A PUMPING MANUAL
for
IRRIGATION
and
DRAINAGE
by JOHN W. WOLFE

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Oregon State College
Corvallis

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Foreword

There are many pumping installations in Oregon doing irrigation or drainage work that have been installed apparently without any engineering assistance whatsoever. The result, in most cases, is an inefficient system which uses too much power for the work it does. The power cost may be three or four times as high as it needs to be.

In addition to the high power cost, some of these pumps are not large enough to supply the amount of water needed. Others, especially those used in sprinkler irrigation, do not develop sufficient pressure to do a good job of irrigating. Adequate pressure and adequate capacity are both important to successful pump operation. Obtaining both requires the application of engineering principles.

The farmer who buys a pump usually cannot escape the cost of the engineering work required for a good selection and installation. If he does not pay it to the dealer as a part of the price of the pump, he will pay for it either in increased power cost or in inadequate pumping results or both.

The dealer who sells the pumping equipment seems to be in the best position to do the required engineering work for the selection. To do so, however, he must have considerable information from the farmer.

This Pumping Manual contains the essential information that a farmer might be required to supply to a dealer from whom he is buying a pump. In addition, it contains enough information for a farmer or farm adviser to check a selection made by the dealer. Thus it is a tool by which a farmer can demand that a dealer make a good selection. He will then be buying a pump not so much for the brand name as for the assurance that it will perform adequately and efficiently. The best-made pump on the market is not satisfactory if it is not selected to fit the job.

Wm. A. Schoofeld

Dean and Director

The author is indebted to F. E. Price, Associate Dean of Agriculture, for reviewing the manuscript and making suggestions which were incorporated into it. In addition, he is indebted to Professor J. B. Rodgers and other members of the Agricultural Engineering Department for consultation and advice on many phases of the subject. In the early stages of preparation, some helpful suggestions were solicited from R. W. Clarke, of Western Regional Sales Company, Portland, Oregon.
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Definition of Terms Used

Hydraulic terms

1. **Quantity** \( (Q) \). Quantity is the volume rate of flow of water. Common units are gallons per minute \( (\text{gpm}) \), cubic feet per second \( (\text{cfs}) \), and acre inches per hour.

2. **Static suction lift** is the vertical distance in feet from the surface of the source of supply to the center line of the pump.

3. **Total static head** is the vertical distance in feet from the surface of the source of supply to the point of free discharge.

4. **Friction loss** \( (H_f) \) is the number of feet lost in head (or pressure) caused by the frictional resistance of the pipe and fittings to the flow of water.

5. **Velocity head**, usually expressed in feet of water, is the pressure required to set the water in motion.

6. **Total dynamic head** is the total static head, plus all friction losses, plus the velocity head, plus the discharge pressure, if any. This is the total head or pressure against which the pump operates. This figure is used in the formula to compute brake horsepower requirement.

Water measurement terms and conversion factors

1. **Inches of water**. This term as used in this bulletin means the depth of water applied to the soil, measured in inches. It is the same measurement used when we speak of an inch of rainfall.

2. **Acre inch**. An acre inch is the quantity of water required to cover one acre of land one inch deep.

3. **Conversion factors**.
   - \( 1 \text{ cu. ft. per second (cfs)} = \text{very nearly } 450 \text{ gal. per minute (gpm)} \)
   - \( 1 \text{ cfs} = 1 \text{ acre inch per hr. (approx.)} \)
   - \( 1 \text{ acre ft.} = 43,560 \text{ cu. ft.} \)
   - \( 1 \text{ acre inch} = 27,154 \text{ gal.} \)

Pumping terms

1. **Water horsepower** \( (\text{WHP}) \) is the theoretical power required to pump a certain number of gallons per minute against the computed total dynamic head at 100 per cent efficiency.

2. **Brake horsepower** \( (\text{BHP}) \) is the actual power required to run the pump. It is greater than the water horsepower because the pump is not 100 per cent efficient.

3. **Pump efficiency** is found by dividing water horsepower by brake horsepower. Over-all pump and motor efficiency can be found by dividing the electrical horsepower input to the motor by the water horsepower of the pump.
A PUMPING MANUAL for IRRIGATION and DRAINAGE

by JOHN W. WOLFE
Assistant Agricultural Engineer

This bulletin contains a discussion of the pumps most commonly used for irrigation or drainage. Emphasis is placed on centrifugal pumps because they are most common. Others included are deep-well turbines, propeller, and mixed-flow pumps. The characteristics and limitations of each are described. Some of the information included is to be used for determining the total head or pressure required and the total quantity of water required of a pump. These values are both necessary before a pump selection can be made. They necessarily include some technical determinations. Consequently, these portions of the bulletin will be a little more easily understood by those who have some knowledge of hydraulics.

Power sources for pumping are discussed.

In general, the Manual is intended as a guide to efficient application of pumping to drainage and irrigation problems of Oregon farms.

Sources of Power for Pumping

Electric motor

Electricity is the most satisfactory source of power for irrigation and drainage pumping plants. Where electricity is available at reasonable rates, the total annual cost is comparable to that of other sources of power. Nebraska tractor tests have shown that farm tractors operating under maximum loads for two hours produce about 10 brake-horsepower hours per gallon of gasoline. Comparing this with an 88 per cent efficient electric motor, electricity at 2¢ per kilowatt hour will just about equal gasoline at 17¢ per gallon. This is the cost for electricity and gasoline only. It does not include oil, repairs, operating costs, interest, or depreciation. In many cases, these last mentioned items make electric power substantially cheaper than gasoline power.

Features which make electric motors especially desirable for pumping are low initial cost, low cost of upkeep, high efficiency, compactness, their constant speed, and especially their dependability. Since the discharge pressure of a centrifugal pump varies as the square of the speed of rotation, it is extremely important especially for sprinkler irrigation systems that this speed of rotation be kept constant. The speed of an electric motor is determined by the number of cycles of the current and the number of poles on the motor.
Therefore, the manufacturer determines the speed of a direct-connected pump and motor. The speed of a belt-driven pump can be governed by the size of the pulleys.

Dependability of a motor is especially important for sprinkler irrigation. Irrigation systems may require performance 24 hours a day during the period of peak use. With an electric motor, there should be no attendance required of the operator between settings of the sprinklers.

Electric motors are also the most suitable source of power for drainage pumps because they lend themselves to dependable automatic operation.

Single-phase motors are often used for loads up to and including 5 horsepower even though a three-phase motor would be more efficient. This is true if three-phase power is not easily available. Single phase can also be obtained in 7½ horsepower, 10 horsepower, and even larger sizes, but the initial cost of the motors is high and their efficiency low. Three-phase power becomes increasingly preferable in the larger sizes, and it is considered almost mandatory above 5 horsepower.

