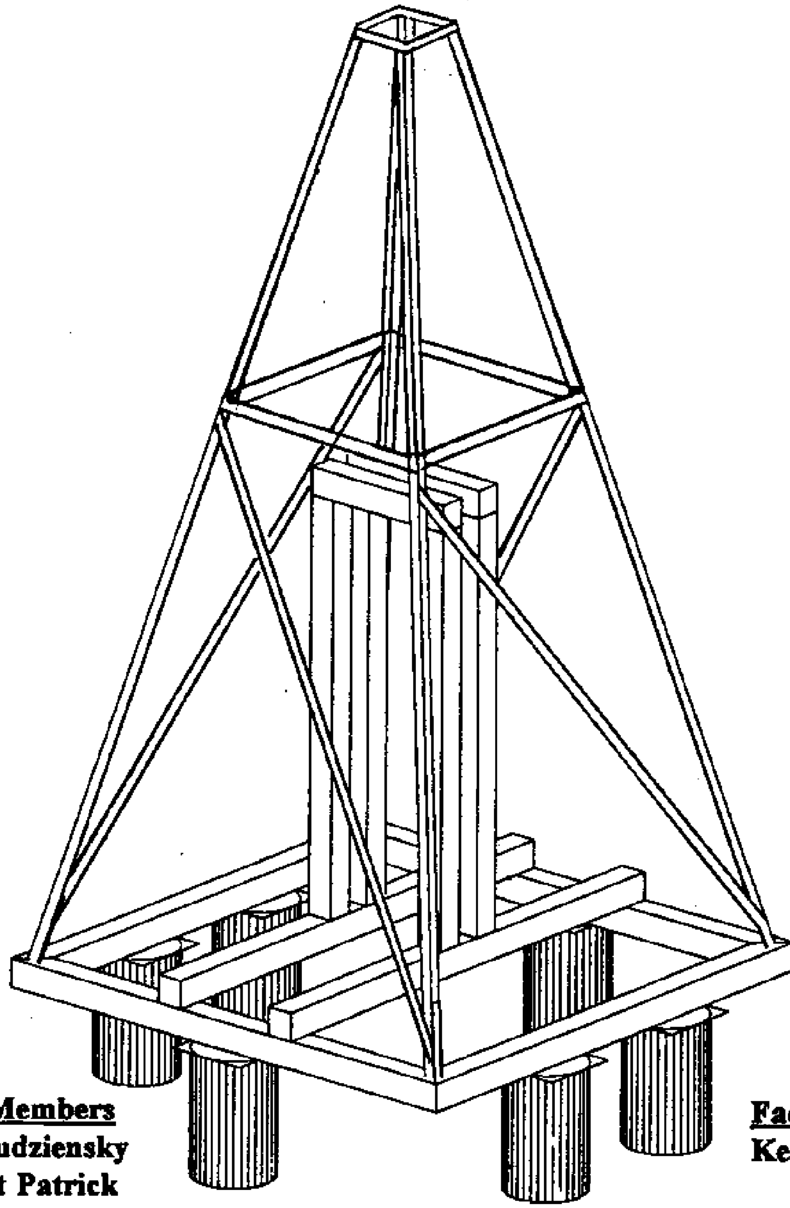
Penetrometer Pushing Platform

UNIVERSITY OF NEW HAMPSHIRE

Tech 797



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Executive Summary

In situ testing of soils provides a more accurate profile than core sampling. A piezo-cone penetrometer provides data in this manner. The insertion of such a device is an intricate procedure. It can be performed on land with the aid of machinery and human supervision. Testing of ocean soils adds many problems since human presence at the penetration site becomes much more difficult.

A solution to the ocean soil penetration difficulties requires several systems. The necessary components include a framework to hold all components, a drive system to push the penetrometer into the soil, and an anchoring system to provide the necessary reaction force so the frame will not move off of the ocean floor.

The frame was built in four sections, which are bolted together before use. This helps facilitate transport to and from the boat. It stands roughly fifteen feet tall when assembled and supports the thirty foot section of pipe attached to the penetrometer. This support is provided at the top of the frame through the use of two rollers. These rollers allow motion in the vertical direction, but constrain the pipe from moving laterally within the frame.

The drive system is run hydraulically, using two pistons to drive the pipe. The difficulty encountered with this configuration is that a clamp is necessary for the pistons to transmit force to the pipe. Therefore, a machine collet was manufactured to fit the specifications of this application. Two other hydraulic pistons are used to activate this clamp.

Six suction anchors of twelve inch diameter are to provide the necessary reaction force. They are bolted to the bottom of the frame so they can be detached for easy transport. They are evacuated by use of a pump on board the boat with a hose running from the anchors to the pump. They are released by reversing the pump and pushing water back into the anchors.

Project Description:

A. Background:

Soils analysis is a major function of Civil Engineering. One method of testing soils is core sampling. This method's validity has been questioned due to the removal of the soil sample from its enviroment to the lab. In situ testing of soils with a piezo-cone penetrometer (figure 1) provides very accurate information regarding the soil's properties. This accuracy of soil analysis is desired for ocean floor soils. Therefore, a device to deploy and recover the penetrometer in the ocean environment is necessary.

B. Considerations:

Several constraints and requirements had to be considered when designing the penetrometer pushing platform:

1) The system must provide up to five tons of force for driving the penetrometer.
2) The system must provide a constant rate of descent of 2 cm/sec for the penetrometer.
3) The system must be deployable from the UNH Gulf Challenger.
4) The system must be self sufficient: i.e., no diver should be required for testing.
5) The system must be able to penetrate soil up to thirty feet down from the ocean floor.

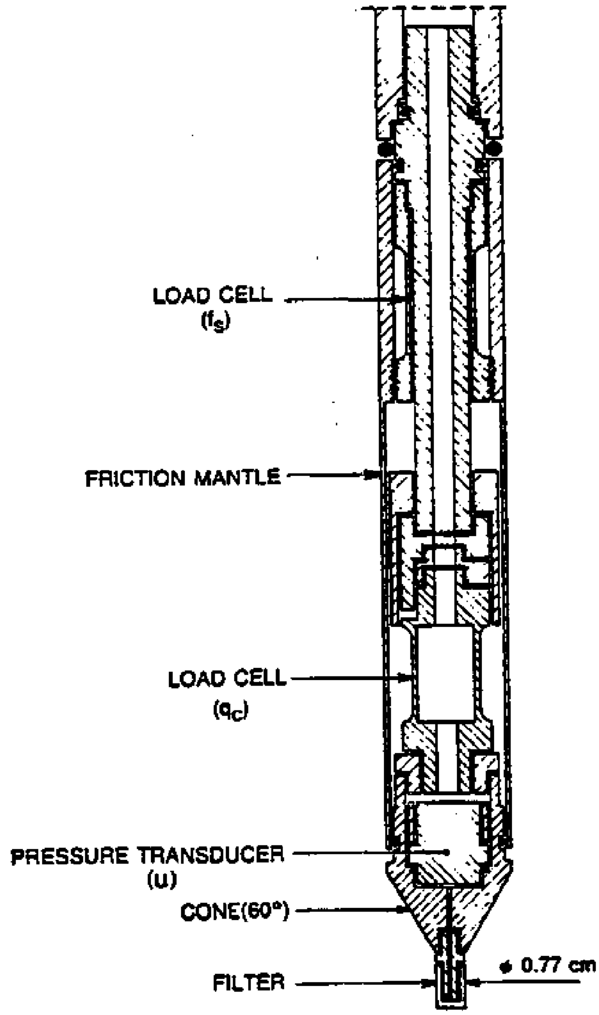


fig. 13 - THE WISSA PIEZOCONE (BALIGH ET AL., 1981)

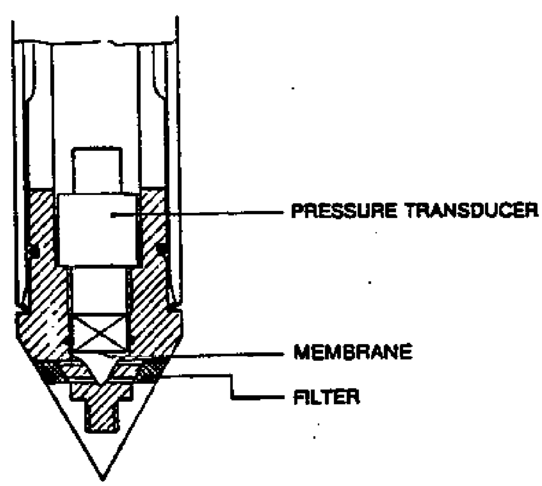


fig. 14 - THE FUGRO PIEZOCONE (AFTER ZUIDBERG ET AL., 1982).

Figure 1

Design Approach:

A systematic approach was taken to the solution of this design problem. The design of the entire system was divided into several subsystems. Group members were split up to work on the subsystems so the work could be divided evenly. The groups stayed in constant contact so that no system would impede another systems use. Also, ideas were shared between groups so that the best design possible could be achieved.

The project was divided into three subsystems initially. These were the frame, drive, and anchoring systems. The frame's purpose is to connect the anchoring system to the drive system and support thirty feet of pipe. The drive system is required to provide the necessary five ton driving force to the pipe attached to the penetrometer. The anchoring system's purpose was to rigidly attach the platform to the ocean floor.

As the project progressed, it became necessary to implement several other subsystems. These included the development of a guide, to insure that the pipe did not move within the frame. A braking device to prevent the pipe from falling back into the hole during extraction, and a method of launch, recovery, and transportation had to be developed.

The time table (Figure 2) shows how the given time to completion was allocated, and how the time was actually spent.

The proposed budget (Figure 3) gives a breakdown of how the funds were distributed.

Project Timeline

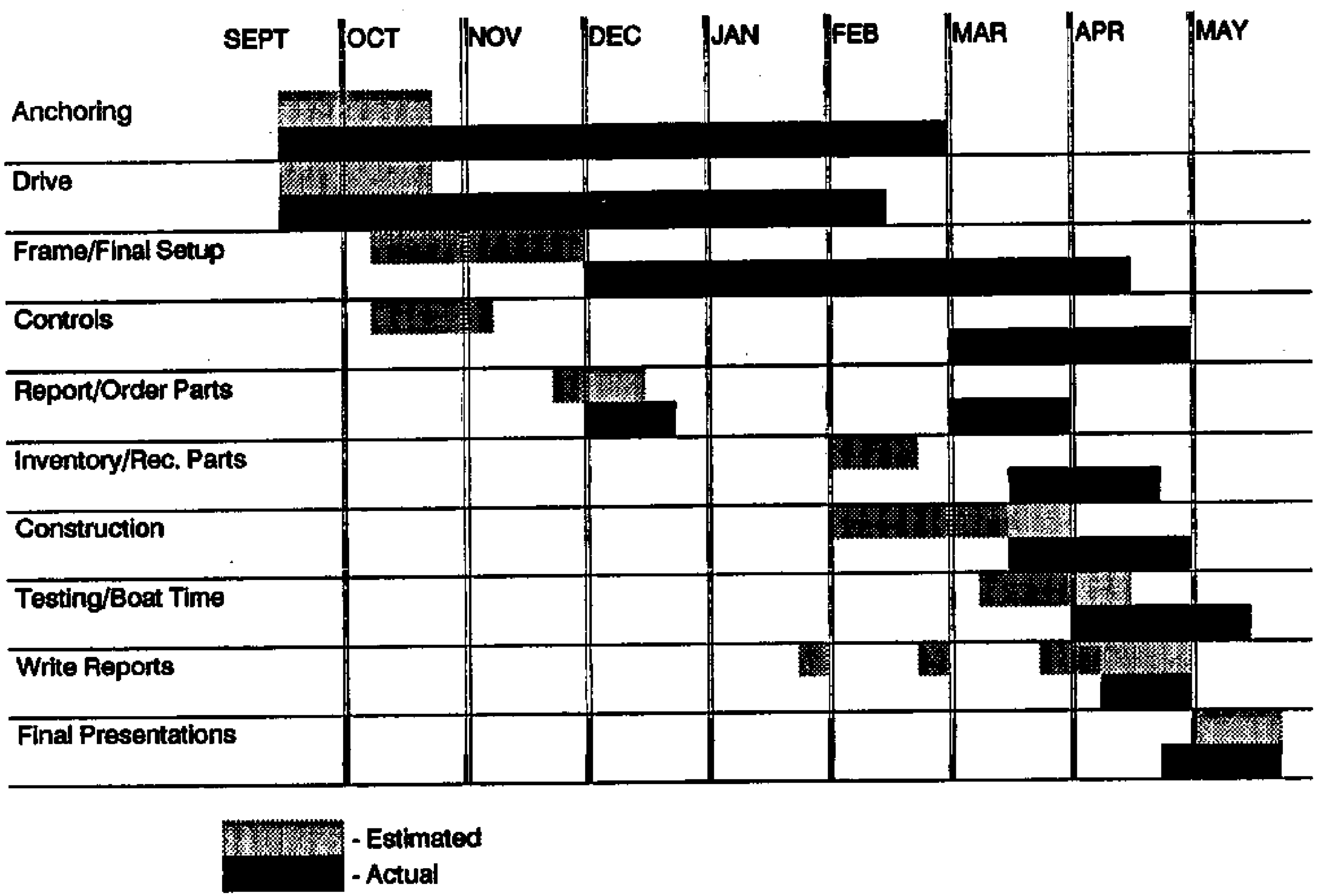


Figure 2

Budget

Division	Need	Price Estimated	Price Actual
	Anchors		$384
Anchors	Pump	$700	0
	Hose/Fittings		150
	Pistons		200
Drive System	Hydraulic gear	700	450
	Control		300
	Materials	400	500
Frame	Fabrication	200	0
	Misc. Hardware	100	300
Support	Photocopies	100	0
	Phone	0	0
	TOTAL	$2,300	$2,284

Figure 3

System Components:

A. Introduction:

Design of this penetrometer pushing platform is broken into systems. First, a frame system is necessary to support the penetrometer and the thirty feet of pipe to be inserted. Next, a drive system to preform the insertion and the extraction of the penetrometer is needed. To keep the platform from rising off the ocean floor during insertion an anchoring system is also needed. Finally, this platform must be easily transported and deployed, therefore a launch, recovery, and transportation system is also considered (figure 4).

B. Anchoring System:

1. Criteria:

The design criteria for the penetrometer pushing platform anchoring system requires that five tons of anchoring reaction force be generated to offset the input force of driving the penetrometer. Numerous preliminary designs were suggested (figure 5). From these choices ideas were narrowed down into two basic designs, augers and suction anchors.

Block Diagram of System Approach

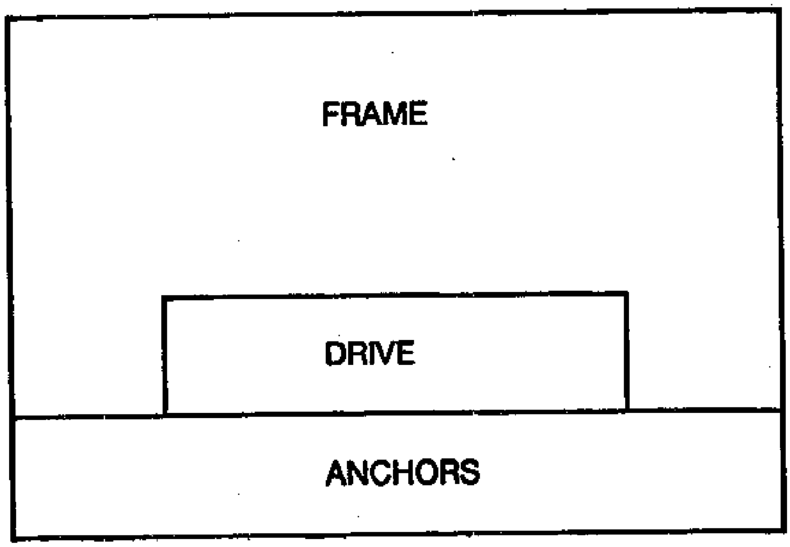
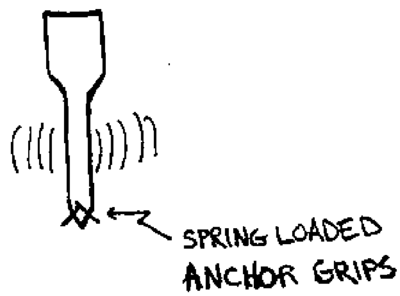


Figure 4

System 1

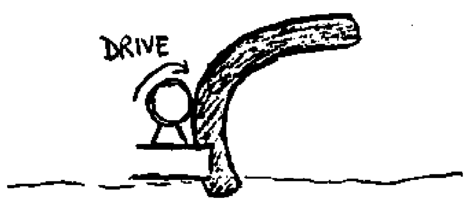
Vibrating Anchors



- Not for use in silty Clay
- Need for underwater motors

System 2

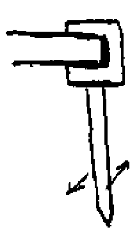
Banana Anchors



- Needs to be used in conjunction with Augers
- Need for many underwater motors

System 3

Drilling Anchors



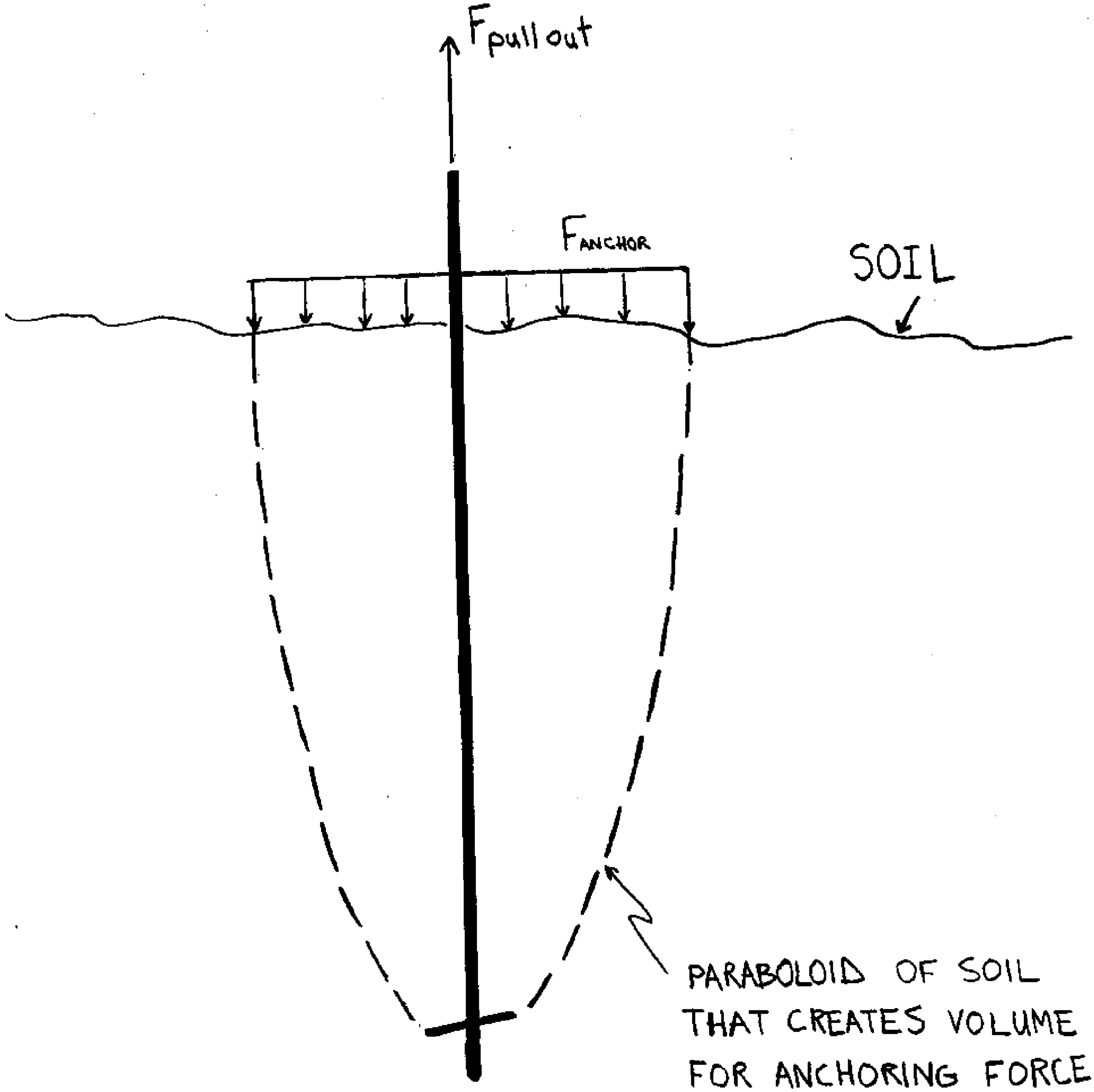
- Soil disruption
- High cost
- Motors underwater

Figure 5

2. Augers:

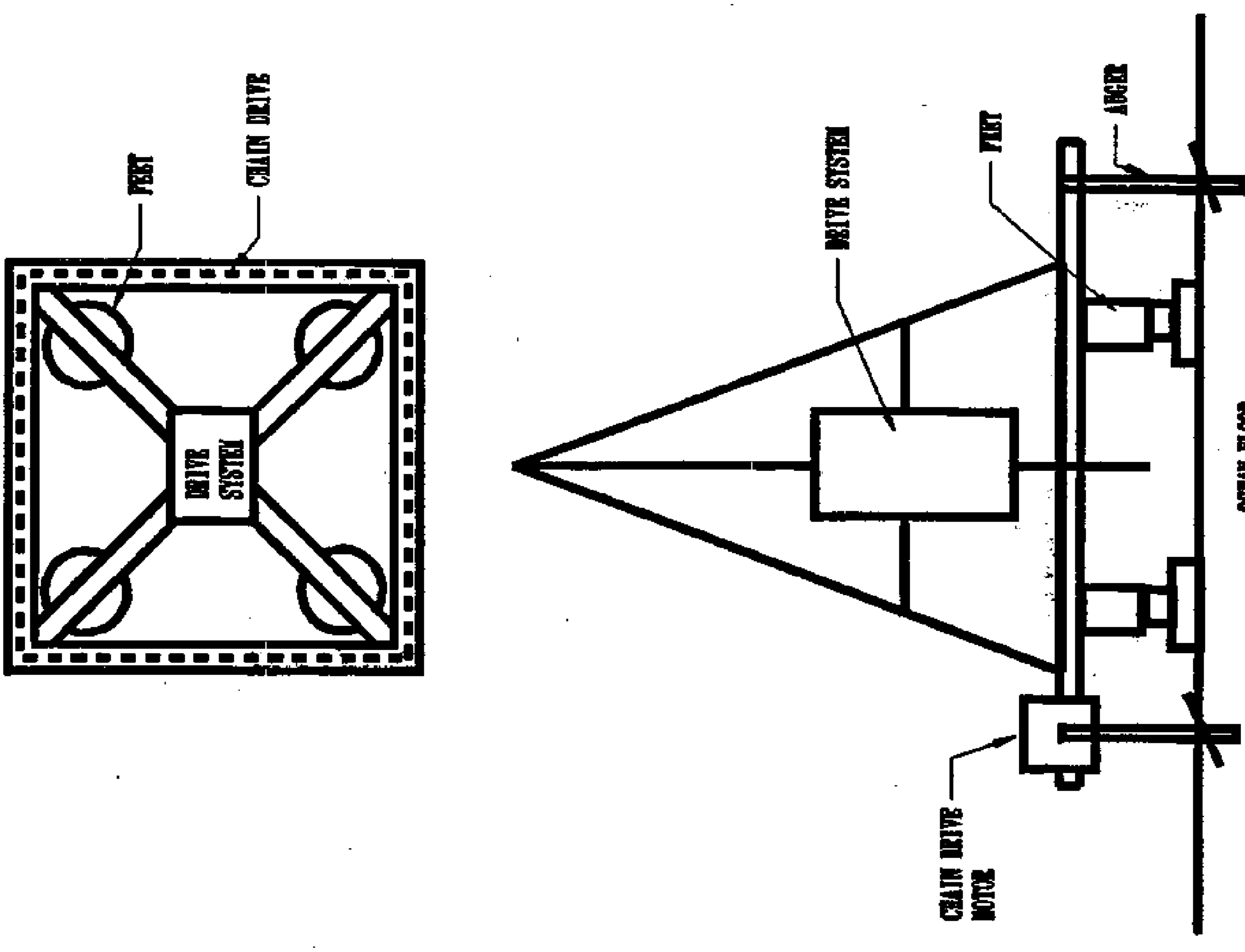
An auger anchor is thin circular shaft with a helical tier at one end (figure 6). When screwed into the ground, it generates anchoring force by holding soil in the form of a paraboloid above the tier. The P^3 design team first viewed a land based penetrometer at Pease International Tradeport in September 1993. This system used augers for anchoring. Due to low soil resistance and ease of accessibility, these augers were screwed into the ground under human power. For an underwater platform human installation is not possible. Although it was concluded that the same approach would work if installation could be conducted by a motor with the output shaft attached to the auger. To keep costs low, a chain drive around the perimeter of the base of the platform was suggested. This would provide rotational motion for all of the augers while requiring only one motor (figure 7). Upon the presentation of this design, concerns were raised about cases when the soil under each auger would not be the same density. This would affect the input torques created by the soil on the chain drive. If these torques were not constant, a moment would be applied to the chain which could produce plastic deformation or failure of the chain or motor. It was then decided that each auger would have its own encased motor and the chain drive would be eliminated. With this decision, the cost of the using augers increased fourfold.

A system was designed to handle the landing of the platform. After deployment, the augers would each be spinning by the rotation of their individual motors. Pistons mounted to the underside of the platform base would hang lower than the augers while uncompressed. Upon landing, these



AUGER ANCHOR

FIGURE 6



Chain Drive

Figure 7

pistons would compress under the weight of the platform. While compressing, the spinning augers would engage the soil and begin generating anchoring force. Upon full compression of the pistons, the augers would reach their full input depth (figure 8).

At this point, actual values of anchoring force generated by augers were investigated. A comprehensive anchoring test was set up and carried out at the Jackson Estuarian Laboratory in Durham, N.H.. Using a three hundred pound scale, a block and tackle arrangement, and a stationary tripod, anchoring forces were measured by pulling augers imbedded in soil out of the mud flats at the Estuary (figure 9). With the results of this test, a mathematical model relating anchoring force to depth of anchor submerged was constructed. This was done by using the relation that the auger anchoring force is generated by holding soil in the form of a paraboloid of volume above the tier. Knowing the length of the augers tested, the volume of the paraboloid was calculated using multidimensional calculus (Appendix A). Using the determined force value from the test, density was calculated using equation 1.

$$p = F / (V g) \qquad (1)$$

After finding the value for density, the submerged length of the auger can be varied, thus changing the volume of the paraboloid and the holding force. This was done to optimize the anchor length (figure 10). Results from this model suggested that either auger or tier extensions or a combination of both would be necessary to meet the anchoring force requirements.

It was now concluded that auger lengths would be to long to hang off of the bottom of the platform without concern of snapping under the impact of

