BICYCLE POWERED WATER PUMP

by

Charles B. Fly

Submitted in Partial Fulfillment

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ABSTRACT

There is a need for pumping systems in underdeveloped countries that could produce several gallons per minute of water at up to 200 feet of lift. A double acting bicycle powered pump was built and tested to provide an economic answer. The pump utilized a fluidic linkage between the pump pistons to allow the pistons to be oriented vertically. This permitted direct connection of the pistons to the surface through music wire drive cables loaded in tension with no down hole linkages or flexure.

The pump did not achieve design performance due to dissolved air and fluid inertia effects in the fluidic linkage. This led to increased seal drag resulting from flexure of the drive cables during the suction stroke. The pump achieved 65% volumetric efficiency at 24 RPM. It delivered 5 gpm at 55 feet of lift. Estimated efficiency was 40%.

The failure mode was analyzed resulting in a recommended improved design for a single acting pump. This new pump utilizes improved valving and a long stroke rolling diaphragm to reduce cable loads.

A final discussion recommends the development of a bicycle powered cable tool drilling unit to reduce the cost of drilled wells for the application of these pump units.

Thesis Supervisor: Professor David Gordon Wilson

Title: Professor of Mechanical Engineering

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INTRODUCTION

There is a marked need for medium head, low volume pumps in underdeveloped countries. Such pumps, however, must satisfy a number of very stringent constraints due to the environment to which they are exposed. The units must be extremely low cost, ultra-reliable and almost elegantly simple. Poverty stricken cultures that possess virtually no technical skills comprise the environment for these pumps. The pumps must be rugged enough to withstand substantial physical abuse in addition to an abrasive environment. They must be simple enough in construction that local people can be dependably trained for the maintenance and operation of these units.

Cost and the need for fuel, lubricants and maintenance eliminate commercially available gasoline powered pump units. Windmills are generally too expensive as commercial units. And, they are usually too complex and fragile when built as low cost units. A particular problem for low cost units is a mechanism to accommodate high winds, their resultant speeds and loads.

This leads to the simplest power source -- human power. Various schemes have been devised for producing and coupling muscle power. But, by far the most applicable, is the bicycle and the drives derived from it. This is due to the worldwide availability of low cost bicycles. It is also due to the almost unparalleled physiological coupling efficiency of the bicycle mechanism.

Simplicity, low cost, and low maintenance in the pump unit is satisfied by rolling diaphragms. These units have virtually infinite longevity relative to sliding seals such as leather cups.

A test pump was built on the principle of coupling a bicycle derived drive to a double acting rolling diaphragm pump. Previous work had been done on diaphragm pump units using a treadle drive [Ref. 1]. From this work it was observed that the drive required inertia to carry it through the motion extremes and a more efficient body motion, Application of a bicycle with the rear wheel converted to a flywheel satisfied this requirement. The second observation was that linkages and/or flexure of the driving elements at the pump were undesirable. Linkages were found to be undesirable due to complexity, wear points and size. A report for the Agency for International Development by the Battelle Memorial Institute [Ref. 2] indicates wells for hand pumps are being drilled by several groups in underdeveloped countries. This makes small size a desirable feature. There is no practical method of placing a linkage in a drilled hole. While flexure of the drive mechanism around a pulley can be reduced in size to fit in a drilled well, the resultant wear and stress on the drive reduces reliability. The work by Asbell et al. [Ref. 1] discusses the problems that were encountered in their pump due to cable breakage.

On the basis of these observations, it was decided to produce a pump which connected directly to the surface via cables loaded in tension. The cables passed through a straight tubular seal adopted from the work of Asbell et al. [Ref 1]. But, most importantly, the cable underwent no flexure. This left a two fold requirement. One, some means must be provided for returning the pistons during the suction stroke. And, two, some vacuum must be applied to the underside of the diaphragm to keep it from reversing during the suction stroke. These two criteria were satisfied by filling a sealed space below and between the two pistons with water. A surface linkage motion was derived that increased the volume below the pistons during the stroke thereby producing a vacuum that would transfer the water between cylinders and return the pistons during the suction stroke.

This fluid linkage did not work. First, dissolved air

air was released from the water. This necessitated raising the pistons off the surface of the fluid to maintain a vacuum at all times. This caused a sacrifice in stroke. This second scheme failed as well. It was mistakenly assumed, during the analysis of this second scheme that the water vapor would act as a compressible vapor, creating enough pressure difference to transfer the linkage fluid. The rate that water and its vapor reach equilibrium prevented the development of any significant pressure. Also, there was enough air to cause a problem in the initial scheme but insufficient air to transfer the water in the second. Consequently the piston on the suction stroke returned until it struck the surface of the transfer water. Here it slowed momentarily as it accelerated the transfer water. This led to an initial flexure of the drive cables. Once this occurred, the seal friction increased dramatically restricting the piston motion as the drive linkage continued in its motion. The flexure continued to increase during the stroke. At slow speeds, around 15 rpm, the piston would eventually catch up with the linkage. Above this speed, the piston would be caught in its downward motion by the upward movement of the drive linkage.

During the course of testing the pump, these causes for failure were not determined. Thus, finally the pump was tested in its existing state and dismantled. The pump did deliver 5 gpm at 55 feet of lift at an estimated efficiency of 40%.

Following the testing program the cause of the failure was determined. Analysis of these causes led to a redesign of the pump unit. This unit differs in three fundamental points from the test pump.

1. It is a single acting pump. By appropriate linkage balancing at the surface, the power requirement of the operator is stabilized. This reduces the down hole system to one cylinder with only one rolling diaphragm -- resulting in a simpler, cheaper pump unit.
2. The proposed pump uses an improved valve scheme that incorporates the valving directly into the cylinder construction. This reduces losses and complexity.
3. The proposed pump is based on long stroke diaphragms developed by Diaphragm Industries, Inc. of Portsmouth, New Hampshire. Use of these diaphragms reduces the physical diameter of the pump and the cable loads for a given displacement.

These changes result in a pump system of minimum cost and complexity.

This pump is a cost effective means of providing water from wells or other sources for use in underdeveloped countries. However, it must be recognized that it is a part of a larger problem. Specifically, the development of water sources costs more than the pumps. In particular, wells cost more. Hand dug wells to these depths are clearly feasible and are dug all of the time. But often, hard strata are nearly impossible to dig through. And, there is always the danger of cave-ins in such wells. What is required is a low cost method of drilling wells. It is proposed that a technique for coupling a cable tool drilling system to a man driven power source be developed. Design criteria for such a system is included.

Together, this pump and a developed drilling scheme would place the development of water sources into the hands of the people of underdeveloped countries.

REPORT FORMAT

Since there are two pumps described here, it is necessary to isolate the development of ideas from the application in each case. Hence, this thesis is divided into three sections.

The first section develops criteria and general rules applied to the design of each of the pump units. Developments cover the methods used but do not discuss the specific details of either pump. Section II discusses the test pump design, the testing program, and the analysis. Section III discusses the long stroke pump unit applying the developments of Section I to the knowledge gained from Section II.

SECTION I

GENERAL PRINCIPLES

DESIGN CRITERIA AND LIMITATIONS

The overall objective of this project is to develop a cost effective man powered water pump suitable for use in wells. The design criteria for this are:

1). The pump should be a physically effective solution to raising water up to 200 feet under the conditions encountered in underdeveloped regions. As such, the unit must be extremely durable and of such a design that local people can be trained to effect field repairs.

2). The pump and drive should be scaled to one man operation, taking into account that people from underdeveloped countries are generally physically weaker than their counterparts from developed countries.

3). The pump should be efficiently adaptable to various depths within its operating range.

4). To accommodate drilled wells and to simplify construction, installation, and transportation;

the pump should be small. Ideally, the pump should be small enough to fit within at least a 6-5/8 inch drilled hole. This is a standard drilling size used throughout the world, particularly for drilling oil wells. The Battelle Institute report on hand pumps indicates that water wells are being drilled in underdeveloped regions, making this point especially significant [Ref. 2].

5). Construction should be simple, using local materials and standard products where ever possible.

6). The cost of construction and installation should be minimized. One hundred dollars per unit is taken as an upper limit.

7). Threaded fasteners loaded in tension should be kept to a minimum. The Battelle report [Ref. 3] indicates that thread failure due to poor materials and fabrication are a major problem in underdeveloped regions.

8). The pump should be single acting. The primary motivation for this is the reduction in the number of diaphragms used. As the diaphragms are the most expensive single element, representing about one-fourth of the total cost, only one can be justified. Suitable balancing at the surface links the pump to the drive system. A second advantage that results from a single acting pump is the smaller diameter of the unit.

9). Bearings, if any, should be simple, replaceable and operable under abrasive conditions with no lubrication.

10). As many elements as possible should be loaded in tension to reduce weight and cost.

Satisfaction of these ten criteria should result in a pump system acceptable to the underdeveloped countries and suitable for implementation by the AID or similar group, the United Nations, or the countries themselves.

PUMP UNIT

Sealing is clearly the major problem in any pump. Under the conditions encountered in underdeveloped areas, this problem is amplified. Rolling diaphragms are uniquely suited to these pumps in that they eliminate sliding (and wearing) surfaces that are positive sealing. Low cost and very loose tolerances further recommend them for this application. A possible alternative is use of plastic or metal corrugated piping for a bellows pump. The cost of such piping is generally prohibitive, however.

Cables loaded in tension appear to provide the simplest drive system provided there is some method to return the piston on the intake stroke. Compressively loaded elements rapidly become too large and unwieldy even at moderate depths. Torsional driving schemes such as applying forces by rotating the pipe stem become very complicated, particularly for small diameter units. Ideally, the cables should be connected directly to the pistons from the surface with no intervening linkages or bends as around a pulley. This eliminates all wear points (except the seal around the wire) and undue stresses on any component that is in the well. This should increase considerably the reliability of the downhole unit.

Two factors significantly influence such a simple design, however. First, some force must be supplied to return the pistons on the intake stroke. Second, the rolling diaphragm cannot exert any vacuum in order to pull water into the pump chamber. Any vacuum on the working side of the rolling diaphragm will cause the diaphragm to be sucked up into the annulus between the piston and the cylinder, often with catastrophic results. A conventional double ended piston rolling diaphragm assembly creates this reverse vacuum by expanding the small volume trapped between the diaphragms. If the trapped volume is increased by changing the spacing between the two diaphragms, then less vacuum pressure can be tolerated before the diaphragm loses its convolution.

Maintaining a vacuum on the underside of the diaphragm can both return the piston and keep the diaphragm properly loaded. Maintenance of a continuous vacuum is impractical due to leakage. Therefore, the following scheme is proposed as shown in Fig. 1. At the lowest point of travel of the piston, all the volume within the cylinder below the diaphragm is filled with deaerated fluid. As the piston is raised, vapor at the saturation pressure fills the space below the piston, as the vapor pressure of fluid at ambient temperatures is less than 1 psia, a nearly pure vacuum for returning the piston and diaphragm is produced. However, if the piston is left at the bottom of the stroke between uses, there is no vacuum to maintain. Additionally, the vacuum pressure is constant from the point that the piston raises off the surface until it returns. As can be seen in Fig. 1, the piston enters and moves through the water. Thus, judicious design is required in order to avoid excessive fluid drag on the piston. This is not a serious problem, however, in that hollow pistons keep the displacement of the fluid to a minimum.

Sealing around the cable is most simply satisfied by running the cable through a close fitting tube and accepting some minor leakage. However, since the cables run straight to the surface from the pump cylinder, the seal tube can be quite long if necessary. The only restriction on the seal tube length is cost and the drag on the drive cable.

Cable endings, such as shown in Fig. 14, provide a simple attachment scheme that avoids the need for set screws or bending the wire. Very low heat can be used to set the solder without altering the wires' tensile or corrosion properties. Attachments of this type were used in the test program with no failures due to the cable pulling out of the solder joint. By pinning through the connectors as shown in Fig. 1, there need be no threaded joints in the drive train loaded in tension.

The final pump design point, and probably the most significant, is the use of long stroke rolling diaphragms. Diaphragm Industries of Portsmouth, New Hampshire has developed a method of producing rolling diaphragms that have half strokes longer than their diameter. A typical rolling diaphragm has at best a half stroke that approaches its diameter. For a given displacement, use of the Diaphragm Industries' rolling diaphragms will result in significantly less cable loads due to less piston area. The advantages of this in terms of weight, size, and cost are immediately obvious. A typical diaphragm that has been discussed with Diaphragm Industries would have an effective diameter of one inch with a thirteen inch stroke. To the author's knowledge, no other company produces such diaphragms.

DRIVE UNIT

Man powered pumps can be driven by a variety of schemes, utilizing a wide range of human power input modes. Any of these is acceptable, provided it satisfies the following criteria.

1. It utilizes an efficient body motion.
2. Some form of inertia is provided in the system.
3. The drive can be constructed at low cost from available materials and equipment.
4. The drive has minimal or no maintenance and lubrication requirements.
5. Some simple means is provided for either stroke or

speed adjustment to adapt the drive to various operators whose power capacity varies.

Due to its universal availability and excellent physiological coupling, the bicycle and drives derived from it offer a good solution to the drive design problem. Further linkages generally provide simpler stroke adjustment schemes than speed changing systems. This is due to a fewer number of parts in general. It is necessary that the power requirements for a given pump be field adjustable by modifying stroke or speed to accommodate variations in operator capacities.

LANDING AND BALLASTING

It is generally undesirable to place a pump directly on the bottom of a well. The walls of uncased wells gradually cave, filling up the hole. A pump set directly on the bottom of a well (particularly a drilled well, due to close fit) can become stuck to such an extent that the pipe stem can be pulled in two without removing the pump. Thus, the pump must be hung from the surface so that it is a few feet above the bottom of the well. Additionally, if the pump is supported on the bottom, it will gradually settle, necessitating continuous adjustments of the drive cables.

This being the case, the effective weight of the pipe and additional weight, if necessary, is used to offset the load on the cables. A second problem arises in the case of double acting pumps. Since the drive cables will necessarily be off center, there will be a torque applied at the bottom of the pipe. This will cause a swinging of the pipe string, accelerating wall failure if the pump strikes the walls. This also upsets cable adjustments. As it is recommended that only single acting pumps be used for other reasons, these problems are obviated.

SECTION II

TEST UNIT

TEST PUMP UNIT

The test pump unit represents an early approach to vacuum generation that ultimately led to the proposed pump design. The assumed method of operation can be derived from Fig. 6 which is a drawing of the assembled pump cylinder. Two of these cylinders were connected by a rigid hose. The drive linkage was designed to create a higher velocity on the ascending piston (pumping stroke) than the descending piston (suction stroke). This would lift the ascending piston off the fluid surface, creating a vacuum. An implicit assumption in the original design was that while the vapor would tend toward its equilibrium pressure, the time required for the vapor pressure to reach equilibrium would be long compared to the time of the pump stroke. The point was not investigated in the original concept as how fast the vapor reached equilibrium did not matter. The vacuum at less than equilibrium pressure would serve to pull the fluid around from the suction cylinder. It would also serve to intake fluid in the opposite cylinder since the pressure drop through the crossover tube was small.

Shop prints of the test pump unit are given in Appendix A-3. The unit was designed around a 3.5 inch diameter by 3.5 inch tall Bellofram [Ref. 7] rolling diaphragm. The diaphragm had a simple flange that was perforated for six 1/4 inch bolts. All of the design dimensions relating to the diaphragm are derived directly from the Bellofram design manual. Although long stroke diaphragms had been sought before constructing the test pump unit, they had not been found at the time the test pump unit was designed. Thus, the test unit was constructed with conventional rolling diaphragms.

The test pump embodied the concepts of straight connection of the pistons to the surface and of the use of solder connections. There were no failures due to pulling the wire out of the joint. The wires did fail due to flexure. The flexure resulted from the release of dissolved air below the pistons and from fluid inertia effects induced in trying to by-pass the problem. A few minutes operation in such a manner would break the cable off just above the solder joint. No cable failures occured inside the pump unit itself. Nor was any problem encountered that resulted from a reversal of the diaphragm.

The cable and seal assembly were fabricated from 1/4 inch cold rolled steel rod. This assembly worked as planned. Drilling the .082 inch diameter holes was very difficult as the drills were small, long, and flexible. Substitution of tubing for the steel rod would resolve this problem while retaining the benefits of this construction.

The two cylinder units were connected with 30 inches of 1-1/4 inch I.D. steel wound hydraulic hose for convenience. The field unit would have been fabricated from pipe.

The diaphragm retainer was separate from the cable attachment system to avoid undue twisting of the diaphragm.

The check valves were constructed from 2 inch close pipe nipples. A plate with several orifices in it served as a base for a rubber flapper seal. This assembly screwed into two inch by one inch reducers on each end. The parts for this unit are shown in Figures 15 and 16.

DRIVE UNIT AND FLYWHEEL

The constructed drive unit consists of four major components:

1. bicycle
2. bicycle support
3. flywheel
4. drive linkage

The relationship of these parts is shown in Fig. 4.

The bicycle was conventional and used as is. There were only two modifications. The first was to cut out the rear brake mounting bracket on the back stays to clear the flywheel. The second consisted of tightening the head tube bearing until the front forks could not turn. This served to make the bicycle part of the triangulation of the drive framework.

The bicycle was supported on four upright two by fours. Each upright had a bracket, shown as Fig. 20, mounted on top with lag screws. The axles of the bicycle sat directly in these brackets. The wheel nuts secured the bicycle to this structure. This assembly was held together by a single two by four acting as a backbone. This structure can be seen in the drawings in Appendix A-2. The only problem with this structure was splitting of the two by fours where the lag bolts held the axle brackets down. The brackets should be redesigned to bolt through the uprights. These brackets should also be redesigned to permit chain adjustments. This can be accomplished with a longitudinal slot at the front axle.

The flywheel was made by wrapping one half inch strapping around the rear wheel. In all, 37 pounds of strapping was added. Four 1/4 inch bolts on 90° spacing held the strapping in place. The flywheel inertia is calculated to be 58 lbm-ft^2. The free wheel assembly was spot welded to lock it. A 12 tooth sprocket on the rear wheel connected to the 49 tooth pedal sprocket giving a good speed ratio. This assembly was pedaled up to an estimated 80 rpm at the pedals, giving an estimated 325 rpm on the flywheel. No appreciable vibration resulted.

