HAWAII DEEP WATER CABLE PROGRAM
PHASE II

CABLE HANDLING EQUIPMENT
CONCEPT STUDY

PREPARED BY WESTERN GEAR MACHINERY CO.

JANUARY 1986

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CABLE PROGRAM

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Prepared by
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Prepared for

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The Ralph M. Parsons Company
Hawaiian Electric Co., Inc.
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FOREWORD

This feasibility study of cable handling equipment requirements for the Hawaii Deep Water Cable (HDWC) program was prepared by Western Gear Machinery Co., drawing on experience in the laying of submarine communication cables and oil pipelines.

An executive summary at the beginning of the report briefly describes the purpose of the study, the work performed, and the significant conclusions and recommendations.

Section 1.0, "Introduction," contains information about the basis of work, scope of responsibility, and contracted parties.

Section 2.0, "Problem Analysis," provides a basic definition of static and dynamic requirements and provides conclusions from an analysis of dynamic tension control.

Section 3.0, "Equipment Analysis," contains a description and analyses of existing equipment and techniques to determine the recommended combination for the HDWC program at-sea test.

Section 4.0, "Conclusions and Recommendations," describes a baseline set of equipment for the program and summarizes requirements for development.

The appendices include information about the authors of the study and a glossary of key terminology.

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EXECUTIVE SUMMARY

The purpose of the Hawaii Deep Water Cable (HDWC) program is to demonstrate the technical feasibility of the use of cable handling equipment under varying conditions between the Hawaiian Islands. This will involve at-sea tests to obtain data to allow demonstration of the technical feasibility of the HDWC program. The testing will involve laying and retrieving a length of high voltage electrical power transmission cable in the Alenuihaha Channel between the islands of Hawaii and Maui.

In this Cable Handling Equipment Concepts study, the HDWC program at-sea test requirements of high cable tension and increased sea states were evaluated against present state-of-the-art cable handling equipment to formulate recommendations for successful completion of the program. Equipment requirements for a baseline commercial program were also evaluated.

Deep channels, steep bottom slopes, and operation during worst case sea conditions are challenges expected during the at-sea test. Successful cable lay operations involve interactions between the cable, equipment, and environmental conditions. The primary determinants of the cable equipment are cable design and vessel response to sea conditions.

The cable proposed for use is based on a single aluminum conductor oil filled (SCOF) construction sealed within a lead sheath and polyethylene jacket. Double armor wire elements wound in opposite directions over the core provide a torque balanced construction.

Using the design sea state for the Alenuihaha Channel and cable dynamic loads, it was determined that the cable equipment must provide heave compensation by limiting energy input to the vessel/cable interface, because the cable dynamic tension levels are above those allowed by the cable parameters.

When HDWC program requirements were compared to the capabilities of present state-of-the-art equipment, it was determined that the level of tension anticipated for the program at-sea tests is approximately twice what existing cable lay equipment is rated.

The program's high tension levels can be met by using a tensioner similar to those used in submarine pipe laying programs. A separate dynamometer must be used to ensure a fast response measurement of line tension. A hydraulic motor drive must be used because it provides better dynamic response, as compared to a D.C. electric motor drive for a comparable power output.

Though cable storage requirements for the HDWC program are similar to the capability of present day power cable lay vessel turntables, their drive system levels must be increased to provide the acceleration and deceleration required to meet speed and equipment frame considerations.

A surge-slack handling method is required to provide a means of non-uniform rate of cable payout or retrieval because of heave compensation requirements. Expected high tension levels could cause accumulative cable twist, and means of removing this would need to be provided.

Non-uniformity of cable payout in heave compensation operation would require automatic integrated controls between the elements of the cable handling equipment, with manual control as the backup mode of operation.

Mathematical models of a linear track type of tensioner were established to study design improvements which would increase the ability of the tensioner to limit variations in cable tension. Simulations were run to determine system response to particular conditions.

It was concluded that a relatively high system stiffness is a major objective in the design of the hydraulic system. A faster responding hydraulic flow control device may be required after full system design parameters are known, including replacing the stepper motor drive with a more powerful servo actuator and the use of a direct operating servo valve.

It was concluded that further study and model refinement must be accomplished when final design and cable parameters are known to determine the need for gain scheduling in relation to cable tension and suspended cable length, to optimize performance throughout the spectrum of operating conditions. Before a machine design is completed, factors such as the limit of slip of the track gripper pads on the cable, the simulation of the cable as a distributed mass system, and dynamic simulation of the overboarding device must be considered. Also, final cable parameters of spring rate, friction, twist, gripping size, and strength must be obtained to ensure the validity of the simulation model.

Analysis of existing cable equipment indicates that presently no complete lay vessel exists that meets all of the criteria required by the HDWC program. One existing vessel, the SKAGERRAK, would with extensive modification be capable of performing the at-sea tests, but there are concerns regarding availability for modification. The conclusion drawn is that an entirely new vessel would best suit the needs of the HDWC program at-sea tests.

Three types of overboarding support devices were considered: active dynamic, passive dynamic, and static. It was concluded that effective heave compensation at the overboarding device by using an active or passive dynamic device is impractical, and that tension compensation must be provided by the cable tensioner. Passive metal troughing or a multiple roller arrangement is recommended, depending on final cable parameters.

Three types of overboarding devices were analyzed, including chute style, multiple roller, and sheave style. The chute was eliminated from consideration due to high cable tension hysteresis (the tension difference between cable payout and inhaul). The multiple roller device's high degree of strain and crushing force per roller would cause unavoidable damage to the cable sheath. At this time, a single 12.3 meter sheath provided with radiused guards is recommended because it is the only device without excess tension loss that is suitable for use with the cable design parameters. The sheave's dynamic effect on cable tensioning capability must be minimized during design.

Two basic types of cable tensioners were studied; capstan and linear. Capstan tensioner styles analyzed included single drum, multiple drum, and sheave style arrangements. The capstan tensioner was eliminated from consideration because a drum of excessive diameter would be required and because the device can not be operated without the use of a complex draw-off and hold back (DOHB) machine to maintain back tension on the drum. Linear tensioner arrangements considered were the pneumatic tire type and the track type. The linear tire tensioner would require an excessive overall system length. The linear track tensioner was found to be the most suitable because it is the only tensioner of practical size that does not violate cable design parameters for squeeze and tension shear per unit length. A linear track tensioner with 26.2 meters of active length based on preliminary PPC #116 cable parameters to achieve a 78.7 MTon tension is recommended. The tensioner overall length would be approximately 31.2 meters.

A tank type of cable storage device had been eliminated from consideration because of the HDWC program's use of torque balanced cable construction. A turntable device approximately 10 meters in diameter, utilizing roller support structure, was recommended for the at-sea test vessel. This turntable would have a 9.4 kilometer capacity when PPC #116 cable is stacked to a height of 3.23 meters.

The above-mentioned turntable would require either a bull wheel sheave type of surge-slack device or a pickup arm surge-slack device. After cable interface considerations of handling tension and cable twist were considered, the bull wheel concept was eliminated from consideration because it would require more space and added structural support than is practical, and would require a cable transporter to provide additional cable inhaul and tension restraint. The pickup arm is considered the better choice because of its smaller deck space requirements and runaway cable safety, because it does not require a cable transporter, and because its disadvantages can be economically addressed by optimum design of the system components. A single pickup arm located over the turntable is the recommended configuration. The pickup arm would be passive during cable payout and under manual control during cable recovery or loading.

The cable lay vessel's integrated control system would supply long period cable lay operational commands, and the tensioner controls would supply the short period internally sensed dynamic commands. Machinery controls would consist of controls for the tensioner, surge-slack device, and turntable. Those controls would control cable displacement during tension compensation, speed changes to compensate for the changing cable coil diameter during loading or unloading, and compensation for overall vessel speed variations.

The control system would use two basic types of sensing devices: general monitoring sensors which have no immediate effect on cable control in the event of failure, and critical control sensors which immediately affect the cable handling process if failure occurs. Critical sensors for the system would include the cable tension sensor, cable speed and footage sensor, tension machine speed sensor, tension machine stability augmentation sensors, pickup arm cable position sensors, and turntable speed sensors.

The cable tension sensor would use the precision deflection method of tension sensing. The cable speed sensor would be a D.C. tachometer and the position indication sensor would be composed of optical sensors reading markers on the cable. The tensioner and turntable speed sensors would be magnetic sensors and the tension machine stability augmentation sensors would be of the pressure feedback type. The pickup arm cable position sensors would be encoder type sensors.

A centralized control system is proposed which would allow separate troubleshooting and maintenance of each machine system. Three types of integrated controls were considered, including a hardwired analog system using discrete components, a microprocessor digital design, and a hybrid computerized system using hardwired components to digitize information. The system tentatively recommended is a hybrid system consisting of analog circuitry and digital machinery. The analog devices would provide the basic machine control loops while the digital devices would provide the data loops and interface with the overall cable lay control. The system would provide memory backup and redundant controls in case of system failure or power outage. The bottom level of the control loop would be operator assumption of direct control, so the basic line of control would require a minimum of instrumentation.

In summary, the risks inherent in the HDWC program result from the necessity of increasing the state-of-the-art of cable laying technology. Following are areas requiring further study and review due to insufficient systems concept and design definition:

- The sensor system for control of the onboard handling equipment requires concept refinement, as this type of sensor system has not previously been applied to cable machinery.

- Controls integration and detailed design definition also require concept refinement.

- Final data regarding the cable/machine interface will be required to perform design evaluation and testing to optimize the tensioner design. It is important, for economic and dynamic response reasons, to minimize the length of the machine.

- Further study to arrive at an optimum tensioner control configuration is required, including coordination to ensure baseline assumptions are representative of the expected configuration and conditions. Gain scheduling needs, cable gripper slip factors, overboarding sheave dynamics, dynamometer sensitivity and other related factors would be considered.

- Sufficient data is required to proceed with fundamental cable parametric studies to establish the tensioner concept design and to define the tensioner gripper block interface with the cable.

- A surge-slack system model should be developed and studied utilizing final cable parameters to determine the slack absorbtion capability of the system, power drive requirements, and other factors such as sensor accuracy.

- Cable vessel guidance and machinery limitations must be determined so detailed cable entry requirements can be set for the overboarding sheave and guards.

- Pickup arm maneuverability to remove cable twist during loading or retrieval should be further studied to ensure adequate turntable coverage.

- Existing turntable designs meet size and capacity requirements for a baseline commercial lay program, but do not operate at the speeds required. Further studies of risks, stopping distances, and power requirements as they relate to bearing design for turntable support systems are required for a baseline commercial system.

- Response characteristics of the selected cable vessel to the design sea state will be needed to determine probable loading parameters to begin the design definition phase of the HDWC program.

1.0 INTRODUCTION

1.1 BASIS OF WORK

The Hawaii Deep Water Cable (HDWC) program is a research and development project to demonstrate the technical feasibility of the use of cable lay equipment under varying conditions between the Hawaiian Islands. The purpose of the cable handling equipment subsystem at-sea test is to obtain data on the cable, on cable equipment and on lay vessel concepts and procedures that will allow determination of the technical feasibility of the HDWC concept of a deep ocean water high voltage electrical transmission as it relates to the Alenuihaha Channel between the islands of Hawaii and Maui. Vessel and cable equipment operational data will be obtained by laying and retrieving a length of cable under various sea state, sea bottom, and operating conditions. This data will be used to validate design models and allow a determination of feasibility.

1.2 SCOPE OF RESPONSIBILITY

Requirements for preliminary design of the cable handling equipment subsystem were developed in this study for the HDWC program at-sea tests. The program requirements of high cable tension and increased sea states were evaluated against present state-of-the-art equipment to formulate recommendations for successful completion of the program.

Cable equipment elements required for a baseline commercial program were also evaluated in this study. The technical feasibility of these elements is to be proven against the operation requirements of the HDWC program. This should result in a reduction of the risks associated with the critical aspects of cable deployment.

This study is limited to cable equipment and does not consider vessel deck arrangement. Cable deployment and retrieval were not included in this study as they depend on vessel deck and cable equipment arrangements.

Assessment of the relationship of cable to the cable handling subsystem equipment was limited to the three candidate cables, PPC No. 113, 116, and 119 that were previously selected for the HDWC program.

Figure 1.2-1 shows a sketch of the proposed equipment as it has been conceived to date. This arrangement, shown installed on a flat barge similar to that used in the past, is based on proven machinery. Only the equipment covered by this study is shown for clarity and basic visualization.

1.3 CONTRACTED PARTIES

This study was prepared under contract to Hawaiian Dredging and Construction Company, using data provided by them and other team member companies including Pirelli Cable, Parsons Hawaii, Makai Ocean Engineering, and contributing consultants such as Bud Schultz and Luis Vega. The relationship of team contributors to WGMC is shown in Figure 1.3-1.

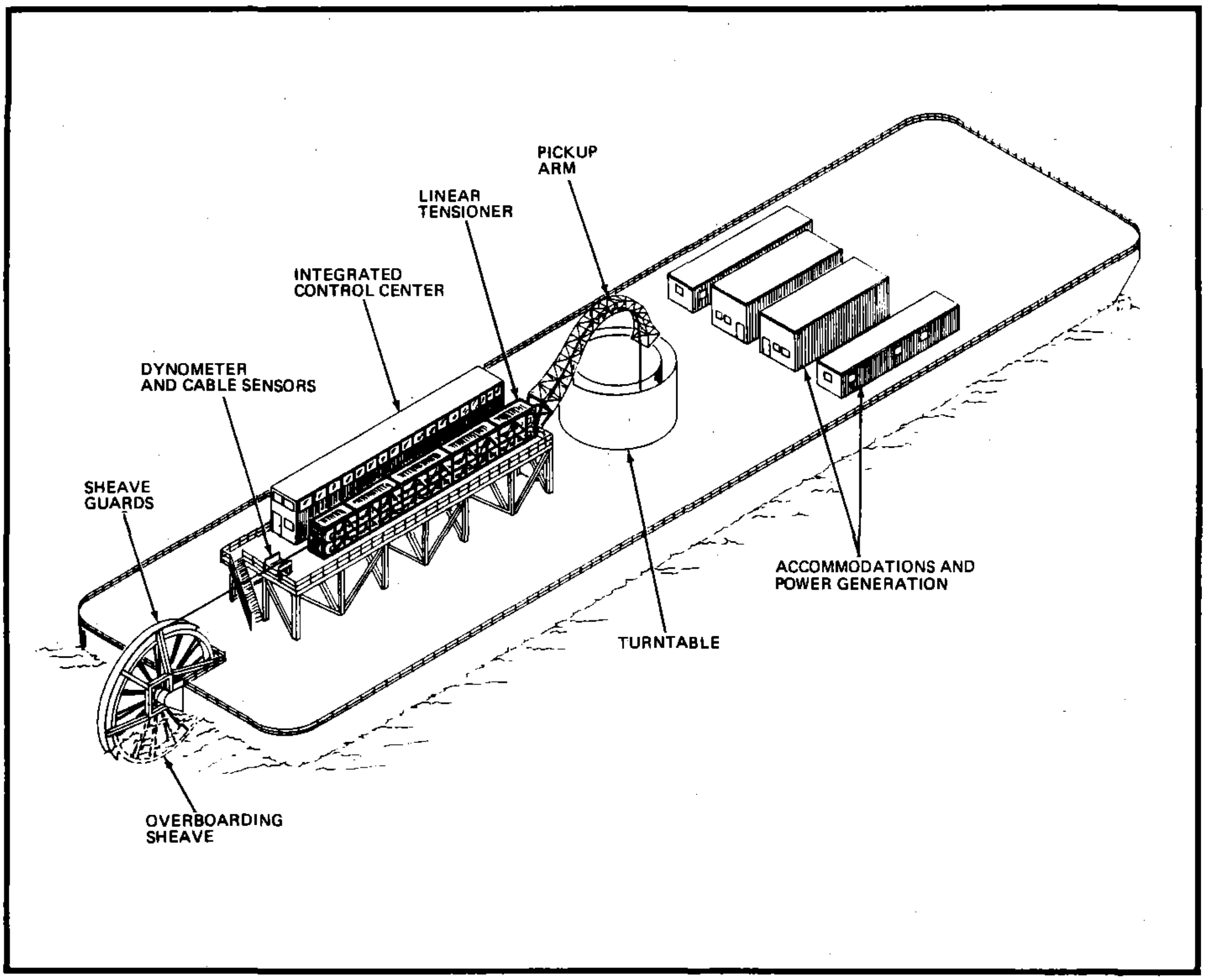

FIGURE 1.2-1 Conceptual Drawing of the Major Cable Handling Machinery for the Hawaiian Deep Water Cable Program

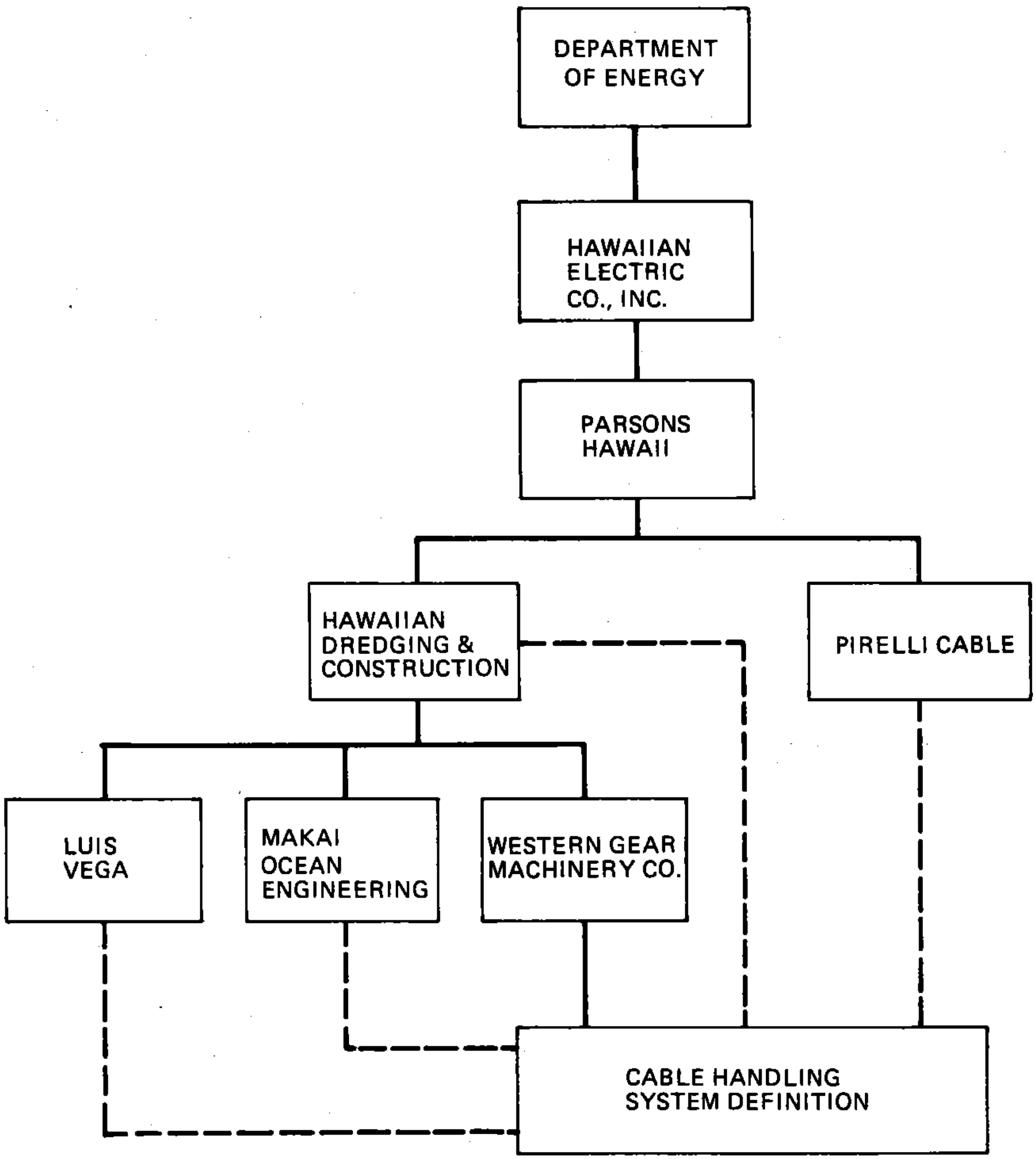

FIGURE 1.3-1 Basic Relationships and Contributors to the Cable Handling Equipment Concepts Study

2.0 PROBLEM ANALYSIS

2.1 GENERAL

The map in Figure 2.1-1 shows the environmental conditions existing in one section of a proposed route for a transmission power cable system between the islands of Hawaii and Oahu. Figure 2.1-2 shows the profile of the expected cable lay route, in which cable must be laid accurately and safely in waters as deep as 2000 M (6562 FT). Deep channels, steep bottom slopes, and cable laying operations during worst case currents with deteriorating sea conditions are challenges which may be encountered. Surveys of currents, bottom conditions, sea states, and other factors which affect operations are being conducted.

The process of laying a cable successfully on the sea bottom involves interactions between the cable, the cable route environmental conditions, and the cable laying equipment. This interrelationship is shown in Figure 2.1-3. In this study the interface between the areas of cable equipment design and cable design has been considered, along with the driving function analysis resulting primarily from sea state conditions. The cable handling equipment must be designed to provide certain defined functions and the forces it imposes on the cable must be within defined allowable limits. The cable, a complex assembly of elements and components, must be handled in a way that does not compromise its integrity or reliability.

The primary determinants of the cable equipment are the design parameters of the cable and the vessel response to sea conditions. These variables are discussed in Sections 2.2 and 2.3, respectively.

2.2 CABLE DESIGN

Figure 2.2-1 shows a cross section through a cable of the general type proposed for candidate cables 113, 116 and 119. The candidate cables are

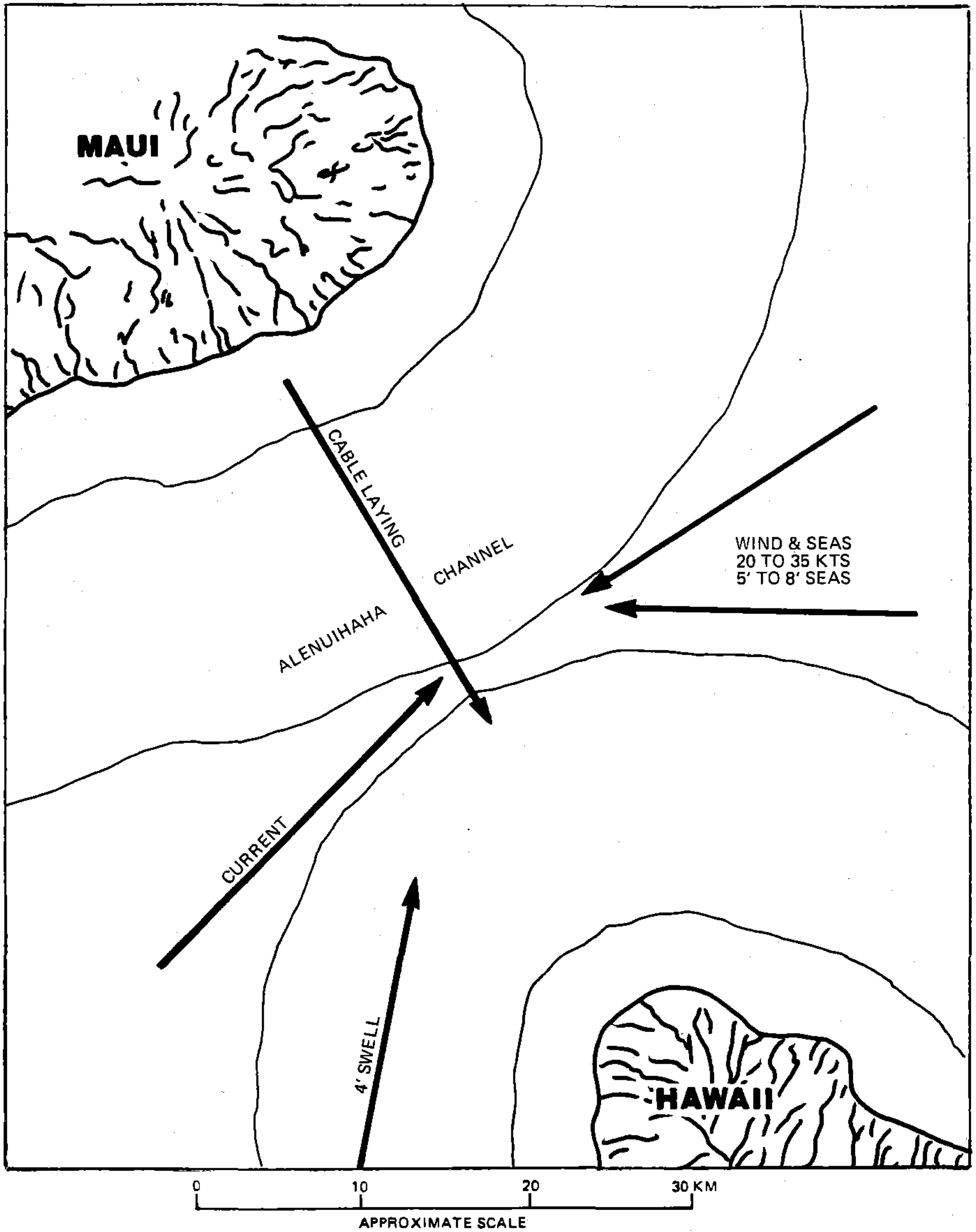

Figure 2.1-1. Map of Alenuihaha Channel

DEPTH AND SLOPE PROFILES
FOR ROUTE OPTION I

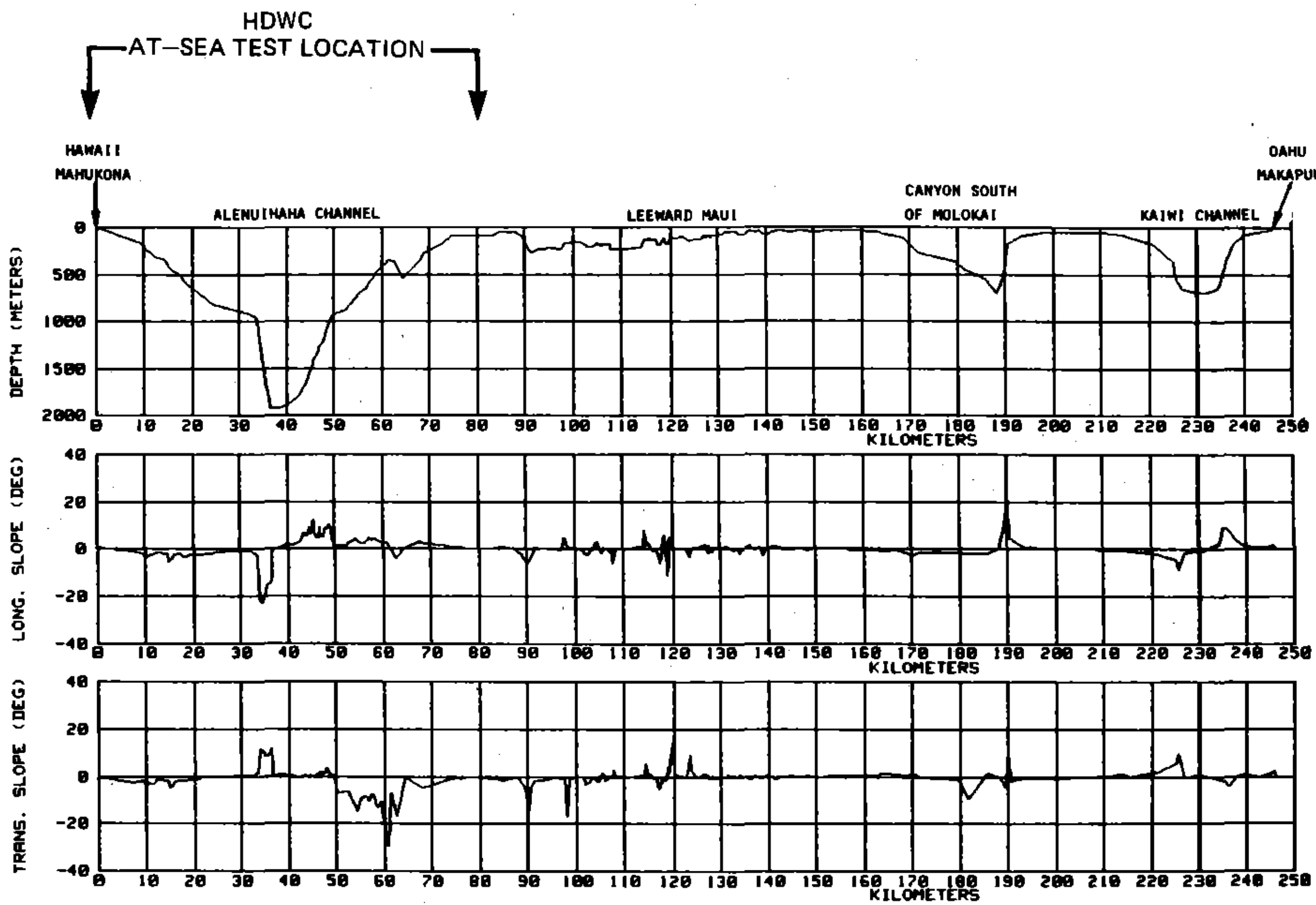

FIGURE 2.1-2 Depth and Slopes Data of the Projected Cable Route from Hawaii to Oahu. Source Makai Ocean Engineering

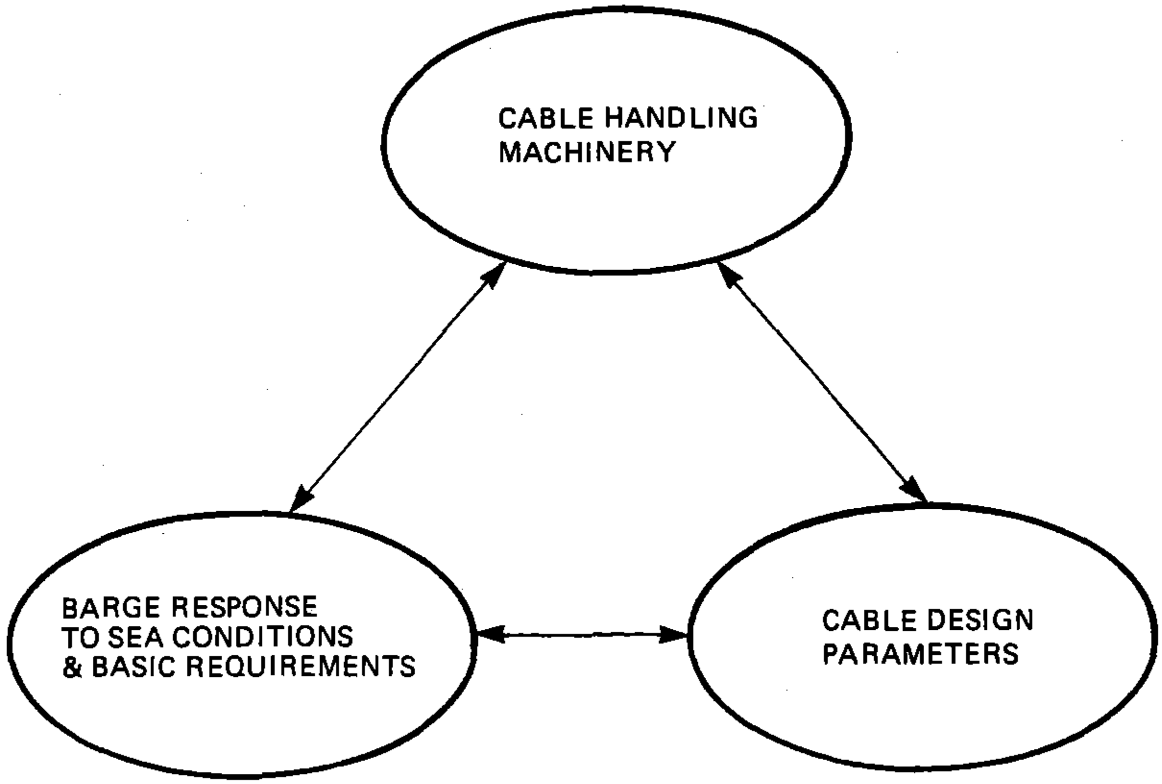

FIGURE 2.1-3 Primary Technical Requirement Interface Relationships - Cable Handling Equipment

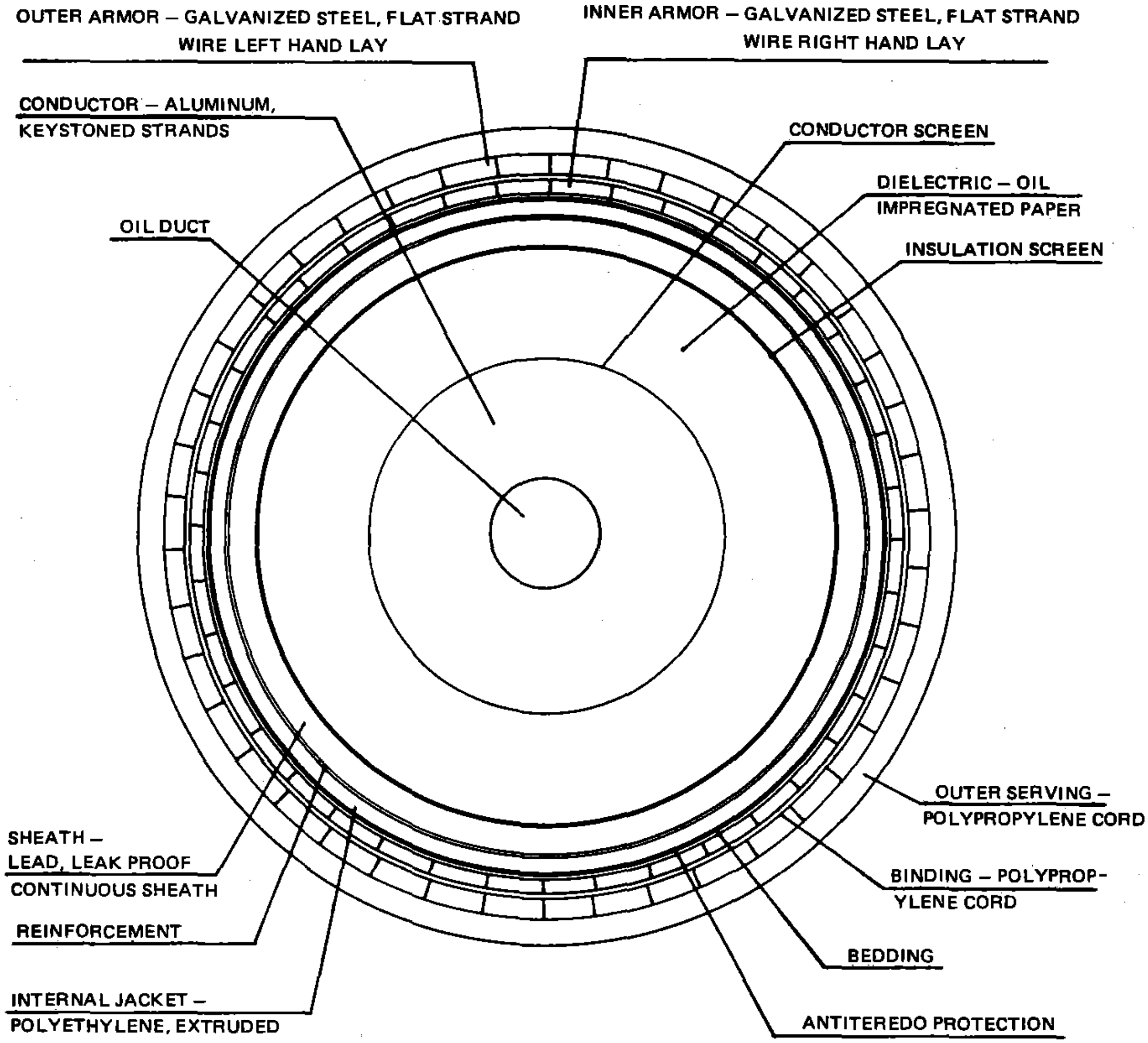

FIGURE 2.2-1 Assumed Cable Section

Note that this cross section is based on existing cables and assumptions of conductor sizing by WGMC.

similar in construction to those used in the Cook Inlet cable lay in Alaska. This handling experience accents the importance of knowing the cable's twist/torque characteristics. The candidate cables use an aluminum conductor in place of copper, which saves weight and reduces loads on the cable and machinery. The use of aluminum also provides a potentially different twist/torque parameter within the cable which relates to the overall cable handling requirements. Figure 2.2-1 shows an assumed cross section based on the Cook Inlet cable, which utilized a copper conductor.

DESCRIPTION OF CANDIDATE CABLES

Preliminary data for the three candidate cables are listed in Table 2.2-1. All three candidate cables use double contra-helically wound flat armor wire cushioned by binding/bedding layers. The armor wire elements are applied over a power conducting polyethylene jacketed core. This design is based on the single aluminum conductor oil filled (SCOF) type construction and is sealed within a lead sheath and polyethylene jacket. The double armor wires provide a torque balanced construction when wound in opposite directions.

To determine if cable twist could occur during cable retrieval, a complete knowledge of the cable's twist/torque characteristics is essential, for design and development of cable handling equipment and in particular for successful cable retrieval under high tension during the HDWC program.

2.3 SEA CONDITIONS

Sea conditions and the resulting vessel response impose requirements on the cable laying equipment. Definitions of sea conditions is probabilistic; therefore the range of probable sea conditions must be defined and based on operating season and forecasting capability. Without these predetermined limits of probable sea conditions, the possibility exists that actual conditions during lay operations could exceed the performance

PRELIMINARY CABLE DATA

CANDIDATE CABLE		113	116	119
CABLE O.D. (mm)		113.9	119.5	124.6
WEIGHT (Kg/m)	DRY	33	36.4	39.8
	WET	23	25.8	28.3
BENDING DIA (m)	w/MAX TENSION	11.6	12.0	12.3
	w/o TENSION	6.8	7.0	7.2
MAXIMUM ALLOWABLE TENSION		72 TON	78.7 TON	84.0 TON
MAXIMUM ALLOWABLE SQUEEZE		5 T/M	5 T/M	5 T/M
EMERGENCY SQUEEZE*		15 T/M	15 T/M	15 T/M
MAXIMUM ALLOWABLE SHEAR		3 T/M	3 T/M	3 T/M

* Emergency squeeze is the level at which the electrical integrity of the cable is known to be compromised. This level is based on the 3-to-1 safety factor.

TABLE 2.2-1 Preliminary Cable Data
Source: Parsons Hawaii

criteria of the machinery. Figure 2.3-1 shows that cable tension variaations increase progressively as a function of worsening sea conditions.