ANCHOR FOOT DETAIL

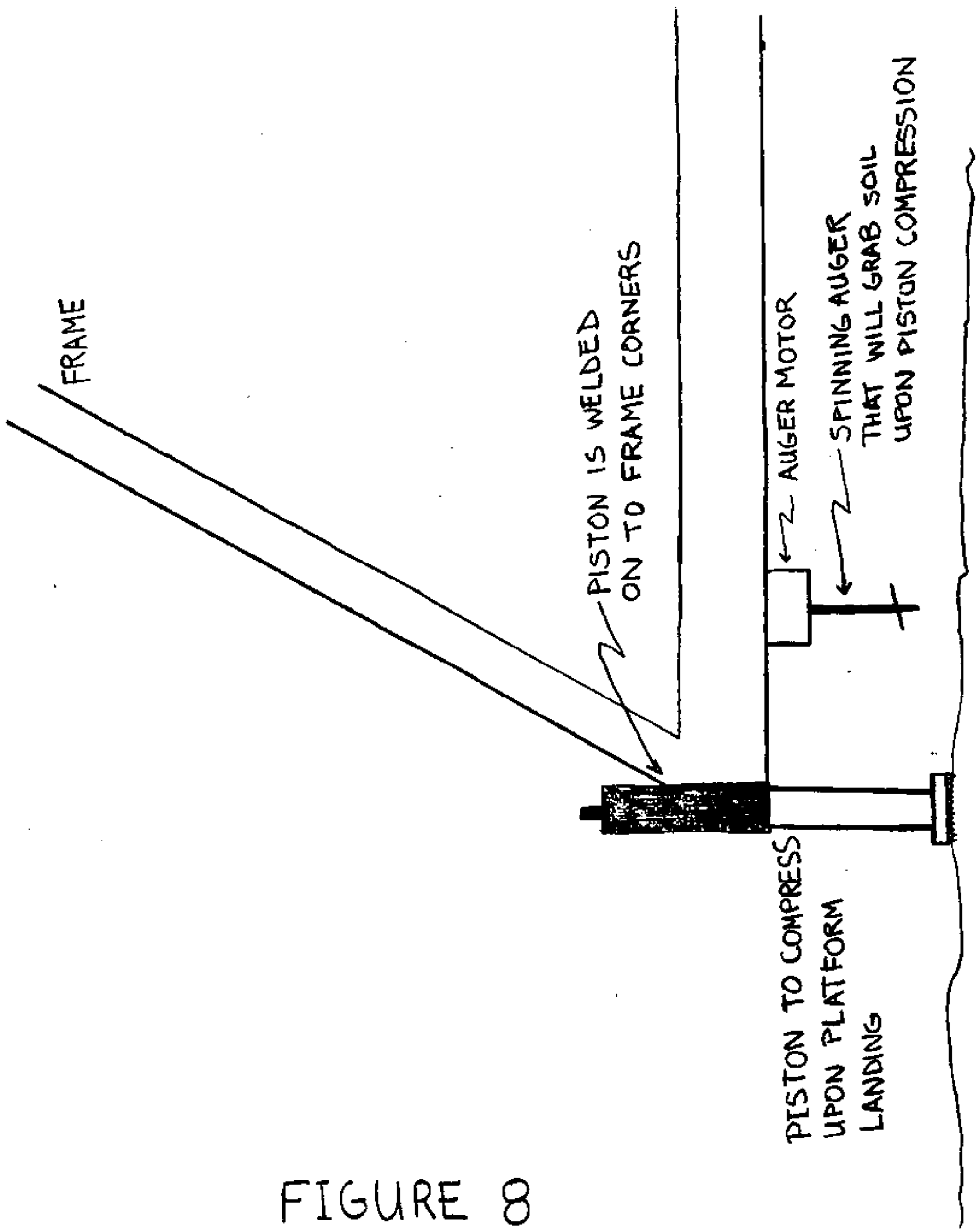


FIGURE 8

AUGER TEST APPARATUS

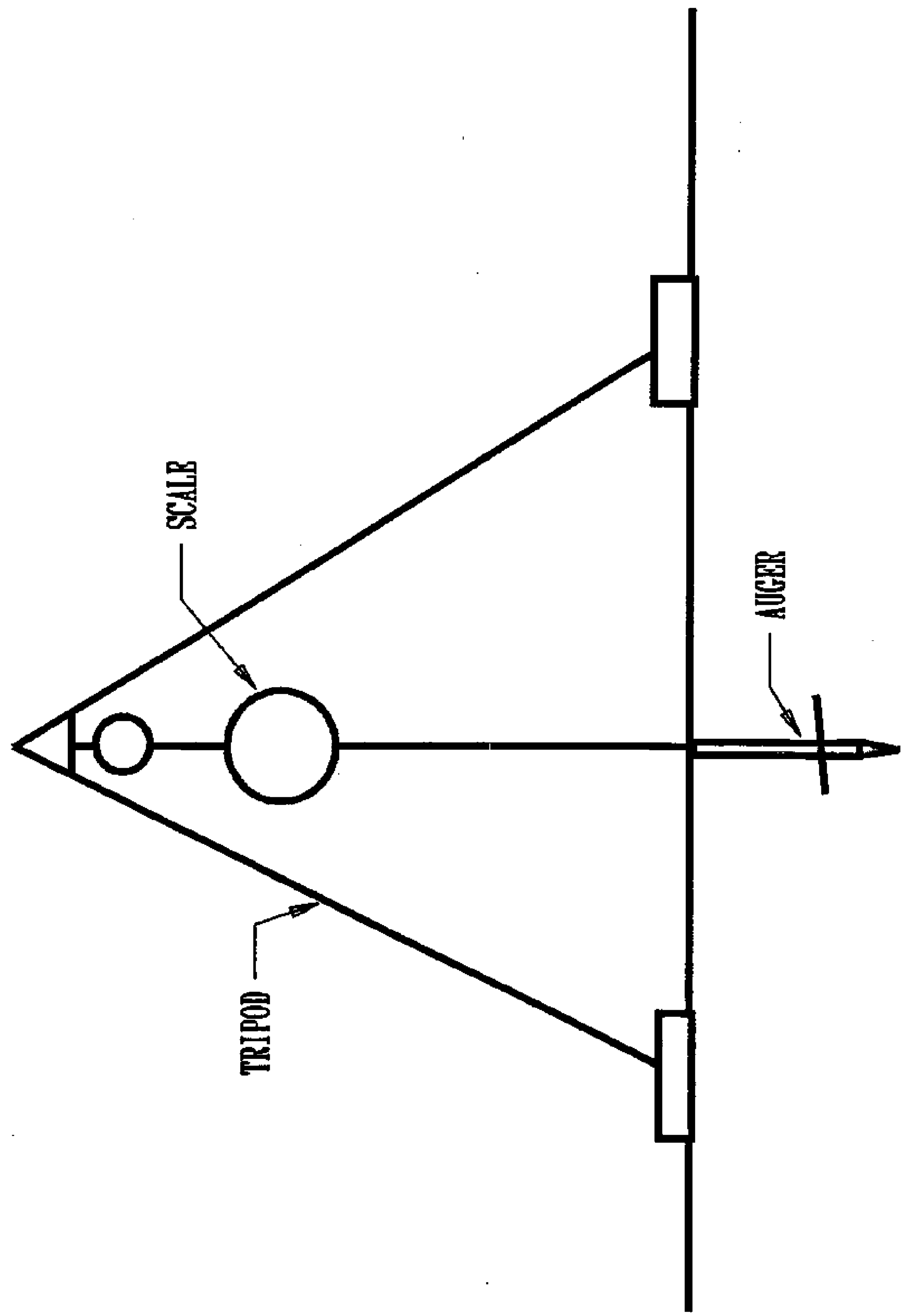


Figure 9

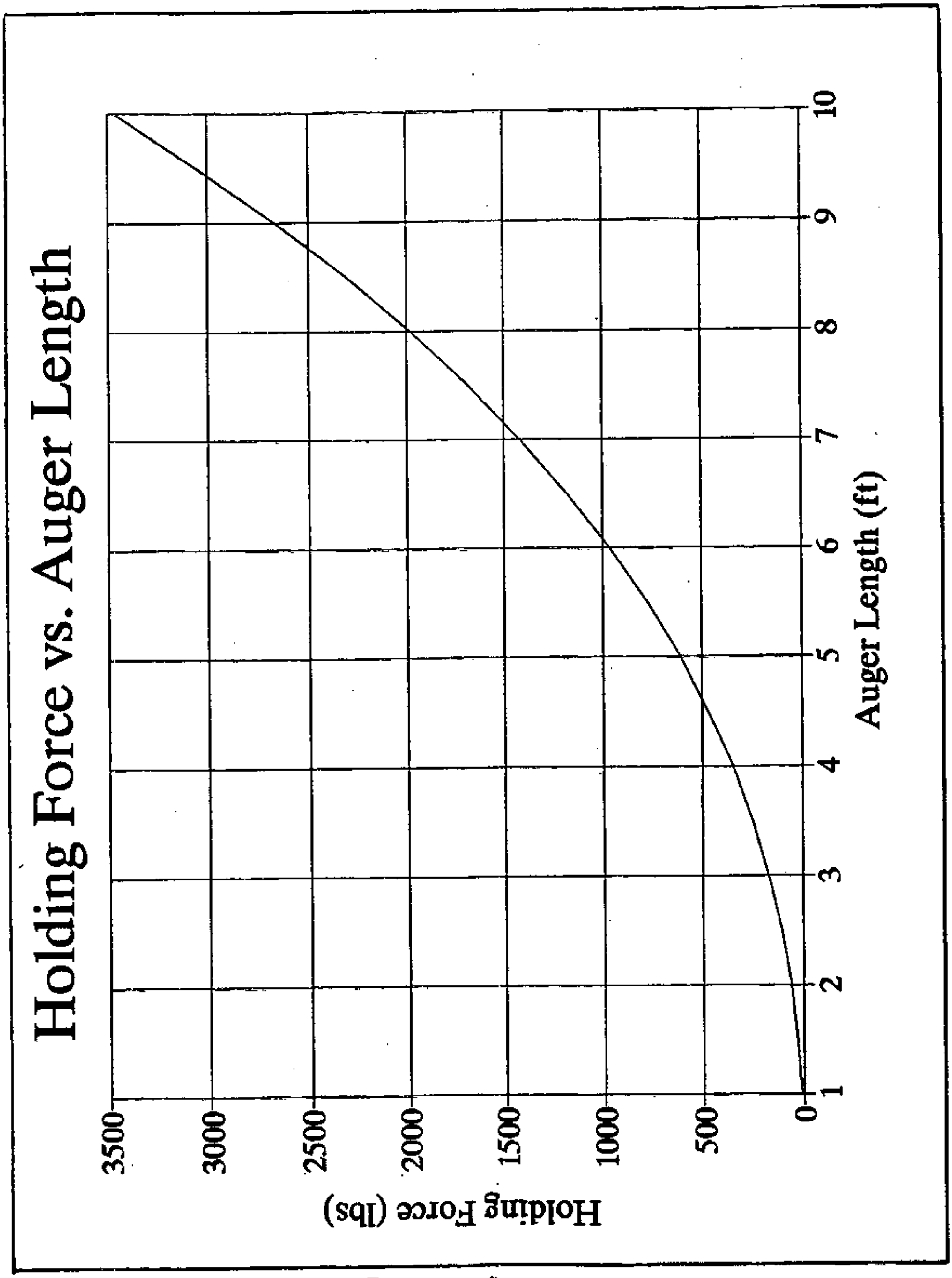

Holding Force vs. Auger Length
Holding Force (lbs)
3500
3000
2500
2000
1500
1000
500
0
Auger Length (ft)
1
2
3
4
5
6
7
8
9
10

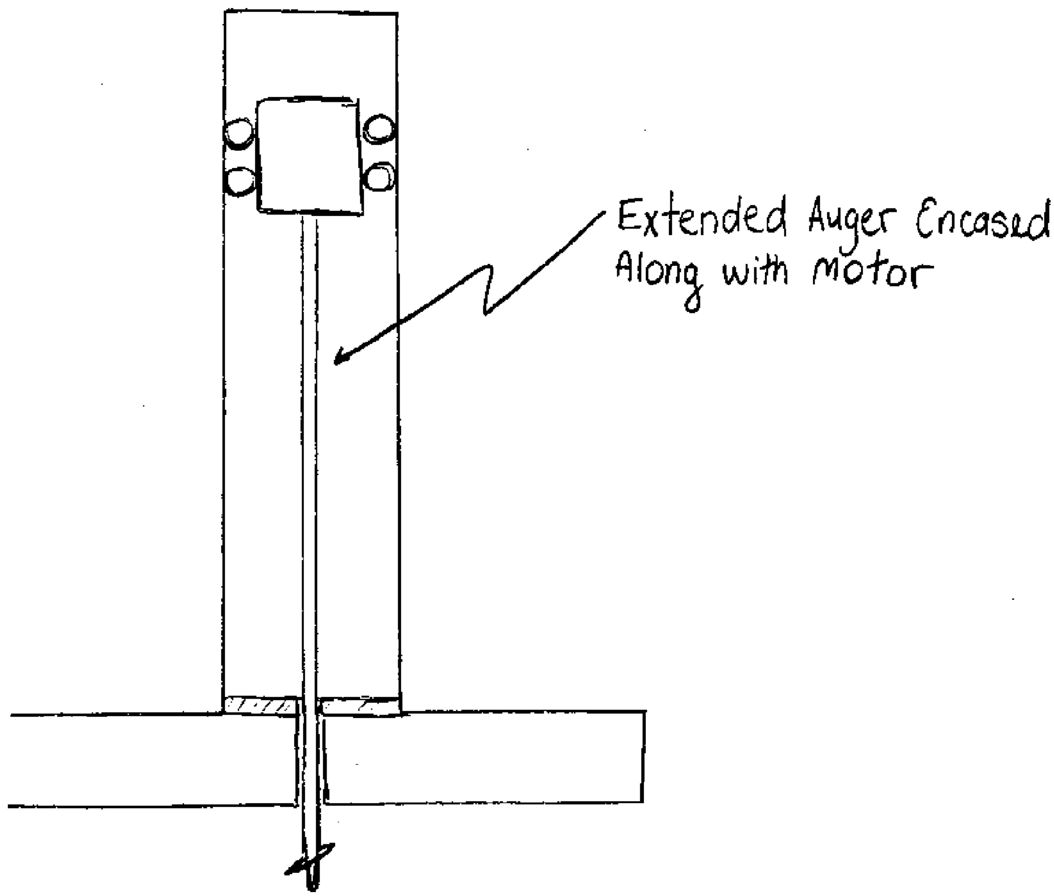
Figure 10

landing. To combat this problem motors sliding on rods were suggested (figure 11). This would allow some of the length of the auger to be retracted into the frame of the platform. The problem here is that the motors must be on sliders and encased to be protected from the muddy environment. Also, auger motion would now have to be both linear and rotational. The idea of four separate augers each operating with its own motor was getting too complex. This lead to the investigation of suction anchors as a new alternative

3. Suction Anchors:

Suction anchors are cylinders that are open on one end. The open end imbeds itself into the soil creating a desired suction area. A positive displacement pump attached by hose to each anchor pumps out the water from the silty soil below (figure 12). This action generates a holding force. To releive the holding force, water is pumped back into each anchor thus breaking the seal. Use of suction anchors is restricted to silty or clay soils that contain a significant amount of water to be pumped away.

After being introduced to the idea of suction anchors by the Sediment Coring Device report done by UNH students in 1988, the idea was investigated further by the P^3 design team. To learn more about the workings of a suction anchor and to calculate the anchoring force values that it could generate two suction anchor pull out test were set up. Similar to the auger pull out tests, they were carried out a the Jackson Estuarian Laboratory. Various sized cylinders were tested. From the results obtained, it was learned that suction anchor holding force is proportional to the volume of the cylinder submerged (figure 13). With this new relation, calculations were done to find



Auger Extensions

Figure 11

Suction Anchor

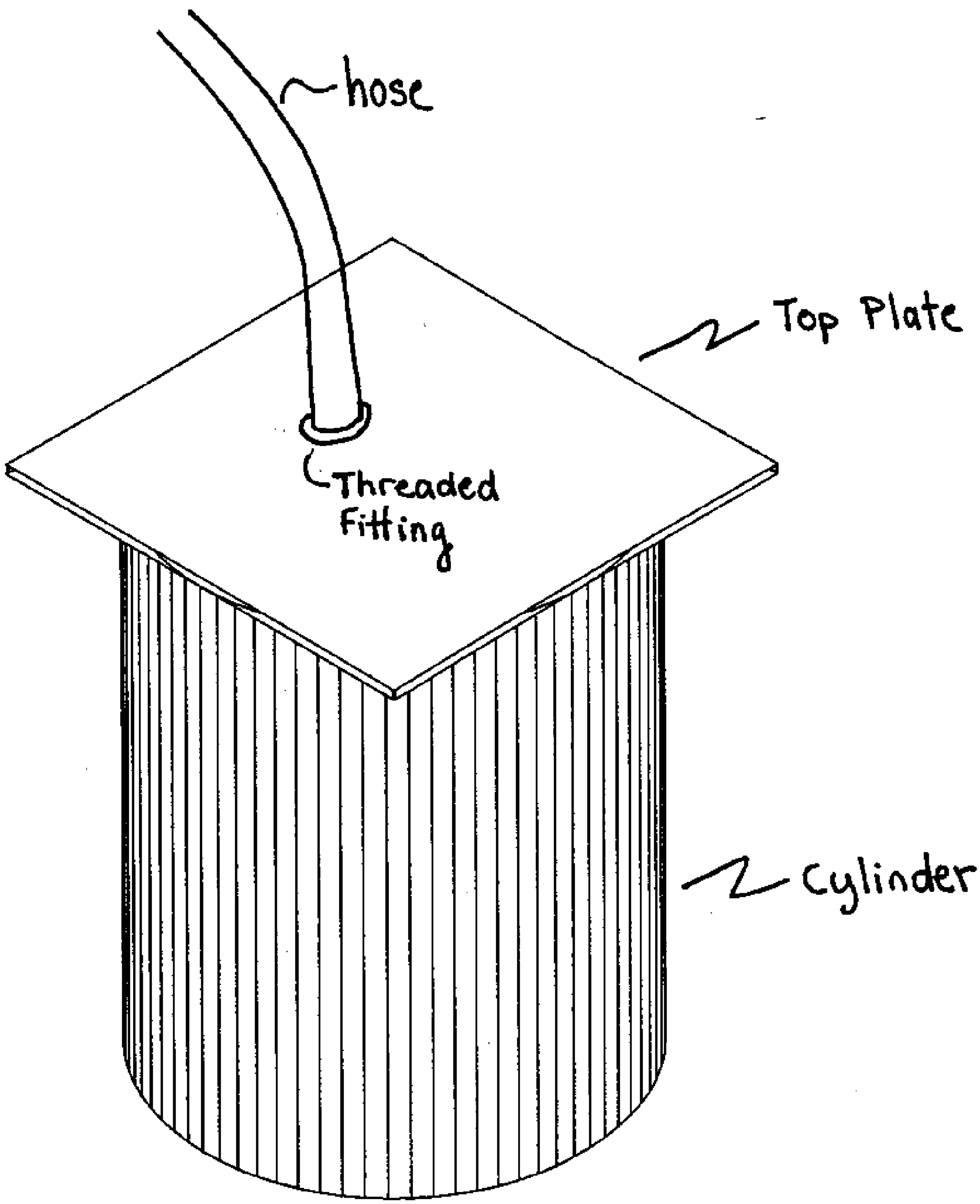
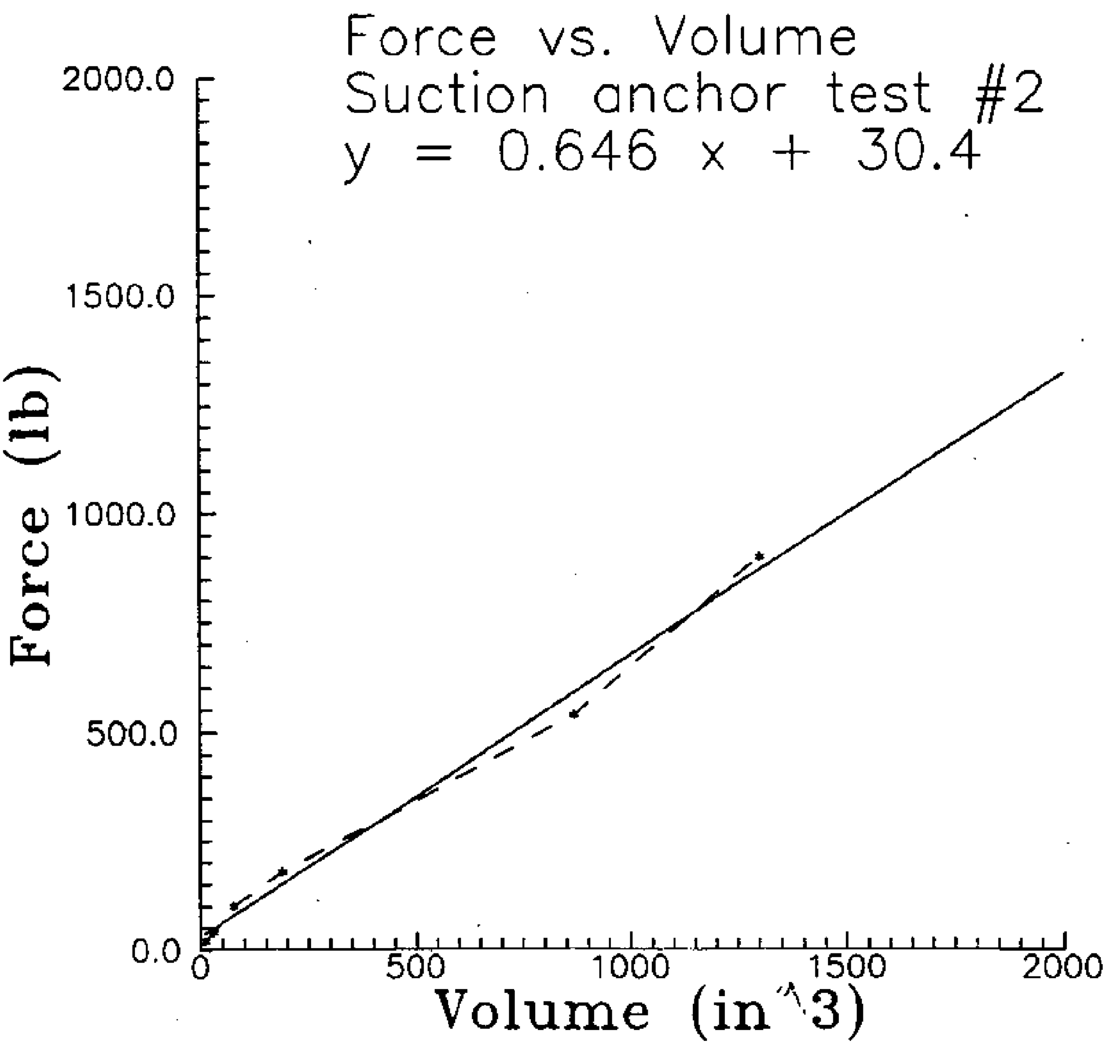


Figure 12



Force vs. Volume
Suction anchor test #2
y = 0.646 x + 30.4
2000.0
1500.0
1000.0
500.0
0.0
Force (lb)
0
500
1000
1500
2000
Volume (in^3)

Figure 13

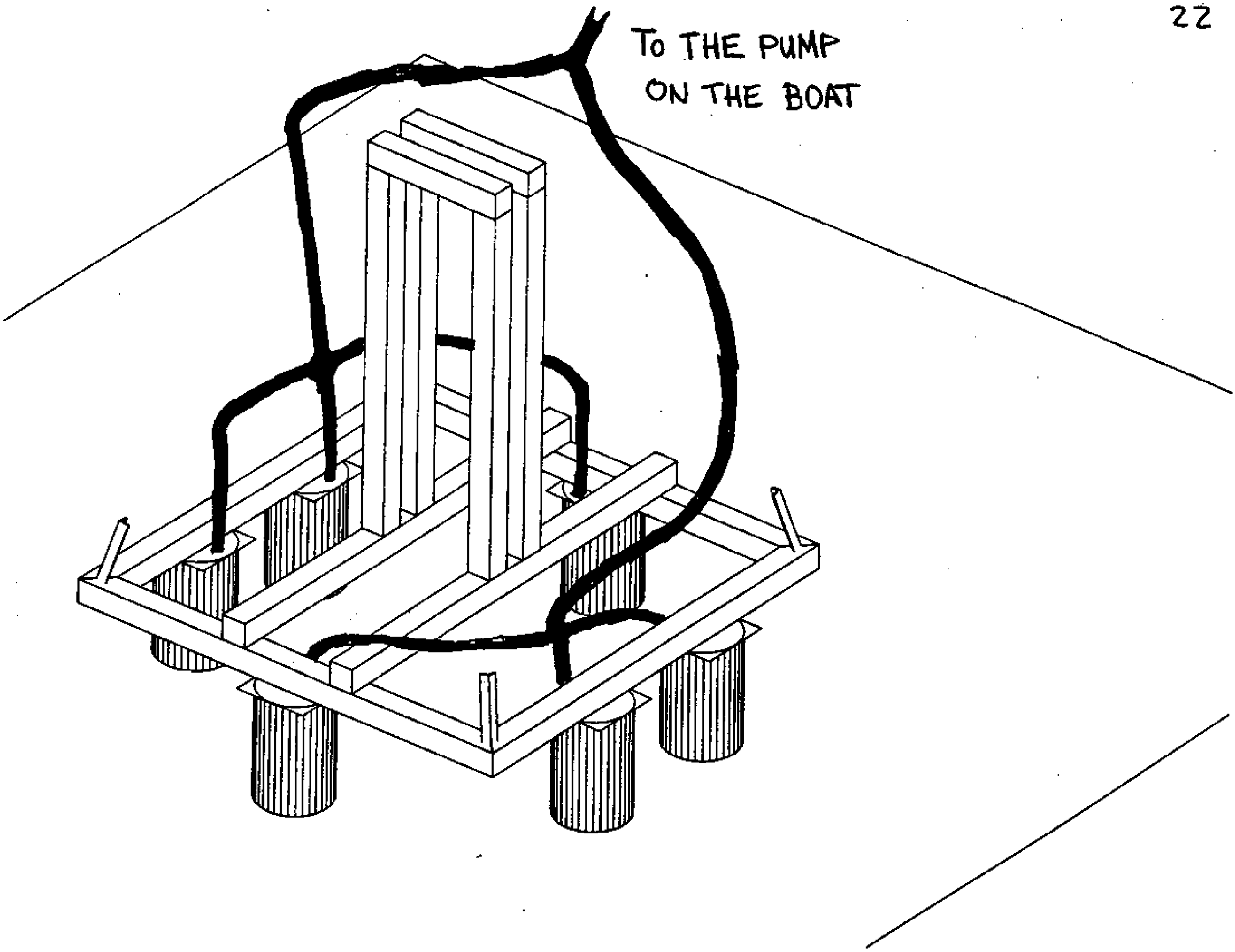
the optimal geometry for this application (Appendix B).

Results showed that six cylinders of twelve inch diameter and fifteen inch length gave the necessary volume to produce the anchoring force required. This also met the stress requirements (Appendix C). These anchors needed a top piece which would overhang along the cylinder edge. This would keep the entire anchor from sinking into the sediment under the weight of the platform. A hose used for suction was attached to the top sheet on each anchor. These six hoses would all come together to one at the top of the platform. Therefore, only one teather of hose would go from the platform to the pump on the boat (figure 14).

Upon analyzing the design alternatives for anchoring, it was decided that suction anchors would be used. Although their use would be limited by a teather of hose to the boat, suction anchors were the most simplistic and low cost way to meet the platforms anchoring requirements.

4. Fabrication:

These anchors were fabricated by rolling 10 gage steel into cylinders, welding the cylinder seams, and then welding the tops on. This work was done at FedCo. Manufacturing in Somersworth , N.H. . A hole was then drilled in the top of each anchor and a threaded fitting was inserted to allow for a standard hose attachment. The rest of the hose was then cut and fitted so that only one piece went from the platform to the pump on the boat. The pump was supplied at no charge by UNH. These anchors were painted for rust protection and bolted to the bottom of the platform base for easy removal.



Configuration of
Anchors & Hose
on the base.

Anchor/Hose Configuration

Figure 14

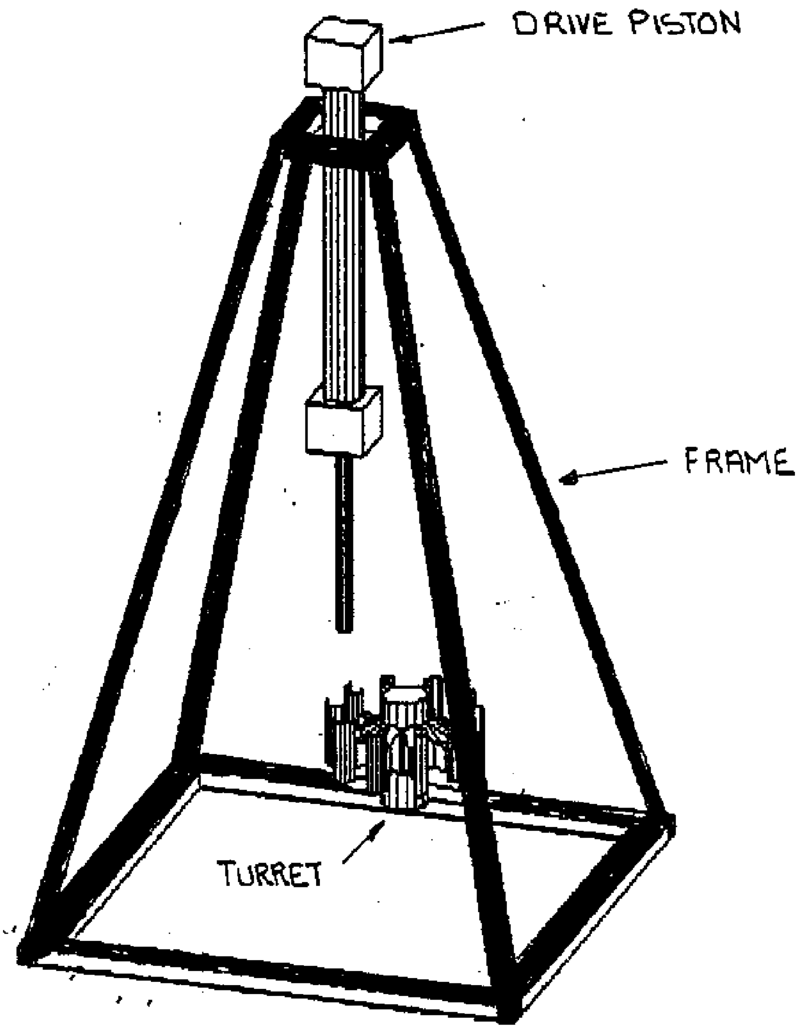
B. Drive System:

1. Criteria

Before drive configurations could be analyzed in any detail, it had to determined whether the drive pipe is to be assembled remotely and automatically, or on the surface by the operators. The plausible drive setups for each case would differ markedly from one another, as well as have a considerable variation of possibilities within each case.

The concept of remotely assembled drive pipe (figure 15) is arguably the most effective and size conservative method of performing the under water penetration operation. Pipe sections would be stored, assembled, driven, pulled out, disassembled, and replaced in the holders, automatically. The highlight of remote assembly is the elimination of the bulky, elongated framing system needed in the case of pre-assembled pipe. A remote assembly system would be the simplest to operate, relative to the user, and easiest to transport and handle. Due to the complexities, however, the scope of this design is far beyond the time and resources allocated for a final product. The pre-assembled drive pipe method is the only realistic alternative, and all drives are considered to fit this scheme.

In examining the many, varied possibilities of performing the penetrometer driving operation, hydraulics and pinch rollers have shown to be the easiest to implement and most effective in use, considering the harsh, dirty environment.



REMOTE PIPE ASSEMBLY

TURRET SYSTEM

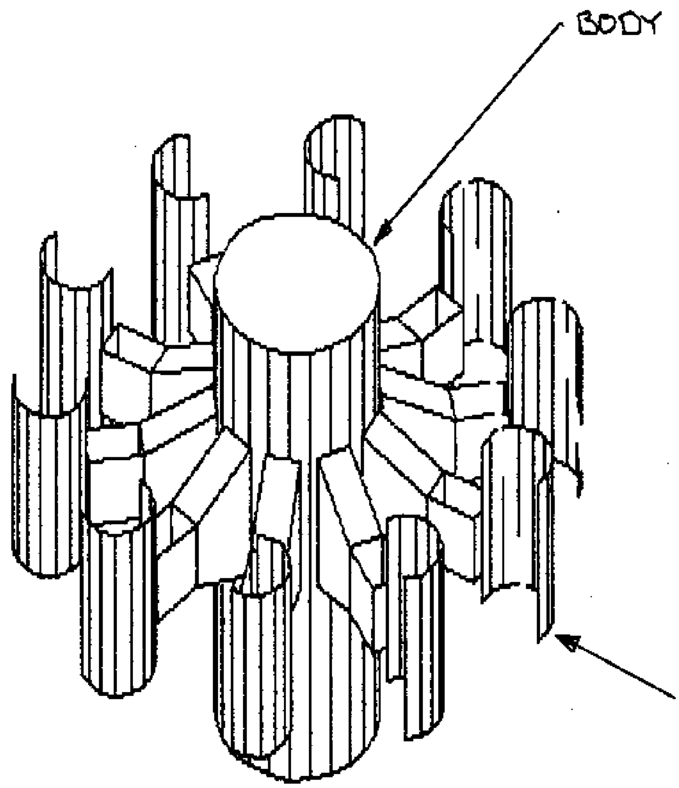
Figure 15

2. Remote Assembly

To perform the operations as described above, a revolving turret system (figures 16-19) could be employed. The idea is to store the three foot pipe sections on the arms of the turret, which would rotate accordingly to repeatedly position pipe sections for assembly and drive, working in reverse for penetrometer retrieval. The holding ability of the arms would be provided by a spring loaded push-rod assembly, actuated by a tapered block located on the shaft of the turret body. The tapered block would be actuated, in turn, by a damped telescoping rod fixed to the drive piston, supplying a constant downward or upward force to the spring loaded, bearing mounted shaft of the turret body. The pipe sections have male and female tapered thread ends, facilitating assembly by means of spinning them together. A small dc motor connected to the end of the drive piston rod would secure a threaded coupling to the pipe section positioned by the turret, the holding arm would release and rotate out of the way, allowing the new section to be fastened to the pipe already in the ground.

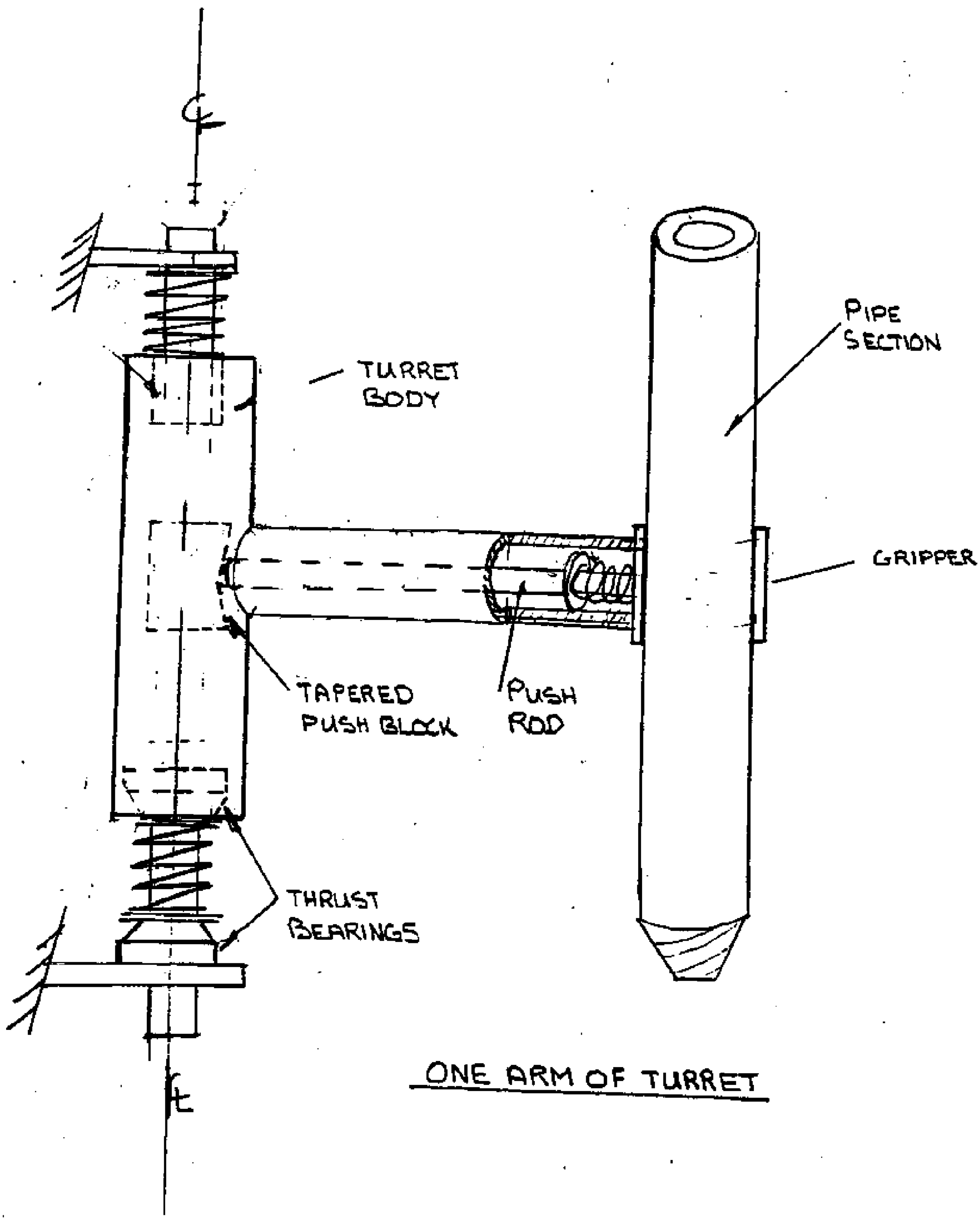
3. Pre-Assembled Pipe Drive:

As opposed to remote assembly, the pre-assembled drive system, chosen as final design, has considerably less moving parts. The idea is to load all thirty plus feet of drill pipe into a frame capable of supporting it in the ocean environment. The relative simplicity of the possible drives (discussed below) needed to push the pre-assembled pipe makes it the only realistic option, considering design time and budget.



