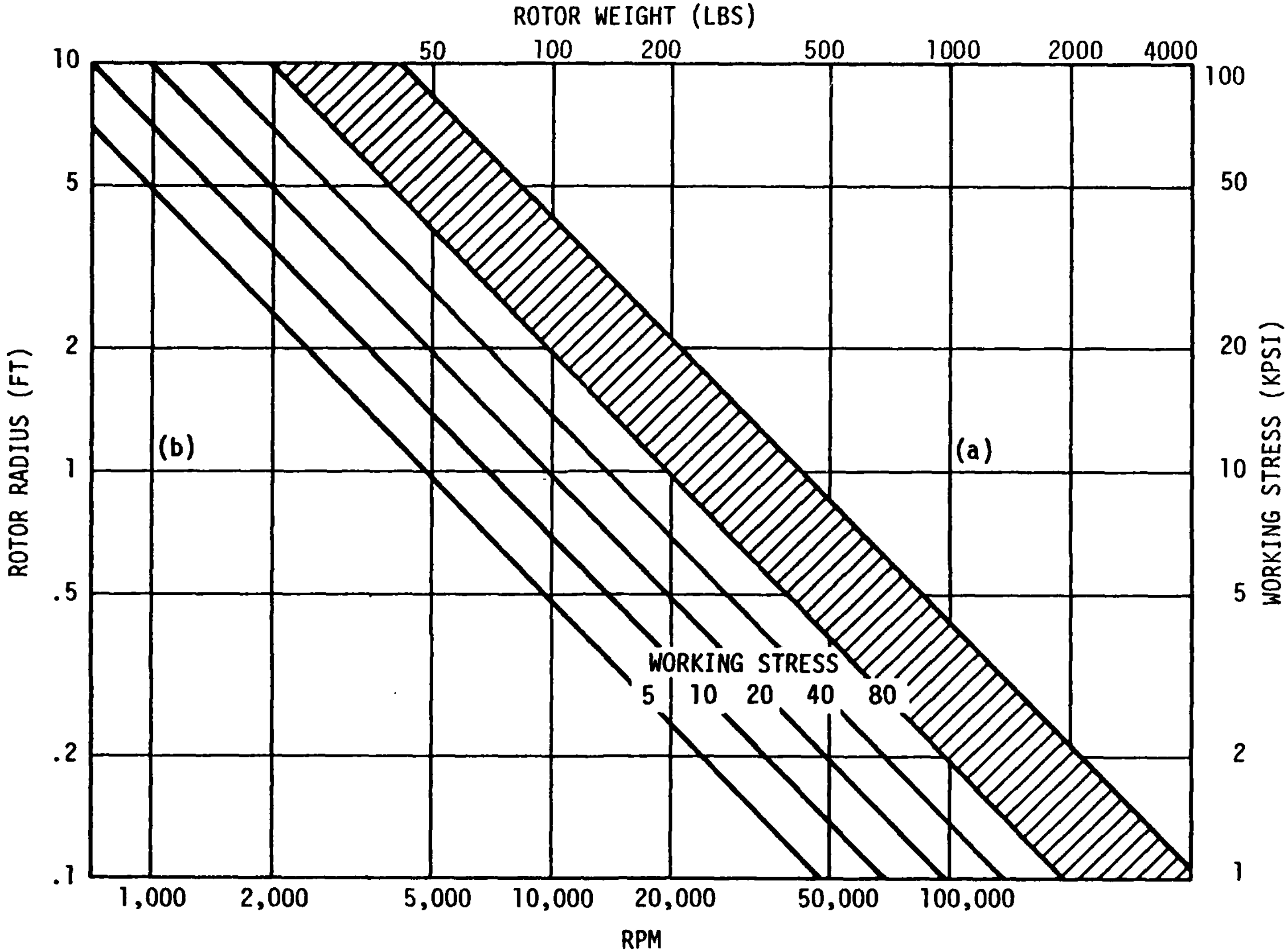

Figure A-1. LINEAR CHAIR Flywheel Performance

Table A-2 (cont'd)