Three-phase motors are inherently more efficient than single phase. Some of them operate above 90 per cent efficiency. Standard motor sizes for three-phase are 1, 1½, 2, 3, 5, 7½, 10, 15, 20, and 25 horsepower.

**Internal combustion engine**

Wherever electric power is not available at reasonable rates, some type of internal combustion engine is used. Selection of an engine should be based on the local cost of fuel, the initial cost of the engine, the maintenance required for constant operation, the expected life of the engine, the recommended operating speed, the ability to maintain a constant speed, and the ability to maintain a constant load. For continuous service, a heavy-duty engine should not be loaded to more than about 90 per cent of the horsepower indicated by the manufacturer. For an automobile or similar engine, the load should not be more than about 50 per cent of rated horsepower.

In large sizes, diesel engines might be used to advantage over gasoline engines, comparing cost per horsepower hour. Most of the diesel engines, however, require considerable belt or gear reduction to obtain a speed high enough for a centrifugal pump.

**Tractor engine**

An economic survey of sprinkler systems was made in 1946 by the Agricultural Economics Department at Oregon State College. The survey revealed that some farmers in Oregon have found their
tractors an inexpensive source of power, since only a small portion of the depreciation need be charged against pumping. When the pump must be operated continuously, however, it is likely that other farm work will also require the tractor. For this reason it is recommended that irrigation systems have their own power source if at all possible.

Hydraulics of Pumping

Static pressure

Static pressure is the pressure caused by the weight of the water itself. This pressure increases with the depth below the free surface. This can be illustrated by the experience of a deep sea diver. The farther below the surface of the ocean he goes, the more pressure his body is subjected to. At 200 feet depth this pressure is exactly twice that at 100 feet depth. The pressure on the diver's body in pounds per square inch when he is 100 feet deep is the weight in pounds of a column of water 1 square inch in cross section and 100 feet high. To determine this, first find the weight of a column of water 1 foot high as shown in the following illustration.

Figure 1. Factors for converting pounds per square inch to pressure in feet of water.
Let us suppose we have a tin container one square inch at the bottom and one foot high. If we fill this container with water and then weigh the water which it contains, we will find the water weighs 0.433 pound. All of this weight is carried by one square inch at the base. Now, if we have a similar container one square inch at the base, but three feet high instead of one foot, and into this container we pour exactly one pound of water, we find the water fills the container to a height of 2.31 feet. Therefore, we can use these conversion factors to change pounds per square inch (psi) to feet of water and vice versa.

One psi equals 2.31 feet of water.

One foot of water equals 0.433 psi.

The deep sea diver, 100 feet under the surface, is subjected to 100 times 0.433 or 43.3 psi pressure.

If water in a pipe line is not flowing, pressure is distributed undiminished throughout its length. It increases with depth below a free-water surface exactly as it does on the deep sea diver. Thus, if the water surface in a storage reservoir were 100 feet above the faucet in a house, a pressure gauge at the faucet would read 43.3 psi if no water is flowing. Similarly, a pump which was used to fill a reservoir and was located 100 feet below it would have to be capable of developing more than 43.3 psi or no water would flow to the reservoir.

**Atmospheric pressure**

Atmospheric pressure is the pressure which enables us to utilize a suction lift on a pump. It is the height of a column of air 1 inch in cross section and as high as the air on the earth's surface goes. At sea level this is 14.7 psi. Or, using the 2.31 conversion factor, it is 34 feet of water.

Theoretically, a pump should be able to operate with a 34-foot suction lift at sea level. In a centrifugal pump, however, air which leaks past the impeller and through small holes in the pipe and pump casing tends to reduce the effective suction lift. About 18 feet at sea level and 15 feet for most inland elevations is all that a centrifugal pump will lift efficiently. This is reduced to 11 feet at 12,000 feet elevation.

**Head loss in pipes**

Head loss means the loss of pressure in a pipe line through which water is flowing. It can be expressed in pounds per square inch or in feet of water. Most of the head loss in a pipe line is caused by the friction between the flowing water and the inside wall
Table 1. Loss of Head in Feet Due to Friction, per 100 Feet of 15-Year-Old Ordinary Iron Pipe.\(^1\)

<table>
<thead>
<tr>
<th>Rate of flow in gallons per minute</th>
<th>Nominal diameter of pipe</th>
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</table>

\(^1\) Pipe coefficients: The values of friction given in this table are for commercial wrought iron or cast iron pipe of 15 years' service when handling soft clear water. In order to be able to use that table for other classes of pipe, the values taken from the table should be multiplied by a coefficient, selected from the list below, to correspond to the required condition:

- New smooth brass and steel pipe (Durand) coefficient \(=0.6\)
- New smooth iron pipe (Williams & Hazen \(C = 120\)) \(=1.0\)
- 15-year-old ordinary pipe (Williams & Hazen \(C = 100\)) \(=1.0\)
- 25-year-old ordinary pipe (Williams & Hazen \(C = 90\)) \(=1.2\)
- Portable aluminum with couplers (Williams & Hazen \(C = 120\)) \(=0.7\)

Courtesy Goulds Pumps, Inc.
of the pipe. Friction loss varies approximately as the square of the velocity in the pipe, or stated another way, it varies as the square of the quantity of water flowing through the pipe for a given pipe diameter. It also varies for every material from which pipes are made. For a given quantity of flow the friction loss is less for a large pipe than for a small pipe. The reason, of course, is that the velocity is less in the large pipe.

Table No. 1 gives the friction loss in feet (of water) per 100 feet of ordinary iron pipe for variable quantities of flow and sizes of pipe. The following example illustrates the use of the table.

Given: A 4" standard pipe 10' long used for pump suction. Pump discharge is 100 gpm.

Required: Head loss due to friction, $H_f$.

Solution: In Table No. 1, under 4" pipe and opposite 100 gpm read 1.22 feet per 100-foot pipe. For 10 feet, $H_f$ would be $0.1 \times 1.22$ or 0.122 feet head loss due to friction.

Minor head losses in pipe lines are caused by pipe fittings, such as couplings, elbows, T's, and valves. There is also a small entrance loss when the water enters the suction pipe. A sudden enlargement or reduction in the pipe size also causes minor losses. These losses can be found from Figure 2. Table 2 can be used in preference to Figure 2 for standard elbows only. The figures obtained from Figure 2 give the equivalent length of straight pipe which will give the same friction loss as the fitting. These figures for all fittings are added to the total length of straight pipe and then the friction loss for the entire pipe system can be obtained from Table 1.