DESIGN SEA STATE

The overall system must be able to successfully lay cable in conditions described in a previous study[1] as the "Design Sea State" for the Alenuihaha Channel. The design sea state, listed below, is the worst-case environmental condition that exists in the Alenuihaha Channel for 75 percent of the year.

Wind:	- Speed	<= 35 knots (40 MPH)
	- Direction (from)	60° ± 30°
Wave Spectrum:		Seas:
	- Sig. Ht. H_s	2.44 M (8 FT)
	- Mean Period T	5.53 sec
	- Direction (see vessel heading, Fig 2.1-1) (from)	69° ± 22°
Current Distribution:		
	- Speed	2.9 KTS (2.3 MPH) @ surface; 2.2 KTS (2.5 MPH) @ 200 M (700 FT); 1.2 KTS (1.4 MPH) @ 400 M (1300 FT); 1.2 KT (1.4 MPH) tidal component
	- Direction(towards)	50°± 15°
Water Depth		900 M (3000 FT) to 2000 M (6600 FT)

CABLE TENSION

The full implication of the sea condition requirement has been analyzed extensively by others[1] and the resultant cable dynamic loads summarized.[2]

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1. Hawaii Deep Water Cable Program, Phase II, Cable Dynamnics Analysis Parameter Study, by E.G.&G. & Makai O.E.

2. Parsons Summary Letter #HDWC-0488, C. A. Chapman to F. A. McHale, dated 3-20-84.

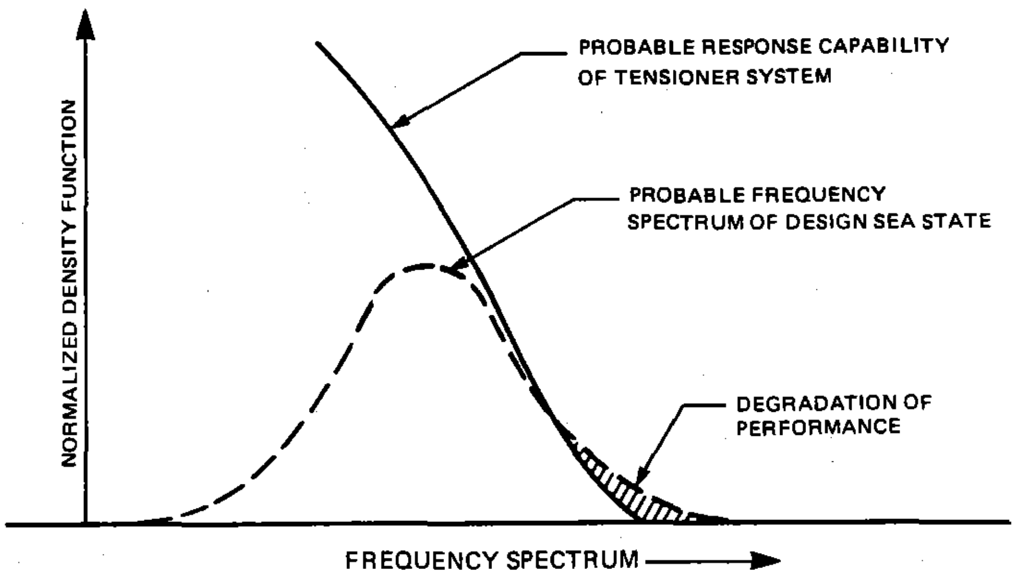

FIGURE 2.3-1 Probailities Relationships of Requirements to Capabilities

There are finite limitations to developmental possibilities for the tensioner, and under some conditions the barge motion will exceed the capability of the tensioner to compensate for tension variations.

The summarized cable tensions for the three candidate cables are shown in Table 2.3-1, given for a 400 FT (122 M) barge with stern cable deployment.

TABLE 2.3-1. Summarized Cable Tensions

CABLE PPC #	MAXIMUM ALLOWABLE TENSION	SUSPENDED CABLE WEIGHT 7000 FT (2133 M)	MAXIMUM DYNAMIC CABLE TENSION
113	72.0 MT (158,760 LBS)	50.2 MT (110,690 LBS)	78.2 MT (172,430 LBS)
116	78.7 MT (173,535 LBS)	55.1 MT (121,495 LBS)	84.5 MT (186,325 LBS)
119	84.0 MT (185,220 LBS)	60.4 MT (133,180 LBS)	91.4 MT (201,535 LBS)

HEAVE COMPENSATION

As can be seen in Table 2.3-1, the dynamic cable tensions are above those allowable. Therefore, the cable machinery must be designed to compensate for the vessel heave induced dynamic loads by limiting the energy input into the cable. The previous study[1] defined the limits of energy input to the vessel/cable interface at various frequencies up to about 3 rad/sec, and the magnitude of total vessel displacement in response to these disturbances in the area of = $\pm$ 15 cm and at a frequency of 1.2 rad/sec.

If the cable equipment can be designed to provide a frequency response of approximately 3 rad/sec with an amplitude as defined, it is theoretically capable of completely eliminating the effects of vessel response to sea conditions on the nominal level of cable tension. The equipment should theoretically be capable of continuing to pay out cable as though sea disturbances were not present.

Major difficulties encountered when meeting heave compensation requirements include: a) sensing devices which function to cause the machinery to respond to sea conditions are not infinitely effective; and b) sea conditions are not finitely definable. As a result, actual tensions will vary to some extent from those projected in Section 2.5.3, "Simulation Results."

2.4 CABLE HANDLING FUNCTIONS

The basic functions of the cable handling machinery can be defined as follows:

Function	Device
1. Overboard or retrieve the cable	Overboard Sheave
2. Control the tension and payout rate of the cable	Cable Tensioner
3. Control the cable on board the vessel	Pickup Arm & Guides
4. Store the cable on the vessel	Turntable

These basic machinery functions are then used combined or individually as required to perform cable laying, retrieval, repair, and other operations. See Figure 1.2-1 for identification and preliminary location of cable machinery devices on board the vessel. All of the above functions have been performed on previous cable lay projects. State-of-the-art cable laying, and, in particular, power cable laying, is discussed in Section 3.1, the Survey of Existing Machinery and Equipment.

To translate the cable handling machinery defined functions into actual hardware design concepts it is necessary to compare this project with the present state-of-the-art. The level of tension anticipated as cable is laid could approach 92 metric tons. Presently, the largest existing cable machinery is rated at about 45 metric tons. The storage requirement for HDWC demonstration cable is similar in magnitude to the capability of present day power cable lay vessels.

CABLE GRIPPING

The cable tensioner concept proposed for HDWC and a baseline commercial system does not present a capacity problem because the tension levels involved are normal in submarine pipe laying programs. The challenge is in gripping the power cable to achieve this level of tension, and in such a manner that will not compromise cable integrity or reliability.

CABLE HANDLING PROBLEMS

Handling cable on the vessel presents new challenges. A means of non-uniform rate of cable payout or retrieval is needed because of the cable tensioner heave compensation requirements. The response rate of the cable storage turntable must meet the long term requirements of the heave compensator. The high cable tensions could also result in accumulated torque/twist forces in the cable, especially during retrieval operations.

A non-uniform rate of cable movement between the tensioner and the turntable would require a means of surge-slack control. Any method of providing this control would require an integrated control system between the tensioner, surge-slack device and turntable which would minimize turntable response required by the non-uniform rate of cable movement. Turntable and surge-slack requirements are discussed further in Sections 3.4.2, 3.4.4 and 3.5.

Cable twist during recovery would result from the expected high tension levels. Any cable exhibits some twist when withstanding tension forces. Torque balanced cable exhibits the least twist, but some minor negative and positive twist occurs under zero to maximum cable tension. Because of machinery inefficiencies and vessel dynamics, each segment of cable would be subjected to a different tension value during the laying and retrieval of that section. These differing tension levels and resultant twist differences could result in an accumulation of twist in the cable segment between the tensioner and the turntable during cable retrieval. Cable handling provisions should be made to remove accumulated twist should it occur during the at-sea test retrieval operations. Accumulative twist handling is discussed further in Section 3.4.

CABLE MACHINERY CONTROLS

Generally, the control of individual cable machinery elements does not pose new requirements, but a new system using specifically designed control equipment will be required. Control systems used on previous power cable lay projects have depended on manual control functions which were adequate for previous steady state cable speed conditions. Non-uniformity of cable payout under cable tensioner heave compensation operation requires that the controls for the cable tensioner, surge-slack device, and turntable cable storage device be automatic and integrated together. Automatic controls are preferred over manual controls for accuracy at high speeds and to avoid operator fatigue during long term continuous operation. Manual control would always be provided as the backup mode of operation.

2.5 TENSIONER DYNAMICS

2.5.1 INTRODUCTION

One critical characteristic of cable handling machinery is the ability of the cable tensioner to limit variations in cable tension which result from vessel response to sea conditions. This requires the massive active parts of the cable machinery to be more responsive in this application than similar machinery in past and present applications. Design improvement approach has been mathematical modeling and functional simulation using proven techniques and programs. Only the linear track type of tensioner was modeled, as it was the tensioner recommended in Section 3.3.

2.5.2 MODEL DESCRIPTION

The baseline cable tensioning device consists of a single track-type machine, similar in function to a linear pipe tensioner. The ability of this device to maintain a constant cable tension depends on the response characteristics of the cable tensioner under expected operating conditions. This was predicted by modeling the cable tensioning system using a standard block diagram approach and inputting this model to the EASY5[1] computer program. Simulations were then run using various gains and inputs to determine the system response at particular conditions.

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[1]EASY5 Dynamic Analysis System is a set of general purpose computer programs that can be used to analyze a wide range of continuous or discrete non-linear dynamic systems. EASY5 is a proprietary software system available through Boeing Computer Services for batch job access.

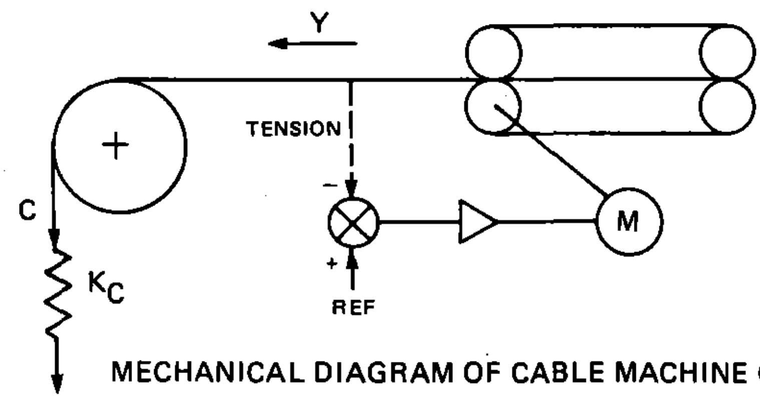
SIMPLIFIED TENSIONER

Figure 2.5.2-1 shows a simplified block diagram of the cable tensioner. This model serves to explain the basic tensioner operation as follows: Beginning in the upper left corner, the difference between the tension command and the actual cable tension (as measured by a separate dynamometer) gives an error signal. When multiplied by the forward loop gain K_Q, a flow command is applied to the hydraulic pump. The pump dynamics introduce a lag to the system, which is estimated to be about 10 radians/second. The difference between the pump flow and the motor flow is the rate at which the oil is being compressed. This quantity integrated is the compressed volume which, multiplied by the bulk modulus and divided by the total oil volume, yields the system pressure. The pressure acting on the effective area of the motor gives the pulling force of the motor on the cable tensioning track. The net force on the cable tensioning track is this pulling force minus the cable tension, minus the damping force due to friction, and results in an acceleration of the track mass. Integrating gives the track speed, which integrated again yields the track displacement. The disturbance input, c(t) is the cable motion (or ship motion relative to the "fixed" cable). The difference between the track and cable displacements is the stretch in the cable. The stretch multiplied by the cable stiffness yields the cable tension, which completes the loop.

This linear, or ideal, system response is valid for gross approximations and simple comparison but it lacks the realism to approximate actual operation.

ACTUAL TENSIONER

The actual cable tensioner contains non-ideal characteristics which were not included in the simple model and which degrade performance. Figure 2.5.2-2 shows a more complex model which was used for the actual analysis. Note that the pump dynamics block has been expanded with a



MECHANICAL DIAGRAM OF CABLE MACHINE CONCEPT

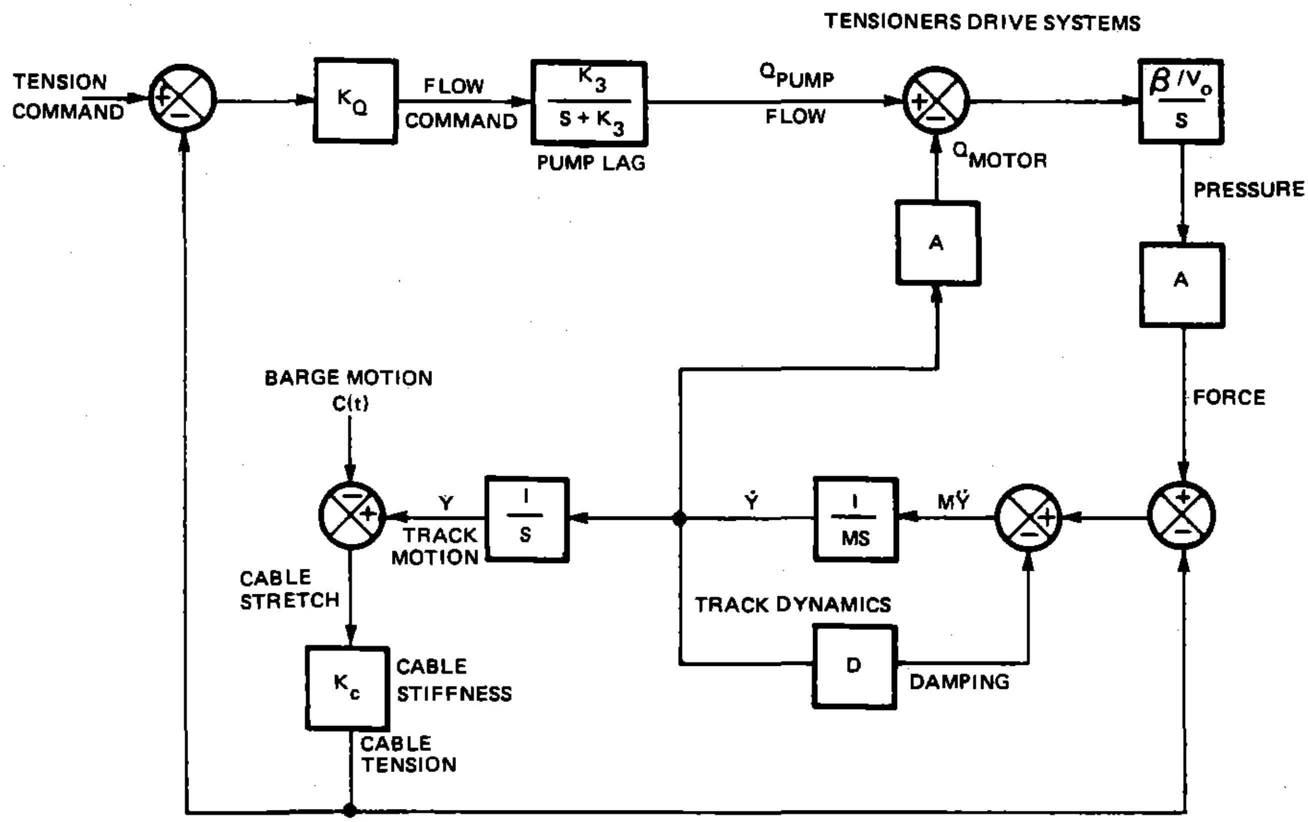
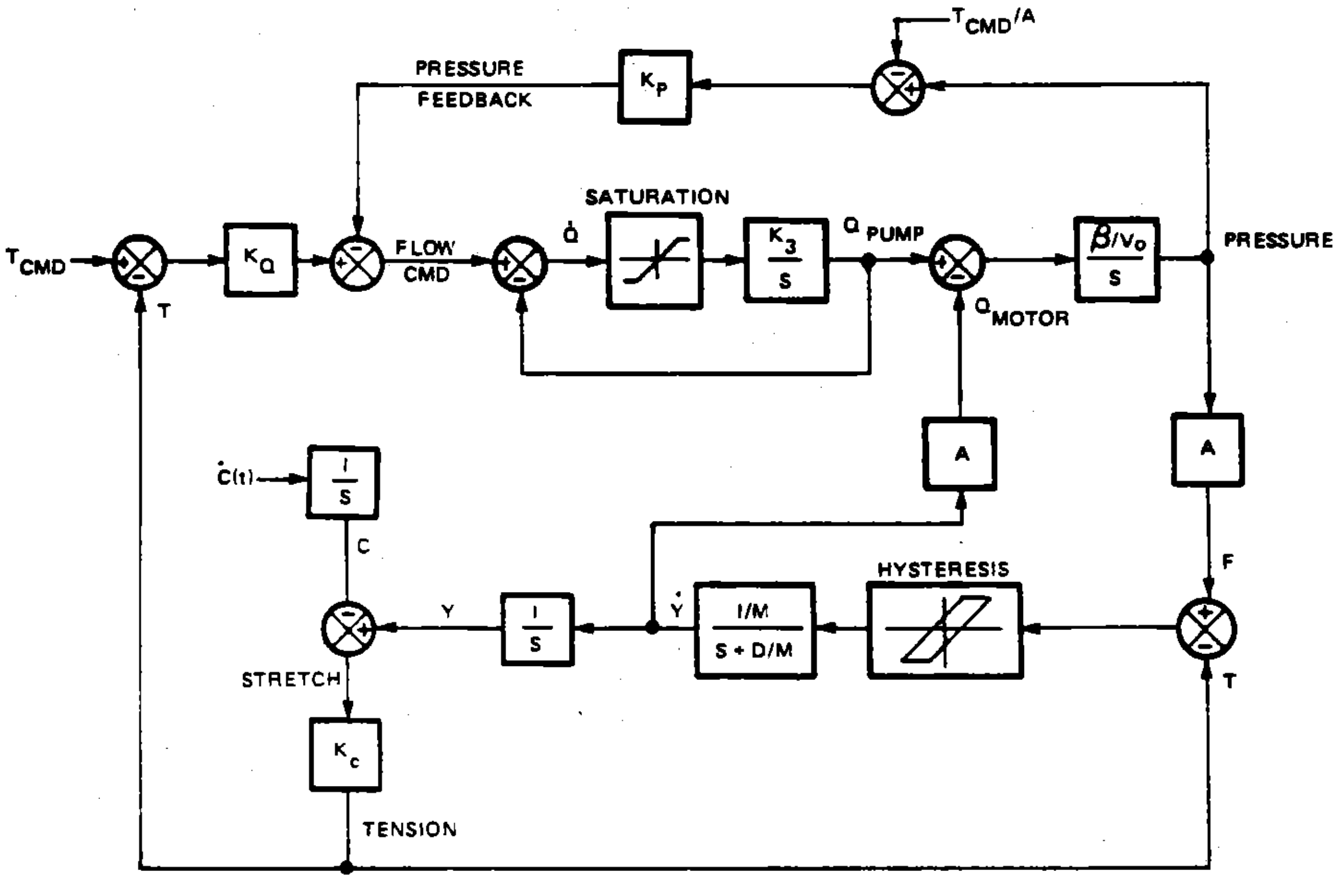


FIGURE 2.5.2-1 Block Diagram of Hydraulic Cable Tensioner

This diagram shows the basic functional relationships of the various elements that comprise a cable tensioner. It is used to yield the characteristic responses of a theorectically ideal machine before "real world" characteristics are introduced and studied.



BASELINE VALUES

T_{CMD} = 100,000 LB – BASIC OPERATING TENSION
K_Q = .5 – DRIVER GAIN
K_P = 10 – DAMPING GAIN
K_3 = 10 RAD/SEC – FLOW CONTROLLER LAG
β/V_0 200 LB/IN^5 – FLUID CHARACTERISTIC
A = 42.99 IN^2 – EFFECTIVE AREA

M = 127 LB-SEC^2/IN – TRACK MASS
D = 500 LB-SEC/IN – ESTIMATED DAMPING
K_C = 7000 LB/IN – CABLE SPRING RATE
SATURATION LIMIT ON Q = 1800 IN^3/SEC
HYSTERESIS = ∓ 7000 LB
$\dot{C}(t)$ = -10.6 COS t – DRIVING FUNCTION (EXTERNAL)

THIS DIAGRAM IS THE BASELINE COMPUTER MODEL AND IS SUFFICIENTLY COMPLEX TO DESCRIBE THE OPERATION WITH REASONABLE ACCURACY.

Figure 2.5.2-2. Block Diagram of Hydraulic Cable Tensioner saturation component added to limit the rate at which the flow can be varied. The saturation level is a function of the hydraulic pump control servo. The stepper motor/rotary valve-type actuator on current tensioners has a limited response rate which is a significant limiting effect when rapid response is required. If the stepper motor were replaced by a faster responding servo-positioner or a servo-valve were used to regulate hydraulic fluid flow in place of the hydraulic pump, more rapid changes in flow would be possible. This would raise the level of the saturation component but would involve some increased cost in economic and reliability terms due to hydraulic complexity. The cable tensioning track mass and frictional damping effects have been combined into a single block and are preceded by a hysteresis loop. This hysteresis is inherent in all machines of this type and arises from friction in the gearing and track components. The hysteresis of a typical large pipe tensioner could be approximately 3.2 Mton (7000 lb) at the cable tension levels expected in the HDWC program. Any significant hysteresis causes an oscillation as the cable tensioner cycles back and forth through the hysteresis loop even though additional compensation is used to inhibit it. The pressure feedback loop shown in the block diagram was added to provide this additional compensation. A pressure transducer senses the system differential pressure and since only fluctuations in pressure about a mean value are required, the steady state pressure is subtracted from it. This is easily accomplished because it is essentially proportional to steady state cable tension. The only other difference between the simplified and more complex models is that the relative cable input motion was changed to a velocity to facilitate modeling of a steady payout velocity (forward cable vessel speed) superimposed on a sinusoidal heave motion. More realism can be added as is described later but this is a reasonable simulation model in that all major characteristics are represented.

The assumed or selected parameter values used in the analysis are based on typical equipment values and are shown in Figure 2.5.2-2. Some values varied from one simulation to another as described later.

TENSION SENSING

One important difference exists between this cable tensioner model and standard pipe tensioner design. The standard pipe tensioner has the entire machine flexibly mounted and pipe tension is sensed by measuring the force between the machine frame and a rigid deck foundation. A load cell is used to measure frame force which is directly proportional to pipe tension on the machine. The block represented by K_Q describes the measuring of cable tension and amplification of the small electrical signal to provide a useful control signal. If this type of cable tension sensing were used in the cable tensioner for the HDWC program, the dynamic response of the sensor itself would become a dominant determinant of machine operation. This is because the mass of the machine must move before a tension signal is developed by the load cell. It was proposed that a separate line tension measuring device (dynamometer) be used which would demonstrate a frequency response considerably higher than the characteristic frequency of the track system. Consequently, cross-coupling effects between the tension sensor and tensioner were ignored at this stage of the design study.

2.5.3 SIMULATION RESULTS

The goal of the modeling and simulation effort was to develop the control system of the cable tensioner so that cable tension could be maintained within 10 percent of the command value. This 10 percent figure is a judgmental value based on previous experience and the difference between maximum and suspended cable tensions (shown in Table 2.3-1). The remainder of the cable's excess tension capability is allowed for other vessel unknown factors and safety margins. Using the pressure feedback scheme described above, this can be accomplished by adjusting the system gains. The resulting cable tension response is much better than 10 percent, especially if the practical limitations of the hydraulics are ignored or assumed to be improved by further design effort as described earlier. Figure 2.5.3-1 shows the approximate practical limit of the system for standard hydraulic components and results in a cable tension variation of 4 percent. The input, or disturbance, here is a sine wave with an amplitude of 152 mm (6 inches) and a period of about 6 seconds, superimposed on a constant, 254 mm per second (10 inch/s) payout velocity. This is equivalent to a vessel heaving while moving forward and is considered the typical environment. The cable tension varies only about 4 percent from the 45.4 Mton (100,000 lb) command value, while the pressure and flow fluctuate at about 3 cycles per second due to system resilience within reasonable limits.

Figure 2.5.3-2 shows the response to identical conditions, using a higher forward loop gain and lower pressure feedback gain. The cable tension following is almost perfect but the large high frequency pressure fluctuations and the rapid changes in flow are beyond the capabilities of the standard hydraulic system. A servo-valve type of control would achieve this performance and has been used in the past, but the increased hydraulic activity is not desirable for hydraulic component reliability reasons.

RESULTS OF GAIN VARIATION

Using figure 2.5.3-1 as a baseline (KQ = 0.5, KP = 10), the effects of varying the gains are shown in Figures 2.5.3-3 to -6. Reducing KQ by half results in little change (for these operating conditions), while doubling it produces a ripple in the cable tension response and has a non-periodic effect on flow and pressure. Reducing kP by one-half gives a slight improvement in cable tension response but increases the amplitude and the period of the pressure fluctuations. Doubling KP results in a large, low frequency ripple in cable tension. The hydraulic pressure fluctuations are greatly reduced in amplitude but are increased in frequency. The hydraulic flow response becomes a small, high-frequency

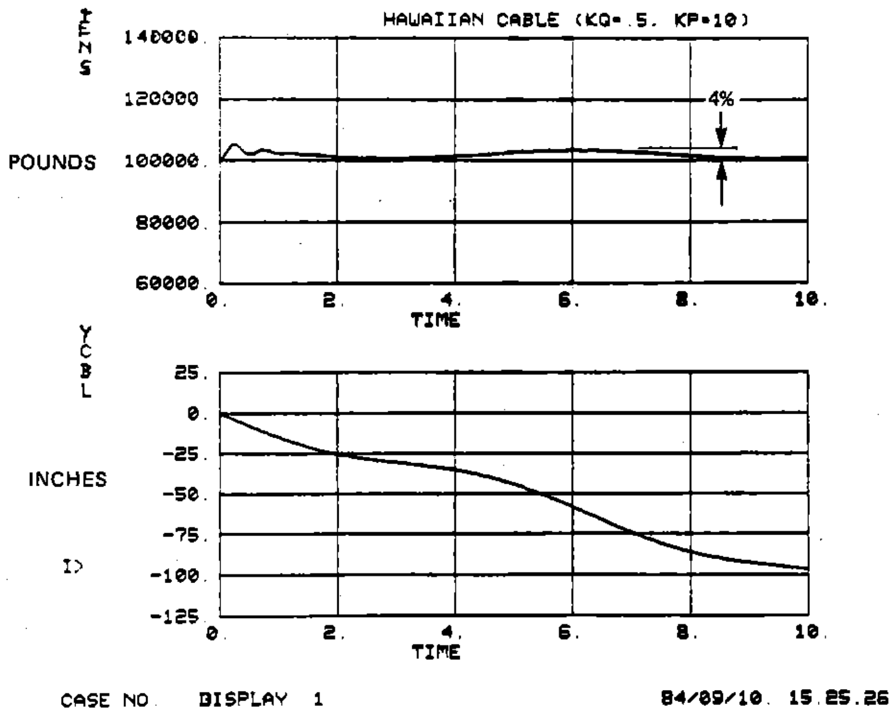
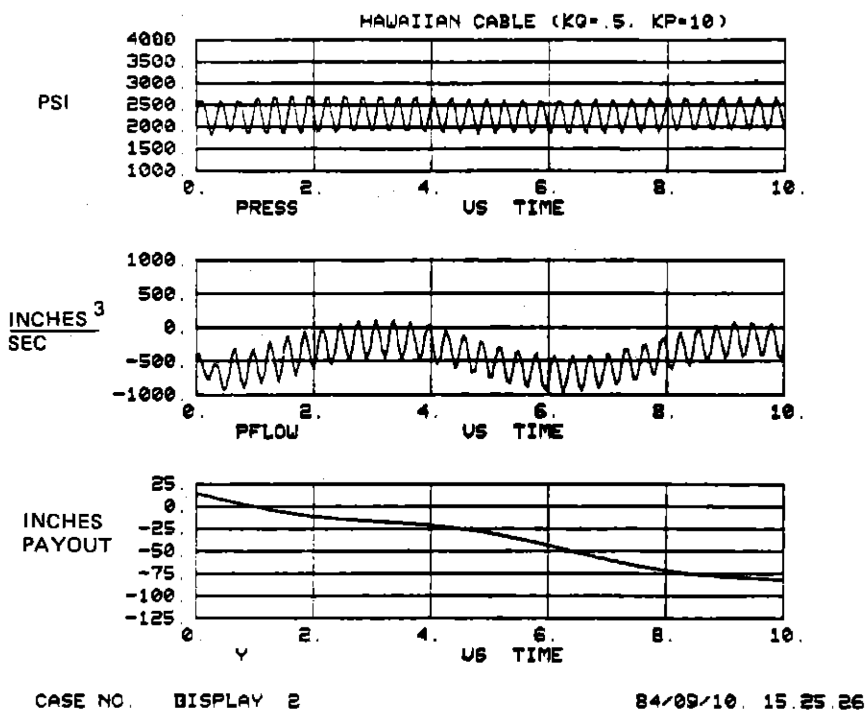
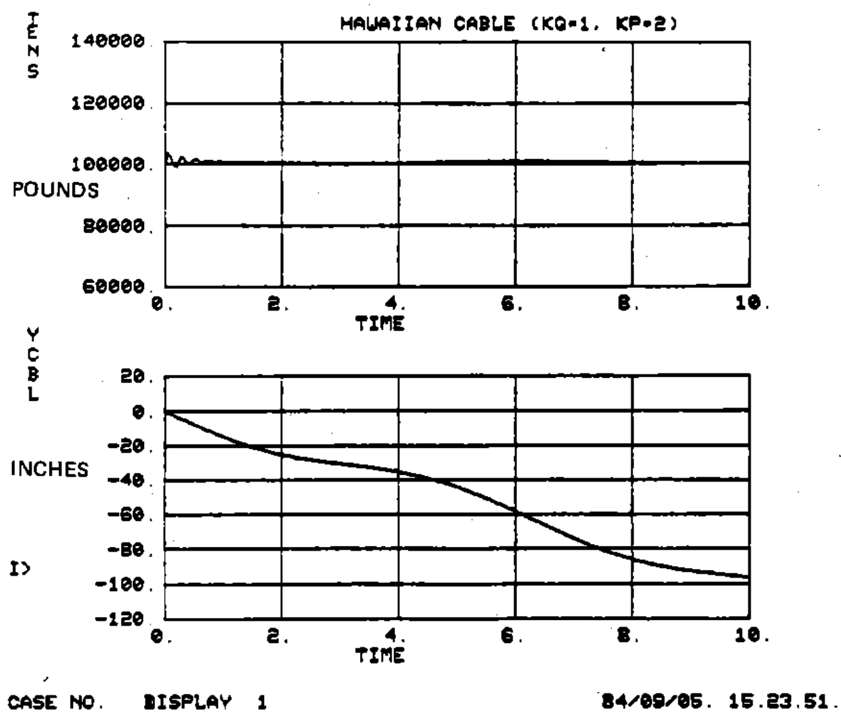


FIGURE 2.5.3-1
Linear Tensioner Responses

This represents an optimum design of hydraulic drive with gains set for best responses at this condition.

The initial waveform results from a step command and represents a limiting condition response.





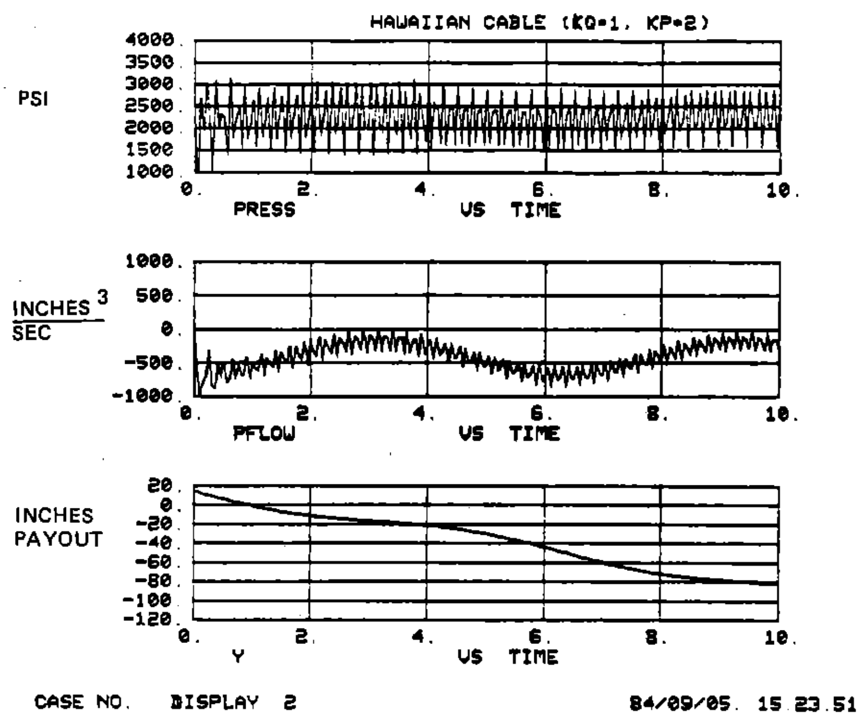
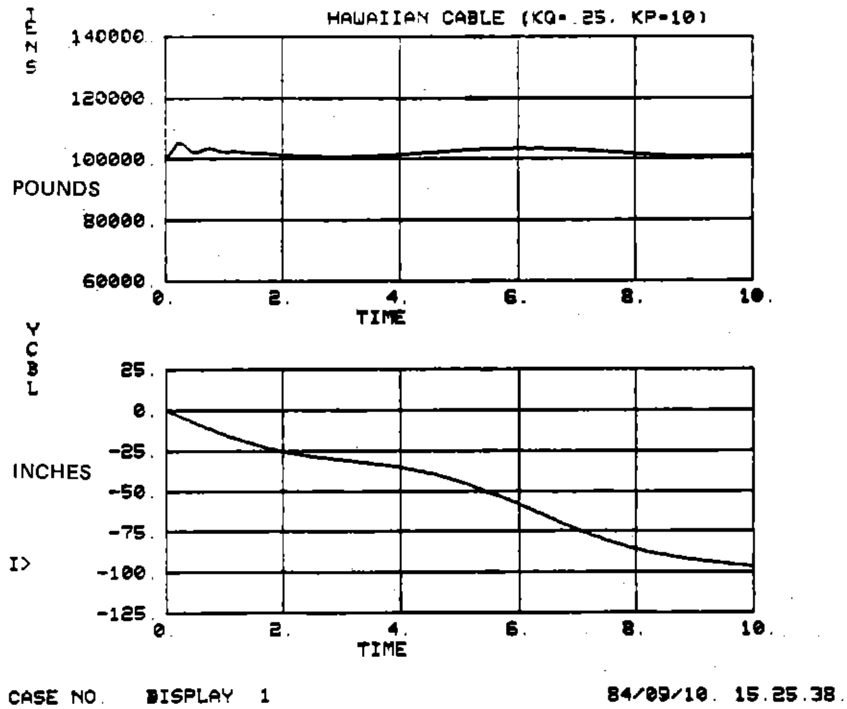


FIGURE 2.5.3-2

The responses here represent the possible improvement of tension by varying gains, but the increased hydraulic activity is not desirable for component reliability reasons.



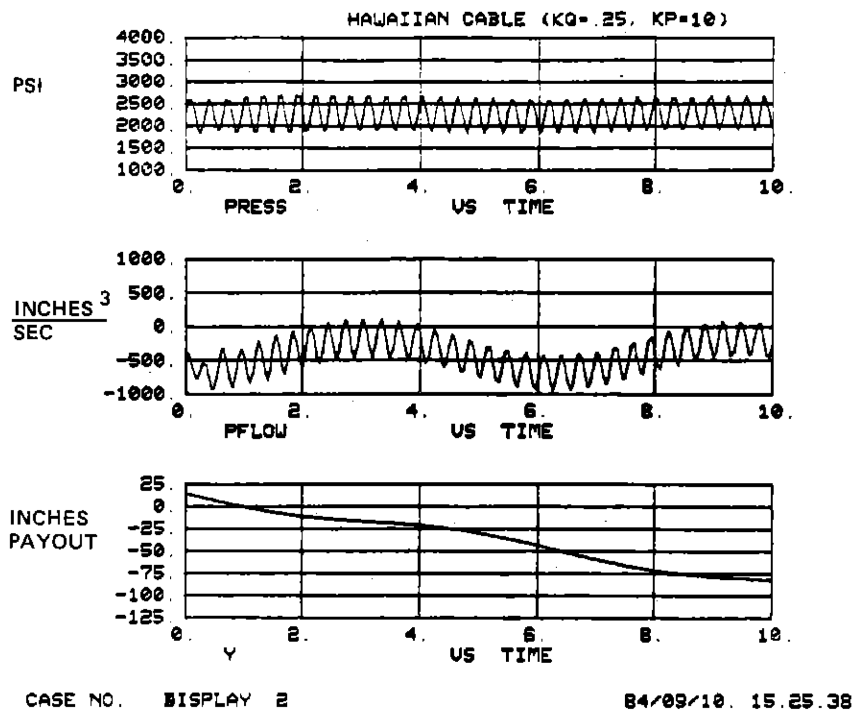


FIGURE 2.5.3-3

This response represents the slight degradation resulting from a reduced loop gain.

