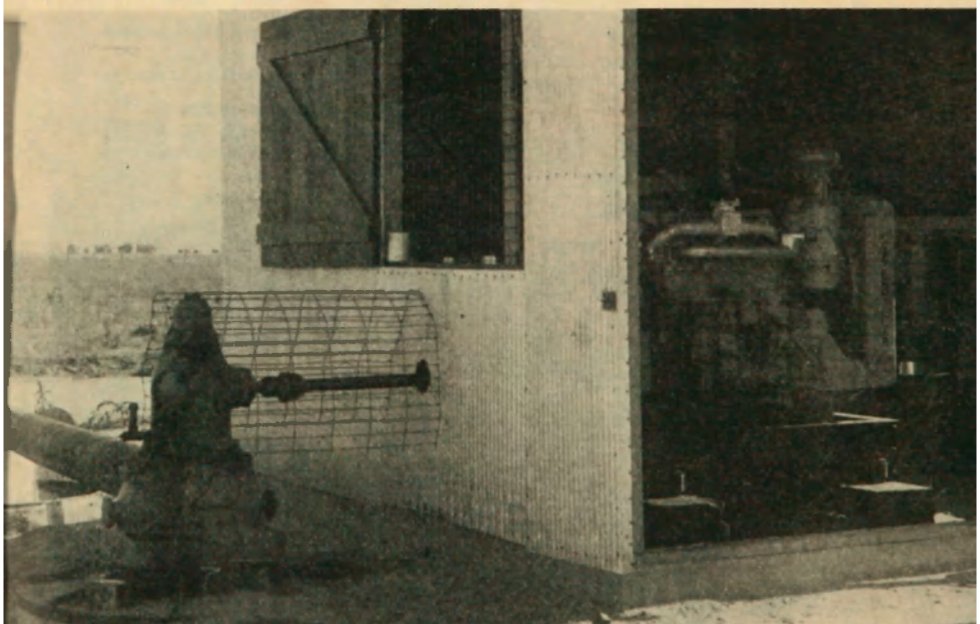
Bulletin 433 September 1936

Equipping a Small Irrigation Pumping Plant

W. E. CODE



Deep-Well Turbine Pump Driven by High-Speed Diesel Engine

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FORT COLLINS, COLORADO

CONTENTS

DEFINITIONS

Acre-foot—Volume equivalent to a depth of 1 foot over 1 acre; 43,560 cubic feet.

Acre-foot-foot—One acre-foot lifted 1 foot.

Brake horsepower—Actual horsepower developed at the shaft by the motor or engine; also input to a pump.

Drawdown—Vertical distance that water recedes in a well due to pumping.

Horsepower—Rate of work equal to 550 foot-pounds per second; 746 watts.

Kilowatt—Electrical unit of power; 1,000 watts; 1.34 horsepower.

Kilowatt-hour—One kilowatt for 1 hour; the electric meter indicates kilowatt-hours.

Over-all efficiency—Ratio of water horsepower to indicated horsepower of an engine or to input to a motor. Ordinarily the term is used in connection with electrically driven plants and includes the efficiency of motor, transmission, and pump.

Pump efficiency—The ratio of water horsepower to brake horsepower.

SUMMARY

THREE principal pump types are used in irrigation; the horizontal and vertical centrifugal, and the deep-well turbine. The horizontal pump is the least costly, but its use is limited by restrictions as to suction lift, and for this reason the vertical centrifugal or the deep-well turbine may be required. If the depth to water is over 25 feet, the horizontal centrifugal becomes inconvenient in operation. The vertical centrifugal has depth limitations because of the large pit required for its installation which the deep-well turbine does not require. The design of the deep-well turbine permits of its pumping from practically any depth.

Modern pumps of standard design have improved very greatly in efficiency during the last 15 years. To obtain the advantage of these high efficiencies the old methods of guessing at the well capacity and the lift must yield to engineering methods of actual well test before purchase of equipment. The characteristics of a centrifugal pump are such that high efficiency occurs only over a limited range of lift and discharge when operated at constant speed. Therefore pumps direct driven by electric motors must be closely fitted to the well characteristics.

It is generally conceded that electricity is the most satisfactory source of power. However, when the time use of a plant each season is short, it is probable that some type of cheap internal combustion engine will prove more economical. High-speed Diesel engines are rapidly entering the field as power sources for irrigation pumps. These engines must be used at least 1,500 hours per season in order that their high economy on low cost fuel can offset the high initial investment. Other kinds of power units available are the tractor and automobile engines and the slow-speed stationary engine, all of the electric-ignition type.

Completion of a plant in a business-like manner and on engineering principles is essential in obtaining maximum benefits. Outlets should be as low as possible to avoid pumping against excess head. Pipes should be of adequate size to avoid excessive loss of head by friction. Shelters should be provided to protect equipment, and provision should be made to prevent any damage to wells by the entrance of surface water.

The cost of pumping can never be very satisfactorily stated. The factors that influence it are numerous and variable. In many cases it has been found that the cost of pumped water compared favorably with that of gravity water. With ordinary farming conditions, lifts exceeding 40 feet should receive mature consideration, for in the past most of the failures have occurred with lifts greater than this. Pumps of high efficiency, low-cost power, and productive soil have, however, permitted the safe extension of this limit in many cases.

EQUIPPING A SMALL IRRIGATION PUMPING PLANT

Containing a Section on

COST OF PUMPING*

W. E. CODE**

THE economic feasibility of a pumping plant depends largely on the wise selection of the machinery with which it is to be equipped. Pumps, especially, should fit the condition for which they were intended. The determination of the pump size requires consideration not only of the capacity of the source but also of the use to which the water is to be put. Electric motors, engines, and connected piping should be of correct size for the economical operation of the plant. Only when all the component parts function at their best is it possible to pump water at the minimum cost. Pumping is not restricted to wells only, as there are numerous installations where a reservoir or canal is the source of water. In either case the selection of the proper type of pump bears an important relation to successful operation.

PUMP TYPES

The pumps in general use in irrigation work operate on the centrifugal principle. In this pump the velocity of the water leaving the impeller is transformed into pressure in the case surrounding the impeller. This case may be of the open or volute type, or may have diffusion vanes, in which case it is called a turbine. Probably the most common type is the horizontal centrifugal, but for use in wells it is rapidly yielding its former popularity to the deep-well turbine pump. The vertical axial-flow or propeller pump does not operate on the centrifugal principle, but lifts the water by vertical thrust. The vertical centrifugal, quite widely used 20 years ago, has nearly disappeared—the deep-well turbine taking its place. Other types used to a very limited extent include the rotary, plunger, bucket, and air-lift pumps.

Horizontal Centrifugal

This pump derives its name from the horizontal position of the shaft, and the nominal size is determined by the diameter of the discharge connection. The pump may be obtained from the 1-inch size, with a capacity of about 25 gallons per minute, to practically

*The edition of Bulletin 387, "Cost of Pumping for Irrigation in Colorado," being exhausted, it is necessary to include cost data in this bulletin. The subject of wells is treated in Bulletin 415, "Construction of Irrigation Wells in Colorado."

**The author wishes to express his appreciation for the valuable aid received from M. R. Lewis and C. H. Rohwer of the U. S. Bureau of Agricultural Engineering in their helpful criticisms of the manuscript of this bulletin. Also thanks are extended to the several commercial companies for supplying information, criticisms, and illustrations.

any size desired. It may be either direct connected or belted, is ordinarily simple in design, and in most cases is the least costly of any pump per unit of capacity. Because of its simple design, ruggedness, and the ease with which adjustments and repairs may be made, it is the first one to be considered for pump irrigation.

There are two main types of horizontal-centrifugal pumps—the side or single-suction, and the double-suction. The former is the more common and the less expensive one used in irrigation. The

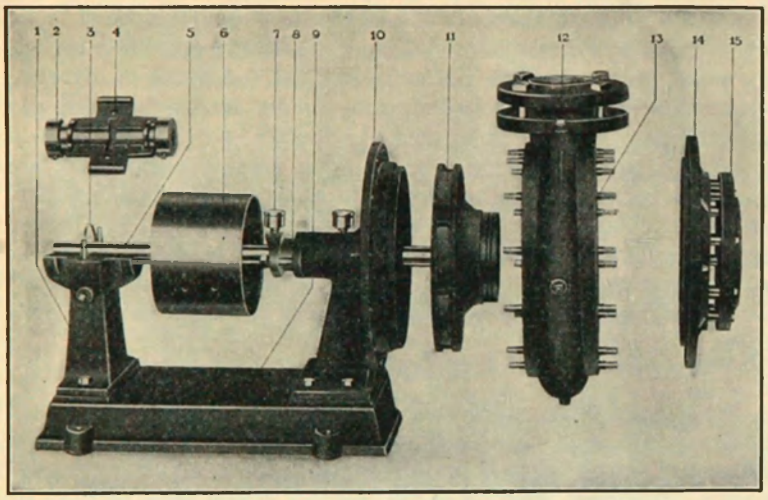

Figure 1.—Horizontal centrifugal pump with enclosed impeller: *1*, pedestal bearing; *2*, thrust collar; *3*, oil ring; *4*, pedestal-bearing cap; *5*, shaft; *6*, pulley; *7*, grease cup; *8*, stuffing-box gland; *9*, base plate; *10*, inboard head; *11*, impeller; *12*, discharge flange; *13*, volute; *14*, outboard head; *15*, suction flange.

latter is generally better built, has a higher efficiency, and is of greater cost. Recently there has been much improvement in design and efficiency of the side-suction pump, and since it is of simpler design, eliminating one packing gland, practice may tend more to this less-costly type in high-class installations. The side-suction pump may be equipped with one of three types of impellers: open, open on one side, and enclosed. Figure 1 shows a side-suction pump equipped with an enclosed impeller. An open and a semi-open impeller are shown in figure 2. The enclosed and semi-open types are to be preferred to the open impeller.

Ordinarily the enclosed impeller will show the highest efficiency because of the lesser clearance between the parts separating the pressure and suction chambers. Efficiency drops when this clearance increases, allowing water to slip by, and many pumps are therefore equipped with wearing rings which may be replaced when the clearance becomes excessive. Semi-open and open-impeller pumps have more slippage when new, which increases as the vanes wear. Pumps with semi-open impellers should have clearance

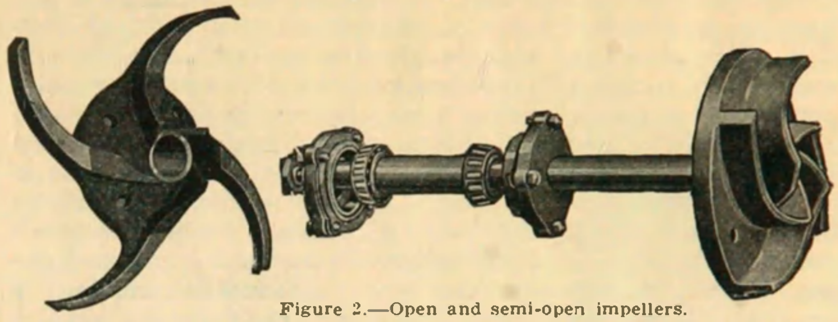

Figure 2.—Open and semi-open impellers.

adjustment. With water entering on both sides of the impeller, the double-suction pump is hydraulically balanced; but with the single-suction pump, pressure builds up behind the impeller hub, and bal-

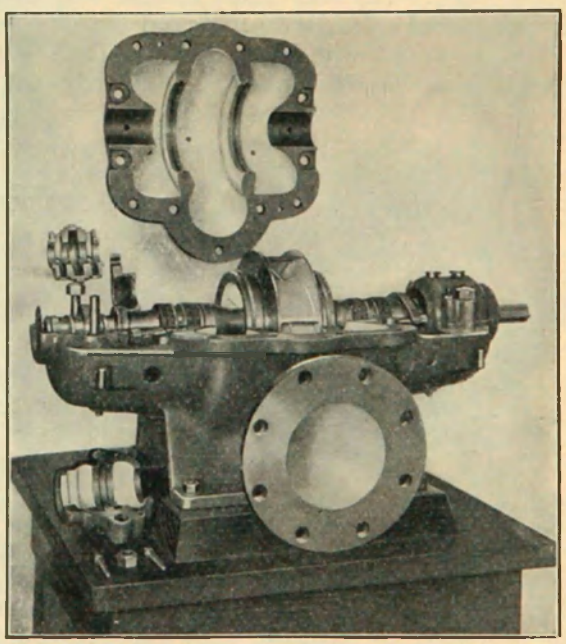

Figure 3.—Horizontally-split, double suction, horizontal centrifugal pump.

ance must be obtained by relieving this pressure. In its simplest form, balance is fairly successfully obtained by means of a hole through the impeller near the hub, admitting suction conditions to the opposite side. Since this balance is never perfect under all conditions, the shaft must carry a thrust bearing.

Some of the newer pumps are built without attempting to balance the impeller hydraulically, the thrust all being taken by bearings. This allows the packing gland to operate under pressure rather than suction. Thrust bearings are also necessary on a double-suction pump because it may become unbalanced due to improper pipe connections to the pump, obstructions lodging in one side of the impeller, or any condition that will allow one side to pump more water than the other. The double-suction pump, shown in figure 3, is usually built with a horizontally-split case which permits of impeller inspection without the removal of attached piping. Figure 4 illustrates a recent type of side-suction pump built in small sizes in which the motor shaft also carries the pump impeller, forming a very compact unit.

Better-class pumps are equipped with shafts enclosed by bronze sleeves which protect the shaft from rusting and preserve smoothness at the packing gland. The shaft must remain smooth with either a grease or water seal in order to maintain the packing in good condition. If not equipped with bronze sleeves, the shaft should be of stainless steel, which is much less subject to corrosion than plain steel.

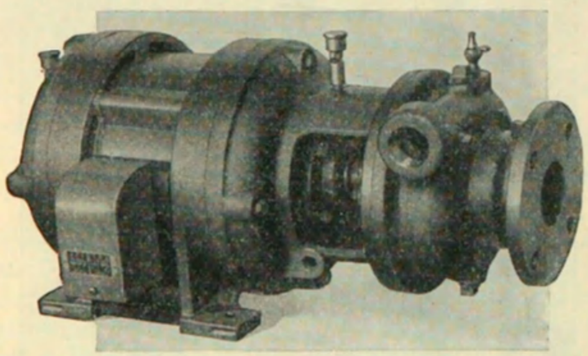

Figure 4.—Horizontal centrifugal pump direct connected to electric motor; impeller and rotor on same shaft.

The fact that the horizontal centrifugal pump must be placed above the water surface from which it draws its supply introduces the limitation of suction lift. Most manufacturers have a base guarantee on the performance of their pumps with a 15-foot suction lift at sea level, although it may be easily possible for them to operate at a 25-foot suction lift, but at less efficiency. As altitude is gained the permissible suction lift decreases, so that at an elevation of 5,000 feet the corresponding permissible lift is 5.6 feet less than at sea level. The allowable suction lift is affected by size, design, capacity, and speed of the pump, and by the total head; and it is important that the manufacturer be informed of the working conditions should the suction lift, including the friction, exceed 15 feet at an elevation of 5,000 feet. Although lifts of 22 to 23 feet have been observed in the vicinity of Fort Collins, it is far from a desirable operating condition from the efficiency standpoint, and not all designs of pumps can reach such a high point.

