TESTING AND ANALYSIS OF THE BANGLADESHI TREADLE PUMP

By

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ABSTRACT

The Treadle pump was invented in 1980 by development workers to provide low cost, low technology and locally manufacturable irrigation for farmers in northern Bangladesh. The twin cylinder steel-bodied suction pump, called the Treadle pump, operated by a walking motion upon two centrally pivoted levers, is well suited to the high water tables of much of Bangladesh.

The increased popularity of the pump, resulting from extensive marketing efforts, with annual sales of over 50,000 units in 1987/88, has prompted suggestions for design alterations, specifically pump material changes to decrease costs and increase sales. A technical field and laboratory study supported by calculations from hydraulic theory was required as a basis upon which to start design alterations.

In laboratory tests, different pump body configurations were found to have negligible impact on overall pump performance characteristics. These findings were supported by results obtained from calculations of friction and turbulent loss. Any worthwhile alterations must rather be justified by cost and manufacturing benefits.

The combination of computer-aided instrumentation and high speed data collection in laboratory testing, theoretical analysis, and field testing in Bangladesh revealed that the losses in pump performance are a result of poor valve sealing, valve opening and closing delays and leakage past the piston seal, resulting in lower discharges than had been previously assumed. The reduced discharge for irrigation and the operator power input limit, measured at the pump, of 55 watts results in a four meter maximum depth of water table to which the standard 89 mm (3.5 inch) pump can be used, with some variation due to irrigation requirements and operator strengths.

Improvements to piston valve and foot valve design to reduce leakage and valve-delay times and the use of a smaller 77 mm (3 inch) cylinder diameter, are achievable and are recommended. These improvements combine to permit operation of the pump to nearly six meters without exceeding the limit of input power or reducing the discharge below the crop irrigation requirements.

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1 INTRODUCTION

This thesis examines the performance of two configurations of steel Treadle pump bodies and piston configurations, including the designs currently marketed for use in Bangladesh. Investigation of the pumps by intensive laboratory testing and data analysis, field tests, and analysis from hydraulic theory provides a basis for evaluating the pump design and alteration alternatives.

An initial study of the pump uses, proposed design alterations and pump theory was conducted in Bangladesh during the summer of 1987, followed by laboratory testing of alternate pump and piston designs at the University of British Columbia at Vancouver, Canada, in mid-1988. Field testing of Treadle pumps, using the currently marketed configuration of pump body and pistons, was conducted in the central, eastern and northern regions of Bangladesh at the start of irrigation for the 1988-89 dry season crops in late November and December 1988.

Included is a history and description of the Treadle pump, to familiarize readers with this technology, and an examination of the pump hydraulics to describe the pump operation in more detail and provide the basis for the analysis of laboratory and field tests. Adescription of the laboratory and field tests and the presentation of the test results, as well as those from the theory based calculations, provides a comprehensive view of the pump performance characteristics. The results also identify the mechanisms affecting pump performance as well as providing the basis for proposed design alterations.

The testing and study included is based on the Treadle pump designs and operation configuration described in section two. This uses centrally pivoted levers to operate the pump, referred to as the "dheki" operation structure, as is primarily used in Bangladesh at this time. The testing parameters and design alternatives examined are limited to those manufacturable with materials and technology available in Bangladesh.

2 BACKGROUND

2.1 TREADLE PUMP DEVELOPMENT AND IMPACT

The Treadle pump was invented in 1980 by Gunnar Barnes and Dan Jenkins, development workers with the non-governmental organizations (NGOs), the Rangpur-Dinajpur Rehabilitation Service (RDRS) and USAid respectively, operating in northern Bangladesh (RDRS 1984).

Bangladesh, located east of India and north-west of Burma, encompassing the Ganges and Brahmaputra river deltas where they enter the Bay of Bengal, illustrated in figure 1, is well suited to human powered irrigation because:

a) Labor is widely available, either by the pump owner or at low cost.

b) The high water table throughout most of the country is within the 10 meter limit of suction pumps, as shown in the water table depth map in figure 2 and,

c) farm plots are typically so small that the quantities of water required daily are within the lifting capacity of a single human operator.

The size of plots irrigated by manual irrigation range in size from 0.25 to 1 hectare (Orr and Islam 1988). High value crops such as vegetables, spices and tobacco are grown as they generally have lower irrigation requirements, less than 6 mm/day on average (EPC 1988 Tables 24-26), and greater financial return than rice crops. Given these parameters, the irrigation water requirements vary from 15 to 60 cubic meters per day for 0.25 and 1 hectare irrigated area respectively. In more understandable terms for the physical effort needed to meet these requirements using manual irrigation, this equates to lifting from 15 to 60 tonnes of water a vertical distance of up to four meters every day during periods of peak irrigation requirements.

The original aim in designing the pump was to provide low cost manual irrigation, to benefit low income farmers in the Rangpur and Dinajpur regions of northern Bangladesh. Since its introduction, use of the pump has spread to many other new regions of Bangladesh previously without irrigation or irrigated by more expensive or less appropriate means. In a recent socio-economic study of the treadle pump, the low cost of 1488 taka ($60 CAD) (EPC 1988, table 29), including installation on an unplasticized Polyvinyl Chloride (uPVC) tubewell, was found to provide 100% investment return in one season with a minimum irrigated area of 0.08 hectares (.16 acres) (Orr and Islam 1988), proving that the aim of the original design has been achieved.

Benefits of Treadle pump irrigation include dry season cropping, previously not possible for lack of irrigation, diversification of crops to modern high yield rice varieties and vegetables, job creation, with 30% of pumping labour being hired, and increased self sufficiency in rice (Orr and Islam 1988).

Figure 1: Geographical location of Bangladesh.

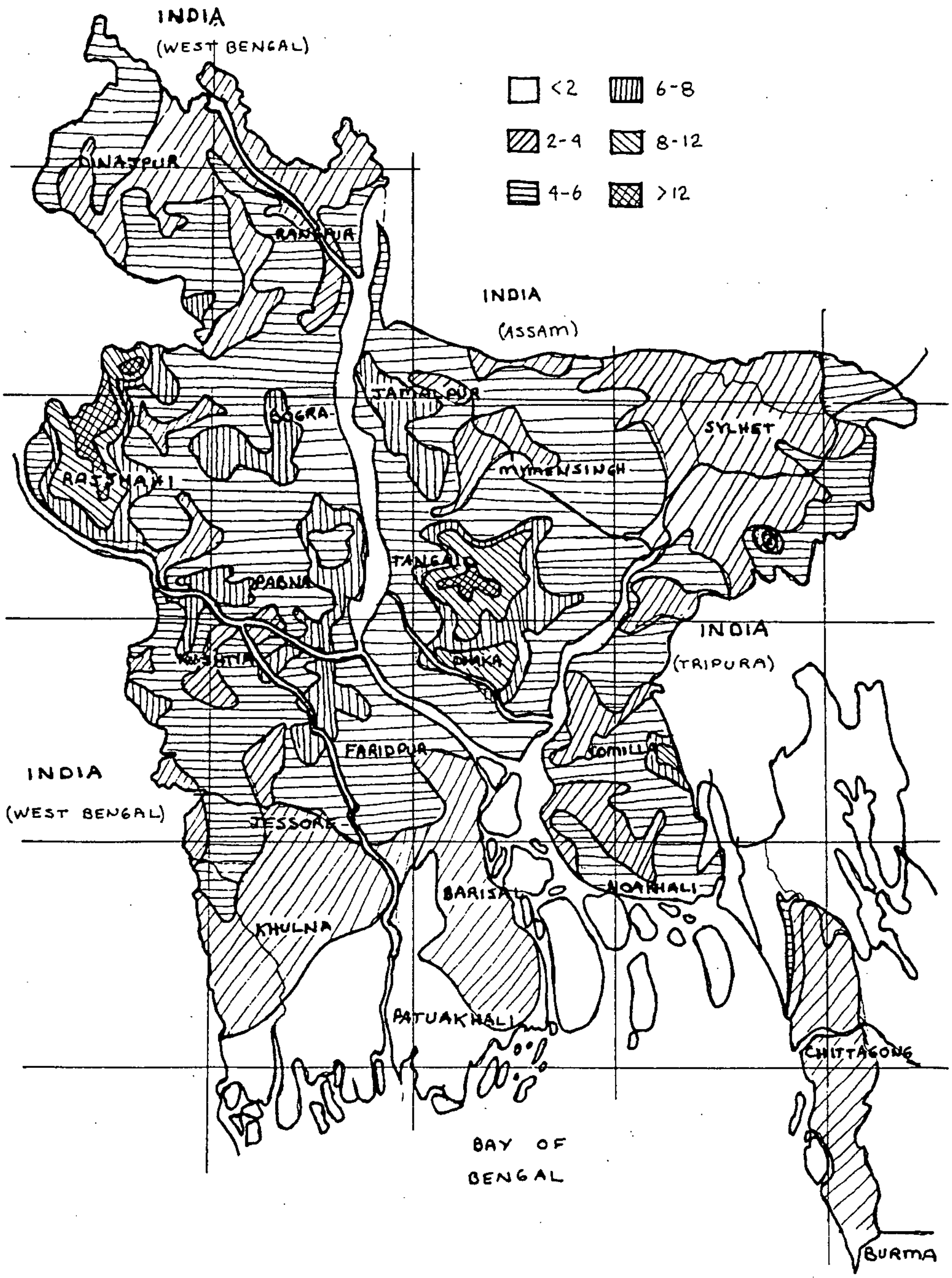

Figure 2: Water table depths in Bangladesh.
(Source: United Nations 1982, Plate 7)

Sales of the Treadle pump have increased dramatically in the past four years as is shown in table 1.

YEAR	IDE	RDRS	OTHERS	TOTAL
1980-85	0	39,870	0	39,870
1985-86	120	10,316	4,000	14,436
1986-87	12,000	3,000	16,000	30,000
1987-88	12,000	3,000	27,000	43,938

Table I: Treadle pump sales of International Development Enterprises (IDE), Rangpur Dinajpur Rehabilitation Service (RDRS) and others. (Source: IDE 1988)

The large and increasing sales of the Treadle pump are attributable to the low initial cost, the ease of maintenance and repair, comfort of operation, the availability of replacement parts and the ability to make some repairs using salvaged parts such as inner tube rubber. Although initially installed on bamboo tubewells, the increased availability and increased market demand for uPVC pipe has resulted in increasing uPVC usage for manual pump tubewells. This has resulted in more consistent well life and increased benefits but at an increase in cost and a greater dependence upon imported materials. In addition to the obvious benefits of the pump, increasingly focused marketing efforts throughout Bangladesh by various Non-Governmental Organizations (NGO's) and Bangladeshi private businessmen have contributed greatly to the availability and quality of the pumps sold, and have further increased sales.

2.2 PREVIOUS TEST RESULTS

The increase in popularity of manual irrigation in Bangladesh and the concern with drinking water supply world-wide, brought on in part by the United Nations declared Decade of Sanitation and Water Supply in 1980, has resulted in increased testing and documentation of pump performance throughout the world. Although the primary focus has been on drinking water pumps, manual irrigation pumps, such as the Treadle, Rower and No. 6 pumps used in Bangladesh, have received attention from regional NGOs, farmers and small business interests.

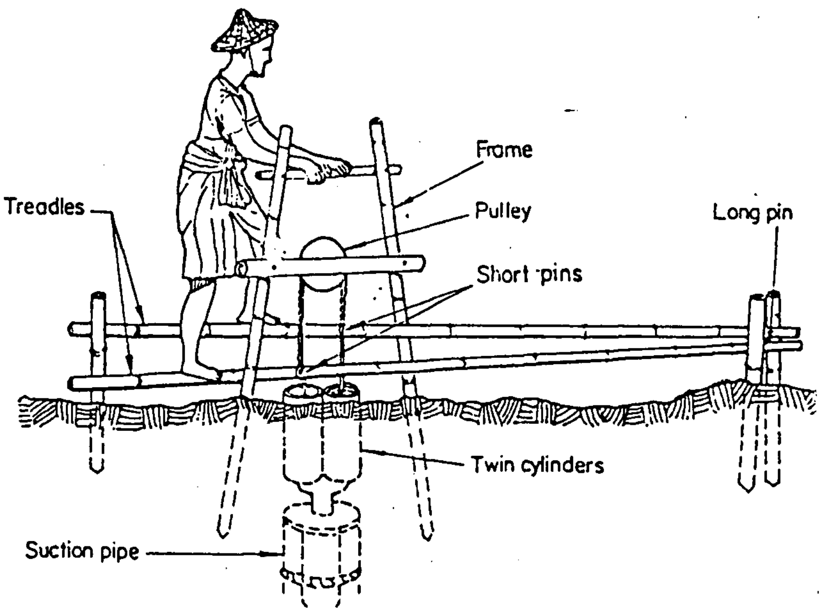

Figure 3: Rope and pulley type Treadle operation.
(Source: Barnes 1985)

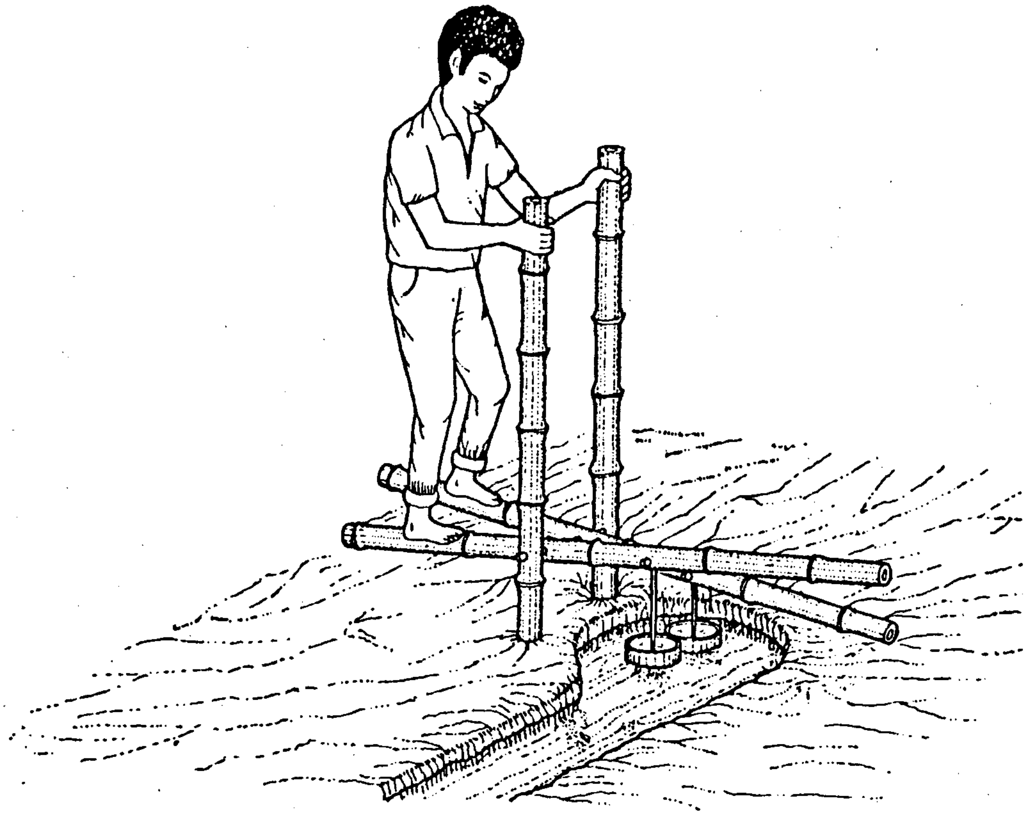

Figure 4: Centrally pivoted "dheki" type Treadle operation. (Source: Stickney, R.E., et al. 1987)

Over the past four years, the Treadle pump has been tested by RDRS, the Mennonite Central Committee (MCC), the International Rice Research Institute (IRRI), the Bangladesh University of Engineering and Technology (BUET), and most recently by Engineering and Planning Consultants Limited of Dhaka, Bangladesh (EPC) on contract with the Bangladesh Rural Development Board (BRDB) and the World Bank.

