NHU-T-93-002 C3

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S.P.O.S.S.

Self Propelled Oil Surface Skimmer

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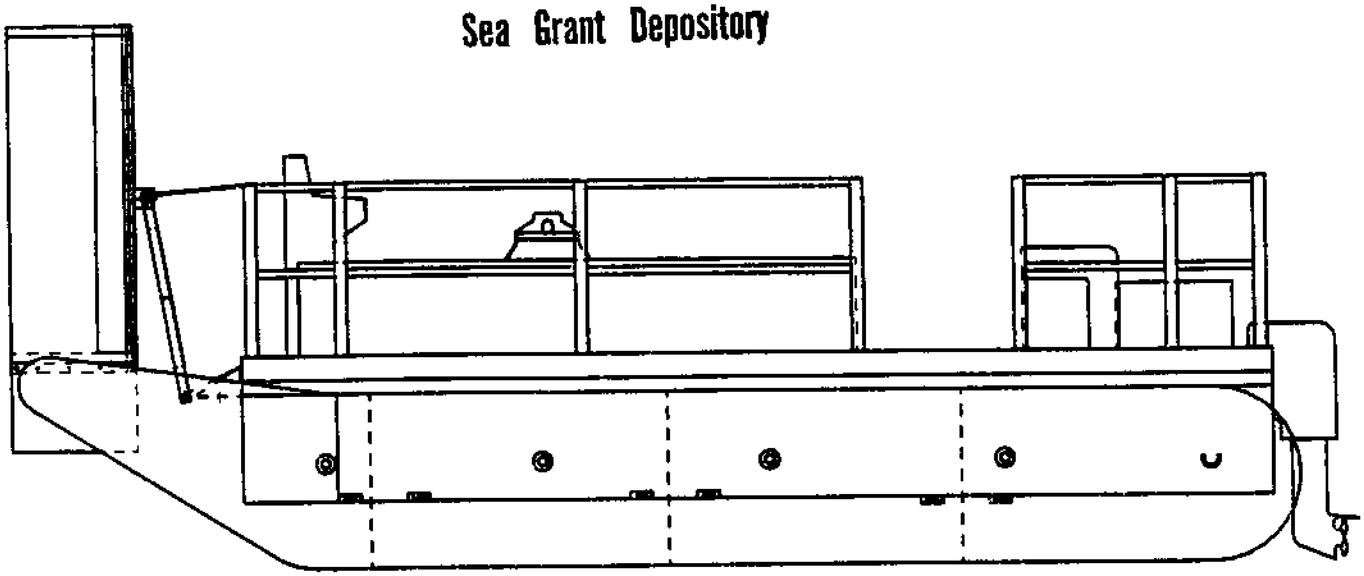

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UNHMP-AR-SG-93-8
#3

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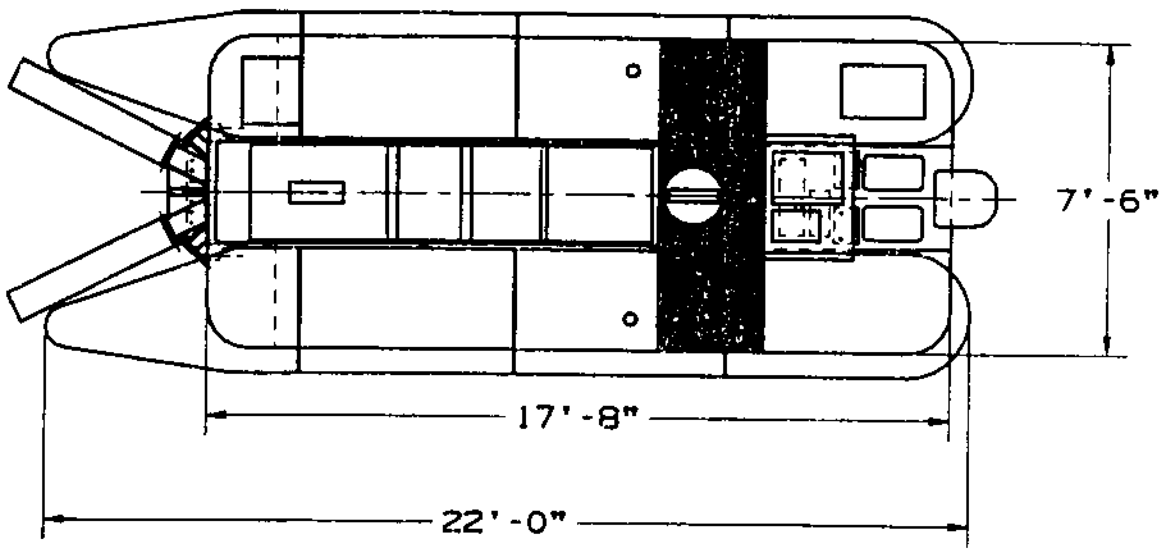
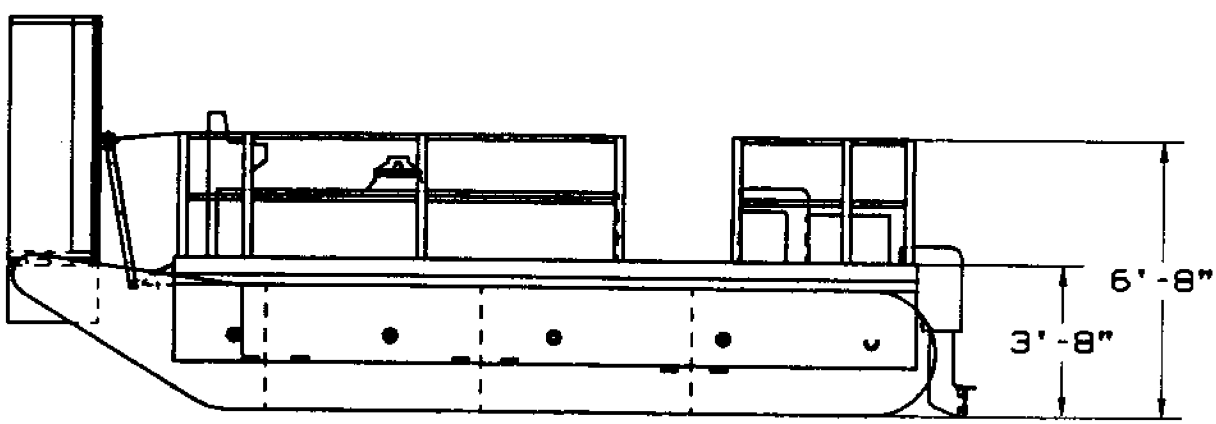
INTRODUCTION:

Oil spills have become a very important environmental concern over the past decade. With an economy based on oil and growing public awareness about the consequences of spills, politicians have taken notice. As a result, it has been mandated that by early 1993, oil containment facilities must submit oil spill contingency plans to the Coast guard. A number of methods for cleaning oil spills have been suggested from controlled burning of spills[1] to laser ignition of arctic spills[2]. In our opinion, this just takes the problem out of the sea and puts it into the air. The size of spills varies immensely from the Exxon Valdez, to the improperly connected transfer plumbing at a containment facility which is quickly noticed. It is precisely this type of smaller oil spill with which this project has concentrated on.

The Self Propelled Oil Surface Skimmer (SPOSS) is designed to respond quickly to all types of oil spills in harbors and small bays where oil terminals and ports are located. The vessel is self-propelled and is equipped with a separate diesel engine which operates all hydraulic components. In order to respond rapidly to oil spills a trailer must be designed for transporting SPOSS to various deployment locations. (See figure I1 and I2) scale = .018



Top View



Side View

Figure I1

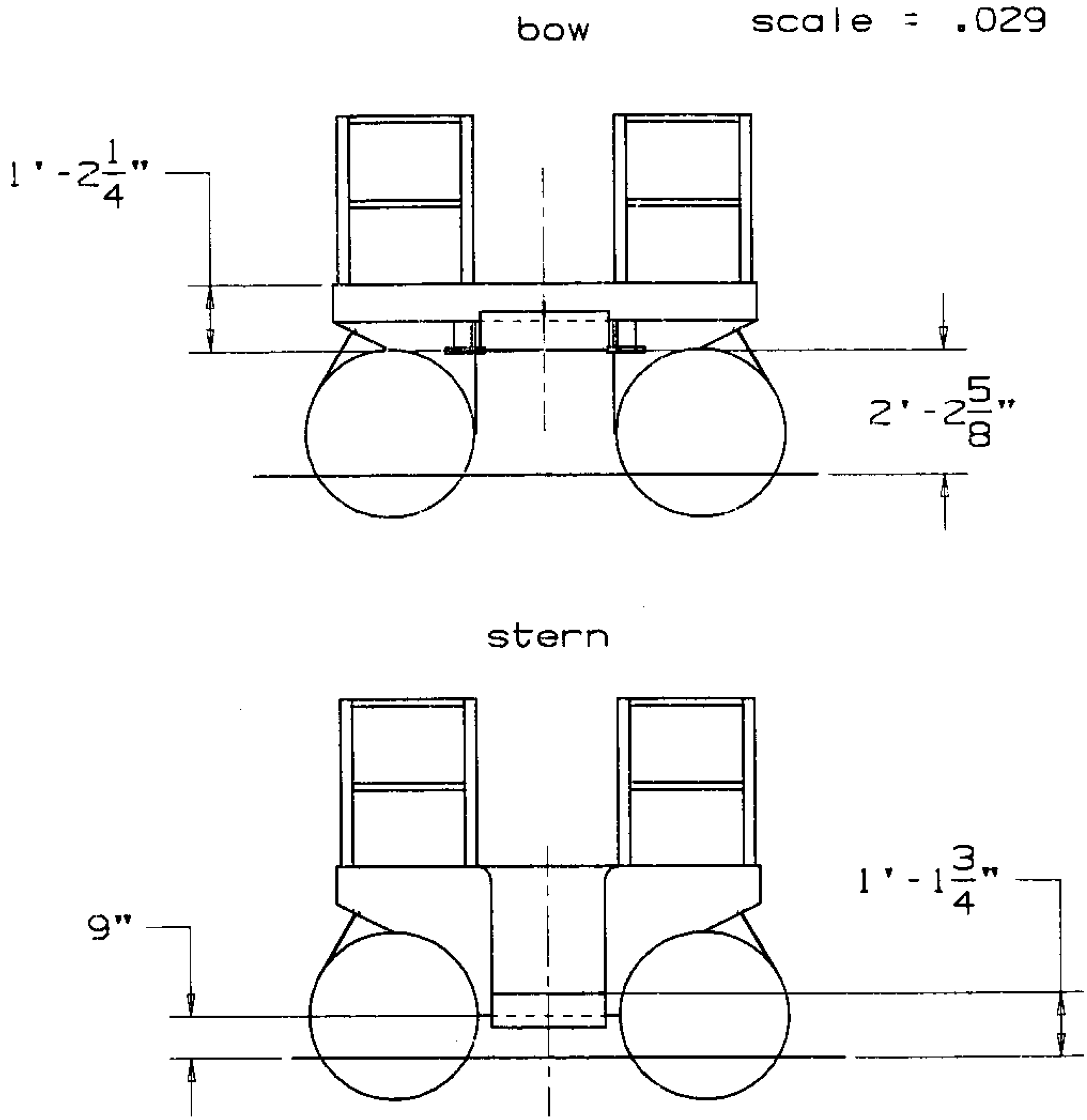


Figure I2

Self Propelled Oil Surface Skimmer Specifications

OVERALL DIMENSIONS & WEIGHT

Length:	22 ft. 3 in.
Beam:	8 ft. 6 in.
Draft-Light:	0 in.
Draft-Loaded:	1 ft. 2 in.
Overall Height: (mast lowered)	8 ft. 9 in.
Displacement Light:	3080 lb.
Displacement Loaded:	4770 lb.

PROPULSION

Type: Diesel outboard
Model: Johnson -857905
Power: 85 HP

CONTROL

Steering: Teleflex cable
Throttle & Gearbox: single lever control of propulsion outboard

HYDRAULIC SYSTEM

Pump Type : Hydura double pump
Model: PVWH -06; PVWH-10
Output: 15GPM @ 1000 PSI
Reservoir: 20 gallons

Motor Type: Char-Lynn motor
Model:104-1001
Output:3/4 Hp

ELECTRICAL SYSTEM

Type:	2 - wire negative ground
Voltage:	12 VDC
Alternator:	50 Amp max.

DIESEL ENGINE

Type:	Horizontal, 4 cycle water-cooled diesel
Model:	ZB600-1-B
Power:	13.8 HP @ 3200 RPM
Displacement:	34.78 in^3

OIL COLLECTION SYSTEM

Type:	Dynamic Inclined Plane (DIP)
Model:	DIP 400 module
Collection Belt:	PVC reinforced with polyester
Belt Width:	15 in.
Maximum Sweep Aperature:	7 ft.
Transfer Pump:	Positive Displ. Moyno 1L-CDQ
Capacity:	33 GPM
On Board Oil Storage:	1 Tank 150 gallons

PERFORMANCE CHARACTERISTICS

Transit Speed:	20 Knots
Skimming Speed:	0 - 3 knots
Oil Recovery Efficiency:	> 90 %
Effective Oil Collection Speed:	0 - 3 knots

DECK LAYOUT:

The vessel's deck is constructed of 3/16 & 1/8 in. marine grade aluminum plate resulting in both a durable and lightweight platform. It contains a 150 gallon oil storage container amidship with plumbing access built in. The tank is vented with check valves and flame proof screens in compliance with Code Federal Regulations[3]. The hull is designed with attachment stanchions built in to accommodate two 22 ft. pontoons with a diameter of 3 ft. A control station is located on the starboard bow with control cable conduit running from bow to stern providing the operator with a single location for navigational operations. This location allows the pilot view of incoming debris during skimming operations. The DIP module is located in the bow of the vessel and requires two persons to raise and lower. Actual control of the belt speed will be accomplished by the manual hydraulic valves.

The vessel's deck is enclosed by 24 in. bulwark for safety purposes and is useful for cable, piping, and tube runs. Lifting pads are provided for the six cleats to add strength for mooring and anchoring. The platform also provides ample room to accommodate the hydraulic components. These components consist of a Char-Lynn motor (located on the DIP module) to drive the inclined plane, a Kubota diesel engine mounted with a Hydura pump, a Moyno pump mounted with a Parker motor, and a hydraulic fluid reservoir all located in the stern of the vessel. Advantages and rationale for decisions on locations of components are that pilots have access to all components and also to influence in the uneven weight distribution created from the DIP module. The vessel's propulsion motor, located in the stern, also has a positive effect on the weight distribution. (refer to figure D1 and table D1)

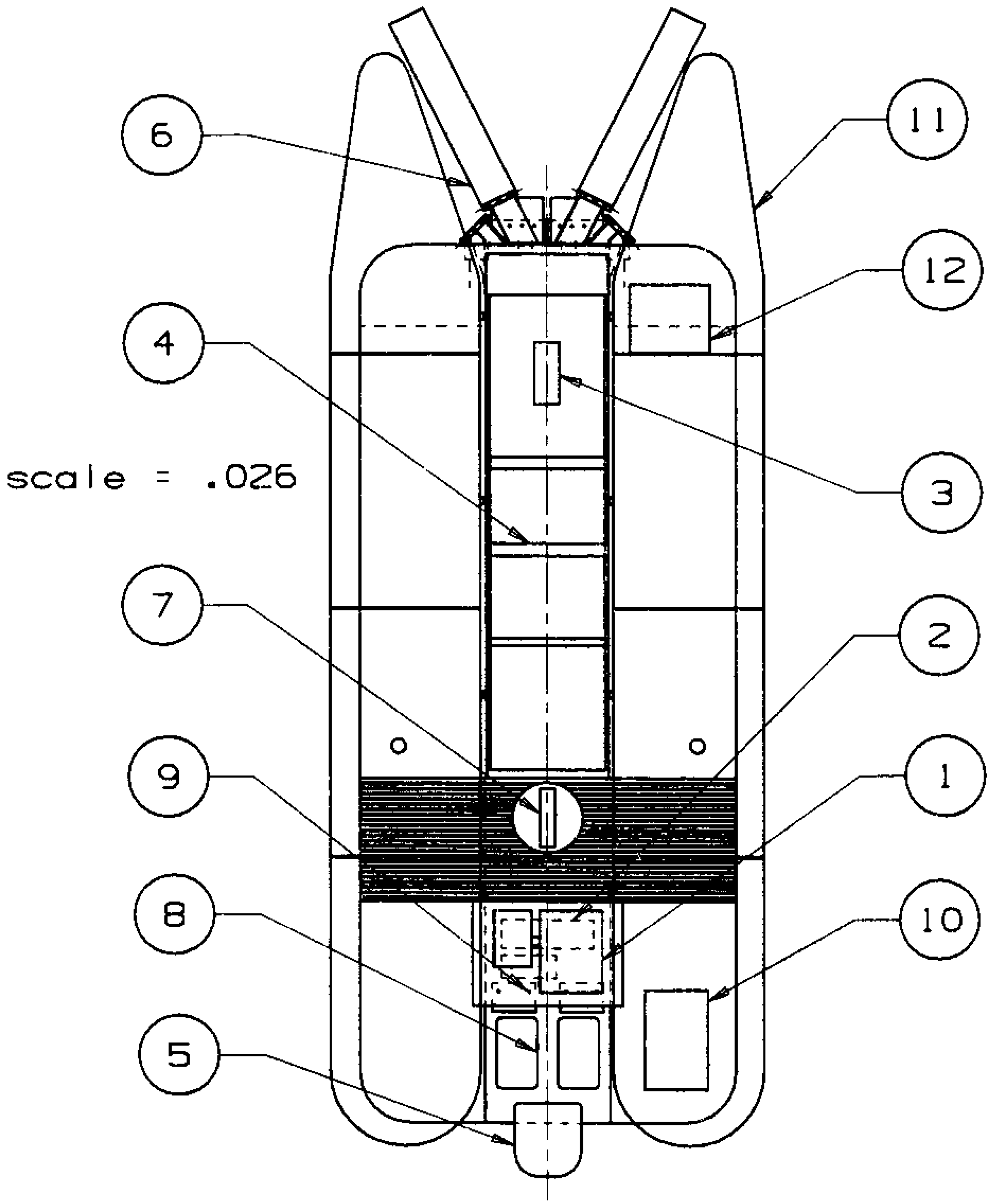


Figure D1

TABLE D1: LIST OF COMPONENTS AND WEIGHTS FOR DECK LAYOUT.