TURRET

Figure 16



ONE ARM OF TURRET

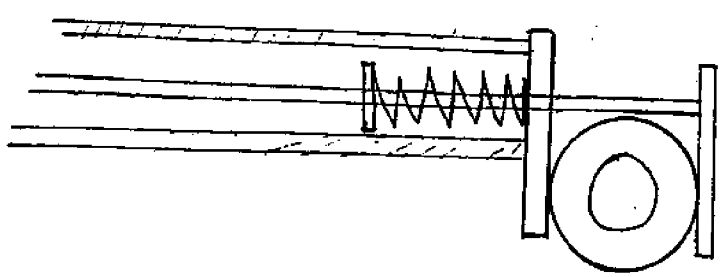


Figure 17

CLAMPING & RELEASING ACTUATION

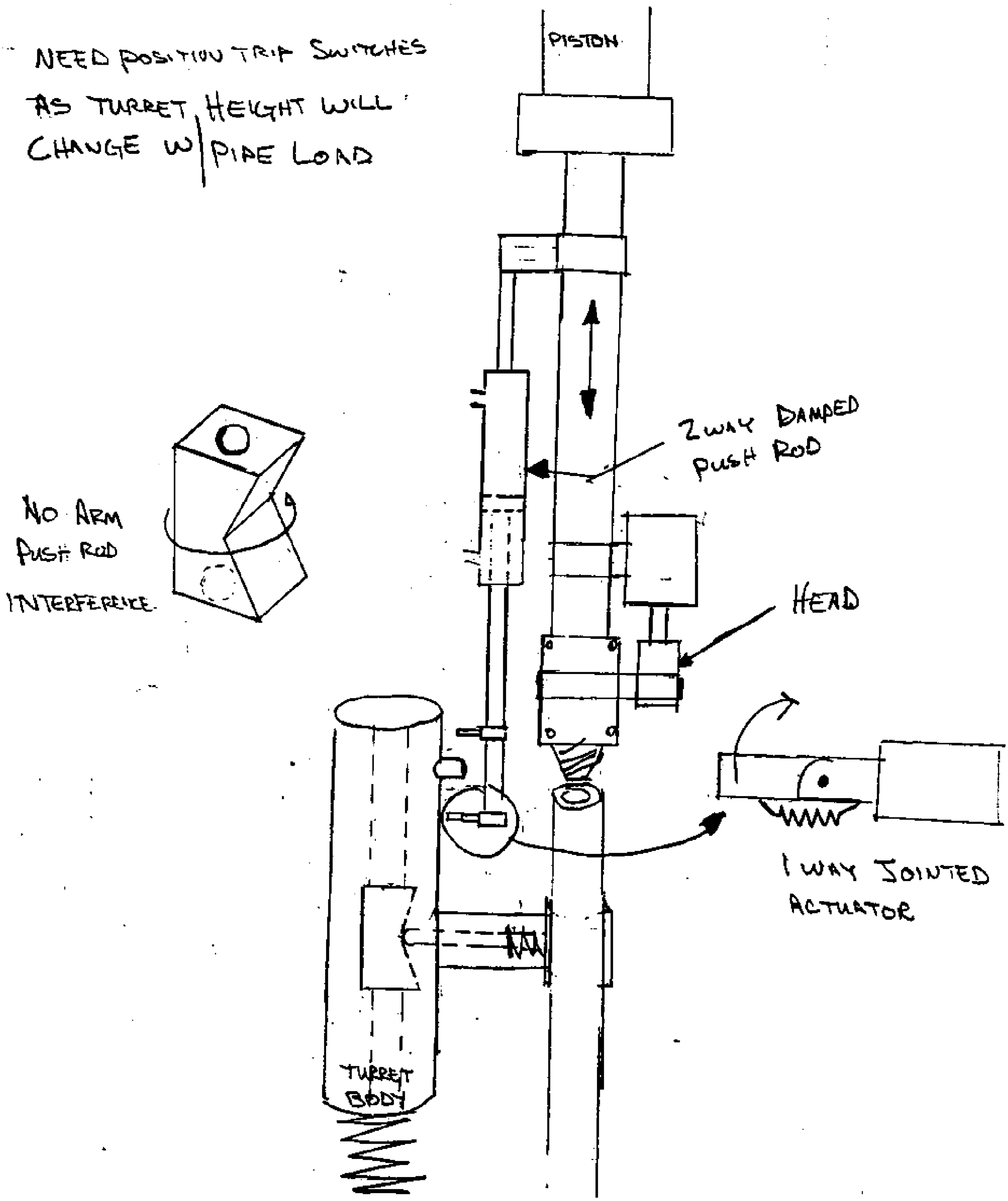


Figure 18

POSSIBLE COUPLING APPROACH

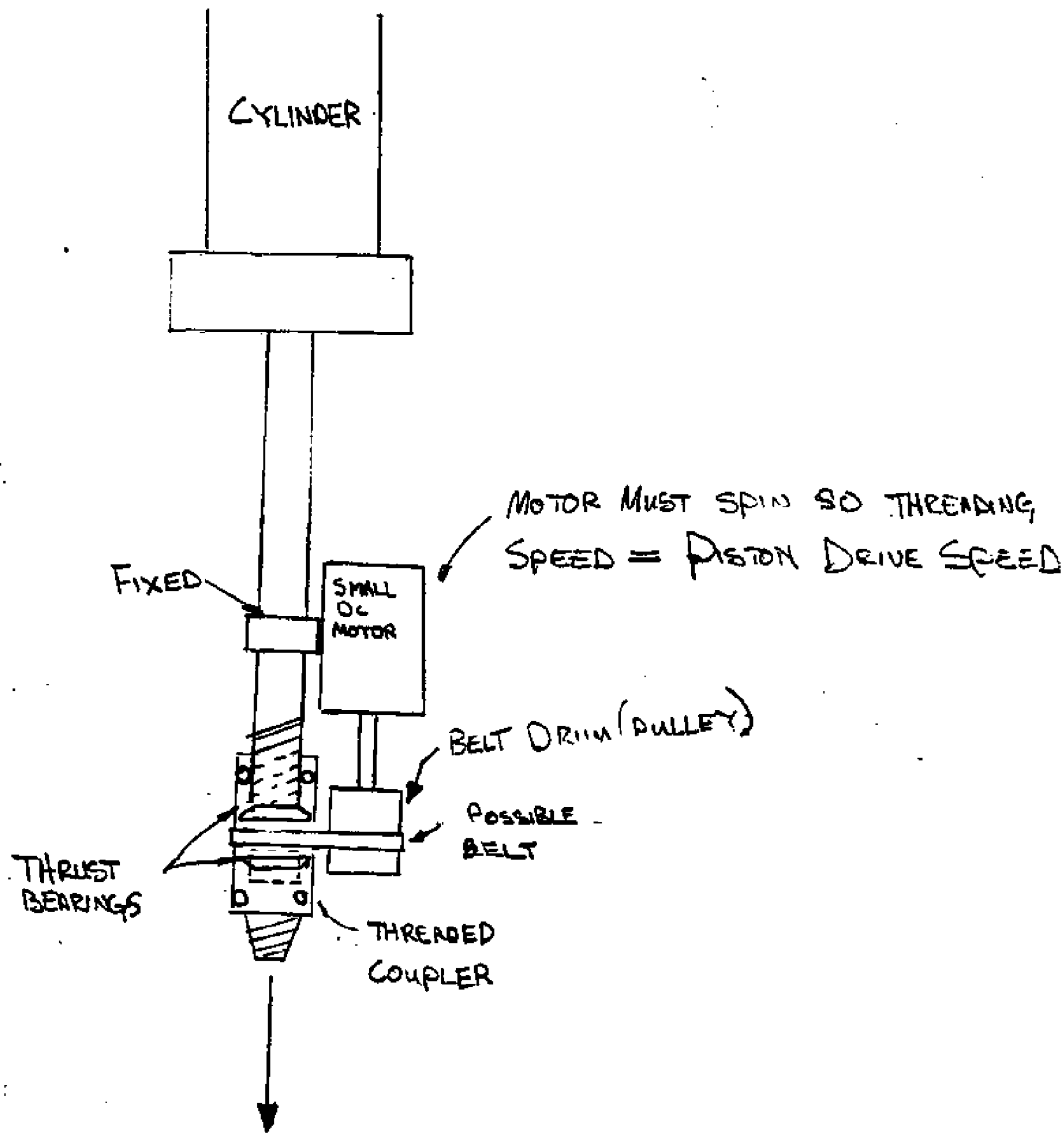


Figure 19

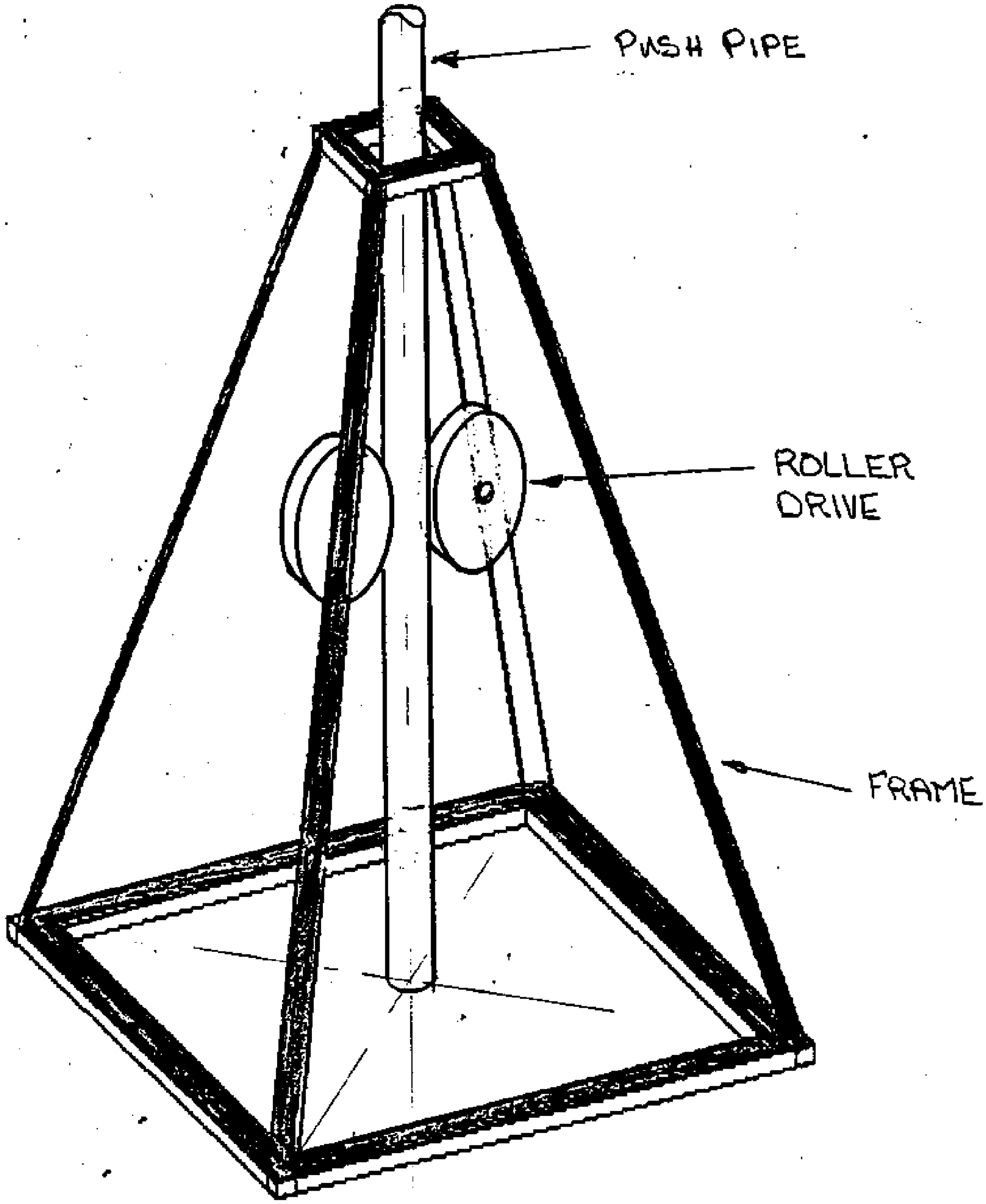
a. Pinch Roller Drive: (Not Used)

The pinch roller drive is shown in figure 20 & 21. This system uses the concept of frictional force to achieve the driving movement. By applying a normal force with pneumatic pistons, which are activated from the boat, a frictional force is achieved with a static coefficient of friction of approximately 0.7 (Lindeburg). The materials considered are steel rollers on the steel drive pipe which is fastened to the penetrometer. The roller surface could be made concave to make a better fit around the pipe. This fit allows for the rollers to distribute the force more evenly. This is essential since damage to the pipe is undesirable.

In considering the force analysis, system complexity, and bulk, four rollers proved to be the most desirable configuration both in size and in assembly. Two dc motors can be used to drive the rollers (Table 1). By using the gearing configuration shown in figure(21) the drive pipe can be pushed in or out of the ground by reversing the direction of the motors. One motor is needed to drive two of the rollers.

The assembly is fastened rigidly to the frame with the use of two steel plates. These plates support the reaction moment which occurs during operation. This moment is transmitted from the shaft of the rollers to the plate. The pneumatic pistons are not required to counteract this moment. Their only purpose is to supply the required normal force. This is important since the pistons are relatively weak with forces which are not in line with the pistons line of motion.

The advantages of such a system is the cost and ease of fabrication. Compared to a hydraulic system the components of the pinch roller assembly



SURFACE PIPE ASSEMBLY

PINCH ROLLER DRIVE

Figure 20

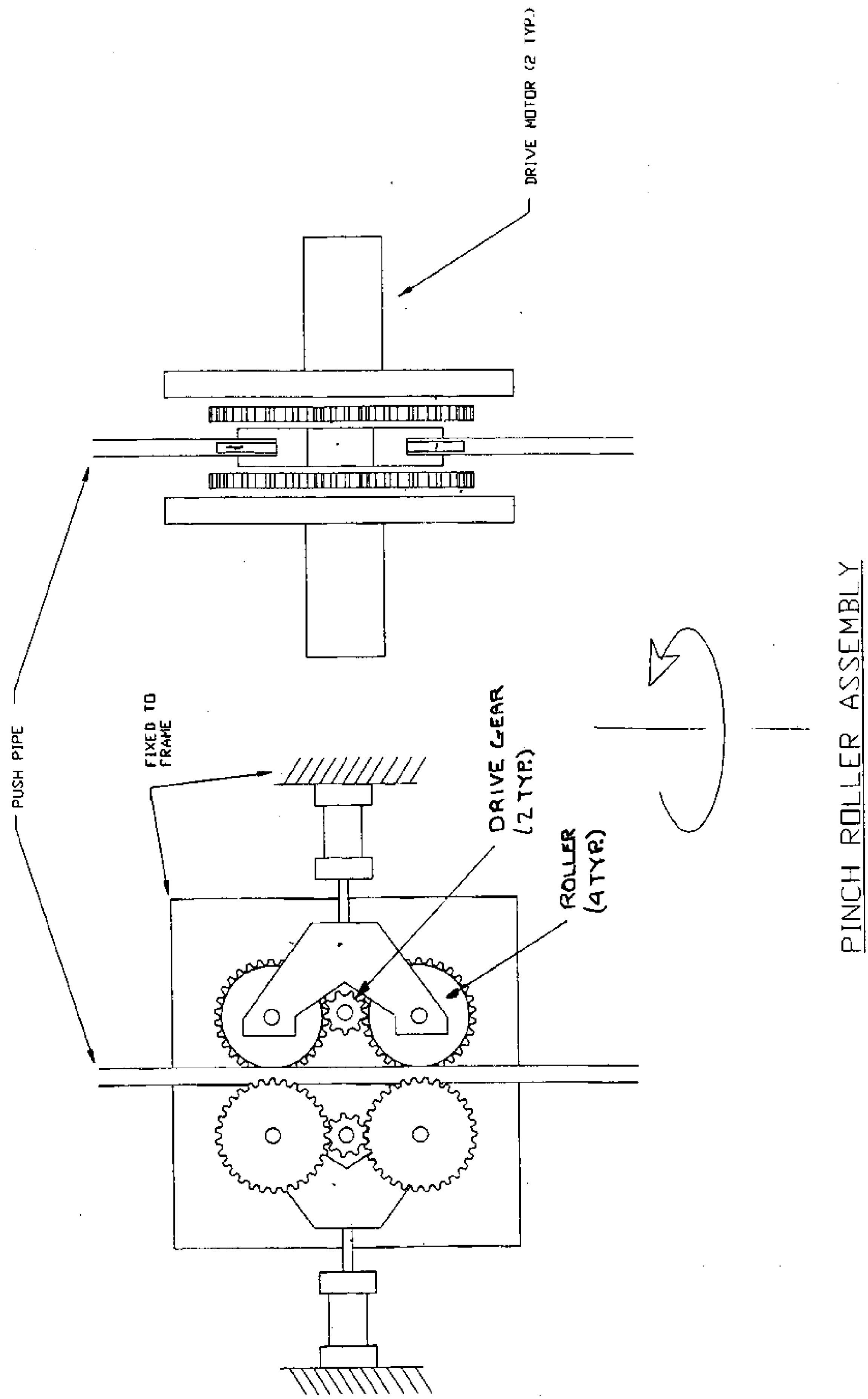

DRIVE MOTOR (2 TYP.)
PUSH PIPE
FIXED TO FRAME
DRIVE GEAR (2 TYP.)
ROLLER (4 TYP.)
PINCH ROLLER ASSEMBLY

Figure 21

Pinch Roller Design for P ^ 3

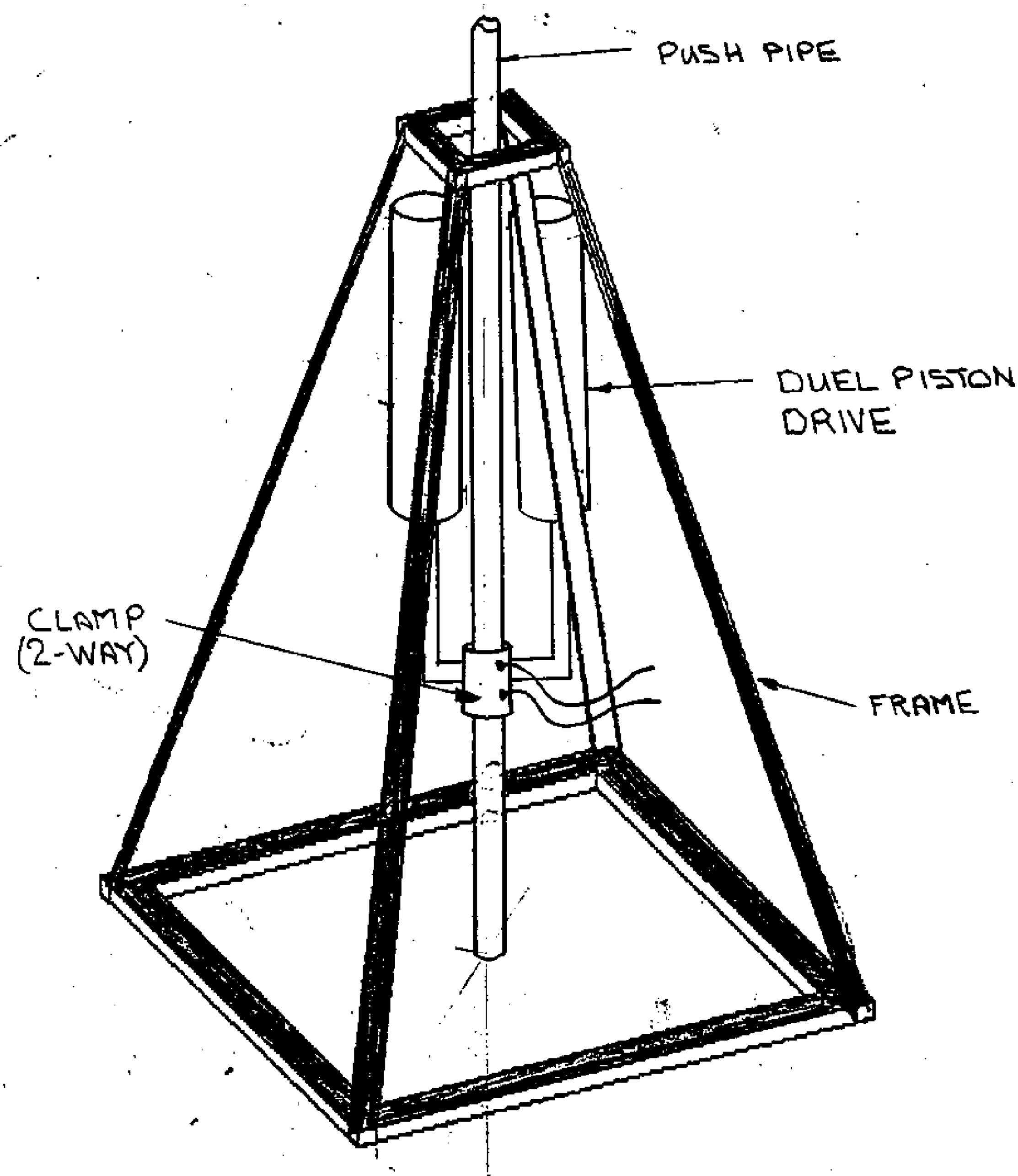
rollers radius ft	rollers number	gear Ratio	Force required lb	one roller' torque lb-ft	one roller' torque lb-in	roller speed rpm	normal force lb	one motor torque lb-in	motor speed rpm
0.125	4	20	10000	312.5	3750	5.012747	3571.429	750	100.2549
0.25	4	20	10000	625	7500	2.506373	3571.429	1500	50.12747
0.125	6	20	10000	208.3333	2500	5.012747	2380.952	750	100.2549
0.25	6	20	10000	416.6667	5000	2.506373	2380.952	1500	50.12747

Table 1

are somewhat less expensive. The fabrication will not require outside consultants since the system is familiar to the project team. The disadvantage of this system is that it is not as reliable as the hydraulic drive. The transmitted drive force is solely dependent upon the frictional force between the rollers and the pipe. If the coefficient of friction varies due to external factors failure could result.

b. Dual Pistons Drive

The method selected for the drive system is the dual piston drive. This incorporates two 3 in. dia., 16 in. stroke pistons that will both push and pull the drive pipe (Figure 22). This system incorporates the simplicity of surface assembly with an easy to design drive. The main difficulty in using such a drive system is the design of the clamp necessary to transmit the pistons driving force. In order for this system to function effectively, a two-way clamp must be used. After careful consideration of clamping needs, a floating four tapered jaw arrangement with a second set of clamping pistons fixed to two actuating plates was conceived (Figure 23). Problems with this design (discussed below) made it necessary to re-evaluate the floating jaw principle, resulting in the collet concept, used in the final design.



SURFACE PIPE ASSEMBLY
DUAL PISTON W/HYDRAULIC CLAMP

Figure 22

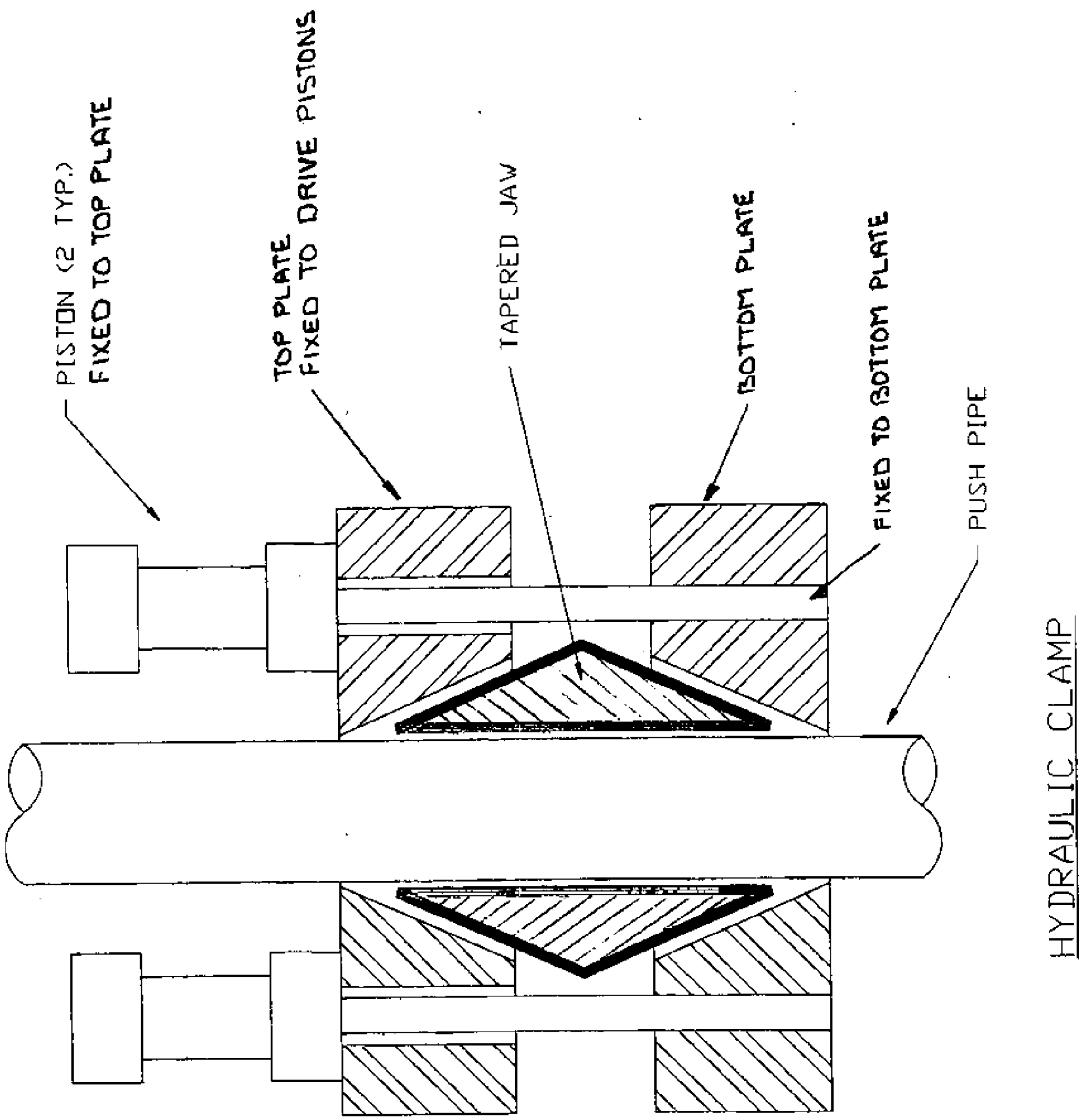

PISTON (2 TYP.)
FIXED TO TOP PLATE
TOP PLATE
FIXED TO DRIVE PISTONS
TAPERED JAW
BOTTOM PLATE
FIXED TO BOTTOM PLATE
PUSH PIPE
HYDRAULIC CLAMP

Figure 23

b. 1) Four Jaw Clamp: (Not used)

As the clamping pistons shown in figure 24 (bottom caps fixed to top plate, rods fixed to bottom plate).stroke, the two plates (with thru holes tapered to match the jaws) squeeze on the jaws, pressing them against the pipe, providing the rigid connection necessary to transmit the force of the drive pistons. The use of four jaws as opposed to two or six or eight, etc. was decided upon based on surface contact uniformity, projected wear, and complexity of machining.

A sixteen degree included angle taper was chosen based on the constraint that eight degrees included was the minimum required to prevent lock-up. A lock-up prevention factor of two seemed reasonable considering the material properties were unknown at the time, due to budget constraints. Tables 2,3 & 4 show that the clamping piston force necessary at sixteen degrees was easily obtainable by the hydraulic components available for the project. A simple stress analysis (Figure 25 and Tables 5-7) shows that the size of each portion of the clamp is not determined by failure, but by the necessary geometries. This allows the clamp to be designed around the other necessary constraints, such as available space. The inherent fault with the floating jaws is that because they are floating, their engagements to the pipe may not occur simultaneously or uniformly, due to environmental factors present at the ocean floor. This may prevent adequate gripping or even cause lockup.

The proposed solution to this foreseen problem was to fix the four jaws together, constraining them to operate as a single unit, providing uniform and simultaneous gripping. This solution led to the idea of a single unit with

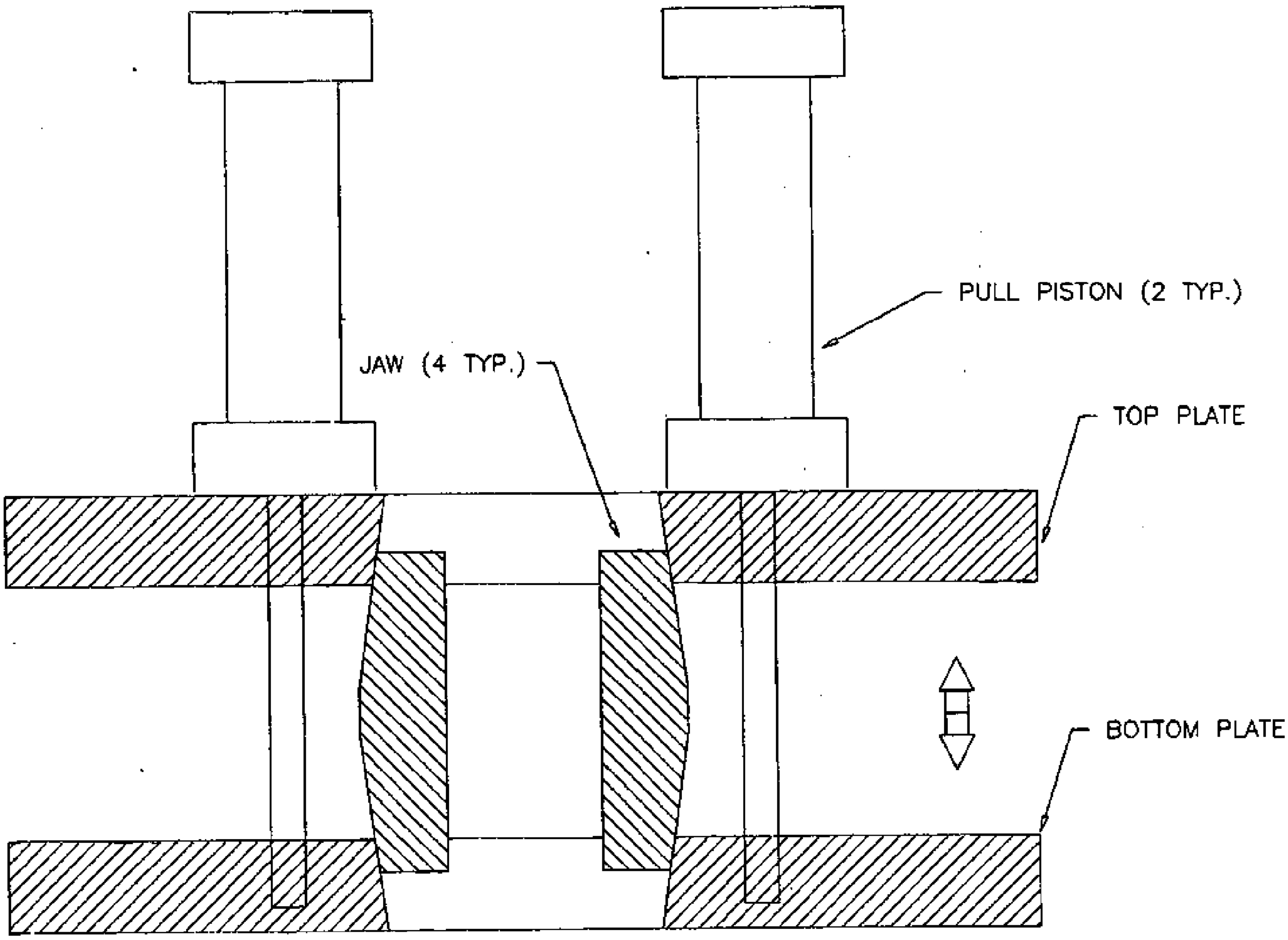


Figure 24
FOUR JAW CLAMP

Figure 24

Clamp Analysis

angle deg	angle rad	clamp force lb	norm force lb	pist force lb
0	0	20000	20000	1.22E-12
1	0.017453	20000	20003.05	349.1013
2	0.034907	20000	20012.19	698.4154
3	0.05236	20000	20027.45	1048.156
4	0.069813	20000	20048.84	1398.536
5	0.087266	20000	20076.4	1749.773
6	0.10472	20000	20110.17	2102.085
7	0.122173	20000	20150.2	2455.691
8	0.139626	20000	20196.55	2810.817
9	0.15708	20000	20249.3	3167.689
10	0.174533	20000	20308.53	3526.54
11	0.191986	20000	20374.33	3887.606
12	0.20944	20000	20446.81	4251.131
13	0.226893	20000	20526.08	4617.364
14	0.244346	20000	20612.27	4986.56
15	0.261799	20000	20705.52	5358.984
16	0.279253	20000	20805.99	5734.908
17	0.296706	20000	20913.84	6114.614
18	0.314159	20000	21029.24	6498.394
19	0.331613	20000	21152.41	6886.552
20	0.349066	20000	21283.56	7279.405
21	0.366519	20000	21422.9	7677.281
22	0.383972	20000	21570.69	8080.525
23	0.401426	20000	21727.21	8489.496
24	0.418879	20000	21892.73	8904.574
25	0.436332	20000	22067.56	9326.153
26	0.453786	20000	22252.04	9754.652
27	0.471239	20000	22446.52	10190.51
28	0.488692	20000	22651.4	10634.19
29	0.506145	20000	22867.08	11086.18
30	0.523599	20000	23094.01	11547.01
31	0.541052	20000	23332.67	12017.21
32	0.558505	20000	23583.57	12497.39
33	0.575959	20000	23847.27	12988.15
34	0.593412	20000	24124.36	13490.17
35	0.610865	20000	24415.49	14004.15