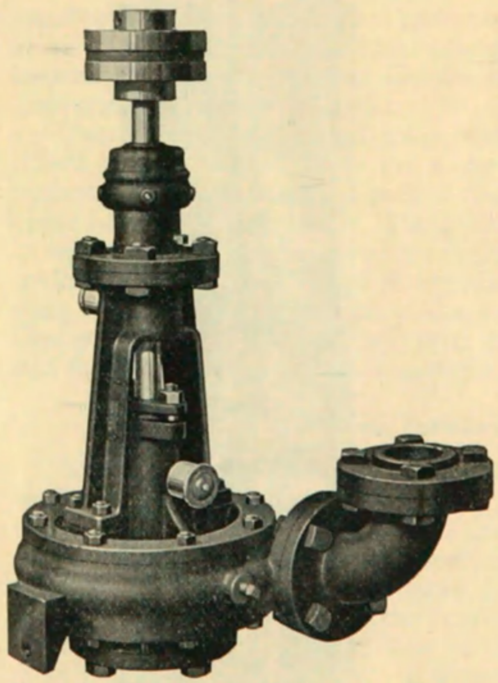

Figure 5.—Vertical centrifugal pump with open drive shaft.

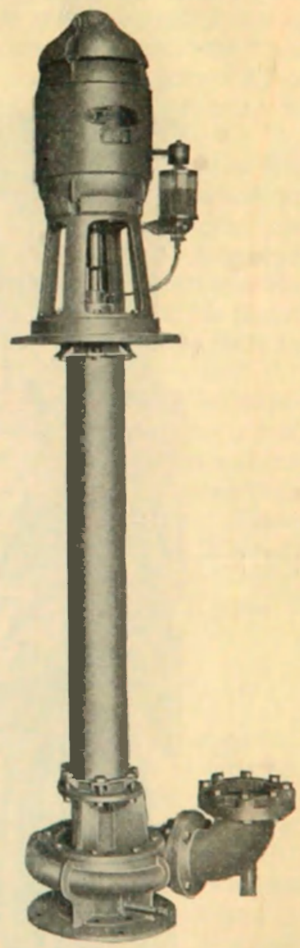

Figure 6. — Vertical centrifugal pump with enclosed drive shaft.

Vertical Centrifugal

The vertical centrifugal pump operates with the driving shaft set vertically. It has the advantages of being either direct- or belt-driven at the ground surface and may be set submerged to start without priming. It can be used in wells where there is a considerable fluctuation of the ground-water table, where the drawdown is excessive for the horizontal pump, and where the water table is at a

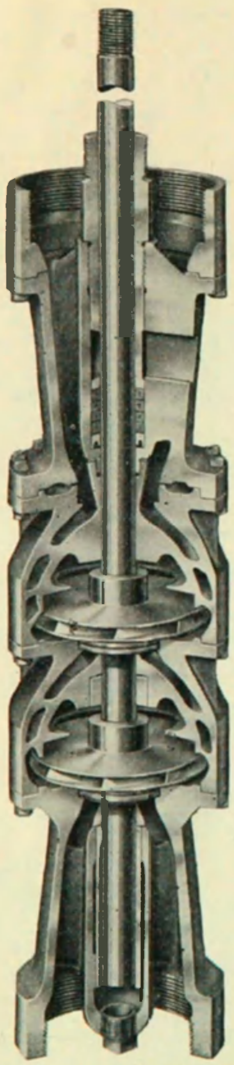

Figure 7.—Deep-well centrifugal turbine pump bowl assembly showing bottom bearing, impeller, intermediate bearing, upper packing-gland, drainport, and last shaft bearing.

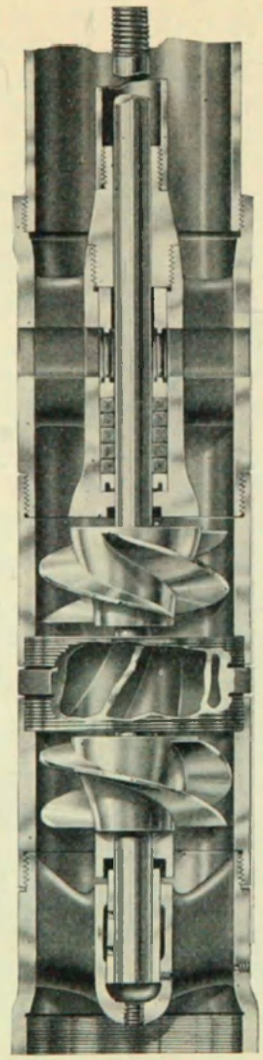

Figure 8. — Axial-flow, deep-well turbine pump bowl assembly; bearing details same as for centrifugal turbine.

depth too great for an inclined-belt drive. The pump is held in place in a steel or timber frame suspended from the top of the well, and the cross bracing carries the shaft-bearing supports at 4- or 5-foot intervals. The shaft may be enclosed in a steel tube, in which case the bearings are carried within the tube. Steel frames are to be preferred to timber because the latter usually become distorted through warping and loosening at joints. The distortion throws the shaft out of line; vibration and worn bearings ensue; and the results are a large power loss in the shaft, low efficiency, and a gradual wrecking of the frame. In more recent designs the pump is supported by a large pipe which obviates the use of a frame. The shaft may or may not be enclosed in a separate cover pipe.

One important caution to be observed is the adjustment of the impeller. It should not at any time be allowed to drag on the bottom part of the case; yet, with an open type of impeller, it should not have excessive clearance because of water slippage. A dragging impeller absorbs a great deal of power and may overload an electric motor to the danger point. The adjustment should be at the thrust bearing at the ground surface, and the operator should have definite instructions as to how it is made.

Two bad features of the usual vertical pump should be mentioned. When placed below the water surface the packing gland cannot be replaced without raising the pump, and tightening the gland bolts is not convenient. The second is that a pit large enough to admit the frame is required for the usual type of pump. There are a few makes of pumps on the market, constructed on a different plan, that do not require quite so large a pit as the ordinary type. One of these of 6-inch size will fit in a 24-inch casing. Figures 5 and 6 show types of vertical centrifugal pumps.

Deep-Well Turbine Pumps

Under this head are grouped all the types that are suspended by the discharge column within which the drive shaft is located. Properly the term should be applied only to those pumps operating on the centrifugal principle and having diffuser vanes in the case. All pumps grouped under this name do not have such vanes.

The deep-well turbine pump is a development of the vertical centrifugal so altered as to reduce the amount of space required in a well and to overcome depth limitations. It was first introduced about 1901. Two main types are recognized which are determined by the character of the flow produced by the impeller. One is the centrifugal type in which the water is discharged at right angles, or nearly so, to the axis of rotation. The other is the axial-flow type in which the water is given an upward thrust with an impeller whose blades are shaped like a ship's propeller. There are numerous modifications which are designed to take advantage of the better characteristics of both types which are called mixed-flow impellers.

The bowl (or bowls, if more than one stage is used) containing the impeller is placed below the water surface in the well at such a depth that there will be little or no suction lift when in operation. The shell of the bowl contains guide or diffuser vanes which receive the water from the impellers, transform most of the velocity head into pressure, and discharge it vertically into the discharge pipe or to the next bowl above. This may be seen in figures 7, 8, and 9.

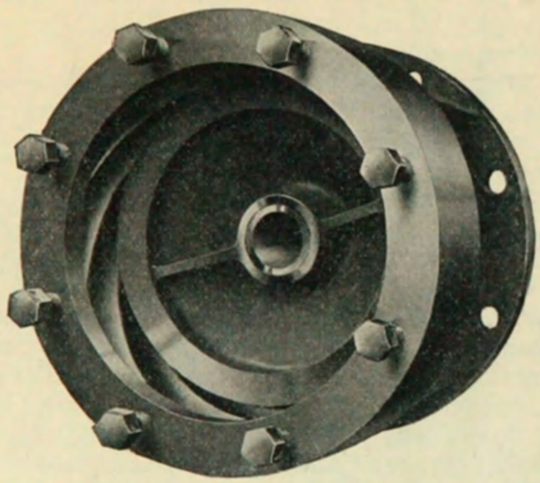

Figure 9.—Lower end of deep-well centrifugal turbine bowl showing diffuser vanes.

The drive shaft is located in the center of the discharge pipe and may or may not be enclosed in a cover pipe, depending on the lubrication system.

When water lubricated, a cover pipe is not essential; but one is required when oil is the lubricant and is preferred by many manufacturers when using water. When an open shaft is used it is held in line by rubber bearings or guides at 10-foot intervals. These bearings are centered by spiders at pipe joints. With a cover pipe, metal bearings are provided at from 3- to 6-foot intervals which are lubricated by means of a drip oiler supplied from an oil reservoir at or in the discharge head. On electric drives oil may be fed from a separate reservoir through a needle valve opened by the current passing through a solenoid; but in many of the newest models the reservoir is contained in the head, and oil is fed automatically only when the pump is in operation. Other oil reservoirs in the head serve to lubricate both the motor and thrust bearings. When the shaft is water lubricated a filtered supply is taken from the discharge and allowed to drip into the cover pipe. To prevent the tube from becoming full of oil or water, thus increasing rotating friction on the shaft, the lower end of the tube is provided with a gland* which allows leakage into a drainport. Into this same drainport the leakage from the packing,* where the shaft enters the bowl assembly, is released into the well. The drainport device prevents high water pressure from forcing oil from the tube and replacing it with water, and further allows water to escape that might enter the tube. No such device is necessary with an open shaft, as the condition requiring it does not arise.

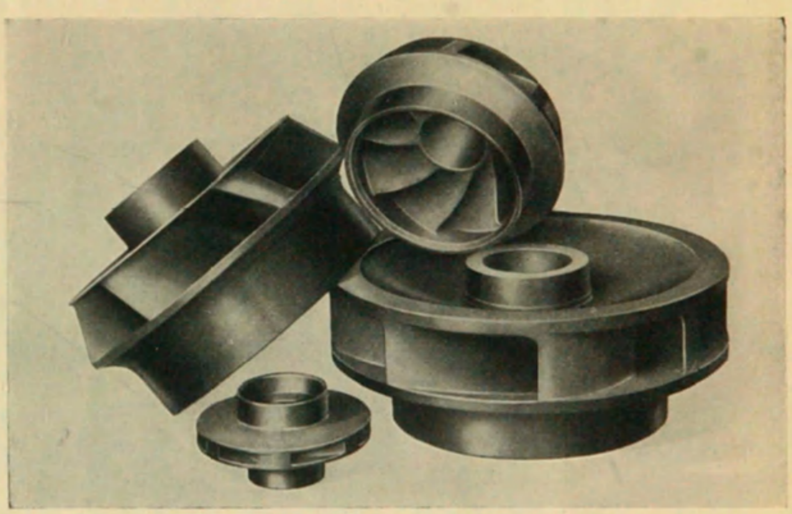

Figure 10.—Enclosed deep-well centrifugal turbine impellers: *upper two,* mixed flow; *lower two,* straight centrifugal types.

In the oil-lubricated pump the bottom bearing may be lubricated by a drip oiler feeding through a long tube from the surface or by grease held in the chamber enclosing the bearing. The former method is uncertain, and many manufacturers now use the latter with excellent results. A non-soluble hydraulic grease is used; and since the bearing is always cold, there is little or no loss of lubricant during many years of operation. Should all the lubricant be lost out of the bearing and become water lubricated, it will stay in satisfactory condition providing the shaft is of stainless steel.

Two bearings are required in the head, whether the pump be belt- or direct motor-driven. One, the thrust bearing, carries the vertical shaft load, the hydraulic down thrust, and some radial load. For the usual loads this is a ball bearing and may be located either

*The latest practice tends toward the use of long metal bearings without packing.

on top of the motor or below it. The other is a radial ball bearing, its location depending on the position of the thrust bearing. Nearly all deep-well turbine pump heads are equipped with a clutch or ratchet which prevents unscrewing the shaft joints, should power be applied accidentally in the wrong direction. The clutch permits reverse rotation; but should the shaft joints start to unscrew, the shaft lengthens and the clutch becomes disengaged. Reversal may easily happen to a pump driven by means of a quarter-turn belt and also through a mistake in the alteration of wiring, if electrically driven. Such a device is a valuable preventive against the possible wrecking of the pump. Back spinning, due to water passing through the

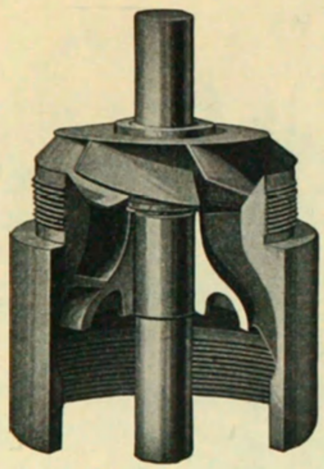

Figure 11. — Semi-open, mixed-flow type of deep-well turbine impeller. This type permits of capacity adjustment.

impellers in a reverse direction, does not tend to unscrew the shafting, as the force is applied in the direction that will tighten the joints.

Those pumps of the screw or propeller type in which the propellers are not all grouped at the bottom but are fastened to the shaft at regular intervals along its entire length present an entirely different problem in lubrication when the shaft is enclosed, as is the usual case. This type of construction is now little used. Numerous methods have been tried, none being quite so certain as in the uninterrupted cover pipe.

Impellers may be of bronze, cast iron, or porcelain-enameled cast iron. Cast iron is from two to three times as hard as bronze, and porcelain is four or five times as hard as cast iron. Bronze will not rust or pit as cast iron does, but as long as the protective coating of enamel remains on the iron the smoothest conditions exist, and a gain of 1 or 2 percent in efficiency over bronze is obtained. Impellers of the centrifugal and mixed-flow types may be enclosed or semi-open in design. These are shown in figures 10 and 11. The enclosed impeller usually gives the highest efficiency but is dependent on the maintenance of the close fit of the lower part of the impeller or skirt against the housing. As wear occurs leakage becomes greater, and consequently there is a small reduction in efficiency. Several manufacturers have developed a horizontal seat to overcome this objection, but experience is lacking at this time as to its success. Impellers without a bottom shroud (semi-open) may be adjusted at any time for a minimum clearance to maintain the efficiency. This feature may be utilized to reduce the pump capacity by raising the impeller and allowing more back slippage in case the well does not furnish an adequate supply, but in so doing there is an accompanying loss in efficiency. It is also useful in reducing capacity when a well is pumped and cleared of sand for the first time.

Deep-well turbine pumps equipped with a drainport, as mentioned on page 13, are subject to another source of deterioration in efficiency at the packing gland. As the packing wears down, leakage which is discharged back into the well increases. Stainless-steel shafting should be used through the pump bowls, because bearings and packing will last longer when the shaft does not become rusty. Stainless steel shafts should be used throughout in water-lubricated pumps using a cover pipe. Where no cover pipe is used non-corrosive steel sleeves should be used at the rubber bearings.

Other Types of Pumps

There are many forms of rotary pumps which displace water through a rubbing or rolling contact motion and are well adapted to certain uses. They are not recommended for irrigation use, as the water usually carries grit which causes rapid wear on the closely fitting parts, with an accompanying rapid loss of efficiency and ability to be self-priming.