LOCATION	COMPONENT	WEIGHT (LBS.)
	HULL	1000
1	KUBOTA & HYDURA	220
2	MOYNO & PARKER	150
3	CHAR-LYNN	30
4	DIP MODULE	1500
5	JOHNSON OUTBOARD	270
6	BOOMS	120
7	FULL OIL TANK	1150
8	FUEL TANKS	100
9	BATTERIES	30
10	HYD. FLUID RESERVOIR	200
11	PONTOONS	----
12	CONTROL PANEL	----

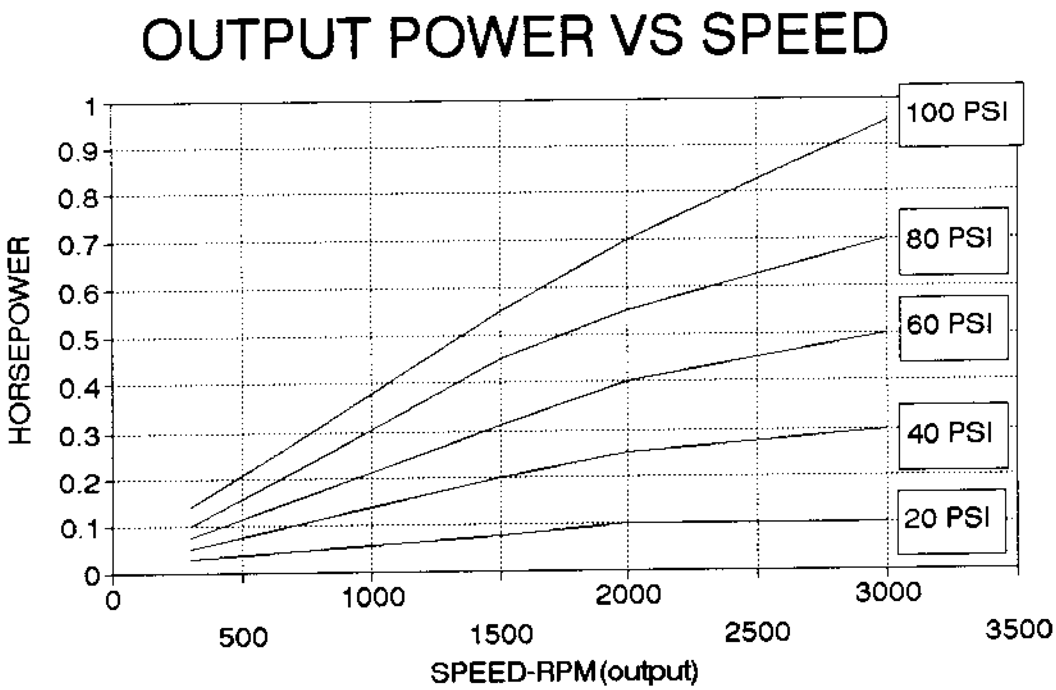
TOTAL **3770 LBS.**

HYDRAULIC SYSTEM (HYD)

Introduction

A system was needed to transmit power from a common source to the DIP module and the oil transfer system. This system had to be safe, efficient, compact, and expandable. Three options were researched to meet this criteria.: electrical, pneumatic, and hydraulic. The first system researched was electrical. This is because the DIP module came equipped with an electrical motor that ran the belt. It was also thought that an electrical generator to provide power would be relatively inexpensive. With the addition of an electric motor to power the Moyno pump, the system would be complete at a low cost. However, this was not the case. The generator was the first problem encountered. Oil spill recovery is a hazardous environment and electrical sparks are absolutely prohibited. This is an inherent problem with any electrical system. The generator would have to be a Coast Guard approved marine grade generator. These generators are priced from $3500 up. All connections and cables had to be hazardous duty grade. All components would be subjected to a corrosive marine environment and don't stand up as well as other systems. For these reasons the electrical system was rejected.

The next option considered was a pneumatic system. A diesel engine would drive an air compressor which would power two air motors. Again, the air motors would drive the DIP and the Moyno pump. The first consideration was the amount of air that would be needed to drive the respective systems. It was decided that to obtain the 3/4 hp needed to replace the electric motor on the DIP, 25 cfm of air flow would be needed (see figure HYD1). Two options were available to get this needed air. Each required a diesel engine with a minimum of 16 hp. One compressor was equipped with an air receiver to which pneumatic lines would be directly connected. This device weighed 700 lbs. The other compressor used a 40 gallon air tank instead


OUTPUT POWER VS SPEED
HORSEPOWER
SPEED-RPM (output)
100 PSI
80 PSI
60 PSI
40 PSI
20 PSI
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
0
500
1000
1500
2000
2500
3000
3500

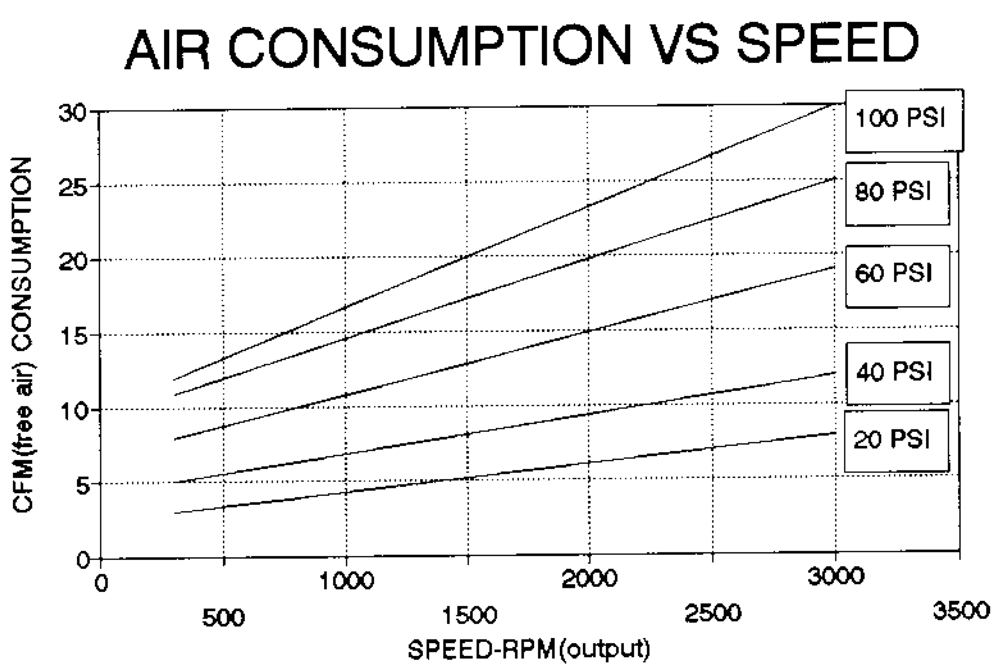

AIR CONSUMPTION VS SPEED
CFM(free air) CONSUMPTION
SPEED-RPM (output)
100 PSI
80 PSI
60 PSI
40 PSI
20 PSI
0
5
10
15
20
25
30
0
500
1000
1500
2000
2500
3000
3500

Figure HYD1

of the air receiver. This compressor weighed 1000 lbs. Because of the extreme weight and the size of these compressors it was decided that this was impractical for a small vessel.

The last system considered was a hydraulic system. This system was able to provide the needed power in a compact form at a competitive price. The system is driven by a compact diesel engine which powers a double hydraulic pump. The double pump setup is actually two independent hydraulic pumps. One pump provides power to a low speed, high torque hydraulic motor which rotates the DIP. The other pump powers a high speed hydraulic motor which operates the Moyno pump. See figure HYD2 for complete hydraulic schematic.

Kubota Diesel Engine

The engine selected was a Kubota Horizontal Diesel Engine ZB600C-1-B (see figure HYD3). This is a compact, lightweight engine. A diesel engine was selected because of the lower flashpoint of diesel fuel as opposed to that of gasoline (100°F for diesel). It would also provide a more economical and durable power plant.

Kubota-Hydura System Layout

The Kubota is a 2-cylinder, 4-cycle water cooled engine that provides 11 continuous hp at 3200 rpm to a horizonal shaft (see HYD, Engine Specifications).[4] Attached to this shaft is a 4.1 DIA pulley that drives a timing belt. This timing belt drives a larger 7.6 DIA pulley which drives the Hydura hydraulic pump. The approximate 2 to 1 reduction is needed so as not to exceed the maximum rpm rating of the hydraulic pump. This belt is kept in tension by an adjustable flat faced idler pulley. Both the Kubota engine and the Hydura Hydraulic pump are mounted on a 22 x 30 inch aluminum plate to avoid unnecessary movement between the two

Hydraulic System for the SPOSS

Max Pres. 1000 psi

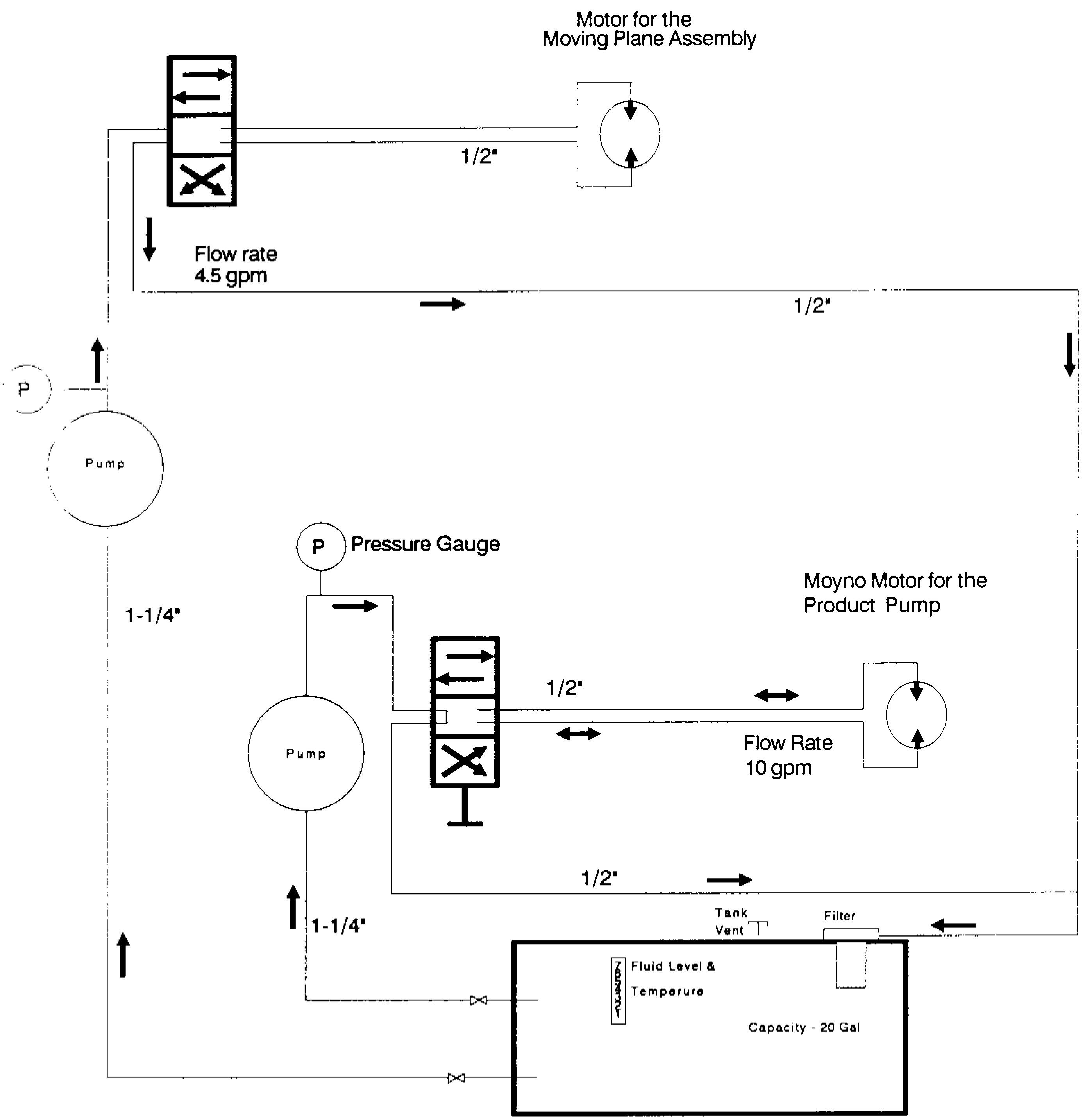
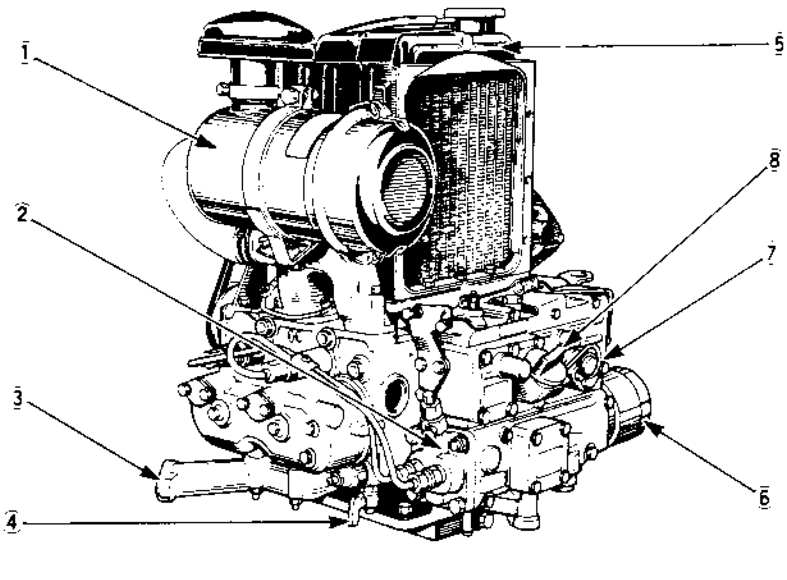
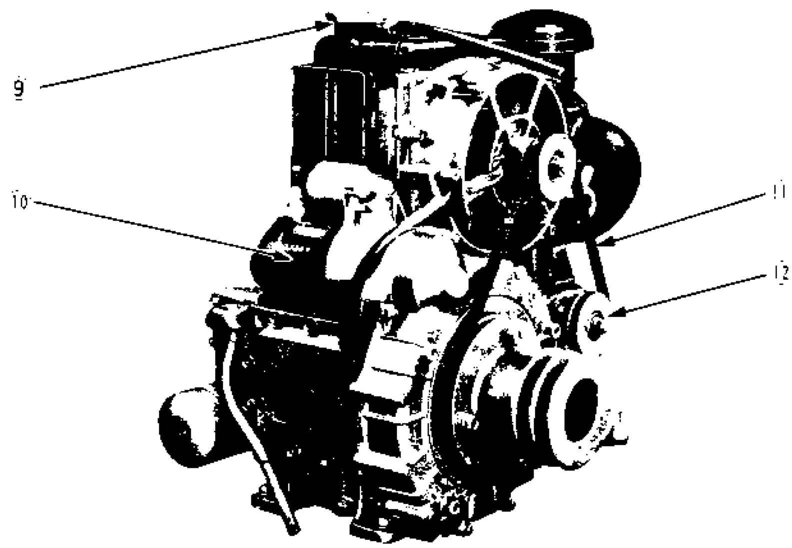


Figure HYD2

Phil A:\NEWHYDRA.GEM

1 .NAMES OF PARTS
1 .NOMS DES PIECES
1 .BEZEICHNUNGEN DER ERSATZTEILE





1	*Air cleaner*	1	*Filtre à air*	1	*Luftfilter*
2	*Injection pump*	2	*Pompe d'injection*	2	*Einspritzpumpe*
3	*Exhaust manifold*	3	*Soupape d'échappement*	3	*Abgassammler*
4	*Water drain cock*	4	*Robinet de vidange d'eau*	4	*Wasserablaufhahn*
5	*Radiator*	5	*Radiateur*	5	*Kühler*
6	*Oil filter*	6	*Filtre d'huile*	6	*Oelfilter*
7	*Oil gauge*	7	*Gauge d'huile*	7	*Oelstandanzeiger*
8	*Oil port plug*	8	*Valve de l'orifice d'huile*	8	*Deckel der Öleinfüllöffnung*
9	*Radiator pressure cap*	9	*Bouchon de pression du radiateur*	9	*Kühlerdruckkappe*
10	*Starter*	10	*Starter*	10	*Anlasser*
11	*Fan belt*	11	*Courroie de ventilateur*	11	*Ventilatorriemen*
12	*Tension pulley*	12	*Poulie de tension*	12	*Spannrolle*

Figure HYD3

HYDRAULICS SPECIFICATIONS

Kubota Horizonal Diesel Engine

Model	ZB600C-1-B
Type	Horizontal, 4-cycle water-cooled diesel
Number of Cylinders	2
Bore X Stroke [in. (mm)]	2.83 x 2.76 (72 x 70)
Total Displacement [$in.^3$ (cc)]	34.78 (570)
SAE Gross Intermit. HP (HP/rpm)	13.8/3200
SAE Net Intermit. HP (HP/rpm)	12.5/3200
SAE Net Cont. HP (HP/rpm)	11.0/3200
Max. Operating Speed (rpm)	3200
Min. No Load Idling Speed (rpm)	800
Governing	Centrifugal ball mechanical governor
Direction of Rotation	Counter-clockwise from flywheel side
Starter (V x kW)	12 x 0.8
Dynamo	12 x 50
Fuel	Diesel fuel No. 2-D (ASTM D975)
Battery	12V,45 AH, or 350CCA(@ 0°F,SAE) equivalent
Dry Weight [lbs. (kg)]	165.6 (75.1)

Table HYD1

components. This will result in reduced need to adjust the belt tension. A protective covering over the timing belt protects the operator from possible injuries while the belt is rotating. The Kubota is a low vibration engine, but to further reduce vibration transmission either to or from the hull, rubber vibration isolators are located at each of the four mounting points on the plate. A fuel connection point is mounted on the plate. A fuel cock is mounted at this point with the fuel line leading from there to the fuel tank. The engine is equipped with an electric fuel pump which will provide fuel from a six gallon fuel tank mounted on the deck. This should power the engine for approximately 6 hours. The assembly is covered with an aluminum cowling to protect it from the marine environment. A terminal block is mounted on the plate for the connection of a control panel.[5]

Electrical System

The control panel consists of a key, coolant temperature gauge, oil pressure gauge and a throttle. The key prevents unauthorized use and a means to pre-heat the two glow plugs prior to start-up. The temperature and pressure gauges allow for the monitoring of the engines' performance during operation. The engine requires a 12 volt starting battery (see HYD, Engine Specifications) and is equipped with an internal dynamo to charge the battery during operation. The engine is also equipped with an auxiliary 50 amp alternator to provide extra electrical power.