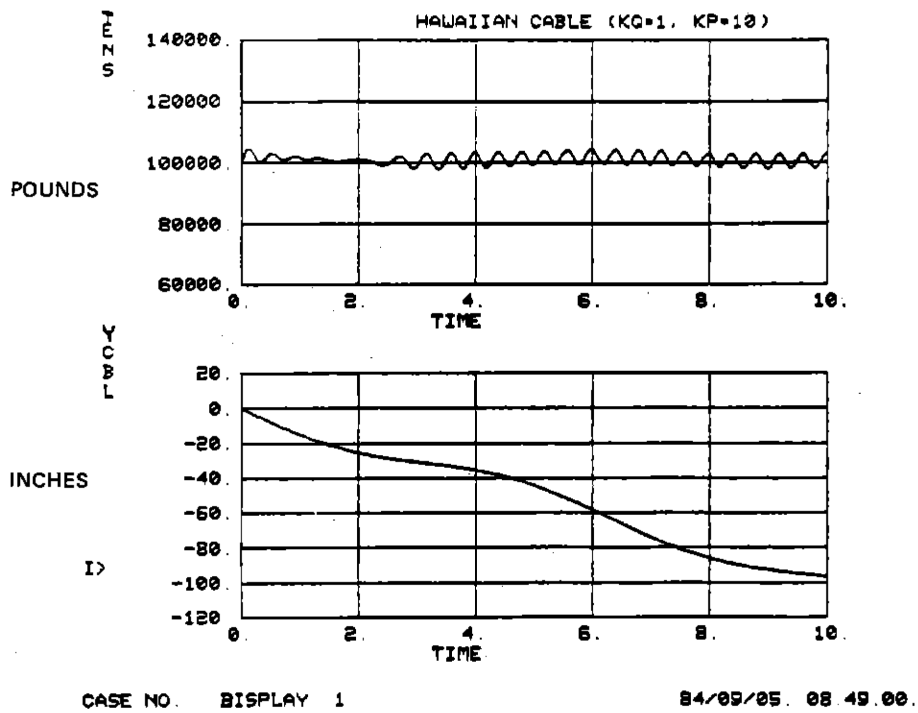
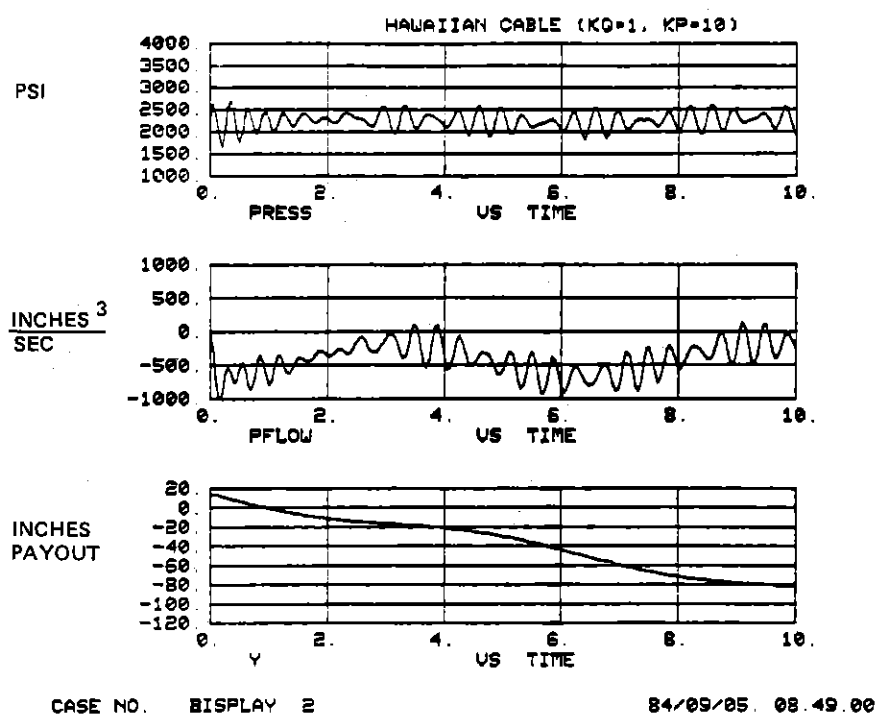


FIGURE 2.5.3-4

A higher gain reduces stability significantly



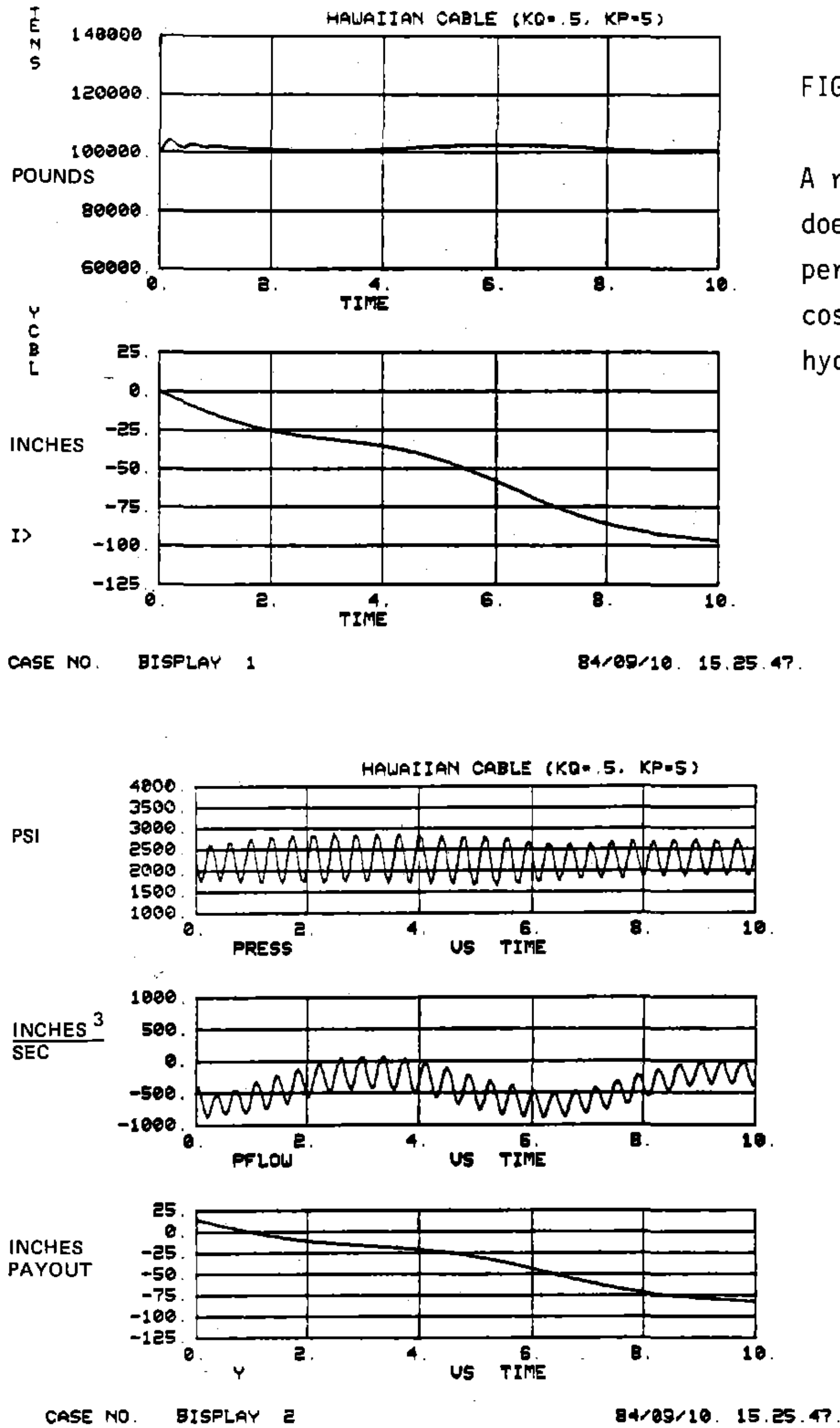


FIGURE 2.5.3-5

A reduction in damping does improve tension performance but at some cost in increased hydraulic fluctuations.

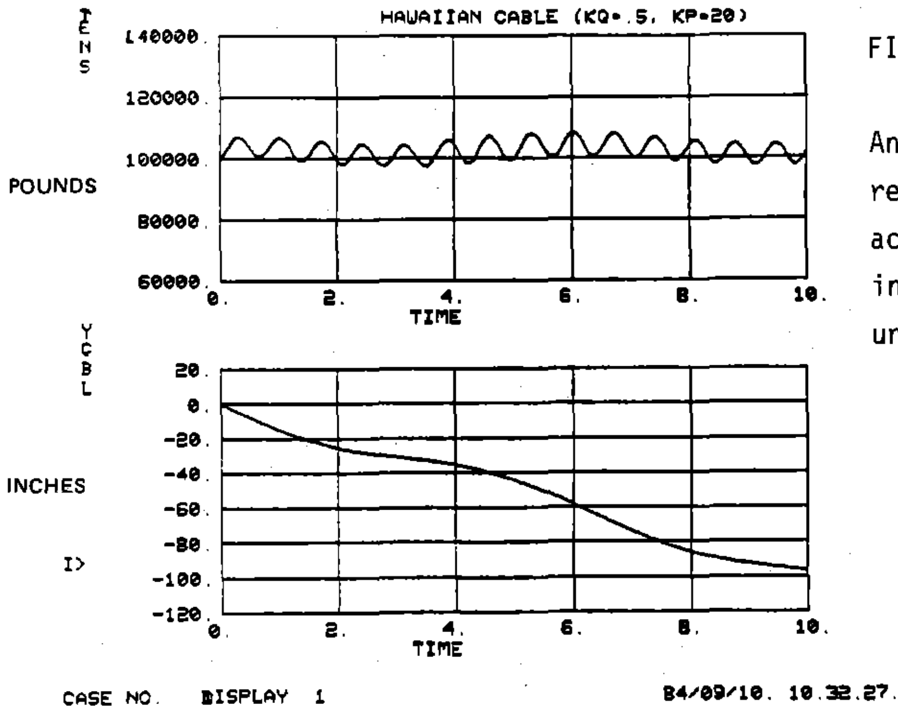
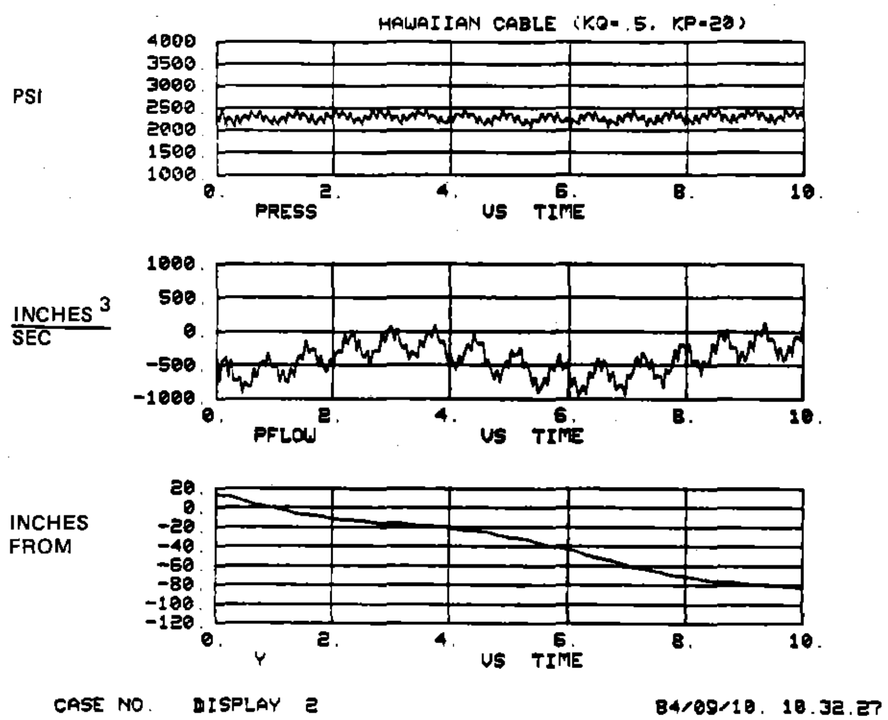


FIGURE 2.5.3-6

An increase in damping reduces hydraulic activity but results in an essentially undamped track mass.



disturbance superimposed on a larger, low frequency oscillation. It appears that there are significant trade-offs which may preclude the possibility of obtaining both a smooth cable tension response and a smooth response of hydraulic pressure and flow with one combination of gains. However, the interplay of the two gains is complex and it is conceivable that further optimization can be achieved. The EASY5 program lends itself to such tasks.

HYSTERESIS EFFECTS

The response while heaving with no forward velocity is shown in Figure 2.5.3-7. The cable tension response is cyclical because of the hysteresis loop which must be overcome every time the command changes direction. This was not the case in the previous simulations where the constant payout velocity exceeded the heave velocity so the net cable velocity never changed direction. The cable tension variations do not exceed 7 percent in this simulation which occurs whenever the net cable payout speed is less than the cable speed required to compensate for vessel heave.

GAIN SCHEDULING

One of the most important factors affecting cable tensioner response is cable stiffness which is directly related to the length of cable suspended, i.e. depth of the water. In shallow water, the relatively short cable length is much stiffer than the long length of cable in deep water. The tension command will also vary with depth so that, as the cable length decreases and the stiffness increases, the commanded tension decreases. Using the baseline gains of Figure 2.5.2-2, the effects of varying the cable tension and stiffness are shown in Figures 2.5.3-8 and 2.5.3-9. At greater depths (increased tension and cable length and decreased stiffness), the cable tension takes on a mild oscillation, and the hydraulic pressure and flow variations are reduced. With a short cable causing high stiffness and reduced tension due to decreased depth,

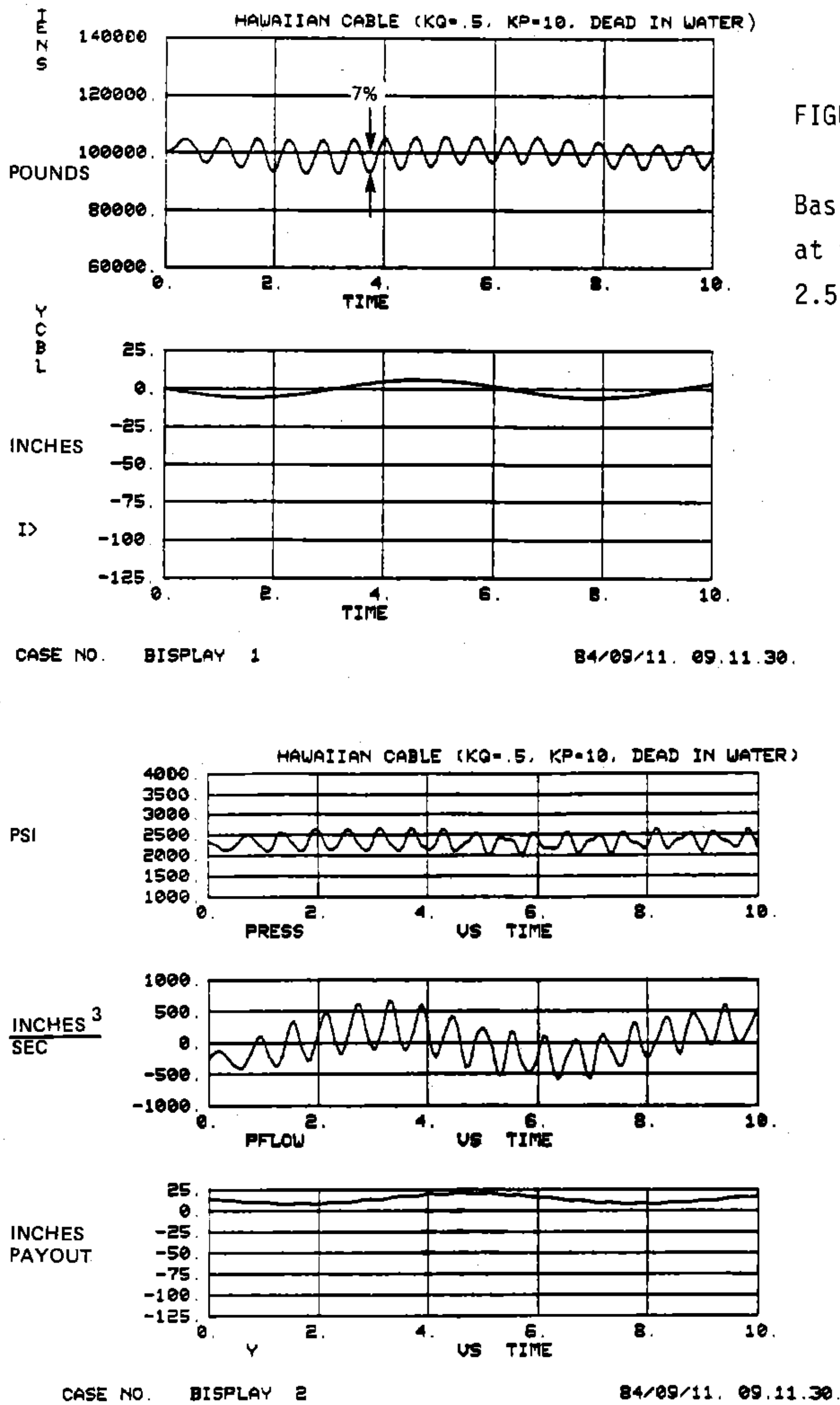
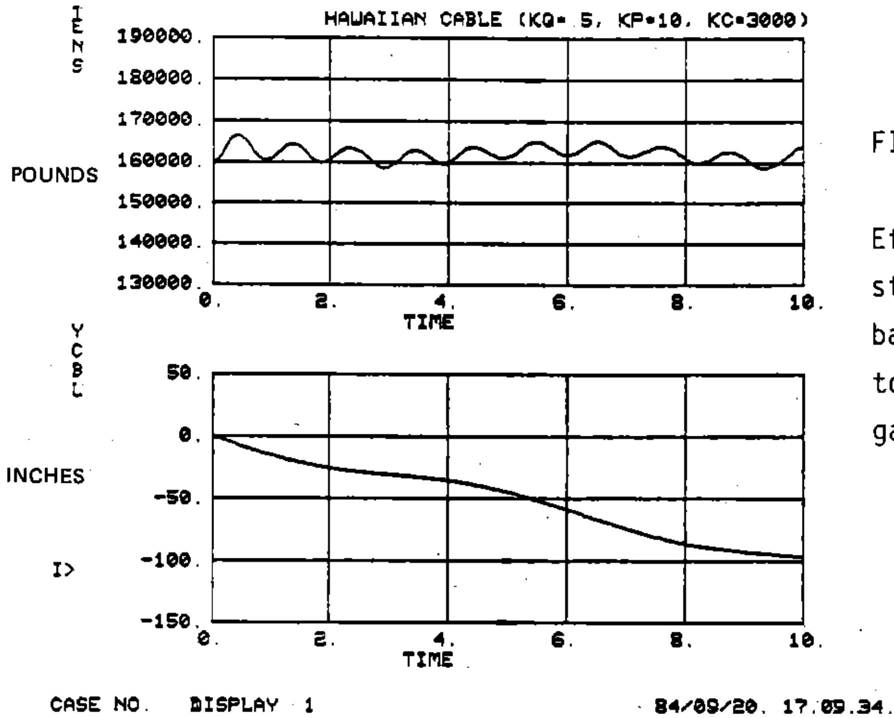


FIGURE 2.5.3-7

Basic tensioner operating at rest (compare to 2.5.3-1).



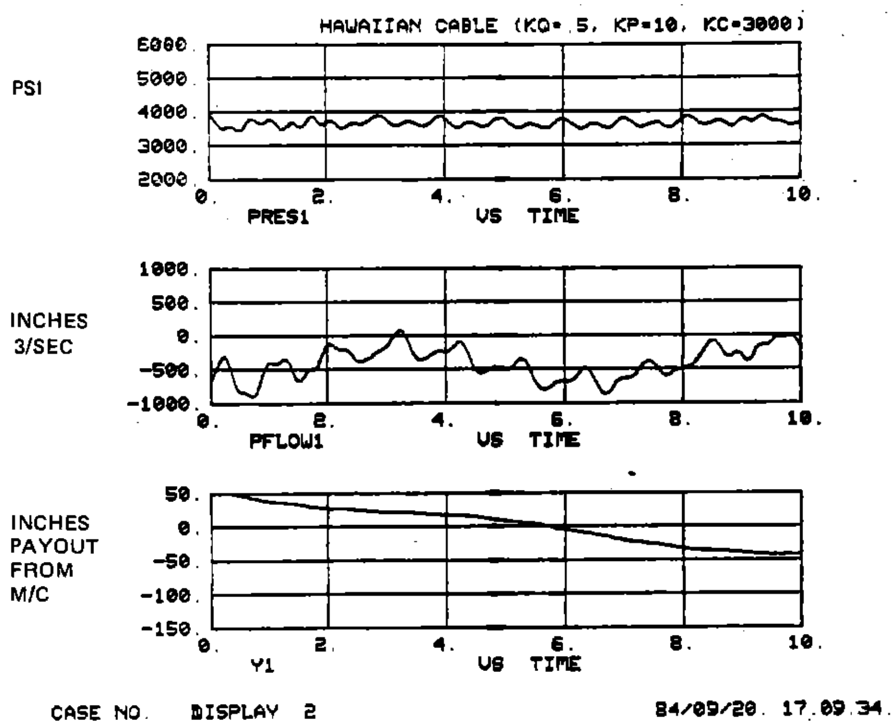


FIGURE 2.5.3-8

Effect of reducing stiffness of cable at basic gains is similar to that of reducing loop gain.

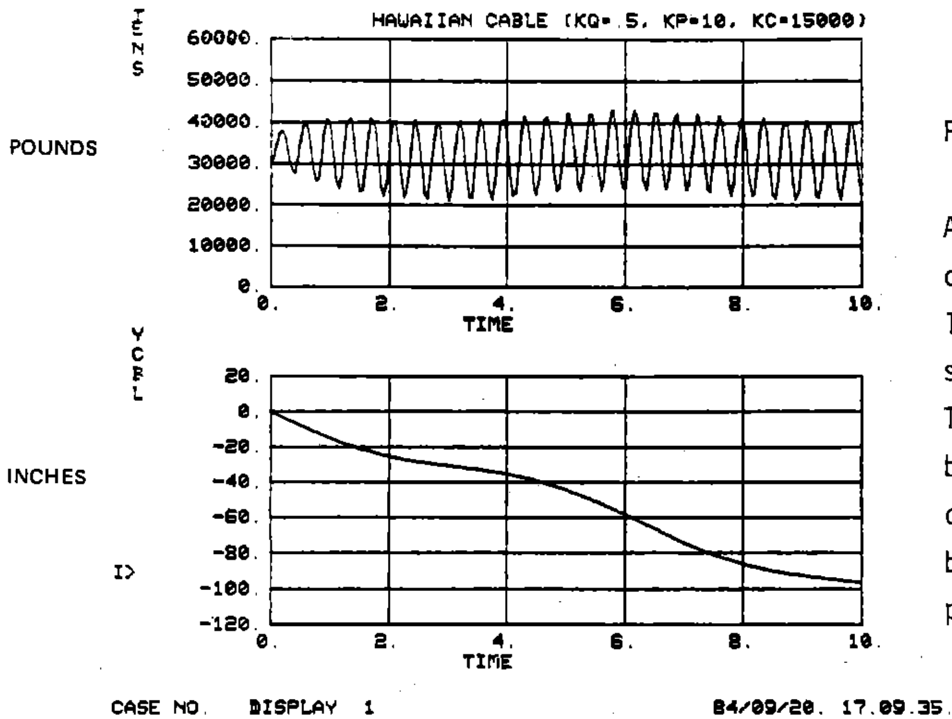
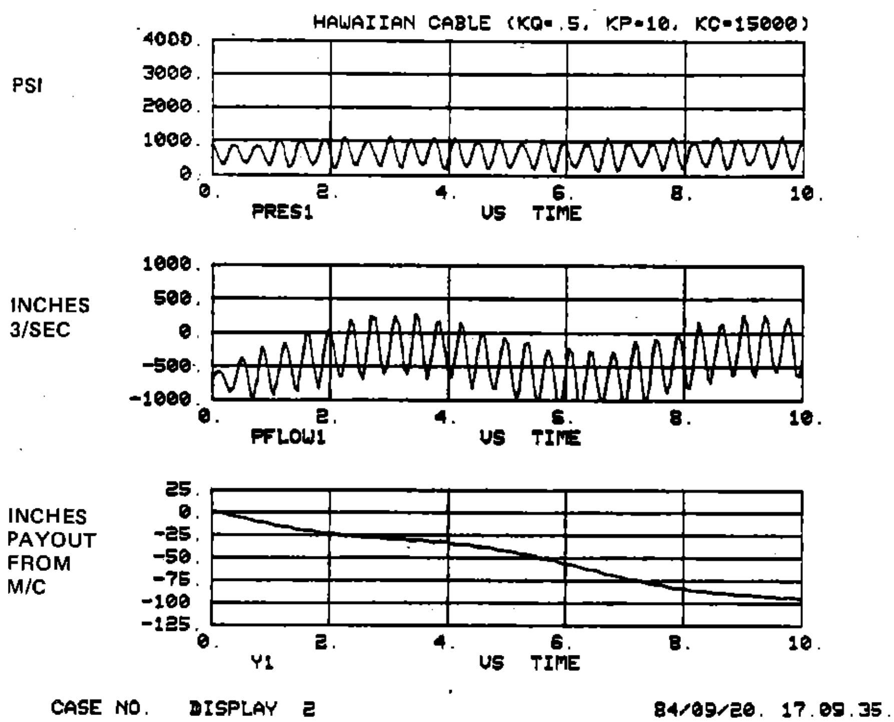


FIGURE 2.5.3-9

A marked increase in cable stiffness increases loop gain and drives system into instability. This appears dramatic but gain and damping changes will produce a better control in a practical system.



the response is oscillatory and eventually becomes unstable as the resultant stiffness is further increased. To maintain optimum response, the gains must be adjusted as the suspended cable length varies by scheduling the gains to the depth of the channel or by another convenient function of the vessel's integrated control system.

Another factor which significantly affects cable tensioner response is the frequency of the input motion (heave frequency). All of the preceding plots were based on an input motion described by 6 x sine(time). Figure 2.5.3-10 shows the effect of doubling the frequency while halving the amplitude [3 x sine(2xtime)]. The cable tension response remains good and the pressure behavior is virtually unchanged. The hydraulic flow is also unchanged except for the requirement of heave compensation at twice the frequency.

SYSTEM FREQUENCY RESPONSE

The complete frequency response is shown in the form of a Bode plot in Figure 2.5.3-11. In this plot, the transfer function was generated with the cable motion as input and the tension as output. As an example from this plot, for a 1 rad/sec sinusoidal input the gain is 47 dB or 224 ($20 \log_{10} 224 = 47$), so that the amplitude of the output (tension) is 224 times the amplitude of the input (cable motion). If the heave amplitude is plus or minus 154 mm (6 inches), then the tension variation is plus or minus $\frac{154}{\text{deg/rad}} \times 224 = .6$ Mton (1340 lb). As the frequency of the input motion increases toward the resonant frequency of 12 rad/sec, the amplitude ratio also increases. However, the amplitude of the input motion will decrease as its frequency increases so this does not necessarily pose a problem. An input motion at 12 rad/sec would have to have an amplitude of 13.7 mm (.5 inch) to result in a cable tension variation of 4.5 Mton (10,000 lb). This would be a severe input motion, equivalent to an acceleration of .25 g at 2 cycles per second.

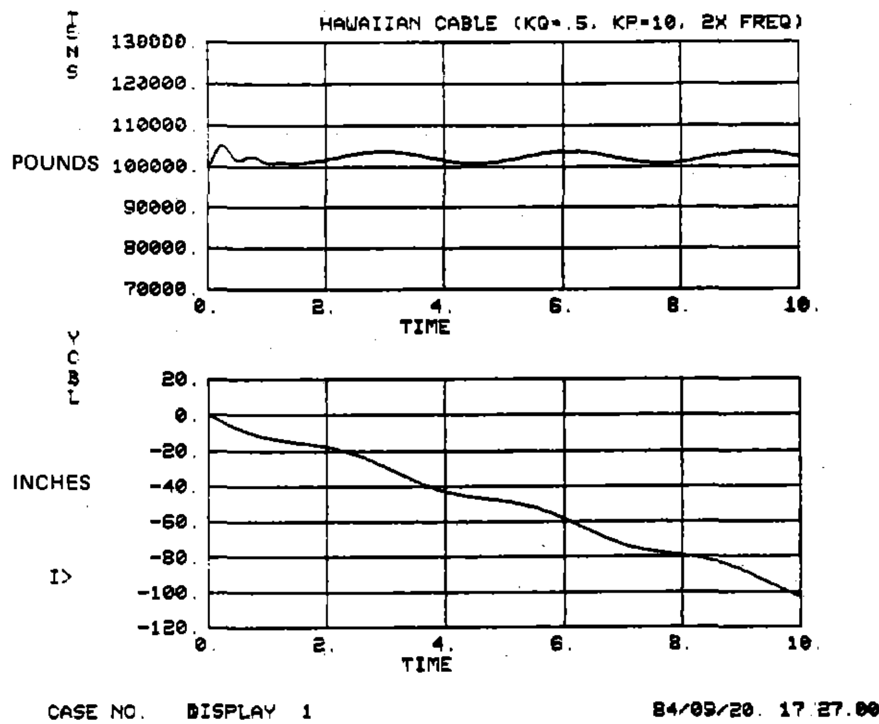
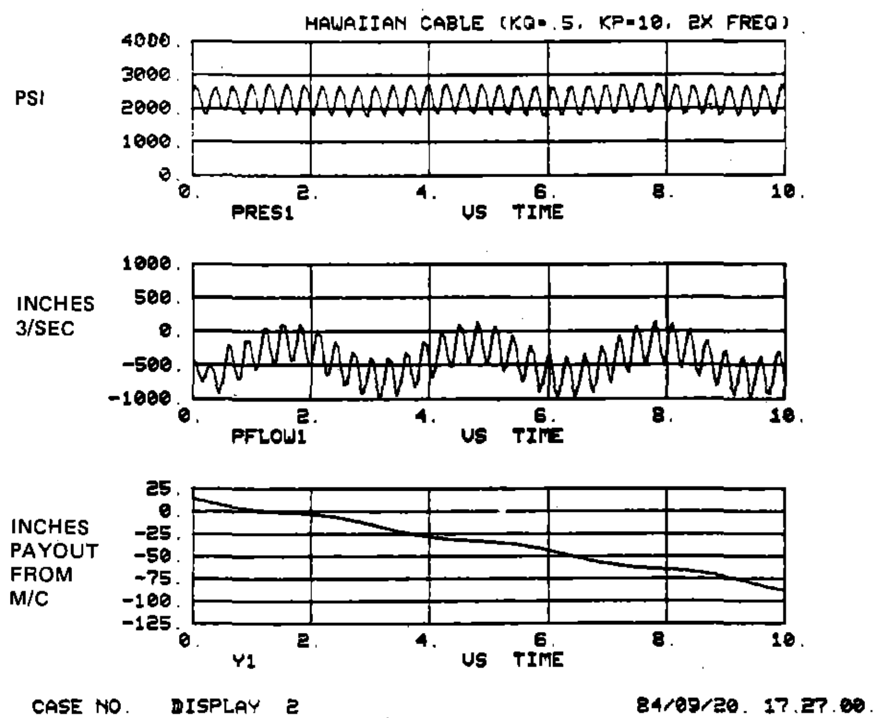


FIGURE 2.5.3-10

This represents the basic tensioner operating in choppier seas and performance is virtually unchanged.



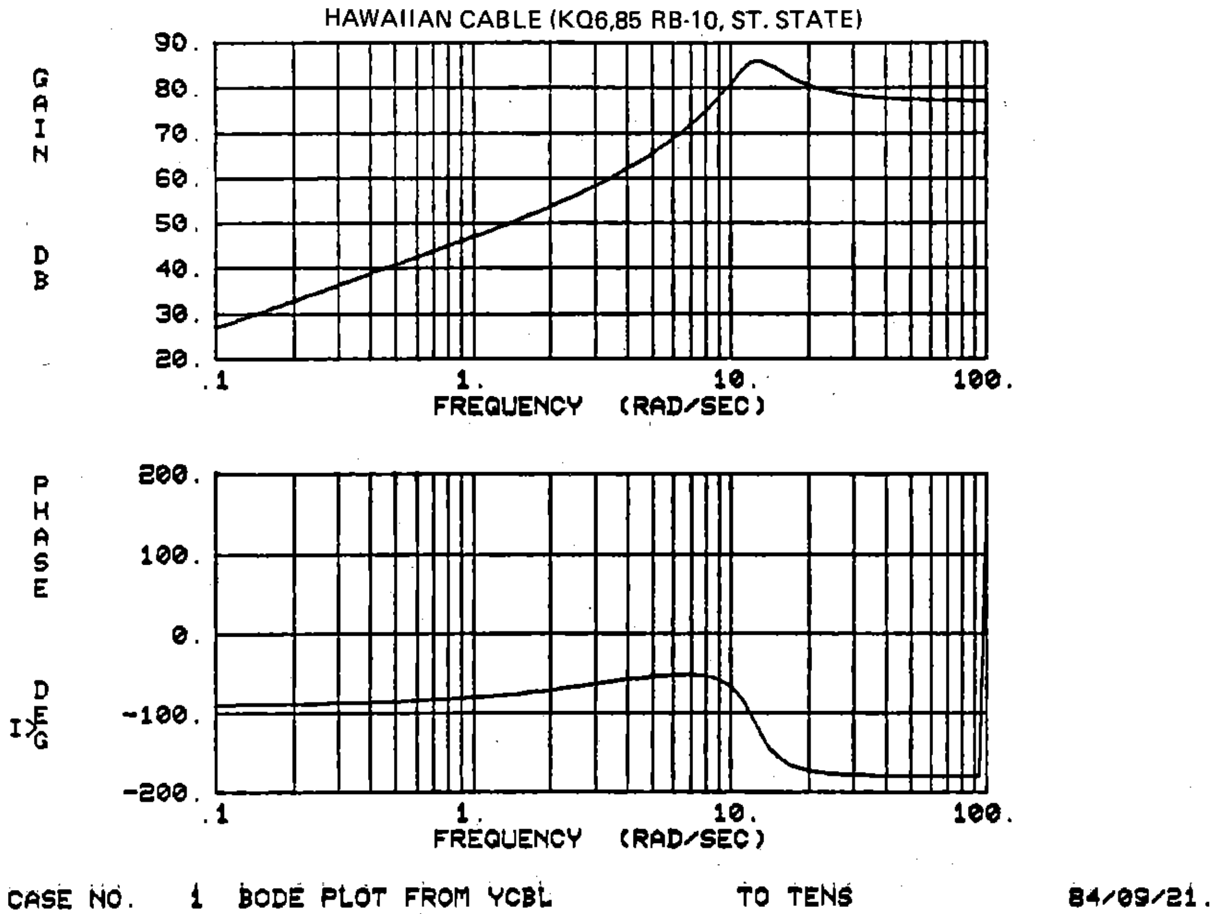


FIGURE 2.5.3-11

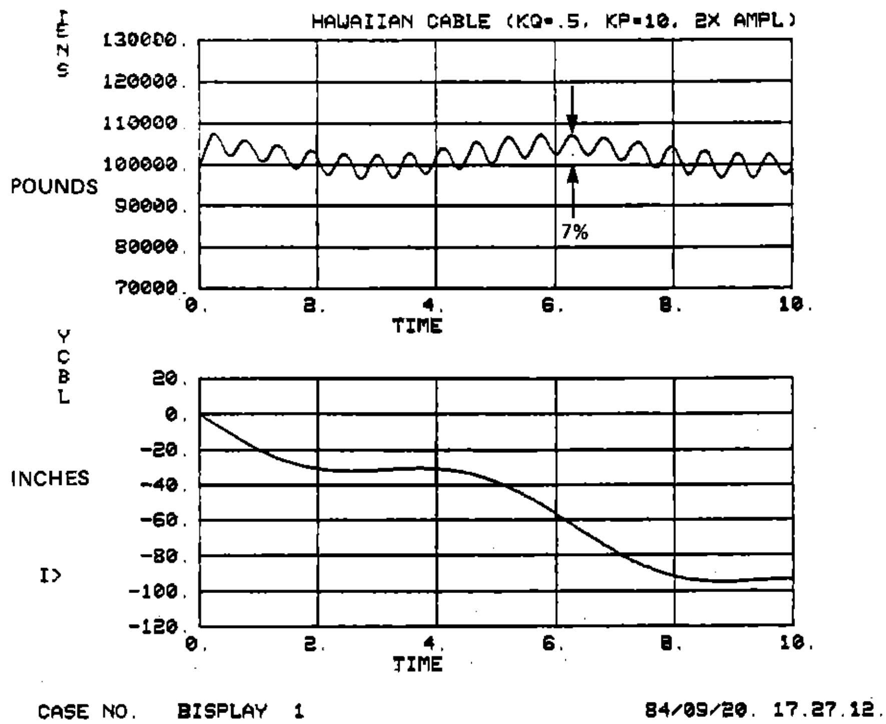
These plots are the basic tensioner response characteristics in terms of tension as a function of rate input versus frequency. Response is satisfactory out to about 12 rad/sec (2 Hz), which was used as the design study goal.

If the amplitude of the heave motion is increased to the point where the cable tensioner is required to reverse direction to maintain cable tension, then the behavior shown in Figure 2.5.3-12 is the result. The oscillatory response is similar to the zero-velocity case (Fig 2.5.3-7) since they result from the same cause: When the cable tensioner reverses direction, it must cross through the hysteresis. Thus, for best cable tension compensation performance the vessel's forward velocity should remain high enough so that the dynamic motion of the cable does not cause a reversal in cable tensioner direction of travel. This means the cable tension variation increases from 4 percent (Fig. 2.5.3-1) at speed to 7 percent (Fig. 2.5.3-7) when the net cable speed is zero. Both conditions have tension variations that are less than the initial goal of 10 percent.

2.5.4 ALTERNATIVE TENSIONER DRIVE

The linear cable tensioner may be powered by either hydraulic or D.C. electric drive motors. Historically, D.C. electric drives have been used to power cable machinery where the application does not require dynamic tension holding or compensation. D.C. electric drives are not suitable for extended operation at low or zero speed due to uneven commutator heating. For this reason, most D.C. electric motor driven cable machinery is speed controlled utilizing brakes to hold loads.

D.C. electric motors have large rotative inertias, as compared to hydraulic motors. This motor inertia reduces the machine's dynamic response when used in a tension holding application. Figures 2.5.4-1 and 2.5.4-2 are response curves of identical linear tensioners; one hydraulically driven, the other D.C. electrically driven. Neither of these curves is applicable to the HDWC program, but is shown only for the hydraulic vs. D.C. electric drive comparison. The primary reason for the 17 to 25 percent tension variance in the electrically driven tensioner is due to the motor armature inertias. The inertia variance can be overcome



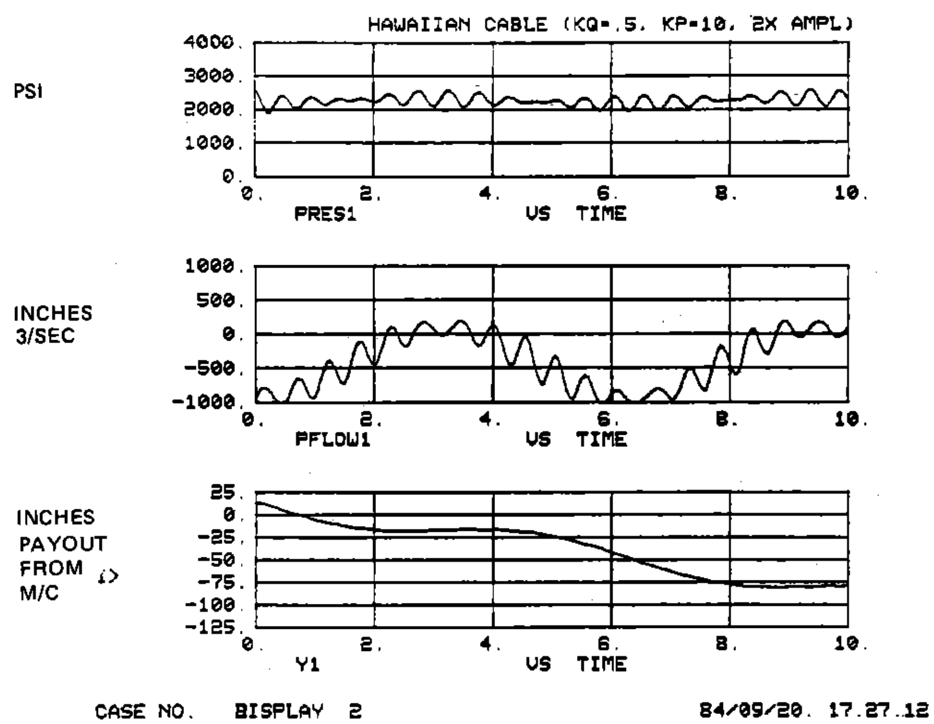


FIGURE 2.5.3-12

Tensioner responses when payout rate reversal occurs.