The drive linkage was fabricated of two by fours, one by twos, and a special pedal adapter. The objective of the linkage was to provide a piston motion that would move the ascending piston faster than the descending piston, thereby creating the driving force to return the descending piston and fill the pump cylinder with fluid. Appendix A-7 gives the position, velocity, acceleration and piston differential motion for the linkage at 24 rpm and 3.75 inch stroke.

Two types of bearings were used in the linkage. The first is a wood bearing made of oil soaked hardwood [Fig. 17]. It fitted around the center tube of the bicycle pedal. This permitted the pedal to rotate independently of the linkage, allowing ankling which aids in reducing operator fatigue. A wooden adapter [Fig. 21] fitted the bottom side of the pedal tube and received 1/8 inch by 1/2 inch bent iron straps that held the assembly together. The adapter then bolted to the drive linkage. No problems were encountered with this system. One observation is in order, though. If the pedal center tube is only pressed into its end pieces, it may rotate in the cage instead of inside the wooden bearing. This would rapidly wear out as only a small land of metal would be taking the bearing action. If such a condition were possible, the tube would have to be soldered or welded to the pedal cage.

The second set of bearings consisted of 1/8 inch Schedule 40 steel pipe operating around 1/4 inch bolts. These bearings were fabricated in two forms. The first [Fig. 17] consisted simply of short pieces of pipe pressed into holes drilled in the wood linkages. The second set consisted of a piece of pipe with flanges welded on each side of the pipe [Fig. 18]. These were secured to the main linkage arm by lag screws. The screws only prevented the bearings from sliding along the beam. Each of these bearings was loaded compressively.

Cable adjustment was accomplished with a 12 inch long 1/4 inch threaded rod. Into this rod was soldered the music wire in precisely the same manner as the piston connector. Nuts and washers above and below the cable connector block locked the adjustment rod into place.

Overall, the drive structure proved entirely adequate and rugged. Except for splitting of the wood on the supports and the need for chain adjustments, the system can be used directly in future projects.

TEST SITE AND PLATFORM

Appendix A-2 contains two views of the test site and platform. The platform was simply a four foot tall wooden structure designed to separate the pump and drive. The upper and lower 1/2 inch plywood floors were stiffened to take out the cable loads. The drive unit was mounted on the upper level with the pumps being placed below. The platform was placed next to a free standing swimming pool which was used as a source of water. A 1-1/2 inch hose of 10 feet length connected to the intake valves.

The output valves connected to a surge chamber that smoothed the output so that the unit would work against a constant pressure. The back pressure was established with an adjustable one inch popoff valve.

A small tub was used during the pumping tests as a measure of the volume pumped. It was calibrated by marking a line with a scribe on the side of the tub. The tub was filled to the line with water and weighed. Deducting the weight of the tub gave the volume of water held by the tub.

TEST PROGRAM

The test program sought data for three objectives:

1. Determine the pumping capacity of the pump unit.
2. Determine an approximate value of the pumping efficiency and delineate the relative magnitude of the individual losses.
3. Determine the general acceptability of several design components including the flywheel, bicycle mounting, drive linkage, straight line connection of the drive cables to the pump pistons, the piston return mechanism, the valves and the solder connectors for the drive cables.

Six groups of experiments were conducted to answer the above objectives:

1. Pumping tests.
2. Pumping deceleration tests.
3. Linkage deceleration tests.
4. Seal friction tests.

5. Valve pressure drop tests.

6. Vacuum leakage tests.

The first test satisfied the first objective. It, together with tests two through five supplied efficiency data. The sixth test was done in an effort to find the cause of failure of the pump unit.

The pump was never made to operate successfully. Nevertheless, the pump did deliver more fluid at higher pressures than the pump constructed by the 2.673 group due mainly to better man-machine coupling.

In an effort to find the cause of failure, several tests and analyses were conducted. First, the amount of air below the pistons was determined and found to be insufficient to cause the complete failure. Next, the inlet system was investigated for pressure drop. This was found to be sufficiently low. In the end, a complete iterative analysis of the inlet and crossover flows, including inertia effects, indicated that the pump should work. Implicit in all these analyses was that the vapor did not change state rapidly relative to the time required for the stroke. Finally, pumping and efficiency tests were run at the best conditions possible. After this, the test pump and drive were dismantled.

The pump could be cycled up to about 50 rpm without breaking the drive cables once the solder fillet had been added. However, the cable flexure was so intense that trying to pump at these rates would have been pointless. The pump tests were done at 24 rpm which resulted in an acceptable cable deflection. Initially, the runs were tried at 15 rpm, but this proved extremely uncomfortable to the operator. These tests, conducted at various pressures from 9 to 24 psi, are shown in Table 1.

The procedure for the pumping tests was as follows:

1. The popoff valve was backed off to zero pressure.
2. The rider began pedalling. A clock with a sweep second hand was used by the rider to regulate his speed.
3. Once a steady cadence was established, the pressure was increased via the popoff valve up to the operating pressure. There were still some minor pressure fluctuations (± 2 psi) even with the surge chamber. Hence, the high and low gage indications were centered around the operating pressure.
4. As soon as a steady cadence was again determined, the test began by switching the flow into the

receiver vessel. The time and number of pump strokes to fill the receiver vessel to a pre-calibrated depth were recorded.

This cycle was repeated for each pumping run. The 9 psi run was performed at 15 rpm and was not repeated at 24 rpm.

The next four test groups, as shown above, were conducted in an effort to arrive at an approximate value of the pumping efficiency. These tests also determine the relative magnitude of the losses, pointing out the main areas for improvement.

Two of the tests, the pumping deceleration test and the linkage deceleration test, were conducted in a similar manner. The operator pedalled the pump unit up to 48 rpm. When a steady rate was determined, the operator removed his feet from the pedals while the linkage was at its upper and lower extremes of motion. The unit, driven by the flywheel, slowed to a stop in a number of pump strokes that was dependent upon the rate that energy was being expended. By starting the deceleration at a speed that was twice that of the pumping tests, the average energy expended per stroke should be approximately the energy dissipated per stroke at the operating speed. This tacitly assumes a linear relationship between energy expenditure and speed. The accuracy of this experiment, due to the flexure of the drive cables, is such that any improvements in the efficiency analysis would probably be unjustified.

The pumping deceleration test took place as described above. The pump was connected to pump water just as it had been during the pumping tests. When the operator removed his feet from the pedals, the pump output was transferred to a receiver vessel. When the pedals stopped, the flow was diverted. The flow continued for a few seconds after the pedals stopped before the popoff valve shut off the flow. This extra flow came from the discharge of the surge chamber. The volume pumped was measured and recorded. Due to the discharging of the surge chamber, the pressure dropped from its operating level as the unit slowed down. It was generally observed that the pressure dropped to roughly 1/3 of its initial value as the pump slowed. Thus, the average pressure is estimated at 2/3 of the operating pressure. This is certainly subject to a significant amount of error. This value is probably no more than 20% accurate. The fluid volume is probably 5% accurate. While this flow from the discharge of the surge chamber affected both the pressure and volume data, attempts to operate the unit without the surge chamber resulted in totally unreadable gage variations. A glycerine filled gage was tried to no avail. A gage dampener was not tried. The results of these tests are recorded in Table 2. The operating pressure was set by adjusting the popoff valve in from zero as was done in the pumping tests. The operator adjusted his speed with a clock as in the pumping tests. The speed was double checked by timing the last 20 strokes before the operator removed his feet from the pedals. This also served as a countdown for the transfer of the hose to the receiver vessel and for the operator to remove his feet from the pedals.

The second series of deceleration tests were conducted to determine the losses due to the linkage. For this series of tests the diaphragms and the transfer fluid were removed from the cylinders. The seals were unscrewed from the top of the cylinders and slid up on the drive cables out of the way. Thus, the pistons simply hung inside the cylinders with the wire coming out through the threaded seal hole. This constrained the linkage to follow the same motions as in pumping. Several decelerations from 48 rpm were run. The speed was simply controlled by the operator with the clock as the load was light. The operator had no trouble maintaining a very steady cadence. An average of 20 pump strokes was required to stop the machine. This is also recorded in Table 2.

The seal friction is quite reasonable, averaging 7 lbf. This appears to be independent of speed over the range encountered in operating the pumps. The procedure for measuring these values was to secure the seal element pulling the wire through the seal with a scale. The wire was attached to one of the pistons below the seal in order to keep the wire straight. The friction increased sharply if the cable were flexed in any manner. If the piston began swinging, the cable could not be pulled through the seal with the scale. As the piston swung back and forth, the wire could be intermittently pulled through the seal at high load each time the piston passed directly below it. This fact accounts for part of the failure of the pump unit, as will be discussed in the analysis section. Steady, smooth tests that did not swing the piston resulted in 7 pound average readings. The results did not change when the seal assembly was under water.

The valve was tested in an effort to locate the cause of the failure of the pistons to descend. If the pressure drop in the inlet system were too high, this could have accounted for the problem. Both the valve and the inlet plumbing were tested. The valve is reported separately as it might be used in subsequent designs. The equation for the valve pressure drop is

$$P = 0.33 \; Q^{0.8}$$

P - pressure drop - psi

Q - flow rate - gpm

The inlet plumbing test was not recorded, but was simply run to verify that the equation presented in Appendix A-5 did adequately reflect the inlet pressure drop. The inlet pressure drop due to friction ranged from 2 to 6 psi over the flow range up to 20 gpm. The flow rate, however, was only visually estimated by the distance it traveled after it left the plumbing.

The final test was a measure of the vacuum holding ability of the pump units. Each piston was raised up to within approximately 1/2 inch of its cylinder top. This created the maximum vacuum possible with the system. Over a period of four days, this condition was maintained. The cable tension was tested by "plucking" the cables twice daily during the four day period. No detectible change in pitch occurred indicating the unit did not leak air at any significant rate.

ANALYSIS

Following the test program and dismantling of the pump and drive units, it was discovered that the pump should not have worked at all with the pistons raised. The reason for this is the extremely rapid rate at which a fluid and vapor system will seek equilibrium. The effect of the rate of equilibriation being so high is that very little pressure difference is generated between the vapor and fluid in either condensation or vaporization. As a result, no significant pressure was developed to accelerate and transfer the fluid below the pistons. Appendix A-6 shows the derivation of the pressure drop in evaporation or condensation. At the point that the bottom of Piston A strikes the surface of the fluid, it begins accelerating the fluid. When the annulus between the cylinder and piston below the diaphragm fills with fluid the piston can readily transfer the fluid to the opposite side. However, in accelerating the fluid, the piston momentarily lags the linkage, causing the initial flexure of the drive cable. As was noted in the seal tests, any deviation from straight motion dramatically increased the seal friction. For most of the remainder of the stroke, the seal friction retards the motion of the piston limiting transfer of the linkage fluid and limiting the intake of fluid into the cylinder. This entire sequence of events is adequately supported by the fact that the cable flexure occurred also with the plumbing disconnected. In this form, no loads were placed on the working side of the diaphragm. Appendix A-5 presents calculations showing the inlet and transfer losses, demonstrating the low value of these pressure drops at the operating speed. While the inlet losses are rather high, they are not significantly high enough to cause the failure of the piston motion. The entire inlet plumbing was flow tested confirming the overall friction drop but no adequately simple system was devised to test the acceleration component. However, in light of the agreement on friction losses, it is reasonable to assume the inertia load is correct.

The entire problem with the test pump could have been solved by using deaerated water. However, it did not seem feasible, due to the construction of the pump unit, to get deaerated water into the pump without entraining air both in solution and as pockets inside the pumps. This problem was bypassed for the loading with regular water since the pumps could be filled and sealed totally under water in the swimming pool.

If a greater spacing between the drive and the pump had existed, thereby limiting the initial flexure upon acceleration of the fluid mass, it is entirely possible that the seal friction would not have risen so sharply and the pump could have been made to function. Nevertheless, this mechanism is clearly undesirable even with longer cables as the shock on the cables would accelerate their breakage.

The second series of tests were aimed at determining an approximate value for the overall efficiency of the pumping unit and to isolate the major losses. The major losses in the test unit occurred in the linkage and seals.

By operating the pump unit, and later the linkage, at roughly twice the speed of the pumping tests, it is possible to obtain an approximate value of the energy expended per stroke at the speed of the pumping tests. This is done by dividing the total energy available less that expended to pumping fluid by the number of strokes required to stop the machine. This procedure tacitly assumes a linear relationship between losses and speed, while obviously, fluid losses vary as the square of the speed. However, in light of the problems encountered in pumping; this is sufficiently accurate. Table 2 shows the series of efficiency runs. Here, the pump was operated between 45 and 48 rpm and then permitted to stop, using the energy of the flywheel. This procedure has been described in the test section. The number of strokes to stoppage, the volume of fluid pumped, and the operating pressure under power were recorded. From the speed, the energy of the flywheel can be determined. The inertia of the flywheel is 58 $lbm\text{-}ft^2$. The energy for each run is shown in Table 2.

As was discussed in the test section, the pressure dropped during the slowdown of the pumping unit. Thus, part of the flow obtained during the stoppage came from the surge chamber. The final pressure in the surge chamber is estimated to be 1/3 of the initial pressure. Table 3 presents the volume differences for an isothermal expansion of the gas in the surge chamber from the initial pressure to 1/3 of that pressure. These volumes are deducted from the delivered volume in Table 2 resulting in the corrected volumes shown. The work delivered is the product of the corrected volume and the estimated average pressure. The remaining flywheel energy must be expended in various losses. The average loss per pump stroke is presented in the final column. The overall average loss is 52 ft-lbf/stroke.

The average linkage loss as derived from the linkage stoppage test is 17 ft-lbf/stroke based on 48 rpm initial speed and 20 strokes to stoppage.

The nominal seal friction accounts for 4 ft-lbf/stroke when no increase in drag occurs due to bending. Under the conditions encountered with the test pump, the seal friction clearly accounted for the major part of the remaining losses. By the analysis of the transfer fluid presented in Appendix A-5 the transfer fluid accounts for approximately .5 ft-lbf/stroke. The intake losses amount to approximately 4 ft-lbf/stroke.

The pumping data is presented in Table 1. The first three columns -- pressure, number of strokes and time -- are the source data. The remaining information is derived. The % filling column is based on the theoretical stroke volume of 39.2 in^3/stroke which requires 67.1 strokes to fill the 11.4 gallon receiver vessel. The net work is the delivered volume times the operating pressure. The energy per stroke is the key value from this table related to efficiency. The fact that the deceleration tests result in lower delivered work per stroke should cause an over estimate of the losses occurring at the operating speed. Table 4 presents the efficiencies resulting from this analysis for each operating pressure. The pump test values were averaged for this table. Significantly higher efficiencies could result for this pump in the event that the initial flexure did not increase the seal friction. But the overall process is clearly undesirable.

A final note on the operation of vapor return pumps is the impact of dissolved air on the system. Dissolved air tends to reduce the internal pressure of the transfer system while simultaneously generating part of the gas compression and expansion pressure difference initially intended to transfer the fluid between cylinders. A drawback to the presence of air is that the pistons must always be operated such that there is sufficient volume within the cylinder system to keep the air at sufficiently low pressures. The only practical way to do this is to use part of the diaphragm stroke as a preload cycle. This also is undesirable.

Based upon this analysis, then the following recommendations and observations result.

1. The vapor pressure vacuum system can return the piston under the proper design conditions. Such conditions would seek to reduce or eliminate the forces imposed upon the piston when it encounters the fluid.
2. The fluid should be deaerated or have a very low

capacity for dissolved air. Correspondingly, the pump should be designed so that all the air below the piston can be purged.

3. The inlet losses should be minimized. In the test pump fairly high losses were encountered due to the need to draw water from the swimming pool to pump through roughly 12 feet of hose and pipe.

4. The valves should be equipped with better flappers to reduce or eliminate back flow. Alternatively, another valve could be designed or procured.

5. The seals are entirely adequate provided no cable flexure occurs. This requires a continuous tensile load on the cable. The seal assemblies should be fabricated from small bore tubing to eliminate the drilling operation.

6. The flywheel mass can be reduced to at least half of its 58 $lbm\text{-}ft^2$ value without adversely affecting operation.

7. The solder joint system for the drive cables worked extremely well and is recommended as it reduces stresses in the cable to a minimum.

8. The linkage proved entirely adequate. Some reduc-

tion in linkage drag can be obtained by separating the members from each other with washers. The linkage members slid against each other during operation. Spacing them apart should eliminate the friction that results.

9. The bearing assembly at the pedal, to permit ankling, is considered very important in that it reduces operator strain, just as it does in regular bicycling. Constraining the pedal to follow the motion of some part of the linkage, as would occur by simply bolting the pedal to a linkage element, particularly should be avoided.

TABLE 1

PUMPING TESTS

Working Pressure - PSI	Pump Strokes	Time - Minutes	Pedal Speed RPM	Flow Rate GPM	Percent Filling	Power - ft-lbf/min.	Power Hp	Work/Stroke ft-lbf
9	79	2.45	16	4.6	85	810	0.024	25
12	84	2.72	15	4.2	80	970	0.027	31
12	86	2.75	16	4.1	78	960	0.029	31
12	96	2.00	24	5.7	70	1,310	0.040	27
12	100	2.08	24	5.5	67	1,260	0.038	26
15	106	2.23	24	5.1	63	1,470	0.045	31
15	101	2.13	24	5.3	66	1,540	0.047	33
18	100	2.12	24	5.4	67	1,860	0.056	39
18	106	2.17	24	5.2	63	1,820	0.055	37
21	102	2.13	24	5.3	66	2,160	0.056	45
21	114	1.98	29	5.8	59	2,320	0.070	35
24	107	2.20	24	5.2	63	2,390	0.072	49
24	107	2.20	24	5.2	63	2,390	0.072	49

67.1 strokes required to fill 11.4 gallon receiver vessel at 100% volume efficiency. Theoretical pump displacement = 39.2 in^3.

