55TH CONGRESS, 2d Session. } HOUSE OF REPRESENTATIVES. { DOCUMENT No. 509.

DEPARTMENT OF THE INTERIOR

WATER-SUPPLY

AND

IRRIGATION PAPERS

OF THE

UNITED STATES GEOLOGICAL SURVEY

No. 14

TESTS OF PUMPS AND WATER LIFTS.—HOOD

WASHINGTON
GOVERNMENT PRINTING OFFICE
1898

UNITED STATES GEOLOGICAL SURVEY

CHARLES D. WALCOTT, DIRECTOR

NEW TESTS

OF

CERTAIN PUMPS AND WATER LIFTS USED IN IRRIGATION

BY

OZNI PORTER HOOD

WASHINGTON

GOVERNMENT PRINTING OFFICE

1898

CONTENTS.

ILLUSTRATIONS.

LETTER OF TRANSMITTAL.

DEPARTMENT OF THE INTERIOR,
UNITED STATES GEOLOGICAL SURVEY,
DIVISION OF HYDROGRAPHY,
Washington, January 24, 1898.

SIR: I have the honor to transmit herewith a paper entitled New Tests of Certain Pumps and Water Lifts Used in Irrigation, by Ozni P. Hood, professor of mechanics and engineering at the State Agricultural College, Manhattan, Kansas, and to recommend that it be published in the series of Water-Supply and Irrigation Papers. The facts herein briefly set forth represent a large body of original experimental work by Professor Hood and assistants. begun in the summer of 1896 and but recently brought to a finish.

While the water lifts have had their origin in the needs of irrigation, devices for pumping have not; with them, indeed, there is no special adaptation to irrigation use. Economy of power must be the essence of such adaptation; time is not, for ample time can be taken to secure the full supply. But with existing designs, as, for example, the deep-mine varieties, the aim usually has been rather to attain rapidity of delivery, even at the cost of wasteful expenditure of power. Among appliances for irrigation, pumps, of whatever kind, are but in a trial stage. Their wide use and the extension of irrigation possibilities that would result can come only through adaptations in design especially economical of power. It should not be left to come by gradual evolution through chance suggestions of practice, but should be deliberately brought about by engineers. Professor Hood's experimental determinations of efficiencies and of the little sources of frictional loss have contributed data which will be useful for this purpose.

Very respectfully,
F. H. NEWELL,
Hydrographer in Charge.

Hon. CHARLES D. WALCOTT,
Director United States Geological Survey.

NEW TESTS OF CERTAIN PUMPS AND WATER LIFTS USED IN IRRIGATION.

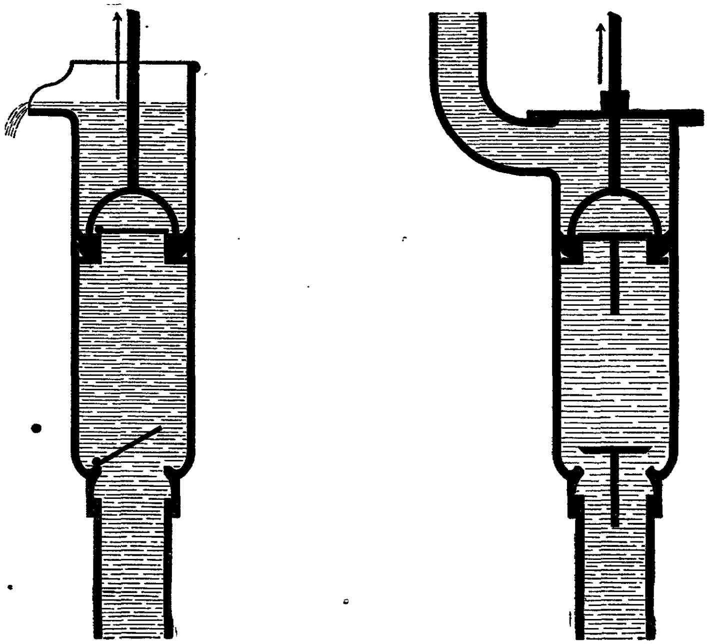
By O. P. Hood.

INTRODUCTION.

The necessity of raising water for irrigation and other purposes has been met by a great variety of water-lifting devices. By far the most familiar of these is the reciprocating pump, ranging in capacity from the simple mechanism for cisterns to the large engines used for public water supply. In order that one may know what it is reasonable to expect from the many forms of pumps, and be able to make an intelligent selection, it is necessary first to become acquainted with some of the problems connected with their design and operation. In this paper it is the author's purpose to state certain of these problems, to give the results of new tests of irrigation pumps and machinery, and to offer notes on several other water lifts in less common use.

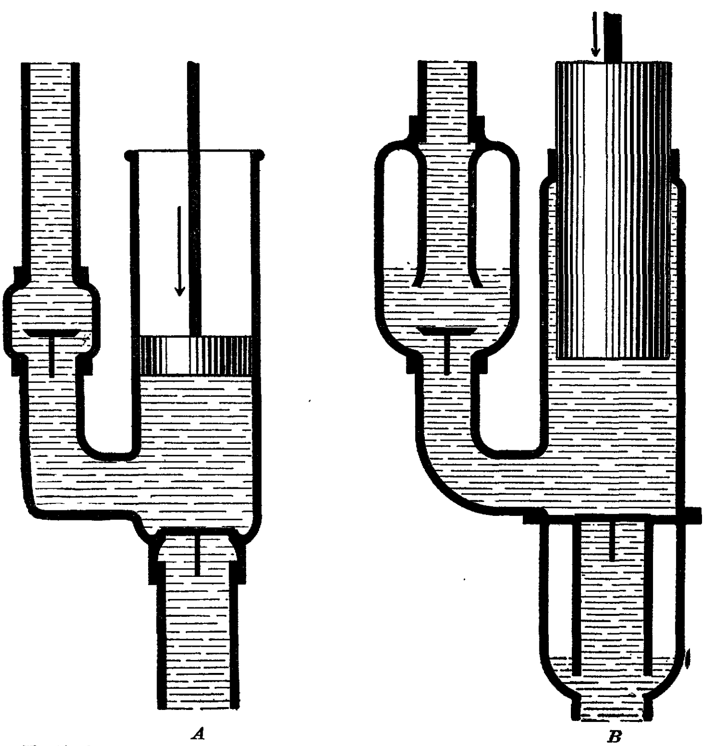
DEFINITIONS.

Of reciprocating pumps there are many forms, and among them there are essential differences. The names commonly used to designate these diverse forms are not in every case self-explanatory; in some cases they may be even misleading. Among these common designations are bucket pump, lift pump, suction pump, combined suction and lift pump, piston pump, plunger pump, and force pump. Common to all these forms is a backward and forward moving piston or plunger. When all the effective work is done while the piston is moving either in one direction or the other the pump is single-acting. When the effective work is distributed between the two motions the pump is double-acting. It is to this alternating movement that the term "reciprocating" applies. Generally speaking, in all reciprocating pumps there are three essential parts—the suction pipe below, the discharge pipe above, and, intermediate in position between these two, the pump cylinder, in which a piston reciprocates.



A B

FIG. 1.—Vertical sections of bucket pump. *A*, ordinary form; *B*, with stuffing box.



A B

FIG. 2.—Vertical sections of piston and plunger pumps. *A*, piston pump; *B*, plunger pump.

A bucket pump is one in which the reciprocating piston has a valve-covered opening through it. Such a pump is shown diagrammatically in fig. 1, *A*. Between the incoming water and the piston is another valve-covered opening.

If the pump parts are so placed that the suction valve comes below the normal water level, it is called a lift pump. If, on the other hand, the suction lift is relatively great and the delivery pipe short, it is called a suction pump. Most pumps, however, by reason of the more even proportioning of these parts, are called combined suction and

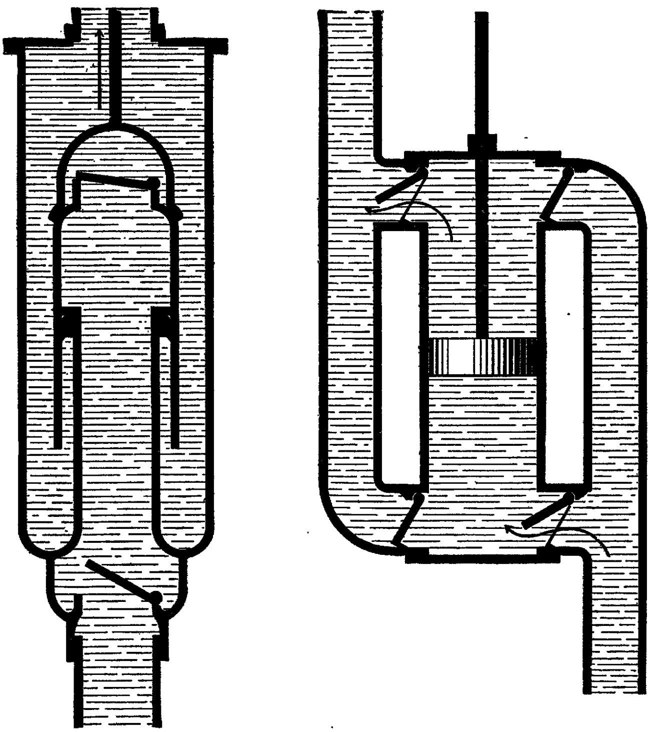


FIG. 3.—Vertical sections of plunger and double-acting pumps. *A*, plunger pump with large valve openings; *B*, double-acting pump.

lift pumps. Fig. 1, *B*, shows a pump in which the pump rod is carried through a stuffing box, whereby it is rendered unnecessary for the delivery pipe to follow the pump rod. A piston pump is one in which the piston carries no valve, the discharge valve being otherwise placed, as in fig. 2, *A*. Fig. 2, *B*, shows a plunger pump, a solid plunger, packed on the outside, replacing the piston, which is packed on the inside. Fig. 3, *A*, indicates one of several methods of obtaining large valve openings. Fig. 3, *B*, shows a double-acting pump, serving as a suction and force pump and acting in either direction of movement of the piston. For irrigation the type most in favor is that shown in fig 1, *A*, operated as a simple lift pump.

SUCTION.

Whenever, within a closed vessel, the normal air pressure is by any means diminished, there results what is commonly called suction. Suction is here but a manifestation of superior pressure on the part of the external air. This atmospheric pressure, though varying continually, is, at sea level, about 14.7 pounds per square inch. A column of water about 34 feet in height exerts an equal pressure. Hence, to raise the piston, shown in fig. 4, it is necessary not only to lift the column of water, L, already above the piston, but to overcome as well the atmospheric pressure, equivalent to an additional column of water, B, 34 feet in height; or to overcome, in total, a pressure equivalent to the weight of a column of water, L+B.

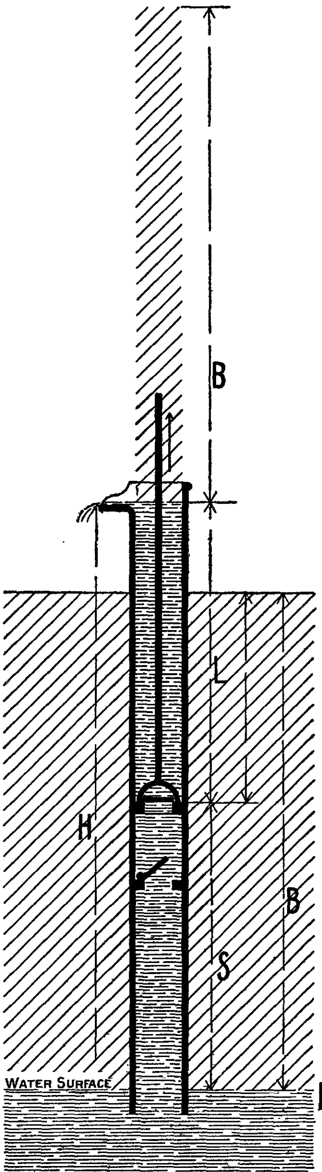


FIG. 4.—Diagrammatic representation of pump problem.

But with the raising of the piston conditions for suction, so called, are thereby at the same time established, i. e., the water surface in the pump tube below the piston is relieved of pressure, whereas outside the tube the atmospheric pressure continues, undisturbed, upon the surrounding water surface. It results that, by reason of this pressure without, water is forced upward in the pump tube; and it continues to follow the piston to a balancing height, which at sea level, as stated, is 34 feet. If the lift of the piston is short of this height the rising water will bear upward against the under side of the piston with a pressure equivalent to the weight of so much of the column as remains, or B—S; less, however, such force as may be used up in overcoming the various resistances during motion. As the piston lift does in fact always stop short of this maximum, the remnant of pressure, B—S, serves as a head of water for effecting flow into the suction pipe and for filling the cylinder. Though the theoretical limit to the height of suction is about 34 feet, a practical limit is reached at from 25 to 30 feet, depending on the perfection of packing, the tightness of joints, and the piston speed. If, however, the piston speed be too great, the head of water (B—S) may not be able to produce a flow sufficiently rapid to follow it, and in consequence there will be incomplete filling of the cylinder during the period of the upstroke.

The water will encounter resistances to flow in the suction pipe, such as those involved in friction against its sides, in lifting the suction valve, and in passing through the reduced valve opening, all of which resistances will tend to diminish its velocity of entry into the cylinder. The less the height of the suction lift the greater will be the head left for overcoming these resistances, thereby admitting of a more rapid rate of filling and a higher piston speed. Again, by enlarging the suction pipe and thereby reducing the sum of the frictional resistances the flow will be increased. It follows that in practice many pumps have suction pipes of larger diameter than their discharge pipes.

The height of suction lift limits the possible piston speed. To show this, a 4-inch by 14-inch pump was arranged to discharge water at a total height of 38 feet, the cylinder being placed successively at various heights in the pipe, from the natural water level up to the limit of suction. The suction and delivery pipes were each 2 inches in diameter. The various speed limits reached under these varied conditions are shown in fig. 22 (p. 35), in the progressive diminution of discharge due to incomplete filling of the cylinder. The upstroke of the piston may be more rapid than the suction uprise of the water following it, in which case there will be separation of the piston from the following water. This separation may under certain conditions lead to a shock. The separation will begin, if at all, with the beginning of the upstroke; but the acceleration of the upstroke has a diminishing rate, while the speed of the following water may have an increasing rate. If the water overtake the piston about midway of its rise, the meeting will be without perceptible shock; if near the end of its stroke, the shock may be considerable. If, on the other hand, the piston has time to complete the upstroke and accomplish a part of its return, the shock may be severe.

It is these recurring shocks which produce what is called pounding. This action is shown in the accompanying diagrams, figs. 5 to 8, which exhibit the pull on the pump rod at various stages of its stroke. In fig. 5, *A*, the line x, from A to B, represents the length of the stroke, the direction AB the upstroke, the direction BA the downstroke. Vertical distances above the line x represent the pull in the pump rod. Thus, when the piston has reached the point C on the upstroke, the pull in the rod is represented by the line CD, and when it has reached the point E the stress is EF. On the return stroke the stress at E is EG, and at C it is CH. The figures here referred to are from card diagrams, automatically drawn by a specially devised dynamometer, which was introduced into the pump rod of pumps in operation, so that they record actual practice. The diagram shown in fig. 5, *A*, gives the results of the test of a 4-inch pump having a total lift of 38 feet, a 7-inch suction lift, 2-inch suction and delivery pipes, and a speed of thirty 14-inch strokes per minute, or an average piston speed of 70 feet per minute. The height of the line *ab* shows the pull necessary merely to support the column of water.

At the starting point, A, of the upstroke, the delivery valve closed, and the sudden pull on the rod which resulted is shown by the long vertical line. The depression immediately following is due to the spring of the pump rod and connections upon receiving the first large stress. The upward movement of the water column having become accelerated, a continually lessening pull is required to produce its discharge, as shown by the rapidly falling upper line. The suction valve closes at B, and the weight of piston and rod still produces a small tension in the rod. With excessive friction, or at high speed, this return-stroke line frequently falls in part below the zero line, showing compression in the pump rod.

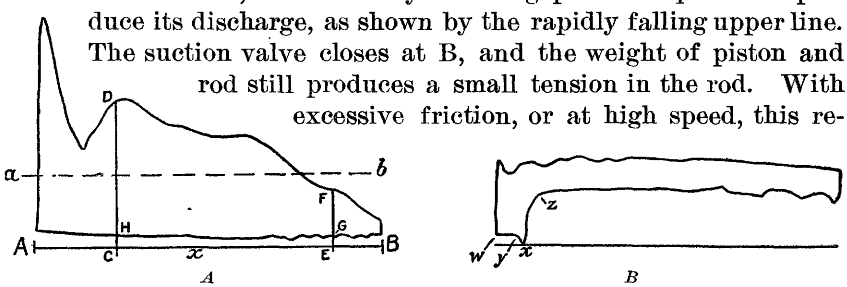


FIG. 5.—Dynamometer diagrams of pump-rod stresses. *A*, general; *B*, with suction lift of 30.8 feet, 30 strokes per minute.

In fig. 5, *B*, the pump was given a suction lift of 30.8 feet, with the same total lift and speed, however, as in fig. 5, *A*. The pull on the pump rod is seen to be very differently varied. On the upstroke this pull is much less, having only a small body of water to accelerate in addition to the weight of the atmosphere which it has to lift. The load remains nearly constant throughout the upstroke, the piston not being aided in its upward movement, toward the end, by the momentum of a rising column of water, as in fig. 5, *A*. On the return stroke the downward thrust is aided by the weight of the water in the delivery pipe as well as by the atmospheric pressure, and is resisted only by friction in the cylinder until the piston reaches the point *z*, where it encounters the slowly rising column of water. The blow of the piston against the rising water momentarily relieves the rod of tension, as shown by the drop in the vertical line in the diagram at *x*. This opposing pressure is at once dimished, however, by the opening of the piston valve. At *y* the suction valve closes and remains closed until opened at *w* on the upstroke.

If the piston speed had been slower the cylinder would have more nearly filled, as shown by the card (fig. 6, *A*) taken at 10 strokes per minute, under which conditions the cylinder had become more than half filled before the shock of meeting. In fig. 6, *B*, the suction lift was 24.7 feet, and at a speed of 30 strokes per minute the cylinder had filled to the point *x*. In fig. 6, *C*, at a speed of 20 strokes, the rising water overtook the more slowly rising piston during the upstroke at the point *z*. In fig. 6, *D*, the speed was but 15 strokes per minute. In this case the following water overtook the piston near the middle of its upstroke, when the difference of speeds was not great, and the diagram in consequence shows at *a* but slightly reduced tension.

It is characteristic of this condition of high suction lift that with the same speed the upstroke of the pump starts more easily than when the cylinder is full, and that pounding may occur at any point in the last three-fourths of the stroke—the nearer its end the more

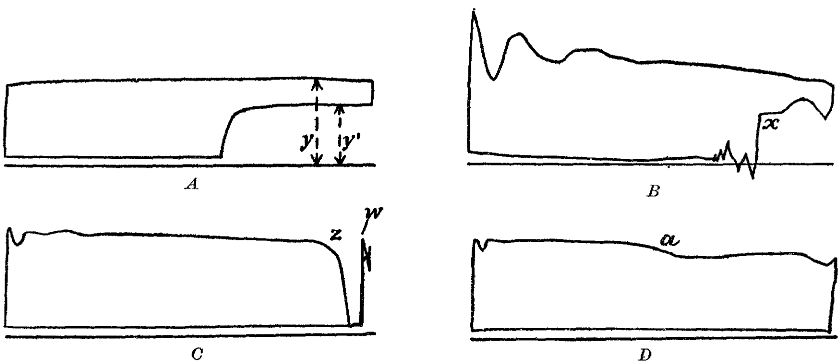


FIG. 6.—Dynamometer diagrams of pump-rod stresses. *A*, suction lift of 30.8 feet, 10 strokes per minute; *B*, suction lift of 24.07 feet, 30 strokes per minute; *C*, suction lift of 24.07 feet, 20 strokes per minute; *D*, suction lift of 24.07 feet, 15 strokes per minute.

severe. A sudden change or reversal of stress in the rod produces disastrous pounding and limits the speed of the pump. It should not be forgotten in this connection that it is weakening of suction head, as it might be called, that is the real cause of pounding, and that the difficulty may arise from undue frictional resistance to flow as well as from excessive suction lift.

It has been a common practice, in order to prevent an intake of sand where the suction pipe is itself the well and not merely a pipe suspended in an open well, to have the whole intake at the foot of the suction pipe through a screen. To the use of the screen there are objections of a serious nature, and the difficulty which is met by its use is best avoided entirely by resorting to the open well. However, if the screen is to be used at all, the total area of its perforations should be at least double the cross-section area of the suction pipe. Even then the screen openings will in time become clogged to such extent as greatly to increase the resistance. Not infrequently in a pump with a screen intake failure to fill is wholly due to a clogged condition of the screen.

The introduction of elbows in the suction pipe is to be avoided wherever possible. The resistance to flow which they offer is considerable. Again, pipes of the smaller sizes are commonly cut with the rolling cutter, which raises a burr on the inside edge, thereby seriously reducing the entrance diameter. It is worth while to remove this burr. The following table shows the reduced cross-section area of a number of sizes of pipe cut with this tool:

Reduction of cross-section areas of pipes of various size cut with the rolling cutter.

	Diameter of pipe.						
	½-inch.	¾-inch.	1-inch.	1¼-inch.	1½-inch.	2-inch.	2½-inch.
Reduced area	*Per cent.* 87	*Per cent.* 77–83	*Per cent.* 83–90	*Per cent.* 80–88	*Per cent.* 85	*Per cent.* 90	*Per cent.* 95

Wherever, as sometimes happens, it becomes necessary to extend the suction pipe horizontally for any distance, such horizontal section should be given a larger diameter than that used in vertical sections. The larger volume of water which the head (B—S, fig. 4, p. 14) must move, together with the added friction of the increased length of pipe and of the necessary turns, reduces the velocity of the incoming water and fixes a lower limit to the possible piston speed.

UPSTROKE.

On the upstroke the pump simultaneously performs two distinct operations: It lifts directly the column of water L (fig. 4, p. 14) and the atmospheric column above it. By thus completely relieving of load the column of water below, it establishes the conditions under which that lower column is driven upward by the atmospheric pressure without. The resistances to upward movement above the delivery valve offered by the water are its hydraulic pressure, its inertia, and its friction. The hydraulic pressure depends on the height H, and is about 0.433 pound per square inch of piston area for each foot of height of water. If the discharge pipe is smaller than the cylinder, the velocity of the discharging water will be correspondingly greater. If the velocity be doubled, the frictional resistances, varying as the velocity squared, will be increased four times. Furthermore, water encounters resistance in making entrance into a pipe; wherefore it is desirable, when practicable, that the discharge pipe should have the same size as the cylinder. If the pump rod be carried through a stuffing box its friction therein will be considerable. There will be frictional loss also at the necessary turns in the delivery pipe, which loss may, however, be minimized by giving to these turns a long radius.

During the upstroke there is acceleration of speed at the outset and retardation toward the close, the rate of acceleration diminishing toward the center of the stroke, and the rate of retardation increasing from that point toward the close of the stroke. The upward start of the water column is thus, in effect, somewhat abrupt, and in abruptly overcoming the inertia of this water column a pull is exerted upon the pump rod much in excess of that due to the weight of the water alone. This inertia resistance also varies as the velocity squared, and becomes a destructive resistance in fast-running pumps.

DOWNSTROKE.

At the finish of the upstroke the suction valve below closes, and the column of water above it remains at rest until the downstroke has been completed and another upstroke begun. During the descent of the piston through this stationary column of water the piston valve, or delivery valve, as it is called, is opened thereby. The resistances encountered on the downstroke are the frictional resistance of the piston packing and of the column of still water through which the piston has to be forced. It will be apparent that no useful work is accomplished on the downstroke; that all, on the other hand, is accomplished on the upstroke. In rising, the piston lifts B+L, and in turn is pressed upon from below by B—S, the difference between these forces being that force necessary to be applied to the rod and equal to H, fig. 4, p. 14. It is therefore evident that the force to be exerted is measured by the total height of the water lift and is independent of the relation of the suction lift to the direct lift.

VALVES.

There are two common forms of valves—the clack valve, which opens on a hinge, and the disk valve, which rises bodily on guides either at its circumference or along a central spindle. Valves of whatever variety should open and close readily, and should afford, when open, a water way as large and as direct as practicable. To open readily they must not be too heavy; on the other hand, for quick closing weight is an advantage. Also, to give free and large water passage they must have considerable amplitude of movement; yet, again, for quick closing such amplitude of movement is a disadvantage. Thus it appears that of necessity valve design is a matter of compromise among opposing conditions and requirements.

Some of the conditions which bear upon valve design are brought out in the following diagrams. Fig. 7 is the record made by an 8-inch pump having large clack valves, the smallest valve-opening area being 70 per cent of the cylinder cross-section area, and even this is an unusually large valve opening. These valves were in fact so large that, in the experiments, at only 25 strokes per minute they did not close until from 7 to 8 per cent of the stroke had been made. On the return stroke the lower valve did not close until the piston had had time to travel from B to C (fig. 7), during which time, of course, water was escaping. This delay in closing and the consequent loss of water occurred at the very moderate piston speed of about 40 feet

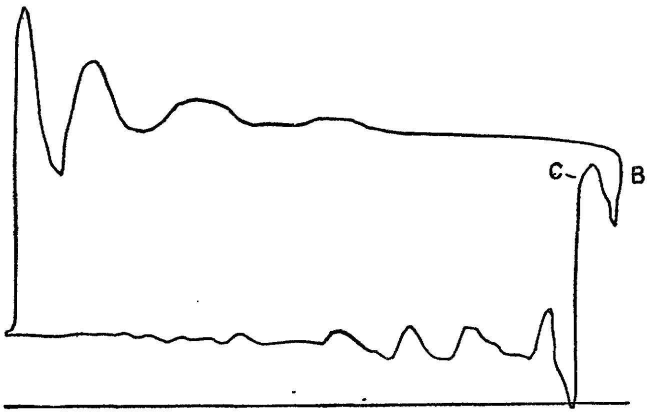


FIG. 7.—Dynamometer diagram of pump-rod stresses from an 8-inch pump having large clack valves. Piston speed, 40 feet per minute.

per minute. Its diagrammatic record exhibits the main defect in large clack valves.

Fig. 8 is the record made by a 6-inch pump using eight small clack valves for delivery valves, having a 14-inch stroke, and operated at a speed of 55 strokes per minute. The aggregate area of the valve openings was 15 per cent in excess of the cylinder cross-section area, yet at this high piston speed of more than 125 feet per minute the delay in the closing of these valves, as shown at A, was very small, being only about 2 per cent of the time interval of the whole stroke. The efficiency of this pump was very high. Thus there is seen to be decided advantage in the substitution, wherever practicable, of several small valve openings for a single large one. Valve lift is thereby minimized and quick closing secured, and without reduction in the aggregate area of valve opening.

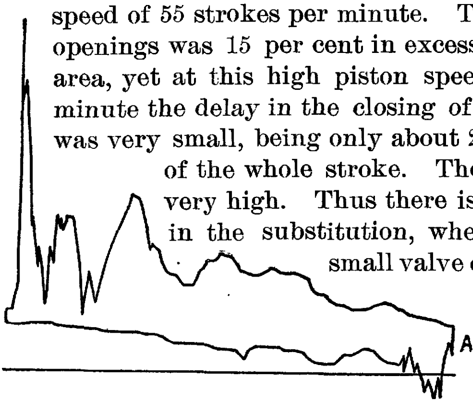


FIG. 8.—Dynamometer diagram of pump-rod stresses from a 6-inch pump having multiple small valves. Piston speed, 40 feet per minute.

That the discharge area around the edge of a clack valve shall equal that of the valve-seat opening, the valve must rise at its outer edge to a height equal to at least half the diameter of the valve opening. The diameter, in inches, of a clack-valve opening should not exceed 100 divided by the speed of the pump in strokes per minute. For example, a pump making 25 strokes per minute should have a clack-valve opening not exceeding in diameter $\frac{100}{25}$, or 4 inches. Thus the clack valve is limited in use either to small pumps, to pumps of low speed, or to pumps of such design that they admit of the introduction of several small valves in substitution for a single large one.

With multiple valves the valve area should be equal to at least 50 per cent of the cylinder cross-section area. A modification[1] of the usual clack valve is presented in fig. 9, *A*, as an illustration of the manner in which some of the objections to this valve may be overcome. This modification consists in mounting upon the back of the large clack a second clack, covering a valve opening in the first. The tendency of the current of water to open widely the larger clack is in part overcome by the yielding of the smaller clack. It is claimed that higher speeds may thus be attained; also that the modification can readily be introduced into any large single clack already in use.

