4

Pumps

4.1 Introduction

Pumps are mechanical devices that impart energy to a fluid. Irrigation pumps lift water from one elevation to a higher level, overcome friction losses during conveyance, provide pressure for sprinkler (trickle emission device) operation, and meter (inject) chemicals into irrigation systems. Irrigation pumps use mechanical energy, usually from electric, gasoline, diesel, liquid petroleum gas, or natural gas motors to increase the potential (pressure) and/or kinetic energy of the irrigation water. In some parts of the world, human and animal power is used to pump irrigation water.

Pumps may be classified as rotary, reciprocating, or centrifugal. Rotary pumps use gears, vanes, lobes, or screws to trap and convey fluid from the inlet to outlet sides of the pump. Pumps that use the back and forth motion of mechanical parts, such as pistons or diaphragms, to pressurize the fluid are known as reciprocating pumps. Centrifugal pumps use the centrifugal force imparted to the fluid by one or more rotating elements (called impellers) to increase the kinetic and pressure energy of the fluid.

Rotary and reciprocating pumps, often called positive displacement pumps, are not normally used to pump irrigation water, primarily because of their relatively low discharge capacity and susceptibility to sediment laden water. They are, however, used to meter (inject) chemicals into irrigation systems. Centrifugal pumps are used for both chemical injection and irrigation pumping. Since the remainder of this chapter deals with irrigation pumping, only centrifugal pumps are considered.

This chapter describes the operation, performance, and selection of centrifugal pumps. Sufficient information is presented to allow the reader to

1. become familiar with the common types of centrifugal pumps used with farm irrigation systems,

2. define the parameters typically used to characterize centrifugal pump performance,
3. describe the performance characteristics of centrifugal pumps,
4. predict the performance characteristics of two or more pumps connected in series or parallel,
5. compute system curves and use them with pump performance curves to determine the head and discharge at which a pump or system of pumps will operate,
6. evaluate the effect of changes in impeller speed and diameter on pump performance,
7. select the most suitable pump or combination of pumps for a farm irrigation system.

4.2 Centrifugal Pumps

A centrifugal pump consists of a set of rotating vanes, called impellers, enclosed within a stationary housing called a casing. Water is forced into the center (eye) of the impeller by atmospheric or other pressure and set into rotation by the impeller vanes. The resulting centrifugal force accelerates the fluid outward between the vanes until it is thrown from the periphery of the impeller into the casing. The casing collects the liquid, converts a portion of its velocity energy into pressure energy, and directs the fluid to the pump outlet.

Centrifugal pumps are either single- or multistage depending on the number of impellers. Single-stage pumps have only one impeller, while multistaged pumps have several impellers connected in series (i.e., the outflow of the first impeller is directed into the eye of the second, the outflow of the second impeller into the third, etc). Centrifugal pumps are classified as either horizontal or vertical according to the orientation of their axis of rotation. Horizontal pumps are sub-classified according to the location of the suction nozzle (inlet) as end-suction, side-suction, bottom-suction, or top-suction. In addition, pumps are also classified by casing and impeller type.

4.2.1 Casings

Centrifugal pump casings are either of the volute or diffuser type. Two typical volute casings are diagrammed in Figure 4.1. These casings have carefully designed volute (spiral) shaped passages for water that increase in cross-sectional area toward the outlet of the pump. The rate of area increase within a volute casing is normally sufficient to reduce the velocity of the fluid as it approaches the outlet. This increases pressure at the outlet. A majority of single-stage pumps have volute-type casings.

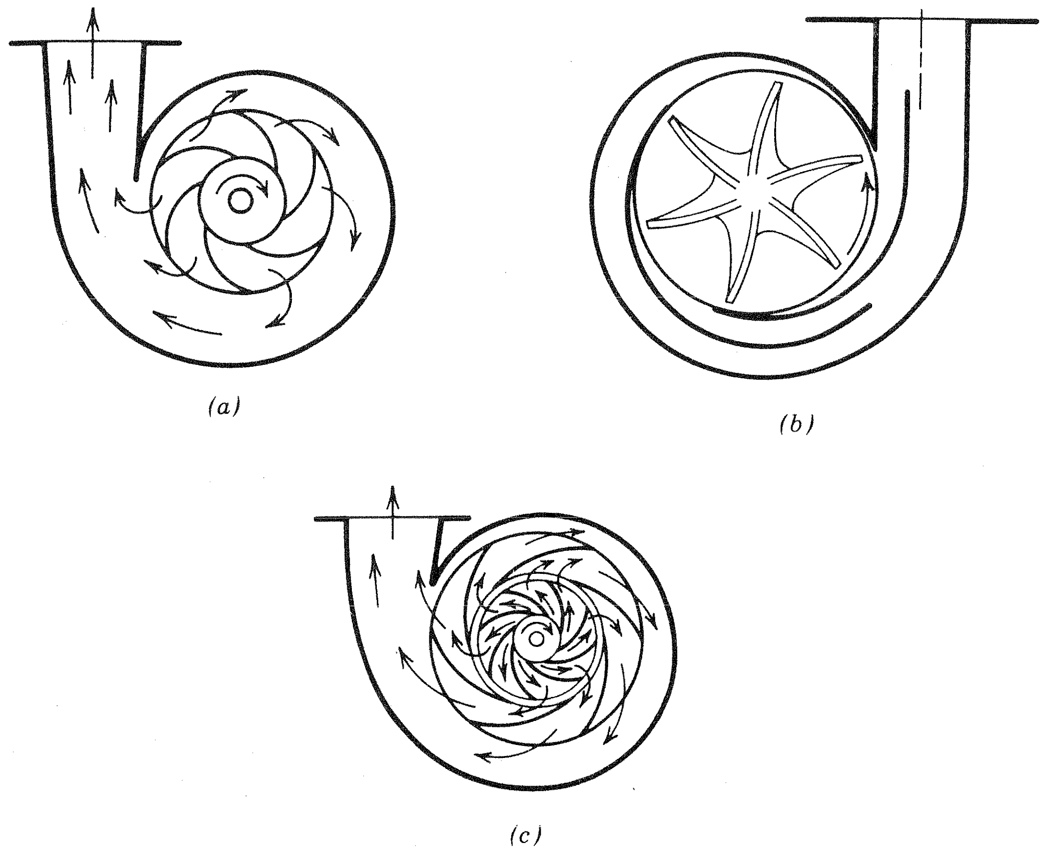


Figure 4.1 Some centrifugal pump casings. (*a*) Single volute casing. (*b*) Double volute casing. (*c*) Diffuser turbine casing.

Casings like the one in Figure 4.1*a*, called single-volute casings, are normally used when pumps operate continuously at or near design capacity (i.e., operated at or near peak efficiency). Double-volute casings (Figure 4.1*b*) should be used when pumps are expected to operate at part-capacity for extended periods of time, because radial forces are more balanced. The double-volute design is also used in situations where the rib that separates the first and second volutes is needed to strengthen the casing. In single volute casings, uniform or near uniform pressures act on the impeller when the pump is operated at design capacity.

Diffuser-type casings have stationary vanes surrounding the impeller that guide water from the impeller into volute or circular shaped casings. Fluid pressure usually rises as it passes through diffusers because of the progressive increase in cross-sectional area between vanes in the direction of flow. Diffuser vanes provide a more uniform distribution of pressure in the casing and are used in multistaged pumps.

