CHAPTER 9

EQUIPMENT LOWERING MECHANICS

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1.0 INTRODUCTION

Improper design considerations while lowering or retrieving equipment at sea can result in substantial cable damage including rupture followed by the total loss of a valuable payload.

The hydrodynamic behavior of the payload at the end of the lowering cable is often unknown or ignored. Yet the shape and weight of the payload has considerable bearing on the cable performance. For example, a payload with a large drag area can substantially add to cable static tension when the cable is hauled
in. Conversely, while paying out the payload may fall slower than the cable itself thereby creating a slack cable condition. This slack condition may result in a subsequent snap load, when the lowering is stopped and the payload impacts the cable, or in a kink in the wire,, or both. Payload spinning, kiting and tumbling can also obviously impair an orderly lowering.

Recurrent causes of cable damage include: loading the cable beyond its yield point (or breaking strength), fatigue failure, and kinking. In this section the mechanisms which create these detrimental conditions are first reviewed. These potential failure mechanisms include: quasi-static tensile loads, wave induced dynamic loads, zero load (slack cable), and impact (snap) loads.

The mathematical concepts which are used to predict and quantify these causes of cable failure are also reviewed and their use illustrated with a few typical examples.

In the fast part of this chapter specific design recommendations are made for improving the payload hydrodynamic behavior. Finally, the operational limits such as maximum length of cable paid out or allowable payout rates are discussed.

2.0 MECHANISMS CAUSING LOWERING CABLE DAMAGE

One way to obtain measurements of oceanographic parameters at great depths is to lower sensing instrument packages with electromechanical cable. Of necessity these cables are kept to small, workable sizes but because of the long lengths deployed their immersed weight often results in very high tension levels. Vessel motion, due to wave action, introduces additional cyclic loads which can, and often do cause, cable deterioration due to flexure fatigue. Kinks can occur in the cable during zero load conditions. These problems often result, at best, in loss of electrical signal due to short or open circuits and at worst in complete failure of the cable and total loss of the instrument package.

While hanging free from the ship, the tension in the cable is the sum of the static load, due to cable and instrument immersed weight, and the dynamic load, due to cable and attached equipment inertia and hydrodynamic resistance. Most of the time static and dynamic effects occur simultaneously. However, for the sake of clarity it will be helpful to consider them separately.

2.1 Static and Quasi-Static Tensile Loads

By and large, cables are designed and built to resist fair amounts of tensile loads. As their working life progresses their original strength is reduced by corrosion, abrasion and normal wear and tear. If the tensile loads come close to the actual strength of the cable, permanent cable damage or even total failure will occur. To prevent this form of failure it is necessary to understand and quantify the mechanisms of tensile loading.

The first factor of tensile loading is plain weight. The weight that the cable must support at its ship end is made of two parts: 1) the weight in water of the payload and 2) the weight in water of the cable itself. Whereas the payload immersed weight remains constant, the immersed weight of the cable increases with the length of cable paid out. In many situations the payload weight is but a small fraction of the cable weight.

The second factor of cable tensile loading is due to hydrodynamic resistance (drag). If, on a calm day, the lowering winch is turning at a constant rate, the resulting steady state motion of the payload and lowering cable through the water will produce a quasi-static loading which, depending on the direction of motion (up or down), will add to or subtract from the static loading due to cable and payload weight. A few words on the nature of hydrodynamlc resistance will help understand how cable and payload drag interact and combine to drastically change the static loading due to weight only.

Simply stated, hydrodynamic resistance is the force experienced by a body when moving through a fluid. This resistance is due to a combination of viscous and pressure effects. These two effects are concurrent. Their relative magnitude depends, however, on the nature of the flow past the body. As long as the flow remains smooth, or laminar, shear stresses predominate and the resistance, or drag, is essentially due to the friction of the fluid on the bodies immersed surface (skin friction drag). On the other hand, when a combination of fluid speed and body shape (blunt bodies) result in a wake past the body the drag force is then essentially due to the pressure difference between the upstream and downstream sides of the body (pressure drag).

To illustrate the point, the force needed to tow a small but long and neutrally buoyant fishing line aft of a sail boat is essentially due to friction drag on the line. On the other hand the force experienced by someone towing a fully submerged bucket from a short rope aft of the same boat is essentially pressure drag.

In applications involving lowering and hauling equipment to and from the sea floor it is fair to say that the hydrodynamic resistance experienced is the sum of the friction drag on the cable and of the pressure drag on the equipment or payload at the end of the cable.

Cable drag is directly proportional to cable length, whereas equipment drag remains essentially constant. When hauling in, the hydrodynamic resistance will increase the static load due to cable and equipment weight. Its maximum contribution of course is at the beginning of the haul when the cable is longest. Methods to calculate drag forces are reviewed in the next section.

2.2 Wave Induced Dynamic Loads

Next we will consider the dynamic loads imparted to the cable as the ship heaves, rolls, and pitches in rough seas.

After lowering the payload to a certain depth (say 2,000 meters) let us secure (stop) the winch. If the cable tension at the head sheave could then be read and displayed, the record would show large fluctuations around a mean. This mean would of course be the immersed weight of the cable paid out and attached equipment. Deviations from the mean are due to dynamic forces imparted on the cable by the motion of the head sheave. As the cable and attached equipment are pulled towards the surface or allowed to plunge back into the sea the cable and the equipment experience both drag and inertia forces.

As previously mentioned, the drag forces are caused by cable and payload instantaneous speed. The inertia forces are caused by cable and payload instantaneous change of speed. Both forces are concurrent. Drag forces reach a maximum when the speed is largest, inertia forces are greatest at the time of maximum acceleration - usually when cable and payload are at rest, at the beginning of a new motion cycle. Here again it may be instructive to briefly look at the nature of the inertia forces. If at some instant the cable and the equipment are hanging still from the ship (zero speed) and at some later but proximate instant cable and payload are pulled upwards at some speed by a ship roll, the tension at the sheave increases. This tension increase is caused by the "inertia" of the cable and equipment which "resent" and resist the instantaneous upward pull.

In general the inertia force can be defined as the force required to change the speed of a body. lts magnitude equals the product of the body mass by the change of speed experienced per unit of time (acceleration).

Fully immersed bodies do trap and entrain a certain amount of water in their motion. This entrained water undergoes the same acceleration as the body itself. The effect is as if the mass of the body had been increased. In fact the actual mass to be accelerated, called the body virtual mass, is the sum of the body mass and of the mass of the entrained water. As a result in the increase in mass, the force needed to accelerate a body in water may be much larger than in air. For example the starting load to accelerate an elevator from rest would be much larger if the elevator was fully submerged (neglecting buoyancy effects). Formulas to calculate inertia forces are presented in the next section.

Now let us go back to our ship and let the winch run again, hoisting the equipment back to the surface at some constant hauling speed. The hydrodynamic resistance due to this additional speed will, at least in the beginning when cable weight reduction is not significant, increase the tension mean and therefore also the tension peaks previously experienced when heaving to, with the winch secured.

The instantaneous tension is now the algebraic sum of four simultaneously occurring effects, namely:

- o the drag due to hauling speed
- o the drag due to wave induced motion
- o the inertia forces to accelerate (or decelerate) the cable and the equipment.

This time varying, wave induced, tension results in cyclic stresses which can cause the wires and/or the conductors of a cable to fail in fatigue.

It is a well-known fact that the number of fatigue cycles to total failure dramatically decreases as the cyclic tension increases. In instrument lowering applications, because of the long lengths of cable required, the tension can reach a very large fraction of the cable strength. Under these conditions, only a few hundred cycles of repeated stresses can severely damage the cable (see Reference 1).

Keeping the wave induced loads and their time of application small will prevent accelerated fatigue cable deterioration.

2.3 Zero Load. Slack Conditions

Zero load can be the prelude to catastrophe. A slack cable can easily jump out of a sheave, can kink, or it can be subjected to severe snap loading. The payloads attached at the free end of the cable may force the cable to unlay and turn on itself. If the cable is allowed to become slack at some later time it will relieve some of the stored torsional energy by forming one or a number of twisted loops at the point of slack. When tension is reapplied the loops are pulled tight, the armor wires and the conductors are then severely bent thus permanently damaging the cable at the point of kink.

Understanding the mechanisms leading to slack conditions is a first step towards the prevention of their occurrence.

If a body heavier than water is allowed to free fall to the sea floor, it will first accelerate and gain speed. As speed increases so does the hydrodynamic resistance on the body. Sooner or later the drag will equal the pull of gravity and the body will continue to fall at a constant maximum speed called “terminal velocity.” This being accepted, let us consider what happens as the equipment is lowered to the bottom.