The advantages of the clack valve are its simplicity and its high efficiency at low speed. Being usually cut out of leather, as shown

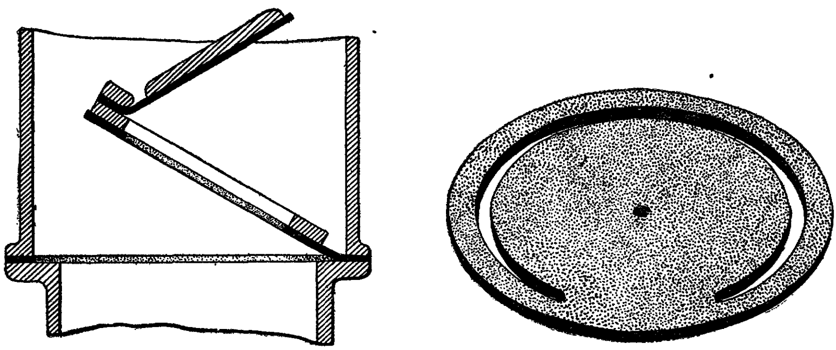


FIG. 9.—Clack valves: *A*, vertical section of double clack valve; *B*, leather suction valve.

in fig. 9, *B*, it is self-packing when closed and is self-hinged. It should be cut from a solid piece of leather, of uniform thickness and without soft spots. The width of the hinge should equal half the valve diameter, and the leather should be placed with the hair side to the seat. A metal disk slightly larger than the valve opening should be attached to the back of the valve, both to add to its stiffness and to give it weight.

Clack valves should have a small angle of lift in pumps running more than 15 strokes per minute. The resistance to flow through clack valves has been determined for the following angular openings:

Resistance to flow through clack valves.

	Degrees of lift.				
	15°	20°	25°	30°	40°
Resistance	*Per ct.* 64	*Per ct.* 53	*Per ct.* 32	*Per ct.* 21	*Per ct.* 10

[1] Barr.

If, for high speed, the valve lift be limited to about 30 degrees, or to an opening that, at its outer edge, is equal to half the valve diameter, the disadvantage of obstruction to flow will be about balanced by the advantage gained of quick closing.

A modification in form is the butterfly valve, in which the hinge is along a diameter line across the center of the leather disk, so that, instead of opening as one piece from a single point at the edge, it opens upward in two halves, back to back, like the wings of a butterfly, and is in effect a double valve. Fig. 27 (p. 48) shows such valves.

The disk valve is commonly but a circular metal plate, with one face adapted to fit a valve seat, and designed for a straight lift, within guides, either as shown in fig. 14 (p. 30), or in fig. 23 (p. 43). As a rule its vertical range is limited mechanically, but occasionally, as in fig. 23, the disk is merely weighted, and control of its range is left to the velocity of the incoming water, to which the weighting has been or can from time to time be adjusted. For low speeds, of about 20 strokes per minute, large disk valves work well, but for higher speeds the valves should be smaller. A valve of more that 5 inches diameter would be considered large. For quick-running pumps multiple valves from 3 to 4 inches in diameter are best.

In irrigation pumps the valve lift should not exceed $1\frac{1}{2}$ inches. At that lift, and at a speed of 30 strokes per minute, about one-eleventh of a second is required for the drop-return of the valve disk, which involves a sacrifice of about 7 per cent of the stroke length. It is well to limit the valve diameter to four times the valve lift; and in the case of irrigation pumps, to 6 inches as a maximum.

The area between the edge of the disk and the cylinder walls should not be less than the valve-seat area. Where the cylinder is not enlarged into a valve chamber the valve area can not exceed about 40 per cent of the cylinder area, because of the necessary grids and lap of valves. It follows that the piston speed should fall well within 100 feet per minute; that is, in a pump with disk valves of this kind, having a 40 per cent valve opening and a 24-inch stroke, a speed of 25 strokes per minute should be considered high. This is a piston speed of 100 feet per minute.

In irrigation pumps the velocity of water through valves should not exceed 250 feet per minute. The speed of the piston at its maximum rate will be 1.57 times that at its average rate. In other words, when the average piston speed is 100 feet per minute there will be a maximum rate at about the middle of the stroke of 157 feet per minute; and this also will be the velocity of the water in the cylinder immediately ahead of the piston, but the water has to be forced through a contracted valve opening, and it must in consequence take on there a correspondingly increased velocity. If the valve opening be 40 per cent of that of the cylinder cross section, as above stated, the increase of velocity over that in the cylinder will then be from

157 feet per minute to 392 feet per minute. Hence an average piston speed of 100 feet per minute would be excessive. Experience with pumps of this class has shown that the best running speed is 60 feet per minute. It is found that beyond this speed the resistance offered by disk valves rapidly reduces the pump efficiency.

If the disk be weighted merely, and not limited in movement mechanically, it may be drawn up so far into the cylinder by the high velocity of the entering water that very late closing will result. The speed of the pump shown in fig. 27 (p. 48) is restricted thus to 50 strokes per minute. Fig. 10 gives the record of this pump when running at 55 strokes per minute. The suction valve was drawn up into the cylinder by the high speed of the water, and did not return to its seat until after half the downstroke had been completed. Seating at this late stage produced serious pounding.

For lifts exceeding 100 feet ball valves become desirable, because of

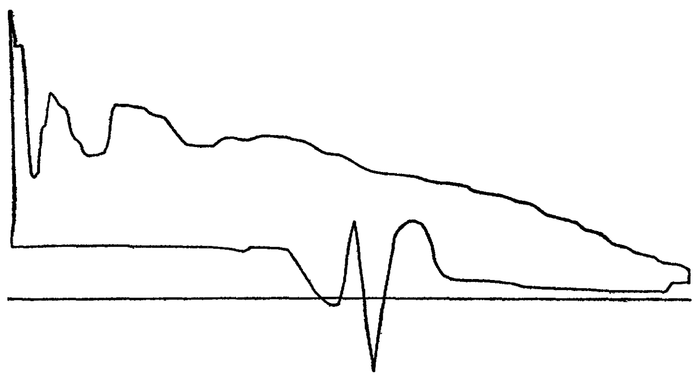


FIG. 10.—Dynamometer diagram of pump-rod stresses, showing late closing of suction valve.

the increased pressure. The ball should be solid and should fit into a metal seat. This form of valve is used in deep-well pumps, where, in order to secure greater strength of parts, it becomes necessary to reduce the valve area to a smaller percentage of the cylinder cross-section area.

PISTON AND ROD.

The piston of a pump is usually packed by means of cup leathers held between disks in the piston. Cup leathers can not well be homemade, as special appliances are necessary for their manufacture, and they had best be bought of dealers. The cup leather should not exceed three-fourths of an inch in width on the wearing surface. The water pressure holds the leather against the walls of the cylinder with considerable friction. This friction amounts, approximately, in pounds to $\frac{DWH}{3}$, D being the cylinder diameter in inches, W the width of the cup leather in inches, and H the lift of water in feet.

In a 6-inch cylinder, with a 20-foot lift, the piston friction will be about 30 pounds, or 12 per cent of the weight of the water lifted.

The piston friction, regarded thus as a percentage, is, approximately, $\frac{W}{D}$. To show the amount of this piston friction, reference is made to fig. 6, *A*. This diagram is from a pump having a 4-inch cylinder, a 14-inch stroke, and running at 10 strokes per minute. The suction lift was 30.8 feet, and the total lift was 38 feet. There was about 8 feet of water above the piston. The height y represents the pull on the rod due to the lifting of the 8 feet of water, the atmospheric pressure, and the weight of piston and rod, and to the overcoming of the friction of the cup leather.

The piston, once having left the water behind, is beyond the influence of the suction valve. The line y' represents the pull due to the load of nearly 8 feet of water, the atmospheric pressure, and the weight of piston and rod; but it does not represent in this case the friction of the piston, which, on the downstroke, tends rather to reduce the pull on the rod and to produce compression. The difference between y and y', therefore, represents twice the combined piston and water friction in the delivery pipe. At this speed the water friction in a straight pipe is very small, and the difference is almost entirely due to piston friction. In this case it amounts to 30 per cent of y. The piston friction was, therefore, 15 per cent of the whole work done, equivalent to about 18 per cent of the weight of water. The pump leather was three-fourths inch wide. By the above formula $\frac{W}{D}$ would give $18\frac{3}{4}$ per cent. The width of the packing leather is said to influence the friction very little in heavy hydraulic work, but it seems to be important with the lighter pressures of ordinary pumping.

Piston friction is a large factor in the problem, varying nearly directly as the lift and inversely as the cylinder diameter. The tendency is to render the efficiency of the larger cylinders the higher. Since a single leather has been found insufficient for the prevention of leakage in pumps working in deep wells, two, or sometimes even three, are used. Where these leathers can not readily be procured they can be made by soaking good harness leather until pliable and then bolting it into a wooden form. To make the form, a board about three-fourths inch thick, having in it a hole of the same diameter as the pump cylinder, is nailed to a stiff backboard, and a cylindrical block, three-eighths of an inch less in diameter, is bolted concentrically within the circular opening. The bolts should be long enough so that the wet and pliable leather, laid over the hole, can be drawn down by the bolts and block, forcing the leather into position. The wrinkled edge of the leather can then be trimmed off, and when the whole is dry, removed and brought to a thin and even edge with a sharp knife. The friction of the piston leather should not be so great that on the downstroke the piston will not return by its weight alone.

Metal rods joined by screw couplings are commonly used in such pumps. By using a large rod a differential action is obtained in the delivery, rendering more uniform the work between the upstroke and the downstroke. A stiff rod is necessary where water is delivered on the downstroke. Wooden rods, because of their elasticity, frequently give less trouble at connections than metal rods, and where delivery pipes are large enough they are sometimes preferred.

PISTON SPEED.

Four factors tend to limit the piston speed: First, the excessive initial shock of lift on the upstroke; second, the tardy closing of valves; third, incomplete filling of the cylinder; fourth, the rapid lowering of general pump efficiency as the speed is increased.

To give to a body of water rapid motion necessitates a greater expenditure of energy than would be necessary simply to balance its static pressure. The acceleration of motion of the water is greatest at the beginning of the upstroke, and, as the rotative speed of the crank increases, the acceleration and the force necessary to produce it also increases. In practice the quantitative importance of this first stress is found to be largely a matter of pump design. If the rod is short and the pump connections and frame very rigid, the stresses will increase to a high limit earlier than when the machine has a certain lack of rigidity. The yielding of windmill towers to the various shocks of pump operation is in those structures a means of keeping down this stress. The extension and the elasticity of the pump rod also tend to soften the shock of starting the water column. The spring of parts is, therefore, in some degree, rather an advantage than otherwise, and the incorporation of a stiff spring into the pump rod would be advantageous.

In some of the pumps tested the pump rod had a length of about 40 feet and the machinery was placed upon an unusually rigid floor. Fig. 11 shows the maximum stress for a number of pumps tested from this floor. The horizontal distances represent average piston speed in feet per minute. With the pump piston at rest, the pump rod supported a certain constant load. With the pump piston in motion, this load no longer remained constant, but varied, being greater at the beginning of the stroke and at the end usually less; and this variation was found to increase as the speed increased. In fig. 11 distances above the base line exhibit this increase of stress. Vertical distances represent the maximum load while running, expressed in percentages of the quiet load. For example, in the test of one 4-inch pump it appeared that at a speed of 40 feet per minute the maximum load on the rod was 175 per cent of the quiet load, while at a speed of 80 feet per minute it was about 200 per cent. In the test of another 4-inch pump, at these same speeds the corresponding percentages were 200 per cent and 400 per cent. At piston speeds of 70 feet per minute the initial stress may be equivalent to from two and one-half to five times the weight of the water. The percentages are greatest where loads are light. Where the parts are not of ample proportions, or where they are loosely assembled, the shocks due to this initial stress soon destroy the machine.

The stresses shown in fig. 11 may in large part be due to the late closing of valves. In fig. 12, delayed and sudden application of the load after the piston had acquired a high speed, at one-tenth of its upward stroke, produced a much more severe shock than would have

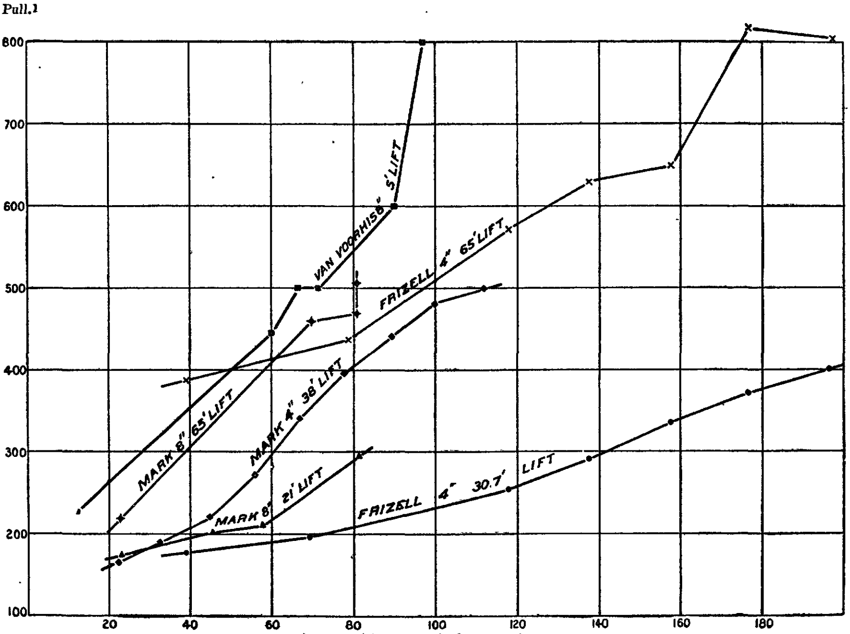


FIG. 11.—Diagram illustrating relation between piston speed and pull.

been felt if the valve had closed earlier. The effect of this slow closing of large valves is shown by the upper line in fig. 11. In the case of the lower valve it produces a blow on the under side of the piston, and frequently the shock of this blow is such as to jar a spray at each stroke from the upper surface of the water in the delivery pipe.

Incomplete filling of the cylinder limits piston speed, as shown in figs. 5, B (p. 16), to 6, D (p. 17). There is no advantage to be gained in running faster than the cylinder can fill. The upward tendency of the water into the cylinder depends rather upon other forces which we have called "head." We have seen that "suction" is the name

[1] Per cent of hydraulic pressure on the piston, representing maximum pull on the pump rod.

for an appearance rather than for an actual fact. The piston merely relieves the lower water column of load, and this column rises really under impulse from other agencies. It becomes the business, then, of the piston to keep the load that had held it down out of its way. Incomplete filling reduces the maximum stress nearly to that of a constant load, and indeed during the first part of the upstroke it results in more quiet running; but on the downstroke, on the other hand, with water and piston moving in opposite directions, it produces serious pounding.

Longer strokes admit of a much more rapid piston speed. This speed is limited by the number of times per minute the valves can open and close without serious shock and by the clear valve area. William Barr, in discussing steam pumps, says of the action of these valves: "The area of clear way through a set of valves in a water end should be not less than 40 per cent of the plunger area for speeds

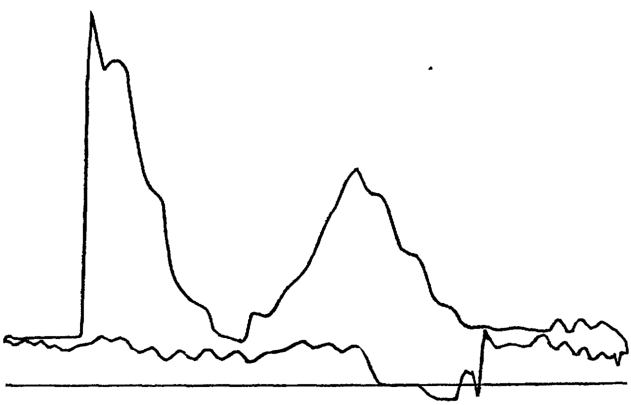


FIG. 12.—Dynamometer diagram of pump-rod stresses, showing excessive initial shock due to late closing of discharge valve.

of 100 feet per minute, 50 per cent for 125 feet, 60 per cent for 150 feet, 75 per cent for 175 feet, and 100 per cent for 200 feet per minute." With the usual windmill speeds, not exceeding 30 strokes per minute, a stroke of 2 feet would give a piston speed of 120 feet per minute, and should be possible with pumps having 50 per cent of clear valve area. Steam pumps, as stated by Barr, have a low efficiency as compared with some of the best forms of irrigation pumps; and while the above speed is possible, the valve area for high efficiency should be larger still. It is impossible accurately to set limits to pump speed, because it depends upon valve design and upon length of stroke, which in turn vary greatly in different forms of pumps.

The individual peculiarities of various makes are well brought out in fig. 11. The 4-inch Frizell pump, which gave the lowest maximum stress, and therefore made record of smooth running, had a valve area of but 27 per cent, while the 8-inch Van Voorhis pump, which had 97 per cent of valve area and a short stroke, was noisy. In general it may be said of a reciprocating pump that a speed of 100 feet per minute is about all that should be expected. The following table shows the highest piston speed employed in the test of a number of irrigation cylinders. For a regular working speed it would be excessive

Speeds of piston in irrigation cylinders.

No. of fig. herein.	Diameter, in inches.	Stroke, in inches.	Speed, in feet per minute.
18	4	14.00	114
27	4	23.70	198
23	6	14.11	141
14	8	13.95	90
32	8	10.00	216

Weisbach says: "Usually the velocity of the piston is not over 78 feet per minute, the common valve ranging between 40 and 60 feet, though piston velocities of 196 feet sometimes occur." In some high-class pumping engines speeds of 480 feet have been obtained.

While it is possible to run at a high speed, it may not, for practical reasons, be desirable to do so. By referring to the diagrams showing the efficiencies of pumps of various design it is seen that all are more efficient at slow speeds. A fall in efficiency of about 15 per cent—up to 30 strokes per minute—is to be expected, even in a good pump. A large pump run slowly will, therefore, work with small power and to better advantage than a smaller pump at a higher speed. For the very best results in connection with a windmill a higher gearing than the usual 3 to 1 would be advisable, the limiting factor being the increased expense of the larger size of pump thus made necessary.

DISCHARGE.

The discharge of a pump per stroke is measured by the capacity of the cylinder within the stroke length, less, however, the amount by which the cylinder may have failed to fill on the upstroke, and less, as well, those losses on the downstroke which will result from back-slip of water past the tardily closing valve and from leakage through the valve after closing. The discharge of various cylinders tested is given in diagrams accompanying the description of each. Reuleaux gives the discharge, in per cent of the geometrical displacement, as 92 per cent at a speed ranging from 27 to 40 strokes per minute, 95 per cent at 50 strokes, and 98 per cent at 60 strokes. At slow speed the leakage reduces the discharge by an amount ranging from 1 to 5 per cent. This loss diminishes, as a percentage, with increase of speed, but with high speed the loss from slow closing of valves rises and soon outweighs it. Weisbach states the loss to be from 5 to 15 per cent, which seems excessive for such cylinders as have been tested by the writer.

Where the velocity of water is high, owing either to a high piston speed or to a small pipe, the discharge per stroke may even exceed the cylinder capacity per stroke. The momentum of the rising water may be sufficient to hold the valves open for a brief interval after the completion of the upstroke, and thus for a moment to continue intake after the beginning of the downstroke. The paradoxical character of this effect is brought out by the discharge curves in figs. 19 (p. 32) and 22 (p. 35), and by the stress diagram, fig. 13. This stress diagram is a record from the work of the pump shown in fig. 18 (p. 32), and indicates that the tension on the rod in that pump, under the conditions imposed, was relieved by the momentum of the rising column of water before the end of the stroke had been reached, and that this momentum impulse continued effective until the lower valve had closed, at A (fig. 13), on the return stroke. This same effect may be observed when the suction pipe is carried to a considerable horizontal distance, increasing the weight of the moving water. Unless a suction chamber is put on in such cases, the suction pipe should increase in area directly as the length increases. Fig. 19 shows an example of the discharge being increased by a more rigid setting.

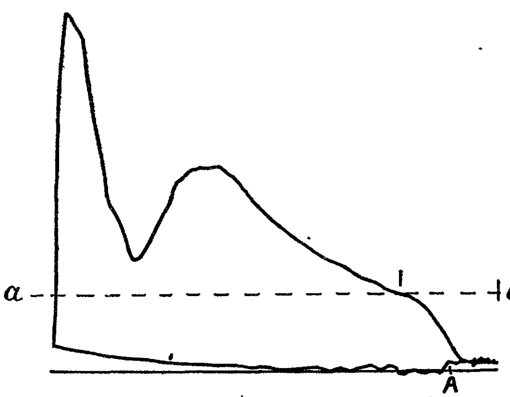


FIG. 13.—Dynamometer diagram of pump-rod stresses, showing excessive discharge.

EFFICIENCY OF PUMPS.

The efficiency of a pump is that percentage of the effort expended which is returned in useful work as water lifted. Efficiency will vary with the design, speed, and adaptation to the particular use. In small steam pumps the useful work done will account for only about 50 per cent of the power developed by the steam. Weisbach states the efficiency to be, "in well-designed pumps, working under favorable conditions, 80 per cent; in pumps of average perfection, 75 per cent; and in ordinary pumps, 70 per cent, and sometimes 65 per cent only." Mr. William O. Weber[1] gives the efficiency of reciprocating pumps as follows:

Efficiency of reciprocating pumps.

Lift (feet)	10	15	20	25	30	35	40	50	60	80	100	120	160	200	240	280
Efficiency	30	45	55	61	66	68	71	75	77	82	85	87	90	89	88	85

[1] Trans. Am. Soc. Mech. Eng., vol 7, p. 598.

To determine the usual range of efficiency for such pumps as are offered for irrigation purposes, and to determine what room there may be in them for improvement, the author has made something more than 1,000 measurements of various designs. The process by which were obtained the stress diagrams previously referred to affords a method for determining the amount of work, useful and otherwise, performed by the pump. By weighing the water discharged at various speeds and lifts a rating curve for each pump was obtained; and the useful work was determined by multiplying the weight of water discharged by the total lift. The diagrams show only the work actually performed by the pump rod, and therefore do not exhibit the ineffectual effort of the mechanism which transmitted power to the rod. The efficiencies are those of the pump cylinder and the pipes alone. They seem to show that some forms of the irrigation pumps tested here compare very favorably with those tested by others. Each pump has a particular sphere to which it is best adapted. No great increase of efficiency over the best now to be had is to be expected in any new form. Many of those now in the market could be much improved. The pumps represented in the drawings are typical, and by noting their performance, as shown in the following diagrams, the general range of efficiency of other pumps may often be closely approximated.

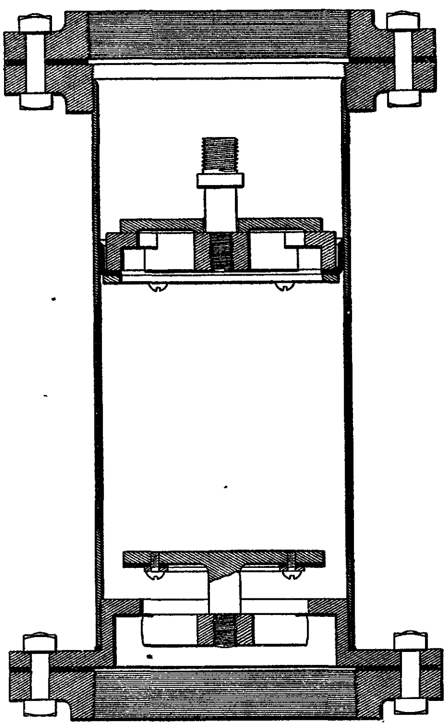


FIG. 14.—Vertical section of 8-inch brass cylinder with disk valves.

The efficiency is seen to rise with increase of lift and to fall with increase of speed. The maximum efficiency is reached at lifts lower than those of Weber's table. Certain pumps may be expected to have an efficiency above 80 per cent at the usual speeds and at moderate lifts. Efficiency with such will but slightly increase, if at all, at lifts of more than 50 feet.

MARK PUMP.

Fig. 14 shows an 8-inch brass cylinder having disk valves with limited movement. It is made by the Mark Manufacturing Company, of Chicago, Illinois, and is specially designed for irrigation work.

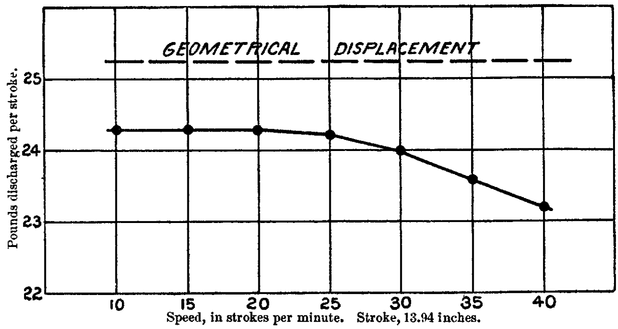


FIG. 15.—Discharge curve of 8-inch Mark pump.

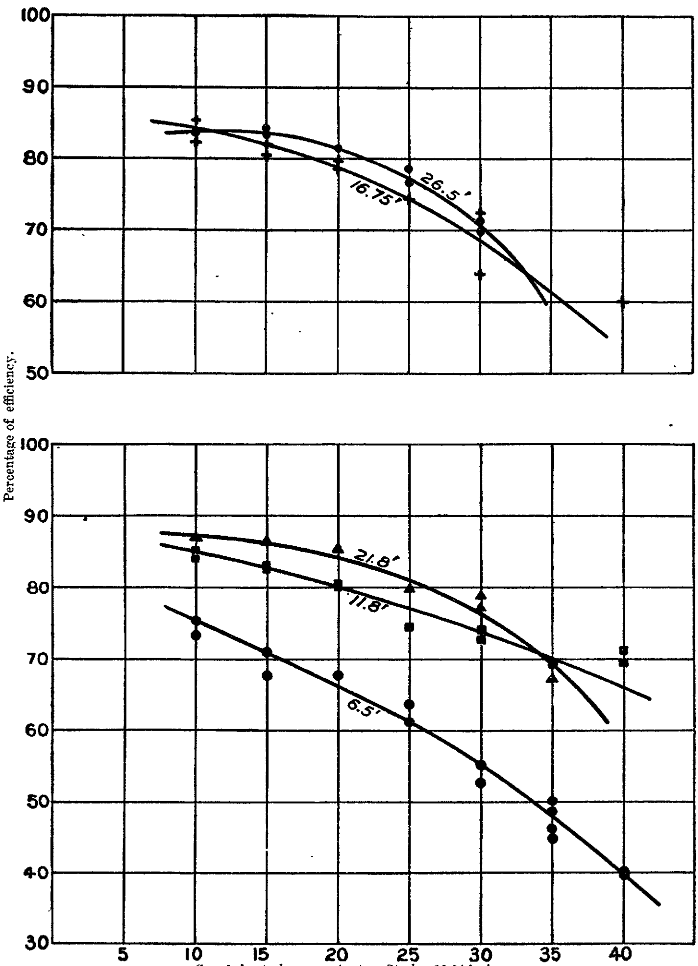


FIG. 16.—Varying-efficiency curve of 8-inch Mark pump, showing effect of varying height of lift and speed. (The line for 11.8 lift has been placed 10 per cent too high in the column 35–40. It should curve downward at the end and not cross the line 21.8.)

The lower valve is limited in movement by a cross attached to a pendent spindle on the valve disk. This valve is leather faced and has a clear opening of 36 per cent of the piston area. The upper valve area

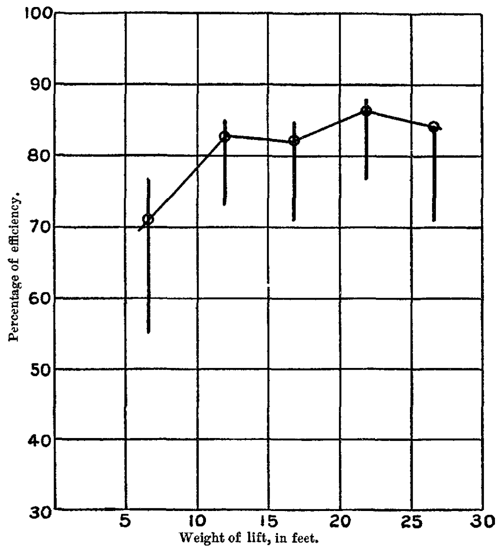


FIG. 17.—Variation of efficiency with varying height of lift of 8-inch Mark pump at speed of 15 strokes per minute.