Maintenance

Only maintenance procedures that are necessary prior to operation will be discussed here, for complete information regarding operation and maintenance refer to Kubota Diesel Engines Operators Manual.[4] A fuel water separator is supplied with the engine to ensure clean fuel for operation. This separator is transparent so the quality and availability of fuel can be easily checked. The engine oil can be checked with a oil dip stick and oil must be changed after every 100 hours of operation. The location of the oil drain plug at the bottom of the engine mandates the position of the engine on the vessel. The system had to be mounted to allow for easy access to this plug (see figure D1).

Hydura Double Hydraulic Pump

Two Hydura axial piston variable delivery pumps are integrally coupled together and driven off a single shaft. The delivery rates of each pump can be controlled individually so each hydraulic motor can be operated separately. This has the advantage of allowing the hydraulic motor that powers the DIP module to continue running while the motor that drives the Moyno pump transfers oil from the DIP module to on-board containment. With this ability, oil-skimming operations can continue while the DIP containment is emptied. Oil skimming operations need only be stopped when both the DIP containment and the on-board storage are full.

The two pumps selected were the Hydura PVWH10 to power the Parker motor and the PVWH06 to power the Char-Lynn motor. Both pumps operate at a maximum speed of 1800 rpm and a minimum speed of 600 rpm (see HYD, Hydura Pump Specs). By varying the angle on the swashplate, the displacement of the pump can be controlled and therefore, the flow rate delivered to the respective motors (see figure HYD5). This allows both the DIP and the Moyno pump to be operated at various speeds. Continuous power input is rated at approximately 13 HP for the specifications given, where the power input actually delivered is 11 HP. This gives a delivery rate of approximately 5.5 gpm at a pressure of 2500 psi for the PVWH06 and 9.5 gpm at 1500 rpm for the PVWH10 (see figure HYD6).

HYDRAULIC SPECIFICATIONS

PVWH Hydura Pumps

	Maximum Disp.	Cont. Press.	Max. Press.	Flow Rate	Max. Speed	Power Input
	in^3/rev	psi	psi	gpm	rpm	hp
06	0.86	3000	3500	5.5	1800	12.9
10	1.35	2000	2500	9.0	1800	13.3

Table HYD2

Motors

Motor	Disp. (in^3/rev)	Max. Flow Rate (gpm)	Max. Speed (rpm)	Max. Pressure (psi)
Char-Lynn	9.6	20	477	3000
Parker	1.27	10	4000	1000

Table HYD3

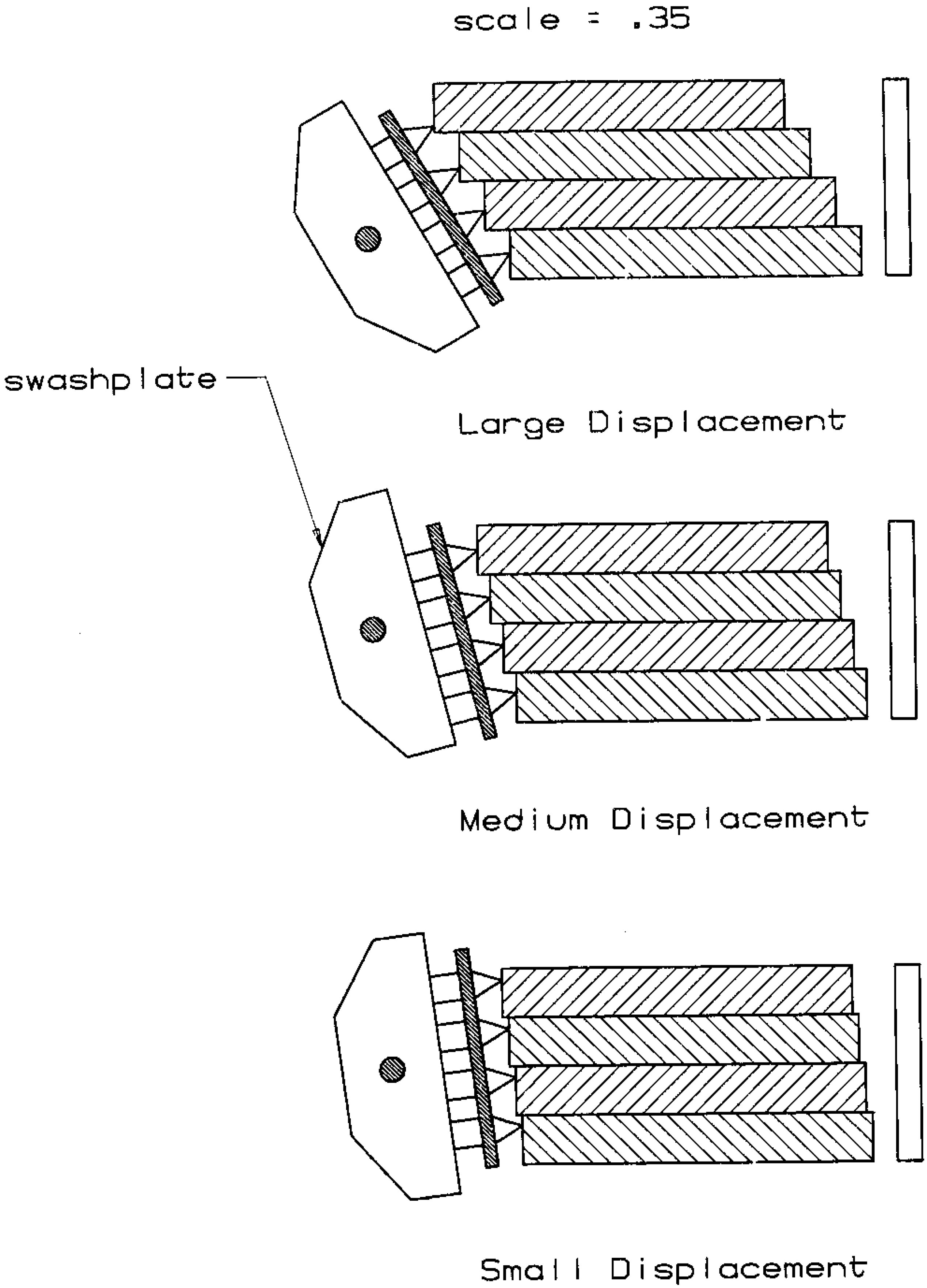
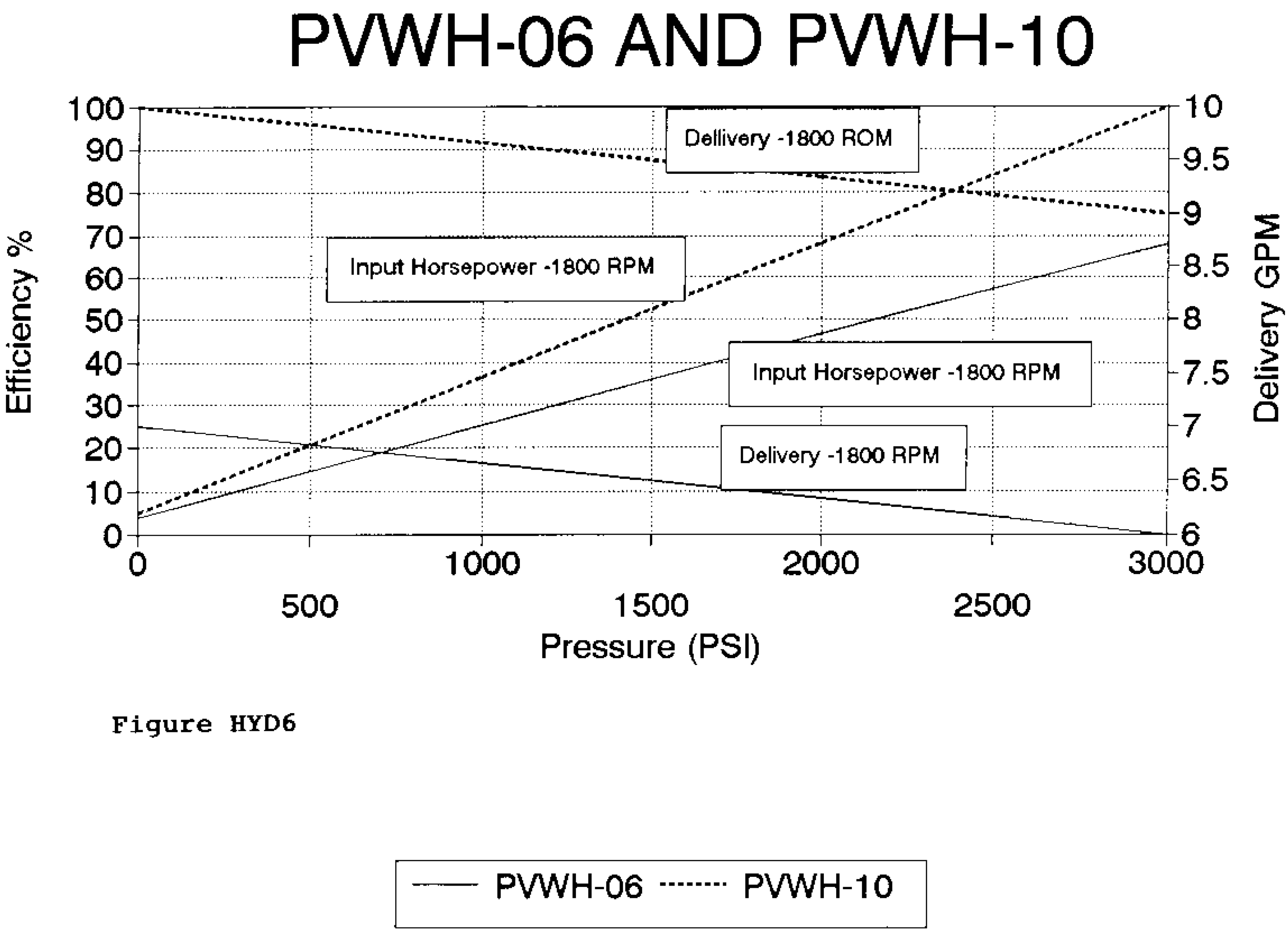

scale = .35
swashplate
Large Displacement
Medium Displacement
Small Displacement

Figure HYD5



PVWH-06 AND PVWH-10
100
90
80
70
60
50
40
30
20
10
0
Efficiency %
10
9.5
9
8.5
8
7.5
7
6.5
6
Delivery GPM
Dellivery -1800 ROM
Input Horsepower -1800 RPM
Input Horsepower -1800 RPM
Delivery -1800 RPM
0
1000
2000
3000
500
1500
2500
Pressure (PSI)
PVWH-06
PVWH-10

Figure HYD6

Hydraulic Plumbing

The low pressure side of the plumbing will begin at the reservoir and fluid will be pumped through 1-1/4 in. flexible tubing from the reservoir to the hydraulic pumps. Each pump will have a separate supply line with an intake pressure of slightly less than 14.7 psia (see HYD, Hydura Specifications).

High pressure plumbing leaving the pumps will be controlled by two series 5000 Husco valves. These valves will provide for both forward and reverse rotation of the motors. This allows the DIP to be operated for oil collection and also for reverse rotation in the event of debris fouling the mechanism. Similarity, the Moyno pump can be fully reversible. The valves are also contain a pre-set pressure relief valve set at 1000 psi. The high pressure lines transmit fluid to the respective motors. Power is transferred to either the DIP or the Moyno pump and the fluid is returned to the reservoir.

Heat is generated in the fluid due to resistance at the hydraulic motors. The flowing liquid also generates heat within the piping and especially at elbows. Depending on the pipe size, one 90° elbow could generate as much heat as several feet of pipe.[6] Any heat generated in the system results in power loss. Fluid temperature control will be discussed in the Reservoir Section. Flexible pipe will be used to deliver fluid as linearly as possible. Areas that are subject to wear will be protected. All pipe will be secured so that in the event of rupture, loose piping will not be a danger. At the connection to the DIP module provisions were made to allow for the unhindered raising and lowering of the DIP.

Hydraulic Motors

Hydraulic motors transform hydraulic working energy into useful mechanical energy. Hydraulic motors used in conjunction with various pumps is termed a hydrostatic drive which can be either closed loop or open loop. Our system is an open loop system. This system has the motor inlet connected to the pump outlet and the motor outlet connected to the reservoir. The motor rotation is stopped or reversed with the Husco valves. The speed of the motor depends on flow rate and motor displacement.

The M2 Series Parker motor used has a displacement of 1.27 in3/rev. with a recommended speed range of 40 to 4000 rpm (see HYD, Motor Specifications). This motor will be operated with a flow rate of 10 gpm and a pressure of 1000 psi. This will result in a shaft speed of 1728 rpm, producing a torque of 202.13 in-lb, and 5.54 HP (see HYD, Calculations). In order to save money and time the Parker motor mount was designed and constructed at UNH (see Figure HYD7). A mounting plate for the Moyno Pump and the Parker motor assembly was also designed and constructed at UNH (see Figure HYD8).

The 2000 series Char-Lynn motor has a displacement of 9.6 in3/rev with a maximum speed of 477 rpm (see HYD, Motor Specifications). This motor will be operated with a flow rate of 4.5 gpm at a pressure of 1000 psi producing a shaft speed of 108 rpm, a torque of 1527.89 in-lb, and 2.62 HP (see HYD, Motor Calculations).

Both motors are internal gear motors with an inner drive gear and an outer driven gear. The outer gear has one more tooth than the inner gear. The inner gear is attached to a shaft which is connected to a load. An imbalance is created by the difference in gear area exposed to hydraulic pressure at the motor inlet. The exposed area of the inner gear increases at the inlet. Fluid pressure acting on these unequally exposed teeth results in a torque at the shaft (see HYD

Parker Motor Mount

<u>Not to Scale</u>

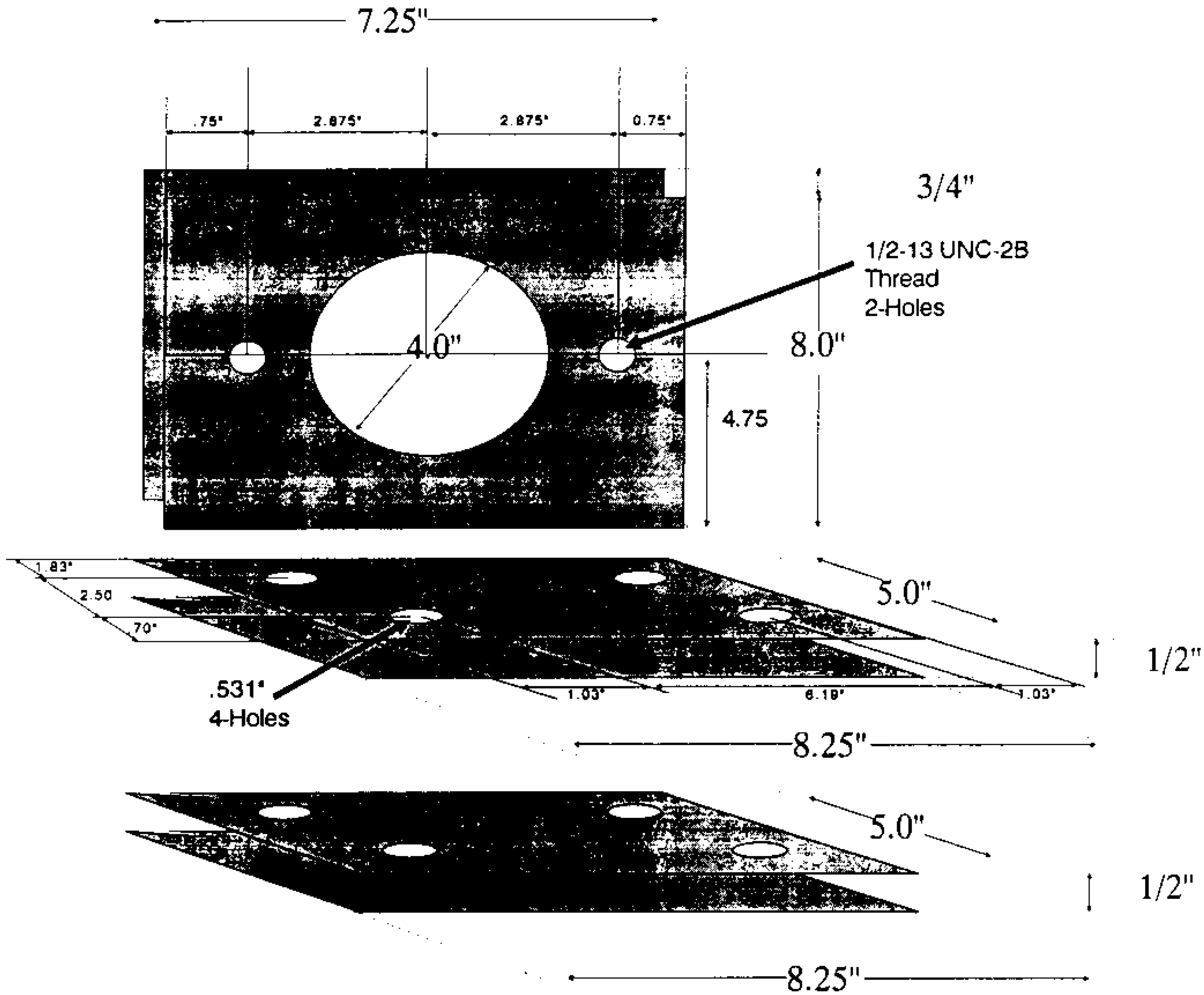
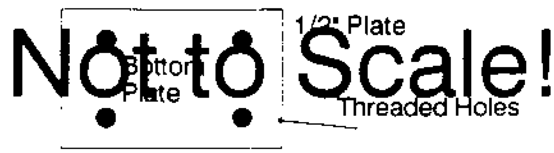
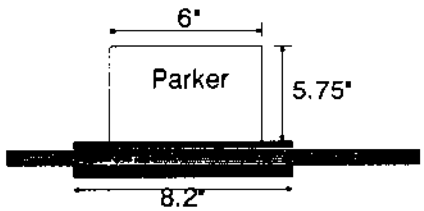