Pump casings are either axially or radially split. Pump casings are made of two or more parts that can be "split" (by removing the bolts that hold them together) to provide access to impellers, bearings, seals, and other internal parts. Pumps with casings that can be split in a plane parallel to the axis of rotation are called axially split pumps. In radially split pumps, the casing split is in a plane perpendicular to the axis of rotation.

4.2.2 Impellers

Impellers are classified as radial, axial, or mixed flow depending on the direction of flow through the impeller relative to the axis of rotation (see Figure 4.2). Liquid enters radial flow impellers in a plane parallel to the axis of rotation and is discharged at 90° to it. Impellers that receive and discharge water in a plane parallel to the axis of rotation are known as propeller or axial flow impellers. Mixed flow impellers receive water in a plane parallel to the axis of rotation and discharge it at an angle that is between 0 and 90° to the axis of rotation.

Impellers can also be classified as single- or double-suction. Water enters single-suction impellers through an inlet located on one side of the impeller and flows symmetrically into both sides of double-suction impellers. A double-suction impeller is, in effect, two single-suction impellers arranged back-to-back in a single casing. Double-suction impellers are normally used with axially split casings because of their hydraulic balance in the axial direction and since they normally have low net positive suction requirements (see Section 4.3.2d). Single-suction impellers are preferred when waters with high concentrations of suspended materials are being pumped, for multistaged pumps, and for small pumps, because single-suction impellers have large waterways that are easier to manufacture. Because of low first cost and ease of maintenance, single-suction impellers are utilized in most end-suction radially-split casing pumps.

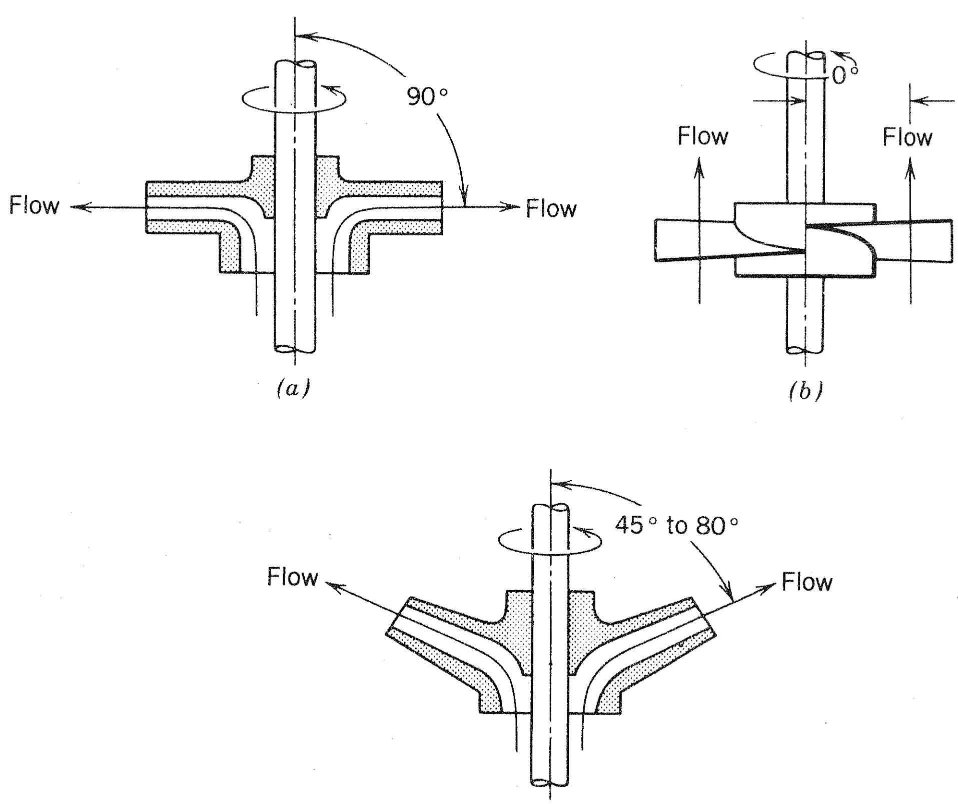


Figure 4.2 Impellers for centrifugal pumps. (*a*) Radial flow impeller. (*b*) Axial flow impeller. (*c*) Mixed flow impeller.

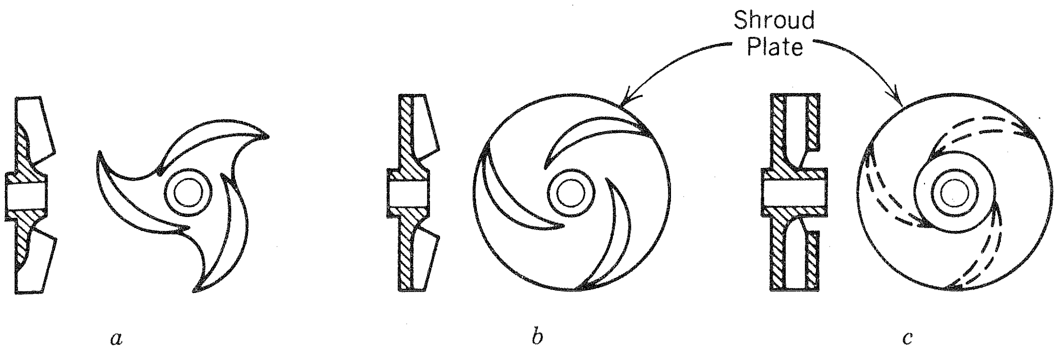


Figure 4.3 (*a*) Open, (*b*) semiopen, and (*c*) enclosed impellers for centrifugal pumps. *Source:* G. O. Schwab, R. K. Frevert, T. W. Edminster, and K. K. Barnes, *Soil and Water Conservation Engineering*, copyright © 1981 by John Wiley & Sons, Inc., New York, p. 370. Reprinted by permission of John Wiley & Sons, Inc.

Impellers are also classified according to their mechanical construction. Enclosed impellers, like the one in Figure 4.3, have shrouds (sidewalls) enclosing the waterways between vanes. Semienclosed (semiopen) impellers have one full shroud (backwall), while open impellers consist only of vanes attached to a hub (without a shroud or sidewall). The vanes of many open impellers are, however, strengthened by ribs or partial shrouds. Open and semienclosed (semiopen) impellers are most suitable for pumping suspended material or trashy water. Enclosed impellers are not normally suitable for use with such waters because the abrasiveness of the suspended materials greatly increases impeller wear and hence, deterioration of pump performance. Enclosed impellers are, however, more widely used with clear waters than open or semiopen impellers because less adjustment is required to maintain efficient operation.

4.3 Centrifugal Pump Performance

4.3.1 Performance Parameters

Capacity, head, power, efficiency, required net positive suction head, and specific speed are parameters that describe a pump's performance. These parameters are discussed in the following sections.

4.3.1a Pump Capacity

The capacity of a pump (Q) is the volume of water per unit time delivered by the pump. In SI units, Q is usually expressed in liters per minute (l/min) or, for larger pumps, cubic meters per second (m^3/s). The corresponding units in the English system are gallons per minute (gpm) and cubic feet per second (cfs).