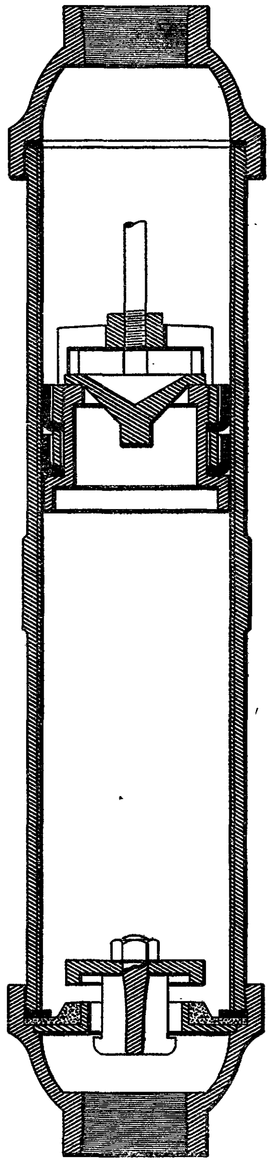


FIG. 18.—Vertical section of a brass-lined cylinder of small Mark pump.

is 34.7 per cent, the area between the edge of the disk and the cylinder walls is 53 per cent,

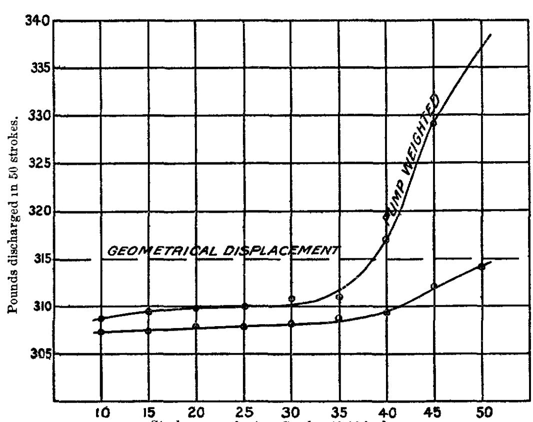


FIG. 19.—Discharge curve of 4-inch Mark pump.

and the area between the edge of the disk and the valve seat is slightly in excess of the valve area. The weight of the pump is $145\frac{1}{2}$ pounds; of the piston and rod alone, $67\frac{1}{2}$ pounds. The pump rod was a three-fourths-inch pipe, and the delivery pipe was 6-inch. This pump is typical of the best design of those having large disk valves. It shows a very high efficiency at slow speeds, but a rapidly falling efficiency as the speed increases. Fig. 15 shows the discharge to be about 96 per cent of the geometrical displacement of the piston at a speed of 25 strokes per minute, but

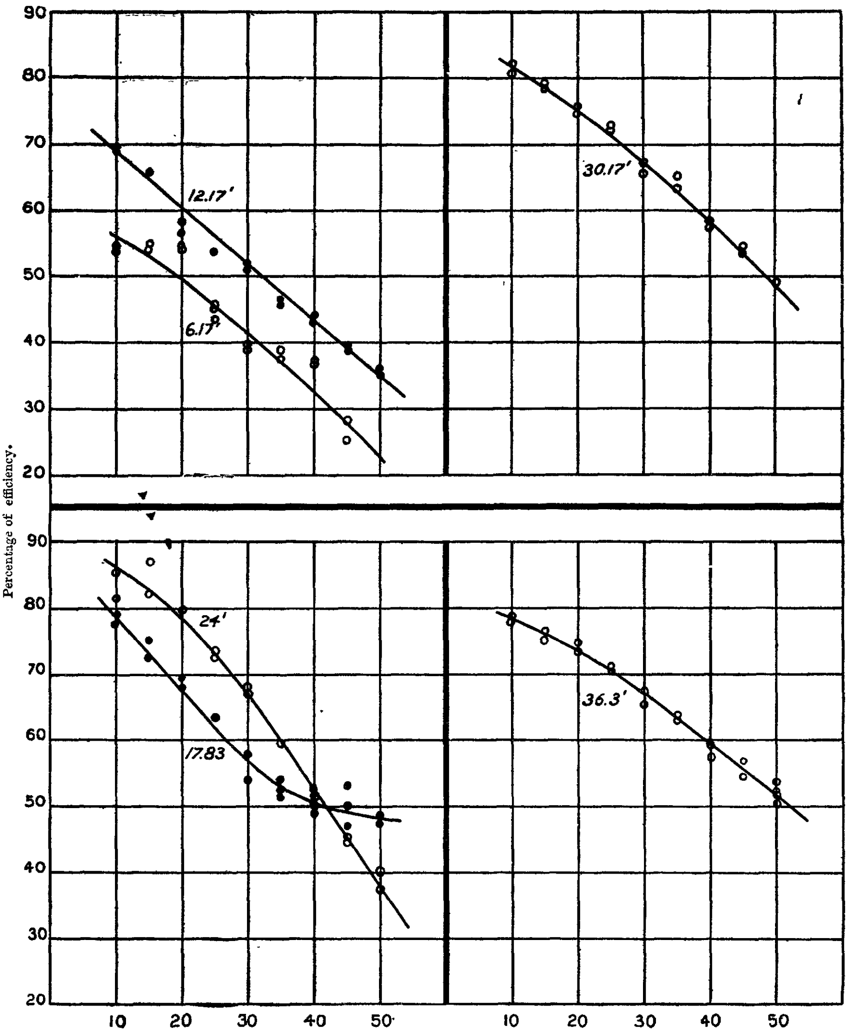


FIG. 20.—Varying-efficiency curves of 4-inch Mark pump, showing effect of varying height of lift and speed. (The lines for 17.83 and 24 should not cross. The diagram is erroneous in this respect. The correct figures for these curves are given on pages 37, 38.)

also shows that it has a falling rate beyond that speed. The efficiency is high for strokes fewer than 20 per minute, and at lifts exceeding 10 feet. Fig. 16 shows the efficiency curves at five lifts, from $6\frac{1}{2}$ to $26\frac{1}{2}$ feet.

In fig. 17 the heavy vertical lines show the range of efficiency at varying speeds at each lift, and the irregular line joining those points represents the efficiencies at a speed of 15 strokes per minute. It would seem from this diagram that a considerable increase of efficiency at a higher lift is not to be expected.

Fig. 18 gives a section of a brass-lined cylinder of the same make, which is typical of the better-made small pumps that provide for a suction and discharge pipe of less diameter than the cylinder. It has a 14-inch stroke, with a metal disk-valve carried in the piston. The upper valve area is 30 per cent of the cylinder area. The lower metal valve seats on a rubber ring. The lower valve area is 18 per cent. The suction and delivery pipes are 2-inch.

Fig. 19 shows two discharge curves of this pump. When rigidly

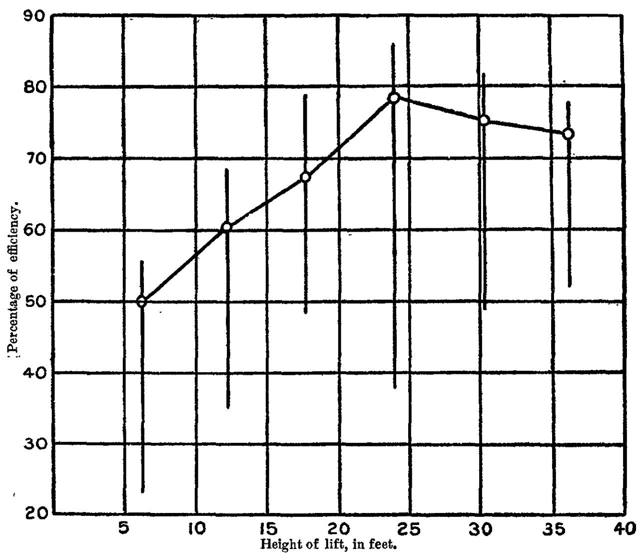


FIG. 21.—Variation of efficiency with varying height of lift of 4-inch Mark pump at speed of 20 strokes per minute.

held, so that the rapidly moving water could not jar the pump and pipe, the discharge was greater than when simply suspended in the well, as usual. The discharge is greater than the geometrical displacement of the piston at speeds above 38 strokes per minute, owing to the high velocity of the water in a small pipe. The efficiencies at six different lifts are given in the tables, and in the diagram, fig. 20. The rapidly lowering efficiency shows the effect of small connections and of small valve area.

From fig. 21 the maximum efficiency seems to be reached at about a 25-foot lift. This pump was used in an interesting test which brought out the effect of varying the suction lift. The total lift—suction and direct together—was 38 feet. The cylinder was placed at different heights in the pipe, giving different heights of suction, ranging from nothing to 30¼ feet. Fig. 22 shows the effect on the discharge of the varying suction lift. In all cases, as the speed increases, the tendency

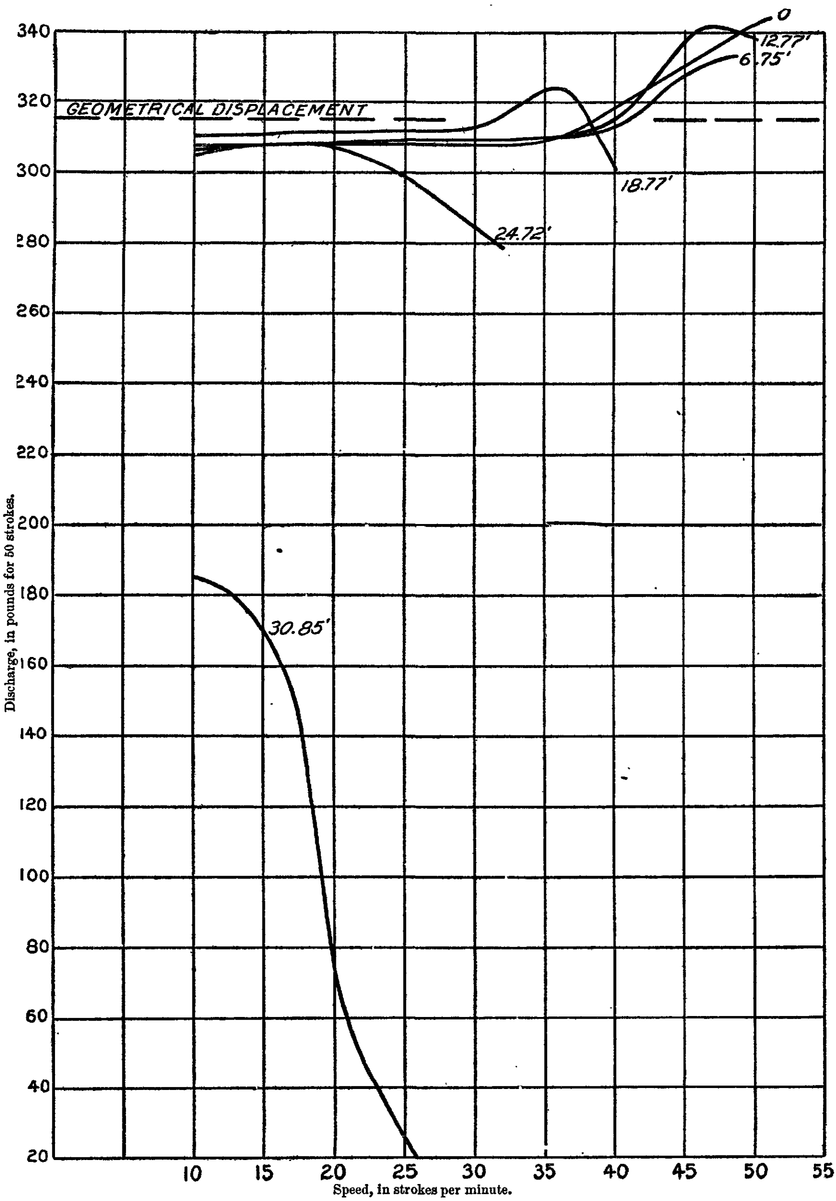


FIG. 22—Discharge curve for 4-inch Mark pump at a total lift of 38 feet and at various suction lifts.

is to increase the discharge; but when the lift gets to be so great that the atmospheric pressure is unequal to the filling of the cylinder the discharge curve rapidly falls. This maximum rise of the curve is seen to occur earlier with each successive increase in the suction lift. Figs. 5, *B* to 6, *D* (pp. 16, 17) give diagrams of stresses in the pump-rod, and may be compared with fig. 5, *A*, which is typical of normal conditions. The tables present the results of suction tests.

Discharge of 8-inch Mark pump stroke, 13.94 inches.

Speed.[1]	Discharge for 15 strokes; lift, 6.5 feet.	Discharge for 15 strokes; lift, 11.8 feet.	Discharge for 15 strokes; lift, 26.5 feet.	Plotted average discharge per stroke.
10	367½	371	356	24.30
10	369½	369	355½	
15	366½	368½	358¼	24.30
15	368	362½	357½	
20	368	369½	358	24.30
20	365½	367	358½	
25	365	369½	355¼	24.266
25	365	370½	358¾	
30	358	367	351½	24
30	358	361½	351¾	
35	346½	358		23.60
35	348	356½		
40	337½	359		23.20
40	352	357		

[1] Strokes per minute.

Figures from efficiency tests of 8-inch Mark pump; stroke, 13.94 inches.

LIFT, 6.5 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on the rod.	Foot-pounds of work given to the rod.	Mechanical efficiency.	Maximum tension on the rod.	Minimum tension on the rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	24.300	157.95	184.72	214.28	73.7	312	72
10	24.300	157.95	179.64	208.38	75.7		
15	24.300	157.95	200	232.36	67.9		
15	24.300	157.95	191.52	222.16	71.1		
20	24.300	157.95	200	232.36	67.9		
20	24.300	157.95	200.16	232.50	67.8		
25	24.266	157.73	213.36	247.50	63.7		
25	24.266	157.73	220.32	255.57	61.3		
30	24	156	244.04	283.08	55.1		
30	24	156	252.52	292.92	53.2	648	— 72
35	23.600	153.40	284.72	330.28	46.4		
35	23.600	153.40	271.16	314.55	48.7	664	— 80
35	23.600	153.40	289.80	336.17	45.6		
35	23.600	153.40	259.32	300.81	50.9	738	—108
35	23.600	153.40	266.94	309.65	49.5		
40	23.200	150.80	325.38	377.44	39.7		
40	23.200	150.80	320.04	371.25	40.5		

LIFT, 11.8 FEET.

10	24.30	286.74	293.60	340.58	84.2	480	70
10	24.30	286.74	290.60	337.10	85		
15	24.30	286.74	297	344.52	83.2		
15	24.30	286.74	298.05	345.74	82.9		
20	24.30	286.74	308.50	357.86	80.2		
20	24.30	286.74	305.55	354.44	80.8		
25	24.266	286.34	311.95	385.06	74.3		
25	24.266	286.34	311.95	385.06	74.3		
30	24	283.20	328.95	381.58	74.2		
30	24	283.20	335.45	389.12	72.8	650	— 25
35	23.60	278.48	346.80	402.29	69.2	700	— 50
35	23.60	278.48	346.80	402.29	69.2		
40	23.20	273.76	384.60	446.14	61.3		
40	23.20	273.76	395.30	458.55	59.4	1,140	—100

Figures from efficiency tests of 8-inch Mark pump; stroke, 13.94 inches—Cont'd.

LIFT, 16.75 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on the rod.	Foot-pounds of work given to the rod.	Mechanical efficiency.	Maximum tension on the rod.	Minimum tension on the rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	24.30	407.03	426.22	494.68	82.3	585	60
10	24.30	407.03	419.85	487.45	85.1		
15	24.30	407.03	435.90	506.08	80.4		
15	24.30	407.03	429.45	498.16	81.7		
20	24.30	407.03	445.50	516.78	78.7		
20	24.30	407.03	441.97	512.69	79.5		
25	24.266	406.47	458.32	531.66	76.4		
25	24.266	406.47	467.92	542.79	74.8		
30	24.00	402.00	477.52	552.93	72.5	900	0
30	24.00	402.00	529.72	619.48	63.9		
35	23.60	395.30	555.30	644.15	61.3	1,170	—150
40	23.20	388.60	558.45	647.80	59.9	1,245	—150
40	23.20	388.60	564.82	655.20	59.3		

LIFT, 21.8 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on the rod.	Foot-pounds of work given to the rod.	Mechanical efficiency.	Maximum tension on the rod.	Minimum tension on the rod.
10	24.300	529.74	536.18	607.67	87.1	825	75
10	24.300	529.74	536.18	607.67	87.1		
15	24.300	529.74	539.47	611.40	86.6		
15	24.300	529.74	539.47	611.40	86.6		
20	24.300	529.74	549.30	622.54	85.1		
20	24.300	529.74	546.00	618.80	85.6	960	75
25	24.266	529.01	582.24	659.87	80.2	997.5	—37.5
25	24.266	529.01	583.65	661.47	79.9		
30	24.000	523.20	595.35	674.73	77.5		
30	24.000	523.20	595.35	674.73	77.5		
30	24.000	523.20	585.52	663.59	78.8		
35	23.600	514.48	684.15	775.37	69.8	1,387.5	—165
35	23.600	514.48	707.17	811.46	67.5		

LIFT, 26.5 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on the rod.	Foot-pounds of work given to the rod.	Mechanical efficiency.	Maximum tension on the rod.	Minimum tension on the rod.
10	24.300	643.95	684.15	775.37	83.5	997.5	45.0
10	24.300	643.95	680.85	771.83	83.6		
15	24.300	643.95	677.62	767.97	83.8		
15	24.300	643.95	674.32	764.23	84.2		
20	24.300	643.95	694.05	786.59	81.8		
20	24.300	643.95	697.35	790.33	81.4		
25	24.266	643.06	720.37	816.42	78.7	1,275	—37.5
25	24.266	643.06	740.77	839.44	76.6		
30	24.000	630.00	786.15	890.97	71.3		
30	24.000	630.00	801.67	908.56	69.9		

Discharge of 4-inch Mark pump at various speeds and lifts.

Speed.	Discharge, in pounds, for 50 strokes for lift:				Average discharge, in pounds, per stroke by discharge curve.
	6.17 feet.	12.17 feet.	24.2 feet.	30.17 feet.	
10	309.5	311	304	307	6.146
10	306.5	309.5	303.5	307.5	
15	306	312	305	309	6.148
15	306	309	305.5	308	
20	306	310.5	308	307	6.152
20	305.5	310.5	308.5	308	
25	303	310.5	309	307	6.156
25	305.5	308	308.5	309.5	
30	308	308	312.5	309	6.16
30	306	307	309.5	307	
35	309	308.5	310	307	6.17
35	309	308.5	311	308.5	
40	306	309.5	311.5	311	6.192
40	306.5	308.5	309	313	
45	300	312	321		6.24
45	301	310.5	328.5		
50		314.5			6.28
50		313.5			

Discharge of 4-inch Mark pump at various lifts and speeds; pump cylinder heavily weighted.

Speed.	Discharge, in pounds, for 50 strokes for lift: 30.17 feet.	Discharge, in pounds, for 50 strokes for lift: 36.3 feet.	Average discharge, in pounds, per stroke by discharge curve.	Speed.	Discharge, in pounds, for 50 strokes for lift: 30.17 feet.	Discharge, in pounds, for 50 strokes for lift: 36.3 feet.	Average discharge, in pounds, per stroke by discharge curve.
10	308.5	305.5	6.16	30	311.5	310	
10	308.5	303.5		35	311	310.5	6.23
15	311.5	308.5	6.192	35	311.5	310.5	
15	312	307		40	314.5	321	6.34
20	311.5	308.5	6.196	40	317	316	
20	312	307.5		45	332	328	6.584
25	312	308	6.198	45	328	329	
25	310.5	309		50	339.5	331.5	6.744
30	312	309.5	6.204	50	341	337	

Figures from efficiency tests of 4-inch Mark pump; stroke, 13.91 inches.

LIFT, 6.17 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
10	6.146	37.92	60.58	70.21	54	81
10	6.146	37.92	59.93	69.46	54.6	
15	6.148	37.94	60.58	70.21	54	
15	6.148	37.94	59.62	69.1	54.9	
20	6.152	37.96	59.68	69.17	54.8	
20	6.152	37.96	60.58	70.21	54	
20	6.152	37.96	59.23	68.65	55.3	107
25	6.156	37.98	72.44	83.95	45.2	
25	6.156	37.98	71.47	82.84	45.8	132
25	6.156	37.98	75	78.12	43.6	
30	6.16	38.01	82.37	95.47	39.8	
30	6.16	38.01	83.97	97.33	39	158
35	6.17	38.07	86.86	100.7	37.8	159
35	6.17	38.07	86.86	100.7	37.8	
40	6.192	38.21	88.5	102.5	37.2	
40	6.192	38.21	89.42	103.6	36.8	
45	6.24	38.5	117.4	136.1	28.2	
45	6.24	38.5	131.8	152.8	25.2	294
48	6.27	38.69	173.3	200.9	19.2	

LIFT, 12.17 FEET.

10	6.146	74.8	93	107.8	69.3	117
10	6.146	74.8	92.6	107.3	69.6	
15	6.148	74.82	97.7	113.4	65.9	159
15	6.148	74.82	97.7	113.4	65.9	
20	6.152	74.87	110.7	128.3	58.3	147
20	6.152	74.87	113.6	131.7	56.8	
25	6.156	74.92	120.8	140	53.5	
25	6.156	74.92	120.2	139.3	53.7	
30	6.16	74.97	124.3	144.1	52	
30	6.16	74.97	126.6	146.7	51	223
35	6.17	75.09	139.1	161.2	46.5	255
35	6.17	75.09	140.1	162.4	46.2	
40	6.192	75.3	146.5	169.8	44.4	
40	6.192	75.3	150	173.8	43.3	300
45	6.24	75.94	166.7	193.2	39.3	
45	6.24	75.94	165.4	191.7	39.6	366
50	6.28	76.4	187	216.8	35.2	
50	6.28	76.4	182.7	211.7	36.1	372

Figures from efficiency tests of 4-inch Mark pump; stroke, 13.91 inches—Continued.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
LIFT, 17.83 FEET.						
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
10	6.146	109.6	119.5	139	79.1	210
10	6.146	109.6	121.5	140.9	77.8	----------
15	6.148	109.6	126.1	146.1	75	----------
15	6.148	109.6	130.5	151.3	72.4	----------
20	6.152	109.7	139	161.1	68	----------
20	6.152	109.7	135.7	157.3	69.7	225
25	6.156	109.8	137	158.8	69.1	264
25	6.156	109.8	148.7	172.4	63.7	270
30	6.16	109.8	162.9	188.8	58.1	375
30	6.16	109.8	175.1	202.9	54.1	411
35	6.17	110	183.5	212.6	51.7	432
35	6.17	110	175.6	208.3	52.8	420
40	6.192	110.4	184.3	213.6	51.7	----------
40	6.192	110.4	189.4	219.6	50.3	489
45	6.24	111.3	211.8	245.5	45.3	555
45	6.24	111.3	213.7	248.7	44.9	----------
50	6.28	112	239.5	277.6	40.3	----------
50	6.28	112	257.5	298.4	37.5	684
LIFT, 24.2 FEET.						
10	6.146	148.7	150	173.8	85.5	228
10	6.146	148.7	157.2	182.2	81.6	234
10	6.146	148.7	147.4	170.8	87	225
15	6.148	148.8	156.5	181.4	82	255
15	6.148	148.8	156.5	181.4	82	258
20	6.152	148.9	161.1	186.7	79.7	285
20	6.152	148.9	161.1	186.7	79.7	285
25	6.156	149	177.3	205.5	72.5	348
25	6.156	149	174.7	202.5	73.6	345
30	6.16	149.1	188.8	218.8	68.1	447
30	6.16	149.1	191.4	221.8	67.2	471
35	6.17	149.3	215.3	249.5	59.8	558
35	6.17	149.3	200.4	232.3	64.2	501
40	6.192	149.8	262.7	304.4	49.2	720
40	6.192	149.8	244.6	283.5	52.8	669
45	6.24	151	243.6	282.3	53.5	665
45	6.24	151	242.4	281	53.7	640
45	6.24	151	275.8	319.6	47.2	----------
45	6.24	151	257.5	298.4	50.6	760
50	6.28	152	267.2	309.5	49	775
50	6.28	152	274.7	318.4	47.7	775
LIFT, 30.17 FEET.						
10	6.146	185.4	195.6	226.7	81.7	300
10	6.146	185.4	191.3	221.7	83.6	305
15	6.148	185.5	198.9	230.5	80.4	360
15	6.148	185.5	200	231	80	345
20	6.152	185.6	206.7	239.6	77.4	380
20	6.152	185.6	206.7	239.6	77.4	390
25	6.156	185.7	208.9	242	76.7	410
25	6.156	185.7	211	244.6	75.9	410
30	6.16	185.8	235.9	273.4	67.9	535
30	6.16	185.8	234.8	272.2	68.2	535
35	6.17	186.1	255.4	296	62.8	650
35	6.17	186.1	257.6	298.5	62.3	675
40	6.192	186.8	260.8	302.3	61.8	675
40	6.192	186.8	265.1	307.3	60.8	700
45	6.24	188.2	283.4	328.5	57.3	830
45	6.24	188.2	288.8	334.7	56.2	320

Figures from efficiency tests of 4-inch Mark pump; stroke, 13.91 inches—Continued.

LIFT, 30.17 FEET.

[Pump heavily weighted.]

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
10	6.16	185.8	195.9	227	81.8	315
10	6.16	185.8	194.8	225.8	82.3	305
15	6.192	186.8	204.3	236.8	78.8	350
15	6.192	186.8	203.2	235.6	79.3	350
20	6.196	186.9	216.3	250.7	74.5	410
20	6.196	186.9	213	246.9	75.7	415
25	6.198	187	222.8	258.2	72.4	490
25	6.198	187	220.6	255.7	72.7	480
30	6.204	187.2	240.3	278.5	67.2	575
30	6.204	187.2	245.7	284.7	65.7	595
35	6.23	187.9	248.9	288.5	65.1	620
35	6.23	187.9	254.3	294.7	63.7	615
40	6.34	191.2	281.2	326	58.6	730
40	6.34	191.2	285.5	330.9	57.8	755
45	6.58	197.6	318.7	369.4	53.7	880
45	6.58	197.6	314.1	364	54.5	845
50	6.744	203.5	356.8	413.6	49.2	1,000

LIFT, 36.3 FEET.

[Pump heavily weighted.]

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
10	6.16	223.6	244.6	283.5	78.9	355
10	6.16	223.6	245.6	284.7	78.5	350
15	6.192	224.8	253.3	293.5	76.5	415
15	6.192	224.8	256.5	297.3	75.6	440
20	6.196	224.9	263	304.8	73.8	465
20	6.196	224.9	259.7	301	74.7	465
25	6.198	225	272.7	316.1	71.1	560
25	6.198	225	273.8	317.3	70.9	570
30	6.204	225.2	287.7	333.5	67.5	680
30	6.204	225.2	296.3	343.4	65.5	700
35	6.23	226.1	307.9	356.9	63.3	760
35	6.23	226.1	306.9	355.7	63.6	760
40	6.34	230.2	335.8	389.3	59.1	875
40	6.34	230.2	343.3	398	57.8	925
45	6.584	239	360.5	417.9	57.2	925
45	6.584	239	367	435.3	54.9	985
50	6.744	244.8	418.5	485	50.4	1,035
50	6.744	244.8	391.6	453.9	53.9	1,000
50	6.744	244.8	403.4	467.6	52.3	1,050
50	6.744	244.8	403.4	467.6	52.3	1,065
50	6.744	244.8	409.9	475.1	51.5	1,000

Plotted averages of discharge per stroke, suction test of 4-inch Mark pump.

Speed.	Discharge, in pounds, per stroke.					
	Total lift, 38 feet; suction lift, 30.85 feet.	Total lift, 37.96 feet; suction lift, 24.77 feet.	Total lift, 38 feet; suction lift, 18.77 feet.	Total lift, 37.96 feet; suction lift, 12.77 feet.	Total lift, 37.92 feet; suction lift, 6.73 feet.	Total lift, 37.92 feet; suction lift, 0.6 foot.
10	185.5	6.11	6.21	6.16	6.13	6.14
15	172.5	6.154	6.226	6.168	6.16	6.15
20	79.5	6.15	6.23	6.176	6.168	6.156
25	25.5	5.996	6.234	6.184	6.174	6.16
30	7.5	5.730	6.256	6.192	6.178	6.164
35			6.46	6.2	6.184	6.184
40			6.024	6.266	6.250	6.34
45				6.754	6.56	6.556
50				6.78	6.68	6.706

Discharges of 4-inch Mark pump at various suction lifts and speeds.

