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Bobbing Crane Heave Compensation for the Deep Towed Fiber Optic Survey System

Michael J. Purcell and Ned C. Forrester - Woods Hole Oceanographic Institution

ABSTRACT

The Fiber Optic Survey System (FOSS) is an underwater vehicle designed for high resolution observation of the seafloor. Towed with the standard 17.3 mm (0.68 inch) fiber optic, electromechanical cable at speeds up to 1 m/s (2 knots), it may be deployed at depths from 100 meters to 6000 meters in wave conditions up to sea state 4. These parameters result in large variations in deployed cable length, towing angle, static cable tension and cable/vehicle dynamic response to wave induced ship motions that are transmitted into the cable. Computer modeling of the cable/vehicle system indicates that certain combinations of towing parameters result in excessive vehicle motion or high dynamic cable tension. This paper addresses the effectiveness of heave compensators, particularly a bobbing crane, in reducing induced motion and cable tension. Both a passive and a combination active/passive system have been considered. Modeling results indicate that the passive system reduces high dynamic cable tensions, but eliminating induced vehicle motion requires an active component.

1 INTRODUCTION

Towed underwater vehicles and instruments are used extensively for exploring the ocean. Examples include biological sampling, sonar mapping of the seafloor and underwater photography. A concern in any towed system is its response to ship motion that is transmitted into the tow cable. Since lateral motion is quickly dampened by drag, the primary concern is axial motion of the cable. Axial cable motion may induce motion of the vehicle that disrupts its ability to perform a task or collect quality data. Worse, generated dynamic tensions may combine with the static tension to exceed the cable breaking strength or result in fatigue failure at the cable termination. As seas increase, the risk of cable failure increases and the tow system may have to be retrieved, resulting in costly delays in completing a survey or scientific study. Integrating a heave compensator with the towed system can reduce the ship induced vehicle motion and dynamic cable tensions. This may allow the system to be deployed during wave conditions that would otherwise require retrieval.

Heave compensators have been used in the ocean drilling industry and in towed systems for some time. Types of heave compensation systems include: constant tension winches, ram tensioners and bobbing cranes. Many of these systems are passive in that they respond to the ship motion and changes in cable tension rather than being driven based on measurements of these values. With a towed vehicle system, an attempt is made to keep the ship tow point stationary or to deploy and retrieve cable to compensate for the ship motion. With the exception of the constant tension winch (Mitchell, 1992), these compensators include pneumatic and/or hydraulic systems that store energy as tension increases. This energy is used to return the compensator to its original position as tension decreases. Initially, the fluid system pressure must be set to support the static load, which is based on the towing parameters. As towing conditions such as deployed cable length change, the system pressure must be modified to balance the new static load. Another important consideration is the range of the heave compensator. Sufficient compensator range is required to ensure that the compensator will not reach a travel limit, which would result in high dynamic tensions. Ship motion is dependent on wave conditions and the ship response to the waves. The required compensator range may be established based on knowledge of ship response and planned maximum sea state.

The type of heave compensator selected often depends on the application's requirements. In some cases the heave compensator may be required only during short term operations such as launch and recovery.

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The ram tensioner with multiple wraps of cable around two sheaves with variable separation is ideal for this application or for short term operations near the seafloor such as borehole re-entry. One of the sheaves is mounted on pistons that slide inside cylinders connected to a pneumatic or hydraulic system. Increases in cable tension are minimized because the moving sheave slides toward the stationary sheave to deploy additional cable as tension increases. In addition, the sheave motion increases pressure in the fluid system, which pushes the moving sheave back to the equilibrium position, retrieving cable, when cable tension decreases. Liu (1983) reports a significant reduction in dynamic tension and elimination of snap loads with a ram tensioner during sea tests simulating a salvage operation. A ram tensioner is less useful for continuous towing operations because it may lead to bending fatigue failures of the tow cable at the ram tensioner.

Much of the work on heave compensation has been focused in the area of ocean drilling (Azpiazu 1983, Butler 1971 and Niedzwecki 1988). Long drill strings suspended from ships are subjected to high static loads and generation of a resonance condition could lead to costly failures. Heave compensators designed for drilling systems must have the capacity to handle the large loads and have sufficient stroke to compensate for extreme ship motions. The design is somewhat facilitated by the continuous vertical orientation of the drill string, to which the compensator is aligned, and by the lack of a size limit on the heave compensator, which may be an integral part of the drill ship. Typically, drilling ship compensators use the ram tension technique with the drill pipe attached to the pistons. A long stroke is required because this design does not have the advantage of multiple wraps of cable, as in tensioners used for towed systems.

The bobbing crane compensator has been studied by Kidera (1983) and Hover (1993 and 1994). Kidera designed a passive bobbing crane compensator for a towed instrument array that had previously been limited to low sea state operation because of vertical motion of the cable. The compensator was optimized for the lightweight instrument array, which maintained a vertical orientation with a depressor. The low weight of the system allowed utilization of a low stiffness system (natural frequency of 25 sec), and a large stroke of 8 m allowed operation in higher sea states. Kidera reports 96% reduction in vessel induced vertical motion in sea state 5. Hover's research into using a passive bobbing crane compensator with a towed underwater vehicle more closely matches our application with FOSS. His investigations showed important parameters that must be considered in designing this type of system and addressed limitations in performance.

2 Fiber Optic Survey System

In the investigation of heave compensation for FOSS, reducing dynamic tensions has been a priority over reducing vehicle motions. Heave compensation for launch and recovery operations has not been considered because the cable handling system will soon include a tension limiting winch for these operations. The unique characteristics of FOSS that complicate the heave compensator design are its large weight and varying towing angle. These are directly opposite the parameters Kidera (1983) faced when designing his bobbing crane system, but the same type of compensator has been pursued for FOSS because it minimizes bend cycling of the cable.

The FOSS heave compensator must have sufficient range and capacity for continuous operation in wave conditions up to sea state 4. The significant wave height (peak-to-peak) in sea state 4 ranges from 1.5 to 2.5 meters. The probable wave period is 9 seconds, but the wave period ranges from 6-17 seconds. The maximum heave motion of the ship was estimated based on a weekly wave (1/70000) that is 2.4 times the significant wave height (U. S. Army, 1977) and maximum ship response amplitude operator of 1.3. This results in maximum significant heave at the fantail of 3.2 meters (p-p) and maximum heave of 7.6 meters (p-p).

FOSS is an open frame vehicle 5.2 meters in length, 1.07 meters in width and 1.83 meters high capable of deployment to 6000 meters. The frame is constructed of stainless steel and accounts for about one third of the air weight of 31 kN. In water FOSS weighs approximately 20 kN. The major sensors on FOSS include 4 video cameras, two electronic still cameras, two 35 mm film cameras, dual frequency side scan sonar, sub-bottom profiling sonar and a CTFM forward looking sonar. The vehicle size and geometry resulted from maximizing the distance between the cameras and lights, the large number of sensors and a focus on maintainability.

FOSS is towed at ship speeds up to 1 m/s (2 knots) with the standard 17.3 mm (0.68 inches) diameter, fiber optic, electromechanical cable. The high cable breaking strength of 200 kN and torque balanced design are achieved with three external layers of steel armor wires. The cable core components include 3 single mode optical fibers and 3 copper conductors for transmission of data and up to 12 kVA of electrical power. The tow cable weighs 10.2 N/m in air and 7.8 N/m in water. The static load at the overboard sheave varies from the vehicle water weight, 20 kN, near the surface to approximately 68 kN with the vehicle deployed to 6000 m. Cable angle at the overboard sheave depends on both speed and depth as shown in Figure 1. It varies from 90° (straight down) to 42° from the horizontal.

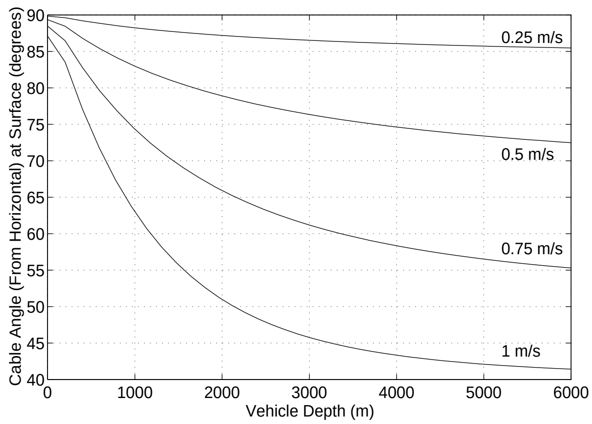


Figure 1. Cable Angle to Horizontal as a Function of Speed and Depth

FOSS may be towed from ships of opportunity and therefore has significant deck support equipment. This includes two modified shipping containers: the vehicle maintenance van, and the mission control module from which FOSS is controlled and all sensor data is viewed and recorded. Other deck equipment includes a palletable 112 kW (150 hp) cable handling system with both diesel and electrohydraulic power supplies and a standard marine crane from which FOSS is towed. A heave compensation crane designed for FOSS must be easily broken down for shipment.