The five tests have varied in focus as follows:

1) The RDRS tests dealt with depth to water table to discharge relationship of the original pump design and larger piston diameter prototypes for use at shallower depths (RDRS 1983).

2) The MCC tests focused on the impact of pump installation parameters, such as tubewell length, diameter and filter length, on the Rower, Treadle and No. 6 pumps from a primarily statistical analysis of a small size test sample (Spare and Pritchard 1981).

3) The IRRI tests focused on the adaptation and initial promotion of the Treadle pump in the Philippines. Comparisons of the power input requirements for the rope and pulley and the centrally pivoted "dheki" styles of Treadle pump installations, shown in figures 3 and 4, indicated no significant differences between them (Stickney et al. 1987). The "dheki" style operation was, however, found to more comfortable to operate, which is also the reason for the large increase in "dheki" style installations over rope and pulley style installations in Bangladesh.

4) The BUET tests concentrated on the relation of power efficiency and discharge to operator characteristics (Bureau of Research Testing and Consultation. 1986). The use of hired urban operators (rickshaw operators) in the laboratory tests, the lack of comparable field tests, the use of observational recording of force transducer readings to calculate power input at the pump, and the lack of

theoretical analysis to support the laboratory data, all reduced confidence in the report's conclusions.

5) The EPC report focused on the field performance of the Treadle, Rower and No.6 pumps. The tests were commissioned to provide a basis for planning of irrigation pump promotion as part of overall agricultural development planning (EPC 1988) for the BRDB and the World Bank. The pumps were field tested for two months, at four locations, to compare the discharge, pump wear and operator input characteristics of each pump type. The final report concluded that the Treadle pump be recommended for manual irrigation installations on the grounds of lowest cost, ease of repair and large discharge for a given amount of operator power. Of particular interest are the discharge results, with an average of 43 litres per minute at a water table depth of 3.36 meters. This is achieved with a maximum power input of 69.73 watts over a 20 minute period or 51.6 watts over 30 minutes of sustained pumping. The power input values were estimated from the operator metabolism parameter of body surface area (EPC 1988, Table 12, sec. 3.18).

The discharge results were much less than those presented by BUET. The two reports report similar power input measurements but the results are not comparable. The EPC results were based on the measurement of the metabolic rate of the operator, which would give the power input to the treadle levers (EPC 1988, sec. 3.18) while the BUET tests measured the average force input to the piston rod giving the power input to the pump (Bureau of Research Testing and Consultation, sec. 3.3). The power at the operator and at the pump can not be equal due to friction losses between the treadle levers, axle pin and between the rope and pulley system used with these tests, as well as unmonitored metabolic losses. Given the differences between the discharge measurements, power input results, the more representative field test conditions and the large number of tests, the EPC report seems to provide a more complete indication of field Treadle pump performance.

All of the Treadle pump tests to date used the original design rope-pulley style treadle lever configuration, shown in figure 3, as opposed to the centrally pivoted "dheki" style configuration used on most installations since 1986, shown in figures 3,4 and 6.

The "dheki" style operation has been adopted by the Bangladeshi farmer in an effort to reduce the number of moving parts subjected to wear and requiring periodic replacement and to increase operator comfort. Unfortunately, the shorter stroke lengths of the "dheki" style results in lower discharge and greater losses with the increases in comfort. The increase in operator comfort and ease of operation has been obtained from the change of pumping style to a shifting of body weight from side to side, pivoting at the hips and without bending the knees, from the the longer bent-knee step-like operation of rope and pulley style. The large popularity of the "dheki" style operation clearly illustrates the greater importance of comfort and ease of operation to the Bangladeshi pump operator compared to the mechanical efficiency of the pump mechanism. This priority of operator comfort over mechanical efficiency in the case of the operation configuration, clearly shows that charateristics of the Treadle pump that must be carefully considered in any design alterations extend beyond the purely technical performance characteristics.

The lower discharge data given in the EPC report is contrary to previous pump performance assumptions. The lack of agreement between theoretical analysis, laboratory and field test results in all of the above reports precludes their use as a basis for design improvements and comparisons of alternative pump designs.

The need for further technical knowledge was identified in 1988 by the MPG, an organization comprised of manufacturers and marketers of manual irrigation pumps in Bangladesh.

3 DESCRIPTION OF THE TREADLE PUMP

3.1 PHYSICAL DESCRIPTION

The Treadle pump is a two cylinder manual suction pump constructed from sheet steel. It is typically mounted on a single 40 mm (1.5 inch) diameter tubewell, also referred to as the rising main or riser. Attached to the bottom of the rising main is a slotted unplasticized polyvinyl-chloride (uPVC) or poly-propylene cloth covered slotted bamboo screen located below the water table, as shown in figure 5.

The pump body consists of two parallel steel cylinders made from rolled and welded sheet attached to the top of a Y or box shaped junction. A short steel 40 mm (1.5 inch) pipe is welded to the bottom of the junction to connect the pump to the uPVC or bamboo tubewell.

Plastic tubewell components predominate in the more recent pump installations in most of Bangladesh due to the superior durability and sealing of the riser, although the cost of the plastic components are greater than the bamboo components.

Two designs of piston were tested:

(a) The original two-plate piston design which consists of a perforated plate mounted on a steel pump rod above a solid plate with a plasticized poly-vinyl chloride (pPVC) cup seal

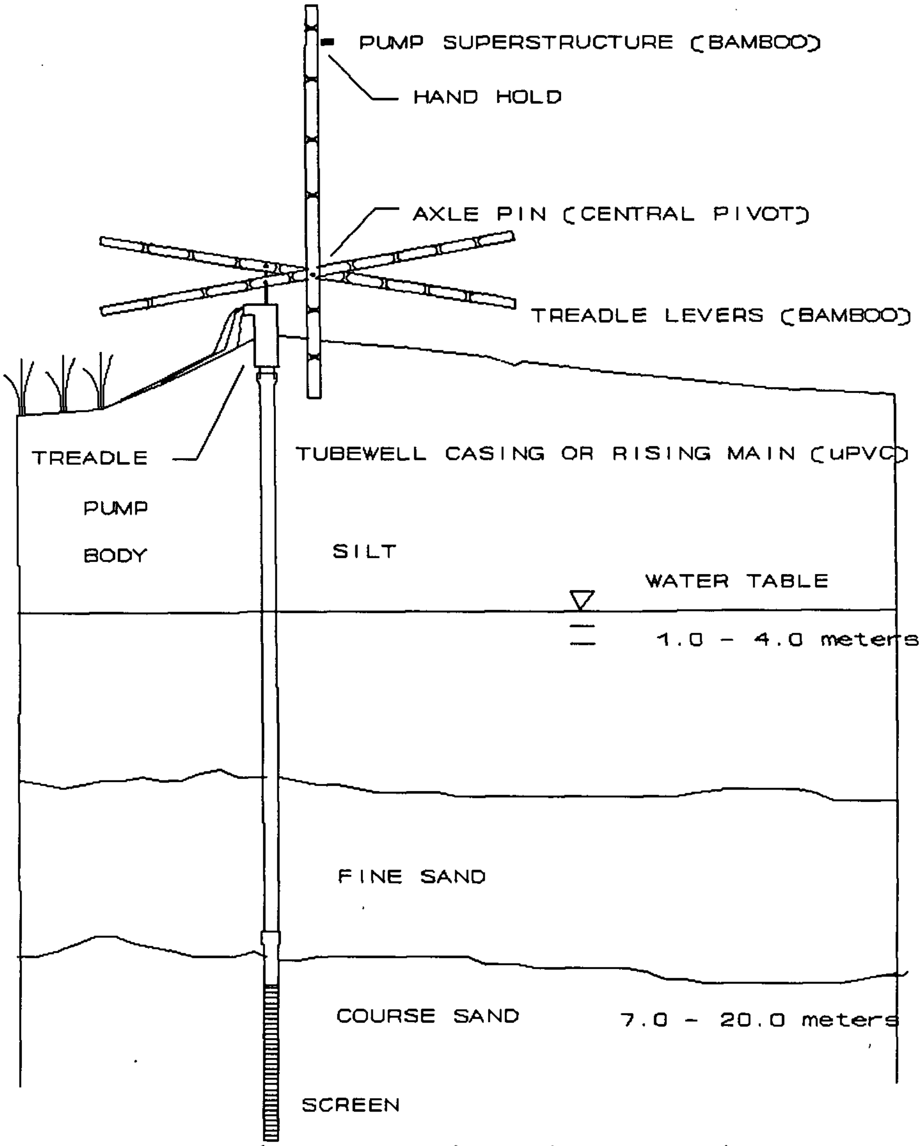

Figure 5: "Dheki" style field installation of the Treadle pump.

placed between them and,

(b) the No. 6 piston, adapted for use in the Treadle pump. The No. 6 piston consists of a molded uPVC poppet valve, used with the same type of pPVC cup seal as used in the two plate design, sandwiched between a threaded upper cage, attached to the pump rod, and a similarly threaded lower plate. The poppet valve, seal and valve seat are configured as shown in figure 6. It is called the No.6 piston because of its use in the widely used cast iron Number 6 drinking water pump, which is similar to the hand pump still used for most manual pump applications in North America.

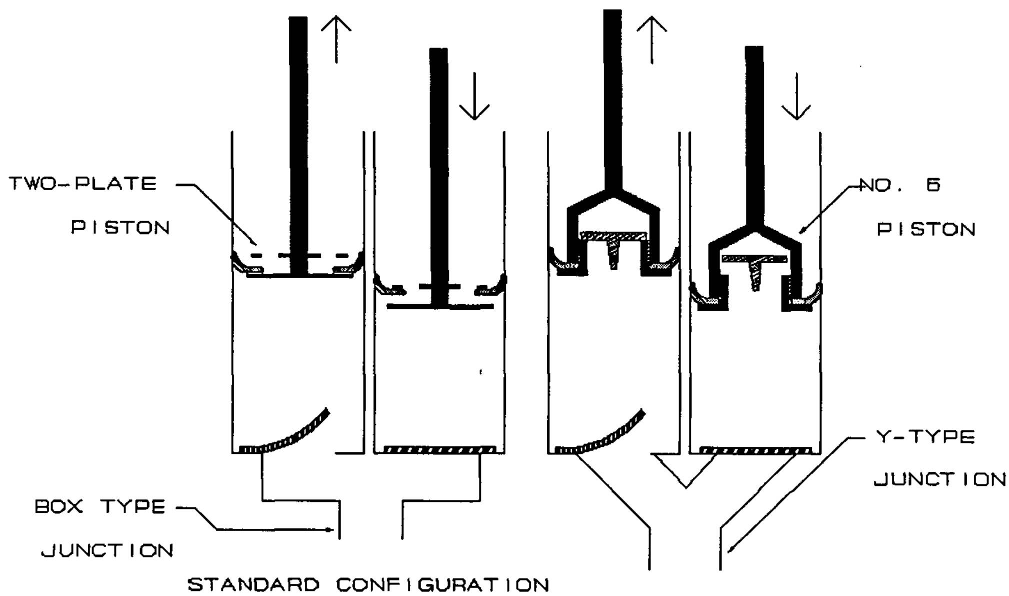

Figure 6: The two piston configurations, the two-plate design and the No. 6 style piston, and the two types of junctions.

Both piston types are connected to the horizontal bamboo rods, called treadles, by a hook and pin arrangement at the top of the piston rod.

The lower valves, or foot valves, consist of weighted inner-tube rubber flaps mounted at the base of the cylinders, where the cylinders join to the smaller diameter junction box. Connection of the flap to the pump body is by nuts and bolts to the cylinder base plate.

The box shaped junction and the two-plate piston design are the standard configuration used in Bangladesh due to the low cost and ease of manufacture.

As shown in figure 7, the pump is operated by the treading action of the operator on the opposite side of the central pivot from the pump. The energy exerted by the operator is used in three ways, to pump water, to overcome friction at the central pivot and the piston rod connections, and to increase the potential energy of the treadle levers by lifting the treadle mass on the pump side of the pivot not balanced by the mass on the operator side. This difference in treadle mass on the pump side of the central pivot is termed the unbalanced treadle mass. If the pump is properly set up, the energy used to raise the treadle lever will be equal the energy required for the downward stroke of the piston and minimal energy is wasted. In actual practice, many of the pumps are installed with increased unbalanced treadle mass to increase the upward pressure on the operator's foot during the return stroke of the piston. This increases operator control over the loose action of the lever and increases the speed of the return stroke. In extreme cases, the levers may be set up so that the end of the levers on the operator's side of the central pivot touch the ground at the lower end of the operator stroke and hit the pump at the upward end. Although this results in wasted energy as a result of hitting the pump and ground and also increases wear on the treadle levers, pivots and the pump body resulting in rough operation, it has the benifit that it decreases the sensitivity and skill required of the operator to obtain the correct balance in the treadle levers and the more optimal smooth operation. This shows that greater simplicity of operation can take precedence over efficiency and smoothness with some operators.

The pump can also be operated using two operators, one on each side of the superstructure, but this is not common practice.