Speed.	Discharge, in pounds, for 50 strokes.					
	Total lift, 38 feet; suction lift, 30.85 feet.	Total lift, 37.96 feet; suction lift, 24.77 feet.	Total lift, 38 feet; suction lift, 18.77 feet.	Total lift, 37.96 feet; suction lift, 12.77 feet.	Total lift, 37.92 feet, suction lift, 6.73 feet.	Total lift, 38.08 feet; suction lift, 0.6 foot.
10	185.5	306	311.5	308.5	306.5	306.5
10		305	309.5	307.5	306.5	307.5
15	172.5	310.5	311	306.5	307.5	307
15		305	312	309.5	306.5	308
20	79.5	308.5	312.5	307.5	309	308.5
20		306.5	310	310	309.5	307
25	25.5	299.5	313	310.5	308.5	307
25		300	311	309	308.5	309.5
30	7.5	290	312.5	308	309	308
30		283.5	313	309	309	307
35			323	311	309.5	311.5
35			323	309	309	307
40			301.5	312.5	312.5	318.5
40			301	314		318.5
45				337.5	328	
45				338		
50				340	334	331.5
50				338		339

Figures from suction test of 4-inch Mark pump.

TOTAL LIFT, 38.08 FEET; SUCTION LIFT, 0.6 FOOT.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
10	6.14	233.8	253.3	293.6	79.6	350
10	6.14	233.8	256.6	297.4	78.6	370
15	6.15	234.2	257.7	298.6	78.4	390
15	6.15	234.2	257.7	298.6	78.4	400
20	6.156	235.4	263.1	304.9	76.8	460
20	6.156	235.4	267.5	310	75.6	470
20	6.156	235.4	264.2	306.2	76.5	470
25	6.16	235.5	275.1	318.8	73.5	530
25	6.16	235.5	276.1	320	73.3	530
30	6.164	235.7	297.8	345.2	68	710
30	6.164	235.7	316.3	366.6	64	760
35	6.184	235.6	322.8	374.2	62.9	780
35	6.184	235.6	340.2	394.3	59.7	860
40	6.34	241.4	359.3	416.4	57.9	910
40	6.34	241.4	368	426.5	56.6	920
45	6.556	249.6	394	456.6	54.6	990
45	6.556	249.6	390.7	452.8	55.1	1,000
50	6.706	255.4	415.6	481.7	53	1,030
50	6.706	255.4	418.8	485.4	52.6	1,030

TOTAL LIFT, 37.92 FEET; SUCTION LIFT, 6.73 FEET.

10	6.13	232.4	245.1	284	81.8	380
10	6.13	232.4	243.5	282.2	82.3	375
15	6.16	233.6	256.5	297.3	78.5	435
15	6.16	233.6	255.5	296.1	78.9	435
20	6.168	233.9	266.4	308.7	75.7	490
20	6.168	233.9	270.7	313.8	74.5	505
25	6.174	234.1	287.1	332.8	70.3	585
25	6.174	234.1	292.6	339.1	69	630
30	6.178	234.2	302.2	350.2	66.9	660
30	6.178	234.2	315.2	365.3	64.1	720
35	6.184	234.5	332.2	385.1	60.9	785
35	6.184	234.5	332.2	385.1	60.9	800
40	6.25	237	359.9	416.9	56.8	870
40	6.25	237	381.5	442.1	53.6	950
45	6.56	248.8	422.4	489.6	50.8	1,005
45	6.56	248.8	404.1	468.3	53.1	960
50	6.68	253.3	468.8	543.3	46.6	1,160
50	6.68	253.3	472	547.1	46.3	1,050

Figures from suction test of 4-inch Mark pump—**Continued.**

TOTAL LIFT, 37.96 FEET; SUCTION LIFT, 12.77 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
10	6.16	233.8	254.4	294.8	79.3	370
10	6.16	233.8	252.2	292.3	80	380
15	6.168	234	256	296.7	78.9	410
15	6.168	234	260.4	301.8	77.5	410
20	6.176	213.8	263.1	304.9	76.8	460
20	6.176	213.8	264.2	306.2	76.5	440
25	6.184	234.7	277.3	321.4	73	560
25	6.184	234.7	282.8	327.7	71.6	560
30	6.192	235	301.3	349.2	67.3	650
30	6.192	235	299.1	346.7	67.8	640
35	6.2	235.4	319.6	370.4	63.5	715
35	6.2	235.4	318.5	369.1	63.7	705
40	6.266	237.9	394.6	457.3	52	915
45	6.754	256 4	414.1	480	53.4	885
45	6.754	256.4	407.6	472.4	54.2	885
45	6.754	256.4	439.1	508.9	50.3	....
45	6.754	256.4	430.4	498.9	51.4	....
50	6.78	257.4	431.8	500.5	51.4	....
50	6.78	257.4	443.7	514.3	50	1,000

TOTAL LIFT, 38 FEET; SUCTION LIFT, 18.77 FEET.

10	6.21	236	239.1	277.1	85.1	310
10	6.21	236	241.2	279.6	84.4	325
15	6.226	236.6	254.4	294.8	80.2	....
15	6.226	236.6	250	289.8	81.6	385
20	6.23	236.7	268.5	311.2	76	465
20	6.23	236.7	264.1	306.1	77.3	440
25	6.234	236.9	271.7	314.9	75.2	485
25	6.234	236.9	271.7	314.9	75.2	490
30	6.256	237.7	302.2	350.2	67.8	550
30	6.256	237.7	302.2	350.2	67.8	555
30	6.256	237.7	303.8	352.1	67.5	570
35	6.46	245.5	350.2	405.9	60.4	725
35	6.46	245.5	340.5	394.7	62.2	765
40	6.024	225.7	365.7	423.9	54	830
40	6.024	225.7	355.9	422.1	55.5	....
40	6.024	225.7	364.6	422.6	54.1	870
40	6.024	225.7	364.6	422.6	54.1	....

TOTAL LIFT, 37.96 FEET; SUCTION LIFT, 24.77 FEET.

10	6.11	231.9	240.1	278.3	83.3	290
10	6.11	231.9	241.2	279.6	82.9	285
15	6.154	233.6	250	289.8	80.6	310
15	6.154	233.6	248.9	288.5	80.9	305
20	6.15	233.5	258.7	299.9	77.8	345
20	6.15	233.5	268.4	311.1	75	365
20	6.15	233.5	269.5	312.3	74.7	370
20	6.15	233.5	273.8	317.4	73.5	375
20	6.15	233.5	259.8	301.1	77.5	345
25	5.996	227.6	291.7	338	67.3	425
25	5.996	227.6	291.5	337.8	67.3	500
25	5.996	227.6	286	331.5	68.6	415
25	5.996	227.6	287.1	332.8	68.4	525
25	5.996	227.6	292.8	339.3	67	405
30	5.73	213.7	279.5	323.9	65.9	585
30	5.73	213.7	270.7	313.8	68.1	475

TOTAL LIFT, 38 FEET; SUCTION LIFT, 30.85 FEET.

10	3.71	141	178.3	206.6	68.2	265
10	3.71	141	178.3	206.6	68.2	265
15	3.45	131.1	165.2	191.5	68.4	265
15	3.45	131.1	175	202.8	64.6	265
20	1.59	60.4	116.8	135.4	44.6	270
20	1.59	60.4	125	144.9	41.7	270
25	.516	19.6	96	113.9	17.21	270
25	.516	19.6	102.6	118.9	16.4	275
30	.14	5.32	109.8	127.2	4.1	285
30	.14	5.32	81.1	94	5.6	295

FRIZELL PUMP.

Fig. 23 represents a piston pump having a valve chamber surrounding the cylinder, and containing eight small clack discharge valves.

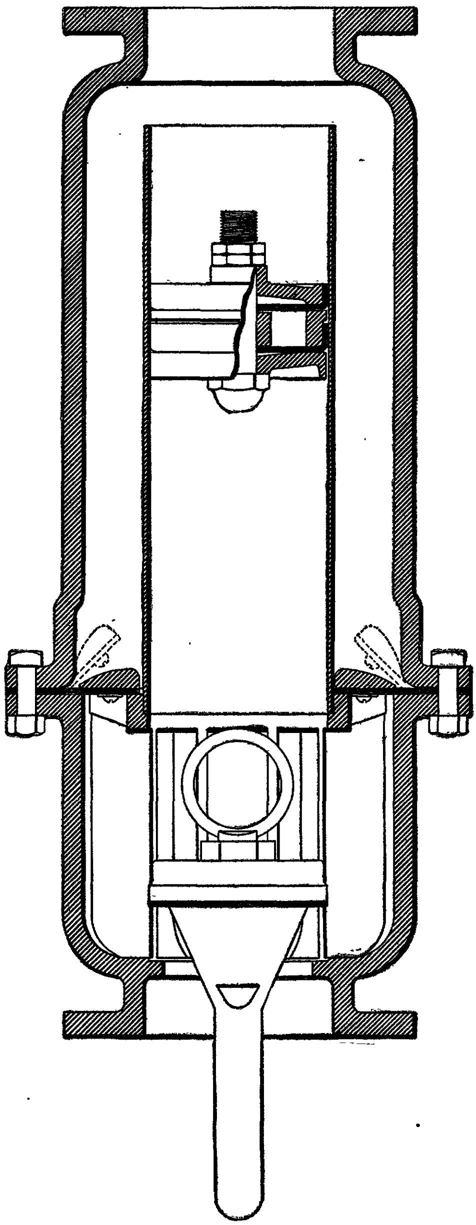


FIG. 23.—Vertical section of piston pump with valve chamber surrounding cylinder.

The suction valve is a heavy disk valve, playing in a special suction chamber, guided at its circumference, and not limited in its rise. It is arranged to withdraw through the cylinder if necessary. On the upstroke water is drawn into the cylinder through the suction valve;

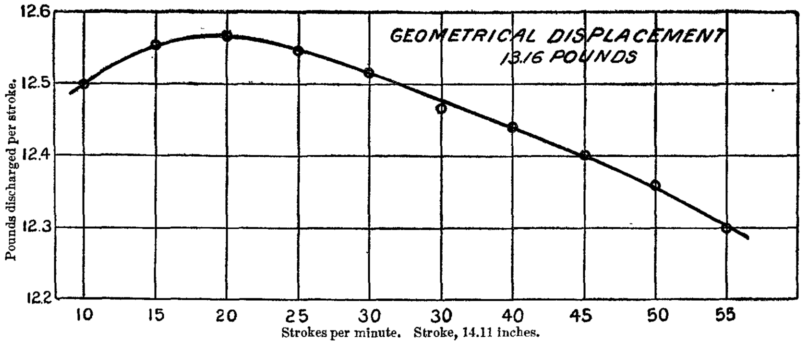


FIG. 24.—Discharge curve of 6-inch Frizell pump.

on the downstroke this water is forced through the delivery valves into the space above the piston, and is lifted on the next upstroke.

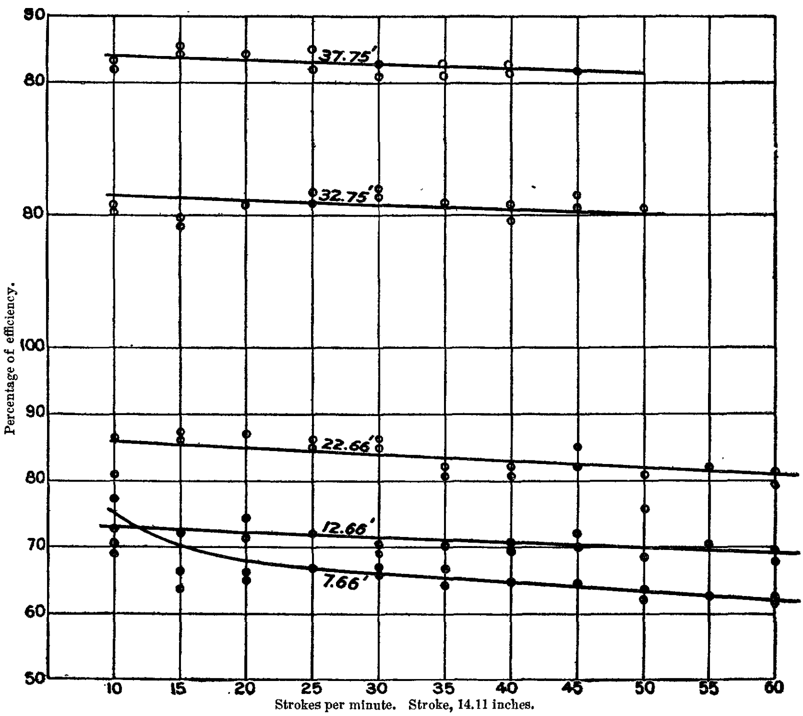


FIG. 25.—Varying-efficiency curves of 6-inch Frizell pump, showing effect of varying height of lift and speed.

The suction-valve area is $59\frac{1}{2}$ per cent of the cylinder area; the cylinder diameter, 5.74 inches; the upper valve seat area, 29.7 inches, or

3.8 inches more than the cylinder area; the weight of the pump, 270 pounds, and the weight of piston, rod, etc., 62 pounds. This pump was designed especially for irrigation purposes, and is known as the "Frizell" pump. It is remarkable for its sustained efficiency at high speeds. So far as tested the speed was limited only by the ability of connections to stand the work. Fig. 24 gives the discharge per stroke, which reaches a maximum of 95½ per cent of the geometrical displacement. Fig. 25 shows efficiency curves for five lifts. The small dif-

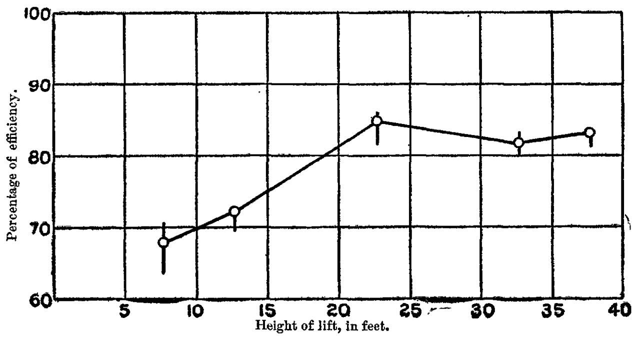


FIG. 26.—Variation of efficiency with varying height of lift of 6-inch Frizell pump at speed of 20 strokes per minute.

ference of efficiency at different speeds is due to the employment of large valve areas and small quick-closing valves. Fig. 26 shows that efficiency varies with the lift, being at its best probably at about 25 feet. The tables present the details of the tests.

Discharge of 6-inch Frizell pump at various lifts and speeds.

Speed.	Discharge in 30 strokes, lift 7.66 feet.	Discharge in 30 strokes, lift 12.66 feet.	Discharge in 30 strokes, lift 32.75 feet.	Plotted averages per stroke.
10	378	367	377	12.50
10	377		378	
15	377	371	380	12.56
15	379	371	383½	
20	377	373½	381½	12.57
20	378½	372½		
25	382	370	380	12.55
25		370	382	
30	375	366½	381½	12.52
30	379	369	376	
35	374	371	377½	12.47
35	371	365	374½	
40	379	370	379	12.44
40	375½	365½	374½	
45	374	365½	376	12.40
45	373	367	373	
50	377½–378	372	370½	12.36
50	380 –379	364	369	
50	371½–374			
55	363	364		12.30
55	371 –370	359		
60	372	361		12.22
60	371 –371	357		

Figures from efficiency tests of 6-inch Frizell pump; stroke, 19.. inches.

LIFT, 7.66 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	12.50	95.75	106.37	124.11	77.1		
10	12.50	95.75	111.70	131.36	72.8		
15	12.56	96.21	124.45	146.35	66.4		
15	12.56	96.21	127.65	150.12	64.1		
20	12.57	96.28	124.45	146.35	65.7		
20	12.57	96.28	123.40	145.12	66.3		
25	12.55	96.13	122.35	143.88	66.8		
25	12.55	96.13	122.88	144.50	66.5		
30	12.52	95.90	123.40	145.12	66		
30	12.52	95.90	121.30	142.65	67.2		
35	12.47	95.52	121.30	142.65	66.9		
35	12.47	95.52	125.53	147.61	64.7		
40	12.44	95.29	125.53	147.61	64.5		
40	12.44	95.29	125.53	147.61	64.5		
45	12.40	94.98	125.53	147.61	64.3		
45	12.40	94.98	124.45	146.35	64.9		
50	12.36	94.68	126.56	148.84	63.6	800	—30
50	12.36	94.68	128.16	150.72	62.8		
55	12.30	94.22	128.16	150.72	62.5		
55	12.30	94.22	128.16	150.72	62.5		
60	12.22	93.60	128.16	150.72	62.1		
60	12.22	93.60	129.26	152	61.5		

LIFT, 12.66 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
10	12.50	158.25	191.45	225.15	70.3	300	65
10	12.50	158.25	192.80	226.70	69.8		
15	12.56	159	186.95	220.25	72.1	310	40
15	12.56	159	187.75	220.79	72		
20	12.57	159.14	181.40	213.33	74.6		
20	12.57	159.14	189.35	223.67	71.4		
25	12.55	158.98	187.20	220.15	72.2		
25	12.55	158.98	188.55	221.74	71.7		
30	12.52	158.50	192.80	226.73	69.9		
30	12.52	158.50	191.45	225.15	70.3		
35	12.47	157.86	191.45	225.15	70.1		
35	12.47	157.86	191.45	225.15	70.1		
40	12.44	157.49	189.35	222.67	70.7		
40	12.44	157.49	191.45	225.15	69.9	750	—75
45	12.40	156.98	190.65	224.21	70	750	—80
45	12.40	156.98	185.10	217.68	72.1		
50	12.36	156.58	187.70	219.15	75.9		
50	12.36	156.58	194.44	228.66	68.4		
55	12.30	155.72	188.55	221.74	70.2	850	—70
55	12.30	155.72	188.55	221.74	70.2		
60	12.22	154.70	192.80	226.73	68.2		
60	12.22	154.70	180.35	222.67	69.4		

LIFT, 22.66 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
10	12.50	283.25	277.50	326.34	86.7	410	60
10	12.50	283.25			80.6		
15	12.56	284.61	279.65	328.87	86.4		
15	12.56	284.61	277.50	326.34	87.2		
20	12.57	284.84	277.50	326.34	87.3		
20	12.57	284.84	277.50	326.34	87.3		
25	12.55	284.38	282.70	332.35	85.6		
25	12.55	284.38	280.60	329.98	86.2		
30	12.52	283.70	282.70	332.35	85.3	500	—85
30	12.52	283.70	284.80	334.92	84.4		
35	12.47	282.56	291.10	342.33	82.6		
35	12.47	282.56	295.35	347.33	81.3		
40	12.44	281.89	295.35	347.33	81	770	—85
40	12.44	281.89	291.10	342.33	82.3		
45	12.40	280.98	291.10	342.33	81.8		
45	12.40	280.98	284.80	332.12	84.6	930	—50
50	12.36	280.08	294.40	346.12	80.9	925	—100
50	12.36	280.08	295.35	347.33	80.6		
55	12.30	278.72	292.10	343.39	81.1		
55	12.30	278.72	291.10	342.33	81.4	1,050	—100
60	12.22	276.90	290.25	341.34	81.1		
60	12.22	276.90	296.20	348.33	79.4		

Figures from efficiency tests of 6-inch Frizell pump; stroke, 19.1 inches—Continued.

LIFT, 32.75 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	12.50	409.38	435.38	512	81.3		
10	12.50	409.38	431.25	507.15	80.7		
15	12.56	411.34	441.60	519.32	79.2		
15	12.56	411.34	444.90	523.20	78.6		
20	12.57	422.06	440.40	517.91	81.5		
20	12.57	422.06	441.60	591.32	81.7		
25	12.55	421.41	429	504.50	83.5		
25	12.55	421.41	435.38	512	82.1		
30	12.52	420.43	425.85	500.90	83.9	720	—30
30	12.52	420.43	431.25	507.15	82.9		
35	12.47	408.39	424.42	499.12	81.8		
35	12.47	408.39	424.42	499.12	81.8		
40	12.44	407.41	430.80	506.62	81.3		
40	12.44	407.41	435.03	510.42	79.8	1,050	—97.5
45	12.40	406.10	425.20	500.04	81.1	1,027.5	—90
45	12.40	406.10	419.47	493.29	82.9		
50	12.36	404.79	425.20	500.04	80.9		
50	12.36	404.79	425.20	500.04	80.9	1,260	—60

LIFT, 37.75 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
10	12.50	471.87	484.25	573.97	82.2		
10	12.50	471.87	484.87	566.68	83.2		
15	12.56	474.14	478.72	562.97	85.3		
15	12.56	474.14	475.50	559.19	84.7		
20	12.57	474.52	476.62	560.40	84.6		
20	12.57	474.52	479.77	564.21	84.1		
25	12.55	473.76	493.52	579.20	81.7	750	0
25	12.55	473.76	486.22	561.79	84.8		
30	12.52	472.63	483	568	83.2		
30	12.52	472.63	495.75	583	81		
35	12.47	470.74	492.52	579.20	81.2		
35	12.47	470.74	483	568	82.8	1,020	—67.5
40	12.44	469.61	489.38	575.51	81.6		
40	12.44	469.61	486.22	571.79	82.3	1,005	—67.5
45	12.40	468.10	486.22	571.79	81.9		

In the smaller size of this same make, fig. 27, a single butterfly valve in the piston replaces the several small clack valves. The cylinder is

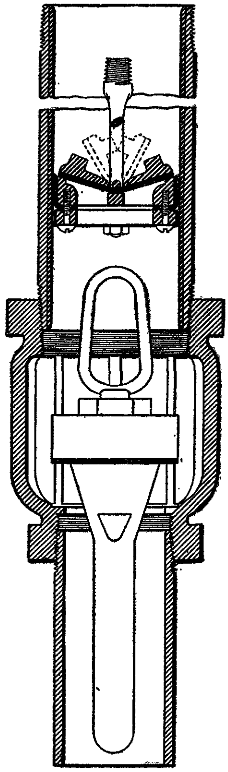


FIG. 27.—Vertical section of Frizell cylinder with butterfly valve.

brass-lined, is 3.73 inches in diameter, and has a 24-inch stroke. The valve area in the piston is 27.4 per cent of the cylinder area; the suc-

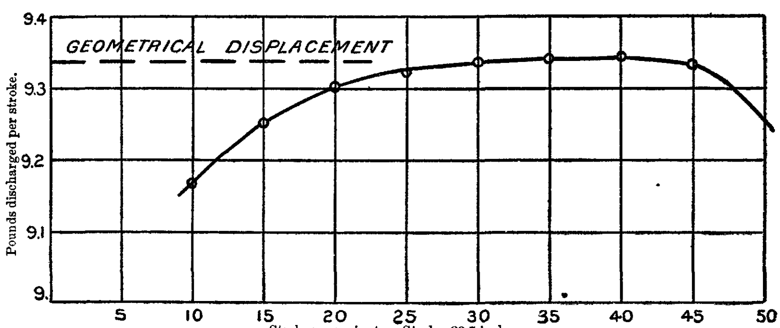


FIG. 28.—Discharge curve of 4-inch Frizell pump.

tion valve, 56 per cent; the weight of the pump, $57\frac{1}{2}$ pounds, and the weight of piston, rod, etc., $39\frac{1}{2}$ pounds. Fig. 28 shows a discharge attaining to 100 per cent at speeds ranging from 25 to 40 strokes per minute. Fig. 29 shows the efficiency at six different lifts. The sustained efficiency of this pump is noticeable. In fig. 30 increase of efficiency with increase of lift appears. This pump is much used in Kansas for irrigation.

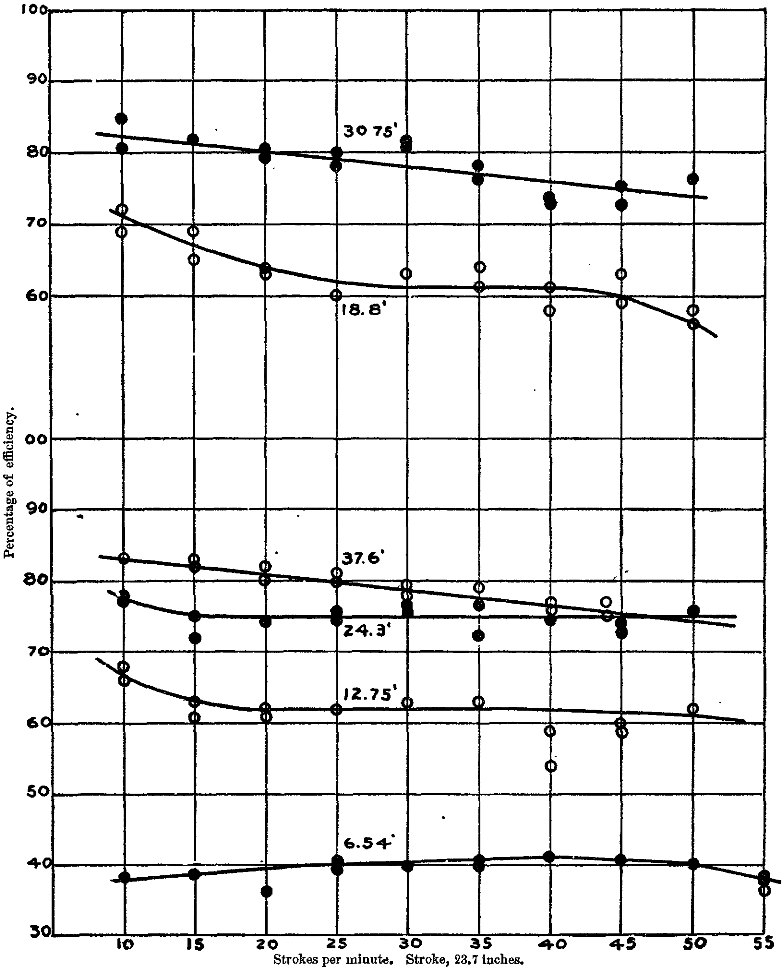


FIG. 29.—Varying-efficiency curves of 4-inch Frizell pump, showing effect of varying height of lift and speed.

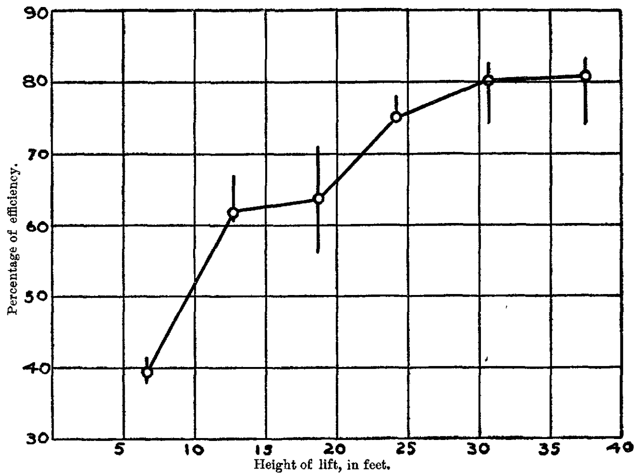


FIG. 30.—Variation of efficiency with varying height of lift of 4-inch Frizell pump at speed of 20 strokes per minute.

Discharges of 4-inch Frizell pump at various lifts and speeds; stroke, 23.7 inches.

