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MOST Project 2

LEVEL II

HYDROGRAPHIC WINCH SYSTEM

by

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Electronics Branch

27 Feb 67

36p.

ABSTRACT. This report contains a complete physical and operational description of a hydrographic winch system developed at NOTS.

Significant design features and operational parameters of the winch are given; the functions of the components of the system are discussed; and the operation of the system in each of five available operating modes is explained with special emphasis placed on the Constant Depth mode which is currently undergoing test and evaluation.

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2 February 1967

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FOREWORD

The hydrographic winch system was designed by the Electronics Branch of the U. S. Naval Ordnance Test Station.

This project was started in June 1964 and will be completed by June 1967.

CONTENTS

INTRODUCTION

NOTS has developed a hydrographic winch for reliable high-speed, high-power, load handling from shipboard.

The winch has the following significant design features:

1. The cable is controlled at all points in the system to avoid both slacking and overtensioning.

2. The winch is hydraulically powered with proportional torque control through variable displacement hydraulic motors operating from a constant-pressure source.

3. The hydraulic energy is controlled with high-bandwidth, closed loop components. Care was taken to keep the system as linear and controllable as possible.

4. The winch is designed to compensate for ship's motion by paying cable in and out as the ship falls and rises. This maintains the load at a nearly constant depth.

5. The system can store significant amounts of energy at points where high peak power may be demanded. This reduces transient power loads on the ship's electrical generators, and increases system response.

6. Energy is dissipated directly by cooling heated oil at low pressure.

7. Complete control of the winch is carried out from a remote console.

GENERAL DESCRIPTION

The NOTS hydrographic winch system (Fig. 1) consists of five units: the traction unit, cable accumulator, storage drum, hydraulic power supply, and the control console.

1. The traction unit (Fig. 2) basically consists of a pair of capstans grooved to take five wraps of cable. It is driven through a gear reduction box by a 150 HP hydraulic motor with proportional torque control. This unit is the "working" unit in the system, developing the torque required to hold and accelerate the load.

Maximum line tension is 8,000 lbs; maximum line rate is 25 ft/sec (15 knots); maximum acceleration (no load) is 250 ft/sec^2; and minimum load position increment is less than 1/2 inch at full load.

2. The cable accumulator (Fig. 3) is a passive cable storage device located between the traction unit and the storage drum.

To achieve maximum load control, the traction unit should be able to accelerate as rapidly as possible. The inertia of the rotating components of the traction unit can be kept low enough so that, with reasonable motor torque, high acceleration can be achieved. The storage drum, however, has an inherent inertia at least ten times that of the traction unit. In order for the drum to accelerate with the capstan, a proportionately greater torque would be required. The amount needed cannot reasonably be provided within the range of the design parameters of this winch, consequently paying cable in and out would result in slacking and overtensioning the cable between the traction unit and the drum. It is therefore necessary to either decrease the acceleration of the traction unit to whatever the drum can manage, or to put a buffer between the units. The cable accumulator serves as the buffer.

The cable accumulator is conceptually simple. It takes in or pays out cable to the traction unit or the storage drum at a nearly constant tension. The rate and amount of cable handled from either side are controlled independently.

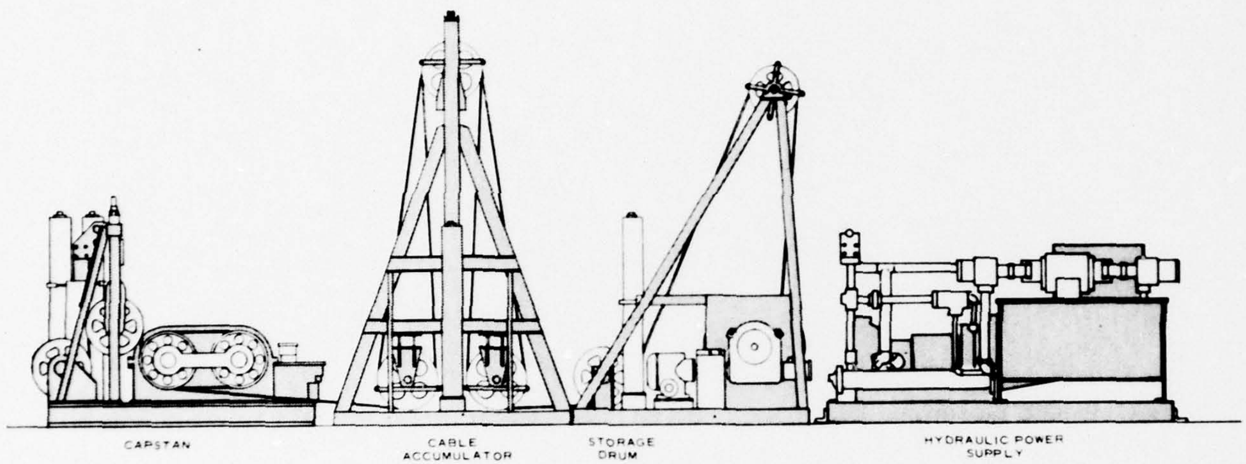


FIG. 1. Hydrographic Winch System.

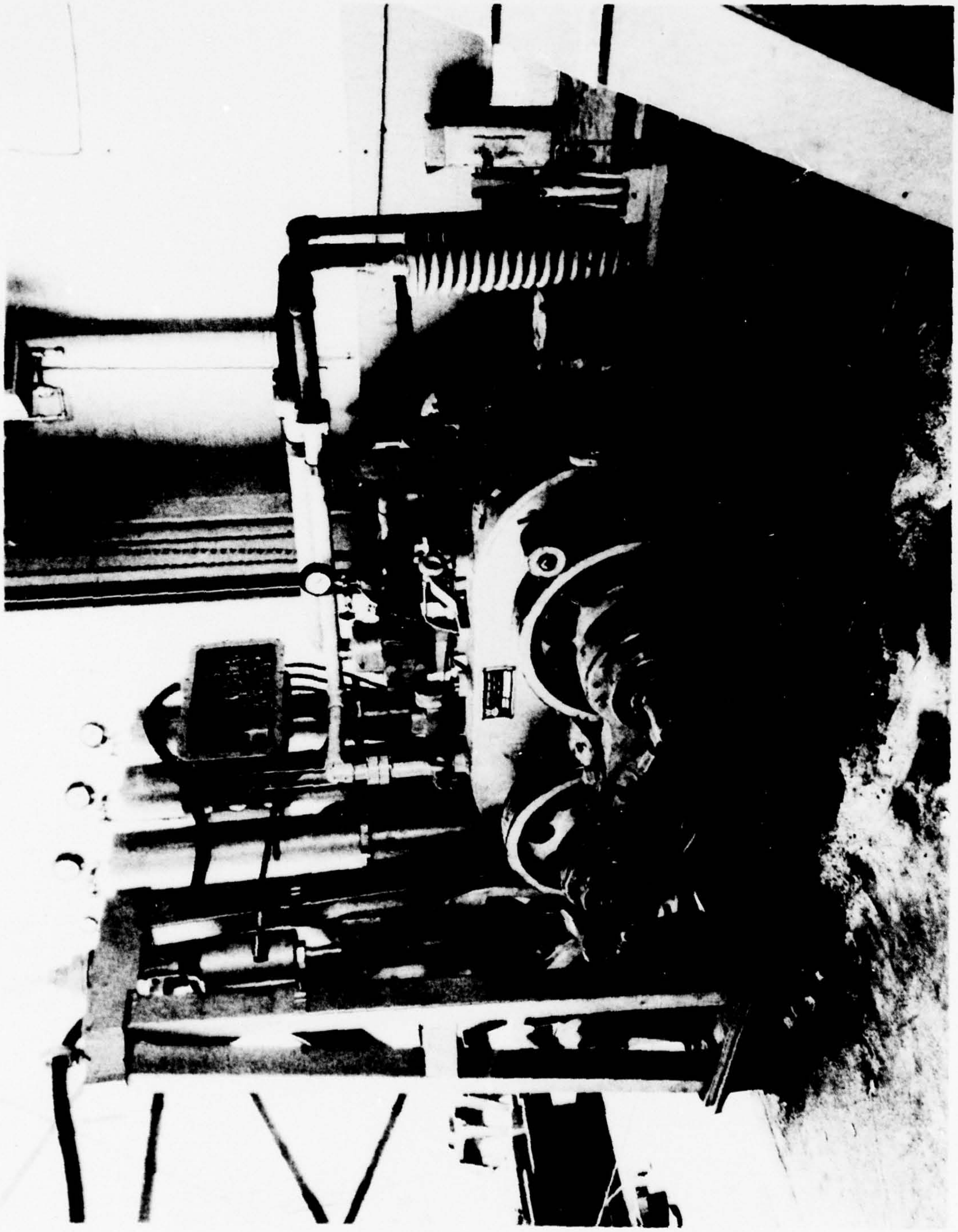


FIG. 2. Traction Unit.