Bucket pumps may be useful for short lifts from ditches or lakes where a constant water level is maintained, because they are not affected by small trash in the water. They require too large a pit into the water to be suitable for wells, and the drawdown is limited to the depth to which a pit can be sunk.

Air-lift pumping requires a depth of water in the well while operating equal to or greater than the total lift; hence it is not practicable in a shallow well. Air under pressure in a separate pipe is liberated near the bottom of the discharge pipe; and the many small bubbles, reducing the density of the water and ascending rapidly, produce a flow at the ground surface. The efficiency of the air lift is always low, seldom exceeding 40 percent; and for this reason its use is limited to the developing of wells and to small diameter wells, where its ability to pump larger quantities than other pumps is important.

Plunger pumps are of too low capacity to be considered practicable for irrigation, except for small gardens where from 5 to 20 gallons per minute may be required. Windmills can be used very advantageously for this purpose when a large-capacity tank is provided for storage. In this manner a large stream can be provided for a short time when irrigating.

ADAPTABILITY OF PUMP TYPES

The horizontal-centrifugal pump is well adapted for pumping from wells when the depth to water does not exceed 25 feet, and the drawdown plus the distance of the pump above the water table does not exceed 15 feet. Although this pump can be placed at any depth below the ground surface in a pit when direct connected to an electric motor, it becomes inconvenient in operation at depths exceeding 25 feet. At depths of 20 feet or less it can be belt-driven from an engine or motor at the ground surface. It lends itself readily to pumping from a battery of wells, because it can be connected to a common suction pipe leading to all the wells.

The horizontal pump is not very well adapted to conditions of a widely fluctuating water table because of the necessity of placing the pump above the highest point to which water may rise. This limits the extent of drawdown in the well because of the suction-lift limitations. The operating conditions of a considerable suction lift and very little discharge lift adversely affect the pump efficiency. This frequently occurs when the water table is within a few feet of the ground surface.

Unless the range in stage is too great, the horizontal pump is well suited to pumping from reservoirs, rivers, and canals. Change in stage may be met with an arrangement of alternative foundations, so that the pump may be placed to operate without an excessive suction.

The conditions for the use of the vertical centrifugal in pumping from wells are: first, where the water table exists at a greater depth than 25 feet, and it becomes desirable to drive with a pulley at the ground surface; second, where the drawdown in the well is excessive; third, where the ground-water table fluctuates considerably. These pumps do not need priming when submerged but require a pit for their installation. They may be used to advantage also on reservoirs or rivers subject to a fluctuating stage, as the point of drive can be placed high enough to prevent its becoming submerged.

The deep-well turbine pump, with its various impeller types, has an almost universal application. It first came into use in Colorado for pumping from wells about 1926, and since that time it has far outnumbered all other kinds of pumps in new and replacement installations. Because of the small diameter it can be installed in drilled wells whose diameter needs to be only slightly greater than that of the bowl diameter. By adding more bowls, changing the type of impeller, or increasing the speed, the capacity for a certain size can be greatly varied and still maintain good efficiency. For lifts of from 5 to 15 feet the propeller pump may be more efficient or better adapted in that it would operate at a higher speed than a centrifugal.

There is practically no limit to the drawdown obtainable in a well so far as the deep-well turbine pump is concerned, as it is quite possible to pump from a depth of 500 feet. Fluctuations in the water table affect only the pump-operating conditions. Due to the keen competition the deep-well turbine has been continually improved, both in efficiency and dependability, and has acquired a remarkably long, trouble-free life. When electrically driven by a direct-connected vertical motor the turbine becomes the most compact and neatest type of pumping plant. Nothing is simpler in operation, as only the push of a button is required to start or stop it after the switch is closed.

CARE OF PUMPS

Neglect of any machinery results in shorter life. Housing for protection from moisture and dust is important. Some motors for turbines are built weatherproof, and such plants may be left in the open when all bearing housings and air ducts are protected from rain and dust. The electric starting equipment, however, must be placed in a rainproof shelter of some sort.

The packing gland of a horizontal centrifugal requires periodic attention. It should never be drawn up with heavy wrenches, as only light pressure should be exerted on the nuts. When unnecessarily tight much extra power is consumed; the shaft or sleeve may become scored, and it may run hot if grease only is used for a seal. When the gland still leaks with moderate tightening new packing of proper size and character should be inserted. Only a thoroughly graphite-lubricated, high-grade cotton or flax square packing of the right size should be used. When the stuffing box is operating under suction and is water sealed, a lantern gland is used between two separate rows of packing, and care should be taken to place the proper depth of packing at the pump end, so that when the gland is tightened the water tube from the pump enters opposite the lantern gland. A small amount of leakage from the gland is to be desired for lubrication rather than to have it so tight as to shut off such leakage.

When a shaft or sleeve becomes rough from rust or from sand carried in the water, or from improper tightening of the gland, a packing gland will not stay in good order for many hours at a time. The same is true if the shaft is not straight, or the bearings are worn. Rough shafts may be removed and made smaller by turning on a lathe, but crooked shafts should be discarded. Water containing sand should not be used for sealing. It is better to use grease only, in such a case, or provide some means for a clear water supply.

Pumps equipped with sleeve bearings should use a lubricating oil whose viscosity is about S. A. E. 20 or 30. Ball bearings for speeds less than 1,800 revolutions per minute should be packed with a soft grease unless designed for oil. Roller bearings are satisfactory for the slower speeds and require the same type of grease lubrication as ball bearings. Dust and grit are potent enemies of bearings. Oil vents are provided with dust caps which not only should be kept closed but clean, so that dirt will not fall into the bearing when the cap is opened. All pump casings are provided with a drain plug at the lowest point, and it is very essential that this be removed in the fall and left out until operations are resumed in the spring.

Should it be observed that a pump is not up to its normal capacity, one or more of four things can be checked without much trouble. They are: the speed, the direction of rotation, increase in lift and the possibility of a partially clogged impeller. If the suction pipe is equipped with a screen the last-named condition is eliminated, but the screen may be found partially clogged.

Shaft bearings of the vertical centrifugal are best lubricated through small pipes by drip oilers at the ground surface, one for each bearing. This is not only the safest method but the surest one, because the oil supply is conveniently regulated. The thrust bearing should have a dust-tight cover, as this type of installation usually is without shelter. About twice a year the impeller clearance should be checked at the thrust-bearing adjusting nut.

Since specific instructions on lubrication usually accompany the installation of a deep-well turbine pump, they should be followed precisely. Substitution of oils may be dangerous, since in some of the latest designs the same oil for lubricating the motor and thrust bearings may be used on the shaft bearings. Oil for the shaft bearings should be thin, about S. A. E. 10 being satisfactory for the low temperatures in the cover pipe. The rate of drip to the shaft bearing should be carefully watched, as too much oil in the cover pipe increases power consumption. The resistance is greater to a spinning shaft if it is immersed in oil than if each bearing were getting just the proper amount of lubricant. Usually the rate of drip is about two drops per minute.

Belt tension should be checked periodically. Too tight a belt throws unnecessary pressure on the bearings and absorbs power. Too loose a belt allows too much slippage. A belt should not be left in tension during periods of non-use, especially if exposed to the weather.

MOTIVE POWER

Internal Combustion Engines

These engines are of three main types: electrical ignition, the most common type; hot spot; and Diesel. All such engines are generally rated at sea level and are affected by elevation, losing about 3 percent in power for each 1,000 feet of altitude.

Electric Ignition

Under this heading come all engines in which combustion takes place by the ignition of an explosive mixture of air and oil vapor by an electric spark. In irrigation pumping nearly all the various kinds of these engines are in use. They include the stationary, tractor, and automobile engines, and are of the four-cycle type. The fuel economy of these engines will vary from 7 to 11 horsepower-hours per gallon of fuel when new, a fair average being between 8.5 and 9.5.

For continuous duty and long life the stationary engine is best suited, because it is of slow speed, and it can be purchased in the size of unit desired. It is usually the most economical to operate; and in sizes over 10 horsepower, kerosene and distillate can be used for fuel. They are easily available in sizes from 1 to 15 horsepower in single-cylinder units, and in more than one cylinder to larger capacities. The large-size engine of the past is seldom offered on the market, being superseded by other kinds. There is usually no difficulty starting these engines by hand when under 15 horsepower, but the larger ones require power starting equipment or the help of several men.

The various tractor manufacturers usually offer their engines as power units for stationary work. Usually they are of two- or four-cylinder construction and range upward from 15 horsepower. Some are designed to burn kerosene or distillate, but ordinarily gasoline is used because of less carbon deposit and less crank-case dilution. They are of higher speed than the previously described stationary type and consequently will not weigh as much per horsepower. Having definite ratings, which in many cases have been checked by the University of Nebraska, they can be selected to meet definite power requirements.

Occasionally new automobile engines are purchased for irrigation pumping, but ordinarily their use is limited to those discarded from automobiles. Much of their useful life has been used up, and they cannot be recommended where reliability and heavy duty are concerned. They can be purchased for very little, and nearly any blacksmith can conjure a means of attaching a pulley to the drive shaft behind the first universal joint. An additional cross member welded to the original chassis furnishes a bearing support for the

TABLE 1—*Horsepower ratings of new Buick, Chevrolet, and Ford engines**

Year	Revolutions per minute 900	1200	1500	1800
Buick				
1928—Standard	25	33	42	48
1928—Master	32	42	52	62
1929—Series 116	28	38	48	57
1929—Series 121-129	36	46	62	72
1930—Series 40	32	44	56	65
1930—Series 50, 60	39	55	66	77
1930—Marquette	24	33	42	48
1931—Series 8-50	27	36	44	53
1931—Series 8-60	33	45	57	68
1931—Series 8-80, 8-90	42	56	72	85
1932—Series 50	29	38	48	57
1932—Series 60	32	48	60	72
1932—Series 80, 90	44	60	76	88
1933—Series 50	28	38	49	59
1933—Series 60	34	48	60	72
1933—Series 80, 90	44	59	74	88
1934-35—Series 40, 50	28	39	49	60
1934-35—Series 60	36	48	62	74
1934-35—Series 90	44	61	76	90
1936—Series 60, 80, 90	39	54	68	82
1936—Series 40	30	41	52	63
Chevrolet				
1928	19	25	31	34
1929	21	28	34	40
1930-31	21	28	34	41
1932 Passenger	22	29	37	44
1932 Truck	22	30	36	44
1933 Standard	21	28	36	48
1933 Master	25	34	42	50
1933 Truck	25	33	41	48
1934 Standard	22	30	36	45
1934 Master	25	34	40	51
1934 Truck	25	33	41	48
1935 Standard	26	35	43	52
1935 Master	26	36	44	54
1935 Truck	26	34	43	51
Ford				
Model B 4-cylinder	24	32	40	45
Model 18 V8	22	30	38	44
Model 40 V8	24	33	41	49

*Buick and Ford ratings are with all accessories. Chevrolet ratings are without fan and generator, which absorb from 4 to 6 percent of the output at these speeds.

end of the shaft. Such a power unit is acceptable for short-season pumping or to permit getting by cheaply for one season.

One usual difficulty encountered with an automobile engine is the operating speed. Since the power output increases with the speed and is seldom known, the engine may be operating at a very uneconomical speed for the pump load. Often it is found that, because of improper pulley sizes, the speed is wrong for the pump to give the best efficiency. The greatest difficulty arises when the engine is direct connected to the pump as is frequently done. It is quite possible that an engine developing plenty of power at speeds used in an automobile will not develop sufficient power at the slower speeds at which pumps operate, therefore requiring the use of the intermediate gear. Table 1 shows the power rating of the engines of a few makes of cars for the past 7 years. It must be realized that there are new engine ratings and that old engines will have less power. In order to assure reliable operation it is recommended that automobile engines not be fully loaded.

Hot-Spot Engines

These engines obtain their power through the rapid burning of a heavy fuel oil which is injected in a solid stream at the proper time against a hot spot in the cylinder head. The hot element may or may not be removable, and in starting it receives its heat from some outside source, usually a blow torch. They are of the two-cycle principle and have a compression pressure considerably higher than that of the electric ignition type. At the end of the compression stroke the fuel is injected into the chamber containing the hot spot in a measured amount regulated by the governor, and combustion takes place as a rapid burning rather than as an explosion. Water or water vapor may be fed into the cylinder, which reduces knocking to a considerable extent; but, as it is detrimental to the cylinder walls, it is seldom used. Near the end of the power stroke the exhaust port in the cylinder wall is uncovered, and at the end of the stroke air, compressed in the crank case, enters another port, scavenging most of the burned gases.

Although some smoke usually comes from the exhaust of these engines, they often have been observed belching forth smoke in great volumes. This means very poor combustion, a waste of fuel, and a rapid accumulation of carbon around the exhaust port. They are sensitive to adjustments and require that mechanical parts be kept in good repair. When the engine is not in perfect adjustment some difficulty may be expected in hand starting and in getting it to rotate in the proper direction. Being of heavy construction and slow speed, usually about 300 revolutions per minute, the cost of this engine is considerably greater than that of the tractor type. The fuel economy is good when the engine is functioning properly and in good mechanical condition, being only slightly under that of the Diesel. For irrigation loads they may be obtained in one and two cylinders, from 15 to 75 horsepower, and are to be favorably considered when the pumping time exceeds 1,500 hours per season.

Diesel Engines

The slow-speed type, of which only two or three are now in use in irrigation in this state, is heavy and costly and must yield to the lighter, less costly, high-speed type. It is desirable, therefore, to discuss here only the high-speed type. Being developed primarily for use in trucks and tractors, these units so far have not much exceeded 100 horsepower in size. The slow-speed type, which is obtainable in both two and four-cycle, has a speed of 300 revolutions per minute or less, whereas the high-speed, so far built only in the four-cycle type, has speeds up to 1,200 revolutions per minute for continuous duty. Several of the larger manufacturers limit the speed for continuous operation to about 900 revolutions per minute. It should be appreciated in comparing engines and prices that the horsepower rating increases as the speed is increased. Commercial experimentation with this engine dates back 10 or 15 years and was limited to but one or two firms. In 1933 the number of manufacturers working on the design and construction of these engines greatly increased, and in 1934 a large number were placed on the market for all kinds of mobile and stationary work. By the end of 1935 some 30 units were installed in pumping plants in this state. It will be seen that these engines have not been proved by time as has the slow-speed type, and no good estimate can be had of their length of life on continuous service.

The principal feature in the development of the high-speed Diesel is in the design of the pre-combustion chamber, better fuel-metering devices, and the use of lighter reciprocating parts which have given the engine a greater flexibility and speed not attained by the old type. The pre-combusion chamber, both because of its shape and because it retains the heat of combustion, greatly facilitates the ignition of the fuel.

Because of the high-compression pressure of about 500 pounds per square inch these engines require special methods for starting. The small sizes, and in some makes up to 30 horsepower, can be started by hand cranking. This is accomplished with relief valves open until some momentum is gained. Starting on some engines is facilitated by the use of an electrically heated plug or a glowing wick. It is desirable that engines above 20 horsepower have a mechanical means of starting. At present there are four principal methods employed: (1) use of an electric motor and batteries, as in the automobile; (2) use of an auxiliary built-on gasoline engine; (3) operation as a gasoline engine for a short time and automatic conversion into a full Diesel operation without attention from the operator; and (4) use of compressed air. Electric starting should be of ample capacity to turn the engine over for a considerable period in anticipation of times when the engine will not start promptly.