Figure 7: Field operation of the Treadle pump in Bangladesh. (Photo by Author)

3.2 PUMP OPERATION

There are four distinct stages in the stroke cycle of the Treadle pump, which consists of a single up and down stroke cycle in both cylinders, with a half cycle phase difference between cylinders. The stage of the cycle can be defined by the position and direction of movement of the piston at any point in time. The four stages, for one cylinder in the pump cycle using the two-plate piston, is shown in figure 8.

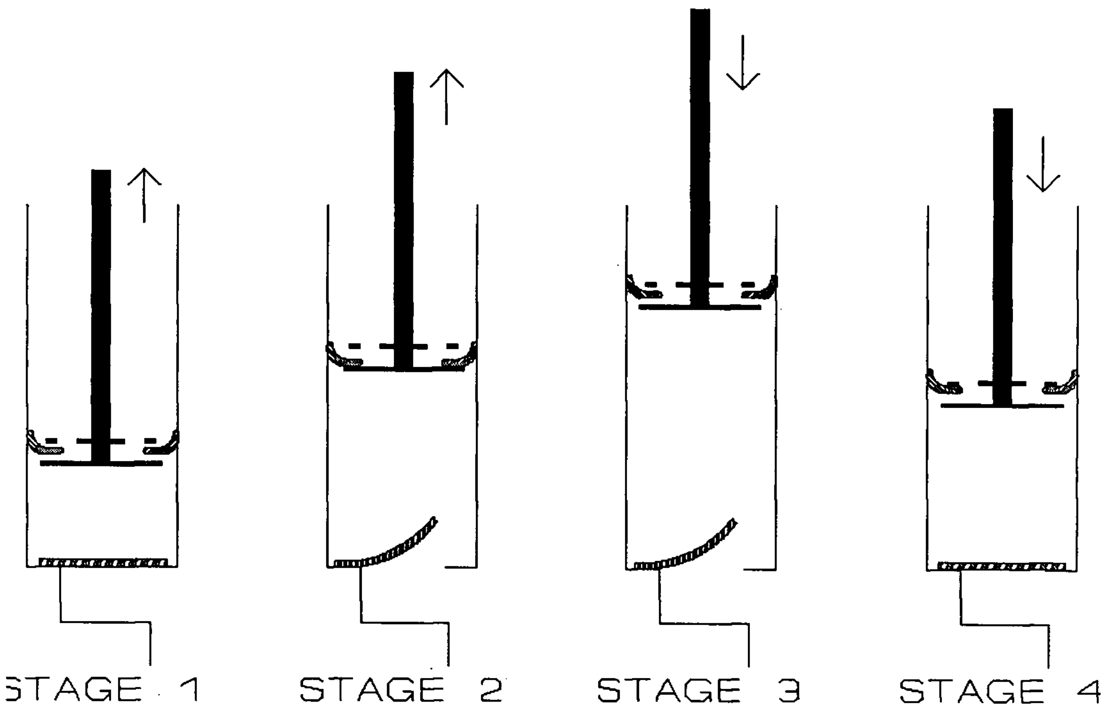

Figure 8: The pump cycle defined by valve positions.

In the <u>first</u> stage, occurring with the initial upward movement of the piston from the lowered position, the piston valve closes with the upward movement of the piston plate to connect with the pPVC cup seal. When no water is lifted above the piston by the piston movement, only the friction force of the cup seal moving against the cylinder and the drag force of the piston moving upward through the water are exerted on the piston. The power input in this stage is very small.

In the <u>second</u> stage, further upward movement causes a decrease in pressure immediately below the piston and the lifting of water above the piston to the discharge spout. The combination of the low pressure below the piston and atmospheric pressure acting on the water table results in a net positive differential pressure, causing flow from the water table through the rising main and into the cylinder. The initial flow, with the closing of the piston valve, causes the foot valve to open and remain open for the duration of the upward stroke. Force is required to produce the low pressure below the piston and to overcome both the piston seal friction force and the drag force through the foot-valve, and this force remains close to constant throughout this stroke stage. The additional force required to accelerate the water in the rising main to equal the motion of the piston and to lift the water to the discharge spout from above the piston are at a maximum at the beginning of this stage with a reduction towards the top of the stroke. The reduction in force is a result of reduced piston and water column acceleration with the slowing of the piston toward the top of the stroke and the reduction of the mass accelerated with the discharge of the water from above the piston. An additional force is required in this stage to accelerate and lift the unbalanced mass of the treadle lever, used as the actuating force in stages three and four, the return stroke. The magnitude of this force depends on the position of the central pivot on the treadle levers. The degree of unbalance results from the quality of installation and, ideally, should nearly equal the force required to move the piston downward during the return stroke. The major portion of work and the maximum power is imparted to the pump in this stage.

In stage <u>three</u>, the downward return stroke begins with the opening of the piston valve. The upwards movement of the valve is limited by the upper perforated plate. This results in a differential pressure across the piston and forces water to flow through the perforated plate and the annulus between the lower plate and the cylinder wall previously sealed by the pPVC cup seal. The downward movement of the water in the cylinder caused by the increased pressure causes the foot valve to close, stopping the reverse flow of water from the cylinder into the rising main. Only the friction force of the cup seal resulting from the piston motion, the drag force of the water through the piston, and the initial drag force through the foot-valve, need to be overcome during this stage. The force for the downward movement of the piston in stages three and four is provided by the unbalanced mass of the treadle lever which was raised in stages one and two.

In stage <u>four</u>, the piston returns to the bottom of the stroke, inhibited only by the resistance of flow through the piston valve and the friction of the cup seal against the cylinder wall. Some water is discharged by displacement from the piston and piston rod.

The second cylinder follows the same cycle as the first, but is offset one half cycle, resulting in a continuous flow of water from the pump.

4 PUMP OPERATION MODEL

4.1 OPERATIONAL CONSTRAINTS

As briefly described in section three, the operation of the Treadle pump utilizes the difference between atmospheric pressure, and the low pressure created below the piston to provide flow from the water table to the pump. Atmospheric pressure, at 20 degrees Celsius, is 101 kPa. When expressed in height of water over a unit area, termed atmospheric head, is equal to 10.23 meters. The pressure differential between atmospheric and that produced below the piston is termed unbalanced head (UH). Ideally, when the depth to the water table equals atmospheric head, the unbalanced head and resulting lifting force are reduced to zero. This results in the theoretical maximum height for the operation of the Treadle pump being constrained to the 10.23 meter limit of atmospheric head.

In practice, the actual vapour pressure of water and the use of a portion of the unbalanced head to overcome turbulence, friction and to accelerate the water column with the piston, the limit of water table depth to which the unbalanced head remains positive is about 7 meters (World Health Organization 1977).

Further constraints on the operation of the Treadle pump include the limits of operator capabilities such as power input, which varies with weight, diet, conditioning, health and environmental conditions such as temperature and humidity. The condition and design of the pump seals and valves, which can influence the amount of sliding friction and leakage past the piston and the amount of suction head attainable, also constrain pump operation.

4.2 CALCULATION OF UNBALANCED HEAD

4.2.1 MINOR LOSSES

Minor losses are those caused by the friction and turbulence of water against components of the pump system. Included are those from when it enters the pump system at the filter until it is discharged at the pump spout.

The losses from turbulent flow through the expansions, contractions and fittings were calculated using tabulated loss coefficients (Roberson and Crowe 1985, 377), calculated as a proportion of velocity head, and added to the total head loss of the system.

Losses resulting from the friction between the water flow and the uPVC riser pipe and the steel pump body were calculated using the Darcy-Weisbach equation with a friction factor of 0.005 derived from the Moody diagram (Roberson and Crowe 1985, 368).

Although there are variations in installation such as riser pipe and filter qualities that can affect the amount losses that occur, the losses in the laboratory test apparatus were kept constant from test to test using a single riser to pump system. This allowed comparison of the measured losses to those calculated.

4.2.2 VALVE TIMING AND LEAKAGE

Valve timing and leakage in a constant differential manual pump system affects pump performance by decreasing discharge for a given energy input.

In the Treadle pump, delays in piston valve closure results in the operator exerting effort to the pump without lifting any water and reduces the volume of water above the piston for discharge. This delay reduces the force required, with the reduction in discharge, but consequently increases the time pumping time required to meet the irrigation requirements. Delays in foot valve closure allow water in the piston to return to the riser pipe reducing the discharge on the following stroke. This has the effect of increasing the energy requirement, as a reduced amount of water remains in the pump as compared to the amount for which energy was expended.

Leakage through the valves and past the piston cup seal also reduces pump efficiency. The leakage past the piston cup seal during the upward stroke results in a reduction of the suction head created below the piston and also reduces the volume of water above the piston to be discharged. From examination of the valve lift distances, the valve opening and closing times, and the total losses through laboratory experiments, the proportion of loss through leakage past the piston seal can be compared to that from valve closing delays.

A measure for the amount of water lost by valve delays and leakage is termed volumetric efficiency, defined as the ratio of the volume of water discharged to the volume swept by the piston in the same time period. Although it is possible to obtain volumetric efficiencies of greater than one, from the momentum effects of a water column, it is not common in manual pumps which have small input power and relatively small water column momentum.

4.2.3 INERTIA

The suction head required to accelerate the column of water in the riser pipe is dependent upon the length of the pipe and the acceleration. As the riser length is fixed and sealed, the acceleration of the water column is a direct result of the acceleration of the pistons, but limited by the total unbalanced head available.

The piston motion characteristics of the Treadle pump were assumed to be periodic with respect to time. To find the general shape and timing of the acceleration curve, an

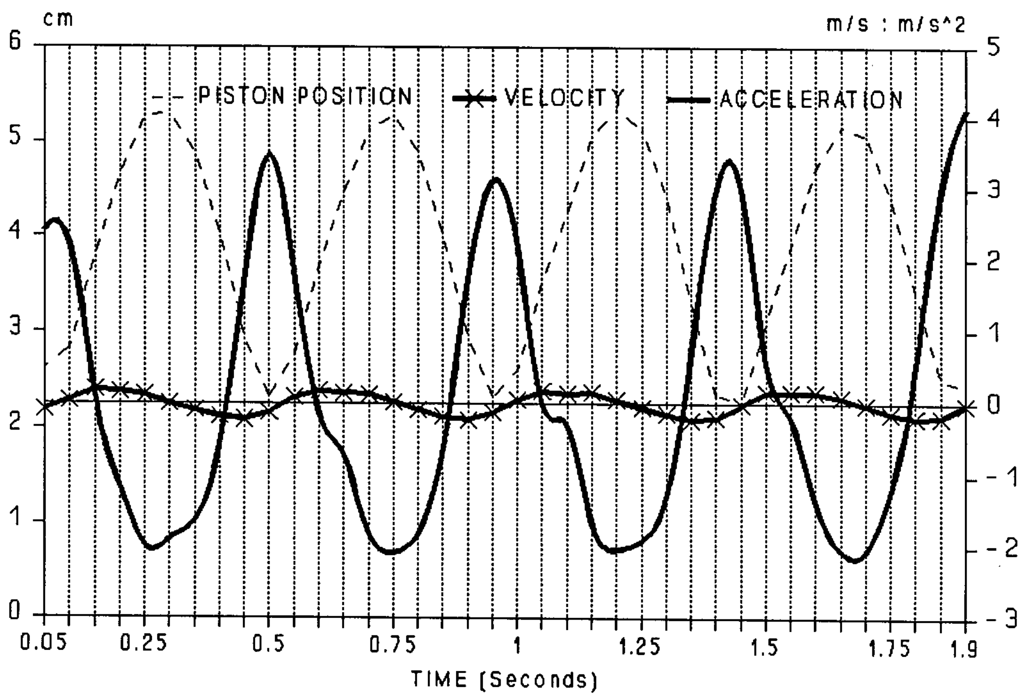

Figure 9: Characteristic piston motion parameters.

approximation of the form:

$$y(t) = y_{avg} + A \sin(wt)$$

where $y(t)$ = piston displacement with respect to time (centimeters)

y_{avg} = median distance of piston travel (cm)

A = Amplitude of piston displacement about y_{avg} (cm).

w = Angular velocity (as if piston was connected to a circular crank mechanism) (radians/second)

t = time (seconds)

was assumed to be sufficient given the motion variations due to operator and pump irregularities. The piston motion, for one piston, measured in the laboratory is shown in figure 9. Irregularities from operator and pump variations are especially evident in the calculated acceleration curve. The small variations in position over time, that are not evident with the resolution of position graph, are revealed. An approximation of the motion shown in figure 9, given below using the assumed sinusoidal form, could be used to mathematically model the pump characteristics.

$$y(t) = 3.6 + 1.5 \sin (14t - 2.6)$$

Although the approximation is not a precise fit to the measured motion parameters, it is sufficiently accurate given the variations in motion from the pump and operator. Mathematically, this should result in a maximum or minimum acceleration, the second derivative of the position equation, when the position of the piston is at a minimum or maximum respectively. At the same minimum or maximum piston position, the velocity, the first derivative of the position equation, should be zero. The plot of <u>laboratory</u> measured position, velocity and acceleration versus time, figure 9, shows close agreement on the maximums and minimums to the those expected from the mathematical derivations. As such, the assumption as to the periodic nature of the Treadle pump motion characteristics was considered valid.

However, for the purposes of accurately predicting the forces in the pump from calculation, inputs for position, velocity and acceleration were taken from the laboratory data rather than from the mathematical expressions. This assured that the calculation of the force inputs could be compared directly with the measured results without any errors induced by breaks or irregularities in the laboratory stroke motion.

4.2.4 MECHANICAL LOSSES

The one mechanical loss accounted for in the pump model was the sliding friction between the pPVC piston seal and the steel cylinder. Variations in the form, consistency and dimensions of the piston cup seals and pump cylinders resulting from the manufacturing process, made analytical calculation of the sliding friction difficult. Given the inexactness of the pump and the inconsistency between pumps, an empirical approximation of 15 newtons, taken from laboratory data of dry pump tests, was adopted.

Other mechanical losses, such as the friction in the hook and pin piston connection to the bamboo treadles, and the friction of the central steel axle treadle pivot against the oscillating treadles, also reduce the power efficiency from the operator to the pump. The separate measurement of these losses was beyond the scope of this thesis but are estimated in the assessment of the total losses occurring in the Treadle pump.