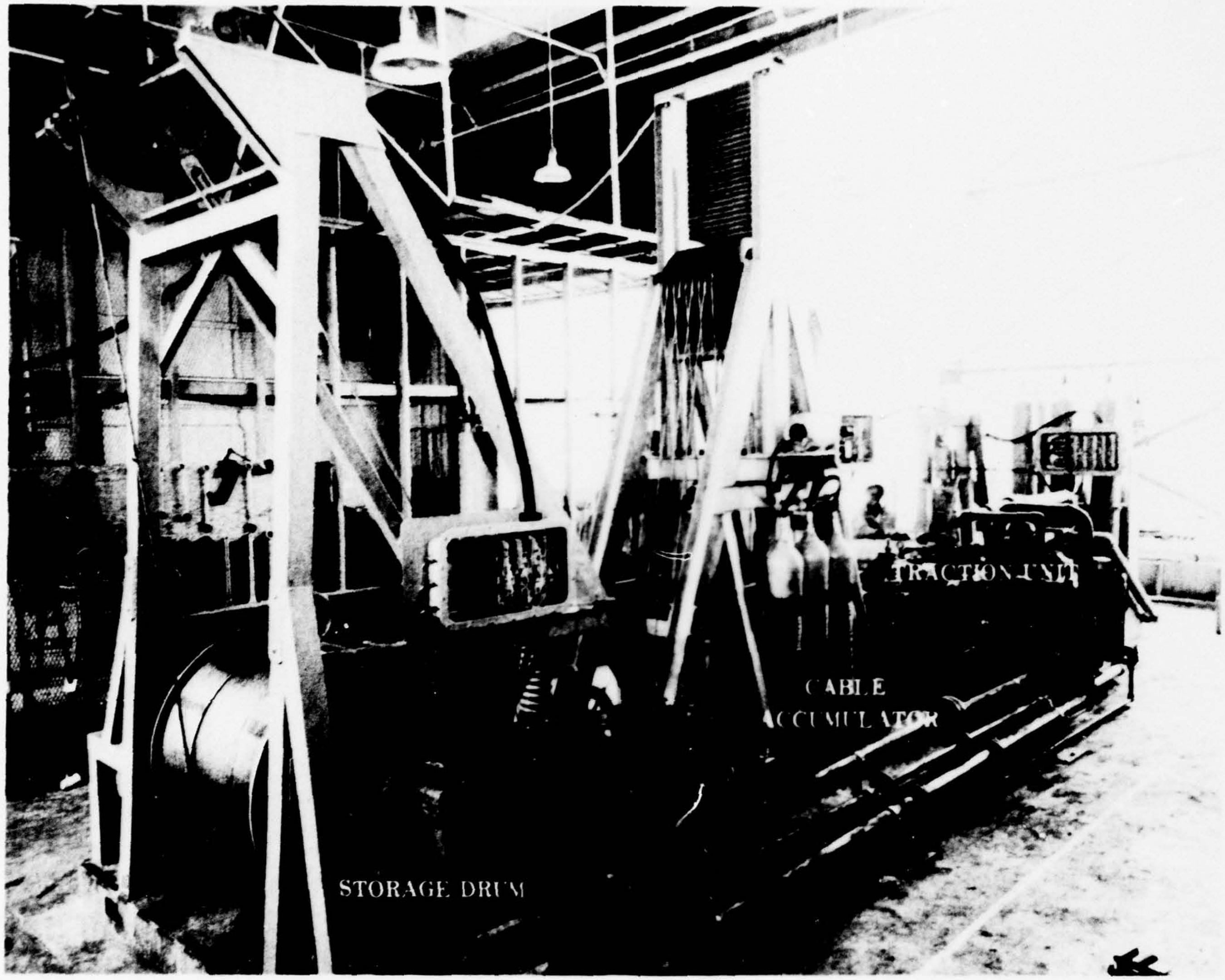


FIG. 3. Storage Drum, Cable Accumulator, and Traction Unit.

During operation, as the traction unit accelerates to pay out cable, cable initially comes from the accumulator. As the accumulator is emptied, the storage drum begins feeding cable in. The drum eventually reaches the speed or position required to hold the accumulator half-full. For hauling in cable, the operation is analogous. Thus, with the cable accumulator to act as a buffer, the winch system can include a large, heavy drum load of cable, while maintaining a high acceleration capability at the traction unit. (A more detailed description of the cable accumulator is given in Appendix B.)

3. The storage drum Fig. 3, consists of a flanged drum which can store 10,000 feet of 1/2-inch cable and is driven through a gear reduction box by a 50 HP hydraulic motor with proportional torque control. A Lebus spooling system and a fleet-angle compensator are used on the storage drum.

4. The hydraulic power supply is the source of all oil used by the traction unit and the storage drum. It contains the oil reservoir, pumps, and heat exchanger required for this system. (A complete description and hydraulic schematic are given in Appendix A).

5. The control console (Fig. 4 & 5) provides complete remote control and monitoring of the winch system. The operator can select any of five available methods of control through the mode selector switch on the console. In addition, the monitor panel gives all pressures, torques, rates, and line tension, as well as load depth and attitude.

OPERATIONAL DESCRIPTION

The system is best described by considering the operation in each of the operating modes. The START, MANUAL, HANDLING, RAISE/LOWER, and CONSTANT DEPTH modes are available to the operator through the mode selector switch on the console.

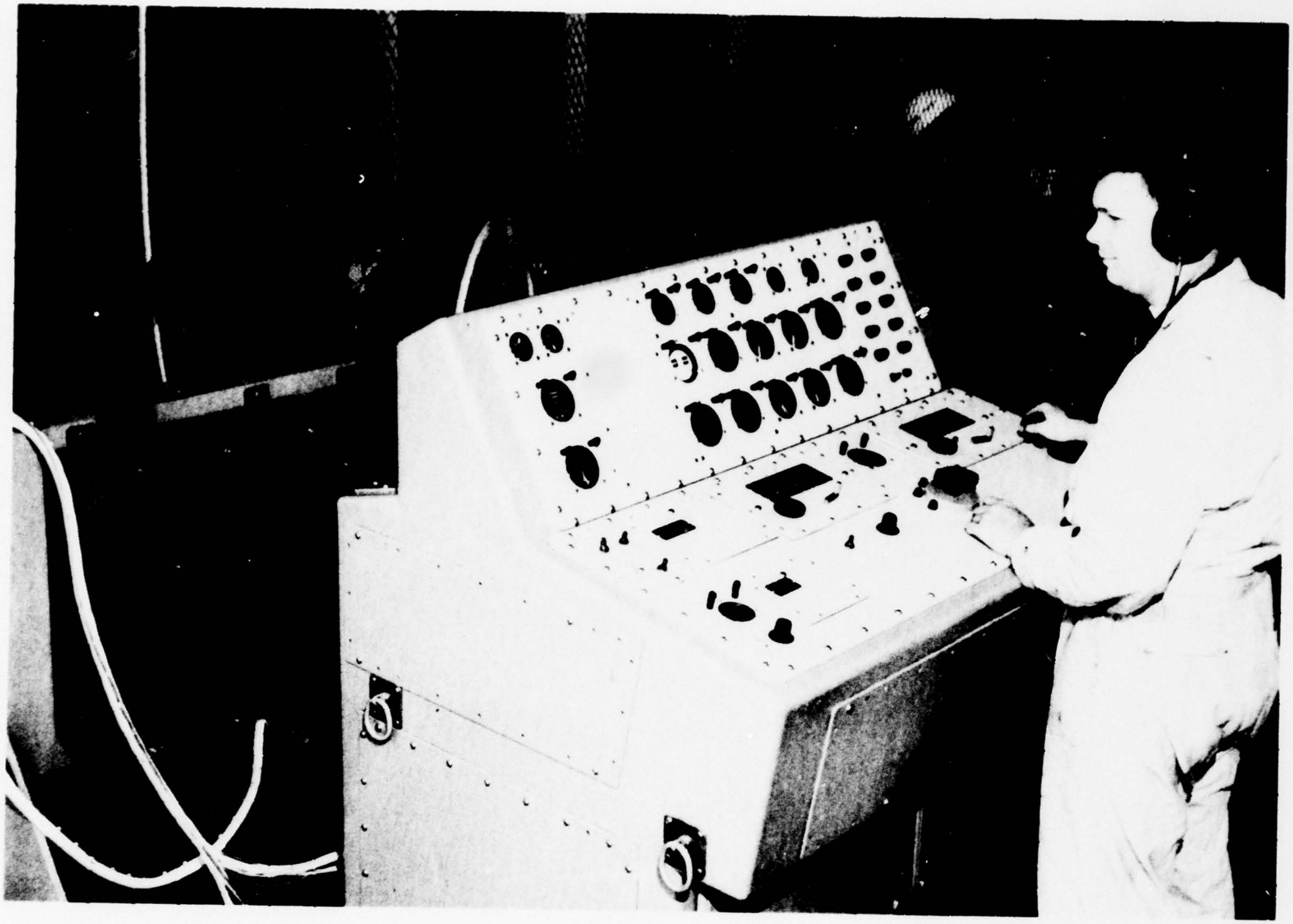


FIG. 4. Control Console.