Assuming the sea to be flat calm and the winch to pay out at some constant and reasonable speed, then the equipment will descend smoothly at the payout speed. But if this speed is increased beyond the equipment's own terminal velocity then the cable will override the equipment and form a slack loop probably full of kinks.

One might be tempted to think that paying the cable at a rate less than the equipment terminal velocity would prevent slack conditions to occur anywhere along the cable. This is not always true. As evidenced in an example presented in the next section, in certain cases a length can be reached where the combined drag on the cable and the equipment entirely negates the gravity pull. The cable will then again become slack, this time at the shipboard end.

Now let us assume a situation where the winch is secured but the ship is rolling heavily. On a down roll the head sheave may well reach speeds high enough to momentarily create slack conditions either at the sheave, or at the equipment end, or at any point in between. Of course such high speeds can also be obtained when paying out from a rolling ship.

Methods for determining conditions of zero tension and points of occurrence will be briefly reviewed in the next section.

2.4 Snap Loads

Cable tension, as we have seen, is the algebraic sum of the external forces acting on the cable, namely the static force due to weight and the dynamic forces of inertia and hydrodynamic resistance.

This dynamic force can be either compressive or tensile. When the compressive component exceeds the static tensile force the cable goes slack. The payload is then allowed to travel on its own until the cable catches it again. Severe snap loads, as high as ten times the immersed weight of the payload (Reference 2) are then imparted to the cable.

It may again be instructive to describe the mechanism which produces snap loads in some detail. Let us assume that an up roll is pulling hard and fast on the cable. The steel cable is rather stiff, having a high modulus of elasticity. The payload has a large virtual mass.

It is heavy and its ugly shape entrains a lot of water. The upper end of the cable moves with the ship. Because of its inertia the payload does not move appreciably yet. The cable is forced to stretch and because of its stiffness the pull on the payload increases at a rapid rate. As a result the payload starts to move faster acquiring upward speed and momentum.

Now comes the down roll. The pull of the cable on the payload diminishes and vanishes as soon as the distance between payload and cable shipboard end equals the relaxed (no load) length of the cable. The payload then starts to travel on its own. It still has considerable momentum and keeps on going upwards, slowing down until the pull of gravity stops it. It then reverses direction of motion. It starts to fall acquiring downwards speed and momentum.

In the meantime the cable is still going down following the ship down roll and giving plenty of time for the payload to gain considerable downwards momentum. Now comes the next up roll. The cable rushes back to the surface. When the distance between the upper end of the cable and the payload position again equals the unstretched length of the cable, the cable starts to pull on the payload. The great force necessary to rapidly stop and reverse the direction of the payload constitutes the snap load.

lf properly timed, that is if the wave frequency is such as to permit the procedure to repeat itself the cable will be subjected to a series of snap loads and probably will break.

A simple mathematical model to predict the occurrence of snap loads and quantify their magnitude is presented in the next section.

3.0 PREDICTING CABLE LOADS

This section will present the formulas and certain simple analytical methods which will permit a reasonable prediction of both the static and dynamic cable loads.

3.1 Immersed Weight. Static Load

The weight of a fully immersed object equals the weight of the object in air less the weight of the water displaced by the object. If the two weights are equal the object is said to be neutrally buoyant. If the air weight of the object is less than the weight of the water displaced the buoyant object will want to come back to the surface.

Example 9.1

What is the static load at the ship due to 2,000 meters of 1/2 inch 3x19 wire rope supporting a cylinder of cast iron 4 feet high by 2 feet in diameter.

Use:

Weight in water of 1/2” 3x19	=	.341 lb/foot
Water density	=	64 lbs/cu foot
Cast iron density	=	450 lbs/cu foot
One meter	=	.28 feet

Solution

Air weight of cylinder	=	π x 4x450	=	5655
Weight of water displaced	=	π x 4x64	=	- 804
Immersed weight of cylinder			=	4851
Immersed weight of cable	=	2000x3.28x.341	=	2237
Static load at ship end of cable			=	7088 lbs

3.2 Hydrodynamic Resistance – Quasi-Static Load

The hydrodynamic resistance of a fully submerged object moving at a constant speed “V” (ft/sec) can be estimated using the formula:

$$D = 1/2^{\rho} C_D \ AV^2 \qquad (9.1)$$

where D is the hydrodynamic resistance or drag (lbs)

ρ is the water mass density = 2 slugs/cu.ft.

C_D is the drag coefficient

A is the object area used to empirically derive the drag coefficient (sq-ft)

Pressure drag coefficients for various body shapes (spheres, cylinders, plates, etc...) have been widely published in the literature (References 3 and 4). Longitudinal drag coefficients for cables and long cylinders have also been extensively studied. Published values vary from .02 for rough cylinders to .0025 for smooth cylinders (see Figure 9-1).

The following example illustrates the use of formula (9.1).

Example 9.2

A biological sampler is lowered to a depth of 2,000 meters with the help of a 3/8 inch 3x19 wire rope. It is then hauled back at a constant speed of 100 meters/mm. Find the tension at the upper end of the cable immediately after the starting transient, using the following characteristics:

Immersed weight of cable	=	.191 lb/ft
Drag coefficient of cable	=	.01
Shape of sampler	=	cone, base up, filled with water
Diameter = 6 ft; Height = 5 ft		
Immersed weight of Sampler = 200 lbs	=	Dry weight = 320 lbs
Drag coefficient of sampler	=	1.0

Also, find the percent increase due to drag over the plain static load.

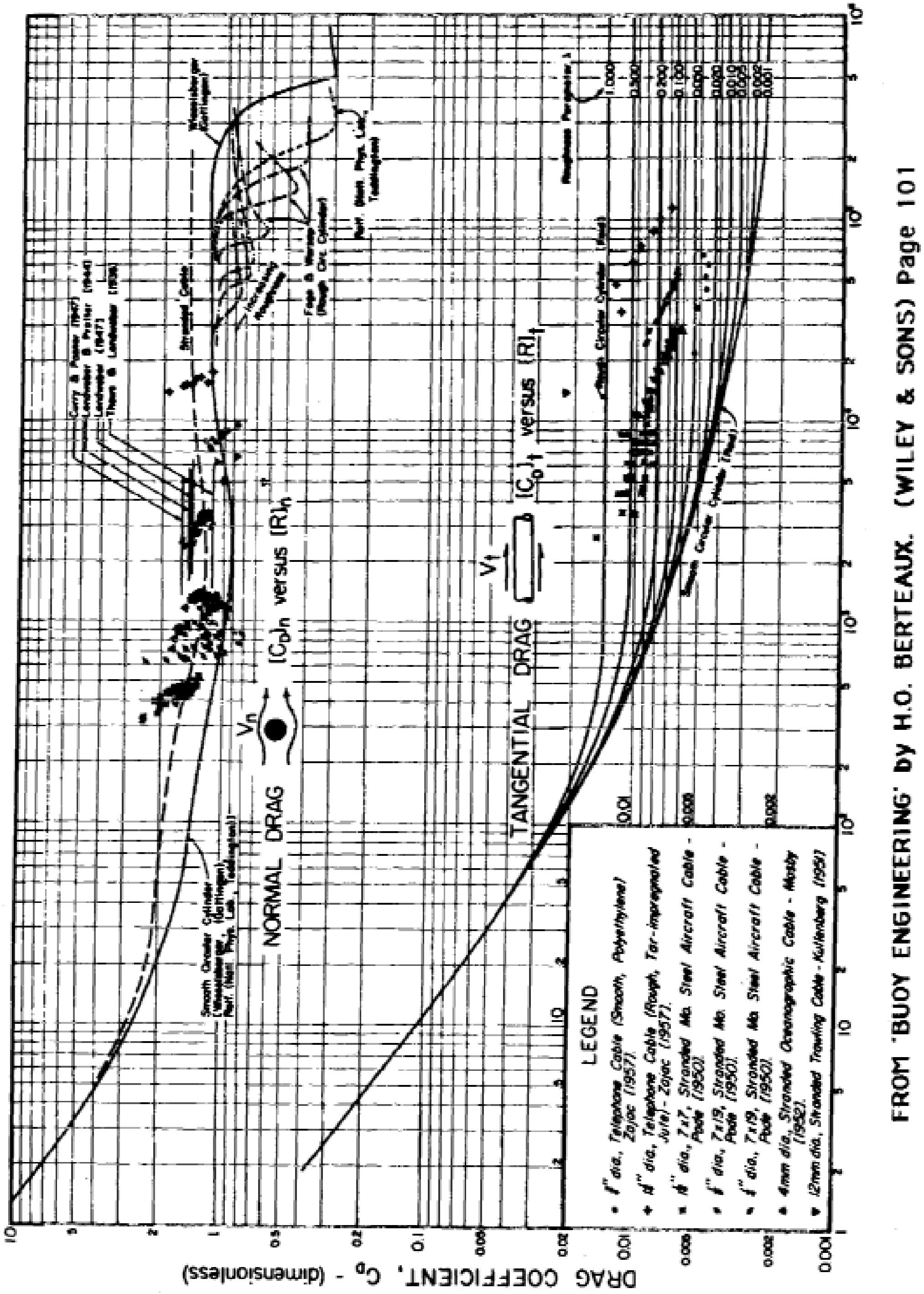
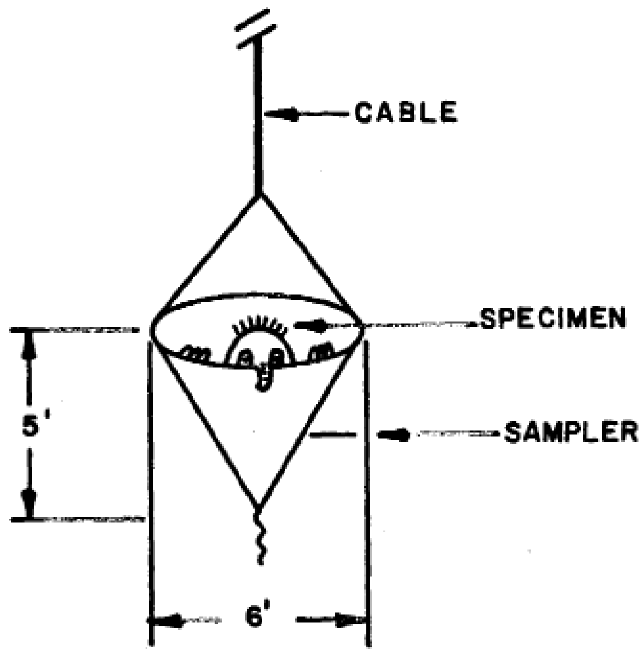