4.3 GOVERNING EQUATIONS

For stage one of the pump cycle, previously described in section 3.2 and shown in figure 8, the force applied to the piston is the sum of the lifting of the piston mass, the sliding friction between the piston seal and the cylinder wall and the drag force from the water flow through the piston. It is assumed that no water above the piston is lifted by the small upward piston movement in this stage which occurs before the piston seal seals against the lower piston plate, i.e. no discharge is produced. The force required to lift the piston is dependent upon the mass of the piston, the unbalanced mass of the treadle lever and the acceleration imparted to them, the coefficient of friction of the seal against the cylinder wall and the velocity of the piston through the water. The equation to calculate the force required is:

$$F = ((M+M_t)*a_p) + (H_d*w*a_p) + F_f \qquad (1)$$

where F = force applied to the piston (newtons)

M = the piston mass (kg)

M_t = unbalanced mass of the treadle lever (kg)

a_p = the acceleration of the piston (m/s^2)

H_d = head loss resulting from the drag forces (m)

F_f = sliding friction force

w = specific weight of water (newtons/m^3)

with

$$H_d = \frac{C_d * V_p^2}{2*g} \qquad (2)$$

where

C_d = drag coefficient

V_p = velocity of the piston (m/s)

g = gravitational constant = 9.81 m/s^2

For stage two, the completion of the upstroke, the piston must produce sufficient suction to overcome the minor losses, accelerate the water column and maintain sufficient unbalanced head to produce flow from the water table to the pump. This suction, expressed in meters of water, is referred to a suction head (H). The force applied to the piston also includes the force required to lift and accelerate the unbalanced treale lever mass, and is calculated using the equation:

$$F = (H*A_p*w) + F_f + ((M+M_t)*a_p) \qquad (3)$$

where

Ap = cross-sectional piston area (m_2)

and

$$H = - WT - h_l - h_d - \frac{a_r * l_r}{g} - \frac{a_p * l_p}{g} \qquad (4)$$

where

H = suction head (m)
AH = atmospheric head (m)
WT = depth to water table (m)
h_l = total minor head losses
h_d = total head loss from drag

$\frac{a_r * l_r}{g}$ = head required to accelerate the water column

$\frac{a_p * l_p}{g}$ = head required to accelerate water in the cylinder

In the third stage, from the end of the upstroke to the closure of the foot-valve, the forces acting on the piston include the friction force of the seal against the cylinder and the drag force, from water escaping through the closing foot valve, at the same rate as the piston downward movement. The force acting on the piston is a result of gravity acting upon the unbalanced mass of the treadle lever raised in stage one and two.

In the fourth stage, the piston continues downward after the closing of the foot-valve. The forces from the sliding friction and the drag of the piston through the water are calculated using the sum of the constant friction force and the drag force from equation (2), with the piston velocity substituted for the riser flow velocity. This force is also supplied by the unbalanced treadle lever mass.

As stated, the forces in stages three and four are not exerted by the operator but rather by gravity acting on the mass of the unbalanced portion of the treadle lever, the piston and the connecting pin. The unbalanced mass of the second treadle is simultaneously being lifted by the operator when the first is exerting downward force.

It was included in the force calculations for the upward portion of the stroke where the force is exerted but because the force transducer was located at the piston rod and measured the force exerted at the piston, not the force exerted by the operator to the treadle levers. The force exerted to raise the unbalanced mass of the treadle lever was included in the force and power input measurements by measuring the force exerted on the downward portion of the stroke by the unbalanced treadle mass. This was added into the power calculations to account for the lifting force not measured on the upward stroke. The method of adding the effort exerted by unbalanced treadle lever mass on the downward stroke does not take into account the force differences between the upward and downward lever acceleration. It was assumed that if the pump is properly installed, the difference between the two is minimal.

From the above equations, calculations for the forces at different piston positions, based on one tenth of a second measurement increments, were conducted with the results shown in figure 9. A water table depth of 3.34 meters, an abrupt inlet, 6.37 meters of uPVC riser pipe with a friction co-efficient of f=0.0005, a box type junction and the two plate standard piston loss characteristics and the motion parameters taken from laboratory tests were used in the calculations.

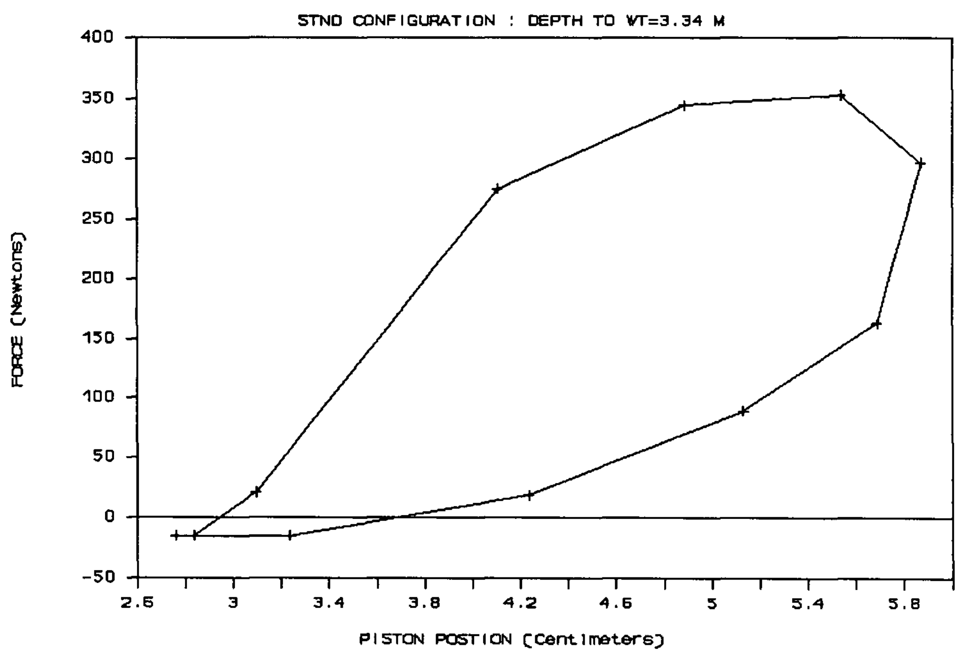

Figure 10: An example of calculated force results with laboratory measured input parameters.

5 LABORATORY TESTING

5.1 APPARATUS DESIGN CONSIDERATIONS

The laboratory apparatus design required a careful mix of strength and sensitivity to balance the imprecise characteristics of the Treadle pump, resulting from manufacture and design for operation in Bangladesh, and the precision required for accurate measurement of performance characteristics. Obtaining a measurement accuracy of twenty newtons, without the destruction of the sensitive measuring apparatus by the large force and displacement fluctuations inherent in the pump, involved the design of a specialized force transducer.

This was achieved through repeated design and testing of alternate designs over a period of three months to obtain the correct balance between sensitivity and durability. The addition of signal amplification and conditioning finally enabled the testing to be conducted within the required parameters. In addition, the use of a highly stable superstructure and wide range input transducers for linear motion measurement was also necessary to ensure accuracy and a high degree of sensitivity.

The modified steel superstructure, pump attachment and well structure increased stability compared to the bamboo commonly used in Bangladesh. Used in conjunction with pumps, seals and pistons unaltered from manufacture in Bangladesh, the force and displacement fluctuations were minimized without compromising the operating characteristics of the pump.

Force inputs to the pump via the pistons were measured using a tubular strain gauge force transducer with reduced

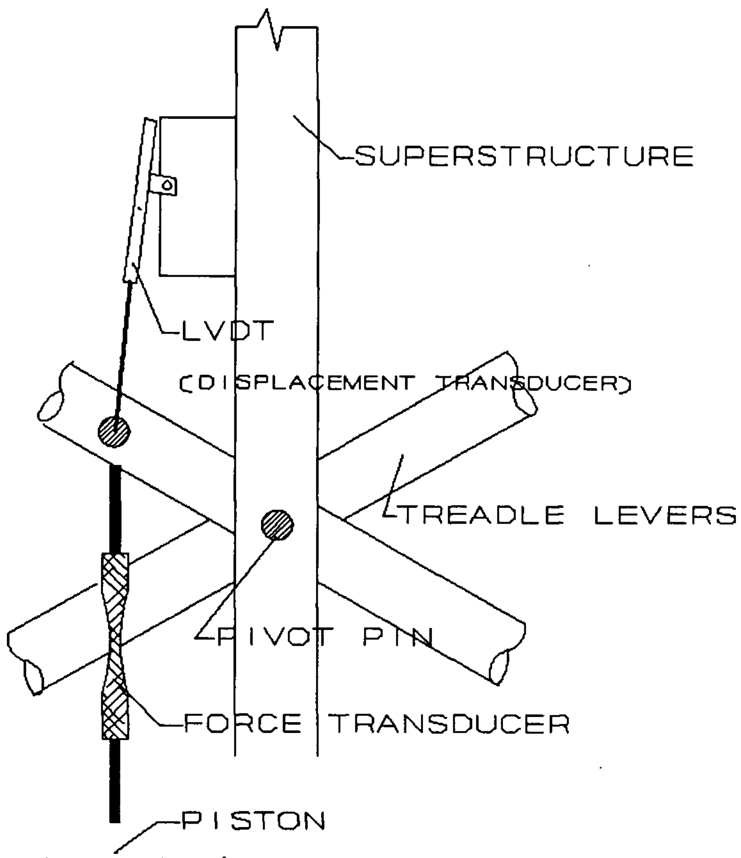

Figure 11: Instrumentation for the laboratory tests.

wall thickness to obtain maximum sensitivity mounted as part of the piston rod as shown in figure 11. The tubular transducer was machined from aluminum, providing maximum stability against bending and buckling from large loads but retaining the sensitivity required to accurately read the small force changes occurring during the pump cycle. Two diametrically opposed strain gauges were affixed to the transducer and connected in opposite arms of the wheatstone bridge circuit.

This cancelled the bending forces resulting from the often eccentric nature of the piston to cylinder sliding forces, the roughness of the hook pin connection of the piston to the treadle and the arced motion of the centrally pivoted treadle lever which resulted in angular displacement of the piston through the pump cycle. The force transducer was also temperature compensated using two 120 ohm resistors mounted beside the transducer on the pump frame and connected to adjacent arms of the bridge circuit from the strain gauges. The bridge ciruit was balanced by the use of a variable resistance circuit connected in series with one of the temperature compensating resistors.

The stroke position was instrumented using a linear variable distance transducer (LVDT) mounted between the right hand side (from the operator) piston pin and the vertical upright posts of the pump structure. It was

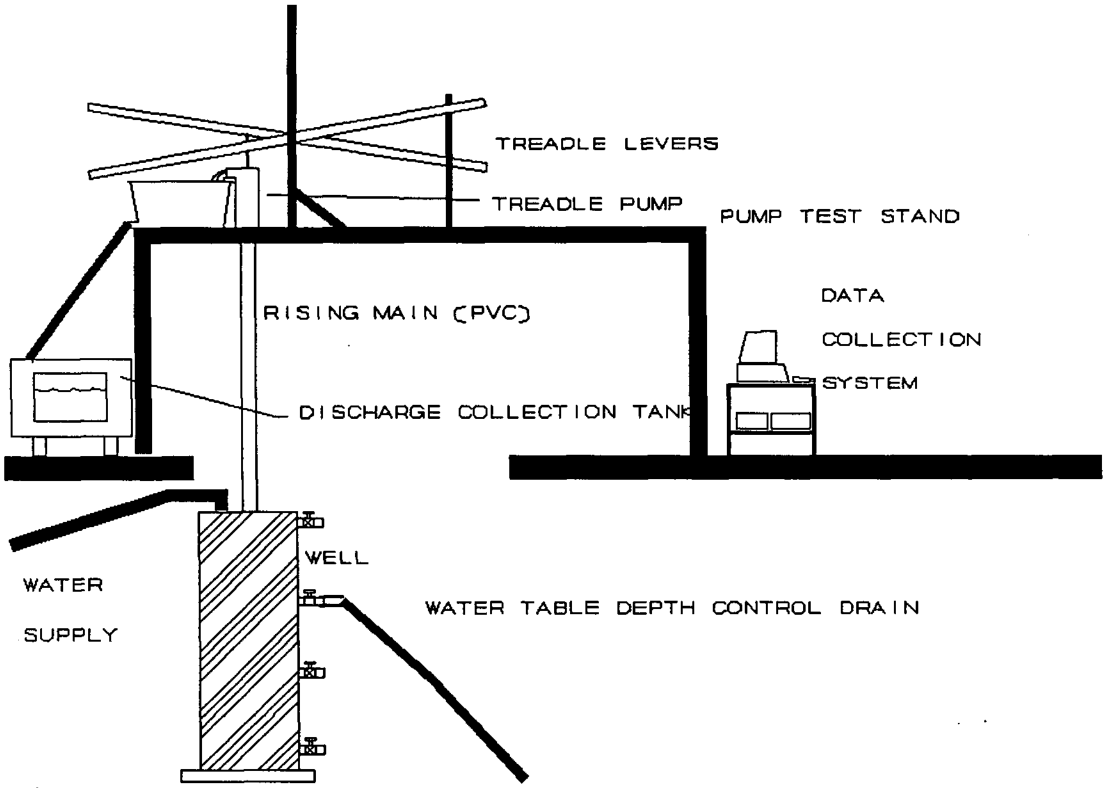

Figure 12: Laboratory apparatus.

mounted with pivot connections at both ends to prevent bending of the transducer due to the arc of the treadle lever throughout the stroke. The LVDT was supplied with a ten volt excitation voltage from a regulated direct current power supply.

The data from the transducers was conditioned using an amplifier to boost the force transducer signal 1000 times, to a level usable by the analog to digital converter. The analog to digital converter allowed for data collection and storage by a personal computer. Using a Basic language computer program, data input timing was controlled and data was collected from the analog to digital conversion board into the computer memory at a rate of 10 readings per second.

The use of a high collection rate enabled the force peaks and delays at the pistion to be detected and analyzed but resulted in very large data sets. Analysis of the data using spreadsheet based programs became very time consuming and was somewhat limited due to the large size of each of the 64 data sets. The duration of each test was limited to five minutes, at the ten hertz collection rate, by the amount of internal memory available in the personal computer and by the storage media where the data was stored for later processing. The Basic language program was also used to convert the data from raw converter readings to data suitable output for later analysis utilizing previously measured calibration factors. The overall testing structure is shown in figure 12.

Using free weights and a measuring scale, the force and distance instrumentation was initially calibrated and the calibration factors re-checked after every four tests.

5.2 TESTING PROCEDURE

The five steps of the procedure for the laboratory tests were:

1) Inspection of pump components and trial pump operation to check for worn out components, any problems such as leakage and to obtain uniform pump operation.

2) The computer data collection channels were zeroed where appropriate and the initial level of water in the discharge collection tank was read and recorded.

3) With data collection started, the zero readings for the force and position transducers were recorded followed by priming of the pump. After a pause, to indicate the beginning of discharge in the collected data set, the pump was operated for between 4 and 6 minutes, during which time the data was collected from the force transducer, the LVDT, the LVDT and strain gauge voltage supplies and the computer timer at a rate of ten readings per second.

