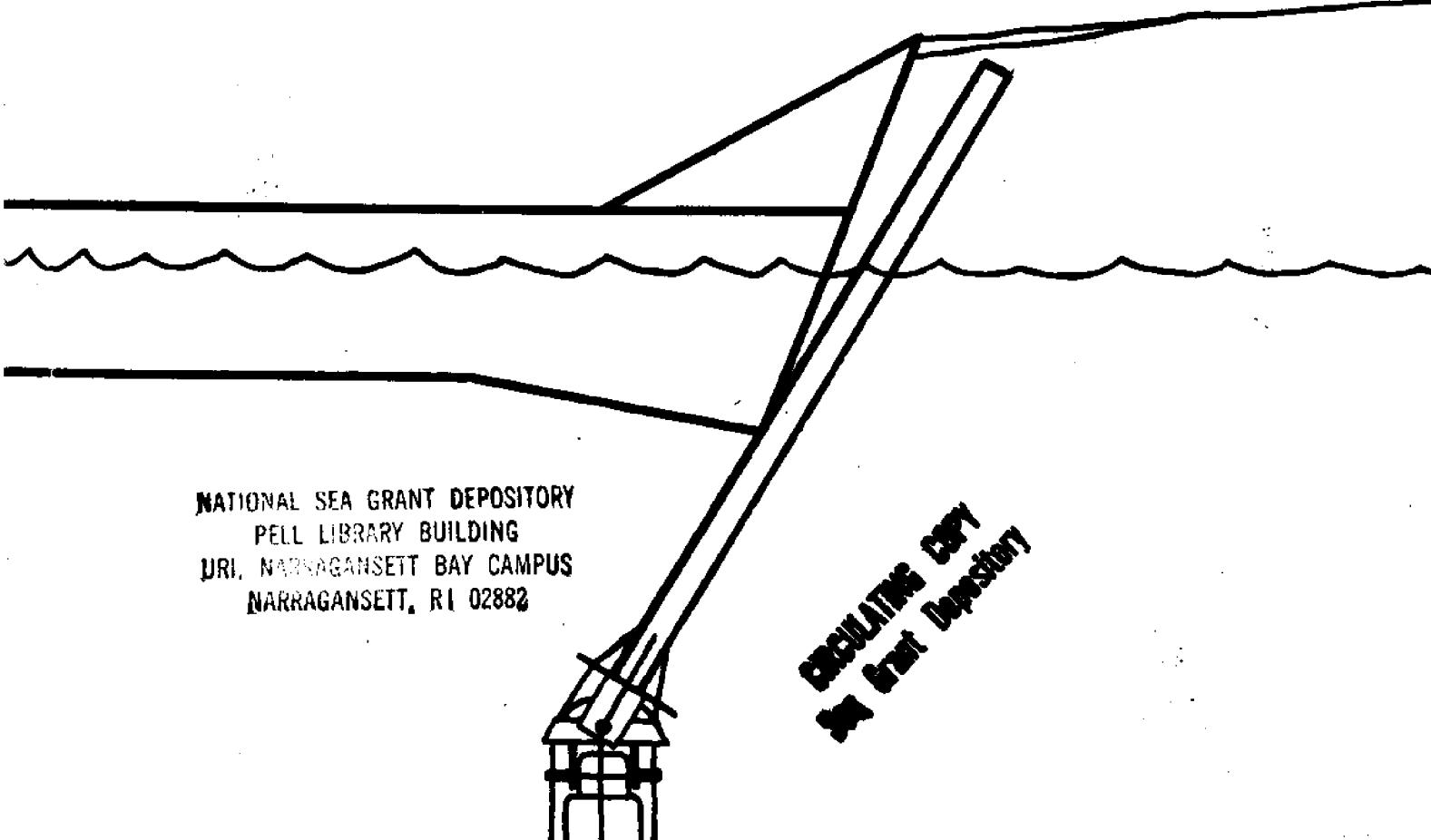


THE DESIGN, DEVELOPMENT, AND TESTING
OF A QUARTER SCALE
COLLISION TOLERANT PILE STRUCTURE

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A REPORT FOR:

TECH 697

OCEAN PROJECTS 1984-85
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ABSTRACT

The Collision Tolerant Pile Structure student project was an extension of a Coast Guard sponsored project conducted at the University of New Hampshire. The Coast Guard sponsored project investigated the problem of damage to rigid pile navigation markers from collisions with towed barges. Basic research, including preliminary design concepts, computer simulations, and small scale models, was conducted during the summer of 1984 to develop a compliant pile navigation marker.

In this project the preliminary research and design was used as a basis for the design, development, and field testing of a quarter scale Collision Tolerant Pile Structure (CTPS). This work included computer analysis, working drawings, fabrication, and installation of the CTPS at a location in the Great Bay Estuary. The CTPS was taken out of the laboratory and into the field under actual conditions where actual collision testing was performed.

ACKNOWLEDGEMENTS

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I. INTRODUCTION

BACKGROUND

Rigid pile structures currently used by the Coast Guard to support navigation markers in shallow waterways are susceptible to collision by towed barges. The expense involved in replacing damaged markers could possibly be avoided by replacing the existing rigid supports with a compliant Collision Tolerant Pile Structure (CTPS). The CTPS is a single pile, hinged at the mud line, on which navigational markers can be mounted. This concept was developed in a previous Coast Guard sponsored project using design requirements stated in Swift and Baldwin (1985) and established by the Coast Guard. These requirements are summarized in Tables 1 and 2. That preliminary concept was tested under laboratory conditions and served as the basis for this student project.

APPROACH

The project goal was to design, construct, and field test a one quarter scale CTPS. Using previously developed computer simulations, design parameters were found for the quarter scale model. Several spring and hinge options were then developed and analyzed. From the design analysis a concept was selected, working drawings were completed, the necessary materials were acquired, and the quarter scale CTPS fabricated. Upon completion of construction the CTPS was tested out of water for stiffness characteristics. The CTPS was then placed in the Great Bay Estuary for the field experiment. This culminated in a collision test where a barge was towed directly over the pile. Finally, a computer simulation of pile verticality response was made using measured hinge stiffness characteristics.

Table 1. Verticality and Collision Requirements-Full Scale.

1) Verticality	5 degrees (design goal)
	10 degrees (allowed)
2) Daymarkers	2 at 36 sq. ft., 7 ft. above MHW
3) Barge Speed	10 Kts. maximum

Table 2. Environmental Condition Requirements-Full Scale.

Condition	Design Goal	Minimum Allowance
Water depth (max)	30 ft.	20 ft.
(min)	10 ft.	15 ft.
Current	3 Kts.	2 Kts.
Wind gusts	60 Kts.	60 Kts.
sustained	50 Kts.	40 Kts.
Wave Height	5 ft.	4 ft.
period	3-5 sec.	3-4 sec.
Bottom Slope	15 degrees	15 degrees
Bottom Composition	Soft Clay	Soft Clay

II. CPTS DESIGN

Before the CPTS components could be designed an understanding was needed of the forces to be encountered. This began with the scaling of the prototype to achieve similitude. Once scaled parameters were obtained it was possible to simulate the dynamics of the model using the computer programs developed by Swift and Baldwin (1985). Chapter II of their report describing the computer models is included in appendix 1. These gave an estimate of what was to be encountered in the quarter scale model. Next a detailed design approach for a spring concept and material selection could begin.

DIMENSIONAL ANALYSIS

The design of the one quarter scale model CPTS requires different scaling factors for dimensionally nonsimilar parameters. The geometric parameters are proportional to L_r ($L_r = \text{length of model}/\text{length of prototype} = 1/4$). Areas are then proportional to L_r^2 , volumes to L_r^3 . Because gravitational and inertial forces were crucial to the pile dynamics Froude scaling was used to scale velocity and time. See Swift and Baldwin (1985). The Froude number

$$\text{Eq. 1) Froude number} = \frac{U}{\sqrt{L_r g}} = \left(\frac{\text{inertial forces}}{\text{gravitational forces}} \right)^{1/2}$$

yields velocity and time as proportional to $L_r^{1/2}$. Forces are proportional to L_r^3 and moments to L_r^4 .

COMPUTER SIMULATION

The computer simulations address the different design criteria while

sharing some basic assumptions. The CTPS was treated as a point hinge-rigid beam system similar to an inverted pendulum. The hinge develops the restoring moment necessary to maintain near vertical operating conditions. The dynamic equation for the CTPS is the time rate of change of angular momentum applied to the hinge.

$$\text{Eq. 2)} \quad I_h \dot{\Theta}_h = \sum M_h$$

The moments taken into consideration result from gravity, wind, current, and wave action.

To have the CTPS meet Coast Guard criteria the pile inclination angle must be small (5 degree goal, 10 degree maximum) under severe operating conditions. This requires a high initial hinge stiffness. Under collision conditions the CTPS must be compliant enough to fold down and not sustain damage. A lower value of spring stiffness at larger angles will accomplish this. With these two conditions under consideration a piecewise linear hinge moment was selected. These characteristics are used in the computer simulations and are shown in Figure 1.

To simulate this behavior a single spring with a high preload is used. The preload is determined from the K_1 value calculated by the computer simulation PILESTIFF. The breakpoint angle of ten degrees is used with the hinge stiffness K_1 and hinge moment arm to find the preload.

In theory with all pile components treated as ideal(rigid) this method would produce linear hinge moment characteristics. An infinitesimal inclination would then result in a moment preload as shown in Figure 2. In practice the pile will travel a finite angle before

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THEORETICAL HINGE STIFFNESS (K1 AND K2)

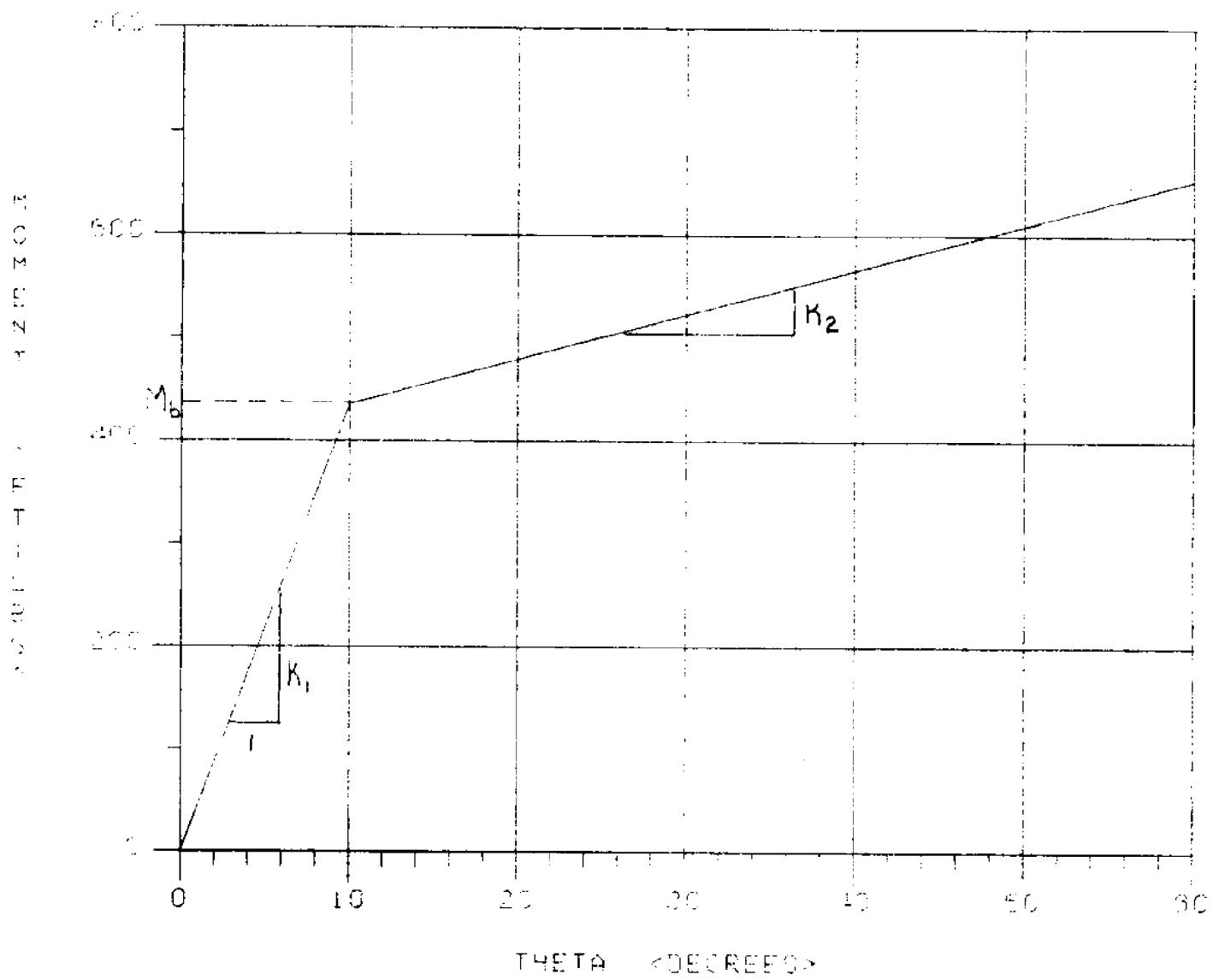


Figure 1. Hinge moment vs. angle indicating piece-wise linear concept.

θ = 30° - 35°

THEORETICAL HINGE STIFFNESS

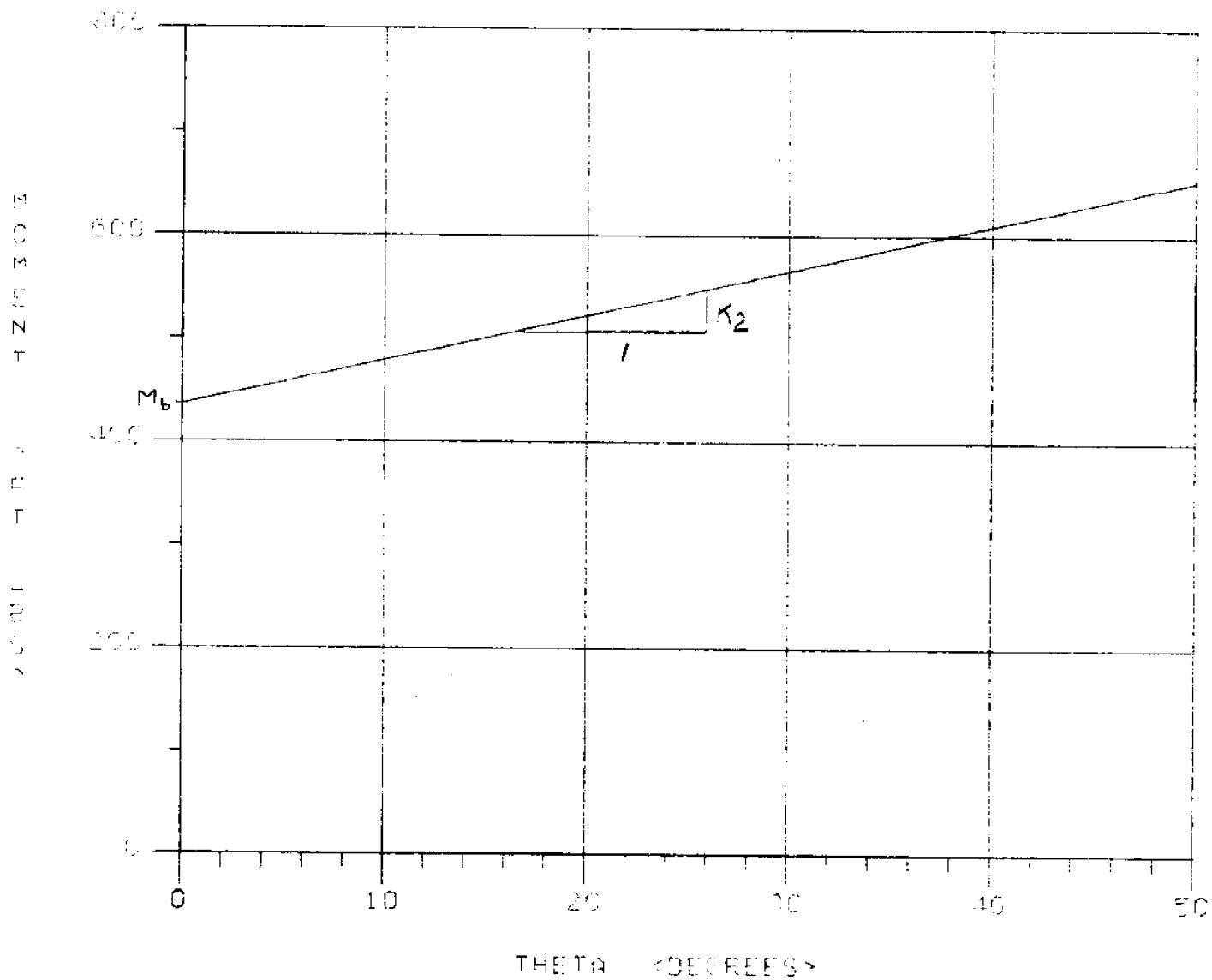


Figure 2. Hinge moment vs. angle with all components assumed rigid.

reaching this moment thus indicating the piecewise linear hinge moment shown in Figure 1. See Swift and Baldwin (1985).

Computer simulation PILESTIFF was used in the initial design phase to calculate the desired hinge stiffness moment K_1 . This value was used in the next phase of design when spring characteristics were needed. To select the proper materials for the model forces on the structure had to be obtained. Collision conditions were simulated by the program COPILE. This simulation resulted in estimated collision forces and moments. These, along with the hinge stiffness results allowed material strength requirements to be formulated. As an estimate the large angle stiffness value, K_2 , was taken as one tenth of K_1 .

Once the CTPS was designed PILESTIFF was run using the actual model weights and dimensions to obtain a new, more realistic design value of K_1 . Operating conditions were simulated using the program OPPILE. This simulation was used to determine whether the model CTPS designs would meet the verticality criteria under worst case weather conditions. Hurricane conditions are simulated by program HURPILE. Under hurricane conditions day markers are assumed to be sacrificed. Wind and wave conditions are different from those used in OPPILE while other conditions are the same. The main difference between HURPILE and OPPILE is the pile is not restricted to small angle conditions for hurricanes.