FIGURE 9-1

DRAG COEFFICIENT vs REYNOLDS NUMBER FOR FLOWS NORMAL AND TANGENTIAL TO SMOOTH AND ROUGH CIRCULAR CYLINDERS

Solution



Total immersed weight of cable	=	.191 x2000x3.2	=	1253 lbs
Weight of sampler	=			200 lbs
Static tension	=			1453 lbs
Hauling speed	=	100x3.28/60	=	5.46 ft/sec

Skin area of cab $= \pi \times \frac{3}{8} \times \frac{2000}{12} \times 3.28$

$= 644$ sq-ft

Cable drag $= 0.01 \times 644 \times 5.46 \times 5.46 = 192$ lbs

Cross section of sampler	=	π x 3 x 3
	=	28.3 sq.ft.
Sampler drag	=	1.0 x 28.3 x 5.46 x 5.46 = <u>844 lbs</u>
Total drag	=	<u>1036 lbs</u>
Tension at cable upper end	=	2489 lbs
Percent increase due to drag	=	1036/1453 = .713 or 71.3%

If "s" is the length of cable paid out, the quasi-static tension T(s) at the ship end can be found using the following expression:

$$T(s) = W_p + W_L s + 1/2\rho C_c \pi D_c s V|V| + 1/2\rho\, C_p A_p V|V| \qquad (9.2)$$

where

W_p = immersed weight of payload (lbs)

WL = immersed weight of cable per unit of length (lbs/ft)

C_c = cable longitudinal drag coefficient

D_c = cable diameter (ft)

C_p = payload normal drag coefficient

A_p = payload normal cross section (sq-ft)

V = constant cable speed, hauling being positive and lowering negative

$|V|$ = absolute value of V

Cable drag is both a function of speed and length. If the amount of cable paid out and the speed of lowering are large enough, the combined cable and payload drag can become as large as the cable and payload immersed weight. The tension in the cable then becomes zero.

For a given lowering speed V, the length of cable necessary to produce a slack condition can be found by setting T(s)=O in (9.2) and solving for s.

3.3 Terminal Velocity. Zero Load

At terminal velocity the immersed weight of the object "W" equals the drag on the object, a condition which is expressed by:

$$W = \frac{1}{2}\rho C_D A V_T^2$$

Therefore the terminal velocity VT of the object is given by:

$$V_T = \sqrt{\frac{2\,W}{\rho\, C_D A}} \qquad (9.3)$$

Example 9.3

Find the terminal velocity of:

1. The biological sampler described in Example 9.2 using a nose down drag coefficient $C_D = 0.2$.
2. The 2000 meters of 3/8" 3x1 9 wire rope combined with the sampler

Use ρ = 2 slugs/cu.ft

Solution

1. Terminal velocity of the sampler.

From previous computations,

Immersed weight of sampler	= 200 lbs.
Cross section	= 28.3 sq-ft

$$\text{Terminal velocity} = \sqrt{\frac{200}{0.2 \text{ x } 28.3}} = 5.94 \text{ ft/sec}$$

(1.81 m/sec)

or 109 meters/min

2. Terminal velocity of cable and sampler combined.

From previous computations,

Immersed weight of cable	=	1253 lbs.
Skin area of cable	=	644 sq ft
Drag coefficient of cable	=	0.01
Terminal velocity	=	

$$\sqrt{\frac{1253 + 200}{644 \text{ x } 0.01 + 0.2 \text{ x } 28.3}} = 10.95 \text{ ft/sec}$$

(3.34m/sec)

This example shows that a payout rate in excess of 109 meters/mm would cause a stack condition in the wire rope lower end. Similarly a downwards speed in excess of 11 ft/sec, which could be easily obtained by a combination of payout rate and ship down roll, would produce a slack condition at both ends of the 1,000 meters length of cable.

3.4 Virtual Mass. Inertia Load

As previously discussed, in order to accelerate a body immersed in water not only must the body be accelerated but also a certain amount of water close to or ahead of the body. As a result the force F needed to accelerate the body in water is greater than the force F required to accelerate the same body in vacuum. This can be expressed by:

$$F^1 = (m + m^1)a > F = ma$$

where m is the body mass

m^1 is the added mass of the entrained water

a is the acceleration.

The added mass is usually computed using

$$M^1 = cm\ \rho\ (Vol) \qquad (9.3)$$

where

C_m is the added mass coefficient, ρ is the water mass density (slugs/fl^3) and Vol is the volume of water displaced by the immersed body (cu-fl).

Added mass coefficients for bodies of different shape (sphere, cylinders, plates, etc.) have been empirically determined for linear and oscillating accelerations. Published values of added mass coefficients pertinent to cable lowering problems can be found in Reference 4.

The virtual mass m_v is the sum of the body mass m and of the added mass m^l.

$$m_v = m + m^1$$

Example 9.4

1. Find the virtual mass of the biological sampler previously discussed.
2. Find the inertia force on the lower end of the cable and the percent increase over the static load at that end if the sampler is accelerated towards the surface from rest to a speed of 8 ft/sec (146 meters/minute) in (a) 8 seconds, (b) 2 seconds.

Use $\rho = 2$ slugs/cu ft

and $C_M = 1.5$

Solution

1. Virtual mass of sampler.

Volume of sampler	=	$1/3\pi \times 3 \times 3 \times 5$	= 47.12 cu-ft
Mass of water in sampler	=	47.12 x 2	= 94.24 slugs

Added mass	=	1.5x47.12x2	=	141.36 slugs
Mass of sampler structure	=	320/32	=	10 slugs
Virtual mass			=	245.6 slugs

2. Inertia force.

Case a. The prudent operator brings the load to full speed in 8 seconds.

The average acceleration is then

$$\frac{8\,\text{ft/sec}}{8\,\text{sec}} = 1\,\text{ft/sec}^2$$

The average inertia force is then 245.6x1 = 245.6 lbs

The percent increase over the 200 lbs of static load due to the immersed weight of the sampler is then

$$\frac{245.5}{200} = 1.23$$

or 123% increase.

Case b. The "other" operator brings the load to full speed in two seconds.

The average acceleration is then

$$\frac{8\text{ ft/sec}}{2\text{ sec}} = 4\text{ ft/sec}^2$$

The resulting inertia force is then 245.6x4 = 982 lbs

The percent increase is

$$\frac{(982)}{200} = 4.92$$

or 492%. This is almost five times the immersed weight of the sampler at rest.