4.3.1b Head

The head (H) is the net work done on a unit weight of water by the pump. It is given by

$$H = \left(\frac{P}{\gamma} + \frac{V^2}{2g} + Z\right)_{\mathrm{d}} - \left(\frac{P}{\gamma} + \frac{V^2}{2g} + Z\right)_{\mathrm{s}} \tag{4.1}$$

where

P = water pressure (kPa, psi);
γ = specific weight of fluid (kN/m^3, lb/ft^3);
V = water velocity (m/s, ft/s);
g = acceleration of gravity (9.81 m/s^2, 32.2 ft/s^2);
Z = elevation head in meters or feet above a datum. (For horizontal pumps, the datum is a horizontal plane through the centerline of the shaft. For vertical pumps, the datum is a horizontal plane through the entrance eye of the first-stage impeller.)

The subscripts d and s identify the discharge and suction sides of the pump, respectively.

4.3.1c Power

The power imparted to the water by the pump is called water power. Equation 4.2 is used to compute water power (WP).

$$\mathrm{WP} = \frac{QH}{K} \tag{4.2}$$

where

WP = water power (kW, hp);
Q = pump capacity (l/min, m^3/s, gpm, cfs);
H = head (m, ft);
K = unit constant ($K = 6116$ for WP in kW and Q in l/min, $K = 0.102$ for WP in kW and Q in m^3/s, $K = 3960$ for WP in hp and Q in gpm, and $K = 8.81$ for WP in hp and Q in cfs).

4.3.1d Efficiency

Pump efficiency (E_p) is the percent of power input to the pump shaft (the brake power) that is transferred to the water. E_p can be computed using Eq. 4.3:

$$E_p = 100\left(\frac{\mathrm{WP}}{\mathrm{BP}}\right) \tag{4.3}$$

where

E_p = pump efficiency (percent);
WP = water power (kW, hp);
BP = brake power (kW, hp).

4.3.1e Required Net Positive Suction Head

The required net positive suction head ($NPSH_r$) is the amount of energy required to prevent the formation of vapor-filled cavities of fluid within the eye of single- and first-stage impellers. These cavities, which form when pressures within the eye drop below the vapor pressure of water, collapse within higher pressure areas of the pump. The formation and subsequent collapse of these vapor-filled cavities is called cavitation. When these collapses occur violently on interior surfaces of the pump they produce ring-shaped indentations in the surface called pits. Continued cavitation and pitting can severely damage pumps, and must be avoided.

The net positive suction head required to prevent cavitation is a function of pump design and is usually determined experimentally for each pump. Manufacturers conduct laboratory tests to determine $NPSH_r$ values for each pump model they manufacture. Cavitation is prevented when heads within the eye of single- and first-stage impellers exceeds the $NPSH_r$ values published by the manufacturers.

4.3.1f Specific Speed

Specific speed (N_s) is an index to pump performance derived using dimensional analysis. It consolidates a pump's speed, design capacity, and head into one term. N_s is computed from

$$N_s = \frac{NQ^{1/2}}{H^{3/4}} \tag{4.4}$$

where

N_s = specific speed (rpm);
N = pump speed (rpm);
Q = pump design capacity (gpm);
H = design head (ft).

Geometrically similar pumps have similar performance characteristics and identical specific speeds regardless of their size. For example, pumps with radial flow impellers generally deliver relatively small discharges at high heads and have low specific speeds ranging between 500 and 2000 rpm regardless of impeller diameter. Similarly, pumps with axial flow impellers normally deliver relatively large discharges against low heads and have large specific speeds ranging between 5000 and 15,000 rpm for all impeller diameters.

4.3.2 Performance (Characteristic) Curves

The operating properties of a pump are established by the geometry and dimensions of the pump's impeller and casing. Curves relating head, efficiency, power, and required net positive suction head to pump capacity are utilized to describe the operating properties (characteristics) of a pump. This set of four curves is known as the pump's *characteristic curves*. Characteristic curves for a typical single-stage

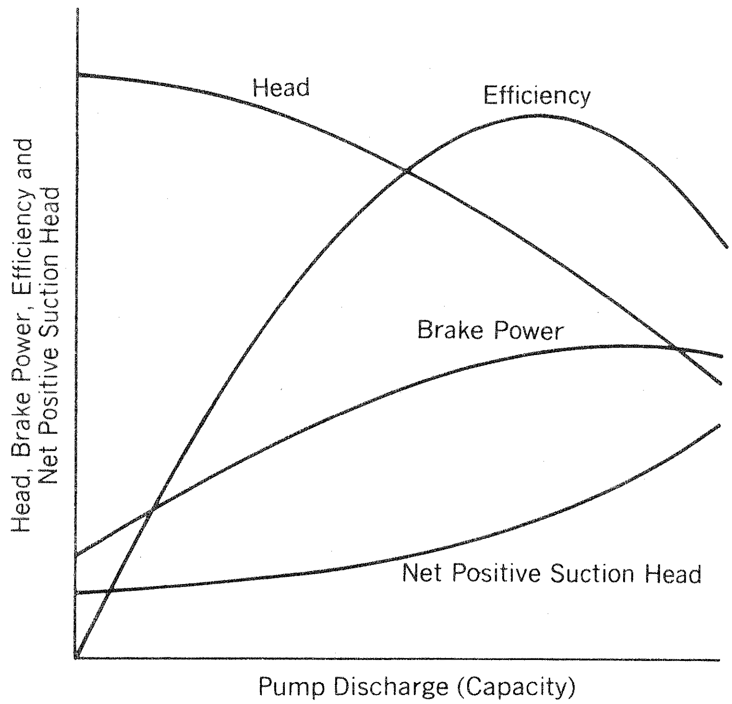


Figure 4.4 Characteristic curves for a single-stage centrifugal pump.

pump are illustrated in Figure 4.4. Characteristic curves for multi-stage pumps are considered in Section 4.3.4.

Pump manufacturers normally publish a set of characteristic curves for each pump model they make. Data for these curves are developed by testing several pumps of a specific model. Some manufacturers' curves represent the average performance of all pumps of a specific model tested, while other manufacturers prepare their curves for the pump having the poorest performance.

4.3.2a Head Versus Pump Capacity

This curve relates the head produced by a pump to the volume per unit time of water being pumped. Generally, the head produced steadily decreases as the amount of water pumped increases. The shape of the H–Q curve varies with specific speed.

Figure 4.5 shows typical H–Q curves for various specific speeds and impeller designs. For radial flow impellers, head decreases only slightly and then drops rapidly as Q increases from zero. Slope changes along H–Q curves for mixed and axial flow impellers are not as dramatic as those for radial flow impellers. Radial flow impellers operating on the flat portion of their H–Q curves work well in situations where head must remain essentially constant as Q fluctuates (as in set-move systems where the number of operating laterals varies during the irrigation season). In situations where a relatively constant Q is desired and H is expected to fluctuate (such as in a well, small stream, or small reservoir), impellers with higher specific speeds will probably perform best.