Coefficient of friction = 0.5

Table 2

36	0.628319	20000	24721.36	14530.85
37	0.645772	20000	25042.71	15071.08
38	0.663225	20000	25380.36	15625.71
39	0.680678	20000	25735.19	16195.68
40	0.698132	20000	26108.15	16781.99
41	0.715585	20000	26500.26	17385.73
42	0.733038	20000	26912.65	18008.08
43	0.750492	20000	27346.55	18650.3
44	0.767945	20000	27803.27	19313.78
45	0.785398	20000	28284.27	20000

☆ Coefficient of Friction = 0.5

Table 2

Clamp Analysis

angle deg	angle rad	clamp force lb	norm force lb	pist force lb
0	0	14285.71	14285.71	8.75E-13
1	0.017453	14285.71	14287.89	249.3581
2	0.034907	14285.71	14294.42	498.8681
3	0.05236	14285.71	14305.32	748.6826
4	0.069813	14285.71	14320.6	998.9545
5	0.087266	14285.71	14340.28	1249.838
6	0.10472	14285.71	14364.4	1501.489
7	0.122173	14285.71	14393	1754.065
8	0.139626	14285.71	14426.11	2007.726
9	0.15708	14285.71	14463.79	2262.635
10	0.174533	14285.71	14506.09	2518.957
11	0.191986	14285.71	14553.1	2776.862
12	0.20944	14285.71	14604.87	3036.522
13	0.226893	14285.71	14661.49	3298.117
14	0.244346	14285.71	14723.05	3561.829
15	0.261799	14285.71	14789.66	3827.846
16	0.279253	14285.71	14861.42	4096.363
17	0.296706	14285.71	14938.45	4367.581
18	0.314159	14285.71	15020.89	4641.71
19	0.331613	14285.71	15108.87	4918.966
20	0.349066	14285.71	15202.54	5199.575
21	0.366519	14285.71	15302.07	5483.772
22	0.383972	14285.71	15407.64	5771.803
23	0.401426	14285.71	15519.43	6063.926
24	0.418879	14285.71	15637.66	6360.41
25	0.436332	14285.71	15762.54	6661.538
26	0.453786	14285.71	15894.31	6967.608
27	0.471239	14285.71	16033.23	7278.935
28	0.488692	14285.71	16179.57	7595.849
29	0.506145	14285.71	16333.63	7918.701
30	0.523599	14285.71	16495.72	8247.861
31	0.541052	14285.71	16666.19	8583.723
32	0.558505	14285.71	16845.41	8926.705
33	0.575959	14285.71	17033.76	9277.251
34	0.593412	14285.71	17231.68	9635.836
35	0.610865	14285.71	17439.64	10002.96

☆ Coefficient of friction = 0.7

Table 3

36	0.628319	14285.71	17658.11	10379.18
37	0.645772	14285.71	17887.65	10765.06
38	0.663225	14285.71	18128.83	11161.22
39	0.680678	14285.71	18382.28	11568.34
40	0.698132	14285.71	18648.68	11987.14
41	0.715585	14285.71	18928.76	12418.38
42	0.733038	14285.71	19223.32	12862.91
43	0.750492	14285.71	19533.25	13321.64
44	0.767945	14285.71	19859.48	13795.55
45	0.785398	14285.71	20203.05	14285.71

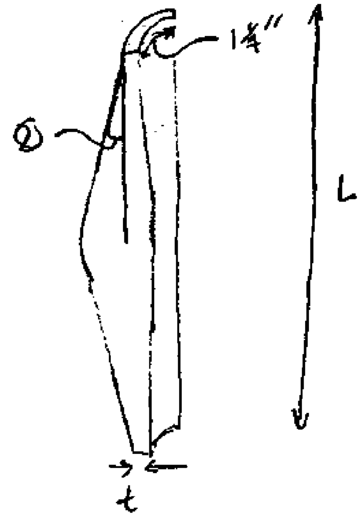
☆ Coefficient of friction = 0.7

Table 3

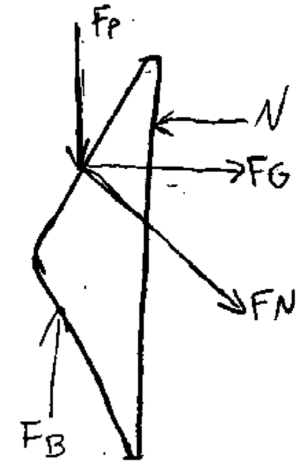
clamp piston analysis

piston force lb	Minimum required pressure for various bores bore in	bore in	bore in	bore in	bore in	bore in
	0.5	1	1.5	2	2.5	3
15000	76394	19099	8488.3	4774.6	3055.8	2122.1
14000	71301	17825	7922.4	4456.3	2852.1	1980.6
13000	66208	16552	7356.5	4138	2648.3	1839.1
12000	61115	15279	6790.6	3819.7	2444.6	1697.7
11000	56023	14006	6224.7	3501.4	2240.9	1556.2
10000	50930	12732	5658.8	3183.1	2037.2	1414.7
9000	45837	11459	5093	2864.8	1833.5	1273.2
8000	40744	10186	4527.1	2546.5	1629.7	1131.8
7000	35651	8912.7	3961.2	2228.2	1426	990.3
6000	30558	7639.4	3395.3	1909.9	1222.3	848.83
5000	25465	6366.2	2829.4	1591.5	1018.6	707.36
4000	20372	5093	2263.5	1273.2	814.87	565.88
3000	15279	3819.7	1697.7	954.93	611.15	424.41
2000	10186	2546.5	1131.8	636.62	407.44	282.94
1000	5093	1273.2	565.88	318.31	203.72	141.47

Table 4



Cross section



$$FN^2 = Fp^2 + FG^2$$

$$FN = \sqrt{Fp^2 + FG^2}$$

Knew this

$FG = N$

$Fp = F_B$

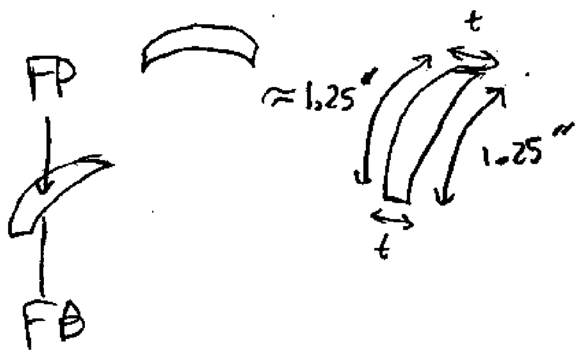
$$\sigma_{des} = \frac{F}{A} = \frac{FG}{L\,(1\tfrac{1}{4}")}(SF)$$

Find it — determine — pick this

$\begin{pmatrix} 1.5 \\ 2 \\ 2.5 \end{pmatrix}$

$L\,(1\tfrac{1}{4}")$

Pick Material $\sigma_y > \sigma_{des}$



Knew this

$$\sigma_{des} = \frac{F}{A} = \frac{FP}{t\,(1.25)}(SF)$$

Find it — determine — pick this

Stress Analysis (Clamp Finger)

Figure 25

Analysis of yielding on top surface of clamp finger

Thickness for pushing forces of (t in inches)
(Forces in lbs.)

Yield stress SF		5000	6000	7000	8000	9000	10000	11000	12000	13000	14000	15000
180000	2	0.04444	0.05333	0.06222	0.07111	0.08	0.08889	0.09778	0.10667	0.11556	0.12444	0.13333
175000	2	0.04571	0.05486	0.064	0.07314	0.08229	0.09143	0.10057	0.10971	0.11886	0.128	0.13714
170000	2	0.04706	0.05647	0.06588	0.07529	0.08471	0.09412	0.10353	0.11294	0.12235	0.13176	0.14118
165000	2	0.04848	0.05818	0.06788	0.07758	0.08727	0.09697	0.10667	0.11636	0.12606	0.13576	0.14545
160000	2	0.05	0.06	0.07	0.08	0.09	0.1	0.11	0.12	0.13	0.14	0.15
155000	2	0.05161	0.06194	0.07226	0.08258	0.0929	0.10323	0.11355	0.12387	0.13419	0.14452	0.15484
150000	2	0.05333	0.064	0.07467	0.08533	0.096	0.10667	0.11733	0.128	0.13867	0.14933	0.16
145000	2	0.05517	0.06621	0.07724	0.08828	0.09931	0.11034	0.12138	0.13241	0.14345	0.15448	0.16552
140000	2	0.05714	0.06857	0.08	0.09143	0.10286	0.11429	0.12571	0.13714	0.14857	0.16	0.17143
135000	2	0.05926	0.07111	0.08296	0.09481	0.10667	0.11852	0.13037	0.14222	0.15407	0.16593	0.17778
130000	2	0.06154	0.07385	0.08615	0.09846	0.11077	0.12308	0.13538	0.14769	0.16	0.17231	0.18462
125000	2	0.064	0.0768	0.0896	0.1024	0.1152	0.128	0.1408	0.1536	0.1664	0.1792	0.192
120000	2	0.06667	0.08	0.09333	0.10667	0.12	0.13333	0.14667	0.16	0.17333	0.18667	0.2
115000	2	0.06957	0.08348	0.09739	0.1113	0.12522	0.13913	0.15304	0.16696	0.18087	0.19478	0.2087
110000	2	0.07273	0.08727	0.10182	0.11636	0.13091	0.14545	0.16	0.17455	0.18909	0.20364	0.21818
105000	2	0.07619	0.09143	0.10667	0.1219	0.13714	0.15238	0.16762	0.18286	0.1981	0.21333	0.22857
100000	2	0.08	0.096	0.112	0.128	0.144	0.16	0.176	0.192	0.208	0.224	0.24
95000	2	0.08421	0.10105	0.11789	0.13474	0.15158	0.16842	0.18526	0.20211	0.21895	0.23579	0.25263
90000	2	0.08889	0.10667	0.12444	0.14222	0.16	0.17778	0.19556	0.21333	0.23111	0.24889	0.26667
85000	2	0.09412	0.11294	0.13176	0.15059	0.16941	0.18824	0.20706	0.22588	0.24471	0.26353	0.28235
80000	2	0.1	0.12	0.14	0.16	0.18	0.2	0.22	0.24	0.26	0.28	0.3
75000	2	0.10667	0.128	0.14933	0.17067	0.192	0.21333	0.23467	0.256	0.27733	0.29867	0.32
70000	2	0.11429	0.13714	0.16	0.18286	0.20571	0.22857	0.25143	0.27429	0.29714	0.32	0.34286
65000	2	0.12308	0.14769	0.17231	0.19692	0.22154	0.24615	0.27077	0.29538	0.32	0.34462	0.36923
60000	2	0.13333	0.16	0.18667	0.21333	0.24	0.26667	0.29333	0.32	0.34667	0.37333	0.4
55000	2	0.14545	0.17455	0.20364	0.23273	0.26182	0.29091	0.32	0.34909	0.37818	0.40727	0.43636
50000	2	0.16	0.192	0.224	0.256	0.288	0.32	0.352	0.384	0.416	0.448	0.48
45000	2	0.17778	0.21333	0.24889	0.28444	0.32	0.35556	0.39111	0.42667	0.46222	0.49778	0.53333

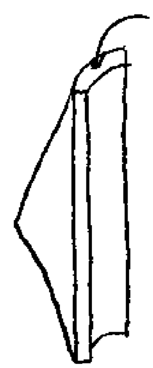


Table 5

clamp tooth

****** *From the analysis it is shown that length is not a limiting factor*

yield stress psi	force lb	sf	L in
180000	5000	2	0.0444
175000	5000	2	0.0457
170000	5000	2	0.0471
165000	5000	2	0.0485
160000	5000	2	0.05
155000	5000	2	0.0516
150000	5000	2	0.0533
145000	5000	2	0.0552
140000	5000	2	0.0571
135000	5000	2	0.0593
130000	5000	2	0.0615
125000	5000	2	0.064
120000	5000	2	0.0667
115000	5000	2	0.0696
110000	5000	2	0.0727
105000	5000	2	0.0762
100000	5000	2	0.08
95000	5000	2	0.0842
90000	5000	2	0.0889
85000	5000	2	0.0941
80000	5000	2	0.1
75000	5000	2	0.1067
70000	5000	2	0.1143
65000	5000	2	0.1231
60000	5000	2	0.1333
55000	5000	2	0.1455
50000	5000	2	0.16

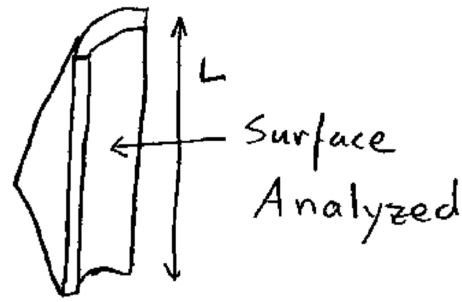


Table 6

Stress analysis on top and bottom plate

yield strength psi	hole diameter in	width in	safety factor	thickness required with various moments: moment lb in 20000	moment lb in 25000	moment lb in 30000	moment lb in 35000	moment lb in 40000	moment lb in 45000	moment lb in 50000
180000	1.9	10	2	0.4057204	0.4536092	0.496904	0.5367177	0.5737753	0.6085806	0.6415003
175000	1.9	10	2	0.4114756	0.4600437	0.5039526	0.5443311	0.5819144	0.6172134	0.6506
170000	1.9	10	2	0.4174829	0.46676	0.51131	0.5522779	0.5904099	0.6262243	0.6600984
165000	1.9	10	2	0.4237612	0.4737794	0.5189993	0.5605833	0.5992888	0.6356417	0.6700252
160000	1.9	10	2	0.4303315	0.4811252	0.5270463	0.569275	0.6085806	0.6454972	0.6804138
155000	1.9	10	2	0.4372172	0.4888237	0.5354796	0.578384	0.6183185	0.6558258	0.6913011
150000	1.9	10	2	0.4444444	0.496904	0.5443311	0.5879447	0.6285394	0.6666667	0.7027284
145000	1.9	10	2	0.4520423	0.5053987	0.5536365	0.5979958	0.6392844	0.6780635	0.7147417
140000	1.9	10	2	0.4600437	0.5143445	0.5634362	0.6085806	0.6506	0.6900656	0.727393
135000	1.9	10	2	0.4684856	0.5237828	0.5737753	0.6197482	0.6625387	0.7027284	0.7407407
130000	1.9	10	2	0.4774099	0.5337605	0.5847053	0.631554	0.6751596	0.7161149	0.7548514
125000	1.9	10	2	0.4868645	0.5443311	0.5962848	0.6440612	0.6885304	0.7302967	0.7698004
120000	1.9	10	2	0.496904	0.5555556	0.6085806	0.6573422	0.7027284	0.745356	0.7856742
115000	1.9	10	2	0.5075913	0.5675044	0.6216699	0.6714802	0.7178425	0.761387	0.8025724
110000	1.9	10	2	0.5189993	0.5802589	0.6356417	0.6865715	0.7339758	0.7784989	0.8206099
105000	1.9	10	2	0.5312127	0.593913 9	0.6506	0.7027284	0.7512482	0.796819 1	0.8399211
100000	1.9	10	2	0.5443311	0.6085806	0.6666667	0.7200823	0.7698004	0.8164966	0.860663
95000	1.9	10	2	0.5584719	0.6243905	0.6839856	0.7387889	0.7897985	0.8377078	0.8830216
90000	1.9	10	2	0.5737753	0.6415003	0.7027284	0.7590334	0.8114408	0.860663	0.9072184
85000	1.9	10	2	0.5904099	0.6600984	0.7231015	0.7810389	0.8349657	0.8856149	0.9335201
80000	1.9	10	2	0.6085806	0.6804138	0.745356	0.8050765	0.860663	0.9128709	0.9622504
75000	1.9	10	2	0.6285394	0.7027284	0.7698004	0.8314794	0.8888889	0.942809	0.993808
70000	1.9	10	2	0.6506	0.727393	0.796819 1	0.860663	0.9200874	0.9759001	1.028689
65000	1.9	10	2	0.6751596	0.7548514	0.8268982	0.8931522	0.9548198	1.0127394	1.067521
60000	1.9	10	2	0.7027284	0.7856742	0.860663	0.9296223	0.993808	1.0540926	1.1111111
55000	1.9	10	2	0.7339758	0.8206099	0.8989331	0.9709588	1.0379986	1.1009638	1.160517 7

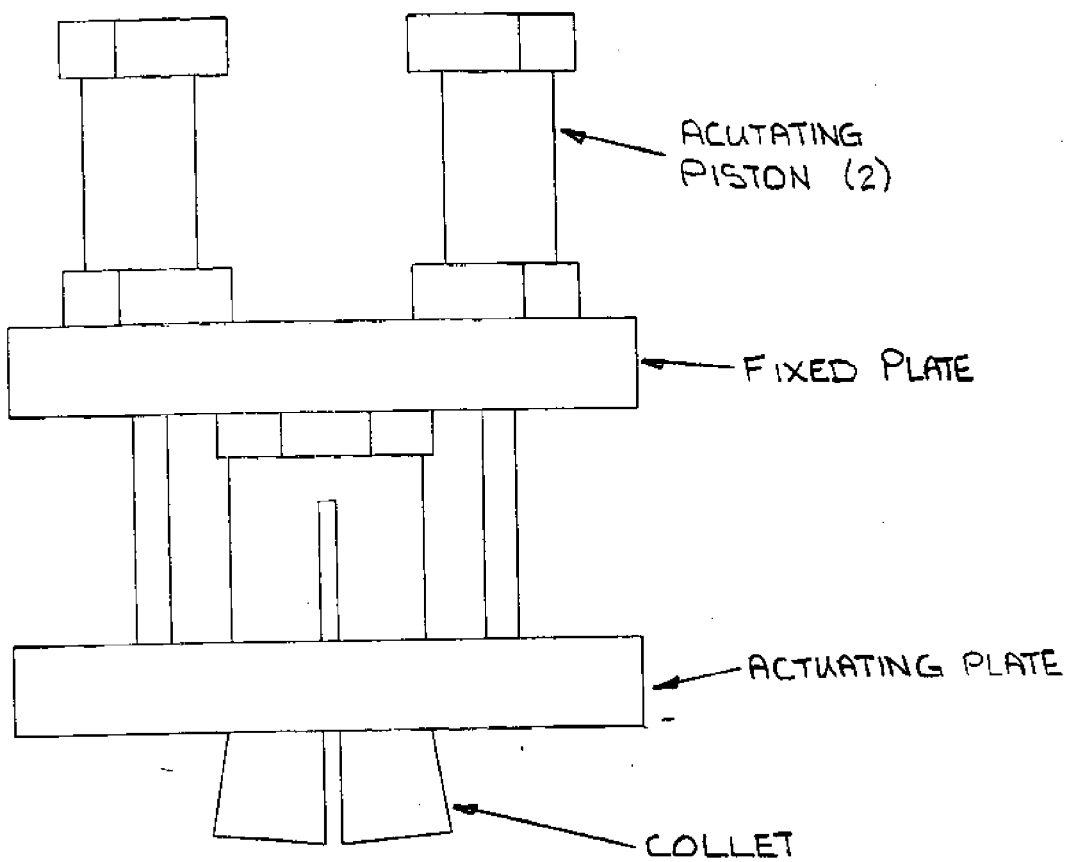
Table 7

gripping surfaces that could be elastically bent into place, similar to a metal cutting tool holder known as a collet.

b. 2) Collet Clamp (Fixture)

As mentioned above, the collet clamp (Figure 26) incorporates the same principle as a metal cutting tool collet. The collet is threaded at one end to mate with matching threads in the top plate, which is fixed to the rod ends of the drive pistons. Two additional 2 in. dia. pistons (with the same configuration as for the four jaw clamp, see description above) provide the needed force and motion to the bottom plate, which intern, squeezes the collet's flexible jaws to the pipe by way of the 16^0 taper angle. Since one 3000 psi, 2.1 gpm pump is used for both sets of pistons, an accumulator is incorporated to maintain pressure to the clamp pistons during the driving or pulling stroke.

Unfortunately, this fixture could not incorporate an off the shelf tool holder as these do not exceed 1.125 in dia., whereas the drill pipe is a full 1.44 in. dia. However, the ratio of dimensions of existing tool holders was used to initiate the design for this application. A stress analysis of the clamp jaws in bending (Table 8) showed a stress of 56 ksi. 17-4 stainless steel, heat treated after machining to roughly 150 ksi yield strength showed to be an optimal material to provide necessary strength as well as salt water corrosion resistance. However, budget constraints again dictated actual design. A donated round of 304 stainless steel in the fully annealed condition (which can



COLLET FIXTURE

Figure 26-a

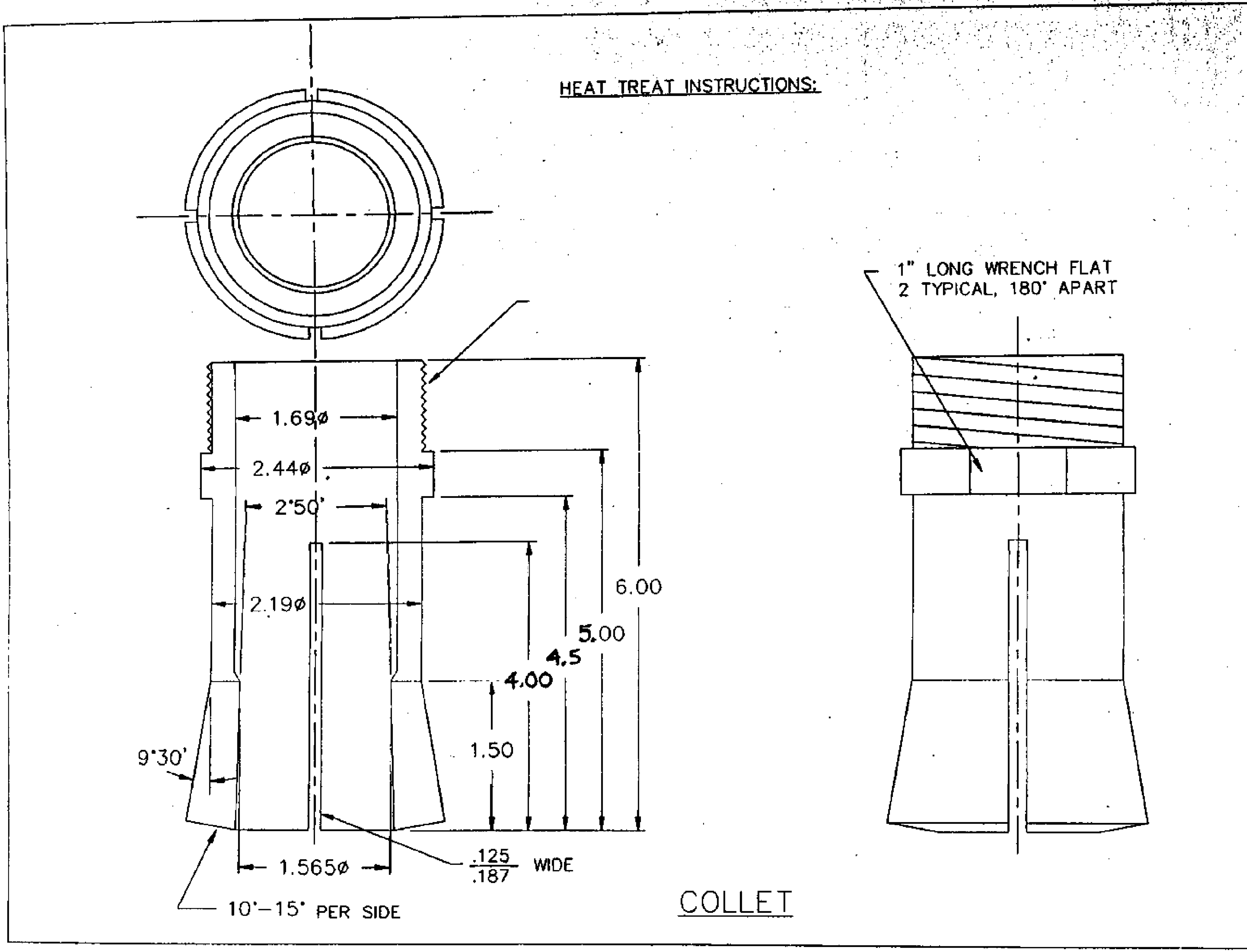

HEAT TREAT INSTRUCTIONS:
1" LONG WRENCH FLAT
2 TYPICAL, 180° APART
1.69ø
2.44ø
2°50'
2.19ø
6.00
5.00
4.5
4.00
1.50
9°30'
1.565ø
.125
.187
WIDE
10°-15° PER SIDE
COLLET

Figure 26-b

Clamp Bending Analysis Feb-24-94

Geometry Analysis (see fig. 1-2)

R2 (in) = 2.5 d2 (in) = 0.0625 alpha (rad) 0.025

R1 (in)	disp (in)	d1-d2 (in)	phi (rad)	phi (deg)	theta (deg)	theta (rad)	stroke (in)
3	0.075	0.0125	0.024995	1.432096	9.5	0.165806	0.448182
3.25	0.08125	0.01875	0.024995	1.432096	9.5	0.165806	0.485531
3.5	0.0875	0.025	0.024995	1.432096	9.5	0.165806	0.522879
3.75	0.09375	0.03125	0.024995	1.432096	9.5	0.165806	0.560228
4	0.1	0.0375	0.024995	1.432096	9.5	0.165806	0.597576
4.25	0.10625	0.04375	0.024995	1.432096	9.5	0.165806	0.634925
4.5	0.1125	0.05	0.024995	1.432096	9.5	0.165806	0.672273
4.75	0.11875	0.05625	0.024995	1.432096	9.5	0.165806	0.709622
5	0.125	0.0625	0.024995	1.432096	9.5	0.165806	0.746971

Stress Analysis For 1/4 of collet

r2 (in)	r1 (in)	Ic (in^4)	Norm F(lbf	centroid(in	bend stress(psi)
0.9	0.845	1.9E-05	2.697571	0.872789	12558.96
0.95	0.845	0.000136	19.29167	0.898524	23758.3
1	0.845	0.000449	63.70914	0.92467	34767.56
1.05	0.845	0.001064	151.1494	0.951196	45601.79
1.1	0.845	0.002099	298.0423	0.978072	56274.47
1.15	0.845	0.003676	522.0099	1.005272	66797.76
1.2	0.845	0.005928	841.8351	1.032771	77182.62
1.25	0.845	0.008995	1277.433	1.060549	87438.96
1.3	0.845	0.013026	1849.829	1.088586	97575.76
1.35	0.845	0.018175	2581.136	1.116864	107601.2
1.4	0.845	0.024607	3494.537	1.145367	117522.7
1.45	0.845	0.032492	4614.274	1.174081	127347.1
1.5	0.845	0.042008	5965.628	1.202992	137080.5
1.55	0.845	0.05334	7574.91	1.232088	146728.8
1.6	0.845	0.066681	9469.454	1.261357	156297
1.65	0.845	0.08223	11677.6	1.290788	165790

Table 8

not be heat treated) with a yield strength of 34 ksi was used. This low yield strength will ultimately result in some plastic deformation of the flexible jaws, but not enough to hinder clamping operation in any way. Another foreseen problem, due to low yield strength, is possible fatigue cracking. In any case, proof of concept was definitely accomplished, and with the correct material, this design will prove to be a very efficient and usable fixture.

The two plates, which are 1 in. thick, 8 in. square 316 stainless steel with a yield strength of 32 ksi, were also donated. As opposed to the collet material, 316 stainless is precisely the material needed for the application. A stress analysis of the plates (Table 9) yielded only 6 ksi, well within the strength of the material. With a suitable clamp design completed, the remaining hydraulic components, lines and fittings, must simply be picked out and purchased.

Initially, the control of the hydraulic drive was envisioned to be completely automatic, but budget constraints forced a simplified, manual approach in which an operator on the boat will initiate each piston stroke in the appropriate sequence.

CLAMP STRESS ANALYSIS
see fig. 1 for description

C =	0.375
HOLE dia:	3.53
a =	1.485
Fr =	5000
M =	4455
Width	8
Modulus	30000000
d =	3.52

PUSH PISTON SPAN (in) = 6.5

PLATE THICKNESS, h (in)	I	MAX STRESS (psi)	DEFLECTION (in)
0.75	0.1575	10607.14	-0.01199
1	0.373333	5966.518	-0.00506
1.5	1.26	2651.786	-0.0015
1.75	2.000833	1948.251	-0.00094
2	2.986667	1491.629	-0.00063

Table 9

C. Frame:

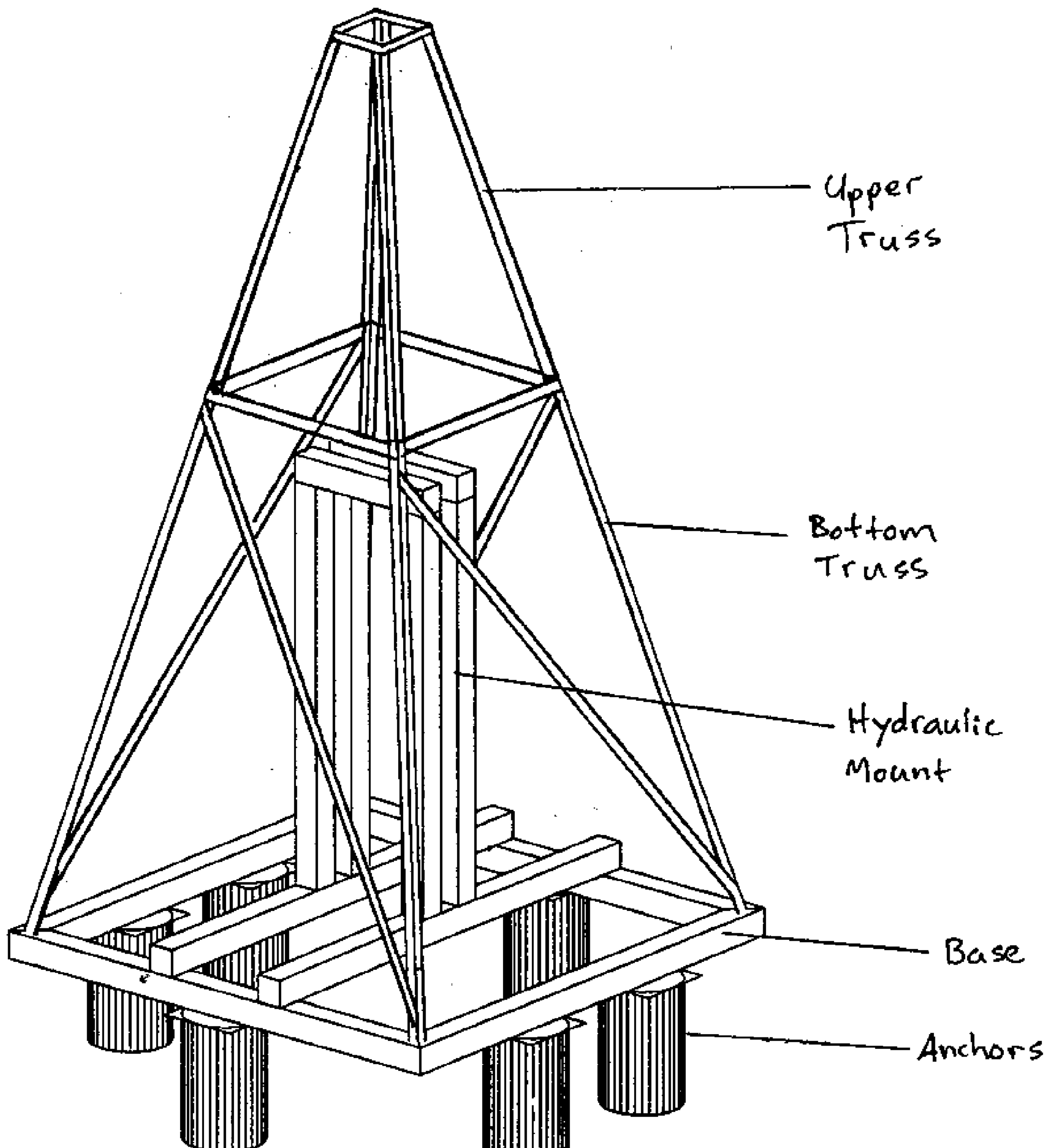
1. Criteria

The frame must incorporate light weight, strength, ease of assembly and deployment, while remaining in the constraints of the budget and the allotted fabrication time. A basic configuration of the frame is shown in figure 27. The primary forces involved are the reaction force required by the drive system (5tons max), the drag on the assembly due to currents, and the weight of the structure during launch and recovery. The size of the base for the frame will not be dependent upon the drag force. Using a current of 1 knot, the minimum base width essential for stability was calculated at less than 1 ft. (see Appendix D). The size of the base is more dependent upon the weight of the pipe it is supporting and the volume requirements of the components involved.

Providing the necessary strength could be easily achieved if weight was not a factor. Our assembly must have a minimum weight to allow for ease in handling. This can be achieved with the use of a light weight truss where members that experience forces of greater magnitudes will be more structurally sound. A finite element analysis was performed on the bottom section of the truss where the frame was tethered for launch and recovery (2 in. angle iron 1/4 in. thick) (figure 28). The analysis shows stresses which are all below 1 ksi (Appendix E). This is well below the yield stress for structural steel.