3.5 All Forces Considered. Steady State Peak Tensions

When the hauling and lowering of equipment is done in a rough sea way the tension is no longer time independent, and inertia as well as drag forces must be considered.

To find, under these conditions, the tension in the cable at the shipboard end it is practical to first assume a zero hauling speed (winch secured). Assuming the travel path of the payload and the cable to be vertical (or nearly so) the dynamic tension T(s,t) at the head sheave can then be evaluated with the help of Morisson's equation in one direction, namely

$$T(s,t)= W\rho + W_L s + l/2\ \rho(C_c\ \pi D_c s + C_p A_p)\ V\left|V\right| + (m+m^l)\frac{dV}{dt} \qquad (9.5)$$

where

s is the length of cable paid out

V is the vertical component of the head sheave speed at time t.

M is the mass of the cable and payload

m^l is the added mass of the cable and payload

and dV/dt is the vertical acceleration of the head sheave at time t.

Expression (9.5) implicitly stipulates that cable and payload rigidly follow the head sheave motion. In other words it treats the cable as a rigid bar. Despite this oversimplification expression (9.5) can be profitably used to calculate maxima of expected cable tension other than snap load.

To this end one first derives the expression of head sheave vertical speed and acceleration as a function of ship geometry, wave amplitude and frequency. These expressions are then introduced in (9.5) and values T(s,t) are computed at discrete time intervals over a full wave period. The time of maximum dynamic tension occurrence can then be found by inspection.

The next step is to add the quasi-static contribution of drag due to hauling speed. A computation is made of the sheave velocity at time of maximum T(s,t). The hauling speed is then added to this particular sheave velocity and the tension due to drag is then computed. Next the acceleration of the sheave is found for the time of maximum T(s,t) and the corresponding inertia force is also computed.

The instantaneous maximum tension is then the sum of the total drag force, the inertia force, and the immersed cable and payload weight.

This simple computing procedure is best implemented with the help of a computer. Reference 5 presents in detail a derivation of head sheave speed and acceleration due to ship heave and roll and a program to evaluate peak tensions due to combined hauling and ship motion.

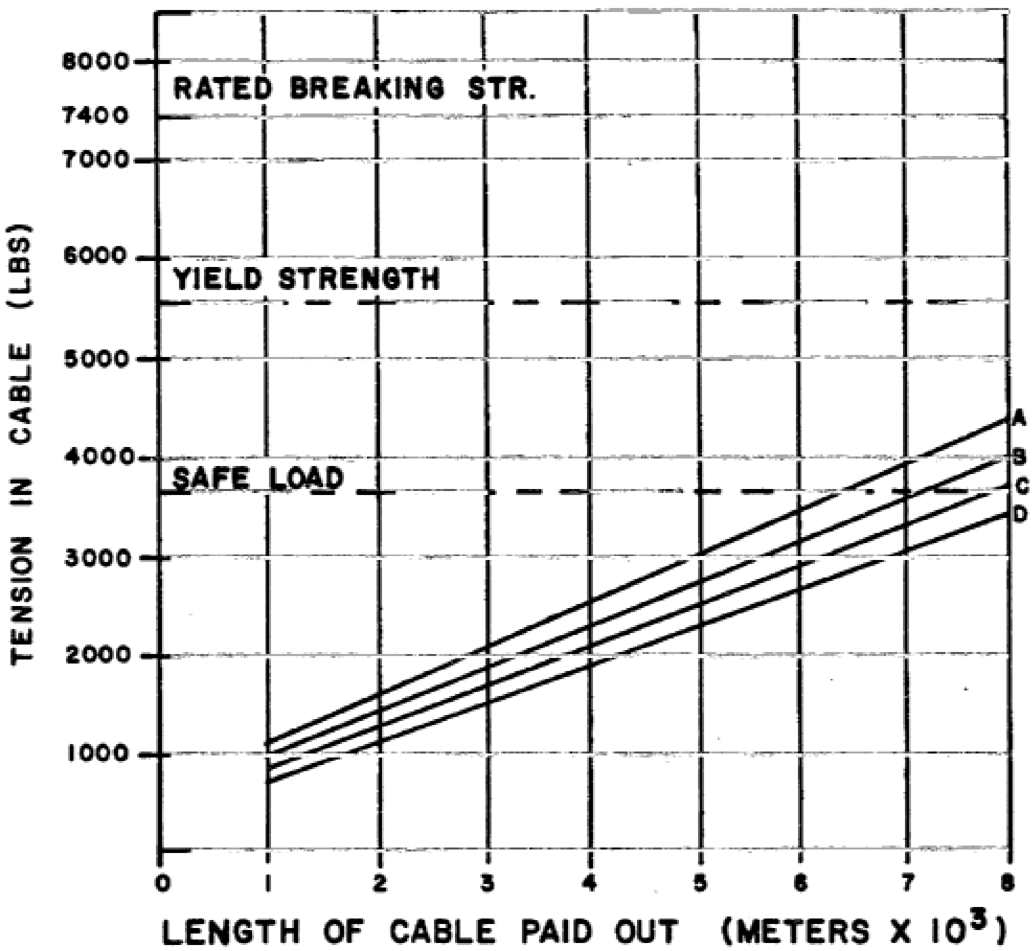
Once these calculations have been performed for a specific vessel the results obtained using this technique should be condensed and presented in a form easy to read. As an example, Figures 9-2 and 9-3 show the peak tensions calculated while hauling a CTD instrument package from the R/V ATLANTIS II under flat calm and sea state 3 conditions.

3.6 Snap Loads

A simple spring mass model (see Figure 9-4) can be used to predict the occurrence of snap loads and compute the ensuing cable tensions. In this model (Reference 2) the following assumptions are made:

- The motion of the payload is entirely vertical (one degree of freedom system).
- The mass of the cable is assumed to be a small fraction of the equipment mass. This would be the case for rather short lengths of cable (hundreds of meters instead of thousands), or if the cable is light (Kevlar line for example), or if the payload entrains a lot of water.
- The cable acts as a linear spring, the tension "T" being directly proportional to the cable elongation "ΔL" i.e.

$$T = k \, \Delta L \tag{9.6}$$



SEA STATE 0
SHIP - ATLANTIS II
PERIOD = N.A.
HEAVE AMPLITUDE = 0.0
ANGLE OF ROLL = 0.0

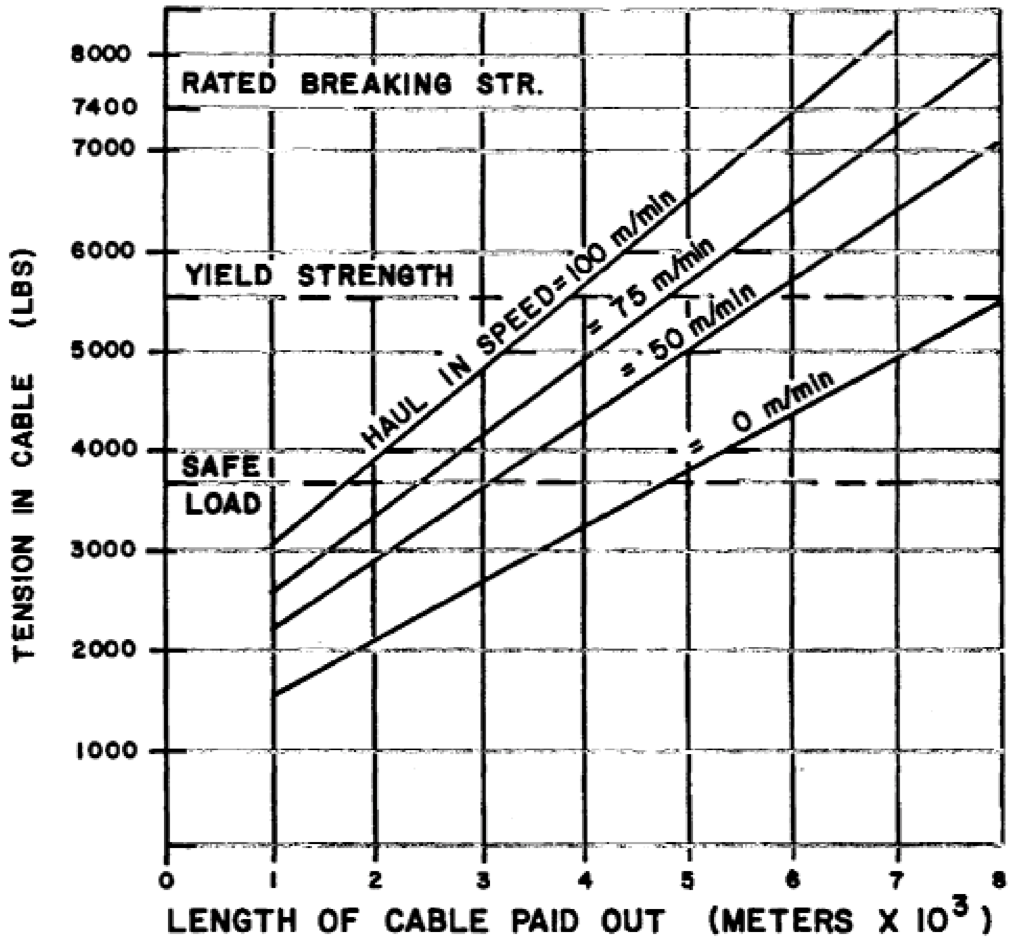
HAUL IN SPEED
A = 100 m/min
B = 75 m/min
C = 50 m/min
D = 0 m/min

CABLE CHARACTERISTICS =
WT/1000' = 145 lb.
DIAMETER = .303"
DRAG COEFF. = .01
RBS = 7400 lb.