The head generated when Q is zero (i.e., when the pump is operating against a closed valve) is the shutoff head (see Figure 4.5). For pumps with steadily declining

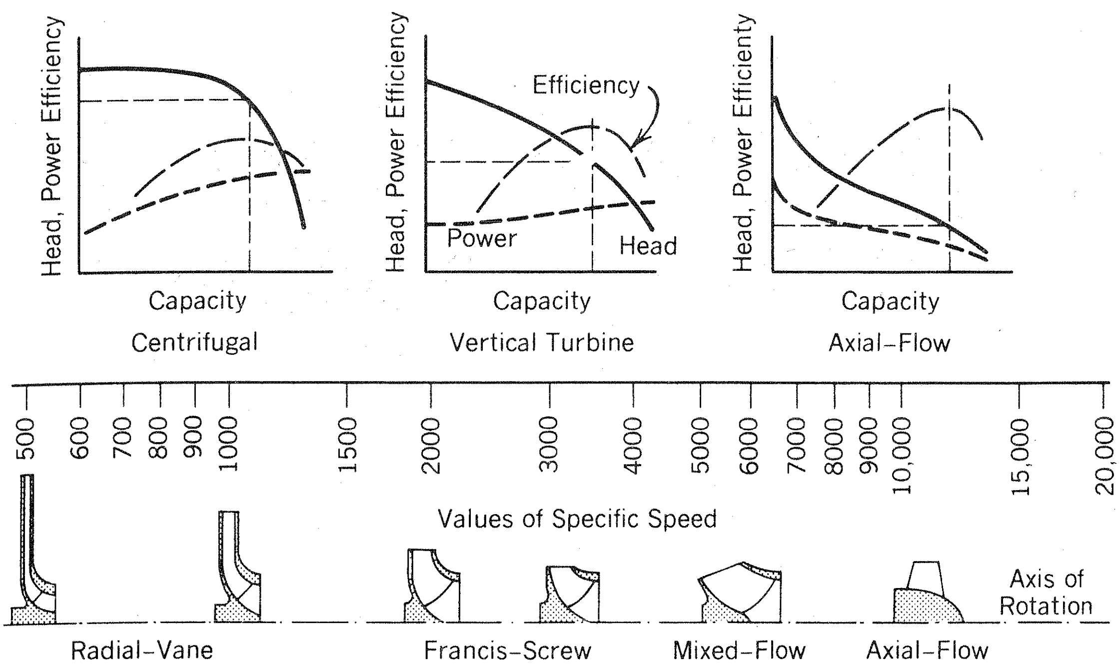


Figure 4.5 Head–discharge curves as a function of specific speed and impeller design. *Source:* R. Walker, *Pump Selection: A Consulting Engineer's Manual,* copyright © 1972 by Butterworth Publishers, Stoneham, MA, 58 pp. Reprinted by permission of Butterworth Publishers.

H–Q curves, the shutoff head is the maximum head and must be known to design piping on the dischargeside of the pump. In such situations, discharge-side piping must be able to withstand the shutoff head when the discharge-side valve is closed. Note that pump efficiency is zero at the shutoff head, since energy is being used to turn the pump.

4.3.2b Efficiency Versus Pump Capacity

An E_p–Q curve for a typical pump is illustrated in Figure 4.4. E_p for a pump steadily increases to a peak, and then declines as Q increases from zero. There is generally only one peak efficiency for a specified impeller. The E_p–Q relationship for a pump is sometimes drawn as a series of envelopes on the H–Q curves of different impeller diameters (see Figure 4.6).

Theoretical efficiencies as a function of specific speed, impeller design, and pump capacity are shown in Figure 4.7. Data in this figure indicate that larger capacity pumps with specific speeds of about 2500 rpm can be expected to have the highest efficiencies.

E_p is also related to the types of materials used in construction, the finish on castings, the quality of machining, and the type and quality of bearings used. For example, impellers with extremely smooth surfaces tend to be more efficient than rougher surfaced impellers.

An E_p–Q curve is usually for a specific number of stages. If a different number of stages is needed for a particular situation, efficiencies must be adjusted up- or downward depending on the number of stages. Manufacturers usually provide

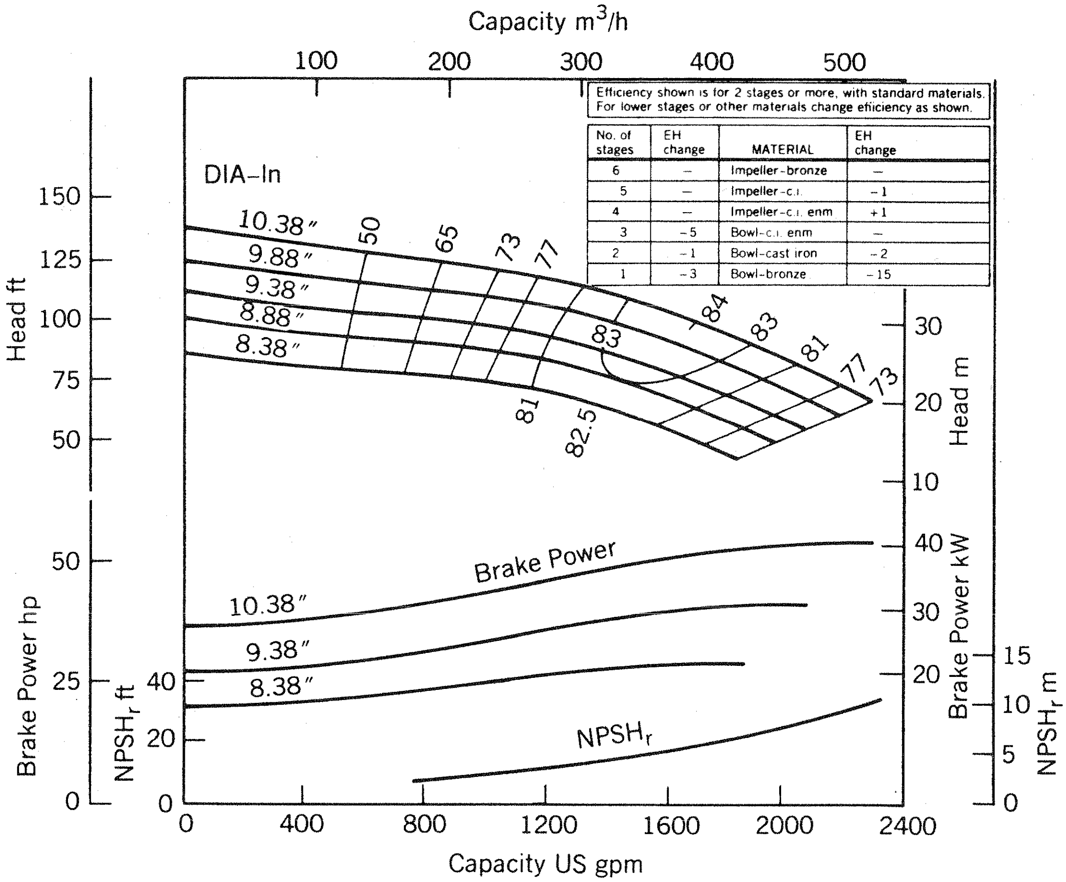


Figure 4.6 Example characteristic curves from a pump manufacturer's catalog.

information for making these adjustments. For example, information in Figure 4.6 indicates that graphed efficiencies must be lowered by three percentage points for a single-stage pump, lowered by one percentage point for a two-stage pump, and would remain unchanged for more than three stages.