3 COMPUTER MODEL

A computer model was used to study the behavior of the FOSS system as presently towed from a fixed crane and to investigate the possible performance improvements available through the use of various compensating systems. A time domain model was selected to account for the non-linear behavior of a bobbing crane compensator and for the quadratic vehicle and cable drag. The model is divided into four main parts: a cable, a vehicle, a source of assumed ship motions and an optional compensator.

The cable and vehicle models are borrowed directly from Hover (1990) and Triantafyllou (1990), where they are described in detail. The cable is modeled as a two dimensional plane system. The initial cable shape and tension as a function of depth are determined in a static solution based on vehicle mass and drag, cable mass and drag, tow speed and any additional water currents (no additional currents were used for this work). The dynamic solution is computed as a perturbation of the static solution, using the vehicle dynamics and motion of the surface tow point as boundary conditions. The cable parameters modeled are shown in Table 1.

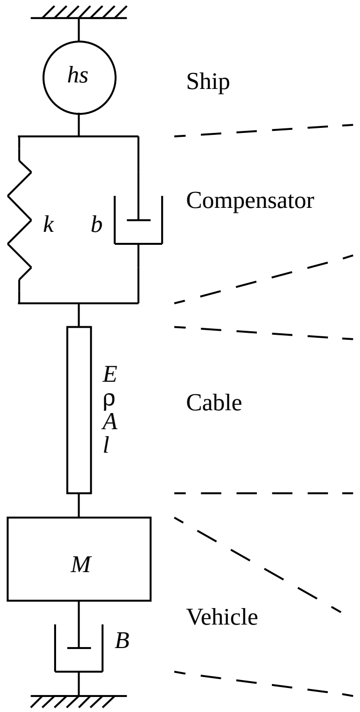
The lateral (normal to the cable axis) motions of the cable are modeled as a series of fourier spatial components added to describe the static and subsequent dynamic shape, the contributions of the components being changed in time. The dynamic analysis includes the tether mass, hydrodynamic added mass and normal drag. Hover (1990) found that 10 spatial frequency components and 30 or more cable segments yield sufficient accuracy for conditions similar to those considered here.

Table 1. Cable Parameters

Parameter	Value	Units
Diameter	17.3	mm
Modulus of Elasticity	62e9	N/m^2
Density	4400	kg/m^3
Normal Drag coefficient	1.6	-
Tangential Drag coefficient	.08	-

The axial (tangential) movement of the cable is modeled as a massless stiff spring. The result of this assumption is that the dynamic part of the tension (the deviation from the static solution) is everywhere the same. This assumption is certainly valid if the natural frequency of the cable (with one end fixed and the other free) is much faster than the excitation (heave) frequency. At shallow depths the cable natural frequency *is* much higher than the wave frequencies; but when FOSS is towed at a depth of 6000 meters at 1 m/s (requiring 8400 m of cable), the resonance period is 9 seconds, clearly in the range of possible wave periods. The effect of this simplification in the time domain model was studied with a simple, linear, one-dimensional, frequency domain model of the system (see Figure 2) in which the cable is represented as a lossless transmission line so that the effect of its distributed mass could be analyzed. This model is based on the Laplace transforms of the vehicle and compensator impedances and uses a standard reflection coefficient representation of the cable (Adler, 1960). The simplified model shows that including the distributed cable mass is not essential for the study of the FOSS system. The resonant amplitudes increase only 15-30% and the resonant frequencies decrease less than 10% with respect to the response using the massless cable in the time domain simulator, although future work could model the distributed mass of the cable for greater accuracy. In fact, the largest departure from the results of the time domain simulation are due to the linearized drag used in the frequency domain model, which supports the value of a model that accounts for the non-linear effects.

The vehicle provides the bottom boundary condition for the cable and is modeled as a body with the parameters of Table 2. The particular motions of the vehicle were not important for the purpose of this analysis so only rudimentary properties of the vehicle are modeled. No effort was made to acquire a detailed characterization of the vehicle behavior. The simplifi-



Variables and Parameters	Impedance	Velocity	Tension
s = Complex frequency h = Ship heave		hs = heave vel.	
k = Compensator spring b = Compensator damping x = Compensator stroke	$Z_{comp} = \frac{s}{k + bs}$	$xs = hs\frac{Z_{comp}}{Z_{top} + Z_{comp}}$	
v = Tether velocity at surf. t = Tension at surface	$Z_{top} = \frac{1 + \Gamma e^{-2sl/c}}{1 - \Gamma e^{-2sl/c}}$	$v = hs\frac{Z_{top}}{Z_{top} + Z_{comp}}$	$t = \frac{v}{Z_{top}}$
E = Elastic modulus ρ = Cable density A = Cable area l = Cable length	$Z_0 = \frac{1}{A\sqrt{E\rho}}$	$c = \sqrt{\frac{E}{\rho}}$ = phase vel.	
Γ = Reflection coefficient H = Vehicle heave T = Tension at vehicle	$\Gamma = \frac{Z_{veh} - Z_0}{Z_{veh} + Z_0}$	$Hs = v\frac{1 + \Gamma}{e^{sl/c} + \Gamma e^{-sl/c}}$	$T = \frac{Hs}{Z_{veh}}$
M = Vehicle mass B = Vehicle vertical drag	$Z_{veh} = \frac{1}{Ms + B}$		

Figure 2. Simplified, One-dimensional, Frequency Domain Model of Ship, Compensator, Cable and Vehicle

cations include: no righting moment, centered drag forces, centered added mass components, a longitudinally centered cable connection, and hydrodynamic added mass equal to the vehicle mass.

The motions of the ship constitute the drive for the simulation and form the boundary conditions for the top of the cable when there is no compensation system. With a compensator, the ship motions drive the base of the compensator and the dynamics of the compensator determine the cable drive. The model allows any combination of surge and heave drive, but for simplicity, we have only used heave inputs. The ship is assumed to have infinite mass in that the forces from the compensator and tow cable do not alter the motion of the ship. As a further simplification, ship pitch has only been considered in terms of its effect on heave. The FOSS system is generally towed from the stern of a vessel, and thus there is a considerable arm that adds additional *heave* due to pitch motions. However, because a typical ship from which FOSS would be deployed only pitches a few degrees in sea state 4, variations in the *orientation* of the crane compensator base and its travel limits are ignored. Of course the ship is also the source of the tow speed, but for this study a constant tow speed has been used and so speed only affects the initial static solution for the cable.

The heave compensator used for modeling is a bobbing crane, pictured in Figure 3 (showing the specific parameters used in Section 6). This crane has the following modeled features and parameters:

Table 2. Vehicle Parameters

Parameter	Value	Units
Weight in air	30250	N
Buoyant force	10234	N
Mass	3084	kg
MMOI about transverse axis	3084	kg-m^2
Longitudinal added mass	3084	kg
Vertical added mass	3084	kg
Longitudinal location of cable connection	0.0	m
Vertical location of cable connection	0.5	m
Front face area	1.55	m^2
Top face area	4.95	m^2
Drag coefficient	1.1	-

- A boom with length, mass, center of gravity, and a pivot at the base).
- Limits on the boom angle range, along with an angle of initial static balance.
- A piston attached to the boom on the line connecting the boom pivot to the overboard sheave center, and the base of the piston connected to a defined point in the plane of the crane. If the actual location of the piston top is not on the pivot-to-sheave line, it can be so modeled by rotating both the piston top and base about the boom pivot until it does coincide with this line.
- The piston has a spring rate, damping and static friction. The piston spring rate is determined from the change in fluid pressure as the piston extends and retracts. The fluid pressure is calculated using the polytropic law, $(PV)^{\alpha} = K$, where K is a constant, α is the polytropic exponent for the back-up volume gas (Azpiazu, 1983) and V is the back-up