Strict attention must be given to the recommendations of the manufacturers as to the grade of fuel oil to be used. Much trouble may arise, or poor fuel efficiency may result, if this admonition is not heeded. Diesel fuel oil should be a distillate containing enough of the higher-gravity fractions to furnish the very necessary lubricating qualities for fuel pump lubrication, yet none so heavy as to cause incomplete burning. If all the fuel does not ignite properly the exhaust will be smoky, a residue will collect on the piston, and undue wear of the cylinder will follow. For a satisfactory fuel for a high-speed Diesel the following specifications are a safe guide:

1. Flash point, minimum 150° F.
2. Viscosity, minimum 35 seconds (Saybolt Universal Viscosimeter at 100° F.).
3. Distillation: Not less than 10 percent should distill below 460° F., not less than 90 per cent should distill below 675° F.; recovery, 95 percent.
4. Sulphur, maximum 2 percent by weight.
5. Sediment and water, maximum 0.05 percent by volume.
6. Carbon residue, maximum 0.25 percent by weight.
7. Ash, maximum 0.01 percent by weight.
8. Pour point, to be specified if fuel is to be used at temperatures below freezing.

Such an oil will flow readily at temperatures above freezing. The important items are numbers 2, 3, and 5. High percentage of recovery means that most of the fuel will be properly burned and hence will be more economical, as less will be required. Great care should be exercised at all times to keep the fuel clean and free from water. To avoid sediment the supply pipe from the storage tank should take off above the bottom, and a drain-out should be provided at the lowest point. Only lubricating oil that meets the specifications and recommendations of the manufacturer should be used.

One very important point is the design of the cooling system. Whatever it is, it should be a closed system; that is, new water should not have to be added continually unless very pure. It must also be so designed that the temperature can be maintained at the point recommended by the manufacturer. Most of the waters in the eastern part of Colorado contain a goodly proportion of dissolved solids. As the temperature of the water rises these salts are deposited in the channels around the engine, practically insulating them and preventing the heat from being properly dissipated.

Some manufacturers supply and practically insist on a radiator and a fan as part of the equipment. Such a system of adequate capacity will provide cooling water at a fairly uniform temperature. However, radiators are costly, and a fan consumes from 1 to 3 percent of the power developed. A very good system is that of pumping the circulating water through a coil immersed in a cold water supply, the regulation being effected by a valve. The coil may be set in a barrel into which cold water is being pumped, or if conditions permit it may be set in the ditch or box into which the pump discharges. Figure 12 illustrates the use of a cooling coil immersed in the water discharged from the pump. By adjustments at valve B practically all the circulating water can be pumped directly into

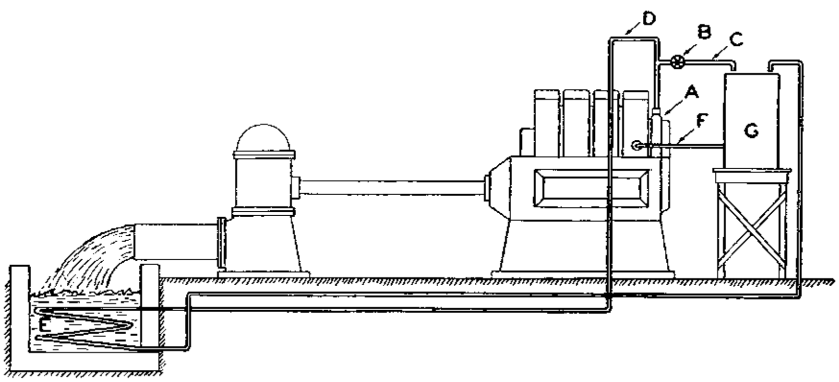

Figure 12.—Cooling system design in which the circulating water is cooled by means of a copper coil placed in the water being pumped: *A*, engine circulating pump; *B*, regulating valve; *C*, 1-inch pipe; *D*, three-quarter-inch pipe (not necessarily underground); *E*, 1-inch copper tube coil 20 to 25 feet long; *F*, inlet pipe to engine; *G*, reservoir of 30 to 50 gallons capacity.

the reservoir, or all through the cooling coil into the reservoir. Only one valve is required to accomplish this because of the added lift, smaller size, and length of pipe in the coil circuit. This system provides the very desirable condition of warm water being taken into the engine and a means of positive temperature regulation. A temperature rise of 20 to 40 degrees is much more desirable than a wide range produced by taking small quantities of very cool water into the engine. All such systems must contain a reservoir to provide for the usual small losses constantly occurring, and to insure an abundant supply of water in case of failure of the irrigation pump. The use of a cooling tower is also recommended, or for a small engine a single large reservoir connected with the circulating pump. It is well worth while in areas where hard waters prevail, to obtain water with a low salt content, even if this should mean hauling it considerable distance.

One of the greatest causes for rapid depreciation of an engine lies in the lack of proper and intelligent care. Such machinery deserves a good covering to keep out moisture and as much dust as possible. All parts should be kept in good repair, for one part out of adjustment may cause damage to others. Savings in operating cost by using cheap or improper lubricants usually prove to be costly economies in the end.

Electric Motors

Two types of motors, the three-phase, squirrel-cage, induction motor and the single-phase induction motor are used in irrigation pumping practically to the exclusion of all other types. The three-phase induction motor, usually for 220-volt current, is the simplest, least costly, most trouble-free, and longest-lived of electric motors and is available in convenient sizes used in pumping. Inherently it is limited to certain speeds according to the windings, those in ordinary use being of approximately 860, 1,160, and 1,760 revolutions per minute for 60-cycle current. Its efficiency will vary from 80 to 90 percent. There is little difference among standard makes as to quality and price, all being built to a more or less uniform standard. The slower-speed motors contain more copper and cost more than the higher speeds for the same horsepower. In the last 8 years these motors have been so made that a compensating starter is not required, but they start with full voltage, or "across-the-line." The size of across-the-line motors is limited by some companies to 50 horsepower, which is seldom reached or exceeded in irrigation pumping in Colorado.

The single-phase motor has an armature and brushes. The brushes require attention from time to time, as they wear out and may require adjustments. Single-phase motors cost more than the three-phase motors of the same size and speed, and because they produce unfavorable power-line conditions power companies usually limit their size to less than 10 horsepower. They should be avoided whenever possible.

No attempt should be made to operate a motor injured by lightning. Lightning striking at the plant may only burn out the transformers, but damage may also occur to starting devices and motor windings. The power company's representative or a competent electrician should be called in to inspect the equipment when such an accident happens.

A motor that has been submerged should not be operated until thoroughly dried out and the bearings carefully cleaned. Drying out may be accomplished in 2 or 3 days in a warm, dry room. The insulation in old motors may be so porous as to require a week's time in which to dry out.

To obtain greatest economy an electric motor should be fully loaded. Some manufacturers permit up to a 10-percent overload with their guarantee, and many such overloaded motors are in operation. More care is required in such cases to provide ample ventilation, as the rise in temperature of an overloaded motor may be excessive and detrimental to the insulation.

DRIVES

Four types of drive are to be found in irrigation work: direct, flat-belt, V-belt, and gears.

TABLE 2—*Maximum horsepower ratings per inch of width of medium quality rubber belting with 180° arc of contact*

No. of plies	Pulley diameter	Speed in feet per minute 1000	1500	2000	2500	3000	3500	4000	4500	5000
	Inches	*Horsepower*								
3	5	0.9	1.3	1.7	2.0	2.3	2.5	2.7	2.9	3.0
	6	1.1	1.6	2.0	2.4	2.8	3.1	3.3	3.5	3.7
	8	1.4	1.9	2.4	2.9	3.4	3.8	4.1	4.4	4.7
4	6	1.2	1.7	2.2	2.6	2.9	3.2	3.4	3.6	3.7
	8	1.5	2.1	2.8	3.3	3.8	4.2	4.5	4.8	5.0
	10	1.8	2.5	3.2	3.8	4.4	4.8	5.2	5.6	5.9
5	10	1.9	2.6	3.4	3.9	4.6	5.0	5.5	5.8	6.1
	12	2.1	2.9	3.9	4.6	5.3	5.9	6.4	6.9	7.2
	14	2.2	3.2	4.2	5.0	5.8	6.4	7.0	7.5	7.9
6	12	2.3	3.2	4.1	4.8	5.5	6.1	6.6	7.0	7.3
	14	2.5	3.4	4.5	5.3	6.2	6.8	7.4	8.0	8.4
	16	2.7	3.7	4.9	5.8	6.7	7.4	8.1	8.6	9.1

The direct drive requires but little discussion. There is no loss of power as with other drives, and hence it is the most desirable. It is limited, however, to those conditions where the speed of the driver is in accord with the prescribed speed for the pump. For horizontal centrifugals the drive is usually through a flexible coupling. On some smaller motor-driven pumps, however, the pump impeller and motor rotor are mounted on the same shaft. The flexible coupling is not intended to compensate for misalignment, except to a very slight degree, and every precaution should be observed to obtain perfect alignment between the pump and driver. In the recent turbine pumps the drive shaft passes through a hollow shaft in the motor, and the two are held together by a key.

The loss of power through flat-belt transmission varies greatly—5 to 20 percent, according to conditions. The points of loss are numerous but largely in slippage. Slippage may be caused by the belt being too loose, too stiff, or too narrow. Other causes of slippage are poor belting layouts such as small pulleys, pulley centers too close together, pulling from the top instead of the bottom, or the belt being placed at too steep a vertical angle. Too great a tension is another source of loss involving unnecessary pressure on the bearings. Quarter turns, half turns, and idler pulleys all add to the inefficiency of flat-belt drives. The relation of pulley size to belt thickness and horsepower capacity is important. When heavy belts are turned about a small diameter pulley the outer fabric fibers are greatly over-stressed, and the belt life is materially shortened. Power ratings are based on tensions when large pulleys are used and are reduced for small pulleys to compensate for increased flexing. Table 2 is given only as a guide for specifications of a medium-quality rubber, flat-belt drive, as the recommendations of a belt expert should be obtained when possible. A flat-belt transmission is shown in figure 13.

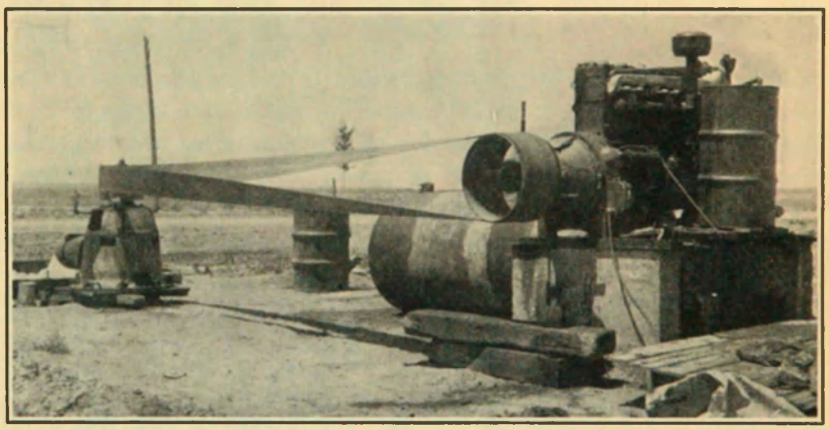

Figure 13.—Flat-belt transmission from an engine. The cooling system is that outlined in figure 12.

V-belts offer many advantages over flat belts. Their efficiency is higher, 95 percent or greater being claimed. Applications of V-belts are usually made by those who have a good understanding of their use, and when properly designed there is practically no slippage nor excessive wear on the belts. The manufacturer lists the correct pulleys and belts for maximum efficiency under nearly all conditions. The driver can be placed very close to the pump, allowing pit installations otherwise impossible, and a reduction of shelter space. They also may be used on a less costly, flat-faced, driven pulley when its diameter is great in proportion to the grooved drive pulley. A large number of belts on a quarter-turn drive is undesirable because the distance between pulley centers changes, and it is not possible for all belts to have the same tension. This condition is one of the reasons why very short pulley centers are not permissible on quarter-turn drives. Applications of V-belt drives are shown in figures 14 and 15.

Within the last few years right-angle geared heads for the turbine pump have been perfected and manufactured on a quantity basis, so that they cost but little more than V-belt or flat-belt drives. Their efficiency may be expected to be 95 percent or greater. The head containing the gears replaces the ordinary pulley head, and the drive shaft may be directly connected to an engine or motor. This drive shaft should always contain a flexible joint similar to the universal joint in an automobile drive shaft, to allow automatically for any differential settlement between the pump head and engine foundation and for any other misalignments. The cover illustration shows the application of a right-angle geared head.

PUMPING-PLANT DESIGN

In the planning of a pumping-plant layout two important considerations are to be kept in mind—efficiency and ultimate cost. Under efficiency such items as the pump, pipe sizes, elbows, tees, structures, pump location, kind of power, and drive are involved. Considerations of ultimate cost involve excellence in materials, and permanence. The length of time of use each season, and whether or

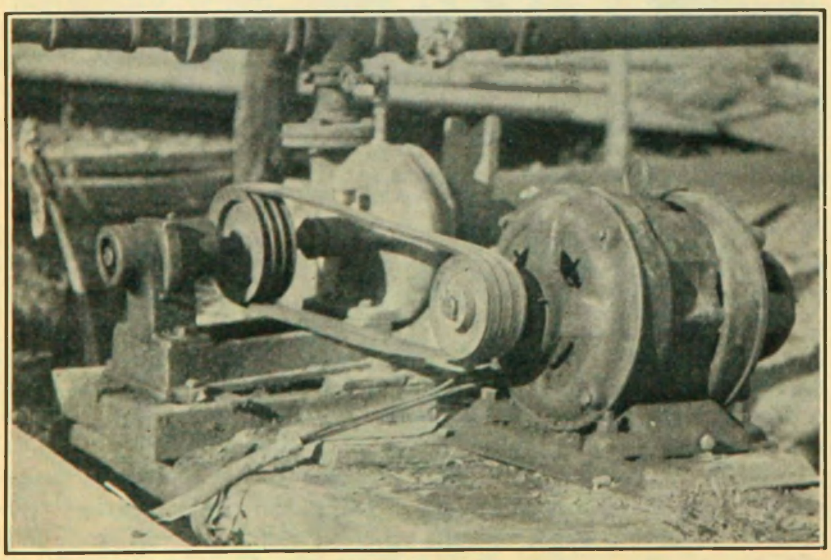

Figure 14.—V-belt drive for horizontal centrifugal pump.