4.3.2c Brake Power Versus Pump Capacity

The brake power (BP) versus capacity curve for a pump is derived from its H-Q and E_p-Q curves. An equation for computing BP from H, Q, and E_p is obtained by solving Eq. 4.3 for BP and substituting Eq. 4.2 for WP. The resulting equation is:

$$\mathrm{BP} = \frac{100\ \mathrm{WP}}{E_p} = \frac{(100)(Q)(H)}{(E_p)(K)} \tag{4.5}$$

where K is the unit constant in Eq. 4.2.

The shape of the BP–Q curve depends on the pump's specific speed and impeller design. Figure 4.5 shows that for radial flow impellers, BP generally increases from a nonzero value to a peak and then declines slightly as Q increases from zero. BP increases steadily from a nonzero value as Q increases for mixed flow impellers. For axial flow impellers, however, BP is maximum when Q is zero and

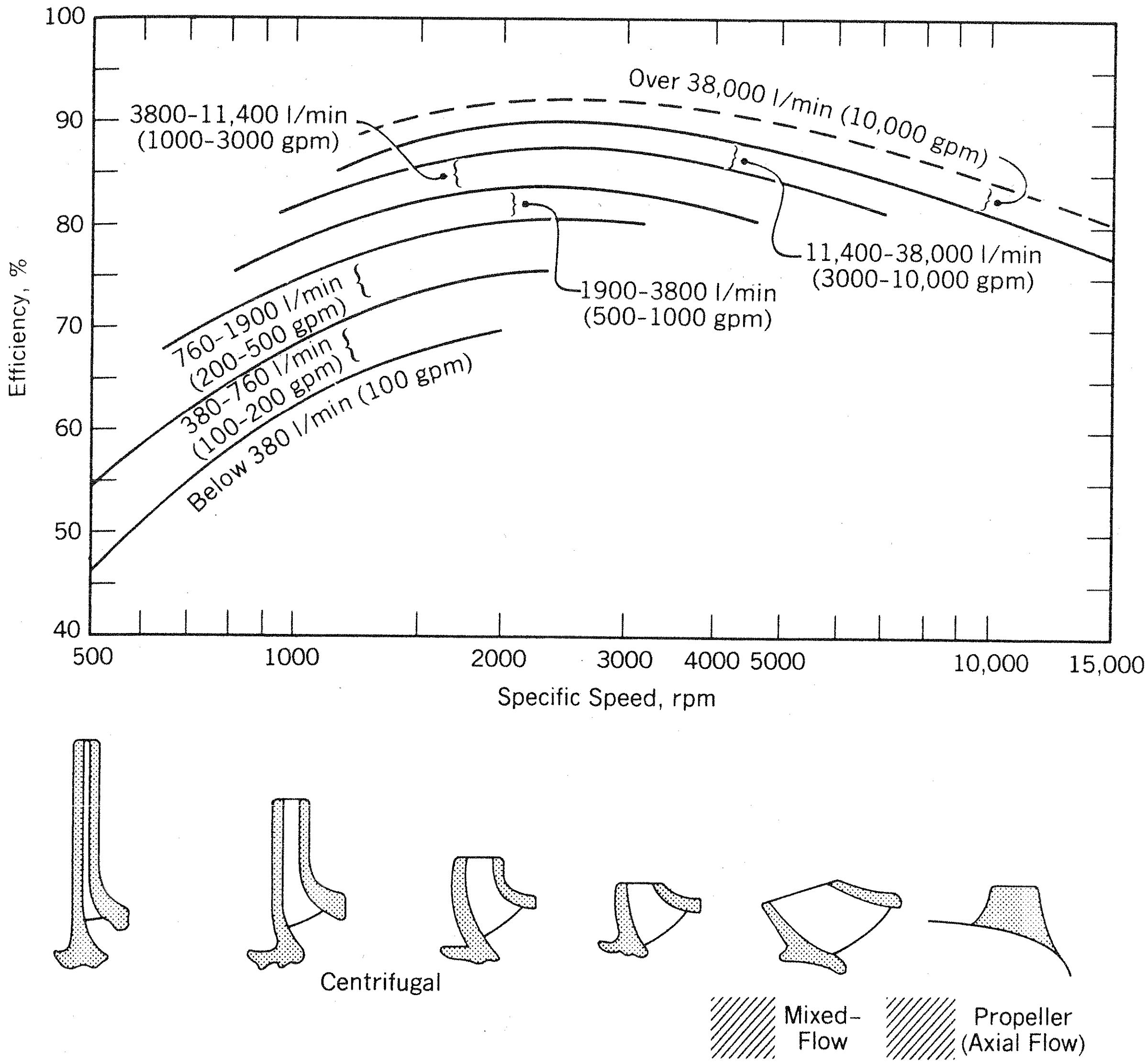


Figure 4.7 Theoretical pump efficiencies as a function of a specific speed, impeller design, and pump capacity. *Source:* G. O. Schwab, R. K. Frevert, T. W. Edminster, and K. K. Barnes, *Soil and Water Conservation Engineering*, copyright © 1981 by John Wiley & Sons, Inc., New York, 368 pp. Reprinted by permission of John Wiley & Sons, Inc.

steadily declines as Q increases from zero. Thus, the discharge side of the pump should be open to the atmosphere when axial flow impellers are started to minimize the start-up load. Similarly, a discharge-side valve should be closed when radial and mixed flow pumps are started.

4.3.2d Required Net Positive Suction Head Versus Pump Capacity

The fourth characteristic curve typically published by manufacturers is the $NPSH_r$ versus Q curve. Figure 4.4 shows that $NPSH_r$ steadily increases as Q increases for a typical radial flow pump. (The actual net positive suction head for a pump installation is discussed in Section 4.4.2.)

4.3.3 Affinity Laws

Changing the diameter and/or speed of an impeller alters its characteristic curves. This allows pump manufacturers to use a single impeller for a variety of head and discharge conditions and pump owners to alter pump performance to match changes in the configuration and/or operation of their irrigation systems.

Changes in impeller performance resulting from changes in pump speed can be estimated using the following equations.

$$Q_2 = Q_1\left(\frac{N_2}{N_1}\right) \tag{4.6}$$

$$H_2 = H_1\left(\frac{N_2}{N_1}\right)^2 \tag{4.7}$$

$$\mathrm{BP}_2 = \mathrm{BP}_1\left(\frac{N_2}{N_1}\right)^3 \tag{4.8}$$

$$(\mathrm{NPSH}_r)_2 = (\mathrm{NPSH}_r)_1\left(\frac{N_2}{N_1}\right)^2 \tag{4.9}$$

where the subscripts 1 and 2 refer to the original and new performance points. The following example demonstrates the use of Eqs. 4.6, 4.7, 4.8, and 4.9.