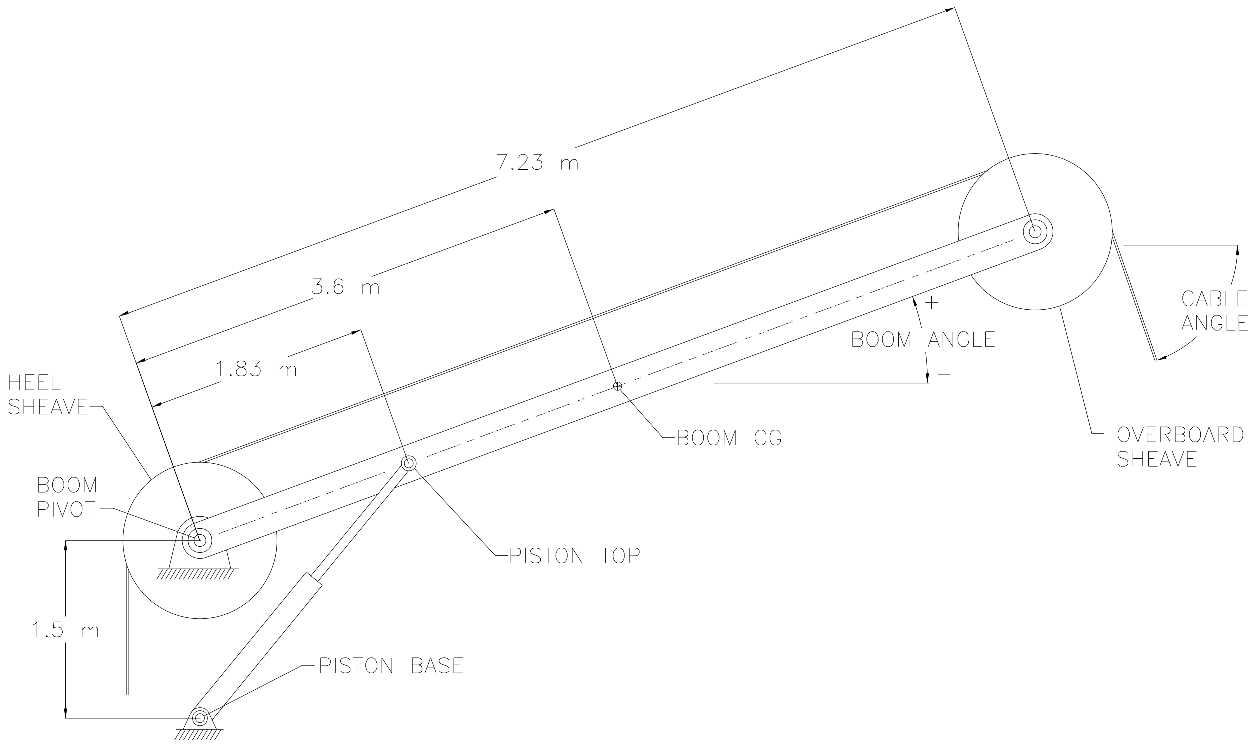


Figure 3. Features of the Model Heave Compensating Crane

volume. The ratio of gas volume to cylinder volume for a particular simulation was based on a maximum desired pressure change between piston mid-extension and minimum extension position. This was controlled with the model variable “piston force fraction” that represents the change in pressure as a fraction of the mid-extension pressure. Piston damping is proportional to velocity squared.

- A heel sheave with a diameter and a position in the plane of the crane near the base of the boom. Because axial motions of the cable are so critical to the effective spring rate seen by the cable and to the resulting heave performance, it is important for the model to properly account for the extension of the cable with boom motion that occurs if the heel sheave is *not* concentric with the boom pivot.
- An overboard sheave located at the end of the boom with a given diameter.

During the initialization of the crane model, the program outputs various useful design data on the crane and system being simulated, including: tether tension as a function of boom position (from which spring rate can be determined), piston parameters and pressures required, and system equilibrium spring rates and resonant frequencies.

The crane model uses rectangular integration for each time step, while the rest of the system is integrated by Newmark’s method as discussed by Triantafyllou (1990). Because the cable must be simulated at time steps as small as 1 to 5 milliseconds due to its rapid tension response, rectangular integration is sufficient for the crane dynamics.

4 UNCOMPENSATED BEHAVIOR

In order to evaluate the simulated heave compensator performance, the behavior of FOSS in an uncompensated mode was modeled. Figures 4a and 4b show peak vehicle heave and peak-to-peak dynamic tension (with maximum and minimum values printed adjacent to corresponding points) for 1.5 meter peak heave input at different wave periods and vehicle depths at a forward speed of zero m/s. Figures 5a and 5b show the same curves for a tow speed of 1 m/s. At long wave periods the cable/vehicle system acts as a stiff link, the vehicle follows the ship motion and the generated dynamic tensions are small for both tow speeds. With short wave periods at large depths the vehicle motion is as much as 75% less than the ship motion and the dynamic tensions are minimal. The motion attenuation results from filtering by the cable/vehicle system because the frequencies corresponding to these wave periods are higher than the cable/vehicle system natural frequency. With short wave periods at shallower depths there is a significant resonance band with amplified vehicle motion and large dynamic tensions. At depths between 500 and 2500 meters the vehicle motions with no forward speed are 50% greater than the ship motions and the peak dynamic tensions approach the weight of the vehicle. At a forward speed of 1 m/s the amplification of vehicle motion and dynamic tensions are slightly less than at no forward speed, but it is probable that the cable will go slack and snap loads will be generated if the vehicle is in this depth range in these wave conditions at any tow speed.

The reduction in vehicle motion and peak dynamic tensions as tow speed increases are evident in these results and also in the passive compensator results of section 6. As towing speed is increased the cable angle rotates away from the vertical axis so that only a fraction of the heave motion is coupled into the cable. However, fast towing should not be considered as a temporary solution in an extreme motion and dynamic loading condition without full knowledge of the ship heave *and* surge response as speed is increased.

Based on these results we have two major areas to compare with heave compensated behavior. The first is the impact of heave compensation on the resonance band with its amplification of motion at the vehicle and high dynamic tensions. The second is the lack of motion attenuation at the vehicle at longer wave periods whose corresponding frequency is less than the system natural frequency.

5 IDEAL COMPENSATOR

Before analyzing a proposed compensator for FOSS, it is instructive to consider the features of an ideal compensator for comparison with characteristics of a practical implementation. Some desirable characteristics are mutually exclusive in practice and thus inevitable trade-offs have to be made.

Ideally the cable is controlled at the surface so as to impart no motion to the vehicle at the bottom. At first this would seem to require that the surface tow point move only with uniform horizontal translation in an inertial frame. But as previously mentioned, lateral motions of the cable are rapidly damped by the relatively large normal drag on the cable; so in depths exceeding a minimum of about 100 meters, it is only necessary to provide zero axial motion of the cable. In contrast to lateral excitations, the cable readily slides

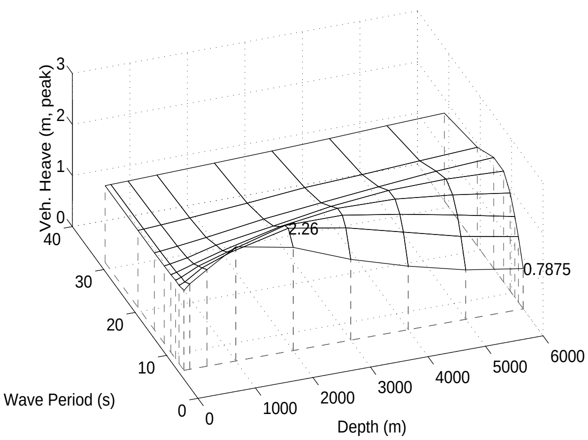


Figure 4a. Vehicle Heave

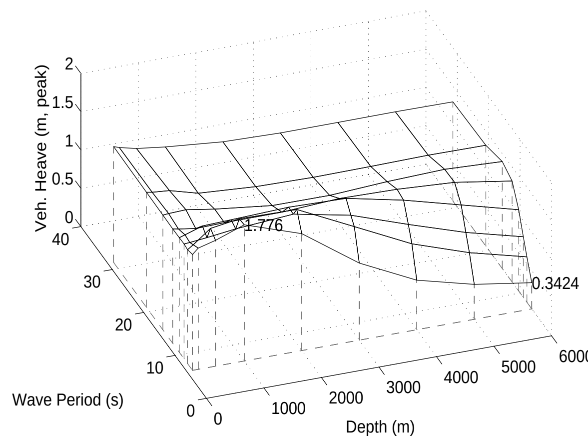


Figure 5a. Vehicle Heave

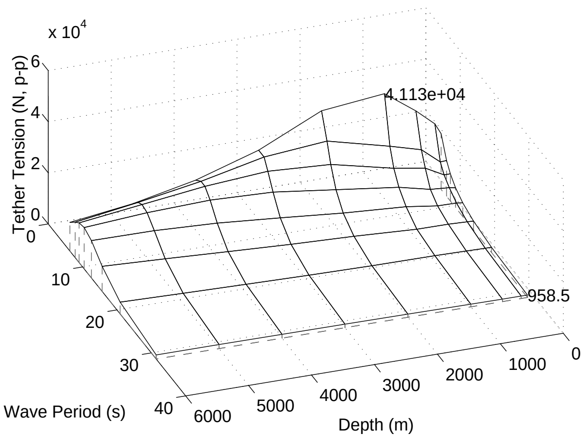


Figure 4b. Dynamic Tension
Uncompensated Response to 1.5 m Ship Heave at Zero Tow Speed

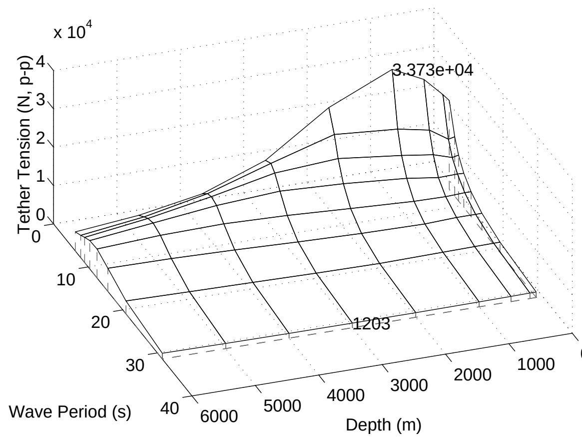


Figure 5b. Dynamic Tension
Uncompensated Response to 1.5 m Ship Heave at 1 m/s Tow Speed

axially and the vehicle is almost as well coupled to the surface in this mode as if the whole system were suspended in air. Thus the first requirement for an ideal compensator is to prevent axial motion of the cable at the tow point; this can be accomplished in practice by controlling the compensator to prevent any changes in cable tension.