Table 2. **Friction of Water in 90° Elbows Given in Equivalent Number of Feet Straight Pipe.**

<table>
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<tr>
<th>Size of elbow, inches</th>
<th>3/4&quot;</th>
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<th>2&quot;</th>
<th>3&quot;</th>
<th>4&quot;</th>
<th>5&quot;</th>
<th>6&quot;</th>
<th>8&quot;</th>
<th>10&quot;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction equivalent feet straight pipe</td>
<td>2.2</td>
<td>2.8</td>
<td>5.5</td>
<td>6.5</td>
<td>8.1</td>
<td>12</td>
<td>14</td>
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</table>

**Velocity head**

Velocity head in pipes is not actually a head loss, rather it is that head which is left over after the losses have been subtracted. It is that head which the water still contains when it finally leaves the end of the pipe. Usually it is small and often it can be neglected when computing the total dynamic head. However, if a relatively large quantity of water is being put through a relatively small pipe, the velocity head might be quite large.
Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Pipe.

Note: For sudden enlargements or sudden contractions, use the smaller diameter, d, on the pipe size scale.

Figure 2. Minor friction losses caused by elbows, valves, and other fittings.
An example may help us visualize what velocity head really is. Consider a horizontal pipe leading from a pump, discharging into the air a few inches above the bottom of a ditch or reservoir or tank. The velocity of the water coming from the pipe causes the water to strike the bottom some distance away from the end of the pipe. Now let us place a 90° elbow on the end of the pipe pointing into the air. The water will now discharge straight up for a certain distance before it falls to the ground. If it were not for the frictional resistance of the air, the water would rise from the pipe a distance exactly equal to the velocity head in feet. This principle is illustrated by Figure 3.

Following is the formula for determining the maximum or theoretical velocity head:

\[ h = \frac{v^2}{2g} \]

Where \( v \) = velocity of water through the pipe in feet per second
Where \( h \) = head in feet (velocity head)
Where \( g = 32.2 \) feet per second per second, acceleration due to gravity.

Values for velocity head for various pipe sizes and rates of flow in gallons per minute are found in Table 3.

If the velocity head formula is used in place of the table, it is necessary first to find the velocity in the pipe. The following formula will serve this purpose:

\[ v = \frac{0.408 \times gpm}{D^2} \]

Where \( v \) = velocity of flow in feet per second
Where \( gpm \) = gallons per minute
Where \( D \) = inside diameter of the pipe in inches
<table>
<thead>
<tr>
<th>Rate of flow in gallons per minute</th>
<th>Nominal diameter of pipe</th>
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<td>545</td>
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<td>9.137</td>
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<tr>
<td>590</td>
<td>9.291</td>
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<tr>
<td>595</td>
<td>9.373</td>
</tr>
<tr>
<td>600</td>
<td>9.455</td>
</tr>
</tbody>
</table>

Table 3. VELOCITY HEAD IN PIPES.1

1Velocity heads in feet.
Computing Horsepower Output

The horsepower output of a pump, or water horsepower as it is sometimes called, can be computed from the following formula:

\[ \frac{gpm \times TDH}{3,960} = WHP \]

Where \( WHP = \) water horsepower output (sometimes called theoretical horsepower)

Where \( gpm = \) gallons per minute discharge from the pump

Where \( TDH = \) total dynamic head in feet of water

When the unit of discharge is cubic feet per second instead of gallons per minute, the following formula may be used:

\[ \frac{Q \times TDH}{8.8} = WHP \]

Where \( Q = \) discharge in cubic feet per second

Where \( TDH = \) total dynamic head in feet of water

The total dynamic pumping head is found by adding together the following values, usually expressed in feet of water:

1. Total static head or the difference in elevation from the source of supply to the point of discharge expressed in feet.
2. Friction loss in pipe and fittings expressed in feet.
3. Discharge pressure expressed in feet of water. To obtain feet of water from pounds per square inch, multiply by 2.31. If the pump has a free discharge, this value is 0. If the pump discharges into a sprinkler irrigation system, however, this value may constitute the largest part of the total dynamic head.
4. Velocity head. This value is small and is usually negligible unless a great deal of water is being forced through a small pipe.

Brake horsepower

Brake horsepower can be obtained by dividing water horsepower by pump efficiency expressed as a decimal. This relationship is expressed by the following equation:

\[ \frac{WHP}{Eff.} = BHP \]
It is an expression of the actual power required to pump, including all losses. It does not, however, include the power losses within the motor or engine.

If we include efficiency in the above equation for water horsepower, we have the following convenient equation for brake horsepower:

\[ BHP = \frac{gpm \times TDH}{3,960 \times Eff.} \text{ (expressed as a decimal)} \]

Example:

A centrifugal pump is delivering 500 gpm from a creek to a field 20 feet above the field. There is a free discharge. Friction loss in the pipe is 5 feet. Pump efficiency is 65 per cent. Find brake horsepower (BHP).

Solution:

\[ TDH = 20 \text{ feet} + 5 \text{ feet} = 25 \text{ feet} \]

\[ BHP = \frac{500 \text{ gpm} \times 25 \text{ feet}}{3,960 \times 0.65} = 4.86 \text{ HP} \]

Capacity required for irrigation

Discharge requirements of a pump in an irrigation installation can be estimated by using the water requirement of the crop during the period of peak use. For example, in a moderate climate alfalfa may require 0.2 of an inch per day during its period of peak use. Since one acre inch equals 27,154 gallons, this rate of use may be converted to 3.8 gallons per minute per acre. If we lose about 30 per cent of the water to evaporation, run-off, or penetration below the root zone, then the actual discharge required of the pump will be 3.8 divided by 0.70 or 5.4 gallons per minute per acre that must be supplied by the pump. This figure is for 24-hour operation. If the pump is to be operated only ten hours per day during this period, then the capacity of the pump must be \( \frac{5.4 \times 24}{10} \) or 13 gallons per minute per acre. Therefore if the pump is to irrigate 10 acres of alfalfa operating ten hours per day, it must be capable of supplying 130 gallons per minute. A few other values for peak moisture use for common irrigated crops and maximum yields may be found in Table 4.
Table 4. Peak Moisture Use for Common Irrigated Crops Producing Optimu yields