TABLE 2

PUMPING DECELERATION TESTS

Working Pressure PSIG	Time for 20 Strokes	Speed RPM	Strokes to stop	Flywheel energy ft-lbf	Estimated Average Pressure	Volume - cu. in.	Surge Volume - cu. in.	Corrected volume - cu. in.	Work Delivered ft-lbf	Percent net Efficiency	Average Work Delivered ft-lbf/Stroke
9	12.5	48	6.2	380	6	150	34	116	58	15	9
15	12.5	48	6.0	380	10	142	40	102	85	22	14
18	13.3	45	5.0	330	12	132	42	90	90	27	18
21	13.9	43	4.2	300	14	115	43	72	84	28	20
24	13.6	44	4.0	320	16	102	43	59	79	25	20

LINKAGE DECELERATION TESTS

The linkage averaged 20 strokes to stop from 48 rpm.

Average loss per stroke is 19 ft-lbf/stroke.

TABLE 3

SURGE CHAMBER VOLUME CHANGES

Initial Pressure PSIG	Initial Volume cu. in.	Final Pressure PSIG	Final Volume cu. in.	Differential Volume cu. in.
9	100	3	134	34
12	89	4	127	38
15	80	5	120	40
18	72	6	114	42
21	66	7	109	43
24	61	8	104	43

ISOTHERMAL EXPANSION $PV = CONST$

$P_o = 14.7$ psia

$V_o = 161$ cu. in.

Surge chamber consists of 48" of 2 inch Schedule 40 pipe.

TABLE 4

ESTIMATED PUMPING EFFICIENCIES

Working Pressure PSIG	Total Energy per Stroke ft-lbf	Pumping Energy per Stroke ft-lbf	Losses per Stroke ft-lbf	Efficiency Percent
15	81	32	49	40
18	87	38	49	44
21	92	40	52	43
24	109	49	60	45

NOTE: 9 psi run deleted due to speed of 16 rpm.

12 psi run deleted due to error in recording deceleration results.

SECTION III

PROPOSED UNITS

PROPOSED PUMP REDESIGN

The cause of failure of the test pump stemmed from two original problems. First, dissolved air came out of solution eliminating the vacuum that would have returned the pistons. Second, in an effort to hold vacuum the pistons were raised by shortening the stroke. In this case, when the piston struck the surface of the water, the piston slowed as it accelerated the transfer fluid. This led to cable flexure and increased seal drag.

The original concept of returning the pistons to the surface of the fluid so that no permanent vacuum exists is valid. What is required is elimination of dissolved air and significant fluid forces acting on the piston. The air can be eliminated by two methods. One, deaerate the fluid below the piston. Two, use a fluid that has very low air absorption. Boiled water is a simple solution provided the water can be transferred to the pump without reabsorption. Oils hold less air than water but the author has no data on how much they do hold.

Provided such a fluid is supplied under the pistons, the pistons will return readily. Fig. 1 shows a proposed design for a single acting pump. The piston is shown at midstroke and the fluid below the piston is illustrated. The piston is hollow which reduces fluid displacement to a minimum. The only fluid force on the piston is simple fluid resistance due to the motion of the piston walls through the fluid. For sufficiently inviscid fluids, including oils, below the piston, the inlet losses should exceed the vacuum before the piston drag becomes too great. At the end of a pumping session the piston is simply left at the bottom of the cylinder with no vacuum below the piston. Guide vanes keep the piston from cocking in the cylinder.

The proposed pump uses one of the Diaphragm Industries' long stroke diaphragms. This reduces the pump diameter to as small as practical or necessary. This pump could be used from 25 feet to 100 feet. The stroke would vary from 13 inches to 3.25 inches respectively for a fluid output of 0.07 horsepower. At 70% efficiency this would require 0.10 horsepower input. The pressure rating of 50 psi restricts the maximum depth to 115 feet.

The intake valve is built into the top of the cylinder. It consists of a flexible rubber sleeve that covers ports drilled around the periphery of the cylinder. Spring loaded fingers as part of the mounting bracket at the top of the sleeve should help seat the valve. The discharge valve is screwed into the top of the cylinder. The seal is contained in the center of the discharge valve. A reducer connects the pump to the discharge line. The drive cable is inside the discharge pipe where it is protected. The flange for the diaphragm is not load bearing. The solder connector is pinned to the piston rather than threaded into the piston. The reason for this and the concern for the flange bolts arises from a statement in the Battelle report [Ref. 3] indicating that very poor thread quality in bolts can be expect ed. Consequently, as many as possible threaded components loaded in tension should be eliminated. To keep the flange bolt loads down, any ballast weight would be applied around the drop pipe. This also reduces the depth of water that the pump needs.

Appendix A-4 contains a letter from Diaphragm Industries stating the feasibility of the diaphragm and its estimated cost. Estimating twenty eight dollars for the diaphragm and fifteen dollars for the bicycle, this assembly could easily fit within one hundred dollars total cost.

Fig. 2 shows schematically the scheme to adapt the bicycle drive to a single acting pump. If the pump linkage is connected to the left bicycle pedal, then the left foot must press with one half the effective pump load during the power stroke. And the right foot must press with one half the effective pump load during the suction stroke. Thus, the rider sees a continuous load. Flywheel inertia is still required, but only about 20 pounds of weight is required on the rim of the bicycle wheel.

This design should provide an extremely cost effective man powered water pump for medium lifts. Conversations with Diaphragm Industries have indicated that it could be possible to extend the capacity of the diaphragms to 100 psi. This would extend the utility of the pump unit.

PROPOSED WELL DRILLING SYSTEM

The Battelle report [Ref. 2] has indicated that the AID and other groups are or have in the past drilled water wells for hand pumps. It also reported a cost of $0.26 per inch diameter per foot of well. The proposed pump requires a 4-1/2 inch hole. For 100 feet this hole costs $117. This cost is quite old and could be low by as much as a factor of 2. This cost alone significantly exceeds the cost of the pump. As such, this fact will seriously affect the dissemination and use of these pumps. A reduction in the cost of drilled wells would greatly enhance the utility of these systems.

One possible and recommended system is to develop a bicycle powered well drilling system. Such a system would necessarily be simple. This would place the development of water sources in the hands of the people that would use it.

Fig. 3 shows schematically how such a system might be configured. The principle of operation is that of a cable tool drilling unit. The tool is suspended from the end of a drilling line. The line is rapidly raised and lowered. Once the proper operating conditions are established, the tool, which weighs 200 to 400 pounds, is springing on the end of the drilling line. This springing causes a very sharp bounce of the tool on the bottom of the hole. However, already having the cable tensioned quickly pulls the tool from the work face preventing sticking. The cutting end of the tool is fashioned in a manner to force the tool to rotate. Rotation of the tool causes a straight round hole to be drilled. The oscillations of the bit keep the cuttings suspended in water at the bottom of the hole. Depending upon conditions, six inches to a few feet can be drilled before the drilling tool must be pulled and the cuttings removed.

The bicycle is used to drive a crank and flywheel. The crank needs to rotate between 20 and 40 rpm in order to set up the proper bouncing reaction of the drilling tool. The walking beam should have a number of stroke adjustments as the frequency is partially dependent upon stroke. Flywheel energy storage of about 5,000 ft-lbf is required. This is a four foot diameter by three inch thick flywheel rotating at 120 rpm. This could be connected to the crank by bicycle chains and sprockets.

The feed spool should also be driven by a bicycle. However, it is only used intermittently during removal of the tool or bailer. During drilling, line from the feed spool is gradually let out as the drilling progresses.

This drilling system could cut from a few feet per day up to perhaps 15 feet per day in soft formations. Only two people would be required to drill a well.

Such a system could probably be built for $500 including a trailer in which to haul the equipment (especially the flywheel). Implementation and introduction of a system like this would place extensive water development power in the hands of the people of underdeveloped countries. Their own initiative and ingenuity would take care of the rest.

CONCLUSIONS

The work described in this paper has led to a pump design that resolves all the problems with piston return that were encountered. Use of long stroke diaphragms, redesign and integration of the valving and conversion to a single acting pump has produced a very efficient cost effective pump design. It is recommended that this design be built and tested. Implementation of pumps along these lines could resolve much of the water problem in underdeveloped regions.

These pumps are only part of the solution of the problem, however. Development of the proposed drilling system would have substantial impact on the overall problem of water development in underdeveloped regions. In particular, this device would place the initiative and capability for the water development with the people who would use the water. Such a situation would increase the acceptability of the pumps and systems. It would also hasten development as the people realized the capacity to create their own water supply.

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5. Idel'chik, I. E. - ibid, p. 266.

6. Schrage, Robert W., A Theoretical Study of Interphase Mass Transfer, 1953, Columbia University, New York. Chapters 1 and 2.

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APPENDICES

APPENDIX A-1

PROPOSED PUMP AND DRILLING SYSTEMS

FIGURE 1 - PROPOSED PUMP - SINGLE ACTING

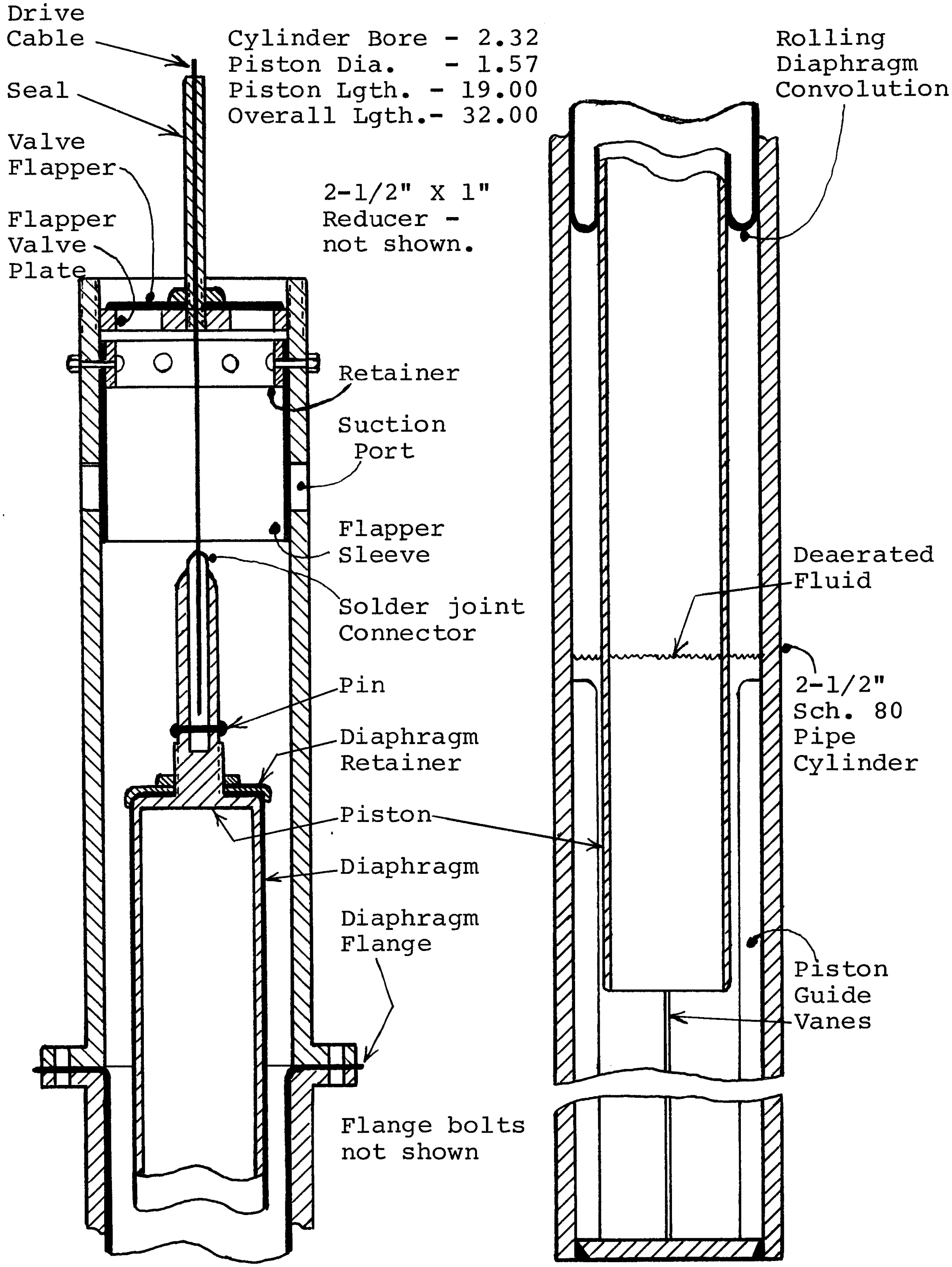
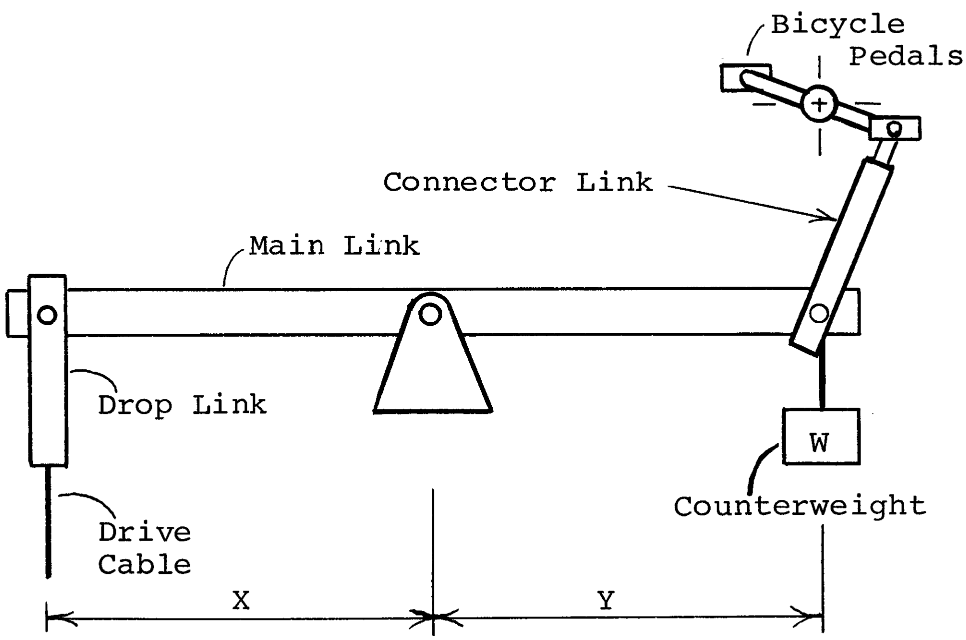


FIGURE 2

SINGLE ACTING LINKAGE



Cable load has 2 components:

P = Pumping load

V = Vacuum load

Counterweight mass W is

$$gW = \frac{X}{Y}\left(\frac{1}{2}P + V\right)$$

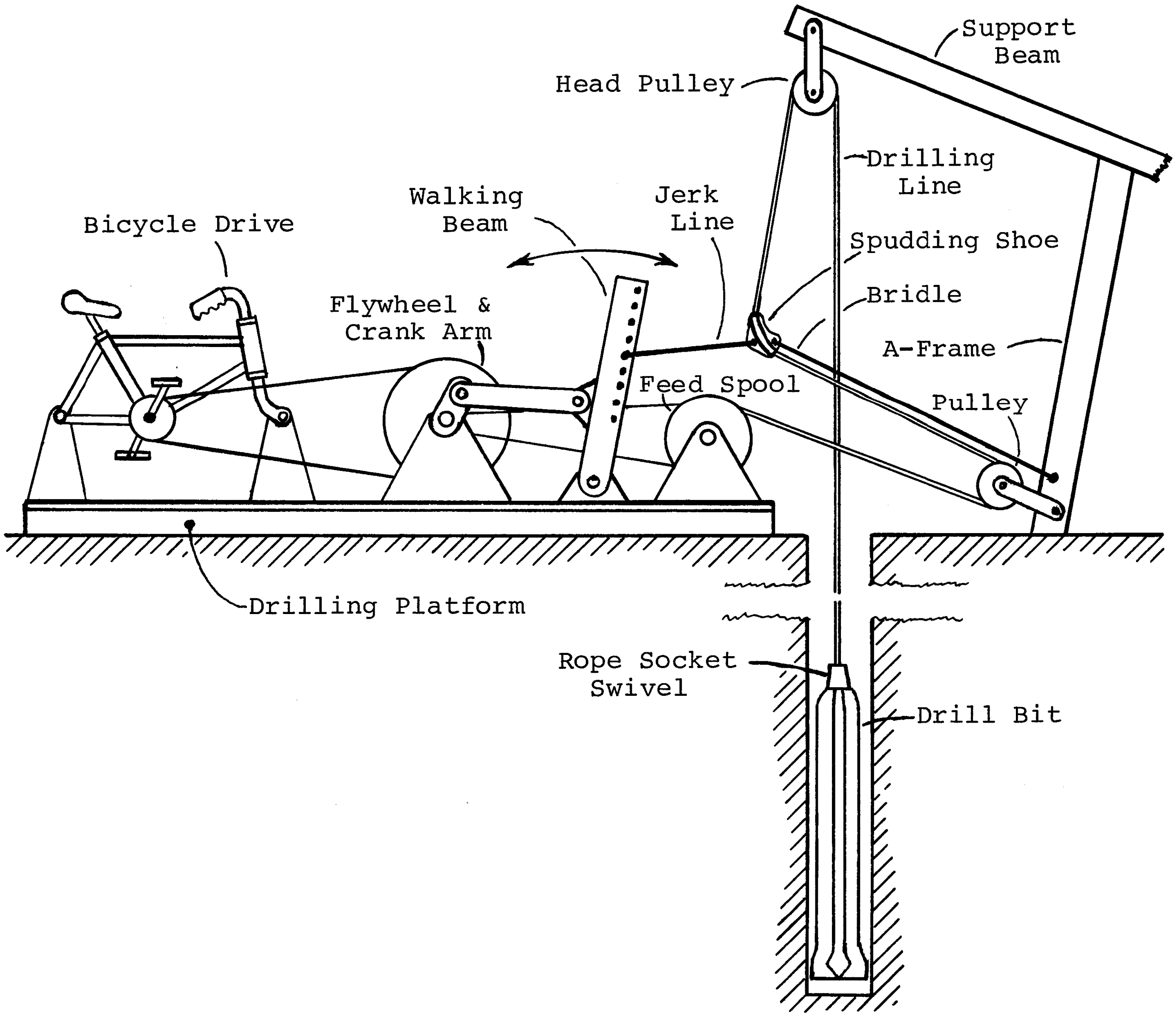


FIGURE 3 - BICYCLE DRIVEN CHURN DRILLING SYSTEM

APPENDIX A-2

TEST SITE

FIGURE 4

TEST SITE AND PLATFORM - SIDE VIEW

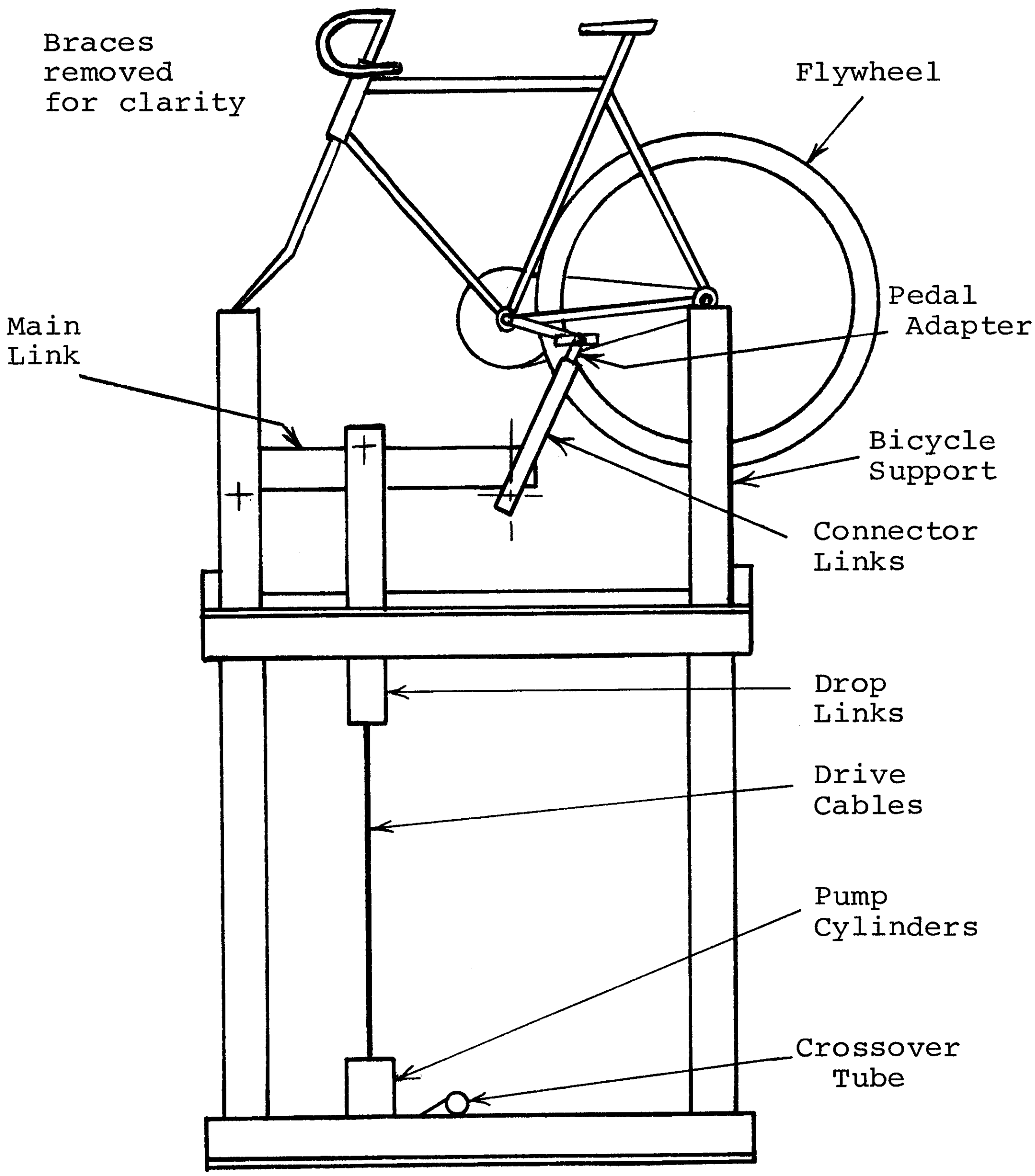
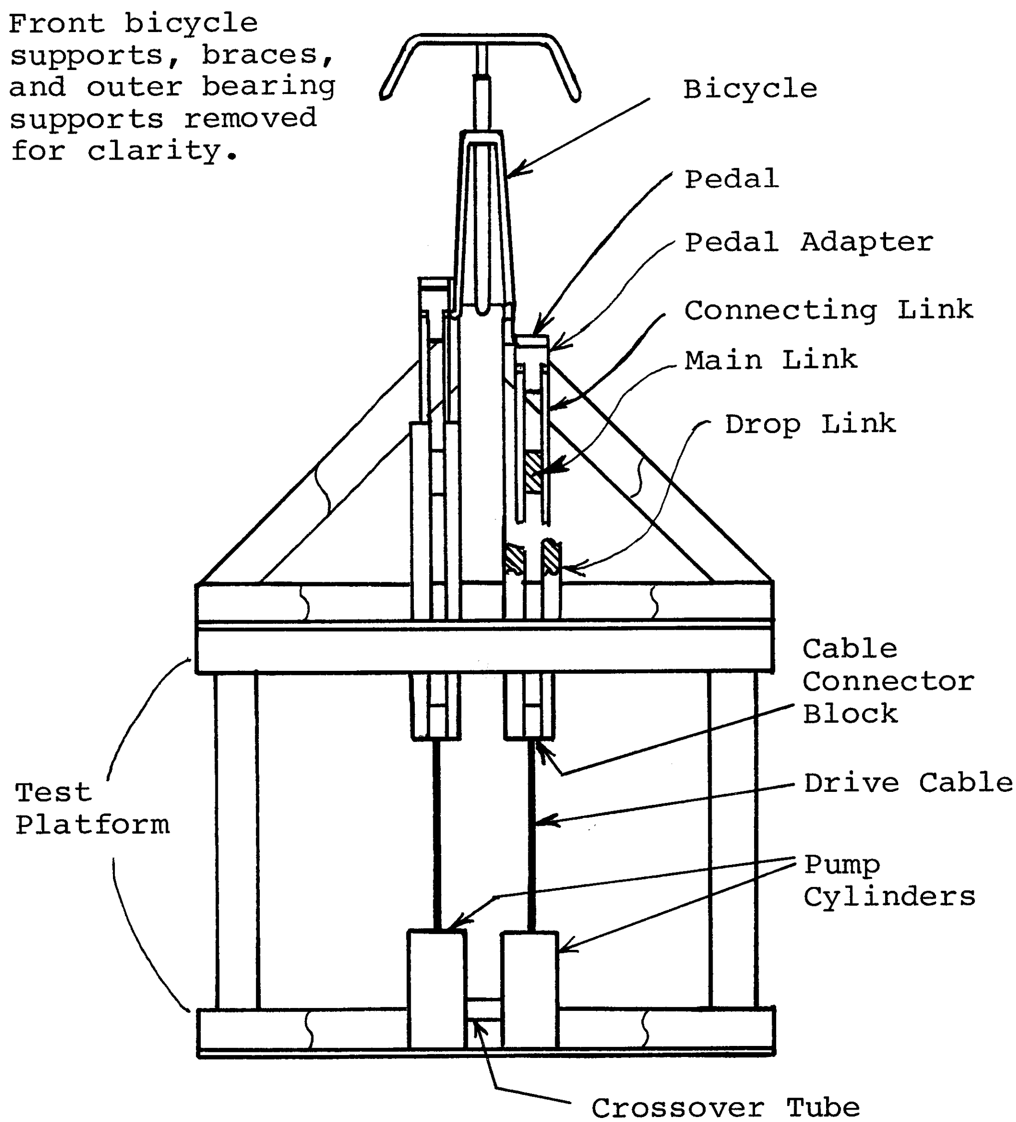


FIGURE 5

TEST SITE AND PLATFORM - FRONT VIEW



APPENDIX A-3

TEST PUMP DRAWINGS

FIGURE 6

ASSEMBLED PUMP UNIT

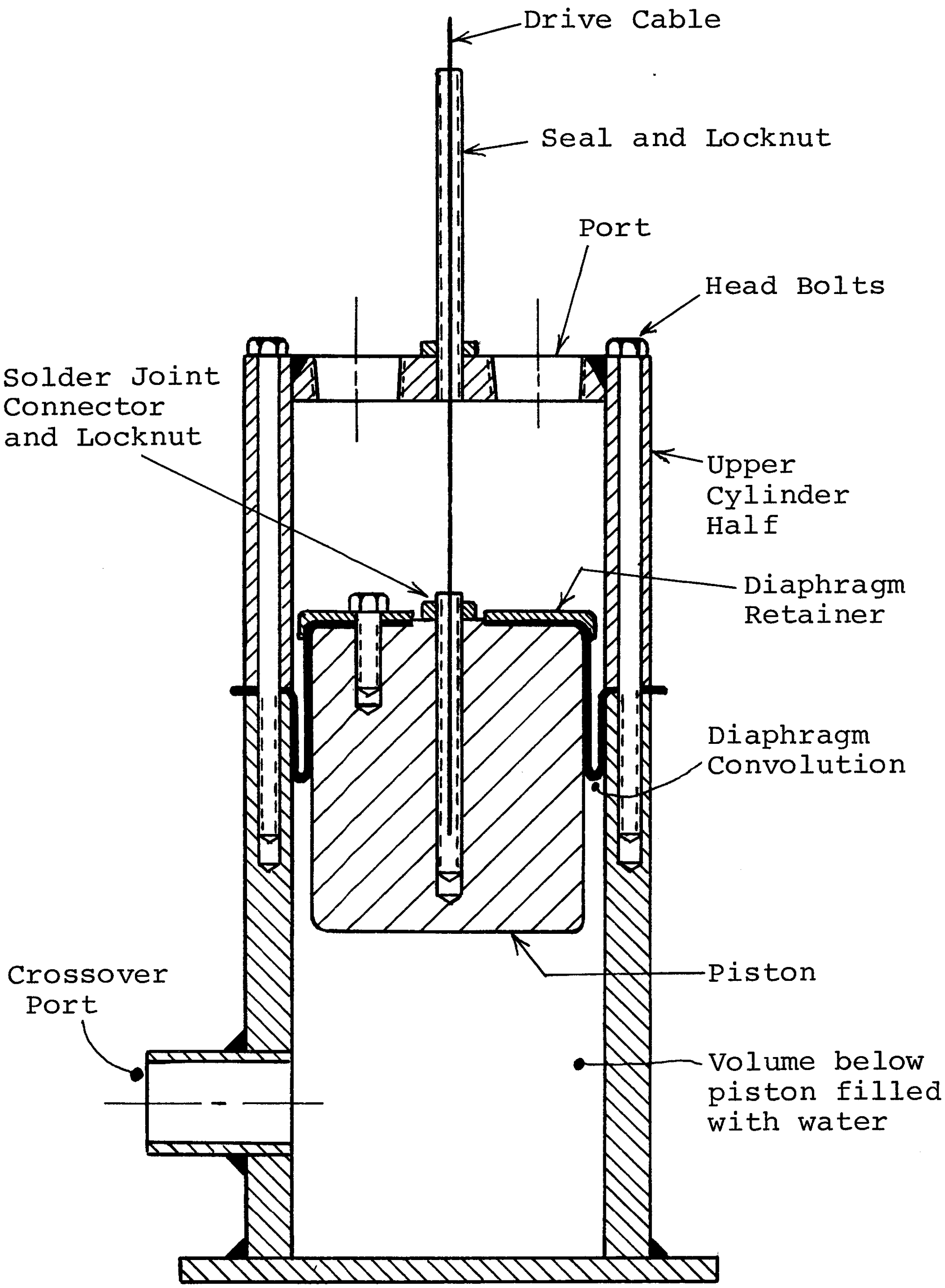


FIGURE 7

UPPER CYLINDER HALF

SIDE VIEW

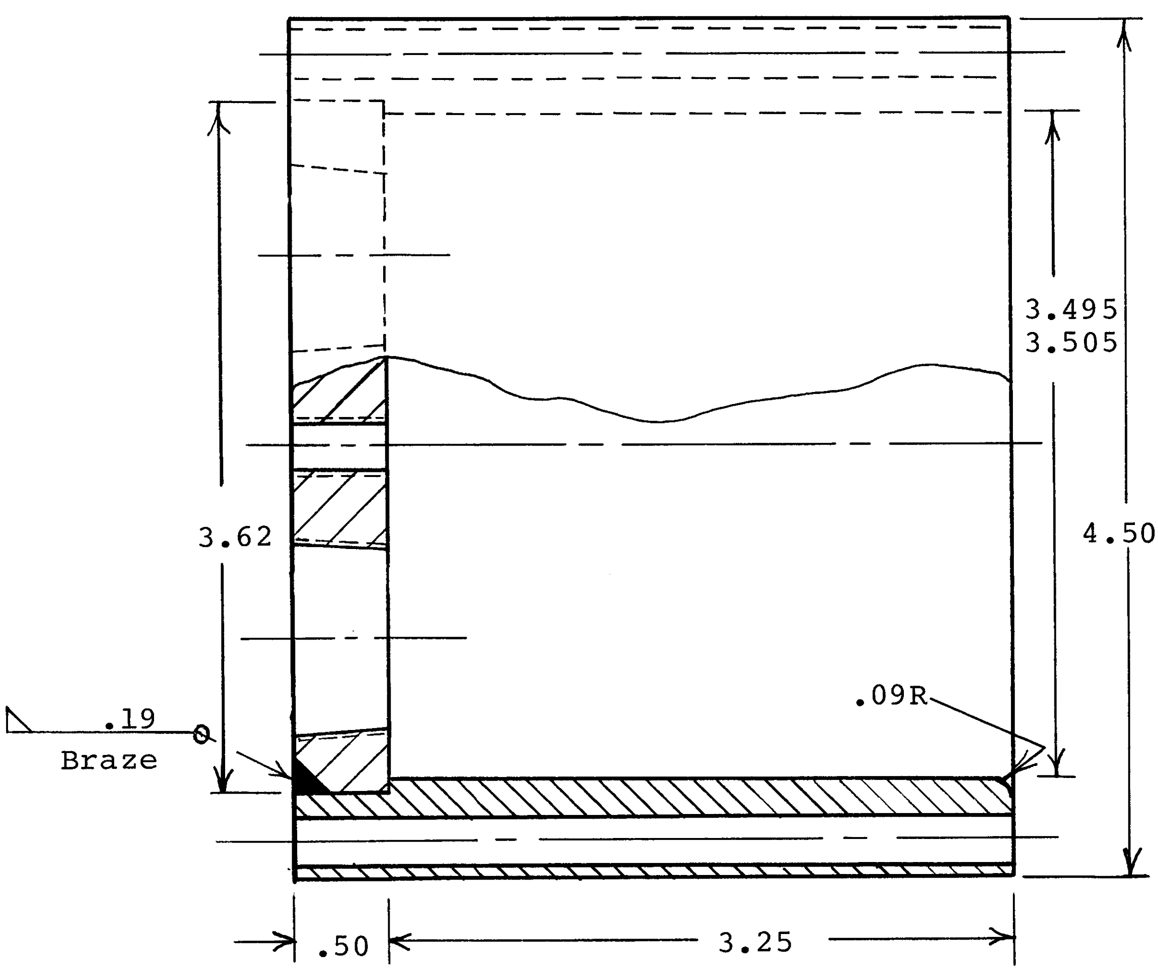
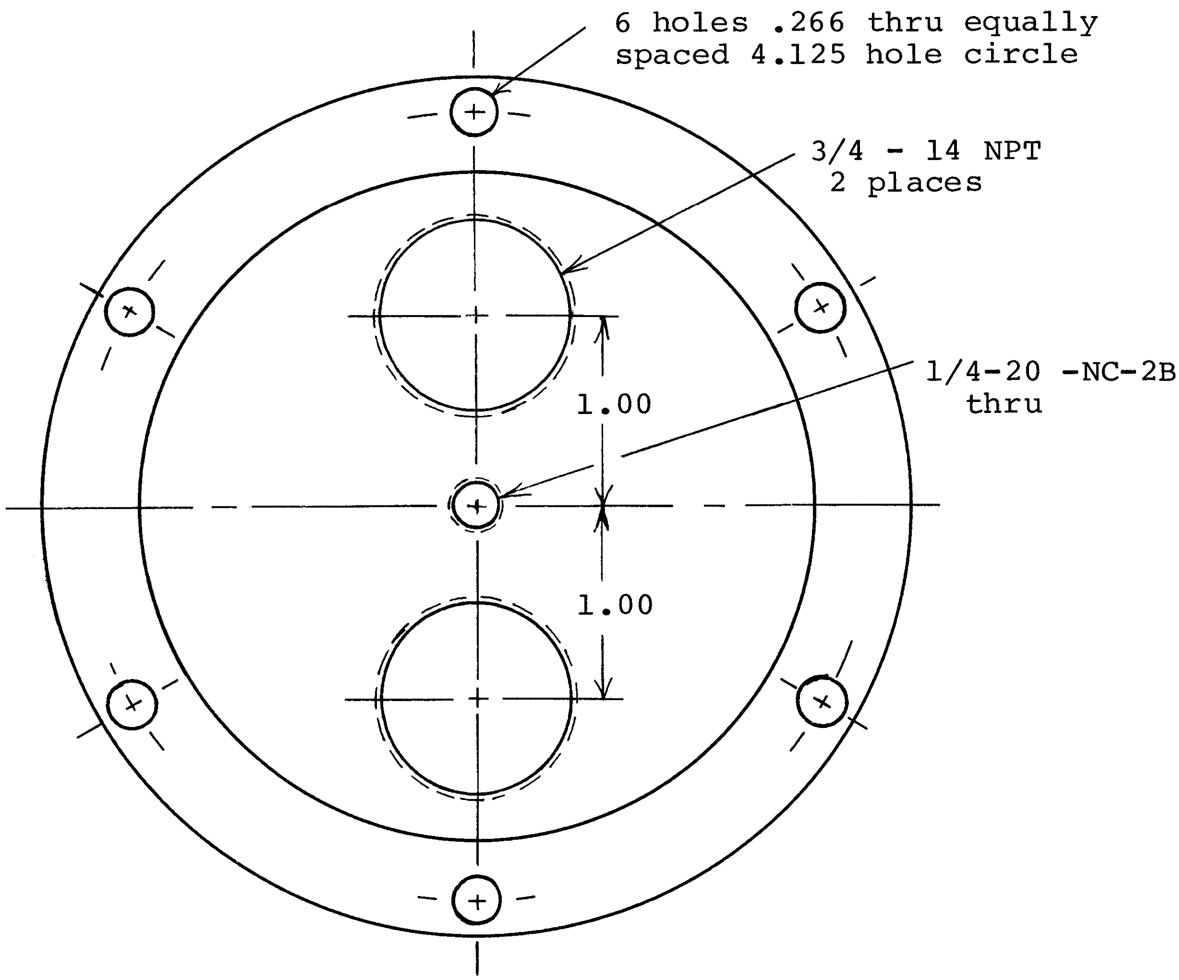


FIGURE 8

UPPER CYLINDER HALF

END VIEW



NOTE: Add all holes after weld.