An ideal compensator will not have any limits in stroke, and will have sufficient velocity and acceleration to keep up with any wave induced ship motion in the maximum sea state for which it is designed. As an example, a constant tension winch can satisfy both the constant tension and unlimited stroke goals. On the other hand, a bobbing crane or ram tensioner will have some limit to its travel and must be operated within a safe fraction of this travel to prevent a large wave from causing the compensator to reach a limit. Contact of a compensator with a stop, even if equipped with a spring or shock absorber, will cause very large tensions to be induced in the cable, defeating a primary purpose of the compensator. Simulation has shown that even an absorber acting over 1/4 of the stroke is insufficient to prevent excessive dynamic cable tensions. It is much safer to reduce compensation over the entire stroke if the goal is to reduce the likelihood of dangerous cable tensions.

Two ways to reduce compensation are to introduce a non-zero spring rate or a non-zero damping (both relative to axial cable motion) to limit compensator travel and return a controlled fraction of dynamic tension to the cable. Following the lead of Hover (1994), who used a more analytical frequency domain technique, Figures 6a, 6b and 6c summarize time domain simulations of the separate and combined effects of adding spring rate and damping to a compensator for a representative wave period (10.5 sec) and deployed depth (1000 meters). In Figure 6c, compensator travel is shown as equivalent boom motion for the compensator of Section 7. Near-zero spring or damping allows complete freedom of the compensator, and no dynamic tension or vehicle motion. As either spring rate or damping reach a maximum, the compensator is effectively locked, and dynamic tension and vehicle motion are equivalent to when the there is no compensator. Note that substantial reductions in vehicle motion occur only for spring rates below 300 N/m and damping below 300 Ns/m. At intermediate values, there may be a spring rate that resonates with the vehicle mass, producing compensator travel, dynamic tension and vehicle motion all in excess of the no compensation case. In contrast, increase in damping does not produce such severe phenomena, and thus damping is the preferred control method. Figure 6b shows a hump in tension with damping near 100 kNs/m; this behavior is *not* predicted by the simplified model of Figure 2 and its origin has not yet been determined.

The adverse behavior with spring rate is due simply to the resonance of the compensator spring with the cable/vehicle mass. This resonance exists at a similar frequency and Q regardless of whether the system is modeled with a simple spring/mass, with a springy but

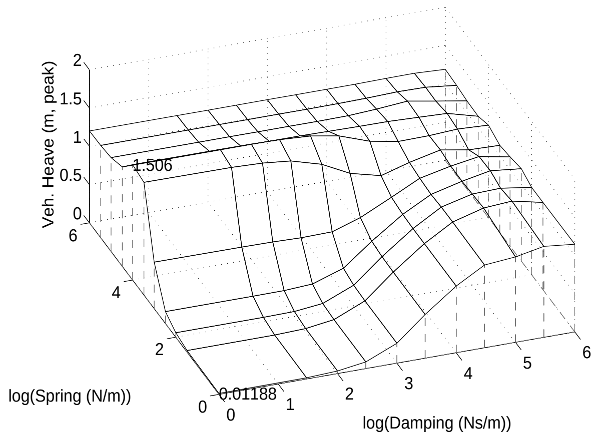


Figure 6a. Vehicle Heave

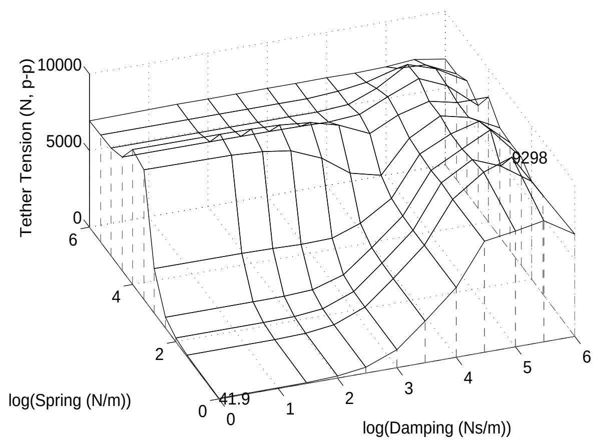


Figure 6b. Dynamic Tension

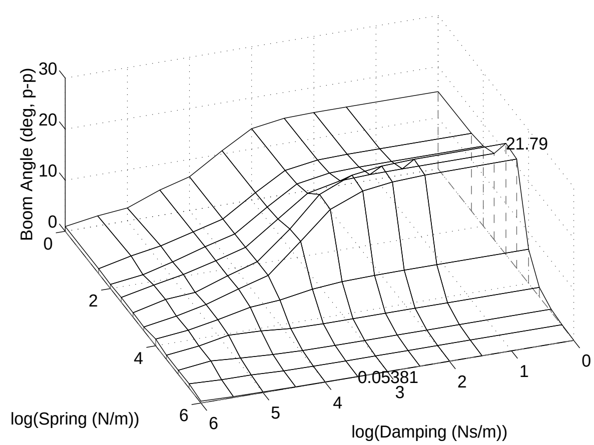


Figure 6c. Boom Travel
Ideal Compensation of Crane Showing Response to Various Spring Rates and Damping with 1 m Ship Heave at 1000 m Depth and 10.5 sec Wave Period

massless cable, or with a cable having distributed mass (as analyzed with the simple model). However, as would be expected, the Q of the resonance is very dependent on the damping in the system, particularly the vehicle vertical drag; thus not all systems may exhibit this behavior.

Another important requirement for a compensator is to avoid cyclic bending of the cable under tension. This is most important for systems that require continuous compensation at stable depths, which would otherwise result in fatiguing the same spot on the cable. Even if the strength member survives, the electrical conductors or optical fibers could be damaged. A bobbing crane compensator with the heel sheave located at the boom pivot and a fixed overboard sheave (in the plane of the crane) subjects the cable (in three locations) to one bending cycle (wrapping and unwrapping of the cable on and off a sheave with boom angle changes) for each wave. Systems that deploy and retrieve cable to provide compensation, such as a ram tensioner, constant tension winch and bobbing crane with sheave not at the boom pivot, may subject the cable to two or more bending cycles per wave, and in some systems this occurs on each of many more sheaves.

The compensator must be able to adjust to intentional changes in the static load due to changes in vehicle depth or tow speed, or due to movement of the vehicle through the air-water interface. For a constant tension winch, depth changes could be accommodated by gradual changes in the tension set point so that the cable is hauled in or payed out without exceeding the winch's speed range, thus preserving compensation throughout the maneuver. For a ram tensioner or bobbing crane, compensator adjustments must be coordinated with winch orders to avoid exceeding the stroke limits.

A related requirement of a limited stroke compensator is to maintain its average operating position and not drift toward a limit. A (conceptually) simple way to do this is to maintain a low spring rate (in the cable axial direction) so that compensator force drops when excess cable is retrieved and vice versa. The effect of this strategy is discussed further in Section 7.

Finally, a compensator should meet the obvious goals of requiring the least possible input power, demonstrating high reliability particularly with regard to cable damage or failure, and being safe to operate. Safety requires special attention due to the large energy involved in deep ocean systems. Special hazards involved are large moving equipment on deck, high tension cables, and large energy storage systems, particularly compressed gas in some implementations. Safety must be evaluated for both normal operations and failure scenarios.

6 PASSIVE HEAVE COMPENSATION WITH A BOBBING CRANE

As part of the passive heave compensation analysis, different crane parameters were evaluated in order to optimize the design for FOSS. The final crane design was selected based on both compensation performance and achievable design parameters. Table 3 shows the final crane parameters used in the model.