INSTRUMENT CHARACTERISTICS =
IMMERSED WT. = 350 lb.
DRAG CONSTANT = 9.72 ft^2
VIRTUAL MASS = 21.0 SLUGS

FIGURE 9-2

PEAK TENSION AT HEAD SHEAVE vs LENGTH OF CABLE PAID OUT



SEA STATE 3
SHIP - ATLANTIS II
PERIOD = 8 SECONDS
HEAVE AMPLITUDE = 3 FT.
ANGLE OF ROLL = 15 DEGREES

CABLE CHARACTERISTICS =

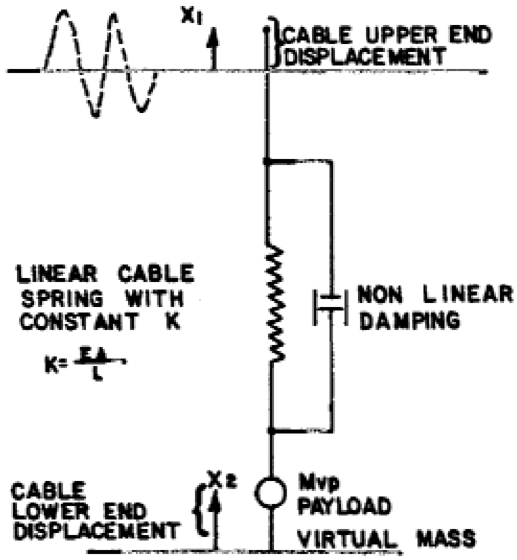
WT/1000' = 145 lb.
DIAMETER = .303"
DRAG COEFF. = .01
RBS = 7400 lb.

INSTRUMENT CHARACTERISTICS =

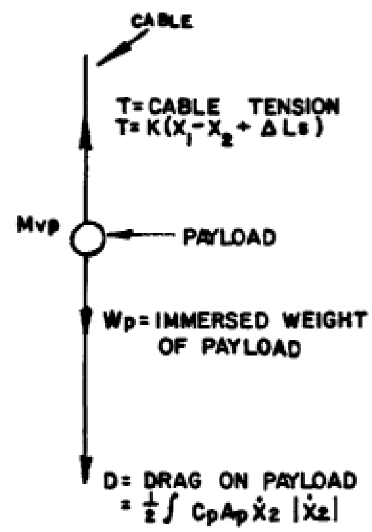
IMMERSED WT. = 350 lb.
DRAG CONSTANT = 9.72 ft^2
VIRTUAL MASS = 21.0 SLUGS

FIGURE 9-3

PEAK TENSION AT HEAD SHEAVE vs LENGTH OF CABLE PAID OUT



CABLE/PAYLOAD SPRING MASS MODEL



PAYLOAD
FREE BODY DIAGRAM

FIGURE 9-4

In the elastic range of cable elongation the spring constant k is given by

$$k = \frac{EA}{L}$$

where E is the cable modulus of elasticity (psi)

A is the cable metallic area (sq.in)

L is the cable unstretched length (ft)

k will then be expressed in lbs/ft.

- Equipment drag is non linear and of the form

$$D = 1/2\, \rho\, C_p A_p\, V\,|V|$$

- Equipment drag is assumed much larger than the cable drag.
- The vertical displacement of the cable upper end can be described by an explicit function of time (such as a sinusoid).

Applying Newton's law to the payload mass m_{v_p} m_{vp} (see Figure 9.4) yields the following equation of motion:

$$k(X_1 - X_2 + \Delta L_s) - W_p - 1/2\, \rho\, C_p A_p \dot{X}_2\, |\dot{X}_2| = m_{vp} \ddot{X}_2$$

where

K = cable spring constant (lb/ft)

X_1 = displacement of cable upper end (ft)

X_2 = displacement of cable lower end (ft)

ΔL_S = cable elongation under pure static loading (ft)

Wp = immersed weight of payload (lbs)

m_{vp} = virtual mass of payload (slugs)

X_2 = instantaneous speed of payload (ft/sec)

X_2 = instantaneous acceleration of payload (ft/sec^2)

Noting that $K\,\Delta L_S = W_p$, this equation of motion reduces to

$$k(X_1-X_2)- 1/2\ \rho C_p A_p X_2 \left| X_2 \right| = m_{vp} X_2 \qquad (9.7)$$

The instantaneous cable tension is then, at the payload end, given by

$$T = W_p + k(X_1-X_2) \qquad (9.8)$$

The motion of the payload mass is governed by equation (9.7) as long as T>O.

If T, as given by (9.8) equals zero, then the payload is no longer pulled by the cable and a new equation of motion will prevail. Applying Newton's law to the payload in free flight yields

$$-W_p - 1/2\rho\ C_p A_p\ \dot{X}_2 \left| \dot{X}_2 \right| = m_{vp} X_2 \qquad (9.9)$$

The system can be assumed to be initially at rest. At time t = 0, the upper end of the cable starts moving upwards. The ensuing motion of the payload is then found by integrating equation (9.7) using suitable numerical integration techniques. The author has found Euler's algorithm to be satisfactory provided the time increments are kept small.

Briefly stated, in this algorithm the acceleration of the payload over the time increment ΔT is given by:

$$\ddot{X}_2 = \text{Sum of the forces}/m_{vp}$$

The speed is then simply

$$\dot{X}_2 = \dot{X}_{2\,\mathbf{(t\text{-}\Delta T)}} + \ddot{X}_2\,\Delta T$$

where $$X_{2(t-\Delta T)} + \ddot{X}_2\,\Delta T$$

Similarly the displacement is then

$$X_2 = X_{2(t-\Delta T)} + \ddot{X}_2\,\Delta T$$

The tension is computed for each time interval, using (9.8). If it becomes positive then equation (9.7) prevails again. Speed and displacements when switching to a new equation are of course those computed in the time increment immediately preceding the switch over.

Here again this computing procedure is best implemented with the help of a computer.

<u>Example 9.5</u>

To illustrate the use of this technique let us consider the response of a particular payload/cable system with characteristics as follows:

- **Cable characteristics**

Type:	= 3x19 wire rope
Size	= .375 inch
Length	= 3000 ft
Immersed weight	= 3000 x .191 = 573 lbs
Modulus of elasticity	= 18,000,000 psi
Metallic area	= .1 sq. in
Strength	= 14,800 lbs

- **Payload characteristics**

Type	= heavy instrument package

Weight in air = 4200 lbs

Weight in water displaced = 2200 lbs

Added mass = 12 slugs

Normal area = 3.14 sq. ft

Drag coefficient = 1.0

- **Input**

  The vertical displacement of the cable upper end is assumed to be given by

$$X_1 = 7 \sin \frac{2\pi}{4} t$$

ie. Displacement amplitude = 7 ft

Period = 4 secs

<u>Solution</u>

After transient, the response of the system--as calculated by a computer program implementing the technique just described--is as shown on Figure 9-5. From this figure one can see that the vertical displacement of the cable lower end varies from -3.5 ft to+14.8 ft whereas the upper end goes from -7 to +7 ft.

The peak tension obtained after the period of slack is 8,100 lbs or four times as much as the static load (2,000 lbs).

Under static load the comfortable cable safety factor is 14,800/200 = 7.4.

Under snap load conditions, this safety factor reduces to a mere 14,800/8100 = 1.83.