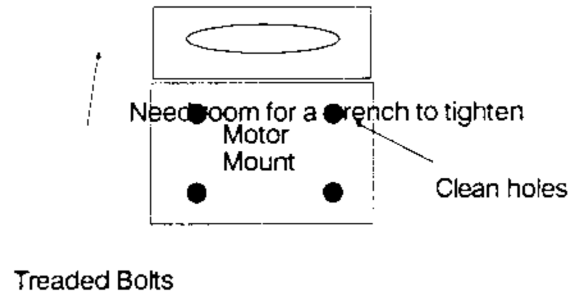
Figure HYD7

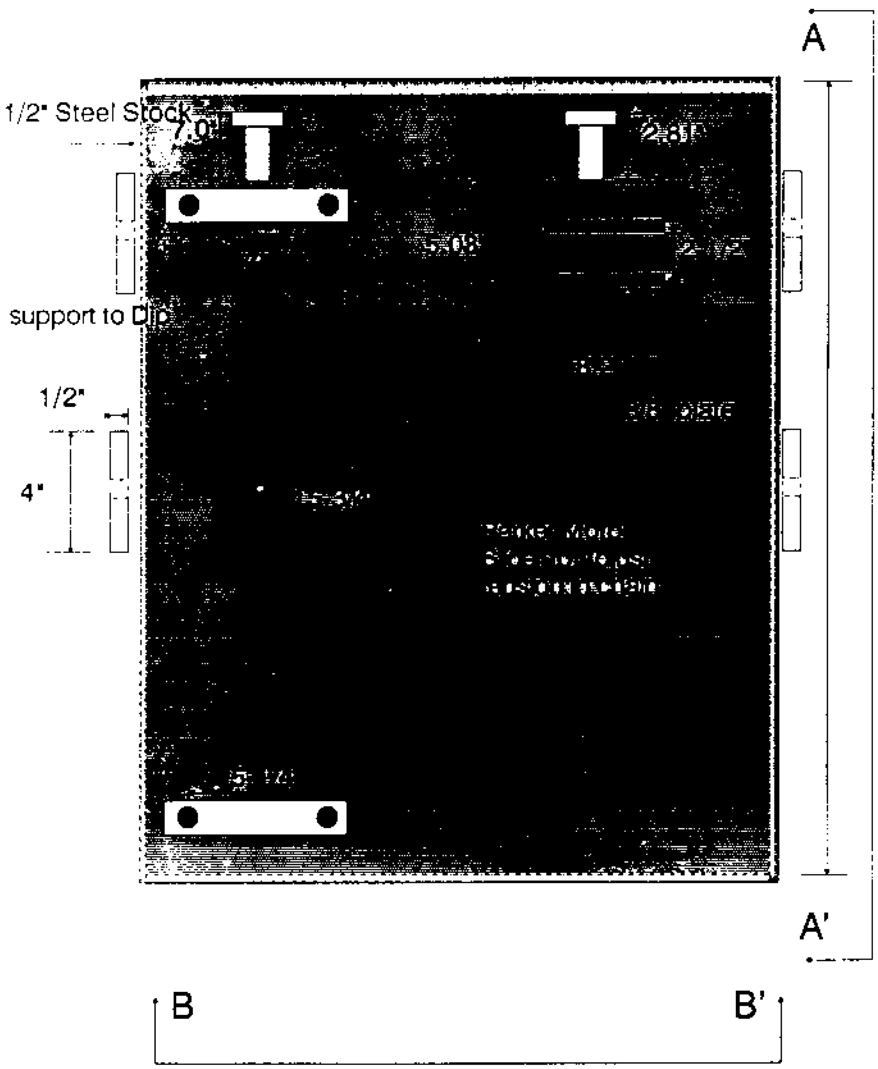
Dip Parker/Moyno Motor Mount

Compartment ~ 20" Wide X 16" Deep









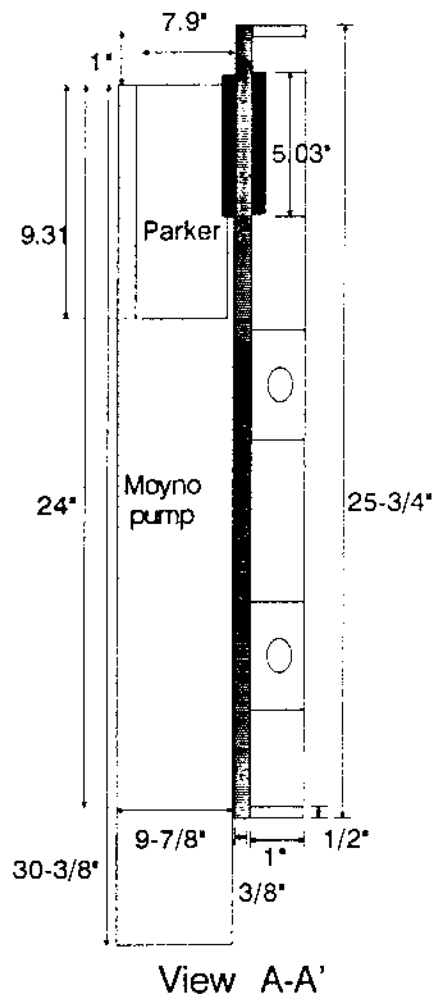


Figure HYD8

HYDRAULIC MOTOR CALCULATIONS

MOTOR SHAFT SPEED

Motor Shaft Speed (RPM) = (GPM x 231)/Displacement

For Parker Motor;

RPM = (9.5)x(231)/(1.27) = 1728

For Char-Lynn Motor;

RPM = (4.5)x(231)/(9.6) = 108

OUTPUT TORQUE

Motor Torque (lb-in) = PSI x Displacement / 2PI

For Parker Motor;

T = (1000)x(1.27)/6.83 = 202.13 lb-in

For Char-Lynn;

T = (1000)x(9.6)/6.83) = 1527.89 lb-in

MOTOR HORSEPOWER

Motor HP = RPM x Torque / 63025

For Parker Motor;

HP = (1728)x(202.13)/63025 = 5.54 HP

For Char-Lynn;

HP = (108)x(1527.89)/63025 = 2.62 HP

Figure 8). The larger the gear, or the higher the pressure, the more torque will be developed at the shaft.

Reservoirs

The function of the hydraulic reservoir is to contain the system's hydraulic fluid. It is constructed of steel walls, a dished bottom; a flat top; drain plug; filler cap; vent; temperature sensor; cleanout covers; and baffle plate. Besides acting as a fluid container, a reservoir serves also to cool the fluid, to allow contamination to settle out, and to allow entrained air to escape.

With fluid returning to a reservoir, a baffle plate blocks the returning fluid from going directly to the suction line. This creates a quiet zone which allows large dirt to settle out, air to rise to the fluid surface, and gives a chance for the heat in the fluid to be dissipated to the reservoir walls.

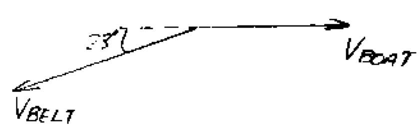
Fluid baffling is a very important part of proper reservoir operation. For this reason, all lines which return fluid to the reservoir should be located below fluid level and at the baffle side opposite the suction line.

The reservoir will be mounted at the same elevation as the Hydura pump. This is because 10% of hydraulic fluid is dissolved air and if the inlet pressure is too low this air will boil out of the fluid at the inlet. This is a dangerous condition for a hydraulic pump. Cavitation would result, indicated by loud noise and possible damage to the pump. The pump could transmit these air bubbles and metallic debris to the motors resulting in further damage. By mounting at the same elevation, advantage can be taken of the head pressure of the fluid to increase inlet pressure thus reducing the chances for cavitation to occur.

OIL RECOVERY:

The oil and debris recovery will be accomplished by the Dynamic Inclined Plane (DIP module) which collects oil from the surface of the water. This is accomplished by directing the oil (with a oil boom sweep system) into the opening of the DIP module. A conveyer belt then draws the oil into a temporary collection well. Due to the buoyancy of oil relative to that of water, the oil rises to the top of the 20 gallon collection area where it can be pumped to a larger, 150 gallon, on board storage tank. (see figure OR1)

The DIP module consists of a 15 in. wide PVC-polyester reinforced belt inclined at an angle of 23 degrees from the horizontal. This belt is driven by a Char-Lynn hydraulic motor. Operating speeds of the vessel will range up to 3 knots in order to maintain a relative velocity of zero at the oil collection point.



$$VelBoat = VelBelt \times Cos23° \qquad \text{(D1)}$$

This derived equation is used to ensure the passage of oil down the plane undisturbed. To prevent the formation of head walls in the recovery of floating oil the speed of the vessel must be below a critical velocity[7]. Previous tests on skimmers of this nature show solid results indicating recovery of 90% oil/water ratio at velocities up to 2 knots[8].

In order to transport SPOSS, manual raising and lowering mechanisms for the DIP module were devised. These mechanisms include two separate winch and pulley systems (rated at 1500lbs) with 3/16 inch stainless steel cable wire located fore and aft of the DIP module (figure OR1) While in rapid transit to the oil spills, where speeds may exceed 25 knots, the DIP module

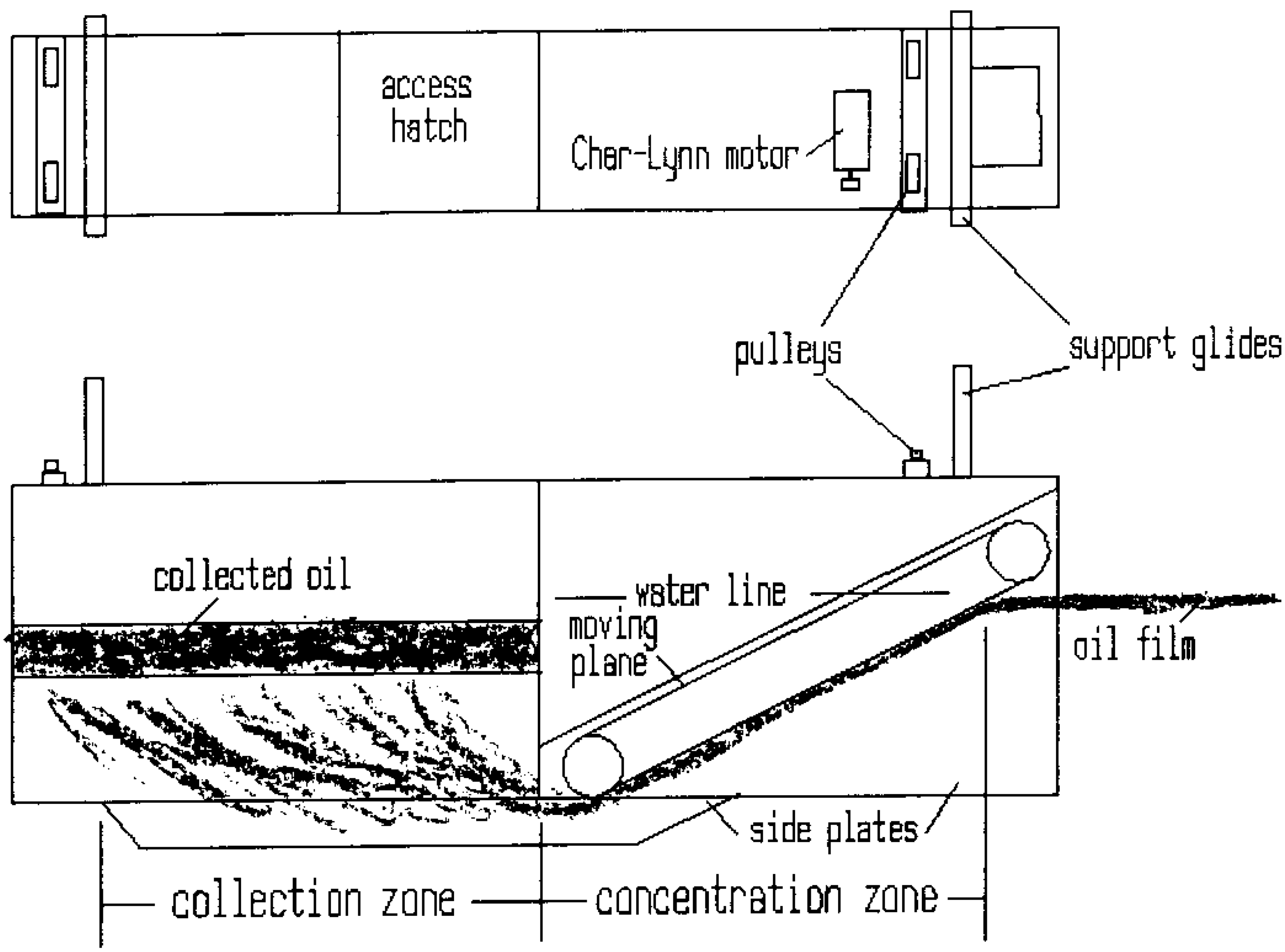

access
hatch
Char-Lynn motor
pulleys
support glides
collected oil
water line
moving
plane
oil film
side plates
collection zone
concentration zone

Figure OR1

will be raised completely out of the water to eliminate drag. During skimming operations the DIP module will be lowered into the water. Braces and glides will support the DIP module in both positions. This two person system permits quick and simple removal of the DIP module allowing the platform to have many diverse uses such as a dive platform or submersible launch platform. A design alternative considered consisted of a hydraulic winch system but was decided against due to cost.

OIL TRANSFER SYSTEM

The oil transfer system incorporated into the SPOSS involves removing the oil from the 20 gallon collection zone of the DIP module to a 150 gallon inboard tank. After filling the inboard tank, the system will pump the oil to a larger overboard storage. The discarded oil can be either pumped to a larger barge that travels with the SPOSS or shuttle back to land based storage facility. The former option will expedite oil transfer rates but will not always be accessible due to availability of resources.

The pump employed for applications of this nature is a progressive cavity, fully reversible Moyno pump which can deliver oil at 220 GPM and pass particles up to 1 in. in diameter. Manual ball valves, located on the oil transfer manifold, are used to control the direction of oil flow between the collection well, the storage tank, and overboard storage.(figure OT1) To offset the weight distribution and to assure ease in access, the Moyno pump is mounted in the stern of the vessel. The Moyno pump will be powered by the Parker motor and is readily accessible to the operator's console. Other design alternatives considered were electrical and pneumatic versions. Neither version met spark safety requirements, Code Federal Regulations, and oil flow specifications[9].

SPOSS
Product System

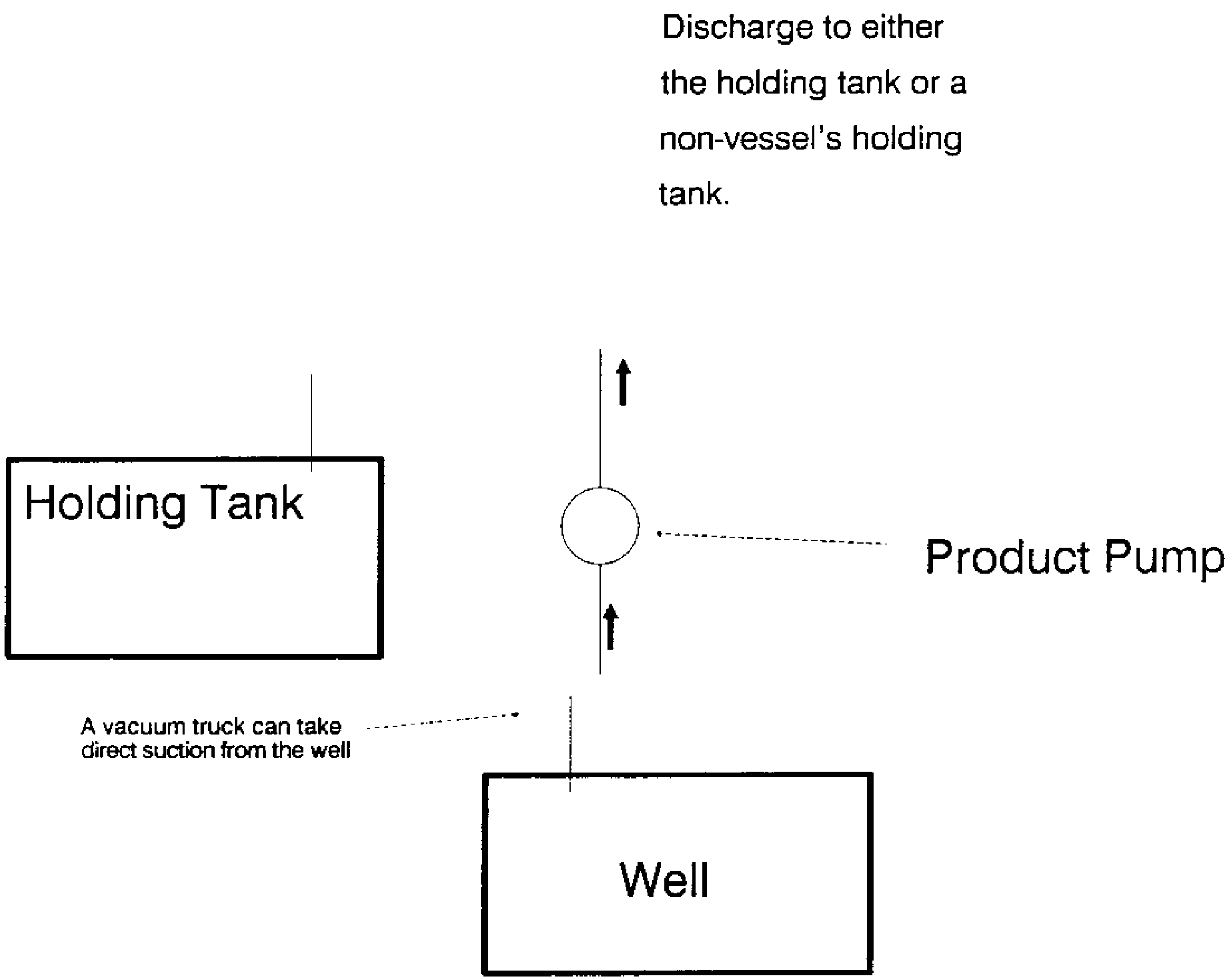


Figure OT1

PONTOONS:

The pontoons are located port and starboard of the vessel and hold 2 - 3 PSI each. The pontoons are partitioned into four cells to add safety from sinking in the case of a puncture. When fully inflated these pontoons are able to support 8000 lbs. each minimizing the slight offset in weight distribution. The pontoons are attached with 12 nylon straps (6 straps per pontoon) rated at 6000 lbs. each. Each strap device is self manufactured consisting of 18 ft. nylon straps rated at 6000 lbs. each and two stainless steel rings. The straps will readily withstand the weight of the SPOSS (3100 lbs.) as well as other forces encountered while in operation. (refer to table D1)

PROPULSION SYSTEM:

The propulsion system for the SPOSS consists of a Johnson gasoline outboard. This unit produces 85 H.P. at a full speed of 4500 RPM. The outboard is mounted on the transom of the hull. The outboard is provided with electric start and remote control for throttle and gearbox. A built-in alternator on the outboard maintains charge on the starting battery. The engine draws fuel from an independent 6 gallon tank. Steering is accomplished from the operator's console by a Teleflex steering cable connected to a 15" diameter steering wheel. The engine speed and propeller rotation direction is controlled by a single lever control at the operator's console. The costs of the propulsion system are shown in table PRO 1.