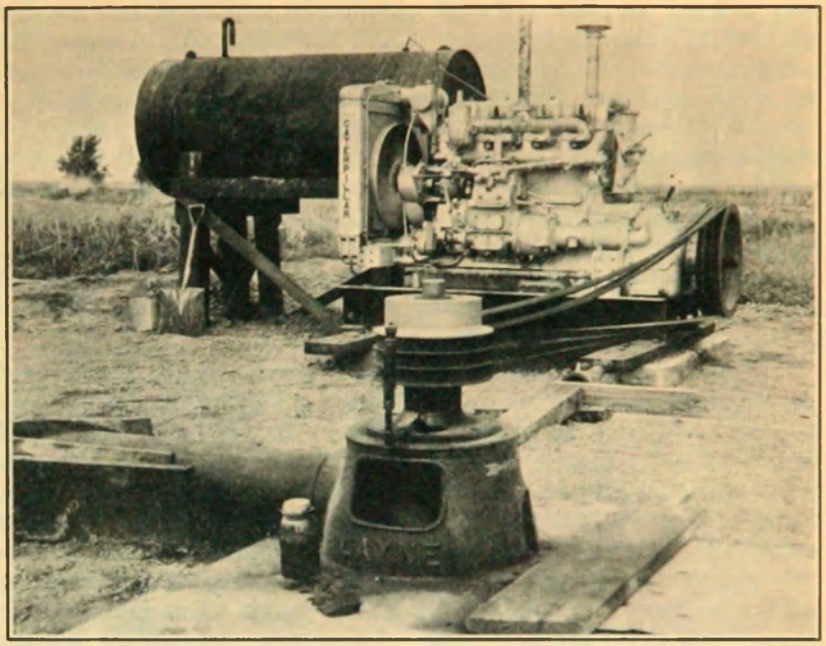

Figure 15.—V-belt drive from Diesel engine.

not the plant is used every season, have an important bearing on the initial investment in the plant. The more it is used, the greater the justification for high quality and highly efficient machinery.

Pump

The pumping conditions are usually the deciding factor in the selection of the pump type; however, other things may be given consideration, such as convenience in operation, and whether the plant is to be operated by a tenant or the owner should more than one type be adaptable.

The horizontal centrifugal pump, as previously mentioned, should be considered first. In a pit well it may be set on timbers inserted in slots cut or formed in the curb material. These timbers or steel beams should be much larger than is required for support of the load only, so as to avoid excessive bending and vibration and to allow for deterioration.

In pumping from rivers with moderately sloping banks the horizontal pump may be mounted on skids on sloping timbers, so that it could be removed quickly from the danger of floods and to facilitate the regular winter removal. With steep banks it may be necessary to build a foundation platform secured to piles driven into the river bed. Although the pump would not be injured by becoming submerged, but would require thorough cleaning of the bearings, drift driving against it might cause breakage. Power units must be set well above the danger line. Intakes should be protected by screens to prevent the entry of trash. A desirable type is that of a box opening towards the river. This opening should be protected by vertical iron or wooden bars slanting inward to facilitate removal of coarse trash with a rake. Behind this a removable screen with openings about 1 inch in diameter should be installed. The total screen opening should be greatly oversize to allow for partial clogging.

Priming may be effected in several ways. When the lift and the pipe line length do not exceed 40 feet, a foot valve of ample size may be used and the system flooded to above the pump from a tank or other outside source. If the foot valve does not leak, the pump will stay primed; but since most foot valves leak sometimes, and are therefore undesirable, provision must be made for priming water to be held in reserve. By means of a check valve, or preferably a gate valve above the pump, priming is effected by pumping out the air from the highest point on the pump. This method is to be preferred to the preceding one. A common pitcher pump is often used for exhausting the air, but a thresher tank pump is much to be preferred because of its greater capacity and easier action, and because it will stay primed. For priming a system of large capacity, power-driven pumps are frequently used. Probably the one best adapted is the rotary air exhauster, but plunger types may also be used. As these may operate at high speed when pumping air, a tank equipped with a gage glass should be installed in the line to permit observation of the position of the water level. The air pump is shut off as soon as the water appears in the gage glass, to prevent water from entering the air pump.

A pit of ordinary size will permit direct connection or V-belt drive from an electric motor. With ample ventilation in a big pit, a gasoline engine might be used if not located more than 5 feet below the top. Deep settings of gasoline engines are dangerous because of the possibility of explosions and leakage of exhaust gases. The safest location for an internal-combustion engine is at the ground surface, in which case a flat belt will be required. With small-diameter wells a pit is commonly dug and concrete lined to the high-water level, at which point the pump foundation is located. Should the depth to water in such case be great, a small pit can be sunk and an offset room constructed at the bottom for a direct-connected or V-belt electric drive. If the drive is from an engine or tractor, then a belt-way should be constructed to allow the belt to run at an angle of about 45°. A vertical belt is to be avoided because of the tension required to prevent excessive slippage. Pump pits should be lined with concrete or bricks, preferably the former. Lumber is not desirable because of its short life in contact with earth. Tightness is required because burrowing animals may leave channels through which irrigation or flood waters may enter the pit and well.

The vertical centrifugal pump is nearly always hung from timbers or steel beams placed across the top of the well. These should always be heavy and well supported. The pull of the belt is parallel

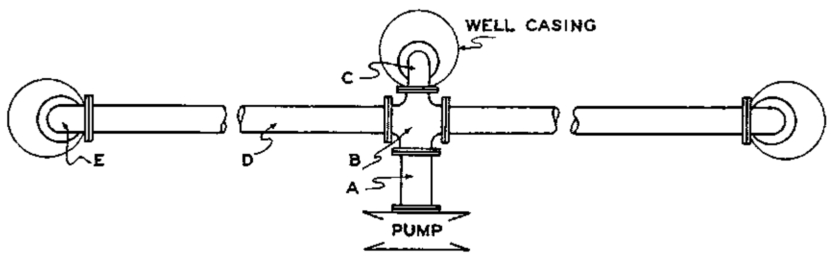

Figure 16.—Suction-pipe layout for battery of three wells, with sizes for various capacities.

Total Capacity	*Recommended Size of Parts*
450 g. p. m.	*C*, 3½ inches, drop pipe 3½ inches; *D*, 4 inches; *E*, 4 inches, drop pipe 5 inches.
600 g. p. m.	*C*, 4 inches, drop pipe 4 inches; *D*, 5 inches; *E*, 5 inches, drop pipe 5 inches.
900 g. p. m.	*C*, 5 inches, drop pipe 5 inches; *D*, 6 inches; *E*, 6 inches, drop pipe 5 inches.
1,200 g. p. m.	*C*, 5 inches, drop pipe 5 inches; *D*, 6 inches; *E*, 6 inches, drop pipe 6 inches.

A cross with the desired outlets is not always available; hence reducing flanges, or if threaded pipe is used, bushings may be required at *B* which should be the same size as the pump inlet. The nipple *A* is especially required for a double-suction horizontal centrifugal pump. Long radius elbows, tapered if necessary, at the drops into the wells are to be preferred because of the lesser friction loss and must be flanged. With the capacities and sizes named, the friction should not exceed 1.8 feet, and the draft will be approximately equal on each well.

to the timbers, and the top part of the frame work is braced to them to resist the pull. The frame may be made up of double planking, cross-braced to increase rigidity, and with horizontal bracing to carry the shaft bearings. Single planking is sometimes used when the frame length does not exceed a board length, but this is not recommended. Every effort should be made to make the frame rigid to prevent vibration. It should not rest on the well bottom, as this will cause it to get out of shape more quickly than when hung from the top. In addition to the bracing above to the foundation timbers, it is well to insert similar bracing from them below to the frame. Steel is far superior to wood for pump frames, as will be recognized in examining the condition of a wood frame that has been in use a number of years and noting the excessive vibration. Bearings on open shafting should be placed not more than 5 feet apart and lubricated through pipes equipped with drip oilers, or pressure-grease connections at the ground surface. Enclosed shafts are to be preferred, as the tube gives the shaft better alignment and facilitates oiling. The thrust bearing should be at the top for ease in adjusting, especially if there is danger of its becoming submerged should it be located just above the pump.

A deep-well turbine pump is usually set on the well casing or on steel beams or special covers spanning the top. If there is any danger of the casing moving after pumping starts, a separate con-

TABLE 3—*Loss of head in feet due to friction, per hundred feet of pipe**

Gallons per minute	Cubic feet per second	Inside diameter of pipe in inches								
		3	4	5	6	8	10	12	14	16
50	0.111	0.74	0.18	0.06						
75	0.167	1.60	0.39	0.13						
100	0.223	2.77	0.68	0.23	0.09					
150	0.334	5.99	1.46	0.49	0.20					
200	0.446	10.66	2.53	0.84	0.35	0.08				
300	0.668		5.46	1.82	0.75	0.18	0.06			
400	0.891		9.44	3.15	1.29	0.32	0.11	0.04		
500	1.114			4.81	1.97	0.48	0.16	0.07		
600	1.337			6.81	2.79	0.68	0.23	0.09	0.04	
700	1.560			9.12	3.74	0.91	0.31	0.13	0.06	
800	1.783				4.81	1.18	0.39	0.16	0.07	0.04
900	2.005				6.02	1.47	0.49	0.20	0.09	0.05
1000	2.228				7.35	1.80	0.60	0.25	0.12	0.06
1100	2.451				8.82	2.16	0.72	0.30	0.14	0.07
1200	2.674					2.54	0.85	0.35	0.16	0.09
1300	2.897					2.96	0.99	0.41	0.19	0.10
1400	3.119					3.41	1.14	0.47	0.22	0.11
1500	3.342					3.88	1.30	0.53	0.25	0.13
1600	3.565					4.39	1.47	0.60	0.28	0.15
1800	4.011					5.49	1.85	0.75	0.35	0.18
2000	4.456					6.71	2.25	0.92	0.43	0.23

*Computed from the Scobey formula, $H=K\frac{V^{1.9}}{D^{1.1}}$ taking $K=0.34$, which is intended for riveted pipe where the rivet heads extend but slightly as in most light-weight pipe. See U. S. D. A. Technical Bulletin 150, The Flow of Water in Riveted Steel and Analogous Pipe.

crete foundation should be cast about the well in such manner that it will be independent of the casing. The pump head should be set level, and the well should be so nearly plumb as to make this possible. If, however, the well is not plumb, and too small to allow the head to be shifted to find a level position, then it should be set so that the shaft in the head follows the direction of the pump. To facilitate pumping into an underground line without using elbows the well can be finished off in the bottom of a shallow concrete pit and the pump head placed to discharge straight into the pipe as shown in figure 19*C*. Such an installation is an alternative of an underground discharge which can be furnished by the manufacturer, but at an increased cost.

The vertical centrifugal and the centrifugal and propeller types of turbine pumps are well adapted to pumping from rivers and reservoirs. These are usually installed on pile piers, with the drive head placed above the high-water zone. The pier must reach out into an adequate depth of water or a channel must be dredged to the site. With an over-all length of 25 feet, these pumps may be removed as a unit in the fall, and it is well to plan the layout to facilitate removal in such manner.

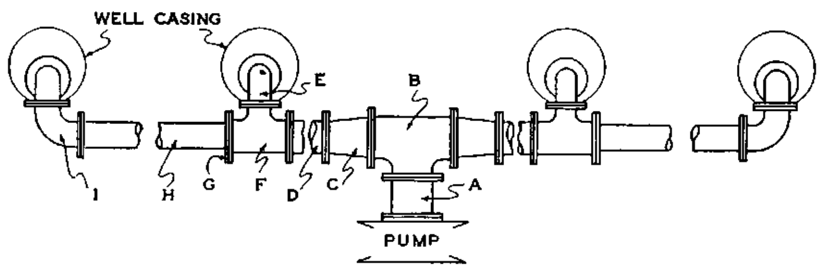

Figure 17.—Suction-pipe layout for a battery of four wells, with sizes for various capacities.

Total Capacity	*Recommended Size of Parts*
600 g. p. m.	*C*, increaser; *D*, 5 inches; *E*, 4 inches; *H*, 5 inches; *I*, 5 inches; drop pipes, 4 inches.
800 g. p. m.	*C*, increaser; *D*, 6 inches; *E*, 4 inches; *H*, 5 inches; *I*, 5 inches; drop pipes, 4 inches.
1,000 g. p. m.	*C*, increaser; *D*, 6 inches; *E*, 4 inches, drop pipe 4 inches; *H*, 5 inches; *I*, 5 inches, drop pipe, 5 inches.
1,200 g. p. m.	*C*, increaser; *D*, 6 inches; *E*, 5 inches; *H*, 6 inches; *I*, 5 inches, drop pipe, 5 inches.
1,600 g. p. m.	*C*, increaser; *D*, 8 inches; *E*, 5 inches, drop pipe, 5 inches; *H*, 6 inches; *I*, 6 inches, drop pipe, 6 inches.

The tee fitting *B* is of the same size as the pump inlet in all cases and should be set about 4 inches higher than the pipe ends. This fixes the size of the nipple *A*, which is required for double-suction pumps. *E* and *I* should be long radius elbows, tapered if necessary, although the use of reducing flanges is permissible for a reduction of 1 inch. Threaded fittings for sizes 6-inch and less may be used except at the pump and for the drops into the wells. With the capacities for the sizes named the friction should not exceed 2.0 feet, and the draft on each well will be approximately equal.

Suction Piping

Ordinarily too little attention is given to the planning of suction pipes. They should be as short as possible and of a larger size than the intake of the pump. At no point should the pipe rise above the pump intake, nor should there be any undulations in horizontal runs that would permit the formation of air pockets. Horizontal runs should be laid on a grade rising toward the pump. It is good design, in connecting a pipe to a pump having a smaller diameter inlet, to use an eccentric increaser. Long suction lines should certainly be investigated as to friction losses, for these may be so great as to seriously limit the pump capacity.

Considerable thought must be given to pipe sizes in a plant where a battery of wells is the source of supply. Here the water must travel through a considerable length of pipe, and as the number of wells increases the problem becomes somewhat complicated. Ordinarily there is no way of determining how much water each well will produce with a given drawdown when all wells are being pumped at one time. About the only plan that can be followed is to assume that each well will furnish an equal amount and design the pipe lines so that the friction loss to the near wells is about equal that for the end wells. To do this smaller drop pipes are required for the near wells. In figures 16 and 17 pipe sizes are shown for three- and four-well systems with varying capacities. With these sizes the friction loss will not exceed 2.0 feet of head. To aid in the design of pipe systems, friction losses per 100 feet of pipe are given in table 3 for sizes commonly used.

Standard pipe or heavy gage (not less than number 10) welded pipe should be used for suction lines. Riveted pipe is not trustworthy from the standpoint of airtightness. There are too many possibilities for air leaks, and rusting seems to be more active at rivet holes. Welding at joints to reduce cost may be resorted to in many places, but the drop pipes should be flanged so that they can be removed in case work needs to be done on the well. In laying suction pipe it should be sloped up to the pump, so that in priming all the air can be removed. A slope of 8 or 10 inches to 100 feet is sufficient, and there should be no in-between high and low spots to trap air. Gate valves on the suction side should be avoided because of the possibility of air leaks at the stem.

Discharge Piping

The conditions of discharge are so varied that only a few representative cases will be discussed. If the discharge pipe length is 10 feet or less there is little head saved in increasing the size over that of the pump outlet. For short lines less than 25 feet long a size should be selected that will not incur a friction loss of over 2.0 feet per 100 feet of pipe. The size of pipe for long lines must be selected on the basis of initial cost against possible power savings over a period of time.* The greater the length of pumping time contemplated each season, the greater is the justification for large-size pipe.