EXAMPLE 4.1 Estimating Changes in Pump Performance Resulting from a Change in Pump Speed

Given:

- original pump speed is 1750 rpm
- original discharge is 1000 gpm
- original head 300 ft
- original NPSH_r is 12 ft
- original brake power is 100 hp

Required:

Q, H, BP, and NPSH_r for a speed of 2000 rpm

Solution:

$$Q_2 = 1000\text{ gpm}\left(\frac{2000}{1750}\right) = 1143\text{ gpm}$$

$$H_2 = 300\text{ ft}\left(\frac{2000}{1750}\right)^2 = 392\text{ ft}$$

$$\mathrm{BP}_2 = 100\text{ hp}\left(\frac{2000}{1750}\right)^3 = 149\text{ hp}$$

$$(\mathrm{NPSH}_r)_2 = 12\text{ ft}\left(\frac{2000}{1750}\right)^2 = 16\text{ ft}$$

Changes in pump performance due to changes in impeller diameter can be estimated using the following equations.

$$Q_2 = Q_1\left(\frac{D_2}{D_1}\right) \tag{4.10}$$

$$H_2 = H_1\left(\frac{D_2}{D_1}\right)^2 \tag{4.11}$$

$$\mathrm{BP}_2 = \mathrm{BP}_1\left(\frac{D_2}{D_1}\right)^3 \tag{4.12}$$

$$(\mathrm{NPSH}_r)_2 = (\mathrm{NPSH}_r)_1\left(\frac{D_2}{D_1}\right)^2 \tag{4.13}$$

Because trimming (reducing) the diameter of impellers may alter impeller geometry, Eqs. 4.10 through 4.13 approximate changes in pump performance. Eqs. 4.10 through 4.13 are most reliable for diameter changes of less than 20 percent. The following example illustrates the use of these equations.

EXAMPLE 4.2 Estimating Changes in Pump Performance Resulting from a Change in Impeller Diameter

Given:

- Information from Example 4.1
- Original diameter 8 in

Required:

Q, H, BP, and NPSH_r for a diameter of 7.5 in

Solution:

$$Q_2 = 1000 \text{ gpm}\left(\frac{7.5}{8.0}\right) = 938 \text{ gpm}$$

$$H_2 = 300 \text{ ft}\left(\frac{7.5}{8.0}\right)^2 = 264 \text{ ft}$$

$$\mathrm{BP}_2 = 100 \text{ hp}\left(\frac{7.5}{8.0}\right)^3 = 82 \text{ hp}$$

$$(\mathrm{NPSH}_r)_2 = 12 \text{ ft}\left(\frac{7.5}{8.0}\right)^2 = 10.6 \text{ ft}$$

Equations 4.6 through 4.13 are known as the affinity laws. These laws and the procedures demonstrated in Examples 4.1 and 4.2 can be used to generate shifts in entire head–pump capacity curves similar to those in Figure 4.8.

4.3.4 Performance Curves for Pumps Operating in Series

Two or more pumps operate in series when they are linked inlet-to-oulet as shown in Figure 4.9. Series hookups are used when the head required by the irrigation

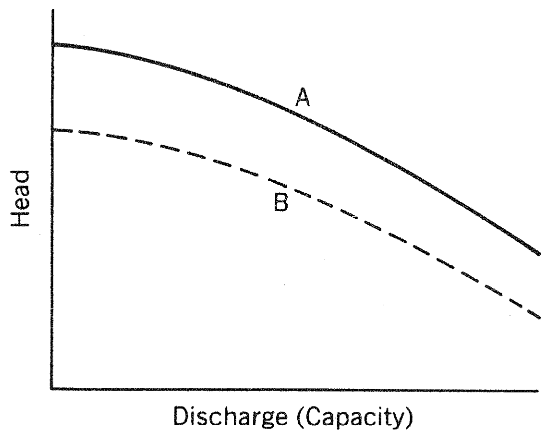


Figure 4.8 Head-capacity curves for a centrifugal pump. Curve B is obtained from curve A by reducing either impeller diameter or speed and applying the affinity laws.

system exceeds that which can be supplied by individual pumps. Figure 4.10 shows the individual head–discharge curves for pumps A and B and the combined head–discharge curve for pumps A and B operating in series. The combined curve is obtained by adding individual pump heads as shown in Figure 4.10.

The brake power and efficiency characteristic curves for pumps operating in series are obtained using Eqs. 4.14 and 4.15, respectively.

$$\mathrm{BP}(Q)_c = \sum_{i=1}^{n} \mathrm{BP}(Q)_i \tag{4.14}$$

$$(\mathrm{E_p})_c = \frac{(100)(Q)}{(K)(\mathrm{BP}(Q)_c)}\left(\sum_{i=1}^{n} H(Q)_i\right) \tag{4.15}$$

where

$\mathrm{BP}(Q)_c$ = combined brake power for discharge Q(kW, hp);
n = number of pumps operated in series;
$\mathrm{BP}(Q)_i$ = brake power required by pump i to produce discharge Q(kW, hp);
$(\mathrm{E}_p)_c$ = combined efficiency (percent);
Q = discharge (l/min, gpm);
$H(Q)_i$ = head corresponding to discharge Q for pump i(m, ft);
K = unit constant for Eq. 4.2.

When pumps are close together, that is, in the same pumping station, it is normally only necessary to provide the $\mathrm{NPSH_r}$ needs of the first pump (i.e., the furthest pump upstream). When pumps are widely separated, however, the head available at the inlet of each pump should be determined and compared to the

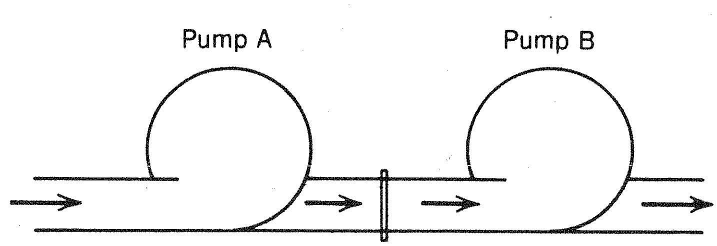


Figure 4.9 Two centrifugal pumps hooked in series. (The arrows indicate the direction of flow.)

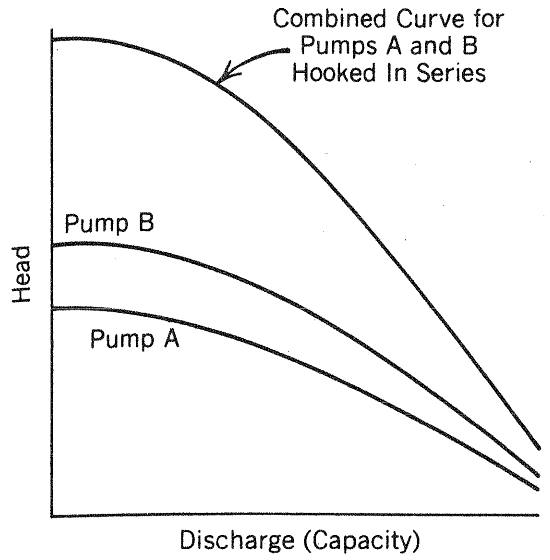


Figure 4.10 Individual head-capacity curves for pumps A and B and the combined head-capacity curve for pumps A and B operating in series.

$NPSH_r$ needs of the pump. It is also important that pressures not exceed those that can be withstood by pump seals and piping.