3.7 Advanced Cable Dynamics

The purpose of this section was to introduce the basic principles which govern cable dynamics. External forces acting on the cable--immersed weight, hydrodynamic drag, and inertia--have been reviewed. Formulas to calculate their magnitude have been given. How these forces interact to produce tension peaks and, equally important, slack conditions in the cable has been explained with the help of simple mathematical models. These introductory concepts will enable the reader to quantify the impact that payload weight and shape as well as hauling speed and ship motion may have on cable tension. They will help predict extreme conditions which after all are the most important ones.

Of course there is much more to the science of cable dynamics. Models treating the cable as a continuum in which deformation waves travel and dissipate have been proposed. Others treat the cable as a multiple degree of freedom system made of a number of point masses connected by linear and nonlinear springs and damping elements. Cable response to deterministic and random input has been investigated both in the time and the frequency domain. Readers interested in this field are referred to the bibliography at the end of this chapter.

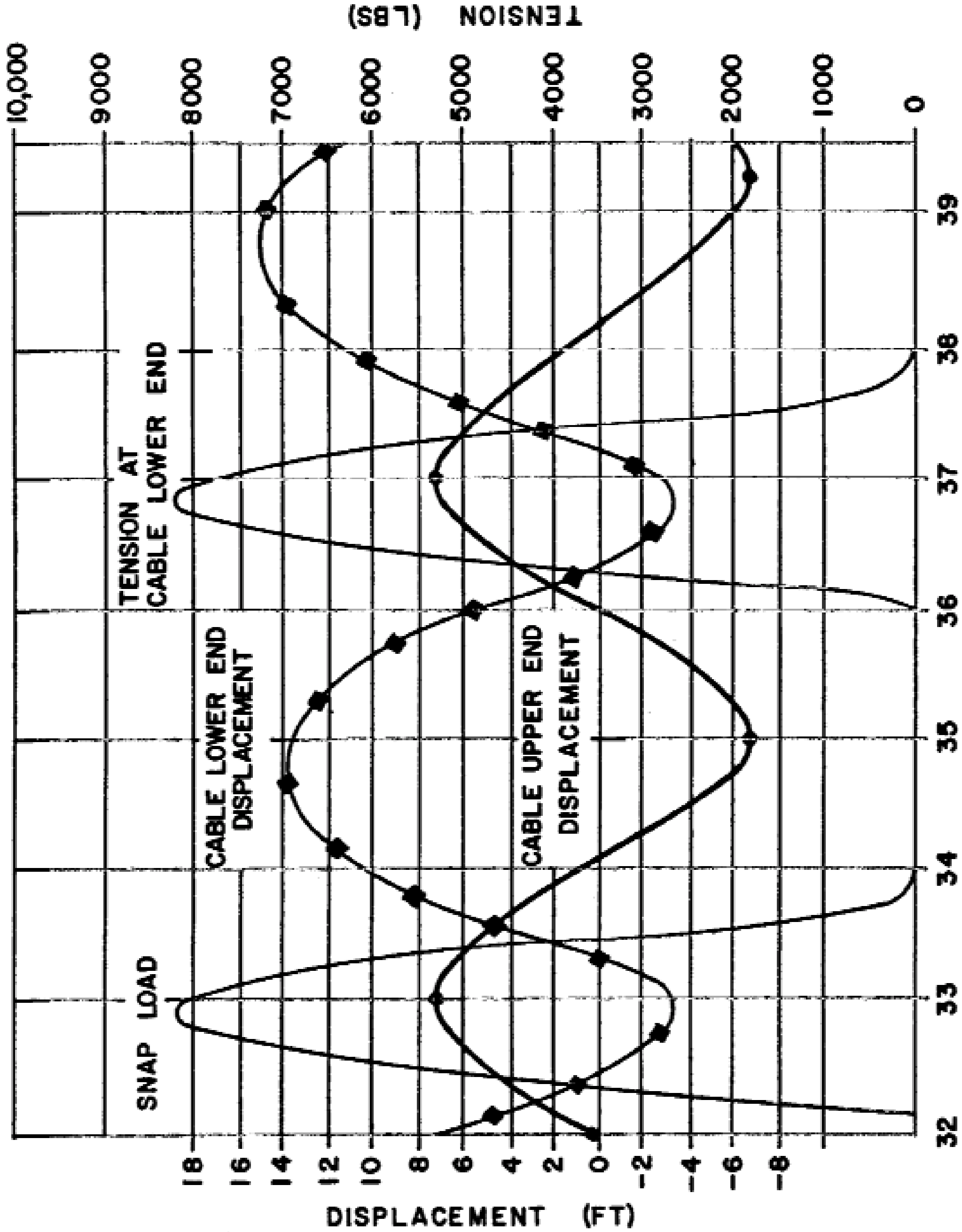
4.0 RECOMMENDATIONS

In this section certain recommendations will be made to improve lowering mechanics and increase cable life expectancy.

4.1 Equipment Design Considerations

Equipment handled underwater is subjected to hydrodynamic forces not present when handled ashore. If the equipment has poor hydrodynamic characteristics it will impart high and unnecessary loads to the cable or cause it to kink and fail. To reduce or better yet suppress these detrimental conditions, underwater payloads should be designed with the following considerations in mind.

4.1.1 Stability. If the payload was permitted to free fall it should do so in a vertical path. In addition it should not tumble, flutter, or spin. A payload falling sideways, or kiting, will pull the cable at large angles from the vertical perhaps causing large cable bites and slack conditions. A tumbling or fluttering payload will jerk the cable at the point of



CABLE LOWER END DISPLACEMENT AND TENSION AS A FUNCTION OF CABLE UPPER END DISPLACEMENT.
(IMMERSED WEIGHT OF PAYLOAD EQUALS 2000 LBS)

FIGURE 9-5

attachment which may fail due to repeated bending. A spinning load can force hundreds of turns in the cable which will kink at the first opportunity.

To fall in a plumb, orderly way, the payload must be statically and dynamically stable. The payload is statically stable if it has a natural tendency to return to an upright steady state vertical flight. The payload is dynamically stable if it returns to its steady state upright vertical flight with oscillations of decaying amplitudes.

Investigating the static stability of a submerged object falling at some constant speed is straightforward and relatively easy. The step-by-step procedure involves:

- assume an initial tilt angle from the vertical
- resolve the drag forces into normal and tangential components
- compute the moments with respect to the body center of gravity induced by the buoyancy and drag forces
- sum the moments

If the resultant moment tends to reduce the initial tilt angle then the object is statically stable at that angle. lf not it will have a tendency to capsize. The process must then be repeated for increasing initial tilt angles.

If an object is found to be statically unstable over a large range of tilt angles (say from 0 to 45°), then it is unfit for cable deployment. Its configuration must be altered until a proper combination of weight distribution and drag righting moment is found for all tilt angles considered. An example of such sensitivity study can be found in Reference 6.

Predicting the dynamic stability of free falling objects involves the simultaneous solution of six nonlinear partial differential equations. The mathematics required for this solution certainly go beyond the scope of this discussion. Interested readers should again consult the bibliography at the end of this chapter.

The following practical considerations if systematically implemented, can greatly improve the flight stability of cable lowered equipment and instrument packages.

- Weight Distribution. Packages should not be too heavy. Their center of gravity should be as low as possible and in all cases well below the center of buoyancy so as to provide a good righting moment.

- Shape. When placing instrumentation on a frame or equipment in some packaging form an effort should be made at reducing the top and bottom drag areas. A slender package will have much less drag as it travels vertically through the water than a fat, chubby one. Furthermore, it will entrain much less water and its added mass will be small.

- Symmetry. Vertical axisymmetry will greatly enhance flight stability. The payload weight should be distributed evenly around the vertical axis. If not the center of gravity will be off the center line and the package will not hang vertically from its point of cable attachment. The payload shape should also be axisymmetrical so that fluid induced forces cancel each other. As demonstrated in tank tests, an acoustic pinger strapped on the outside of an instrument package frame causes the package to tilt and kite sideways as it sinks.

- Spin. In certain cases, equally distributed appendages can have the proper shape or inclination to induce a torque on the lowered equipment. This torque will force the equipment to spin, It is often possible to observe the spin of a load at the beginning (or the end) of its lowering. If detected, the condition causing the load to spin should be corrected.

- Control Surface. Control surfaces can sometimes be used to advantage to stabilize an otherwise tumbling payload. Vertical fins located in the upper part of the package can provide a good righting moment, however, off center loads equipped with vertical fins will steadily kite sideways. Furthermore, if one or several fins are bent, the load will spin. Horizontally mounted circular flaps are known to be very effective for stabilizing blunt cylinders. They are less sensitive to off-center loading. Their drawback is to reduce the cylinder terminal velocity.