Rim thickness, inches	0.043
Material	steel
Stress @ 3450 rpm, psi	10,000
Stress @ 1725 rpm, psi	2,500

Spin up time occurs in two parts. First is the time to spin up from zero to N_2 rpm; second is the additional time to accelerate from N_2 to N_1. Only the second time applies to recovery after an emergency. The spin up time is given by:

$$t = \frac{E}{\eta \times 33000 \times Hp},$$

where t is the spin up time in minutes,

E is the rotor energy in ft-lbs,

η is the spin-up effeciency, and

Hp is the horsepower available.

At N_2 rpm, $E_2 = \Delta E/3$ and N_1 rpm, $E_1 = 4\,\Delta E/3$. Thus

$$t_2 = \frac{\Delta E}{3 \times \eta \times 33000 \times Hp} = \frac{777,770}{3 \times .7 \times 33000 \times 5} = 2.24 \text{ min.},$$

and $\Delta t = 3t_2 = 6.73$ minutes. The total time to spin up from a dead start is $4t_2 = 9$ minutes. Note that the transmission efficiency affects spin-up time twice - the energy wasted during spin-up, and the added energy that must be available from the wheel to waste during hauldown. The spin-up transmission must be powered by a variable speed motor, be able to slip, or vary its effective ratio.

Similarly, the hauldown transmission must be able to engage the flywheel smoothly without dissipating its energy through slippage.

Finally, the question of the power required to sustain the flywheel at speed must be addressed.[1] Table A-3 summarizes the calculations for the horsepower losses on a 4-ft flywheel in air, with a close-fitting housing. Nineteen horsepower are required to sustain the rotor speed against viscous drag in the housing. If the housing does not conform to the rotor , 31 horsepower are required, because the flywheel becomes a centrifugal pump, sucking fluid along its axis and expelling it radially around the rim.

TABLE A-3. FLYWHEEL WINDAGE LOSSES

Parameter	Symbol	Formula	Value	Unit
Radius	R	-	2	feet
Speed	N	-	3450	RPM
Angular Velocity	ω	$\pi N/30$	115π	rad/sec
Density	ρ	-	.00205	$\text{lb-sec}^2/\text{ft}^4$
Kinematic Viscosity	ν	-	2.07×10^{-4}	ft^2/sec
Reynolds Number	Re	$\omega R^2/\nu$	6.98×10^6	-
Moment Coefficient	Cm	$1.17\times.0622\times Re^{-0.2}$	.00311	-

[1] Schlichting, H., "Boundary Layer Theory," 6th Ed., McGraw Hill Book Co., New York, 1968, pp 606-610.

Table A-3 (cont'd)

Parameter	Symbol	Formula	Value	Unit
Rotor Moment	M	$1/2\rho\ Cm\ \omega^2 R^5$	20.3	ft-lb
Rotor Power	H	$M\omega/550$	13.3	horsepower
Transmission efficiency	η	-	70	percent
Required Power	Hp	H/η	19.0	horsepower

It is apparent that the selected parameters (w = 416 lbs, R = 2 ft, and N = 3450 Rpm), while suitable for energy storage, require an unacceptable power to compensate for windage. These parameters may be traded off using the relations:

$$W_1\ R_1{}^2\ N_1{}^2 = W_2\ R_2{}^2\ N_2{}^2, \text{ and}$$

$$\frac{R_1{}^{4.6}\quad N_1{}^{2.8}}{Hp_1} = \frac{R_2{}^{4.6}\quad N_2{}^{2.8}}{Hp_1} .$$

Thus, if we reduce R from 2 to 1 foot radius, but retain N = 3450 rpm

$$\frac{R_2}{R_1} = 0.5,$$

$$W_2 = \left(\frac{R_1}{R_2}\right)^2 W_1 = 4 \cdot W_1 = 1632 \text{ lbs}$$

$$Hp_2 = \left(\frac{R_2}{R_1}\right)^{4.6} Hp_1 = (.5)^{4.6}\ (19) = 0.78 \text{ horsepower.}$$

Consideration of windage losses has forced the geometry to shift from a rotating disk to a rotating cylinder with steeply conical ends. The purpose of the conical ends, however, was to maintain roughly constant stress throughout the rotor. The stress in a low-windage rotor is so small that its variation can be neglected. The design of the rotor as a simple disk reverts to the equations with which the subject was introduced.