4.3.5 Performance Curves for Pumps Operating in Parallel

Pumps operate in parallel when they obtain water from a common source and discharge it into a single outlet as in Figure 4.11. Parallel operation of two or more pumps is a common method of meeting variable discharge requirements, since brake power and energy consumption can be minimized by operating only those

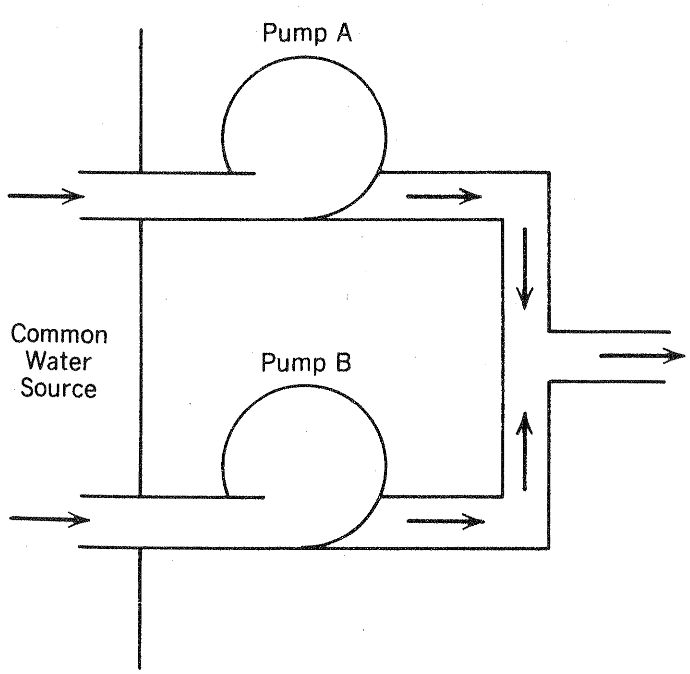


Figure 4.11 Two centrifugal pumps operating in parallel. (The arrows indicate the direction of flow.)

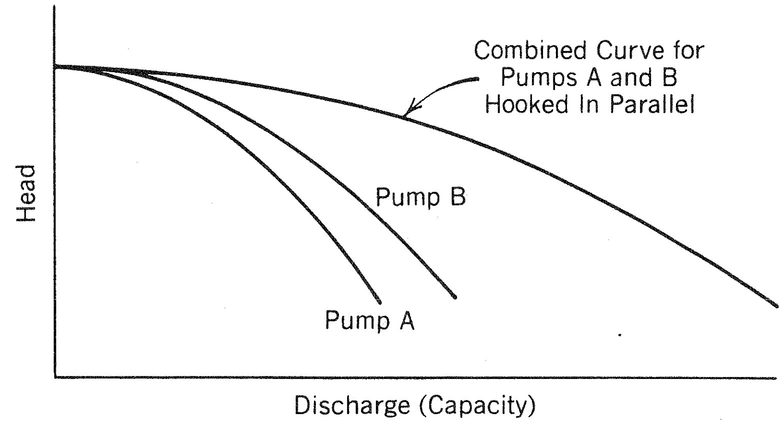


Figure 4.12 Individual head-capacity curves for pumps A and B and combined head-capacity curve for pumps A and B operating in parallel.

pumps needed to meet the demand. Figure 4.12 includes the individual head-discharge curve for pumps A and B operating in parallel. The combined curve is obtained by adding individual pump discharges as shown in Figure 4.12.

The brake power and efficiency characteristics curves for pumps operating in parallel are obtained using the following equations

$$\mathrm{BP}(H)_c = \sum_{i=1}^{n} \mathrm{BP}(H)_i \tag{4.16}$$

$$(\mathrm{E_p})_c = \frac{(100)(H)}{(K)(\mathrm{BP}(H)_c)} \sum_{i=1}^{n} \mathrm{Q}(H)_i \tag{4.17}$$

where

$\mathrm{BP}(H)_c$ = combined brake power for head H(m, ft);
n = number of pumps operated in parallel;
$\mathrm{BP}(H)_i$ = brake power required by pump i to produce head H(kW, hp);
$(\mathrm{E_p})_c$ = combined efficiency (percent);
H = head (m, ft);
$Q(H)_i$ = discharge corresponding to head H for pump (l/min, gpm);
K = unit constant for Eq. 4.2.

4.3.6 Pump Operating Point

A centrifugal pump operates at combinations of head and discharge given by its H–Q characteristic curve. The particular H–Q combination at which a pump is operating is the pump's operating point. Brake power, efficiency, and required net positive suction head for the pump can be obtained once the operating point has been determined.

The operating point depends on by the head and discharge requirements of the irrigation system. A system curve, which describes the H–Q requirements of the irrigation system, and the H–Q characteristic curve of the pump are used to determine the operating point. As shown in Figure 4.13, the operating point is the head and discharge at the point where the system and pump H–Q curves intersect

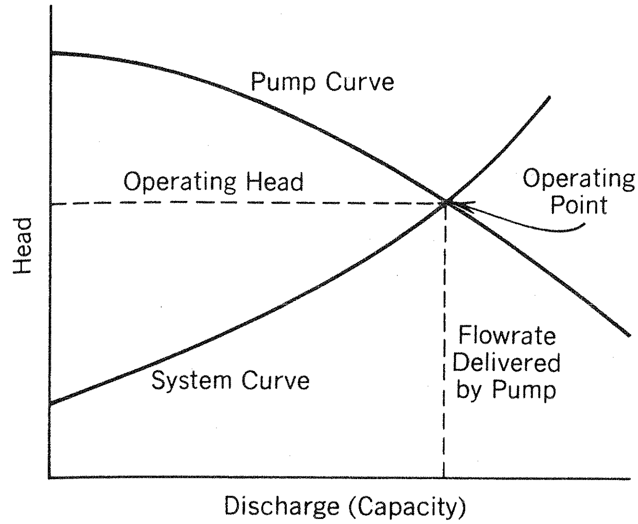


Figure 4.13 Typical system and pump head–capacity curves. The point of intersection of these curves is the operating point.

(i.e., where the H–Q requirements of irrigation system equal the H–Q produced by the pump).

System curves are constructed by computing the heads required by the irrigation system to deliver different volumes of water per unit of time. System head is computed using the equation

$$H_s = \mathrm{SL} + \mathrm{DL} + \mathrm{DD} + H_l + M_l + H_o + \mathrm{VH} \tag{4.18}$$

where

H_s = system head (m, ft);
SL = suction-side lift (m, ft);
DL = discharge-side lift (m, ft);
DD = water source drawdown (m, ft);
H_l = head loss due to pipe friction (m, ft);
M_l = minor losses through fittings (m, ft);
H_o = operating head (m, ft);
VH = velocity head (m, ft).

The suction- and discharge-side static lifts are independent of system flow, while DD, H_l, M_l, H_o and VH all increase with increasing Q. H_s is independent of the pump with the exception of friction loss that occurs in the column pipe of vertical turbine (diffuser-type) pumps.

The suction-side static lift, SL, is the vertical distance between the center line of horizontal pumps and the static water surface elevation of the water source (determined when the pump is not operating). For vertical pumps, SL is measured from the top of the discharge pipe rather than the centerline of the pump. SL is positive when the water source is below the pump and negative when the water source is above it. The water surface elevation for a well is the static water level in the well. The static water level of a water source is normally assumed to be constant even though there are fluctuations in static water levels as streams, lakes, reservoirs, and wells respond to changing hydrologic conditions. Considerable judgment is needed to choose the proper water level.