Speed.	Discharge for 50 strokes; lift, 24.3 feet.	Discharge for 50 strokes; lift, 24.8 feet.	Discharge for 50 strokes; lift, 37.8 feet.	Discharge for 30 strokes; lift, 12.75 feet.	Discharge for 30 strokes; lift, 6.54 feet.	Plotted average discharge per stroke.
	Pounds.	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	
10		451.5	462	275.5	274	9.168
10		457.5	465	275.5	274	
15	460	463.5	466	275.5	277	9.25
15		464	468	275.5	278.5	
20	465	464.5	467	278	281	9.302
20		464	468	278	281.5	
25	467	463.5	467	278.5	280	9.322
25		463.5	467.5	278.5	280	
30	466	469	467	279.5	279	9.334
30		465	467	279.5	280	
35	472	468	468	279.5	281	9.34
35		463.5	466.5	281	280.5	
40	467	466.5	466	281	284	9.42
40		466.5	467	280.5	280	
45	467	462.5	467.5	280.5	280	9.332
45		465	467.5	280	282	
50		449.5	443	277.5	279	9.254
50				279	279	
55				171	253	
55					267	

Figures from efficiency tests of 4-inch Frizell pump; stroke, 23.7 inches.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
LIFT, 6.54 FEET.							
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	9.168	59.96	79.38	156.77	38.2	120	33
10	9.168	59.96	79.99	158.01	37.9		
15	9.2592	60.56	79.15	156.33	38.7		
15	9.2592	60.56	78.96	155.94	38.8		
20	9.2724	60.64	84.31	161.51	36.4	135	9
20	9.2724	60.64	83.75	165.41	36.6		
25	9.31	60.89	77.91	153.81	39.6		
25	9.31	60.89	76.04	150.18	40.5		
30	9.327	61.00	76.66	151.40	40.2	177	9
30	9.327	61.00	77.91	153.87	39.9		
35	9.34	61.08	75.82	149.74	40.7		
35	9.34	61.08	77.08	152.23	40.0	195	6
40	9.35	61.18	75.00	148.11	41.2	201	— 15
40	9.35	61.18	75.00	148.11	41.2		
45	9.33	61.04	76.23	150.55	40.5	253.5	— 30
45	9.33	61.04	76.65	151.38	40.3		
50	9.25	60.53	75.82	149.74	40.4	249	— 24
50	9.25	60.53	78.75	155.50	39.9		
55	8.66	56.68	75.82	149.44	37.8	222	— 30
55	8.66	56.68	75.41	148.90	38.0		
LIFT, 12.75 FEET.							
10	9.168	116.89	89.63	177.00	66		
10	9.168	116.89	87.07	171.97	68		
15	9.260	118.05	94.90	187.43	63		
15	9.260	118.05	98.20	193.11	61		
20	9.270	118.22	97.17	191.91	61		
20	9.270	118.22	95.77	189.24	62		
25	9.310	118.70	97.22	192.00	62	175	30
25	9.310	118.70	96.60	190.78	62		
30	9.327	118.91	94.63	186.88	63		
30	9.327	118.91	97.80	184.82	63		
35	9.340	119.08	95.22	188.09	63		
35	9.340	119.08	96.45	190.78	63	250	11.5
40	9.350	119.17	109.85	216.95	54		
40	9.350	119.17	100.70	198.88	59		
45	9.330	118.99	99.57	196.66	60	325	— 11.5
45	9.330	118.99	101.40	200.26	59		
50	9.250	117.99	95.77	189.24	62	350	— 95
50	9.250	117.99	96.90	191.54	62		
55	5.700	72.67	99.30	196.11	37	375	— 75
LIFT, 18.8 FEET.							
10	9.168	172.36	126.75	250.33	69	180	36
10	9.168	172.36	120.33	237.65	72		
15	9.260	174.07	135.21	267.04	65		
15	9.260	174.07	133.50	263.66	69		
20	9.270	174.32	139.05	274.62	63		
20	9.270	174.32	136.89	270.35	64		
25	9.310	175.03	148.32	292.93	60	240	24
25	9.310	175.03	142.83	282.09	62		
30	9.327	175.35	139.47	275.45	63		
30	9.327	175.35	139.47	275.45	63		
35	9.340	175.59	141.90	280.25	64	288	3
35	9.340	175.59	145.32	287.00	61		
40	9.350	175.86	146.64	289.61	61	336	6
40	9.350	175.86	151.68	299.56	58		
45	9.330	175.45	150.42	297.08	59	360	— 9
45	9.330	175.45	138.57	278.67	63		
50	9.250	173.98	154.62	305.37	56	405	— 30
50	9.250	173.98	151.26	299.49	58		
55	5.700	107.16	150.42	297.08	36	450	—120
55	5.700	107.16	156.32	308.75	34		
58	5.430	102.14	159.72	315.36	32	450	—120
58	5.430	102.14	156.33	308.75	33		

Figures from efficiency tests of 4-inch Frizell pump; stroke, 23.7 inches—Continued.

LIFT, 24.8 FEET.

Speed.	Discharge per stroke	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	9.168	227.36	149.28	294.82	77.1		
10	9.168	227.36	147.48	291.27	78.0		
15	9.260	229.65	161.28	377.69	72.3		
15	9.260	229.65	155.01	306.06	75.3		
20	9.270	229.89	155.82	307.74	74.7		
20	9.270	229.89	156.66	309.40	74.3		
25	9.310	230.89	156.66	309.40	74.6		
25	9.310	230.89	154.14	304.34	75.8		
30	9.327	231.11	154.14	304.32	75.8	348	— 36
30	9.327	231.11	152.46	301.10	76.7		
35	9.340	231.63	154.14	304.42	76.8		
35	9.340	231.63	162.48	319.23	72.5		
40	9.350	231.88	157.59	311.24	74.4	398	— 24
40	9.350	231.88	158.73	312.65	74.1		
45	9.330	231.38	161.10	319.00	72.5	471	— 36
45	9.330	231.38	158.73	312.65	74.0		
50	9.250	229.40	153.99	304.13	75.4		

LIFT, 30.75 FEET.

Speed.	Discharge per stroke	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
10+	9.168	280.89	176.45	348.50	80.5	250	15
10+	9.168	280.89	175.05	345.70	81.2		
10	9.168	280.89	176.00	347.80	80.7	240	40
10	9.168	280.89	167.60	331.00	84.8		
15	9.26	284.72	176.55	348.70	81.6		
15	9.26	284.72	176.55	348.70	81.6		
20	9.27	285.12	179.25	354.00	80.5	280	40
20	9.27	285.12	181.15	357.80	79.3		
25	9.31	286.29	180.65	356.80	80.2		
25	9.31	286.29	184.85	365.10	78.1		
30	9.327	286.81	180.15	355.80	80.6	370	0
30	9.327	286.81	177.35	350.30	81.8		
35	9.34	287.20	190.15	375.60	76.4	425	0
35	9.34	287.20	185.90	367.20	78.2		
40	9.35	287.65	197.45	390.00	73.7	490	— 25
40	9.35	287.65	198.60	392.20	73.3		
45	9.33	286.98	199.70	394.40	72.7	540	— 5
45	9.33	286.98	193.25	381.70	75.1		
50	9.25	284.58	188.70	372.70	76.3	590	—125

LIFT, 37.6 FEET.

Speed.	Discharge per stroke	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
10	9.168	344.61	208.32	411.35	83		
10	9.168	344.61	208.32	411.35	83		
15	9.26	348.14	213.30	421.26	82	275	50
15	9.26	348.14	211.56	417.84	83		
20	9.27	348.64	214.98	424.58	82		
20	9.27	348.64	220.74	435.93	80		
25	9.31	350.06	220.74	435.93	80	360	36
25	9.31	350.06	218.94	432.40	81		
30	9.327	350.70	224.16	442.70	79.4		
30	9.327	350.70	225.18	444.73	78		
35	9.34	351.18	224.34	443.07	79	410	0
35	9.34	351.18	231.00	456.22	77		
40	9.35	351.73	231.66	457.36	76		
48	9.33	350.91	229.56	453.38	77		
48	9.33	350.91	234.30	462.89	75		

This pump was arranged as a force pump, as shown in fig. 1 (p. 12), and tested at three different lifts, as shown in fig. 31. At a lift of about 24 feet the efficiency is seen to be reduced from about 75 per cent to about 52 per cent upon the addition of a stuffing box and two turns in the discharge pipe. This drop in the efficiency exhibits the harmful effect of added complications. The gland was carefully made, and as lightly packed as possible, to prevent leakage. Adding this reduction of about 20 per cent to the efficiencies found for the two high lifts of 47 and 70 feet, respectively, the efficiency of the pump

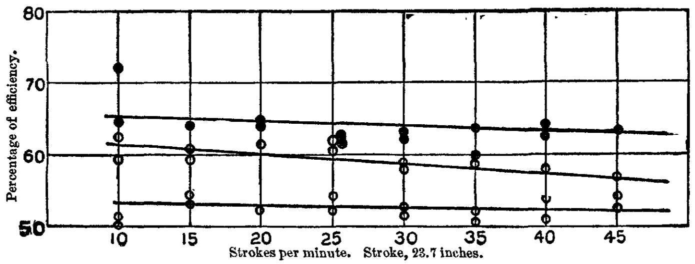


FIG. 31.—Efficiency of 4-inch pump used as a force pump.

when used as a simple lift pump, as in the first trials, is brought up to the neighborhood of 80 per cent.

Figures from efficiency tests of 4-inch Frizell pump, used as a force pump with a stuffing box; stroke, 23.7 inches.

LIFT, 24.3 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	9.168	222.48	221.08	436.63	51.2	280	24
10	9.168	222.48	225.52	445.40	50		
15	9.260	225.02	223.32	441.06	53.2		
15	9.260	225.02	218.88	432.28	54.4		
20	9.270	225.26	217.76	430.07	52.3		
20	9.270	225.26	218.88	432.28	52.1		
25	9.310	226.32	211.08	416.88	54.2	308	—32
25	9.310	226.32	217.76	430.07	52.6		
30	9.327	226.60	217.56	429.56	52	376	—32
30	9.327	226.60	216.60	427.78	52.9		
35	9.340	227.12	221.08	436.63	52.1		
35	9.340	227.12	220.96	448.56	50.6		
40	9.350	229.55	217.68	449.08	51.1		
40	9.350	229.55	215.44	425.47	53.9		
45	9.330	224.69	215.44	425.47	52.8	520	—88
45	9.330	224.69	213.44	431.82	52.3		

Frizell 4-inch pump; stroke, 23.7 inches; used as a force pump and delivering water against air pressure equivalent to the lift noted.

LIFT, 47.37 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	9.168	434.29	348.60	695.78	62.4		
10	9.168	434.29	369.60	729.96	59.4		
15	9.26	438.60	376.20	734.66	59.7		
15	9.26	438.60	360.30	711.59	61.6		
20	9.27	439.21	361.32	713.61	61.5		
20	9.27	439.21	360	711	61.7	444	— 18
25	9.31	441.01	366.66	724.16	60.9		
25	9.31	441.01	360	711	62		
30	9.327	441.82	378.30	747.14	59.1		
30	9.327	441.82	383.58	757.48	58.3		
35	9.34	442.44	380.33	751.15	58.9	636	—108
35	9.34	442.44	380	750.50	58.9		
40	9.35	442.90	385.44	761.23	58.1		
40	9.35	442.90	385.32	761	58.2		
45	9.33	441.96	402.24	794.34	56.9	756	— 90
45	9.33	441.96	413.32	816.30	54.1		

Frizell 4-inch pump; stroke, 23.7 inches; used as a force pump and delivering water against air pressure equivalent to the lift noted—Continued.

LIFT, 70.5 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.	Minimum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*
10	9.168	646.34	495.75	895.77	72.1		
10	9.168	646.34	510	1,007	64.5		
15	9.26	652.83	515.33	1,101.11	59.2		
15	9.26	652.83	514.50	1,016.22	64.2		
20	9.27	653.54	508.35	1,003.90	65		
20	9.27	653.54	513.22	1,013.62	64.4		
25	9.31	656.36	536.25	1,059.09	61.9		
25	9.31	656.36	534.15	1,054.96	62.2		
30	9.327	657.55	532.13	1,050.95	62.5		
30	9.327	657.55	527.85	1,042.50	63		
35	9.34	658.47	521.53	1,030.02	63.8	780	— 75
35	9.34	658.47	515.33	1,101.11	59.8		
40	9.35	659.18	532.13	1,050.96	62.7		
40	9.35	659.18	516.66	1,020.32	64.6	825	—165
45	9.33	657.77	525.60	1,038.06	63.3		
45	9.33	657.77	525.60	1,038.06	63.3		

VAN VOORHIS PUMP.

Fig. 32 is a pump specially designed for slow speeds and low lifts, yet capable of running at high speeds. It is locally known in Kan-

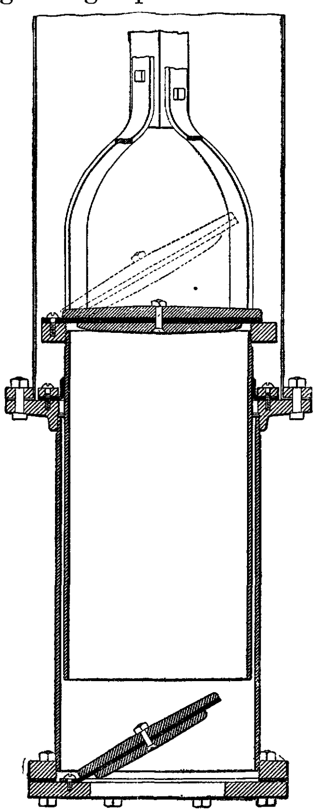


FIG. 32.—Vertical section of pump designed for slow speeds and low lifts.

sas as the Van Voorhis pump. The piston is replaced by a hollow plunger the upper opening of which is covered by a large clack valve. The plunger is packed with a suitable cup leather on the outside.

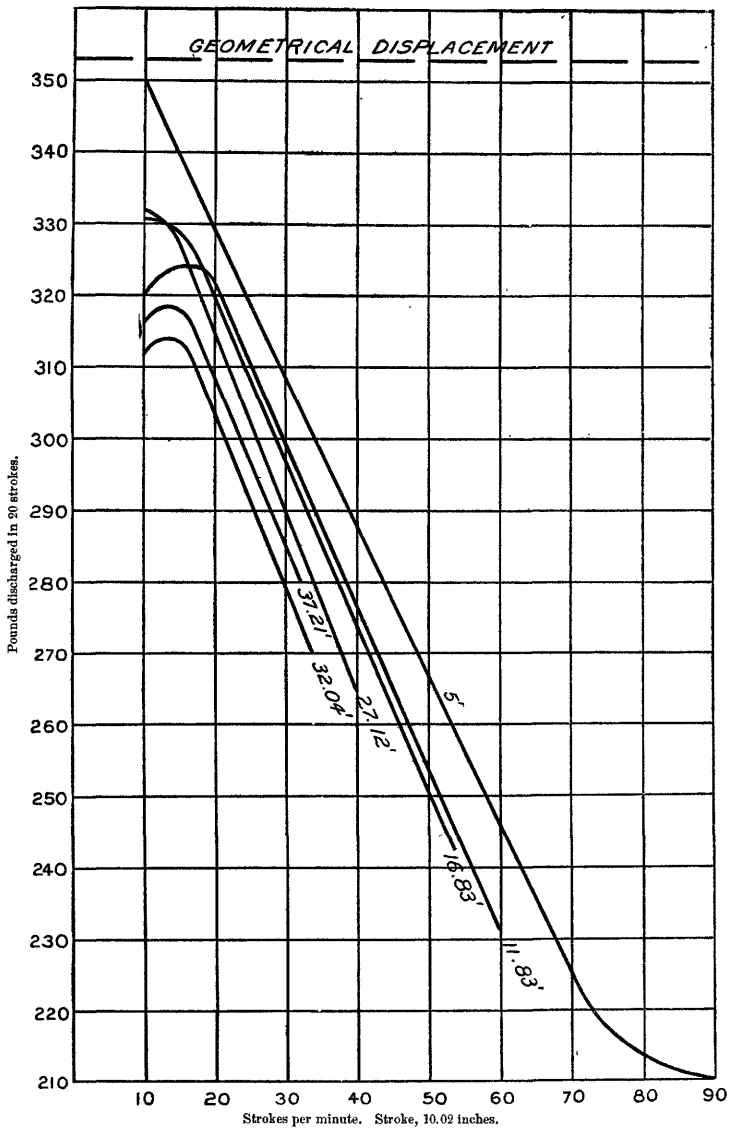


FIG. 33.—Varying-discharge curves of 8-inch Van Voorhis pump, showing effect of varying lift and speed.

The suction valve also is a large clack valve. The valve area is very great, being 70 per cent for the lower valve and 94 per cent for the upper valve in an 8-inch pump. The weight of the pump was 146 pounds, and of the plunger, rod, etc., 101.5 pounds. This design has many advantages for very slow running, which, however, quickly disappear when a moderate piston speed is reached. The discharge

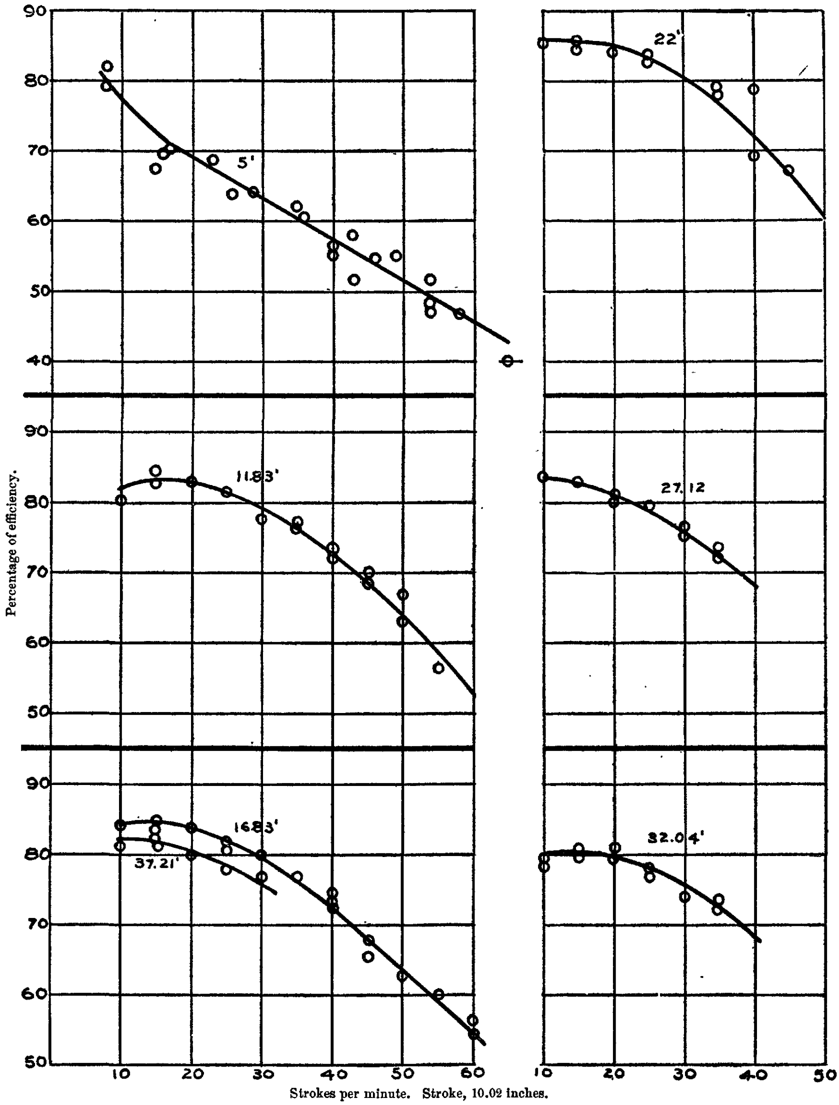


FIG. 34.—Varying-efficiency curves of 8-inch Van Voorhis pump, showing effect of varying lift and speed.

curves, shown in fig. 33, and varying for each lift, give the results for the large valves. Fig. 34 shows the efficiencies for six different lifts, and fig. 35 the change of efficiency with increase of lift.

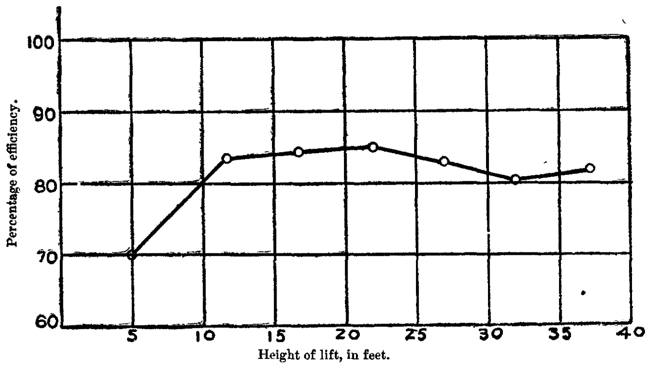


FIG. 35.—Variation of efficiency with varying height of lift of Van Voorhis pump at speed of 15 strokes per minute.

Discharge of Van Voorhis 8-inch pump at 5-foot lift; stroke, 10.02 inches.

Speed.	Discharge in 20 strokes.	Discharge in 20 strokes.	Plotted average; discharge per stroke.	Speed.	Discharge in 20 strokes.	Discharge in 20 strokes.	Plotted average; discharge per stroke
	Pounds.	*Pounds.*	*Pounds.*		*Pounds.*	*Pounds.*	*Pounds.*
10	348.5	348.5	17.55	50	262	259.5	13.35
15	344.5	346	17.05	55	249	253	12.80
20	332	328	16.50	60	244	241.5	12.35
25	315	315.5	15.97	65	227	235	11.60
30	301	296	15.42	68	220	238	11.35
35	296	294	14.85	70	222	219.5	11.05
40	289	290	14.30	75	221.5	220	10.90
45	281	291	13.77	80	213	215	10.67
47	373		13.55	85	215.5	208.5	10.55

Discharge of Van Voorhis 8-inch pump at various lifts and speeds.

Speed.	Discharge, in pounds, for 20 strokes for lift:				
	11.83 feet.	16.83 feet.	27.12 feet.	32.04 feet.	37.21 feet.
10	320.5	328	331	306	316.5
10	318.5	332.5	332.5	308	315.5
15	323	329.5	329.5	313	319
15	325.5	328	326	317.5	318
20	326.5	320.5	317.5	298	309
20	322.5	321	315	302.5	309
25	309.5	306.5	300	293	299
25	308.5	310.5	300.5	290.5	296
30	294	297	288.5	290	280.5
30	301.5	301	286	282.5	268.5
35	290	284.5	285		
35	290.5	286	284		
40	275.5	256			
40	270.5	275			
45	272.5	243			
45	267.5	251			
50	257.5				
50	249.5				
55	247.5				
55	239				
60	232.5				
60	223.5				

Plotted average discharge per stroke.

Speed.	Discharge, in pounds, per stroke for lift: 11.83 feet.	16.83 feet.	22 feet.	27.12 feet.	32.04 feet.	37.21 feet.
10	15.975	16.525	16.562	16.6	15.55	15.8
15	16.1	16.45	16.412	16.375	15.7	15.925
20	16.05	16	15.887	15.775	15.2	15.45
25	15.575	15.425	15.275	15.125	14.575	14.85
30	15	14.85	14.675	14.5	14	14.25
35	14.425	14.3	14.1	13.9	13.35	
40	13.85	13.725	13.487	13.25		
45	13.275	13.175	12.912	12.65		
50	12.7	12.6	12.3	12		
55	12.125	12.05				
60	11.55	11.5				

Figures from efficiency tests of 8-inch Van Voorhis pump; stroke, 10.02 inches.

LIFT, 4.875 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds work given to rod	Mechanical efficiency.	Maximum tension on the rod.	Minimum tension on the rod.	Stroke finished at closure of the upper valve.	Stroke finished at closure of the lower valve.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*	*Pounds.*	*Per cent.*	*Per cent.*
8	17.65	86.04	130	108.55	79.20	235	+ 85		
8	17.65	86.04	125.50	104.79	82.10				
35	14.85	72.39	140.95	117.68	62			4	12
40	14.30	69.71	147.65	123.28	56	515	+ 20	5	16
40	14.30	69.71	151	126	55.30			5	16
54	12.80	62.40	158.30	132.18	47.20	780	— 50	11	21
54	12.80	62.40	156.65	131.63	47.40			11	21
54	12.80	62.40	155.30	129.67	48.20			11	21
54	12.80	62.40	157.70	131.69	47.30	675	— 35	11	21

LIFT, 5.166 FEET.

15	17.05	88.09	154.30	128.84	67.60				2
16	16.95	87.57	150.80	125.90	69.55				3
17	16.85	86.96	148.65	124.12	70				5
20	16.50	85.25	150	125.25	68	285	40		6
23	16.20	83.70	145.70	121.66	68.80	280	50.	.6	6
26	15.85	81.89	153.30	128	63.97			.9	9
29	15.50	80.08	150	125.25	63.92			2	11
36	14.75	76.20	150	125.25	60.80	490	0	4	15
36	14.75	76.20	151.30	126.33	60.30			4	15
43	14	72.33	167.75	140.07	51.60	550	0	6	16
43+	13.90	71.81	147.65	123.28	58.20			6	17
46	13.65	70.52	154.30	128.67	54.80	575	15	7	19
43+	13.90	71.81	147.65	123.28	58.20			5	17
49	13.35	68.97	150	125.25	55			5	20
54	12.80	66.13	153.30	128	51.60	650	0	10	21
54	12.80	66.13	153.30	128	51.60			10	22
58	12.35	63.81	164.04	136.89	46.60	876	— 72	12	22
65	11.60	59.93	179.46	149.84	40	600	0	12	
72	11.05	55.25	194.32	162.25	34	1,200	— 75	17	27
84	10.55	54.51	285.30	238.22	22.80			20	
85	10.50	54.24	226.16	188.64	27.80	1,500	—150	21	37
90	10.40	53.73	264.12	220.37	24.70	1,200	—160	20	
100	10	50	283.72	236.90	21.10	1,560	—135	22	

Figures from efficiency tests of 8-inch Van Voorhis pump; stroke, 10.02 inches—Continued.

LIFT, 11.83 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
10	15.975	188.9	281.1	235.8	80.5	417
10	15.975	188.9	281.1	235.8	80.5	422
15	16.1	190.4	276	230.2	82.7	427
15	16.1	190.4	269.9	225.4	84.5	437
20	16.05	189.8	274	228.8	82.9	475
20	16.05	189.8	273.7	228.5	83	500
25	15.575	184.3	272.5	226.9	81.2	525
30	15	177.4	272.4	227.4	78	660
30	15	177.4	266.2	222.3	79.8	720
35	14.425	170.6	264.6	221	77.2	750
35	14.425	170.6	266.9	222.9	76.5	780
40	13.85	163.8	272.2	227.3	72	1,060
40	13.85	163.8	266.9	222.8	73.5	895
45	13.275	157	274.5	229.2	68.5	1,070
45	13.275	157	268.4	224.1	70	1,015
50	12.7	150.2	269.1	224.7	66.8	1,135
50	12.7	150.2	285.4	238.2	63	1,265
55	12.125	143.4	303.9	253.7	56.5	1,410
55	12.125	143.4	303.9	253.7	56.5	1,380
60	11.55	136.6	308.5	257.6	53	1,357
60	11.55	136.6	303.2	259.5	52.6	1,432

LIFT, 16.83 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.
10	16.525	278.1	395.8	330.5	84.1	565
10	16.525	278.1	396.6	331.2	83.9	565
15	16.45	276.8	394.7	329.6	84.1	595
15	16.45	276.8	391.2	326.6	84.7	590
20	16	269.3	386.6	322.8	83.4	650
20	16	269.3	387.3	323.4	83.2	650
25	15.425	259.6	383.8	320.5	81	710
25	15.425	259.6	380	317.3	81.8	710
30	14.85	249.9	375.3	313.4	79.7	875
30	14.85	249.9	374.6	312.8	79.9	------
35	14.3	240.6	374.2	312.5	77	940
35	14.3	240.6	373.5	311.9	77.1	1,000
40	13.725	231	370.4	309.3	74.7	1,030
40	13.725	231	375.3	313.4	73.7	1,050
40	13.725	231	372.3	310.9	74.3	1,070
40	13.725	231	374.2	317.8	72.7	1,010
45	13.175	221.7	391	326.5	67.9	1,224
45	13.175	221.7	405.9	338.9	65.4	1,309
50	12.6	212	404.8	338	62.7	1,245
50	12.6	212	405.9	338.9	62.5	1,395
55	12.05	202.8	404.7	337.9	60	1,372
55	12.05	202.8	404.7	337.9	60	1,330
60	11.5	193.5	426.4	356.1	54.3	1,515
60	11.5	193.5	410.4	342.7	56.4	1,447

LIFT, 22 FEET.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.
10	16.562	364.3	510.8	426.5	85.4	720
10	16.562	364.3	510.1	425.9	85.5	715
15	16.412	361	504.7	421.4	85.6	770
15	16.412	361	510.8	426.5	84.6	740
20	15.887	349.5	497.7	415.6	84.1	855
20	15.887	349.5	492.2	411	85	890
25	15.275	336	486.8	406.5	82.6	945
25	15.275	336	480.7	401.4	83.7	------
30	14.675	322.8	483.9	404	79.9	1,035
30	14.675	322.8	482.2	402.7	80.1	1,140
35	14.1	310.2	474.6	396.3	78.2	1,215
35	14.1	310.2	468.4	391.2	79.3	1,160
40	13.487	296.7	501.5	418.8	70.8	1,380
40	13.487	296.7	513.8	429	69.1	1,507
45	12.912	284	505.3	421.9	67.3	1,402
45	12.912	284	505.3	421.9	67.3	1,402
50	12.3	270.6	531.6	443.9	60.9	1,687
50	12.3	270.6	533.2	445.3	60.7	1,650

Figures from efficiency tests of 8-inch Van Voorhis pump; stroke 10.02 inches—Continued.

LIFT, 27.12 FEET.

Speed.	Discharge per stroke.	Foot pounds of useful work.	Average pull on rod.	Foot pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
10	16.6	450.2	646.3	539.7	83.4	832
10	16.6	450.2	646.5	539.8	83.4	900
15	16.375	444.1	642.5	536.5	82.7	969
15	16.375	444.1	640.9	535.1	83	975
20	15.775	427.8	634.7	530	80.7	1,027
20	15.775	427.8	638.2	532.9	80.2	1,035
25	15.125	410.2	617.3	515.4	79.5	1,207
25	15.125	410.2	619.9	517.7	79.2	1,170
30	14.5	393.2	619.9	517.7	75.96	1,395
30	14.5	393.2	615.7	514.1	76.4	1,380
35	13.9	367	611.5	510.6	71.8	1,500
35	13.9	367	600	501	73.2	1,447

LIFT, 32.04 FEET.