MAT'L: Cast iron

FIGURE 9

PUMP CYLINDER LOWER HALF

SIDE VIEW

MAT'L: Cyl. - Cast iron

Base plate - 1020 HR

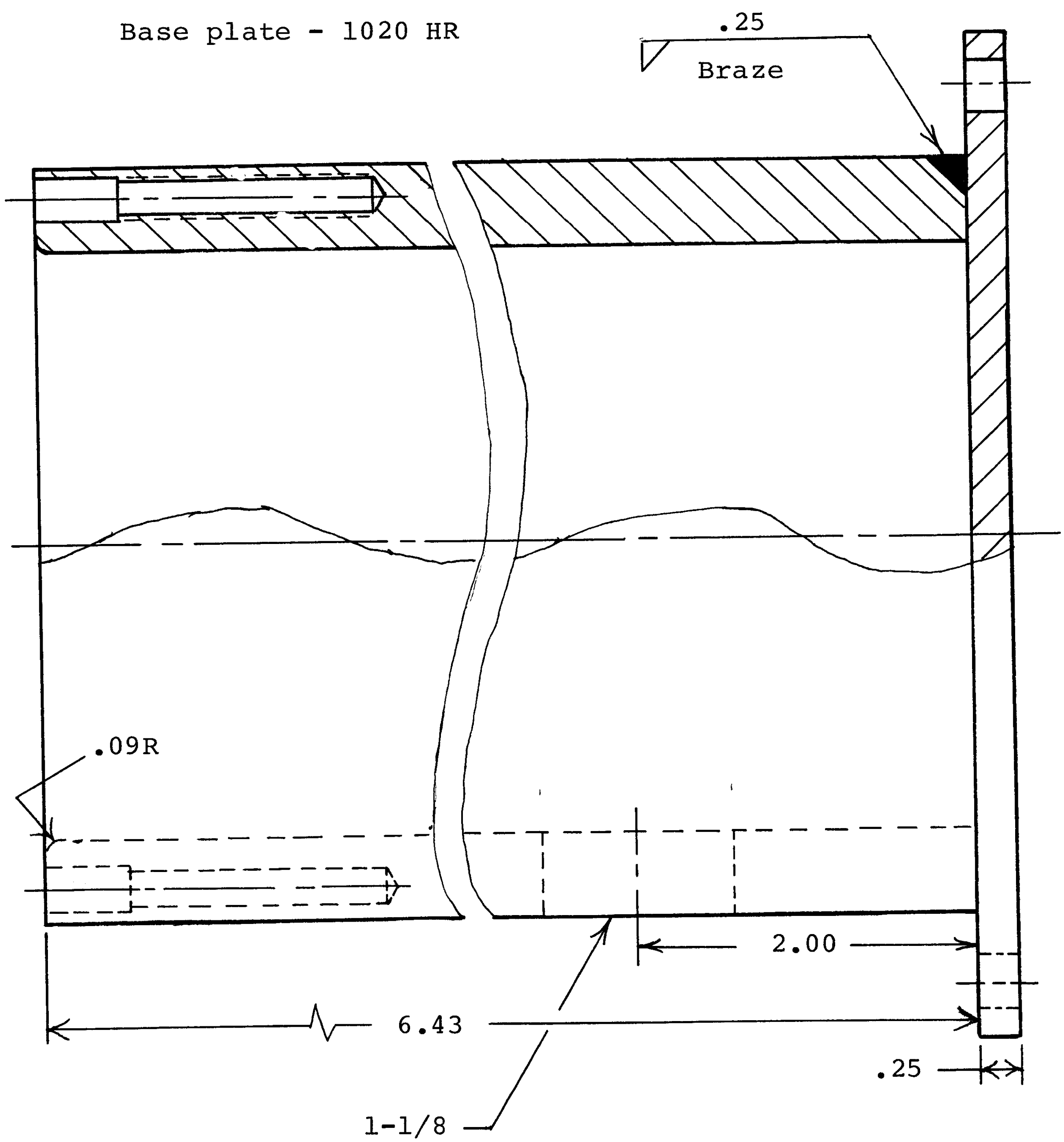


FIGURE 10

PUMP CYLINDER LOWER HALF

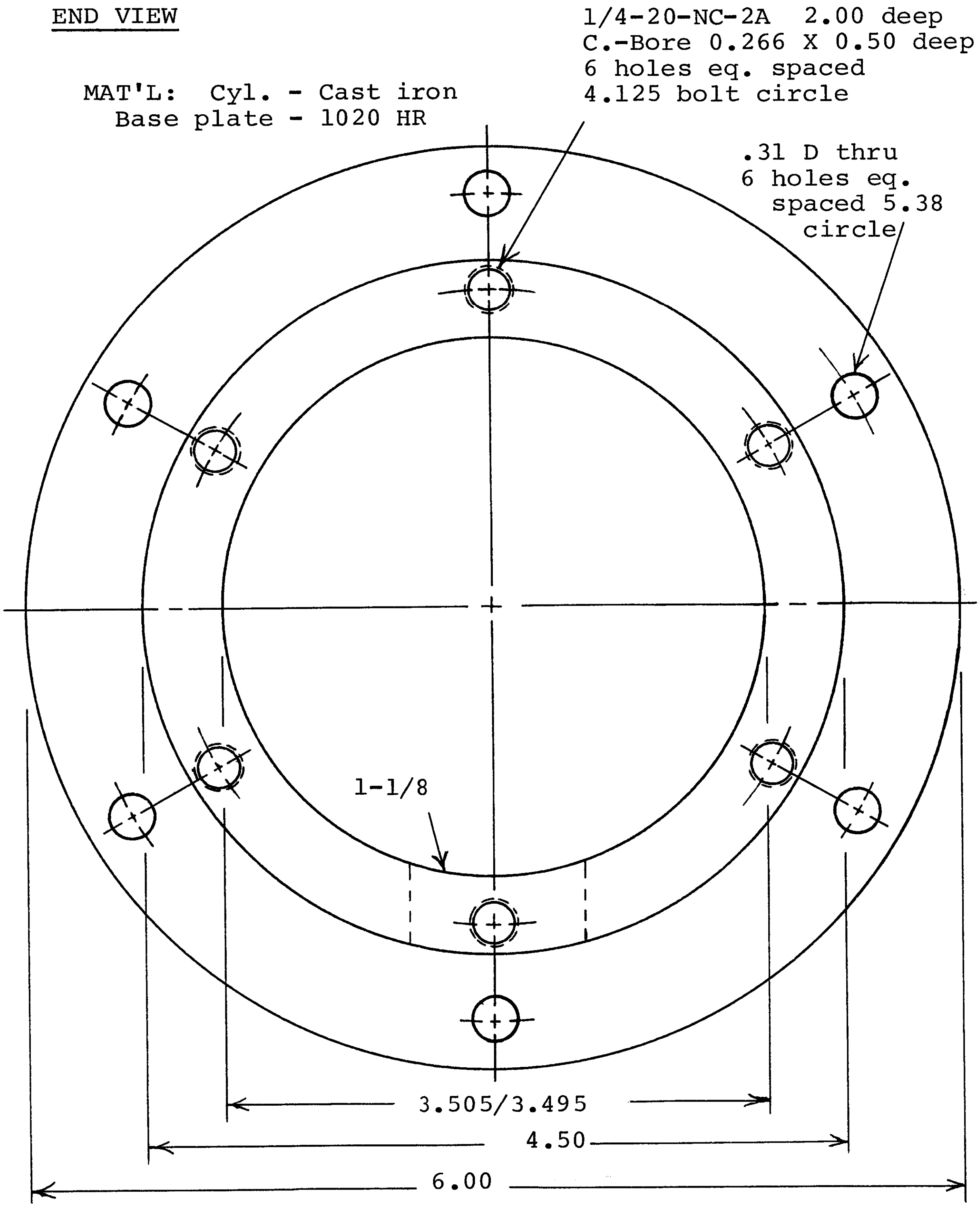
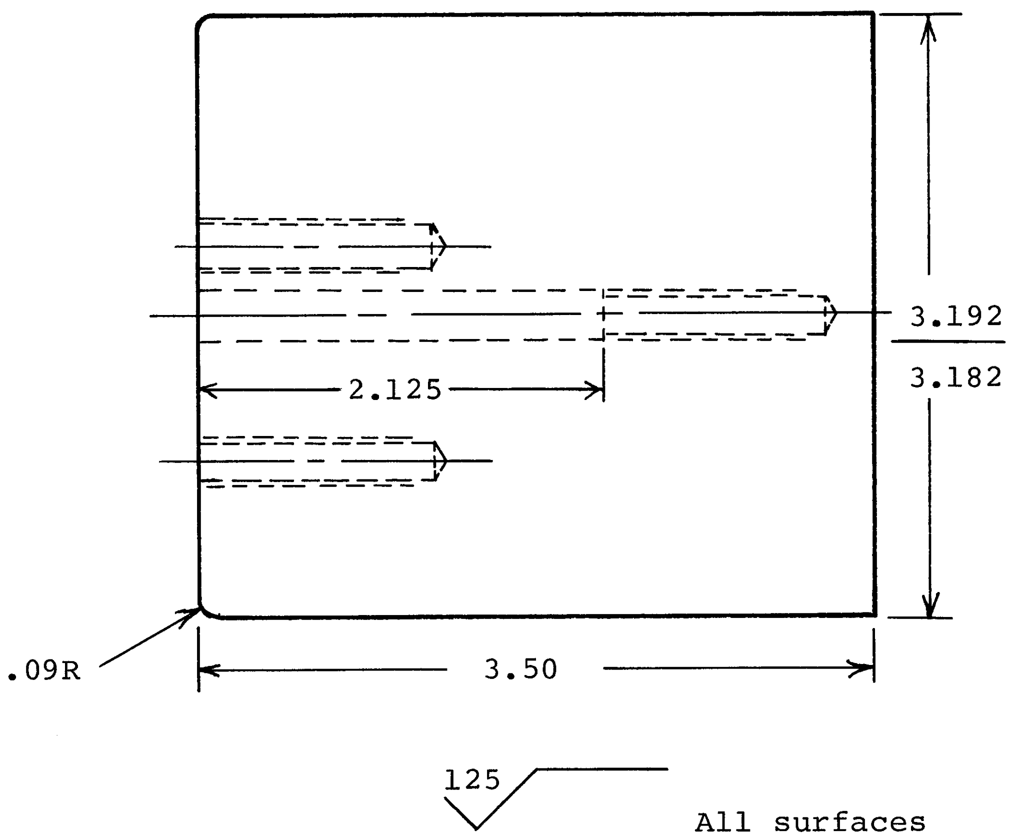