A.4.3 ELECTRO-HYDRAULIC DRIVE.

Electro-mechanical and electro-inertial drives have been considered. The effectiveness of electro-mechanical drives is limited primarily by electrical problems as discussed in A.4.1. The electro-inertial drive is limited by the complexity of the transmission systems required for spin-ups and haul-down, as well as the costly tradeoff of rotor weight and windage horsepower against an evacuated, submersible rotor chamber.

Hydraulic motors operate over a wider range of speed and power than electric motors. Let the mechanical transmission assumed in the preceding sections be replaced by a 3 component hydraulic transmission: fixed displacement pump directly coupled to an electric motor, fixed displacement motor, and high pressure fluid accumulator. For normal haul-down, the pump will drive the hydraulic motor directly. For emergency operation, the accumulator will be discharged through the motor. Following a "crash dive", time must be allowed for the pump to re-charge the accumulator.

Select a Vickers PFB5 fixed displacement, in-line, piston pump. This pump delivers 4.75 gallons per minute at 1800 rpm. In order to charge an

accumulator to 3000 psi, the peak horsepower will be

$$Hp = \frac{\dot{V} \cdot P}{\eta \cdot 33000}$$

where:

$\dot{V}$ is pump rate, cubic feet/min

P is pressure, pounds per ft^2, and

η is pump efficiency.

$$Hp = \frac{4.75 \text{ gpm} \times 231 \frac{in^3}{gal} \times \frac{1 \ ft^3}{1728 \ in^3} \times 3000 \ \frac{lb}{in^2} \times 144 \frac{in^2}{ft^2}}{.8 \times 33000 \frac{ft\text{-}lb}{Hp\text{-}min}}$$

$$= 10.4 \text{ horsepower @ 1800 rpm.}$$

Now, select a Vickers MHT-150-12 low speed, high torque, constant displacement hydraulic motor. This motor delivers, theoretically, 150 ft-lb torque per 100 psi drive pressure, and uses 0.525 gallons per revolution. For normal haul-down, it is connected directly to the pump, so that 4.75 gpm are forced through it, turning it at 9.05 rpm.

Let the winch drum be 20 inches in diameter, with 6 inch flanges. Then the average radius is 13 inches. In order to haul-down at 0.5 feet/sec, its rotation must be

$$N = 60 \frac{U}{2 \pi \bar{R}}$$

$$= \frac{60 \frac{sec}{min} \times 0.5 \frac{ft}{sec} \times 12 \frac{in}{ft}}{2 \times 3.14 \times 13 \text{ in}}$$

$$= 4.41 \text{ rpm.}$$

Thus, the hydraulic motor must drive the drum through a 2.05:1 reducer. Select 2:1 for convenience. This gives a nominal positioning speed of 0.51 ft/sec.

Now, check the line speed and hydraulic pressure for normal haul-down with a full drum. When the drum is full, the line speed is

$$U_\ell = \frac{N_m}{R_g} \times \frac{2\pi R_x}{12} \times \frac{1}{60},$$

where N_m = 9.05 rpm is the hydraulic motor speed,

R_g = 2 is the gear reduction, and

R_x = 16 inches is the full-drum radius.

$$U = \frac{9.05 \times 2\pi \times 16}{2 \times 12 \times 60} = 0.63 \frac{\text{ft.}}{\text{sec}}$$

As expected, the line speed is slightly above average. The hydraulic pressure will be

$$P = \frac{T}{1.5} = \frac{F \times R_x}{\eta R_g \times 1.5},$$

where F is the line force and

η is the motor efficiency.