HINGE DESIGN

Design requirements for the hinge include omnidirectional motion, freedom to move ninety degrees off vertical, and restoration to a vertical position.

One design considered was the grooved ball and socket hinge shown in

Figure 3. This hinge is omnidirectional and is capable of moving ninety degrees off vertical by moving into the nearest groove opposite the direction of impact. By running stays from a spring inside the pile, over the ball and terminating them on the socket a moment arm equal to the radius of the ball was created. Thus, a restoring moment was created equaling the spring force times the moment arm. This design was abandoned though, when it appeared that friction and environmental fouling might create problems and building a quarter scale model would not be practical without further testing on a smaller scale.

Another design considered was the flat plate, central stay hinge shown in Figure 4. This hinge was omnidirectional, since it could ride on the collars in any direction. It was also free to travel ninety degrees off vertical and created a restoring moment by the force of a central spring and the moment arm created by the collars. This design was also abandoned though, when it appeared that fiction, environmental fouling and jamming could be problems. It also appeared that it would not be practical to build a quarter scale model without further testing on a smaller scale.

A third design considered was the double axis universal hinge shown in Figure 5. This design was omnidirectional and capable of traveling ninety degrees off vertical, as were the others. The spreader arms, which are in the axis planes, were added to created moment arms. By running stays from a central spring, out over the spreader arms and terminating them on a base plate a restoring moment equal to the spring force times the spreader arm distance was created. It was also necessary to add sheaves under the spreader arms to maintain a constant moment arm at any angle of inclination. Since testing of a small scale version of

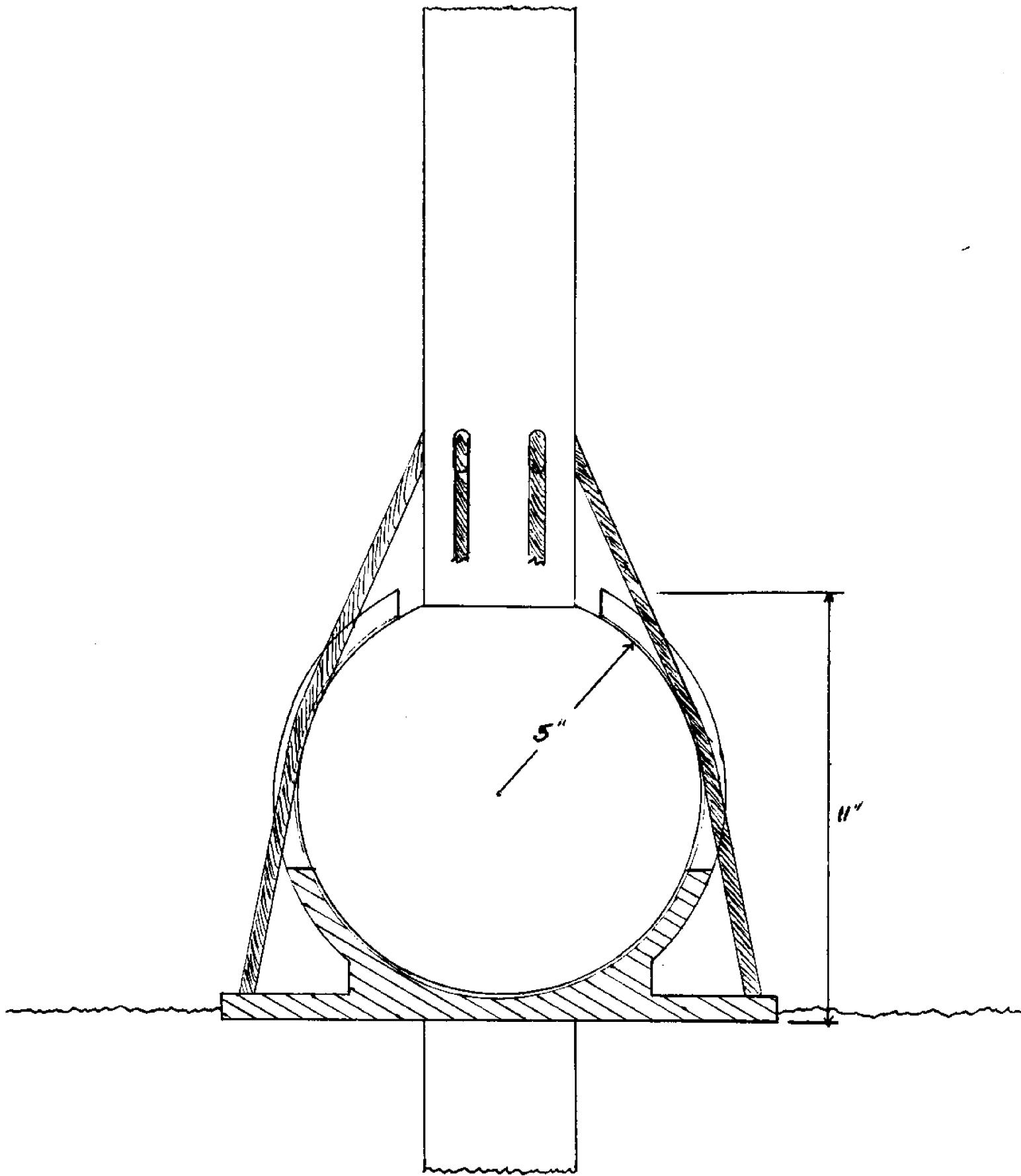


Figure 3. Ball and socket hinge concept.

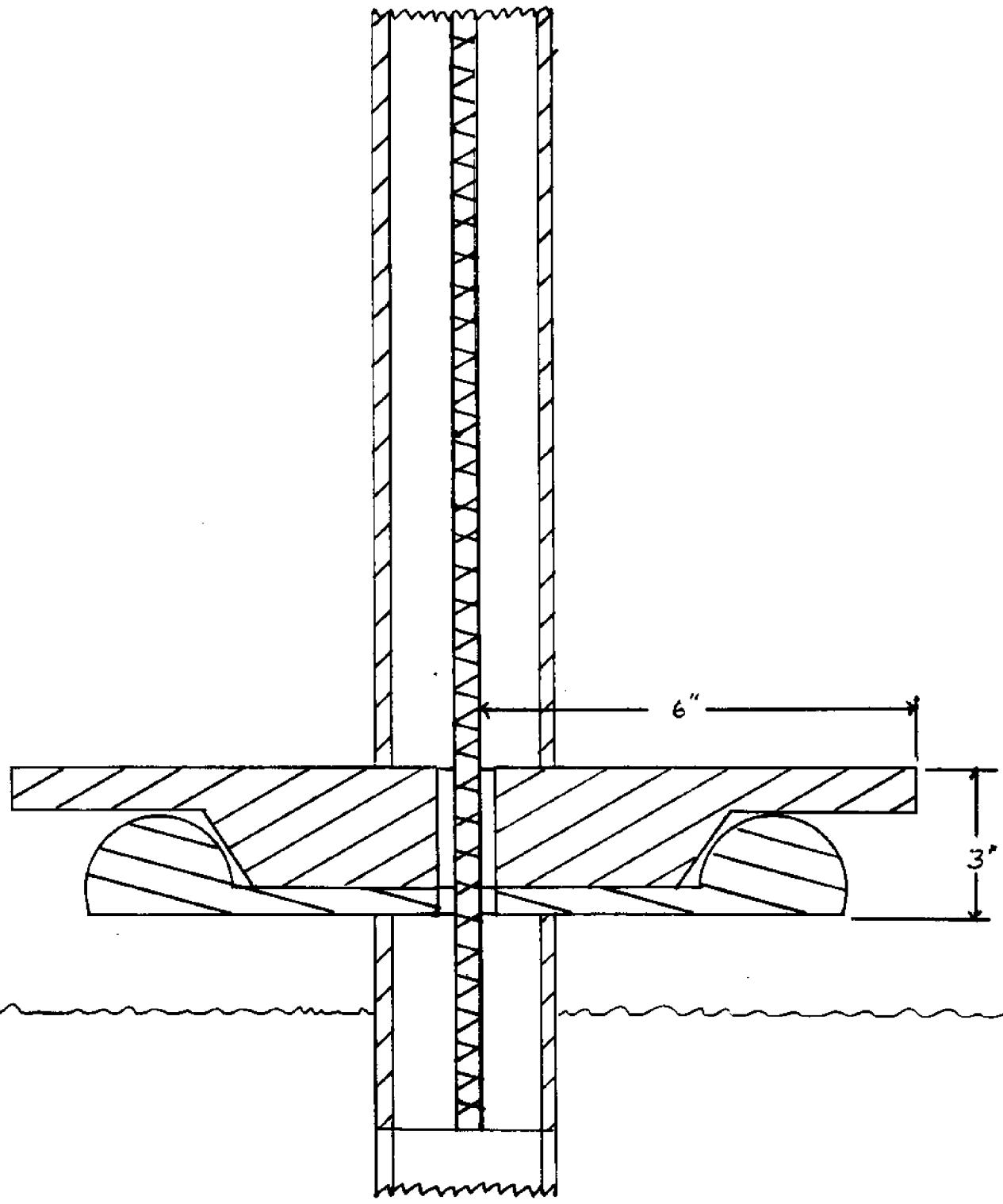


Figure 4. Flat plate central stay hinge concept.

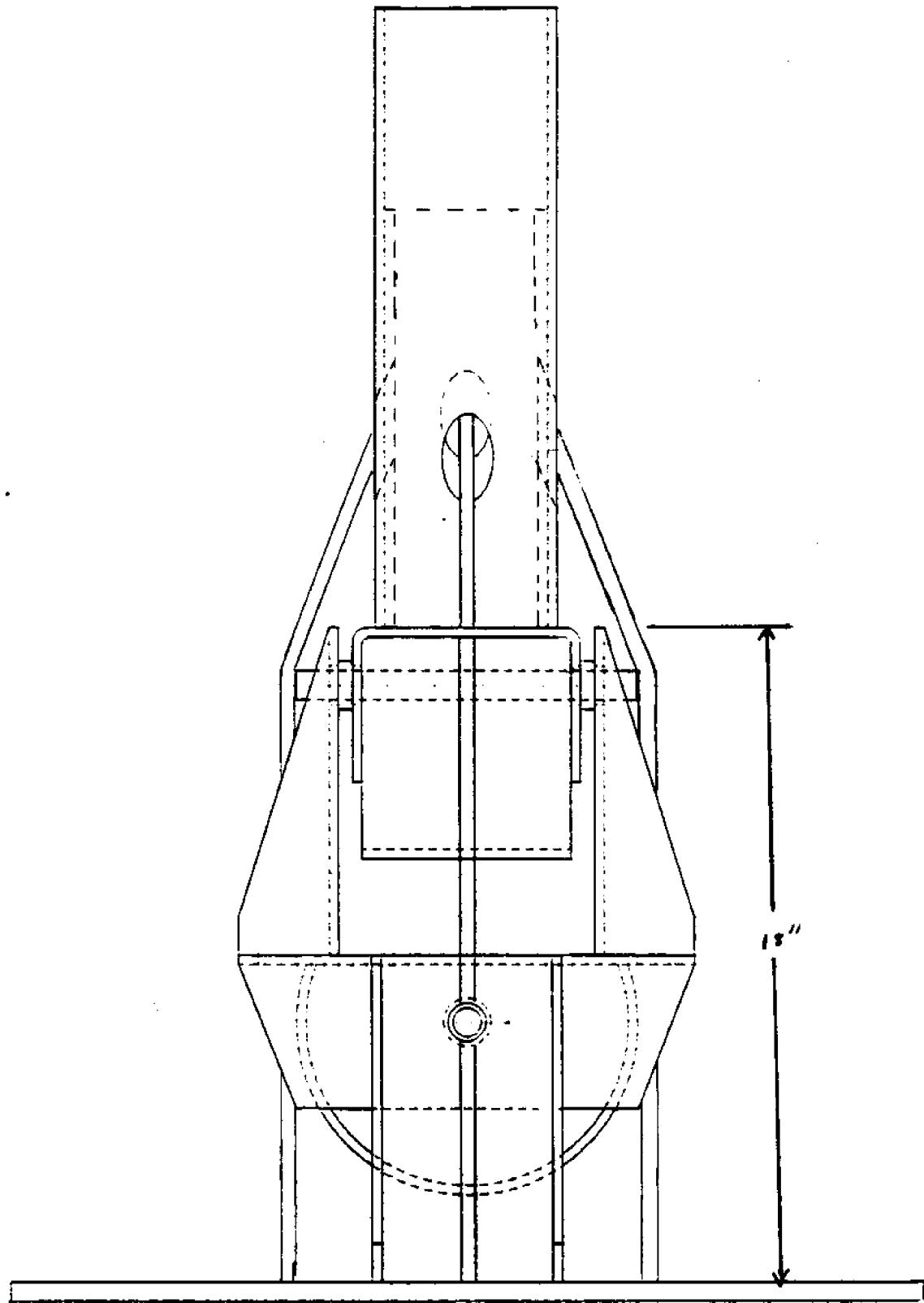


Figure 5. Universal joint hinge concept.

this design had already been completed by Swift and Baldwin (1985) with successful results it appeared that this would be the best design to use on a quarter scale model).

Fabrication

The hinge was designed to be built with available structural steel components. The major parts were a 2 foot square base plate, several feet of 5 inch square pipe for uprights and stay spreaders, a foot of 9 inch diameter pipe for hinge sheaves, and a foot and a half of 4.5 inch diameter pipe to locate the pile on the hinge. A beam deflection calculation showed that the hinge uprights would not deflect under the preload or the barge impact. The hinge pins were made from stainless steel to avoid corrosion problems on the bearing surfaces. They were made .75 inches in diameter, to eliminate any flexing which might cause the hinge to bind.

Friction at the stay/hinge contact points was found to be the major problem on the small scale model of the hinge. On examination of the hinge, five friction points were found (refer to Figure 6):

- 1) The stay guide collar inside the pile.
- 2) The holes where the stays exited the pile.
- 3) The stay holes at the ends of the spreader arms.
- 4) The sheave surfaces when the pile was off vertical.
- 5) The stay holes through the ends of the axis pins.

To reduce friction in the spreader arm and axis pin holes, anodized teflon impregnated aluminum bearings were designed as shown in Figure 7. Friction at the holes where the stays exited the pile was eliminated by placing the stay guide collar precisely so the stays would rest on the collar and spreader arm bearings without touching the edges of the pile

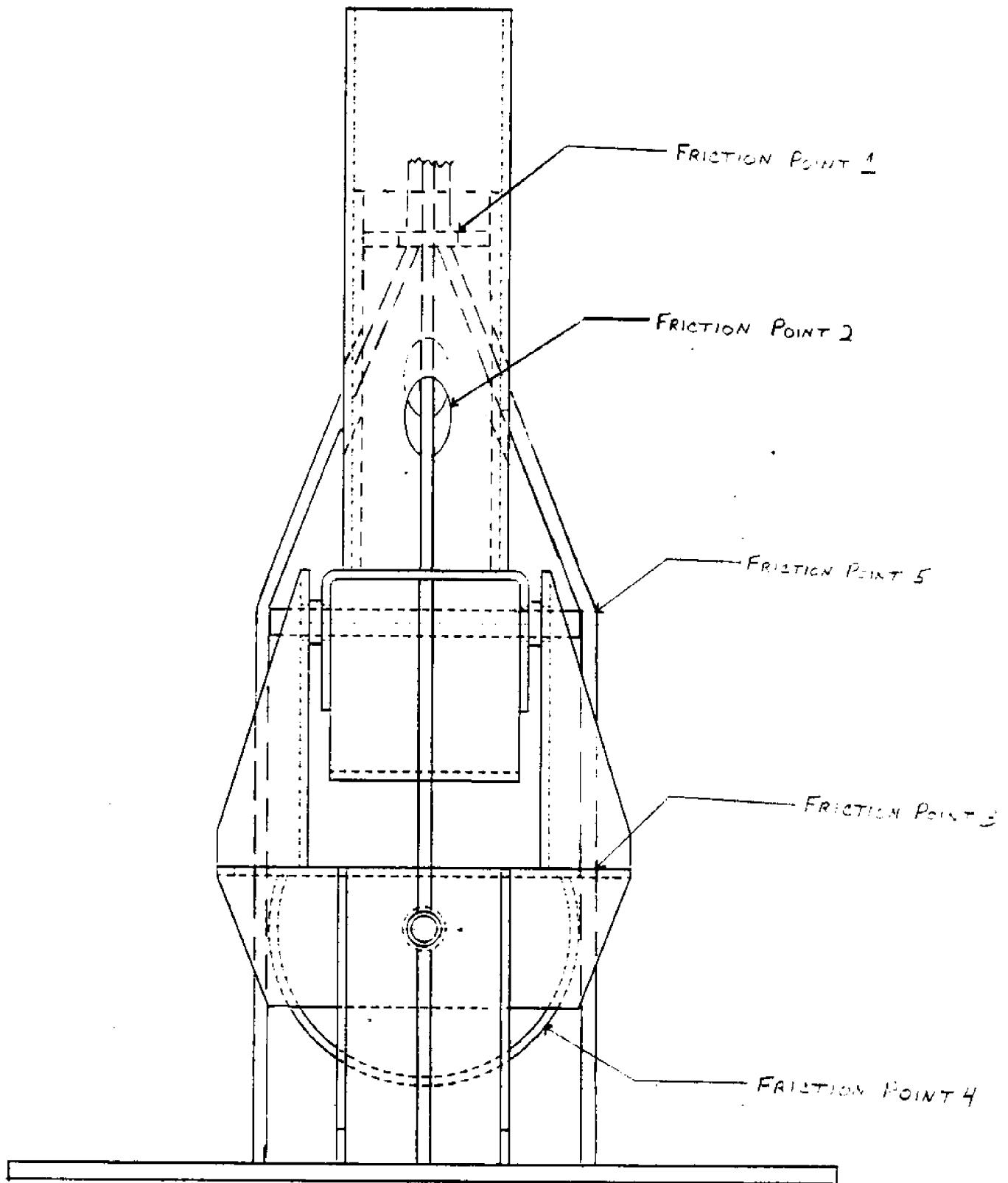
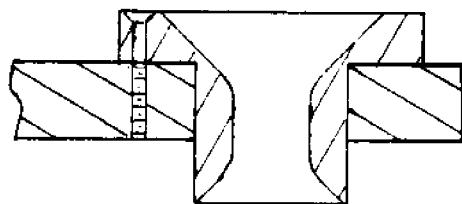


Figure 6. Known friction points in universal joint hinge.