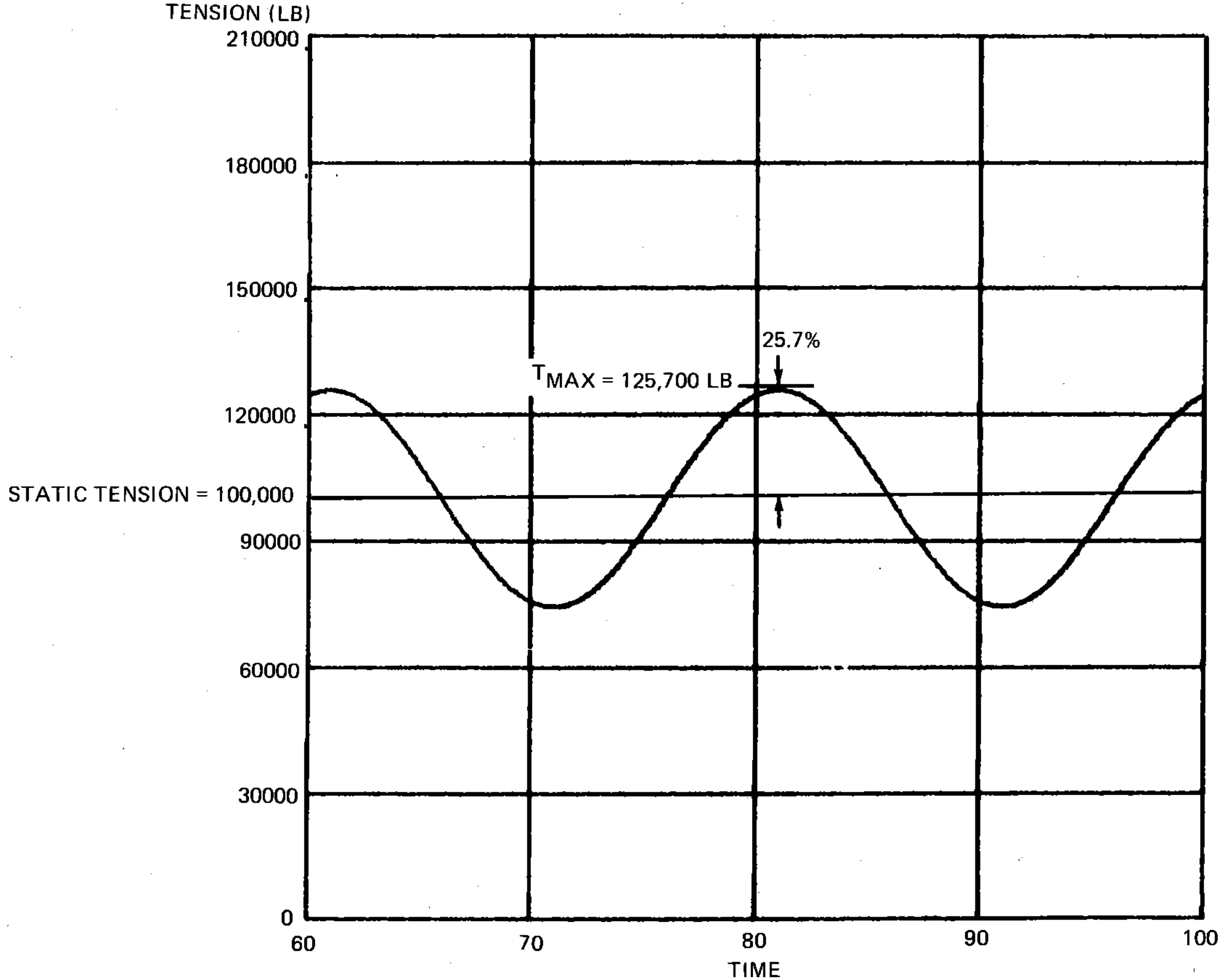


Figure 2.5.4-1. Simulation

Variation in cable tension due to heave Ampl. = 6 inch and period = 20 sec.
Cable laying system with Electric Drive System (D.C. Motor)

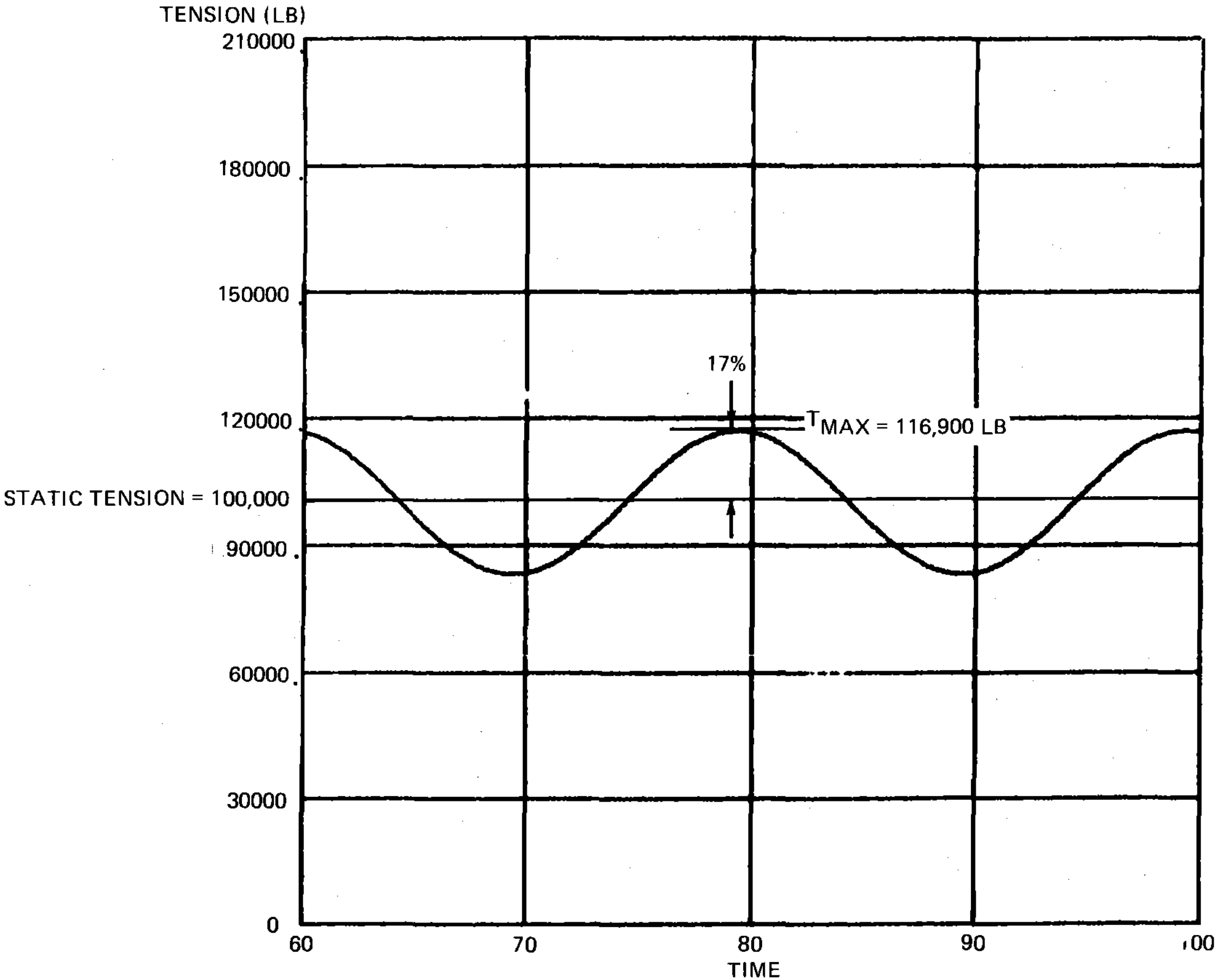


Figure 2.5.4-2. Simulation

Variation in cable tension due to heave Ampl. = 6 inch and period = 20 sec.
Cable laying system with Hydraulic Drive System.

by the addition of inertia compensation to the control system. Inertia compensation is a form of gain scheduling, which is related to the motor speed rate of change. While this improves dynamic response, it requires extra capacity in oversized motors, motor controllers, and power generators.

The D.C. electric motor driven cable tensioner was not chosen for the HDWC program at-sea tests due to its dynamic response and poor tension holding at zero speed applicability.

2.5.5 DYNAMICS STUDY CONCLUSIONS

SYSTEM REQUIREMENTS FOR SUITABLE RESPONSE

Up to this point, the simulation study program has primarily investigated the achievement of a dynamic response that will provide the tension compensation (within 10 percent) required by the HDWC program. A secondary task was to identify major difficulties or risks. Following are guidelines for the machine design:

1. A separate dynamometer must be used to ensure a fast response measurement of line tension.

2. A relatively high system stiffness is a major objective in the design of the hydraulic system.

3. A faster responding hydraulic flow control device may be required and various alternates should be investigated, ranging from replacing the stepper motor drive with a more powerful servo actuator to the use of a direct operating servo valve. No risk is associated with these alternatives, though there are difficulties associated with the large capacity hydraulic supplies required if servo-valves are considered.

4. Further study and model refinement must be completed to determine the need for gain scheduling in relation to cable tension and suspended cable length to optimize performance throughout the complete spectrum of operating conditions.

5. The dynamics study must be refined to consider other factors before a machine design is completed, including the limit of slip of the track gripper pads on the cable, the simulation of the cable as a distributed mass system, and the dynamic simulation of the overboarding device (see Section 3.2.3). Figure 2.5.5-1 is a modified block diagram which includes a simple mass simulation of the cable.

6. Final cable parameters of spring rate, friction, twist, gripping size, and strength must be obtained prior to completion of the machine design, to ensure that the validity of the simulation model is maintained.

Conclusions and recommendations for all of the cable handling equipment subsystem components are summarized in Section 4.0.

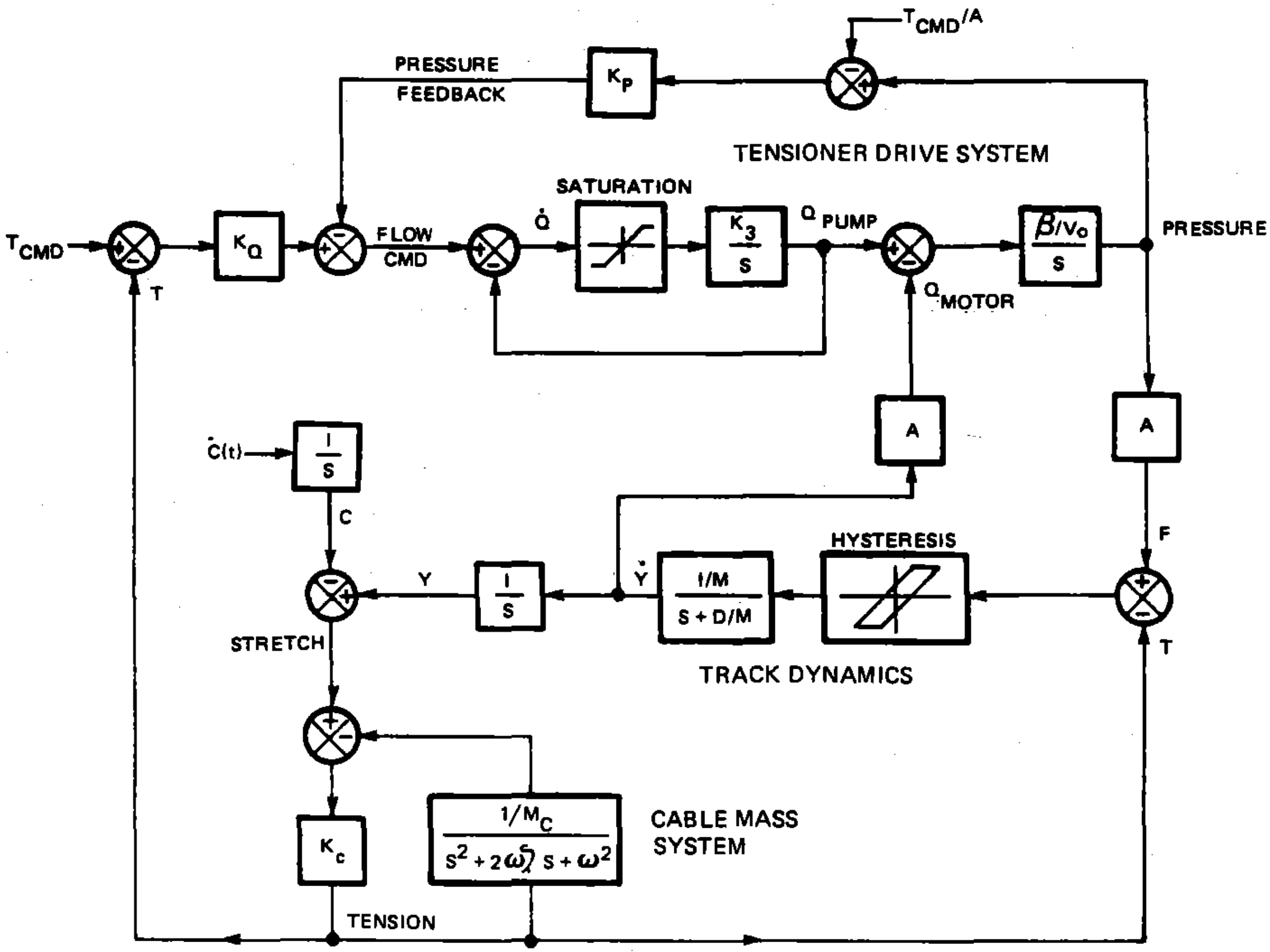


FIGURE 2.5.5-1 Functional block diagram of tensioner with a simple mass simulation for the cable.

3.0 CABLE HANDLING EQUIPMENT ANALYSIS

3.1 SURVEY OF EXISTING EQUIPMENT

The purpose of this survey of existing equipment is to explore the possibility of using currently existing cable laying machinery and cable vessels for the proposed HDWC program.

At this point, there are no complete lay vessels in existence that meet all the criteria required by the HDWC program. However, the technology base is available to build and equip such a lay vessel. The decision to be made is whether any existing vessel or machinery can be modified efficiently and economically to meet the needs of the project. Only one vessel is capable, with modification, of accomplishing the Hawaii inter-island cable lay and at-sea test lay. There are serious questions with this vessel in regard to suitability and availability to be modified.

The vessel considered for the HDWC program is the SKAGERRAK (see Figure 3.1-1) owned by the NVE State Power System of Oslo, Norway. This vessel was designed to lay power cable and has been used to lay lines between Denmark and Norway in the Baltic Sea and more recently in the Straits of Georgia. The ship, commissioned in 1976, is 99.7 meters long and 32 meters wide. It has an electrically driven storage turntable with an outside diameter of 29 meters and an inner diameter of approximately 12 meters. The turntable can accommodate up to 7,000 tons of cable with a maximum payout speed of 1.2 rev/min. Mooring winches, cranes, and the 5 meter diameter overboarding sheave are adequate for miscellaneous use on the HDWC program. The main overboarding sheave, cable tensioning and cable handling machinery, however, are insufficient for the HDWC program at-sea tests. The overhead pickup arm above the turntable is adequate for the cable load but does not have the flexibility to provide the cable twist control required for this project. The pretensioner,

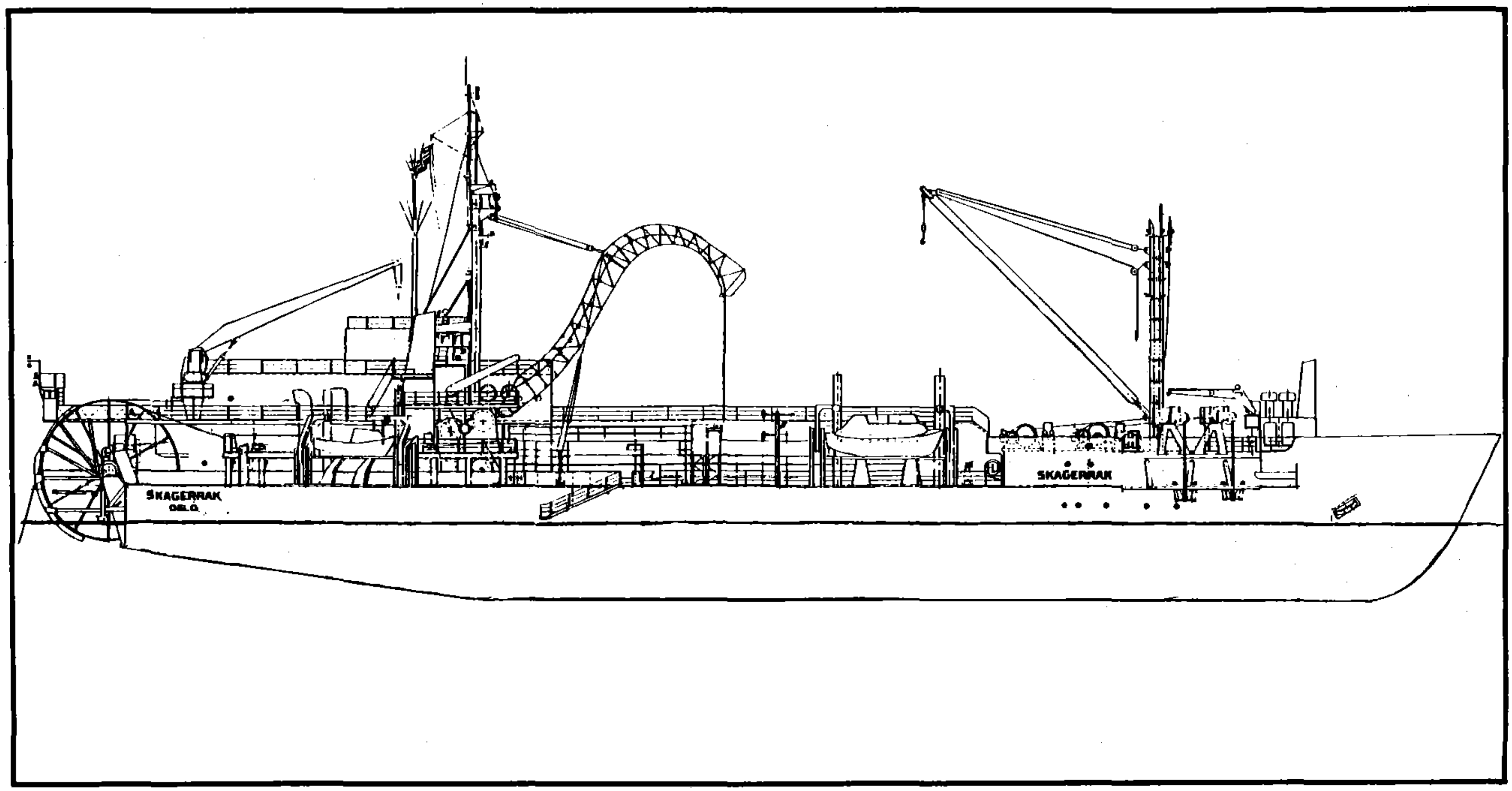


Figure 3.1-1. Skagerrak

cable capstans and linear cable engine are also insufficient in capacity to develop the tension required to lay cable in the Alenuihaha Channel.

Preliminary studies indicate that a linear tension machine ranging in length from 29 to 33 meters (reference section 3.3.3) would be required to handle the overboarding tension needs with the preliminary tension shear parameters of the cable. Given this length and the general outline of the SKAGERRAK, one method to accommodate the tensioner would be to construct an overboarding sheave off the bow instead of using the stern system as it exists. Currently there are only 15-24 meters available between the turntable and overboarding sheave at the stern. Such a modification has not been discussed with the owners of the ship and the Norwegian Power Board may not agree to such extensive modification because the vessel works well for its intended use.

Another power cable laying vessel was considered; the mothballed SUSITNA (Figure 3.1-2), owned by Chugach Electric of Alaska and used to lay the 230 KV Cook Inlet cables. This vessel is not recommended for the proposed HDWC program at-sea tests primarily because of its 15 meter by 50 meter size. The vessel's response to the required design sea state would be excessive, requiring more cable tension compensation than the cable tensioner could provide. Though the vessel has a small linear tensioner (5 MTON), the equipment of interest is the hydraulically driven turntable, which has the capacity (500 tons) that will be required for the HDWC program at-sea test. This is further discussed in Section 3.4.

This study attempts to evaluate the equipment required within the limitations of existing vessels. The conclusion drawn is that an entirely new barge or vessel is required to best meet the HDWC program's at-sea test requirements.

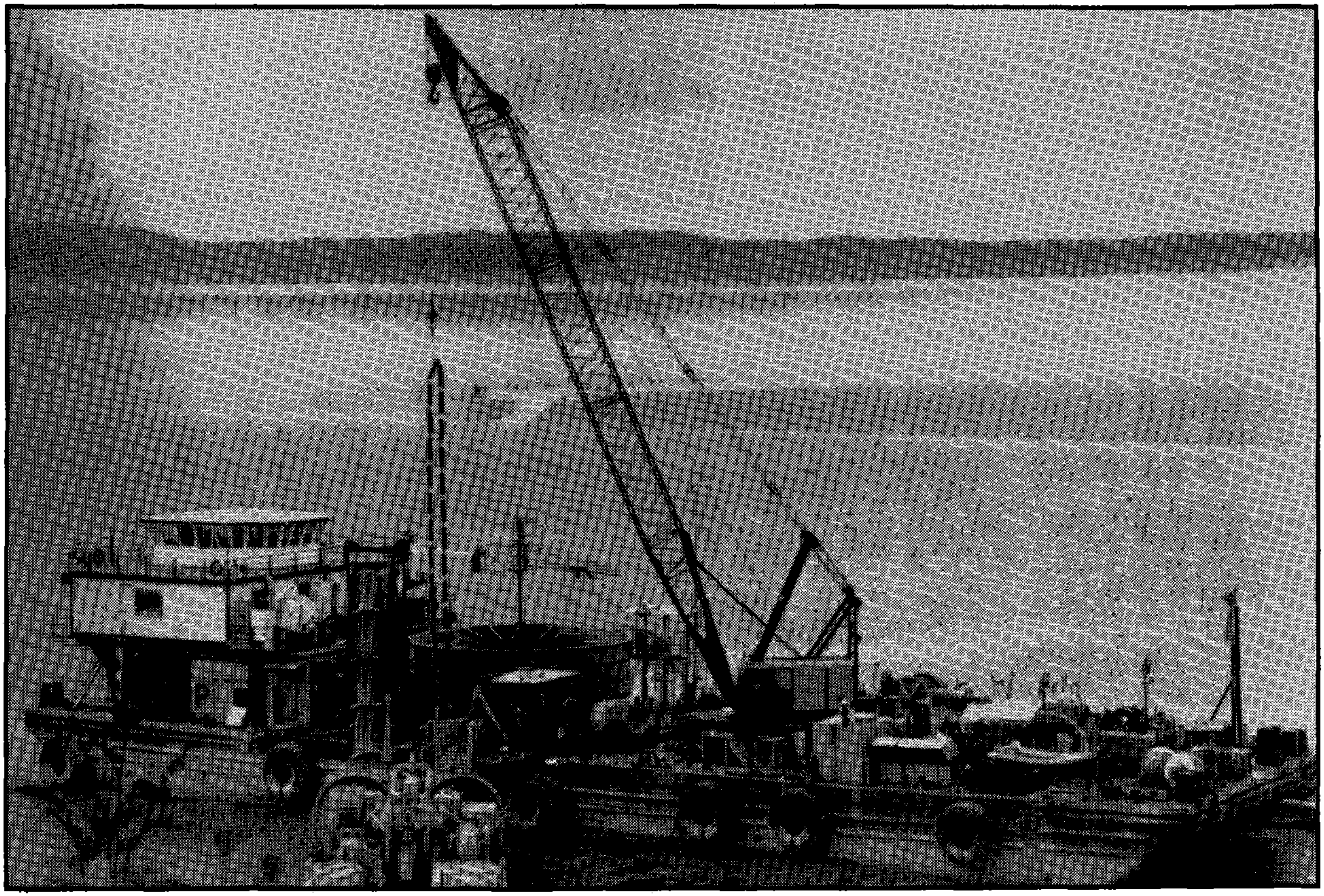
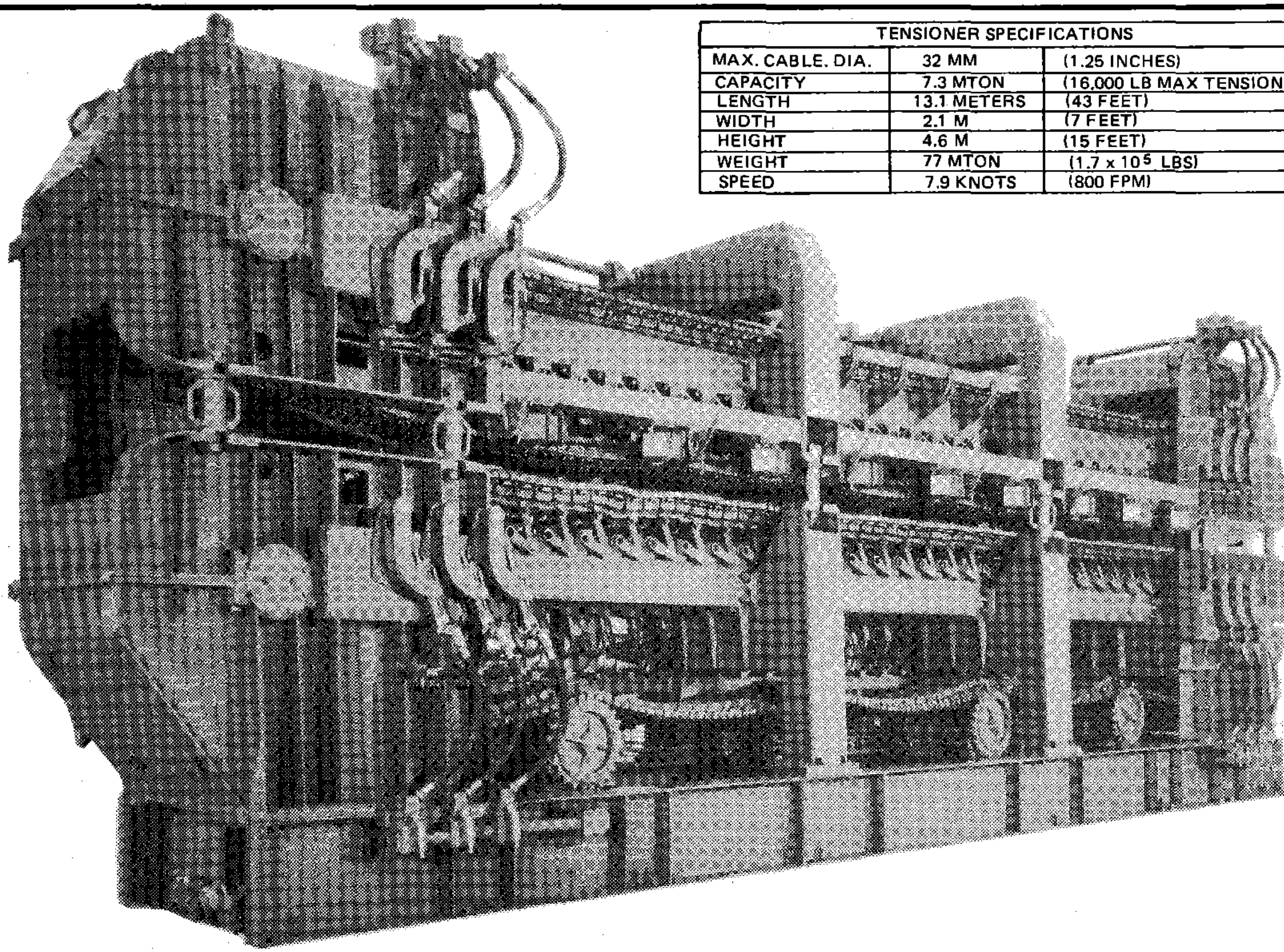


Figure 3.1-2. Susitna

MACHINERY

As modern submarine cable and pipe laying technology has evolved, tensioning and handling machinery has been developed to meet the increasingly complex and exacting requirements of the industry. These machines have been used in systems developing up to 204 Mton of tension. Figures 3.1-3 through 3.1-6 depict four of these tensioner systems.



TENSIONER SPECIFICATIONS		
MAX. CABLE. DIA.	32 MM	(1.25 INCHES)
CAPACITY	7.3 MTON	(16,000 LB MAX TENSION)
LENGTH	13.1 METERS	(43 FEET)
WIDTH	2.1 M	(7 FEET)
HEIGHT	4.6 M	(15 FEET)
WEIGHT	77 MTON	(1.7 x 10^5 LBS)
SPEED	7.9 KNOTS	(800 FPM)

Figure 3.1-3. Long Lines Cable Tensioner

T-ARC 7 CABLE SYSTEM		
MAX TENSION	45.4 MTON	(100,000 LB)
OVERBOARDING SHEAVES	4.1 METERS	(162 IN. DIA.)
CABLE DRUM DIAMETER	4.6 M	(180 IN) SELF FLEETING
CABLE SIZE	216-12 MM	(8.5 - 1.25 IN)
SPEED	6.9 KNOTS @ 4.5 MTON 1.5 KNOTS @ 45.4 MTON	(700 FPM @ 10,000 LB) (150 FPM @ 100,000 LB)

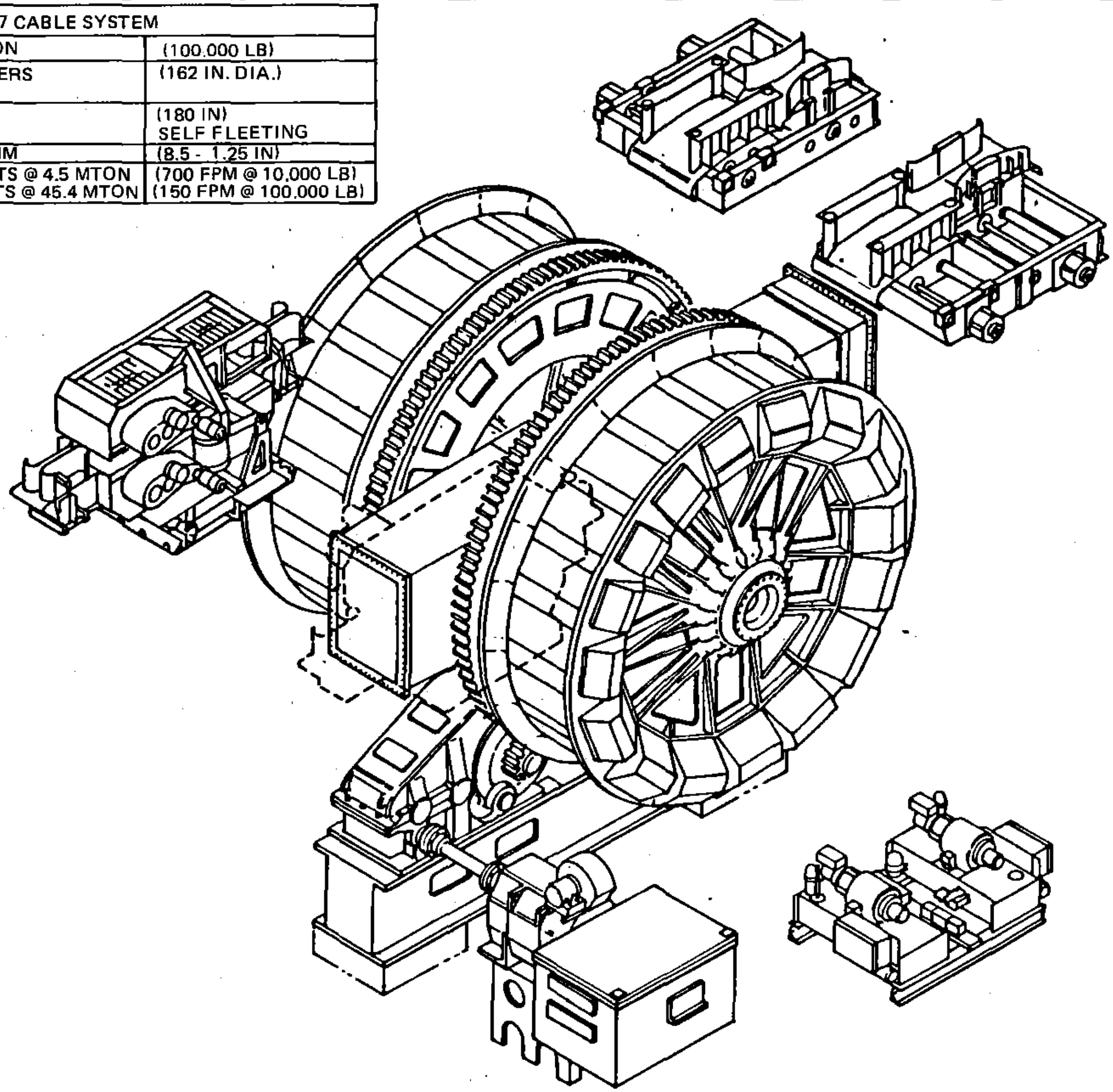


Figure 3.1-4. Cable System Aboard USNS Zeus

WHEEL TYPE PIPE TENSIONER (TYPICAL)		
PIPE SIZE	1016 - 457 MM	(40 IN. MAX. - 18 IN. MIN.)
MAX TENSION	24.5 MTON	(54,000 LB)
SPEED	0.8 KNOT	(80 FT/MIN.)
LENGTH	8.6 METERS	(340 IN)
WIDTH	4.3 M	(168 IN)
HEIGHT	4.5 M	(177 IN)
WEIGHT	55.3 MTON	(122,000 LB)

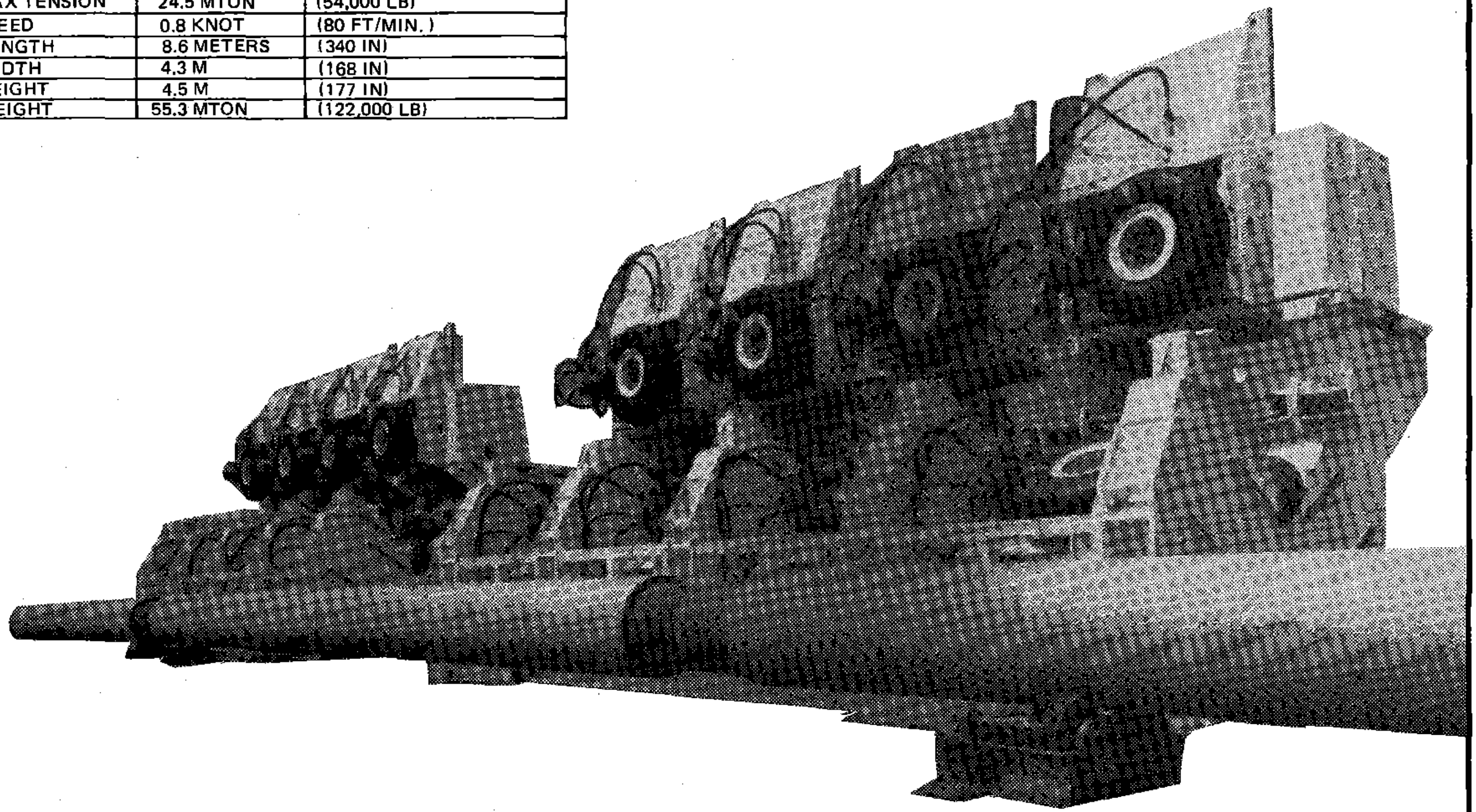


Figure 3.1-5. Wheel Type Pipe Tensioner

LPT SERIES PIPE TENSIONERS (TYPICAL)			
	MAX TENSION	RATE	PIPE SIZE
LPT 40S	18.1 MTON (40,000 LB)	0.8 KNOTS (80 FPM)	1270/76 MM (50 IN MAX/3IN MIN)
LPT 150S	68 MTON (150,000 LB)	0.8 KNOTS (80 FPM)	1830/152 MM (72 IN MAX/6 IN MIN)

SIZE		
	LPT 40S	LPT 150S
WEIGHT	49.9 MTON (110,000 LB	93.4 MTON (206,000 LB)
LENGTH	6.1 METERS (240 IN)	8.6 METERS (337 IN)
WIDTH	3.3 METERS (137 IN)	4.5 METERS (176 IN)
HEIGHT	4.3 METERS (168 IN)	6.3 METERS (247 IN)

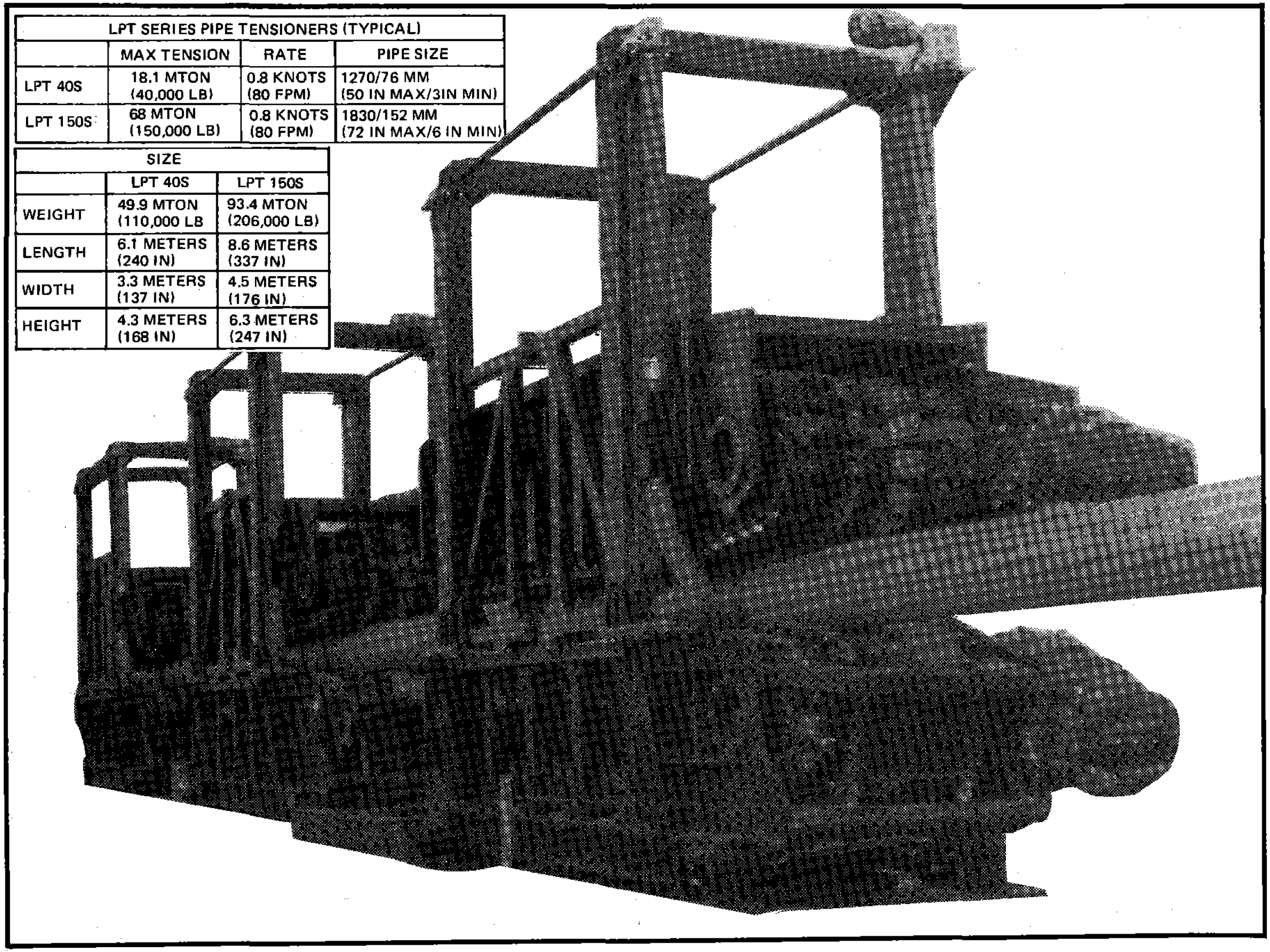


Figure 3.1-6 Track Type Pipe Tensioner

3.2 OVERBOARDING DEVICE

The overboarding device guides cable onto and off of the cable vessel and controls the bending of cable approximately 90° from sea to on board cable handling and tensioning machinery. This device and its supporting structure must function in a near sea surface (splash zone) environment with adequate strength to withstand dynamic cable imposed forces and forces resulting from operational sea states. The device and its supporting structure must also be capable of surviving, without damage, forces imposed by seas at vessel survival sea states experienced during transit.