Table 3. Crane Model Parameters

Parameter	Value	Units
Boom rotation:		
below horizontal	20	degrees
above horizontal	70	degrees
Boom length	7.23	m
Boom mass	2700	kg
Boom inertia	52800	kg-m^2
Boom center of gravity from pivot (1/2 boom length)	3.6	m
Piston pivot on boom from boom pivot (1/4 boom length)	1.83	m
Piston base location:		
vertically below pivot	1.5	m
horizontally aft of pivot	0.0	m
Piston force fraction (see Section 3)	0.15	-
Piston damping	50000	Ns^2/m^2
Piston static friction	1000	N
Polytropic exponent (α) for air	1.44	-
Inboard sheave location:	centered at boom pivot	
Sheave diameters	1.27	m

Figure 7 shows a proposed crane design for the FOSS vehicle that is slightly different from the model configuration shown in Figure 3. The model allows a

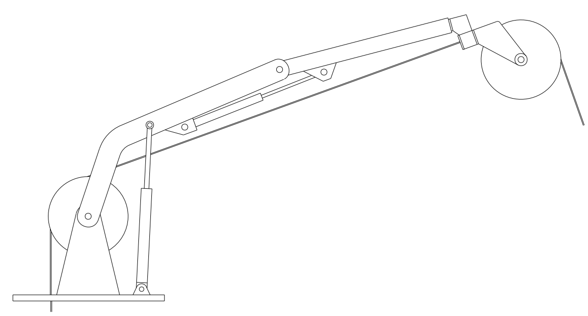


Figure 7. FOSS Bobbing Crane Configuration

straight boom that may be positioned without concern for the cable path, but a real crane requires a forked and curved boom that is clear of the cable and sheaves over the entire rotation range. The change in boom shape for the actual crane also requires a shift in the main piston end positions. Both piston ends have been rotated a fixed angle about the boom pivot position. They remain the same distance from the boom pivot, and the piston extension and induced torque on the boom versus boom rotation are the same as in the model configuration. The proposed crane has other features, such as a hinged boom and slewing capability, for deck operations.

Based on the uncompensated behavior of FOSS, the two main needs of the heave compensation system are elimination of the resonance band between 500 and 2500 meters at short wave periods and overall reduction in vehicle motion. Figures 8a, [illegible] and [illegible]

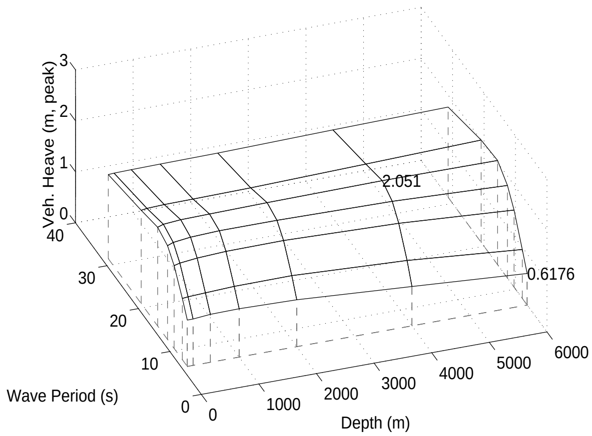


Figure 8a. Vehicle Heave

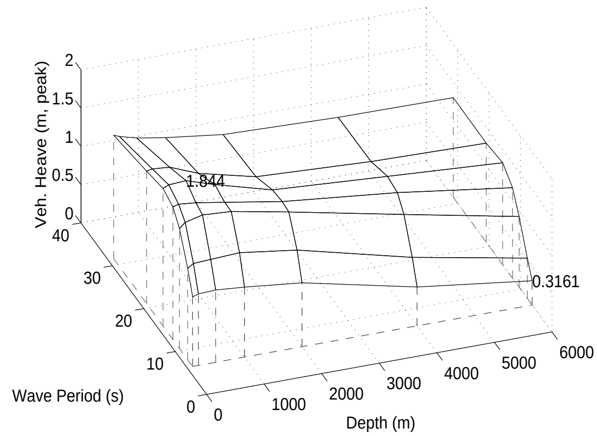


Figure 9a. Vehicle Heave

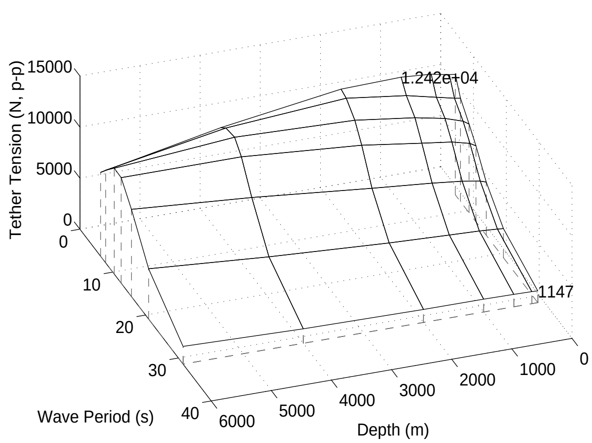


Figure 8b. Dynamic Tension

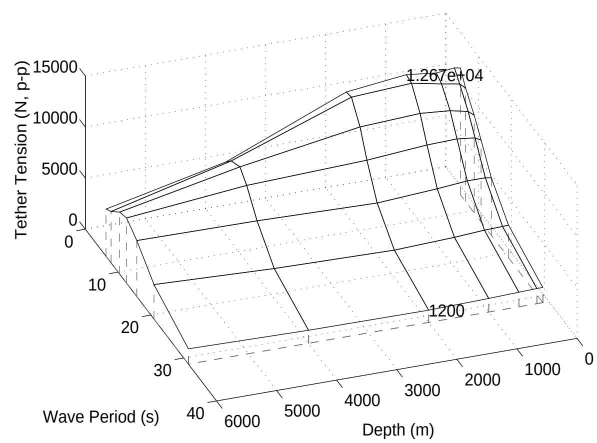


Figure 9b. Dynamic Tension

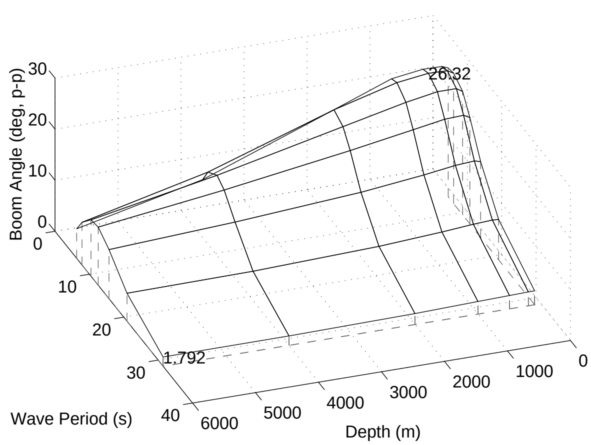


Figure 8c. Boom Travel
Passive Compensated Response to 1.5 m Ship Heave at Zero Tow Speed

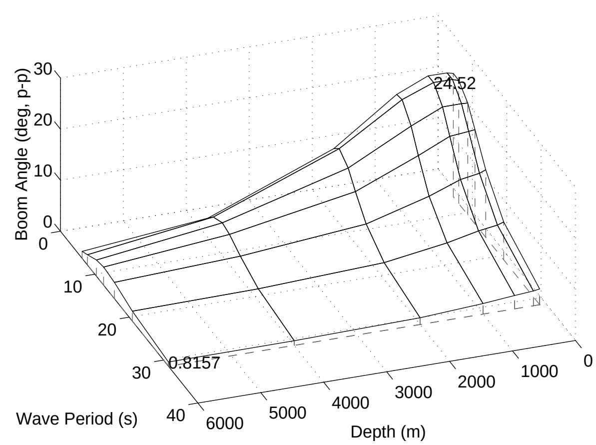


Figure 9c. Boom Travel
Passive Compensated Response to 1.5 m Ship Heave at 1 m/s Tow Speed

show vehicle heave, dynamic tension and boom rotation with heave compensation at various wave periods and depths for the zero forward speed condition. Figures 9a, b and c show the same curves at a tow speed of 1 m/s. In both cases the high dynamic tensions and amplified vehicle motions in the resonance band have been reduced significantly. The maximum peak-to-peak dynamic tension is 30% of the uncompensated maximum value. The maximum uncompensated vehicle heave (1.5 times the ship motion) that accompanied the high dynamic tensions is reduced by more than 50%.

At longer wave periods the bobbing crane compensator is less effective. The typical uncompensated behavior is low dynamic tensions and vehicle motion equivalent to ship motion. With heave compensation there is little change in dynamic tension and a slight increase in vehicle motion. In fact, at a tow speed of 1 m/s the maximum compensated vehicle heave (1.85 m peak) is greater than the maximum uncompensated vehicle heave (1.78 m peak). This increase is attributed to a slight shift in the resonance band with the addition of the crane to the cable/vehicle system. The crane compensator does not reduce the vehicle motions in longer waves because the dynamic tensions are not great enough to rotate the boom. Without boom rotation there is no motion compensation. In these conditions there is little risk of a cable failure, but video imaging and sonar surveying of the sea floor may be degraded.

6.1 Crane Spring Rate

A typical curve of effective crane spring rate versus boom angle from horizontal is shown in Figure 10.

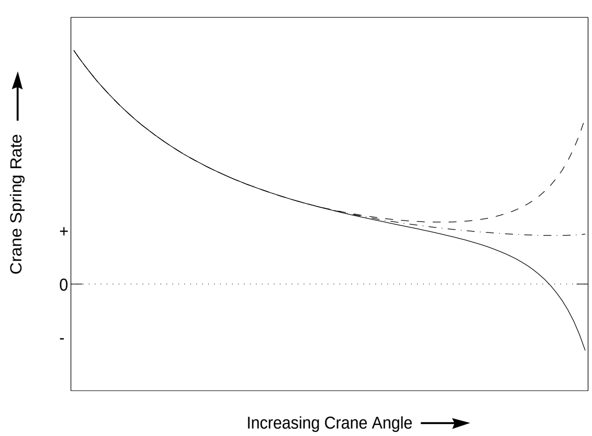


Figure 10. Typical Crane Spring Rate Referenced to Cable Axial Direction

The spring rate is defined as the additional cable tension that generates one meter of motion in the axial direction of the cable, and is calculated for incremental motions. The general shape is created by the non-linear geometric relationships that change the net torque on the boom applied by both the cable and the piston and that change the fraction of motion aligned with the cable axis as the boom rotates. It is characterized by a sharp gradient in spring rate at low boom angles, a relatively flat spring rate at middle boom angles, and a sensitive spring rate at high boom angles that may sharply increase or sharply decrease with possible negative spring rates.