Speed.	Discharge per stroke.	Foot pounds of useful work.	Average pull on rod.	Foot pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.
10	15.55	498.2	760.4	634.9	78.4	1,057
10	15.55	498.2	760.4	634.9	78.4	1,080
15	15.7	503	753.5	629.2	79.9	1,132
15	15.7	503	746.5	623.3	80.7	1,110
20	15.2	487	734.9	613.6	79.3	1,252
20	15.2	487	721	602	80.9	1,350
25	14.575	467	725.6	605.9	77	1,320
25	14.575	467	718.9	600.2	77.8	1,440
30	14	448.5	724.5	604.9	74.1	1,455
30	14	448.5	712.1	594.5	75.4	1,447
35	13.35	427.7	706.2	589.6	72.5	1,567
35	13.35	427.7	696.3	581.4	73.5	1,567

LIFT, 37.21 FEET.

Speed.	Discharge per stroke.	Foot pounds of useful work.	Average pull on rod.	Foot pounds of work given to rod.	Mechanical efficiency.	Maximum tension on rod.
10	15.8	587.9	868	724.8	81.1	1,177
10	15.8	587.9	865.7	722.8	81.3	1,177
15	15.925	592.5	867.2	724.1	81.8	1,185
15	15.925	592.5	864.5	721.8	82	1,185
20	15.45	574.9	857.3	715.9	80.3	1,230
20	15.45	574.9	859.1	717.3	80.1	1,260
25	14.85	552.5	845.5	704.5	78.2	1,305
25	14.85	552.5	847.8	707.9	78	1,380
30	14.25	530.2	825.7	689.5	76.9	1,380
30	14.25	530.2	823.1	687.3	77.1	1,380

COOK PUMP.

The Cook cylinder, made by the Cook Well Company, of St. Louis, Missouri, has been developed especially for deep-well pumping, and the 4-inch size is shown in fig. 36. The design admits of the withdrawal of both the piston and the suction-valve device for inspection or repair. The actual diameter is 3.75 inches. Though furnished with double packing leathers and having a valve area of 21 per cent of the cylinder area, its efficiency in deep wells is notable. Both valves are rubber disks, guided by a central stem. The cylinder is brass lined and highly polished, and friction is thus greatly reduced. However, on account of the high pressures with which this pump deals, double cup leathers for packing are required, by which, on the other hand, friction is increased. Built especially for high lifts, it is not to be expected that it will develop a high efficiency at the low lifts prevailing in irrigation work.

In testing this pump it was first operated in an open well with a 34.8-foot lift. The efficiency varied from about 78 to 58 per cent at

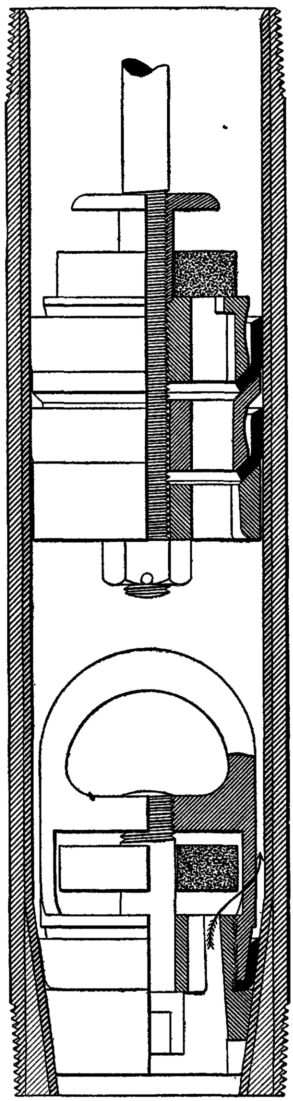


FIG. 36.—Vertical section of Cook cylinder.

speeds of 12 to 40 2-foot strokes per minute. The pump was then provided with a stuffing box, carefully made and packed, and was made to deliver water against an air pressure equivalent to the previous total lift of 34.8 feet.

The mean pull on the pump rod was found to be increased about 30 pounds by the friction in the stuffing box, and the efficiency, in

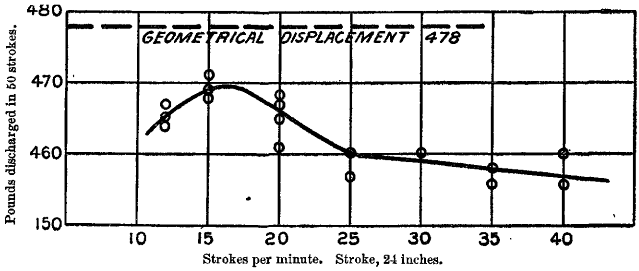


FIG. 37.—Discharge curve for 4-inch Cook deep-well pump.

consequence, lowered from the 78 to 58 per cent of the previous test to 67 to 53 per cent. At the higher lifts tried the same method was

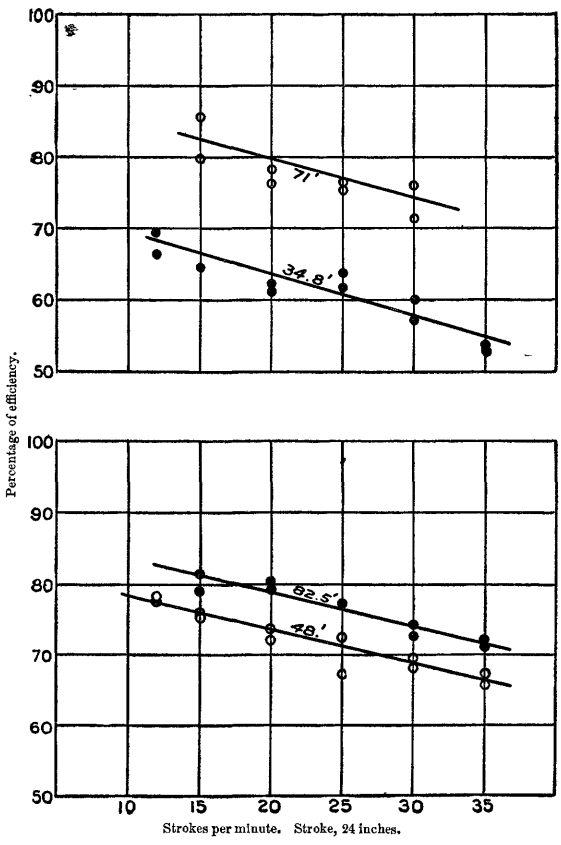


FIG. 38.—Varying-efficiency curves of 4-inch Cook pump, showing effect of varying lift and speed.

followed, and the efficiency, at an 82.5-foot lift and at speeds ranging from 15 to 35 strokes per minute, was found to range from 81 to 72 per cent. Allowing for the increase of friction resulting from the addition of a stuffing box, the efficiency of the pump in an open well and at the same depth would probably be 85 to 77 per cent. Fig. 37 gives the discharge curve, and fig. 38 the efficiency curves, at differing lifts, for this pump, fitted with the packed stuffing box and rod. There are exhibited a quick action of the valves, a remarkably low maximum stress, and slight compression in the pump rod at speeds above 20 strokes per minute.

There is here seen to be marked advantage in the special adaptation of parts to special conditions.

Discharge of 4-inch Cook pump; stroke, 24 inches.

Speed.	Experimental discharge per 50 strokes.	Plotted average discharge per stroke.	Speed.	Experimental discharge per 50 strokes.	Plotted average discharge per stroke.
	Pounds.	*Pounds.*		*Pounds.*	*Pounds.*
12	465	9.3	20	465	
12	464		25	457	9.2
12	467		25	460	
15	469	9.38	30	460	9.18
15	471		30	460	
15	468		35	458	9.16
15	468		35	456	
20	468	9.32	40	456	9.14
20	461		40	460	
20	467				

Figures from efficiency tests of 4-inch Cook pump; stroke, 24 inches.

LIFT, 34.8 FEET; OPEN WELL.

Speed.	Discharge per stroke.	Foot-pounds of useful work.	Average pull on rod.	Foot-pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
12	9.3	323.6	207.6	415.6	77.8	280
12	9.3	323.6	213.9	428.1	75.6	285
15	9.38	326.4	205.3	410.9	79.4	315
15	9.38	326.4	203.9	408.2	79.9	305
20	9.32	324.3	215.9	432.1	75	350
20	9.32	324.3	212.5	425.4	76.2	335
25	9.2	320.1	225.6	451.6	70.8	385
25	9.2	320.1	225.6	451.6	70.8	410
30	9.18	319.5	246.3	493	64.8	510
30	9.18	319.5	243.1	486.6	65.6	525
35	9.16	318.8	257.6	515.6	61.8	575
35	9.16	318.8	253.4	507.3	62.8	635
40	9.14	318.1	273	546.4	58.2	650
40	9.14	318.1	270.1	540.7	58.8	650

LIFT, 34.8 FEET, USING STUFFING BOX.

12	9.3	323.6	243.1	486.5	66.5	315
12	9.3	323.6	232.7	465.7	69.5	310
15	9.38	326.4	251.4	503.2	64.8	350
15	9.38	326.4	251.4	503.2	64.8	375
20	9.32	324.3	264.5	529	61.2	400
20	9.32	324.3	260.4	521.3	62.2	415
25	9.2	320.1	250	500.4	63.9	420
25	9.2	320.1	259.1	518.6	61.7	435
30	9.18	319.5	266.7	533.8	59.8	565
30	9.18	319.5	279.2	558.8	57.1	560
35	9.16	318.8	297.2	594.9	53.5	650
35	9.16	318.8	300	600.4	53	675

Figures from efficiency tests of 4-inch Cook pump; stroke, 24 inches—Continued.

LIFT, 48 FEET, USING STUFFING BOX.

Speed.	Discharge per stroke.	Foot pounds of useful work.	Average pull on rod.	Foot pounds of work given to rod.	Mechanical efficiency.	Maximum pull on rod.
	Pounds.		*Pounds.*		*Per cent.*	*Pounds.*
12	9.3	446.4	284.8	570.1	78.3	380
12	9.3	446.4	288.1	576.7	77.4	375
15	9.38	450.2	295.4	591.2	76.1	400
15	9.38	450.2	298.6	597.7	75.3	415
20	9.32	447.4	302.2	605	73.9	460
20	9.32	447.4	309.2	618 9	72.3	450
25	9.2	441.6	327.3	655.1	67.4	515
25	9.2	441.6	304.9	610.2	72.3	490
30	9.18	440.6	322.7	645.9	68.2	565
30	9.18	440.6	316.2	632.8	69.6	545
35	9.16	439.7	325.7	651.9	67.4	640
35	9.16	439.7	331.9	664.4	66.1	655

LIFT, 71 FEET, USING STUFFING BOX.

15	9.38	666	416.6	834	79.8	565
15	9.38	666	389.3	779.2	85.4	520
20	9.32	661.7	433.9	868.4	76.2	585
20	9.32	661.7	421.4	843.4	78.4	595
25	9.2	653.2	431.3	863.2	75.6	625
25	9.2	653.2	428.1	856.8	76.2	630
30	9.18	651.8	429	858.6	75.9	750
30	9.18	651.8	454.7	910.2	71.6	765

LIFT, 82.5 FEET, USING STUFFING BOX.

15	9.38	773.9	475	950.7	81.4	590
15	9.38	773.9	488.2	977.1	79.2	590
20	9.32	705	478.5	957.7	80.3	705
20	9.32	705	485.3	971.5	79.1	700
25	9.2	759	489.6	980	77.4	780
25	9.2	759	488.9	978.5	77.5	825
30	9.18	757.3	509.7	1,020	74.2	830
30	9.18	757.3	518.7	1,038	72.9	840
35	9.16	755.7	529.2	1,059	71.3	815
35	9.16	755.7	523.7	1,048	72.1	1,020

SUMMARY.

To pump large quantities of water with small power economically, it is necessary that the pump speed be slow, preferably below 20 strokes per minute; that the pump itself be large and of long stroke, and that the valve area be at least 30 per cent of the cylinder cross-section area.

For the more rapid-running pumps the valve area should be still larger.

The efficiency of pumps rightly called good pumps may vary from 20 to 85 per cent, depending on the lift and the piston speed.

A fall of 25 per cent in the efficiency of a pump in the usual range of windmill speed is not uncommon.

A pump having a variation of only 5 per cent is possible.

For wells about 20 feet deep there is no need for using a pump of less than 75 per cent efficiency for average speeds.

Two pumps may compare very favorably at a certain lift, and much less favorably at some other lift not greatly different.

A pump having an efficiency of over 80 per cent at piston speeds up to 100 feet per minute is possible at a 20-foot lift.

Some pumps can be run at a piston speed of 180 feet per minute and maintain an efficiency above 70 per cent for lifts above 20 feet.

A pump with very large clack valves, on a very low lift, may give an efficiency from 80 per cent at slow speed to 50 per cent at a piston speed of 80 feet per minute.

Pumps at slow speed and at high speed alike discharge a little less water per stroke than at medium speed—perhaps from 1 to 2 per cent.

A pump with small delivery pipe may at high speed deliver a considerably larger amount of water than that computed from the cylinder capacity.

With pumps having an ample supply of water the speed may be limited by failure of the lower valve if its movement is not limited.

Valves of limited movement are preferable for fast speeds.

The addition of a stuffing box to any form of pump may seriously reduce the efficiency.

The initial stress on the upstroke at the higher speeds is the destructive element in windmill operation. The use of a device for reducing this stress is well worth consideration. At piston speeds of only 70 feet per minute this sudden jerk may be equivalent to two or three times the weight of the column of water lifted.

A method of pump testing is possible which would determine the behavior of valves and give the measure of the resistances at various stages of the stroke in such manner as to admit of quantitative statement.

RESISTANCE TO ROTATION OFFERED BY VARIOUS CRANK-DRIVEN PUMPS.

The resistance offered to any driving mechanism by a single-cylinder pump varies greatly at different stages of the pump stroke. If connection with the pump be through the usual crank, the resistance offered by the crank will be slight at each dead center. Piston and rod, if of sufficient weight, will afford some motive power on the downstroke. The useful work is done during the upstroke, and the larger portion of it during the first part of the upstroke, excepting when the speed is very slow. In fig. 39 horizontal distances represent positions of the crank in its circular course during a single revolution. The position in each case is located from the lower dead center as an initial point. The heights of the vertical lines represent the resisting moment at that point. Neglecting the effect of angularity of the connecting rod, the smooth sinusoidal curve represents the resistances to turning, assuming the load to be uniform on the upstroke and to be diminished to about one-eighth on the downstroke. This diminished load is the weight of the piston, rod, etc. The shaded area above the zero line represents the work done in raising the piston, rod, and water; and that below the line, from 180° to 360°, represents the work returned by the falling piston and rod on the downstroke. The outside line and the heavier-shaded portion show the actual resisting moments, as computed from the diagram, fig. 40, from the work of an 8-inch pump at a lift of 22 feet and a speed of 25 strokes per minute. The additional work required for acceleration of the water, etc.,

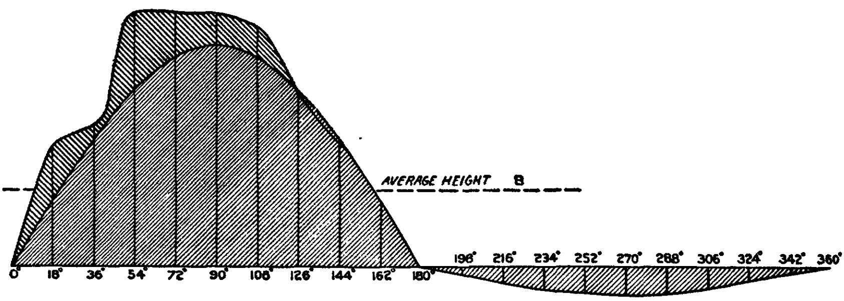


FIG. 39.—Diagram of resistance to rotation of crank driving a single-cylinder lift pump.

is shown by the heavier-shaded area. This great variation in the turning resistance renders it impossible, without considerable loss of efficiency, effectively to use certain forms of motors.

If a horse be used to operate a pump of this kind, by means of a sweep or other suitable turning device, during a stroke of the pump, he will meet with resistances such as shown by the vertical lines in fig. 39. The pump must not be so large that the maximum moment

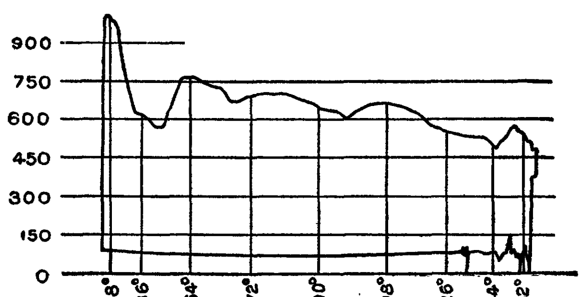


FIG. 40.—Dynamometer diagram of actual resistance at various positions of crank.

can not be reached and overcome at a point somewhere between 50° and 90° of the revolution. Even then, for at least half the time, the horse will be doing no work at all, and for 60 per cent of the time he will be loaded below the average. The line AB gives the average resistance. Furthermore, it appears that the maximum load reaches to more than three and one-half times that of the average load. This fault may be remedied in part by the use of a fly wheel of moderate weight. The unevenness of loading will continue to be so considerable, however, as to render the work of the horse laborious; whereas with equalized load he could do even a larger amount of work with comparative ease. A windmill encounters the same difficulty, as may be seen from the ineffectual efforts it makes in light winds to pass

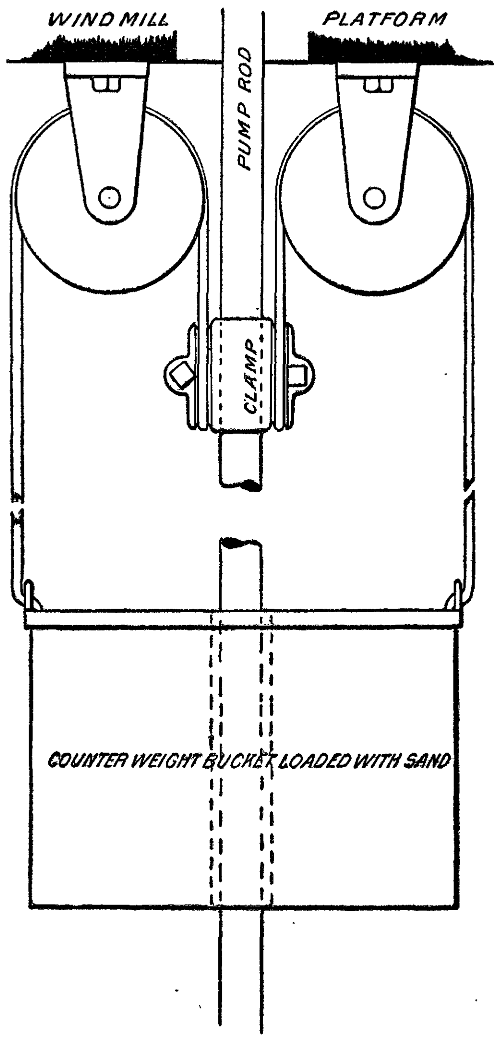


FIG. 41.—Method of counterbalancing a windmill pump.

this point of maximum load. It becomes apparent, too, that if the load were evenly distributed the mill would find the light wind power sufficient for light running. By the use of a counterweight a partial distribution at least can be effected.

A modification of the horse-power sweep sometimes to be seen is a wooden spring bar so attached as to offer a spring resistance during the light-pull interval and to assist correspondingly on the heavy pull. It has decided merit. As applied to the pump rod of a windmill the modification might take the form of a weight encircling the rod and given a direction of motion opposite to that of the rod by means of pulleys, as shown in fig. 41. This device should be attached to back-geared mills only, and should have its range of action fixed so high in the tower as to leave the pump rod free to act in compression without bending. The counterweight should equal one-half that of the load on the piston and should range as low in the tower as practicable.

Differential pumps equalize the work between the upstroke and

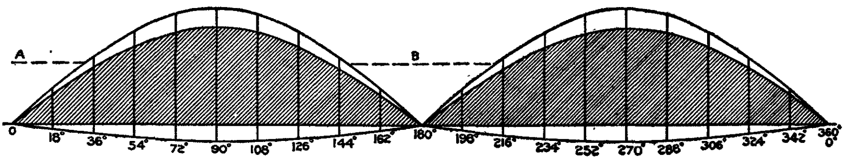


FIG. 42.—Diagram of resistance to rotation of a shaft carrying two cranks driving a duplex pump.

the downstroke by employing a very large piston rod. If the cross-section area of the piston rod be one-half that of the cylinder, one-half of the cylinder capacity will be discharged on the upstroke and the remaining half by displacement on the downstroke. Work is thus equalized between the upstroke and the downstroke. Since, however, the work required on the downstroke, which has now become in fact a down thrust, necessitates the use of stiff rods in order to avoid compressive bending, there is disadvantage in the practice where long rods are required, as in deep wells or for windmills with

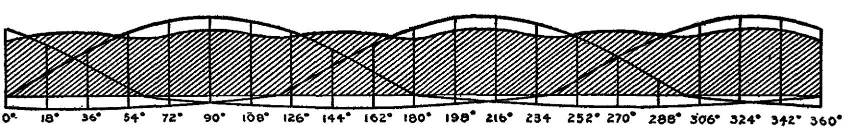


FIG. 43.—Diagram of resistance to rotation of a shaft carrying three cranks driving a triplex pump.

high towers. For such long thrust rods guides must be provided at frequent intervals to preserve the alignment and insure direct action, and frictional resistance is thus introduced.

Again, there will be enlargement of the pump rod for a short distance only at the water surface in the discharge pipe. This enlargement is partially withdrawn from the water on the upstroke, thereby vacating a space to be filled and thus reducing the discharge, and it is lowered into the water on the downstroke, thereby forcing upward an equal volume of water by displacement and thus compensating for the previous loss. Yet it does not effect an equalization of work between the two strokes. The full cylinder capacity is lifted on the upstroke nearly to the full height of the discharge, and on the downstroke but half of this is displaced from the height of the discharge opening, requiring little work on the downstroke. It is evidently a mistake to regard this device as an equalizer of work. The enlargement, to be fully effective, should continue down to the piston.

Compound pumps having four valves and discharging on both the upstroke and the downstroke, or two cylinder pumps in which one cylinder makes an upstroke during the downstroke of the other, would produce a resistance diagram such as shown in fig. 42. This diagram

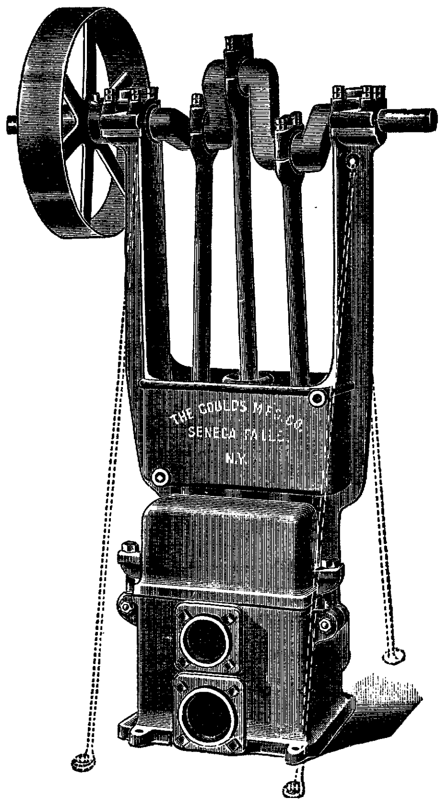


FIG. 44.—Gould triplex power pump.

represents the crank resistance of a pump discharging the same quantity of water per revolution of the crank as the pump represented in fig. 39 (p. 66). The curve below the zero line exhibits the effect of the weight of piston and rod, and the curve above the resistance due to lifting piston, rod, and water. The shaded portions show the resistances at each point in the revolution, the piston weights balancing each other. In this double-cylinder form the maximum resistance is about 1.6 times the average resistance, as against $3\frac{1}{2}$ times with the single-cylinder form. At two points in each rotation the resistance will be zero, but it does not fall below zero, as with the single-cylinder form. A pump of this kind can be more easily driven by horse power than one with a single cylinder, but it is, on the other hand,

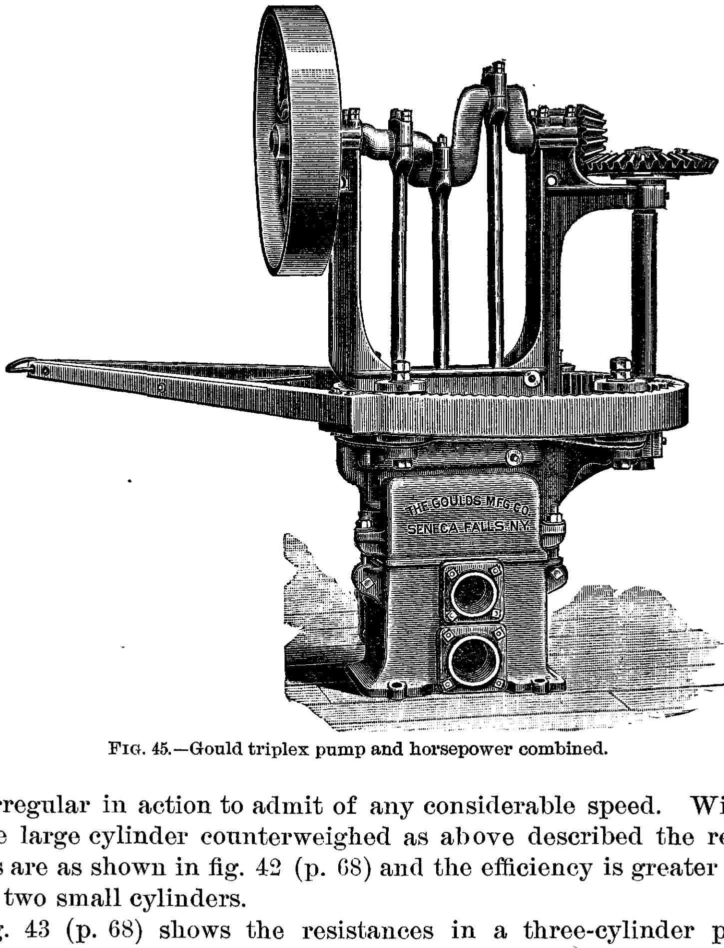


FIG. 45.—Gould triplex pump and horsepower combined.

too irregular in action to admit of any considerable speed. With a single large cylinder counterweighed as above described the resistances are as shown in fig. 42 (p. 68) and the efficiency is greater than with two small cylinders.

Fig. 43 (p. 68) shows the resistances in a three-cylinder pump arranged to discharge the same quantity per revolution as did the

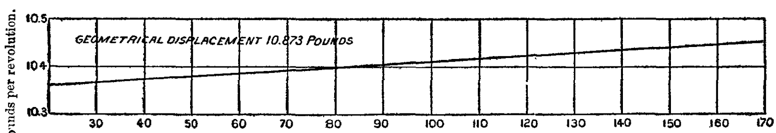


FIG. 46.—Discharge of Gould 4-inch by 8-inch triplex pump.

two-cylinder and the one-cylinder forms. The three pistons are driven from three crank elbows in the shaft set at 120° each to each, as shown in fig. 44 (p. 69). The resistance offered by this device is very uniform, as appears from the shaded portion of the diagram. The maximum resistance is but 6 per cent above the average resistance, and a moderate fly wheel renders the motion entirely steady.