Table Pro 1: List of Propulsion Components and Prices.

COMPONENT	QUANTITY	PRICE
Johnson 85 HP outboard	1	Donated
6 gallon Fuel Tank	2	31.00x2=$62.00
Teleflex Steering Cable	1	95.00
Throttle & Dir. Control	1	120.00
	TOTAL	**$277.00**

ELECTRICAL SYSTEM:

The SPOSS's electrical system is a 2-wire 12 volt DC system. Power is supplied from a deep cycle marine battery located on deck in a weatherproof box. Battery charging and additional power is provided by a 50 amp max alternator on the vessel's hydraulic power pack. The alternator was installed so that auxiliary power would be available for future accessories, such as spot lights. The Electrical power distribution is provided by a control panel mounted on the operator's console. Individual circuits on this panel are controlled by three way switches with built in circuit breakers, thereby providing overcurrent protection. Further circuit protection is provided by a fuse installed in the main line between the battery and the control panel. A second weatherproofed battery is required for starting the Johnson outboard motor. Having two batteries on the vessel provides added safety in the event that one of the batteries loses it charge. The costs of the electrical system are shown in table ELEC 1.

Table ELEC 1: List of Electrical Components and Prices.

COMPONENTS	QUANTITY	PRICE
Battery	2	50.00x2=$100.00
Control Panel	1	$31.95
	TOTAL	**$131.95**

SAFETY:

In order for the SPOSS to be coast guard approved for operation various safety features are required. Navigation lights on the starbird and port sides, green and red respectively, are required along with a white stern light which is mounted above the boats highest point. An air horn for alerting others of your location, an ABC rated fire extinguisher, and coast guard approved life-vests are also required. In order to send distress signals smoke and light flares along with high altitude (1,000 ft.) rocket/parachute flares are essential. The costs of all this safety equipment is listed in table SAF 1.

Table saf 1: List of Safety Components and Prices.

COMPONENT	QUANTITY	PRICE
Navigation Lights	3	10.50x3=$31.50
Air Horn	1	$12.00
Fire Extiguisher	1	$35.00
Throw Cushion	1	$9.60
Life Jacket	4	15.00x4=$60.00
Smoke Flare	3	13.95x3=$41.85
Light Flare	3	23.10x3=$69.30
Parachute Flare	3	37.95x3=$113.85
	TOTAL	**$373.10**

OIL BOOM SWEEPS SYSTEM (OBSS)

The SPOSS is equipped with a pair of oil boom sweeps, figure OBSS 1. As shown in figure OBSS 1 the four main components of the oil boom sweeps system are the boat fastening points, support arms, boom strip, and boom. The function of the sweeps are to increase the oil skimming aperture from 2 feet to 7 feet. The sweeps are connected to the bow of the vessel with a quick release connection. The sweeps are pulled up from the waters surface when not in use and tied off as shown figure OBSS 2.

The functional requirements of the sweeps are to be: lightweight, corrosion resistant, strong enough to operate at three knots, responsive to wave dynamics, easy to manufacture, inexpensive, designed flexibly, easily deployed and retrieved, and effective in the collection of oil. In order to satisfy these requirements the bow of the boat was drawn on CAD, equations and materials were researched. The machinists working at U.N.H.'s Morse Hall machine shop were consulted on the ease of manufacturing the different design alternatives considered. Drawing the boat on CAD as accurately as possible was very important because different design options could be considered very quickly, in terms of weighing out the pros and cons of each.

After drawing the bow of the boat on CAD the next step in the design process was to determine what material should be used. A material's cost, corrosion resistance, and strength to weight ratio were the deciding criteria. The material with the lowest cost, highest corrosion resistance, and highest strength to weight ratio would be the ideal material. These requirements would lead to the design of the lightest, most economical and longest lasting OBSS. Strength to weight ratio is defined as tensile strength divided by density. The tensile strengths of steel, aluminum, and stainless steel are 100 ksi, 33 ksi, and 90 ksi, respectively[10]. While the densities of steel, aluminum, and stainless steel are .28 lb/in^3, .097 lb/in^3, and .31 lb/in^3, respectively[10].

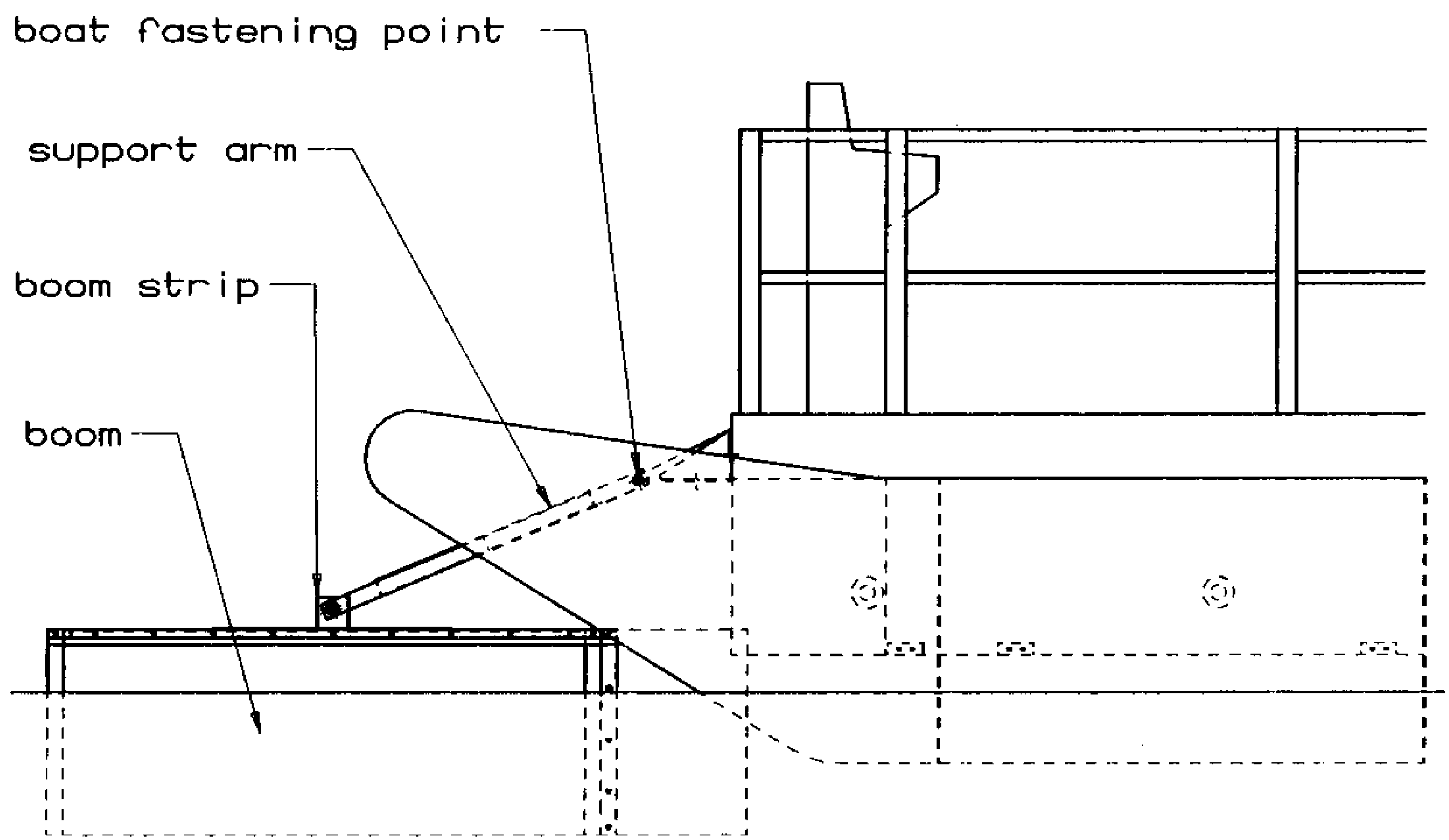
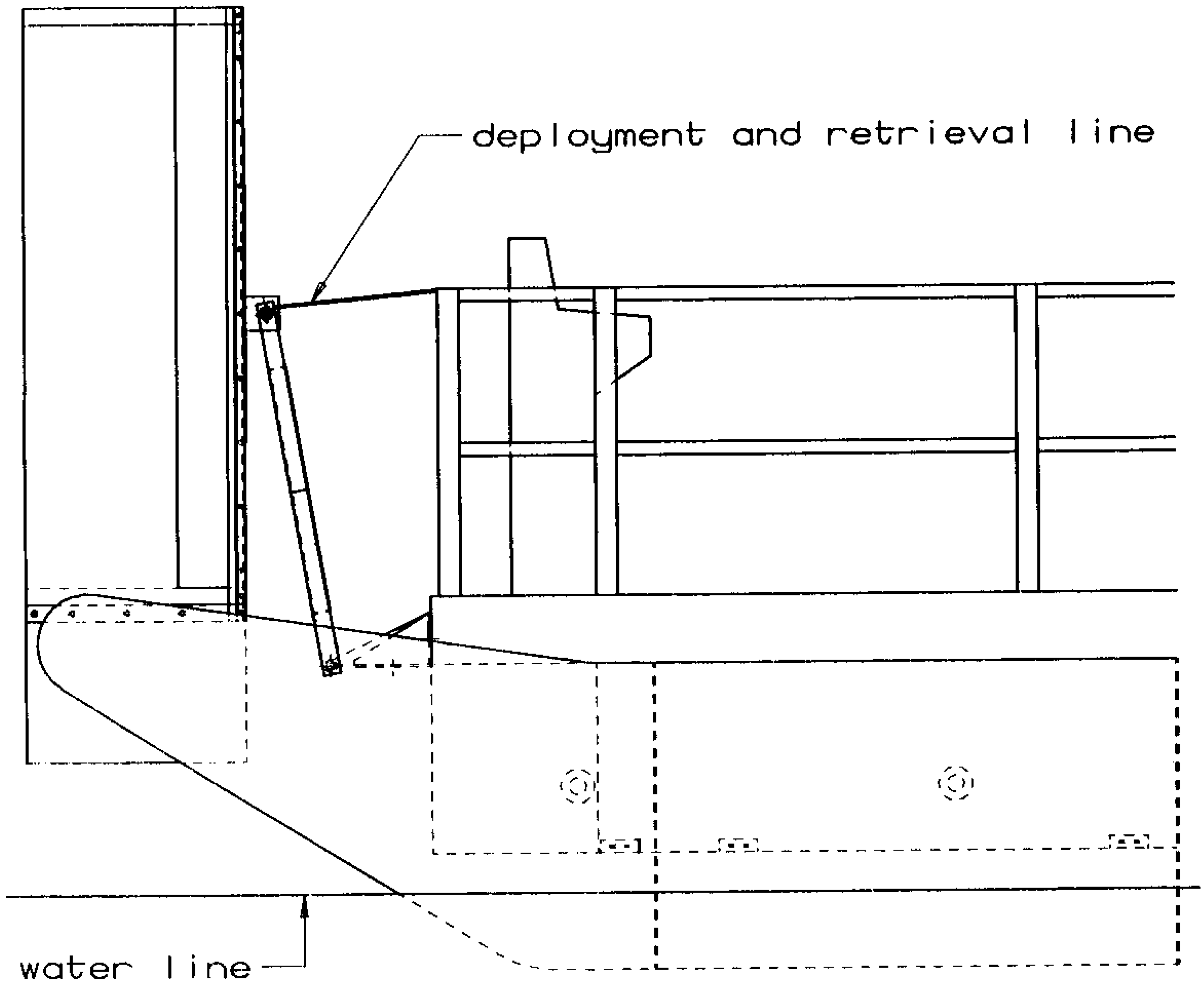


Figure OBBS 1: Main components of the Oil Boom Sweeps System (side view).

scale = .045

Figure OBSS 2: OBBS shown in storage position (side view).

scale = .055



The strength to weight ratios of steel, aluminum, and stainless steel are therefore 357, 340, and 290. The cost per pound of steel, aluminum, and stainless steel are \$.30, \$2.00, and \$3.30 respectively[11]. In terms of corrosion resistance steel is the worst, while aluminum and stainless steel would perform approximately equally as well.

If steel was chosen it would have to be painted which would quickly wear away at the pivot points of the OBSS, leaving the surface exposed for the formation of rust. Aluminum and stainless steel will not deteriorate like steel, making them ideal for a marine environment. Even though steel is the cheapest and has the highest strength to weight ratio it was not chosen because of the corrosive environment the OBSS operates in. Deterioration would cause the OBSS to have too short of a life span. Aluminum was chosen because it has a higher strength to weight ratio and is cheaper than stainless steel while providing approximately the same corrosion resistance. After choosing aluminum Northstar Steel was called in order to see what series aluminum they sell. Northstar Steel sells 6061 series aluminum which has a yield strength of 33 ksi[12].

Once the 6061 series aluminum was chosen the OBSS could be designed. The material had to be chosen first in order to insure that the dimensions of the OBSS were substantial enough to prevent the occurrence of stresses exceeding the yield strength of the aluminum. The primary force equation[12],

$$F_d = (C_d * \rho * V^2 * A)/2 \qquad \text{equation 1}$$

where:

F_d = drag force on boom
C_d = coefficient of drag
V = velocity of water normal to boom
ρ = density of water
A = area of boom in contact with water

, was used to determine the drag force acting on the boom. C_d was found to equal 2.0[13]. The density of salt water at 48 degrees F is equal to 2 lbf/ft^3 [(12)]. The normal velocity to the boom was determined using the trigonometric function:

$$V_n = V_b * \cos(\theta_1) \qquad \text{equation 2}$$

where:

V_n = velocity normal to boom
V_b = velocity of the boat
θ_1 = acute angle boom makes with the boat in the horizontal plane.

The worst case boat velocity of 3 knots (5.06 ft/sec, 3.45 mph) was ultimately used as V_n to add an early safety factor. This value could actually be reached if the vessel's pilot forgets to slow down when making a turn. θ_1 is equal to 67 degrees. The area is equal to the boom length (6') times the skirt height (1.5') which equals 9 ft^2. The skirt length was determined after talking with Slickbar Products Corporation and JBF Scientific. Applying equation 1 at 3 knots yields a boom force of 461 lbf. This force value is the key value in determining whether or not the various components of the system would fail, because it is the worst case.

A failure analysis was conducted at the boat fastening point because this is the area were the largest force acts. The force is the largest here because the support arm acts as a lever arm. From the free body diagram, figure FBD 1, the unknown reaction forces where determined from summing the forces in the X and Y directions and summing the torques about points a and b. The resulting equations are as follows:

$$\Sigma F_x = 0 = R_x - F_b + S_x \qquad \text{equation 3}$$
$$\Sigma F_y = 0 = R_y - S_y \qquad \text{equation 4}$$
$$\Sigma T_a = 0 = -F_b(SA) + w/2(R_y+S_y) \qquad \text{equation 5}$$
$$\Sigma T_b = 0 = w/2(R_y+S_y) - SA(R_x+S_x) \qquad \text{equation 6}$$

where:

R and S = reaction forces
SA = support arm length
w = width of the fastening tube.

Using these equations with w, SA, and F_b equaling 9", 42", and 461 lbf, respectively, R_x, S_x, R_y, and S_y were found to equal 0 lbf, 461 lbf, 2,151 lbf, and 2,151 lbf. R_x equals 0 lbf because of

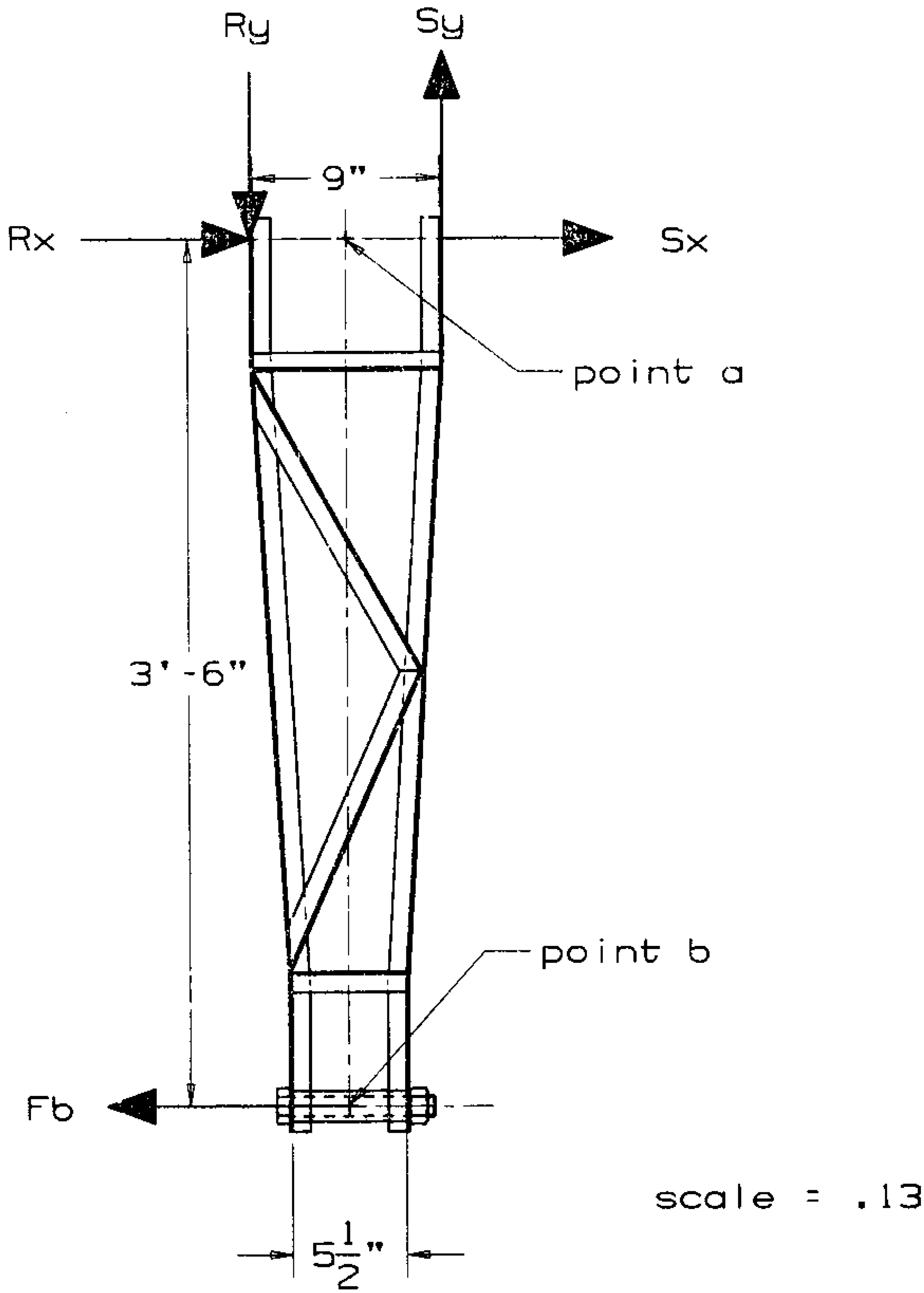


Figure FBD 1: Free body diagram of the boom support arm, showing reaction forces.

the way in which the support arm is fastened to the fastening tube.