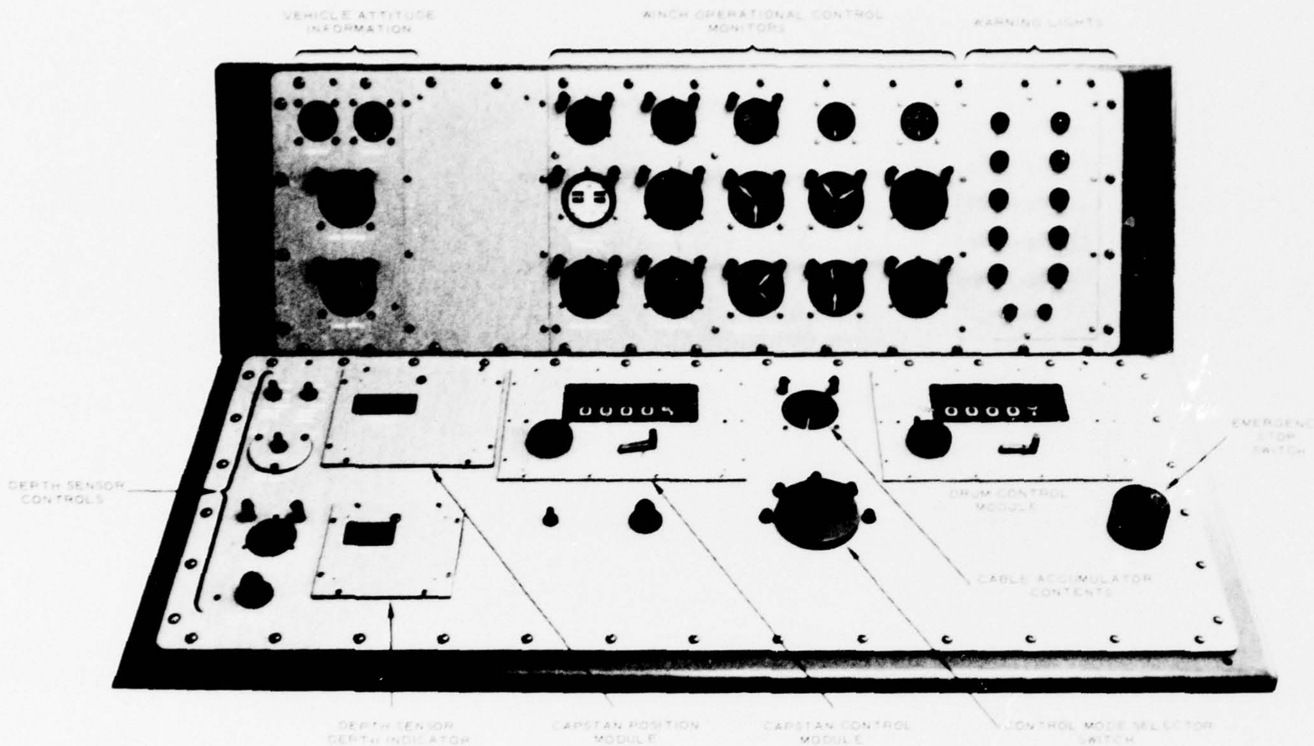


FIG. 5. Top View of Control Console.

START MODE

In the START mode, the system is held stationary. The brakes on the traction unit and storage drum are applied, and the control system orders the units to hold their position. The electrical and hydraulic power are turned on, and the system is automatically checked out for operational readiness. If, during any other mode, a system malfunction occurs, or the operator pushes the EMERGENCY STOP switch, control is automatically switched back to the START mode. When pressures and voltages are correct, the operator may switch to the MANUAL mode.

MANUAL MODE

In the MANUAL mode, the operator can order the traction unit or the storage drum, independently, to pay any desired amount of cable in or out through handwheel inputs on the console. This mode is generally used to set the contents of the cable accumulator to half full. It is the only mode in which the operator has direct control of the storage drum. From the MANUAL mode, control is switched to the HANDLING mode.

HANDLING MODE

The HANDLING mode is used when the load is to be handled carefully, over distances of about 100 feet or less, and at low speed, as when the load is being handled on deck, or being deployed or retrieved near the ship. The operator has control of the traction unit, and can order cable payed in or out, at a maximum rate of about 4 ft/sec. The storage drum is automatically controlled by the cable accumulator, and goes to whatever speed and position are required to keep the cable accumulator half full. The operator has no control over the storage drum in this or the following modes. For high speed, long distance moving of the load, control is switched to the RAISE/LOWER mode.

RAISE/LOWER MODE

The RAISE/LOWER mode is set up for long distance, high speed raising and lowering of the load. The storage drum is still under control of the cable accumulator and requires no attention from the operator. The traction unit control is similar to that in earlier modes, with these differences:

(a) The operator may change the position order rapidly with a slew switch, or slowly with the handwheel as in the MANUAL and HANDLING modes.

(b) When the traction unit receives a large position order, the operator can adjust the maximum line rate up to 25 ft/sec.

(c) The line tension is measured at the traction unit and fed back through a threshold device to the traction unit rate control. If the line tension drops below a set level, the traction unit will be slowed, or even reversed, if necessary, to maintain line tension. If, in spite of this control, the line tension drops to zero (slack line), the control is automatically switched back to the START mode. The brakes are set, and the system is ordered to remain at rest. This safety feature also means the operator cannot switch control to the RAISE/LOWER mode from the HANDLING mode unless there is tension in the cable.

CONSTANT DEPTH MODE

In the course of taking data from instrument packages suspended in the ocean, it is often desirable to cancel out the motion of the ship on the surface so that the load remains relatively still. A fifth operational mode, the CONSTANT DEPTH mode, is currently being designed and tested for the NOTS hydrographic winch to enable it to perform this function.

Control of the winch in this mode is different from the previous control schemes, but is conceptually quite straightforward: In order to hold the load at a constant depth, the winch must pay out cable as the ship rises on a wave, then take cable in as the ship falls, with the amount and rate of cable motion just compensating for ship's motion.

The Constant Depth operation (Fig. 6) is as follows: The traction unit is under control of the depth sensor through the control console. The operator can make no direct inputs to the system. The depth sensor is attached firmly to the winch cable at a depth of approximately 300 feet. This depth is great enough that surface waves cannot cause a significant error in the depth sensor output, yet shallow enough that the mechanical delay down the winch cable will not disrupt system stability. The depth transducer is capable of resolving a change in its depth to within less than one inch in 300 feet.

As the ship rises on a wave, the depth sensor will be pulled up by the cable. This produces an error signal which causes the traction unit to pay out cable in order to return the sensor to its nominal depth. As the ship comes down, the sensor will drop, causing the traction unit to take in cable, again returning the sensor to its nominal depth. Holding the depth sensor at a constant depth relative to the mean ocean surface will maintain the load at a relatively constant depth. The system is designed to operate under conditions up to at least sea state 6 (wave heights to 30 feet) with sensor motion attenuated from the ship's motion by a factor greater than 100.

During the CONSTANT DEPTH mode, the storage drum is controlled by the cable accumulator, but in a different manner from the HANDLING and RAISE/LOWER modes. The storage drum is not ordered to turn unless the cable accumulator is nearly full or nearly empty. This means that as the traction unit pays cable in and out, compensating for ship's motion, cable is taken from and stored in the accumulator and the drum remains still.