Such pumps admit of the best use of any constant power, as that of a horse or a small engine. In Gould's triplex pump (fig. 44), a horse will encounter a nearly uniform resistance and can work to the best advantage. A 4 by 8 inch pump of this make was tested by the

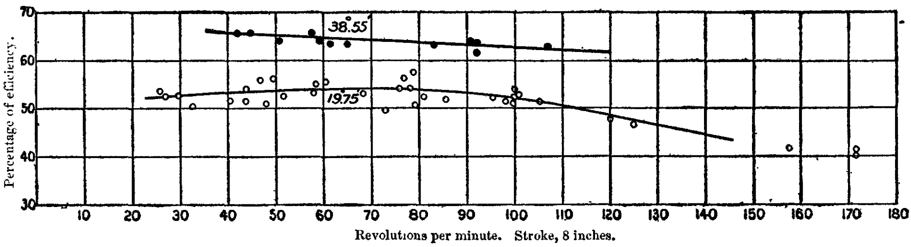


FIG. 47.—Varying-efficiency curves of Gould triplex pump, showing effect of varying speed, and at two different lifts.

author at three different lifts. The crank shaft is arranged to be run by a pulley belted from any source of power, or by means of the encircling internal master wheel and the vertical shaft carrying bevel gears, as in fig. 45, and operated by means of a horse attached to a sweep on the master wheel. The very uniform resistance of this kind of pump, as shown by the diagram, fig. 48, which represents the belt pull or resistance while at work, has opened a large field for it

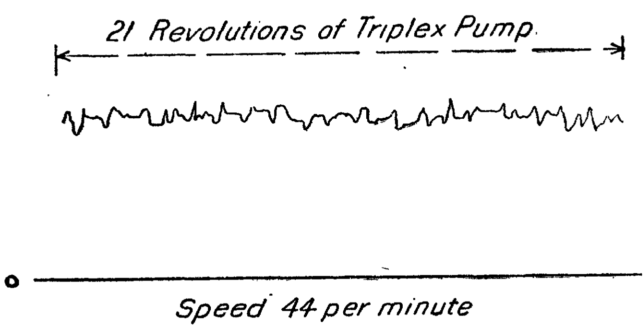


FIG. 48.—Dynamometer diagram showing uniformity of resistance afforded by triplex pump.

in connection with various motors. While not so efficient as a single-cylinder pump of equal capacity, yet it applies energy so effectively as to render it the more advantageous form for many uses. The efficiency of the one tested is seen to be about 60 per cent—not greater than that of many homemade water lifts. Its durability and compactness, however, entitle it to very favorable consideration in comparison with any such homemade device.

Discharges of Gould triplex pump.

LIFT, 19.75 FEET.

Speed, in revolutions per minute.	Discharge per minute by experiment.	Average discharge per minute by discharge curve.	Average discharge per revolution.	Speed, in revolutions per minute.	Discharge per minute by experiment.	Average discharge per minute by discharge curve.	Average discharge per revolution.
	Pounds.	*Pounds.*	*Pounds.*		*Pounds.*	*Pounds.*	*Pounds.*
30	331	308	10.2	75	784	783	10.4
30	321	308	10.2	79	816	825	10.4
36	353	373	10.3	82	876	857	10.4
47	476	490	10.4	90	947	941	10.4
52	539	542	10.4	99	1,035	1,035	10.4
56	581	583	10.4	111	1,175	1,161	10.4
57	644	595	10.4	114	1,208	1,192	10.4
60	628	625	10.4	114	1,213	1,192	10.4
65	672	678	10.4	133	1,390	1,390	10.4
70	728	730	10.4	156	1,630	1,631	10.4
74	771	772	10.4	166.8	1,720	1,745	10.4
LIFT, 38.55 FEET.							
40	416	416	10.4	84	898	853	10.4
45	482	468	10.4	90	964	941	10.4
77	827	803	10.4				

Figures from efficiency test of Gould triplex pump 4 by 8 inches.

LIFT, 19.75 FEET.

Speed, in revolutions per minute.	Discharge per minute.	Foot-pounds of useful work per minute.	Foot-pounds of work given to pump per minute.	Mechanical efficiency.
	Pounds.			*Per cent.*
25.9	265	5,234	9,715	53.8
27	276	5,451	10,370	52.5
29.5	303	5,984	11,429	52.3
32.8	338	6,675	13,283	50.2
40.5	420	8,295	16,008	51.8
43.8	455	8,986	17,569	51.1
43.8	455	8,986	16,588	54.1
46.9	487	9,618	17,213	55.8
48	500	9,875	19,405	50.9
49.6	515	10,171	18,133	56.1
51.8	540	10,665	20,260	52.6
58.2	607	11,988	22,410	53.5
58.8	612	12,087	21,952	55
60.6	639	12,620	22,771	55.4
68.2	712	14,062	26,391	53.2
73.2	764	15,086	30,330	49.7
76	794	15,681	29,063	53.9
78.2	815	16,096	29,809	54
79.2	825	16,292	33,309	48.9
81.1	848	16,748	32,072	52.2
85.7	895	17,666	35,743	51.8
95.3	997	19,690	37,750	52.1
98.3	1,026	20,263	39,710	51
100	1,047	20,678	40,175	51.4
100	1,047	20,678	40,413	51.1
101	1,055	20,836	39,430	52.8
105.3	1,100	21,725	42,409	51.2
120	1,255	24,786	51,698	47.9
125	1,305	25,773	55,202	46.7
157.9	1,650	32,587	78,199	41.6
171.4	1,790	35,352	86,142	41
171.4	1,790	35,352	87,937	40.2

Figures from efficiency test of Gould triplex pump 4 by 8 inches—Continued.

LIFT, 38.55 FEET.

Speed, in revolutions per minute.	Discharge per minute.	Foot-pounds of useful work per minute.	Foot-pounds of work given to pump per minute.	Mechanical efficiency.
	Pounds.			*Per cent.*
42.1	439	16,923	25,801	65.6
44.8	465	17,925	27,167	65.9
50.8	530	20,431	31,765	64.3
57.7	600	23,130	35,144	65.8
59.4	618	23,824	37,177	64
61.2	640	24,672	39,027	63.2
65.2	675	26,022	41,058	63.3
83.3	870	33,538	52,978	63.3
90.9	950	36,622	57,212	64
92.3	965	37,201	58,517	63.5
92.3	965	37,201	60,065	61.9
107.1	1,120	43,176	69,324	62.2

VARIOUS WATER LIFTS.

On the Western plains devices for raising water other than the reciprocating pumps are not common. The centrifugal pump, which, if driven at a high and uniform speed, has an efficiency of about 66 per cent at a lift of about 16 feet, has its field limited to tracts where water is to be found at comparatively shallow depths and to applications where motive power giving uniform and high speed may be economically employed. It is not well adapted to horse or windmill power.

Early forms of water lifts are mentioned and illustrated by Weisbach. While these are of interest, they are not likely to be used by American farmers, who are impatient of slow and laborious methods. The following statements are abstracted from Mechanics of Pumping Machinery, by Weisbach and Hermann:

Men, bailing with buckets holding about one-third of a cubic foot each, can lift 15 buckets per minute, 3 to 4 feet high, for a length of time daily equivalent to six hours of steady labor. This amounts to about 390,000 foot-pounds per day, or a quantity of water sufficient to cover an acre one-half inch deep. With deeper wells the sweep or weighted lever will yield about the same result in foot-pounds daily per man. At a crank handle an ordinary laborer can exert a force of 15 pounds in steady work, or can labor steadily at a rate of approximately 3,300 foot-pounds per minute. This is about the best manner of employing human labor. From a 10-foot well a man with a winch and bucket, properly proportioned, can cover an acre with one-half inch of water in a day. Approximately the same can be done with a hand pump suitably proportioned to the lift. The Dutch scoop, which consists of a long-handled scoop shovel suspended from an overhead support by a rope and swung by three men, will lift water 3¼ feet and throw it 6½ feet horizontally. One man working thus alone can do about as much work as by the bailing method, but three men working together on a single scoop will do more than five men bailing separately.

BUCKET LIFTS.

A very old device for raising water consists of a vertical rope or chain carrying buckets, the chain running on a power-driven sprocket wheel at the top and maintained in proper position by a loose sprocket suspended in its lower loop. The modern development of the sprocket wheel and of the cast-link belt, in which links may be had specially adapted to receive sheet-metal buckets, has brought the bucket lift

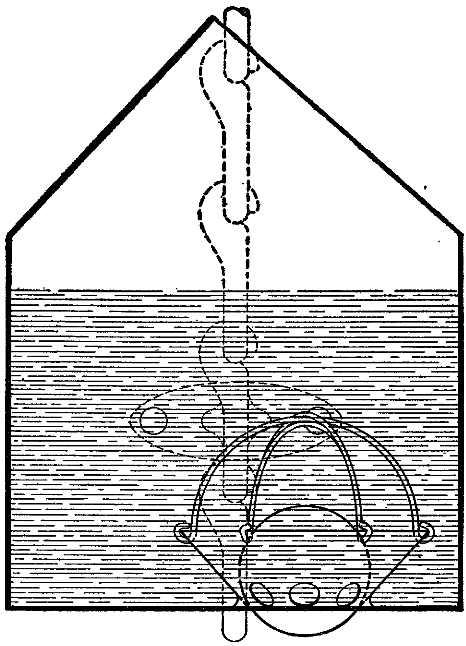
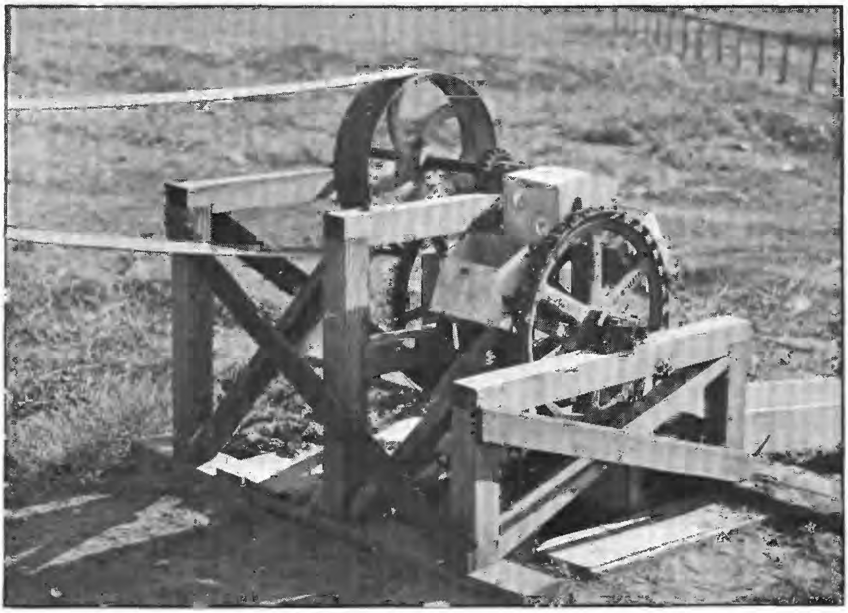


FIG. 49.—Vertical section of bucket of water elevator.

into renewed use. A modern form is shown in Pl. I, *A*. (See also fig. 49.) Its great advantages are uniformity of resistance and steady delivery. Its disadvantages are: large size for a given capacity, necessity imposed for using a large well, and difficulty in securing a perfect delivery. The buckets must be so made that in dipping into the water air will not be entrapped, and so that when delivering air may again readily enter. It has so far not been found practicable entirely to prevent loss from spilling of a portion of the water lifted. Furthermore, for emptying, the lift has to exceed somewhat the height of the discharge point. From these two inherent defects results a noticeable failure in efficiency.

SEAMAN BUCKET LIFT.

A lift designed for irrigation work, and made by Seaman & Schuske, of St. Joseph, Missouri, was tested, and the result appears



A.—VIEW OF LINK-BELT WATER ELEVATOR.

B.—VIEW OF BOYCE WATER LIFT.

in fig. 50. This device consists of a series of galvanized-iron buckets, riveted to lugs carried on a common link belt. The sprocket chains, at each side of the bucket, are carried on two sprocket wheels. Each bucket is provided with a clay-ball valve, carried in a wire cage at the bottom of the bucket. The leather clack valve is a very old form for this use. With the rise of the bucket the valve—of whatever design—covers a circular aperture in the bucket bottom, and

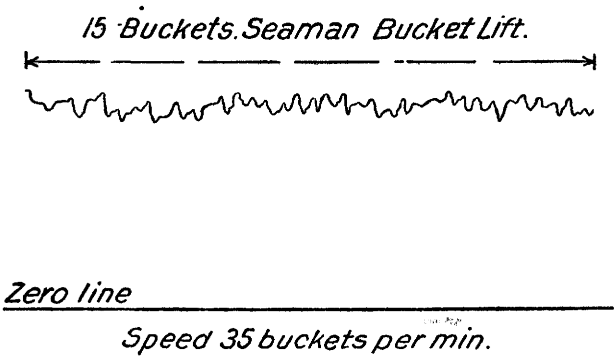


FIG. 50.—Dynamometer diagram showing uniformity of resistance afforded by Seaman bucket lift.

opens in turn with the descent into water, thus obviating the possibility of entrapping air. In the pendent loop of the chain two sprocket wheels on a shaft run without bearings. A gear on the main sprocket shaft is driven by a pinion, the power for which may be derived from any source.

The uniformity of the resistance offered by this lift is shown in fig. 50, taken from a recording dynamometer when the device was delivering about 300 pounds of water per minute, from a depth of 18.8 feet.

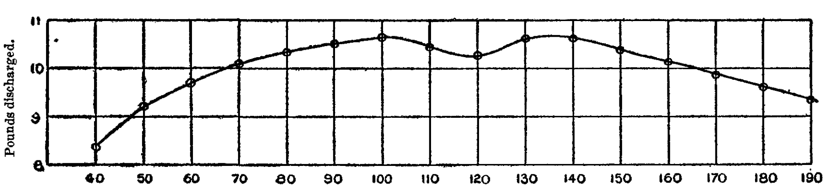


FIG. 51.—Discharge per bucket of Seaman bucket lift at various speeds.

Fig. 51 gives the discharge per bucket at various speeds, and fig. 52 the total discharge. The efficiency, as shown in fig. 53, is about 60 per cent at the favorable speed of 100 buckets per minute. A peculiar drop appears in the discharge record at speeds ranging from 100 to 130 buckets per minute. At very slow speed the discharge is less, due to leakage around the valve, and due, further, to failure of the buckets to empty into the discharge chute. At higher speed the valve loss becomes small, but the buckets soon pass the discharge chute before fully emptying. At still higher speed centrifugal force aids greatly in discharging the bucket and temporarily increases the discharge. As speed still further increases, however, swaying of the chain causes loss, by slopping, from the rising buckets.

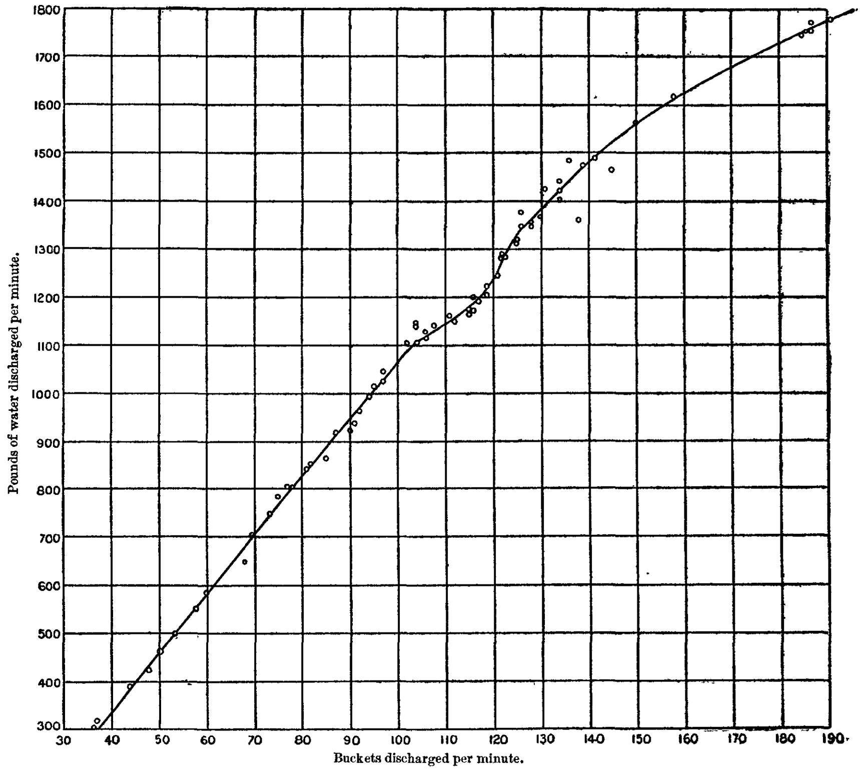


FIG. 52.—Total discharge of Seaman bucket lift at various speeds.

This pump has been successfully used in connection with a windmill. It can be recommended wherever a large open well is available, and where the depth to water is moderate, say from 20 to 35 feet. It

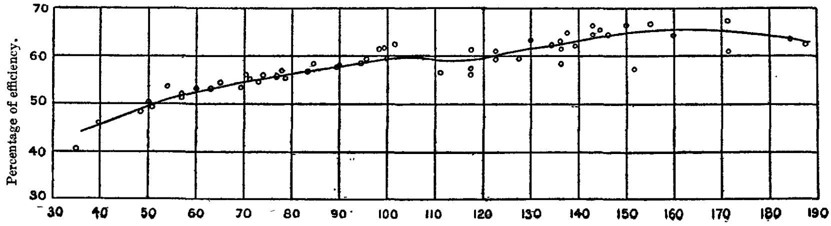


FIG. 53.—Efficiency curve of Seaman bucket at lift of 18.8 feet.

is best adapted to small powers of from 1 to 5 horsepower. The power necessary to operate such a pump is about double that represented by the useful work done, in water raised and discharged.

Discharges of Seaman bucket lift at various speeds.

Speed, in buckets per minute.	Discharge per minute by experiment.	Average discharge per minute by discharge curve.	Average discharge per bucket.	Speed, in buckets per minute.	Discharge per minute by experiment.	Average discharge per minute by discharge curve.	Average discharge per bucket.
	Pounds.	*Pounds.*	*Pounds.*		*Pounds.*	*Pounds.*	*Pounds.*
38	315	310	8.158	117	1,191	1,193	10.19
44	389	385	8.75	118	1,205	1,205	10.21
48	423	433	9.02	119	1,223	1,218	10.23
58	555	558	9.62	121	1,247	1,254	10.31
68	651	682	10.03	122	1,286	1,273	10.43
73	750	743	10.17	122	1,290	1,273	10.43
75	768	770	10.26	123	1,295	1,290	10.48
77	805	793	10.3	124	1,281	1,307	10.54
78	803	804	10.31	125	1,317	1,321	10.57
81	841	841	10.38	126	1,348	1,335	10.59
85	866	889	10.45	126	1,379	1,335	10.59
87	918	912	10.48	128	1,358	1,360	10.62
90	923	948	10.53	128	1,351	1,360	10.62
91	937	960	10.55	130	1,380	1,382	10.63
92	963	972	10.56	130	1,367	1,382	10.63
94	974	996	10.59	131	1,427	1,395	10.64
94	993	996	10.59	133	1,412	1,415	10.64
95	1,114	1,008	10.61	134	1,404	1,426	10.64
97	1,025	1,032	10.63	134	1,420	1,426	10.64
97	1,044	1,032	10.63	134	1,442	1,426	10.64
98	1,014	1,044	10.65	136	1,437	1,447	10.64
99	1,094	1,056	10.66	138	1,362	1,465	10.61
99	1,024	1,056	10.66	139	1,477	1,475	10.61
102	1,108	1,092	10.70	141	1,490	1,492	10.58
104	1,107	1,114	10.71	144	1,534	1,513	10.5
104	1,146	1,114	10.71	145	1,461	1,520	10.48
106	1,123	1,127	10.63	149	1,559	1,548	10.4
106	1,127	1,127	10.63	150	1,561	1,556	10.37
106	1,114	1,127	10.63	158	1,618	1,610	10.12
108	1,140	1,138	10.53	178	1,685	1,718	9.65
111	1,149	1,151	10.37	185	1,745	1,750	9.46
111	1,161	1,151	10.37	186	1,752	1,755	9.42
112	1,151	1,155	10.31	186	1,697	1,755	9.42
115	1,170	1,175	10.22	186	1,720	1,755	9.42
115	1,175	1,175	10.22	187	1,757	1,758	9.4
116	1,200	1,185	10.21	187	1,752	1,758	9.4
116	1,175	1,185	10.21	191	1,779	1,779	9.31

Figures from efficiency tests of Seaman bucket lift.

USEFUL LIFT, 18.8 FEET.

Speed, in buckets per minute.	Discharge per minute.	Foot-pounds of useful work per minute.	Foot-pounds of work given to machine per minute.	Mechanical efficiency.	Lift.
	Pounds.			*Per cent.*	*Feet.*
35.3	275	5,170	12,675	40.8	18.8
39.7	330	6,270	13,564	46.2	19
48.4	440	8,272	17,008	48.6	18.8
50	460	8,648	17,245	50.1	18.8
50.4	465	8,835	17,883	49.4	19
54	508	9,652	17,931	53.8	19
57.1	547	10,393	20,257	51.3	19
57.1	547	10,393	20,025	51.9	19
60	584	11,116	20,921	53.1	19
63.1	621	11,799	22,132	53.3	19
65.2	648	12,312	22,488	54.7	19
69.7	705	13,395	25,218	53.1	19
70.6	713	13,547	24,143	56.1	19
71.4	724	13,756	24,859	55.3	19
73	743	14,117	25,825	54.6	19
74	757	14,383	25,644	56	19
77	794	15,086	27,388	55	19
77.9	804	15,115	26,422	57.2	18.8
78.9	815	15,322	27,563	55.6	18.8
83.3	870	16,356	28,740	56.9	18.8
84.5	884	16,619	28,319	58.6	18.8

Figures from efficiency tests of Seaman bucket lift—Continued.

USEFUL LIFT, 18.8 FEET—Continued.

Speed, in buckets per minute.	Discharge per minute.	Foot-pounds of useful work per minute.	Foot-pounds of work given to machine per minute.	Mechanical efficiency.	Lift.
	Pounds.			*Per cent.*	*Feet.*
89.5	942	17,898	30,937	57.8	19
90	948	17,822	30,721	58	18.8
94.4	1,000	18,800	32,235	58.3	18.8
95.9	1,020	19,176	32,415	59.1	18.8
98.3	1,048	19,912	32,124	61.9	19
98.3	1,048	19,702	32,014	61.5	18.8
100	1,068	20,078	33,674	59.6	18.8
101.7	1,090	20,492	32,620	62.8	18.8
107.1	1,133	21,300	35,529	59.9	18.8
111.1	1,152	21,657	38,076	56.8	18.8
117.6	1,200	22,800	37,226	61.2	19
117.6	1,200	22,560	40,160	56.1	18.8
117.6	1,200	22,560	39,211	57.5	18.8
122.5	1,284	24,139	40,623	59.4	18.8
122.5	1,284	24,139	39,453	61.1	18.8
127.6	1,355	25,474	42,633	59.7	18.8
130	1,382	26,258	41,502	63.2	19
134.8	1,435	26,978	43,386	62.1	18.8
136	1,448	27,512	43,607	63	19
136.4	1,450	27,260	46,750	58.3	18.8
136.4	1,450	27,260	44,405	61.3	18.8
137.9	1,463	27,797	42,816	64.9	19
139.5	1,480	27,824	44,809	62.1	18.8
143	1,505	28,595	43,584	65.6	19
143	1,505	28,595	44,442	64.3	19
144.6	1,516	28,804	44,198	65.1	19
146.3	1,530	29,070	45,058	64.5	19
150	1,556	29,564	44,541	66.3	19
151.9	1,570	29,830	52,101	57.2	19
154	1,584	30,096	45,002	66.8	19
160	1,622	30,818	48,075	64.1	19
171.4	1,685	32,015	47,628	67.2	19
171.4	1,685	32,015	47,436	67.5	19
171.5	1,687	31,715	52,199	60.7	18.8
184.6	1,748	33,212	51,996	63.8	19
187.5	1,760	33,440	53,337	62.7	19

SAME USING TWO VALVES IN EACH BUCKET.

96	1,032	19,401	35,447	54.7	18.8
100	1,069	20,204	36,738	55	18.9
100	1,069	20,078	36,604	54.8	18.8
102.5	1,098	20,642	37,205	55.4	18.8
103.4	1,108	20,941	37,109	56.4	18.9
111.1	1,152	21,657	39,361	55	18.8
115.4	1,178	22,190	40,960	54.1	18.8
122.5	1,283	24,120	42,151	57.2	18.8
125	1,323	24,872	42,830	58	18.8
130.4	1,383	26,138	44,058	59.3	18.9
133.3	1,420	26,838	45,895	58.4	18.9
142.8	1,505	28,294	50,222	56.3	18.8
142.8	1,505	28,294	50,770	55.7	18.8
142.8	1,505	28,444	48,856	58.1	18.9
151.9	1,570	29,516	50,724	58.2	18.8
162.1	1,634	30,719	54,713	56.1	18.8
169	1,673	31,452	54,839	57.2	18.8
184.6	1,749	32,881	62,924	52.2	18.8
187.5	1,760	33,088	63,337	52.2	18.8

LIFTING BY ANIMAL POWER.

It is somewhat surprising that horses are not more used for lifting water. In small operations, or where a farmer is experimenting with a view to irrigation, horse power would seem to be the resource readiest to hand. It is but little recognized how much work may be got from a horse if a suitable device be provided. Farm horses, like their masters, work especially hard during special seasons. Their rate is higher then than it could be if they were worked continuously. If the following estimates at first sight appear low, it should be understood that they are made on the assumption of a uniform daily rate. It is assumed that a good work team of two horses will pull 300 pounds steadily during a day of eight hours. In plowing, this effort is frequently doubled. For horses worked continuously at a straight-away pull, 125 pounds each for medium-weight horses is good average work. For occasional short periods a horse can throw one-third of his weight into his effort, but he can not do so at frequently recurring intervals. The maximum load for a horse, if recurring at short intervals, should not exceed double the average fair load, and the more frequently it recurs the less it should be. The natural gait of a work horse is about 3 feet per second. If loaded to pull 125 pounds at this gait, his work would be equivalent to 22,500 foot-pounds per minute, or about two-thirds of a horsepower.

If a horse could be worked steadily at some device the efficiency of which would be 66 per cent, he could lift, in a ten-hour day, $\frac{3.3}{L}$ acre-feet of water, L being the height of lift in feet. In many cases the lift will be about 15 feet, giving an output per horse of 0.22 acre-foot, and if the depth of irrigation be 3 inches, accompanied with the usual waste, the water for about three-fourths of an acre per day could easily be supplied by two horses. With storage and with intervals of ten days between irrigations a team of two horses could care for from 5 to 7 acres. While to the enthusiast these figures will not appear large, they are large enough to justify the use of horses in small undertakings. The assumed efficiency of 66 per cent is, in fact, low rather than high, and a device for the utilization of horsepower that shall attain to this percentage ought to be possible of achievement by mechanical skill.

The foregoing is a statement of what it is possible to do with horses if suitable machinery be provided; but the majority of existing devices will not do so well as this. As shown by the stress diagram for a single-cylinder pump (fig. 39, p. 66), the maximum stress is more than three and one-half times what would be the average stress. As the maximum effort of a horse should not rise to more than double his average effort, and as the average resistance in the single-cylinder pump is 28 per cent of the maximum resistance, the actual average load put on the horse can equal only 28 per cent of twice his average effort, or 56 per cent of his possible average effort. In order to get full work out of a horse, therefore, it would be necessary, at some period of each pump stroke, for him to make an effort more than three and one-half times his most efficient average effort. This would be too great irregularity of exertion.

A horse can not do full work on an unbalanced single-cylinder pump. It is imperative that recourse be had to some device which will more nearly equalize his efforts. The use of a sweep, by reason of the circular course a horse has to follow, involves a loss of 20 per cent of efficiency, as compared with the possible efficiency in a straight-away pull. At each point of his actual course he will be pulling at an angle with his most effective course, which at any point will always be a tangent to the circular path. Tests of the usual sweep horse-power show an efficiency of about 80 per cent.

A horse can deliver to a pumping device, through the medium of a sweep, 80 per cent of 80 per cent, or 64 per cent of his best effort. If the pumping device have an efficiency of 60 per cent, such as shown by the bucket lift, the water actually lifted will represent 80 × 80 × 60 per cent, or 38.4 per cent straight-away average effort of the horse. In the operation of a bucket lift of this kind,

$$\frac{8640}{L} = p,$$

in which L is the total lift of water in feet, and p the number of pounds discharged per minute for each horse employed.

To determine the proper speed for the bucket lift, let S be the length of sweep in feet, L the lift of water in feet, l b d the dimensions, respectively, of the buckets used in inches, and n the number of buckets emptied during one revolution of the sweep; then, for each horse

$$\frac{10{,}400\,S}{L\,l\,b\,d} = n.$$

Omitting from consideration what might be done by overtaxing the horse, a larger output than this is to be looked for only through increase of efficiency in the mechanism of either the sweep or the pump. This low efficiency, in which but 38 per cent of the power expended by the horse appears as returned in useful work, lies at the root of the nonuse of horsepower for these light water-lifting operations. As already stated, the difficulty does not to the writer appear insuperable.

SOME INDIAN DEVICES.