<table>
<thead>
<tr>
<th>Crop</th>
<th>Cool climate</th>
<th>Moderate climate</th>
<th>Hot climate</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Total usage per day</td>
<td>Rate of usage per acre</td>
<td>Total usage per day</td>
</tr>
<tr>
<td></td>
<td>Inches</td>
<td>Gallons per minute</td>
<td>Inches</td>
</tr>
<tr>
<td>Alfalfa</td>
<td>.15</td>
<td>2.8</td>
<td>.20</td>
</tr>
<tr>
<td>Pasture</td>
<td>.12</td>
<td>2.3</td>
<td>.16</td>
</tr>
<tr>
<td>Grain</td>
<td>.15</td>
<td>2.8</td>
<td>.20</td>
</tr>
<tr>
<td>Potatoes</td>
<td>.10</td>
<td>1.9</td>
<td>.12</td>
</tr>
<tr>
<td>Beets</td>
<td>.12</td>
<td>2.3</td>
<td>.15</td>
</tr>
<tr>
<td>Deciduous orchard</td>
<td>.15</td>
<td>2.8</td>
<td>.20</td>
</tr>
<tr>
<td>Orchard with cover</td>
<td>.20</td>
<td>3.8</td>
<td>.25</td>
</tr>
</tbody>
</table>

1Continuous flow required per acre at 100 per cent irrigation efficiency. Divide this value by estimated irrigation efficiency.

Capacity required for drainage

The capacity required for a drainage pump can be estimated from the number of acres to be drained. On tile drainage systems in western Oregon, main-line tile is often selected large enough to remove one-half inch of rain water in 24 hours. This is about the rate that water will seep down through the soil and into the tile system. Sometimes, however, there is not an adequate outlet at the lower end of a farm for tile drainage. In this case, the tile can be discharged into a sump and a drainage pump installed to lift the water to the ground surface elevation on the other side of the property line. A drainage coefficient larger than one-half inch should be used for this installation because the pump must take care of not only that water which would seep into the tile lines but also that which would normally be removed as surface run-off during that period. From their experience in northern Mississippi, where the total normal yearly precipitation is 3 feet, King and Lynes\(^1\) states that many existing installations are designed to remove 3 inches of water in 24 hours. This figure may be adequate for Oregon for all rains in all years except those of exceedingly heavy precipitation. Expressed in other units, this rate would be 56.5 gallons per minute for each acre in the drainage area. This figure should not be used if the average fall is greater than 25 feet per mile in the watershed or if the watershed is larger than one or two square miles. The discharge from a large watershed or one containing rolling or steep land should be determined by more accurate hydrologic data.

Where drainage pumps are required to pump over or through a dike against tide water or flood stage in a river, a much higher

\(^1\)See Bibliography, page 39.
ESTIMATED FLOWS FROM PIPES

WHEN PIPE IS FULL & Y=6'', Q=1.157D^2X (G.P.M.)
WHEN PIPE IS FULL & Y=12'', Q=0.818D^2X (G.P.M.)

USE EITHER FOLDING RULE OR TEMPLATE WITH "Y" TO 6" OR 12".
FOR SLIGHTLY INCLINED PIPES, MEASURE "X" PARALLEL TO PIPE.
"Y" VERTICALLY.
RESULTS OBTAINED FROM THIS SOLUTION ARE APPROXIMATE.

EXAMPLE: 8" PIPE FLOWING FULL, X=40", Y=6". START IN SCALE A AT 40" WHERE Y=6" CONTINUE THRU 6" IN SCALE C TO 6.58 C.F.S. OR 2962 GALLONS PER MIN. IN SCALE C.

Figure 4. Estimated flow from horizontal pipes.
Table 5.

SOIL CONSERVATION SERVICE

UNITED STATES DEPARTMENT OF AGRICULTURE

ESTIMATING FLOW FROM VERTICAL PIPE OR CASING

THE APPROXIMATE FLOW FROM VERTICAL PIPES OR CASINGS CAN BE DETERMINED BY MEASURING THE MAXIMUM HEIGHT \( H \) IN INCHES TO WHICH THE WATER JET RISES ABOVE THE PIPE, AND INSIDE DIAMETER OF THE PIPE \( D \) IN INCHES.

THE FLOW IN GALLONS PER MINUTE IS GIVEN IN THE FOLLOWING TABLE FOR DIFFERENT SIZES OF STANDARD PIPE AND FOR DIFFERENT HEIGHTS OF THE WATER JETS.

<table>
<thead>
<tr>
<th>HEIGHT (( H )) INCHES</th>
<th>NOMINAL DIAMETER (( D )) OF STANDARD PIPE - INCHES</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2</td>
</tr>
<tr>
<td>2</td>
<td>29</td>
</tr>
<tr>
<td>2.5</td>
<td>32</td>
</tr>
<tr>
<td>3</td>
<td>35</td>
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<td>3.5</td>
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<td>4.5</td>
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<td>5</td>
<td>47</td>
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<td>5.5</td>
<td>49</td>
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<td>6</td>
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<td>6.5</td>
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<td>35</td>
<td>134</td>
</tr>
<tr>
<td>40</td>
<td>144</td>
</tr>
</tbody>
</table>

FOR OTHER PIPE SIZES AND HEIGHTS OF JETS, USE THE FORMULAE:

\[
\text{GAL. PER MIN.} = 5.68 \times C \times D^2 \times \sqrt{H}
\]

\[
\text{CU. FT. PER SEC.} = 0.0126 \times C \times D^2 \times \sqrt{H}
\]

WHERE \( D \) = INSIDE PIPE DIAMETER IN INCHES.

\( H \) = JET HEIGHT IN INCHES.

C = A CONSTANT VARYING FROM 0.67 TO 0.97 FOR PIPES OF 2 TO 6 INCHES IN DIAMETER AND HEIGHTS OF 6 TO 24 INCHES.

COURTESY U.S. GEOLOGICAL SURVEY, MAY 1948

SOUTHWEST REGION

ENGR HANDBOOK SEC. II 6-L.12009-2.21(2)
PUMPING MANUAL

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drainage coefficient must be used. One reason for this is the seepage that comes back through the dike when the water surface is high. Also the pump is operating against its highest head at this time. For large installations refer to U. S. Department of Agriculture Technical Bulletin No. 1008, entitled “Design and Operation of Drainage Pumping Plants.”

Computing Discharge of Existing Installation

For existing installation, pump discharge can be measured or estimated by any one of several methods. If a horizontal pipe is discharging its water six inches or more above the ground, the pump discharge may be estimated from Figure 4. If it is discharging vertically into the air, it may be estimated from Table 5. If the pump can be discharged into a stock tank or other container, the rate can be determined by computing the volume of the container and measuring the time it takes the pump to fill the container. In the case of a sprinkler system, water from a sprinkler can be caught in a pail for a measured period of time and the contents measured. A sprinkler should be selected near the center of a lateral to get an average discharge.