Three types of overboarding device supports were considered; active dynamic, passive dynamic, and static. An active dynamic support is a powered movable support which isolates cable tension from barge heave and pitch motions by using components such as a hydraulic cylinder to compensate for motion between the cable vessel and overboarding device. A passive dynamic support is a non-powered structural support which compensates for vessel heave and pitch motions. A static support is a simple structure with no compensating motion capability beyond a small degree of flexibility.

DYNAMIC RESPONSE CONSIDERATIONS

The use of the overboarding device for heave compensation is theorectically simple but practically difficult. This method is acceptable for supporting drill risers on off-shore drill rigs but is limited in its dynamic response capabilities. The support is an air spring with a relatively low spring rate and high mass. This results in poor frequency response. Active concepts for drill rigs have been investigated but the difficulties involved in assuring a reliable system with the high power required are possibly insurmountable. For the cable lay vessel a resilient or active mounting for the sheave or other overboarding device faces similar difficulties and it is practically impossible to achieve a response capability of 3 rad/sec with a mass of possibly 45 Mtons while carrying a line load of as much as 70 Mtons.

It is concluded that effective heave compensation at the overboarding device is impractical and any compensation required for the HDWC program must be provided by the cable tensioner.

Three styles of overboarding devices were considered: chute, multiple roller, and sheave. A chute device is a simple structural guide over which the cable slides (Figure 3.2.1-1). A multiple roller type uses a series of rollers over which the cable travels from the cable vessel into the sea (see Figure 3.2.2-1). A sheave overboarding device is a single large diameter roller over which cable travels from the cable vessel into the sea.

3.2.1 CHUTE STYLE OVERBOARDING DEVICE

Advantages and disadvantages of a chute style overboarding device are as follows:

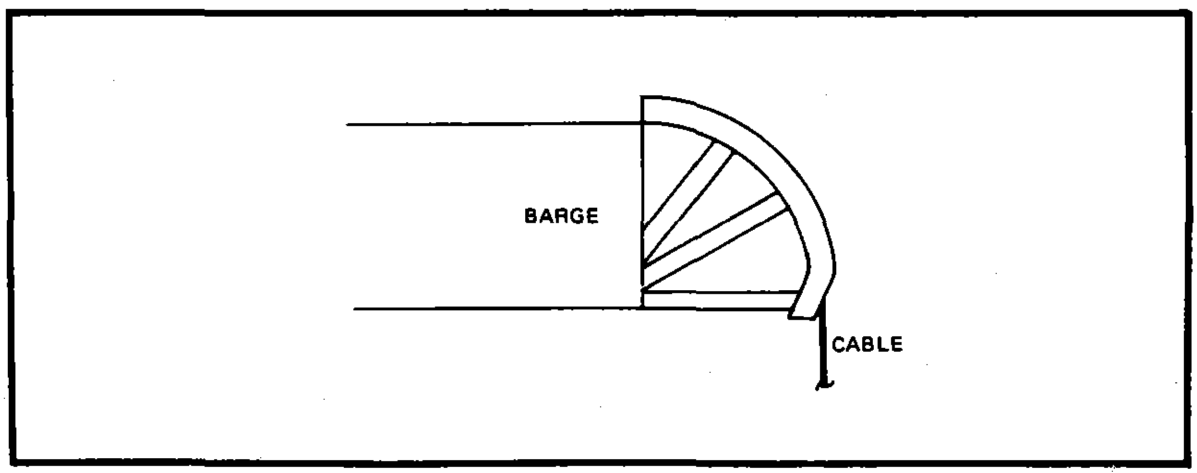


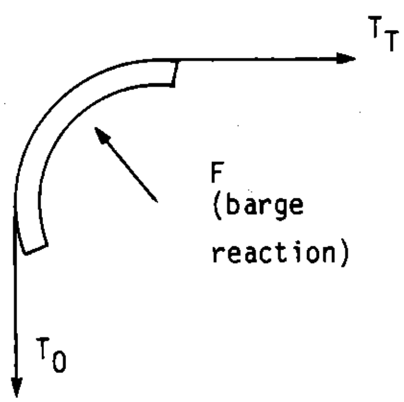
FIGURE 3.2.1-1 Overboarding Chute-Type Device
Simplified diagram of a cable chute type of overboarding device.

Advantages

- It has a simple structure with no moving parts that can be easily maintained using bolt-on friction surfaces.
- Damage resulting from abnormal sea conditions or accidents can be readily repaired.
- The overall profile will be low, presenting a small lateral area to generate high wave forces. This lower profile will reduce the need for elevated tensioning equipment and improve barge stability, roll, and wind area.
- Allows off-perpendicular cable travel relative to vessel centerline.

Disadvantages

- It is difficult to provide material that has a low level of friction that will not rapidly wear through. If a metal surface is used, a coefficient of .15 can be assumed and the total friction losses can amount to 22% of the overboarding tension, as derived below:



T_T = tensioner load

T_0 = overboarding tension

μ = coefficient of friction

θ = wrap angle (rads)

$$\frac{T_0}{T_T} = e^{\mu\theta}$$

If T_0 = 55 Mton

T_T = 43 Mton

Losses = 12 Mton (22%)

With total energy losses of (12 Mton times the payout rate) the wear on the chute is expected to be unacceptably high.

The cable drag would not be predictable within an estimated 25% because the friction coefficient is dependent upon variables including cable and chute contact surface condition (wet or dry), material, contact pressure, and cable sliding velocity.

Due to high cable tension hysteresis (the tension difference between cable payout and inhaul), the chute style of overboarding device is not considered practical.

3.2.2 <u>MULTIPLE ROLLER DEVICE</u>

To bend the cable 90°, each roller of a multiple roller device theoretically bends the cable through an angle of $(\frac{90}{n})°$ (n = number of rollers in 90°) at a radius equal to that of the roller. Thus, to provide a practical multiple roller overboarding device, the permissible cable bend radius usually will be violated. The rollers are located on a curve whose radius is determined by the roller diameter and the spacing between rollers. To reduce the cable bend angle over each roller requires that more rollers be provided which increases the radius upon which the rollers are located unless the roller diameter is decreased. See Fig 3.2.2-1.

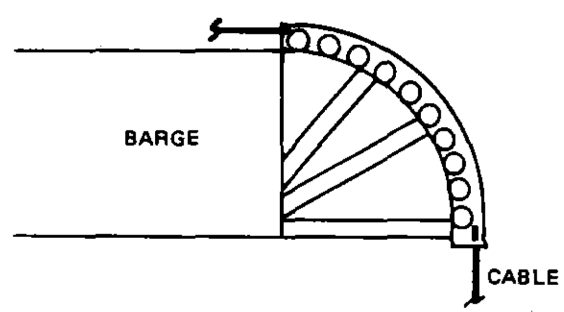
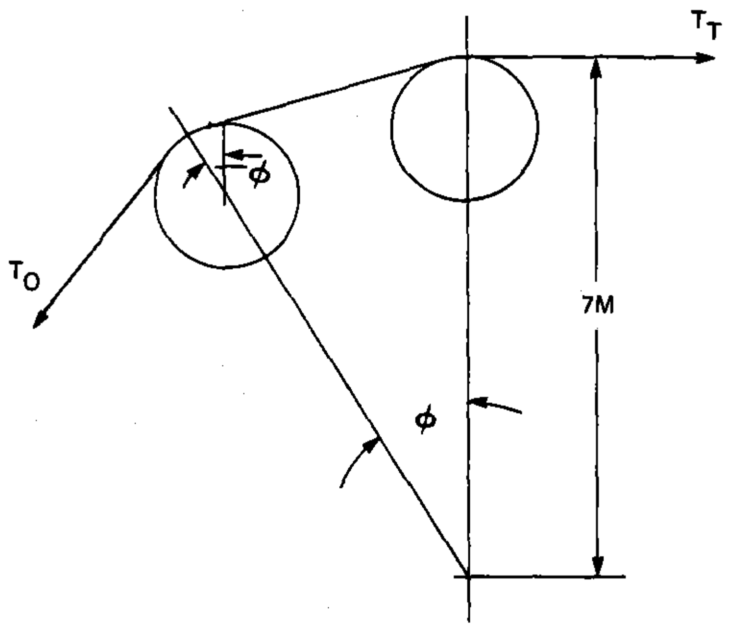


FIGURE 3.2.2-1 Multi-roller Overboarding Device
Diagram of a roller chute overboarding device.

When the cable is bent through a series of small angles over a series of rollers or wheels, the point loading and cable crushing forces are relatively high at the contact points.



Assuming the use of 0.5 meter diameter rollers spaced closely on a 7 meter radius, the angle Ø will be approximately 4.2 degrees and the total contact arc will be 37 mm long. With a line tension of 55 Mton, the reaction through the roller will be 8.05 Mton, resulting in an equivalent crushing force of 217 Mton per meter which is far above the preliminary cable data. Even if the resilient polypropylene outer serving is considered to distribute the loading, the effective crushing forces would be approximately 100 Mton/meter, which remains unacceptable.

The effect of the total 8.4 degree strain per roller on the cable section shown in Figure 2.2-1 would result in a strain on the order of .41 m/m in the lead sheathing, which would cause considerable local yielding and damage to the sheath.

3.2.3 SHEAVE STYLE OVERBOARDING DEVICE

The advantages and disadvantages of sheave style overboarding device are as follows:

Advantages:

1. Permissible cable bend radius would not be violated. The diameter of the sheave could be manufactured with a root diameter of 12.3 meters, as required by the candidate cable #116. Cast steel rim segments could be fabricated together to form the sheave rim, which could then be connected to a cast or forged steel hub. The means of connection could be structural steel pipes serving as spokes fabricated to the hub and rim.

2. Anticipated cable bearing loads would not be violated. Allowable flat bottom sheave loads are not known but it is anticipated that loads generated at the maximum allowable tension on the permissible cable bending diameter would be allowed. A flat bottomed sheave groove would result in sheave-to-cable bearing loads of 8.9 Mton/meter of cable length (at cable tension of 55 Mton) on a 12.3 meter diameter sheave. If the sheave groove were a 90° V shape, sheave to cable bearing load would be reduced to 6.3 Mton/meter on two lines of load contact. (See Figure 3.2.3-1(a).) Using a radiused groove would distribute the sheave to cable bearing load over the contacting area between sheave and cable. A full radius groove, as shown in Figure 3.2.3-1(b), would limit the size of cable end attachments which could be passed over the sheave. Also a sheave groove of this configuration will violate the permissible cable bend radius if the cable were to enter the sheave at an angle other than perpendicular to the sheave axis.

Disadvantages:

1. Large mass and poor dynamic performance. If the overboarding sheave is approximately 28 Mtons with a diameter of 12.3 meters, the inertia

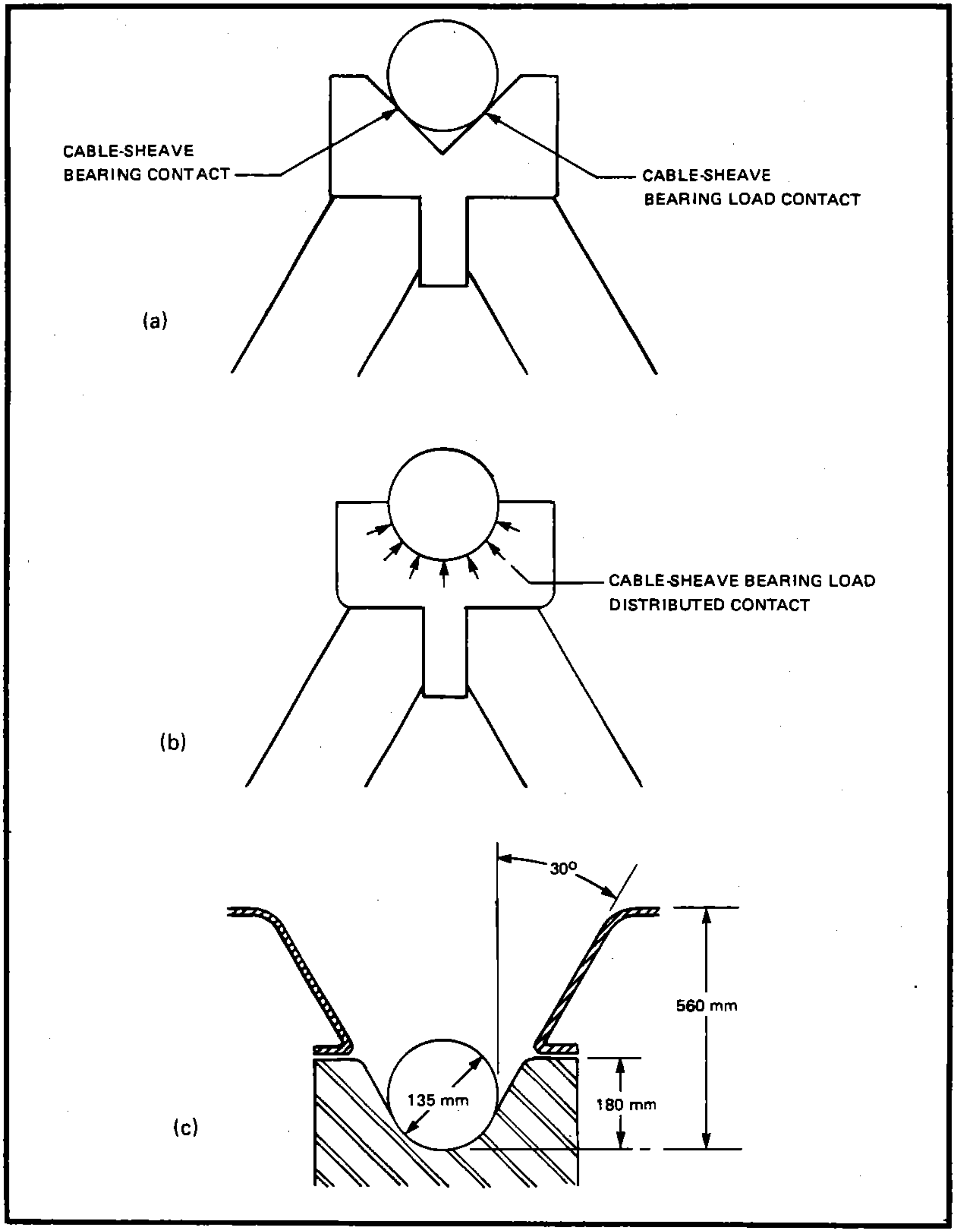


FIGURE 3.2.3-1 Possible Overboarding Sheave Groove Configurations

will be equivalent to at least 633 Mton.m^2, and the frequency response will probably be relatively low so the tension measuring device will lag behind the actual line tension. Since tension is closely related to local acceleration, a lead or anticipation signal can be obtained from a locally mounted accelerometer and this would help dynamic response capability (see 2.5.6).

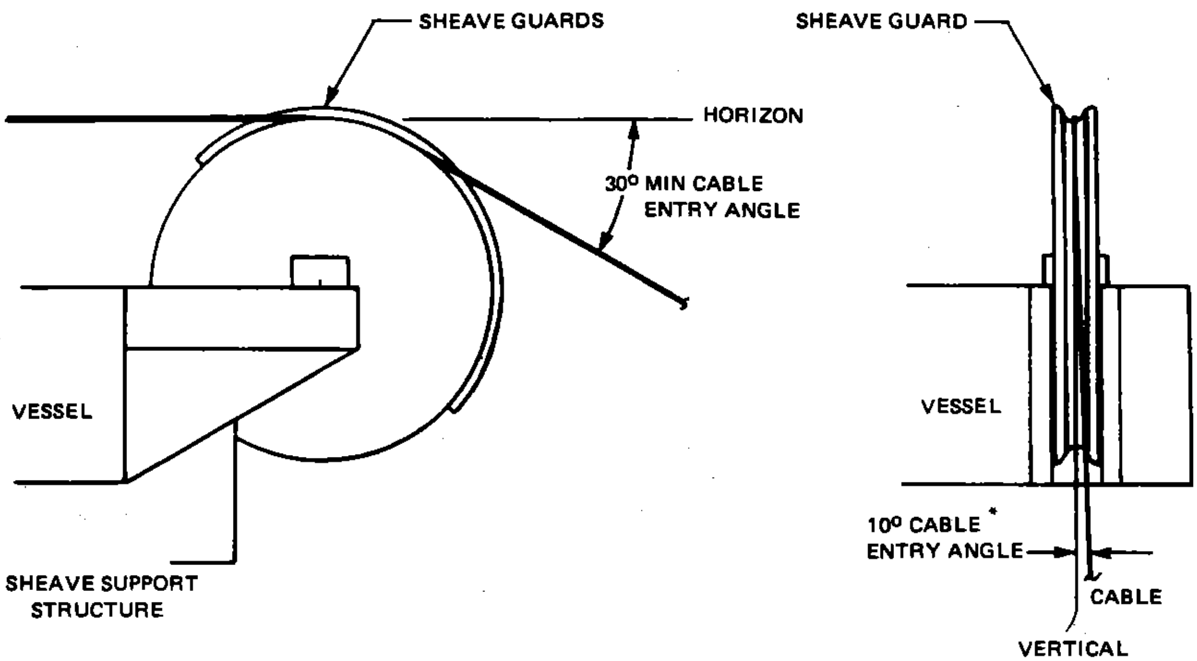
2. During cable retrieval, radiused or V shape sheave grooves will impart a twist into the cable whenever cable entry is not perpendicular to the sheave axis. Cable twist requires special handling of the cable.

   If sheave groove and sheave guards are made as shown in Figure 3.2.3-1(c), then the sheave could pass required cable attachments up to 290 mm in diameter. Cable entrance angles up to 10° in vessel roll direction (see note on Figure 3.2.3-2) could be tolerated without violating permissible cable bend radiuses. Cable entry angles up to 30° off horizontal could also be tolerated.

   A variation of the Figure 3.2.3-1(c) groove configuration, with a flat bottom and radiused sides, could be used if allowable cable loads permit. This groove would allow greater flexibility in sheave use with different cable diameters and larger cable end attachments.

   SHEAVE GUARDS

   Sheave guards retain the cable in the sheave groove and provide cable guidance into and out of the sheave groove. Sheave guards must provide the cable with guidance but not violate any allowable cable bend diameter. Sheave guards must also withstand cable side loading and the sea forces imposed at survival sea states. An alternate to separate stationary guards, as shown in Figure 3.2.3-1(c), is to extend the sheave outer rim diameter and width. Sheave rim outer diameter would be increased 760 mm and the width increased by 460 mm,



*10° limit is based on the Figure 3.2.3-1(c) sheave guard configuration. Increasing the angle requires increased size and configuration of the sheave guards in order to prevent the cable's bend radius limits from being violated.

FIGURE 3.2.3-2 Cable Entrance Capabilities Overboarding Sheave resulting in a significant increase in the mass of the sheave. This would result in further reduction of dynamic response. Also, increasing sheave mass would increase drag through the sheave bearings since bearing loads are increased. Thus the minimum allowable sheave diameter should be used, along with separate sheave guards.

3. Frictional Losses

The forces required to turn the sheave are those resulting from bearing losses, losses at the cable due to side forces, and dynamic forces resulting from non-steady state conditions. Figure 3.2.3-3 shows the sheave loading for deep water laying.

a. Bearing friction

(1) Plain bearings:

- 15.6 Mton hysteresis at 55 Mton cable tension with a bearing friction coefficient of 0.16.
- Positive grease lubrication such as a mechanical device driven by sheave rotation assuring lubrication film could conceivably reduce friction coefficient to .05. Equivalent to 5.3 Mton hysteresis for the same load conditions as above.

(2) Antifriction bearings:

- Reduces cable hysteresis because the rolling friction coefficient will be approximately .002. Equivalent to .2 Mton hysteresis for same load conditions as above.
- Antifriction bearings require positive sealing.

b. Cable side loading from vessel roll and ocean current acting on cable:

- Increases cable drag. For example, if 6 Mton side load and cable to guard friction coefficient is 0.2, this results in 1.2 Mton of cable hysteresis.

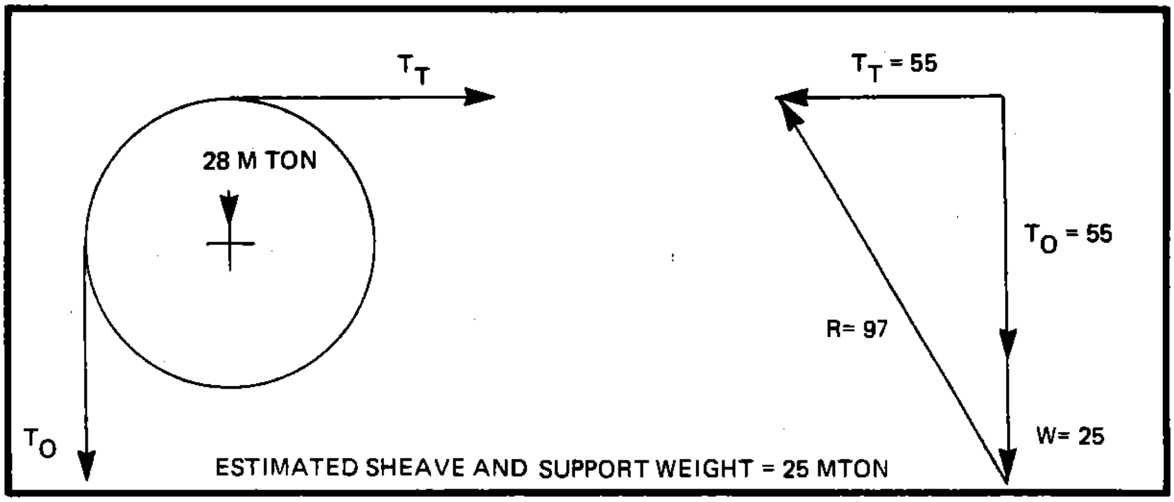


FIGURE 3.2.3-3 Sheave Bearing Forces

The sheave bearing reactions will be 97 MTons with the cable tension shown.

- Side loading could be significantly reduced by orienting the cable vessel so the side loading vector is zero. See Figure 3.2.3-4.
- The sheave must be capable of withstanding side forces imparted by cable operating forces, rolling and survival sea states; thus thrust bearings are required to retain the sheave. If plain thrust washers are used, cable hysteresis resulting from 6 Mton side force results in approximately .25 Mton of cable tension hysteresis, assuming a friction coefficient of .16.

c. Cable hysteresis due to sheave imbalance:
   - If the sheave is out of balance at rim 100 KG, then cable hysteresis is $\pm$ 100 KG occurring each revolution of sheave.

3.2.4 SUMMARY

Overboarding of the cable is a critical requirement influencing the success of the cable lay program. At this time the sheave is the recommended approach as it is the only device without excess tension loss and is suitable for use with the candidate cables. Sheave design must take into consideration mass and inertia and minimize their effects which result in higher dynamic tension variations. Further coordination with the cable manufacturer will be required during the design definition phase of the program to ensure that all applicable allowable cable parameters are known.

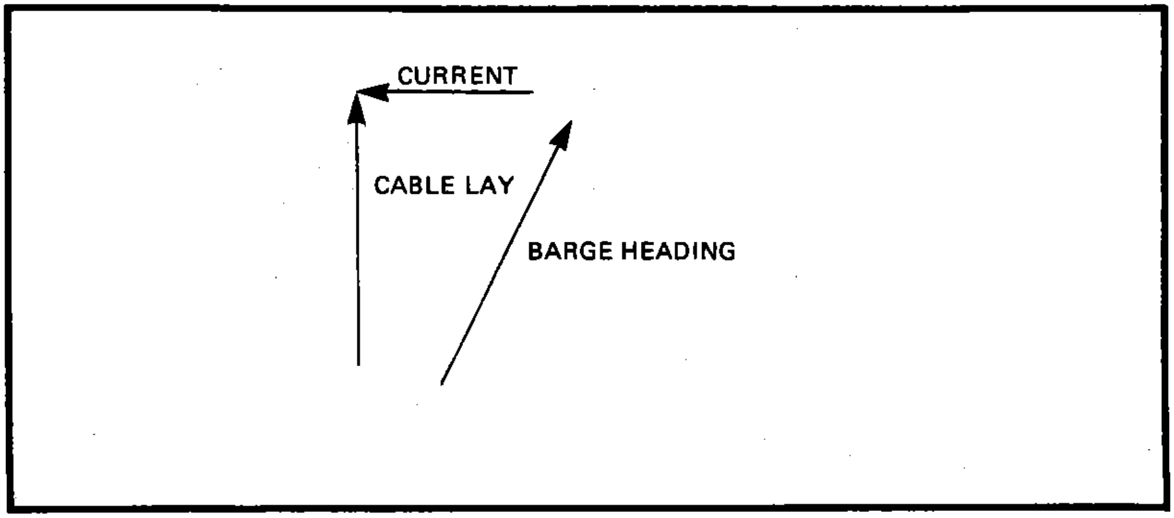


FIGURE 3.2.3-4 Geometry of Vessel Motion

With cable vessel oriented as shown, the side loading on the sheave will be minimized.

3.3 TENSIONER

3.3.1 OPERATIONAL CONSIDERATIONS

The cable tensioner provides the total cable tension required by applying a tension gradient along a suitable length of cable. The tension gradient is controlled by machine design. Tension must be achieved without violating the cable's maximum tension or tension shear and squeeze per unit length parameters which are determined by the cable design. The tensioner also provides tension to pull the cable on board the cable vessel during cable loading and retrieval. Other additional line tensioning requirements are those required to lower anchors and/or other ground tackle required during cable lay start, abandonment or recovery operations.

The cable tensioner can be either a linear or capstan configuration. The capstan type tensioner can have either a single or multiple drum or sheave arrangement. The single device is referred to as a cable drum and the multiple as a traction sheave.

CABLE DRUM TENSIONER

The cable drum is a smooth drum around which the cable is wrapped several times. Two ancillary devices are required for proper operation. The first is a draw-off and hold-back machine commonly called the DOHB. The second is a fleeting device which moves the cable laterally across the face of the cable drum, one cable diameter per drum revolution.

The DOHB maintains a predetermined level of back tension on the drum, allowing its "capstan" effect to amplify low side tension to the high side tension required. Bi-directional DOHB machines are either a wheel or linear type. Single direction payout-only machines can be simple frictional drag devices.

The following summarizes the considerations influencing the suitability of a cable drum tensioner:

BASIC CABLE DRUM:

Advantages

1. Variable cable diameter capability allowing ground tackle line handling
2. Simple machine
3. High tension to size ratio
4. Good dynamic response due to low machine hysteresis

Disadvantages

1. Long length of active cable
2. Cable bent under tension
3. No squeeze pressure control
4. Basic size dictated by largest cable's bend diameter.
5. Complex ancillary machine required to provide applied or hold back tension

When used with fleeting knife additions:

Advantages

1. Simplest fleeting method; high reliability
2. Low (60° to 90°) dynamic angle of wrap loss

Disadvantages

1. Highest drum hysteresis due to sliding losses
2. Irregular cable body (splice, shackle, etc.) capacity by special knife positioning procedure
3. Sliding contact with cable to drum and fleeting knife
4. Probable twisting of cable during fleeting

When used with fleeting ring:

Advantages

1. Lowest drum hysteresis due to no sliding losses on fleeting device

Disadvantages

1. Greatest (180°) dynamic angle of wrap loss
2. Greater mechanical complexity; lower reliability
3. No irregular cable body capacity without additional devices
4. Probable twisting of cable during fleeting

The fleeting device moves the cable laterally to provide space for the oncoming turn or wrap of cable, and must also fleet or move all the wraps of the cable from the oncoming cable wrap to the high tension or overboarding wrap during payout operations. The fleeting movement required is the opposite direction for cable retrieval operation.

Three types of fleeting devices were considered; fleeting knife, fleeting ring, and self-fleeting drum. The fleeting knife is a stationary smooth guiding shoe which forces the cable wraps to move sideways as the cable moves past it due to drum rotation. Two fleeting knives are required per cable drum, one for payout (see Figure 3.3.1-1) and another for retrieval. The retrieve knife can be located on the opposite side of the cable or approximately 180 degrees on the drum. The fleeting ring (see Figure 3.3.1-2) is a floating ring mounted concentric with the drum at an offset angular axis rotating freely with the drum. The fleeting ring is driven by the side friction of the cable it fleets.

SELF-FLEETING DRUM

A self-fleeting drum is a drum type device on which the drum surface is comprised of multiple lateral conveyance devices. The lateral conveyance devices are driven a fixed amount of movement per drum revolution. This fixed amount is slightly greater than the maximum diameter of cable to be tensioned.

A cable positioner, or fairleader, is used to guide the oncoming cable (instead of a fleeting knife or ring). The fairleader is also used to maintain retrieved cable position on the self-fleeting drum, and to remove the accumulative type fleeting errors generated by the slightly non-circular, polygon drum shape and discontinuous fleeting surfaces.

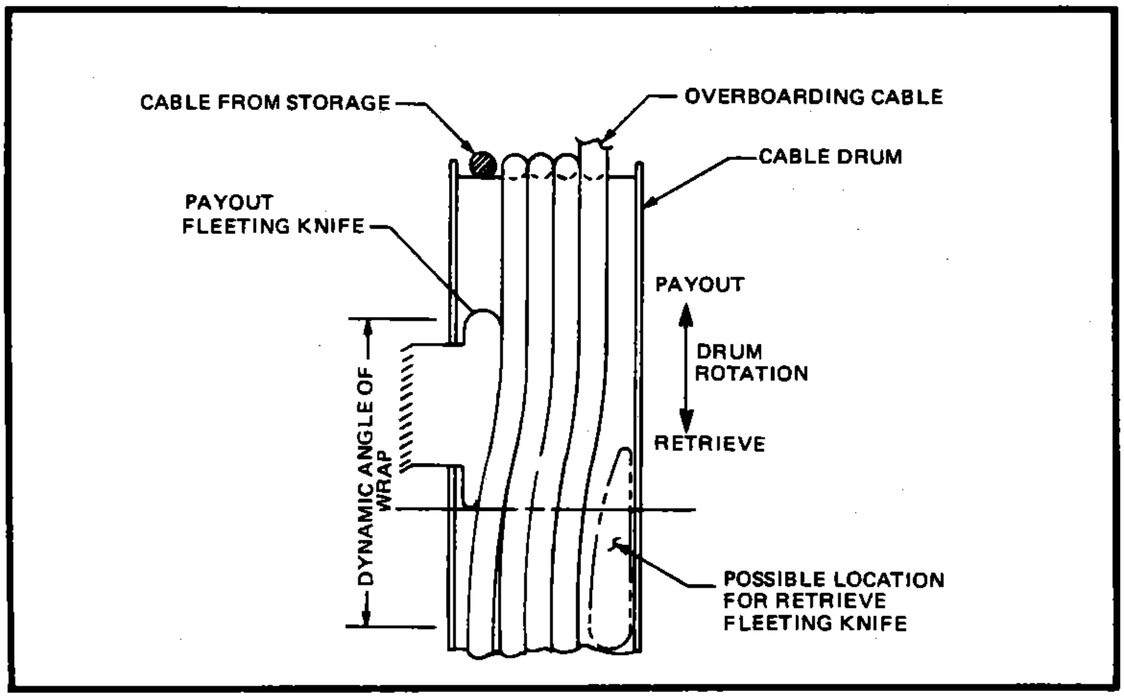


FIGURE 3.3.1-1 Typical Arrangement of Fleeting Knives on Drum Tensioner

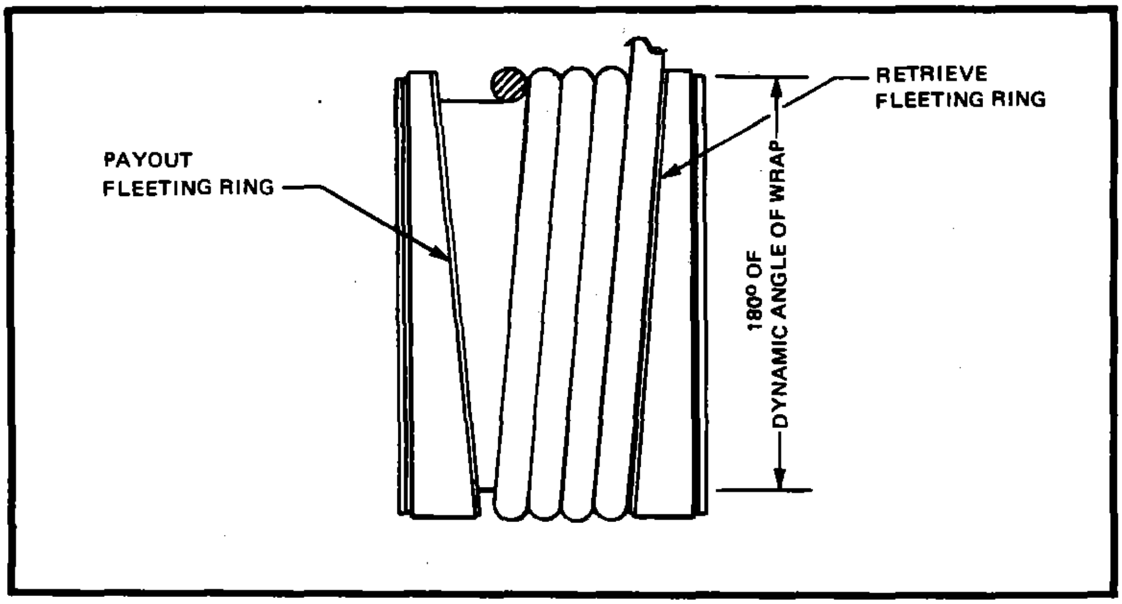


FIGURE 3.3.1-2 Arrangement of Fleeting Rings on Drum Tensioner

Advantages	Disadvantages
When used with self-fleeting feature:	
1. No dynamic angle loss	1. Greatest complexity, lowest reliability
2. Nonuniform cable and/or irregular cable body capability without requiring attention.	2. Ancillary device required to maintain retrieved cable position on drum.

TRACTION SHEAVE TENSIONER

The traction sheave device consists of two drums or sheaves around which the cable is wrapped several times. One drum is canted at a slight angle with respect to the other, which laterally displaces the cable is at least one cable diameter when returned to the other sheave. No fleeting devices are required, but guidance devices must be provided to ensure accurate cable feeding. Following are the advantages and disadvantages of the traction sheave tensioner:

Advantages	Disadvantages
1. Simplest drum type machine	1. Fixed cable diameter capability
2. No dynamic angle loss	2. Long length of active cable
3. Good dynamic response due to low machine frictional losses	3. Cable bending and unbending under tension more than once when on traction sheaves
4. High tension to size ratio	4. Complex ancillary machine required to provide applied or hold back tension
	5. No squeeze pressure control
	6. Basic size dictated by cable's bend diameter under tension
	7. Limited or no ground tackle handling capability.