These force values divided by the area of contact between the support arm and the fastening tube provided the stress values which were compared to 6061 aluminum's yield strength. For the y component reaction forces the area is equal to the thickness of the support arm (.125") times half the circumference of the bore (r=.5") through the support arm. These thickness and bore sizes were chosen after talking with Northstar Steel about material sizes in stock. Therefore the y component contact area equals .19625 in^2, which results in a stress equal to 11 ksi. This stress value results in a factor of safety equaling 3. The X component's area of contact is equal to the cross sectional area of the fastening tube in contact with the support arm. The fastening tube is a 1.5" OD by .25" wall thickness tube. The resulting area of contact area is .9813 in^2, which results in a stress equaling .47 ksi. This stress results in a factor of safety equaling 70. It is apparent from the factor of safeties that the y component has a much greater chance of failure.

The extension of the pontoons from the bow, figure OBSS 3, made it geometric impossible to attach the OBSS to the existing quick release fastening tubes. This brought about the need to design a new quick release fastening platform, which could be mounted on the existing platform with minimal boat modifications. The design solution is shown in figure OBSS 4. The platform had to be made into two pieces to allow for the placement of a welded reinforcement plate and to allow for independent adjustability of the two boom support systems. The reinforcement plate and the new platform are shown installed in figure OBSS 5. The reinforcement plate had to be installed because the existing platform flexed with a load of approximately 50 lbf in the area of the new platform. The fastening tube was made out of a stock 1.5" OD tube with a .5" wall thickness.

The design of the support arm shown in figure OBSS 6 was chosen because it incorporates

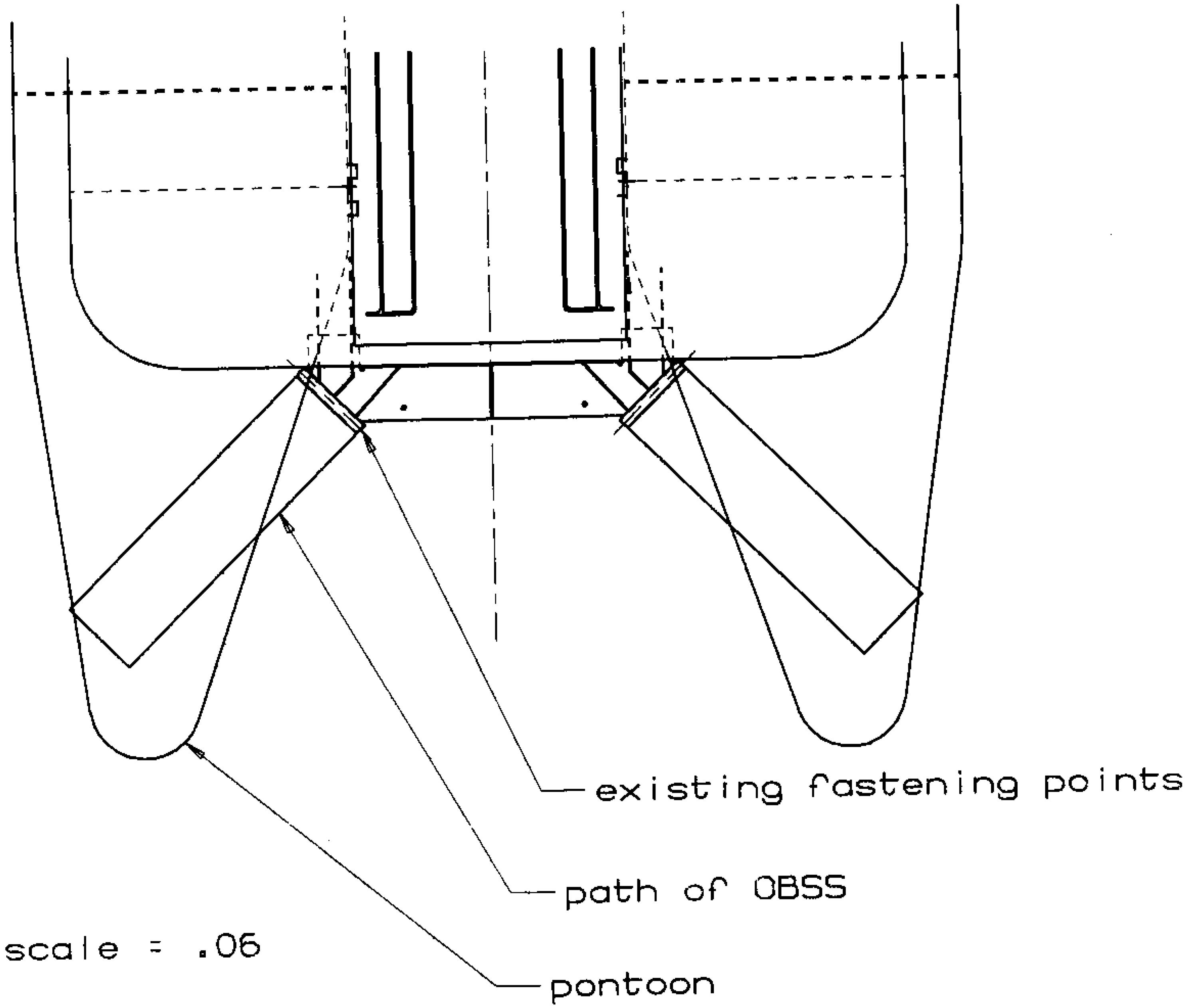


Figure OBSS 3: Geometric impossiblity of existing fastening points (top view).

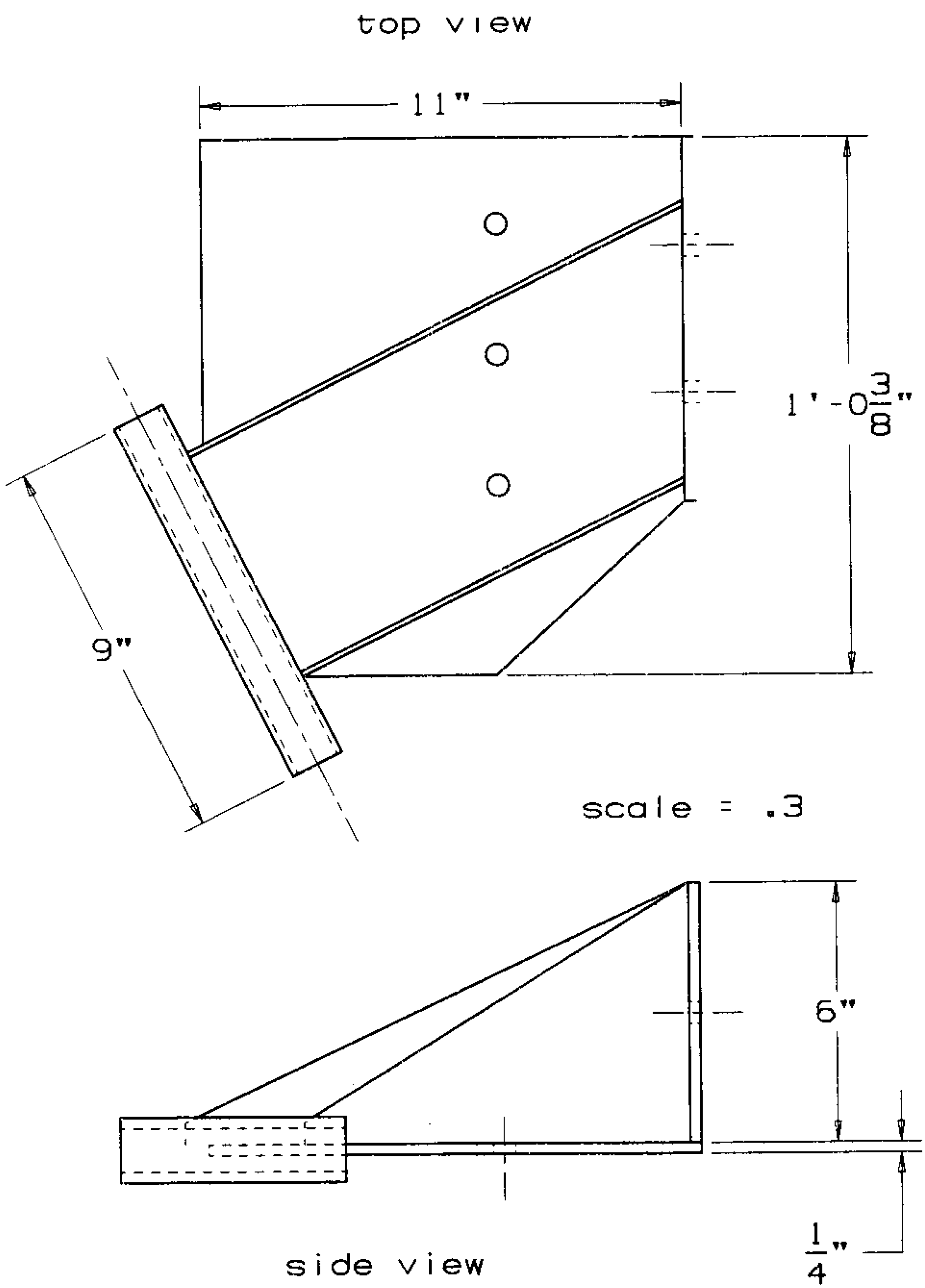


Figure OBBS 4: Fastening point platform provides a solution to the geometric constraints.

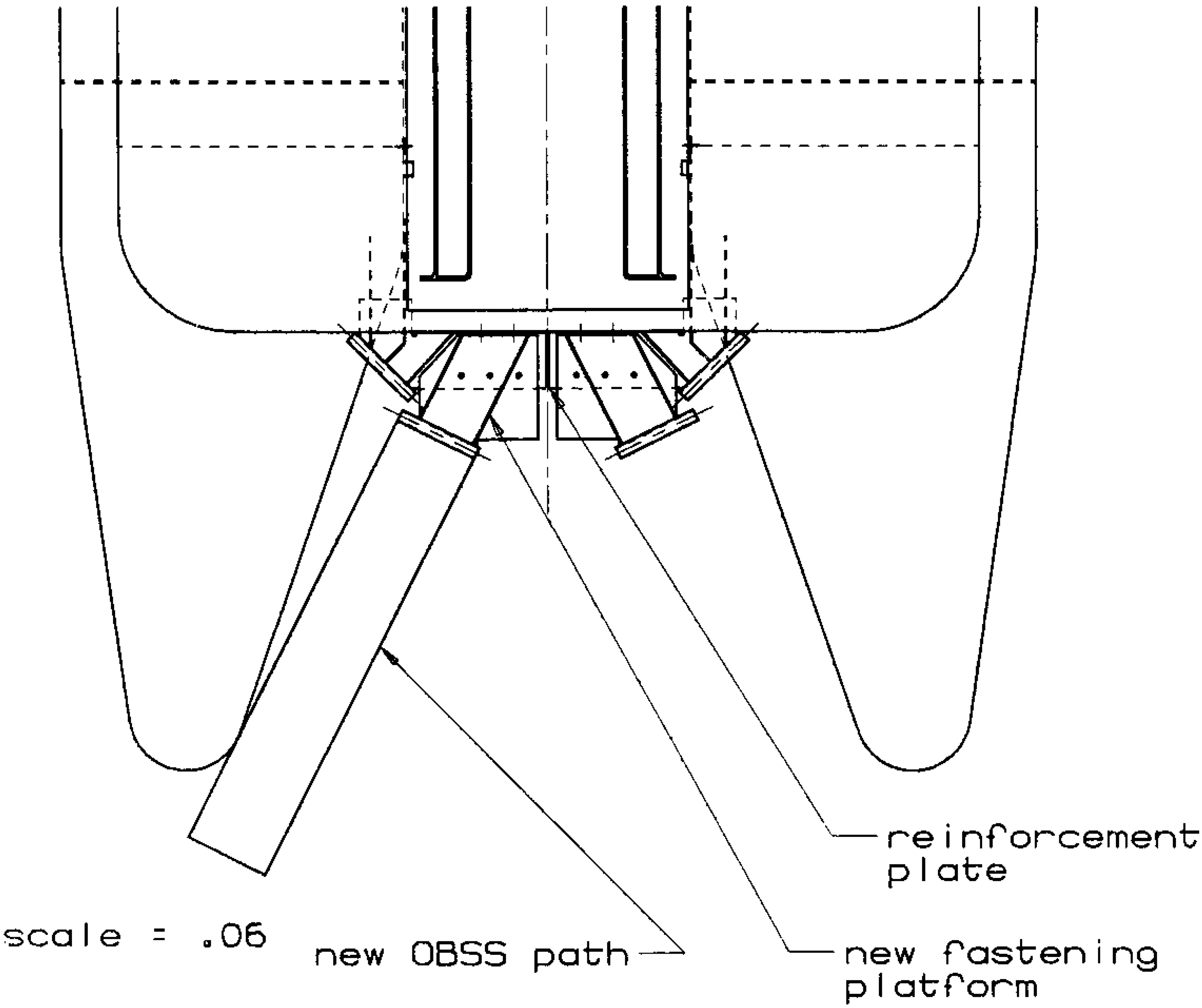


Figure OBSS 5: Installation of reinforcement plate and new fastening platform (top view).

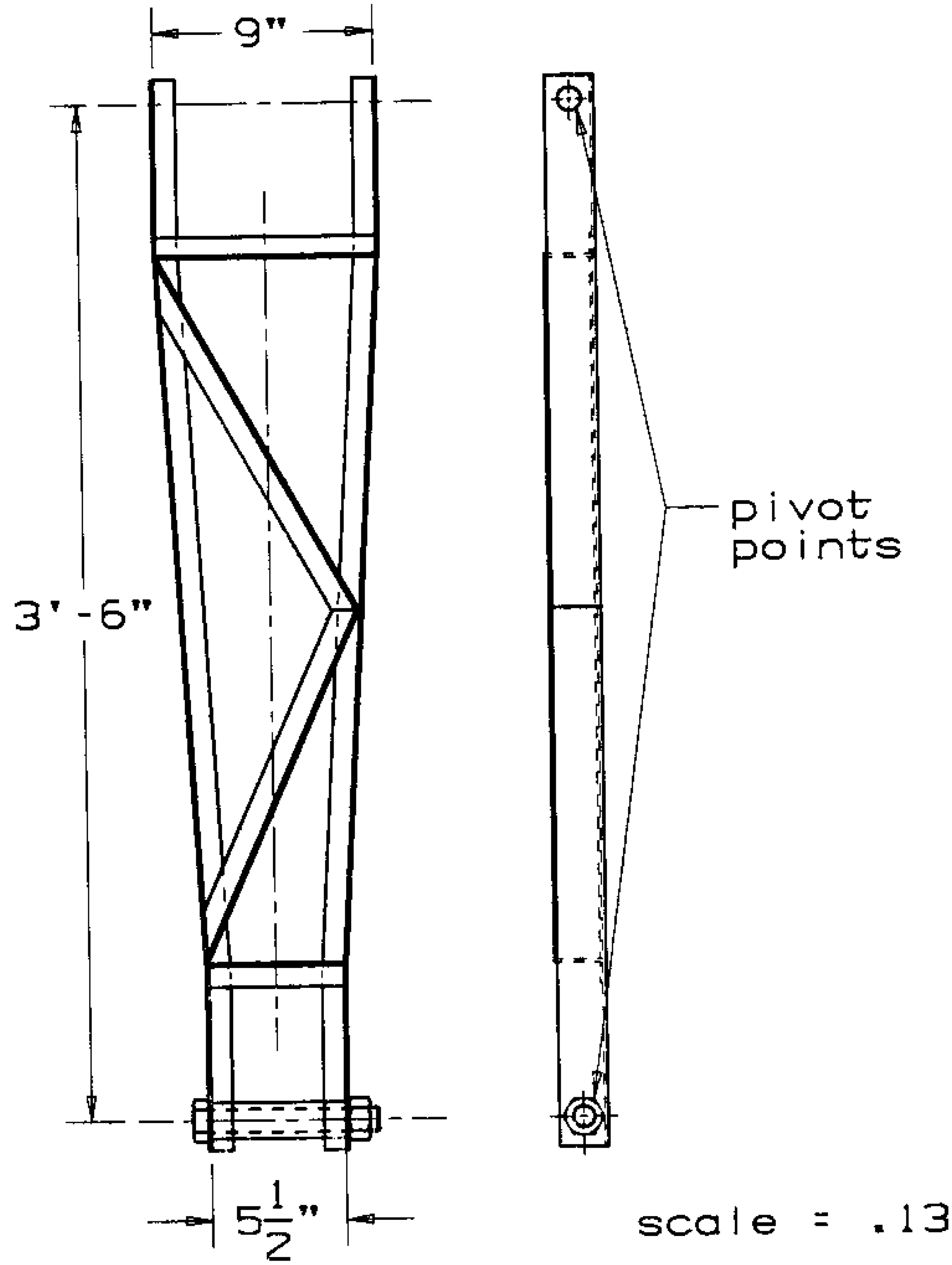


Figure OBBS 6: Boom support arm top and side views.

the use of a framework reinforced with cross bracing, meets the geometric considerations of the boat, allows for the OBSS to respond to the dynamics of waves, and meets the strength requirements. In order to ensure constant contact with the water, given wave dynamics, the OBSS pivots at two points.