4) On completion of pumping, the data collection program was stopped and the data down loaded to a data file on floppy disk for conversion from raw readings and analysis using a spreadsheet based program.

5) The water level in the discharge collection tank was read and recorded along with the number of readings taken by the data collection system and the elapsed real time for comparison and error checking of the computer collected data.

The total time that data could be collected in one test was constrained by the internal memory limits of the computer used and by the size of data file storable on a single data storage diskette and readable by the spreadsheet program used for data analysis. Collection times longer than five minutes were found to exceed these constraints.

5.3 DATA REDUCTION AND ANALYSIS

The data files resulting from the data collection computer program were transferred to a spreadsheet program using floppy disks. Using the spreadsheet functions, the data was imported, reduced and analyzed using the following steps:

1) The first portion of collected data, taken with the pump at rest, was averaged to obtain a raw data reading equating to the zero point for each collected variable. These initial zero readings were also used to check for deviations in data caused by variations in excitation voltage supply or circuitry voltage variation. If the average of the data collected with the pump at rest varied more than 10% from the average of the complete data set, normalized to the average supply voltage, then the test was repeated. Usually only the initial test in a given series was prone to large supply voltage and subsequent data variations due to circuitry heating within the voltage supply, force transducer amplifier and strain gauge bridge circuits. Small variations in the data set about the average were permitted, as the area under the force-displacement curve remained constant. The small positive

and negative variations cancel each other out in the calculation of the overall area encompassed by the curve.

2) If the data was within the acceptable limits for variation, then the zero readings were recorded and the data collected during zeroing, priming and between the completion of pumping and the stopping of data collection, was removed.

3) The force and position data was then converted to appropriate units from the raw data readings, using the calibration factors and zero point data.

4) Calculations for work and power were then calculated on a time incremental basis from the piston position, time and force inputs. The calculation of power used to lift the unbalanced treadle lever mass during the piston upstroke, later providing the force for the downstroke, was calculated from the force measured at the piston during the downstroke and added to that calculated for the upstroke. The stroke rate, total swept volume and average stroke length was calculated using the entire data set.

Using the discharge data, calculated from the readings taken on the calibrated discharge collection tank, the piston position and timing values, the volumetric efficiency, power output and, using the average of the calculated power values, power efficiency results were calculated.

6 FIELD TESTING

6.1 APPARATUS DESIGN CONSIDERATIONS

The apparatus design for the field tests was constrained by four factors.

Firstly, the limited accessibility of the test sites. All equipment and personnel reached the sites by a combination of river boats, motorcycles and walking.

Secondly, the unavailability and unsuitability of sophisticated test equipment such as that used in the laboratory due to extreme environmental conditions and the lack of transportability, repair, component fabrication facilities and power availability.

Thirdly, the variability of pump installations in terms of overall condition, set-up dimensions and operation parameters and fourthly, the limit to disruption of pumping activities for testing in terms of both time and alteration of the pump field situation. Any undue disruption could have had the effect of minor crop damage due to non-irrigation, loss of income for the hired labour operating the pump if hired labour was used, and undue expenditure by local farmers for food and drinks in their efforts to befriend and impress the testers.

The resulting apparatus design utilized two, thirty litre measurement pans, an electrical water level sensor circuit attached to a tape measure, a two hole offset clamp to attach a pen to the piston rod, a clipboard/paper holder and four additional personnel.

As in the laboratory tests, parameters for discharge, stroke length, stroke number and test time needed to be measured. Discharge was measured by alternate filling and dumping of the two measuring pans, stroke length was measured by repeatedly sliding the clipboard held paper along the holder and measuring the resulting traces from a pen or marker mounted to the piston rod. This method for measured stroke length was altered somewhat from pump to pump depending upon the ground surface surrounding the pump, the pump installation, the applicability of using the clipboard holder and the difficulties in affixing the pen or marker to the piston rod. In a few cases, the stroke length was measured by observation of the stroke against a measuring tape as the installation did not allow for affixing a marker to the piston rod without its rapid destruction. Stroke number was counted by one of the additional personnel and the test was timed from start to stop by a stop watch.

The apparatus was simple in design and was found to be very adaptable to variations in test situations and easily understood by the test personnel. The apparatus was labour intensive, by North American standards, but was appropriate for use in Bangladesh, where labour is readily available and less expensive than the cost with increased apparatus complexity. Given the conditions for the testing, the apparatus was found to be appropriate.

6.2 TEST LOCATIONS AND PROCEDURE

Forty-one Treadle pumps were tested in the field tests, ten in the Jamalpur and Sherpur regions in north central Bangladesh, and thirty-three in the Kushtia and Pabna regions in western Bangladesh as shown in figure 11. Research into pump design and alternatives was conducted in the Kurigram region in Northern Bangladesh and in the Shonargoan region just south of Dhaka.

The test procedure varied somewhat from site to site depending upon how the pump was situated and set up in the field and the degree of co-operation from the farmer owner-operator or hired operator. In general, the procedure was as follows:

1) With permission from the owner and operator, soil below the pump spout was removed, if necessary, to permit placement and removal of the discharge measurement pans below the discharge spout.

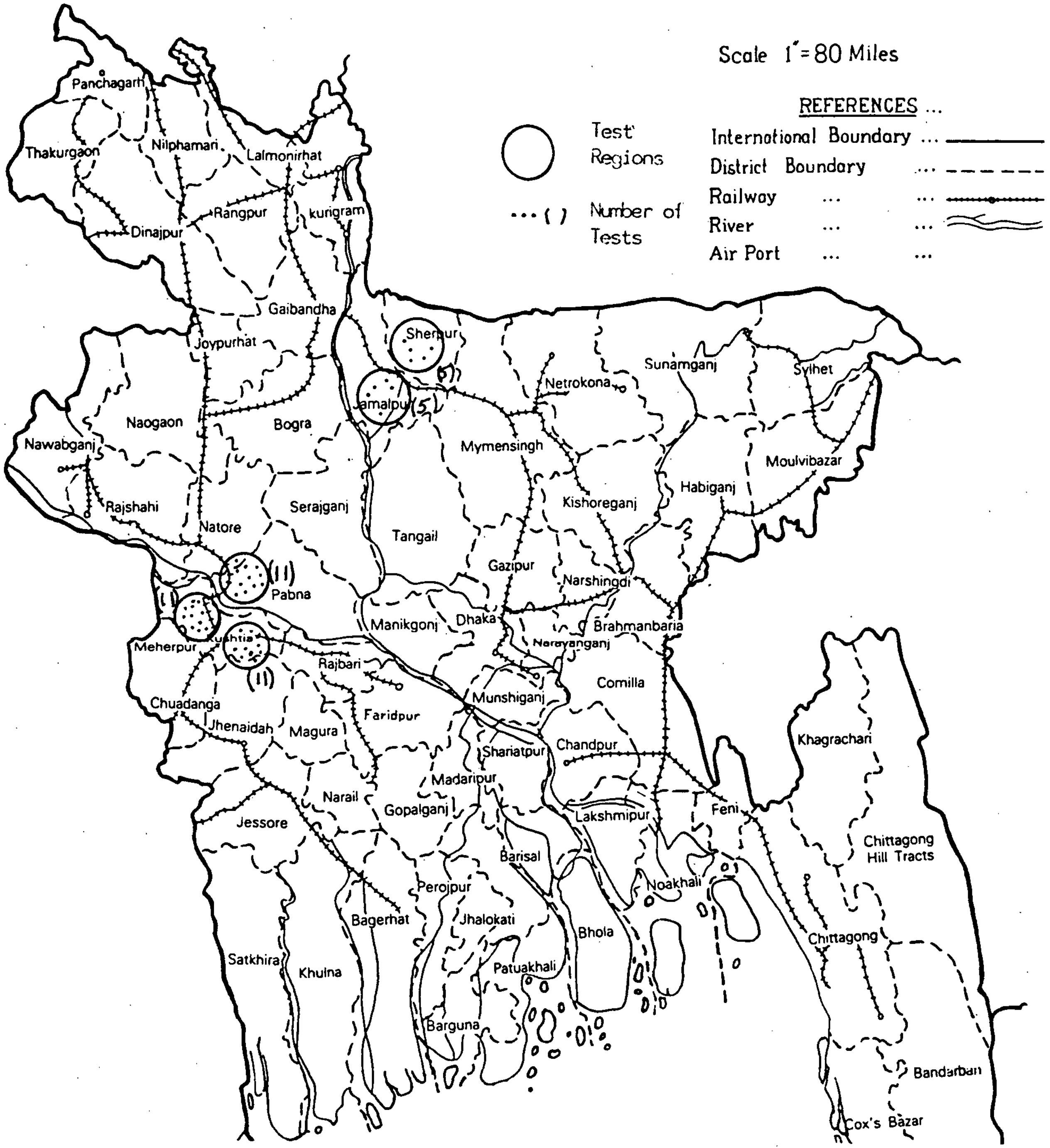

Figure 13: Location of field tests.

2) The owner of the pump was asked about the age of the pump, dates of most recent component replacements, depth and type of riser and filter, and if any problems had been encountered with the pump. The pump operator, often the owner, was asked his or her age, weight and number of hours the pump was operated per day during the dry season to gain insight into operators typically using the pump. The responses were recorded to provide information to assist in design alteration considerations and to signal any significant deviations from typical pump usage which might affect the data collected.

3) The pump cylinders, pistons, cup seals and foot-valves were inspected for signs of undue wear and improper operation. If the components were in such poor condition as to be non-representative of normally used and maintained pumps, then the test was abandoned and the test moved to the next site.

4) After five to ten minutes of pumping to obtain as normal a sustainable pumping rate as possible, the depth to water table was measured using the electrical measuring tape lowered into the rising main through the foot-valve and junction box. The measuring of the water table depth after the initial pumping and draw down resulted a depth that would be consistent with sustained pump operation.

5) The clamping mechanism that attached the pen or marker to the piston rod was attached on the upper portion of the

piston, so as not to interfere with pump operation, and a pen was also attached at right angles to the piston.

6) The operator was instructed to pump as if the test was four hours in duration, to obtain usual sustainable pumping rates. After five minutes of pumping or when the stroke rate slowed and became consistent, the timing and stroke counting was started and the discharge measuring pan was placed under the spout.

7) At least four traces of the stroke motion were made, spaced at about one minute intervals, the discharge measuring pans were removed and emptied when full with as little spillage as possible and the timing and stroke counting were continued. The minimum test duration was four minutes and the maximum was ten minutes.

8) If the stroke rate and length were consistent over the first four minutes of testing, then the data collection was stopped at the next full measuring pan. If not, then the test was continued for as long as 10 minutes with incremental data values recorded to obtain the data for the most consistent pumping.

9) Following completion of the test, the data was recorded on data collection sheets for future analysis. If the pump was mounted on an uPVC riser and filter, then the tests were repeated using the author as the operator for comparison to the laboratory data.

10) Upon completion of the testing, the owner was advised of any repairs or maintenance required on the pump to improve performance and the pump set up was returned to as near its original configuration as possible, unless asked to do otherwise, before leaving the test location.

6.3 DATA ANALYSIS

The data from the field tests had large variations due to differences in pump condition, operator size and motivation and turnover in collection personnel. In analyzing the data, only severe out-liers, those not physically possible, were removed.

The stroke lengths were measured using the vertical distance from the trough to the peak on the trace. The multiple traces for each test were averaged to obtain the stroke length used in calculations to determine the total swept volume necessary for calculating the volumetric efficiency of the pump.

The discharge volume was calculated from the number of discharge pans filled and the respective volumes of the pans. The pan volume was adjusted based on observation of how full the pans were filled during the test and on how much unmeasured discharge occurred on the switching of the pans. Typically this adjustment ranged from a seven to twenty percent reduction in the measured discharge, as the pans were rarely completely full on removal and little water was lost during the switching of the pans. The variation in adjustment was dependent upon the performance of the personal removing and replacing the pans during a particular test, which was recorded for each test.

The volumetric efficiency was calculated using the ratio of the total swept volume, calculated from the number of strokes and the stroke lengths to the measured discharge.

Using the depth to water table and the discharge data collected, the input power to the pump was estimated based on the depth-discharge-power output results from the laboratory tests.

7 RESULTS

7.1 PUMP FORCES

The force displacement curves, generated from the laboratory tests, are useful for measuring the changes in forces applied to the piston throughout the four stages of the piston stroke. These changes are a result of variations in friction, acceleration of the water column in the riser pipe, formation of the unbalanced head, and the use of the potential energy stored in the raised treadle lever mass to provide the energy for the downward piston stroke. Pump configuration, piston and foot-valve design also cause force variations that are of particular interest in pump design.

Taking a detailed look at the initial portion of the force curves, the large forces at the start of the upstroke, evident in figures 14 and 15, result from the large forces required to accelerate the water column in the riser pipe and the water above the piston from rest, or nearly so, to equal the velocity of the piston at that same point in the stroke. In comparing tests using the standard two plate piston and the No.6 piston, shown in figures 14 and 15 respectively, two differences in acceleration forces are evident.

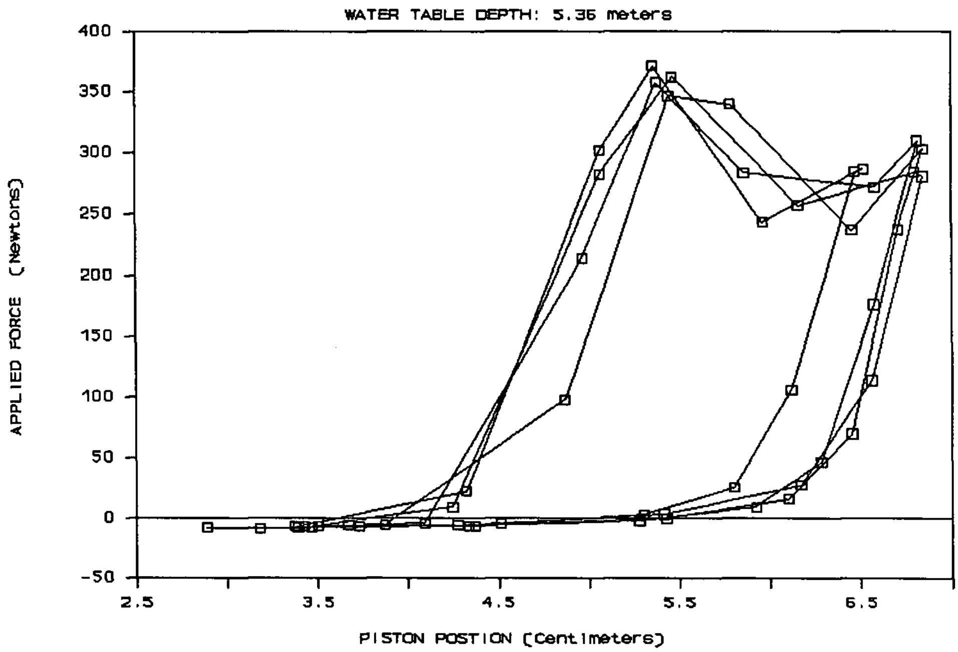

Figure 14: Force-displacement loop using the two-plate type piston.