LINEAR TENSIONERS

The linear type tensioner can be either a track or pneumatic tire type. Both types of machines grip the cable by squeezing it between powered gripping surfaces.

TIRE TYPE

The pneumatic tire type grips the cable by squeezing it between two pneumatic tires. The tires are normally mounted vertically with a horizontal axis. The lower tire is fixed with the upper tire moveable, allowing the tires to have variable squeeze capability and to be separated for cable loading. Each tire is powered and multiple sets can be used to develop the level of tension capability required. Tension capability must be empirically developed by testing, using the tires, tire tread, and the actual cable to be handled. Additional components such as cable guides are incorporated to properly position the cable relative to the tires. See Figure 3.3.1-3.

TRACK TYPE

The linear track type of cable tensioner uses track devices instead of tires. Squeeze loading devices are applied to the track support rollers, providing the force to grip the cable. The track assembly is designed to accommodate the friction devices or grippers which generate the tension force. Some range of gripper movement is normally provided to accommodate variance in cable diameter. One track frame assembly can be moved, permitting the tracks to be opened to allow loading or unloading of the cable.

Following are the significant advantages and disadvantages of the linear type machines:

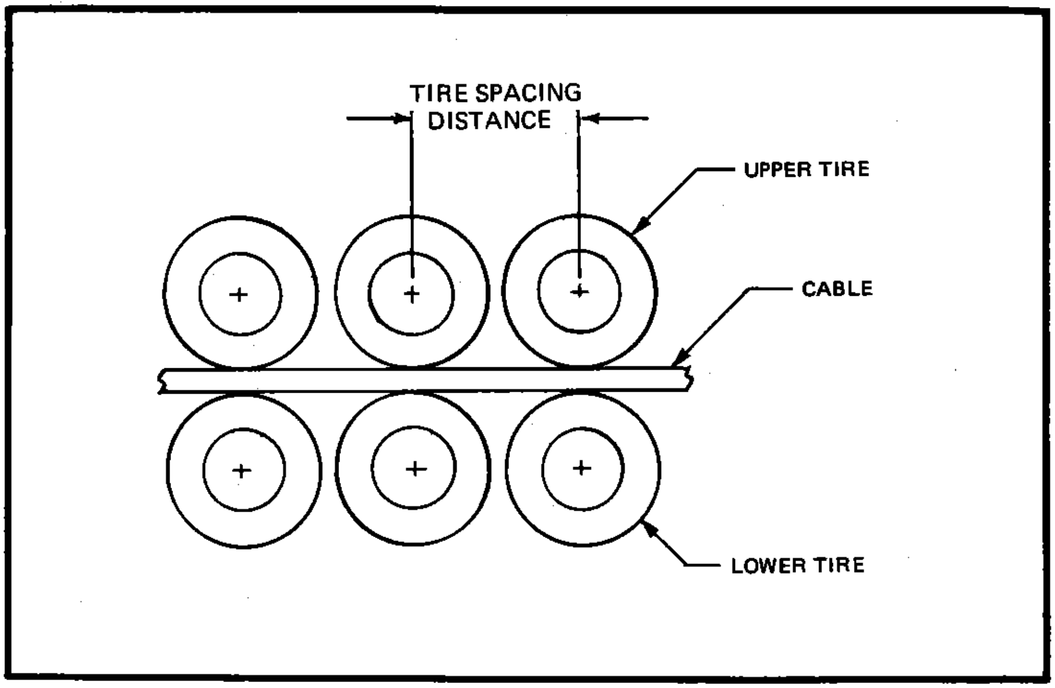


FIGURE 3.3.1-3 Diagram of the Basic Principle of a Wheel Type of Linear Tensioner.

Normally the upper wheels are movable to adjust gripping force and accommodate different cable diameters. (See also Figure 3.1-5)

Advantages	Disadvantages
LINEAR TIRE TYPE TENSIONER	
1. No cable bending under tension 2. Allows control over cable squeeze pressure 3. Good dynamic response due to low machine frictional losses 4. Low length of active cable	1. Low tension to overall length ratio 2. Two drive trains are required for each tire set. 3. Gripper geometry is fixed 4. Ancillary cable guide devices are required 5. Mandatory load sharing required between all tire drive train motors
LINEAR TRACK TYPE TENSIONER	
1. No cable bending under tension 2. Control over cable squeeze pressure 3. High tension to overall machine length ratio 4. No ancillary devices are required 5. Single drive train per track 6. Gripper geometry optional 7. Minimum length of active cable	1. High linear hysteresis due to track components 2. Machine complexity increased due to track components

3.3.2 CABLE INTERFACE

The major cable interface considerations related to the cable tensioner are the cable bending diameter, squeeze or crushing forces, tension shear per unit length and the coefficient of friction with the cable jacket. Cable interface will be discussed separately for circular machines (i.e., drum type) and linear machines.

Drum Type: All circular machines are tension amplifiers (low tension side multipliers) which only amplify tension levels applied by another device. This relationship between tension applied, T_S, and tension developed, T_L, is given by the formula:

$$\frac{T_L}{T_S} = e$$

where T_L = tension developed
T_S = tension applied
μ = coefficient of friction
θ = angle of wrap

(see section 3.2.1)

The angle of wrap is that which applies to the coefficient of friction used and is different for static and dynamic conditions, with static friction normally higher. Applying dynamic friction for dynamic angle of wrap becomes complex because cable dynamics and kinetic energy transfer due to cable elongation then apply along with the varying friction values. Because of this, the drum is normally rated, using only the static angle of wrap and the static coefficient of friction, even though the disregarded dynamic variables affect the drum's performance, to a minor extent.

Static angle of wrap is the angle through which the cable rotates about the cable drum with no relative motion between cable and drum surface. All remaining angles are those where motion exists, such as that caused by the fleeting knife or ring. There is no angular contact loss on a fleeting drum because all cable contact is on moveable laterally conveying drum surfaces.

The crushing or squeeze force on the cable is proportional to the tension developed by the drum. This squeeze force is developed over the width of the cable and is approximately expressed by the following formulas:

$$P = \frac{2\,T_L}{W\,D}$$

where P = cable squeeze pressure
T_L = tension developed
W = cable width
D = drum diameter
F = cable squeeze/unit length

and

$$F = \frac{2\,T_L}{D}$$

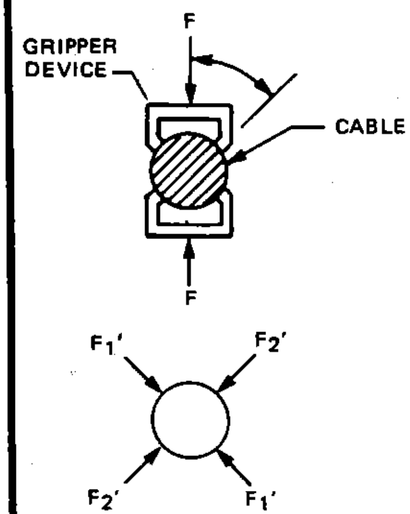
As can be seen, the only control over the cable squeeze force at a certain tension is the cable drum diameter.

The most severe stress on a cable drum is in the area of contact with the drum's fleeting mechanism. This is especially true with a fleeting knife device because the force required to slide the cable causes additional stress.

Linear Type: The forces acting on a cable during its passage through a linear machine are not self-generated and thus are more predictable. Cable tension is dependent on the cable's coefficient of friction and the squeeze force generated by cable tensioner machine. Therefore, the coefficient of friction must be accurately determined for a range of squeeze pressures and pad configurations. This is especially true when an appreciable texture on the cable jacket material allows some degree of mechanical interference. This interference results in an increase in coefficient of friction that is non-linear with the normal or squeeze force.

The cable's tolerance to squeeze forces and its tension per unit length must also be realistically estimated or based on actual test data. Squeeze force can be maintained at the allowable limit; therefore maximum machine performance will be limited by the cable's coefficient of friction or the tension shear capability. Tolerance to squeeze force and its values must be known for single and multiple axis squeeze. Multiple axis squeeze results in greater tension capability with a fixed coefficient of friction (as shown in Figure 3.3.2-1).

When cable parameters are known, the tension shear limit per unit length of the cable will probably be the limiting function based on past cable machine experience. The linear tensioner's capability of varying the squeeze force and the track type's multipe axis grippers can customize the friction capability to meet the expected cable's shear limit. Therefore the machine length will be inversely proportional to the permissable cable tension shear parameter.



Ø	F	F'	$F_1' + F_2'$
0°	1	.5	1
30°	1	.578	1.15
45°	1	.707	1.41
60°	1	1.0	2.0

F = Machine Applied Squeeze Force

where $F' = \dfrac{.5F}{\cos}$

F^1+F^2 = Total Squeeze Force on the Cable

FIGURE 3.3.2-1 Effect of Multiple Axis Grippers on Effective Squeeze Force Capability

Suitable shaping increases the effective squeeze force on the cable.

3.3.3 TENSIONER SELECTION

Cable drums and linear tensioners were evaluated using preliminary data as follows:

DRUM TENSIONER

The main advantages of cable drums are simplicity and flexibility of use, allowing almost any line or cable to be tensioned. The major disadvantage is the requirement for an additional DOHB device to provide the applied or hold back tension. The complexity of this additional machine is proportional to the amount and accuracy of hold back tension required.

Cable parameter control is difficult to achieve on a cable drum. Cable bending diameter must be equal to or less than the drum diameter. The cable's tension shear per unit length limit can only be controlled by varying cable squeeze or coefficient of friction. Neither of these parameters can be directly controlled on a cable drum because friction geometry is fixed and maximum squeeze is controlled by final tension and drum diameter.

The use of preliminary cable values (from Figure 2.2-2) of 3 metric tons per meter for tension shear and 5 MT/M for squeeze result in the cable drum parameters shown in Table 3.3.3-1. The cable drum diameters are larger than practical if 5 metric tons per meter squeeze is used as the limiting parameter.

Preliminary cable data on squeeze allowed an intermittent maximum squeeze to reach 15 MT/M at which point the tension shear per unit length is the limiting parameter. Knowing that maximum squeeze occurs only in the location where the cable reaches its maximum tension, it may be permissible to increase cable squeeze. Maximum squeeze of 10 MT/M would transfer the limiting cable parameter to the tension shear per unit length which would result in excessive drum diameters as shown in Table 3.3.3-1.

GIVEN PARAMETERS		CALCULATED DRUM PARAMETERS		
Cable PPC#	Maximum Tension	Drum Diameter	Maximum Cable Squeeze	Maximum Tension Shear
113	72.0 Meters	11.6 Meters	12.4 Mton/Meter	3.8 Mton/Meter
113	72.0	14.4	10.0	3.0
113	72.0	28.8	5.0	1.5
116	78.7	12.0	13.1	4.0
116	78.7	15.7	10.0	3.0
116	78.7	31.4	5.0	1.5
119	84.0	12.3	13.7	4.1
119	84.0	16.8	10.0	3.0
119	84.0	33.6	5.0	1.5

TABLE 3.3.3-1 Cable Drum Diameters With Candidate Cables and Design Constraints

The cable drum diameters listed in Table 3.3.3-1 are also applicable to a traction sheave tensioner. It has been eliminated from consideration because two sheaves of large diameter are required and it cannot be operated without DOHB equipment.

LINEAR TENSIONER

A linear type of tensioner allows the use of the required cable parameters. Tension shear and squeeze limits compute to a .3 coefficient of friction, which can be achieved in a track type tensioner using multiple axis grippers, but is near the upper limit normally achieved in a linear tensioner when tensioning a synthetic-jacketed cable. The active length for a linear tension machine can be computed using the cable's maximum tension and tensioner shear limit. The machine active lengths, computed using preliminary data, are shown in Table 3.3.3-2.

The active length of a linear tire type tensioner is approximately 25 percent of tire spacing (see Figure 3.3.1-3). This results in a tire type tensioner of 80, 87.3, and 93.3 meters long, respectively, for the cables shown in Table 3.3.3-2. All of these machine lengths are excessively large because the combined length of the overboarding device, tensioner, surge-slack device, splicing area, and turntable would probably exceed the available vessel deck space. This eliminates the tire type machine from consideration.

The active length of a linear track type tensioner is close to its overall length. The track turnaround sprockets are the only additions to overall length in excess of the active length. Adding five (5) meters to the active length results in a track type tensioner 29, 31.2, and 33 meters long, respectively, for the cables shown in Table 3.3.3-2.

CONCLUSIONS

The linear track type of tensioner is the most suitable because it is the only tensioner of practical size that does not violate the preliminary cable design parameters of squeeze and tension shear per unit length.

GIVEN PARAMETERS				
Cable PPC #	Maximum Tension	Maximum Shear Limit	Tensioner Active Length	Tensioner Overall Length
113	72 MT	3 MT/M	24 M	29 M
116	78.7	3	26.2	31.2
119	84	3	28	33

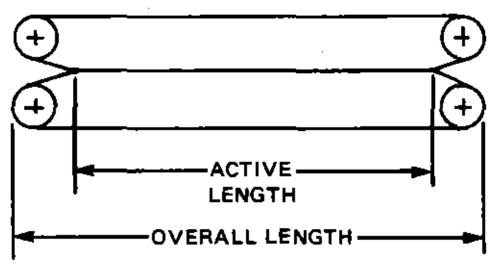


TABLE 3.3.3-2 Linear Track Type Tensioner Dimensions

3.4 STORAGE DEVICE

3.4.1 OPERATIONAL CONSIDERATIONS

The purpose of the storage device is to store the cable after it is loaded on board the cable vessel and to payout and retrieve cable in a controlled manner, preventing cable damage caused by kinks, excessive twisting, or cable bends sharper than the minimum bend diameter.

ALTERNATIVES

Cable storage devices can be tanks or turntables. The tank method is the coiling of cable into a circular or oblong shape in an open volume. Turntable storage is the winding of cable onto a rotating table forming a circular coil.

The following description of tank stowage has been provided for background information. Cable coiled stowage has specifically been ruled out because of the torque balanced cable construction which will be used in the HDWC program. The tank storage container is usually a circular tank-like structure with solid or open ring side walls. The diameter of the internal ring is sized to meet or exceed the cable minimum bend diameter. The outside ring determines the tank's capacity and provides support to the outermost coils of cable. Loading and storing cable into a cable tank imparts a 360° twist into each coil of cable as it is laid down in the tank. This twist is removed as the cable is removed from storage, allowing the cable to be delivered in an untwisted state. The internal ring is shaped like a truncated cone which prevents cable coil from wrapping around and jamming. A device called a "crinoline" aids the smooth uncoiling of cable from the cable tank. See the section on surge-slack devices for further discussion of the crinoline.

The turntable storage device is a circular shaped tub similar to a circular cable tank. The internal ring could be cone shaped but straight sides are sufficient. Cable is loaded on at one location while the turntable rotates, winding the cable into coils with no twist. A turntable must be used when cable must be stored with no twist. During cable loading or payout the speed of cable payout must be matched to turntable rotation at the radius of cable touch down or pick-up. The turntable rotation rate changes constantly, even when cable speed remains constant, because the cable coil circumference changes as the cable winds or unwinds (see Section 3.6.5).

TWIST CONTROL

The turntable can twist or untwist the cable as it is loaded into the turntable by controlling the cable's touchdown or landing point. Rotating the cables touchdown point one complete revolution on the turntable surface will remove or add a 360° twist on the cable, depending on which direction the touchdown point is rotated. Normally this feature is used only during cable loading or retrieval operations, when the cable twist condition between low tension and high tension or coiled shore storage to turntable may be different and may accumulate if not removed. This cable twist must be manually monitored by observing a painted stripe on the cable's outside jacket and removed by varying the cable's touchdown location on the turntable.

Other handling considerations are cable inelastic twisting, nonlinear torque balance, and slight torque inbalance between storage and retrieve cable tension. Inelastic or permanent twist cannot be released without physically inducing a straightening torque. All cable handling, especially temporary or shipping storage where the cable may be coiled down, must be evaluated for possible inelastic cable twisting. If inelastic twisting has occurred, it must be removed when the cable is loaded into the lay vessel's turntable. The twist can be removed by positioning the pickup arm above the turntable during cable loading. The pickup arm would slowly vary the cable's touchdown location so the cable's painted stripe poisition in the turntable would remain constant. Failure to remove the twist could cause severe cable handling problems because cable tensioning during cable retrieval would cause cable twist to accumulate between the cable tensioner and turntable. This problem occurred in the first Cook Inlet cable of the 230 KV torque balanced series and removing the twist was a difficult operation.

CABLE TWISTING CONSIDERATIONS

Removal of all twist in the cable during laying would not guarantee the elimination of twist accumulation during cable retrieval. The torque balance of the cable is not a perfect linear function; positive or negative twist can occur in the same cable but at different levels of cable tension. Therefore, it is expected that there will be sections of the cable with no twist accumulation and other sections with relatively high twist accumulation during recovery under varying but declining tension levels. This can be expected to occur even when all cable is laid in a "zero" twist configuration.

Loading of the cable onto the turntable from a shipping vessel or from a storage area where it was coiled down can present difficulties. Special handling must be considered if the coiled storage is of a diameter where inelastic twist occurred or storage time and temperature allowed appreciable lead sheath jacket creep to occur. Loading of torque balanced cable that contains twist memory (because of lead sheath or other cable element yielding) must be performed in a manner that removes the twist prior to delivery to the lay vessel's turntable. This could best be done by subjecting the cable to tension while conveying from storage to the lay vessel tensioner. This loading level of tension is expected to be approximately 5 to 10 percent of the cable's maximum allowable tension.

The purpose of the loading tension is to use the cable's torque balance characteristic to remove the lead sheath's inelastic twist before delivery to the lay vessel's tensioner. Further untwisting may be required at the turntable to return the cable's twist indicating stripe to the untwisting position.

It is proposed that handling tests be conducted to prove methods, develop procedures, and ensure that cable parameter response is as anticipated. It is expected that some modification in procedures and methods will be required to minimize the difficulties in handling cable untwist operations during HDWC demonstration cable retrieval.

3.4.2 CABLE INTERFACE

Critical considerations for cable interface with storage devices are minimum bend diameter, crush resistance, and torsional characteristics. Splicing arrangement and access to cable end for testing will not be considered at this time.

Normally, minimum cable bend diameter is controlled by the inner diameter of the turntable's center cone or hub. Other areas of cable bending are at the point of cable entry or exit on the stored coils in the turntable. This bend diameter is determined by the cable's flexure strength and must be considered to determine if a bending control device is required at the point of cable entry onto the turntable.

Cable crush strength will determine the maximum height to which cable can be stacked in the storage device. Preliminary cable data indicates that the safe height for cable stacking is approximately 4.0 meters. Advantages to increasing the cable stack height to the maximum allowed, especially with a large capacity turntable, include reduction of the required turntable outside diameter but maintenance of the same cable storage capacity. This in turn reduces the turntable's rotative inertia and reduces the power capacity requirement for its drive system. This relationship of storage height to drive power is shown in Table 3.4.2-1.

Turntable Outside Diameter	Turntable Depth	Turntable Capacity	Turntable Drive Power
23 Meters	4 Meters	104.2 KM	1346 KW
19 Meters	6 Meters	101.9 KM	1046 KW
16 Meters	9 Meters	103.3 KM	891 KW
*12 Meters	3.5 Meters	10.1 KM	69 KW
*10 Meters	4 Meters	11.5 KM	66 KW

Note: Drive power based on an assumed acceleration from zero to 2 knots in 1.0 minute. Turntable support frictional losses based on 10 KG per MTon. Capacity based on PPC #116 cable. *Turntable size suitable for HDWC program at-sea testing.

TABLE 3.4.2-1 Turntable Capacity and Power Requirement vs Diameter and Depth

Data regarding the cable's torsional character- istics are required to determine the proper relationships with the loading and pickup arm and to give insight into possible handling problems. The surge-slack section (3.5) contains a discussion of cable twist characteristics and their relationship to the pickup arm.

Cable characteristics required to finalize the turntable and handling system selection and conceptualization are listed below:

1. Torsional resistance
2. Torsional elastic limit
3. Torque balance versus tension
4. Cable flexure strength
5. Weight
6. Minimum bend diameter
7. Cable support or crush strength

3.4.3 SUPPORT CONSIDERATIONS

The support of a fully loaded turntable requires consideration of cable stack height, support bearing frictional loss, control interface, and vessel power generation capacities. These factors must be evaluated in terms of system requirements and compromises made to determine the optimum turntable. These factors are proportional to turntable size and capacity. Therefore, their impact is less for the at-sea portion of the HDWC program than for a baseline commercial system.

The relationship between cable stack height and turntable drive power, with the other factors held constant, can be seen in Table 3.4.2-1. Control interface refers to turntable response to sensor requirements for acceleration and deceleration to maintain payout rate due to stored coil diameter changes. Control interface also refers to the additional acceleration and deceleration rate needed to accommodate the changes in cable lay speed due to vessel speed, bottom slope or slack requirements. The resultant turntable drive power requirement due to cable considerations can be a significant part of total vessel power requirements. Since the vessel's power generation capacity or supply must meet various demands, the turntable acceleration estimates may have to be reconsidered after power requirements and generation capacity for the vessel are known.

Frictional loss in the turntable support bearings is a significant part of the total turntable drive power requirement. In Table 3.4.3-1 the separate frictional and acceleration power requirements are listed.

Using a pneumatic or liquid compliant film bearing with a coefficient of friction value as low as .001 reduces Table 3.4.2-1 values to those shown in Table 3.4.3-2. The use of a compliant fluid film or water slide type bearing involves advantages and disadvantages as follows:

<u>Advantages</u>	<u>Disadvantages</u>
<u>Roller Bearing Support</u>	
1. Passive support contains no no power or control elements.	1. Precision mounting required to ensure load sharing between rollers.
2. Rapid stopping time.	2. High concentrated load at support elements require additional support structure in turntable and vessel foundation.
3. High reliability, units based on proven, mature technology. Used on the SKAGGERAK.	3. Depth required for support elements and structure is large.
4. Normal corrosion protection required by support elements.	

From Table 3.4.2-1		Frictional Power		Acceleration Power (1 min)
1346 KW	=	949 KW	+	397 KW
1046 KW	=	774 KW	+	273 KW
891 KW	=	681 KW	+	210 KW

NOTE: Frictional value is based on rolling friction of 10 KG per metric ton and acceleration time as noted. Rolling wheel support radius is taken at radius of gyration.

TABLE 3.4.3-1 Turntable Power Requirement for Rolling Support

Ref From Table 3.4.2-1	Total Slide Bearing Power		Sliding Power		Acceleration Power (1 min)
1346 KW	492 KW	=	95 KW	+	397 KW
1046 KW	350 KW	=	77 KW	+	273 KW
891 KW	278 KW	=	68 KW	+	210 KW

NOTE: Frictional value is based on .001 coefficient of friction. Bearing support radius is taken at radius of gyration.

TABLE 3.4.3-2 Turntable Power Requirement Water Slide Bearings

Advantages	Disadvantages
Slide Bearing Support	
1. Immobile unless activated.	1. Requires power input to function.
2. Distributed load on all support elements and structure.	2. Stoping time is greater.
3. Bearing elements and related structure require little depth.	3. Reliability dependent on continuous power availability.
	4. Require axial displacable center support for dynamic positioning.
	5. Slide bearing contact surface must be suitable for salt water splash zone corrosion resistance.

The advantage of low drive power requires study because it requires maintaining continuous power input to the slide bearing. The reliability of the slide bearing depends on its input power and water pumping units as all other components are passive, serving only to contain the pressurized sea water. The 550 Kilo Pascal (80 psi) pressure of the sea water does the actual load support. Other disadvantages that are listed are initial design concerns only. Therefore, the advantage of the slide bearing's low drive requirement is achieved at the disadvantage of increased risk due to lower reliability.

Rolling element support is the recommended turntable support method for the HDWC program at-sea tests, even though rolling element power requirements are higher than those of slide bearing support. Rolling element power requirements for the at-sea test are low (see Figure 2.4.2-1) in comparison to those that would be required for a baseline commercial system. While a power cable turntable utilizing a slide type bearing support has been built, this method will require further study prior to its use on a baseline commercial project. Slide bearing areas requiring study and clarification are speed, capability, reliability, and economic considerations.

3.4.4 TURNTABLE DRIVE POWER REQUIREMENTS

The storage of candidate cable PPC #116 with capacities and drive power requirements for various turntable sizes is shown on Figure 3.4.4-1. Drive power is shown for roller and slide support mechanisms under three different acceleration times. Acceleration times of less than one minute are not practical because of the economic impact of generating this power, and providing the power control, drive motor, and drive train (including the reducer) for this amount of turntable drive power. Actual acceleration requirements must be studied further, using tension compensation displacement and cable lay speed as part of the overall system operating dynamics study. Drive power is a major concern for storage lengths in excess of 80 kilometers or turntable diameters greater than 20.5 meters, such as those that would be required for a baseline commercial system.

Existing turntable drive systems are designed to operate under steady speed conditions, taking into consideration that higher speeds and accelerations would not be immediately required at the beginning of a cable lay, but would be required later when the turntable was at practical capacity. For example, the SKAGGERAK turntable (refer to Section 3.1) is a 470 KW (630 HP) system rated at 7000 MTon and 1.2 RPM. At its rated conditions, the turntable has essentially no excess power capacity for acceleration, and has a maximum speed of 45.2 M/Min (1.47 KTS) when operating at the minimum storage diameter. At 50 percent capacity, this turntable can accelerate to full speed in approximately 45 seconds.

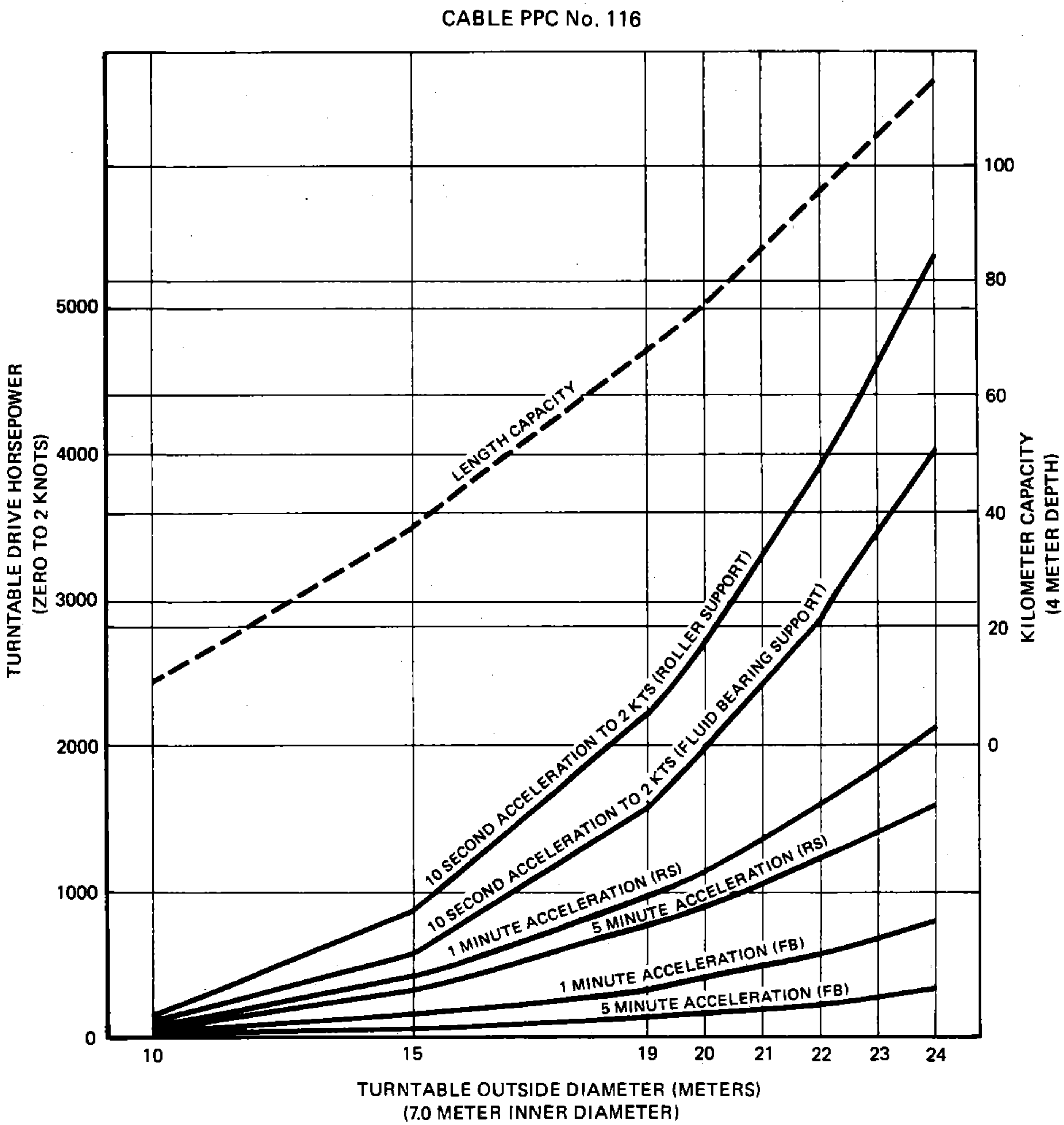


FIGURE 3.4.4.1 Turntable Diameter vs. Kilometer Capacity

Large capacity turntables are feasible but require that compromises be made between maximum cable lay speed when maximum lay speed is required, acceleration time requirements, and vessel power generation capacity. Cable stack heights greater than 4 meters, if allowed by cable design, can be used to reduce turntable diameters and power requirements while retaining capacity. Drive power requirements for the HDWC program at-sea test are not an area of concern, as can be seen from the values shown in Table 3.4.2-1.

3.4.5 DRIVE CONSIDERATIONS

Drive for the turntable may be hydraulic or variable electric drive, depending on level of power input. Note that the SKAGGERAK turntable is electrically powered, while the SUSITNA uses a hydraulic drive. The baseline proposal for the commercial program uses a modular type of drive system and multiple identical drive units force summed to provide the total power required. This method lends itself to development of a proven unit and adjustment of the available power by increments. It also provides a reliability advantage in that an excess number of units will be available over much of the operation as cable is unloaded and power requirements diminish. It is expected that control interface with either hydraulic or electric drive can meet the response rate required. The turntable drive power requirements for the HDWC program at-sea test are shown in Tables 3.4.6-1 amd 3.4.2-1.

3.4.6 AT-SEA TEST TURNTABLE

The HDWC program at-sea test requires a turntable capacity of 9.1 KM (30,000 FT). This turntable capacity can be provided by a turntable with an outside diameter of 10 meters with the inside minimum bend as required by the candidate cable (see Figure 2.3-1). Cable stowage depth would be four (4) meters or less dependent on the candidate cable selected. Turntable size and drive requirements can easily be met using rolling element support as discussed in Section 3.4.3.

The turntable on the mothballed cable vessel SUSITNA is suitable for use on the HDWC program at-sea tests. The SUSITNA turntable is rated for 500 tons (453 MTon) on its rolling element support. The turntable stowage bin dimensions are 10 M (33 FT) O.D. X 3 M (10 FT) I.D. X 2.7 M (9 FT) deep. The turntable stowage depth can easily be increased to provide the required at-sea test stowage capacity. Turntable capacity is shown in Table 3.4.6-1, along with depth modifications required and drive horsepower required.

Cable PPC #	Cable Weight	Turntable Capacity	Turntable Depth	Power Required
113	317 MTon	9.6 KM	3.05 M	53 KW
116	342 MTon	9.4 KM	3.23 M	56 KW
119	366 MTon	9.2 KM (30,180 ft)	3.96 M	54 KW

NOTE: Drive power is based on an assumed acceleration from zero to two knots in 1.0 minute. Turntable frictional losses are based on 10 KG per MTon.

TABLE 3.4.6-1 Capacity and Drive Power Requirements of the Existing Turntable on Cable Vessel SUSITNA

The existing SUSITNA turntable drive is capable of meeting the above power requirements but maximum speed capability is unknown because existing drive component data is not available. Turntable load capacity is satisfactory, regardless of which candidate PPC cable (#113, 116, or 119) is selected. The cable previously stored and laid by the SUSITNA in the Cook Inlet of Alaska was similar to the proposed cables.

3.4.7 SUMMARY

A turntable suitable for a commercial application requiring 80 to 100 kilometers of capacity differs from the HDWC program at-sea test's required capacity of 9.1 KM (30,000 ft). The feasibility of the stowage device for the baseline commercial project is well demonstrated (in terms of capacity) in the turntable on board the SKAGGERAK (see section 3.1). This large turntable has the capacity but power and control are insufficient because of the cable tension heave compensation requirements. Therefore, the goal of the HDWC turntable is to prove its feasibility when used in a cable handling subsystem arrangement that provides tension compensation. No turntable size has been selected for the baseline commercial application because the range of cable parameters expected must be known first. All turntable drives should be modular in design, using available equipment to increase reliability and designed to decrease space requirements.

3.5 SURGE-SLACK DEVICE

3.5.1 OPERATIONAL REQUIREMENTS

The principal purpose of the surge-slack device is to provide cable take-up or payout when the payout rate of the storage device does not match that of the cable tensioner. The surge-slack device can provide other functions depending on its arrangement with the storage device. These could include cable guidance, support, level winding, twist removal, and a convenient sensor platform to sense cable touchdown point for control purposes on the turntable. The configuration of the surge-slack device is dependent on the cable storage device it serves and on ships deck area requirements.

Tank cable stowage is mentioned here for background information and it is recognized that torque balanced cable coiled stowage has been specifically disallowed by the cable manufacturer. The tank method of cable storage has the simplest form of surge-slack provision which is provided directly by the cable uncoiling. Thus the prime purpose of the surge-slack device is cable guidance to prevent more than one coil lifting off the stored cable during the short periods of rapid cable acceleration and payout. More than one coil lifting at one time can cause the formation of a kink which would damage the cable. The cable guidance device used in a cable tank storage method is commonly referred to as a "crinoline." The crinoline is usually constructed of tubular members in a lattice construction for light weight. The crinoline is suspended a short distance above the stored cable in the cable tank by winches or other elevating devices, and lowered periodically as the stored cable is removed.

A turntable method of cable storage requires a more complex method of surge-slack control. The surge-slack device must act in conjunction with the cable storage turntable so that one or both devices are actively controlled. Sensors can be used to determine cable position and their outputs or manual controls can be used by the control systems to maintain proper cable length between turntable and cable tensioner.

BULL WHEEL

One method of surge-slack control with turntable cable storage is commonly referred to as the "bull wheel." The bull wheel is a sheave around which the cable is pretensioned in a loop. The loop length is varied to provide proper cable length (see Figure 3.5.1-1). Cable pretension is developed by the force "F" on the sheave using a device such as a constant tension winch or hanging weight. The pretensioning of the cable requires a raised center cone in the turntable around which the cable slips, developing the required tension restraint.

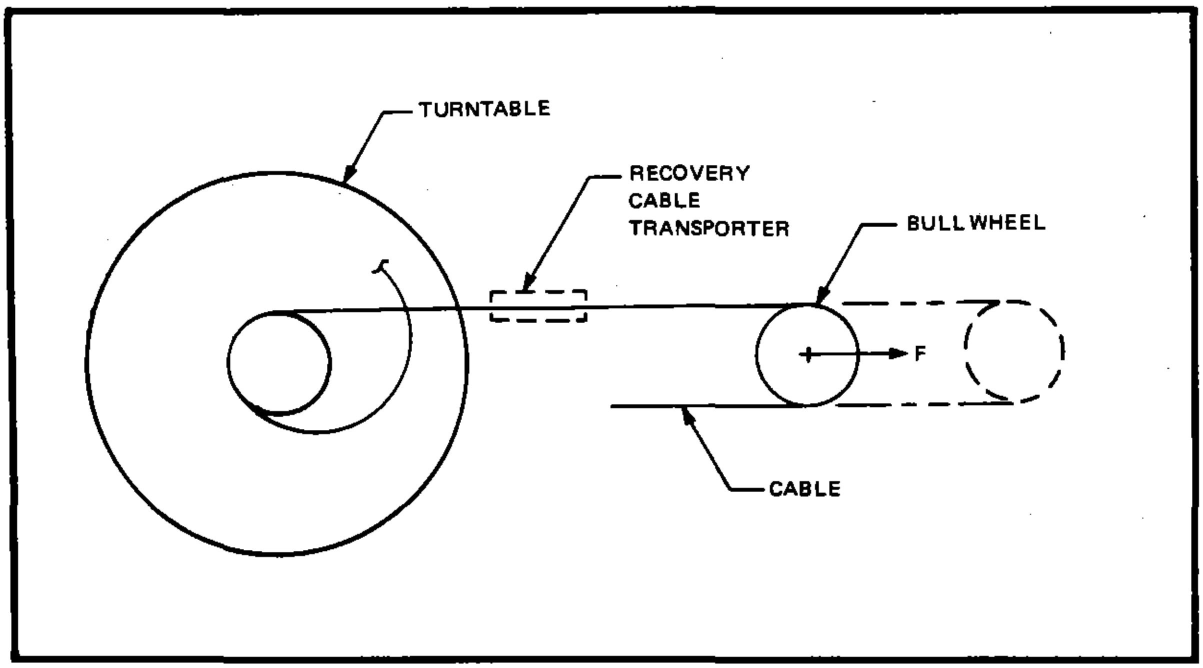


FIGURE 3.5.1-1 Bull Wheel Slack Take-up Concept.
This is a passive take-up device which has been used on the SUSITNA.

Following are the primary advantages and disadvantages of the bull wheel surge-slack device:

Bullwheel Advantages

1. Positive takeup of slack or payout within limits of travel.

2. Can provide large amount of cable takeup which aids manual type of control.

3. Takeup action does not impart twist to cable.

4. Provides a convenient sensing platform for turntable speed regulation input.

Bullwheel Disadvantages

1. Sheave and guide track plus tensioning device require large amount of deck area.

2. Extra cable guidance required.

3. Active control device to provide tension on sheave is required.

4. Additional mass and active control detract from system dynamic response.

5. Additional device required to guide cable and provide twist control during cable recovery.

6. Requires an extended turntable center cone or other device to provide tension restraint.

7. Positive end limits on cable takeup mandates protection against cable runaway by sufficient device strength or other means.

8. Cable transporter and pickup arm required for cable guidance into the turntable during loading and retrieval operations.

PICKUP ARM

A second method of surge-slack cable displacement control with a turntable is an overhead pickup arm. The pickup arm is usually an open lattice boom-like structure mounted so that it overhangs the turntable (see Figure 3.5.1-2). The surge-slack takeup is provided by the cable varying its own touchdown point circumferentially on the turntable. The anticipated angular change in touchdown point for cable displacement control is small but does cause a proportional amount of twist in the cable which is discussed in section 3.5.2. The pickup arm can be pivoted from side to side about a vertical axis to aid in cable positioning from inside to outside turntable diameter, especially with the large diameter turntable that may be required for a baseline commercial project.