It is interesting to note that, with the exception of the depth sensor and some extra electronics in the feedback system, neither special equipment nor modifications to the basic winch were required for the Constant Depth mode of operation.

The flexibility offered by the various modes of control enables the operator to handle the load under all conditions with precision and safety.

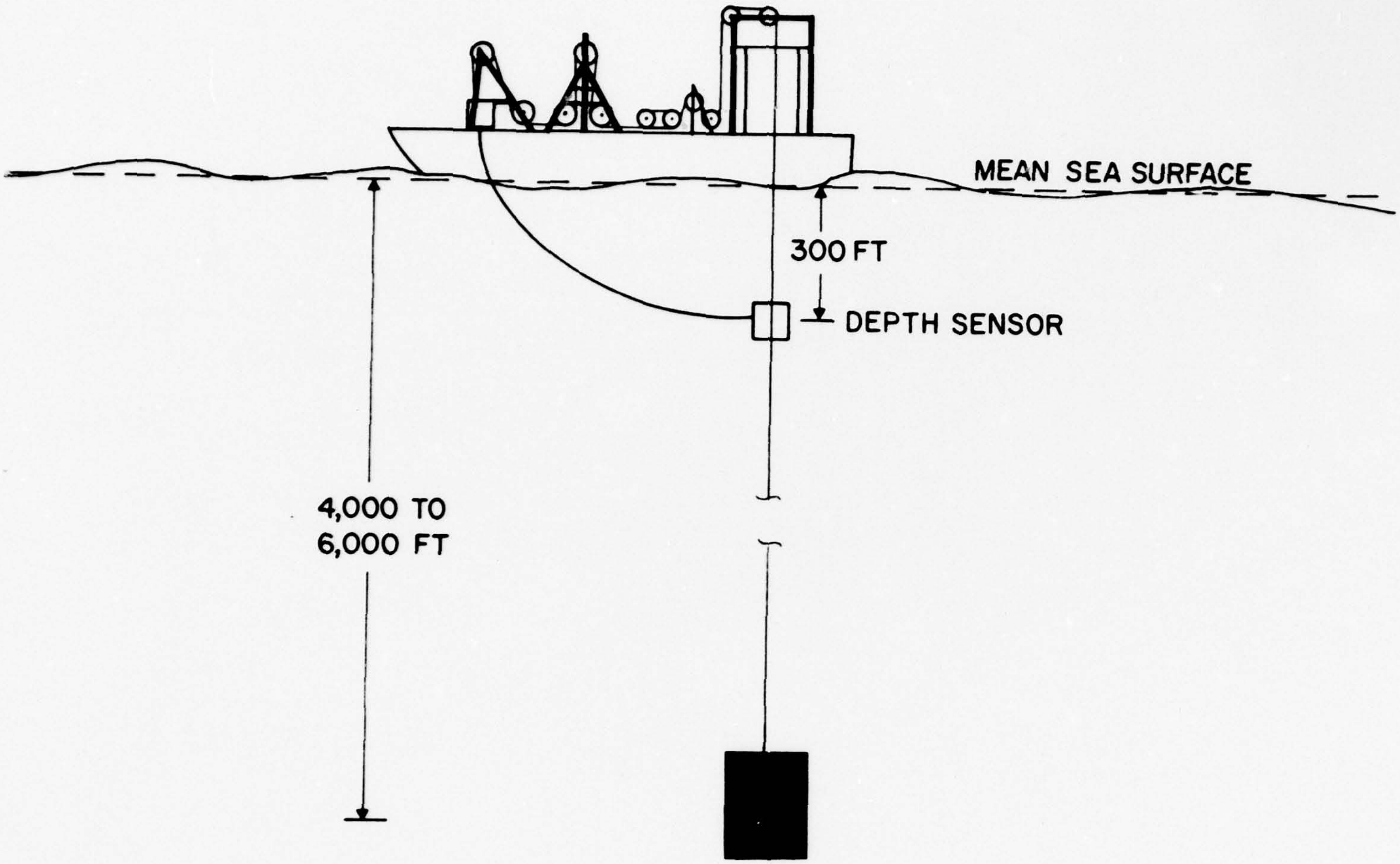


FIG. 6. Constant Depth Mode.

CONCLUSIONS AND COMMENTS

Operation of the NOTS hydrographic winch has demonstrated that a generally useful, high-speed, high-power winch can be built which is capable of:

1. high line rates and acceleration, provided by the large drive and inherently low inertia of the traction unit;

2. complete control of the cable in the system, provided by the cable accumulator acting as a buffer between the traction unit and storage drum, and cable tension feedback into the control system from the load side;

3. reduced transient power loads on the ship's electrical system, through use of the energy storage capability of the hydraulic system.

In addition, the winch can be equipped with controls to automatically compensate for ship's motion with cable motion, so that the suspended load will remain nearly stationary.

It is hoped that some of the design features of this winch will be of use to others. With that in mind, the following general comments are included.

1. The high power, high bandwidth, linear control of energy necessary in this type of system can be achieved readily with commercial hydraulic components.

2. The use of variable displacement hydraulic motors operating from a pressure source makes energy storage and dissipation easy, and offers the most flexible, controllable system for this application.

3. The position-order control system, with maximum rate control, gives the operator the closest control of the load under all conditions.

4. For maximum flexibility, accuracy, and safety, any winch which contains both a traction unit and a storage drum should also include a cable accumulator between the two units.

Appendix A

HYDRAULIC POWER SYSTEM

A hydraulic schematic of the NOTS hydrographic winch is shown in Fig. A1. Referring to the numbered components in the schematic, its operation is as follows:

HYDRAULIC POWER SUPPLY

First, consider the units included in the hydraulic power supply in the center of the picture:

The constant pressure pump, 14, set to deliver 2,100 psi, is driven by a 150 HP electric motor. This pump supplies the main driving energy to the system through the check-valve, 13, which protects the drive pump from high pressure transients in the piping system and keeps accumulator, 9, charged to peak pressure. The pressure reflief valve, 10, is to limit output line pressure in case of a malfunction in the drive pump's pressure regulator; it normally carries no flow. The drive pump is supercharged by a fixed displacement pump, 12, at a pressure of 90 psi, set by the relief valve, 16. The supercharge pump draws oil from the reservoir, 17, through a filter, 15.

All oil flowing through pressure relief valves in the system is returned to the reservoir through the water-cooled heat exchanger, 11. The heat exchanger can dissipate 250 HP at 120 degrees oil inlet temperature, with 60 degree cooling water. The final unit on the power supply is the small fixed displacement pump, 18, which supplies oil to the servo valves at a well-regulated pressure of 2,000 psi, set by the relief valve, 19.

The traction unit and drum are thus supplied with oil from three constant pressure sources: the main drive supply at 2,100 psi; the supercharge supply at 90 psi; and the servo valve supply at 2,000 psi.