A stress analysis was performed on the frame members which transmit the reaction force from the drive system to the anchors (table 10). Using the method of superposition and the equation for stress:

$$\sigma = My/I \qquad \text{Eq 2}$$



Sectioned Bolted Frame

Figure 27

FEA MESH FOR Bottom Truss

(Boundary Conditions are shown)

FD = Fixed Displacement
LF = Lifting Force

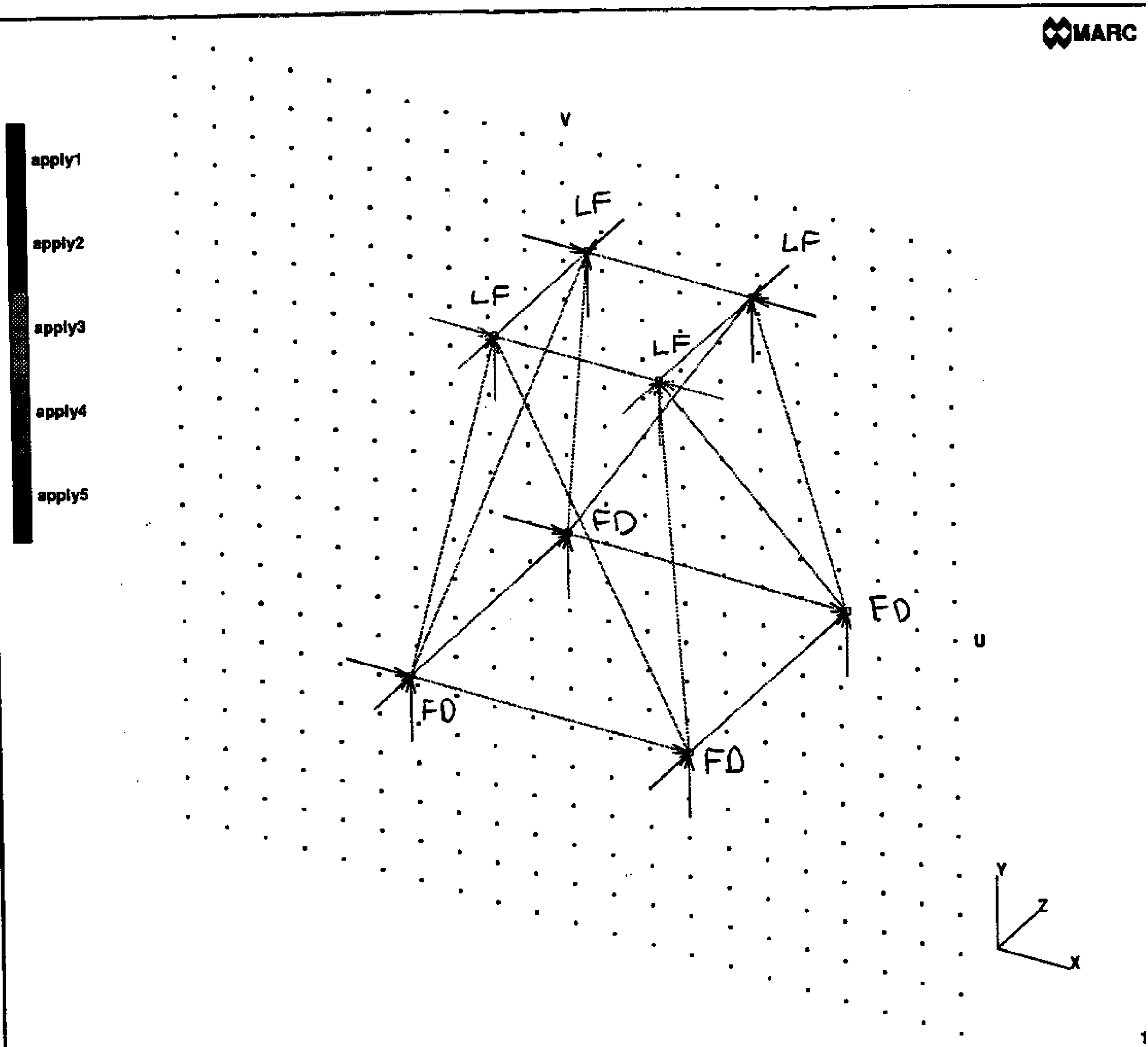


Figure 28

Maximum Beam Length Analysis - Hollow Rectangular Beam

Beam Dimensions (in)								
Width		4						
Height		4						
		Length(in)	36	48	60	72	84	96
		Mo	21388.89	29166.67	36833.33	44444.44	52023.81	59583.33
Thickness	Izz	Stress	S.F.=	4				
(in)	(in^4)		(psi)	(psi)	(psi)	(psi)	(psi)	(psi)
0.125	2.54425		-20787.6	-34063.7	-47689.2	-61489.4	-75389.4	-89351.8
0.25	4.853841		-10896.3	-17855.3	-24997.4	-32231.1	-39517.1	-46835.8
0.375	6.943665		-7616.86	-12481.4	-17474	-22530.5	-27623.7	-32739.7
0.5	8.828125		-5990.95	-9817.11	-13744	-17721.1	-21727.1	-25751
0.625	10.52114		-5026.92	-8237.38	-11532.3	-14869.5	-18230.9	-21607.3
0.75	12.03613		-4394.18	-7200.54	-10080.8	-12997.9	-15936.1	-18887.6

Table 10

where σ is the stress, M is the moment, y is the location from the centroid, and I is the moment of inertia through the cross section (see Appendix F) the maximum stresses were calculated with a safety factor of 4. From the analysis 1/4 in. thick 4X4 in. steel square tubing was found to be optimal when taking into account the cost factor.

The frame must also be easy to transport, deploy, and retrieve and still be able to support the 30 ft. length of pipe necessary for penetration. The structure will be required to be around 12 to 15 ft. tall. This requires that the frame assembly transforms into a more desirable configuration when not in operation. There were three proposed methods for achieving this. These are the hinged frame, the collapsible frame, and the sectioned bolted frame.

2. Hinged Frame:

The hinged frame is shown in figure 29. The hinged frame folds over with the requirement that a fastener must be employed to secure the sections together. The hinge allows the assembly to be quick and easy. This is essential when performing tasks at the air water interface. Other positive aspect of this design include relative ease in fabrication and its cost efficiency. The negative characteristic of the structure is that it only shifts the volume to a different section by folding the top over. This makes the frame bulky and cumbersome in transportation.

3. Collapsible Frame:

The collapsible frame is shown in figure 30. This system collapses the top portion of the frame into the bottom section when it is not in use. This can be achieved using some type of driving force to lift the top portion. This

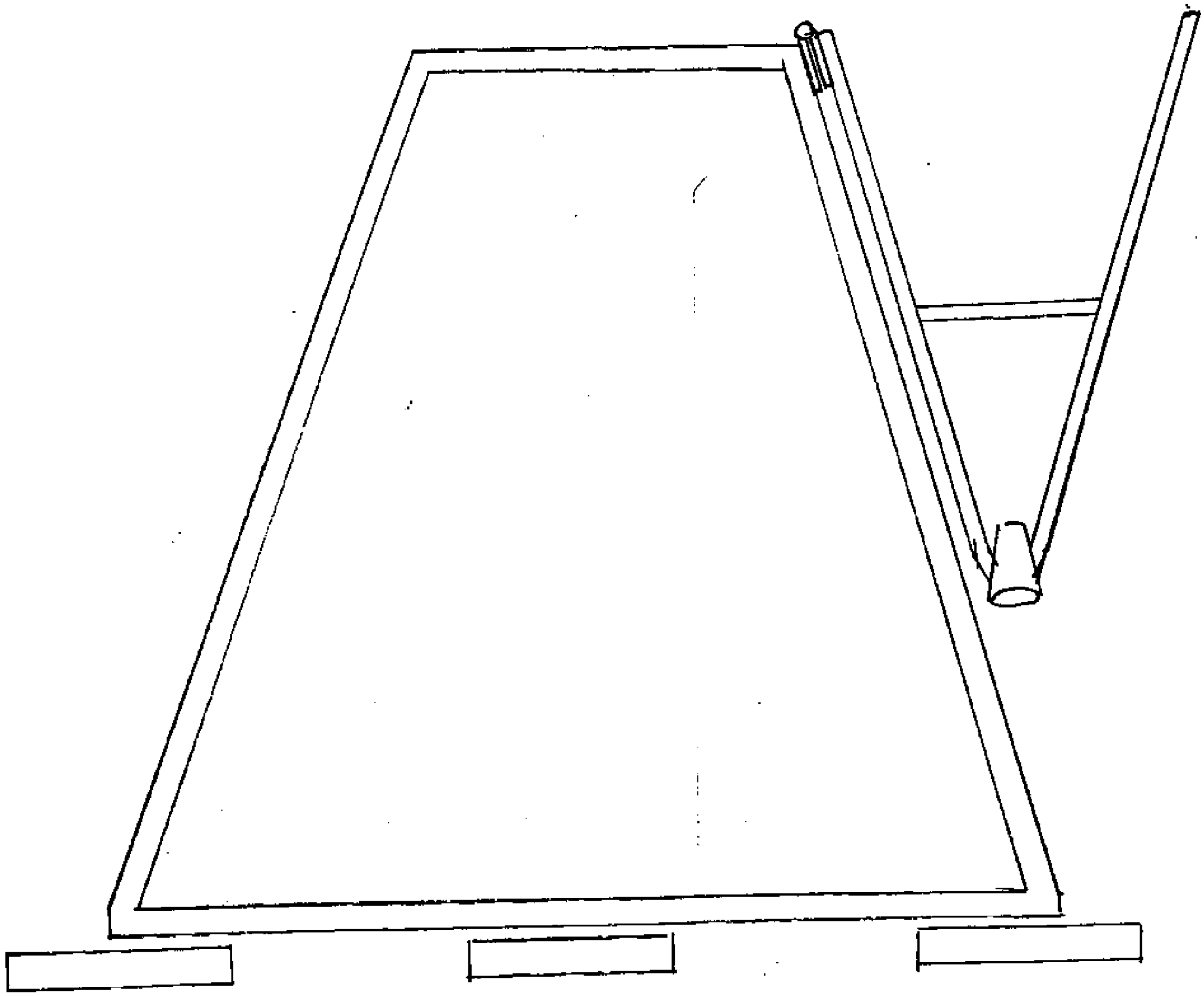


FIG 29 HINGED FRAME

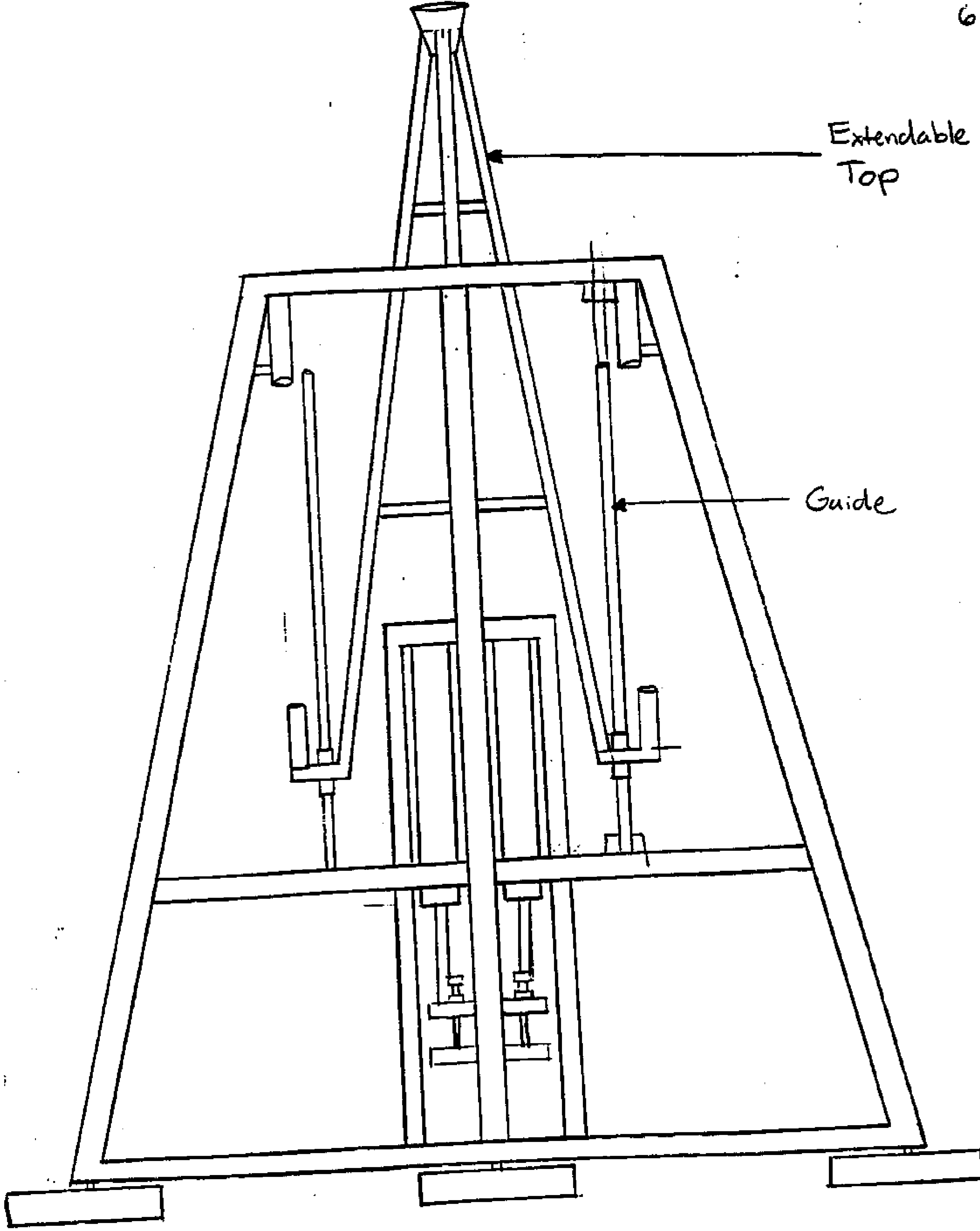


FIG. 30 EXTENDABLE FRAME

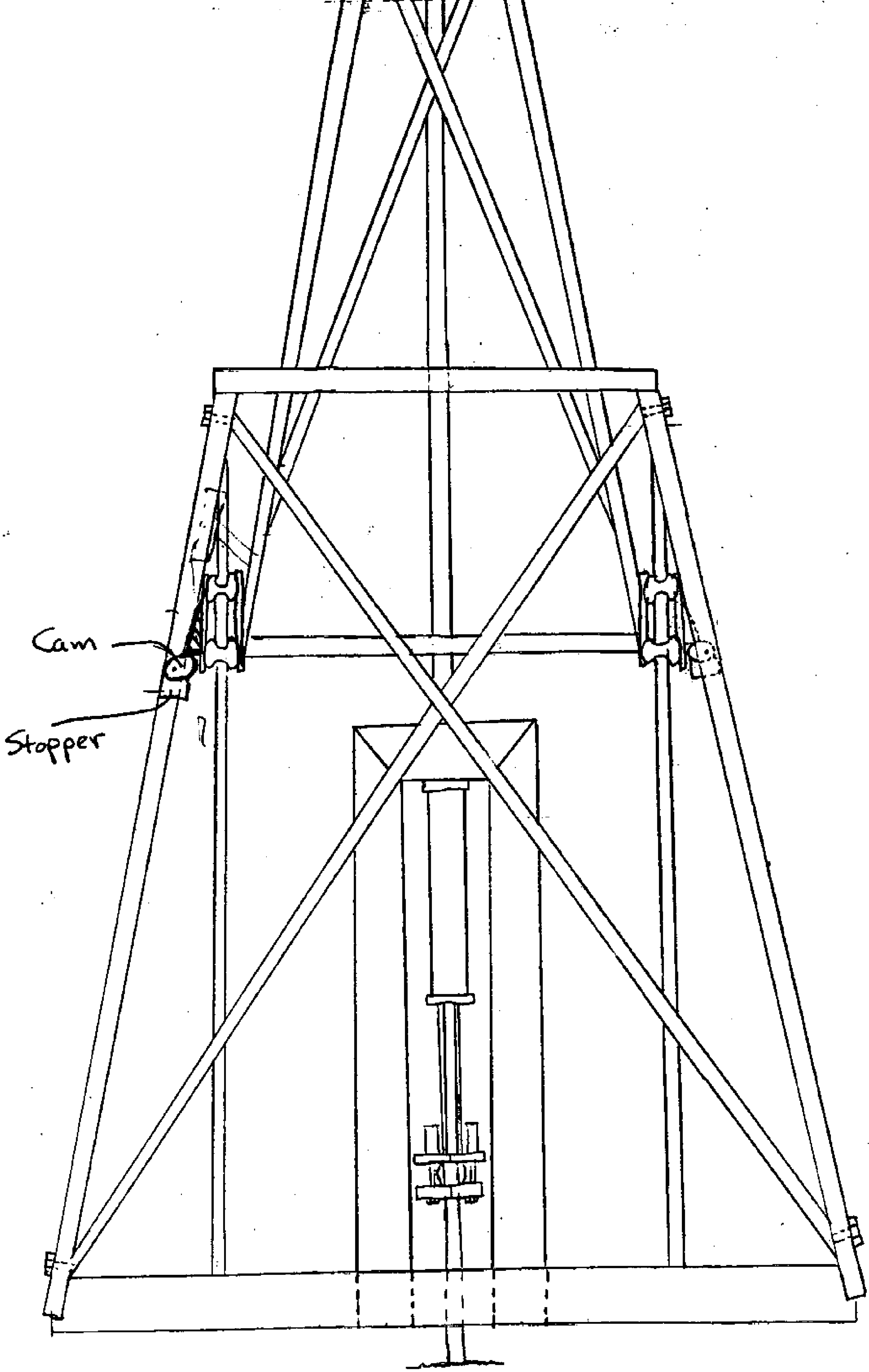
will be relatively easy since the top portion is not required to support much force, thus this section will be relatively light. This top portion then locks into place and with the use of proper geometry the moments experienced can be transmitted to the bottom portion of the truss. Three different methods for achieving this linear motion are the buoyant force, spring force, and an external force.

If the top portion of the frame is positively buoyant it will rise into place when lowered into the water. The buoyancy could be achieved by placing a material such as syntactic foam inside the top portion of the truss. The problem with this design is that it requires guides with a good enough tolerance to prevent binding.

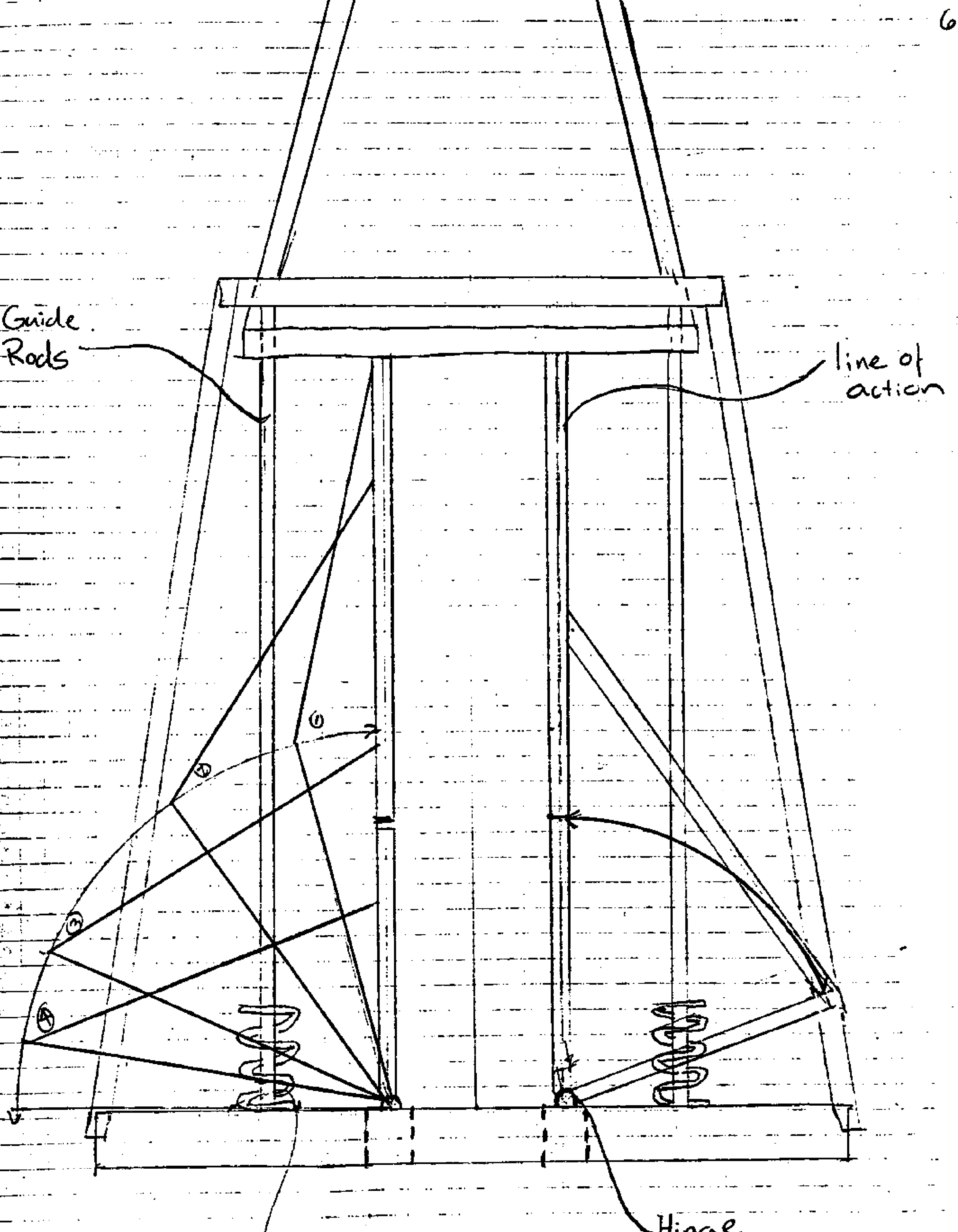
A spring force could also be used to lift the truss. By preloading the springs with the top section fastened to the base, the top section can be accelerated upward when the fasteners are removed. The difficulty in this design is that it requires large springs which will be difficult to compress for storage of the structure.

The external force method requires that the winch on the boat pull the top section into place before the system is deployed. The winch will be required to hold the bottom section while lifting the top. This method is the most optimal of the three since it is the most reliable.

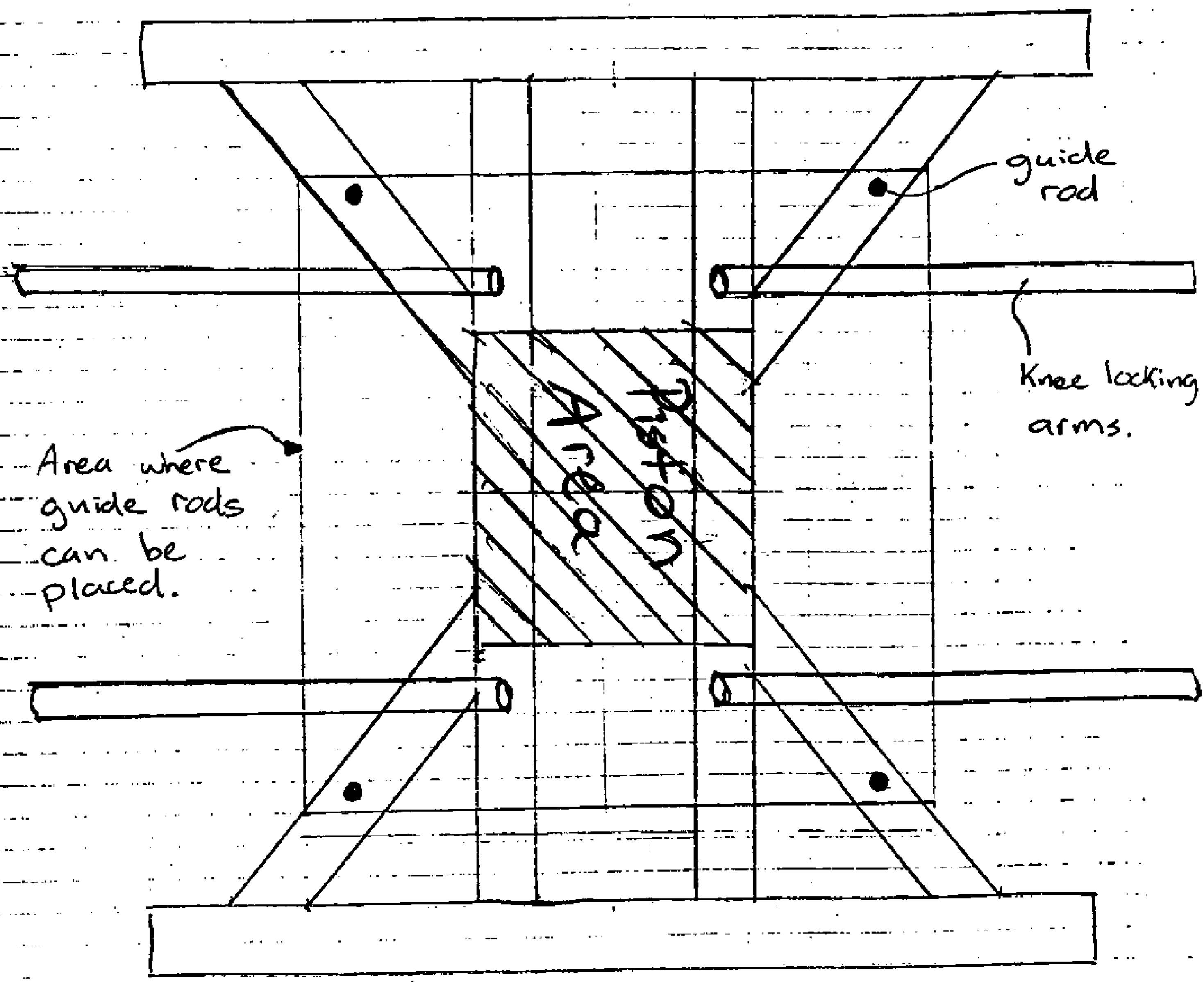
Two designs were analyzed to lock the top section into place once it is raised. These were the cam locking system (figure 31) and the knee locking system (figures 32 to 35). The cam locking system uses a rotating cam that is restricted in its rotation towards the base. Once the top frame is over the cam it cannot move downward until the cams are rotated back manually with the weight of the top section relieved by the winch.



Cam Lock
Fig 31



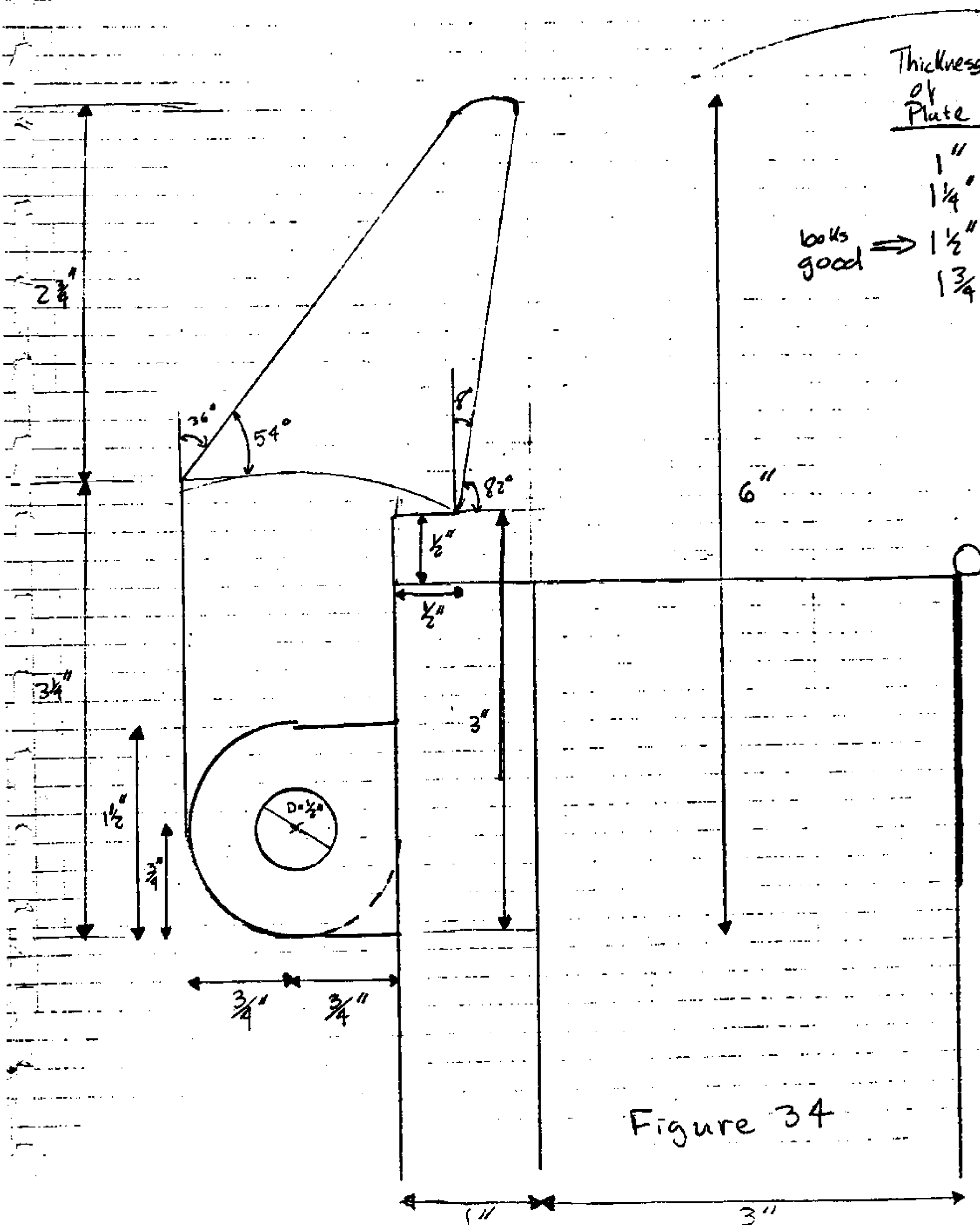
Collapsible Frame
Figure 32



Top View of Knee Locking System

Figure 33

Knee Locking Clamp
(Side View)



Thickness of Plate	taper angle (allo
1"	82°
1 1/4"	77°
looks good ⟹ 1 1/2"	72°
1 3/4"	67°

Figure 34

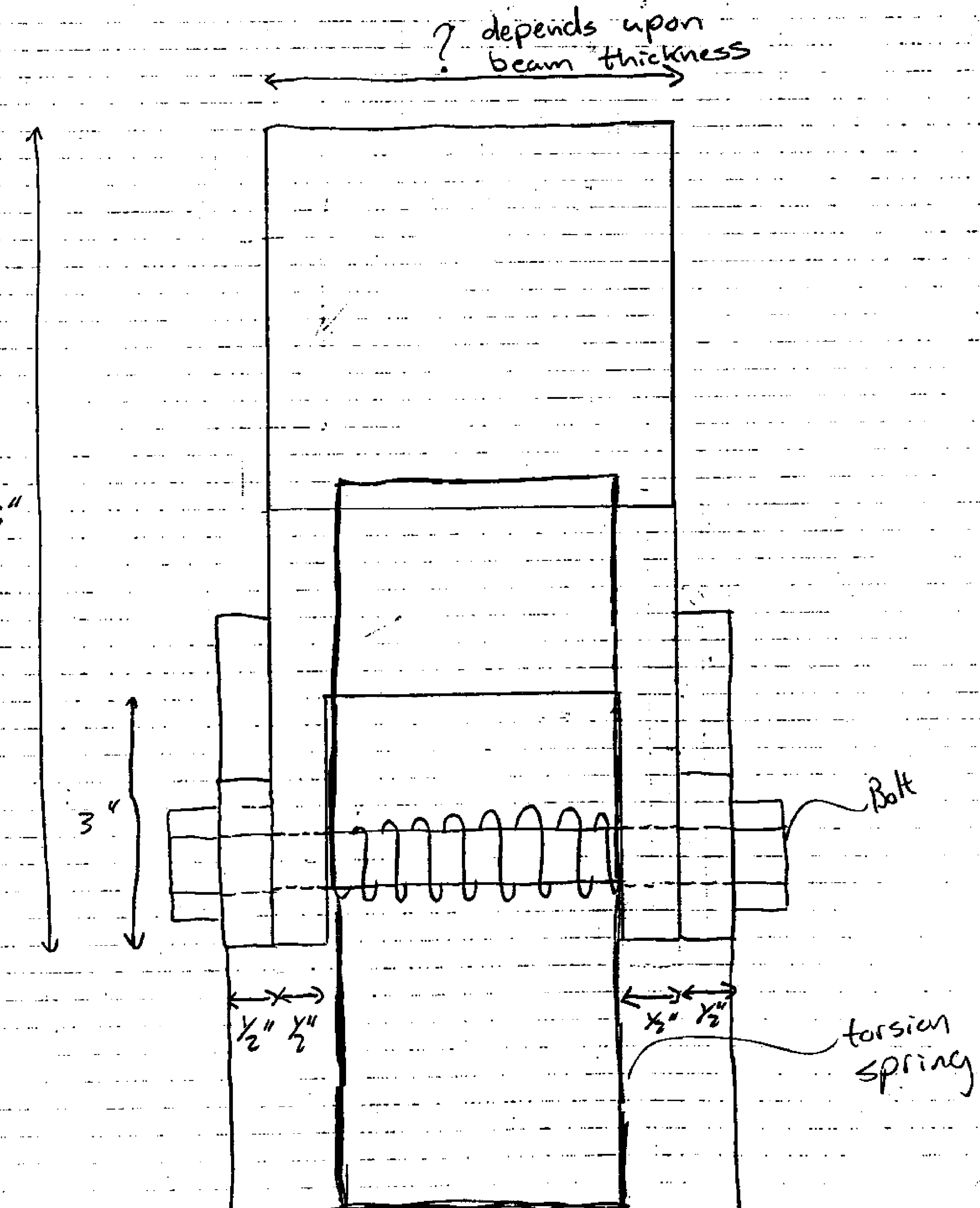


Figure 35
Knee Locking Clamp
(Back View)

The knee locking system uses two lengths of steel which are clamped rigidly when parallel with each other. This system also requires a manual release of the clamp.