The sharp gradient at low boom angles is generated by increasing piston torque, decreasing cable induced torque and minimal motion along the cable axis. Piston torque rises because of increases in both cylinder pressure and the component of piston torque perpendicular to the boom as boom angle decreases. Cable induced torque is a maximum when the boom is perpendicular to the towing cable. As the boom rotates in either direction away from this angle, both the component of tension perpendicular to the boom and the moment arm decrease. Large increases in tension are necessary to balance the increasing piston torque. In addition, the component of overboard sheave motion in the axial cable direction is greatest when the boom and tow cable are perpendicular; this component also decreases as the boom rotates away from this angle.

The spring rate curve becomes flatter when the angle between the boom and the tow cable approaches 90°. In this condition the rate of change of cable induced torque and the component of boom rotation along the cable axis are reduced.

At high boom angles the spring rate curve is very sensitive to the geometry. Small components of the piston force and cable tension contribute to torque on the boom and the component of crane motion along the cable axis approaches a minimum value. Small shifts in geometry may require large changes in cable tension to balance the boom. It is these conditions that generate the sharp gradients at the high boom angles. A condition that must be avoided is a negative spring rate. If the angle between the tow cable and the boom becomes small (<30°, but also dependent on piston angle) the geometry requires *increases* in cable tension to maintain a torque balance. This unstable condition results in the crane being forced against its upper limit.

The other compensation systems mentioned previously do not have the geometry problems that exist with the bobbing crane. Ram tensioners remain parallel to the tow cable or drill pipe. Instead of dealing with torques, as in the bobbing crane, these compensators react to forces that are aligned with the direction of compensating motion.

The bobbing crane compensator spring rate curve shifts vertically and horizontally depending on towing conditions and crane parameters. The impact of balance position, piston position and fluid system variables on crane spring rate were studied. The piston position is constant once the design is finalized and therefore must be optimized for all towing conditions. Both balance position and working pressure are variables that can be changed to minimize stiffness as towing conditions change. Damping, dependent on friction and restriction of flow, received minor attention during the passive analyses. Using estimates of minimum achievable damping in a crane system, the stiffness dominates compensator performance.

6.2 Crane Balance Position

During heave compensation operations, the crane bobs up and down about a balance position. This position is dependent on the nominal piston force and the static cable and boom loads. Ideally, a balance position that generates a spring rate curve of minimum value should be selected. A second important criteria is maintaining adequate margin from a boom rotation limit. Figure 11 shows four spring rate curves for dif-

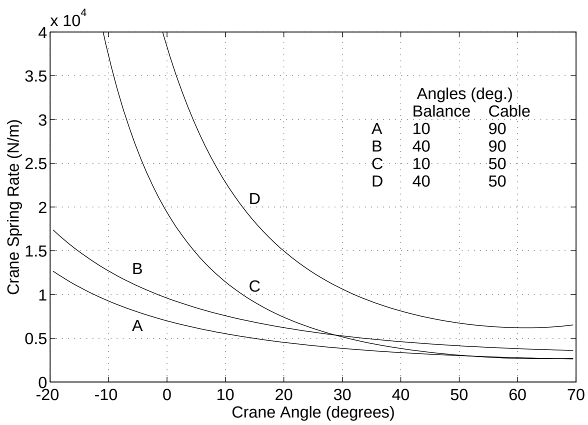


Figure 11. Effect on Spring Rate of Balance Position and Cable Towing Angle (Depth = 3000 m)

ferent combinations of two balance angles and two cable angles (tow speed dependent) at a depth of 3000 meters. The values, 10° and 40°, define the range of boom angles used for the balance position. This provides a rotational margin of 30° to the limits within the boom rotation range of 20° below to 70° above the horizontal. However, it should be noted that the effective compensation distance produced by any given boom rotation is dependent on the change along the cable axis not the actual distance that the overboard sheave moves.

Figure 11 indicates that the minimum spring rate value and flattest spring rate curve are achieved at the upper limit of the balance angle range, 40°. The effect of balance angle on spring rate curve is most evident for a towing angle of 50°. The spring rate value is 12 kN/m when balanced at 10° and 8 kN/m when balanced at 40°. The effect of balance position is reduced when FOSS is suspended vertically rather than towed at a forward speed. The spring rate value decreases only 1 kN/m, from 5.5 kN/m at 10° to 4.5 kN/m at 40°. This is attributed to the angular relationship of the piston and the cable. If the piston and the cable are parallel (as in the ram tensioner), the spring rate gradient becomes dependent on fluid system (see section 6.4) rather than geometry and a very flat curve is achievable. With the bobbing crane configuration this is not feasible, but it is evident from all four curves in Figure 11 that the slope of the spring rate curve decreases as the piston becomes more aligned with the cable. This happens at much lower boom angles when FOSS is suspended vertically rather than towed at a forward speed. It should also be noted that these realistic spring rate values are significantly higher than the desirable range of <300 N/m discussed for an ideal compensator in section 5.

For the results presented in Figures 8a, [illegible], [illegible], [illegible], [illegible] and [illegible], the balance position was set at 10° for cable angles (from horizontal) greater than 80°, perpendicular to the cable for angles between 50° and 80° and at 40° for cable angles less than 50°. Slightly better results would be possible for the zero forward speed, low wave height condition if the balance angle were increased, but the effective upward compensation distance for this condition when balanced at 40° is less than 3 meters. With possible peak heave motions of over 3 meters in worst sea state 4 conditions, it will be necessary to lower the boom balance angle. Maintaining the optimum balance position as sea state and towing conditions change is important in ensuring that the crane does not reach a limit. The crane should be outfitted with sensors that indicate cable tension, towing angle and boom rotation. With these inputs automatic control of balance position is achievable.

6.3 Piston Position

Positioning of the boom piston is critical in the bobbing crane design. The required piston force for balancing the boom and the motion of the piston as the boom rotates are dependent on the angular relationship between the boom and the piston. This is set by the piston end positions relative to the boom pivot point. Spring rate at high boom angles is particularly sensitive to piston position. Shifting the horizontal position of the piston base less than 0.2 m may shift the spring rate curve gradient from sharply negative to sharply positive. A position which ensures that the negative spring rates do not exist in the boom rotation range is necessary and must be verified at all towing angles. Accomplishing this results in slightly higher spring rates over the entire operating range.

6.4 Fluid System Variables

The critical variables in the fluid system are the balance pressure and gas volume. These values are bounded to some extent by equipment capabilities and the need to vary them with changes in towing conditions. The major components of the fluid system include the main piston (may actually be two pistons located on the sides of the boom for cable clearance), the accumulator, the accumulator back-up volume, the compressor, and the vent. These components are shown in Figure 12.

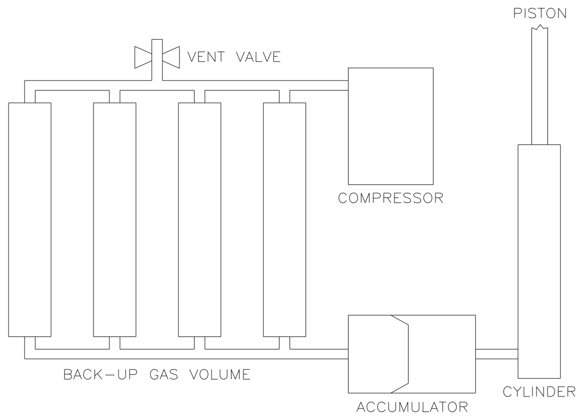


Figure 12. Typical Fluid system Components

The balance pressure is the pressure required to position the boom at the balance angle perpendicular (or approximately, see Section 6.2) to the towing angle. It is primarily dependent on the static cable tension and the towing angle. In our design we have ensured that the balance pressure plus pressure increases from compensation remain under 20 MPa (3000 psi). The balance pressure at three depths and three towing speeds are shown in Table 4. Changing

Table 4. Balance Pressure for Various Conditions

Depth (m)	Tow Speed (m/s)	Pressure (MPa)
100	0.0	5.45
100	.5	5.46
100	1.0	5.49
3000	0.0	9.08
3000	0.5	9.71
3000	1.0	13.29
6000	0.0	12.81
6000	0.5	14.03
6000	1.0	18.87

balance pressure is accomplished with the compressor and vent valve connected to the back-up gas volume. The effect of the balance pressure on compensator spring rate is shown in Figure 13 at three depths and a constant cable angle. The spring rate is highest for the deep system because of the higher static load and required higher balance pressure. Since the change in system pressure is a fraction of the balance pressure, it is higher when the balance pressure is higher. The increased system pressure results in greater changes in piston torque for a given piston motion, which in turn must be offset with a greater change in cable tension for the boom to rotate. Without the cable tension changes, the boom does not rotate and compensator performance is minimal.