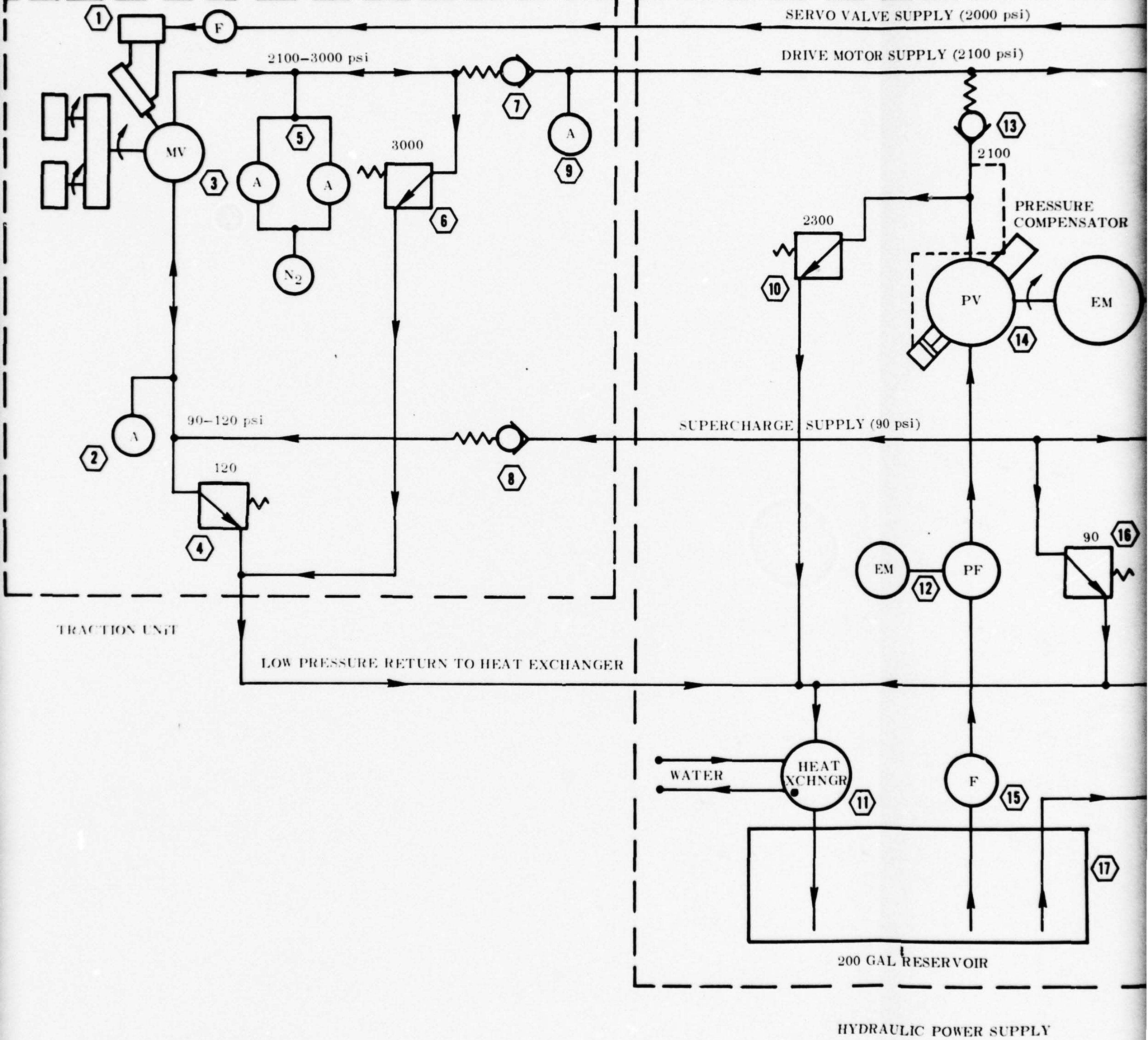
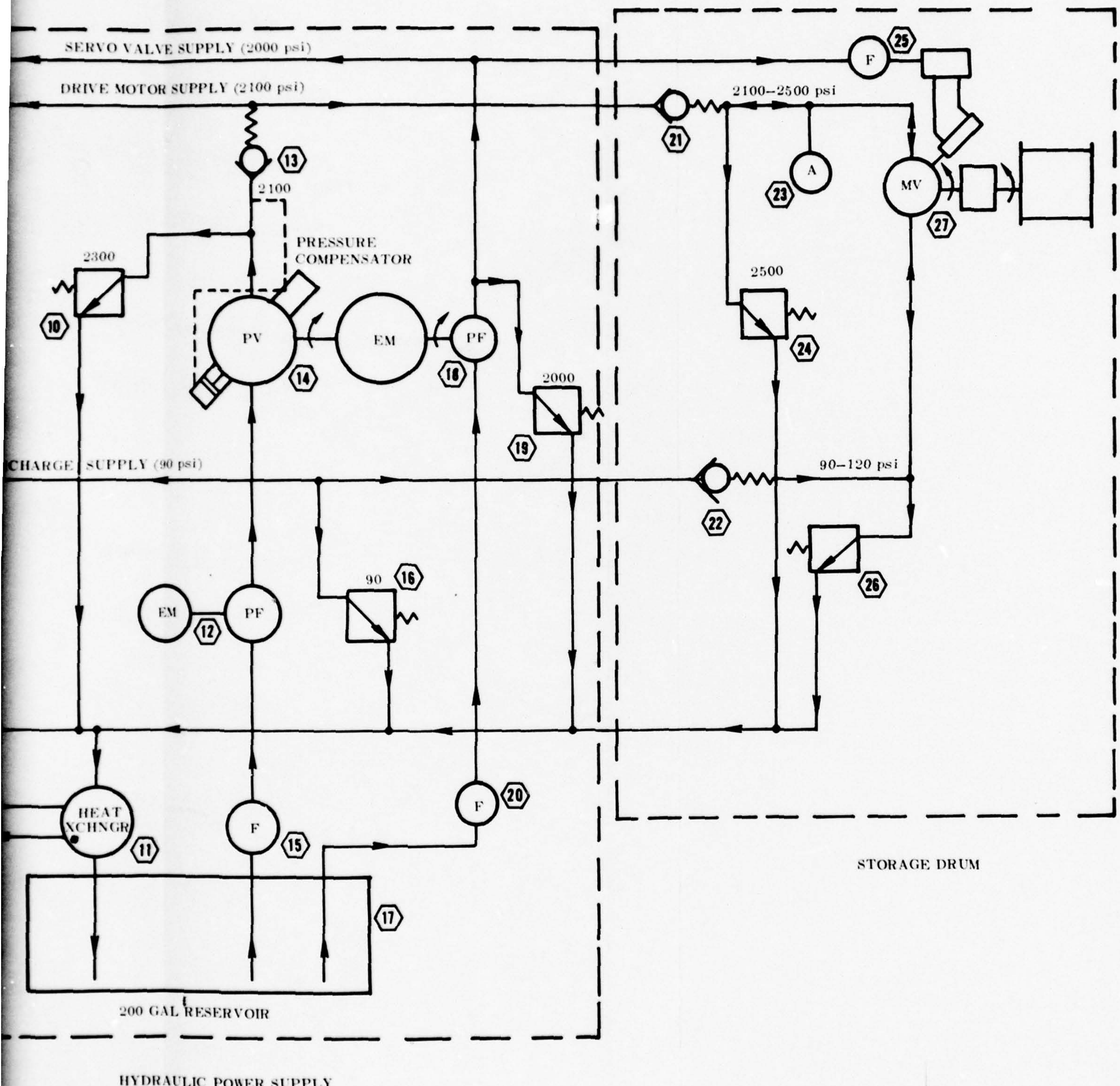


FIG. A1 Hydraulic Schematic



IG. A1 Hydraulic Schematic

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All oil returned from these units to the reservoir passes through the heat exchanger at essentially atmospheric pressure.

TRACTION UNIT

The hydraulic-operation of the traction unit can now be described in detail.

The traction wheels are run by the variable displacement hydraulic motor, 3. The output torque of the motor is proportional to the product of its displacement and the pressure drop across the motor. The flow through the motor is determined by the product of the motor displacement and its shaft speed. The motor displacement is set through the servo valve, 1, and is determined by the feedback control system.

A normal operational sequence proceeds as follows: As the load is lowered, the motor develops torque to match the weight of the load, minus its drag. Flow through the motor is from the low side to the high side (it acts as a pump). The oil is supplied to the motor on the low side through the check-valve, 8, from the supercharging pump. Oil being pumped out of the high side of the motor cannot flow back through the check-valve, 7, into the supply, so flow is initially into the accumulators, 5. The pressure on the high side will increase to 3,000 psi when pressure-relief valve, 6, will open and route the rest of the pumped oil back through the heat exchanger to the reservoir. This is the flow path through which the potential energy of the load is dissipated as the load is lowered. When the load reaches the desired depth, the traction unit stops. The system condition is, then: accumulators, 5, charged to 3,000 psi; accumulator, 9, charged to 2,100 psi; check-valve, 7, reverse biased (no flow); accumulator, 2, charged to 90 psi.

To lift the load, flow through the motor is from the high side to the low side, with the initial high-pressure oil supplied from the accumulators, 5. On the low side, the oil cannot flow back through the check-valve, 8, so the pressure rises to 120 psi when relief valve, 4, opens and returns the oil to the reservoir through the heat exchanger.

In the CONSTANT DEPTH mode, as the traction unit pays short amounts of cable in and out to compensate for ship's motion, the traction unit draws very little energy from the power supply. Energy to haul in cable is drawn from the accumulators, 5, which can store enough energy to haul in 200 feet of cable. The energy recovered from the system when cable is payed out is put back into the accumulators. Due to real losses in the machinery such as friction, leaks, and drag, the pressure in the accumulators, 5, will eventually try to drop below 2,100 psi. At this point, check-valve, 7, opens and a charge of oil is sent into accumulators, 5, from accumulator, 9, and the supply pump. Except for these short recharging times, the supply pump, 14, delivers no oil to the system; therefore, it requires very little power.