The collapsing frame is easy to assemble, launch, and recover. This is mainly due to the fact that there is relatively no manual assembly. The downfalls of this design include its need for close tolerances, its cost, its lengthy fabrication, and its difficulty in transportation to the boat. Since the frame is one piece it requires heavy machinery to move it while on land.

4. Sectioned Bolted Frame

The sectioned bolted frame was the system which was chosen for the application. The sectioned frame is shown in figure 27. This design is separated into four substructures with the anchors also removable. These substructures are fastened together with high strength stainless steel bolts. The goal was to make each section relatively light to allow for easy transport to the boat. With this design it is possible to transport the frame using an average size pickup truck. A detailed step by step instruction manual on how the frame is transported, assembled, and launched and recovered is located in Appendix G.

The cons of this design include its need for close tolerances, the requirement for manual assembly, and its difficulty in launch and recovery. The difficulty in deploying the structure is that it is required that it be fully assembled on the deck. The frame is of greater height than the winch on the UNH research vessel when the top section is fastened into place. After consulting with Paul Pelletier, boat captain of the Gulf Challenger, we were informed that the frame could be launched when in full assembly with the use of special harnesses.

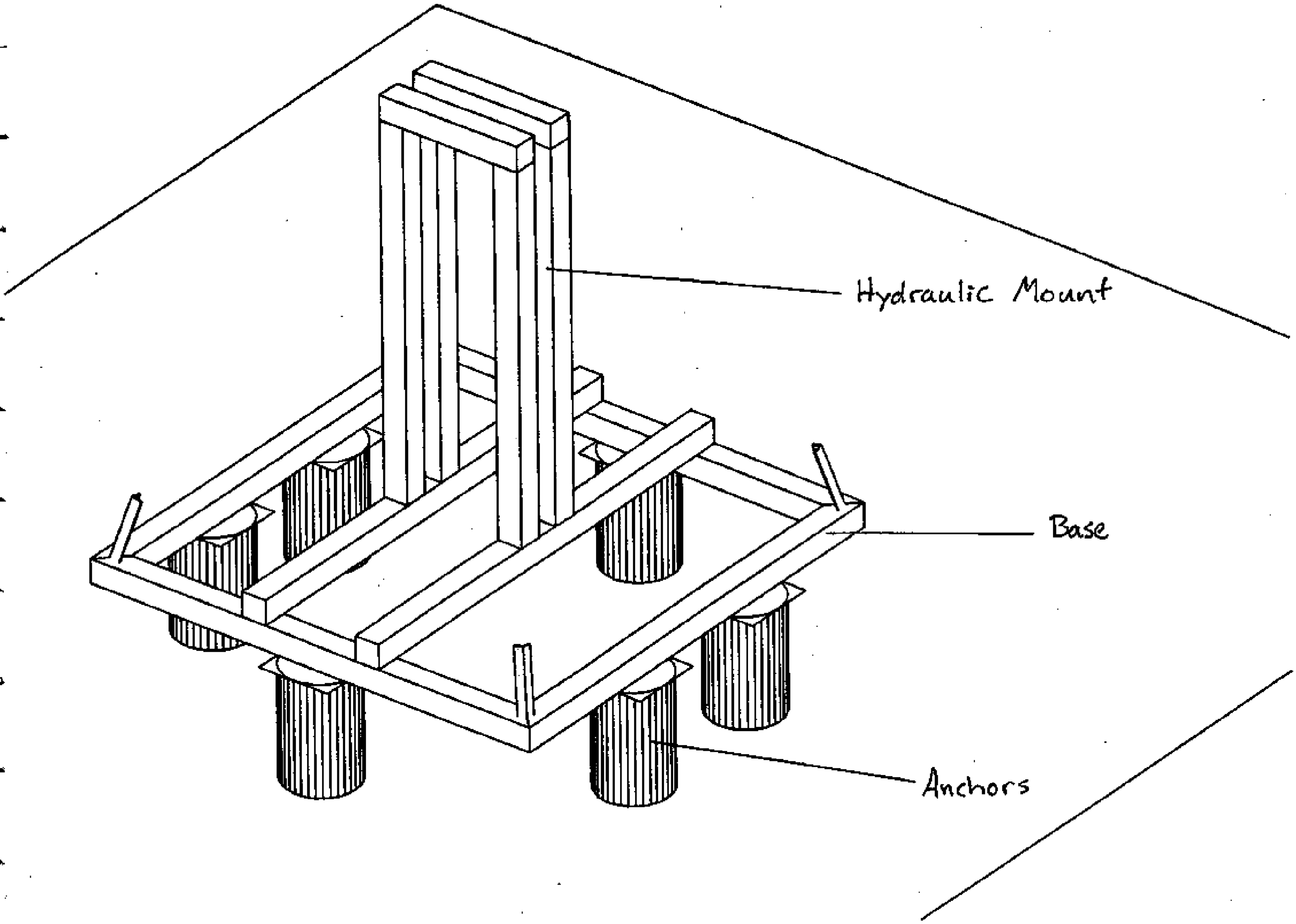
Although there are some cons to this design the benefits far outweigh them. First, as was previously mentioned, the frame is easy to transport when in sections. These sections also facilitate work on separate areas during fabrication and allow for easy replacement if a location is damaged. The most important factor for choosing this design was its cost efficiency. Due to the limited amount of parts involved it was easier to stay in the constraints of the budget. Therefore the sectioned bolted frame was the design which was chosen.

5. Fabrication

The base of the structure was constructed first (see figure 36). Care was taken when welding the joints, that the members were level with each other. After this was accomplished the bottom truss was adjusted to fit the base by first bolting the truss together loosely and then once the fit was satisfactory the joints were welded and the cross sections of angle iron were added. These sections can be fastened together with stainless steel bolts at the ear overlap shown in figure 37.

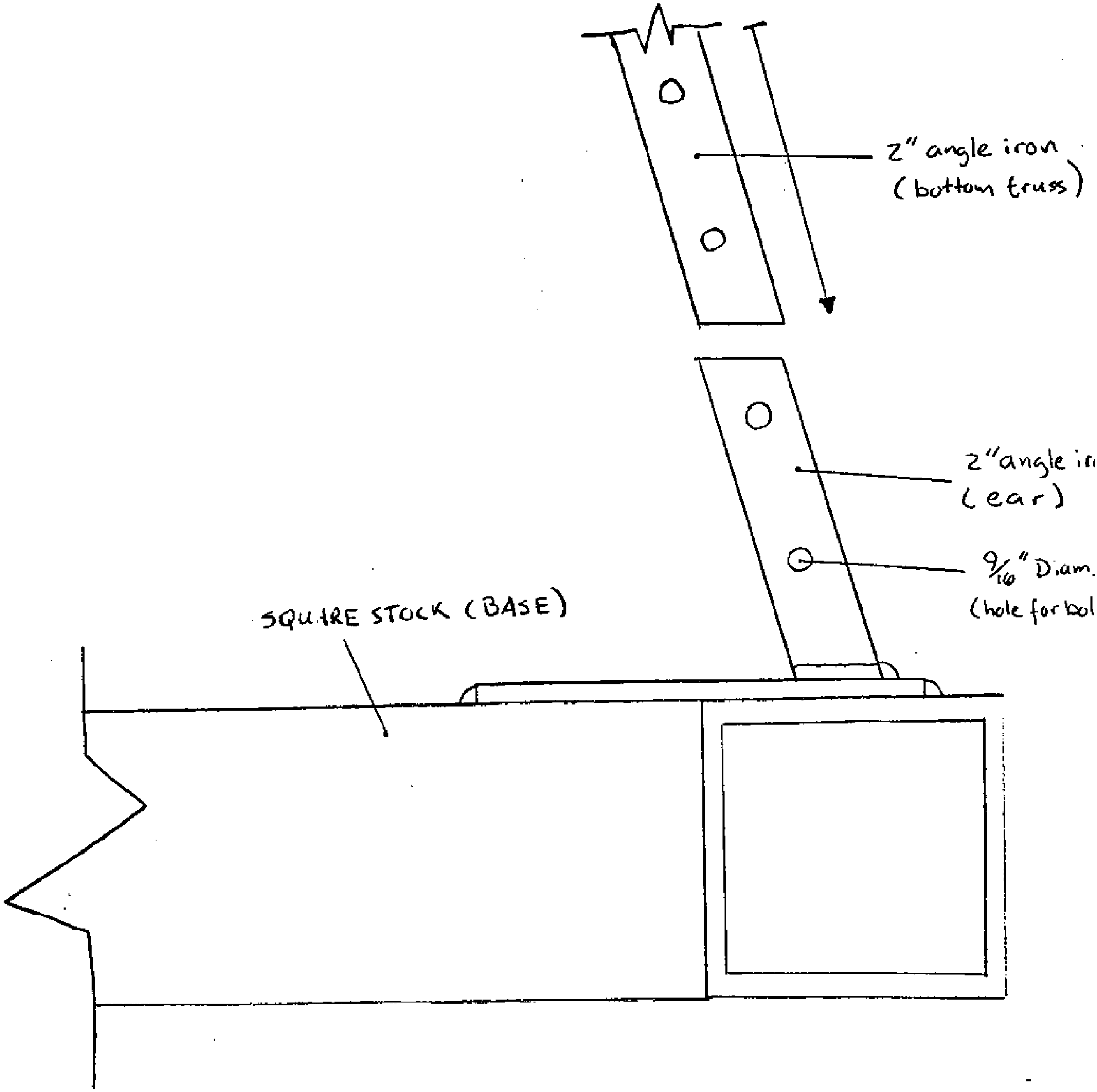
The piston mounts were also adjusted according to the base dimensions. The joint which fastens the mounts to the base is shown in figure 38. All other joints were welded. It was important to clamp the members before welding to prevent movement from thermal expansion which would effect the tolerances necessary for the drive.

The upper truss was fabricated by fitting it to the lower truss. This was important since they overlap and bolt together. This joint is shown in figure 39.



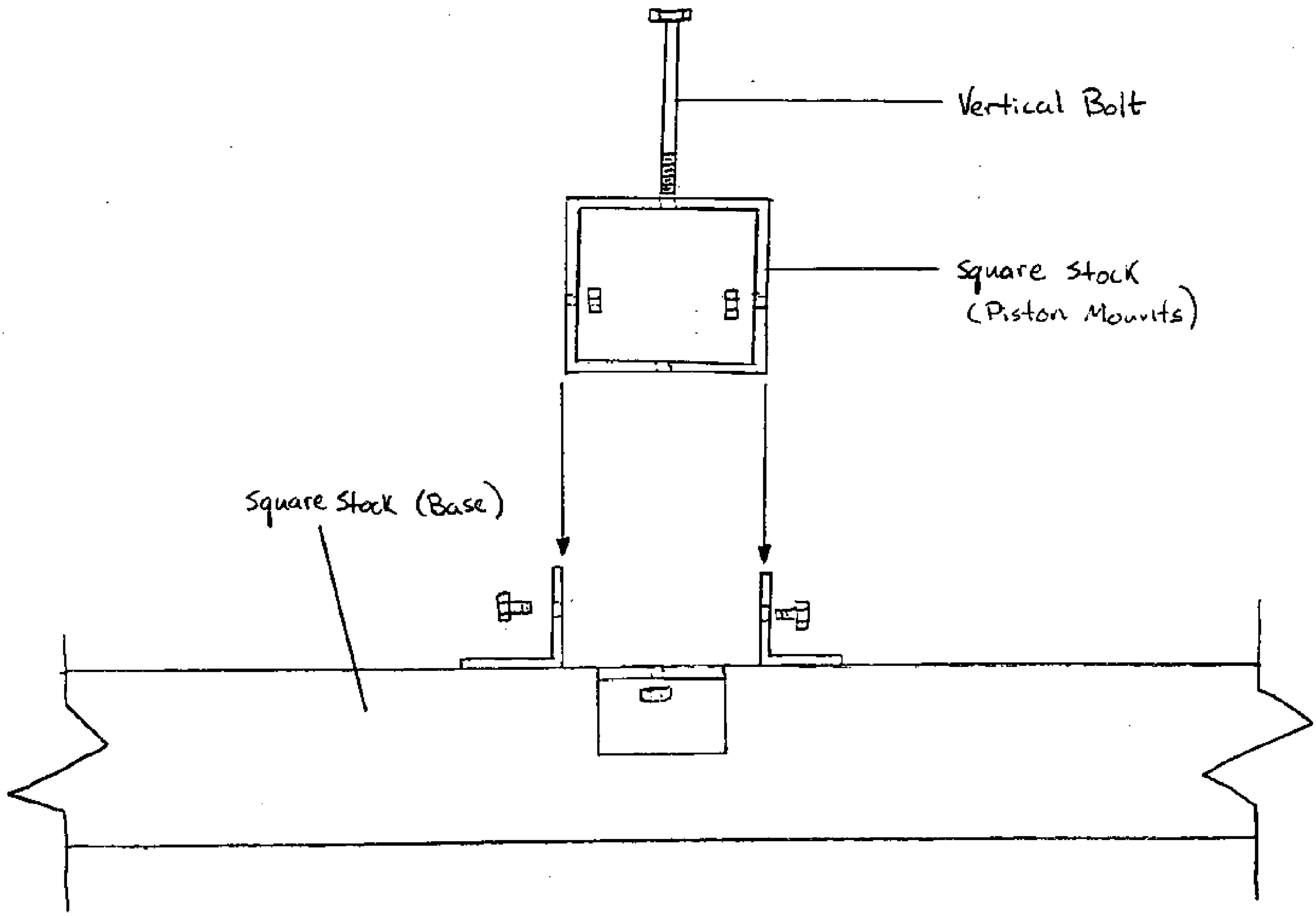
Frame: Square Stack Assembly

Figure 36



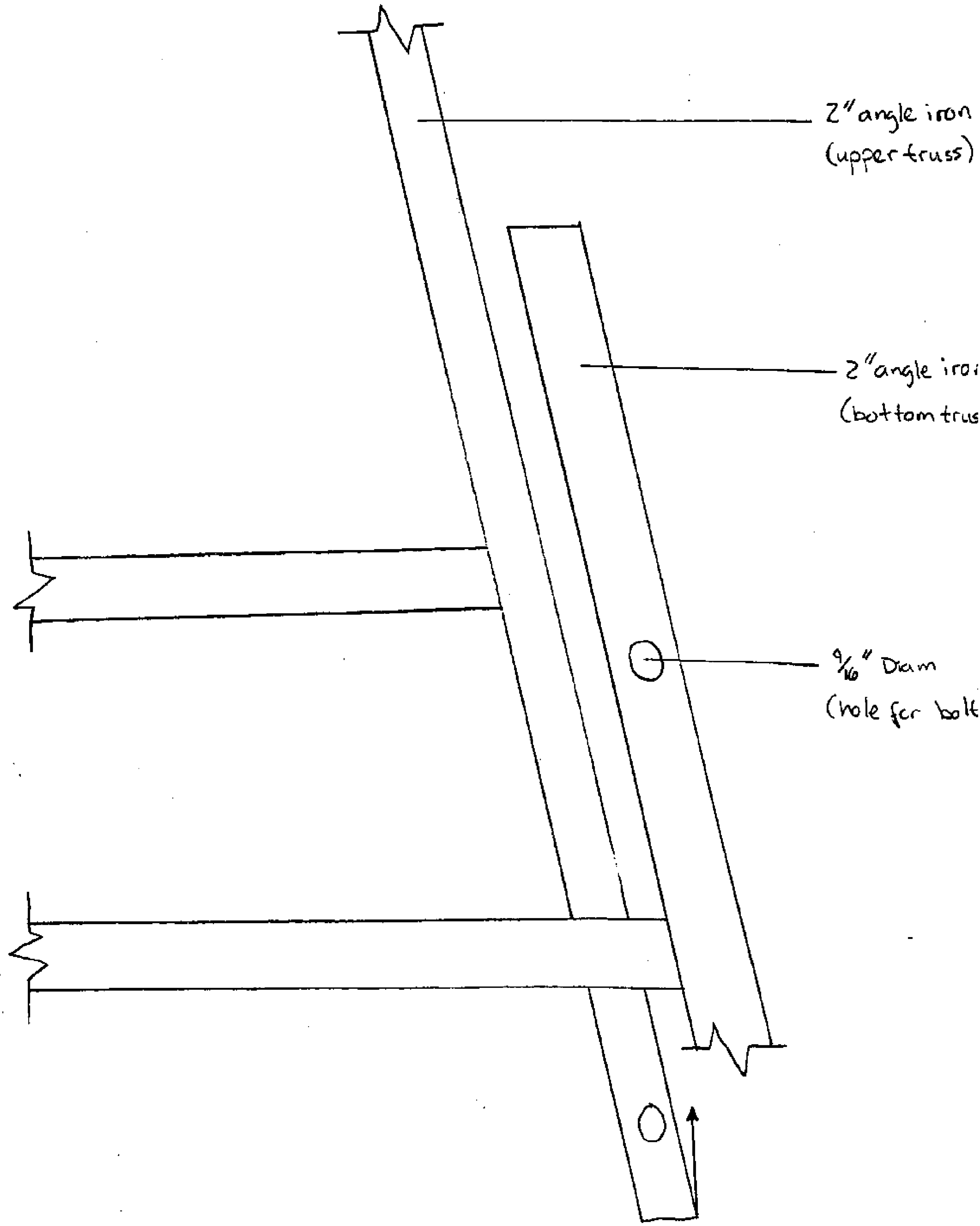
EAR OVERLAP

Figure 37



Piston Mount - Base Joint

Figure 38



Upper Truss - Bottom Truss Joint

Figure 39

E. Guide System:

1. Criteria:

The interface between the drive system and the frame assembly posed a problem due to deflections of the push rod. The shaft would not line up with the top of the frame when it was extracted from the ground. The solution to this problem was to install a set of rollers at the top of the frame and devise a system to guide it to the desired position. Two concepts were analyzed to perform this task. A funnel system attached to the top of the frame and a sheath to guide the pipe the entire distance from the break to the top of the frame.

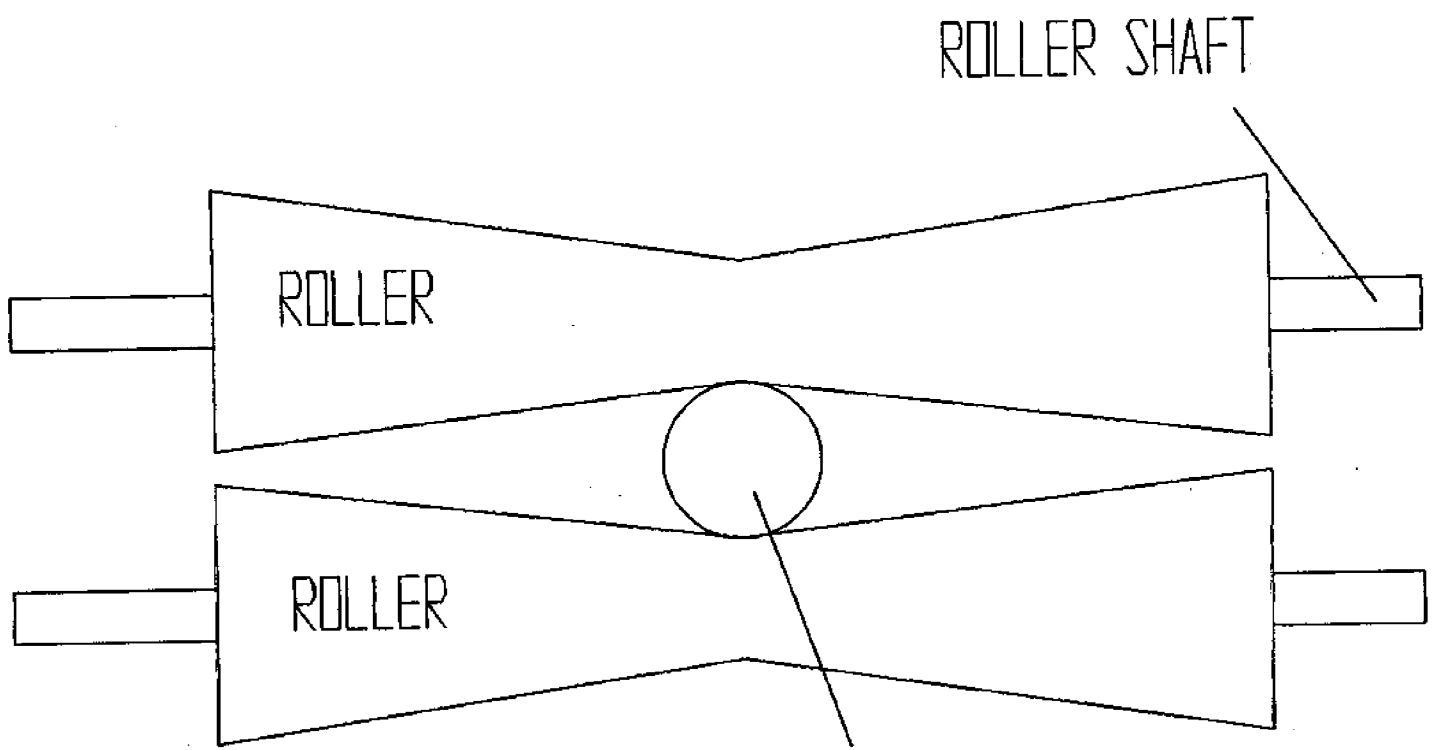
The rollers chosen to hold the pipe would keep the pipe perpendicular to the base of the frame. Boat trailer rollers were used to reduce friction at the interface, provide a small amount of breaking force, and geometrically constrain the pipe in a desired location (see figure 40). These rollers provide guidance in two directions.

2. Funnel:

The funnel system was simple to design for maximum deflections of the pipe. Light weight materials such as .125 inch plate steel could be used since the loads were small. The problem with this design was that it made the top frame section hard to manually maneuver. The added weight made the center of gravity higher and the section would tip easily when moving. It would also be difficult to attach to the frame and would require more welding and cutting. The time for construction and the cost of materials proved this method to be undesirable (see figure 41).

ROLLER ASSEMBLY

FIGURE 40



FUNNEL GUIDE SYSTEM

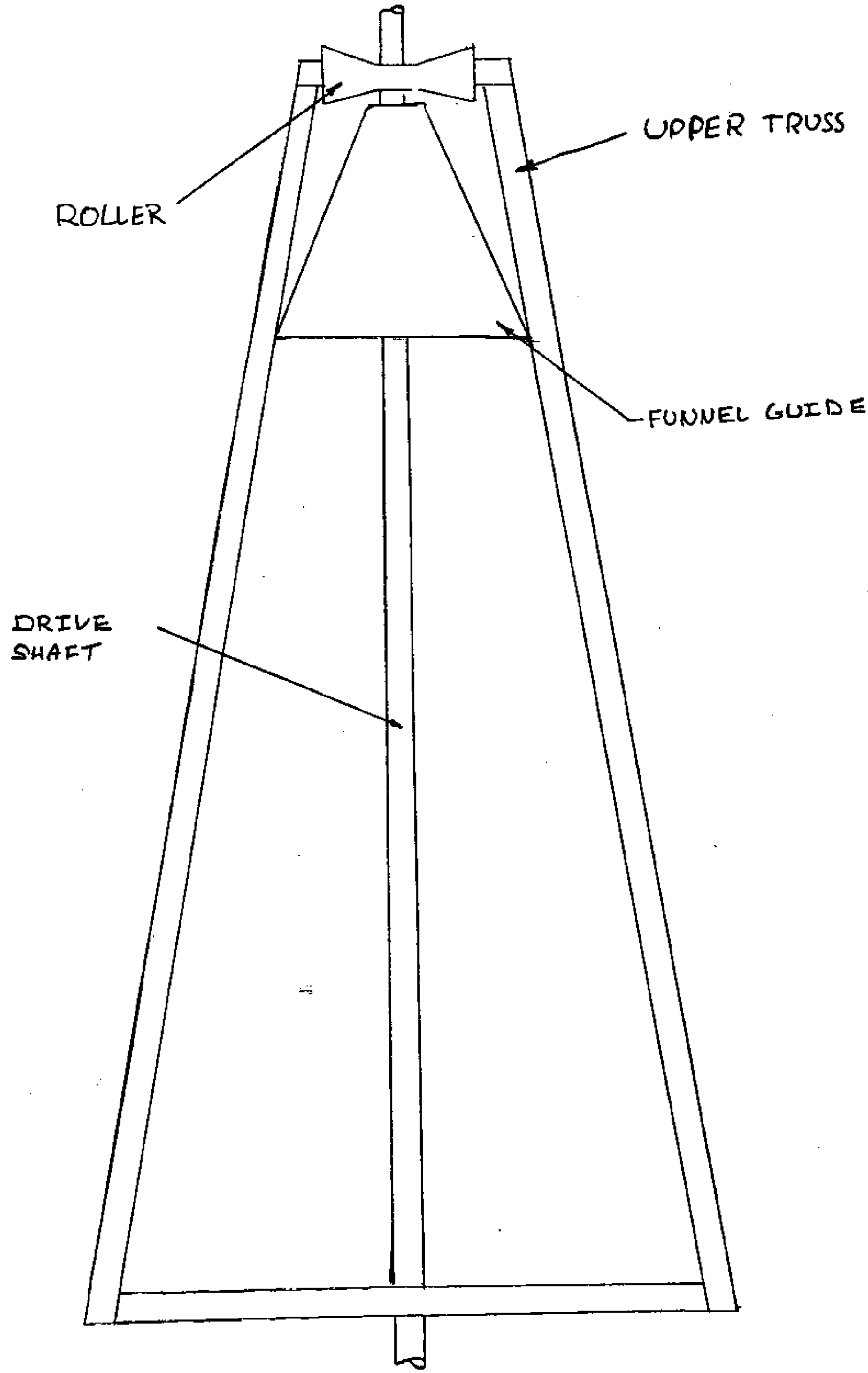


Figure 41

3. Sheath:

The second method for pipe guidance was a sheath that would constantly guide the pipe from the brake to the top of the frame. This would prevent the pipe from making large deflections while inside the frame assembly. It could also be made of lightweight materials such as PVC. This would not make the upper portion of the frame more cumbersome as did the funnel design.

4. Fabrication:

The construction of the PVC sheath would be simple. It would not require expensive machinery or construction techniques. The PVC tube could be attached to the frame using wire or elastic cables to provide some flexure of the system and allow the guide to be removed or replaced easily (see figure 42). This also makes the system easy to integrate with the rest of the frame assembly.

SHEATH GUIDE SYSTEM

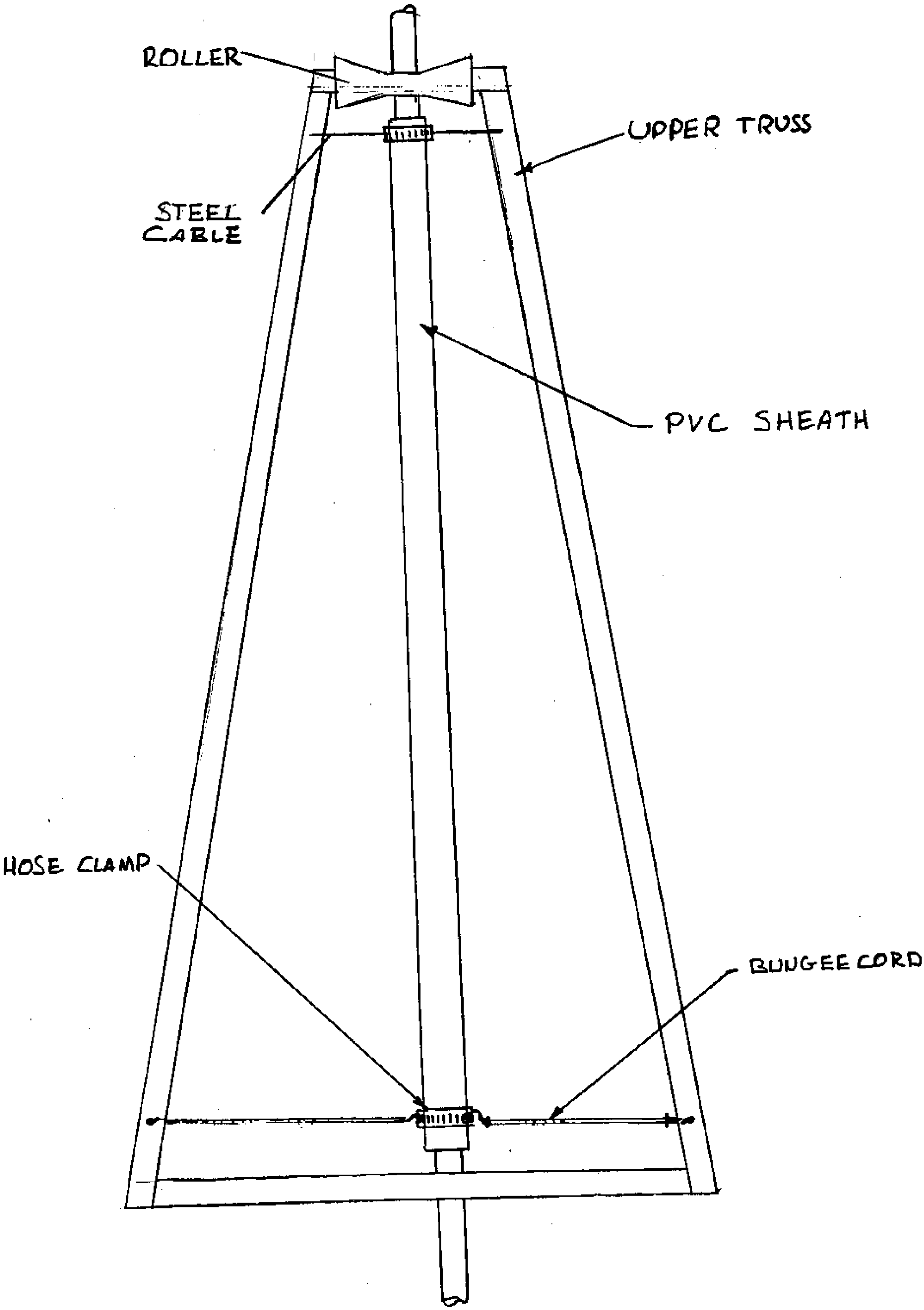


Figure 42

F. Brake:

1. Criteria:

A brake design was necessary to support the pipe while it is being extracted from the ground. From the time the clamp disengages to the time it reengages the pipe, it is unsupported. Therefore, a brake is needed to keep the pipe from falling back into the hole.

Ideally, the brake would engage only when necessary, supporting the pipe while it is being pulled out only. Also, it should not resist motion in both directions, only downward motion. It should be able to support at least the weight of the pipe, which is roughly 300 pounds.

2. Bolted Plate:

The first, and most simple, design was two plates bolted together with the pipe in between (figure 43). The tension necessary in the bolts was to be calculated, and the bolts tightened to provide enough friction to support the pipe. This design provided a very simple alternative which was low in cost and easy to fabricate. Unfortunately, this design resisted motion in any direction. The force of this brake had to be overcome no matter which direction the pipe was being pushed. Calculations supporting this design can be seen in Appendix H.