STAY SPREADER CABLE GUIDES

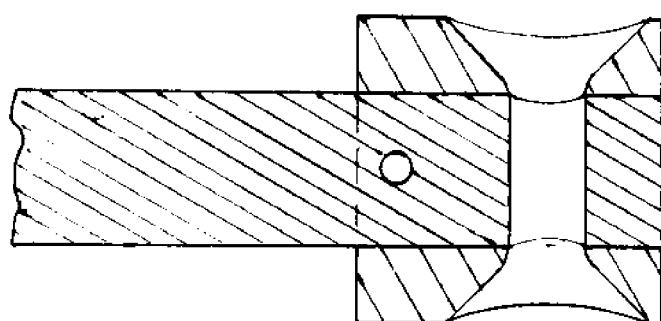


Figure 7. HINGE PIN CABLE GUIDES

exit holes. No easy solution was seen for the friction problem at the stay guide collar or on the sheave surfaces. It was felt that the best solution to these friction problems would be to have a smooth surfaced stay material, which would act as a bearing surface.

SPRING DESIGN

Computer model results showed that the spring component must meet certain performance criteria in order for the entire CTPS to meet the design criteria. In order for the CTPS to meet the verticality requirements, the initial stiffness constant K_1 must be approximately 2500 ft-lbs/rad (or equivalently, a 1200 lb spring preload). The spring must also be able to maintain this preload for 5 years and create a large angle stiffness constant K_2 which will provide the restoring moment for pile recovery after a collision. Also, the spring must be able to sustain large impulse loads resulting from collisions and continuous loads resulting from current, waves, and wind.

Several spring concepts were considered for the pile. Those having the potential for meeting the requirements are identified. Compression spring concepts considered were: solid rubber, a coil spring, and an air bag system. Rubber in tension was considered.

COMPRESSION SPRINGS

One benefit of using a compression spring for the CTPS is the ability to eliminate compressive forces on the pile. A compression spring could be designed as an integral part of the hinge thereby restricting the loads to this segment of the CTPS. Despite this the various concepts for compressive springs were eliminated for use in our

model but are mentioned here for further reference.

Solid rubber was considered using two different approaches. A rubber column with a smaller outside diameter than the inside of the pile was rejected due to instability. This could possibly be avoided by using many shorter columns in series. This approach was rejected due to the uncertainty of uniformly compressing each section and difficulty posed in fabrication. Steel coil springs were also investigated. It would have been necessary to make a special single spring having the characteristics required. Stock items could have been used in series to meet the criteria. Both ideas were rejected due to excessive weight and potential stability problems.

The final compression concept considered was an air bag system (see Figure 8). This concept has the practical advantage of convenient testing of the spring preload after the CTPS is installed. This means that a service vessel could come alongside, measure the pressure in the system, and if necessary inflate to maintain the proper preload. This system was rejected for our model for several reasons. Existing devices were not designed to be stacked as would be required for our purposes. Most importantly there was no air bag in production that would fit into our scale model. Also, cost was beyond budget constraints.

TENSION SPRINGS

The two concepts considered for a tension spring used rubber. The same material, Natsyn 45 from Delford Industries in New York, would be used in each case. One method used a single strand terminated at each end. This would have required a strand with a minimum diameter of three inches. Several methods for making terminations were examined including:

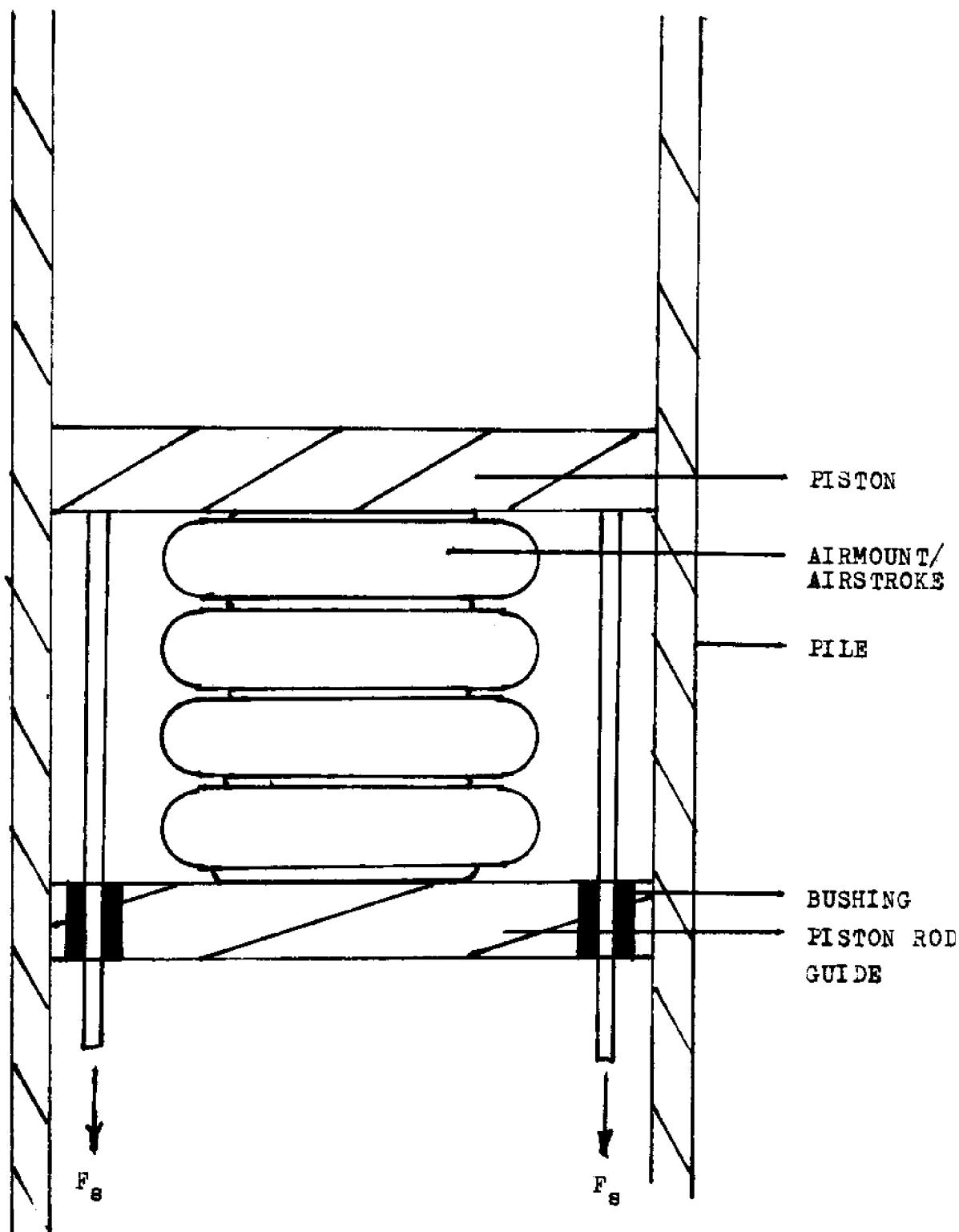


FIGURE 8. Air Bag System Concept

thimbles, Kelim grips, and one which would involve drilling into the end of the strand and inserting a stay with a ball on its end. None of these terminations were feasible so the single strand concept was dropped.

The other concept involving tension was to use multiple strands of the synthetic rubber (see Figure 9). To obtain the same spring rate as the single strand the total cross-sectional area of all the cords had to be equal to that of the single strand. The first idea was to use separate strands terminated at both ends. The problem with this was that the terminations would take up space needed for cords. A continuous strand would minimize the number of terminations and was developed as the choice for the CTPS model. A relatively small diameter cord would mean more strands would be needed and the chance of spring failure due to the loss of a single strand would be minimized. This was to be accomplished by using uncured rubber at each end of the spring to prevent a broken strand from unravelling the entire loop. The material used for the spring, Natsyn Duro 45, gives 140 lbs. force for 100 percent elongation based on a 1 inch diameter strand. An elongation of 150 percent was determined to be a maximum for our purposes as we were concerned about creep and permanent set at elongations beyond this. Using 3/8 inch diameter cord a 2 ft. spring consisting of 70 strands provided a preload of 1200 lbs. and had additional elasticity to give the large angle stiffness K2.

Fabrication

Fabrication was a simple process of fixing the spring ends and wrapping the cord. First one end of the loop was tied to the spring end with a knot wrapped in tape. One layer was then wrapped. Then uncured rubber was glued to the cords over the spring end. Another layer was

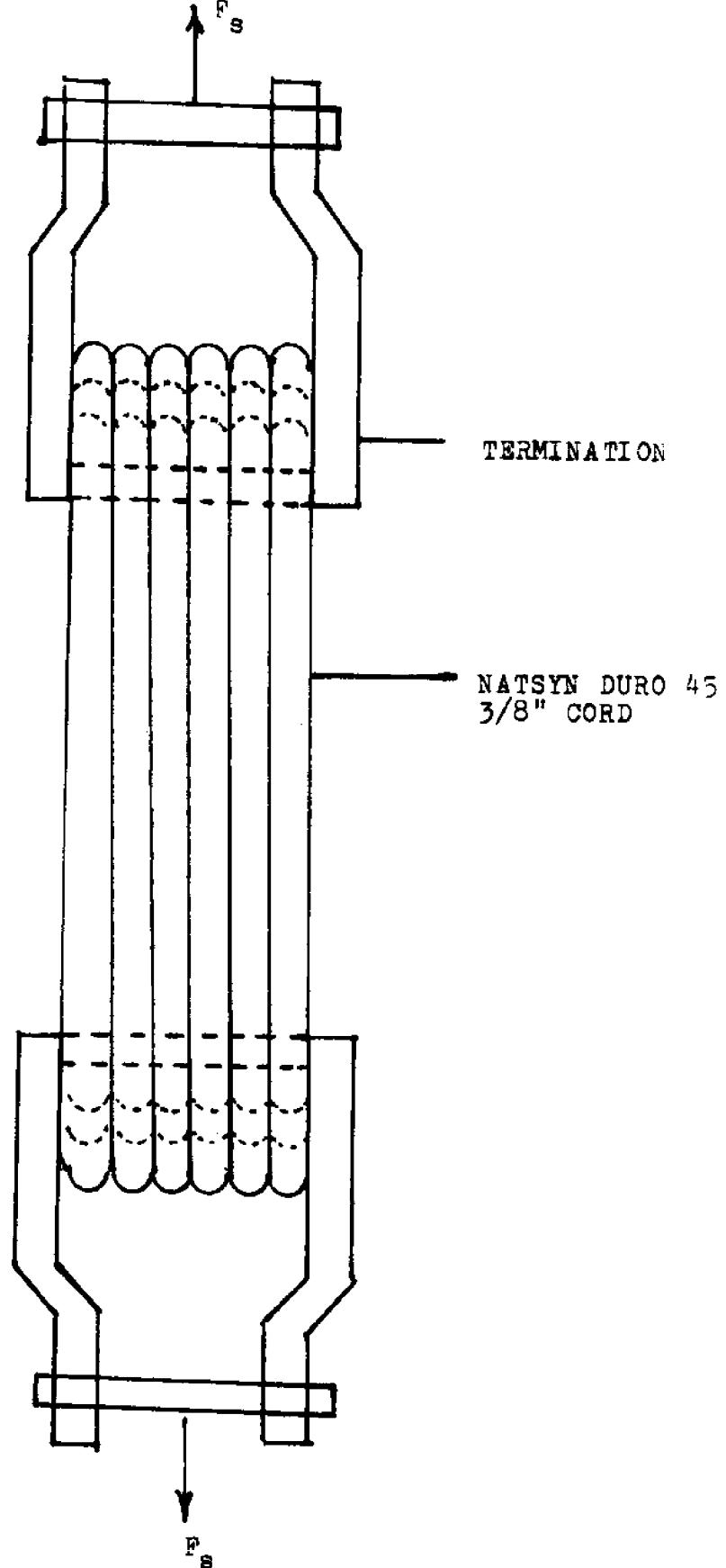


FIGURE 9. Continuous Wrap Tension Spring

wrapped and the procedure repeated until the spring had 70 strands.

PILE DESIGN

The pile had to withstand a constant compressive stress from the spring preload, a force applied at the top of the pile to pull it over for bench tests, and an impact force close to the hinge without buckling or excessive deflection (see Figure 10). It was treated as a cantilever beam with point static and impact forces to determine the deflection:

Eq. 3)

$$\delta_{\max} = \frac{Pa^2(3L - a)}{6EI}$$

The pile was treated as a member with one end free and one end clamped for the buckling case. The critical load was calculated using:

Eq. 4)

$$P_{\text{crit}} = \frac{\pi^2 EI}{4L^2}$$

The combined stress due to the point loads and the spring was highest at two points on the base and was calculated using:

Eq. 5)

$$\sigma = \frac{M_i R}{I} \pm \frac{P}{\pi(r_b^2 - r_i^2)}$$

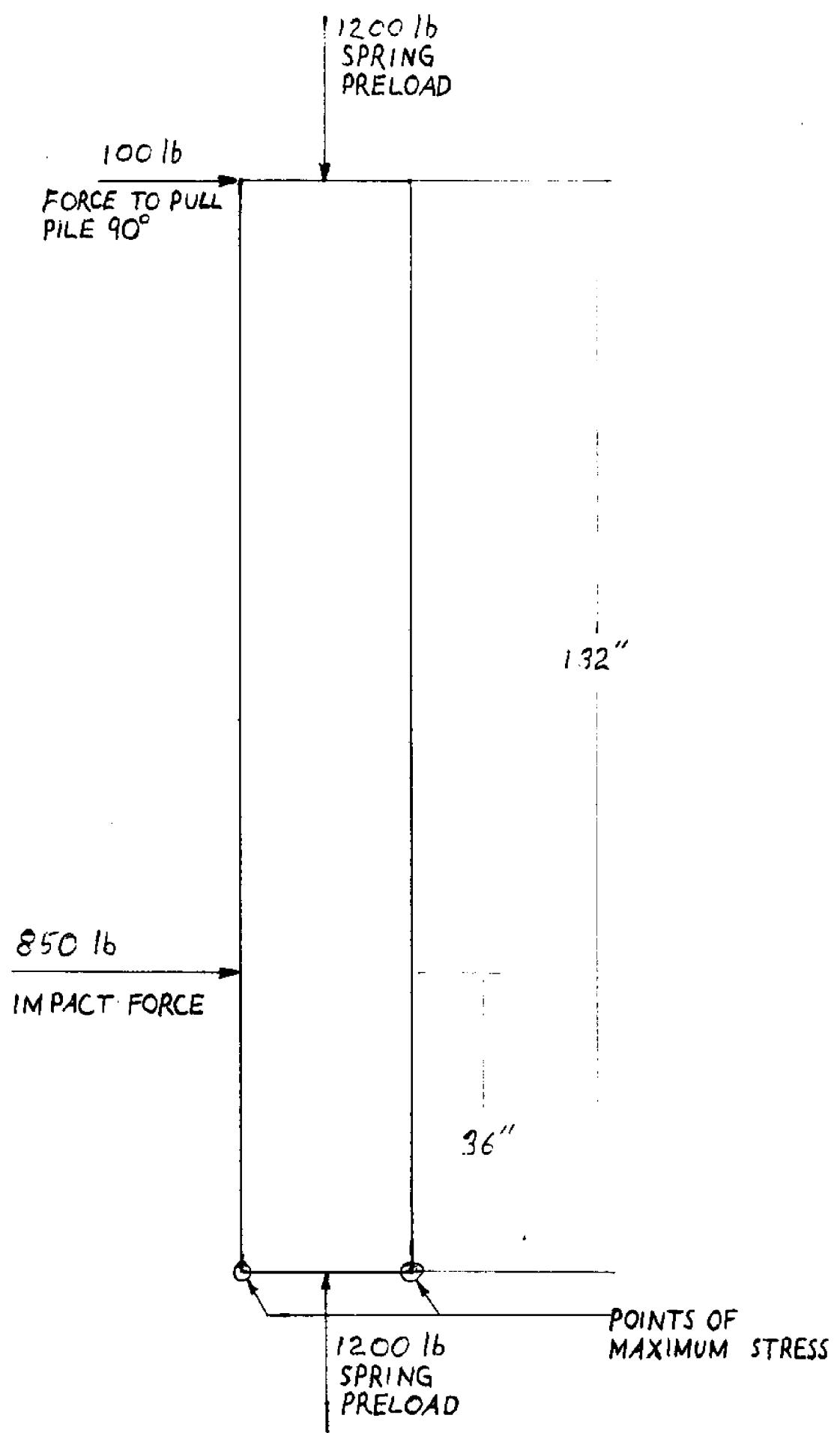


Figure 10. Forces exerted on pile due to collision.

Two requirements for the pile were scaled weight and diameter. Dimensional analysis of the proposed full scale 18 in. schedule 40 steel pipe provided a model weight of 4 lb/ft and a diameter of 4.5 inches. Steel was soon ruled out because the only pipe meeting the weight requirement had walls only 0.083 in. thick and buckling seemed likely. Schedule 80 PVC pipe was considered because it was light enough, but stress analysis showed it would have excessive deflection and would fail by buckling. Aluminum seemed the best material for the pile since it could meet the modeling requirements and could withstand all the stresses with a large safety factor. The tubing selected was 5 in. diameter with a 3/16 in. wall of 6061-T6 alloy. This alloy has good resistance to corrosion in seawater, but the pile was anodized to protect it further.

PRESTRESSING

The method for prestressing the spring had to be simple, so anyone could set the spring tension accurately in the field. The alternatives considered for setting the preload were a 'come-along' winch with a load cell in series and a boat winch with a torque wrench. The 'come-along' was ruled out because it needed a winching point separate from the pile and the load cell required special equipment, both of which could be hard to get in the field.