STORAGE DRUM

The hydraulic operation of the storage drum is analagous to that of the traction unit. The only difference is that the accumulator, 23, is for surge protection, and is not large enough to store a significant amount of energy since the drum requires no large peak energy inputs.

Appendix B

CABLE ACCUMULATOR

The cable accumulator in the NOTS winch is a device which takes in or pays out cable, from either side, at a nearly constant tension. The unit is a generalization of the model shown in Fig. B1.

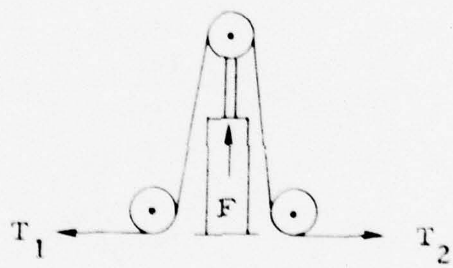


FIG. B1. Simple Cable Accumulator Model.

In this model, the two sheaves at the bottom are fixed and the top sheave can move up and down. The top sheave is supported on a hydraulic piston, which exerts an upward force, F. If the weight of the top sheave and its support is W, then the cable tension is

$$T = \frac{(F - W)}{2}$$

when the system is at rest. As the top sheave moves up one foot, it will pull in two feet of cable, from either or both sides. As the sheave comes down one foot, it lets two feet of cable return. Thus, if the tension in the cable increases above T, cable is taken from the accumulator; if the tension drops below T, the accumulator will take cable in.

The accumulator actually used in the winch (Fig. 3) is simply a row of 9 of the simple 3-sheave models placed side-by-side. The top block of sheaves has a total travel of 5 feet, so the dynamic cable capacity of the unit is 90 feet.

It is important to note that no accumulator of this type can maintain a constant tension in the cable, due to the mass and the inertia of the movable components. If the top sheave and support have mass M and sheave moment of inertia I, and the cable has acceleration A, then the dynamic cable tension can be found from the following equations:

$$(F - W) - (T_1 + T_2) = \frac{MA}{2}$$

or

$$T_1 + T_2 = (F - W) - \frac{MA}{2}$$

due to the vertical acceleration, and

$$R\,(T_1 - T_2) = \frac{2IA}{4\pi R^2}$$

or

$$T_1 - T_2 = \frac{2IA}{4\pi R^3}$$

where R is the sheave radius, to accelerate the sheave in rotation.

In operating units, operating at high acceleration, these tension error terms can amount to hundreds of pounds. An additional tension error will exist as a function of top sheave position, if the pressure in the hydraulic cylinder is not kept constant. In the NOTS unit, this pressure is held quite accurately by the scheme shown in Fig. B2.

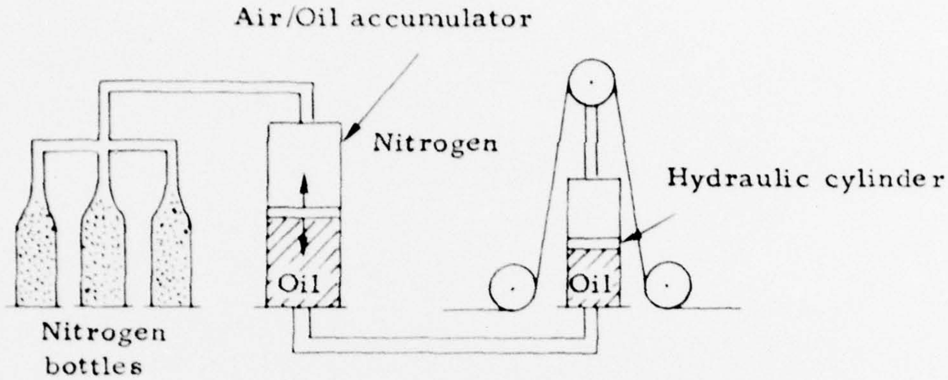


FIG. B2. Pressure Source for Accumulator

The oil pressure in the cylinder is balanced by the nitrogen pressure in the accumulator and in the storage bottles. As the piston travels from top to bottom in the cylinder, the amount of oil (or nitrogen) displaced is small compared to the total volume under pressure, hence the pressure change is small.

These considerations are included to show that a cable accumulator of this sort is inherently incapable of operating accurately as a "constant tension" device.

Appendix C

WINCH CONTROL

TRACTION UNIT

The block diagram of the traction unit control system is shown in Fig. C1. The control is based on a position input with rate and position feedback. With the prime mover being a very linear torque source and the losses in the rotating components being small, the loop is a second order system that has excellent controllability.

Overall Loop

In operation, the actual position of the unit (number of feet of cable payed out) is subtracted from the operator's position order input to give the position error, ρ. The position error is limited at a value just large enough to give full motor torque at zero speed. It is this error limit which enables the operator to control the maximum speed of the cable. The limited position error, ρ', is then summed with the rate feedback term (capstan rpm) to produce the actuating error term, ϵ.

The error, ϵ, causes the valve driver amplifier to excite the servo valve which ports oil into the stroke cylinder, thus causing it to vary the motor stroke in the sense of, and at a rate proportional to, ϵ. The position feedback from the stroke cylinder is subtracted from ϵ in the valve driver amplifier until the two just balance; the servo valve stops oil flow; and the stroke cylinder comes to rest at its new position. This minor loop thus adjusts the motor torque in proportion to the actuating error, ϵ. Depending on the characteristics of the servo valve, this loop can be essentially second order. In order to get the best response from the entire control loop, it is almost essential that this minor stroke loop be very fast relative to the response required of the overall loop.

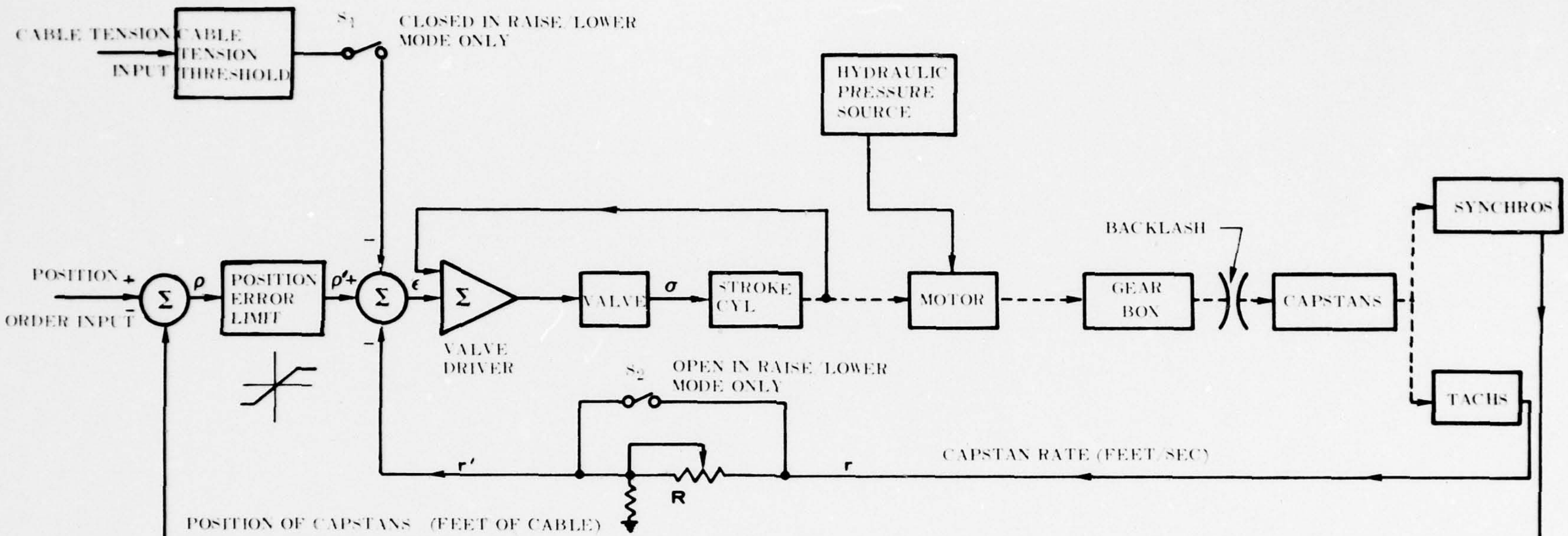


FIG. C1. Traction Unit Control Block Diagram.