The line force depends on the line speed, as developed before

$$F = 4000 \text{ lb (buoyancy)} + 5.37\, U^2 \text{ (drag)}$$

$$F = 4000 + 5.37\, (.63)^2 = 4002 \text{ lb.}$$

Thus, the hydraulic pressure is

$$P = \frac{4002 \times 16/12}{.8 \times 2 \times 1.5} = 2223 \text{ psi.}$$

Since this is within the capability of the pump selected, normal haul-down can be accomplished. Of course, the power required when the drum is only partially full is less than calculated above.

It remains to size and evaluate the accumulator required to provide emergency haul-down capability. The volume of fluid required is based on a bare-drum haul-down, since that case requires the most turns of the motor.

$$\text{turns} = \frac{D \times R_g \times 12}{2\pi R_b} ,$$

where D = 120 ft haul-down distance and

R_b = 10 inches bare drum radius,

so that:

$$\text{turns} = \frac{120 \times 2 \times 12}{2 \times \pi \times 10} = 45.8$$

The corresponding fluid volume is:

$$\forall = .525 \ \frac{\text{gal}}{\text{rev}} \times 45.8 \text{ rev} = 24.0 \text{ gallons}$$

The minimum pressure in the accumulator occurs when a full-drum emergency haul-down is just completed. In this case, the haul-down force is:

$$F = 4000 + 5.37\ (10)^2 = 4537 \text{ lbs,}$$

and the pressure is

$$P = \frac{4537 \times 16/12}{.8 \times 2 \times 1.5} = 2520 \text{ psi.}$$

Let the accumulator have a total volume V_1 and be filled with gas compressed to pressure $P_1 = 1000$ psi. When the accumulator is fully charged, this gas is compressed to volume V_3 and pressure 3000 psi, so that

$$\frac{V_1}{V_3} = \frac{P_3}{P_1} = 3.$$

During emergency haul-down, the gas expands from V_3 to V_2 and the pressure drops to $P_2 = 2520$ psi. Thus $V_3 = V_2 - \Delta V = V_2 - 24$ gallons.

Since $P_2 V_2 = P_3 V_3$,

$$V_2 = \frac{\Delta V}{1 - \frac{P_2}{P_3}} = \frac{24}{1 - 2520/3000} = 150 \text{ gallons}$$

$$V_3 = 150 - 24 = 126 \text{ gallons}$$

$$V_1 = 3 \cdot V_3 = 378 \text{ gallons.}$$

Recharge time is

$$\frac{24 \text{ gal}}{4.75 \text{ gpm}} = 5.05 \text{ minutes.}$$

Select a Superior C858-SA32-462 accumulator as an example for the purpose of evaluating the implication that $V_1 = 378$ gallons. This accumulator has a 20 gallon capacity at 3000 psi, so 19 of them are needed. Each one is 10 ft long by 9 inches in diameter and uses 700 lbs of aluminum. The aggregate weighs 13300 lbs.

In order to reduce the size of the accumulator, we add a second Vickers MHT-150-12 hydraulic motor on the winch. This motor is coupled directly to the accumulator, and engages the winch drive only for emergency haul-down. The original hydraulic motor is connected in the original manner - to the pump for normal operation and to the accumulator for emergency operation.

During emergency haul-down each motor makes 46 turns, requiring 24 gallons, so the total emergency flow is now 48 gallons. But each motor now supplies only half the torque, so that the supply pressure can be half (the motor piping to the accumulator is parallel, not series!)

$$P_2 = 2520/2 = 1260 \text{ psi}$$

$$V_2 = \frac{2 \times 24}{1 - \frac{1260}{3000}} = 82.8 \text{ gallons}$$

$$V_3 = 82.8 - 48 = 34.8 \text{ gallons}$$

$$V_1 = 3\ V_3 = 104.3 \text{ gallons}$$

The surprizing conclusion: twice as much fluid can be delivered from an accumulator about one-fourth the size. This is because the delivery pressure has been halved, so that the accumulator is nearly empty at the end of the haul-down.