The boat winch would be easy to mount on top of the pile and torque wrenches are readily available. By measuring the diameter of the cable wrapped on the winch drum, the torque needed for the required preload can be calculated. The torque wrench was then used in place of the winch handle to operate the winch to stretch the spring. Once the spring was preloaded, the cable was locked with a special cable clamp,

and the winch removed (see Figure 11). The torque wrench method of prestressing did not work as well as expected due to spring friction inside the pile and cable friction at the top plate of the pile.

Prestressing in the final design was calculated by measuring the amount of cable wound onto the winch. A separate load deflection test was performed on the the spring material. This allowed calculating the preload value by measuring the amount of cable wound on the winch.

Stay terminations and adjustments were required to complete the system. The full inside diameter of the pile was needed for the spring. Therefore it was necessary to locate the stay adjustments at the base plate. The top of the stays were secured on a ring attached to the center of the spring termination clamp to prevent jamming (see Figure 12). The bottom ends of the stays were terminated using eye bolts for adjustment to equalize the stay tensions.

STAY SELECTION

When selecting a stay material it was kept in mind that the material should have a high modulus of elasticity, have a smooth surface to reduce friction, be able to support the spring force load and be as flexible as possible. Nylon and dacron ropes were eliminated from consideration right away because of these four criteria.

Kevlar rope was considered as an alternative material for the stays. It was found to be as strong as steel cable of the same diameter and more flexible. Its smooth surface and high modulus made it appear to be the ideal material. Upon further research though, it was discovered that Kevlar strength characteristics significantly decrease when bent or terminated. With the hinge concept it was seen that the maximum load on

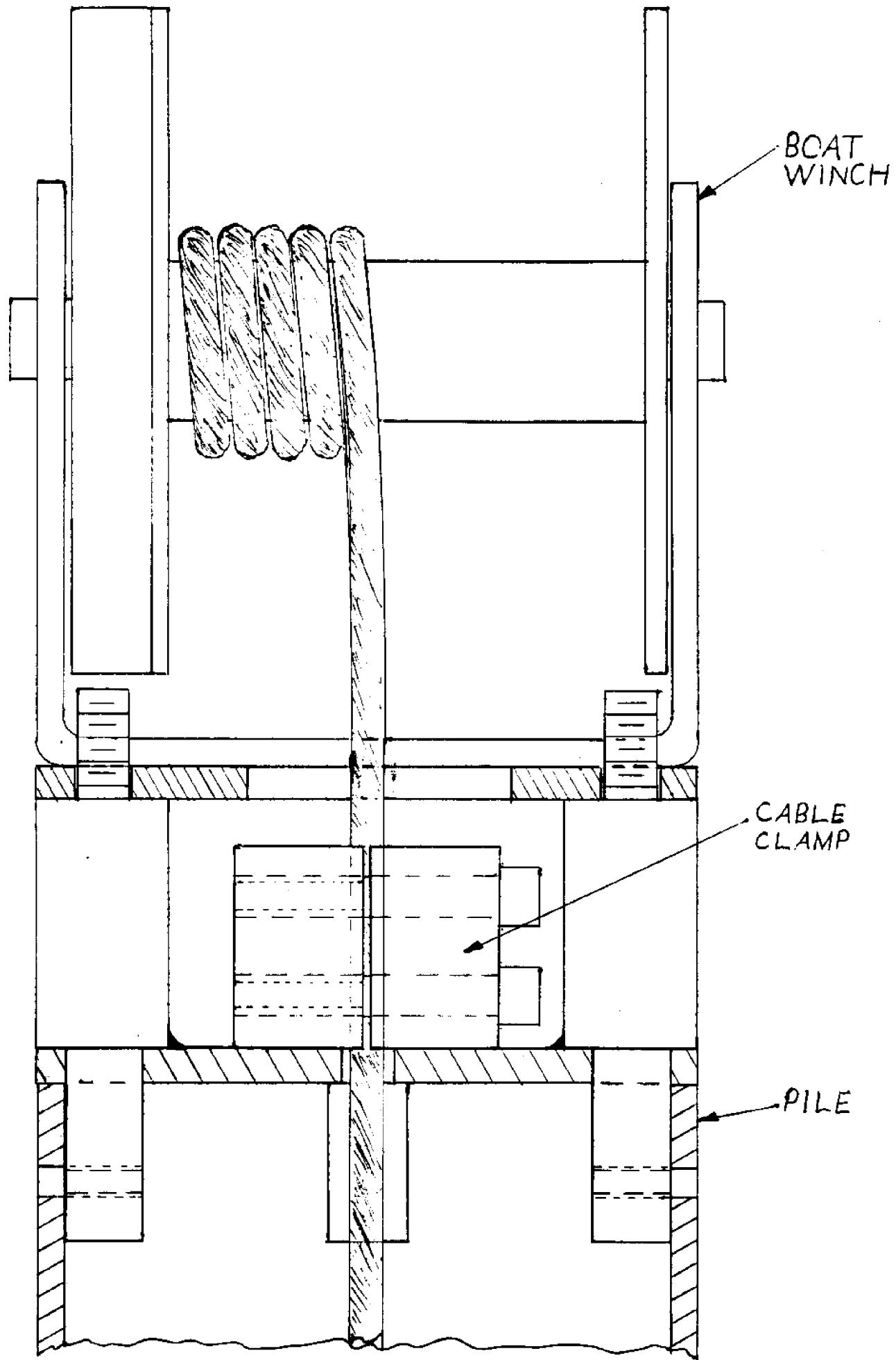


Figure 11. Boat winch prestressing concept.

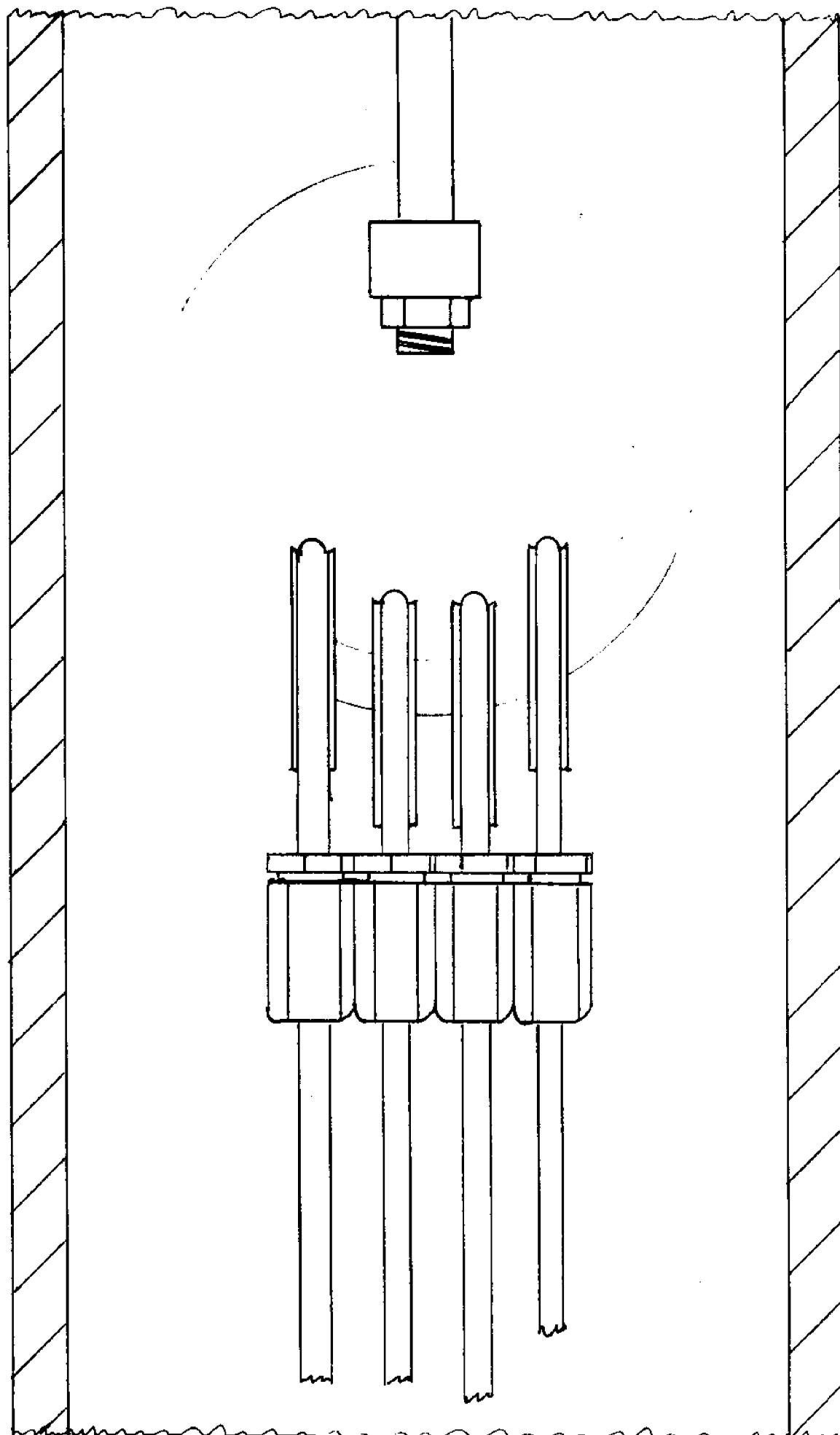


Figure 12. Upper stay termination.

the stays would occur when they were bent over the sheave surfaces. For this reason, Kevlar was eliminated as a possibility.

Polypropolene/steel core cable was also considered. The polypropolene coating would act as a smooth bearing surface and the steel core would provide the high modulus of elasticity needed. The only problem found with the polypropolene/steel core cable was its strength characteristics. The polypropolene/steel core cable was found to have only one half the tensile strength of an equivalent diameter steel cable. For this reason, it was also eliminated as a possibility.

Nylon coated steel cable appeared to be the best possibility. It was as strong as any other material of the same diameter and its modulus of elasticity was high. The nylon surface was smooth and seemed to be the answer to the friction problems on the stay guide collar and sheave surfaces. Allowing a large safety factor, quarter inch 7 X 19 stainless steel cable was selected. Although both types of cable were fairly flexible, 7 X 19 cable was selected over 7 X 7 cable since it was found to be more flexible. The stainless steel cable was selected over the galvanized cable with the feeling that it would hold up better in salt water.

III. PRELIMINARY TESTING AND DESIGN DEVELOPMENT

The pile was assembled, mounted to a work bench, and the spring pretensioned. The pile was then pulled over without instrumentation as a qualitative test. Three problems were apparent:

- 1) The nylon coating peeled off the stays as it slid over the hinge sheave.
- 2) The stay terminations in the pile were jamming the lower end of the spring.
- 3) The pile was not returning to the vertical position.

The major cause of the pile not returning to vertical seemed to be the stay terminations sliding around the ring and jamming against the inside wall of the pile. This prevented the spring from returning to its original length. To resolve this problem a new stay termination was designed and installed (see Figure 13). Also, the nylon coating was removed from the stays since it was not helping reduce friction.

The pile was reassembled and retested, but still did not return to vertical, even though the spring was no longer sticking. After several tries we noticed that the stays were cutting through the stay guides and into the sheaves. Three possible solutions to this problem were: rollers on the hinge sheaves, a 'creeper' mounted on the stays which would roll along the sheaves, or drill blanks welded on the sheaves. Rollers and 'creepers' were eliminated because they would take too long to build, add considerable complexity, and might not last in saltwater. It appeared drill blanks welded onto the sheave would provide a hardened and ground

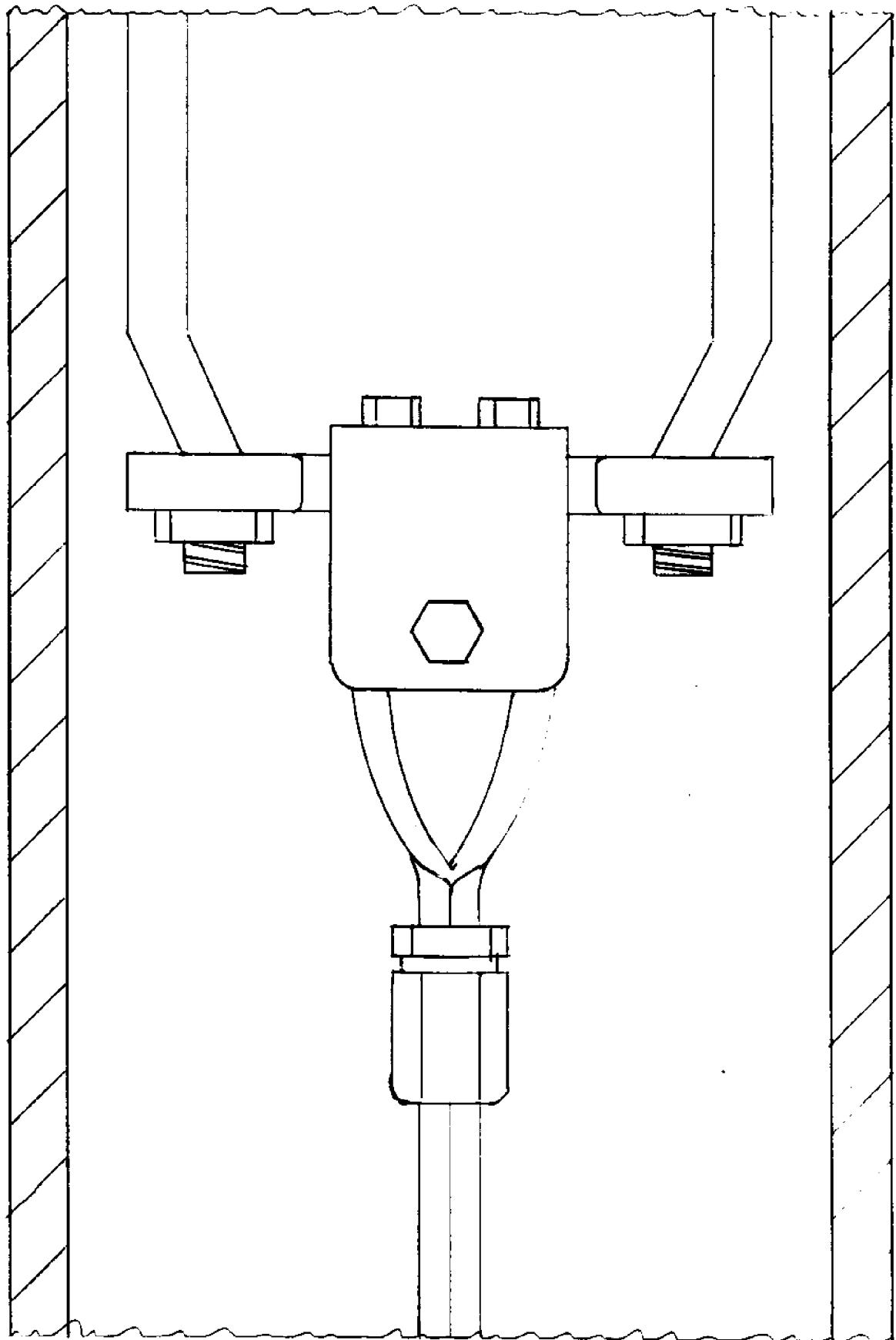


Figure 13. Redesigned upper spring termination.

surface with lower friction for the stays to slide over. The blanks were welded on the sheaves, but unfortunately when the pile was pulled over the contact points between the stays and drill blanks had created point loads and cable strands started breaking. At this point it appeared that the friction problem of the stays on the sheaves could not be significantly reduced and ways of preventing stay movement on the sheaves should be examined.

The original hinge sheaves moved along with the pile. As the pile was pulled over the sheaves slid along the stays. By keeping the sheave stationary, while the pile moved, the stays would just lay over the sheave. The modified hinge concept in Figures 14 and 15 was developed with this in mind. The redesign was demonstrated in a small scale model with no obvious problems. Since the geometry of the modified hinge was similar to the original, reconstruction used most of the original components. The center section was unchanged and the new base and top yoke were salvaged from the old base and top section. Once the new hinge was rebuilt the pile was reassembled. A qualitative test proved the modification successful and final tests could begin.