A horizontal pump will ordinarily be equipped with a gate valve just above the discharge nozzle. It may be attached to the pump if flanged, or on a short nipple if threaded. A gate valve is required in conjunction with priming and serves as a possible means of regulating the discharge. A check valve is required just beyond

*See Appendix A for a more thorough discussion of the selection of pipe size.

the gate valve when pumping into a long pipe line, in order to prevent a large quantity of water from returning to the well. Depending on the quantity, this return to the well may be undesirable from several standpoints: It may cause the well to overflow; it may overspeed the motor in reverse; and the power required to fill the line each time is wasted.

Water hammer is always set up as a check valve closes, the intensity increasing with the total lift and the length of the line. In substantial metal pipe no form of relief is required for the usual conditions; but the pipe line should be securely anchored in concrete near the valve, in order to protect the pump from any movements set up. In figure 19*C*, should water hammer be severe, a concrete block of from 8 to 10 cubic feet in volume should be cast about the increaser, with several feet of earth separating it from the wall of the building. A check valve cannot be placed in a tile or concrete pipe line if the length exceeds about 100 feet, or the head exceeds about 10 feet, without some form of relief from water hammer.

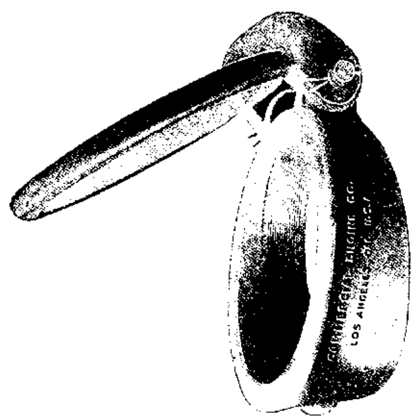

Figure 18.—Flap valve for installation on a pipe end.

Usually a standpipe is inserted beyond the check valve to effect this. The height of the standpipe must be equal to the lift and the pipe friction combined, plus some freeboard. When the height of the standpipe is not over 8 or 10 feet it is probably more economical to construct it of concrete.

With such a structure it is possible to use a flap valve (figure 18) on the end of the pipe entering the box, instead of the more expensive line check valve. Because of the capacity a freeboard of 2 or 3 feet only is required for lines less than 1,000 feet in length. When the height is considerable the standpipe should be of steel, and of a size not less than the line pipe diameter; and it should be securely guyed. A freeboard of 4 or 5 feet is required for lines 1,000 feet long. Figure 19 illustrates some of the conditions ordinarily found. Should the standpipe be connected to the pump by a relatively long, large, thin pipe similar to the condition in figure 19*A*, then an air vent should be installed behind the check valve. This is necessary to prevent the pipe collapsing due to the partial vacuum formed when the pump is stopped. Air vents also should be installed at

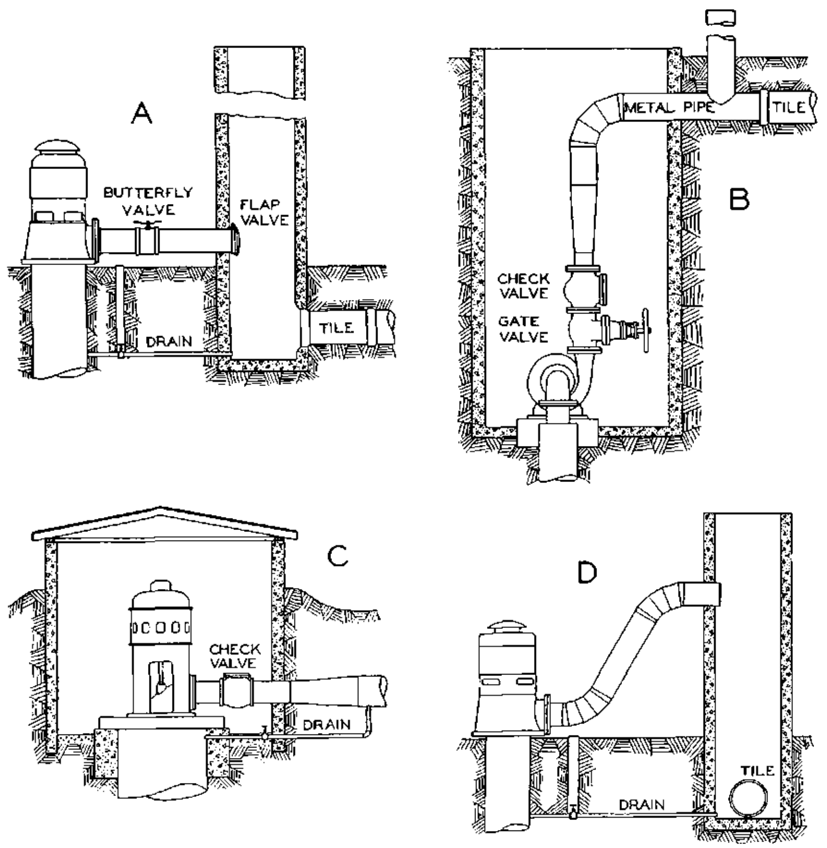

Figure 19.—Common types of installations involving pipe lines. Plan *D* is used for low heads, the lower edge of the discharge pipe being placed so as to keep the line full of water when not shut down and submerged when pumping. It is not adapted to a pipe line having outlets at various elevations. Check valves (plan *C*) are necessary on short lines.

high points in pipe lines that cross hills or ridges. They can be made of 1½-inch or 2-inch pipe and must rise above the hydraulic grade line to prevent an overflow. The question is often asked if it is not "easier on the pump" to discharge into a box instead of directly into a pipe line. The answer is no; actually more power is required. Energy is lost as the water enters the box, and a similar amount of energy is required to start it off again through the pipe line.

Ordinarily all pipe should be placed underground. Steel pipe, when exposed to the sun, will increase in length considerably when hot, and therefore should contain expansion joints. Lacking these the pipe will move laterally and, when cooling, inserted joints might pull apart. Certain couplings using a rubber gasket compressed with bolts (figure 20) permit of lateral and longitudinal movement and are useful as expansion joints and to supply a flexible

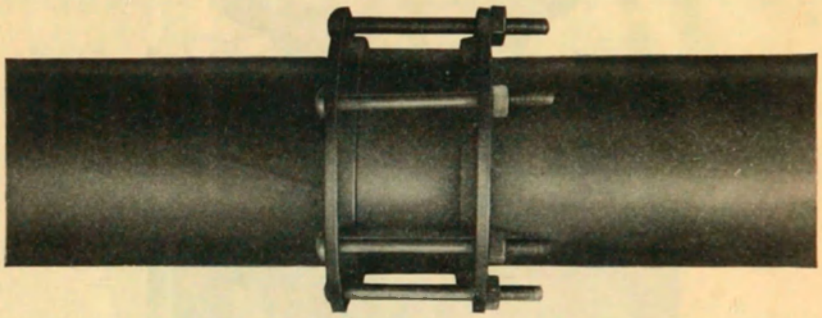

Figure 20.—A steel sleeve-type coupling that permits some flexibility in alignment and change in pipe length.

joint between the pump and a rigid structure. Such a joint should be inserted in the discharge pipe, shown in figure 18*A*, if the pipe is heavy and the length under 6 feet. These couplings require no threading of the pipe ends and are less costly than a flange connection. A type of surface outlet used to a large extent in California, commonly known as an alfalfa valve, provides a very economical and convenient means of controlling water at intermediate points in a pipe line (figure 21). Ground movement due to frost action is responsible for many failures in clay and cement pipe, breakage usually occurring at the joints. To escape this form of injury it is necessary to cover the pipe to a depth of at least 4 feet.

It should be kept in mind that the point at which the pump discharges should be as low as possible to avoid pumping against unnecessary head. In most cases water can be discharged at the water line in the ditch. If there is any danger of canal or flood water entering it, the discharge opening should be above that line, or a flap valve attached. Well casings for turbine pump installations are usually finished off from 3 to 6 inches above the surrounding ground surface to prevent the entry of any surface water. This same safety can be obtained by building a concrete wall around the top of a well casing and passing the discharge pipe through the wall. Such a joint must be made water tight on the outside of the pipe.

Another method of avoiding excess lift at the ground surface is by turning down the end of the discharge pipe so that it discharges below the water surface with a siphon action.

In connecting pipes of different diameters a tapered section should be used in order to recover the head due to the change in velocity. This velocity head is lost if a sudden enlargement is used.

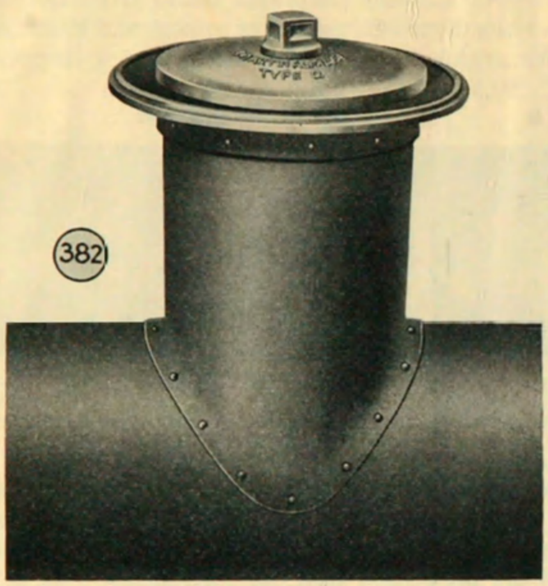

Figure 21.—A simple horizontal valve known as an alfalfa valve.

Siphon Pipes

Siphons between wells have been in use successfully for many years in Weld County but are little used elsewhere in Colorado, principally because of lack of information regarding them. Their principal use is in connecting a battery of wells when a deep-well turbine or vertical centrifugal pump is used. These pumps do not lend themselves easily to direct connection to a common suction pipe. Siphons will work theoretically up to the limit of the air pressure available. This limit is equal to 28 feet of water at an altitude of 5,000 feet, varying approximately 1 foot for each 1,000-foot change in elevation. Practically the limit is about 24 feet, and under certain circumstances pipe friction must be deducted from this. In ordinary installations, if the length of the drop pipe above the water surface in the pumped well does not exceed the practical limit for that altitude, the siphon will work. If the conditions are such that this may be exceeded, then the discharging end should open under the water surface in a bucket placed within the required distance below the elbow (figure 22). A valve at the end of the pipe will accomplish the same purpose but is inconvenient of adjustment.

Substantial threaded pipe or heavy welded pipe only should be used in siphon lines as in suction lines. The cost of installation of such lines in deep trenches or in tunnels constitutes a large proportion of the total, and it is false economy to use anything but long-life pipe. The lines must be laid straight and in such a manner that the priming end will be the high point, in order that all the air

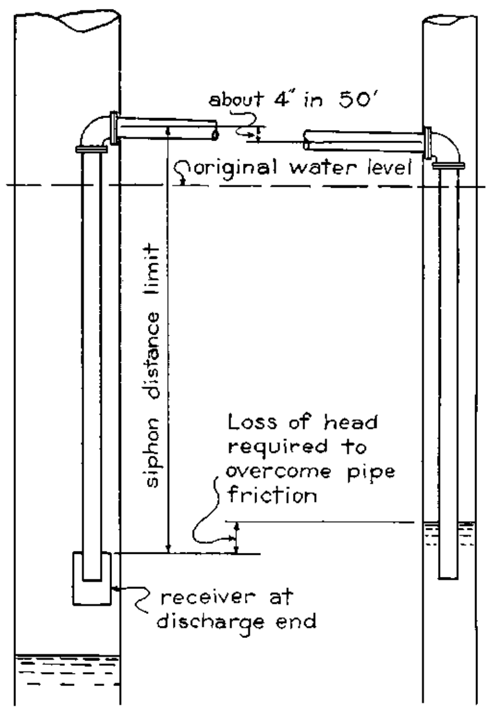

Figure 22.—Method used to prevent a siphon from exceeding the siphon limit.

may be withdrawn by the priming pump. The siphon, when primed, will start to operate when water is pumped from the main well. The water surface in the siphoned well will be drawn down to a point which will be higher than that in the pumped well by an amount equal to the friction losses in the pipe line. This loss can be computed from table 3. It will be seen that if the length of the drop pipe in the pumped well does not exceed the limit previously mentioned, and the drop pipe into the siphoned well is of similar length, the end of the latter cannot become uncovered, and prime will be held. Ordinarily the priming end is located at the pumped well for convenience; but if the line is long, the friction great, and the lift near the limit, flow conditions will be better if the high end is at the siphoned well. No trouble should occur with siphon lines less than 100 feet in length horizontally.

Siphon pipes may lose prime in several ways other than by leaks. Insufficient submergence of the intake may allow air to be taken in through vortexes. Free discharge on the outlet end may cause a condition of greater flow in the discharging drop pipe than in the intake drop pipe, resulting in a separation of the water column. This can be corrected by placing the discharging end in a bucket, as previously mentioned. High velocities tend to carry air bubbles out, whereas low velocities allow air to collect at a high point which, if not removed, will cause loss of prime. Since all natural waters contain air in solution which is released when subjected to reduced pressure, it is wise not to impose high siphon lifts on a line operating under low velocity.

CENTRIFUGAL PUMP CHARACTERISTICS

The reason a centrifugal pump actually pumps water is that, as the impeller is rotated, water is thrown out at high velocity at the periphery by centrifugal force. As water leaves the outer rim of the impeller it is collected in the pump case, and other water moves up through the suction opening to take its place in an uninterrupted flow. In the pump case the velocity is converted into pressure which, in good design, is accomplished with remarkably high efficiency. Of course some velocity remains or the water would not move out. It will be recognized that no water will be pumped until the speed of the impeller is high enough to produce a pressure equal to the lift. Beyond this point more water will be pumped as the speed is increased, but with a rapidly increasing power requirement. A point is reached in the speed of an impeller of specific diameter at a definite lift and discharge where the ratio between the actual work performed and the power required reaches a maximum. This is the point of maximum efficiency. Also for every speed of an impeller of specific diameter there is a maximum efficiency.