Recharge time, of course, has doubled, to 10.1 minutes.

The Vickers hydraulic motor is rated for continuous duty from 0 to 250 rpm. During emergency haul-down the nominal speed is about

$$N = 20 \times 9.05 = 181 \text{ rpm.}$$

The actual speed vs time profile is determined by the force balance of line tension due to buoyancy, inertia, and drag and drum torque due to hydraulic pressure.

APPENDIX B

THREE LEG MOORING CONCEPT

Appendix B

THREE LEG MOORING CONCEPT

Figure 4-5 illustrates a three-leg mooring concept. The subsurface buoy which joins and supports the three mooring legs also supports the haul-down mechanism. The subsurface trimoor is a proven device for providing a rigid platform in the sea. For the LINEAR CHAIR, it allows the haul-down winch to be replaced by a simple capstan, with back tension produced by the weight of the bight hanging beneath the capstan. One advantage of the capstan is that no conductor breakouts with their associated slip rings need be supplied.

B.1 CAPSTAN ANALYSIS.

The analysis of the capstan system follows the winch analysis of section A.4.3, with the provision that the effective capstan radius is constant, while the effective drum radius varied with the amount of line wound on the drum. Indeed, if we accept the average drum radius, 13" for the capstan radius, then much of the analysis is repetitive. These calculations are summarized in Table B-1. Capstans require that the line paying out from the drum be kept under tension in order to make the line grip the surface of the drum without slipping. But since the LINEAR CHAIR array is being hauled vertically, the back tension weight serves to reduce the horsepower needed at the capstan. Let the back tension weight be 500 lbs. Then the weight does work on the capstan during haul-down, at the rate

$$Hp = \frac{W \times U}{550 \times \eta} = \frac{500 \times .51}{550 \times .8} = 0.58 \text{ horsepower.}$$

TABLE B-1. NORMAL HAUL-DOWN WITH CAPSTAN

Pump	Vickers PFB5
capacity	4.75 gallons/minute
at	3000 psi delivery
	1800 rpm
Motor	Vickers MHT-150-12
displacement	.525 gallons/rev.
torque/pressure	1.5 ft-lb/psi
Capstan	
diameter	26 inches
gear reduction	2:1
line speed, normal operation	0.51 ft/sec
efficiency	0.8
Back tension	500 lbs
Power	
line	4.66 horsepower
back tension	0.58 horsepower
net	4.08 horsepower

The net power to drive the capstan is therefore:

$$Hp \cong 4.66 - 0.58 \approx 4.08 \text{ horsepower.}$$

For emergency haul-down of 120 feet, the number of capstan turns required is:

$$\text{turns} = \frac{D \cdot R_g}{2 \pi R} = \frac{120 \times 12 \times 2}{2 \times \pi \times 13} = 35.26.$$

Since the motor displaces 0.525 gallons per turn, the accumulator must deliver:

$$G = t \times 0.525 = 35.26 \times .525 \approx 18.51 \text{ gallons.}$$

The net torque required is:

$$T = \frac{F \times R}{\eta \times R_g}$$

$$T = \frac{(4000 + 5.37\ U^2 - 500)}{0.8 \times 2} \times 13/12 = \frac{4037 \times 13}{.8 \times 24}$$

$$= 2733 \text{ ft-lbs,}$$

and the pressure input for the Vickers motor is:

$$P = T/1.5 = 1822 \text{ psi.}$$

Applying the accumulator equations as in section A.4.3 gives

$$V_2 = \frac{\Delta V}{1 - P_2/P_3} = \frac{18.51}{1 - 1822/3000} = 47.14 \text{ gallons,}$$

$$V_3 = P_2/P_3 \times V_2 = \frac{1822}{3000} \times 47.14 \cong 28.63 \text{ gallons,}$$

and

$$V_1 = P_3/P_1 \times V_3 = \frac{3000}{1000} \times 28.63 = 85.89 \text{ gallons},$$

where V_1 is the total accumulator capacity, $V_1 - V_2$ is the oil reserve after emergency haul-down, and $V_1 - V_3$ is the fully charged oil capacity.