The motor, now a nearly linear torque source, turns the capstans through the gearbox, thus moving the cable. The rotation and the speed of rotation of the capstans are sensed by the synchros and tachometer, and are fed back to close the main loop. (To avoid needless stability problems in this loop, it is well to keep all mechanical backlash to a minimum.)

Rate Control

The maximum rate at which the traction unit will pay cable in or out is determined by the position error limit and the rate feedback gain. A large position order input will saturate the error limit, producing ρ' volts. With the capstans not turning (and no cable tension feedback) the actuating error, ϵ, is just equal to ρ'. As the motor accelerates in response to this input, the rate feedback term, r', increases and $\epsilon = \rho' - r'$. The speed will continue to increase until ϵ decreases to a value just large enough to sustain the load velocity, with no acceleration; the speed is stable at this point.

In the MANUAL and HANDLING modes, the maximum capstan rate is set at about 4 ft/sec and is not variable. In the RAISE/LOWER mode, however, switch, S_2, which shorts out attenuator R, is opened and the operator can set the maximum rate. The operation of this control is straightforward. With S_2 open, the variable attenuator makes $r' = kr$, where k ranges from 1 to 1/6. The load speed at which ϵ goes to zero is then up to 6 times the nominal value, or about 25 ft/sec. This maximum speed is, of course, a function of the weight and/or the drag of the load. Heavier loads will sink faster and rise slower, draggier loads will sink and rise slower.

Cable Tension Control

Operating at high speed in the RAISE/LOWER mode, it is quite possible that the line may go slack (if, for example, the downward velocity of the ship, added to the cable pay-out speed, exceeded the sink-rate of the load). To prevent this, the cable tension is monitored and fed into the control loop through S_1. There is no feedback until the tension goes below about 750 lb. The feedback then increases and is added into the term for ϵ in the sense to slow the capstans. As the tension decreases further, the capstans will reverse. If, in spite of this, the cable should go slack, the system is returned to the START mode.

STORAGE DRUM

The block diagram of the storage drum control system is shown in Fig. C2. Its operation is very similar to that of the traction unit.

Overall Loop

In the MANUAL mode, switch S_1 is closed, and the loop operates under position control exactly like the traction unit. There is no provision for rate control of the drum by the operator, since it is never used in a high-speed, position controlled operation.

Cable Accumulator Control

In the HANDLING, MANUAL, and RAISE/LOWER modes, switch S_1 is open, S_2 is closed, S_3 is in position A, and the storage drum is completely controlled by the cable accumulator. The cable accumulator contents error signal is zero when the accumulator is half full. When the accumulator is either side of half full, the error signal is summed into the term for the actuating error, ϵ, in the sense and amplitude to cause the drum to return the accumulator to half full.

In the CONSTANT DEPTH mode, switch S_3 is in position B, and the drum receives an actuating error signal from the accumulator only when the accumulator is nearly full or nearly empty.

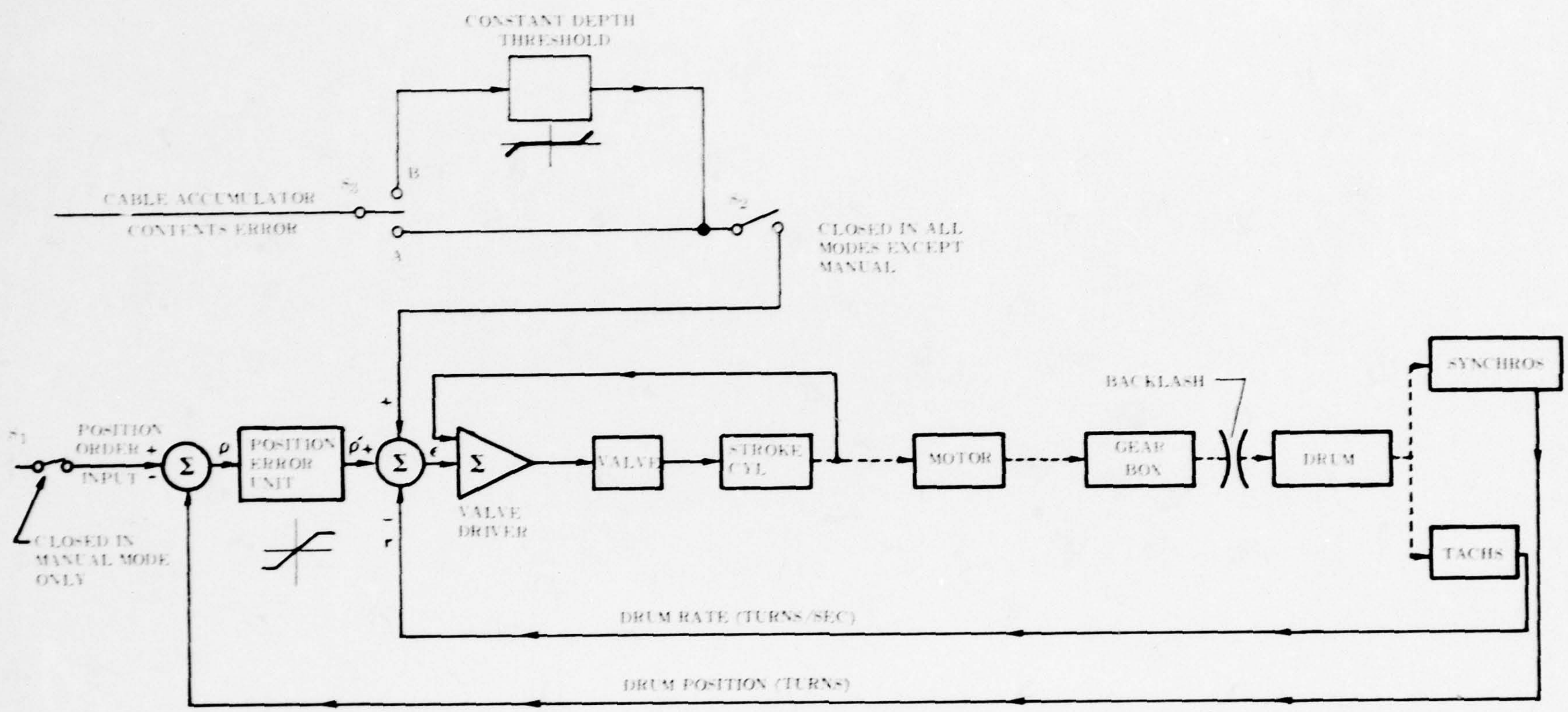


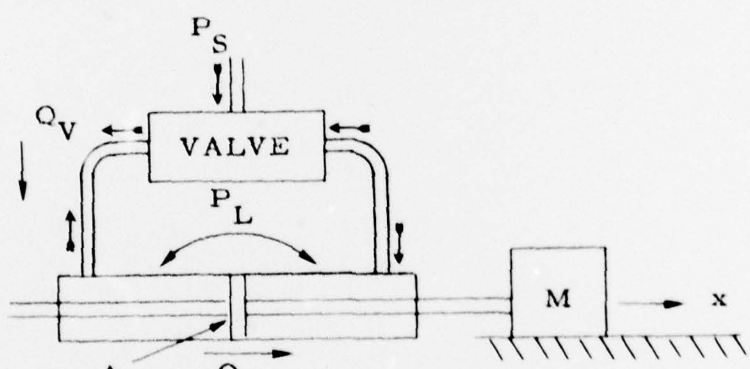
FIG. C2. Storage Drum Control Block Diagram.

Appendix D

NON-LINEARITIES IN THE RESPONSE OF A SERVO VALVE - CYLINDER MECHANISM

In designing a hydraulic system in which oil under pressure is directed in a controlled manner, by a servo valve for instance, into a hydraulic cylinder to move a load, the dynamic loading effects of accelerating the load must be considered.

The mechanism which controls the stroke of the hydraulic motors on the NOTS winch is of this type. It was desired that this stroke control have maximum frequency response, consistent with good controllability. The dynamic response which may be expected from a system of this type is given below.



The values used in the calculation of this response are defined as follows:

P_S = supply pressure

P_L = pressure drop across the load

Q_V = flow through valve

Q_L = flow through cylinder

M = effective mass of load

A = piston area

The following equations describe this system:

$$Q_V = Q_O \left[\frac{P_S - P_L}{P_S} \right]^{1/2} \tag{1}$$

gives the valve flow as a function of the load pressure, P_L, the supply pressure, P_S, and the unloaded valve flow, Q_O;

$$P_L = \frac{M\ddot{x}}{A} \tag{2}$$

gives the load pressure as that required to accelerate the load;

$$Q_L = \dot{x}A \tag{3}$$

is the flow into the cylinder; and we know that

$$Q_V = Q_L \tag{4}$$

if the system is closed.