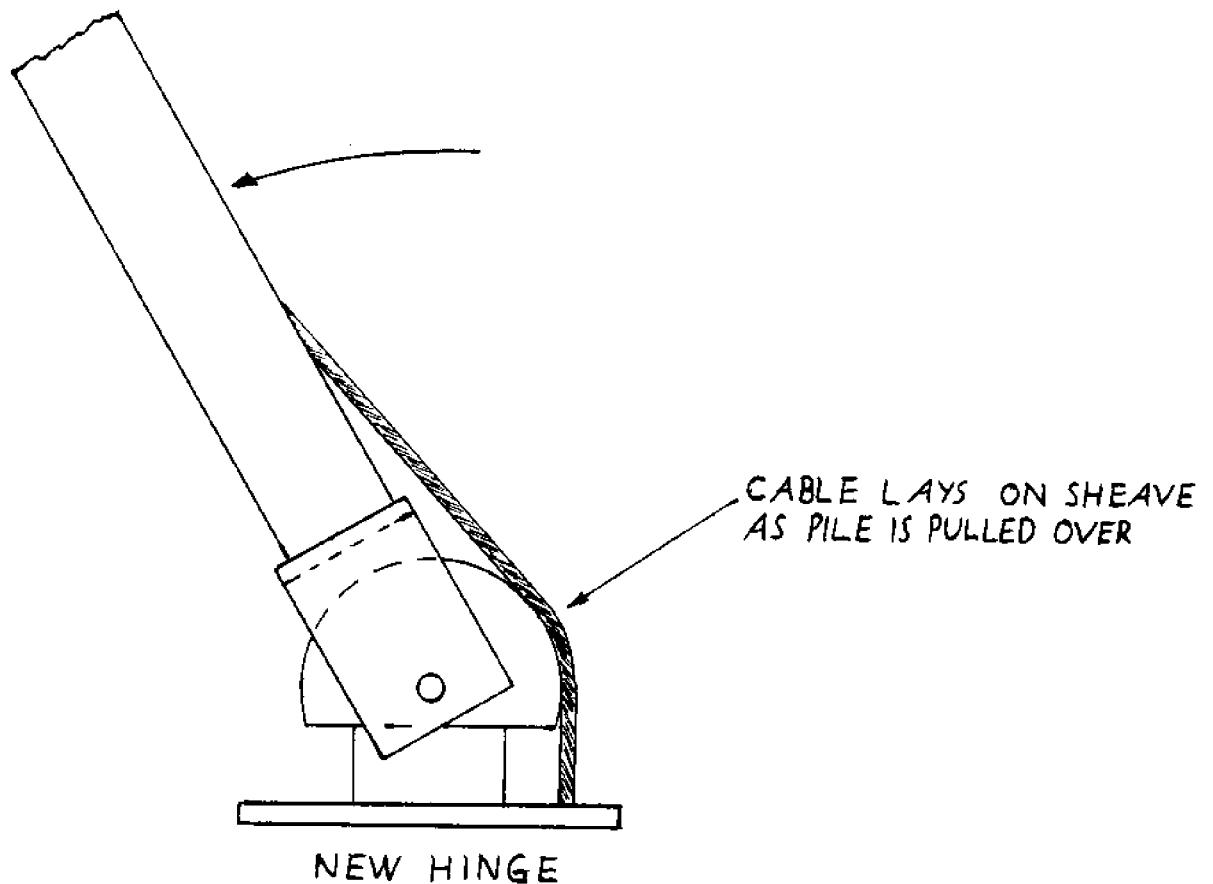
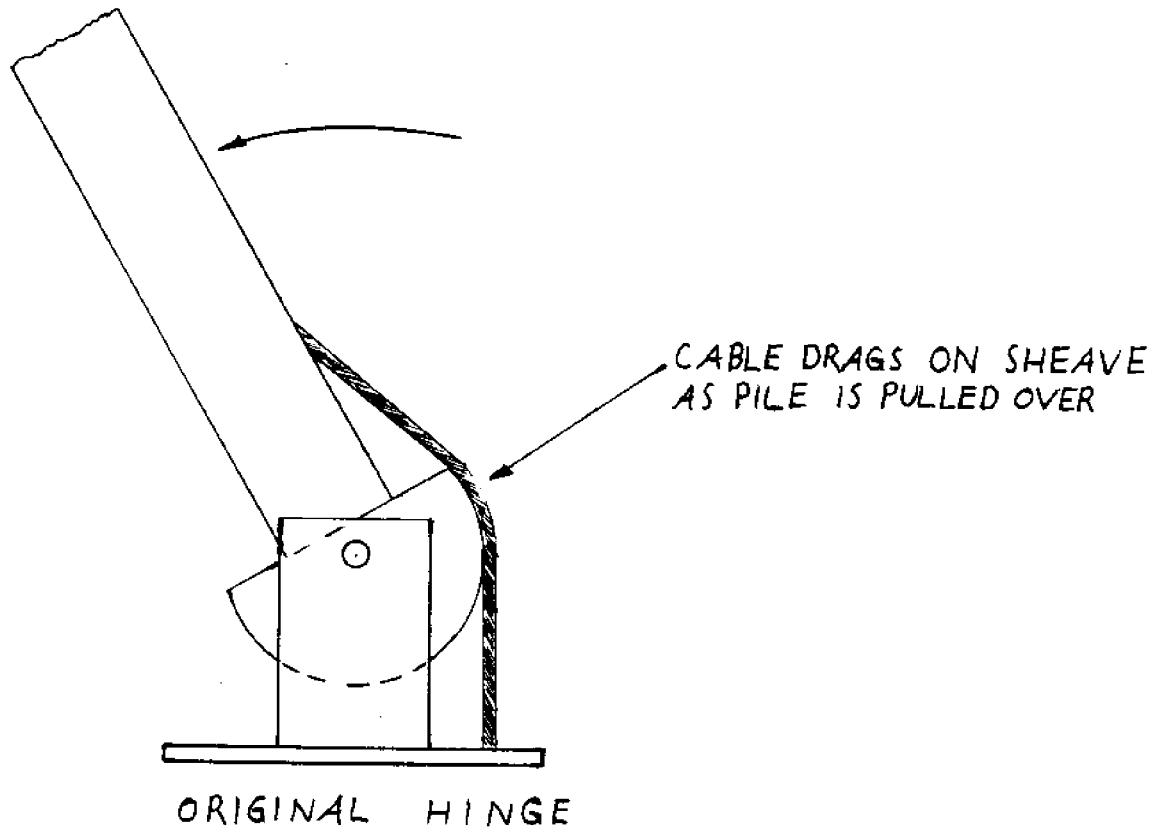


Figure 14. Moving sheaves vs. stationary sheaves.

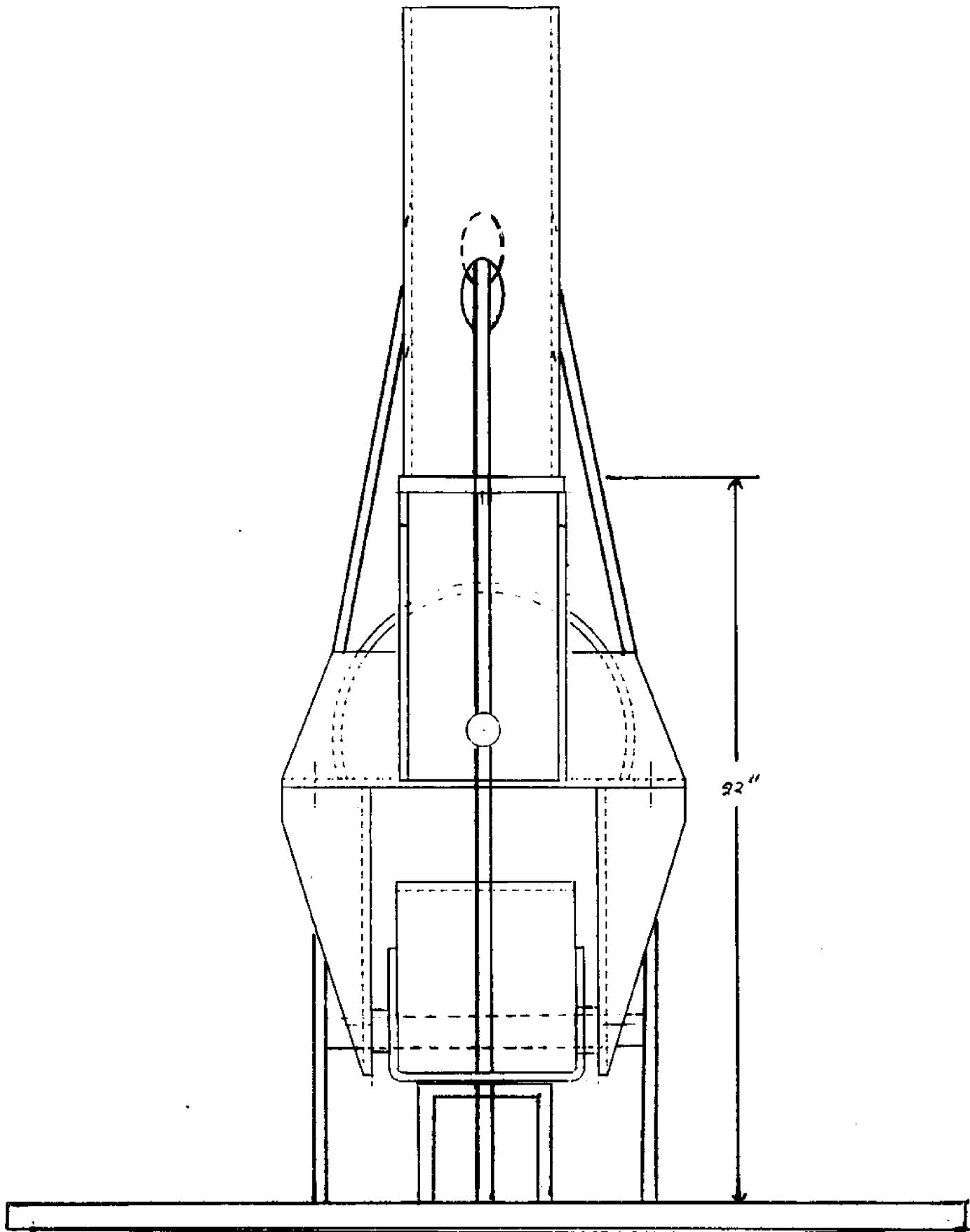


Figure 15. Modified hinge concept.

IV. MOMENT/ANGLE EXPERIMENTS

With the CTPS components constructed and the system assembled it was necessary to determine the structure's stiffness characteristics. Static "bench" tests were performed to determine hinge moment (pile restoring moment) vs. angle relationships. These provided plots of hinge moment vs. angle, which represent the K1 and K2 values (hinge stiffness). This experimental data was then compared with design values.

TEST PROCEDURE

The pile spring was prestressed to the required preload of 1200 lbs. predicted by the computer simulations. To obtain the proper preload, a single element of synthetic rubber was calibrated. A force vs. elongation curve was obtained from the Instron tensile machine in the materials laboratory. Since the entire spring consisted of these parallel elements of synthetic rubber, the spring was calibrated using the curve for a single rubber strand. The spring was then elongated to the preload using the boat winch.

With the spring prestressed, the pile was physically pulled over through a series of angles. The perpendicular force required to do this was recorded. The results produced the hinge moment vs. angle data that was plotted.

The bench test set up is shown in Figure 16. The prestressed pile was clamped on a large, stationary bench. A line was run from the top of the pile to a movable pulley mounted on an I-beam and then down to the

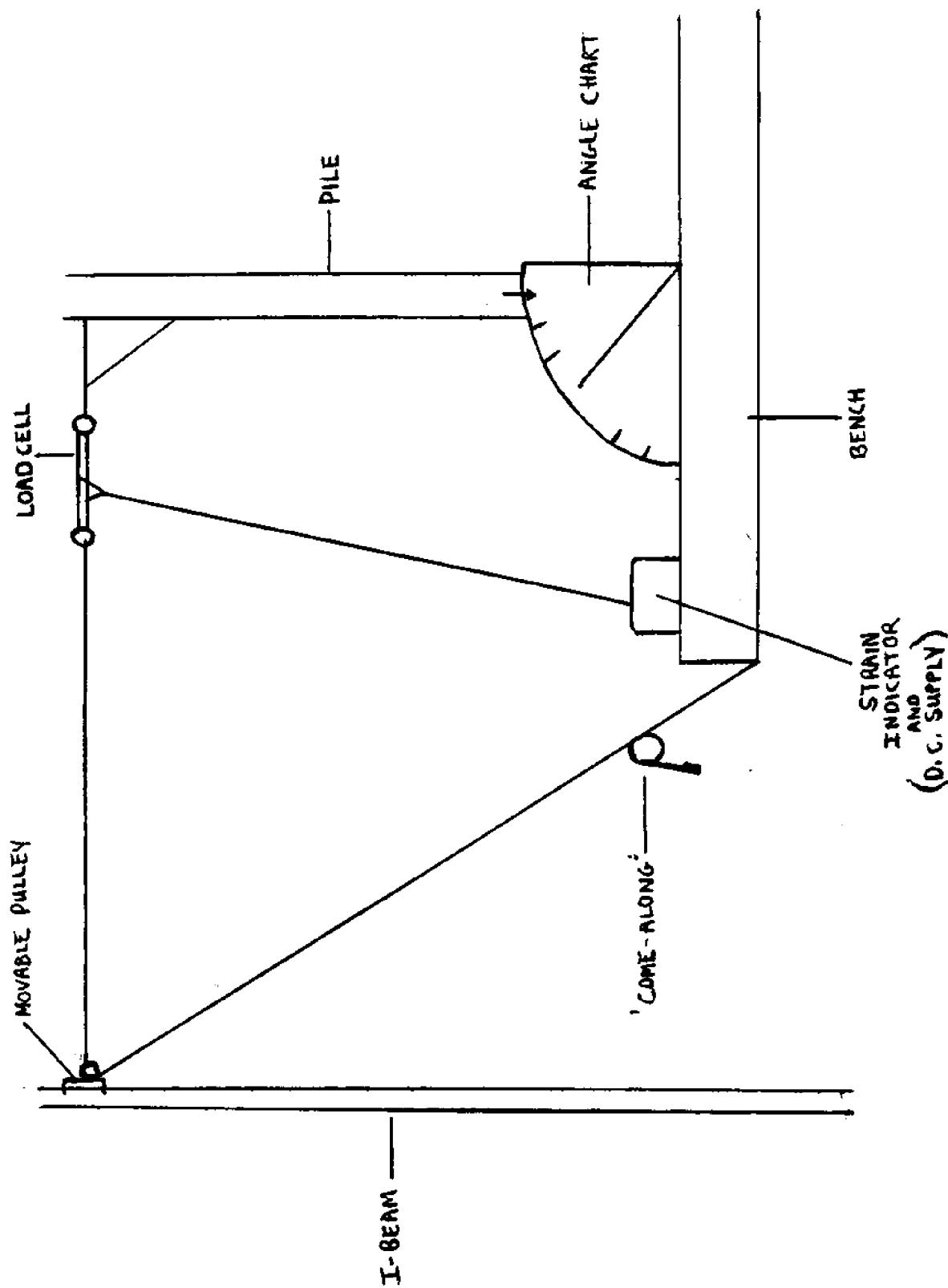


Figure 16. Bench test set-up.

bench. The movable pulley allowed the line's angle to be changed to ensure that the force always remained perpendicular to the pile.

A strain gage load cell was attached in series with the line to measure the perpendicular force. The output of the load cell was fed to a strain indicator box, which also served as the D.C. supply for the cell. The resulting strain readings were proportional to the perpendicular applied force. An angle chart was placed in front of the pile as a means of measuring the displacement angle during the pull down. The data was recorded for angles of 2 degree increments until 10 degrees then for 10 degree increments until the pile was horizontal.

The static pull down was performed in all three major modes of motion. That is, rotation about the upper axis, rotation about the lower axis, and oblique motion. The perpendicular force recorded was then converted to a moment by multiplying by the respective moment arm (upper axis, lower axis, oblique axis). Gravity (weight) also contributed a moment (M_g) about the hinge which was added to the measured moment (M_T) to obtain the total hinge moment (M_h).

Eq. 6)

$$M_h = M_T + M_g$$

RESULTS

The data from the bench test demonstrates the piecewise linear hinge moment resulting from compliance and friction in the CTPS components. This is shown in the plots in Figures 17, 18, and 19. The straight lines represent a least squares fit of the data. A line was fitted to the data up to 10 degrees. Another line was fitted to the data after ten degrees and allowed to have an intercept on the moment axis. The intersection of the two fitted lines was taken as the breakpoint (see Figures 17, 18, and

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TOP HINGE MOMENT VS. ANGLE

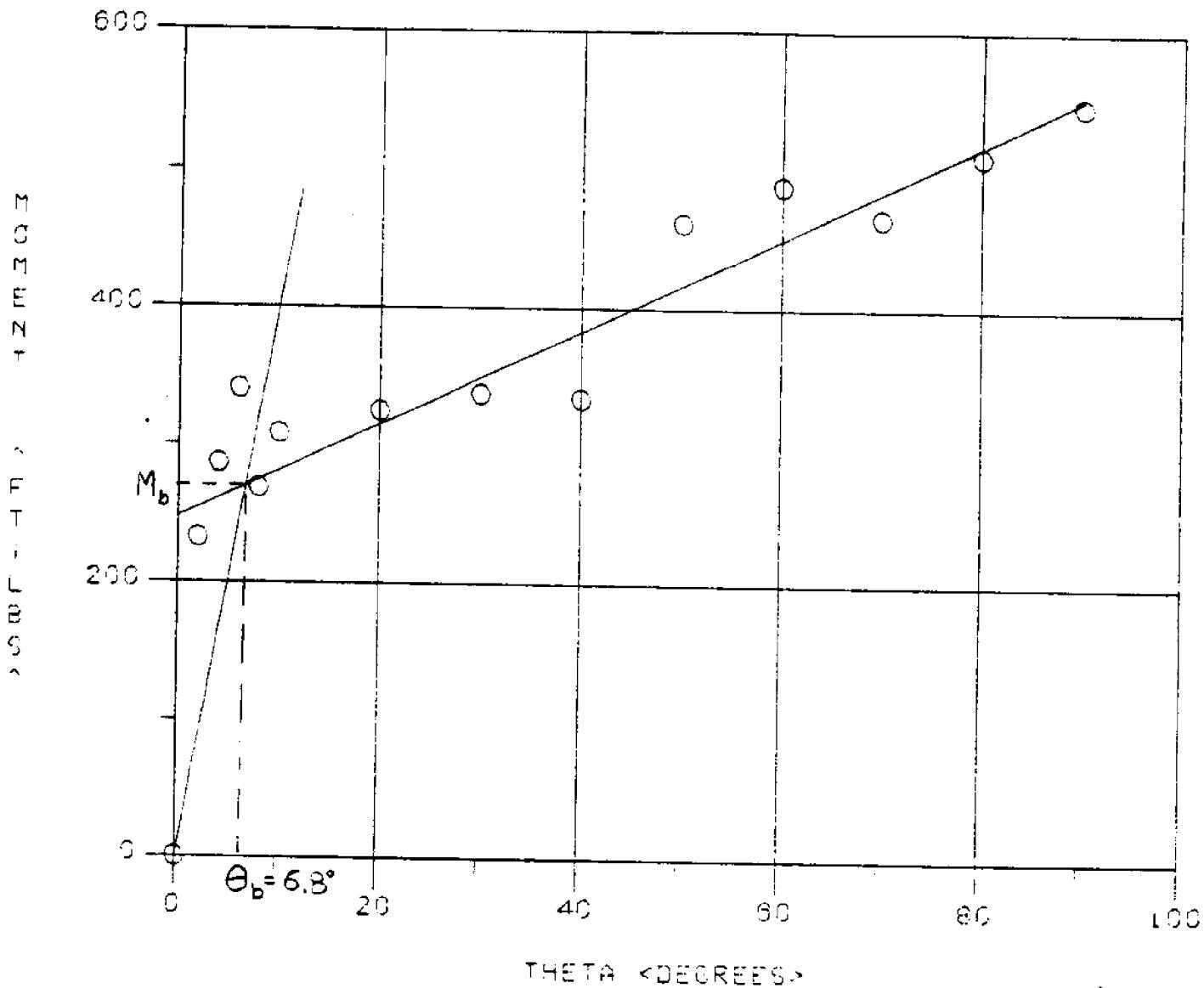


Figure 17. Hinge moment vs. angle about the top axis.
Piecewise linear curve. ($F_1 = 40.34$ ft.-lb./deg.
 $K_2 = 3.396$ ft.-lb./deg.)