The characteristics of a deep-well turbine pump operating at a speed of 870 revolutions per minute are shown in figure 23. At the left side a scale of heads or lifts is shown and at the bottom a scale of discharges. This applies to the curve marked "Field Head-Capacity." On the right side is a scale of efficiencies which, with the discharge scale, applies to the curves marked "Efficiency." Also on the right side, but lower down, is a scale of horsepowers which applies to the curve "Brake Horsepower." To read the curves a certain discharge is selected, and wherever that vertical line crosses a curve the quantities corresponding are read on the right- or left-hand scales. For example, the characteristics are desired for a discharge of 420 gallons per minute. Where the vertical line for this quantity crosses the head capacity curve a head of 59.5 feet is found; on the field efficiency curve 68.0 percent; and on the brake horsepower curve 9.7 horsepower. This is not the most desirable

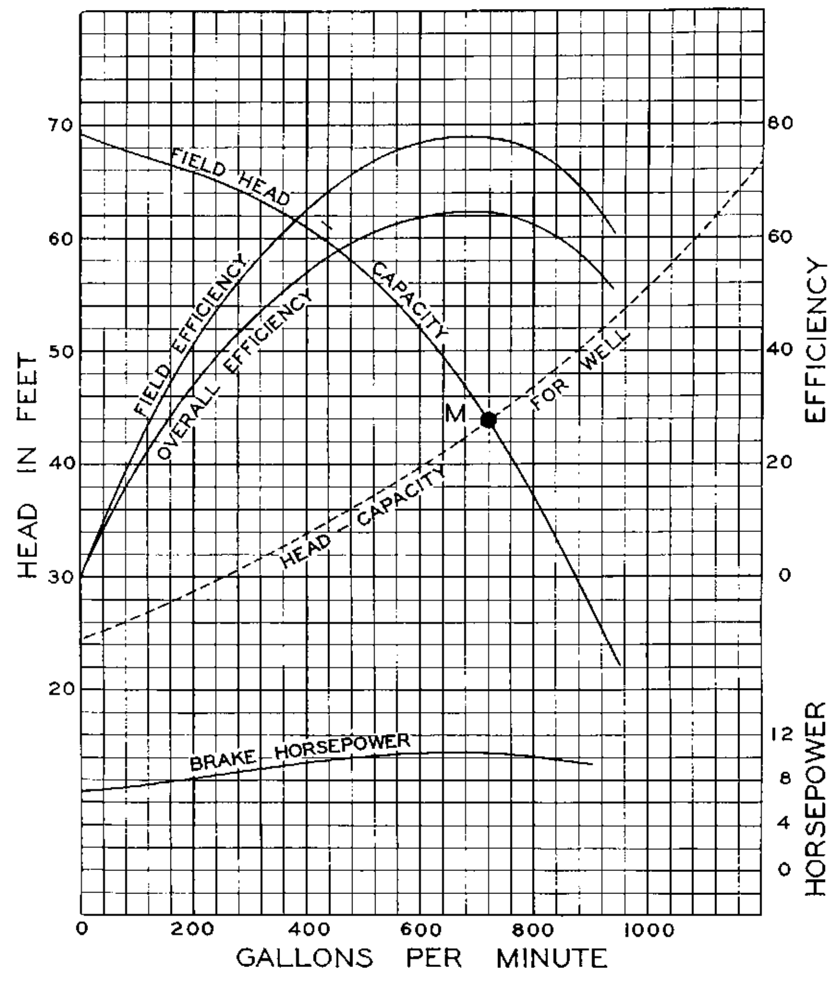

Figure 23.—Characteristics of a number 14 deep-well turbine pump operating at a speed of 870 r.p.m. Dashed line represents the well characteristic. Its point of origin is at a head of 24.5 feet, which is the depth to the static water level from the center of the discharge pipe. The plant capacity, 720 g. p. m., is shown at point *M*, where the two head curves intersect.

point of operation for this pump, for it will be seen that the efficiency curve is higher to the right; and it is near this high point, or at 700 gallons per minute and a head of 45.5 feet, that the pump should operate. A feature of good pump design will be noted on the horse-power curve in that its high point is in line with the maximum efficiency, which means that if selected here, no matter how the water conditions may change, the power requirements will be less. This is known as a non-overloading type of impeller. Another impeller with different shape of vanes or wider, or of different diameter, will cause a change of shape and location of these curves, as will also a change in speed. It is from such curves that the pump manufacturer selects the pump for the condition under which it is to operate.

If the conditions are guessed at, and a pump selected that is of too-great capacity for the well, there are a few ways to remedy the situation. If direct connected, which allows for no speed change, then within reasonable limits the impeller diameter may be reduced by turning in a lathe, or another impeller may be substituted. If belted, a change in pulleys will allow for operating at a slower speed. These changes may not result in as good an operating efficiency as originally planned and can be expected to cover a reasonable range of conditions only. Another method of reducing capacity is that of throttling with a valve. This increases the head pumped against, and as seen from figure 23 the discharge will be less; but the power required is reduced very little, and the resulting efficiency is less. This method, then, is to be avoided as less desirable than the first-mentioned methods. The importance of well testing is shown in the foregoing explanation of centrifugal-pump characteristics. No pumping machinery should be purchased before such a test is made if the high efficiencies of today's pumps are to be realized.*

THE MEANING OF EFFICIENCY

The farmer is chiefly concerned with what it costs to deliver water at the ground surface. This involves the lift, pump, motor or engine efficiency, power or fuel rates, capital costs, and length of time of seasonal operation. A plant may show a low cost for lifting an acre-foot of water 1 foot because it is efficient, but the total lift may make the cost of operation quite high. On the other hand a low-lift plant may be inefficient and show a high cost per acre-foot-foot, and the cost of operation be equally high. The desired ultimate result is a plant planned so that water can be delivered at a given location at the lowest possible cost. This can be effected

*Well testing is discussed in Bulletin 415 of this station, "Construction of Irrigation Wells in Colorado," p. 35.

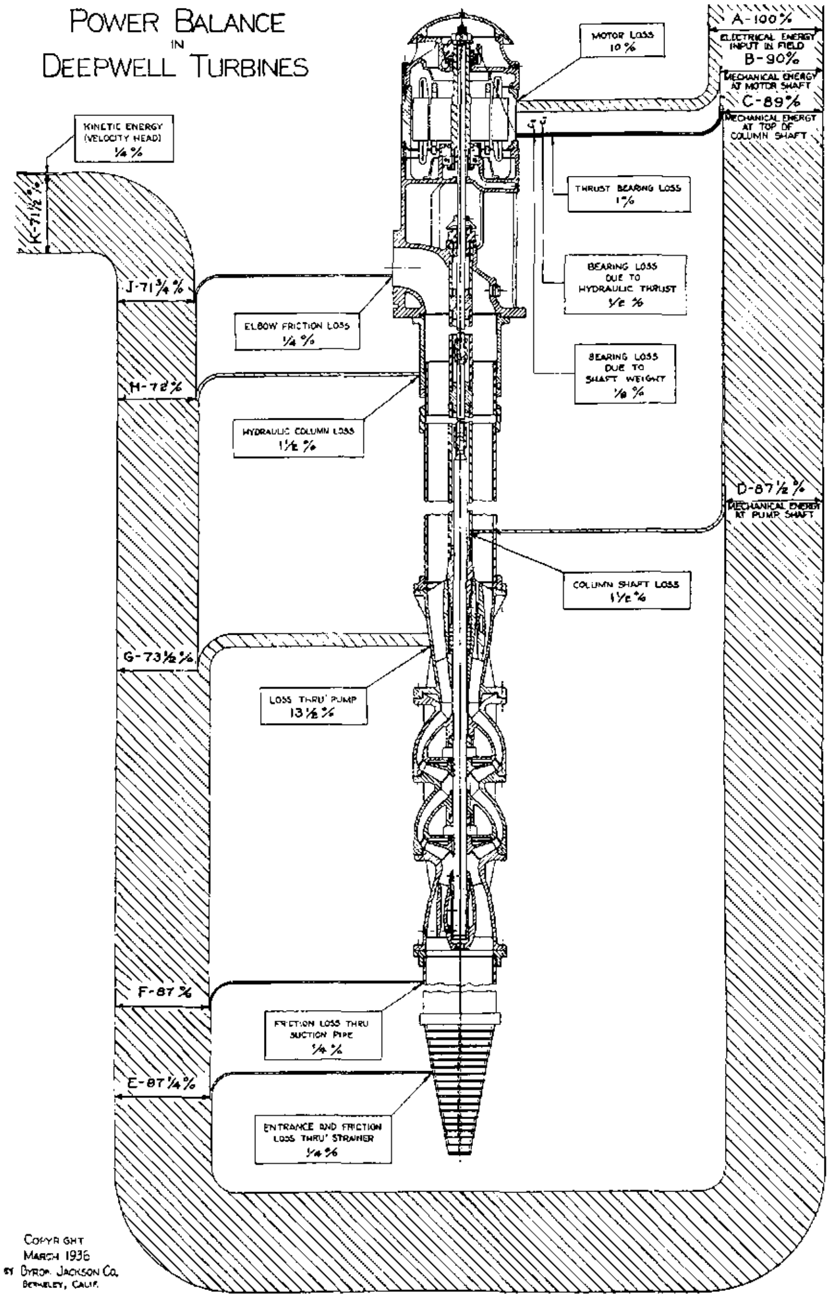

Figure 24.—Sequence of losses as they occur in a deep-well turbine pump.

only by the wise selection of equipment and an attention to details that will make the whole plant efficient.

The beginnings of plant efficiency lie in the well. If by larger diameter, greater depth, or better drilling methods a well is obtained that has a less drawdown per unit of discharge, then the total lift will be reduced, with a consequent reduction in the power required. False economies here may result in a handicap of increased lift that cannot be overcome by the best pump efficiency and lowest power rate. Certain small savings in lift can be made in the manner of discharge in relation to the water level in the discharge box or ditch, as discussed on page 37. The influence of pipe-size of long discharge lines is discussed on page 34. The power required to raise water varies directly as the lift, therefore a 10-percent saving in lift will result in a 10-percent power saving, which illustrates the importance of this item.

The term efficiency is more familiar to most people in its application to pumps. Specifically it means the figure in percentage obtained by dividing the power represented in the water being lifted, called water horsepower, by the actual power required. There are several applications of the term, and it should be made clear to the purchaser which efficiency is meant. In a deep-well turbine pump the term "bowl efficiency" may be used, which means the efficiency of the pumping unit attached directly to the drive head or to a short section of column. The term "field efficiency" takes into consideration the friction of the water passing through the column pipe and the horsepower loss in shaft rotation; hence it is less than the bowl efficiency. Field efficiency is affected by the size of the column pipe, being reduced through using a pipe too small in order to reduce cost. The term "over-all efficiency" includes the efficiency of the electric motor and is the one that should be specified to the purchaser, because it is the one most easily determined by test. Field and over-all efficiency curves are shown in figure 23. The diagram in figure 24 clearly illustrates the various losses as they occur in a deep-well turbine pump. With 100 percent to start with in the form of electric energy, the first loss occurs in the motor, which is shown here as 10 percent. With small motors this would be more nearly 15 percent. Then, with the other losses shown in order amounting to 28½ percent, the energy represented in the water being lifted is 71½ percent of the initial energy. As an illustration of the terms used and their values, assume a pump with a purported field efficiency of 68 percent pumping 750 gallons per minute against a head of 38 feet and driven by a 10 horsepower motor having an efficiency of 85.0 percent.

$$\text{Water power} = \frac{750 \times 38}{3950} = 7.22 \text{ horsepower}^{*}$$

$$\text{Power required} = \frac{\text{water horsepower}}{\text{pump efficiency}} = \frac{7.22}{0.68} = 10.61 \text{ brake horsepower}$$

$$\text{Motor input} = \frac{\text{brake horsepower}}{\text{motor efficiency}} = \frac{10.61}{0.85} = 12.48 \text{ b. hp.} = 9.31 \text{ kilowatts}$$

$$\text{Over-all efficiency} = \frac{7.22}{12.48} = 0.85 \times 0.68 = 58 \text{ percent}$$

The motor is indicated as having a 6 percent overload, which is permissible. If on test the power required is found to be 11.0 horsepower, then the field efficiency would become 65.6 percent (7.22÷11.0).

It may be said that the field efficiency of a good turbine pump need not fall below 60 percent for the usual irrigation pumping conditions, providing the pump is correct for those conditions.

Horizontal centrifugal pumps are considered somewhat differently. The efficiency rating is for the pump part only, and because there is no space limitation in the design of the case the best efficiency may be slightly better than for deep-well turbine bowls. Efficiencies less than 60 percent for any kind of horizontal pump above the 3-inch size should not be considered, even in low-cost pumps. Piping must be designed to have ample capacity in order to utilize the benefits to be obtained with high pump efficiency.

As applied to electric motors efficiency means the percentage of horsepower input that is delivered. This varies according to the motor size, speed, and load, and slightly as between makes of motors. In the sizes of from 3 to 10 horsepower the efficiency may be expected to range between 80 and 85 percent; above 10 horsepower, between 85 and 90 percent. Motor efficiency may be expected to remain practically constant throughout the life of the machine, providing the bearings are in good order and the insulation is not damaged. Vertical motors may be shown to be 1 or 2 percent less efficient than horizontal motors because of the inclusion of the thrust bearing.

The efficiency of an internal-combustion engine is largely relative. The gasoline engine utilizes less than 20 percent of the heat energy available in the fuel, while the Diesel makes use of about 30 percent. It is customary to speak of the efficiency of a gasoline engine in terms of the number of brake horsepower-hours developed from a gallon of fuel, while in Diesel engines the unit

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*Water horsepower $= \frac{\text{g.p.m.} \times \text{lift}}{3{,}950}$

is the pound of fuel. Gasoline engines have an output of from 7 to 11 horsepower-hours per gallon, with 8 or 9 as a good average. Diesel engines reach a maximum output of about 1 horsepower per 0.45 pounds of fuel, or about 16 horsepower-hours per gallon of fuel. The relative fuel efficiency volumetrically between the Diesel and gasoline engines is about 2 to 1, and the fuel cost differential is about 100 percent, which makes the fuel operation cost of the Diesel about one-fourth that of the gasoline engine.

FITTING THE PUMP TO CONDITIONS

There are two principal features involved in obtaining the right pump for the well. One depends on the characteristics of the well and the other on the adjustment to the demand for water. Both are important; but the second, often ill considered, has considerable flexibility and deserves close study. In general, and particularly in regard to electric pumping, the longer a plant is operated each season the lower all unit costs will be, except that of attendance and perhaps depreciation. The farmer should estimate his requirements for a month of maximum demand, planning on almost continuous operation. The procedure suggested is given in this example: Given a farm of 160 acres for which to provide in 1 month a 4-inch irrigation for 60 acres, two 4-inch irrigations on 50 acres, and three 3-inch irrigations on 50 acres, which amounts to a total of 90.83 acre-feet. This figure divided by the number of days of operation desired gives the number of acre-feet daily to be pumped. Thus, if 26 days is the time desired, then 3.49 acre-feet must be pumped per day, or at a rate of 1.76 cubic feet per second (791 gallons per minute).

The well characteristics are now examined for the purpose of determining the drawdown for approximately 800 gallons per minute. If the well test is properly made—that is, the drawdown obtained for several rates of discharge—a curve can be drawn, and it will be a simple matter to compute the drawdown for any other discharge (figure 23). With the drawdown known, the distance from the ground surface plus any lift above ground yields the total lift. This total lift and the discharge are the necessary data to provide the pump manufacturer, in order that he can select the correct pump—the one giving the best efficiency.

Although the foregoing is the best method of attack on this problem, there are other features to be considered when the pump is to be electrically driven. Should the conditions work out in such a manner that the brake horsepower computes just slightly more than the capacity of a certain motor size, thus requiring the next larger size, then it may be possible and certainly advantageous to reduce the load so as to fully load the smaller motor. To illustrate: Should the manufacturer's curves require a brake horsepower of 16.9, requiring a 20 horsepower motor for the conditions as computed, then those conditions should be altered so as to bring down the brake horsepower to about 15½, which will allow the use of a 15-horsepower motor with a permissible overload. To do this may require a slight impeller alteration or a different impeller. A small sacrifice in the quantity of water delivered will have to be made, but a considerable saving in the cost of electric power will be made with the smaller motor.

Another point which often comes up in shallow-well areas is that of lowering the pumping level to near the bottom of the well in order to procure the maximum yield. Since the last few feet of drawdown produce but a slight increase in yield, it is often desirable to allow a greater depth to remain in the well, thus reducing the total lift and permitting the use of the next smaller motor size.