Combining equations (1) - (4) and rearranging gives

$$\ddot{x} + \left(P_S\right)\left(\frac{A}{M}\right)\left(\frac{A}{Q_O}\right)^2 (\dot{x})^2 - \left(\frac{A}{M}\right) P_S = 0 \tag{5}$$

This is a form of the Ricatti equation, which may be solved by putting

$$u = e^{K_1 x}$$

which gives

$$\ddot{u} - K_1 K_2 u = 0$$

where

$$K_1 = P_S \left(\frac{A}{M}\right) \left(\frac{A}{Q_O}\right)^2$$

$$K_2 = \left(\frac{A}{M}\right) P_S$$

For the initial conditions

$$x = \dot{x} = 0 \text{ at } t = 0$$

we get, for

$$\omega = (K_1 K_2)^{1/2}$$

$$x = \left(\frac{1}{K_1}\right) \log (\cosh \omega t) \qquad (6)$$

$$\dot{x} = \left(\frac{K_2}{K_1}\right)^{1/2} \tanh \omega' t \qquad (7)$$

for the step response of the system.

A plot of these two functions against $\omega' t$ is given in Fig. D1. This plot shows that by the time $\omega' t$ equals one, the system is essentially at steady state.

Working back, the coefficient

$$\left(\frac{K_2}{K_1}\right)^{1/2} = \frac{Q_O}{A}$$

which is the limiting value of piston rate, is just the rate expected for no acceleration ($P_L = 0$).

For the NOTS winch traction unit, the actual values are:

$P_S = 2,000$ psi

$M = 1.2 \frac{\text{lb sec}^2}{\text{in}}$

$A = 4 \text{ in}^2$

$Q_O = 40$ gpm

giving

$$\omega' = 8(10^2) \text{ sec}^{-1}$$

Thus if a "time constant" were described for this non-linearity, it would be on the order of 10^{-2} sec.

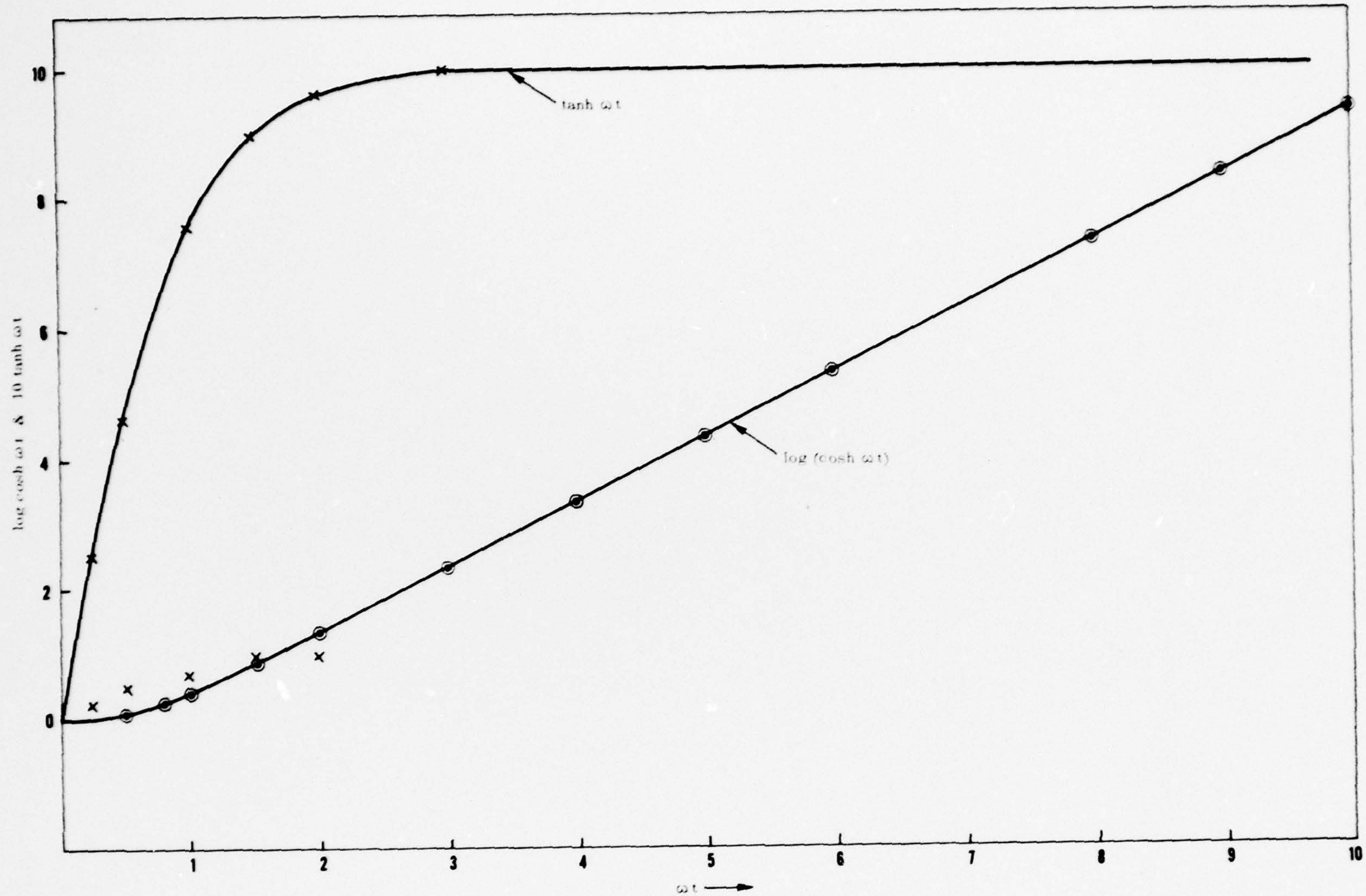


FIG. D1. Normalized Position and Velocity Response of a Mass Loaded Hydraulic Piston.

Although this is seemingly quite fast, the important thing to note is that the response to the input is non-linear, and in the manner of non-linearities, it gets worse as the required response (amplitude and/or frequency) increases. Limits on the "linear" operating range of this system may be heuristically determined in the following manner:

From the initial set of equations, this expression may be derived

$$\dot{x} = \left(\frac{Q_O}{A}\right) \left(\frac{P_S - \frac{m\ddot{x}}{A}}{P_S}\right)^{1/2} \tag{8}$$

Now let the input to the system, namely the unloaded servo valve flow, vary as

$$Q_O = Q \sin \omega t$$

giving a sinusoidal output described by

$$x = X \sin \omega t$$

$$\dot{x} = X\omega \cos \omega t$$

$$\ddot{x} = -X\omega^2 \sin \omega t$$

Now if the input amplitude is raised to, say,

$$Q_O' = 2Q \sin \omega t$$

then the peak values of $\dot{x}$ and $\ddot{x}$ should double. But from Eq. (8), an increase in $\ddot{x}$ decreases the pressure drop across the valve, so although the valve port ($\sim Q_O$) is twice as large, the flow cannot be twice as great; hence $\dot{x}$ cannot be twice as great at all times. Thus the output cannot linearly track the input. Note that a change in frequency has an even more pronounced effect, by the same reasoning.

In cases where

$$\frac{M\ddot{x}}{A} << P_S$$

the non-linearity will be negligible for most purposes. For example, if the maximum loading is kept to

$$\frac{M\ddot{x}}{A} \leq \frac{1}{10} P_S$$

then, for the numbers used earlier for the NOTS winch,

$$X\omega^2 \leq 700$$

For X = 0.4 inch (10% of full stroke)

and $\omega = 2\pi f$

this gives

$$f \leq 7 \text{ cps}$$

Since the velocity and acceleration are not in phase, their peak values do not occur at the same time and this estimate may be overly restrictive. It does serve to indicate, however, a method which, with appropriate values, will indicate the onset of the non-linearity.

For the purposes of designing a system of this sort for maximum response, these suggestions are pertinent:

(a) The response increases as $P_S^{1/2}$, so higher pressure does help, although slowly.

(b) If the servo valve has the capability of delivering more flow at the supply pressure than would normally be required by the system, the feedback loop can be set up to linearize the system still further, especially if the rate of the piston ($\dot{x}$) can be used as a feedback term.