FIGURE 11

PISTON

SIDE VIEW

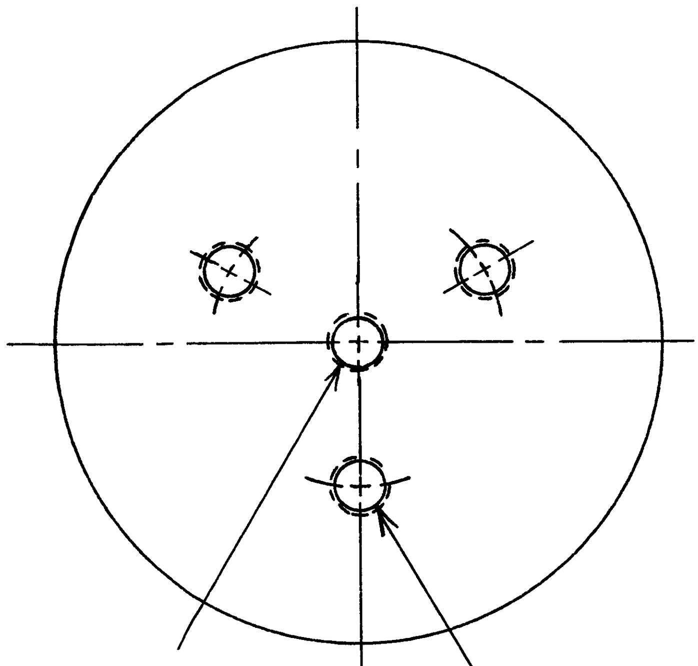


MAT'L: Cast iron

FIGURE 12

PISTON

END VIEW



1/4-20-NC-2A *3.25 depth
C-Bore .266 X 2.125

1/4-20-NC X 1.50 Depth
1.500 Bolt Circle
3 Eq. Spaced

FIGURE 13

DIAPHRAGM RETAINER

MAT'L: Steel

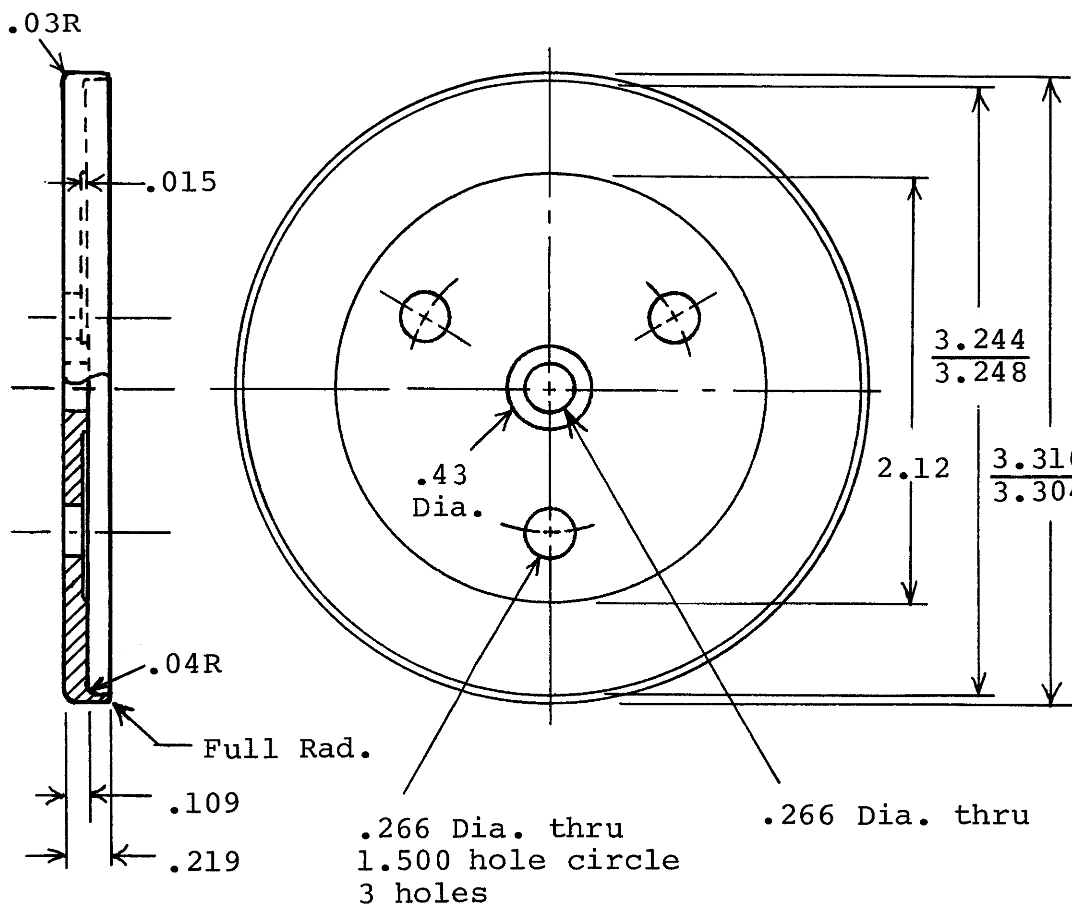


FIGURE 14

PISTON CABLE & SCREWS ASSEMBLY

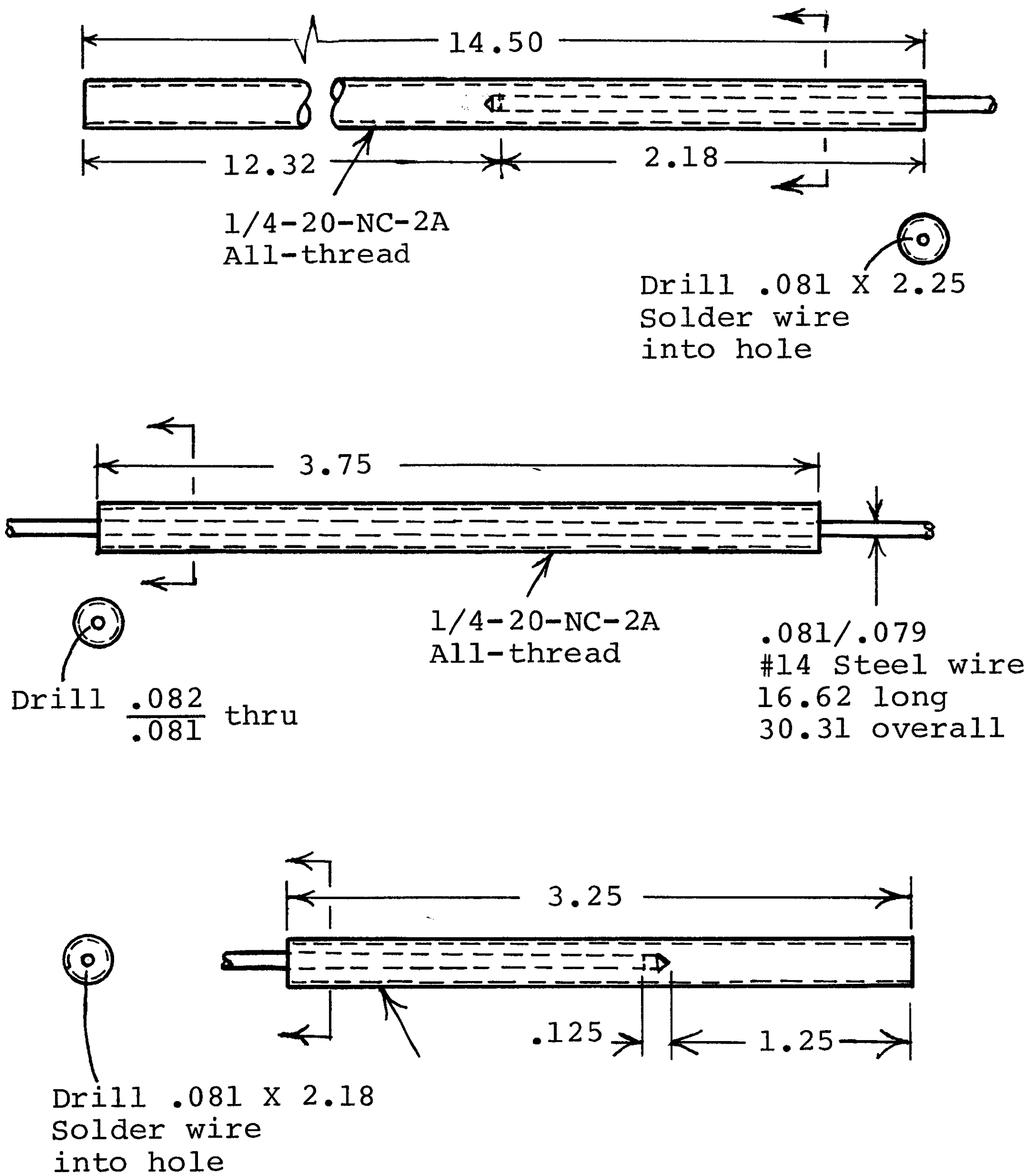


FIGURE 15

VALVE PLATE

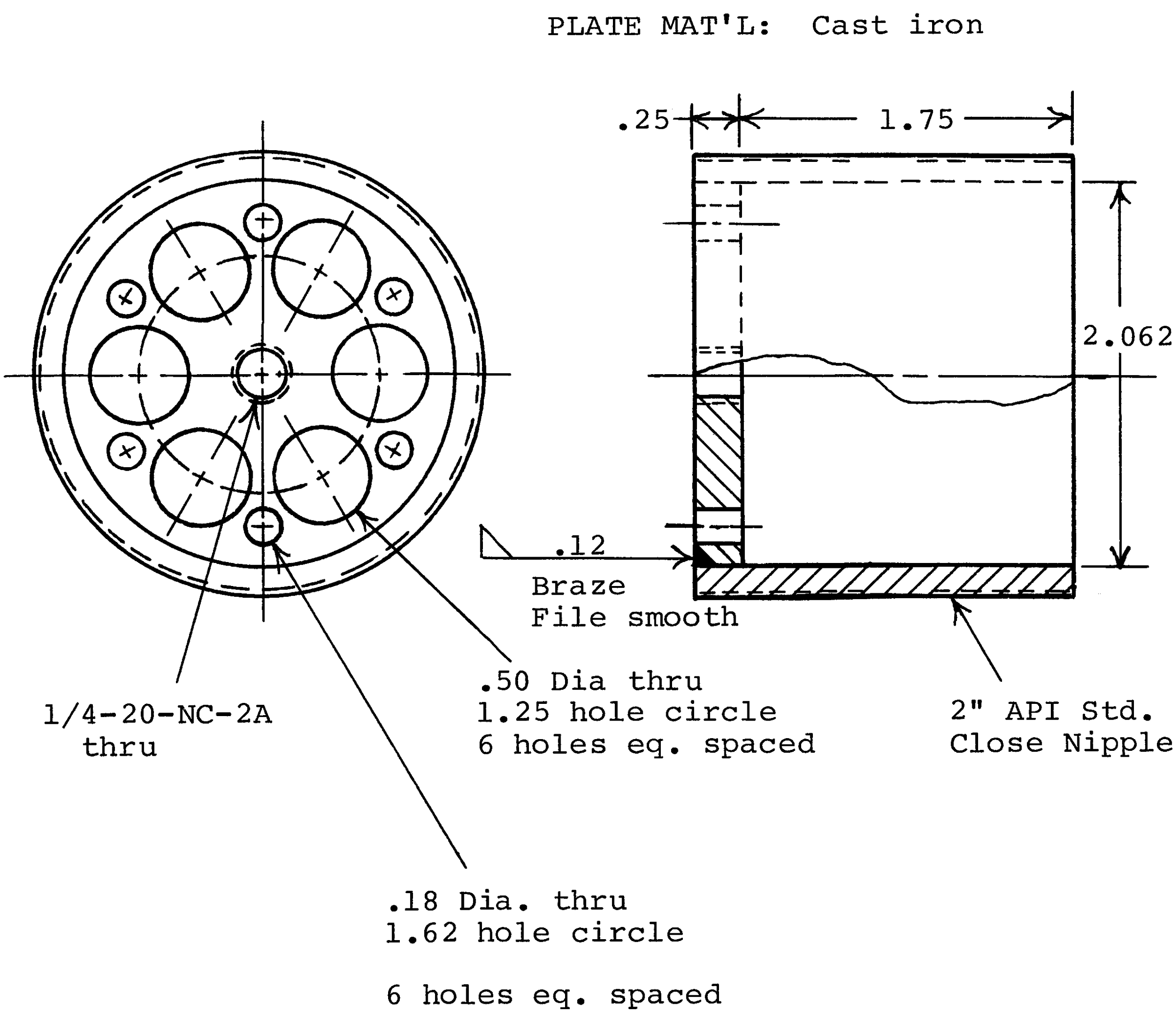
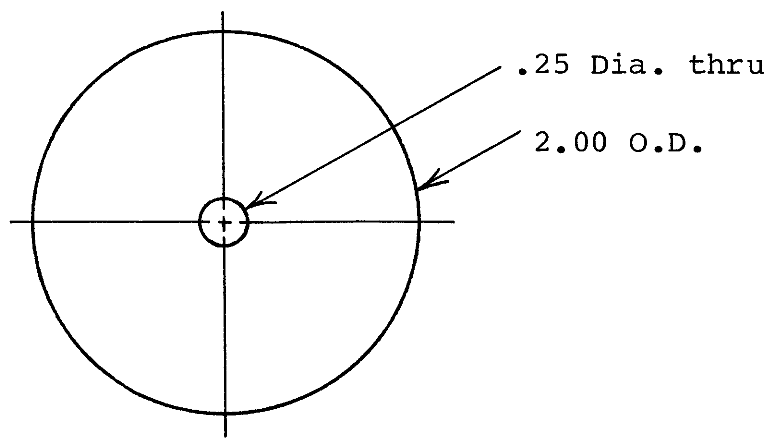
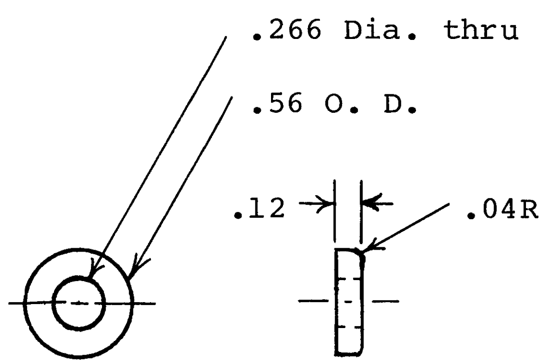