A more complete analysis of the method of pump selection and the steps involved are given in Appendix B, page 54.

COSTS

The pump customer has such a wide range of types, makes, and quality of equipment to select from that very little aid can be extended here from which to make estimates. The cost of a horizontal centrifugal pump will depend on its size, type, and quality. In the 6-inch size the price will vary from $150 to $400. A turbine with belted head varies in cost according to the lift, size and number of bowls, and the size of column pipe. There is no great disparity of price as between standard makes. For pumping 750 gallons per minute against a 20-foot lift the cost may be expected to be about $600, and $900 for a 60-foot lift.

Electric motors vary in price more according to speed than size. A 10-horsepower, 220-volt, three-phase, 1,760 revolutions per minute, squirrel-cage induction motor may cost $175; but for an 860-speed the cost will be $245, which is about the price of a 20-horsepower, 1,760-speed motor. Vertical motors, usually having built-in pump thrust bearings and clutches, cost about one-half more than horizontal motors. Single-phase motors are also more costly than three-phase motors.

Single-cylinder, electric-ignition engines will cost from $35 to $45 per horsepower. Tractor engines, being of higher speed and lighter in weight, will cost less than half of these prices. Hot-head engines cost from $70 to $90 per horsepower, and the high-speed Diesels from $35 to $90, according to size, the larger sizes having the lesser unit rate.

In an endeavor to ascertain costs on pumping this office made some intensive studies in 1929 and 1930. The plants selected were of average performance; none was highly efficient, and a few would be considered poor. It was found that, using gasoline at 14 to 16 cents per gallon, the average fuel cost was 9.0 cents, and the total cost 36 cents per acre-foot raised 1 foot. Among the electrically driven plants two different rates were in effect. With the lower one the average cost for current was 6.7 cents per acre-foot-foot, and the total cost was 13.6 cents. The costs under the higher rates were found to be 10.5 cents and 16.6 cents, respectively, per acre-foot-foot.

In a well-designed, efficient plant these costs will be materially lower. The following actual examples of modern plants are selected to show how low costs may be. These two plants are not strictly comparable, as well B cost considerably more than well A. Pump A is probably less efficient than pump B.

Plant A:

The well is 14 inches in diameter and 139 feet deep. The water stands at about 56 feet, and while pumping 1,100 gallons per minute the lift is 77 feet. The equipment consists of a turbine pump belted (V-belts) to a 50-horsepower, high-speed Diesel engine. The costs for the season 1935 in pumping 503 acre-feet are as follows:

Cost of plant:			
Well	$ 600.00		
Pump	840.00		
Engine (installed)	1,990.00		
Shelter	75.00		
	$3,505.00		
Fixed Charges:			
Interest on $3,505.00 @ 5%		$ 175.25	
Taxes estimated		50.00	
Depreciation on engine 12%		218.88	
Depreciation on pump 8%		67.20	
Depreciation on well and shelter 3%		20.25	
		$ 531.58	$ 531.58
Operating cost:			
Fuel, 470 gallons @ 7½c, 5,714 gallons @ 7¾c		$ 478.08	
Distillate, 30 gallons @ 8c		2.40	
Lubricating oil, 195 gallons @ 65c		126.75	
Other oils and greases		13.91	
Sales tax on above @ 2%		12.42	
Engine repairs, anticipated annually		100.00	
Pump repairs, anticipated annually		25.00	
Attendance, 250 hours @ 35c		87.50	
		$ 846.06	846.06
			$1,377.64

Total cost per acre-foot	$ 2.74
Total cost per acre-foot-foot	0.036
Operating cost per acre-foot	1.68
Operating cost per acre-foot-foot	0.022
Operating time	2,487 hours

Plant B:

The well is 48 inches in diameter and 66 feet deep. The water stands at 30 feet normally, and while pumping 890 gallons per minute the lift is 55 feet. The equipment consists of a turbine pump direct driven by a 15 horsepower motor. The over-all efficiency on test was 65 percent. The costs for the season 1935 in pumping approximately 410 acre-feet are:

Cost of plant:			
Well	$1,056.00		
Pump and motor	1,150.00		
Shelter	40.00		
	$2,256.00		
Fixed charges:			
Interest on $2,256.00 @ 5%		$ 112.80	
Taxes estimated		30.00	
Depreciation on pump and motor 8%		92.00	
Depreciation on well and shelter 3%		32.88	
		$ 267.68	$ 267.68
Operating cost:			
Electric current, 35,374 kilowatt-hours		$ 578.75	
Lubricating oil		2.00	
Anticipated pump repairs, annually		25.00	
Attendance estimated		20.00	
		$ 625.75	$ 625.75
			$ 893.43

Total cost per acre-foot	$ 2.18
Total cost per acre-foot-foot	0.040
Operating cost per acre-foot	1.53
Operating cost per acre-foot-foot	0.028
Operating time approximately	2,500 hours

For the purpose of estimating the probable cost of pumping for a contemplated installation the foregoing methods may be used. Aid in computing the fuel or electric costs may be obtained from table 4, keeping in mind that an over-all efficiency of between 60 and 65 percent may be obtained in a carefully designed plant of over 500 gallons per minute capacity. In the case of the electrically driven plant, knowing the lift and discharge, the electric current consumption may be obtained by multiplying the quantity under the proper efficiency and opposite the discharge by the lift expressed as a decimal part of 100. This will give the number of kilowatts

TABLE 4—*Power required to lift various quantities of water 100 feet*

Gallons per minute	Acre-ft. in 24 hours	Water horse-power for 100-ft. lift	Kilowatts at stated over-all efficiencies in percent				
			50	55	60	65	70
100	0.44	2.53	3.78	3.43	3.15	2.91	2.70
150	0.66	3.80	5.67	5.15	4.72	4.36	4.05
200	0.88	5.06	7.56	6.87	6.30	5.81	5.40
250	1.10	6.33	9.44	8.59	7.87	7.26	6.75
300	1.33	7.60	11.33	10.30	9.44	8.72	8.10
350	1.55	8.86	13.22	12.02	11.02	10.17	9.44
400	1.77	10.13	15.11	13.74	12.59	11.62	10.79
450	1.99	11.39	17.00	15.45	14.17	13.08	12.14
500	2.21	12.66	18.89	17.17	15.74	14.53	13.49
600	2.65	15.19	22.67	20.61	18.89	17.44	16.19
700	3.09	17.72	26.44	24.04	22.04	20.34	18.89
800	3.54	20.26	30.22	27.47	25.18	23.25	21.59
900	3.98	22.79	34.00	30.91	28.33	26.15	24.29
1000	4.42	25.32	37.78	34.34	31.48	29.06	26.98
1100	4.86	27.85	41.56	37.78	34.63	31.96	29.68
1200	5.30	30.38	45.33	41.21	37.78	34.87	32.38
1300	5.74	32.92	49.11	44.65	40.92	37.78	35.08
1400	6.19	35.45	52.89	48.08	44.07	40.68	37.78
1500	6.63	37.98	56.67	51.52	47.22	43.59	40.48

used per hour. The total number of hours per season will have to be estimated from the number of acre-feet that will be pumped. The second column will aid in determining this time. The average cost per kilowatt of electric current will be obtained from the rate schedule prevailing in the territory. Three rate schedules are here given, covering the major part of the pumping territory. It will be noted that each rate involves the size of the motor.

1. Public Service Company of Colorado

First 100 kilowatt-hours per rated horsepower per season......$0.05
Next 200 kilowatt-hours per rated horsepower per season...... .03
Next 200 kilowatt-hours per rated horsepower per season...... .02
All additional power.. .01½

2. Home Gas and Electric Company and Colorado Central Power Company

First 100 kilowatt-hours per horsepower per season	$0.05
Next 200 kilowatt-hours per horsepower per season	.03
Next 200 kilowatt-hours per horsepower per season	.02
Next 1,000 kilowatt-hours per horsepower per season	.01½
All additional power per kilowatt-hour	.01

3. Southern Colorado Power Company

First 200 kilowatt-hours per horsepower per season	$0.055
Next 100 kilowatt-hours per horsepower per season	.033
Next 500 kilowatt-hours per horsepower per season	.022
All additional power per kilowatt-hour	.0166

Discount for prompt payment, 10 percent.

The season extends over the 7 months of irrigation. If power lines do not reach to the plant location, the power company may require a minimum seasonal use of power annually for from 3 to 5 years to cover the cost of the extension.

In the determination of fuel cost for engines it will be necessary to divide the water horsepower shown in the third column of table 4 by the pump efficiency, which gives the brake horsepower. The brake horsepower divided by the number of brake horsepower per gallon of fuel that the engine will deliver will yield the fuel consumption per hour.

It must be kept in mind that fuel and electric costs do not constitute the total. Fixed charges, such as depreciation, interest on the investment, and additional taxes, may easily equal or exceed operation costs. These may be computed in the manner indicated in the two examples of cost analysis just previously given.

PURCHASE AGREEMENT

To prevent misunderstandings between the manufacturer and customer a contract in writing and properly signed should be entered into. This should set forth, in addition to the terms of the sale, a complete description of the pump and accessories, such as starters, switches and the wiring; and the performance at one or two points near the anticipated operating point. In the statement of performance it should be clearly shown what efficiencies are guaranteed. When the pump is direct connected, gear, or V-belt driven the purchaser should by all means insist on an over-all efficiency guarantee. In lieu of this performance guarantee a curve, correctly labeled, is often submitted, which is still more desirable from the standpoint of subsequent testing. The customer should make sure what he is buying; verbal agreements generally have two interpretations when trouble arises, and often lead to disagreeable arguments. The contract should cover, when possible, both the pump and the motor or engine, in order to avoid divided responsibility. The purchaser then has but one company to deal with in cases of dispute.

APPENDIX A

THE economical size of pipe may be determined by a computation of the annual overhead cost of the pipe and the cost of power to pump against the friction. An illustration of the method of attack is given in the following example:

A pumping plant has a capacity of 900 gallons per minute and requires a discharge pipe 50 feet long. The friction loss would be 3.0 feet for a 6-inch pipe, 0.74 foot for an 8-inch, 0.25 foot for a 10-inch, 0.10 foot for a 12-inch, and 0.045 for a 14-inch pipe. The installed cost of 14-gage galvanized pipe per foot is estimated as follows: 6-inch, 80 cents; 8-inch, 95 cents; 10-inch, $1.10; 12-inch, $1.25; and 14-inch, $1.45; and the total cost in each case will be $40.00, $47.50, $55.00, $62.50, and $72.50. It is assumed that the life of the pipe is 20 years, that the money was borrowed at the interest rate of 6 percent, and that it is to be repaid in equal annual payments over the 20-year period. The average annual overhead cost will be one-twentieth of the combined initial cost and the total interest. It is assumed also that the overall efficiency of the plant is 55 percent and the cost of electricity 2 cents per kilowatt-hour. The results are tabulated for 1,700 hours and 300 hours of pumping time.

TABLE 5—*Variation in pumping cost due to use of various sizes of discharge pipe*

Pipe size Inches	Total annual cost	Cost of power due to pipe friction		Total costs	
		1,700 hrs.	300 hrs.	1,700 hrs.	300 hrs.
6	$ 3.20	$31.44	$ 5.55	$34.64	$ 8.75
8	3.80	7.54	1.33	11.34	5.13
10	4.40	2.62	.46	7.02	4.86
12	5.00	1.05	.19	6.05	5.19
14	5.80	.47	.08	6.27	5.88

From table 5 it will be seen that the 12-inch size is the most economical for the 1,700-hour pumping time, and that the 14-inch size costs too much in proportion to the power savings effected. The saving effected by the 12-inch size as compared with the 10-inch is largely theoretical, as the difference in friction head loss, 0.25 and 0.10 foot, is hardly measurable in the field. The difference between the 10-inch and 8-inch sizes, however, is significant, amounting to $4.32; hence the logical choice is the 10-inch. When the pumping time is but 300 hours it is probable that the 8-inch pipe is the proper choice.

APPENDIX B

A FURTHER and more technical discussion on the selection of a pump is placed here. It is based entirely upon a good well test, one in which enough points were obtained to enable the drawing of a characteristics curve. Figure 23 contains such a curve drawn from an actual test of a well sunk 54 feet below the standing water level, which in turn was 24.5 feet below the point of discharge.

From figure 23 the lifts and discharges are entered in table 6. From these figures, and assuming that a different pump would be furnished for each condition having a constant over-all efficiency of 60 percent, the motor size and power input to the motor are computed. It is further assumed that 320 acre-feet of water will be pumped, sufficient for a 160-acre farm. The power rate applied is that of the Home Gas and Electric Company previously given. The table shows the increasing costs as the pumping rate is increased. This is due to the greater drawdown and the less favorable electric rate because of a shorter operating time.

To show the effect of efficiency, other factors remaining the same, the second part of the table is computed on an over-all

TABLE 6—*Quantities involved in pumping 320 acre-feet of water from the same well under different conditions; over-all efficiency 60 percent*

Discharge g. p. m.	Lift ft.	Size motor required hp.	Input to motor kw.	Pumping time days	Electricity used kw-hr.	Cost
700	42.9	10	9.46	103.4	23,480	$384.80
800	46.6	15	11.73	90.7	25,478	479.78
900	50.8	15	14.39	80.4	27,787	502.87
1,000	55.5	20	17.47	72.4	30,345	603.45
1,100	60.7	25	21.02	65.8	33,212	685.68
1,200	66.5	30	25.12	60.3	36,374	770.61
The same conditions but with an over-all efficiency of 55 percent						
700	42.9	15	10.31	103.4	25,589	480.89
800	46.6	15	12.80	90.7	17,802	503.02
900	50.8	20	15.70	80.4	30.317	603.17
1,000	55.5	20	19.06	72.4	33,107	631.07
1,100	60.7	25	22.93	65.8	36,229	730.93
1,200	66.5	30	27.40	60.3	39,675	820.13

efficiency of 55 percent. The additional cost is 5 to 6 percent except in cases of pumping 700 and 900 gallons per minute, where a larger motor is required, resulting in an increase of about 20 percent in power cost.

The rate of pumping is ordinarily determined by the amount to be applied in a month of maximum demand and the amount of time to be devoted to irrigation. Full-time pumping will bring about the lowest costs, but this is not always possible. An owner is seldom able to devote all his time to spreading water, hence he cannot operate a plant more than one-half time. Some labor aid is usually provided, which allows a longer time of operation. Let it be assumed that an 8-inch depth is to be applied in one month. This may be the depth necessary for a light soil and amounts to 106.7 acre-feet, requiring 26.8 days when pumping at the rate of 900 gallons per minute. A 6-inch average application would be more nearly the requirement for an average soil, amounting to 80 acre-feet and requiring 20.1 days at the 900 gallon per minute rate, 22.6 days at the 800 gallon per minute rate, and 25.9 days at the 700 gallon per minute rate. The lowest rate of discharge is adequate and should be the one selected. The saving in power for the smaller rates of discharge is obvious; and further, the smaller capacity pump will have a lower initial cost.

When the approximate rate has been decided upon the head-capacity curve for the pump that appears to fit may be plotted on the same diagram with the well curve. Where the two curves intersect is the point of head and discharge at which the pump will operate. This is shown at point *M* in figure 23.