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BOTTOM HINGE MOMENT VS. ANGLE

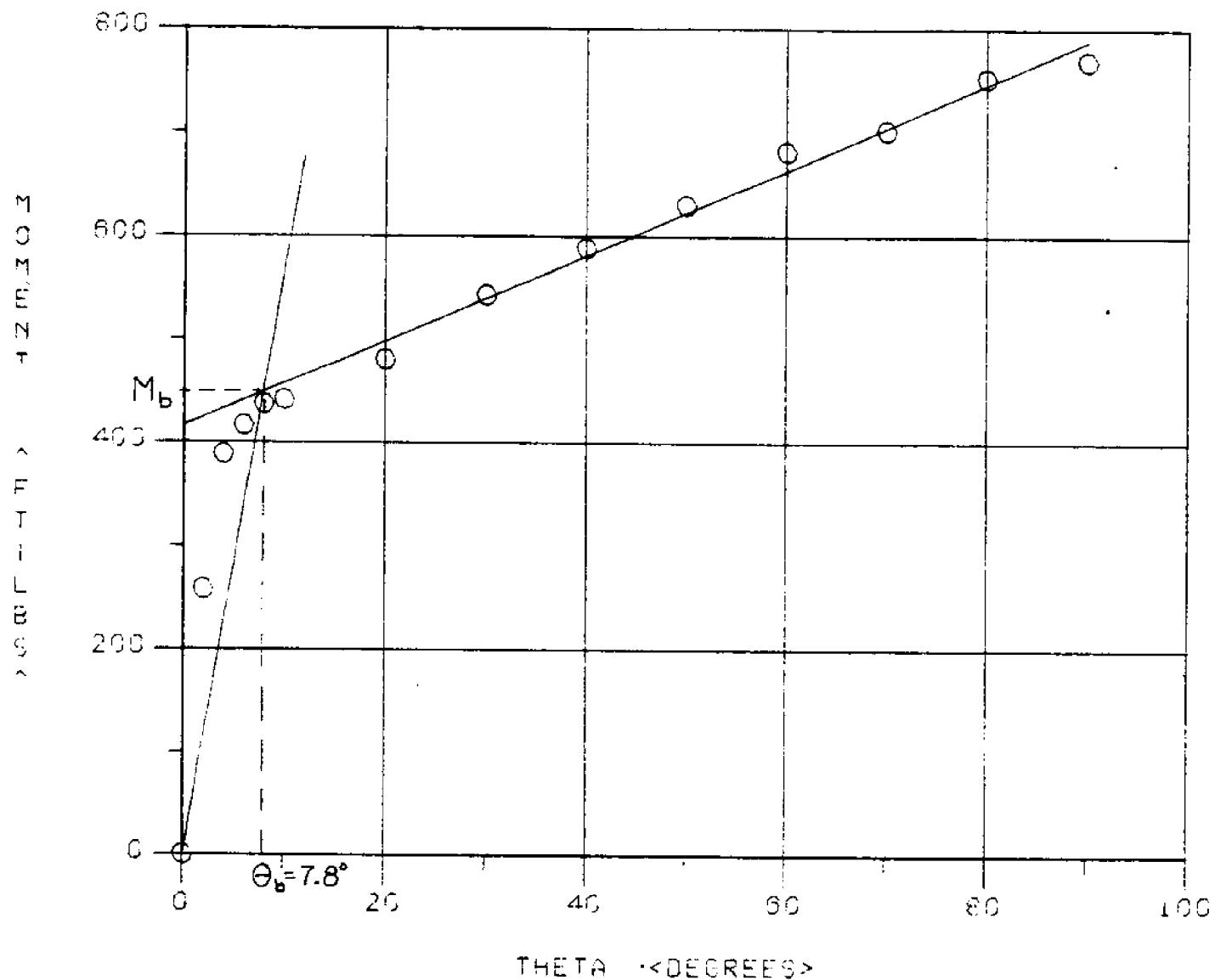


Figure 18. Hinge moment vs. angle about the bottom axis.
Piecewise linear curve. ($K_1 = 56.71$ ft.lb./deg.
 $K_2 = 4.14$ ft.lb./deg.)

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OBLIQUE HINGE MOMENT VS. ANGLE

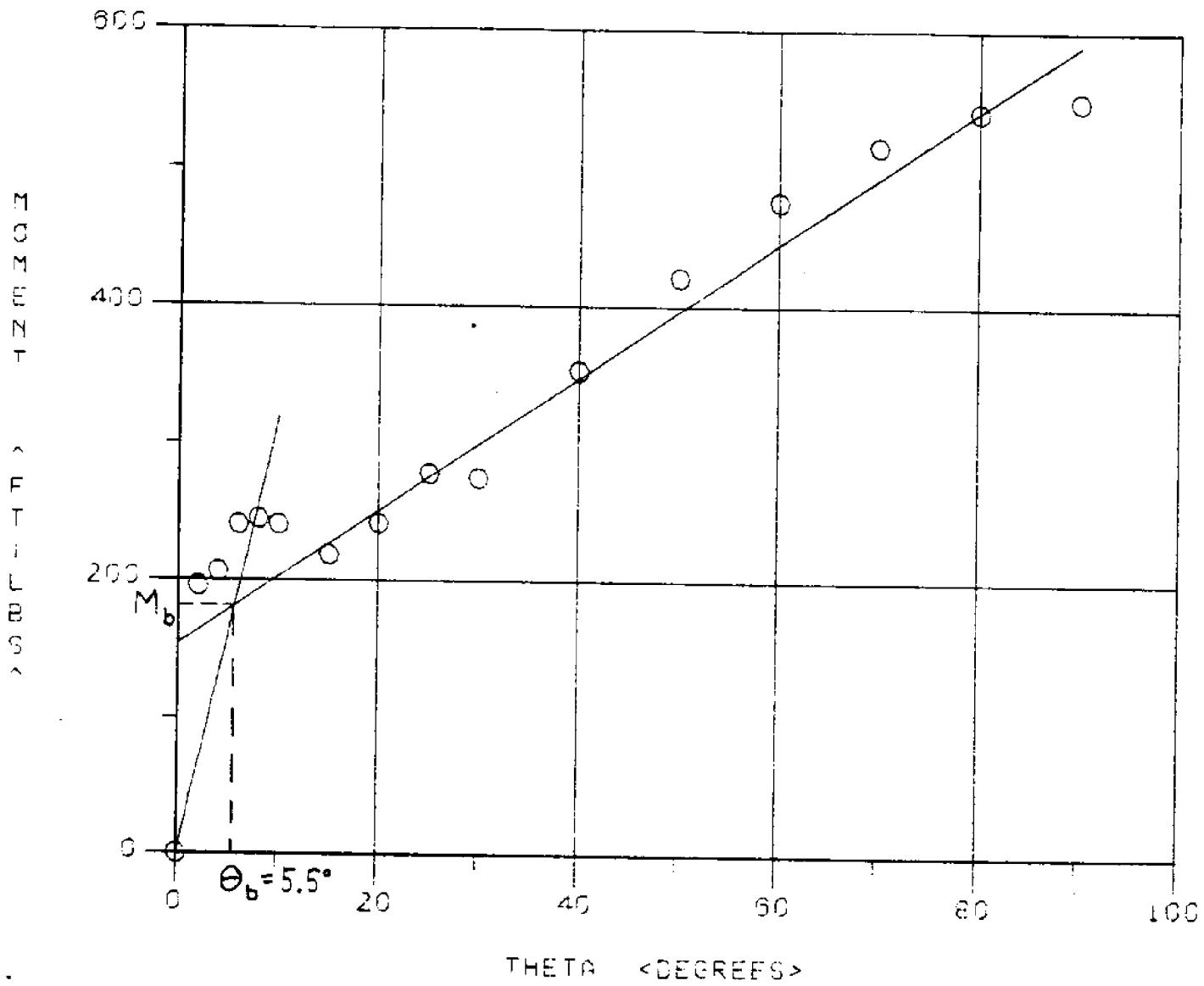


Figure 19. Hinge moment vs. angle about the oblique axis.
Piece-wise linear curve. ($K_1 = 31.84$ ft.-lb/deg.
 $K_2 = 4.863$ ft.-lb./deg.)

19). Thus, the actual breakpoint angle was determined.

In each hinge direction the breakpoint angle varies as do the slopes K_1 and K_2 . The breakpoint angles indicated are less than the 10 degree operational verticality limit. Nevertheless, these experimental values of hinge stiffness and breakpoint angle were a close match to the theoretical values used in the CTPS design process. The values found from the lower axis direction give the closest fit to the theoretical values (see table 3). Differences in stiffness in the three directions was due to friction and uneven stay tensioning.

The bench test indicated that friction was still a problem in the final design. Three locations were pinpointed where friction posed a problem. These are:

- 1) The stay centering collar in the hinge
- 2) The stay guides in the upper hinge axis
- 3) The spring rubbing against the inside of the pile

The most serious of these is where the stays rub against steel. At both the centering collar and the hinge pin stay guides the cable is forced to make an abrupt bend. The normal force involved with a bend of 17 degrees in a cable loaded with 1200 lbs. is 350 lbs. Using the friction equation:

Eq. 7)

an estimation of the friction was made. If a coefficient of friction of 0.2 is assumed, the sliding friction force would be 70 lbs. When this occurs in two places, as it did about the bottom axis, the force is 140 lbs. This is a significant force and may even be a conservative estimate. Design refinements should therefore concentrate on reducing friction at these points.

**Table 3: Lower Hinge Axis Design and Experimental Stiffness
Constants and Breakpoint Angle**

	Design	Experimental	% Difference
K1	2720 ft-lbs/rad	3249 ft-lbs/rad	16.3%
K2	272.0 ft-lbs/rad	237.0 ft-lbs/rad	12.9%
θ_b	10 degrees	7.8 degrees	22.0%

V. FIELD EXPERIMENT

INSTALLATION

With a modified and working quarter scale model, the next step was the installation of the system. Several aspects of pile installation were addressed. These included site location, base construction and installation, and pile mounting.

Site location was dependent on several factors. Accessibility, environmental conditions, and method of installation are among these. It was desired to parallel the installation methods for a full scale CTPS as closely as possible. Swift and Baldwin (1985) detail a practical method for full scale CTPS installation. However, some site locations that were considered would have made this method impractical. As a result, some alternative methods of installation were considered. Mielke (1985) outlines these alternatives as well as the final site selection, method of installation, base design and construction.

The site location chosen was Adams' Point, Durham. This location proved the most advantageous and allowed for an installation method similar to that of the full scale CTPS. Specifically, a single pile base driven into the mud (Figure 20). With the base installed, the hinge installation was simply a matter of transporting the CTPS to the base via a boat and bolting the structure to the base plate.

The installation was designed to occur at low tide to give access to the base. At the chosen location the base was in water between 1 ft. and 1.5 ft. deep. With an expected tidal change of 6 ft. to 6.5 ft. at this location, the water depth met the design requirement of 7.5 ft at high

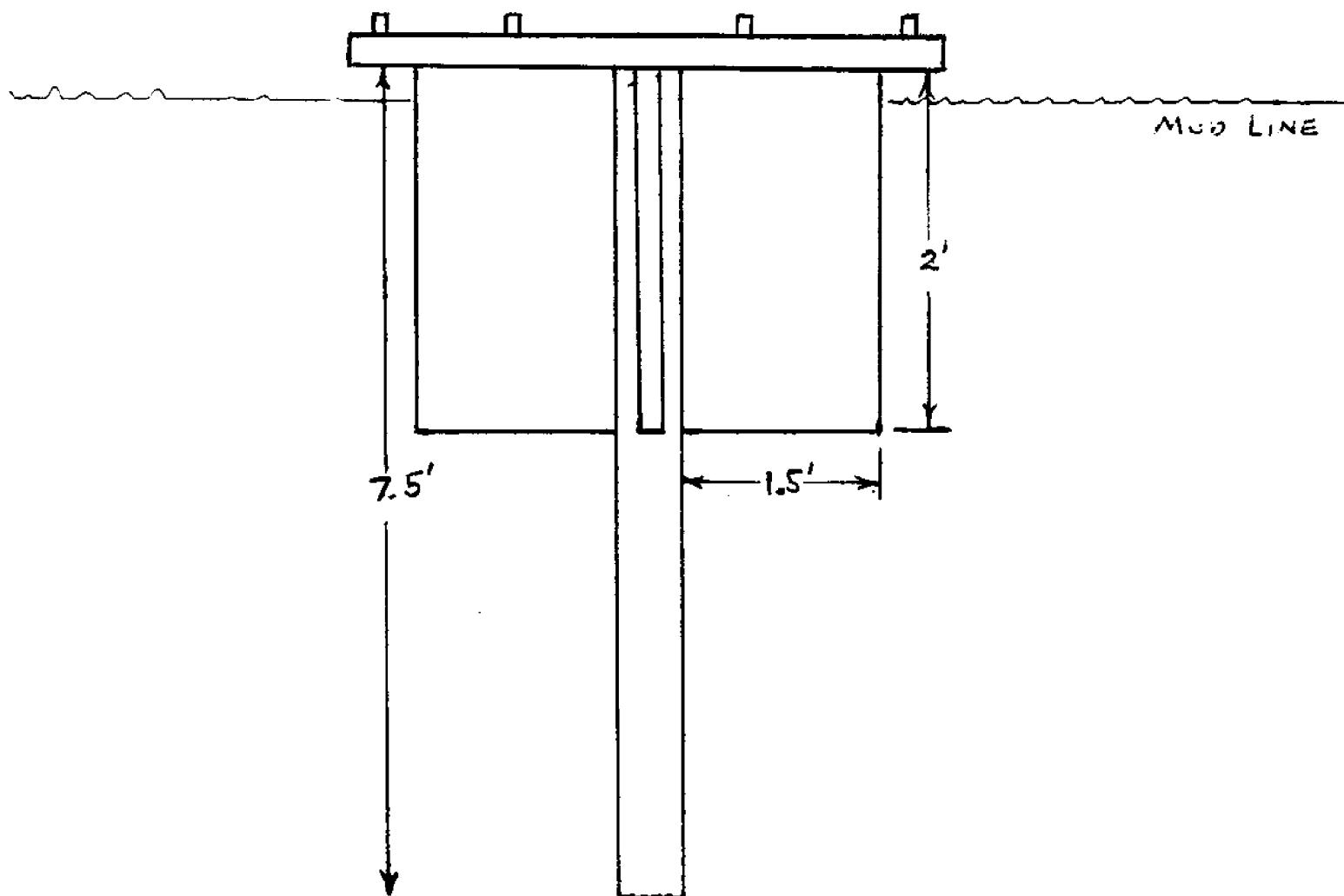


Figure 20. Pile base.

tide.

BARGE CONSTRUCTION

To simulate a scaled collision a barge was built (see Figure 21). During the collision sequence it is assumed the barge maintains a constant speed. Therefore it was necessary to design and build a barge with enough inertia to accomplish this. A barge scaled by length would have certainly met this requirement but would be too large to maneuver. The design used was based on a payload of ten thousand pounds, a scaled estimate from collision experiments outlined in Swift and Baldwin (1985). Bow angle and impact points were chosen to simulate a scaled collision. The barge was then constructed with plywood and framing lumber. Water was used as the ballast and kept was from surging by baffles.

A test run was performed at the installation site to test the towing capabilities of the Jere Chase and the seaworthiness of the barge. This test indicated the barge needed modifications to prevent water from flowing between the baffles creating an unequal weight distribution. Modifications were made to finalize the barge construction phase of the project.