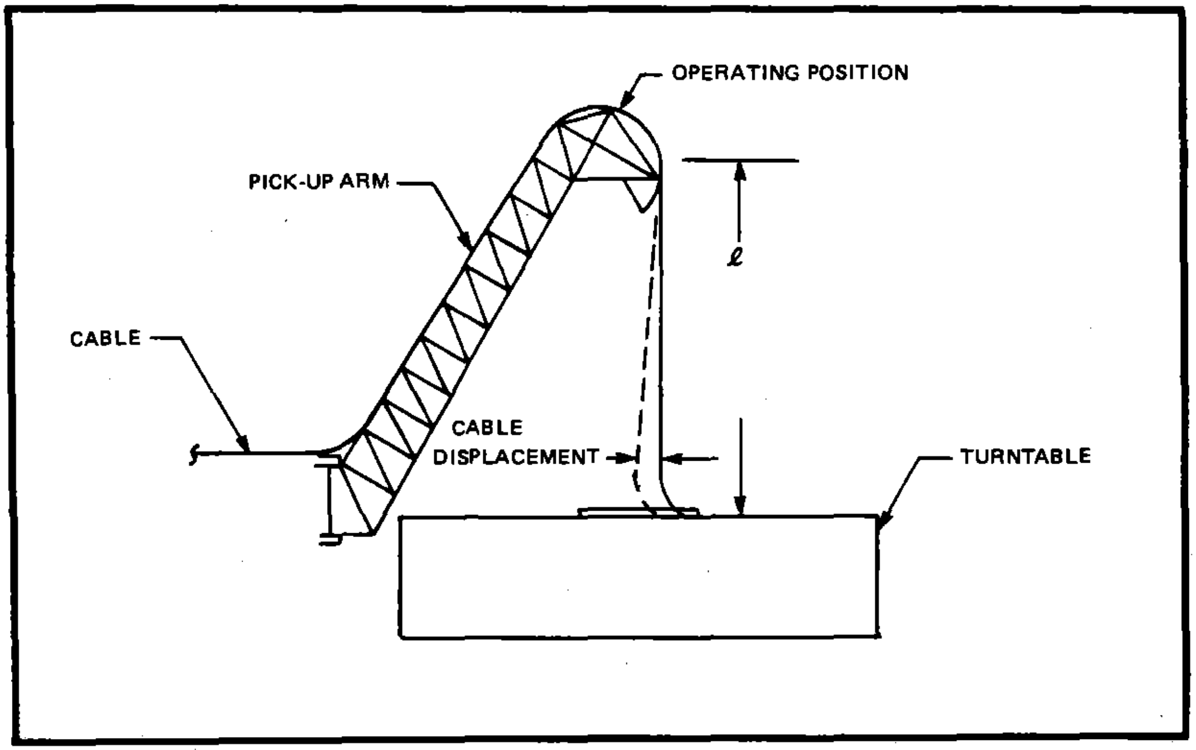


FIGURE 3.5.1-2 Pickup Arm Concept of Slack Takeup Device. Variations in relative payout rates are compensated by the cable pickup point moving around the turntable.

Following are the primary advantages and disadvantages of the pickup arm surge-slack device:

Pickup Arm Advantages

1. Passive takeup or payout, no active control devices required.

2. No additional mass or active controls enhances dynamic response.

3. Provides cable guidance to turntable during loading and unloading without additional devices.

4. Provides a means to control cable twist during cable recovery and/or cable loading.

5. Overhead boom configuration does not require limited deck area.

6. Cable run-away does not jeopardize takeup arm or turntable, because cable would be withdrawn in a manner identical to a fixed cable tank.

Pickup Arm Disadvantages

1. Cable twist is associated with cable takeup or payout.

2. Cable takeup or payout is limited by cable twist characteristics.

3. Sensing of cable position for turntable speed controls is more difficult than for the bull-wheel concept.

4. Cable transporter required with small diameter cable and/or extended distance between turntable and cable tensioner.

3.5.2 CABLE INTERFACE CONSIDERATIONS

The critical cable interface considerations with the surge slack device are low level handling tension, minimum bending diameter, and cable twisting. Handling tension is cable tension required to overcome guidance trough friction and elevation changes. Cable twisting may be caused by the gripping or guiding of the cable or imparted into the cable by the pickup arm method of surge-slack takeup (as discussed in section 3.5.1).

Twisting of submarine power cable can be of two types, elastic and inelastic. Elastic twist is that in which no permanent deformation occurs and the cable returns to its normal state upon removal of the twisting force. Inelastic twist is that in which permanent deformation does occur and the cable will not return to its normal state when the twisting force is removed. In all cases, cable twist should be held to within the elastic range. No firm twist data on the cable is presently available but estimated values have been developed.

Estimated cable twist data was developed using an assumed cross section of a representative size cable (see Figure 2.2-1). The torsional stiffness was calculated to be 775 NM per degree per meter of cable length (270,000 lb-in/1°/inch). Maximum torque, i.e., elastic torque limit, on cable was 653 NM (5778 lbs-in) at lead sheath yield with torque applied with cable at 2.3 Mton (5000 lbs) tension. Assuming a linear relation, the elastic twist constant of the cable is (653/775) = .843 degrees per meter (.021°/inch). A linear elastic behavior was assumed for the lead sheath material with a yield point of 6895 KPa (1000 lbs/in^2). The lead sheath was the first material to reach yield stress.

All intentional twisting of the cable must be within the elastic range of the cable's torsional stress. Therefore, the height of the pickup arm

"ℓ" above the turntable (see Figure 3.5.1-2); must provide sufficient cable length so the cable twist that results from cable touchdown variance is within the allowable cable parameter. The required height "ℓ" can be determined using the take-up length required to compute the angular variance on the turntables with the following formula:

$$\frac{L\ 360}{D \pi C} = \ell$$

L = take-up length
D = minimum cable bending diameter
π = pi
C = cable twist constant (.843°/M)

The "ℓ" required for PPC #116 cable (Table 2.3-1) for a heave compensation take-up of $\pm$ 15 CM is 2.9 M (9.6 FT) using Table 2.3-1 and the estimated cable twist constant values.

The height "ℓ" provides only the dynamic take-up that the cable experiences while stationary, such as that resulting from the tensioner's heave compensation operation. Additional cable twist can be accommodated during longitudinal cable movement as a result of the cable torque developed. The resultant cable torque will "spin" or cause the cable to slip in a rotational manner on its guidance rollers or troughing. The maximum cable torque that can be applied safely is that equal to the cable's elastic torque limit. The spun cable length can be computed using the following formulae:

$$T_s = .5 \mu \ wd^2$$

$$L_s = \frac{T - \mu W \ell d/2}{T_s}$$

T = cable's elastic torque limit
W = cable weight per unit length
d = cable diameter
μ = coefficient of friction
T_s = torque to spun cable
L_s = spun cable length in cable diameter
ℓ = vertical cable length

Length L_s is the total length of cable that can be subjected to twist that occurs during take-up on a pickup arm device. The total take-up allowed can then be calculated using the cable twist constant to relate that cable length to degrees of turntable rotation allowed, per the following formula:

$$\frac{(\ell t\ L_s)CD\pi}{360} = L_t$$

where L_t = allowed take-up length
D = turntable minimum storage diameter

Using the estimated data for elastic torque limit and cable twist constant along with an assumed .3 for "μ" and the "ℓ" determined earlier, L_t is 5.1 M (16.7 FT) of cable take-up. As L_t is the plus or minus allowed, the total assumed cable take-up is 10.2 M (33.4 FT) prior to exceeding the cable's estimated elastic torque limit of 653 NM.

CABLE TWIST CONTROL

During cable loading or recovery operations, cable twist states between low cable tension and high cable tension or coiled storage to turntable may differ and can accumulate. See cable twisting considerations in Section 3.4.1. To control cable twist, the accumulated twist is removed by adding or subtracting a full or partial cable turn on the turntable. This is done using the pickup arm to rotate the cable's touchdown point in relation to the turntable, as previously discussed in Section 3.4.1, Twist Control.

One successful method of monitoring cable twist is by watching a longitudinal painted stripe on the cable's outside jacket. Another method used with exposed armor cables is to measure the nominal gap between the armor wires. Either of these methods will allow visual monitoring of accumulated cable twist and will allow cable twist removal by manually controlling the cable machinery.

3.5.3 CABLE TENSIONER/TURNTABLE INTERFACE

There are several cable handling subsystem operational interface considerations with which the surge slack device must contend. The first is the amount of cable movement required by the tensioner's heave compensation. Second is the amount of additional length or decoupled cable required to accommodate the different acceleration or deceleration rates between the tensioner and the turntable. Third is the failure mode of each individual component and the resultant cable requirements. Fourth is the dynamic response of the surge-slack device and its effect on the cable and its take-up requirements.

All of these considerations are affected by the performance of the vessel, along with its maneuvering and control capability. This becomes especially true with certain cable handling subsystem failure modes. Prior to surge-slack device design, the vessel's parameters must be known, so certain operational compromises, such as reducing vessel acceleration and lay speed at shallow depths, can be made to reduce the cable handlng subsystem performance required during component failures. Failure modes or periods that have significant vessel performance considerations are:

- o Cable tensioner or turntable failure and stopping in an emergency stop while paying out cable at maximum speed.

- o Cable laying speed in shallow depths, where vessel maneuvering may not assure cable integrity.

- o Surge-slack device failure during tensioner heave compensation operation.

- o Tensioner, surge-slack device, or turntable failure during maximum change in cable laying speed.

EQUIPMENT FAILURE CONSIDERATIONS

Both the cable tensioner and the storage turntable must be equipped with brakes for emergency stops and for holding position during shutdown periods. These brakes provide the worst case condition for the surge-slack device, as it must provide cable control as required to prevent cable damage.

Turntable stopping time is the primary factor that must be considered to determine surge-slack device response. The frictional losses of a turntable are considerable, especially if rolling elements support is used, and this shortens the stopping time when compared to the acceleration time. Table 3.5.3-1 gives the stopping time for several turntables.

TURNTABLE OUTSIDE DIAMETER	DRIVE POWER	TURNTABLE CAPACITY	COAST STOP*	BRAKE STOP
Rolling Element, 1% Friction				
23 Meters	1346 KW	104.2 KM	9.1 SEC	3.7 SEC
10 Meters	69 KW	11.5 KM	4.6 SEC	2.1 SEC
Fluid Bearing, .1% Friction				
23 Meters	494 KW	104.2 KM	90.7 SEC	14.5 SEC
10 Meters	17 KW	11.5 KM	46.1 SEC	11.1 SEC
SUSITNA Turntable (see Section 3.4.6)				
10 Meters	56 KW	9.4 KM	4.6 SEC	2.1 SEC

*No brakes used.

NOTE: Drive data is based on one (1) minute acceleration time to two (2) knots using PPC #116 cable. Stopping times are from two (2) knots with turntable at maximum capacity and RPM, with brake capacity equal to drive power.

TABLE 3.5.3-1 Turntable Stopping Times

As shown in Table 3.5.3-1, the fluid bearing supported turntable would require additional braking capacity along with more surge-slack take-up as compared to a rolling element supported turntable.

Vessel data will be required prior to final design, but for the purpose of further discussion it is assumed that vessel control and maneuvering will be sufficient to provide cable safety during a cable handling subsystem emergency shutdown. With this assumption, the worst case cable handling subsystem failure would be with the tensioner going to an emergency stop from a cable speed of 2 knots. In this failure mode, the turntable would have to perform its own emergency stop with the surge-slack device accommodating the cable during the turntable's stopping time (see Table 3.5.3-1). The cable length that must be accommodated by the surge-slack device can then be calculated using the average speed and the stopping time. Using the data in Table 3.5.3-1, this length is 2.2 M (7.1 FT) for the SUSITNA turntable when using brakes.

Failure of the surge-slack device's active components must also be considered. This type of failure is not time-critical, as both the tensioner and turntable are still operating. It would require some type of passive device in the system for a bull wheel device to provide take-up for heave compensation and more critical control over total system speed up and slow down. This type of failure applies to a pickup arm only during cable loading or retrieval operations.

CABLE TAKE-UP

Each of the cable handling subsystem equipment interface considerations require an amount of surge-slack cable take-up, with each amount being additive to the others.

The amount of the cable take-up required for cable tensioner heave compensation is the input to the tensioner discussed in Section 2.5.3. This amount of cable take-up is required regardless if the cable is stationary or under continuous payout or retrieval, and may change at different depths, depending on vessel-to-cable tension heave parameters.

The second required amount of cable take-up is that required to accommodate the acceleration difference between the cable storage turntable and the tensioner. The primary acceleration rate of the tensioner is that required to maintain cable tension during heave compensation operation. The secondary tensioner acceleration or speed rate change is that required to fulfill the vessel's acceleration or cable lay speed rate of change as controlled by the vessel's integrated control system. The tensioner's primary acceleration rate will be one or more orders of magnitude greater than its secondary acceleration; therefore, it will not be a limiting factor in determining this cable take-up amount. This amount of cable take-up will be reduced to zero if the turntable's acceleration rate equals or exceeds the required cable lay speed rate of change.

Cable handling subsystem decelerations must be considered in a similar manner. The cable stowage turntable deceleration rate is different from its acceleration and is dependent on its support method and drive size, as shown in Table 3.5.3-1. It is assumed that the cable tensioner's maximum braking effort or tension is limited by and equal to the cable's maximum allowable tension (see Table 2.3-1) and is dependent on the amount of cable suspended (i.e. water depth).

WATER DEPTH	SUSPENDED CABLE WEIGHT	TENSIONER BRAKE STOP
2134 METERS (7000 FT)	77.7 MTON	3.3 SEC
2000 METERS (6562 FT)	72.8 MTON	3.1 SEC
1500 METERS (4921.5 FT)	54.6 MTON	2.3 SEC
1000 METERS (3281 FT)	36.4 MTON	1.6 SEC
500 METERS (1640.5 FT)	18.2 MTON	.8 SEC
100 METERS (328.1 FT)	3.6 MTON	.2 SEC

NOTE: Tensioner efficiency and overboarding device losses have not been included. Stop time is from 2 knots.

TABLE 3.5.3-2 Tensioner Braking Stop Times kept substantially lower than that indicated by the estimated data in Section 3.5.2.

The non-twist feature of the bull wheel can also be a disadvantage, which could be overcome by the addition of a cable transporter device, which would provide the tension restraint normally provided by the turntable's extended center cone. This would free the cable in the turntable, allowing twist to be added or deleted by turntable rotation at the rate of 360° of twist over the length of cable per turntable rotation. The use of a bull wheel would not provide cable twist control during cable recovery, so an overhead pickup arm would still be required for cable guidance onto the turntable.

ADVANTAGES/DISADVANTAGES OF PICKUP ARM

The major advantages and disadvantages of the pickup arm are opposite to those of the bull wheel. Advantages include less deck space utilized and runaway cable safety, but the cable twisting must be evaluated using final cable twist parameters. The allowable pickup arm take-up amounts must then be analyzed to determine if adequate cable is available under the various equipment failure modes. The interconnecting of tensioner and turntable control is required to closely coordinate the starting and stopping of cable movement to minimize the pickup arm's take-up required.

3.5.4 SUMMARY

The pickup arm concept is recommended for the HDWC at-sea test based on the adequate cable take-up lengths calculated and the rapid deceleration of its small turntable. These preliminary values of take-up length also appear to be sufficient for a baseline commercial system when using a rolling element turntable support system.

3.6 CABLE HANDLING MACHINERY CONTROLS

3.6.1 GENERAL

The overall requirements of the control system for the cable handling machinery are:

1. To insure that the cable is laid with positive tension at the sea bottom touchdown point.

2. To insure the cable is handled in a controlled manner so that allowable cable parameters are not violated.

3. To provide status and malfunction indications that ensure the program control has adequate data to direct the lay operation.

The basic task of the control system is to direct the cable handling subsystem to successful completion of the HDWC program at-sea tests. Section 2.0 discusses the overall task of the HDWC program and the approach used for the solution of cable tension compensation.

The proposed method of cable handling equipment control addresses the interface of vessel and machinery controls. This interface and basic control philosophy are shown in Figure 3.6.1-1.

CONTROL RESPONSIBILITY

The proposed machinery control in normal operation will respond only to a command which may be an actual or a modified tension level. It will not sense the sea bottom condition or determine the appropriate level of cable tension. These parameters will be measured and controlled by the vessel's cable lay integrated control system.

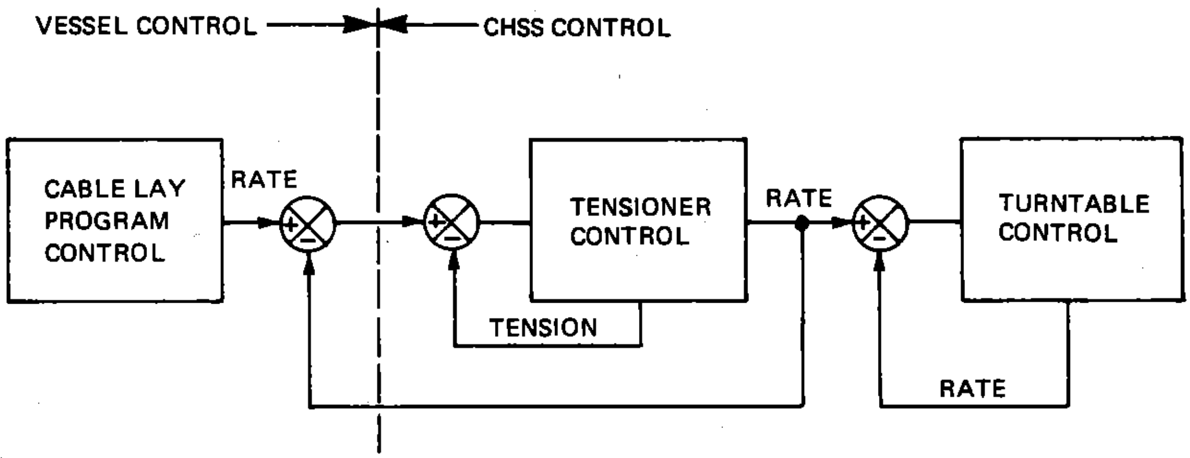


FIGURE 3.6.1-1 Basic Control System

The proposed normal mode of system control is similar to standard control schemes with the exception of the active tension loop. The command to the tensioner control will either be the actual tension required or a tension signal modified to ensure that a commanded rate is obtained. The controls are complex dynamic systems and detailed explanation is required to fully describe their operation. This figure is simplified for illustration purposes.

In summary, the cable handling machinery controls are driven by the short period internally sensed dynamics and the long period external commands resulting from the overall cable lay operation. Consequently, the short period commands are developed and supplied entirely by the linear cable tensioner controls and the long period commands are supplied by the cable vessel's integrated control system. Two-way communication between both control systems with dynamic interaction and compatibility at all times is required for proper equipment operation.

Figure 3.6.1-2 is a diagram showing the on-board control functions.

FAILURE CONSIDERATIONS

A primary consideration is the effect of failures or malfunctions of any part of the systems involved. The cable handling subsystem controls will be designed to operate under various conditions of system degradation including the loss of outside commands. The dynamic studies in section 2.5 show that the frequency response requirements of the control system are such that it would be possible for all of the machinery to be manually controlled as a routine option or in an emergency. Therefore, the possibility of serious damage to the cable as a result of system malfunctions could be greatly reduced by the assumption of control responsibility by the operators on board the lay vessel. However, continuous manual control under worst case conditions would be physically and mentally exhausting for an operator if continued for long periods of time.

MACHINERY CONTROLS

Three primary pieces of machinery comprise the cable handling machinery controls:

1. The cable tensioner (an active control element).
2. The surge/slack control device (may be active or passive).
3. The storage turntable (an active device, but by virtue of its large mass and the limited power supply capabilities, will be limited in its dynamic response).

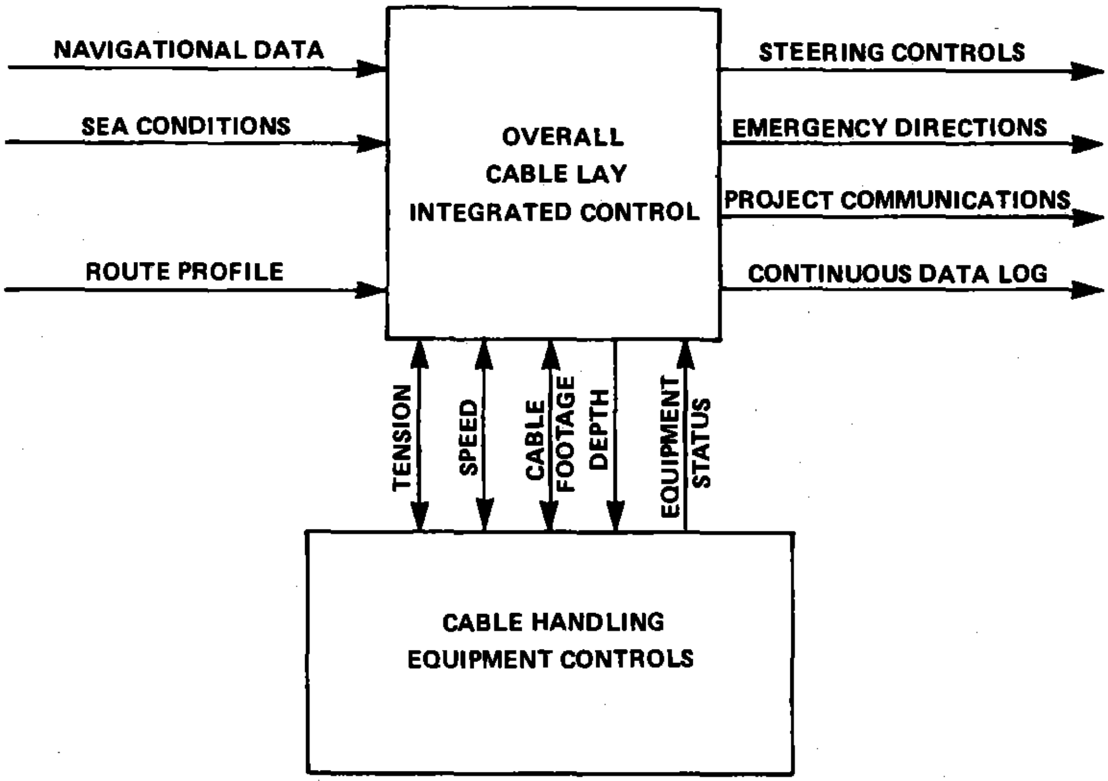


FIGURE 3.6.1-2 Cable Lay On Board Controls

The on board controls comprise two or more fundamental systems. The cable handling equipment controls will be separate from the overall controls but with constant data transmission to and from the overall control system.

These three machinery units, working together, must be capable of handling the worse case cable displacement during tension compensation and speed change operations. Cable displacement during tension compensation will cause a short-term difference in speed rate between the cable tensioner and the turntable. In addition, the turntable must continuously change its rotational rate to compensate for the cable's stored coil diameter change during load or unloading of the cable. A third long-term rate change that occurs and affects this equipment is the overall speed variation in vessel/cable lay operations. These three factors must be assessed to determine their magnitude prior to the design of the control system and surge/slack device.

The overall control functions of the cable laying operation are shown in Figure 3.6.1-3.

3.6.2 CABLE LAYING CONTROL

Though the requirements of this program may dictate a different approach to achieve successful positioning of cable on the bottom of the channel, a discussion of traditional methods of cable laying follows.

The quotes below, taken from the Integrated Control System study for the HDWC made by Makai Ocean Engineering, describe methods used in the past for control of the cable lay.

> In laying power cable, the vessel position and therefore the bottom cable position is normally controlled by mooring a vessel and winching it across the body of water or by dynamically positioning the vessel with some form of propulsion. In shallow water, high current situations, the mooring system is normally used and in deeper water the propulsion methods are used. Examples of moored cable laying is the Hong Kong and Northumberland Straight deployments and an example of the dynamic

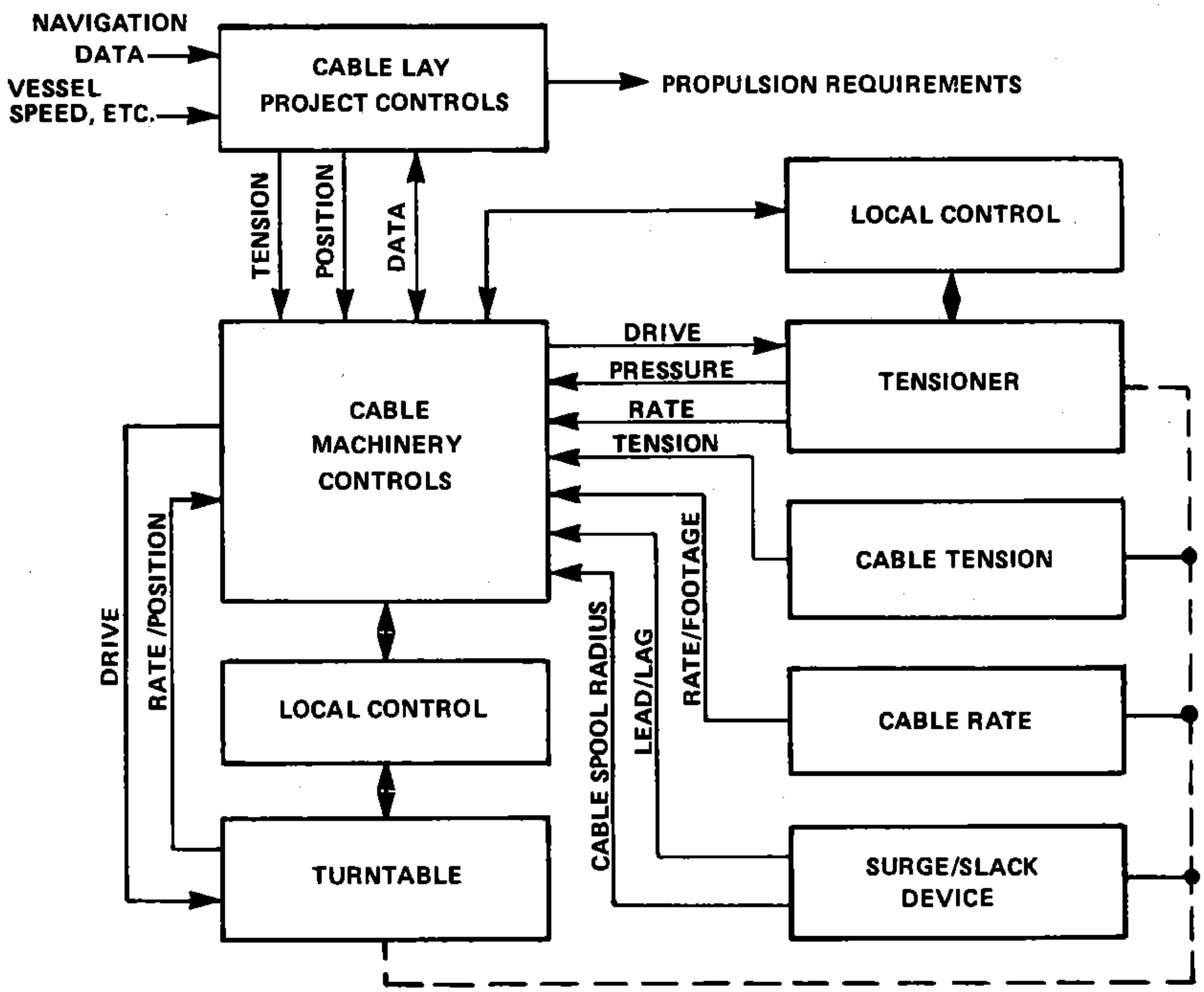


FIGURE 3.6.1-3 Cable Handling Equipment Subsystem Controls

This diagram shows the basic functional relationships of the various cable handling support system machines.

positioning laying would be the Pirelli cable lay to Sicily in 305 m (1000 ft) of water or the connection of the main island of Honshu in Japan to Hokkaido by Sumotomo, also in 305 m (1000 ft) of water.

The proper tensioning of cable on the bottom is conventionally achieved by knowing the depth of water and maintaining a corresponding tension at the surface with a tensioning machine. As the vessel moves, cable tension increases and the tensioning machine pays out more cable. Communication cable, however, is laid quite differently. Communication cable is not normally laid on the bottom under tension because of the desire to avoid any freespan on the bottom. Surplus cable is laid resulting in a zero bottom tension. Such a laying technique requires precise navigation of the laying vessel and this is normally achieved by paying out a small steel taut wire in conjunction with the communications cable. The length of taut wire deployed is precisely measured and its rate of payout governs the communications payout speed.

3.6.3 CABLE TENSIONER CONTROLS

Investigation of the dynamics of the cable tensioner resulted in a simple control loop (shown in Figure 3.6.3-1) which offers the potential for satisfactory performance within the constraints investigated. However, to develop a control for the actual machinery, it will be necessary to continue investigations to develop detailed characteristics.

The dynamic studies of the cable tensioner have concentrated on ensuring that a linear type of machine can be refined to provide the dynamic responses necessary to accommodate the vessel responses to sea conditions. This requires that the cable tensioner must respond to spectral-analysis-derived frequency responses of 2 to 3 cycles per second

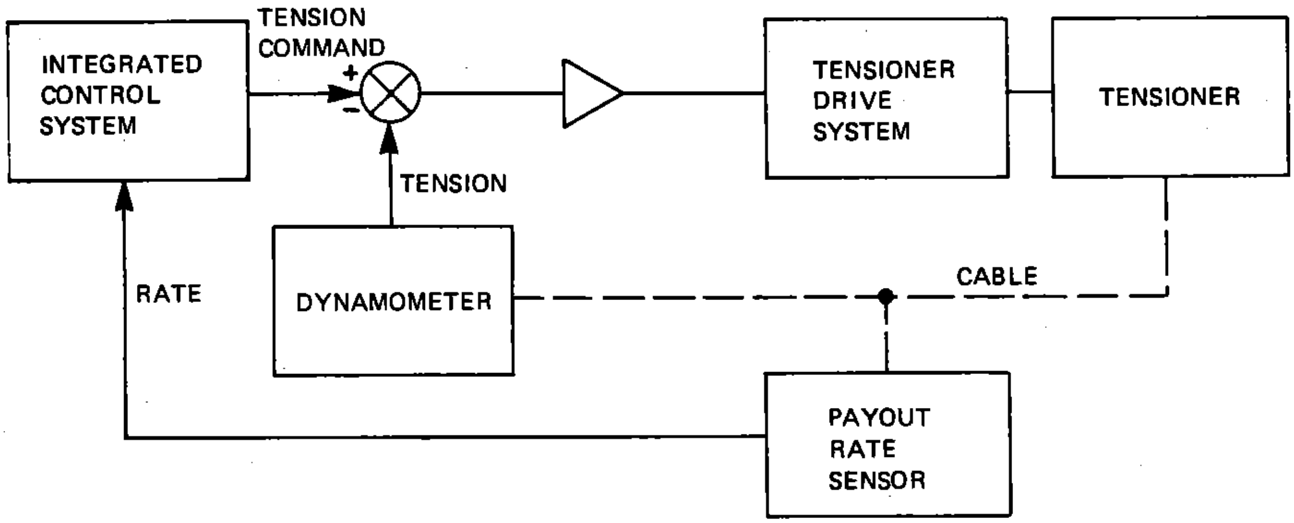


FIGURE 3.6.3-1 Simplified Tensioner Control

with limited amplitude and must respond to lower frequencies at much greater amplitudes. Though this is the characteristic response of this type of machinery, previous applications have not required these high control frequencies. The following factors must be considered when designing the hardware.

CABLE STIFFNESS

The forward loop gain, or sensitivity of the cable tensioner controls to the variations in tension, will vary as a function of the stiffness of the cable being laid. Cable stiffness is a function of cable elongation under tension and is inversely proportional to the suspended cable length. In the dynamics studies, the cable stiffness has been represented by single values which were estimated to be realistic based upon studies of the proposed cables. However, the cable represents a distributed mass system and the apparent stiffness of the cable will vary as a function of the frequency of the driving functions, i.e., the sea states. This complex relationship can best be quantified by empirical measurements of a vessel cable system. The effect will be that the forward loop gain of the controls will be a variable, dependent upon the frequency of the driving commands; thus the cable tensioner controls will require further refinement. The resulting control system may require adaptive characteristics and features which allow the shaping or signal conditioning to be changed to suit the detailed characteristics encountered during the at-sea tests.

For example, to re-establish a linear gain to within defined limits, a relatively simple first or second order analog shaping filter can easily be added.

As the cable stiffness varies as a function of depth of the channel, the forward loop gain will change dramatically in a long term and it will probably be necessary to shape the gain of the controls as a function of average tension level.

DAMPING

A third variable is the damping requirement of the system. All powered servo controls have a tendency towards instability and require damping. This damping could be the natural friction present in mechanical or hydraulic components or it may be necessary to derive and amplify characteristics of the system to artifically suppress high accelerations which are the primary cause of instability. This is demonstrated by the mathematical models simulating the linear tensioning machine as discussed in Section 2.5. In the past the stability of the machine has been augmented by a rate signal derived from the track drive mechanism but this was found to be ineffective at the higher loop gains necessary for this machine and application. The use of differential pressure from the hydraulics was a more effective damping signal because it provides more lead or anticipation. However, in order to use pressure for damping purposes, it is necessary to eliminate all steady state pressure from the signal, otherwise the pressure signal would result in a steady state error in the level of tension achieved. This highpass filter or washout (washing out the D.C. component of the signal) can be mechanized with simple analog circuitry and has been successfully used in the past.

Considering the above, it is apparent that the relatively simple linear control schemes that have been investigated must be modified and increased in complexity. This is necessary to accommodate the requirements stated in Section 2.5.3 under "gain scheduling" which are functions relating to the heave and cable depth disturbances. This does not represent an area of risk as the modern high speed aircraft controls used on late model commercial and military aircraft possess these control characteristics.

REDUNDANT CONTROLS

The potential complexity of the control systems raises the question of failure effects and the need for redundancies to obtain a level of reliability that is compatible with HDWC program objectives. Because the control frequencies are within the capability of a human operator, all that would be required in the cable tensioner controls would be a dual loop in certain areas which would indicate a potential malfunction. This would alert the operator who would assume control while technicians determine and correct the possible malfunction.

Figure 3.6.3-2 shows the basic informational and control channels required for the tension controller.

The cable tensioner controls would provide two modes of control, tension and speed. Both modes are closed loops so that the cable tensioner would precisely follow the commands provided. For automatic speed control either the machine speed or the cable speed could be used for the feedback signal. For the tension mode of control the feedback signal would be the output from the cable tension sensing dynamometer.

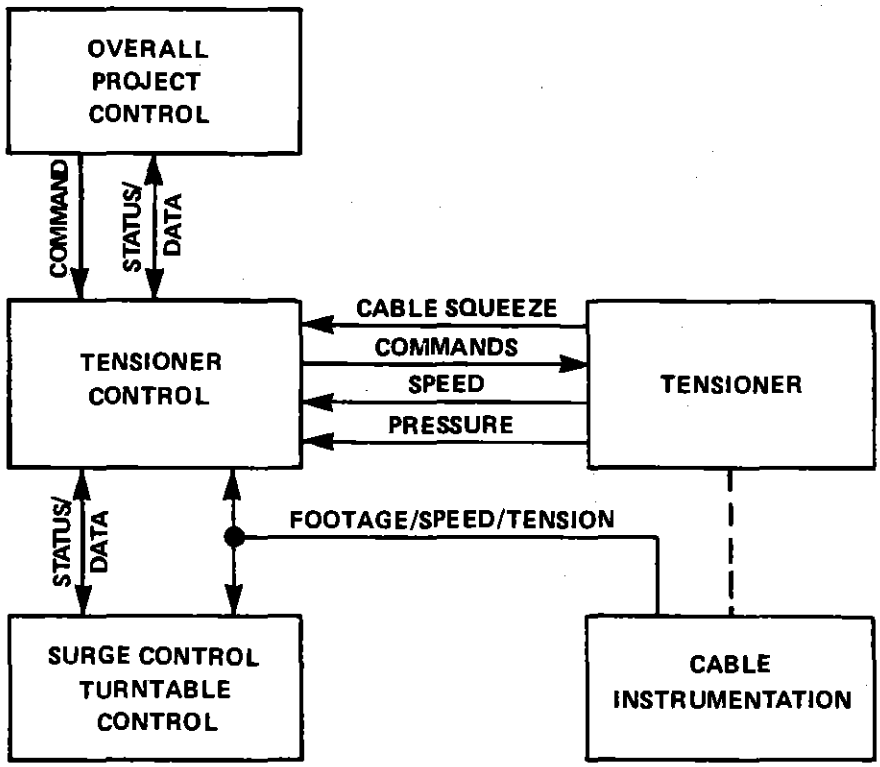


FIGURE 3.6.3-2 Basic Tension Control Functions

This functional arrangement shows the interrelationships necessary for control of the tensioner.

POWER SOURCES

Motive power source choices for the linear cable tensioner are limited to hydraulic or D.C. power. The proven characteristics of hydraulic controls are well known and most high power machinery controls are of this type. D.C. power (as previously discussed in Section 2.5.4) will be investigated to determine if the performance characteristics required can be provided should this drive be utilized in the cable storage turntable.

The stability augmentation functions required of the system will be similar regardless of the motive power system used. The derivation of the back EMF signal from the DC drive motor is analogous to the pressure signal provided by a hydraulic system and consequently similar circuitry can be used regardless of the motive power device.

3.6.4 SURGE SLACK DEVICE

The design of the pickup arm is considered an integral part of the turntable system design and will be strongly influenced by the allowable cable parameters. The surge-slack device is discussed in Section 3.5, but at this point the pickup arm design could follow one of three basic concepts:

1) Passive device - This design would be for a manually moveable device using deck machinery; however, during cable laying operation, it would remain in a fixed position, where all take-up would be provided by the varying cable touchdown point.

2) Manual Active device - This design would include a power unit and controls which would be used to manually move the pickup arm across the turntable diameter and raise or lower the arm as required to move the cable's touchdown point on the turntable. This method would provide all of the capability of the passive device and, in addition, would provide cable twist control during loading or during cable retrieval operations.

3. Automated Active device - This design will be similar to the manual active device but will be incorporated into the turntable automated controls to move the pickup arm in response to the turntable speed and will require a sensing device.

For the HDWC program, a passive device would be sufficient during cable laying operation as the preliminary twist data indicates that a fixed take-off point will not unduly stress the cable. However, cable twist probability and control during the cable retrieval operation require that the manual active device be used. Cable twist cause and control are discussed in Section 3.5.2. For a baseline commercial system where the turntable diameter could be approximately 24 meters, an active device should be considered.