If emergency haul-down is accomplished using two hydraulic motors acting in parallel, as described in section A.4.3, then each turns 35.26 turns, requiring a total of 37 gallons but at a pressure of only 911 psi. Since this is below the 1000 psi precharge pressure, set P_2 = 1100 psi, which will cause the emergency haul-down speed to exceed 10 ft/sec slightly. Then

$$V_2 = \frac{37}{1 - \frac{1100}{3000}} = 58.42 \text{ gallons},$$

$$V_3 = \frac{1100}{3000} \times 58.42 = 21.42 \text{ gallons, and}$$

$$V_1 = 3 \times V_3 = 64.26 \text{ gallons.}$$

It is apparent that the trimoor concept using a capstan substantially reduces the accumulator requirements for emergency haul-down. Indeed, it is logical to consider the back tension weight as a counter weight, and increase its size beyond that required to prevent line slip. A thorough analysis to determine the optimum weight cannot be performed herein, but the problem can be bounded by considering the extreme case.

In the extreme case, the back tension weight is equal to the working strength of the cable. Since the LINEAR CHAIR array buoys' buoyancy is equal to the cable working strength, there is essentially no net torque on the capstan brake under static conditions. Normal-speed depth adjustment requires virtually no power, since the drag is only about 2 pounds at 0.5 feet/second.

The components identified in Table B-1 are substantially too large for the emergency haul-down task in this case. In order to preserve numerical consistency for comparison, they will still be assumed. At 10 feet/second haul-down, the drag is 537 lbs, so the torque required of the hydraulic motor is 364 ft-lbs. One motor supplies this at a pressure of 242 psi; there is no need for a second hydraulic motor. If the precharge pressure is 150 psi and the full charge pressure is 3000 psi, the accumulator volume must be at least 32.5 gallons. Using two 20 gallon tanks, the full charge pressure need be only 1000 psi.

For steel weighing 480 lb/ft^3, a sphere of 4.64 ft diameter will weigh 4615 pounds in air and 4000 pounds in seawater.

B.2 FREE FALL CONCEPT.

If the immersed weight of the back tension counterweight exceeds the net buoyancy of the two buoys, then the capstan is used to haul the counterweight up, instead of dragging the buoys down. In this case, it is worthwhile to inquire whether the emergency drawdown can be accomplished in free fall. No accumulator at all would be required; simply release the capstan.

The acceleration time must be included in a free fall analysis. In the absence of capstan torque, the equations of motion of the array are

$$\frac{dV}{dt} = \frac{W_w - 2B_b - (K_w + K_c + 2K_b)\, V^2}{M_w' + 2M_b'} \text{, and}$$

$$\frac{dx}{dt} = V,$$

where:

V is the falling velocity,

X is the distance fallen,

t is the time since release,

W_w is the immersed weight of the counterweight

B_b is the net buoyancy of a buoy,

K_w, K_c, K_b are the drag factors of the counterweight, cable, and buoy, and

M'_w, M'_b are the apparent masses of the counterweight and buoy.

Inasmuch as the drag varies as the square of the velocity, the acceleration equation is non linear, so a numerical integration is used. The solution is iterated in order to determine the size of surface buoy that will fall 120 feet in 12 seconds using an immersed weight of 4000 lbs. Figure B-1 shows falling speed and distance fallen versus time for this case. The buoys are found to be restricted to 4.4 feet in diameter, producing a net 1500 pounds bouyancy each.