FIGURE 16

VALVE FLAPPER & WASHER



MAT'L: 1/16" Sheet Neoprene

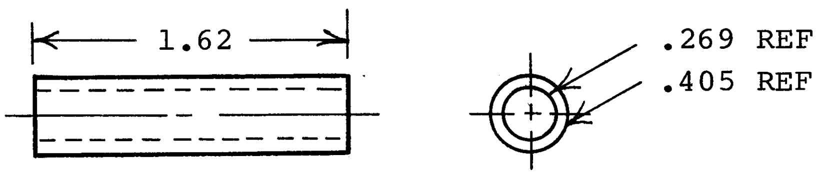


MAT'L: Steel

FIGURE 17

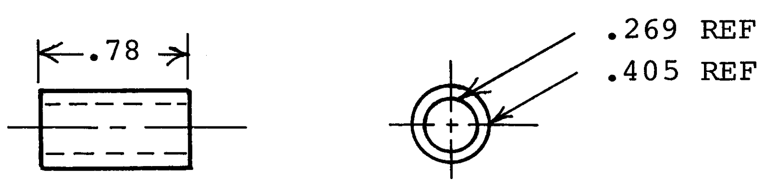
BEARINGS AND BEARING BLOCK

FRAME BEARING



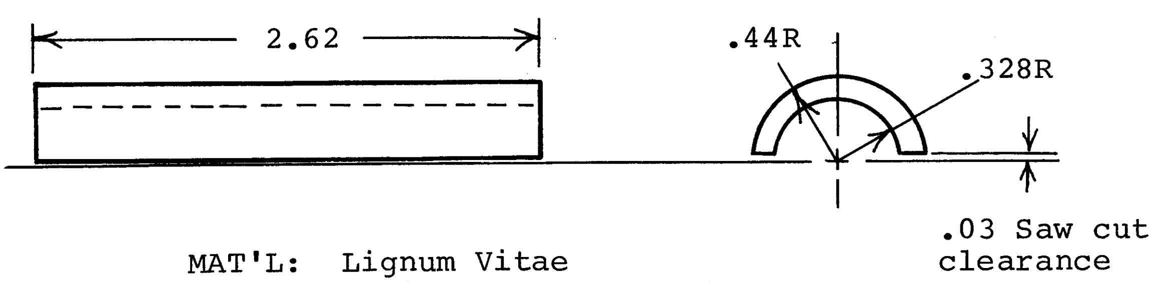
MAT'L: 1/8" Pipe Sch. 40

CONNECTOR BEARING



MAT'L: 1/8" Pipe Sch. 40

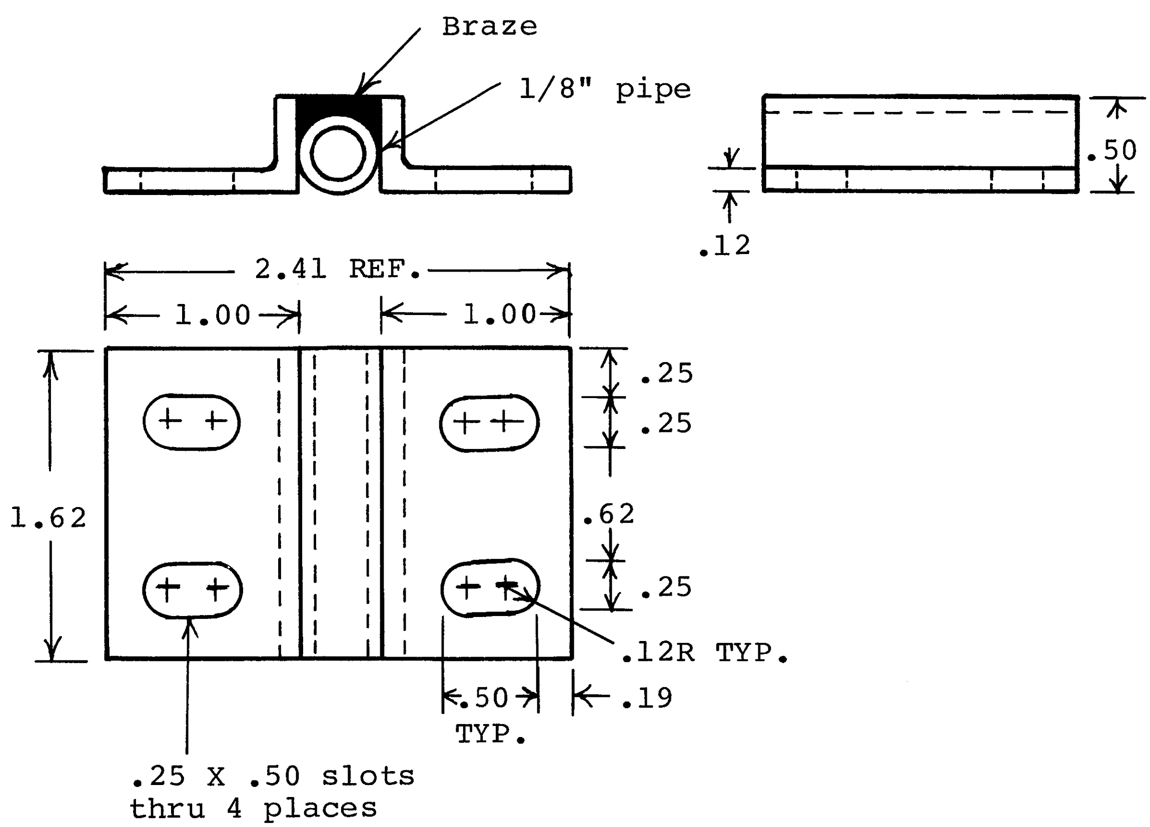
PEDAL BEARING BLOCK



MAT'L: Lignum Vitae
available J. C. Tarbell
or asphalt soaked hardwood

Figure 18

DRIVE BEARING

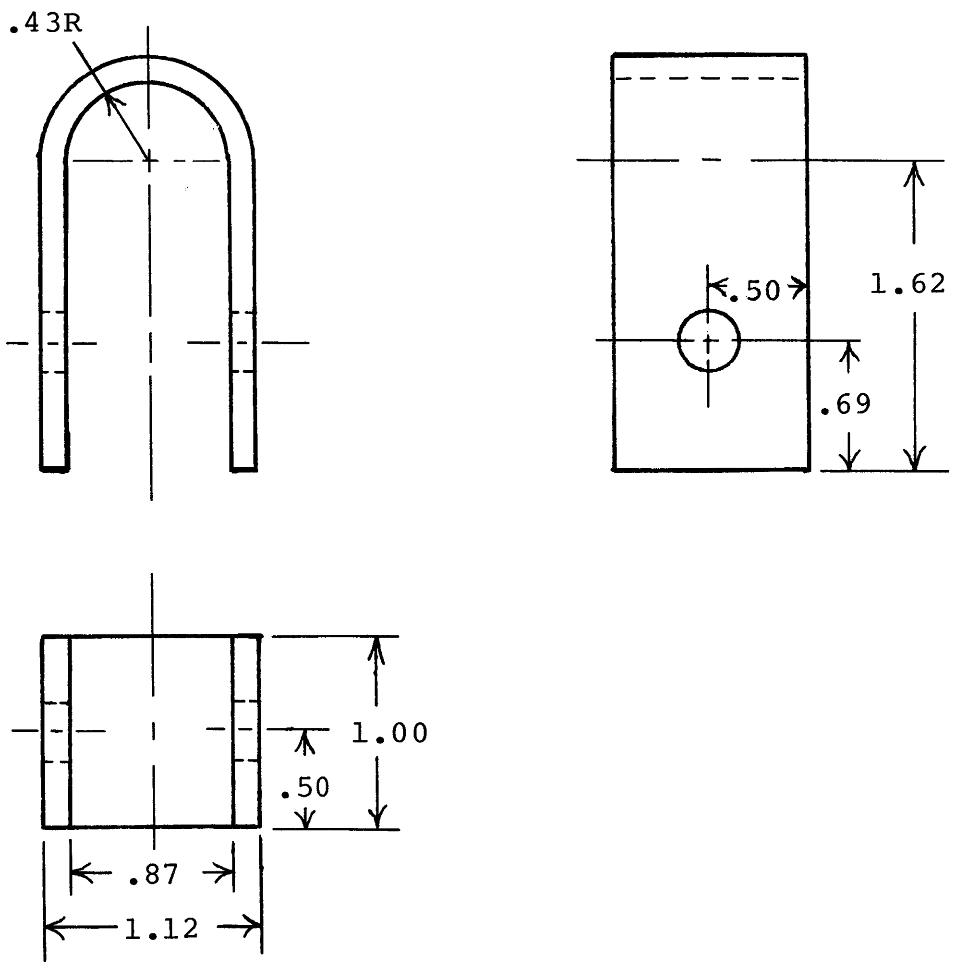


MAT'L: 1/2 X 1 X 1/8 Angle

1/8" Sch. 40 pipe

FIGURE 19

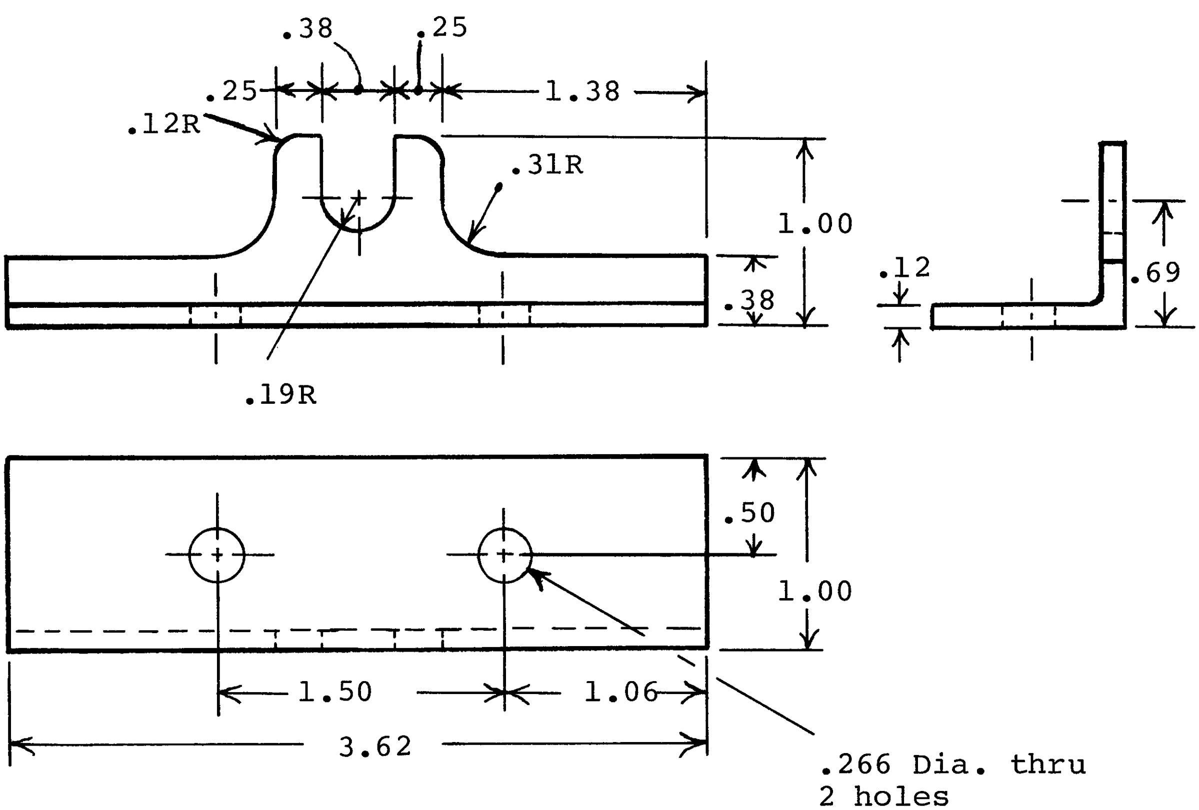
PEDAL STRAP



MAT'L: Low carbon steel strap

FIGURE 20

BICYCLE SUPPORT



MAT'L: 1 X 1 X 1/8 Angle

FIGURE 21

LINKAGE PEDAL ADAPTER

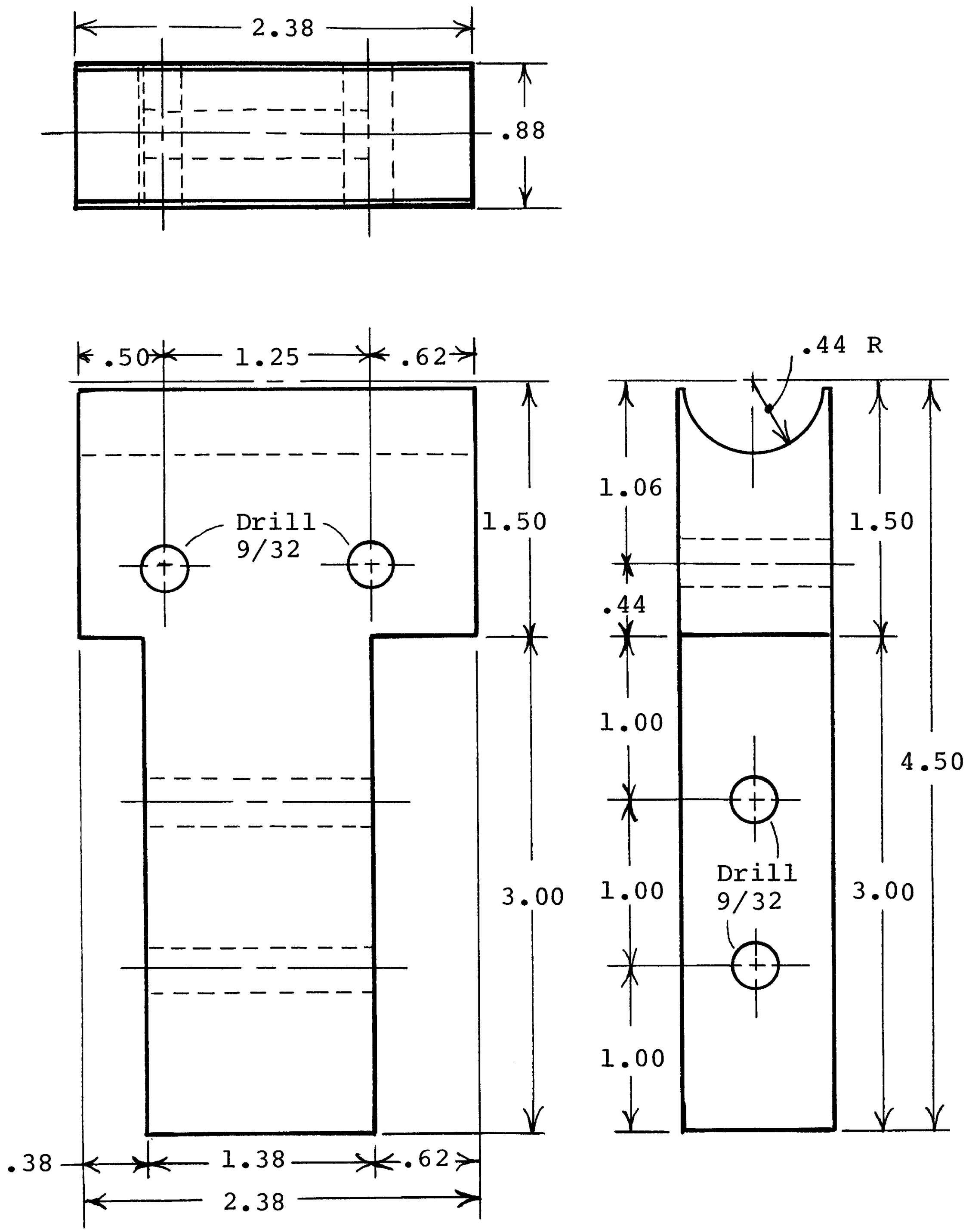
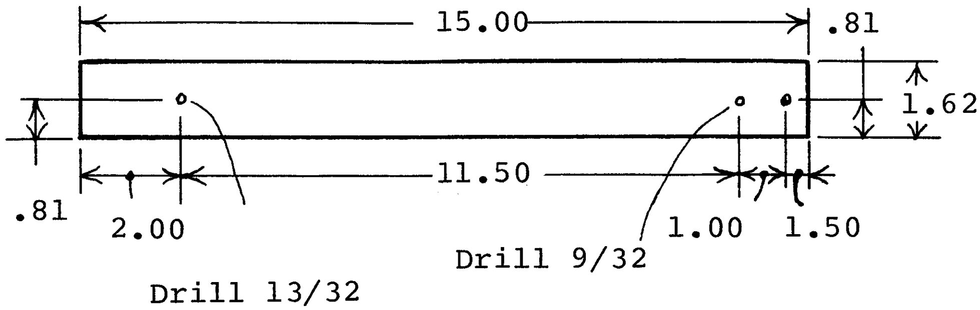
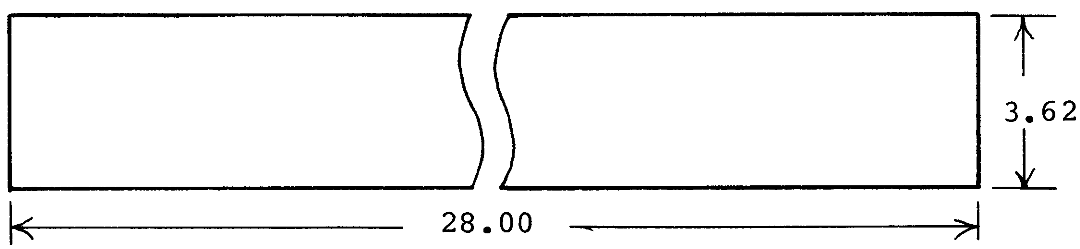


FIGURE 22

LINKAGE COMPONENTS - SHEET 1





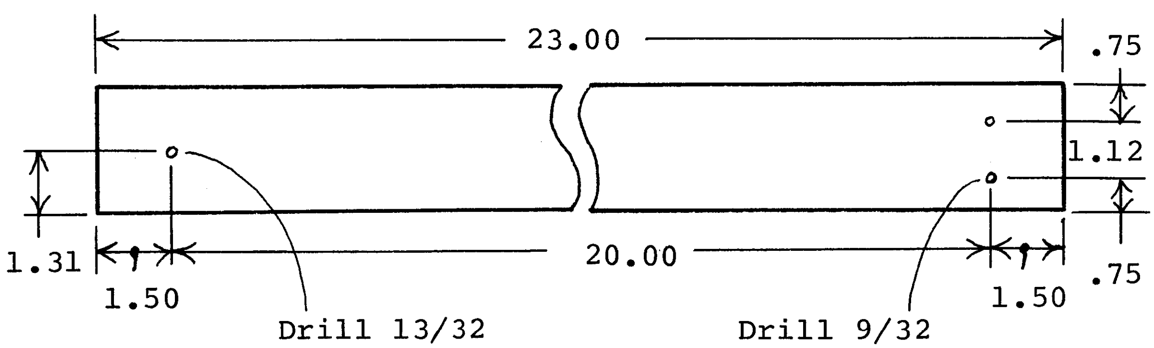
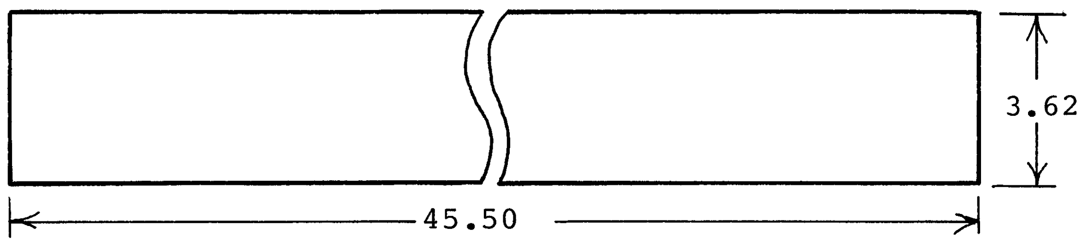


FIGURE 23

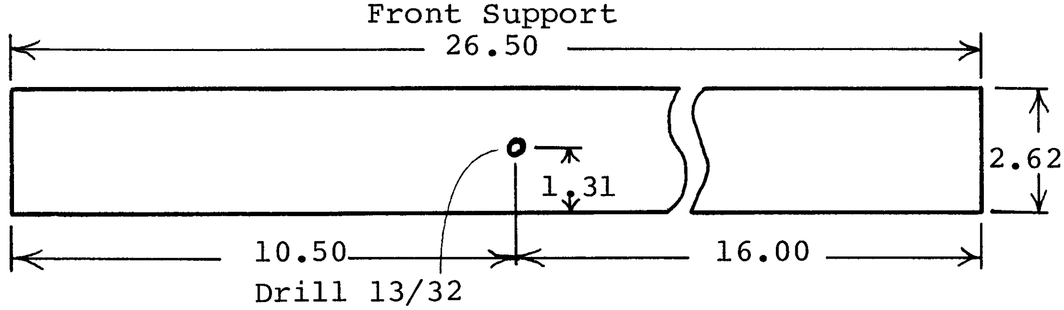
LINKAGE COMPONENTS - SHEET 2

Backbone



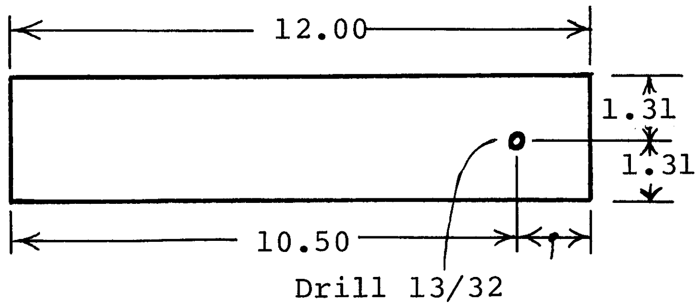
1 required - Material: 2 X 4 Pine

Front Support



2 required - Material: 2 X 4 Pine

Front Outer Bearing Support



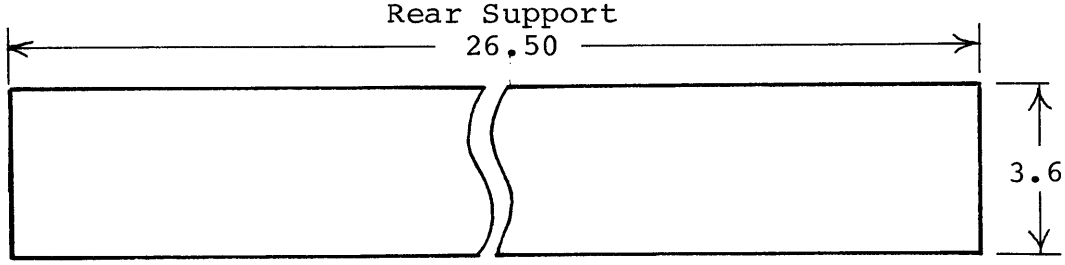
2 required -

Material: 2 X 4 Pine

FIGURE 24

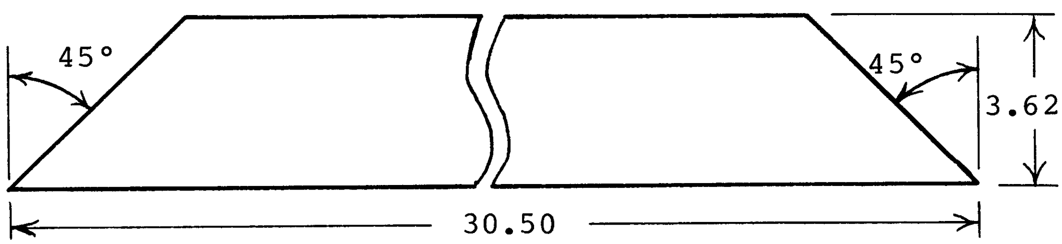
LINKAGE COMPONENTS - SHEET 3

Rear Support



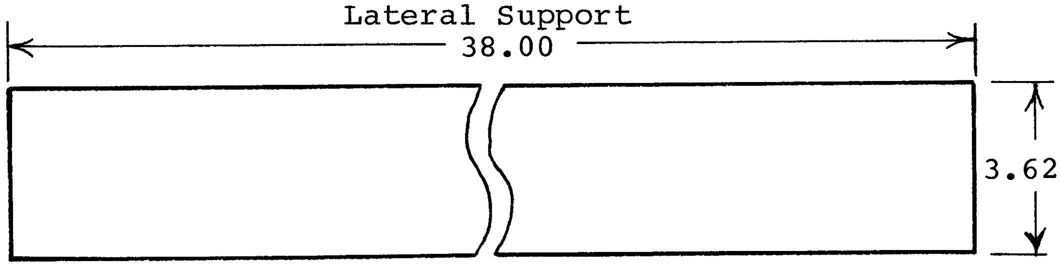
2 required - Material: 2 X 4 Pine

Lateral Brace



4 required - Material: 2 X 4 Pine

Lateral Support



2 required - Material: 2 X 4 Pine

Longitudinal Brace

2 required - Material: 2 X 4 Pine

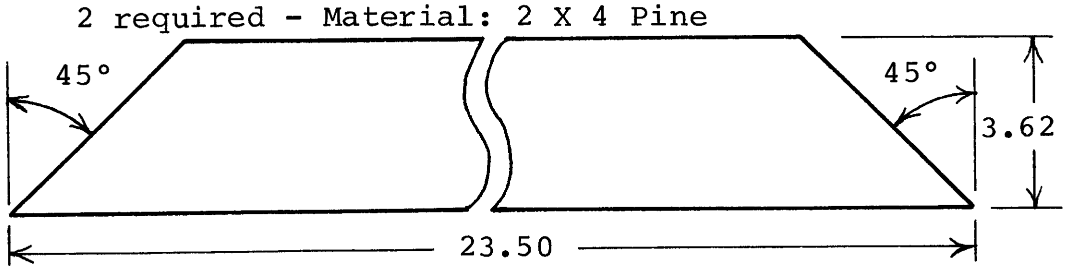
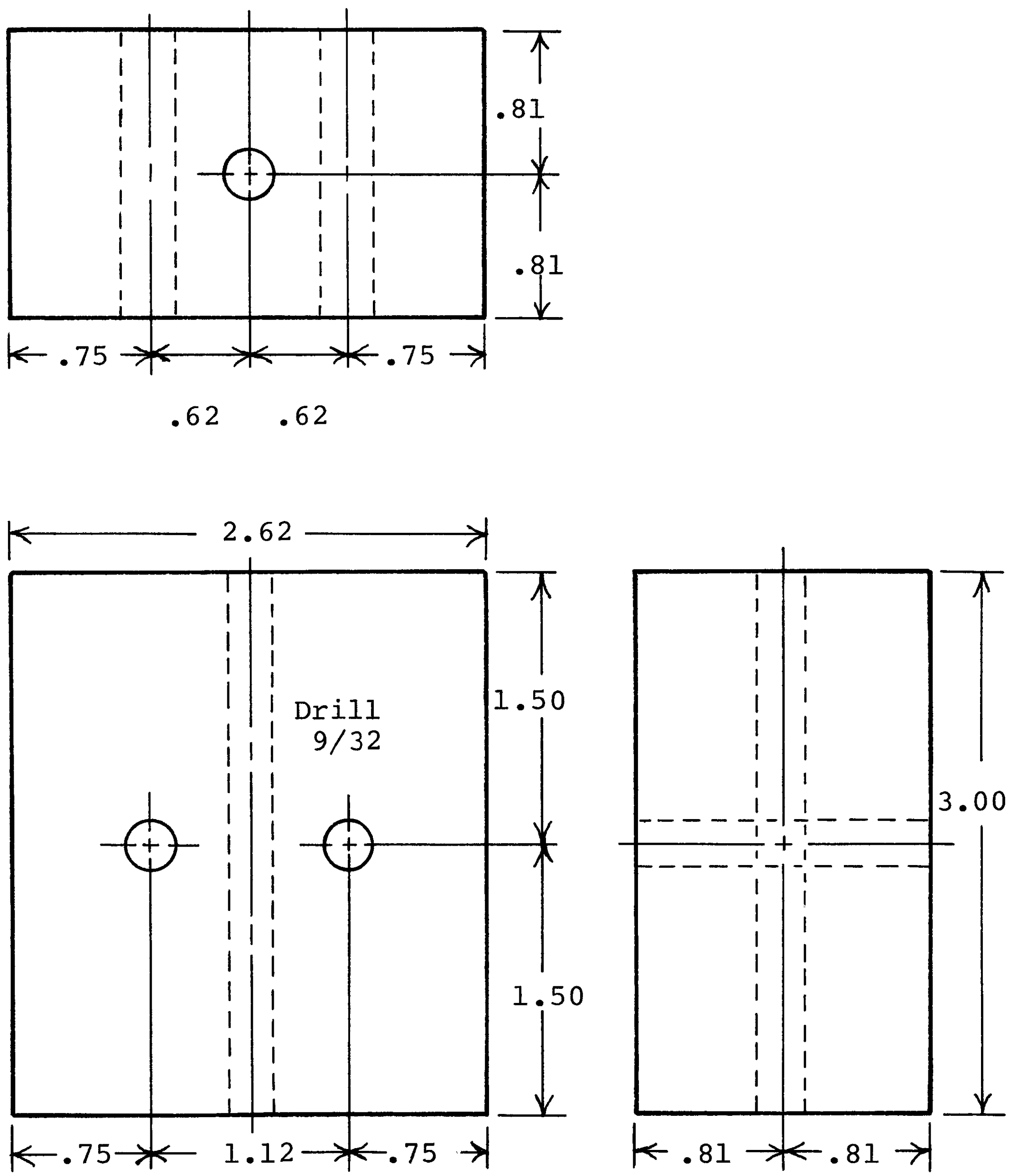


FIGURE 25

CABLE CONNECTOR BLOCK



2 required - Material: 2 X 4 Pine

APPENDIX A-

Letter

from

Diaphragm Industries, Inc.