4.1.2 Terminal Velocity. As previously explained to maintain tension in the lowering cable the speed of the payload fall must always exceed the speed of the cable. A payload with a small terminal velocity will therefore impose limits on the payout rate and/or the sea state in which the lowering operation can take place. If such operations are repetitive - as in the case of oceanographic profiling instrumentation - the ship time consumed in performing the lowering operation or in waiting for favorable weather becomes prohibitively expensive.

Achieving a reasonable fast terminal velocity should therefore be an important design consideration. The equipment designer should not hesitate at clamping some lead or steel blocks at the bottom of the payload to increase its weight. Doubling the instrument weight in most cases would have but a small effect on the lowering cable safety factor. Reducing the drag area and profiling the bottom of the equipment package will also increase the terminal velocity.

4.2 Equipment Handling Considerations

Now that the equipment has been properly shaped and trimmed, an investigation should be made of the operational limits necessary for its orderly and safe deployment.

4.2.1 Depth Limits. Maximum tension occurs at the head sheave. As previously outlined this tension depends on the length of cable paid out, the weight and shape of the payload, the prevailing sea state and the hauling speed.

Whatever the actual condition of use may be, this tension should not be permitted to exceed a value corresponding to a safety factor of two for most applications, and in no case larger than the yield strength of the cable (about 75% of cable breaking strength for most data logging cables). To help plan safe lowering, predictions of tension levels should therefore be readily available. If for example one had graphs of peak tension versus cable length for different hauling speeds and sea states of the type shown in Figures 9-2 and 9-3, then the maximum allowable cable length could be explicitly and rapidly established. In this case, the maximum length that the cable can (or should) have for a given sea state and a given hauling speed can be easily found from the intersection of the particular tension curve with the safe load (50% of RBS) line or the yield strength (75% of RBS) line.

4.2.2 Winch Speeds Limits. Critical and/or repetitive lowering operations should certainly avoid slack cable conditions. Calculations of winch speeds which would cause the cable to become slack should be made as a function of sea state and length of cable paid out. These predictions should be made for every type of equipment lowered. They should be available in a convenient and tabular form. Limits on payout rates should then be set accordingly.

Measurements of oceanographic packages hydrodynamic behavior have been made both on scale models and on actual instruments being lowered from rolling ships (Reference 7). Cable slack conditions followed by severe snap loads at the cable lower end have been observed and reported (Reference 8).

4.3 Motion Compensation. Limits on deployment depths to avoid high dynamic stresses and on payout rates to prevent slack conditions should be considered temporary measures. The alternative is to consider motion compensators which can greatly reduce or suppress the undesirable effects of ship motion. The following considerations on motion compensation were prepared by J.D. Bird (Reference 9).

Motion compensating handling systems for shipboard applications can be categorized in at least two different ways. The first is a mechanical classification that is descriptive of the basic hardware utilized as the primary compensating element:

- Ram tensioners (the term tensioner is retained here since these units were first commonly used in tensioning applications).

- Bobbing booms, and

- Controllable winches.

The second category is a control law classification that is based upon the primary input signal used in the compensation strategy. The two major strategies are:

- Tension activated, and
- Motion activated

Two or more of these basic techniques can also be combined into more complicated systems where certain characteristics of each are desired. One example might be the high frequency rashness of a ram plus the larger amplitude capability of a controllable winch.

The three basic hardware approaches are shown schematically in Figure 9-6. In evaluating alternative hardware approaches, a number of parameters can be examined that will facilitate fair comparisons between approaches that differ significantly in their method of operation. These include total weight, deck space required, power consumption, complexity, cost, cable wear and fatigue during the compensation process, and frequency response. Most of these are fairly straightforward and self explanatory. Frequency response is an important control system parameter that is a complex function of several other parameters including torque available for control, effective inertia at the load, total system compliance and control actuator response. The effective inertia and system compliance define a natural frequency, above which it is difficult to. achieve effective control response. The torque available is one measure of the limit on the rate at which the control can be applied to the system. The servo-control activator can usually be chosen to have response characteristics above these other limits. The effective inertia of the system is an aggregate measure of the inertias of all moving components reflected to a common point such as the drum or load. Reducing the effective inertia increases the natural resonant frequency and responds faster to limited applications of torque. Low effective system inertia is therefore one of the most important characteristics of the hardware that results in systems with better frequency response and wider overall band width.

The Ram Tensioner is a hydraulic cylinder with a sheave or sheaves attached at the end of the piston. The cylinder can be mounted in any orientation that permits the cable to be fair lead from the winch, around the ram sheaves and to the overboarding sheave. As the ram piston is extended the cable is hauled in at the overboarding sheave. Cable is payed out when the piston is retracted. By making multiple passes around the ram sheaves, the cable compensation amplitude can be several times the piston stroke. Ram tensioners have a relatively low effective inertia, however they have fixed maximum amplitude, and subject the cable to relatively high wear and fatigue.

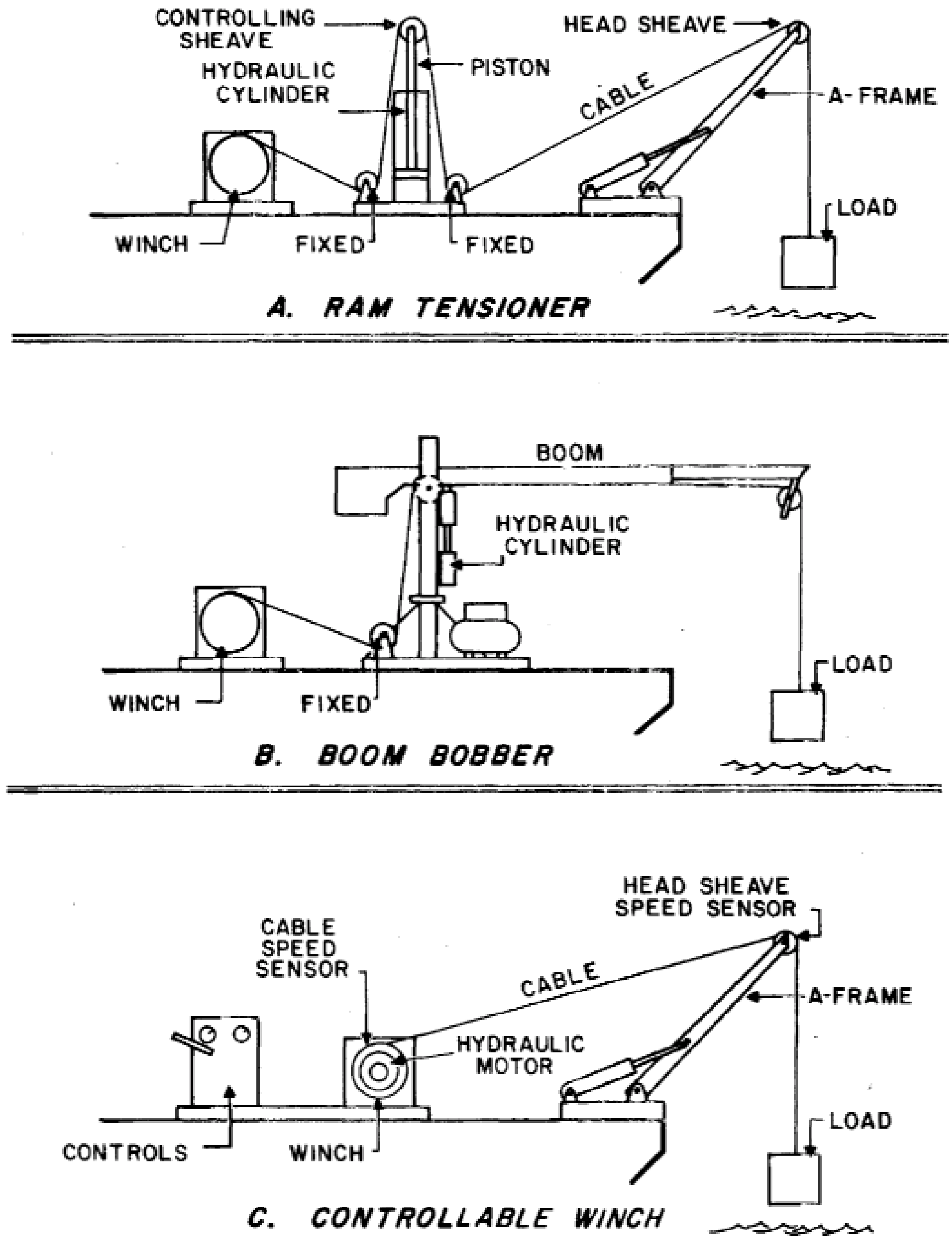


FIGURE 9-6

MOTION COMPENSATION HARDWARE ALTERNATIVES

The Boom Bobber is a cantilevered arm free to pivot at one end with an overboarding sheave fixed at the other end and a hydraulic cylinder located somewhere along its length to support the weight of the arm and the cable suspended payload. As the piston is extended or retracted, the overboarding sheave is raised and lowered respectively. Boom bobbers have a relatively high effective inertia due to the required mass of the moving boom structure and therefore have poor frequency response in servo-controlled applications. Like the ram tensioner, they also have limited compensation amplitude. With careful reeving, however, cable wear can be held to a minimum.