Firstly, the peak accelerating force occurs much more rapidly after the valve closing in the standard two-plate piston compared to the smaller slope following valve closure with the No. 6 piston. However, the maximum forces resulting from acceleration of the water columns are close to equal, given operator variations, and also occur at the same position in the stroke. The difference in the slopes in the initial portion of the curves is a result of the combined effect of the faster valve closing time of the No.6

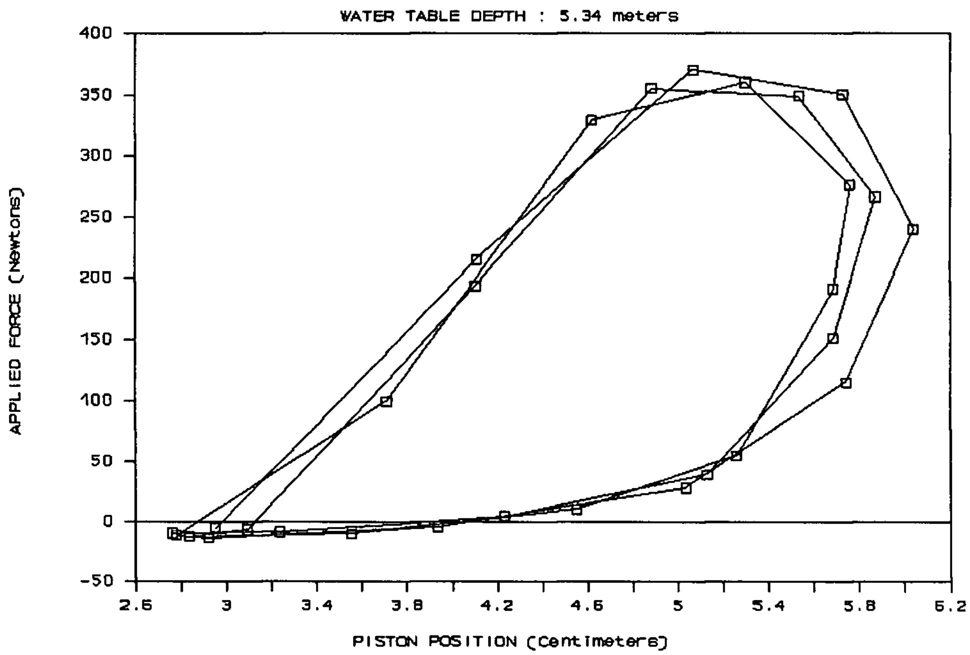

Figure 15: Force-displacement loop using the No.6 type pistons.

piston, indicated by the rapid increase in piston force earlier in the stroke, and the effect of greater initial leakage through and by the No. 6 piston, which is indicated by the flatter slope of the force curve for the No. 6 piston. The equivelent maximum forces indicates that the two pistons seal equally by the end of the stroke.

Secondly, the standard piston shows a sharp decrease in force of 100 to 150 newtons at mid-upstroke followed by an increase in force of 50 to 100 newtons at the completion of the upstroke. This is not evident with the No. 6 piston, which has a more constant force throughout the stroke. Two explanations of this force variation are:

1) That the water column and piston rod exhibit elastic properties resulting from the rapid valve closure and resulting shock forces. This elastic effect is much like a stretching and subsequent rebounding. The elastic effects of a manual pump water column was documented, to explain similar force variation behavior, in the testing of deep set hand-pumps (Yau 1985). Although the documented force variation is similar to that measured on the Treadle pump, the much shorter piston rod of the Treadle pump and the resulting elastic effect of the pump rod is negligible, compared to the ten meter rod used in the analysis by Yau. The results are still useful however. The report (Yau 1985) concluded that the magnitude of the large force increase was required for the design of pump components to withstand the shock loading, but that the overall area under the force displacement curve did not to change. Similarly for the analysis of the Treadle pump force displacement curves, the reductions in piston forces were found to be effectively cancelled by the increases, measured over the entire stroke length, resulting in a consistent work input measurement.

2) The force variation could also be caused by the deformation of the piston seal in the two plate standard piston. This would cause binding of the seal against the cylinder wall as the piston cup seal would deform under the maximum combined forces of head loss, head requirements and acceleration. The unrestricted piston seal is able to rotate about its leading edge, the upper circumference, from the combined moment created by the downward forces of friction, unbalanced suction head and acceleration forces acting on the outer edge of the seal and the upward lifting motion of the piston acting on the inward edge. The large and sudden acceleration force, evident with the standard two plate piston, is followed by a sudden reduction in force, caused by poor sealing and loss of suction from seal deformation. The increase in force a few centimeters later in the stroke could be caused by the cup seal re-sealing against the cylinder and regaining the suction forces. The re-sealing occurs as the downward deforming forces on the seal are reduced and the shape memory inherent with temperature set plastics, such as pPVC, return the seal to its original un-deformed shape. In comparison, deformation of the seal by rotation about its circumference edge is not possible with the No. 6 piston as the seal is solidly fixed between two threaded fittings.

Given that the force reductions and increases through the stroke are not constant with the fixed riser length used in the laboratory tests, and that the No. 6 piston did not show this characteristic curve when maximum sealing and steep acceleration force curves occurred in some tests, the second explanation, that of the deformation of the piston seal, is the most probable. This is not conclusively provable with the test apparatus and results of this thesis.

More importantly, to the improvement of the pump design, the force curve when using the No.6 piston is smoother with less shock loading on the piston than with two-plate design, due to shorter valve closing times and probably less seal deformation. The smoother force transitions also lessen the sudden loads on the pump superstructure, extending the working life and improving structure stability.

7.2 VALVE CLOSURE DELAYS

The opening and closing of the piston and foot valves are evident from the tension or compression forces measured at the piston rod. Examination of the time and distance travelled by the piston from the beginning of a stroke stage to the beginning of associated forces, indicates the delays in valve opening and closing. This is useful in analyzing the proportion of volumetric losses attributable to valve delays as opposed to leakage past the piston seal.

The piston valve closure delay, shown by the more horizontal portion of the force curve at the beginning of the stroke, is about two times greater with the two-plate piston than with the No. 6 piston. The average 1.0 cm stroke length delay of the two-plate piston valve effectively reduces the stroke length used for pumping by 25% in the laboratory tests and an estimated 8% in the field tests. The difference between the laboratory and field tests is a result of the longer stroke lengths of the Bangladeshi operators in the field tests. The proportion of the stroke used for valve closure becomes smaller with the longer stroke. In either case, the valve delay results in reduced discharge and lower volumetric efficiency. The 0.5 cm delay with the No.6 piston reduces the useable stroke length by 12.5 percent and an estimated 4 percent in the laboratory and field tests respectively.

Field Tests	Piston	Foot-valve	Total
Standard Piston	8	13	21
No. 6 Piston	4	13	17
Lab. Tests			
Standard Piston	25	40	65
No.6 Piston	12.5	40	52.5

Table II: Percent loss from piston and foot-valve delays.

It should also be noted that the looseness of the piston to treadle lever connection often results in a longer treadle lever stroke length than piston stroke length. As such, the strokes seemed longer to the operator than those measured.

Delays in foot-valve closing are indicated by the downward distance travelled by the piston before compressive forces which indicate movement of water through the piston rather than through the foot-valve, are exerted on the piston. As seen in figures 14 and 15, the delay in foot-valve closing is large, taking up as much as 1.6 cm of stroke length. This delay represents an average of 40% of the stroke length in the laboratory tests and an estimated average of 13% in the field tests.

The cumulative effects of valve closure delays are given in Table II.

As previously mentioned, the differences in operating styles in the field tests result in the variation between the losses calculated from the laboratory and averaged field test values. The primary cause for the difference is the three times longer average stroke length of 13 cm for the Bangladeshi operated field tests as opposed to the 4 cm length for the author operated laboratory tests. The longer stroke lengths used by the Bangladeshi farmers in the field tests reduces the overall effect of valve closure delays. Although the percentage losses are not as great in the field tests, the closure delays still represent a major loss in the short strokes as compared to the full cylinder length strokes of the original rope and pulley system shown in figure 3. The shift to the "dheki" operating system that has resulted in the shorter stroke lengths was initiated by Bangladeshi farmers to increase operator comfort. They found that the weight shifting, non-knee bending style of operating the "dheki" system was more comfortable than the stepping action of the rope and pulley system. In either system, the operator will move towards or away from the pump to achieve the most comfortable stroke length and power input requirement. With the non-knee bending style of the "dheki" operation, the changes in stroke length are limited by the hip joint motion and as such the range of stroke length is small. For the analysis of the pump, it was assumed that the stroke length is unlikely to increase beyond the extremes measured in the field. For the most part, the longer stroke lengths in the field tests over the laboratory tests are a result of many hours of pumping experience by the Bangladeshi operators. The laboratory operator, operating pumps in Bangladesh, consistently had shorter stroke lengths, further indicating that the difference lies with the operator rather than the pump installation.

7.3 THE EFFECT OF WATER TABLE DEPTH ON APPLIED FORCE

As the force applied to the piston is the sum of providing the unbalanced head to provide water flow, overcoming losses, accelerating the water column and increasing the potential energy of the treadle lever, an increase in water table depth with no increase in overall riser length, affects only the force required to create the unbalanced head needed for water flow. The increase in force, as shown in figure 16, from 230 to 350 newtons, a difference of 120 newtons, with an increase from 3.44 to 5.36 meters respectively, is, for all intents and purposes, equal to the theoretical difference of 123 newtons calculated using equation 1 and 2 from section 4.3, for the same depth increase. This shows that the acceleration and head losses are dependent upon riser length and not the water table depth.

As the area under the force displacement loop equals the work done, the increase in force with an increase in water table depth, increases the work done provided the stroke length and stroke rate are constant.

7.4 THE EFFECT OF WATER TABLE DEPTH ON DISCHARGE AND VOLUMETRIC EFFICIENCY

The effect of water table depth on volumetric efficiency was found to be a reduction of volumetric

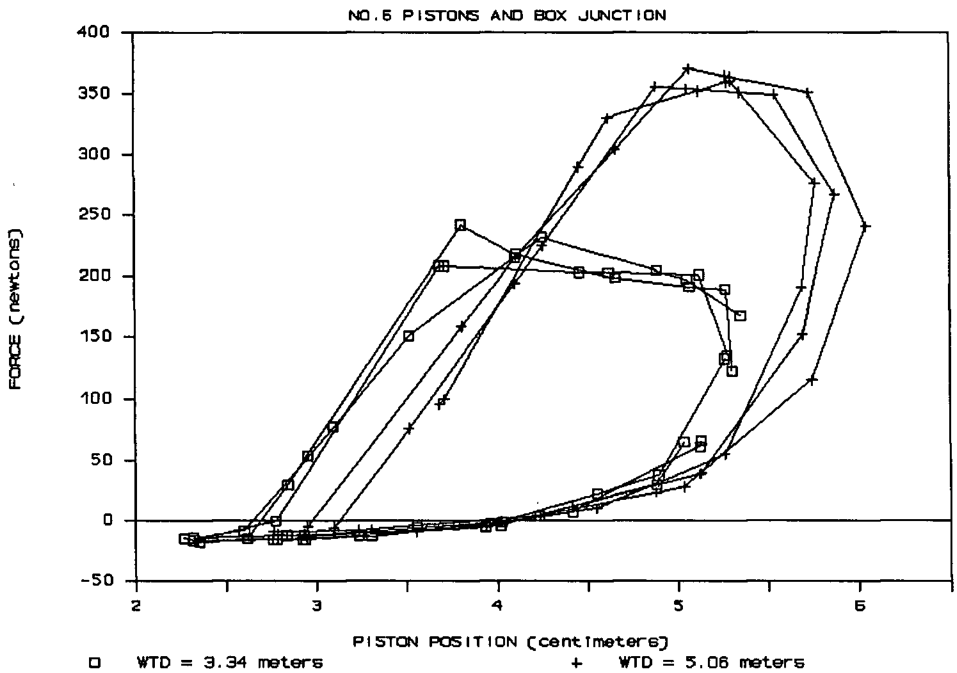

Figure 16: Variations in force with water table depth.

efficiency with increasing depth, as shown in figure 17. This indicates that valve and seal leakage, responsible for poor volumetric efficiency, is a function of water table depth and thus the magnitude of the suction below the piston.

The curves presented for the field tests represent averaged volumetric efficiencies from the raw test results. The values vary from 35 to 73 percent, with averages of 55% percent over a range of water table depths from 2.05 to 4.13 meters. The wide variation was a result of a wide range of

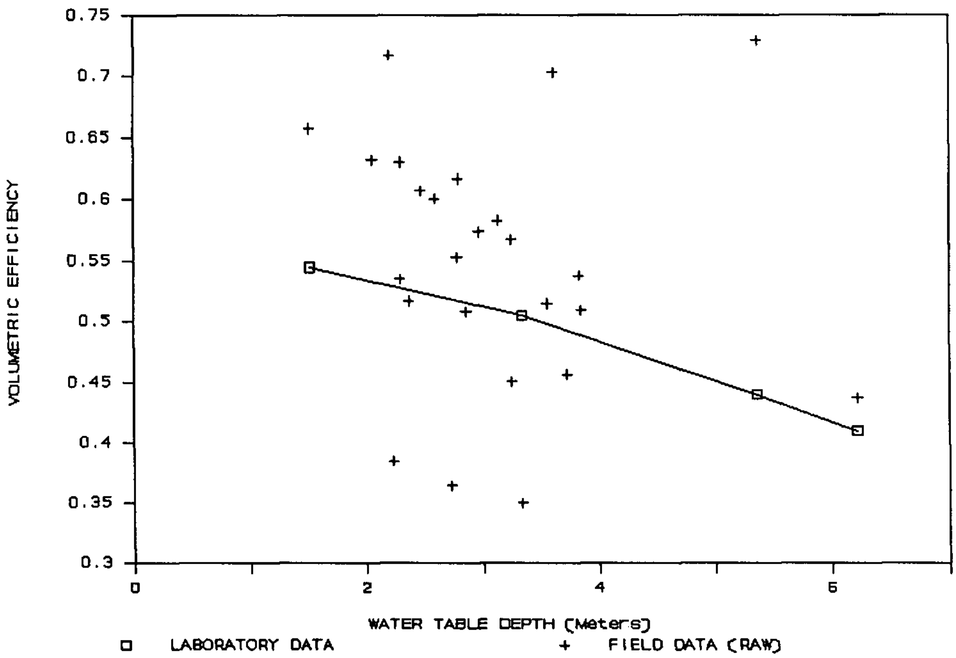

Figure 17: The effect of water table depth on volumetric efficiency.

pump conditions, operational stroke lengths and cylinder conditions, resulting in variations in foot valve and piston seal leakage.