The support arm was chosen to be made out of 2"X1"X.125" angle stock to ensure that the yield strength was not exceeded and for ease of manufacturing given the milling machinery available for our use. The support arm pivots at both of its ends. The end where it attaches to the bow is 9" wide with a 1" bore. These dimensions allow for the support arm to fit on the outside of the bow fastening point and the passage of a 1" OD bolt in order to tie the pivot point together. The other end of the support arm connects to the boom strip pivot point. The decided dimensions for this interface were 5.5" outer width with a 1" bore for a tieing bolt. The total length of the support arm equals 42". This value was found using the following trigonometric function:

$$SA = HFP/\cos(\theta_2) \qquad \text{equation 7}$$

where:

θ_2 = the acute angle between the support arm and the boat in the vertical plane, under calm conditions

HFP = the height the boat fastening point is above the water minus the height the pivot point at the boom strip is above the water line

SA = support arm length.

The boom support strip, figure OBSS 7, like the support arm was designed in an attempt to minimize weight while maximizing strength and yet to still maintain design flexibility. The support arm attaches to the boom strip at its center point. This was done in order to minimize the torque acting about this pivot point, by having the torques about the point cancel out. Two

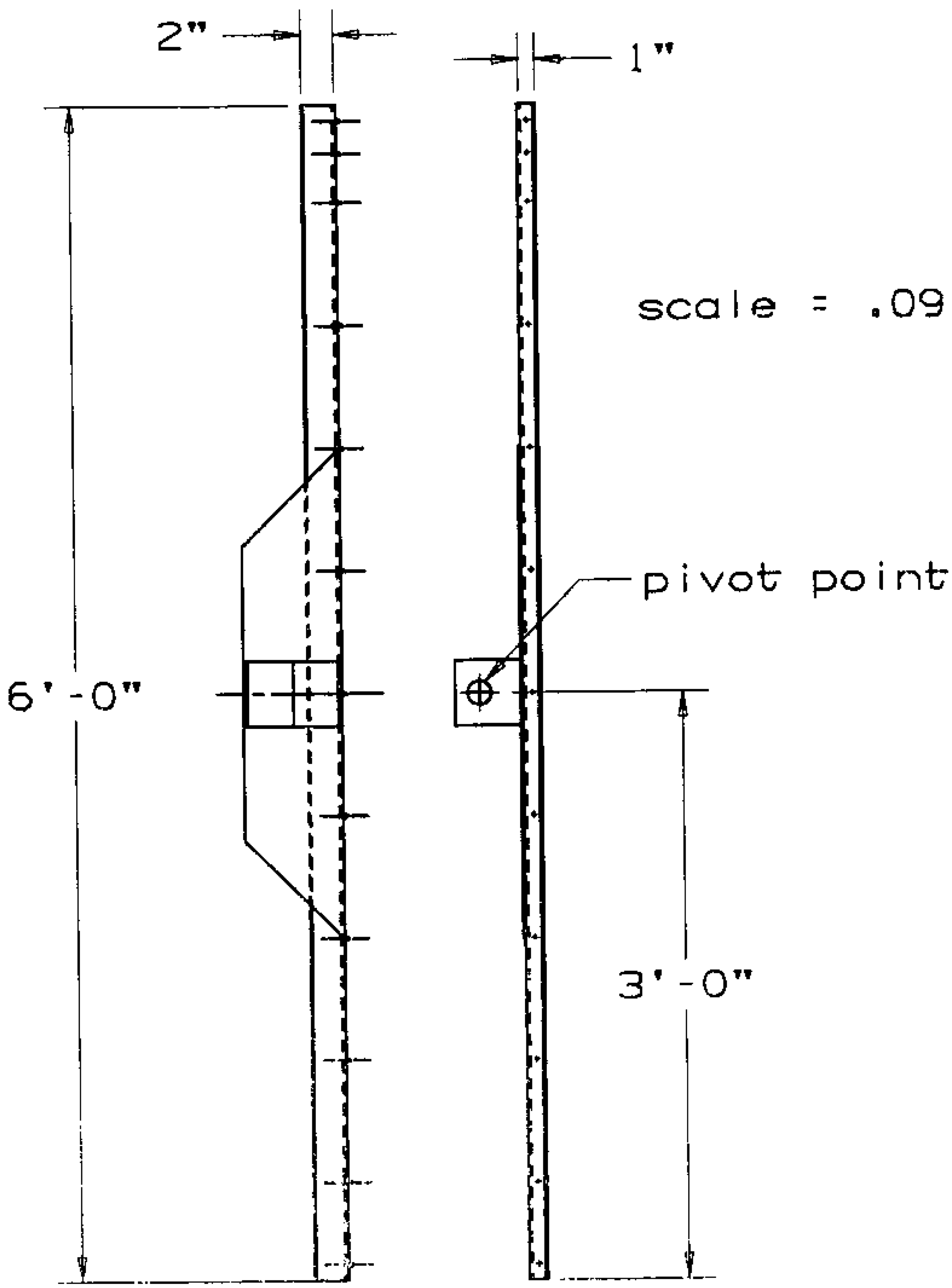


Figure OBSS 7: Boom support strip top and side views.

pieces of angle stock were used in order to clamp the boom to the OBSS. The clamping material was chosen to be 2"X1"X.125" angle stock for the same reasons as the support arm. One piece of the angle stock was then welded to a .25" thick by 6" wide by 3' long aluminum plate in order to provide rigidity to the angle stock. The plate also acts as a platform upon which to mount the support arm to boom strip pivot interface. The pivot interface was constructed by welding two 4"X3"X.25" angle stock with a 1" bore to the plate. The 1" bore allows for a 1" diameter bolt to pass through itself, the support arm, a 1.5" OD by 5.25" long tube, the support, and itself (in that order). Finally the pivot point is tightened with a nut.

The oil booms themselves, figure OBSS 8, were designed to ensure proper buoyancy force to support the support arm and boom strip while effectively containing the oil. The required buoyancy material for the boom was determined by Slickbar Products Corporation after giving them each OBSS's forecasted weight of 60lb. The effective containment of the oil was already partially considered in the design of the two pivot OBSS. The second consideration is in ensuring constant contact between the boom and the DIP module despite wave dynamics. The skirt length of the boom was chosen to be 18' long with a ballast chain attached to the bottom, after talking with Slickbar Products Corporation, to ensure that oil won't flow under the boom.

A heavy rubber material much like a door mat was chosen to fill the wave induced play. The amount of play required was determined by varying HFP + or - 1 ft while keeping SA constant. The two extreme cases were examined, using trigonometric functions, in terms of how the horizontal leg of the triangle changes. From these calculations a rubber mat length of 16" was chosen. The rubber mat was chosen because: it can be easily attached to the boom and the boom strip, is self supporting, can easily be trimmed to proper shape after testing, will not harm the DIP module if it makes contact with the belt, and is inexpensive and replaceable. The rubber

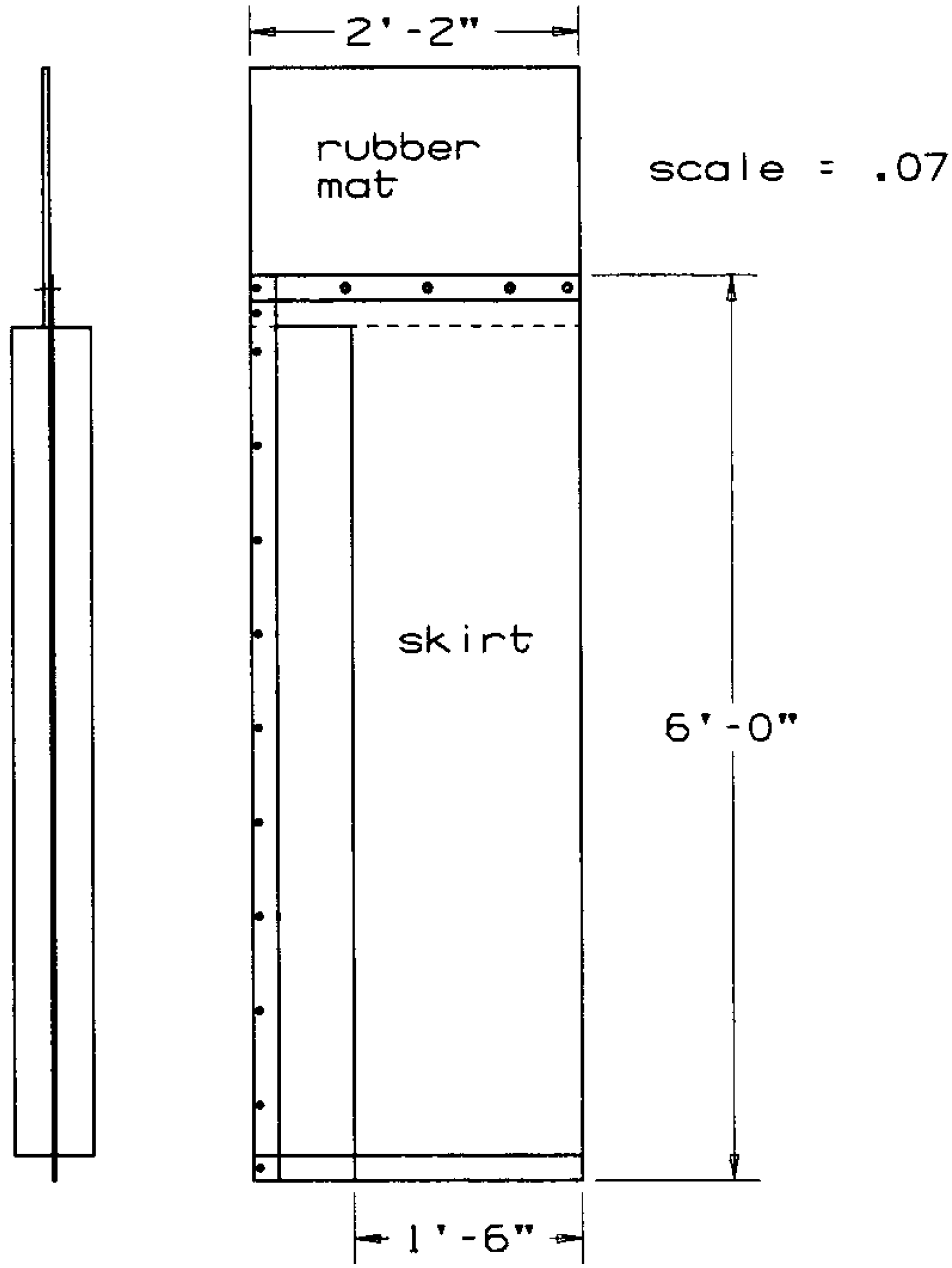


Figure OBSS 8: Side and top views of boom and attached rubber mat.

filler's qualities of being inexpensive and easily replaced are especially important because the mat will most likely make contact with the DIP belt at extreme conditions. Its ductile property will also prevent the destruction of the expensive DIP belt. The cost of the OBSS is shown in table OBSS 1.

Table OBSS 1: Price list of the OBSS.

Aluminum Material	**Quantity**	**Price/unit**
Flat Plate 6' X 6" X 1/8"	1	$53.75
Angle Stock 2" X 1" X 1/8"; 16' lengths	4	$13.50
Angle Stock 4" X 3" X 1/4"; 16" long	1	$24.50
Tubing 1.5"OD; .25" wall thickness; 18" long	1	$20.00
Boom 6X18"	2	Donated

total $129.75

TRAILER DESIGN & CONSTRUCTION

Initially the trailer for SPOSS was unwittingly deemed a secondary concern in the over-all scheme of this project. It was quickly realized, however, if over-looked in terms of design and construction the trailer can represent the deciding factor in obtaining or falling short of the ultimate goal; designing and testing a self propelled oil skimming system.

Prior to 'shopping around' for a trailer pertinent variables were deliberated over which would dictate exactly what type of trailer would best suit our purpose. The total skimmer weight, including pumps, engines, fuel, pontoons, etc was first taken into consideration. From table T1 it can be seen that the estimated overall weight of the skimmer in transit was calculated to be 3,120 lbs. This calculation employed conservative values so as to insure a

Device	Weight (lbs)
Skimmer	1000
Dip Module	700
Hydr. Fluid Tank	200
Moyno Pump	100
Outboard Engine	270
Pontoons	300
Hydr. Pumps/Motor	220
Booms	150
*Extras	140
Total	3080

* Consists of batteries, gas tanks & Char-lyn motor

Table T1: Assessment of over-all skimmer weight

reasonable safety factor. The over-all boat length was then measured to give some idea of what size trailer would be needed. This measurement was taken from one end of the pontoon to the other which spanned 22 feet. Next, the option of using a bunk or a roller support for the skimmer was debated over. The bunk design was ultimately chosen for two reasons: first, a roller design presents less contact area for the pontoons which could result in creep and render permanent deformation. Second, the coefficient of friction amidst the pontoons and rubber is greater than that between the pontoons and carpet-like material of the bunk. Therefore, the deployment and recovery of SPOSS would be smoother with a bunk design. Choosing between a single or tandem axle presented a slight problem. Single axle trailers provide simplicity and maneuverability. Tandem axle trailers, on the other hand, tend to track better and offer added security in the case of a flat tire. It was decided that a single axle trailer would be purchased which could always be renovated to accommodate another axle if need be. Purchasing a galvanized trailer was unanimously agreed upon due to the subjectivity of the trailer to salt water To be in compliance with the laws of NH we had to invest in a trailer equipped with brakes due to the fact that our GVWR (Gross Vehicle Weight Rating) exceeds 3,000 lbs.

Once equipped with these parameters the search for a plausible trailer commenced. After looking through countless want-adds and visiting numerous used-trailer lots a 20 foot, single axle trailer with rollers was considered. Seeing the potential for renovation it was purchased for $700.00 (Fig. T1).

The next objective was to look into what regulations must be abided by when towing a vessel. New Hampshire does not allow the towing of any vessel with a width in excess of 102 inches without a special permit. In light of our infringement upon this regulation a special permit was acquired from the Commission of Transportation in Concord. The trailer was then registered through UNH.

The first obstacle encountered when discussing renovations was the trailer's insufficient width

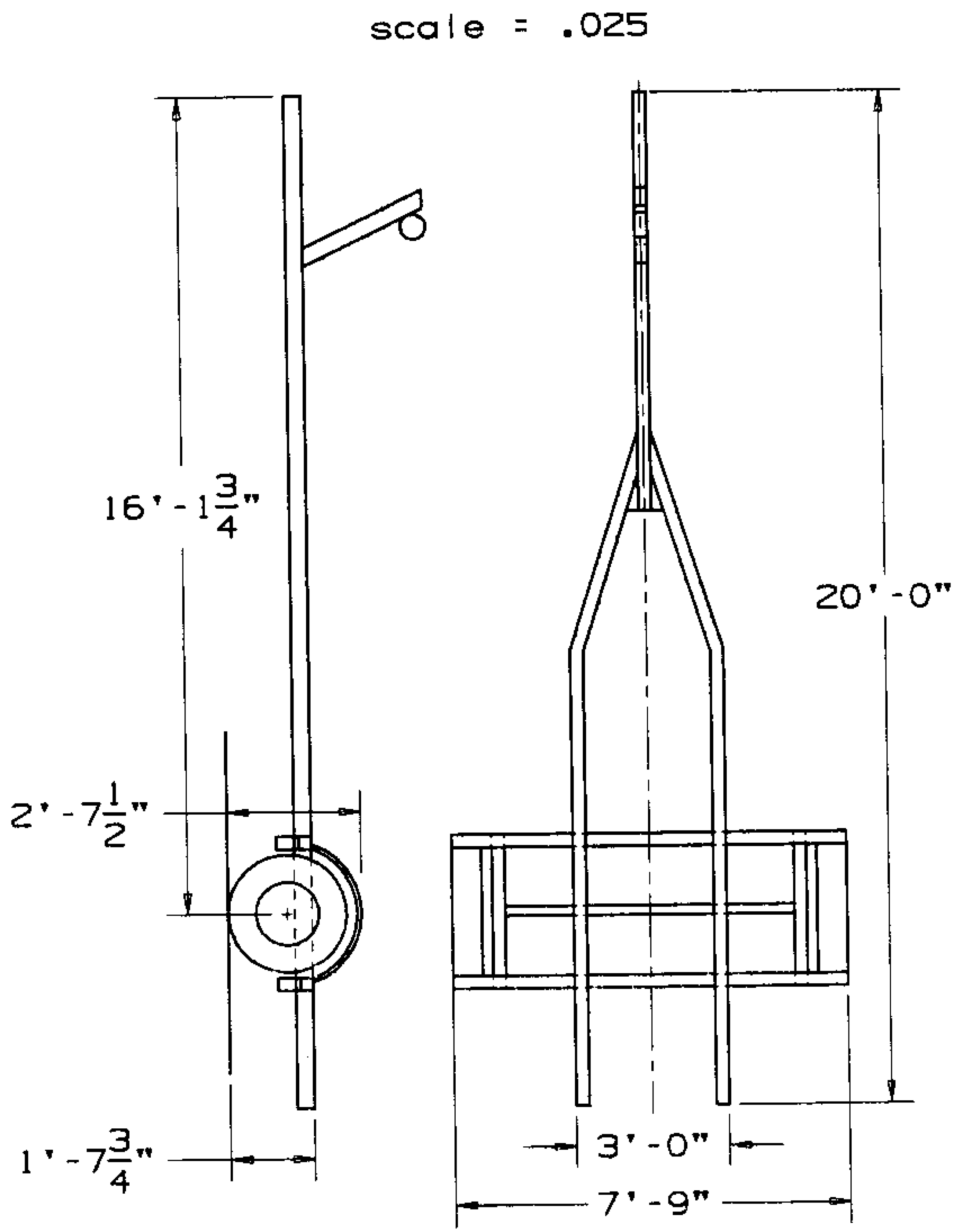
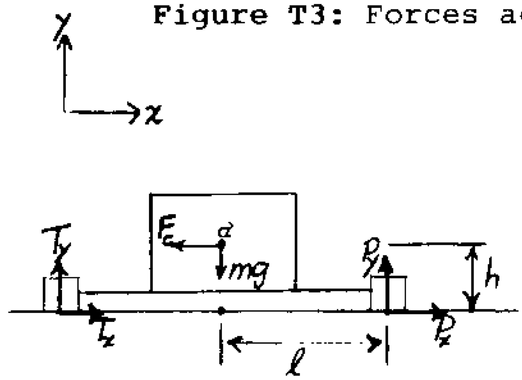


Figure T1: Original trailer

to accommodate the skimmer. This temporary dilemma was reconciled with the proposed construction of a platform that would lie just above the wheel fenders. The unprosperous lever arm produced by this complement was to be curtailed with the addition of another axle. The adjunction of this axle would also alleviate some of the stress endured by the original axle. These renovations resulted in the fabrication of the trailer shown in Fig. T2.