Controllable winches are mechanically the simplest of the three approaches since, presumably, a winch is required in the system for normal cable handling. Winches have medium to low effective inertia depending upon the particular design. Since the effective inertia of the drive motors at the winch drum is increased by the drive gear ratio squared, high speed motor drives generally have higher effective inertias than slower speed, direct drive motors. A major advantage of the controllable winch is that the amplitude of compensation is limited only by the length of the suspension cable. Cable wear with a compensating winch is moderate, when compared to the ram tensioner and the boom bobber.

Current literature tends to group motion compensation strategies as either “Active” or “Passive.” This may be an unfortunate choice of terms. Active systems are thought of as ones that add energy while passive systems do not. Active systems are thought of as possessing feedback elements while passive systems do not. For these reasons, passive systems are considered to be inherently stable, but this may not always be the case. If a passive system has a spring-mass resonance near the peak of the ship motion spectrum, responses can grow uncontrollably.

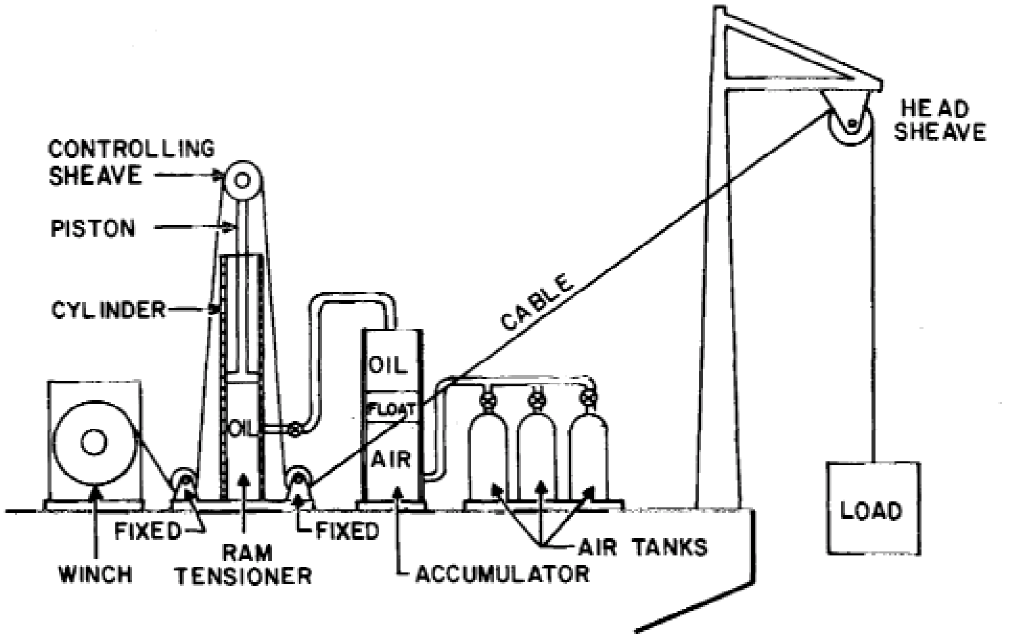
Systems which are basically classified as passive, such as boom bobbers, often contain rather complex servo-control systems to maintain nominal tension bias to compensate for a changing suspended load such as increasing cable length. For these reasons, the more appropriate and less ambiguous grouping of “Tension Activated” and “Motion Activated” compensation system is proposed here.

Tension activated systems respond to changes in wire rope tension by hauling in or paying out the line in such a manner as to reduce these loads. The most common examples of this type of system are boom bobbers and ram tensioners supported by hydraulic accumulators (Figure 9-7a). There are some basic characteristics associated with tension activated systems. They exhibit an effective spring constant which is in series with the spring-mass system of the cable and payload. In order to minimize the change in tension for a given ship displacement, this effective spring constant must be relatively low. The natural frequency of the cable-payload system is generally above the significant heave spectrum of the ship, except for very deep casts. The addition of another spring often aggravates the problem by moving the system’s natural resonant frequency nearer to the ship’s heave frequency. The solution to this dilemma is to provide some damping (a natural byproduct of hydraulic oil moving through piping and accumulators) and to soften the spring sufficiently to lower the natural frequency below the ship’s range of significant heave energy. The resulting soft spring constant provides poor position control, since small changes in tension result in large displacements.

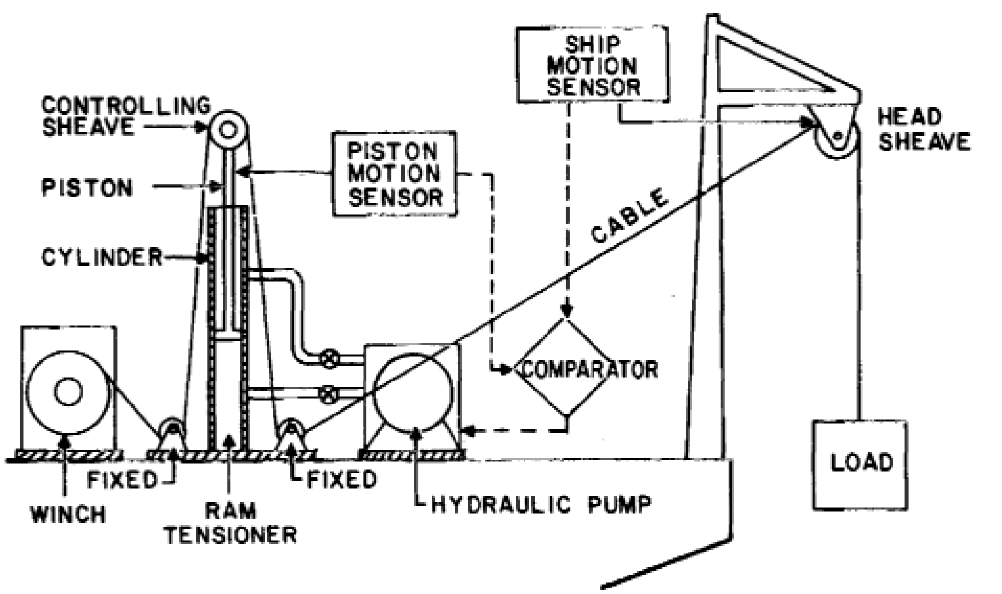
Tension activated systems actually do very little to directly control the position of the payload. Some researchers have experienced reductions in motion of payloads with properly tuned tension activated systems, but this was primarily due to damping and a shift in the systems natural resonance away from the predominant frequency of the ship motion energy. The primary function of tension activated systems is to reduce the magnitude of tension fluctuations in the suspension cable. For this purpose, they are relatively effective.

Motion activated systems (Figure 9-7b) , on the other hand, deal with the problem at its source. If the upper end of the suspension cable can be held stationary in inertial space, the unwanted energy cannot be transmitted to the payload. This point is particularly well made by Clifford L. Trump in his paper, “Effects of Ship’s Roll on the Quality of Precision CTD Data” (Reference 10). Motion activated systems, however, do not utilize the rather easily monitored tension input. Measurement of the vertical motion of the suspension point requires elaborate instrumentation. For this reason, motion activated systems generally have more complex servocontrol elements and multiple feedback loops. System stability becomes more of a concern when high gains are used to provide the required frequency response and accuracy.

Finally, a third category of motion compensators must also be considered. These are “Tension-limiting” devices such as shock absorbers and slip clutches. They are, however, probably



A. TENSION ACTIVATED RAM TENSIONER (PASSIVE)



B. MOTION ACTIVATED RAM TENSIONER (ACTIVE)

FIGURE 9-7 A & B

best categorized as a subset of Tension activated devices although they do little or nothing during normal operations and only begin to function during overload conditions to serve, much in the capacity of an electrical fuse, to prevent the cable from breaking under excessive overloads.

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