The laboratory results for the same configuration of pump as used in the field, but tested over a greater range of water table depths, indicated volumetric efficiencies ranging from 55 to 41 percent at 1.51 to 6.22 meters respectively. The small reduction in volumetric efficiency through the test range was primarily due to increased leakage and piston and foot-valve delays, as seen by the force-displacement curves. This indicates that the volumetric efficiency characteristics of the pump are more dependent on the motion characteristics of the piston rather than an increase in water table depth.

The longer average stroke in the field tests produced reduced losses due to valve closure delays. The variation in stroke length between operators also caused increased data spread as the longer strokes decreased the proportion of the stroke taken up by valve opening and closings. This effectively increased the volumetric efficiency of the pump. The similarity of the volumetric efficiencies in the laboratory and field tests indicated much greater losses past the piston buckets in the field tests as the proportion of losses due to valve opening and closing were less with the longer stroke length.

7.5 POWER INPUT REQUIREMENTS

The power for the Treadle pump, being supplied by human operators, varies with the condition and surroundings of both the pump and operator. The laboratory tests, using the same pump configuration as tested in the field, were conducted using shorter strokes at roughly the same stroke rate, resulting in lower discharges. The shorter stroke length used in the laboratory was a result of the pump operation being controlled according to operator comfort, as

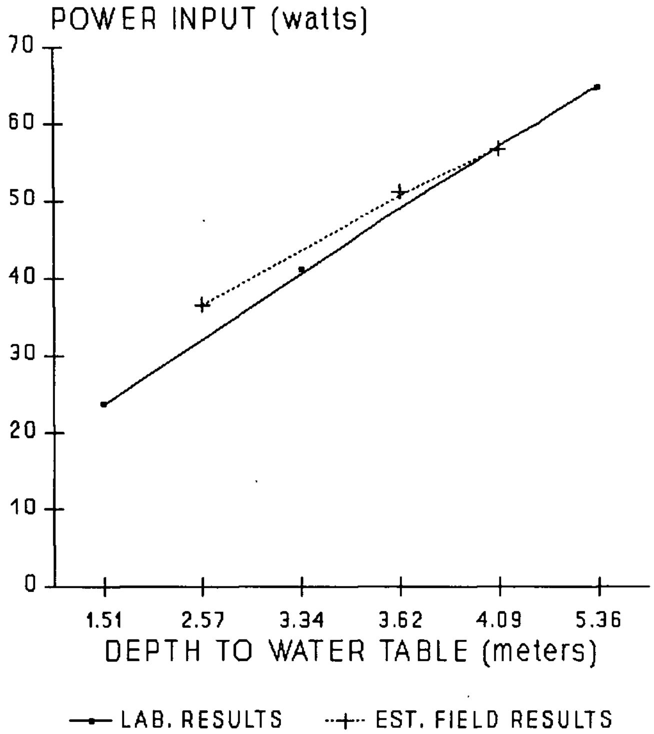

Figure 18: Laboratory and estimated field power input requirements.

The laboratory measured and estimated field test power requirements both indicate that there is an upper maximum power output ability for operators of the Treadle pump. This value was measured as 50 watts at the pump head with some variation with operator and pump installation. Adding the losses resulting from friction in the superstucture not measured by the laboratory instrumentation, the actual limit for power input is closer to 55 watts. Although it is possible to produce much more power than this over a short period of time, the often malnourished Bangladeshi operator, is done by the Bangladeshi farmer. The difference between what was comfortable to the North American laboratory operator and the Bangladeshi farmer operators was the cause of the differences in stroke length. Assuming the losses remained constant between the laboratory tests and the field tests, the power input should be proportional to the discharge in both tests. Using the laboratory discharge and power input measurements, including the power required to raise the treadle lever estimated by the amount of work done by the lever during the piston downstoke, estimates were made for the power input from the field tests, shown in figure 18, using the field discharge measurements. The estimates do not fully take into account the inefficiencies of the superstructure, which include the friction at the central axle pivot and at the piston to treadle lever connection, occurring between the operator and the pistons where the data was collected.

As with the measured laboratory data, the field power inputs rise sharply with an increase in depth to the water table, as a result of the increased force required create the unbalanced head required for water flow from the water table. Corresponding to this increasing power requirement, the pump discharge decreases due to increased losses from pipe friction and leakage through the valves and piston seals as described in section 7.4.

pumping for periods of at least 30 minutes duration and a total daily pumping requirement often between 8 and 10 hours per day, will not exceed this value. The testing by EPC (EPC 1988), which measured power input using operator metabolic measurments, states maximum power operator output values as 69.73 and 51.7 watts for pumping durations of 20 and 30 minutes respectively, closely agrees with the results obtained by calculation from the laboratory and field tests. This limit of power has implications to area irrigable from a given depth. The area must remain above the minumum found by Orr (Orr and Islam 1988) to be the minimum for economic viability and yet remain within the power capabilities of the operator.

For example, the minimum discharge required to supply an average farmer's 0.24 Ha (0.65 acre) crop, with a consumptive use of 6 mm/day, using one pump and a maximum pumping time of 8 hours per day, is 28 l/min. The maximum depth to water table still able to meet the irrigation requirements would be 3.8 meters and would require a constant power input of 40 watts, using the laboratory based field estimates of input power requirements and field discharge results.

Regions where marketing of the pump has been less successful have water table depths of 4 meters and greater. The difficulty in maintaining the 28 litre/minute irrigation requirement without exceeding the power input limit of 55 watts at these depths, is for the most part, the cause for the poor sales.

7.6 PUMP AGE EFFECTS ON DISCHARGE AND VOLUMETRIC EFFICIENCY

The age of the treadle pumps field tested in Bangladesh had little effect on the discharge and volumetric efficiency characteristics measured as shown in figure 19.

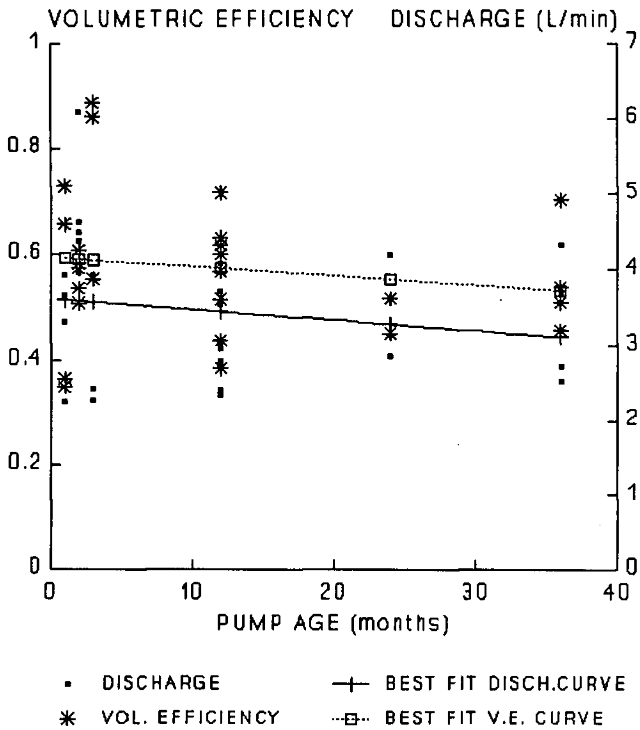

Figure 19: Effect of pump age on volumetric efficiency and discharge from field tests.

The large variations in owner maintenance and care of the pumps observed and tested, overshadowed any long term effects from age as can be seen by the large degree of scatter in the data. Variations of component wear were widely observed, with some pumps requiring replacement piston buckets and foot-valve flaps after two or three months and others not until after 2 or 3 years. Major wear in the steel pump body was comparatively minimal by visual inspection compared to the obvious wear of the cup seals and valves.

7.7 EFFECTS OF PUMP BODY CONFIGURATION ON VOLUMETRIC EFFICIENCY, POWER REQUIREMENTS AND DISCHARGES

Effects from alternate pump configurations, the combinations of the two piston and two pump body types, were most evident in the laboratory results for force inputs, volumetric efficiencies, discharges and input power requirements as shown in figure 20.

The volumetric efficiency is much lower when the No. 6 piston is used as compared to the standard two plate piston, with either pump body at the same water table depth. This is attributable to poor sealing of the poppet valve in the No. 6 piston. The result of this leakage is a loss of suction capability and the partially the lower discharges indicated in figure 20.

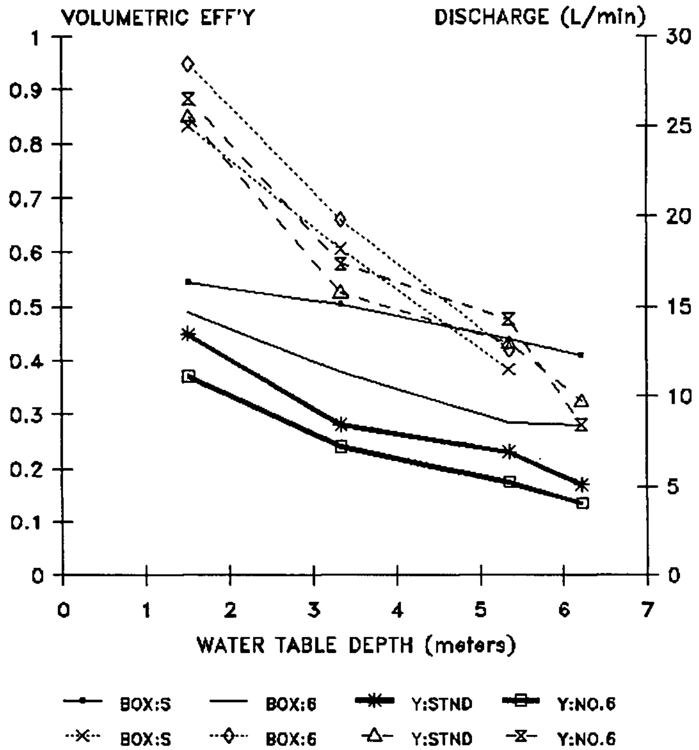

Figure 20: Effect of pump configuration on laboratory measured volumetric efficiency and discharge.

The effects of pump body type are similar to those of the pistons, but the reasons for the poorer performance of the Y-style pump are less obvious. As shown in figure 20, the volumetric efficiencies and discharges for the Y-style pump are all lower than for a similarly configured box manifold pump. The lower volumetric efficiency when using the same set of pistons in each configuration, gives a clear indication that the foot-valve on the Y-style pump tested in the laboratory was operating poorly, as no other change was made which could affect leakage losses. As a result, little can be said about the effects of the Y-style Treadle pump, except that any increases in performance attributable to the manifold design, either Y-style or box, must be minimal to be overshadowed by variations in foot-valve performance.

The alternate pump configurations were not field tested so no comparison is possible to the performance the pumps would have in Bangladesh. Given the poor laboratory test results, an assumption of similar field performance would be reasonable given the close agreement in laboratory and field results of the configuration widely used in Bangladesh, the box manifold and two-plate piston.

8 DISCUSSION

8.1 VALIDATION OF THEORETICAL MODEL BY LABORATORY TESTS

The governing equations presented in section 4.3 produced a force displacement curve that compares closely with the laboratory results as shown in figure 21,

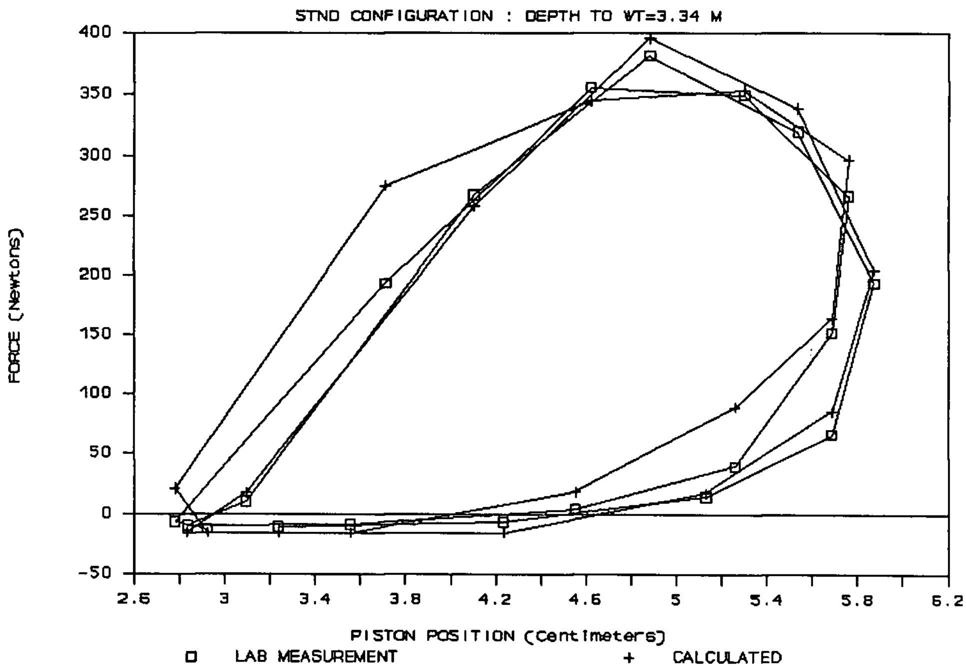

Figure 21: Comparison of calculated and laboratory results.

which is for a model with the box-type manifold Treadle pump using the standard two-plate piston. Values for mechanical friction and piston motion characteristics used in the calculations where taken from the laboratory test data, not from the derived motion equations. The close agreement of the laboratory and calculated force curves shown in figure 21 is largely due to using measured piston motion rather than the theoretical values. This is because the measured values account for irregularities present in the laboratory tests that would not be accounted for in the theoretical motion values.

The laboratory and theoretical results for the other three pump configurations are also in close agreement. This is to be expected because the alterations from the standard configuration are primarily in valve delay and leakage parameters, which were derived from the laboratory data.