COLLISION TESTING

The main objective of the collision tests was to qualitatively observe and record the collision of the barge with the CTPS. Some other initial objectives were to quantify the barge impacts with actual measurements and to record the pile's response to environmental conditions. Due to time constraints, impact measurements and environmental response measurements were sacrificed despite the initial

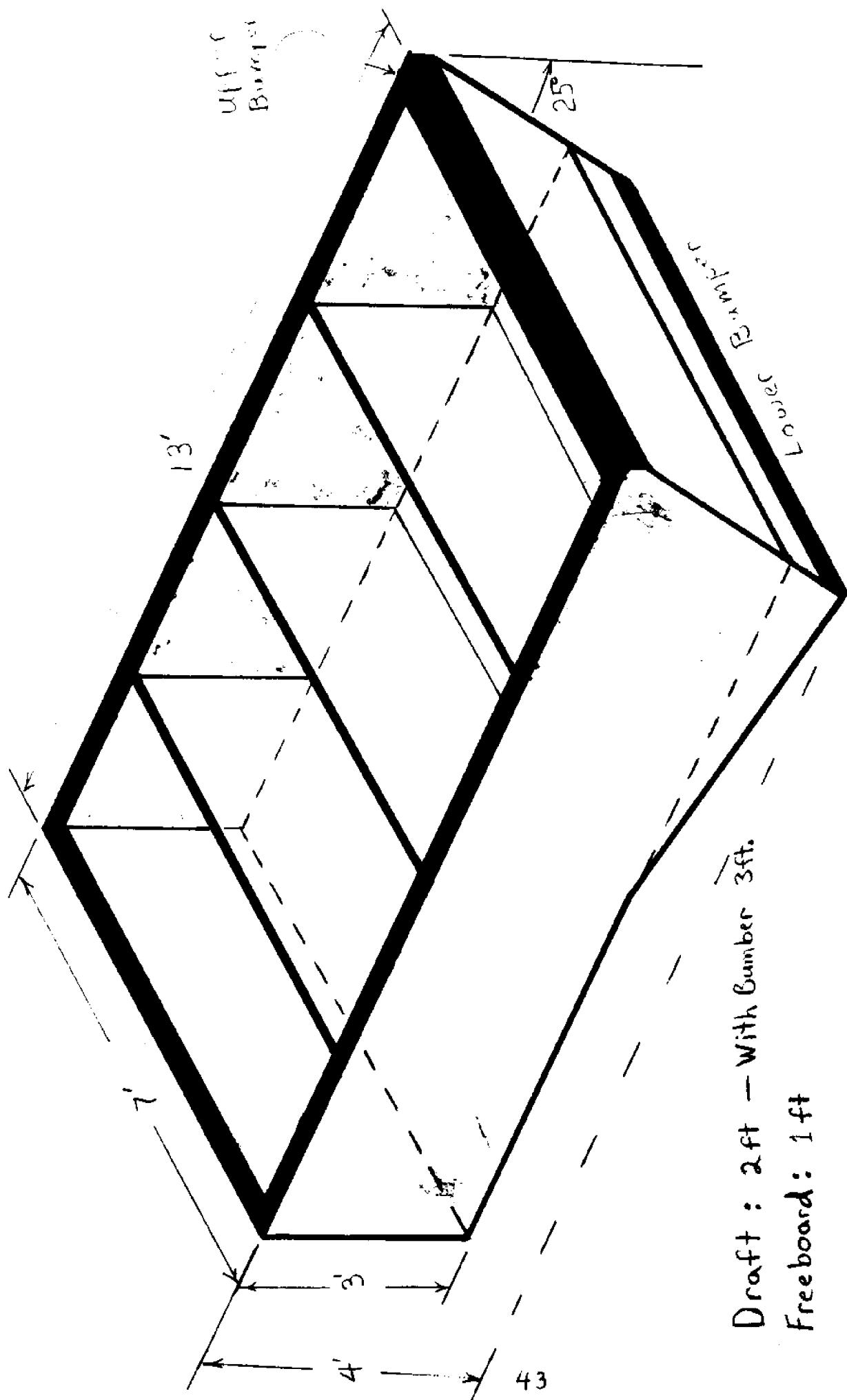


Figure 21. Test barge dimensions.

work done with them. These objectives will be pursued in a continuation of this project in the summer of 1985.

TEST PROCEDURE

The initial set up for the collision test was essentially the same as in the barge test run. The barge was flooded with water to the two foot line to give the necessary mass and draft. The forward compartments were flooded approximately one third that of the rear compartments. This was a necessary weight modification in order to correct for the downward moment exerted on the barge by towing.

The Jere A. Chase towed the barge through several runs at the pile. The initial runs were at a slow speed as a precaution. In each consecutive run the barge speed was increased. In the final runs the barge speed was nearly that of the design requirement. Scaled design speed was 5 knots and the experimental speed was close to 4 knots.

The barge was towed so that collision with the pile would occur in the worst case scenario. In the worst case, collision occurs such that the pile moves with respect to the upper axis so that the impact is closest to the hinge. Of the collision runs made, full pile recovery occurred 50 percent of the time. Other times only partial recovery occurred. The pile returned to about 35 degrees from vertical. This was a result of the friction problems observed in bench testing.

These collision tests were recorded by means of video and still photographs. A video camera and recorder were aboard the Jere A. Chase to record the collision sequences. Still photographs were taken from an outboard motor boat that provided several perspectives of the collision tests.

RESULTS

The results of the collision test are observational. The pile sustains an impact without physical damage and returns to the vertical after many of the collisions. Friction forces prevent full pile recovery the remaining collisions. A collision sequence of still photographs is shown in Figure 22.

FIGURE 22. Collision Sequence (A) Upper bumper; (B) lower bumper; (C, D) Restoration



(A)



(B)



(C)



(D)

VI. COMPUTER SIMULATION OF OPERATIONAL CONDITIONS

Computer simulations for model CTPS environmental response were run using experimental values for hinge stiffness and breakpoint angle about the lower hinge axis. Values used in the simulation are given in table 4. OPPILE was used with worst case high water conditions and for low water conditions. This gave the expected model response shown in Figure 23. The model CTPS response at natural frequency, which was determined by program PILEFREQ, is also plotted in Figure 23. The plot indicates the CTPS model will stay within the ten degree design criteria.

Hurricane conditions were also plotted using the simulation HURPILE with the experimental values as input. This storm condition is shown in Figure 24 compared to response from OPPILE. This plot indicates the CTPS model stays close to ten degrees even under hurricane conditions.

Table 4. Quarter scale model parameters used in computer simulations

	OPPILE	OPPILE	OPPILE	HURPILE
	High Water	Low Water	Resonance	-----
Stepsize (sec.)	.15	.15	.15	.15
Max. time (sec.)	5	5	5	5
K1 (ft.-lb./rad.)	3249	3249	3249	3249
K2 (ft.-lb./rad.)	237	237	237	237
Breakpoint (deg.)	7.8	7.8	7.8	7.8
Load wgt. (lbs.)	99.5	99.5	99.5	99.5
Pile wgt. (lbs.)	40	40	40	40
Pile length (ft.)	11	11	11	11
Pile dia. (ft.)	.4167	.4167	.4167	.4167
Length-load (ft.)	3.2	3.2	3.2	3.2
Length-board (ft.)	9.38	9.38	9.38	9.38
Depth-hinge (ft.)	6.88	1.88	6.88	9.0
Total depth (ft.)	7.5	2.5	7.5	9.75
Wind vel. (ft./sec.)	50.6	50.6	50.6	88.4
Current (ft./sec.)	2.53	2.53	2.53	2.53
Wave hgt. (ft.)	1.25	1.25	1.25	1.5
Wave per. (sec.)	2.5	2.5	1.57	2.5
Wave length (ft.)	29.5	20.6	12.7	30.8

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OPPILE PLOT (OPERATING CONDITIONS)

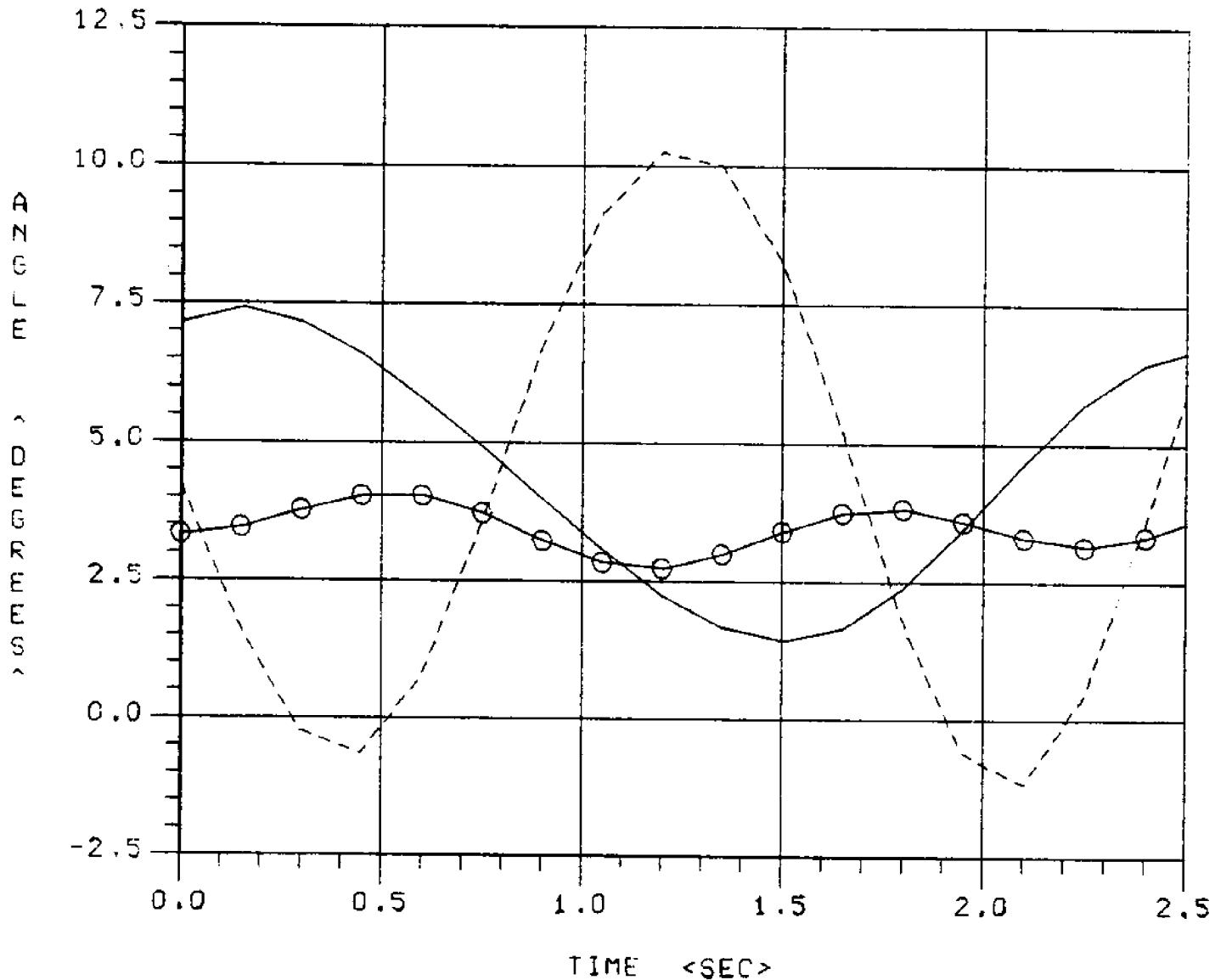


Figure 23. Pile motion during operating conditions. Shown is the steady-state response. High water (7.5 ft.) is represented by (—), low water (2.5 ft.) is represented by (○○○), and resonant conditions (1.57 sec. wave period) is represented by (----)

5-May-85

HURPILE PLOT (HURRICANE CONDITIONS)

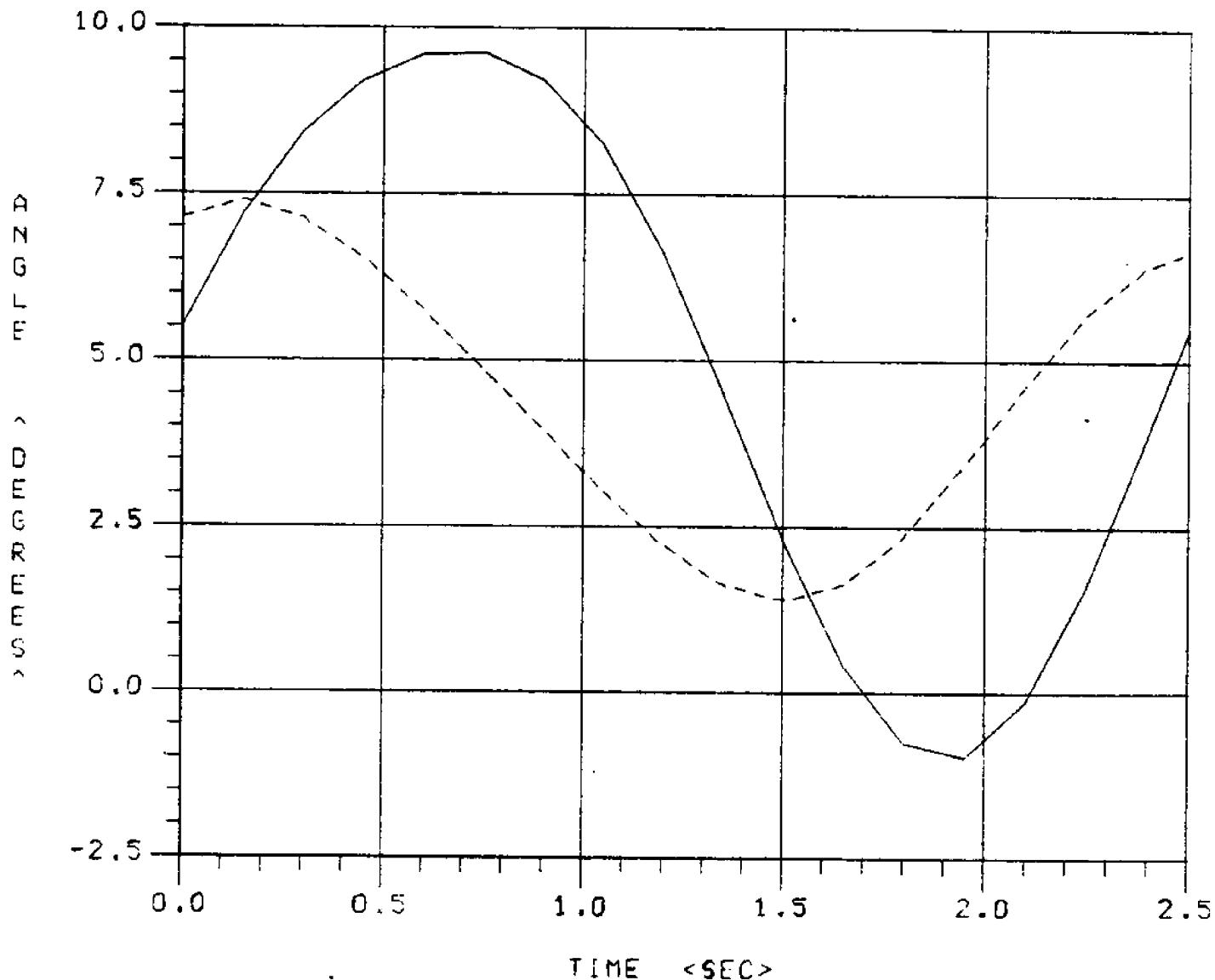


Figure 24. Pile motion during hurricane conditions. Shown is the steady-state response. Hurricane conditions are represented by (—) and normal operating conditions are represented by (----).

VII. CONCLUSIONS

The bench testing of the CTPS model demonstrated the feasibility of the hinge design. The necessary hinge stiffness and articulation has been achieved. The variation in stiffness in the three directions of motion results from sliding friction forces and stay tension imbalance. It was also shown that the breakpoint angle occurs at approximately the design goal of 10 degrees. The tests also indicate that the friction problem has not been completely resolved. The points where friction posed problems have been identified and need closer attention.

The collision experiments demonstrated the CTPS model's ability to withstand severe impacts with no physical damage and recover to a vertical position. Friction forces prevented the pile's full recovery a number of times. In these cases the pile's partial recovery showed that although friction is still present, it is not an insurmountable problem.

Computer simulations were made for environmental conditions the CTPS would be exposed too. Simulations were done for normal operational conditions and hurricane conditions. These simulations used the quarter scale model's actual dimensions and experimental stiffness characteristics. The model's ability to meet the verticality criteria of 10 degrees under operating conditions and severe weather conditions was demonstrated. Though no actual environmental experiments were performed, the simulations clearly indicate the environmental design criteria can easily be met. Actual environmental experiments are expected to be conducted using the model CTPS in the summer of 1985.

RECOMMENDATIONS

The design, development, and testing of the quarter scale CTPS has

demonstrated the feasibility of the CTPS concept. The tests performed on the model show that the concept is a solution to the problem of pile damage from collisions. We recommend that a further investigation of the friction problem encountered in this CTPS design be conducted. We also recommend that the "air bag" spring concept be developed for a full scale prototype. Although not applicable to this quarter scale model, investigation showed the concept has great potential.

VIII. REFERENCES

Swift, M.R. and Baldwin, K.C. (1985) "The Design and Model Testing of a Collision Tolerant Pile Structure", Published report submitted to the U.S. Coast Guard R&D Center, Avery Point, Groton, CT.

Mielke, D.J. (1985) "The Design, Development, and Construction of a Collision Tolerant Pile Structure Base", Submitted to the University of New Hampshire Mechanical Engineering Department (Report pending).

IX. APPENDIX

1. Swift, M.R. and Baldwin, K.C. (1985) "The Design and Model Testing of a Collision Tolerant Pile Structure", A report submitted to the U.S. Coast Guard R&D Center, Avery Point, CT. Chapter II, pp. 9-19.

II. PILE DYNAMICS MODELING

MODELING APPROACH

Computer programs were developed for modeling the dynamics of pile systems under conditions referred to in the design criteria. Since the overall CTPS design criteria given in Tables 2-4 refer to distinct and very different conditions, several computer models were developed. Programs were written for operating conditions in which small angle verticality restrictions must be met, hurricane conditions in which forcing and angular motion may be larger, and collision conditions in which barge contact is the dominant feature. All models, however, share some common assumptions and, as a consequence, many dynamic equations are the same for all applications. General features of the modeling approach are discussed here, while the specifics of individual computer programs are detailed in the following subsections. Program listings are given in Appendix A.

The CTPS is considered to be a flexible hinge-rigid beam-mass system such as that shown in Fig. 2. The hinge is omnidirectional and possesses restoring moment stiffness. Weight and current forcing are external loads common to all major computer models, while wind, wave and barge contact forcing may or may not be present depending on the application. In all models, the directions of current and (when present) wind, wave and barge motion are assumed collinear corresponding to the worst case situation.

The governing dynamic equation for the hinge-beam-mass system considered is the time rate of change of angular momentum equation applied at the (fixed point) hinge,

$$I_H \ddot{\theta} = \sum M's \quad (1)$$

where I_H = moment of inertia about the hinge, θ = angle of pile with respect to the vertical, (") indicates two derivatives of () with respect to time t , and M refers to moments applied about the hinge. (All terminology used is summarized in the NOMENCLATURE section).