Complete Diaphragm Actuators
Diaphragm Assemblies and Hardware
Prototypes or Production Quantities

DIAPHRAGM INDUSTRIES, INC.

[redacted]

1001 ISLINGTON STREET   PORTSMOUTH, NEW HAMPSHIRE 03801   AREA CODE 603 — 436-7040

June 29, 1978

C. Fly
Fly International
P.O. Box 30400
Amarillo, Texas   79120

Dear Mr. Fly:

As per our telephone conversation, I feel we can manufacture a diaphragm with a 13" stroke, 6.5" from the center.

The diaphragm is to be made from Buna-N and Dacron fabric, to withstand 50 PSI from rubber side. However, because the diaphragm will be made from a sleeve material, the head section will have to be tied in with the Dacron insert.

The approximate cost will be $28.00 each, plus $595.00 for tooling (single cavity mold).

Sincerely yours,

Tom Powell

TP:gps

NITRILES (BUNA N)   SILICONES   VITON   HYPALON   SBR   NEOPRENE   BUTYL   POLYURETHANE   DACRON   NYLON   COTTON   GLASS   TEFLON

APPENDIX A-5

FLUID LOSSES IN TEST PUMP

SUCTION LOSSES

Fig. 26 shows schematically the various components of the suction line system. One such assembly was used for each cylinder. The construction consisted of 10 feet of 1-1/4 inch cloth braid two ply rubber hose connected to the valve by a king nipple and reducer. The valve was connected to the rest of the system by two short pieces of one inch pipe and a one inch 90° elbow. The pressure drop for this is

$$P = \frac{\rho V_1^2}{2g}\left[K_{entrance} + f_1 \frac{L_1}{D_1} + K_{fitting}\right]$$

$$+ \frac{\rho V_2^2}{2g}\left[f_2\left(\frac{L_2 + L_3}{D_2} + \left(\frac{L}{D}\right)_{elbow}\right) + K_{expansion}\right]$$

$$+ 0.33 \quad Q^{0.8}$$

$$+ \frac{\rho}{g} \frac{dV_1}{dT} L_1$$

$$+ \frac{\rho}{g} \frac{dV_2}{dT}\left[L_2 + L_3\right]$$

where

P = pressure drop psi

ρ = density 0.0361 lbm/in

g = 386 lbm-in^2/sec^2-lbf

Q = instantaneous flow rate - gallons/min.

subscript $_1$ refers to 1-1/4 inch hose

subscript $_2$ refers to 1 inch pipe

L_1 = 120 inches

L_2 = 11 inches

L_3 = 3 inches

$K_{fitting}$ = 0.03

$K_{entrance}$ = 0.75

$K_{expansion}$ = 1

$\left(\frac{L}{D}\right)_{elbow}$ = 31

f_1 = 0.02

f_2 = 0.02

D_1 = 1.25 inches

D_2 = 1.049 inches

D_c = 3.34 inches

V_c is the velocity of the piston (in/sec) and $\frac{dV_c}{dT}$ is the acceleration of the piston (in^2/sec).

$$V_1 = V_c \left(\frac{D_c}{D_1}\right)^2$$

$$V_2 = V_c \left(\frac{D_c}{D_2}\right)^2$$

$$\frac{dV_1}{dT} = \frac{dV_c}{dT} \left(\frac{D_c}{D_1}\right)^2$$

$$\frac{dV_2}{dT} = \frac{dV_c}{dT}\left(\frac{D_c}{D_2}\right)^2$$

$$Q = \frac{\pi D_c^2}{4} V_c * \frac{60 \frac{sec}{min}}{231 \frac{in^3}{gallon}}$$

Substituting and reorganizing yields

$$P = AV_c^2 + BV_c^{0.8} + C\frac{dV_c}{dT}$$

where

$$A = 0.0187 \frac{lbf\text{-}sec^2}{in^4}$$

$$B = 0.687 \frac{lbf}{in^2}\left(\frac{sec}{in}\right)^{0.8}$$

$$C = 0.103 \frac{lbf\text{-}sec^2}{in^3}$$

To a good approximation the piston velocity is a sine function

$$V_c = 1.875 \ \omega \sin \omega T \qquad in/sec$$

$$\frac{dV_c}{dT} = 1.875 \ \omega^2 \sin \omega T \qquad in/sec^2$$

The maximum pressure drop occurs at $T = \frac{0.3\pi}{\omega}$

TABLE 5

MAXIMUM SUCTION PRESSURE DROP

Speed RPM	Total Pressure Drop PSI
10	1.2
20	2.4
30	3.9
40	5.8
50	7.9
60	10.4

TRANSFER FLUID LOSSES

Fig. 26 shows schematically the major elements of the crossover tube system. The pressure drop is given by

$$P = \frac{\rho V_c^2}{2g}\left[f_c\left(\frac{L_1 + L_2}{D_c}\right) + K_{entrance} + K_{exit}\right] + \frac{\rho V_p^2}{2g}\left[f_p\left(\frac{L_3}{D_p}\right)\right] + \frac{\rho}{g}\frac{dV_c}{dT}\left(L_1 + L_2\right) + \frac{\rho}{g}\frac{dV_p}{dT}L_3$$

where

P = pressure drop psi

ρ = density 0.0361 lbm/in^3

g = 386 $lbm\text{-}in/sec^2\text{-}lbf$

subscript c refers to cylinders

subscript p refers to crossover tube

$L_1 + L_2$ = 2.8 inches

L_3 = 30 inches

D_c = 3.5 inches

D_p = 1.125 inches

f_p = 0.03

f_c = 0.04

$K_{entrance}$ is given in Idel'chik, page 280 [Ref. 4].

$$K_{entrance} = \frac{P_{entrance}}{\frac{\rho V_c^2}{2g}} = A' \left[1 + \left(\frac{V_p}{V_c} \right)^2 \right]$$

$$A' = 0.9$$

$$V_p = V_c \left(\frac{D_c}{D_p} \right)^2$$

$$K_{entrance} = 85$$

K_{exit} is given in Idel'chik, page 266 [Ref. 5].

$$K_{exit} = \frac{P_{exit}}{\frac{\rho V_c^2}{2g}} = A' \left[1 + \left(\frac{D_c}{D_p} \right)^4 \right]$$

$$A' = 1.0$$

$$K_{exit} = 95$$

The following relations apply

$$V_p = V_c \left(\frac{D_c}{D_p} \right)^2$$

$$\frac{dV_p}{dT} = \frac{dV_c}{dT}\left(\frac{D_c}{D_p}\right)^2$$

After manipulation

$$P = AV_c^{\ 2} + B\frac{dV_c}{dT}$$

$$A = 0.0119 \ \frac{\text{lbf-sec}^2}{\text{in}^4}$$

$$B = 0.0274 \ \frac{\text{lbf-sec}^2}{\text{in}^3}$$

As in the suction analysis, the piston velocity is approximately a sine function.

$$V_c = 1.875 \ \omega \ \sin \omega T \qquad \text{in/sec}$$

$$\frac{dV_c}{dT} = 1.875 \ \omega^2 \cos \omega \ T \qquad \text{in/sec}^2$$

The peak pressure drop occurs at $T = \dfrac{0.3 \ \pi}{\omega}$

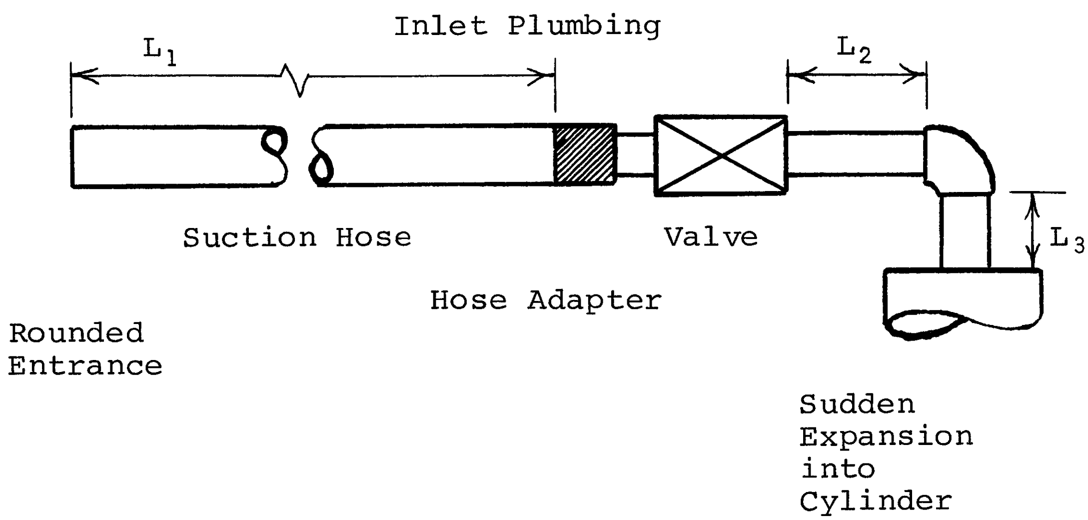
TABLE 6

TRANSFER FLUID PRESSURE DROP

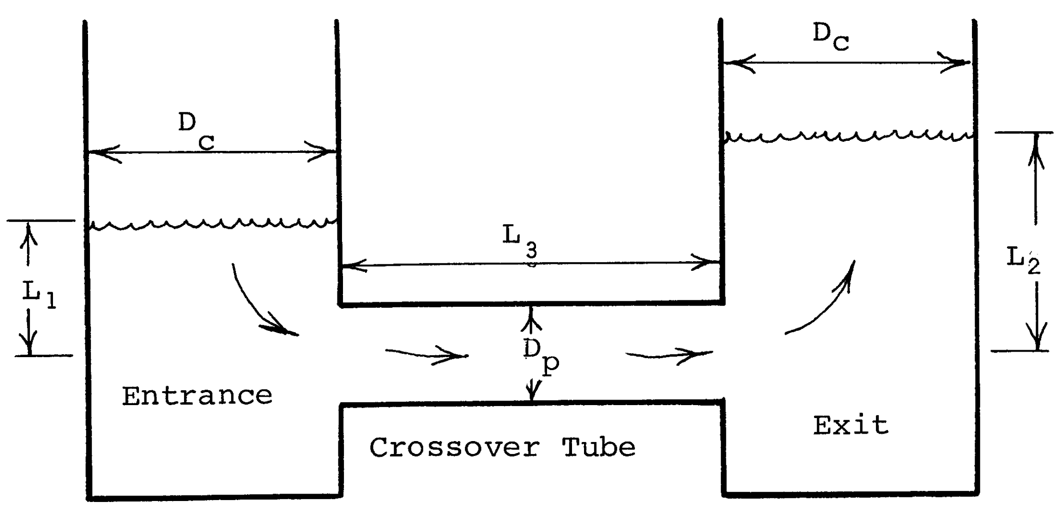
Speed RPM	Pressure Drop PSI
10	0.1
20	0.3
30	0.6
40	1.0
50	1.6
60	2.3

FIGURE 26

INLET AND CROSSOVER SCHEMATICS



CYLINDER CROSSOVER PLUMBING



APPENDIX A-6

VAPOR TRANSFER RATES

Schrage presents the following relation for interphase mass transfer in the absence of non-condensible gases.

$$\frac{\omega}{\omega_s} = \sigma \left[1 - \left(\frac{\rho_o}{\rho_s}\right) \left(\frac{T_o}{T_s}\right)^{0.5} \Gamma \right] \qquad (1)$$

where

ω = mass transfer rate - lbm/in^2-sec

ω_s = absolute rate of vaporization - lbm/in^2-sec

σ = condensation coefficient, empirical

ρ = density lbm/in^3

T = temperature °R

Γ = correction factor

Subscript o refers to vapor phase at condition of mass transfer. Subscript s refers to equilibrium conditions.

Γ is a correction factor that accounts for the bulk momentum of the vapor relative to the interphase surface.

$$\Gamma = e^{-\phi^2} - \phi \sqrt{\pi} \left[1 - \text{erf} (\phi) \right] \qquad (2)$$

where

$$\phi = B_o V_o \tag{3}$$

and $$B_o = \left(\frac{M}{2RT_o}\right)^{0.5} \quad \text{sec/ft} \tag{4}$$

V_o = bulk vapor velocity - ft/sec

M = molecular weight of vapor - $\frac{\text{lbmass}}{\text{lbmole}}$

R = gas constant - 1,545 ft-lbf/lbmole-°R

ϕ is a measure of the bulk velocity of the vapor relative to the speed of sound in the vapor.

ω_s is the absolute rate of vaporization which is the mass flow rate that occurs at a bulk velocity equal to the speed of sound in the vapor.

$$\omega_s \qquad \frac{\rho_s}{2\sqrt{\pi}\, B_s} \tag{5}$$

and $$B_s = \left(\frac{M}{2RT_s}\right)^{0.5} \tag{6}$$

also

$$\omega = \rho_o V_o \tag{7}$$

and

$$\phi = \frac{1}{2\sqrt{\pi}} \left(\frac{\omega}{\omega_s}\right) \left(\frac{T_o}{T_s}\right)^{-0.5} \left(\frac{\rho_o}{\rho_s}\right)^{-1} \tag{8}$$

For the purposes of analyzing the mechanism of the fluid transfer in the pump it is assumed that the system is isothermal, i.e.

$$T_o = T_s \tag{9}$$

Equation 8 is now solved for $\frac{\omega}{\omega_s}$ and equated to equation 1. This produces

$$\rho_o = \rho_s \frac{\sigma}{\sigma \Gamma + 2\sqrt{\pi}\,\phi} \tag{10}$$

For $\phi < 10^{-3}$, an approximate equation for Γ is

$$\Gamma = 1 - \phi\sqrt{\pi} \tag{11}$$

For $10^{-3} < \phi < 0.05$ the error function is approximated by

$$\mathrm{erf}\,(\phi) = 1.13\,\phi \tag{12}$$

Thus

$$\Gamma = e^{-\phi^2} - \phi\pi^{0.5}(1 - 1.13\,\phi) \tag{13}$$

In this case the mass transfer rate is very low so the approximation given by equation 11 is sufficient. V_o is simply the piston velocity V_c. From this and the operating temperature, roughly 60° F., ϕ can be determined and thence ρ_o.

By the ideal gas law

$$\frac{P_o}{\rho_o} = \frac{RT_o}{M} \tag{14}$$

and

$$\frac{P_s}{\rho_s} = \frac{RT_s}{M} \tag{15}$$

where P = vapor pressure psia

and M = molecular weight $\frac{\text{lbmass}}{\text{lbmole}}$

P_o (condensing) - P_o (vaporizing) is the pressure difference acting on the transfer fluid. Table 7 presents this pressure difference as a function of the speed of operation for $\sigma = 1$ and $\sigma = 0.1$.

TABLE 7

VAPOR TRANSFER PRESSURE DROPS

Speed RPM	Vaporizing ρ_o $\frac{lbm}{ft^3}$	Vaporizing P_o PSIA	Condensing ρ_o $\frac{lbm}{ft^3}$	Condensing P_o PSIA	Differential Pressure PSI
$\sigma = 1$					
10	0.000829	0.2567	0.000829	0.2568	0.0001
20	0.000828	0.2566	0.000829	0.2568	0.0002
30	0.000828	0.2566	0.000829	0.2568	0.0002
40	0.000828	0.2566	0.000829	0.2569	0.0003
50	0.000828	0.2566	0.000829	0.2569	0.0003
60	0.000828	0.2566	0.000829	0.2570	0.0004
$\sigma = 0.1$					
10	0.000826	0.2559	0.000831	0.2575	0.0016
20	0.000823	0.2551	0.000834	0.2584	0.0033
30	0.000821	0.2542	0.000837	0.2592	0.0050
40	0.000818	0.2534	0.000840	0.2601	0.0067
50	0.000815	0.2526	0.000842	0.2610	0.0084
60	0.000813	0.2518	0.000845	0.2618	0.0100

APPENDIX A-7

TABLE 8 - LINKAGE MOTIONS

	Ascending Piston			Descending Piston			
Angle °	*Position Inches	Velocity in/sec	Accel-eration in/sec^2	Position Inches	Velocity in/sec	Accel-eration in/sec^2	**Position Difference Inches
15	0.17	1.89	16.62	3.69	-0.52	-5.47	0.11
30	0.49	3.39	12.63	3.59	-1.10	-5.63	0.33
45	0.94	4.42	8.31	3.42	-1.75	-6.21	0.61
60	1.45	4.95	4.07	3.17	-2.50	-7.07	0.87
75	1.97	5.00	0.11	2.83	-3.38	-7.83	1.05
90	2.46	4.64	-3.34	2.40	-4.36	-7.56	1.11
105	2.88	3.97	-5.93	1.87	-5.23	-4.83	1.00
120	3.23	3.12	-7.40	1.28	-5.65	+1.30	0.85
135	3.47	2.24	-7.66	0.73	-5.27	9.54	0.45
150	3.64	1.41	-7.13	0.29	-3.98	16.57	0.17
165	3.72	0.69	-6.35	0.36	-2.08	19.92	0.01
180	3.74	0.06	-5.73	0.00	0.00	19.49	-0.01

STROKE = 3.75 inches   SPEED = 24 RPM

* Position is measured above lowest point of travel.

** Positional Difference is given by

Position (asc. piston) - [stroke - position(desc. piston)]