If a pump is discharging into a ditch, the quantity can be measured quite accurately by placing a weir in the ditch. This method and others are described in U. S. Department of Agriculture Soil Conservation Service Circular No. 843, entitled “Measuring Water.”

Volute Centrifugal Pumps

How they work

Centrifugal pumps are of two kinds: volute pumps and turbine pumps. In common usage, however, these two are referred to as centrifugal pumps and turbine pumps.

The basic principle of centrifugal pumps is illustrated in Figure 5. It is easy to see that a bucket on the end of a rope whirled at arm’s length will develop enough pressure in the water to push a stream of water out of the pipe in the bottom. In a centrifugal pump the vanes on the impeller correspond to the man’s arm and the rope. The volute on the pump contains the water just as the bucket does. The principle involved is exactly the same in both cases. As the impeller turns, it forces water to its outer edge or tip. The water lies in the volute under pressure until it travels around to the
Basic Principle of Action which when applied to pump becomes:

- Pump discharge = Hole in bucket
- Impeller = Arm and rope
- Casing = Bucket

Figure 5. Basic principle of centrifugal pumps.

pump's outlet where it is discharged under pressure. The suction pipe introduces the water at the center of the impeller. Figure 6 shows the parts of the volute centrifugal pump.

Types of volute centrifugal pumps

Centrifugal pumps are designed either as horizontal or vertical depending on their mount. The terms apply to the position of the shaft. Horizontal pumps are usually used for irrigation work because they make a compact, easily accessible unit. Vertical pumps require a long shaft. The motor can be thus kept above high water while the pump is operating down in the water.

Three different types of impellers are commonly used. They are open, semienclosed, and enclosed (see Figure 6). The open impeller consists mostly of curved vanes attached to a central hub.
CONSTRUCTION OF VOLUME CENTRIFUGAL PUMP

![Diagram of a side-suction, horizontal, volute centrifugal pump.](image)

**Diagramatic View of a Side-Suction, Horizontal, Volute Centrifugal Pump.**

Figure 6. Three types of impellers in common use.
They are essentially nonclogging, relatively low in efficiency, but are cheap and easy to construct. The semienclosed impeller has a disk covering the side of the impeller opposite the suction. It has a little higher cost and a little higher efficiency, and it operates fairly well in sand. The enclosed impeller has a disk on both sides of the vanes. Of the three, it has the highest cost, the highest efficiency, and usually has the longest life and lowest maintenance. The enclosed impeller, however, clogs more easily.

Centrifugal pumps are classed as either single-suction or double-suction depending on whether water enters from one side or both sides. The single-suction impeller is subject to end thrust toward the suction line. This thrust is sometimes reduced by drilling holes through the impeller called "hydraulic balance." These holes will transmit some of the suction to the backside of the impeller. The double-suction pump is actually two single-suction pumps with impellers fastened together. End thrust is eliminated since the suction line enters both sides. This principle is best adapted to large capacity horizontal pumps.

The parts of a pump and how they work

The volute is the spiral-shaped casing surrounding the impeller. It increases in size and in clearance from the tip, or outside edge, of the impeller in the direction of rotation of the impeller. The purpose of the spiral is to accommodate a greater quantity of flow as the water nears the pump discharge.

Unlike positive types of pumps, centrifugal pumps have relatively few wearing parts. Bearings are usually bronze or Babbit and may be lubricated by oil, grease, or water. It is important that the manufacturers' specifications for the lubricants be followed closely. The bearing next to the volute is often provided with a water seal to prevent the pump from losing its suction. This is accomplished by diverting a small flow of water from the pressure side of the pump to the packing on the bearing. The water fills the space between the packing and the shaft and keeps out the air. It also serves to cool and lubricate the shaft and bearing.

In order to maintain the highest efficiency using the enclosed-type impeller, it is necessary to provide a seal between the high pressure and the low pressure of the impeller to prevent bypass. This is accomplished on the volute pump with a wearing ring. The ring, made of softer material than the impeller, can be easily replaced and at low cost. It saves the wear on the expensive impeller. It must be replaced promptly, however, when it becomes worn or the pump will operate at a reduced efficiency.
Impellers commonly used on irrigation pumps are "backward curves." The amount or degree of curve is selected to give the required or desired characteristics relative to horsepower, head, discharge, or efficiency.

Pump size is commonly designated by the size of the discharge. For example, a 3" pump is one threaded to take a 3" pipe at the discharge. If it is connected directly to an electric motor, the horsepower of the motor is also quoted to designate size.

**Shutdown head**

Since volute centrifugal pumps are nonpositive in action, the discharge valve may be closed completely while the pump is operating at rated speed without doing damage to the pump. However, the shutdown head is the maximum head which the pump will produce. The shutdown head can be computed approximately from the following formula:

\[ H_s = \frac{v^2}{2g} \]

Where \( H_s \) = the maximum theoretical shutdown head in feet of water

Where \( v \) = the tip speed of the impeller in feet per second

Where \( g \) = acceleration due to gravity or 32.2.

The tip speed of the impeller or \( v \) in the above equation can be found from the following equation:

\[ v = \frac{d \times rpm}{229} \]

Where \( d \) = diameter of impeller in inches

Where \( rpm \) = revolutions per minute of pump impeller.

Combining the two above equations, shutdown head can be very closely calculated by the formula:

\[ H_s = \left( \frac{d \times rpm}{1,840} \right)^2 \]

**Pumping head**

The maximum pumping head is limited by the frictional resistance resulting when the velocity at the impeller becomes excessive. Below this maximum the head varies directly as the diameter of the impeller and directly as the square of the speed in rpm. It also
varies directly as the number of stages. That is, two stages make twice the head, three stages three times, and so forth.

The following computations illustrate the relationship between head and speed. If a pump develops a head of fifty feet at 1,200 rpm, theoretically it will develop at 1,800 rpm:

\[
\left( \frac{1,800}{1,200} \right)^2 \times 50 = 112.5 \text{ feet head.}
\]

Actually it will be slightly less because of increased friction.

According to Frost,\(^1\) the maximum practical head for a centrifugal pump is about 150 to 200 feet per stage. Pump manufacturers build pumps that develop heads somewhat higher, but any large increase may decrease efficiency.

**Suction lift**

The maximum practical suction lift is about 15 feet for a centrifugal pump. They will lift a little higher but the pump efficiency is greatly reduced beyond 15 feet, or a maximum of 18 feet at sea level. This includes the friction loss in the suction pipe. Therefore, the suction pipe should be relatively large to reduce this friction. That is the reason that the intake fitting on a pump is often larger than the discharge fitting. For sprinkler systems a rule of thumb sometimes followed is to use the same size suction pipe as the first section of mainline or supply line pipe in the sprinkler system.