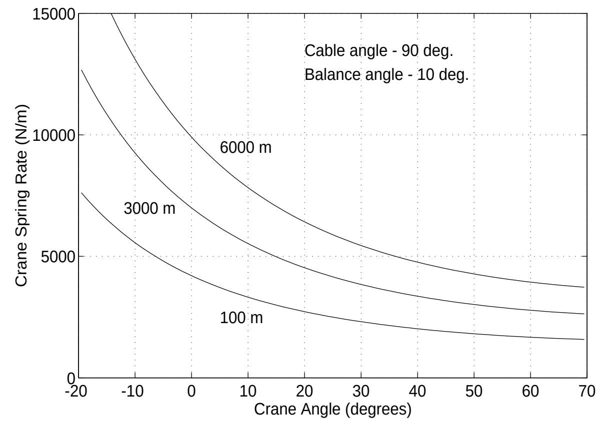


Figure 13. Effect on Spring Rate of Balance Pressure

In addition to the balance pressure and geometric factors, the spring rate is dependent on the ratio of the cylinder volume to the back-up gas volume. If the cylinder volume is small relative to the back-up volume there is less change in pressure and therefore a flatter spring rate curve. However, the benefit from increasing the back-up gas volume reaches a point of diminishing returns because the spring rate is primarily governed by geometry. In addition, there are practical limitations on the back-up volume such as storage space and the need to change system pressure. With a slewing crane, the back-up volume should be located on the crane platform, and the compressor and vent must be sized to allow the system pressure to balance changes in static load during deployment and retrieval at rates up to 100 meters/minute. The gas volume in the analyzed design is 0.3 m^3, five times the cylinder volume; this corresponds to a ±15% change in pressure from the piston mid-extension value.

6.5 Other Factors

The dependence of compensator performance on balance pressure generated some thought into reducing its required magnitude. Two possible techniques are balancing the boom and reducing the weight of the vehicle. An unbalanced boom generates a torque on the boom based on its mass, center of gravity and boom angle. If this is offset, then the balance pressure may be reduced. The boom mass torque represents a greater portion of the total torque at shallow depths (40%) than at greater depths (17% max) where it has little effect on crane spring rate. A secondary benefit of a balanced boom is reduced ship induced boom movement not directly caused by changes in cable tension. However, with the limited effect at greater depths, a desire to limit crane size and a requirement that the crane slew during towing operations, a balanced boom will not be pursued for the FOSS system.

Reducing the vehicle weight will also reduce the balance pressure. Like a balanced boom the lighter vehicle has more impact on spring rate near the surface where it generates most of the cable tension. An important concern when reducing the vehicle weight is the weight to drag ratio. It is critical that the vehicle terminal velocity be greater than maximum ship vertical component of velocity, especially if the system will be used without a compensator. Lightening the vehicle for compensated operations and then ballasting for uncompensated operations is one method of dealing with this issue.

Another variable that effects bobbing crane compensator performance is boom length. A longer boom generates more boom motion per unit of rotation, thus equivalent compensation motion is accomplished with a reduced boom rotation range. Reducing the boom rotation range allows the piston position to be shifted to generate lower crane spring rates without the risk of negative spring rates. However, crane and deck strength requirements limit changes in boom length and thus achievable reductions of spring rate.

The passive bobbing crane compensator is effective at minimizing vehicle motion and dynamic tensions present in resonance conditions, but will not reduce vehicle motions that occur simultaneously with low dynamic tensions because of a high spring rate that is inherent to the bobbing crane geometry.

7 ACTIVE COMPENSATOR

In order to further improve the performance of the compensator, it is necessary to gain more control over the compensator motion in response to cable tension. What is needed is a means to directly regulate the tension in the cable as a way to overcome the geometric limitations that govern the effective spring rate of the passive bobbing crane. If the cable tension is measured, this information can be used in a feedback system to actively control the boom position in order to zero the tension variations. Such a system can produce very small motions of the vehicle at the end of the cable, but cannot directly maintain the boom in a desired mean position nor can it be easily "de-tuned" to limit boom travel in high sea state conditions.

If the cable angle leaving the overboard sheave and the boom angle are also measured, they can be used together to compute the linear motion of the cable in the direction of the cable axis. Knowledge of the cable motion, together with the tension, allows the boom to be controlled to produce any desired cable tension as a function of axial cable "length" and speed, thus creating an artificial spring and/or damper of any value. If a low synthetic spring rate of 100 to 200 N/m of axial tether motion is created, it will keep the boom centered about a desired angle, while still allowing substantial reduction of vehicle motion at all depths. This performance is shown in Figures 14a, 14b and 14c for a spring rate of 100 N/m and zero damping (the active compensator achieves zero damping by overcoming any inherent real damping in other system components). Heave at the vehicle is shown to be less than 0.2 meters amplitude for a 1.5 meter surface amplitude with all wave periods shorter than 16 seconds, cable tension varies only a few hundred Newtons peak-to-peak, and boom travel does not exceed 35° peak-to-peak. Greater reductions in heave can be accomplished through the use of lower spring rates, if adequate centering of the boom can be maintained.

When high sea states are encountered, boom travel must be limited at the expense of compensator performance. Use of square law damping avoids the resonances associated with high spring rates (see Section 5) and simultaneously avoids excessive penalties during intervals of moderate waves. Figures 15a, 15b and 15c show the response to the same heave as Figures 14a, 14b and 14c with the synthetic damping increased from zero to 3000 Ns^2/m^2 and the spring rate at 200 N/m, values suitable for boom control with significant heave of 3 meters amplitude (6 meters crest to trough). Heave performance degrades to 0.7 meters at the vehicle for 16 second and faster waves, while tensions rise to 10 kN peak-to-peak for 6 second waves and boom motions are reduced to 20° peak-to-peak. With 3 meter heave shown in Figures 16a, 16b and 16c, vehicle heave is about 1.6 meters in 16 second waves, and tension is less than 20 kN peak-to-peak and boom travel less than 40°, both in 6 second waves.

The power required by an active compensator is dependent on the implementation of the system. In the limiting case, the cable does not move axially and the only power input is that to accelerate the mass of the boom as it rotates about its pivot, and even this requires zero mean power. However, it is still necessary to support the static tension of the cable. If the tension is supported by the same device that provides active compensation (e.g. a winch or the main cylinder of a bobbing crane), the power can be very high indeed. For example, consider the case of 2.5 meter amplitude seas with a 1 rad/sec wave frequency (6 second period); the peak winch rate is 2.5 m/s and, with FOSS and 6000 m of cable supported, the static tension is 67 kN for a *peak* power of 167 kW (225 hp). Assuming no energy recovery during payout, the *average* power over a cycle is about 50 kW (70 hp). Energy recovery with a traditional hydraulic system would be virtually nil, and even with an electric system, assuming 80% efficiency in each direction, only 64% of the energy could be reused each cycle.