Among devices employed in India for raising water from wells by animal power a number have been tested as to their mechanical efficiencies, and have been reported upon to the Madras government in Bulletin No. 32 of the department of land records and agriculture, under the title, Experiments with Water Lifts. The following copious extracts from this interesting report, together with figs. 54, 55, and 56, which are selected reproductions from its plates, show what may be accomplished with these rude devices of the Indian farmer, to whom the cost of modern machinery is, as a rule, prohibitive.

The experiments conducted by the committee had the following objects in view: The determination of—

(1) The quantity of water and the effective height it was lifted in a given time by bullocks of known weight, and working in a way that did not unduly fatigue them.

(2) The quantity of work actually obtained from the animals in the same time.

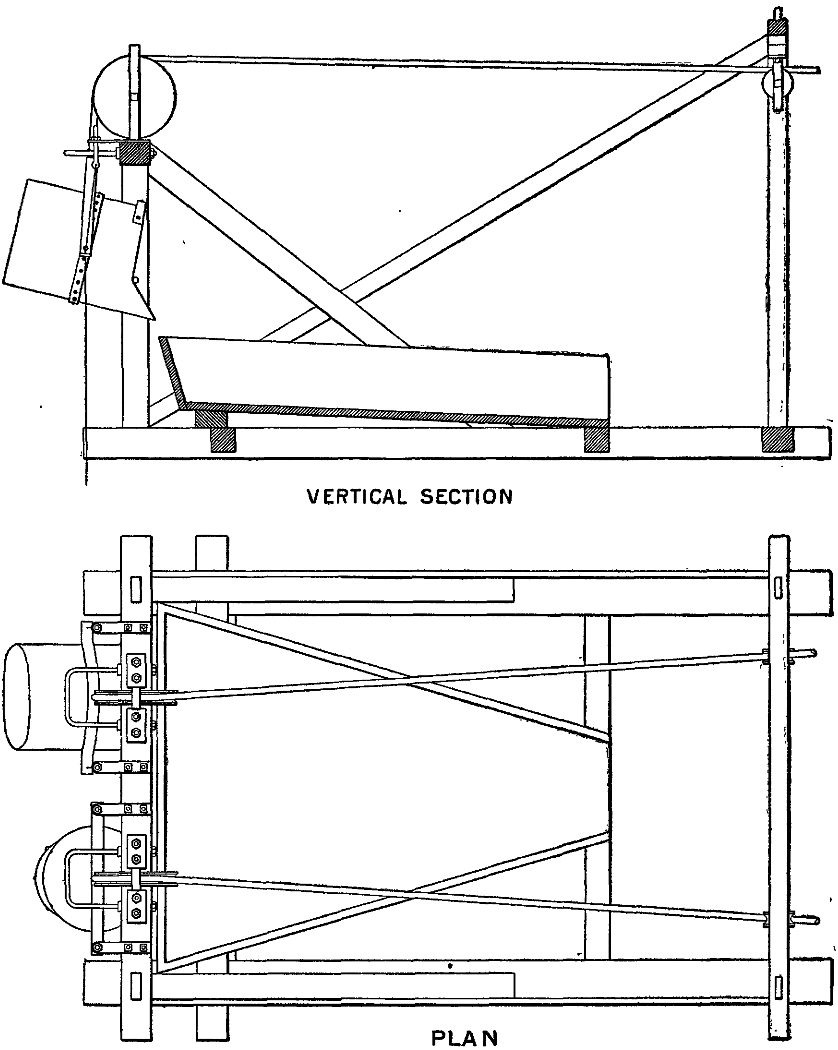


FIG. 54.—Plan and vertical section of Stoney water lift.

(3) The quantity of work which might have been obtained from the animals in the same time had the working of the lifts been continuous.

The first quantity divided by the second gives the mechanical efficiency of the water lift, and the first quantity divided by the third gives its absolute efficiency as a machine for utilizing animal power in a given way.

(4) The quantity of work represented in water lifted per hour divided by the weight of the bullocks in pounds.

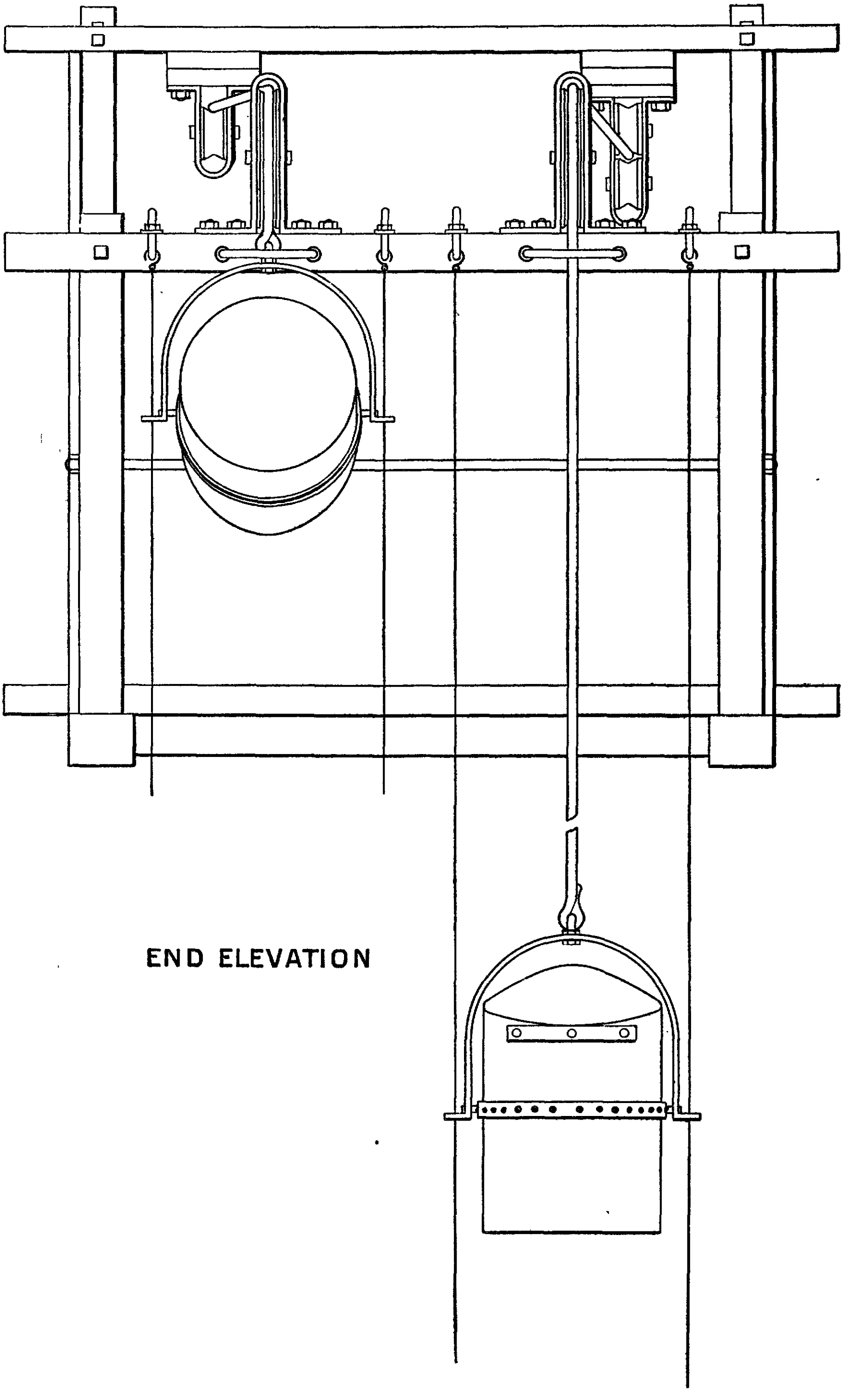


FIG. 55.—End elevation of Stoney water lift.

This yields a constant which, on the assumption that the animals employed in working the various lifts are all equally strained, enables a comparison to be made between very different types of water lift and very different methods of applying animal power. In some lifts only the draft of the animals is utilized. In others, as for instance the ordinary single mhote, both draft and weight are utilized, whilst in a third class only the animals' weight is made use of, as in Mr. Subba Rao's oscillating platform; and between such very different methods the constants yielded by (4) seems to be the best method of making comparisons of the actual value of different machines for lifting water. It must be remembered

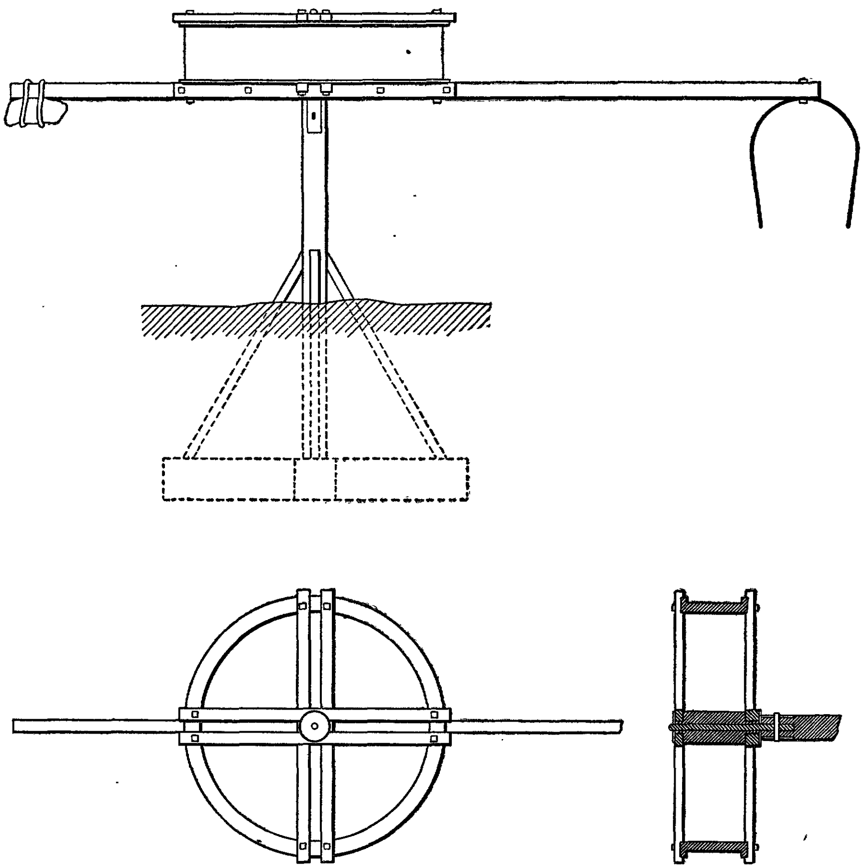


FIG. 56.—Whim used with Stoney water lift.

that though the mechanical efficiency of a water lift may be high, yet the mode of getting the work out of the animals employed may be bad, and the actual value of the constants yielded by (4) depends on the assumption that the strength of animals is proportional to their weight, which is probably roughly true for animals in good working condition and in the prime of life, and on the accuracy with which it is possible to estimate whether two animals working in different ways are exerting themselves to their full working capacity. The figures therefore obtained are only to be taken as offering a rough method of comparing the different lifts, and as a guide in estimating their probable value as machines for lifting water.

Stoney's water lift.—The principal feature in this lift is the employment of buckets of wrought iron, suspended in a stirrup by two adjustable pivots attached to the bucket very slightly above the center of gravity of the bucket when full of water. The mouth of the bucket is inclined as shown in the drawing, and the lower ends of the stirrup are turned outward and encircle steel wires which are suspended in the well from screw eyebolts attached to the framing above. The wires are fastened by some convenient means to the bottom of the well and act as guides to the bucket, ascending and descending, and prevent it from either turning round or swaying to and fro, and thus striking either the sides of the well or the second bucket. On the bucket being lowered into the water it turns horizontally and rapidly fills with water, and on being drawn up assumes a vertical position and rises steadily out of the water, till the discharging level is reached, when the upper side of the inclined mouth comes into contact with an iron bar fixed across the framing of the lift, and the stirrup, continuing its upward motion, causes the bucket to revolve about the point of contact of the bucket with the iron rod, and thus discharge its contents into the delivery trough. The lift was worked by carrying the ropes which hold the buckets over guide pulleys to a whim turned by either a pair of bullocks or a single bullock. Two buckets were attached and the ropes arranged so that as one bucket ascended the other descended, and the dead weight of the buckets was balanced. The whim consisted of a drum built of wood and carried by an iron spindle on the top of a post firmly built into the ground. The bullocks worked at the end of a long arm, the circumference swept out by which was 3.85 times the circumference of the drum. The whim is worked alternately in one direction and the other, the cattle being made to turn round while the bucket is discharging its contents. The lift was provided with two sets of buckets of a nominal capacity of 30 and 25 gallons (English), respectively. In the following table is exhibited the data regarding their capacity:

TABLE I.

		Weight, empty.	Weight of water when full.	Weight of water delivered by buckets.
		Pounds.	*Pounds.*	*Pounds.*
30-gallon buckets	No. 1	101	296	280
	No. 2	98	295	280
25-gallon buckets	No. 1	68	245	230
	No. 2	67	237	230

The capacity of the lift was tested by working it with a single Nellore bullock weighing 1,146 pounds on three different days, but owing to insufficiency of water in the well no test could be continued for more than two hours forty-two minutes. In Table II the results obtained are shown:

TABLE II.

Date.	Time started.	Time stopped.	Number of buckets raised.	Lift at beginning.	Lift at end.	Mean lift.	Gallons raised per hour.	Foot-pounds of useful work done.
				Ft. in.	*Ft. in.*			
July 16, 1895	8-37	9-37	*a* 88	22 0		22.625	2,594	586,900
	9-37	10-07	*a* 51		23 3			
	2-04	3-04	*a* 91	21 10½				
	3-04	4-04	*a* 86			23.24	2,489	575,500
	4-04	4-46	*a* 63					
July 17, 1895	2-00	3-00	*a* 85	22 2		22.87	2,391	546,900
	3-00	3-51	*a* 73		23 6½			
July 18, 1895	2-29	4-29	*b* 167	22 2		22.75	1,886	429,060
	4-29	5-29	*b* 79		23 4			

a 30-gallon buckets. *b* 25-gallon buckets.

The mechanical efficiency of the lift when just moving is 83.6 per cent, and at the ordinary working speed, 79 per cent. It was found that the speed at which the bullock walked when exerting a draft of 92 pounds was, as the mean of a number of observations, 3.64 feet per second, and that in 162 minutes he raised 240 buckets of water, lifting each bucket 23 feet. Walking at this speed without stopping, which without doubt the animal could have easily done, he could have lifted 401 buckets, so that he was only usefully employed for 59.7 per cent of the time. The absolute efficiency of the lift was therefore 79 by 59.7 per cent or 47.2 per cent. This calculation neglects to take into account the extra pull which is necessary to tilt the bucket to make it discharge, which was found to amount to 122 pounds and which was exerted through about 3 feet. This quantity would only affect the result very slightly and the decrease in efficiency would diminish as the height to which the water has to be raised increases.

A device of this kind is among the possibilities of home manufacture in any American community. The results given were obtained from actual trial and could be duplicated here by the use of horses.

A device called an improved single mhote was also tested and described. A water carrier holding 31 gallons was provided, at its bottom opening, with a large leather pipe. This carrier or bucket was suspended by a rope from a pulley, and the free end of the leather discharge pipe was separately suspended and held up by a second rope passing over a pulley below the bucket pulley at the ground surface. The discharge end of the leather pipe was ordinarily held above the bucket, and in this position permitted no discharge. Upon the ascent of the bucket and its upturned discharge pipe or leather tube the tube would be drawn out along the ground by its controlling rope and discharge effected as soon as the bucket passed above the ground surface. The two ropes were connected in such manner that they were operated together, the pull being exerted by bullocks descending an inclined plane which had been excavated below the ground level.

The improvement on the ordinary single mhote is effected by attaching a rope to the draft rope and carrying it on to the large drum of a kind of windlass erected at the end of the inclined plane and at a considerable height above the level of the end of the run. Cords wound round two smaller drums, one on each side of the large drum, carry weights, which it was found almost balanced the weight of the empty bucket, so that at the end of a lift, as soon as the bucket was empty, the draft rope automatically rose in the air and the bullocks were able to turn round and walk up the inclined plane in a natural, easy manner instead of being forced backward, as is the common plan. The improvement effected is undoubtedly a very great one, as not only is the weight of the empty bucket practically balanced, but the animals are also spared the cramped and unnatural backward walk up a steep incline, which probably tires them more than their exertions in drawing the bucket out of the well. The experiments made on this lift were not very extensive, but the following results were obtained and are worthy of record. The mhote was worked by two bullocks weighing 732 pounds and 616 pounds, respectively, or in the aggregate 1,348 pounds. The bucket, which was of iron and fitted with a leathern discharge trunk, weighed 43 pounds and when full held 31 gallons of water, but the mean quantity lifted, as measured into a tank, was 24.2 gallons per lift, the rest being spilt or lost by leakage. With the bullocks employed the rate of working was 90 lifts per hour, and the height of the lift being 23 feet, the total quantity of work usefully done amounted to 500,940 foot-pounds per hour. The draft exerted by the bullocks down the inclined plane was found to be 383 pounds. The useful work done in a single lift was 24 × 10 × 23 or 5,570 foot-pounds, whilst the bullocks exerted a pull of 383 pounds through 25½ feet, the bucket having to be raised an extra 2½ feet to enable it to discharge its contents, and the work done is equal to 9,760 foot-pounds. The bullocks then had to return up a gradient of 1 in 5.28 feet, in doing which they expended 6,510 foot-pounds of energy in lifting their own weight against the action of gravity. The total amount of work done by them in a single lift was, therefore, 16,570 foot-pounds and the useful outturn 5,570 foot-pounds, so that the efficiency of this method of lifting water is not greater than 34.3 per cent. As compared with the ordinary single mhote, the great advantage which this lift possesses is that it allows the bullocks to turn round at the bottom of the inclined plane and ascend walking forward instead of backward, an advantage which it would be difficult to express numerically, but the balancing of the bucket diminishes the draft by about 40 pounds and increases the efficiency of the lift by about 6.3 per cent, not, perhaps, a very large amount, but still by no means a negligible quantity.

A see-saw lift described is unique, and might be developed into a practical device:

In this form of water lift the bullock is made to walk along a platform supported on a roller, and by his weight it is caused to oscillate up and down. Two ropes are attached to one end of the platform and wound round two small drums forming part of a windlass, round the large drum of which a rope working an ordinary single mhote is passed. The platform is not supported in the middle, but at some distance therefrom, so that the working end of the platform greatly preponderates and the bullock has to walk to the free end of the platform to tilt the longer segment up and lower the bucket into the well. The platform is 24 feet long and the supporting roller is fixed 15 feet 3¾ inches from the working end. It was not possible to weigh the platform, and calculations of its weight, based upon the quantity of timber used in it, can only be approximate. It was, however, carefully measured up, and assuming that the teak wood, of which it was constructed, weighed 45 pounds per cubic foot, the weights of the two sections are 1,450 pounds and 850 pounds. To diminish the shock when the free end falls and the bucket is lowered into the water, 230 pounds of iron rails are fastened underneath the platform by a short chain, so that just before this end of the platform reaches its lowest position the rails rest on the ground and their weight ceases to act, and the platform comes to rest more gently than would be the case if the velocity of descent continued to accelerate to the very end. The ropes from the platform were wound round drums, the circumference of which was 3 feet 2½ inches as measured by unwinding one coil of the rope, and the mhote rope was worked from a drum 7 feet 10 inches in circumference so that the motion of the working end of the platform was multiplied 2.44 times. With the bucket empty and the platform horizontal the load at the free end could be varied from 160 pounds to 362 pounds without disturbing the equilibrium, whilst with a load of 247 pounds in the bucket, equal to 24.7 gallons of water, the platform remained horizontal, though the load at the working end varied between 584 pounds and 275 pounds. Taking the mean between the two extreme values to be the actual weight required to balance the platform, it is possible by taking moments about the center to determine the only force acting on the platform which was not measured, viz, the weight of the empty bucket and ropes acting with a lineage of 2.44 to 1. With the bucket unloaded, the weight works out as 65.4 pounds and when loaded 62 pounds, a remarkably close agreement.

The lift was worked during the trial by a bullock weighing 700 pounds and a man weighing 117 pounds. The rate of working was 81 lifts per hour from a well 18 feet 1 inch deep. The average quantity of water brought up by the bucket, as measured into a tank, was 23.5 gallons, and the useful work done per hour amounted to 344,210 foot-pounds. The bullock and the man together were much heavier than was really necessary, and they did not use the full length of the platform, so that it is difficult to estimate the work done by them in working the lift, but the mechanical efficiency of the lift on the day of trial can be ascertained by multiplying the fall of the front end of the platform by the force required to set it in steady motion when lifting a bucket full of water. The total height the bucket had to rise to discharge its contents was 22 feet, and the end of the platform therefore fell 9 feet and the work done was $584 \times 9 = 5,256$ foot-pounds. To raise the platform back to its initial position the free end then falls 5.18 feet and the load on it is 362 pounds, and the work done is equal to 1,875 foot-pounds. The total work, therefore, done in a single lift is 7,131 foot-pounds, and the useful work given to the water is 4,245 foot-pounds, so that the mechanical efficiency, when just working, is 59.6 per cent; at the normal rate of working it is much lower, probably not more than 50 per cent.

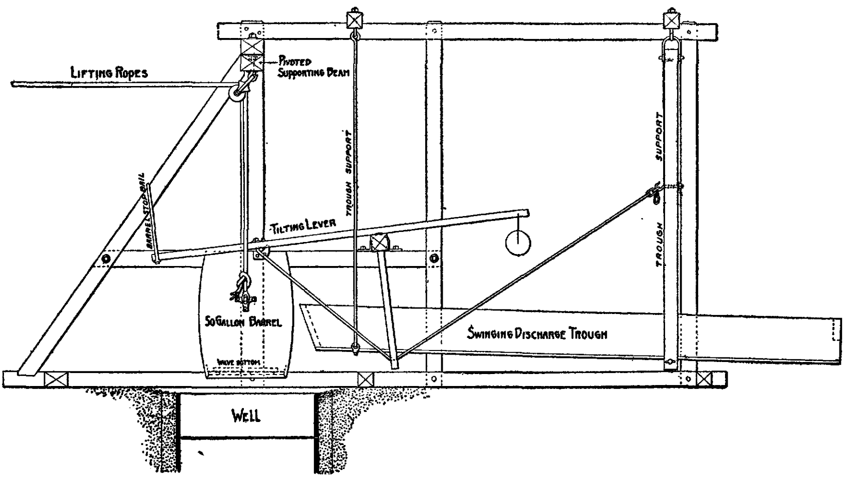


FIG. 57.—Construction of Boyce water lift.

The conclusion reached by the engineer, Mr. A. Chatterton, who made the tests, was that "the most efficient way of utilizing animal power is to make the animal raise himself against the action of gravity, and then, in some way, convert the potential energy stored in the animal's body into work."

AN AMERICAN DEVICE.

An American water lift, devised by Mr. A. Boyce, of Augusta, Oklahoma, was exhibited at Garden, Kansas, in 1896. Figs. 57, 58, and Pl. I, *B* (p. 74), show its construction and general mode of operation. Its very considerable merit lies in the readiness with which it may be constructed on the farm, and in its cheapness. The general plan of the device may be described as follows:

The running gear of an ordinary farm wagon, with the two wheels of one side removed, has its axles fixed radially to a circular track, the diameter of which is made to exceed slightly the total height of the water lift. The axles are bolted to beams which extend to the center of the circular track, where they are pivoted to a low post.

Horses hitched in the usual manner draw this two-wheeled device around the circle. A vertical post, rigidly braced on the wagon frame and extending to a height somewhat above the backs of the horses, carries at its upper end a vertical pin, to which ropes are attached by means of a loose link and extended to a lifting device at the well. As the wagon travels around the track, the framework, as a whole, acts as a crank, the throw of which is equal to the height of lift of the water. The ropes extend horizontally from the top of the traveling post to the well framework, so that the horses pass beneath them. By simple multiplication of ropes more than one well may be drawn upon. Where, however, there is but a single well, a counterbalance upon the wagon must be used to prevent overturning. In the well is a bucket, hung at either side on gimbals, attached a little above its center of gravity when full, so as barely to insure stability. In the bottom of the bucket two large clack valves admit of the ingress of water. The supporting ropes, one on each side, pass through pulley blocks over head and thence horizontally to the crank link on the wagon post. These pulley blocks hang from the ends of a short beam, pivoted at its center so that it may always maintain a position normal to the continually changing direction of the ropes.

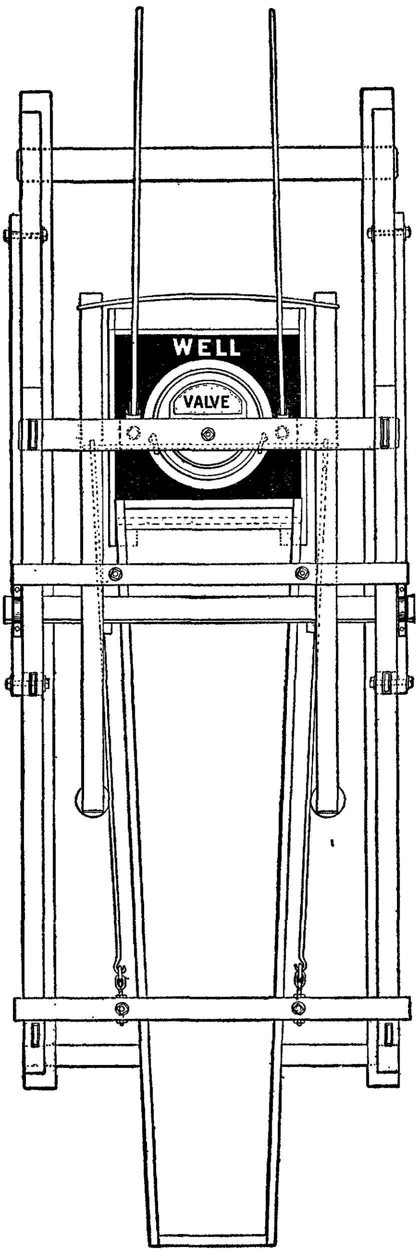


FIG. 58.—Construction of Boyce water lift.

The mechanism for emptying the bucket is designed to receive its full discharge and at the same time to avoid shock. A swinging trough is suspended at four points, by rods, from the framework overhead. As the bucket reaches the summit of its rise it encounters a cross rod, on a tilting lever, as shown in fig. 58. The tilting of this lever draws forward the trough supports by means of an adjustable connecting rod and swings one end of the trough under the tilting bucket. The bucket is prevented from tilting too far by a bail on the tilting lever.

This device is simple and very direct in operation. If applied to a single well, it has the fault of being single-acting, and horses can not be worked to their full capacity without overstrain. The resistances for one revolution would be, practically, as shown in the diagram, fig. 39 (p. 66). To render this work more nearly uniform three or more wells should be drawn upon at the same time. Three wells placed at the points of an equilateral triangle, with the sweep operating at its center, could be served nearly as easily as one. The mechanism is so direct acting that an efficiency of at least 80 per cent could be expected, as with the similar device of Mr. Stoney.

In the case of a single well the full weight of the loaded bucket will come upon the horses at that moment when their direction of pull falls in line with the rope. If a team of two horses can pull 400 pounds for a short distance, the weight of bucket and water may be 80 per cent of that amount. A 50-gallon oil barrel makes a good bucket. The barrel will weigh, when wet, about 70 pounds, and when three-fifths full will have the weight given above.

If three wells be used the resistance will be so uniform that an average pull of 200 pounds, for two horses working in a circle, may be taken as a basis for determining the proper dimensions for the buckets. If the machine has frictional resistances of but 20 per cent the water in each bucket may weigh 160 pounds, and in one revolution 480 pounds, or about 57 gallons, can be discharged. The number of revolutions per minute will depend upon the size of the circle described by the horses, and will be $\frac{57}{L}$ if the diameter of the circle is made equal to the height of the lift. The gallons discharged per minute will be $\frac{3313}{L}$, and the acre-feet, per day of ten hours, $\frac{6.1}{L}$. A mechanical advantage may be secured by attaching the crank post to the sweep at a point nearer to the center than the point to which the horses are attached. It will thus describe a smaller circle. In consequence larger buckets may be used.

This device is adaptable to any number of horses and to any number of wells. Mr. Boyce makes the statement that, "using two buckets drawing from one well, one bucket ascending as the other descends, I can raise 200 gallons per minute from a well 27 feet deep, with a small team of [two] horses." While this is heavy work for two horses, and probably could not be maintained continuously for ten hours, it shows what could be done on occasion. No accurate tests of efficiency have been made, so far as I know.

A device of this kind is well worth the attention of beginners in irrigation who wish to avoid a large initial outlay for machinery.

INDEX.