In addition to reducing the pump capacity, excessive suction lift causes noisy operation, vibration, and pitting of the impeller and other parts of the pump. This phenomenon is called cavitation. It is caused by small bubbles of water vapor which form under the high suction. When the bubbles pass into the high pressure side, they collapse very rapidly. Water rushes in where the bubbles were. It is the force of this rushing water that causes the noisy operation and wears the holes in the parts.

**Pump discharge**

Except for the restrictive effect of the pump casing on increased discharge, the discharge varies directly as the width of the impeller, directly as the diameter of the impeller, and directly as the speed of the impeller. These relations are more nearly true for small increments than for large increments.

In the above example for pumping head variations, if the pump discharge was 400 gpm at 1,200 rpm, then the discharge at 1,800 rpm

\(^1\)See Bibliography, page 39.
would be \( \frac{1,800}{1,200} \times 400 \text{ or } 600 \text{ gpm} \). If an impeller one inch in width is substituted for a like impeller one-half inch in width, it will approximately double the capacity of the pump. If an impeller nine inches in diameter is substituted for a like impeller eight inches in diameter, the capacity will be increased approximately by the fraction \( \frac{9}{8} \).

**Power requirement**

Centrifugal pumps with "backward curved" impellers are characteristically nonoverloading. That is, if the head against which the pump is operating is increased, such as would happen if the discharge valve were partially closed, the horsepower requirement of the pump would decrease. As a result if there is any danger of overloading a centrifugal pump, it would be when it is operating at a head too low rather than at a head too high. For this reason it is sometimes necessary to close the discharge valve before the pump switch is turned on.

If the pump speed is changed, the power requirement is changed in proportion to the cube of the speed. Again using the above example, when the pump speed was increased from 1,200 to 1,800 rpm, the power requirement was increased by the fraction, 

\[ \left( \frac{1,800}{1,200} \right)^3 \text{ or } 3.37 \text{ times.} \]

**Pump efficiency**

Centrifugal pumps characteristically have a high efficiency over a relatively wide range of operating conditions. Each pump, however, has a certain head and discharge where it operates at its highest efficiency. If the head is changed from this value, either increased or decreased, it will cause the efficiency to decrease. For this reason pump specifications used in selecting a pump include operating head, discharge requirements, and minimum acceptable pump efficiency. It is not sufficient to buy a pump solely on the size of the discharge pipe and the horsepower of the motor. Such a selection could result in one that could not deliver a drop of water if the discharge head were higher than the shut down head of the pump. If it did deliver water, the efficiency likely would be low. It is false economy therefore to buy a second-hand pump for an extensive irrigation or drainage job unless it can be determined that the operating efficiency will be fairly high.
Figure 7. Typical performance curves of centrifugal pumps.

Figure 8. Effect on head and discharge rate when speed is increased from 1,200 to 1,800 rpm.
Other factors which affect pump efficiency are the manufacturer's design, the mechanical condition, and the speed of operation. Wear on the vanes or suction rim of the impeller or on the wearing rings will reduce the pump efficiency.

**Pump performance curve**

The skeleton curves in Figure 7 are intended to show the characteristic shape of the head-discharge curve, the efficiency curve, and the brake-horsepower curve of a centrifugal pump. It can be seen that the discharge increases as the head is decreased, but that the relationship is not a straight line function. At shutdown head, the efficiency is zero because no work is being done. From there the efficiency rises to a maximum and then reduces to zero at zero head. The brake-horsepower curve is above zero at shutdown head because of the frictional resistance in the pump. For most pumps it increases slightly with increased discharge to a point past the point of maximum efficiency and then tends to level off or even start down.

Figures 8, 9, and 10 are head discharge curves. They illustrate the effect of various operating conditions.

**Priming centrifugal pumps**

Because of their nonpositive action, centrifugal pumps have to be primed. They will not lift water from a source of supply unless the pump casing and the suction pipe are both full of water. The only installation where priming is no problem is one in which there is a positive head on the suction side of the pump. This would cause the pump to remain full of water after it had been shut off.

Damage can result to the pump if it loses its prime and continues to run. Since at least one of the bearings is usually dependent upon water for lubrication and cooling, these bearings will burn out if the pump is not shut off when it loses its prime. It is advisable, therefore, if the water supply is not dependable, to install a pressure switch or float switch some place in the system which will shut off the motor in case the pump loses its prime.

Methods of priming often include the use of a foot valve on the bottom end of the suction pipe. The pump can then be primed from an outside supply by closing the discharge valve and opening the air vent in the casing. If no outside supply is available, a priming pump may be used by closing the discharge valve and the air vent. A third possibility exists if there is water under pressure in the discharge pipes. This water could be bypassed around a discharge check valve.
Figure 9. Effect on head and discharge rate when one stage is added to a one-stage centrifugal pump.

Figure 10. Effect of impeller width on head and discharge.
PRIMING SETUP FOR CENTRIFUGAL PUMP WITH HORIZONTAL BOTTOM DISCHARGE

CLOSE DISCHARGE GATE VALVE KEEP AIR VENT VALVES CLOSED, OPEN VALVE IN LINE TO PRIMING PUMP. EXHAUST AIR FROM PUMP AND SUCTION PIPING UNTIL WATER FLOWS FROM PRIMING PUMP. CLOSE VALVE IN PRIMING LINE, START PUMP, OPEN DISCHARGE GATE VALVE.

Figure 11. One possible arrangement for a priming pump on a centrifugal pump.

If steam, compressed air, or water under pressure are available, they can be used to create a suction for priming by employing the jet principle. The jet would be installed above the level of the pump so that its suction could fill the casing.

Dry vacuum pumps can be used for priming. They require the use of air release valves or air separators to keep the water out of the vacuum pump. They can be installed for automatic operation to insure that the prime will not be lost.

Advantages of centrifugal pumps
1. High efficiency obtainable
2. High discharge rate possible
3. Simplicity and cheapness
4. Ease of installation
5. Adaptability to different speeds
6. No excessive pressure with valve closed
7. Almost all end thrust can be eliminated
8. Nonoverloading with increased heads.

Limitations of centrifugal pumps
1. Available head per stage is limited
2. Suction lift is limited to 15 feet maximum at average conditions
3. Susceptible to losing prime
4. There is danger of damage to pump if it loses prime
5. May be some danger of overloading if head is decreased
6. Requires more space than a turbine
7. Efficiency will drop if operating conditions differ greatly from those for which the pump was selected
8. Wearing rings must be replaced when worn
9. Water seal is required on some bearings to hold suction

Trouble check list¹

Investigation shows that the majority of troubles with centrifugal pumps result from faulty conditions on the suction side. Except for mechanical trouble, nine times out of ten, this is where to look for the cause.