To determine the feasibility of this design, basic force calculations were performed. A FBD (Free Body Diagram) of the trailer was drawn showing the forces that would be acting upon it while in the process of turning (Fig. T3)[14].

Figure T3: Forces acting upon trailer/skimmer while cornering



T_x, P_x: Tangential Forces
a: Center of Gravity
w: Weight
F_c: Centrifugal Force
R: Radius of Curvature
V: Velocity of Trailer
m: Mass of trailer & boat (GVWR)
a_c: Centripetal Acceleration
T_y, P_y: Normal Forces

$$F_c = ma_c$$

$$a_c = v^2/R$$

$$\left.\begin{aligned} T_x &= \mu T_y \\ P_x &= \mu P_y \end{aligned}\right\} \text{Law of Friction}$$

Assuming there is no motion in the vertical direction

$$\sum F_y = 2T_y + 2P_y - mg = 0 \qquad \text{(T1a)}$$

$$mg = 2T_y + 2P_y \qquad \text{(T1b)}$$

a 2 is placed before the normal forces because of the tandem axle design.

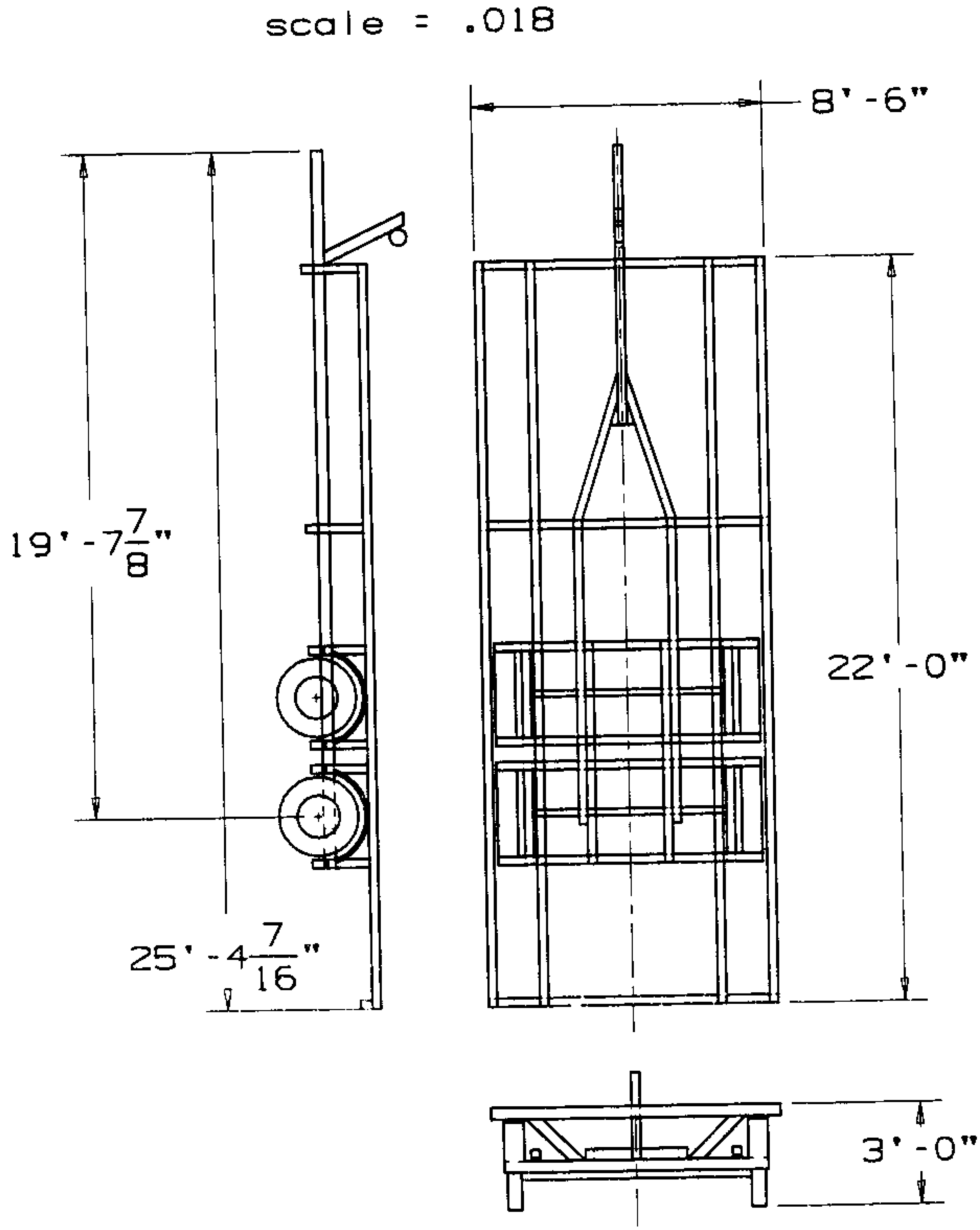


Figure T2: Renovated trailer

Assuming there is no motion in the radial direction

$$\sum F_x = -F_c + 2T_x + 2P_x = 0 \tag{T2a}$$

$$F_c = 2T_x + 2P_y \tag{T2b}$$

Applying the laws of friction and making some simple substitutions yields

$$\mu(2T_y + 2P_y) = mv^2/R \tag{T3a}$$

$$\mu mg = mv^2/R \tag{T3b}$$

$$v_2 = \mu gR \tag{T3c}$$

(T3c) gives the conditions required for slip to occur. If there is no rotational motion about point 'b'

$$\sum M_b = 2T_y l - F_c h - 2P_y l = 0 \tag{T4a}$$

$$F_c h = 2T_y l - 2P_y l \tag{T4b}$$

In the case of tipping, $P_y=0$. Applying this condition with the substitution of equation (T1b) into (T4b) results in

$$2T_y = mg \tag{T5a}$$

$$(mv^2/R)h = mgl \tag{T5b}$$

$$v^2 = (l/h)gR \tag{T5c}$$

Equation (T5c) gives the conditions required for the onset of tipping.

Using equations (T3c) and (T5c), the graph in Fig T4 was drawn to give the maximum permissible speed associated with various radiuses of curvature before tipping or sliding would occur.

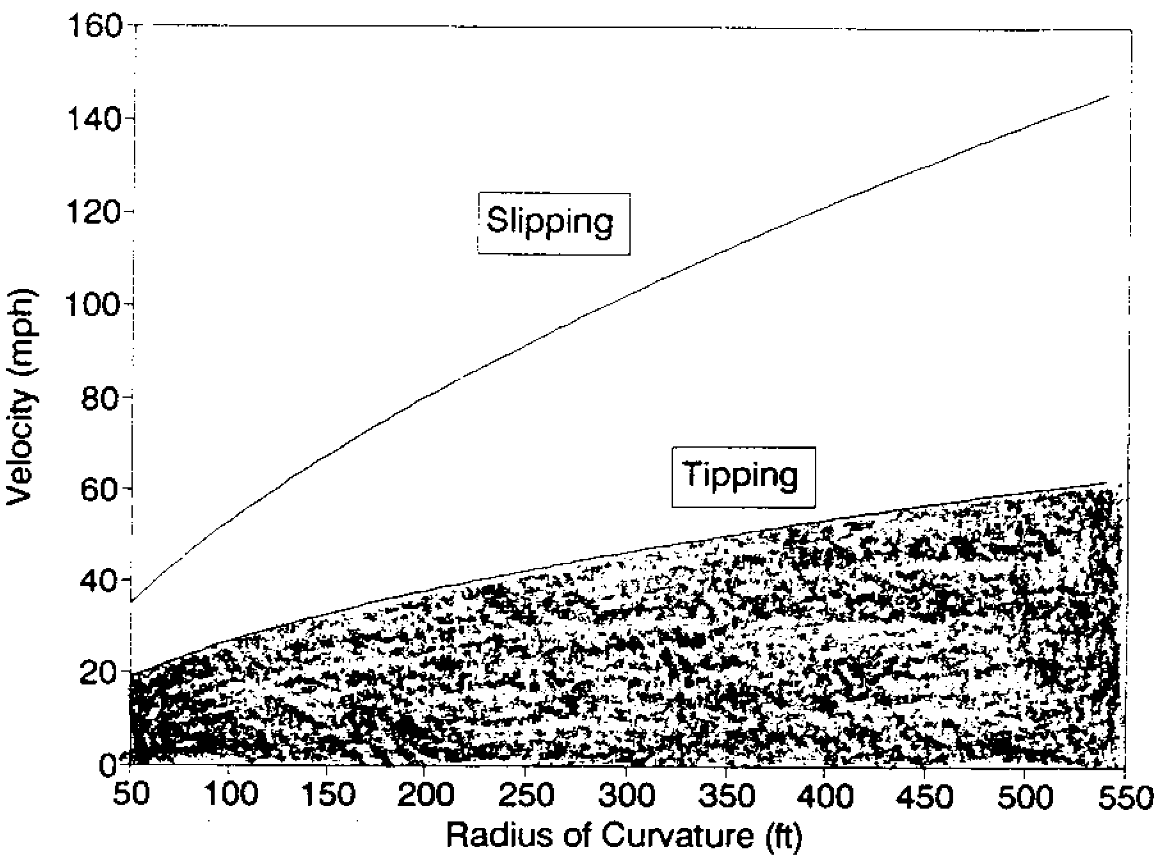


Figure T4: Permissable velocities for various radiuses of curvature

It is appearant from this graph that the trailer will tip before it slides. This scenario will vary depending on the value used for the static coefficient of friction (This value will change with different road conditions i.e. snow, rain, dry, etc). A coefficient of 0.4 was used in the above calculations[15]. The shaded area of the graph represents the safe zone.

Another factor in determining the feasiblility of such renovations is the over-all cost. The economic analysis of the fore-mentioned renovations is presented in Table T2. It is worth noting that the cost of labor for welding was determined for a minimal estimate of 10 hours.

Trailer parts & labor	Cost
78" Axle	89.00
Equalizer	80.00
Double eye spring (2)	102.00
Hubs (2)	72.00
Tires & Rims (2)	212.00
Tires (2)	140.00
Galvanized steel (176')	456.00
Brake System*	350.00
Labor-welding ($10/hr)	360.00
Total	1,861.00

*includes lines, master cylinder, hub, shoes, wheel cylinders

Table T2: Cost analysis for trailer renovations

Harking back to the force analysis on the trailer it becomes evident that this design is impractical. To ensure a safe ride the towing vehicle would have to be driven at absurdly slow speeds according to Fig. T4. The average radiuses of curvature that the trailer will experience ranges from 100 to 150 ft.[16]. This limits the trailer speed to about 20 mph if tipping is to be avoided. To invest the sum of money calculated in Table T2 for renovation of the trailer would be ludicrous considering the uncertain reliability of the design as well as the fact that a brand new trailer would only cost a few hundred dollars more. The inadequate support of the bow on the trailer could result in teetering about the tongue. This increases the probability of capsizing while in transit. The weight distribution of the skimmer shows a majority of the weight to be focussed in the rear (Fig. D1). This could lead to insufficient tongue weight. In light of these adverse predicaments we had no other choice but to disregard the design and pursue other alternatives.

Due to the impending of deadlines accompanied with the unsuccessful renovations, purchasing a new trailer from a dealer was concurred upon. A multitude of trailer catalogues, including price lists, were received via mail from nearby dealerships. After deliberating over numerous designs the field was narrowed down to two trailers. The dealer of each respective trailer was contacted and detailed specifications (spec sheets) were requested. Personal contact with the dealers was also made so that questions could be answered and haggling attempted. After weighing the pros and cons of each trailer it was decided that the GPTT2440TB by Soreland'r would best meet our needs. A picture of this trailer as well as its specifications are presented in Figure T5 and Table T3, respectively.

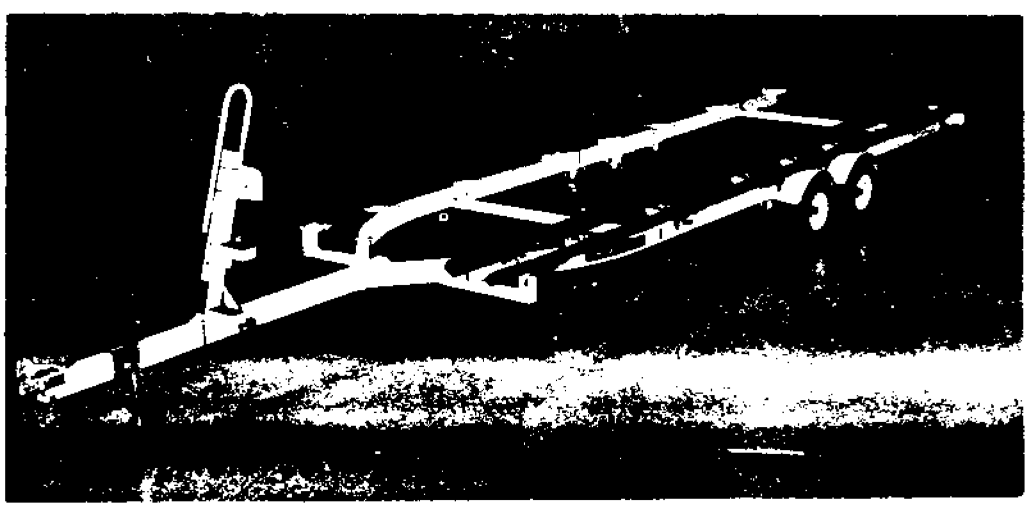


Figure T5: GPTT2440TB Soreland'r

An optional hot-dipped galvanized finish was also seen as a necessity. This price is only $139.00 more than what it would have cost to renovate the original trailer. Unfortunately, due to the bureacratatic policies that must be followed to receive money the trailer was not purchased.

G P T 2 4 4 0 T B
General Data
Boat Length: 24'
Carrying Cap: 4000 lbs
Weight: 970 lbs
Tire size: 20.5x8.0x10D
Wheel Style: Standard
Frame
Size: 3"x5"x12ga.
Width: 73"
Overall Width: 92"
Overall length: 28'3"
Tongue
size: 3"x5"x7ga.
Length (std.): 80"
Coupler size: 2" Class III
Axle
Axle size: 2"x2"x$\frac{1}{4}$"
Hub size: 1 3/8" x 1 1/6"
Bunk
Dimensions: 2"x4"
Length: 20'
Brakes
Drum size: 8$\frac{1}{2}$"

Table T3: Trailer Specifications

Work on a new design has begun but this proposal will most likely be carried into next years Senior Project. A car trailer will be rented to transport SPOSS to the test sight.

SELF PROPELLED OIL SKIMMER SYSTEM
ORIGINAL BUDGET PROPOSAL

DIP VOSS SKIMMER**	$40,000.00
a) Transportaion Cost	
i) Transportation of SPOSS	$ 350.00
ii) Travel Expenses	$ 200.00
VESSEL	
a) Zodiac Pontoons	$ 5,000.00
b) Deck	
c) Assorted Hardware (cleats, Anchor, Line, Navigation Lights, etc.)	$ 1,000.00
PROPULSION SYSTEM	
a) Engine	
i) 1990 85 Hp Johnson**	$ 3,775.00
b) Electric Auxilliary	
OIL OFF-LOADING PUMP AND PLUMBING	$ 1,000.00
A) Generator	$ 800.00
NAVIGATION EQUIPMENT	
a) VHF radio**	$ 350.00
CREW SAFETY EQUIPMENT	
a) USCG type 1 life jacket (3)	$ 200.00
b) Parachute & smoke flares	$ 150.00
CONTROLS	$ 500.00
TRAILER**	$ 3,500.00
TOTAL	$ 9,250.00

SPOSS' UPDATED BUDGET

ITEM	PRICE
Hardware	$ 298.68
Char-Lynn Hydraulic Motor	$ 246.00
Husco 5000 series Valve w/ Spool	$ 522.00
Schroeder Return Filter	$ 150.00
Vescor Reservoir	$ 260.00
Parker Hydraulic Motor	$ 489.30
Moyno Progress Cavity Pump	$1,727.00
Hydra Pumps	$1,850.00
Pontoons	$1,500.00
Kubota Diesel Engine	$1,498.00
Electric Fuel Pump	$1,265.00
Auxillary Alternator	$ 262.00
Boom	$ 129.75
Propulsion System	$ 277.00
Electrical System	$ 131.95
Safety Equipment	$ 373.10
Travel Expenses	$ 200.00
Total	**$11,179.78**

CONCLUSION

Due to inaccessible funds and unforeseen problems encountered when ordering parts the completion of SPOSS at this time was unachievable. Subsidies from various outside companies were not received until early March which delayed the ordering process. This delay also forced the postponing of the assemblage of other dependent components. With the recent arrival of integral systems i.e. the Kubota double hydraulic pump and 85 HP outboard engine a tentative testing date has been set for May 8.

Our chief concern is to test the efficiency of the recovery system. Popcorn, because of its buoyancy and harmless effects on the environment, will be deployed about the testing field as a substitute for oil. A Possible circumstance being considered that may hender testing is controlling the test discharge (popcorn) in a wave environment. The data colleceted from this test will be incorporated into the design of a more efficient SPOSS.

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