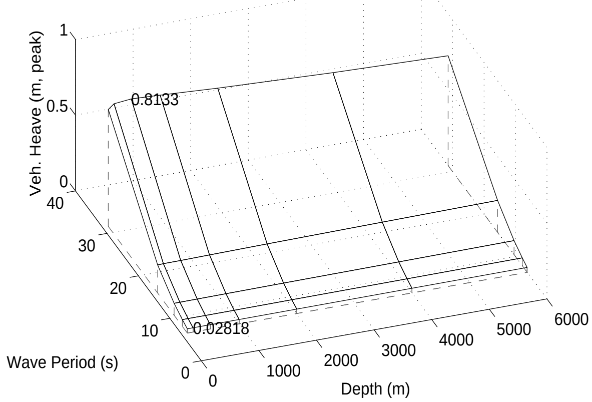


Figure 14a. Vehicle Heave

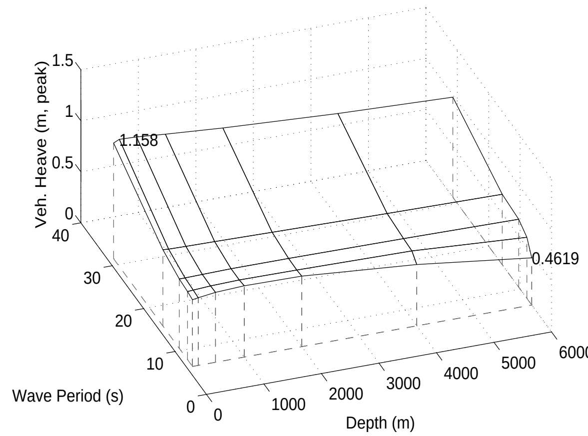


Figure 15a. Vehicle Heave

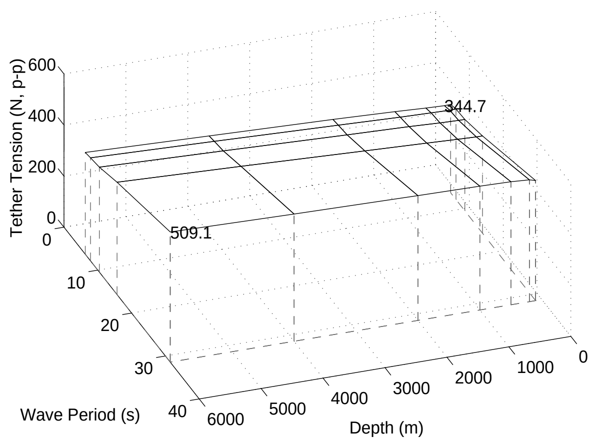


Figure 14b. Dynamic Tension

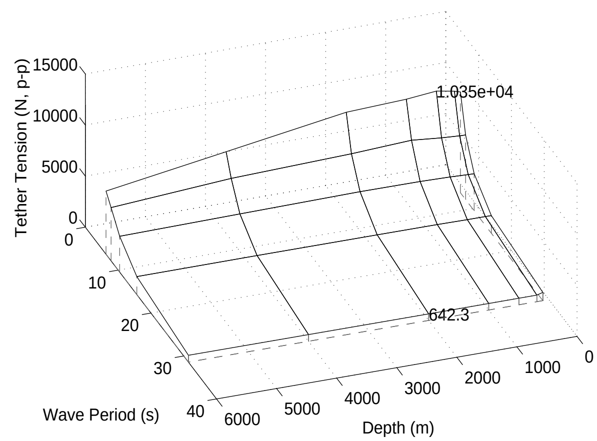


Figure 15b. Dynamic Tension

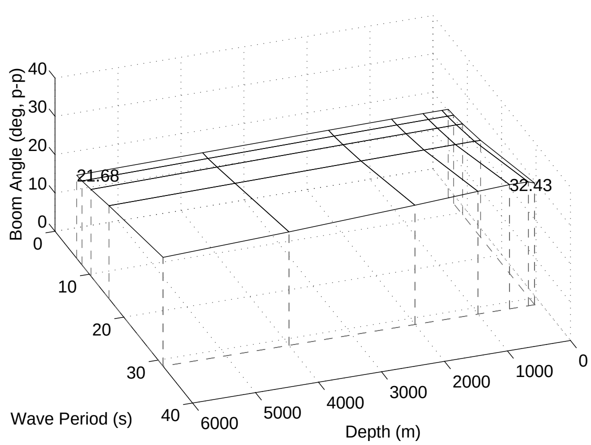


Figure 14c. Boom Travel
Active Compensated Response to 1.5 m Ship Heave at Zero Tow Speed with Spring Rate of 100 N/m and Zero Damping

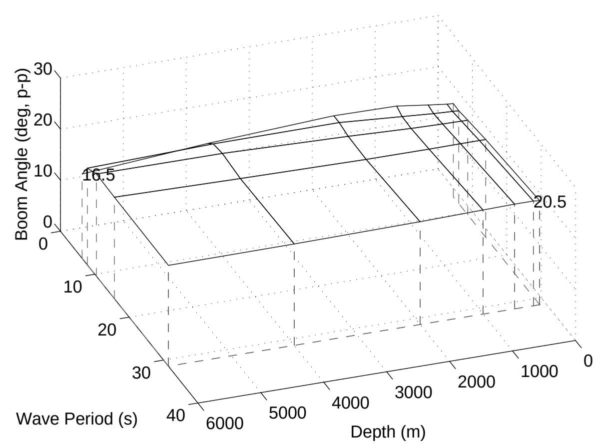


Figure 15c. Boom Travel
Active Compensated Response to 1.5 m Ship Heave at Zero Tow Speed with Spring Rate of 200 N/m and Damping of 3000 N/(m/s)^2

This power can be considerably reduced by using the passive compensation system discussed in Section 6 to support the static load, and adding a parallel active system just to modify the dynamics of the crane while the passive portion stores and returns the bulk of the energy. If the active portion is implemented as a typical hydraulic servo system, with a servo valve throttling oil from a constant pressure source to the two sides of a piston, the power consumed equals the flow rate times the system pressure, regardless of whether the piston is actually doing work or not. Under this assumption, and using the passive crane of Section 6 and the 2.5 m wave condition, the peak power for active compensation is 20 to 40 kW (depending on how much excess system pressure is assumed) and the average power is 12 to 25 kW. For comparison, the real peak power (force times velocity of the piston) is only 10 kW and the average (allowing for 100% recovery of returned energy) is only 1 kW. Thus this active system could be implemented with a practical electrical actuator, although a hydraulic system may be better suited to the environment on the fantail of a ship.

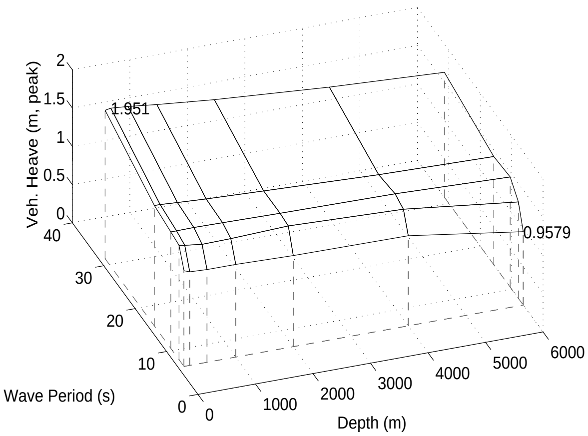


Figure 16a. Vehicle Heave

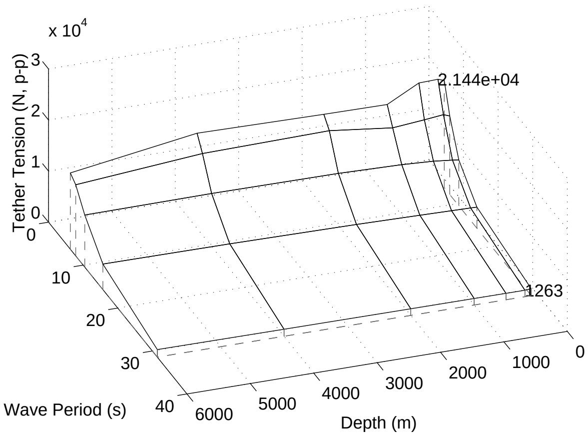


Figure 16b. Dynamic Tension

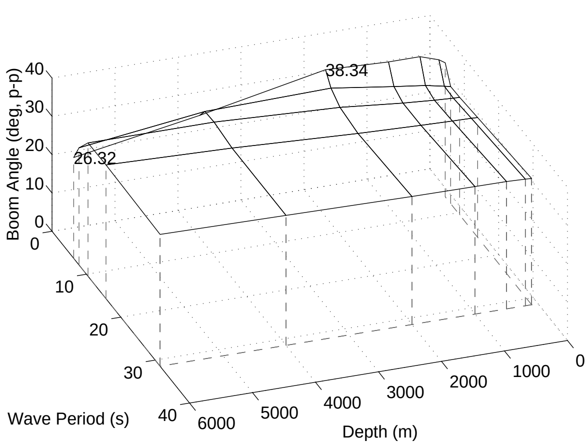


Figure 16c. Boom Travel
Active Compensated Response to 3 m Ship Heave at Zero Tow Speed with Spring Rate of 200 N/m and Damping of 3000 N/(m/s)^2

It is interesting to note that the active power required by such a system, when set for damping of 3000 Ns^2/m^2 in the same 2.5 m sea, is *less* than for the unconstrained case. This occurs because most of the power is required to overcome the overly stiff spring that the passive system provides, and when restrained, the boom moves less and thus lower passive system forces and travels are encountered. The power for a hydraulic implementation drops by about a factor of two, while the peak real power stays about the same. The real average power actually becomes negative at about -2.5 kW, representing the power absorbed by a damper.

8 SUMMARY

A bobbing crane heave compensator has been selected for FOSS based on its reduced cyclic bending of the cable compared to competing implementations. The principal defect of the passive version of this compensator is its geometrically limited effective spring rate. The spring rate can be minimized by careful design of the boom and piston geometry. The resulting high minimum spring rate has been shown to significantly reduce the maximum dynamic tensions that can be excited in the tow cable under certain depth and wave conditions, but it does not allow the vehicle heave motions to be reduced to the extent desirable for best sonar and camera imaging.

Adding an active drive system to the passive crane will allow a significant reduction in vehicle heave for a modest expenditure of energy. Furthermore, the system can be designed so that failure of the active system would generally reduce the system to its passive mode, rather than cause total failure.

The first implementation of the FOSS compensator will be a passive only crane, because it *can* reduce the risk of cable failure due to excessive cable tension. This crane will be constructed with appropriate features to allow the active drive components to be added in the future if operational experience indicates that an active system is desirable.

ACKNOWLEDGMENTS

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