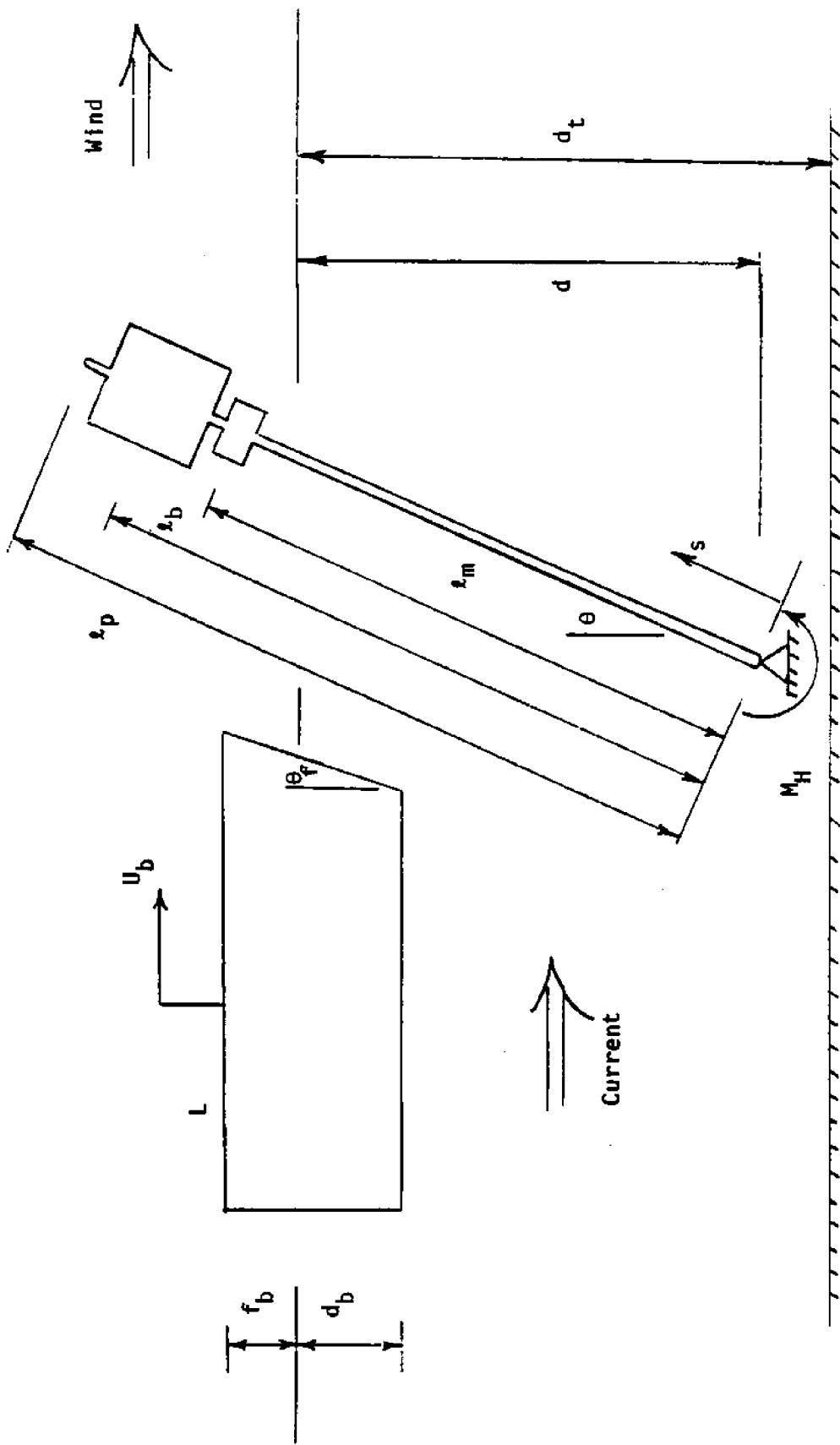


Fig. 2. CTPS nomenclature. The CTPS consists of a flexible hinge, a rigid beam pile, a mass near the top representing the load, and the daymark boards. Barge and environmental parameters are also indicated.

The flexible hinge will be constructed to provide a restoring moment to the CTPS. It is desired to have the hinge be very stiff at small angles to meet the verticality requirement under operating conditions. Yet hinge moment stiffness at the large angles encountered during collisions should be limited in order to reduce maximum pile bending moment. The piece-wise linear behavior of hinge moment $M_H = M_H(\theta)$ shown in Fig. 3 has these characteristics and is used in the models. The hinge's behavior at small angles is determined by the initial stiffness (slope) k_1 , while properties at large angles are additionally influenced by the large angle stiffness (slope) k_2 . It is desirable to have $k_1 \gg k_2$, and the breakpoint angle (point of slope change) should be at the limit of the CTPS's operating range.

The upsetting gravitational moment, M_G , due to pile and load weight has the mathematical form

$$M_G = \ell_m \sin\theta W_L + 1/2 \ell_p \sin\theta W_p \quad (2)$$

where lengths ℓ_m and ℓ_p are shown in Fig. 2 and W_L and W_p are load and pile weights, respectively. Hollow piles are assumed free-flooding, and the restoring moment effect due to buoyancy is neglected.

The moment load induced by relative water movement, M_C , is evaluated using a form of Morrison's equation,

$$M_C = \int_0^{\ell_s} s \left[\frac{1}{2} \rho_w C_w d_p u_r^2 + C_m \left(\frac{\pi d_p^2}{4} \right) \rho_w \dot{u}_r \right] ds \quad (3)$$

where s = pile coordinate shown in Fig. 2, ℓ_s = submerged length, ρ_w = water density, C_w = drag coefficient of the pile in water, C_m = inertia coefficient of the pile, d_p = pile diameter, and u_r = relative velocity normal to the pile. The relative velocity includes steady current, motion of the pile itself and wave fluid velocity. Wave motion is taken as that of a regular (single frequency), small amplitude (linear) surface wave. Thus the wave fluid velocity components are

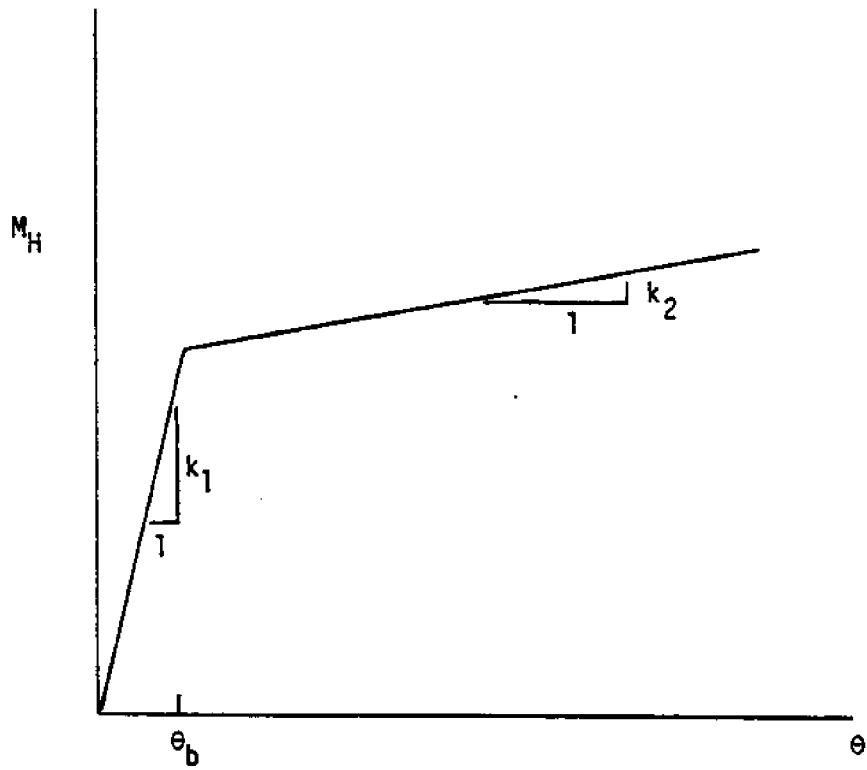


Fig. 3. Piece-wise linear hinge moment behavior. The initial stiffness k_1 is the slope at small angles; k_2 is the slope at large angles. The break point angle is θ_b .

$$u_w = \frac{\sigma H_w}{2} \left(\frac{\cosh k(d_t + y)}{\sinh k d_t} \right) \cos(kx - \sigma t)$$

and

$$v_w = \frac{\sigma H_w}{2} \left(\frac{\sinh k(d_t + y)}{\sinh k d_t} \right) \sin(kx - \sigma t) \quad (4)$$

where σ = wave radian frequency, H_w = wave height, $k = 2\pi/\lambda$, λ = wavelength, d_t = water depth and x, y are horizontal, vertical coordinates with their origin at the mean water level directly above the hinge.

The overturning moment acting on the pile as a result of (steady) wind, M_w , is evaluated using a drag coefficient approach. An approximate expression for wind moment can therefore be written in the form:

$$M_w = 1/4(\ell_p^2 \cos^2 \theta - d^2) \rho_a C_a d_p U_a^2 + 1/2 \ell_b \cos \theta \rho_a C_b A_b U_a^2 \quad (5)$$

in which distances d and ℓ_b are shown in Fig. 2, ρ_a = air density, C_a = drag coefficient of the pile in air, C_b = drag coefficient of daymark boards, A_b = area of boards, and U_a = wind velocity. The first term on the right hand side (RHS) is set to zero should the pile become entirely submerged, and the second is zero when the boards are sacrificed.

During a collision, the barge contact force contributes a moment about the hinge, M_B , which is of the form

$$M_B = F_B \ell_c \sin \theta_c \quad (6)$$

where F_B = barge contact force, ℓ_c = distance from hinge to point of contact and θ_c = angle between pile direction and direction of barge force.

The general pile dynamic expressions given by Eqs. 1-6 serve as the basis for modeling the specific conditions stated in the design criteria. Equation specialization, solution approaches and computer programs based on the mathematical theory are discussed in the following subsections for each application.

PRELIMINARY ANALYSIS

Two computer models were developed to assist in the initial determination of CTPS design parameters and to provide a preliminary assessment of the system's static and dynamic characteristics. The computer program PILESTIFF calculates the initial hinge moment stiffness k_1 (see Fig. 3) necessary to meet a specified verticality requirement under static equilibrium conditions. The program PILEFREQ computes a specified pile system's undamped natural frequency.

PILESTIFF is helpful in the early stages of design when a trial value for k_1 is needed. This can be obtained by ignoring the oscillations induced by waves and solving the corresponding static equilibrium problem. Under static conditions the left hand side (LHS) of Eq. 1 is zero, while the RHS includes moment contributions given by Fig. 3 and Eqs. 2,3 and 5 (no barge contact). The wave fluid velocity contribution, given by Eqs. 4 to the relative velocity (u_r) in Eq. 3 is, however, zero. For small angles, the resulting moment equilibrium equation is easily rearranged to provide the following formula for k_1 :

$$k_1 = \frac{1}{2} \rho_w \frac{W_p}{\ell_p} + \left[\frac{1}{4} (\ell_p^2 - d^2) \rho_a C_a d_p U_a^2 + \frac{1}{2} \rho_b \rho_a C_b A_b U_a^2 + \frac{1}{4} d_p^2 C_w d_p U_c^2 \right] / \theta. \quad (7)$$

The user of PILESTIFF specifies the static angle desired, all other CTPS design parameters, and the wind, current and weight loading. PILESTIFF then uses Eq. 7 to compute the value for k_1 .

PILEFREQ calculates the CTPS undamped natural frequency which is useful in identifying potential resonant situations from wave loading or other sources of periodic forcing. The angular momentum equation is used in which wave excitation and fluid damping (Eq. 3), wind forcing (Eq. 5), and barge contact (Eq. 6) are neglected from the RHS. The remaining hinge moment (Fig. 3) and weight moment (Eq. 2) terms are linearized yielding the harmonic oscillator equation from which the natural frequency ω_0 is easily identified as

$$\omega_0 = [(k_1 - \ell_m w_\ell - \frac{1}{2} \ell_p w_p)/I_{HT}]^{1/2} \quad (8)$$

where $I_{HT} = \frac{1}{3} m_p \ell_p^2 + \ell_m^2 m_\ell + \frac{\pi}{12} \rho_w C_m d_p^2 d^3$, m_p = mass of pile and m_ℓ = mass of load.

PILEFREQ computes natural frequency (and period) using Eq. 8 and CTPS design parameters supplied by the user.

OPERATING CONDITIONS

The computer program OPPILE was developed to model pile dynamics during operating conditions. The principal use of this model is to determine whether the CTPS designs under consideration meet the verticality requirement specified in Table 2. Thus the program predicts inclination angle and hinge moment response for specified pile dimensions, hinge stiffness and load conditions.

The model is based on the angular momentum equation, Eq. 1, with piecewise linear hinge stiffness behavior (see Fig. 3) and loads due to weight, current, wave, pile motion relative to the fluid, and wind as provided by Eqs. 2,3 and 5. Since the operating restrictions require that the pile be nearly vertical, small angle approximations are used and the boards are assumed not to submerge.

Coupling with lateral motion is neglected. The possibility of transverse excitation by vortex shedding was investigated, and motion due to this source was found negligible for full scale CTPS's meeting the design criteria. Reynolds numbers ($\sim 6 \times 10^5$) exceed the critical Reynolds number so that the wake is fully turbulent with little coherent vortex street structure as discussed by Weigel (1964) and others. Lift coefficients (C_L) in this range, as reported for example by Sarpkaya (1976) and Schewe (1982), drop to the range $.03 < C_L < .20$. In addition the natural frequency of pile oscillation was found to be much slower than the shedding frequency, so the system's dynamic response is very small.

The specialized dynamic equations are solved using a Runge-Kutta numerical technique. At each time step, the along-pile integrations required by Eq. 3 are completed using the trapezoidal rule. No stability problems were encountered,

and accurate results were obtained for time steps less than a tenth of the wave period.

The program user must supply pile and hinge design parameters, wind and current velocities, water depth, and wave height, period and length. A utility program, WAVELENGTH, was written to assist the user in specifying consistent wave parameters. Specifically the program calculates wavelength for user specified depth and wave period by solving the following transcendental equation from small amplitude wave theory:

$$\lambda = \frac{gT^2}{2\pi} \tanh (2\pi d_t/\lambda) \quad (9)$$

where λ = wavelength, g = gravitational acceleration and T = wave period.

Wavelength λ is computed using the Newton-Raphson iteration method.

When input to OPPILE is complete, the program calculates and prints out time series for inclination angle θ and hinge moment M_H . Steady state response is generally achieved within 3 wave periods.

HURRICANE CONDITIONS

The computer program HURPILE was developed to model pile dynamics during hurricane conditions such as those described in Table 3 of the design criteria. HURPILE is actually very similar to OPPILE. The main difference is that HURPILE is not limited to small inclination angles. The pile may be entirely submerged and rotate up to $\theta = 90$ deg without loss of accuracy. The large angle capability, however, necessitates approximately three times the computer time. Another important characteristic is that the boards are assumed to have been sacrificed. From the user's point of view, input and output format are virtually identical to that of OPPILE.

COLLISION CONDITIONS

The computer programs COLPILE and RECPILE were developed to model pile dynamics during collision conditions. COLPILE was used to predict angular position, barge loads and hinge reactions during a head-on (worst case) barge collision. RECPILE predicts pile motion and hinge moment as the pile recovers to an upright position after the CTPS has been overrun. A discussion of COLPILE appears immediately below and is subsequently followed by a description of RECPILE.

The complete collision, as modeled by COLPILE, consists of a sequence of processes. Initially there is impact of the top of the barge bow with the pile, then sliding of the pile occurs along the top of the bow and the bow face. Next there is impact of the bottom of the bow rake with the pile followed by sliding along this point. Lastly, the tip of the pile slides along the barge bottom before being released. The program analyzes these processes in chronological order.

The major assumption throughout the collision analysis is that because the barge is so massive, it's motion is essentially unaffected by the collision. Barge speed therefore remains constant. In addition, it is assumed that the boards are sacrificed, and wind and wave forces are considered negligible in comparison with collision forces.

The constant barge speed condition enables the pile kinematics to be analyzed independently of the forces involved. Since the horizontal velocity component of the pile contact point must equal the barge speed, θ , $\dot{\theta}$ and $\ddot{\theta}$ may be determined as function of time from the problem geometry.

Next, CTPS dynamics are analyzed using the rate of change of angular momentum equation, Eq. 1, in which the moment sum includes the hinge moment M_H , the gravitational moment M_G , the fluid force moment due to current and relative pile motion (no waves) M_C , and the barge moment M_B . Using the kinematic results, M_H , M_G and M_C are evaluated using Fig. 3, Eq. 2 and Eq. 3, respectively. Similarly the rate of change of angular momentum term, $I_H \ddot{\theta}$, in Eq. 1 is calculated from the barge kinematics. Eq. 1 is then used to compute barge moment M_B .

Note that unlike OPPILE and HURPILE, Eq. 1 is not "solved" in the usual sense of evaluating $\theta = \theta(t)$. The constant barge speed constraint and geometry determines angular position independently, and Eq. 1 is simply used to calculate the unknown barge moment, M_B , term.

Using the M_B result and collision geometry, Eq. 6 is applied to compute the barge contact force F_B . Having determined F_B , the linear momentum equations (Newton's Second Law) are applied vertically and horizontally to evaluate the hinge reaction forces R_V and R_H , respectively.

While the collision analysis described above is theoretically correct throughout the sequence of collision processes, it is convenient to modify the approach somewhat at the two instants of impact. At these times, the weaker non-impulsive loads due to fluid motion and the hinge moment are neglected, and COLPILE uses the impulse-momentum form of the reduced equations of motion. Eq. 1, for example, becomes

$$(I_H \dot{\theta})_{t_f} - (I_H \dot{\theta})_{t_i} = \sum_i^{t_f} M dt \quad (10)$$

where t_i and t_f are initial and final times, respectively, bracketing the impact process and the RHS includes only impulsive moments. The same steps previously outlined are taken yielding results for barge moment impulses and barge and hinge reaction force impulses.

The program user must supply COLPILE with CTPS design parameters, barge dimensions and speed, current velocity and water depth. The user also specifies the interval between times for which output is desired. Results consist of angle, hinge moment, barge moment (or moment impulse), barge force (or force impulse), and reaction forces (or force impulses).

The program RECPILE is used to model pile recovery after being released from beneath the barge bottom. Initial position is specified by the user (from COLPILE results), then RECPILE calculates angular motion and hinge moment as the pile returns to the upright position. RECPILE is actually a modification of

HURPILE thus making use of HURPILE's large angle capability. Changes include omitting wind and wave excitation and allowing the user to specify the initial conditions for θ .

The user of RECPILE must input the CTPS design parameters, current and water depth as well as the initial inclination angle. The program responds by calculating and printing out time series for θ and M_H .