A major consideration in the pickup arm design and controls is the amount of slack cable movement required between the tensioner and turntable during normal and emergency operation.

PICKUP ARM CONTROL

The pickup arm control will be integrated with the turntable controls. Sensors mounted on the tip of the pickup arm above the turntable measure the cable angle coming off the turntable. These are feedback sensors for the automated turntable controls, indicating turntable lead or lag and cable spool radius. Figure 3.6.4-1 is a simplified functional diagram of the automatic turntable controls. End travel limit sensors will be provided which emit a signal to provide warning of an out-of-limit condition. This signal could be used to change turntable speed or to trigger another control response. There is no requirement for active controls for the pickup arm for the HDWC program at-sea tests; however, the active pickup arm which could be required for a baseline commercial system would be relatively straightforward in design and could consist of readily available components.

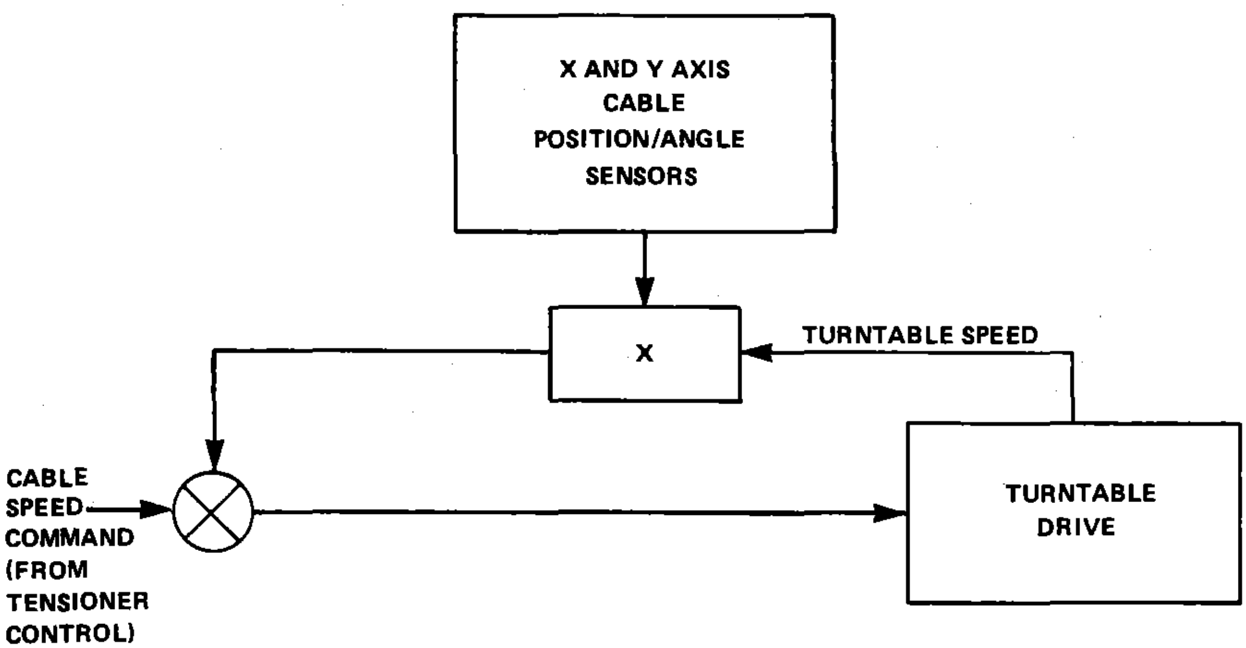


FIGURE 3.6.4-1 Simplified Diagram of Turntable Control

The turntable control is basically a speed control that varys in response to the relatively long term variations in the cable pick-off lead/lag relationship (X axis sensor) and the very long term spooled radius (Y axis sensor).

3.6.5 STORAGE DEVICE

The minimum bend radius of the PPC #116 cable requires a turntable inner diameter of approximately 7 meters, the outer diameter, which depends on desired cable storage capacity, could be 10 to 24 meters. Therefore, the turntable will have to vary in speed to maintain a given cable payout speed, see Table 3.6.5-1. To maintain a constant 2 knot cable speed, the turntable must rotate at 2.8 Rev/Min when the cable is leaving at the 7 meter inner diameter and rotate at 0.818 RPM when the cable is at the 24 meter outer diameter.

The turntable automated control is basically a closed loop speed control (as shown in Figure 3.6.4-1) using three control signals; cable angles at the pickup arm end (both X and Y axes), and turntable speed data. The cable speed command originates from the cable speed sensor on the cable. Since the cable tensioner will be responding to vessel dynamics, this signal may require filtering to provide a smooth continuous average cable speed, though the turntable will not respond to any high frequencies because of its inherent low dynamic response. The signal must then be conditioned to provide the desired turntable speed depending on where the cable is leaving the turntable. This signal is most effectively obtained from a cable lateral angle (Y axis) sensor on the pickup arm.

The turntable speed will be compared to the speed command which will be a function of the payout rate. Any differences will be transmitted as commands to the drive system to accelerate or decelerate the turntable.

The position sensors on the pickup arm which transmit lead or lag signals are the secondary feedback sensors for the closed loop speed circuit. The geometry of pick-up arm sensors and the dynamics of operation must be further studied but both the X and Y axis angles of the cable provide information to the turntable control.

CABLE LAY SPEED	7 METER INNER DIAMETER	10 METER MAX DIAMETER	15 METER MAX DIAMETER	24 METER MAX DIAMETER
2 Knots	2.8 RPM	1.97 RPM	1.31 RPM	0.818 RPM
1 Knot	1.4 RPM	0.98 RPM	0.654 RPM	0.409 RPM
0.5 Knot	0.7 RPM	0.49 RPM	0.327 RPM	0.204 RPM

TABLE 3.6.5-1 Turntable Speeds for Constant Payout Rates

The turntable controls for the HDWC program at-sea test and a baseline commercial program will be similar but the commercial system will introduce special considerations because of greater drive power requirements. The differences are quantifiable but the primary constraint will be the overall program economics which may limit the power available and hence the acceleration capability.

The basic turntable control system must allow manual control at all times so an operator can override or correct a perceived error build-up as indicated by the touchdown point of the cable in the turntable.

3.6.6 CABLE MACHINERY SENSORS

The cable machinery control system uses sensing devices of two basic types: general monitoring sensors and critical control sensors. The general monitoring sensors are those associated with each power unit, providing coarse operating performance indications. If a failure occurs, there would be no immediate effect on cable control. The control sensors are those critical to cable machinery control and if any of these sensors were to fail, the cable lay process would immediately be affected. The critical sensors for cable machinery are:

1. Cable Tension Sensor
2. Cable Speed and Footage Sensor
3. Tension Machine Speed Sensor
4. Tension Machine Stability Augmentation Sensors
5. Pickup Arm Cable Position Sensors
6. Turntable Speed Sensor

CABLE TENSION SENSOR

The purpose of the tension sensor is to measure the tension developed in the cable. The tension sensor cannot measure the overboarding cable tension directly because of the friction losses of the overboarding device. Such losses are additive to tension sensed during payout and subtractive during inhaul of the cable. Figure 3.6.6-1 illustrates this problem.

Proven methods of tension sensing include measuring foundation forces at the overboarding sheave or the cable tensioner or measuring the force to deflect the cable by a calibrated offset. Measurement of forces in the overboarding device is not practical because of the severe environment in which the instrumentation must operate. However, a theoretical advantage is that a more direct tension measurement can be obtained.

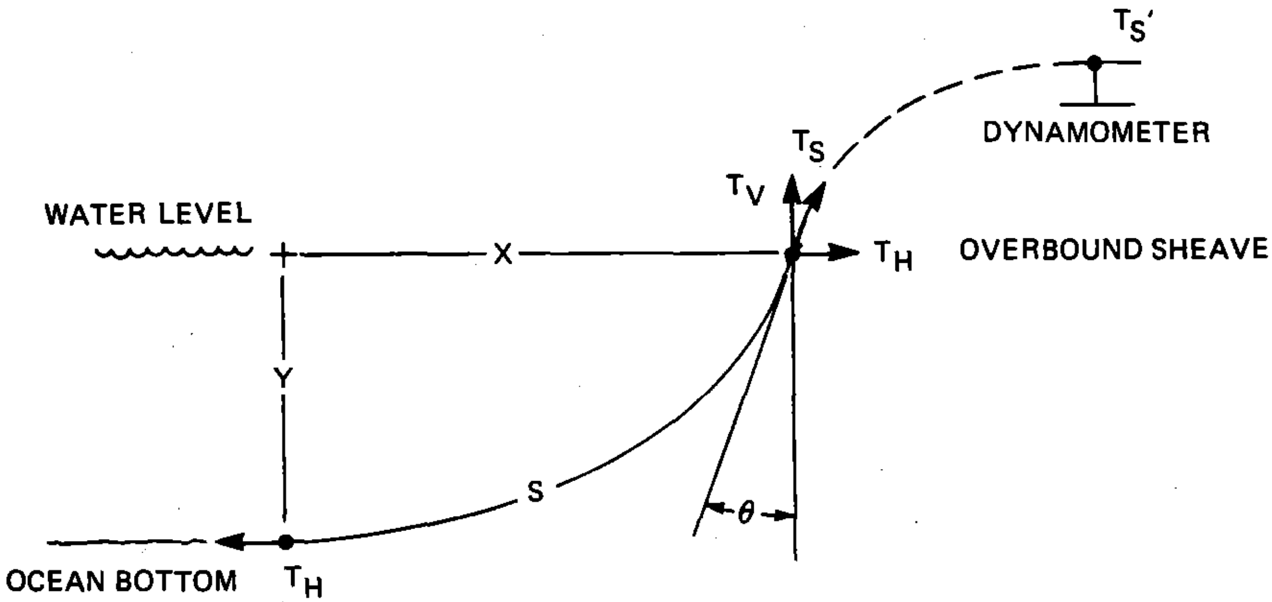
Tension sensing by direct measurement of foundation forces in the cable tensioner is a method used on pipe tensioners and draw off-hold back tension machines. Though this method provides an economic method of tension sensing, it limits the dynamic response capability of a tensioner due to the large frame mass, and cross coupling effects with the track drive system.

The precision deflection method measures the resultant force of a known angular bend in the cable from which the cable tension is then calculated. Dynamic response is high because the machine mass can be minimized when a curved sliding surface is used for cable contact. Cable weight affects tension accuracy, especially at low tension levels, and is minimized by ensuring that the length of suspended cable is kept to a minimum without violating the cable's minimum bend radius.

PRECISION DEFLECTION METHOD

The precision deflection method of tension sensing is the most practical choice because it offers the best dynamic response, along with the benefits of small size and high repeatability.

The precision deflection dynamometer table has little mass and is isolated from the tensioner machine dynamics. Typical load cell accuracy



Y = Distance from touchdown point to water surface (datum)
X = Distance from datum to lay vessel position
T_H = Bottom touchdown tension
T_S = Cable tension at surface
T_S' = Cable tension at dynometer
T_V = Cable weight
S = Length of cable from lay barge to touchdown
θ = Angle of cable from vertical at surface

PAYOUT DIRECTION $T_S' \cong T_S - 1800$ LBS

PICKUP DIRECTION $T_S' \cong T_S + 1800$ LBS

(OVERBOARDING LOSSES 0.8 M TONS
ASSUMED)

FIGURE 3.6.6-1 Fundamental Cable Tension Relationships at Zero Relative Velocity is 0.25% and expected tension signal accuracy is 1% at full scale and 3.5% at tension levels below 15% of full scale. The installation geometry of the dynamometer is critical because it measures a small cable bend angle relative to the cable axis (angle θ, see Fig. 3.6.6-2). For the highest accuracy, the load cell axis must bisect the cable bend angle over the dynamometer table (angle θ, see Fig. 3.6.6-2). The dynamometer design will include end tables to decrease cable weight sensitivity so that sagging cable will introduce minimal error.

Cable sliding over the dynamometer will experience a tension loss as a result of the sliding friction. This friction will cause a tension difference that can be expressed as follows:

$$T_o = T_i - \mu F$$

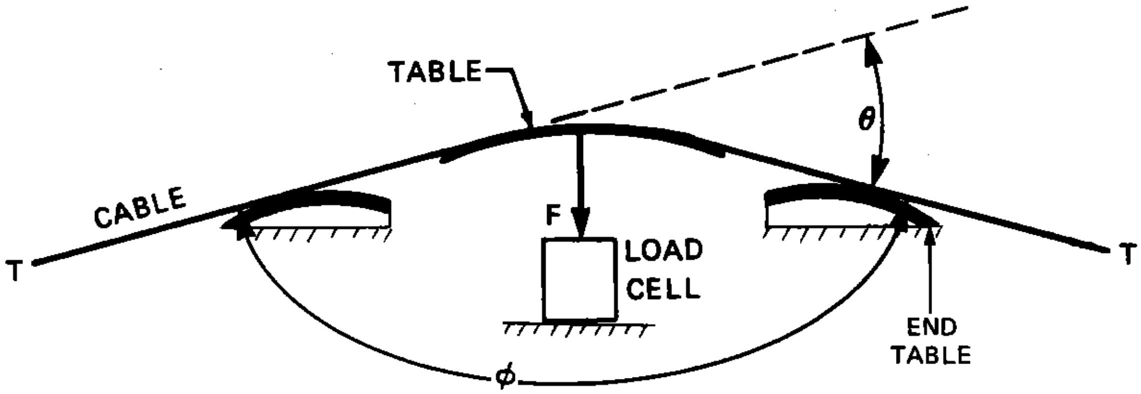
T_o = tension outboard
T_i = tension inboard
μ = coefficient of friction
F = normal force (see Figure 3.6.6-2)

Using a .15 coefficient of friction on the hard anti-wear slide coating during payout and the relationship shown in Figure 3.6.6-2:

$$T_{measured} = .5\ T_o + .5\ T_i \quad \text{or} \quad = .5(T_i - .15(.1\ T_i)) + .5\ T_i = .993\ T_i$$

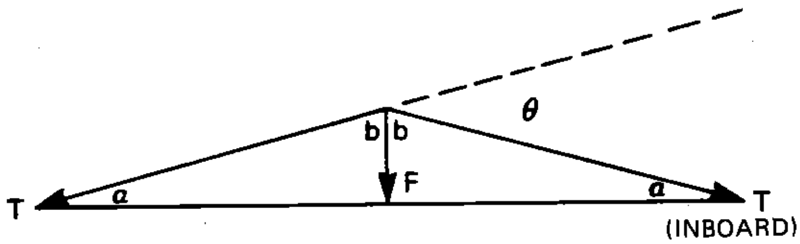
resulting in tension measured to be equal to 99 percent of the applied tension. This error is the cause of the accuracy at full scale stated earlier in this section. Other angular errors are introduced at low tension levels due to the cable's bending stiffness affecting its shape.

The overboard line tension will be different from the dynamometer tension because of the losses in the overboard sheave. Frictional losses are discussed in Section 3.2.3. In the cable payout direction, overboard tension will be higher than dynamometer tension; in the cable pickup



Typical values : $\theta = 5° - 44'$
$\phi = 174° - 16'$

Force "F" bisects angle ϕ



$$180° = \phi + \theta$$
$$b = \phi/2$$
$$\alpha = \theta/2$$

$$\sin\alpha = \frac{1/2\,F}{T}$$

$$F = 2T\sin\alpha$$

$$\boxed{F = 2T\sin\left(\frac{\theta}{2}\right)}$$

IF $\theta = 5° - 44'$

$$F = 2T\sin\frac{5° - 44'}{2} = .1T$$

FIGURE 3.6.6-2 Cable Tension Dynamometer Relationships direction, overboard tension will be lower than dynamometer tension. The tension losses of the overboarding device are in addition to the dynamometer losses.

CABLE SPEED/FOOTAGE SENSOR

This is a specially designed, wheel-driven device riding on the cable. Four elements are involved; the line rider mechanism or meter wheel, a speed sensor, a position sensor and an optical sensor.

The speed sensor could be a high accuracy D.C. tachometer or the speed and position sensors could be combined into a single rotary encoder, which would produce signals for speed, direction, and position. Though the latter method has advantages, there is an advantage to keeping the speed and position sensors independent - if a sensor were to fail, both speed and position data would not be lost at the same time. The preferred method to sense speed is to use a magnetic sensor which counts marks on the meter wheel. This device would provide 0.15% linearity from minimum to full speed with an instantaneous accuracy of 0.5%. Considering possible intermittent meter wheel slippage on the cable, overall speed sensor accuracies could be expected on the order of 1.5%.

The position sensor could be a rotary encoder which would give 0.1% accuracy; however, the meter wheel will have an accumulative slippage error. Therefore, expected accuracy would be approximately 1.5%. Cable footage count must be precise and accurate. To limit cable error build up, it is recommended that cable markers be placed on the cable which can be sensed by optical sensors allowing periodic length update. This would limit error build-up to that which can accumulate between markers. If markers were placed every 300 meters. maximum error would be about 4.5 meters. The marker outputs would be used to update the footage count continuously.

Cable tensioner track speed is not the same as cable speed. There will always be a certain degree of cable slip through the machine and a separate wheel driven sensor will be used for more accurate sensing of cable speed and length.

The cable speed/footage sensors can be mounted on either side of the tensioning machine. Cable length on the high tension side of the cable tensioner will differ from that on the low tension side due to cable stretch. It is recommended that cable speed and length be measured on the high tension side and that the overall control system compensate for the cable predictable error factor of stretch in the basic control equations.

TENSIONER & TURNTABLE SPEED SENSORS

These two sensors will follow the same accuracy and installation guidelines as the cable speed sensor. The preferred magnetic sensor, a proven off-the-shelf item, would provide a 0.15% linearity and an overall accuracy of 0.5%.

TENSION MACHINE STABILITY AUGMENTATION

Pressure feedback sensors are available with 0.25% accuracy, linearity and dynamic band pass capabilities. If a D.C. drive were to be used, an equivalent signal such as back EMF must be derived from circuitry.

PICKUP ARM CABLE POSITION SENSORS

These will be specifically designed sensor systems to measure the cable angle changes near the tip of the pickup arm. These sensors will provide data on turntable lead or lag to cable payout/pickup rate and will compensate for cable spool diameter. The sensors could be cable contacting devices using resolvers or encoder type sensors. Optical sensors are available which can sense 0.5% within a 12° field of view. Further design studies of envelope and reliability requirements must be made to determine which type of sensor is best for this application.

A dynamically positioned pickup arm might be proposed for a baseline commercial system. Once the pickup arm performance requirements (in terms of expected cable angle swing that results from the take-up lengths allowed) are known, the best type of sensor for use in this application can be determined.

3.6.7 CONTROL SYSTEM INTEGRATION

Control of the cable lay operation will be located in a centralized command center where all cable handling and lay vessel decisions will be made based on visual and instrument data. The command center on cable lay ships is usually located in a centralized elevated location which provides good visibility for overall surveillance of vessel operations. Critical machinery areas can be monitored by video cameras which might require special lighting. The two basic sub-control elements will be the lay vessel controls and the cable machinery controls.

The cable machinery controls are the central link for integration of all cable handling equipment and systems instrumentation (see Figure 3.6.1-1). Due to the complexity of the equipment, it is important that the operator maintain visual contact with turntable and tensioner operation during all cable lay operations. On site inspection with communication to the control center is also required. Because under certain conditions it will be necessary to operate one or more of the cable handling devices manually from a local control source, the controls will incorporate two basic types of consoles; the master console in the command center and a series of local consoles. The master console is the centralized control link for all machinery elements and sensors and should be located in an environmentally controlled area. The local console is a remote unit which allows a local operator to operate a piece of equipment on site and would be interlocked with the master to prohibit dual control.

DATA LOGGING SYSTEM

Because control at the centralized command center requires knowledge of all cable machinery operational parameters, a data logging system will be required. Figure 3.6.7-1 shows the types of parameters to be recorded.

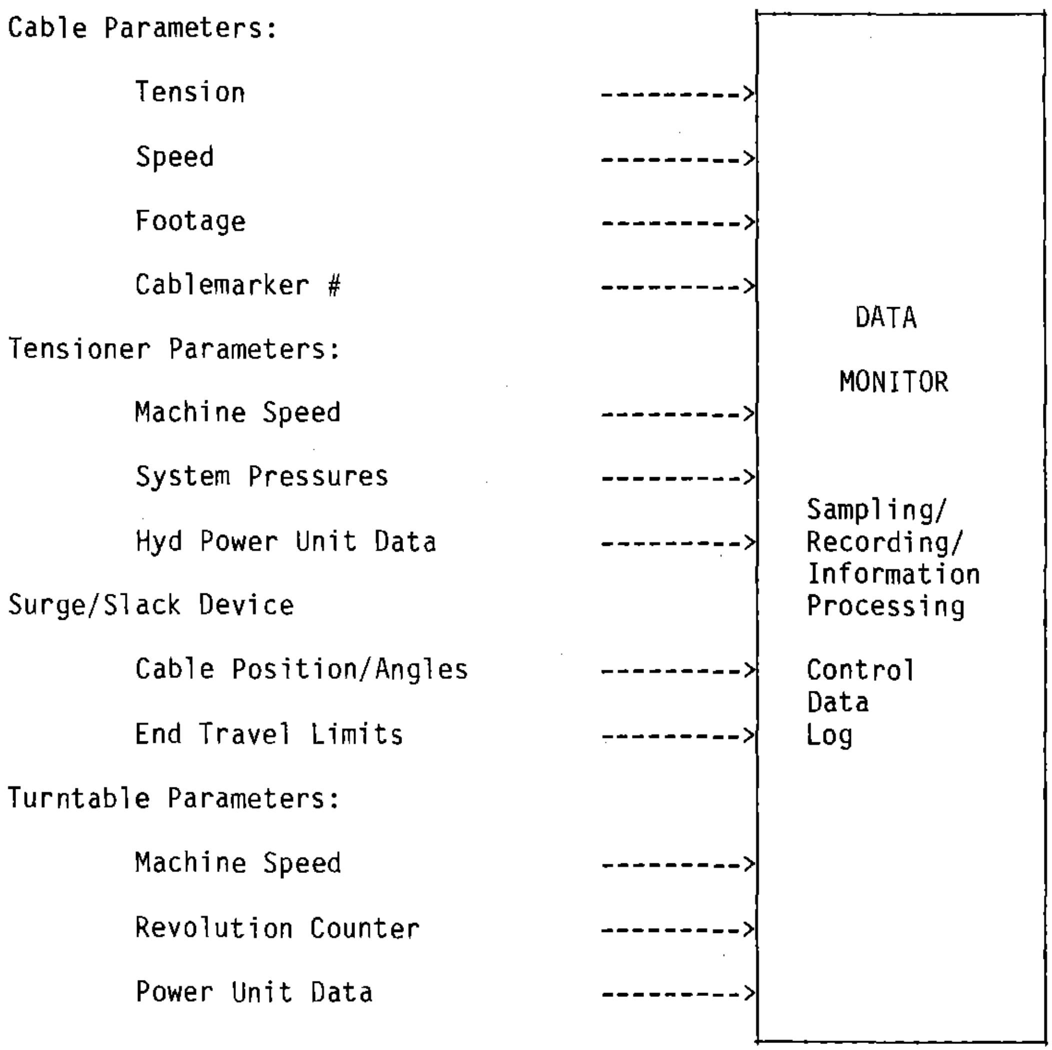


FIGURE 3.6.7-1 Data Logging System

Summarized information logging requirements.

3.6.8 CONTROL HARDWARE DESIGN

The packaging design will feature modularize components for ease of maintenance and will incorporate self-test modes to verify performance of modules. The hardware design must address environmental constraints such as temperature, humidity, vibration, and salt air contamination. The control design must be kept as simple as possible, because reliability data demonstrates that systems comprised of a minimum number of components are more reliable. Consequently the basic control will be manual with a minimum amount of electronics. The "normal" mode of control will revert to manual. All higher level controls will be additive in nature.

Three basic hardware types can be used in cable machinery controls; analog, digital, or a hybrid of both. To determine which type is suitable, system size, response, complexity, cost, and flexibility for future changes and adaptation are factors used in the evaluation process.

A hardwired analog system uses discrete electrical components. All elements are fixed and permanently wired with limited flexibility and adjustment capabilities. The system would be suitable for less complex systems, as it is cost effective and uses few parts. Larger systems require more complex control functions, and more interaction is required between the control elements.

As related to current cable machinery design, a redundant A and B hard-wired system might require an estimated 84 modularized single function printed circuit cards as follows:

Tensioner Controls	36 cards
Turntable	22 cards
Interface Cards	10 cards
Instrumentation Unit	16 cards
	84 cards

A digital microprocessor based design might be advantageous, considering the system complexity and interactions required. Since the HDWC program involves advancement of the state-of-the-art, changes can be expected as a result of system testing and operations. The ease of altering software compared to changing hardwired controls should be considered. However, microprocessor reprogramming would require an on-board computer.

A computerized system uses a limited number of hardwired components to digitize (convert to computer compatible format) all information so that software programming can be used to manipulate data that controls the machinery. Within the computerized system there are two basic types of equipment to be considered for machinery control. Basic differences are tabulated as follows:

Microprocessor

1. Allows for high power manipulation and math functions, and greater latitude in self-test programs.

2. Higher response times, currently in the range of 1 ms, are achieved.

3. Specially designed input/output interface to machinery control elements may be required.

4. The services of computer programmers to alter software may be required because each microprocessor uses machine language unique to its manufacturer.

Programmable Controller

1. The programmable controller uses higher level language, which allows field personnel to make programming changes.

2. Slower response time (currently in the range of 25 ms) is caused by the complexities of high level language.

3. Limited manipulation and math functions are due to high level language.

4. The programmable controller incorporates a computer buss, which interfaces with similar higher level computers and could simplify the controls interfaces.

5. Input/output interface modules for digital and analog signals are incorporated for machinery control elements.

6. Programmable controllers are available prepackaged in industrial packages.

A hybrid hardwired computer system would probably be the most suitable for the machinery control loops, stabilization circuits, and for simplified local control for emergency back-up. The hardwired analog system is the fastest responding system and, since most sensing and control elements are analog, the simplest machinery control is likely to be purely analog. However, overall project control is probably most easily assembled using digital hardware.

Regardless of which course is taken for the hardware design, solid state electronics will be utilized. Therefore the power source will need to be an isolated regulated source, especially if high noise output SCR power units are used anywhere on the vessel. Proper shielding, grounding, and EMI protection must be incorporated.

The digital equipment does not provide memory retention during power loss, and vital information must be retained in such instances. Therefore, battery back-up for computer memory is required. An uninterruptable power source is also required so that controls, sensors, and monitors will remain operative during vessel power loss.

REDUNDANCY

All critical path areas must have some form of redundancy or alternate methods of control to provide safe and continuous operation. A critical path area is one in which, if failure occurs, the cable laying system will cease to operate or create an unsafe condition. Not all components can be made redundant but some general guidelines follow:

a. Electronic Controls

Dual redundant A and B systems, only one providing control but both on line at all times, could be used. The switching could be automatic for some failure conditions but usually a manual command decision would be made for changing from one system to the other. This changeover would occur after a period of manual control precipitated by an alarm triggered by differences between the A & B systems.

b. Sensors

Certain elements must be duplicated, such as a dual bridge load cell in the cable tension sensor and dual channels for surge control sensors.

c. Power Units

A limited level of redundancy is normal in the design of power supplies. A complete loss of control can be avoided by allowing a level of degradation of performance and for field maintenance to restore full performance.

3.6.9 SUMMARY

The controls are driven primarily by dynamic considerations and further work must be completed in that area. Once determined, the exact nature of gain scheduling requirements will have a strong bearing on how they can be mechanized. Similarly, the pressure feedback, speed measurement, and most other data requirements cannot be defined in hardware terms until further definition is provided during machinery design. Certain areas of further study, such as sensor selection, are dependent on the finalization of equipment concepts.

The study of the machinery controls has led to the following conclusions:

1. A centralized controls system has been proposed which will be designed to permit troubleshooting and maintenance of each machine system separately.

2. The control hardware will probably be a hybrid system consisting of analog and digital devices. The analog circuitry would provide the basic machine control loops while the digital devices would provide data loops and interfaces with the overall cable lay controls.

3. The controls will be designed with adaptive characteristics that allow changes to be made by authorized personnel in the field during the program.

4. The basic controls will be integrated into the overall cable lay control system so that operation will be logical and user-friendly.

5. Levels of redundancy will be determined based on an analysis of probable failure modes and effects. The bottom level of all controls will be the operator manual assumption of direct control. Therefore, the direct manual operator control loop will be the basic line of control requiring an absolute minimum of logical input data and instrumentation.

4.0 CONCLUSIONS AND RECOMMENDATIONS

4.1 CONCLUSIONS

See Figure 1 which illustrates a possible arrangement of the equipment. All conclusions pertain to the HDWC program's at-sea test only.

- A single 12.3 meter diameter sheave is required to meet the selected candidate cable parameters. The sheave is to be provided with radiused guards to provide controlled cable entry and exit from the sheave. The sheave's dynamic effect on cable tensioning capability must be minimized during the sheave design.

- The cable tension will be measured by a single, slide type controlled bend device. The expected accuracy of this device is 1% at full scale and 3.5% at tension levels below 15% of full scale.

- The cable speed and length are to be measured by a wheel type, rider device operating on the tensioned cable. Reduction of accumulated errors by periodic length updates can be accomplished by optical sensing of cable markers. Considering possible intermittent wheel slippage, accuracy will be 1.5% between updates, with an instantaneous speed accuracy of 0.5%.

- Cable tension compensation and generation of tension levels approaching 80 MTon can be achieved without violating the PPC #116 cable paameters. A linear tensioner with 26.2 meters of active length to achieve a 78.7 metric ton tension is required. The machine's overall length will be approximately 31.2 meters. All dimensions are based on preliminary cable parameters.

Methods of compensating for the dynamic cable loading effects of vessel responses to sea conditions have been analyzed in section 2.5. The result is that the necessary tensioner responses needed to adequately sense and limit the effects of high frequency sea state disturbances can be obtained. Section 2.5.5 contains basic guidelines for the machine design.

- Cable guidance will be provided by passive metal troughing or a multiple roller arrangement. Troughing or rollers will be determined by the load support required and the allowable cable parameters.

- The surge-slack device will be a single pickup arm located over the turntable. The device will be passive during cable payout and under manual control during cable recovery or loading operations.

- Cable storage will be provided by a single turntable with an approximate diameter of 10 meters. The turntable capacity will be a nominal 9.4 kilometers (PPC 116) when the storing cable is stacked to a height of 3.23 meters.

  An alternate turntable, which is suitable for 9.4 KM (31,000 ft) of PPC #116 cable, is that on board the mothballed cable vessel SUSITNA. The turntable is 10.06 meters outside diameter and would require modification to increase its cable storage depth from 2.44 M as shown in Table 3.4.6-1.

- The requirements for an integrated control system have been considered and an overall control philosophy has been agreed upon. The cable machinery will be driven primarily by the integrated control system for long period effects, and the cable handling subsystem controls will have internal tension and rate control loops to provide control during short period disturbances. Basic guidelines for the control system design are given in Section 3.6.9.

4.2 RISK ASSESSMENT

Risks inherent in the HDWC program result from the necessity to extend the state-of-the-art of cable laying technology to meet conditions which are more severe than those imposed on previous cable lay operations.

The approaches tentatively recommended for the cable handling subsystem equipment design were selected because of their low inherent technical risk. More study and critical reviews are required in the areas listed in Section 4.3 to ensure that new risks are not introduced due to oversight.

The sensor system for the control of the on-board handling equipment, including the lead/lag sensor which determines the relative speed difference between the turntable and the cable tensioner, is an area requiring concept refinement. Sensors of this type have not been previously applied to cable machinery, but the technology and hardware does exist. This is a recognized technical challenge but not a serious risk.

The controls integration and detailed design definition, such as the mechanization of redundancies and malfunction detection circuits, are also an area requiring concept refinement, but these are essentially proven techniques, and may be applied to the specific problem areas.

The engineering risks associated with the concept refinement and design of a new cable tensioner are minimal. However, it is important for economic and dynamic response reasons to minimize the length of the machine. This will require final data regarding the properties of the cable/machine interface. Consequently, design evaluation and testing must be conducted to optimize the machine design. A thorough understanding of the cable machinery interface is a critical requirement for the successful extrapolation of present capabilities to those of the HDWC program.

4.3 RECOMMENDATIONS

- Further studies of the linear cable tensioner controls should be conducted to arrive at an optimum configuration including the need for gain scheduling. Close coordination with other team members will be required to ensure that baseline assumptions or other data used are representative of the expected configurations and conditions.

  The above study will include considerations of the cable gripper slip factors, the dynamics of the overboarding sheave, the sensitivity of the dynamometer, and other cable machinery factors which will increase the validity of the modelling techniques and analysis.

- The linear cable tensioner gripper block interface with the cable can not be defined without final cable parameter data. However, definition of certain parameters may indicate that some limited testing is required to establish the concept design. Sufficient data is needed to proceed with fundamental parametric studies.

- Develop a system model of the surge/slack control problem to permit drive power estimates for the turntable. This requires input of the cable vessel's maneuvering capability and a synopsis of vessel operational limits to test cable handling subsystem equipment response to selected equipment failures. This model will consider the maximum probable variations in cable tensioner payout and turntable payout, the slack absorption capability of the pickup arm arrangement, and other related factors including sensor accuracy. Cable parameters such as twist and flexibility are required to determine the slack absorption capability of the system.

- o Cable vessel guidance and machinery limitations must be determined so detailed cable entry requirements can be set for the overboarding sheave and guards.

- o Pick-up arm flexibility or maneuverability requirements must be studied to ensure adequate turntable coverage. This is essentially a geometry study to ensure that the cable, with its attendant weight and flexibility, can be maneuvered to remove cable twist during loading or retrieval operations.

- o The turntable bearing arrangement and type will require further study for the baseline commercial program. While existing turntable designs exceed the size and capacity required, they do not operate at the speeds projected for the commercial cable lay and consequently sizable drive power increases will be required (see Figure 3.4.4-6). Considering that the total economics of power requirements do include other supporting vessel elements, it is important to develop good representative drag loss figures for rolling element support bearings. The concept of hydrodynamic bearings, which are nearly frictionless and have been used as turntable supports, has been studied. It is recommended that future proposals and studies on the baseline commercial system consider the risks, stopping distances, and inherent advantage of low power that such bearings provide when used as turntable support systems.

- o A definition of the cable vessel response characteristics to the design sea state will be required to determine probable loading and equipment environment. These load parameters are required to begin the design definition phase of the cable handling subsystem machinery development. It is recommended that these parameters be determined immediately after the candidate vessel has been selected.

APPENDIX A - ABOUT THE AUTHORS

John M. Franchuk. Mr. Franchuk, Senior Engineer in marine systems, is experienced in the mechanical design and development of marine deck and submarine equipment including controls. Since 1976 he has been heavily involved in development of civilian and military cableships including design and development of cable and pipe handling machinery. He was project engineer for the concept, design and application of the cable machinery modernization effort on the C.S. Longlines. In addition, he was project engineer for the cable handling systems aboard the current four-vessel T-ARC cableship fleet operated by the U.S. government. He developed the concept and design for a cableship full load self-test system and its field application. He is the designer and co-patent holder of the WGMC draw off-hold back (DOHB) cable tension machine. Educational background includes U. S. Navy electronic training and a BSME from North Dakota State University. Mr. Franchuk is a licensed professional engineer in the state of Washington and has been with WGMC since 1968.

Al Smith. Mr. Smith, Engineering Group Supervisor, is responsible for special projects and marine systems at WGMC. He has been involved with the direction, design and development of high performance equipment such as flight control systems and automated machinery. His areas of expertise include dynamic and stress analysis, hydraulics, electronics and data systems. Educational background includes a BSME equivalent from Guildford Technical College, England. Mr. Smith has been with WGMC since 1976.

Wes Severe. Mr. Severe is a senior electronic/electrical engineer with extensive experience in electronic/electrical machinery control design, test and installation. Emphasis has been in marine machinery applications. He was involved in the development and design of the Deep Ocean

Mining Heavy Lift system aboard the HUGHES GLOMAR EXPLORER. Since 1976 he has had extensive experience in developing controls for both civilian and military cable laying projects. These include the T-ARC cableship fleet for the U. S. Navy. Educational background includes a BSEE from California State University at Los Angeles. Mr. Severe has been with WGMC since 1971.

Doug Skiles. Mr. Skiles, senior mechanical engineer, has worked extensively in development and design of marine deck, pipelaying and cable handling machinery. He is the designer of the self-fleeting drum conveyor modules used on the U.S. Navy T-ARC cableship fleet. Educational background includes a BSME from Washington State University. Mr. Skiles has been with WGMC since 1967.

APPENDIX B - GLOSSARY OF KEY TERMINOLOGY

At-Sea Tests - This testing will involve laying and retrieving a length of cable in the Alenuihaha Channel to obtain data to demonstrate the technical feasibility of the HDWC program.

Cable Storage Device - This element of a cable lay equipment system stores, pays out, and retrieves cable in a controlled manner.

Cable Tension Hysteresis - The difference in tension between cable payout and inhaul.

Critical Path Area - An area in which, if failure occurs, the system will cease to operate or will create an unsafe condition.

Design Sea State - A set of parameters for the worst-case environmental conditions that exist in the Alenuihaha Channel for 75 percent of the year (see Section 2.3).

DOHB Device - This is a draw off and hold back machine used with a capstan or drum type of tensioner to maintain back tension on the drum.

HDWC Program - The Hawaii Deep Water Cable program, for which this study was prepared, will demonstrate the technical feasibility of a cable lay in the Alenuihaha Channel between the islands of Maui and Hawaii.

Heave Compensation - Compensation for vessel's heave induced dynamic loads by limiting the energy input into the cable. Necessary when maximum dynamic cable tensions are above those allowed (see Section 2.3).

Overboarding Device - This element of a cable lay equipment system is used to guide cable on to and off of the cable lay vessel. It also controls bending of the cable.

SCOF - This cable design uses a Single Conductor Oil Filled construction. The cable proposed for the HDWC program will use an aluminum conductor insulated and sealed within a lead sheath and polyethylene jacket with dual armor wire layers for tension and torque resistance (see Section 2.2).

SKAGERRAK - A cable lay vessel evaluated for use in the HDWC program at-sea tests. Permission by owners to effect needed modifications is unlikely (see Section 3.1).

Surge-Slack Device - This element of a cable lay equipment system provides cable take-up and payout when the rate of the storage device does not match that of the tensioner.

SUSITNA - A cable lay vessel evaluated for use in the HDWC program at-sea tests. Not recommended because of size and equipment considerations (see Section 3.1).

Tensioner - This element of a cable lay equipment system provides cable tension by applying a tension gradient along a suitable length of cable. It also provides the tension required to pull the cable on board during retrieval operations. The tensioner recommended for the HDWC program would also provide heave compensation.