3. Brake Shoes:

The second idea for a brake design was rigging two brake shoes to provide the necessary force (figure 44). They were set up so that they would resist motion downward, but would not engage during an upward stroke. This way, the drive would only have to overcome this force on the way down, not on the way back up. This design provides a closer approximation of the ideal, while still retaining simplicity and few moving parts. Calculations supporting

Bolted Plate:

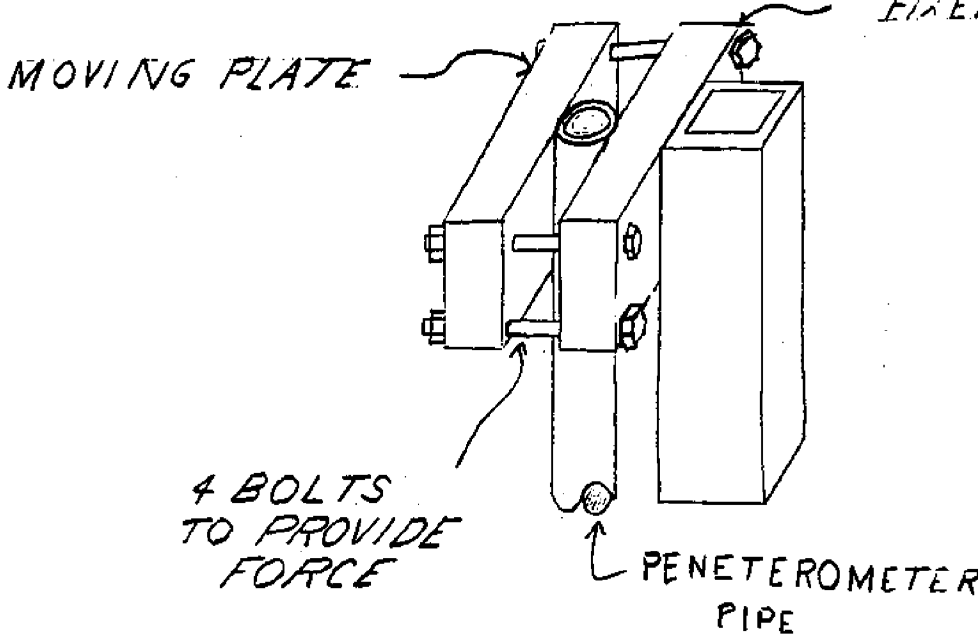


Figure 43

Brake Shoes:

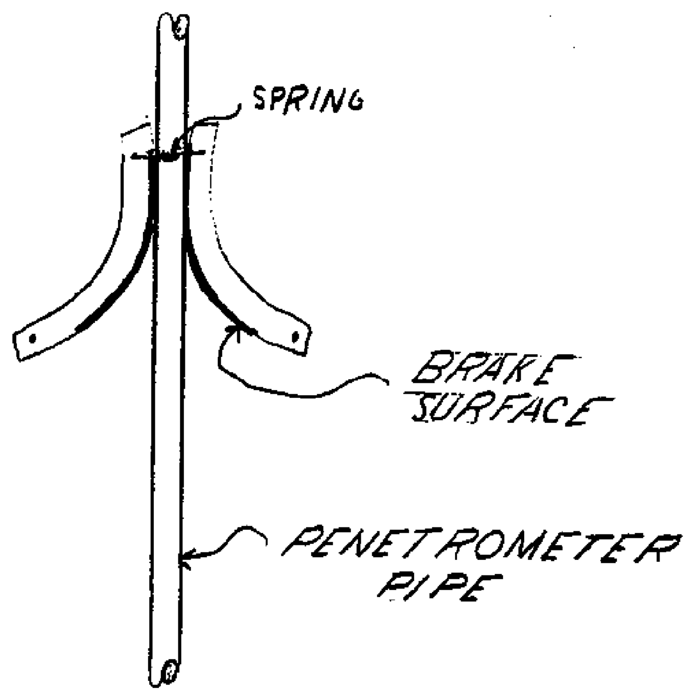


Figure 44

this design can be seen in Appendix I.

4. Fabrication:

The brake had not been constructed at the writing of this report. Design difficulties and problems due to geometry prevented the final design from being finished. It was planned to work with the brake shoe design, but the necessary spring force was much too high, and could not be easily fulfilled.

G. Launch, Recovery, and Transportation

1. Criteria:

The transportation of P^3 is a difficult matter. The system in its entirety weighs close to1700 pounds. This is too large for most of the mechanical or civil engineering departments loading equipment to move. The maximum height of the system is approximately 16 feet. This made it essential for the system to be easily assembled and disassemble into smaller, more manageable parts.

Five major sections make up the system. The heaviest being the drive/piston assembly, with associated framework, weighing close to 650 pounds. This is the only section that requires mechanical assistance. A half ton winch is necessary to move this section. The base, top frame section, bottom frame section, and anchors can be moved and loaded by three to four people. P^3 can easily be transported to a loading dock with the aid of a half ton pick-up truck. The sections are attached together with half inch stainless steel bolts.

2. Design Choice

The UNH Gulf Challenger was used to deploy P^3. All sections can be loaded directly off of a truck onto the ship's deck at the Portsmouth Fish Pier. The Challenger equipment consists of a 12 foot A-frame with a capacity of 6000 pounds and a 4000 pound winch. This is ample for the launch and recovery of P^3. The maximum height of the A-frame would not clear the top of the system so the winch has to be attached, off-center, to the top of the drive section using a bridle. An addition bridle, wrapped around the base, is necessary to restrain the platform from swinging off vertical when raised.

H. Final System Configuration:

As mentioned before, the penetrometer pushing platform was broken into systems. The frame system used is a four sectioned bolted frame. The drive system chosen is a two-way hydraulic clamp with two larger pushing pistons. The anchoring system is a combination of six suction anchors, hose and a pump. To guide the pipe vertically into the rollersc, a PVC sheath has been employed on the upper truss of the frame. Also, a bolted plate system has been added to grip the pipe during retraction from the soil. The bolted frame can be easily broken down into four pieces which can be handled by five people. The anchors are bolted on for simple removal. The relative ease of assembly of this platform facilitates launch and recovery.

System Testing:

Aside from individual systems testing, three other major tests were planned. The first test was a preliminary test of the anchors and the frame. The second test planned was a land hydraulic test. The third was a full test of all systems on the boat.

The first test was designed to determine any difficulties in using the platform on the UNH Gulf Challenger. The frame was loaded onto a truck, driven to Portsmouth, and loaded on the boat. It was assembled and taken to the test site. The top of the platform was then extended, the hose connected to the anchors, and the platform picked up by the winch on the boat. Some difficulties were encountered when attempting to deploy the platform. Because of its large size, the winch had to push against the top of the frame structure. Due to this load, the frame was forced to tip more than possible in the limited space available. Therefore, a rope was rigged to prevent this tipping. Once this difficulty was overcome, the frame was deployed. Once in place, the anchors were evacuated to lock them down, the water was pumped back in to break the seal. The frame was raised back onto the boat, taken back to shore, and disassembled. It was then loaded back up on the truck and brought back to the high bay.

The second test scheduled was a dry land test. Due to hydraulic fittings being delayed, this test was not performed. If the test had not been delayed, it would have displayed the function of the hydraulic drive system. The collet's frictional capabilities and the drive's pushing would have been tested.

Finally, a full test of all of the systems was planned. This was to be a combination of the other two tests. The frame would have been deployed again from the boat. This time, the frame would have all hydraulics hooked up. The systems would have been tested, possibly with divers present, and then used to drive a pipe into the ground.

At the time of the writing of this report, the last two tests were not performed. However, it is planned to do these tests before May 11th.

Summary & Recommendations:

A. Anchors:

Given that the anchoring force generated is time dependant, future tests would require the pump to be turned on at intervals to ensure evacuation at all times. During the test, after the anchor release was initiated, the winch began pulling up on the platform. The winch encountered significant resistance while trying to raise because not all the anchoring force was relieved as planned. Once one of the anchors breaks the suction seal during release all the water being pumped out will go to that anchor. This is because that anchor provides the path of least resistance to water flow.

The anchoring test was a success, but in the future, a manifold that could direct flow to blow out the anchors one at a time should be used. Solenoid valves could also be employed, but the cost would be significant. This would assure that all the anchoring force would be relieved before the platform was raised. Implementation of this manifold would require six hoses from the surface to the anchors. Due to time and budget constraints, this can not be achieved at this time. For now, a block and tackle setup can be applied on the winch of the boat to allow the pull up force to be greater than the anchoring force.

B. Drive:

Although not fully tested, the clamp and drive system appear to function as designed. As mentioned in the body of the report, the collet material was not entirely sufficient for the application. Also mentioned previously was the correct material which should be used in future fabrication. In an ideal case, all components used in the drive could be replaced with higher quality equivalents. Most significantly, the manual controls should be replaced by an automated system to ensure proper sequential function as well as greatly reduce user input.

C. Frame:

After putting the sectioned bolted frame through a test it was determined that the system was a success. The test was used to show if the planned methods for launch, recovery, assembly, and transportation were valid. The frame facilitated transportation with the use of a pickup truck. Two trips were necessary. The facilities at the fish pier in Portsmouth N.H. were sufficient for loading the frame on the Gulf Challenger.

The frame needed to be harnessed in two sections to allow for launch. This process was rather simple and painless. No deformations in the frame were observed after it was recovered.

From the three designs the sectioned, bolted frame was optimal based on its ease in transportation. More research could be performed to reduce the weight of the structure. This could be achieved using other materials such as aluminum. When making this transition one must be aware of the corrosive properties between the steel pistons and the aluminum frame. Materials such as aluminum were avoided in this design due to the cost factor.

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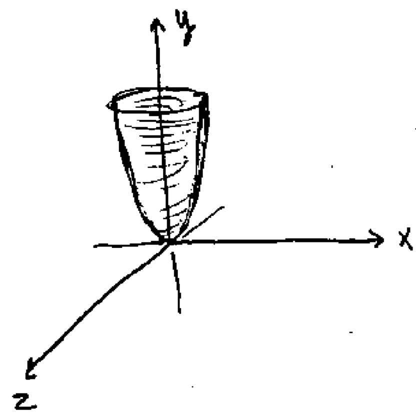
Anthony Hall
Paul Lavoie
Ingersol Rand
Phil Coyne & Chris Doane
Tracey Harvey & Carole McCarthy
Heather Talbot
Gerald Sedor
Ken Houtler & Co.
Paul Pelletier
John Pavlos
FedCo Manufacturing
Mike Gosz
Wally Fournier
Todd Gross
Godfrey Savage

Appendix A

Auger Holding Force

Mathematical Model For Anchoring Force:



$$y = x^2 + z^2$$

① find Volume

$$V = \int_{-z}^{z} \int_{-\sqrt{y-x^2}}^{\sqrt{y-x^2}} \left(y - x^2 - z^2\right)\sqrt{x^2+z^2}\; dz\, dx$$

$$V = \int_0^{2\pi} \int_0^{r} (y - r^2)\, r \; r\, dr\, d\theta$$

$r^2 = x^2 + z^2$
$r = \sqrt{x^2+z^2}$

$$V = 2\pi \left[\frac{yr^3}{3} - \frac{r^5}{5}\right]\Bigg|_0^{r}$$

Note where $x = z$
$r = \sqrt{y}$

② Find Force

$$V\,(\rho_{avg})\, g = F$$

③ Using Known Data (4 foot anchors $\longrightarrow$ 650 lbs)

$$650 = (32.2)(\rho)\, 2\pi \left[\frac{4(\sqrt{4})^3}{3} - \frac{(\sqrt{4})^5}{5} \right]$$

$$\underline{\rho_{avg} = .752989 \text{ slugs}/\text{ft}^3}$$

$$\underline{\rho_{avg} = 24.24 \text{ lbs}/\text{ft}^3}$$

④ Try New Anchor lengths

<u>8 Feet</u>

$$F = (32.2)(.75298)\, 2\pi \left[\frac{8(\sqrt{8})^3}{3} - \frac{(\sqrt{8})^5}{5} \right]$$

$$\underline{F = 3676.9 \approx 3700 \text{ lbs}}$$

Appendix B

Suction Anchor Geometry

Steel → standard 10 gage = .123 in

A

$d = 12$ in

Force, Volume

$y = 0.646X + 30.4$

$8000 = 0.646(x) + 30.4$

$X = 12336.842 \text{ in}^3$

$V_T = \pi r^2 h$

$V_T = \pi(6)^2 h = 12336.842$

$h = 109$ in

6 → 18.1" in sub – too big

Try F = 6000 lb

$X = 9240.86$

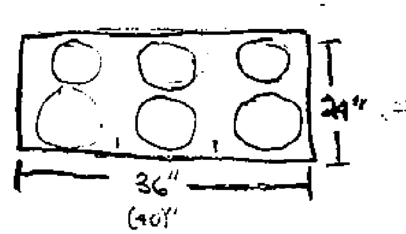
$h = 81.7$

6 → 13.6" sub

6 → 15" should be okay

B

Area For Tops:



circle

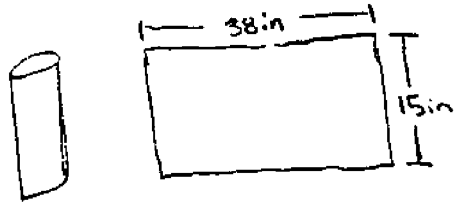
$\pi r^2 = \pi(6)^2 = 113.1 \text{ in}^2$

For six $= 678.5 \text{ in}^2 \approx \text{Area}_{ACTUAL} = 30 \cdot 40 = 1200$

Volume For (6) Actual (TOPS) anchors: (10 Gage)

$V_A = (1200 \text{ in}^2)(.123 \text{ in}) = 147.6 \text{ in}^3$

Find total Volume For Anchors:



$\pi d = \pi\, 12 = 37.699 \approx 38\text{ in}$

$t = .123\text{ in } (10\text{ gage})$

$$V_{1\,ANCHOR} = 38 \cdot 15 \cdot .123 = 70.11\text{ in}^3$$

$$V_{6\,ANCHORS} = \underline{420.66\text{ in}^3}$$

Total Volume Estimate:

$$V_{TOTAL} = 420.66\text{ in}^3 + 147.6\text{ in}^3 = \underline{568.26\text{ in}^3}$$

$$(568.26\text{ in}^3)\left(\frac{1\text{ ft}^3}{12\text{ in}^3}\right)^3 = .3288\text{ ft}^3$$

Price Estimate

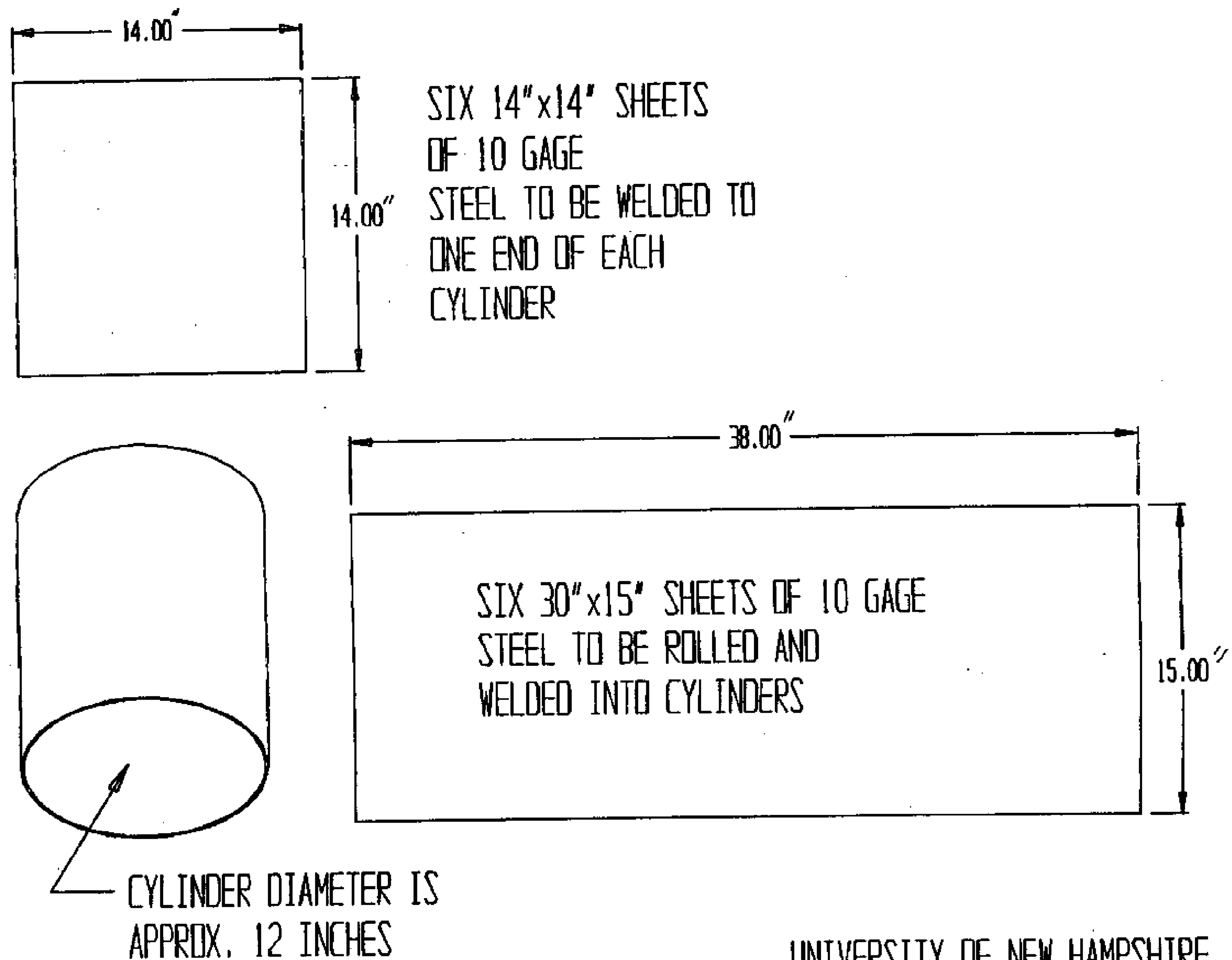
steel \$.25/lb

$$\rho = \left(7.85\text{ g/cm}^3\right)\left(\frac{1\text{ lbm/ft}^3}{1.602\times10^{-2}\text{ g/cm}^3}\right)$$

$$\rho = 490\text{ lbm/ft}^3$$

$$\left(490\text{ lbm}/\cancel{\text{ft}^3}\right)\left(.3288\text{ ft}^3\right) = \underline{161.11\text{ lb}_m}$$

$$\boxed{\text{COST} = \$40.27}$$



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MARCH 7, 1994

Appendix C

Suction Anchor Stresses

Buckling Analysis For Suction Anchor

$$P_{crit} = \frac{C E I}{L^2}$$

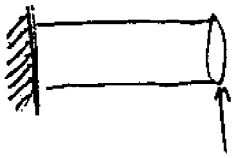
$I = 3.995 \times 10^{-3}\ ft^4$
$L = 1.25\ ft$
$C = 39.5$
$E_{STEEL} = 28 \times 10^6\ psi = 4.032 \times 10^9\ lb/ft^2$

$$P_{crit} = \frac{(4.032 \times 10^9)(39.5)(3.995 \times 10^{-3})}{(1.25)^2}$$

$$\boxed{\begin{array}{l} P_{CRIT} = \underline{407 \times 10^6\ lbs} \\ P_{USED} = \dfrac{2000\ lbs}{6} = \underline{333\ lbs} \end{array}}$$

Bending Stress Calculation:

$D_o = 12 \text{ in} = 1 \text{ ft}$



$\sigma = \frac{My}{I}$ $\quad y = 6 \text{ in} = .5 \text{ ft}$ $\quad D_i = 11.754 \text{ in} = .979$

$l = 1.25 \text{ ft.}$

$M = (2000^{\text{lbs}})(1.25 \text{ ft}) = 2500 \text{ lb-ft}$

For 10 gage
$t = .123 \text{ in}$

$I = \frac{\pi D_o^4}{64} - \frac{\pi D_i^4}{64} = 3.995 \times 10^{-3} \text{ ft}^4$

$\sigma = \frac{(2500)(.5)}{3.995 \times 10^{-3}} = 312.87 \times 10^3 \text{ lbs/ft}^2$

$\sigma \cdot \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) \Rightarrow \underline{2172.7 \text{ psi}}$

SF of 2

$\sigma_{SF} = \underline{4400 \text{ psi}}$

$\sigma_y = 43 \text{ Ksi}$ $\quad \rightarrow$ OK ✓

Appendix D

Drag Calculation

Drag Calculation

$$R = \frac{\rho V_\infty D}{\mu}$$

(for cylinder)

Assume: $T = 20°C \quad \rho = 998\ Kg/m^3$

$\mu = 1.00 \times 10^{-3}\ N \cdot s/m^2$ for H_2O

$V_\infty = 1\ \text{Knot} = \frac{1852\ m}{hr} = 0.5144\ m/s$

$D = 1.44'' = \frac{1.44''}{39.7\ in/m} = 0.03627\ m$

$$R = \frac{(998\ Kg/m^3)(0.5144\ m/s)(0.03627\ m)}{1.00 \times 10^{-3}\ N \cdot s/m^2} = 18619.97$$

From figure 8.10 <u>Fluid Dynamics</u> $C_D \approx 0.4$

$$D_c = C_D \left(\tfrac{1}{2}\right) \rho V^2 \ell d$$

$\ell = 30\ ft = 9.1436\ m$

$d = 0.03627\ m$

$$D_c = 0.4\left(\tfrac{1}{2}\right)(998)(0.5144)^2(9.1436)(0.03627)$$

$= 34.05\ N \Rightarrow$ for pipe

Approximation of Drag on Frame

- find area normal to flow

$$A = 4(8)(12) + 4(8)(12) + 2(10)(12) + 2(10)(12) + 2(3)(12) + (5(2) + 5(2))12 + 25(25)$$

beams at base

angle iron

extras

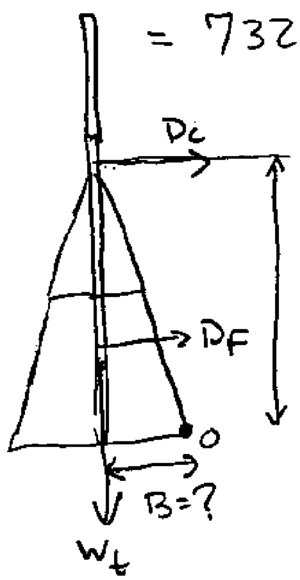
$$A = 2185\ in^2 = \frac{2185\ in^2}{(39.7\ in/m)^2} = 1.386\ m^2$$

there are two sides $\rightarrow|$ to flow so

$$A_{tot} = 1.386 m^2 (2)$$
$$= 2.773 m^2$$

$C_D = 2$ for Normal flat plate or square rod

$$D_F = C_D (1/2) \rho V^2 (2.773) = 2(1/2)(998)(0.5144)^2(2.773)$$
$$= 732.21 N$$



$\updownarrow \frac{1}{3}H$ Assume $H \approx 12 ft$

$$\Sigma M_o = 0 = W_t B - D_c (15') - D_F (4')$$
$$W_t B = D_c (15) + D_F (4)$$

Find weight (Approximate)

W_t = Base + sides + hydraulics + Anchors + rod

Base = $6 \times 6 \times 15.7$ lb/ft $\times 2$ = 1130 lb

Sides = $[10(4) + 3(4)]\, 4.7$ lb/ft + $[5(4)\, 1.49$ lb/ft$]$

= 274.2 lb

hydraulics $\approx$ 160 lb

Anchors $\approx$ 40 lb $\qquad W_t \approx 1904.6$ lbF

rod $\approx$ 300 lb $\qquad \approx 8532.6$ N

$$8532.6 (B) = 34.05 (15) + 252.99 (4)$$
$$B = 0.178 ft \qquad \text{total width} = 0.357 ft$$

$\therefore$ Drag due to water will not effect Frame. It is negligable.

Appendix E

Finite Element Analysis

Sections of FEA Output File

```
program sizing and options requested as follows


element type requested***************************      9
number of elements in mesh***********************     16
number of nodes in mesh**************************      8
max number of elements in any dist load list***       16
maximum number of boundary conditions**********       36
load correction flagged or set*****************
number of lists of distributed loads***********        5
stresses stored at all integration points******
tape no.for input of coordinates + connectivity        5
no.of different materials     1 max.no of slopes       5
maximum elements variables per point on post tp       33
number of points on shell section *************       11
new style input format will be used************
maximum number of set names is*****************       10
number of processors used *********************        1
vector length used ****************************        1


end of parameters and sizing
************************************************
************************************************
```

Defining

1

```
key to stress, strain and displacement output


        element type    9


2-node, 3-d truss

stress and strain are uniaxial

displacements in global directions
  1=u global x direction
  2=v global y direction
  3=w global z direction


workspace needed for input and stiffness assembly        7985


internal core allocation parameters
degrees of freedom per node (ndeg)  3
coords per node (ncrd)  3
strains per integration point (ngens)  1
max. nodes per element (nnodmx)  2
max.stress components per int. point (nstrmx)   1
max. invariants per int. points (neqst)  1


flag for element storage (ielsto)  0
```

```
von mises ,heic ..lter..
isotropic  hardening rule
   e          nu          rho        alpha       yield       yield2
 0.300E+08  0.300E+00  0.100E+01  0.000E+00  0.100E+21  0.100E+21
from element      1 to element     16 by      1
```

geometry

```
  egeom1     egeom2     egeom3     egeom4     egeom5     egeom6
 0.100E+01  0.000E+00  0.000E+00  0.000E+00  0.000E+00  0.000E+00
a list of elements given below
     1      4      5      6      7      8      9     12
  egeom1     egeom2     egeom3     egeom4     egeom5     egeom6
 0.194E+01  0.000E+00  0.000E+00  0.000E+00  0.000E+00  0.000E+00
a list of elements given below
     2      3     10     11
```

fixed disp

```
fixed displacement =  0.000E+00  0.000E+00  0.000E+00
a list of degrees of freedom given below
     1      2      3
a list of nodes given below
     1      2      5      6
```

```
fixed boundary condition summary.
total fixed degrees of freedom read so far =    12
```

b.c. number	node	degree of freedom	magnitude	b.c. number	node	degree of freedom	magnitude
1	1	1	0.000E+00	2	1	2	0.000E+00
3	1	3	0.000E+00	4	2	1	0.000E+00
5	2	2	0.000E+00	6	2	3	0.000E+00
7	5	1	0.000E+00	8	5	2	0.000E+00
9	5	3	0.000E+00	10	6	1	0.000E+00
11	6	2	0.000E+00	12	6	3	0.000E+00

Boundary Conditions

point load

```
read from unit     5
 0.422E+03 0.500E+03-0.422E+03
a list of nodes given below
     3
-0.422E+03 0.500E+03-0.422E+03
a list of nodes given below
     8
-0.422E+03 0.500E+03 0.422E+03
a list of nodes given below
     9
 0.422E+03 0.500E+03 0.422E+03
a list of nodes given below
     7
```

Loads Applied

```
total workspace needed with in-core matrix storage =      8621


load increments associated with each degree of freedom
summed over the whole model

distributed loads
0.000E+00 0.000E+00 0.000E+00

point loads
0.000E+00 2.000E+03 0.000E+00

start of assembly
 time =       0.55


start of matrix solution
 time =       0.60


singularity ratio    1.0993E-01


end of matrix solution
 time =       0.63

1
MARC-SGI K5-2,  03/29/94, output for increment    0.    job1


element with highest stress relative to yield is    4  where equivalent stress is 0.517E-17  of yield



  tresca     mises     mean    p r i n c i p a l   v a l u e s         p h y s i c a l   c o m p o n e n t s
intensity intensity   normal   minimum intermediate maximum      1          2          3          4          5          6
                      intensity

element     1  point   1       integration pt. coordinate=      0.748E+01    0.420E+02   -0.748E+01
section thickness = 0.100E+01
stress  5.137E+02 5.137E+02 1.712E+02 0.000E+00 0.000E+00 5.137E+02 5.137E+02
strain  1.712E-05 1.398E-05 0.000E+00 0.000E+00 0.000E+00 1.712E-05 1.712E-05

element     2  point   1       integration pt. coordinate=      0.360E+02    0.000E+00    0.000E+00
```

Stress and Strain at Element

```
section thickness = 0.194E+01

element    3  point   1       integration pt. coordinate=     0.720E+02    0.000E+00   -0.360E+02
section thickness = 0.194E+01

element    4  point   1       integration pt. coordinate=     0.647E+02    0.420E+02   -0.748E+01
section thickness = 0.100E+01
stress  5.172E+02 5.172E+02 1.724E+02 0.000E+00 0.000E+00 5.172E+02 5.172E+02
strain  1.724E-05 1.408E-05 0.000E+00 0.000E+00 0.000E+00 1.724E-05 1.724E-05

element    5  point   1       integration pt. coordinate=     0.361E+02    0.840E+02   -0.150E+02
section thickness = 0.100E+01
stress  3.330E+02 3.330E+02-1.110E+02-3.330E+02 0.000E+00 0.000E+00-3.330E+02
strain  1.110E-05 9.063E-06 0.000E+00-1.110E-05 0.000E+00 0.000E+00-1.110E-05

element    6  point   1       integration pt. coordinate=     0.150E+02    0.840E+02   -0.361E+02
section thickness = 0.100E+01
stress  3.346E+02 3.346E+02-1.115E+02-3.346E+02 0.000E+00 0.000E+00-3.346E+02
strain  1.115E-05 9.106E-06 0.000E+00-1.115E-05 0.000E+00 0.000E+00-1.115E-05

element    7  point   1       integration pt. coordinate=     0.361E+02    0.840E+02   -0.573E+02
section thickness = 0.100E+01
stress  3.346E+02 3.346E+02-1.115E+02-3.346E+02 0.000E+00 0.000E+00-3.346E+02
strain  1.115E-05 9.106E-06 0.000E+00-1.115E-05 0.000E+00 0.000E+00-1.115E-05

element    8  point   1       integration pt. coordinate=     0.573E+02    0.840E+02   -0.361E+02
section thickness = 0.100E+01
stress  3.330E+02 3.330E+02-1.110E+02-3.330E+02 0.000E+00 0.000E+00-3.330E+02
strain  1.110E-05 9.063E-06 0.000E+00-1.110E-05 0.000E+00 0.000E+00-1.110E-05

element    9  point   1       integration pt. coordinate=     0.647E+02    0.420E+02   -0.647E+02
section thickness = 0.100E+01
stress  5.169E+02 5.169E+02 1.723E+02 0.000E+00 0.000E+00 5.169E+02 5.169E+02
strain  1.723E-05 1.407E-05 0.000E+00 0.000E+00 0.000E+00 1.723E-05 1.723E-05
1
 MARC-SGI K5-2,  03/29/94, output for increment    0.     job1

        tresca     mises     mean   p r i n c i p a l   v a l u e s          p h y s i c a l   c o m p o n e n t s
      intensity intensity  normal   minimum intermediate maximum      1          2          3          4          5          6
                           intensity


element   10  point   1       integration pt. coordinate=     0.360E+02    0.000E+00   -0.720E+02
section thickness = 0.194E+01

element   11  point   1       integration pt. coordinate=     0.000E+00    0.000E+00   -0.360E+02
section thickness = 0.194E+01

element   12  point   1       integration pt. coordinate=     0.748E+01    0.420E+02   -0.647E+02
section thickness = 0.100E+01
stress  5.135E+02 5.135E+02 1.712E+02 0.000E+00 0.000E+00 5.135E+02 5.135E+02
strain  1.712E-05 1.397E-05 0.000E+00 0.000E+00 0.000E+00 1.712E-05 1.712E-05

element   13  point   1       integration pt. coordinate=     0.435E+02    0.420E+02   -0.647E+02
section thickness = 0.100E+01
stress  2.222E+00 2.222E+00 7.408E-01 0.000E+00 0.000E+00 2.222E+00 2.222E+00
strain  7.408E-08 6.048E-08 0.000E+00 0.000E+00 0.000E+00 7.408E-08 7.408E-08
```

```
element   14  point   1       integration pt. coordinate=     0.287E+02    0.420E+02   -0.748E+01
section thickness = 0.100E+01
stress  2.226E+00 2.226E+00-7.421E-01-2.226E+00 0.000E+00 0.000E+00-2.226E+00
strain  7.421E-08 6.059E-08 0.000E+00-7.421E-08 0.000E+00 0.000E+00-7.421E-08

element   15  point   1       integration pt. coordinate=     0.748E+01    0.420E+02   -0.435E+02
section thickness = 0.100E+01
stress  2.223E+00 2.223E+00 7.410E-01 0.000E+00 0.000E+00 2.223E+00 2.223E+00
strain  7.410E-08 6.051E-08 0.000E+00 0.000E+00 0.000E+00 7.410E-08 7.410E-08

element   16  point   1       integration pt. coordinate=     0.647E+02    0.420E+02   -0.287E+02
section thickness = 0.100E+01
stress  2.225E+00 2.225E+00-7.418E-01-2.225E+00 0.000E+00 0.000E+00-2.225E+00
strain  7.418E-08 6.057E-08 0.000E+00-7.418E-08 0.000E+00 0.000E+00-7.418E-08
1
```

n o d a l   p o i n t   d a t a

i n c r e m e n t a l   d i s p l a c e m e n t s

```
 1  0.00000E+00  0.00000E+00  0.00000E+00        2  0.00000E+00  0.00000E+00  0.00000E+00
 3 -1.33536E-03  1.45154E-03 -1.77362E-03        5  0.00000E+00  0.00000E+00  0.00000E+00
 6  0.00000E+00  0.00000E+00  0.00000E+00        7  1.77086E-03  1.43951E-03 -1.30118E-03
 8 -1.80559E-03  1.45976E-03  1.33256E-03        9  1.29842E-03  1.44773E-03  1.80279E-03
```

t o t a l   d i s p l a c e m e n t s

```
 1  0.00000E+00  0.00000E+00  0.00000E+00        2  0.00000E+00  0.00000E+00  0.00000E+00
 3 -1.33536E-03  1.45154E-03 -1.77362E-03        5  0.00000E+00  0.00000E+00  0.00000E+00
 6  0.00000E+00  0.00000E+00  0.00000E+00        7  1.77086E-03  1.43951E-03 -1.30118E-03
 8 -1.80559E-03  1.45976E-03  1.33256E-03        9  1.29842E-03  1.44773E-03  1.80279E-03
```

displacements

total equivalent nodal forces (distributed plus point loads)

```
 1  0.00000E+00  0.00000E+00  0.00000E+00        2  0.00000E+00  0.00000E+00  0.00000E+00
 3   422.00        500.00      -422.00           5  0.00000E+00  0.00000E+00  0.00000E+00
 6  0.00000E+00  0.00000E+00  0.00000E+00        7   422.00        500.00       422.00
 8  -422.00        500.00      -422.00           9  -422.00        500.00       422.00
```

reaction forces at fixed boundary conditions, residual load correction elsewhere

```
 1  -87.423      -496.36       88.340            2   87.429      -500.00       88.071
 3  0.00000E+00  5.68434E-14  1.70530E-13        5  -88.988      -500.00      -88.346
```

```
                 time =       0.81


                 singularity ratio    1.0993E-01


                 end of matrix solution
                  time =       0.83


                 maximum residual force  at node    9 degree of freedom    1 is equal to    0.568E-13
                 maximum reaction force  at node    6 degree of freedom    2 is equal to    0.504E+03
                 convergence ratio                                                          0.113E-15
1
 MARC-SGI K5-2,  03/29/94, output for increment    1.    job1


                 total transient time = 1.00000E+00

       tresca     mises    mean  p r i n c i p a l   v a l u e s          p h y s i c a l   c o m p o n e n t s
     intensity intensity  normal   minimum intermediate maximum      1        2         3         4         5         6
                          intensity


 element    1  point   1        integration pt. coordinate=     0.748E+01    0.420E+02   -0.748E+01
 section thickness = 0.100E+01
 stress  5.137E+02 5.137E+02 1.712E+02 0.000E+00 0.000E+00 5.137E+02 5.137E+02
 strain  1.712E-05 1.398E-05 0.000E+00 0.000E+00 0.000E+00 1.712E-05 1.712E-05

 element    2  point   1        integration pt. coordinate=     0.360E+02    0.000E+00    0.000E+00
 section thickness = 0.194E+01

 element    3  point   1        integration pt. coordinate=     0.720E+02    0.000E+00   -0.360E+02
 section thickness = 0.194E+01

 element    4  point   1        integration pt. coordinate=     0.647E+02    0.420E+02   -0.748E+01
 section thickness = 0.100E+01
 stress  5.172E+02 5.172E+02 1.724E+02 0.000E+00 0.000E+00 5.172E+02 5.172E+02
 strain  1.724E-05 1.408E-05 0.000E+00 0.000E+00 0.000E+00 1.724E-05 1.724E-05

 element    5  point   1        integration pt. coordinate=     0.361E+02    0.840E+02   -0.150E+02
 section thickness = 0.100E+01
 stress  3.330E+02 3.330E+02-1.110E+02-3.330E+02 0.000E+00 0.000E+00-3.330E+02
 strain  1.110E-05 9.063E-06 0.000E+00-1.110E-05 0.000E+00 0.000E+00-1.110E-05

 element    6  point   1        integration pt. coordinate=     0.150E+02    0.840E+02   -0.361E+02
 section thickness = 0.100E+01
 stress  3.346E+02 3.346E+02-1.115E+02-3.346E+02 0.000E+00 0.000E+00-3.346E+02
 strain  1.115E-05 9.106E-06 0.000E+00-1.115E-05 0.000E+00 0.000E+00-1.115E-05

 element    7  point   1        integration pt. coordinate=     0.361E+02    0.840E+02   -0.573E+02
 section thickness = 0.100E+01
 stress  3.346E+02 3.346E+02-1.115E+02-3.346E+02 0.000E+00 0.000E+00-3.346E+02
 strain  1.115E-05 9.106E-06 0.000E+00-1.115E-05 0.000E+00 0.000E+00-1.115E-05
```