1. No Water Delivered
   (a) Priming—casing and suction pipe not completely filled with liquid
   (b) Speed too low*
   (c) Discharge head too high—check vertical head (particularly friction loss)
   (d) Suction lift too high (suction pipe may be too small or long causing excessive friction loss). Handling cold water, total lift including friction loss in suction pipe should not exceed 15 feet, or in any case be more than \( \frac{3}{4} \) total head. Check with gauge.
   (e) Impeller or suction pipe or opening completely plugged up.
   (f) Wrong direction of rotation.
   (g) Air pocket in suction line.
   (h) Stuffing box packing worn—or water seal plugged—allowing leakage of air into pump casing.
   (i) Air leak in suction line.

¹List reprinted here through the courtesy of Goulds Pumps, Inc.

*When connected to electric motors, check whether motor is across the line and receives full voltage and whether wiring is correct. When connected to steam turbines, make sure that turbine receives full steam pressure.
(j) Not enough suction head for hot water or volatile liquids. Check carefully as this is a frequent cause of trouble on such service.

2. **Not Enough Water Delivered**

(a) Priming—casing and suction pipe not completely filled with liquid
(b) Speed too low*
(c) Discharge head higher than anticipated—check, particularly friction loss
(d) Suction lift too high (suction pipe may be too small or long causing excessive friction loss). Handling cold water, total lift including friction loss in suction pipe should not exceed 15 feet nor in any case be more than \( \frac{2}{3} \) total head. Check with gauge.
(e) Impeller or suction pipe or opening partially plugged up
(f) Wrong direction of rotation
(g) Air pocket in suction line
(h) Stuffing box packing worn—or water seal plugged—allowing leakage of air into pump casing.
(i) Air leak in suction line
(j) Not enough suction head for hot water or volatile liquids. Check carefully as this is a frequent cause of trouble on such service
(k) Foot valve too small
(l) Foot valve not immersed deep enough
(m) Mechanical defects:
   - Wearing rings worn
   - Impeller damaged
   - Casing packing defective.

3. **Not Enough Pressure**

(a) Speed too low*
(b) Air in water
(c) Measure impeller diameter
(d) Mechanical defects:
   - Wearing rings worn
   - Impeller damaged
   - Casing packing defective
(e) Wrong direction of rotation
(f) Be sure pressure gauge is in correct place. Not on top of casing.

* When connected to electric motors, check whether motor is across the line and receives full voltage and whether wiring is correct. When connected to steam turbines, make sure that turbine receives full steam pressure.
4. **Pump Works for a While and Then Quits**
   (a) Leaky suction line
   (b) Stuffing box packing worn—or water seal plugged—allowing leakage of air into pump casing
   (c) Air pocket in suction line
   (d) Not enough suction head for hot water or volatile liquids. Check carefully as this is a frequent cause of trouble on such service
   (e) Air or gases in liquid
   (f) Suction lift too high (suction pipe may be too small or long causing excessive friction loss). Handling cold water, total lift including friction loss in suction pipe should not exceed 15 feet nor in any case be more than \( \frac{1}{3} \) total head. Check with gauge.

5. **Pump Takes Too Much Power**
   (a) Speed too high
   (b) Head lower than rating: pumps too much water
   (c) Liquid heavier than water. Check viscosity and specific gravity
   (d) Mechanical defects:
      - Shaft bent
      - Rotating element binds
      - Stuffing boxes too tight
      - Pump and driving unit misaligned
   (e) Wrong direction of rotation.

6. **Pump Leaks Excessively at Stuffing Box**
   (a) Packing worn or not properly lubricated
   (b) Packing incorrectly inserted or not properly run in
   (c) Packing not right kind for liquid handled
   (d) Shaft scored.

7. **Pump Is Noisy**
   (a) Hydraulic noise-cavitation, suction lift too high. Check with gauge
   (b) Mechanical defects:
      - Shaft bent
      - Rotating parts bind, are loose or broken
      - Bearings worn out
      - Pump and driving unit misaligned.
Turbine Pumps

Construction

Turbine pumps use a bowl instead of a volute to change the velocity head to pressure head. This is the chief difference between turbines and volute pumps. Inside the bowl are vanes which guide the water from the tip of the impeller to the suction openings of the impeller above. Each bowl assembly consists of an impeller, a case, and a bearing. For deep wells several bowls or stages are placed one on top of another to obtain the required discharge head. They are placed in the water at the bottom of the pump shaft. In irrigation work it is frequently called “deep-well turbine” because of its adaptability to pumping from deep wells with small casings.

Impellers used are either enclosed or semienclosed. See Figure 12 for detail. Enclosed impellers may permit higher efficiency, but they require a seal to prevent bypassing of water from the high-pressure side to the low-pressure side. Wear occurs at the suction opening or the “eye” of the impeller. Semienclosed impellers have the outer edge of the vane open. No seal is required but there is wear on the edges of the impeller vanes. They do handle sandy water better than enclosed impellers.

Because of the long shaft required for a deep well it is necessary to place bearings intermittently along the shaft to prevent vibration and to keep the shaft aligned. These are lubricated either by water or oil. If oil-lubricated they must be sealed against water. A thrust bearing on top of the shaft carries the total weight of the impellers and shaft and the thrust of the pressure head against the impellers.

Power is supplied to the unit at the surface. Often a direct-connected motor is used for speeds up to 3,500 rpm. Belt or gear drive may be used either with an electric motor or with an internal combustion engine.

The submersible pump is a variation which has its power unit located below the impeller in the well. A specially designed electric motor is required in order to fit inside the well casing. Water can be sealed out of the motor by a mercury column. The motor has the same diameter as the bowl but it is much longer than an ordinary motor. Electric lines leading to it are waterproofed and lead-covered and placed outside the discharge pipe. The motor is cooled by the water surrounding it. This arrangement gets away from the inefficiency and the crystallization problem of a long shaft.

Turbine impellers are placed below the static water level in the well. They must also be low enough so that the drawdown does not exceed the suction limit below the impellers. In practice, they are usually placed below the lowest drawdown.
(A) TURBINE PUMP IMPELLERS

- Impeller
- Rubber seal
- Curved vanes
- Semi-open impeller
- Vanes for impeller below the one shown.
- Closed impeller
- Enclosed end-seal
- Sealed by rim of impeller and case.
- Seal between suction and discharge is unnecessary.

(B) DEEPWELL TURBINE PUMP
WITH OPEN SHAFT AND BEARINGS.