8.2 COMPARISON OF LABORATORY AND FIELD TEST RESULTS

The inconsistency of field operators and pumps made the comparison between laboratory and field test results difficult. Averaged field results, used for comparison at 0.2 meter increments of water table depth, showed that the shorter stroke lengths and the equal or slower stroke rates used in the laboratory, as an estimation of how a Bangladeshi farmer would operate the pump over long durations, produced lower discharge than for similar field configurations. The shorter stroke lengths used in the laboratory tests were a result of what was comfortable for the operator. The Bangladeshi farmers are more comfortable with the slightly longer stroke length but they still prefer the much shorter stroke of the "dheki" style superstructure to the full length strokes of the rope-pulley superstructure. The field test operators pumped with longer strokes and often a faster stroke rates than were assumed for the laboratory tests, but tended to pump for shorter durations, about 1/2 hour rather than continuous pumping, as they reached the limit of their power output capabilities.

The shorter pumping duration, the motivation of irrigating the sole source of income for the family group, and availability and affordability of food and medicine may result in higher power inputs than anticipated from the laboratory tests.

The volumetric efficiency results were widely spread in the field data but the averaged results agreed closely with the laboratory data. This close agreement in volumetric efficiency and thus losses, allowed for the calculation of input power for the field tests to be scaled from the discharge measurements from the laboratory tests.

The close agreement between the theoretical calculations and the laboratory results, and the similar values for volumetric efficiency in the laboratory and field results, provide a basis for design alterations valid for field application based on the theoretical and laboratory results.

8.3 IMPLICATIONS OF THE RESULTS ON PUMP RE-DESIGN

Over the past few years, since the start of the rapid increase in popularity and sales of the Treadle pump, many suggestions have been made to alter the pump design. Part of the purpose of testing the Treadle pump was to form a basis for evaluating these suggestions. Some of pump re-design suggestions have included changes to:

1. Pump body materials, including PVC, concrete, fiberglass, juteglass and cast iron.
2. Piston configurations, including retaining the standard two-plate piston or using the No.6 piston tested.
3. Cylinder size, with 76 mm (3 inch), 64 mm (2 1/2 inch) and 51 mm (2 inch) diameters suggested.
4. Cylinder length, such as a 203 mm (8 inch) or 254 mm (10 inch) cylinder to replace the current 356 mm (14 inch) length.
5. The style of pump to riser junction, with the original box-style and the Y-style suggested.

The test results have indicated that, in the case of altering pump material, the friction force of the seal against the cylinder would change marginally with a cylinder material change, as the pressure of the seal against the cylinder would remain constant and the coefficient of friction between the seal and any of the materials listed does not vary greatly. With the marginal scope for improvement from a hydraulics view, the reasoning for a change in pump materials must rather be based upon economic or manufacturing benefits. In brief, given the current development strategy to promote small scale decentralized manufacturing of treadle pumps in Bangladesh and the more pragmatic reasons including the availability of machine tools for working with steel and the availability, wide acceptance and re-sale value of steel products throughout Bangladesh, the use of steel for the treadle pump provides the best solution in terms of the development and economic criterion at the present time. The use of steel also reduces the reliance on new, more expensive imported materials, products and centralized production technology required for many of the alternate materials.

In the case of piston design alteration, both of the two piston configurations tested, the two-plate piston and the No. 6 piston, have advantages. The two-plate piston is more widely used, easily fixed and provides better piston valve sealing than the No. 6 piston, which benefits from wide availability, shorter valve delays and a non-rusting pPVC construction. Optimally, a combination of the small closing time of the No.6 and positive sealing of the two plate design should be used.

In the case of the cylinder size, discharge and power input requirements are primarily affected by any alteration. A decrease in cylinder diameter would reduce the discharge in comparison to the present 89 mm (3.5 inch) design. This would allow use of the pump to a greater depth without exceeding a power input limitation of 55 watts, which corresponds to a hydraulic output of 25.3 watts, in normal use. As a rule of thumb, an increase in water table depth accessibility of 1.4 meters per 13 mm (1/2 inch) reduction in cylinder diameter. The 13 mm reduction in diameter is accompanied by a 27% decrease in discharge, assuming losses are the same as those occurring in the current pump design, with the 55 watt input limitation.

For example, using a discharge requirement of 25 L/min as the minimum to meet irrigation requirements, the maximum depth of application for a 102 mm (4 inch) cylinder diameter Treadle pump would be 3.86 meters, limited by power input. The 89 mm (3.5 inch) treadle was found to have a maximum water table depth application of 4.09 meters, limited by discharge below the requirement and near the limit of power input. A 76 mm (3 inch) Treadle pump would have a maximum application depth of 5.88 meters, limited by discharge below the irrigation requirement. A 64 mm (2.5 inch) Treadle would have a maximum of 4.3 meters, limited by discharge below the requirement.

From these calculations, the optimum cylinder diameter, assuming operating losses, power inputs and pumping operating parameters remain unchanged from those measured in the field and laboratory testing, is 76 mm (3 inches). This size would allow Treadle pump usage to a maximum water table depth and remain within the limits of power input and irrigation requirements. Any increase in efficiency would increase the power output for the 55 watt limit of power input and increase the optimum cylinder size within the 6 meter water table limit.

In the case of cylinder length, a reduction would reduce the pump cost by about 8 Taka (0.32 CAD) per centimetre. Although on average only 140 mm of the 356 mm (14 inch) cylinder is used during the pump stroke, the portion of cylinder used varies between installations. Field trials conducted by IDE with the 305 mm (12 inch) cylinder pump, found that the reduction in length required greater accuracy of superstructure placement at installation. This reduced pump popularity due to the limit of installation variation possible.

In the case of the junction style, from the calculated results, it was determined that the power losses resulting from turbulence through the manifold account for less than 12% of the total head losses, not including valve delay and leakage losses, at maximum water velocity in the piston and riser, at a head of 3.34 meters. The alteration in type of junction manifold from the box-style to the Y-style suffers from increased manufacturing complexity and cost with little or no savings from reduced losses. The benefit from the smoother junction would reduce suction head requirements by 0.03 meters at maximum flow. This represents a 6.4% reduction in hydraulic losses but is only 1% of the total head losses in the whole system, including valve and leakage losses, and is not worth the increase in cost and manufacturing difficulty.

10 CONCLUSION

In conclusion, the Treadle pump performance can be best improved by design alterations to the piston and foot-valves as indicated by the low discharges and volumetric efficiencies measured in the laboratory and field tests, and by a reduction in cylinder diameter to permit pump use in areas with greater water table depths.

The comparatively small losses caused by turbulence and friction indicate that alterations to the pump material and configuration, as shown in the laboratory and theoretical results, are not warranted on the basis of improved pump performance, but must be decided from economic and development strategy benefits.

The primary problems identified from the laboratory and field testing of the "dheki" style Treadle pump installations were the leakage through the piston and foot-valves and the large valve closure delay times. Although the longer stroke lengths of the field tests reduced the effect of the valve delays, even at the maximum comfortable stroke lengths recorded the minimum loss resulting from valve delays is 17%. For example, if both the piston and foot-valve closure delay times where halved, the decrease in leakage represents a potential increase in volumetric efficiency of 46%, resulting in increased discharge and use to greater water table depths.

The water table depth to which the Treadle pump is usable is limited by either insufficient discharge or large input power requirements, depending upon cylinder size and depth to water table. The current 89 mm (3.5 inch) cylinder diameter pump is limited to a water table depth of 4 meters, due to the sustainable power input limit of 55 watts indicated by the field and laboratory results. The difficulty of selling the pumps in regions with water table depths deeper than 4 meters and the EPC test results also indicate a water table depth limit of 4 meters. A reduction in cylinder size to 76 mm (3 inch) would increase the water table depths accessible to the Treadle pump within the limitations of power input and maintain enough discharge to meet irrigation requirements. The 76 mm pump would be most suitable for water table depths in the 4 to 5.5 meter range, to assure that the power input required does not exceed 55 watts. Any further reduction in diameter past 76 mm would reduce the discharge below the irrigation requirements for the 0.24 Ha (0.6 acre) minimum irrigated area required for pump repayment. The use of the widely available 89 mm (3.5 inch) pPVC piston cup seal, manufactured for use in the ubiquitous No.6 pump, and the poor availability of the 76 mm (3 inch) pPVC seal reduces the desirability of the 76 mm pump at the present time, but more work in manufacturing alternatives may yield more options.

Overall, the study of the piston force characteristics and the use of variations between field and laboratory results to analyze the pump design, clearly shows the benefits of the combined use of laboratory, analytical and field results as the basis for design alterations. The study also indicated the strength of the design and the innovation by the aid workers who originally developed the Treadle pump, as very little alteration is required to optimize the operation of the Treadle pump.

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APPENDIX 1: AVERAGED FIELD TEST DATA

TESTS CONDUCTED IN THE JAMALPUR AND KUSHTIA REGIONS OF BANGLADESH

TEST #	OPERATOR AGE (yrs)	WEIGHT (lbs)	PUMP AGE (yrs)	W.T. DEPTH (m)	STROKE LENGTH (cm)	DISCHARGE (l/min)	VOL. EFF'Y	STROKE RATE (/min)	POWER OUTPUT (watts)
21	27	125	1	2.05	12.32	39.22	0.66	39.02	13.14
42	40	110	1	2.20	16.06	33.12	0.35	47.51	11.91
11	28	130	1	2.23	9.57	36.59	0.73	42.26	13.34
16	18	90	12	2.29	13.75	30.30	0.44	40.61	11.35
17	25	150	12	2.29	9.02	23.32	0.63	32.95	8.73
19	30	150	12	2.37	11.11	35.35	0.72	35.76	13.70
18	25	95	12	2.47	15.62	27.27	0.38	36.55	11.01
22	31	100	12	2.59	9.79	37.15	0.63	48.48	15.73
39	25	100	2	2.73	14.85	43.83	0.54	44.42	19.56
32	18	95	24	2.78	15.73	41.93	0.52	41.55	19.06
41	18	90	2	2.79	12.98	45.00	0.61	46.00	20.53
13	22	120	12	2.86	12.54	40.82	0.60	43.67	19.09
40	40	130	1	2.88	18.92	22.52	0.36	26.31	10.61
43	37	125	3	2.97	11.99	39.22	0.55	28.04	19.04
35	50	120	12	3.13	11.66	29.51	0.62	33.05	15.10
9	40	125	2	3.25	15.73	60.89	0.51	61.33	32.35
10	25	150	2	3.25	8.47	32.97	0.67	46.81	17.52
36	18	100	2	3.56	13.97	39.60	0.57	39.80	23.05
37	30	120	2	3.61	13.64	46.23	0.58	46.85	27.29
27	28	126	12	3.73	6.27	23.96	0.57	54.25	14.61
29	14	80	24	3.83	12.76	28.57	0.45	40.00	17.89
33	25	130	12	3.85	10.12	27.86	0.51	43.10	17.54
31	35	125	36	3.97	9.13	36.92	0.70	46.34	23.97
28	23	115	36	4.01	6.82	27.19	0.46	70.33	17.83
23	24	110	36	4.13	16.28	43.23	0.54	39.77	29.19
24	25	150	36	4.13	9.24	25.17	0.51	43.09	16.99
25	45	110	3	4.28	5.94	24.19	0.86	38.13	16.93
26	25	150	3	4.28	5.72	22.73	0.89	36.06	15.90

APPENDIX 2: AVERAGED LABORATORY DATA

-TESTS CONDUCTED AT THE UNIVERSITY OF BRITISH COLUMBIA, CANADA

NOTE: THE FIRST TWO DIGITS OF THE TEST CODE INDICATE CONFIGURATION
01:BOX MANIFOLD, 2-PLATE PISTON
02:BOX MANIFOLD, NO. 6 PISTON
03:Y-MANIFOLD, 2-PLATE PISTON
04:Y-MANIFOLD, NO. 6 PISTON

TEST CODE	HEAD (m)	STROKE LENGTH (cm)	DISCHARGE (l/min)	VOL. EFF'Y	RATE (/min)	POWER OUTPUT (watts)	POWER INPUT (watts)	POWER EFF'Y
T01021	1.51	4.04	25.06	0.68	73.41	6.11	26.55	0.23
T01022	1.51	3.35	25.04	0.41	73.72	6.15	23.64	0.26
T01041	3.34	3.56	24.07	0.60	45.52	13.15	50.56	0.26
T01042	3.34	4.24	12.36	0.41	57.07	6.69	31.88	0.21
T01061	5.36	3.30	16.63	0.58	70.20	14.58	64.80	0.23
T01062	5.36	1.91	6.27	0.30	87.94	5.49	35.44	0.16
T01081	6.22	4.32	34.54	0.55	60.00	34.51	95.85	0.36
T01082	6.22	4.84	20.85	0.41	41.81	21.17	81.44	0.26
T02021	1.51	4.08	42.60	1.08	77.99	10.56	45.92	0.23
T02022	1.51	2.78	28.44	0.49	91.72	7.57	37.85	0.20
T02041	3.34	3.28	24.53	0.43	69.30	13.50	42.19	0.32
T02042	3.34	2.94	15.13	0.33	63.84	8.28	30.68	0.27
T02061	5.36	2.79	16.14	0.34	67.73	13.92	51.57	0.27
T02062	5.36	3.02	9.03	0.23	53.50	8.00	40.00	0.20
T02081	6.22	4.69	28.70	0.33	74.00	30.16	115.99	0.26
T02082	6.22	3.61	12.21	0.23	58.30	12.13	86.63	0.14
T03021	1.51	4.57	35.54	0.54	58.17	8.75	29.18	0.30
T03041	3.34	3.68	15.77	0.28	61.81	8.71	39.60	0.22
T03061	5.36	1.88	12.97	0.23	60.08	11.44	54.46	0.21
T03081	6.22	4.21	9.72	0.17	54.56	9.94	99.44	0.10
T04021	1.51	4.94	25.84	0.33	63.55	6.53	40.84	0.16
T04022	1.51	3.72	27.19	0.41	72.13	6.87	42.92	0.16
T04041	3.34	4.49	15.67	0.21	65.66	8.71	67.01	0.13
T04042	3.34	4.11	19.10	0.27	68.35	10.55	70.35	0.15
T04061	5.36	6.71	17.15	0.18	56.14	15.51	129.28	0.12
T04062	5.36	4.31	11.44	0.17	63.17	9.53	105.87	0.09
T04081	6.22	7.65	5.80	0.10	39.20	5.61	93.58	0.06
T04082	6.22	4.77	11.03	0.17	55.88	10.91	121.21	0.09