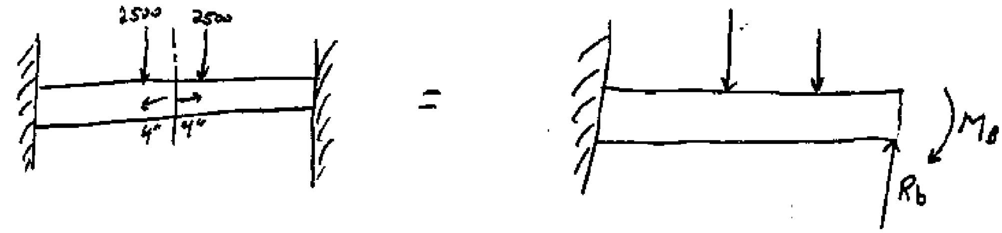
Stress and Strain

```
element    8  point    1        integration pt. coordinate=     0.573E+02    0.840E+02   -0.361E+02
section thickness = 0.100E+01
stress  3.330E+02 3.330E+02-1.110E+02-3.330E+02 0.000E+00 0.000E+00-3.330E+02
strain  1.110E-05 9.063E-06 0.000E+00-1.110E-05 0.000E+00 0.000E+00-1.110E-05

element    9  point    1        integration pt. coordinate=     0.647E+02    0.420E+02   -0.647E+02
section thickness = 0.100E+01
stress  5.169E+02 5.169E+02 1.723E+02 0.000E+00 0.000E+00 5.169E+02 5.169E+02
strain  1.723E-05 1.407E-05 0.000E+00 0.000E+00 0.000E+00 1.723E-05 1.723E-05
1
 MARC-SGI K5-2,  03/29/94, output for increment    1.    job1

         tresca     mises     mean  p r i n c i p a l   v a l u e s        p h y s i c a l   c o m p o n e n t s
       intensity intensity  normal  minimum intermediate maximum     1         2         3         4         5         6
                            intensity


element   10  point    1        integration pt. coordinate=     0.360E+02    0.000E+00   -0.720E+02
section thickness = 0.194E+01

element   11  point    1        integration pt. coordinate=     0.000E+00    0.000E+00   -0.360E+02
section thickness = 0.194E+01

element   12  point    1        integration pt. coordinate=     0.748E+01    0.420E+02   -0.647E+02
section thickness = 0.100E+01
stress  5.135E+02 5.135E+02 1.712E+02 0.000E+00 0.000E+00 5.135E+02 5.135E+02
strain  1.712E-05 1.397E-05 0.000E+00 0.000E+00 0.000E+00 1.712E-05 1.712E-05

element   13  point    1        integration pt. coordinate=     0.435E+02    0.420E+02   -0.647E+02
section thickness = 0.100E+01
stress  2.222E+00 2.222E+00 7.408E-01 0.000E+00 0.000E+00 2.222E+00 2.222E+00
strain  7.408E-08 6.048E-08 0.000E+00 0.000E+00 0.000E+00 7.408E-08 7.408E-08

element   14  point    1        integration pt. coordinate=     0.287E+02    0.420E+02   -0.748E+01
section thickness = 0.100E+01
stress  2.226E+00 2.226E+00-7.421E-01-2.226E+00 0.000E+00 0.000E+00-2.226E+00
strain  7.421E-08 6.059E-08 0.000E+00-7.421E-08 0.000E+00 0.000E+00-7.421E-08

element   15  point    1        integration pt. coordinate=     0.748E+01    0.420E+02   -0.435E+02
section thickness = 0.100E+01
stress  2.223E+00 2.223E+00 7.410E-01 0.000E+00 0.000E+00 2.223E+00 2.223E+00
strain  7.410E-08 6.051E-08 0.000E+00 0.000E+00 0.000E+00 7.410E-08 7.410E-08

element   16  point    1        integration pt. coordinate=     0.647E+02    0.420E+02   -0.287E+02
section thickness = 0.100E+01
stress  2.225E+00 2.225E+00-7.418E-01-2.225E+00 0.000E+00 0.000E+00-2.225E+00
strain  7.418E-08 6.057E-08 0.000E+00-7.418E-08 0.000E+00 0.000E+00-7.418E-08
1
                              n o d a l   p o i n t   d a t a


                              i n c r e m e n t a l   d i s p l a c e m e n t s
```

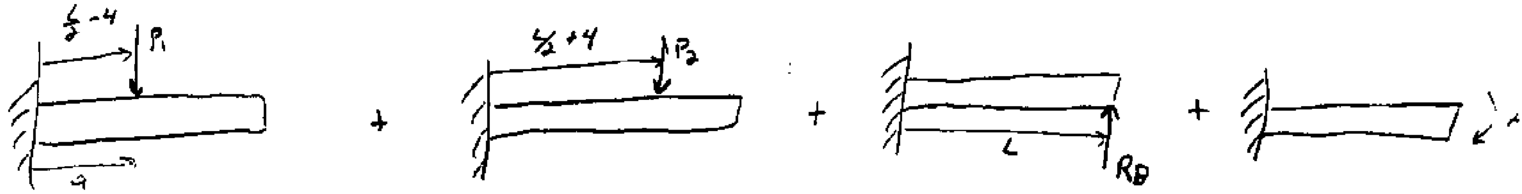
Appendix F

Base Stress Analysis

Analysis of 6' Beams on Base



Use Super position:



pg 531 - Crandall, Dahl, Lardner

An Introduction to the Mechanics of Solids

$$\delta_B^{P_1} = \frac{Pa^2(3L-a)}{6EI}$$

$$\delta_B^{P_1} = \frac{P_1(\frac{L}{2}-4)^2}{6EI}\left(\tfrac{5}{2}L+4\right)$$

$$\delta_B^{P_2} = \frac{P_2(\frac{L}{2}+4)^2}{6EI}\left(\tfrac{5}{2}L-4\right)$$

$$\delta_B^{R_B} = \frac{(R_B)L^3}{3EI}$$

$$\delta_B^{M_B} = \frac{(M_B)L^2}{2EI}$$

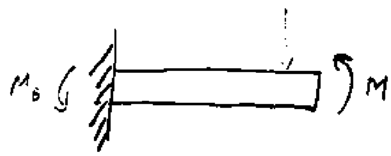
$$\delta_B^{P_1} + \delta_B^{P_2} - \delta_B^{R_B} + \delta_B^{M_B} = 0$$

$$\frac{P_1(\frac{L}{2}-4)^2}{6EI}\left(\tfrac{5}{2}L+4\right) + \frac{P_2(\frac{L}{2}+4)^2}{6EI}\left(\tfrac{5}{2}L-4\right) - \frac{(R_B)L^3}{3EI} = -\frac{M_B L^2}{2EI}$$

$$M_B = \frac{2}{L^2}\left(\frac{R_B L^3}{3} - \frac{P_1(\frac{L}{2}-4)^2}{6}\left(\tfrac{5}{2}L+4\right) - \frac{P_2(\frac{L}{2}+4)^2}{6}\left(\tfrac{5}{2}L-4\right)\right)$$

This value substituted into spreadsheet to calculate moments for various lengths.

Now, calculate maximum moment (at center)



$$M = M_B - P_1 \cdot \left(\tfrac{L}{2} - 4\right)$$

Using moment calculation from above to calculate M_B, plus equation from above to get M, the moment at the center.

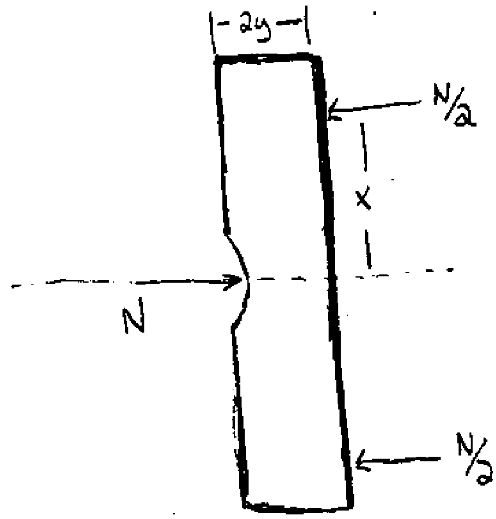
Next, apply $\sigma = \frac{My}{I}$ to get max stress in the beam.

Once again, this was substituted into a spreadsheet to get stresses for different beam lengths and cross sections.

Appendix G

Bolted Plate

Friction Collar



assuming

$\mu = .5$

$mg_{PIPE} = 300\text{ lbs}$

SF=2

$F_F = SF(300) = 600$

$F_F = \mu N = (.5)N = 600$

$N = 1200\text{ lbs}$

$N/2 = 600\text{ lbs}$

$2y = h$

b = Z dist into paper

$$\sigma = \frac{My}{I} = \frac{600(x)(y)}{\left(\frac{bh^3}{12}\right)}$$

$\sigma_{y\,STEEL} = 43\text{ ksi}$

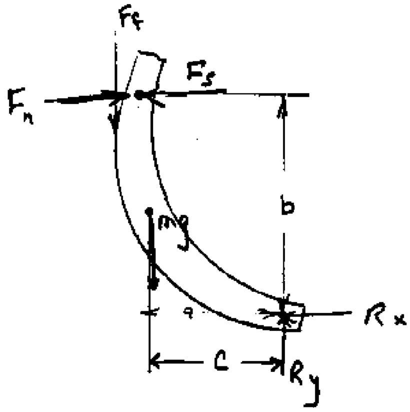
F= 600 lb

SF =5

b=3.0 in	y (in)	h (in)	Stress (psi) x= 2.44 in	with safety factor included x= 2.94 in
	0.15	0.3	162666.7	196000
	0.3	0.6	40666.67	49000
	0.45	0.9	18074.07	21777.78
	0.6	1.2	10166.67	12250
	0.75	1.5	6506.667	7840
	0.9	1.8	4518.519	5444.444
	1.05	2.1	3319.728	4000
	1.2	2.4	2541.667	3062.5
	1.35	2.7	2008.23	2419.753
	1.5	3	1626.667	1960
	1.65	3.3	1344.353	1619.835
	1.8	3.6	1129.63	1361.111
	1.95	3.9	962.5247	1159.763
	2.1	4.2	829.932	1000
	2.25	4.5	722.963	871.1111
	2.4	4.8	635.4167	765.625
	2.55	5.1	562.8604	678.2007
	2.7	5.4	502.0576	604.9383
	2.85	5.7	450.6002	542.9363
	3	6	406.6667	490
	3.15	6.3	368.8587	444.4444
	3.3	6.6	336.0882	404.9587
	3.45	6.9	307.4984	370.5104
	3.6	7.2	282.4074	340.2778
	3.75	7.5	260.2667	313.6
	3.9	7.8	240.6312	289.9408
	4.05	8.1	223.1367	268.8615
	4.2	8.4	207.483	250
	4.35	8.7	193.4205	233.0559
	4.5	9	180.7407	217.7778

Appendix H

Brake Shoes



$F_f \equiv$ Force of Friction
$F_n \equiv$ Normal Force
$F_s \equiv$ Force needed from Spring
$mg \equiv$ weight of brake
$R_x, R_y \equiv$ Reaction Forces at mount
$\mu =$ Coefficient of Friction $\simeq 0.13$

$F_f = \mu F_n$

$F_n = F_f/\mu$

$\Sigma F_y \Rightarrow \quad F_f + mg = R_y$

$\Sigma F_x \Rightarrow \quad F_n = F_s + R_x$

$\Sigma M \Rightarrow \quad F_s \cdot b - F_n \cdot b + F_f \cdot a + mg \cdot c = 0$

$$F_s = \frac{\frac{F_f}{\mu} \cdot b - F_f \cdot a - mg \cdot c}{b}$$

Now, need to find geometry... must find a, b, c to fit constraint $b > \mu a$ to prevent locking up.

also, $F_f \geq \frac{1}{2}$ (weight of pipe) $\simeq 150$ lb

For geometry of brake shoes, $F_s > 900$ lb.
$\Rightarrow$ too big

$$F_s = \frac{\left(\frac{150}{.13}\right) \cdot 3.45 - 150 \cdot 5.25 - (1) \cdot 5.25}{3.45} = \underline{924 \text{ lb}}$$

For approximation purposes, mass of brake = 1 lb and c = a.

Even with these overestimations, spring force is too high.

Appendix I
Instruction Manual

P^3 Instruction Manual

Assembly

P^3 consists of five basic parts: the frame base, the lower truss, the upper truss, the drive assembly and the anchor system. For easy assembly, bring all separated parts to the loading sight. Follow the instructions to prevent time loss and danger to the operator. There is only one order to the assembly.

1. Set up anchors on ship's deck according to their respective positions on the square frame base. The anchors are labeled A through F on each anchor cylinder and on its position on the frame base. The threaded hose fittings should always be on the interior of the frame (see figure 45). Anchors are custom to fit at their respective location on the frame.

2. Lower the square frame base onto the deck over the anchors using a bridal to keep it relatively level and parallel to the ship's deck. Be sure to lower it consistent with the anchors. The anchors can then be bolted to the frame base. Use the 24 sets of stainless steel bolts and washers(fine threads).

3. The third part to attach is the drive assembly. This is the part that houses and supports the piston drive. Attach the winch to the assembly being careful not to damage the brake system on the frame. The frame base position to which it is placed is numbered 1 to 4. Be sure to use this system to line up the drive assembly to ensure that the holes will line up for it to be bolted together. Use 8 stainless steel nuts and bolts to attach this section. Remember to bolt the underside of the square stock to the frame using the 5 inch carriage bolts. (see figure 46).

4. Next, place the <u>**Top**</u> frame section over the drive assembly using the winch. It will latter be pushed up in place once the lower frame section is

Base Anchor Assembly

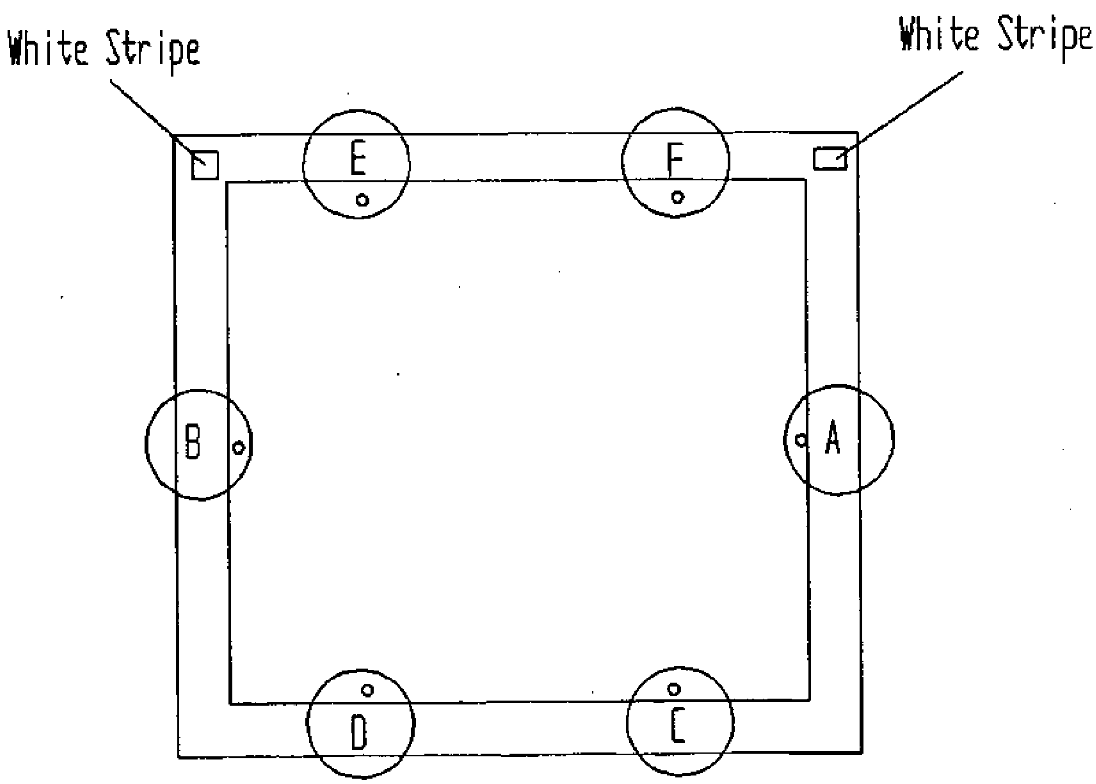


Figure 45

DRIVE ASSEMBLY

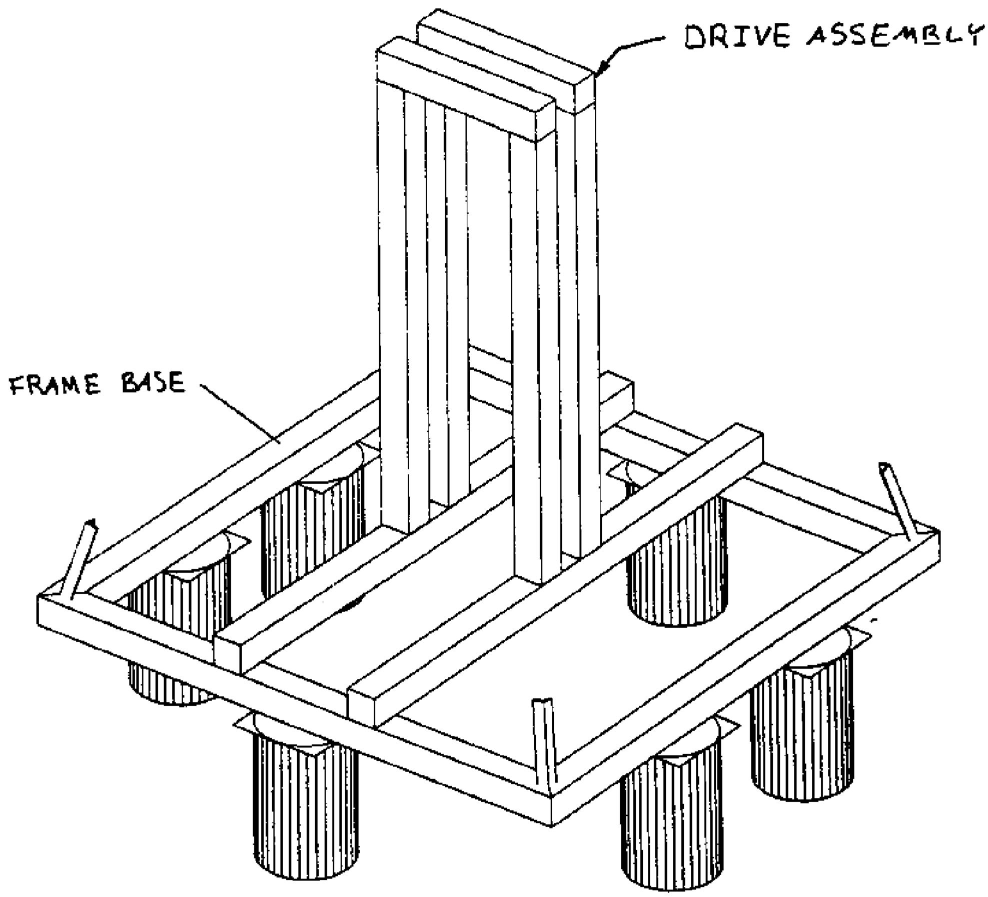


FIGURE 46

bolted and secured. Do not set up the guide sheath yet. It will be in the way when you try to place it over the drive assembly. It should be neatly tied to one corner of the top frame but not obstructing any of the connecting holes.

5. Next, slide the **bottom** frame section over the top frame section and attach to the frame base. It should be lined up with the stripes of white paint located on the bottom of the lower frame section and on the ears of the frame base. This is important to ensure that ALL holes line up and all bolts will fit through the holes (see figure 47-48).

6. The top part of the frame can now be raised into position and bolted. Be sure to locate and line up the white stripes of paint as before in step 5. The frame is now stable.

7. The PVC guide sheath on the top frame section can now be positioned over the shaft break. Attach the bungee cords to there respective holes on the top frame section.

8. Attach all hoses (hydraulic and water) to the system at this time. The hydraulics will be needed for the assembly of the pipe. Be sure to keep them from being tangled. Attached the hydraulic pump and controls to the system and make sure they are in working order. The water pump should be tested as well.

9. Use the hydraulic clamp and piston drive to feed the pipe up through the PVC guide. Assemble as many pieces as it takes to see the pipe sticking out the top of the frame. Make sure the male side of the pipe is sticking downward or else the penetrometer can not be attached.

10. Next, push the measurement cable used for the penetrometer up through the pipe that has been assembled. Push the entire cable through so that the end can be connected to the penetrometer and attached to the threaded

TRUSS ASSEMBLY 1

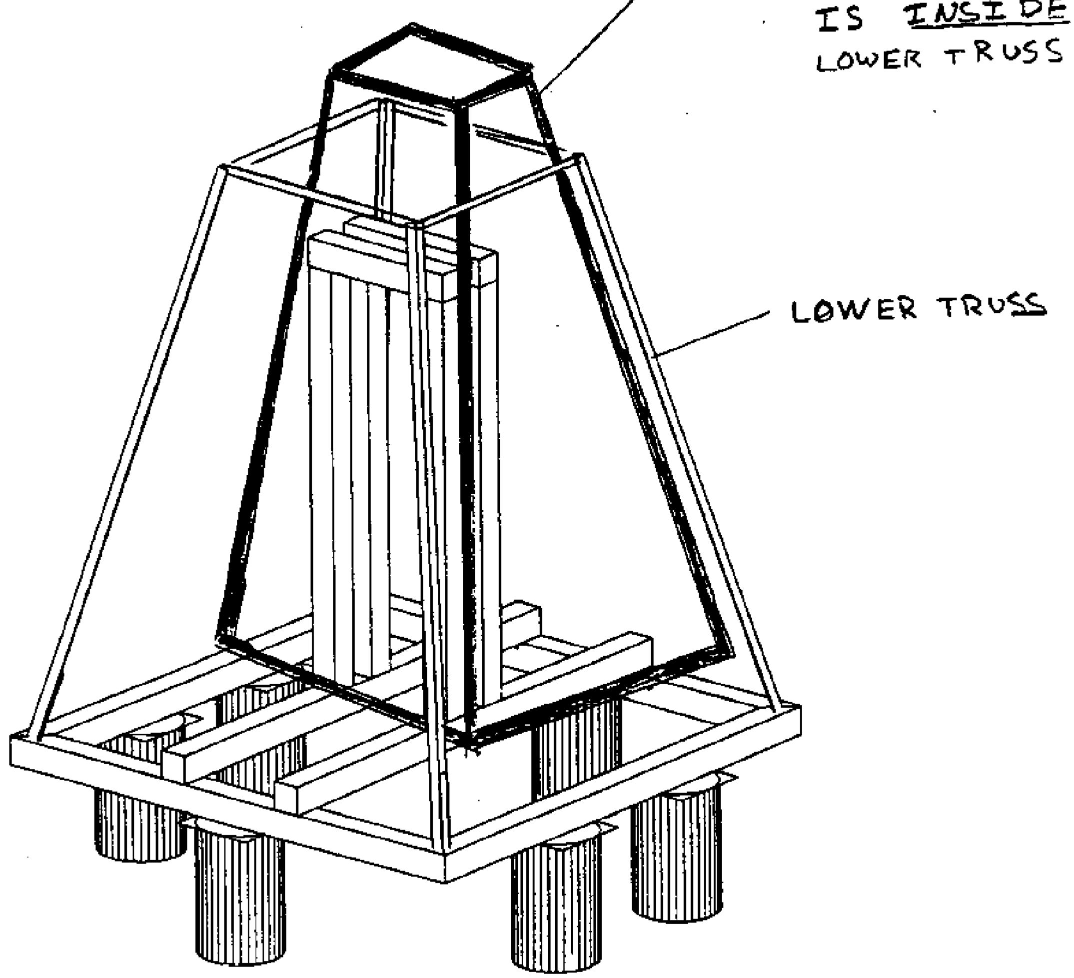


FIGURE 47

TRUSS ASSEMBLY 2

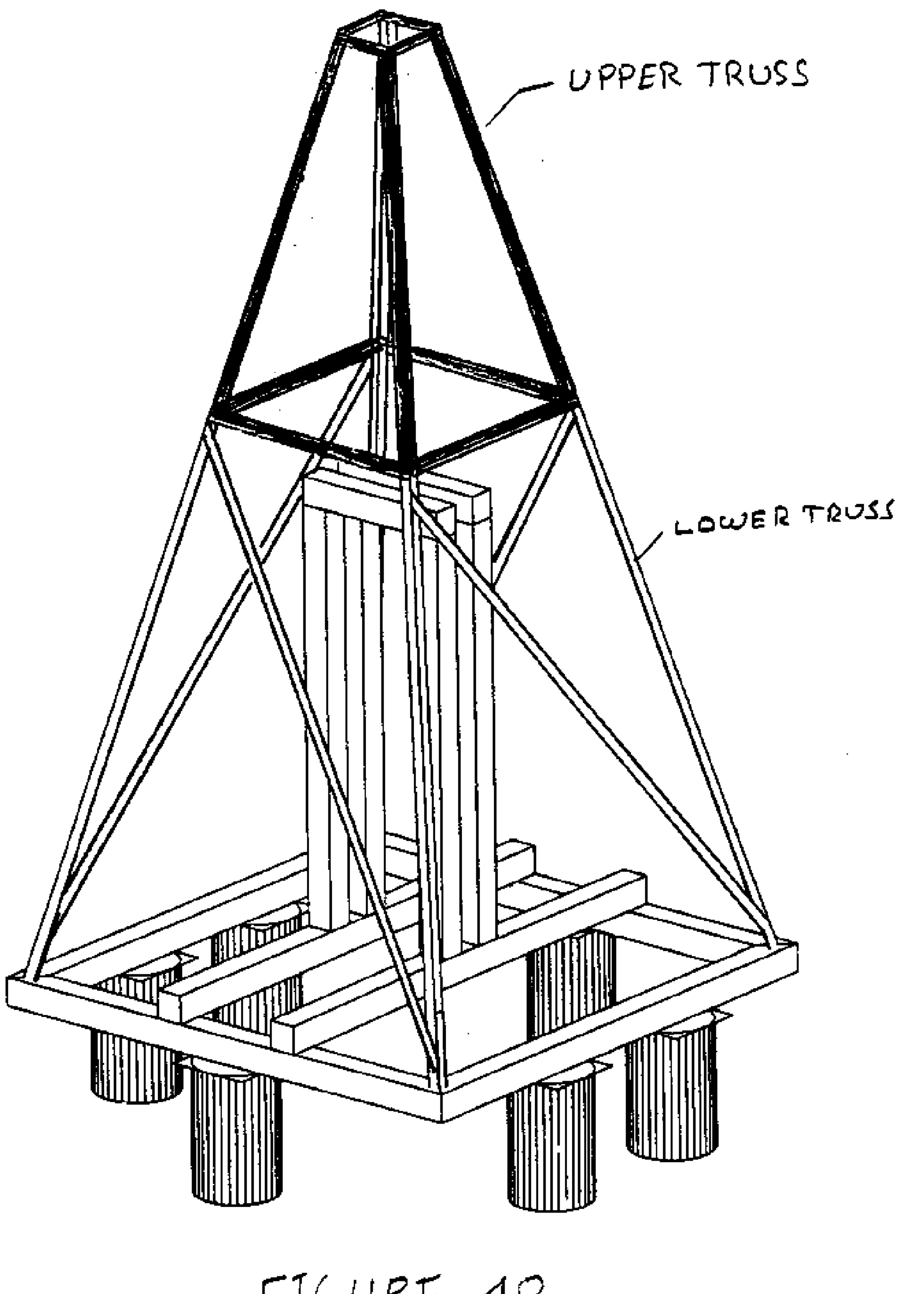


FIGURE 48

pipe near the drive pistons. The other side of the measurement cable can be connected to the remaining sections of pipe and its computer interface as normal. The system is now ready to be deployed.

Deployment and soil Testing

P^3 is designed to be used in silty clay bottoms. The anchors will not hold where the ground sandy since suction anchors are dependent on the cohesiveness of the soil. Test sights should be chosen with this in mind. Damage to the devise could result from its misuse. This devise is specifically designed for deployment on the Gulf Challenger. Other uses may require modifications to assembly and launch systems.

To launch the device, two bridle are needed. One is used to connect the device to the winch which provides the necessary force to lift P^3 off of the deck. The lift bridal will be attached to the drive assembly. The second bridal will be used to prevent the device from swinging inward as it is lifted off of the deck. A rope around the bow side of P^3 and attached to the transom will provide this constraint. This should be used to launch and recover the research device.

After P^3 is overboard, the remaining pipe pieces can be attached as it is lowered into the water. The dive platform on the Gulf Challenger will be used to get close to the top of the frame. This must be done with caution since many cables act as umbilical to the device and will be close to the operator who attaches the remaining sections of pipe. The device may then be lowered to a depth according to the lengths of the umbilical. Be sure to leave enough slack in order to let the boat drift depending on the currents.

The next step is to attach the system to the ocean bottom. Simply activate the pump and evacuate the anchors. This may take up to five minutes. The pump will have to be primed if the hoses are dry. Attachment will be complete when the appearance of "dirty" water is being pumped out. This means that most of the water in the anchors has been removed. For longer test turn on and shut off the pump every few minutes to prevent the pump from burning out and maintain a solid anchoring force.

The shaft is now ready to be driven. 22 piston strokes are needed to drive the 30 foot section of pipe into its maximum depth. It is important not to exceed this amount to prevent the loss of the penetrometer.. Extra pipe pieces should be used as a safety factor to prevent loss. At each drive stroke the clamping pistons should be activated for penetration. The clamping pistons should be released to set up the drive pistons for the next push. Measurements of the penetrometer should be taken using normal methods of data collection. The drive pipe can be removed using a similar technique of alternating strokes until the pipe is removed from the ocean bottom. Several tests can be performed per day.