Figure 12. Turbine pump impellers.
Turbine pumps are very similar to volute pumps in their operating characteristics; consequently, this discussion contains only instances where the characteristics differ.

**Head**

The maximum head per stage is limited because of the well diameter. The maximum practical head is about 30 to 60 feet per stage. Additional head is obtained by adding stages. If discharge remains the same, three stages will build up three times as much head as one stage. Enough stages can be built up to pump successfully from wells several hundred feet deep.

**Discharge**

Like volute pumps, turbine pumps are capable of producing a high discharge. Bypassing is a little more serious in a turbine pump because it has no replaceable wearing ring, and it is in an inaccessible position in the well. Wear is therefore more of a factor in reducing the discharge.

Discharge can be reduced slightly on semienclosed and on some enclosed impellers by raising the position of the impeller. In this case, bypass is intentionally increased to reduce the discharge. It is accomplished by raising the entire shaft.

**Efficiency**

Efficiency curves for turbine pumps are very similar to those of volute pumps if they are operated at the designed speed. Turbines cannot operate, however, at a high efficiency over as wide a range of speed as can volute pumps. The reason for this is that a high efficiency is possible only if the vanes in the bowl are in line with the flow of water as it leaves the tip of the impeller. If the speed of the impeller is changed, the direction of flow of water leaving the impeller is also changed. This will cause turbulence against the vane and will result in reduced efficiency.

**Deterioration of turbine impellers**

Three processes which deteriorate turbine impellers are galvanic action, erosion, and cavitation. Galvanic action is a corrosive action between two different metals in the presence of impure water. Corrosion increases bypassing and increases friction, thus lowering the pump efficiency. It may be necessary to select an impeller material on the basis of impurities in the water. Since the impellers are located under the water surface, this action continues even when the pump is not in operation.
Erosion of the impeller may occur if the water contains sand or other abrasives. If possible, erosion-resistant material should be selected for the impeller when these conditions are known to exist.

Cavitation is a wearing away of the impeller vane due to high velocity of the water. It is a phenomenon not associated with erosion or galvanic action. It can be very critical if the pump is operated at excessive speed. Critical speed for turbines is about 4,000 rpm.

**Advantages of turbines**

1. Adapted to high lift, especially for deep wells
2. High discharge rate possible.
3. High efficiency.
4. Adapted to different water levels.
5. Not much chance of losing prime.
6. No excessive pressures developed when discharge valve is closed.

**Limitations**

1. Higher initial cost and maintenance than volute pumps.
2. Difficult to install and repair.
3. Thrust must be carried by shaft and bearings.
4. Bearings on shaft require a water seal if they are lubricated by oil.
5. Subject to more or less erosion and corrosion.

**Propeller and Mixed-Flow Pumps**

**Uses**

Wherever a large quantity of water is to be pumped against a rather low head, especially one less than 3 feet, a higher efficiency can be obtained from a propeller pump than from a centrifugal pump. For slightly higher heads but still below those usually produced by centrifugal pumps, mixed-flow pumps may be used. Both of these pumps are used extensively in drainage work. They are especially adapted to lifting water over dikes or from a drainage ditch to a shallower drainage ditch or for draining lakes or swamps. They are also used for irrigation work such as pumping from a large canal into a lateral or pumping from a lake or river where the adjoining land is only slightly above water level. When irrigating land adjacent to a small stream, a pump of this type could be used in lieu of a gravity diversion farther up stream. The gravity diversion would probably be cheaper, however, unless a ditch right were difficult to obtain.
Construction

Propeller-type pumps have a rotating screw similar to the blades on a propeller used on ships. Each blade of the impeller helps to impart a velocity to the water in the direction of the shaft. Figure 13 shows the shape of the impeller blades and their position in relation to the rest of the pump. The axis of the pump is vertical and the motor or pulley is mounted above the pump. The mixed-flow type impeller is also shown in Figure 13.

Mixed-flow type pumps have impeller blades somewhat similar to the propeller or screw pump, but are designed to give an outward thrust to the water in addition to imparting a velocity upward. In other words, the shape of the mixed-flow impeller is between that of the propeller type and the deep-well turbine impellers. Likewise, its operating characteristics are between the other two pumps mentioned. It has just enough outward thrust on the blades to develop a little centrifugal pressure in the bowls. Therefore, it can develop a little higher pressure than the propeller pump.
Characteristics of propeller and mixed-flow pumps

The outstanding characteristics of these pumps are simplicity, straight-line motion, variable speed operation at high efficiency, ability to pump sand with the water, high discharge rates, low lift per stage, and low cost per unit of water discharged. They are non-positive in action.

Their operating characteristics are very similar to centrifugal pumps. One slight difference is that a slight increase in the pumping head causes a larger decrease in the quantity of water delivered. Therefore, if a discharge valve is closed slightly the quantity of water delivered drops very rapidly, and at the same time, the discharge pressure rises only slightly. Another difference is the slope of the BHP curve. Unlike the centrifugal pump, the horsepower on the propeller pump increases with increasing head. Overload is likely to occur when the discharge valve is nearly closed instead of open, in contrast with the centrifugal pump.

The best performance for propeller pumps usually occurs when they are operating against 3 to 6 feet of lift per stage. If additional stages are needed for higher lifts, they should be added at from 3 to 6 foot intervals along the shaft instead of being placed together at the bottom of the shaft. The mixed-flow type pump can work satisfactorily with 2 or 3 stages at the bottom of the shaft. However, even with the mixed-flow type of pump the efficiency is reduced with each stage that is added. If more stages are required to develop the required head, it would probably be better to select a turbine-type centrifugal pump. One pump manufacturer finds that the efficiency of a two-stage propeller pump is only 0.925 times the efficiency of a one-stage pump. For a two-stage mixed-flow pump, their factor is 0.88. They seldom use more than two stages.

The efficiency of a propeller pump is usually higher when operated at less than one-half of the shutdown head. In other words, if the shutdown head is 8 feet, the point of highest efficiency would probably be about 3 or 4 feet.

Propeller pumps are satisfactory for speeds above the critical speed of centrifugal pumps. Their speed can be varied widely for increased or decreased discharge without affecting the efficiency or head. Since the movement of the water is in the direction of the axis of the screw and there is a large free passage, very little frictional resistance to flow takes place.

Figure 14 shows a typical performance curve of a single-stage propeller-screw pump.
Figure 14. Performance curves on single-stage propeller-screw pump.

Bibliography

ERRATA

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Formula at bottom of page shown as \( H_s = \frac{(d \times \text{rpm})^2}{1,840} \)

Should read \( H_s = \left( \frac{d \times \text{rpm}}{1,840} \right)^2 \)