The design tradeoff is now apparent. A free-fall system eliminates the entire accumulator system, with its extra piping, valves and control logic, but at an as yet undetermined cost of reduced mooring vertical stiffness.

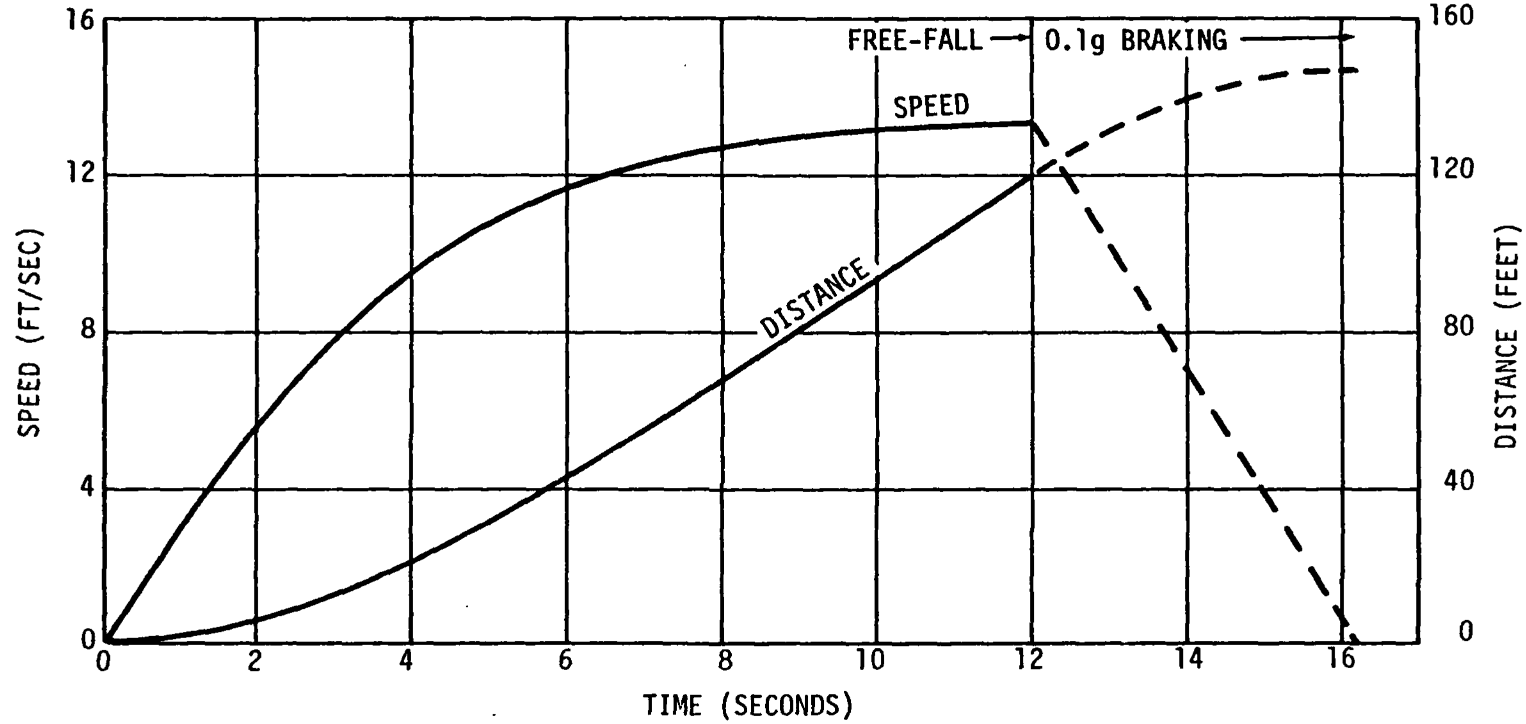

Figure B-1. Emergency Free Fall Drop Speed and Distance vs Time

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