Lift on the discharge-side of the pump, DL, is the elevation difference between the point of delivery and the centerline and top of the discharge pipe of horizontal and vertical pumps, respectively. DL is computed using

$$\mathrm{DL} = \mathrm{Elev}_u - \mathrm{Elev}_p \tag{4.19}$$

where

Elev_u = elevation of the point of delivery (m, ft);
Elev_p = elevation of pump as defined for horizontal and vertical pumps (m, ft).

Drawdown, DD, is the decline in the water surface elevation of the water source due to pumping. For large surface bodies of water, drawdown is extremely small and is normally neglected. Drawdown versus Q relationships for wells and surface sources can be determined by test pumping the source at several pumping rates. Equations 3.7 and 3.8 can be used to estimate drawdown in unconfined and confined wells, respectively.

Head losses due to pipe friction (H_l) and minor losses (M_l) in fittings as a function of Q are determined using Eq. 5.12 through 5.15. The relationship between system operating head and discharge is normally established by the pressure-discharge relationships of individual sprinklers, the number of sprinklers, and pressure variations within the irrigation system. The systems H_0-Q relationship must be evaluated for the entire range of operating conditions. Velocity head is computed using

$$\mathrm{VH} = \frac{Q^2}{KD^4} \tag{4.20}$$

where

Q = system discharge (l/min, gpm);
D = diameter of discharge pipe at pump (cm, in);
K = unit constant ($K = 435.7$ for VH in m, Q in l/min, and D in cm. $K = 385.9$ for VH in ft, Q in gpm, and D in in).

EXAMPLE 4.3 Determining the System Curve for a Sprinkle Irrigation System

Given:

- irrigation system with 100 sprinklers
- $Q_s = 1.41\ P^{0.5}$ (Q_s is sprinkler discharge in gpm; P is operating pressure in psi)

- 2000 ft long, 8 in diameter, PVC supply line ($C = 150$)
- minor losses are 10 percent of pipe friction
- water supply is a large reservoir
- water is lifted 200 ft to field

Required:
system curve

Solution:
quantifying the terms in Eq. 4.18

$$SL + DL = 200 \text{ ft}$$

$$H_l = \frac{(0.285\,C)^{-1.852} L Q^{1.85}}{D^{4.87}} \qquad (5.15 \text{ and } 5.16b)$$

$$= \frac{(0.285(150))^{-1.852}(2000)Q^{1.85}}{8^{4.87}}$$

$$= 7.63(10)^{-5} Q^{1.85}$$

$$M_l = (0.1)H_l$$

where

C = Hazen-Williams coefficient (from Table 5.7);

L = length of pipe (m, ft);

Q = flowrate (l/min, gpm);

D = diameter of pipe (mm, in).

Operating head (H_0):

$$Q = (1.41P^{0.5})100 = 141P^{0.5}$$

$$P = \left(\frac{Q}{141}\right)^2$$

$$H_0 = (2.31)\left(\frac{Q}{141}\right)^2 = 1.16(10)^{-4} Q^2$$

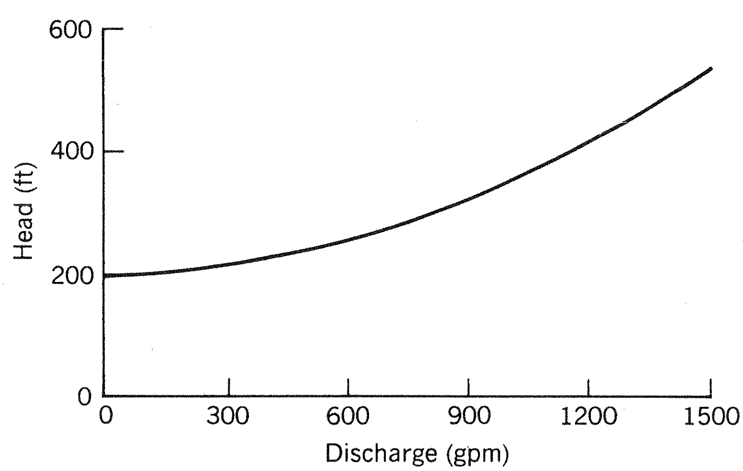
$$VH = \frac{Q^2}{385.9(8)^4} = 6.33(10)^{-7} Q^2$$

$$H_s = 200 + 7.63(10)^{-5}Q^{1.85} + 0.763(10)^{-5}Q^{1.85} + 1.16(10)^{-4}Q^2 + 6.33(10)^{-7}Q^2$$

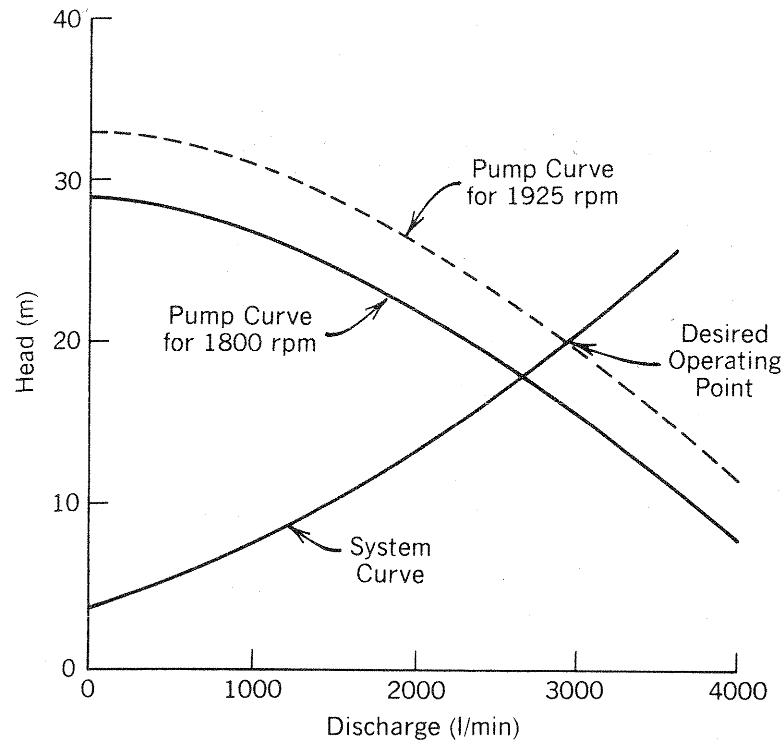
$$H_s = 200 + 8.39(10)^{-5}Q^{1.85} + 1.17(10)^{-4}Q^2$$

Tabular solution of this equation is as follows.

Q (gpm)	H_s (ft)
0	200
100	202
200	206
300	214
400	224
500	238
600	254
700	273
800	295
900	319
1000	347
1100	377
1200	410
1300	446
1400	485
1500	526



Operating points can be altered by changing either the H–Q curve for the pump or the irrigation system. Pump curves are altered by changing pump speed or impeller diameter as per the affinity laws. The following example illustrates a procedure, based on the affinity laws, for obtaining a new operating point.



EXAMPLE 4.4 Obtaining a New Operating Point by Changing Pump Speed (or Impeller Diameter)

Given:
head–discharge curve for pump operating at 1800 rpm and system–head curve in the following diagram

Required:
pump speed to achieve an operating point of 2900 l/min and 20 m of head

Solution:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

$$\left(\frac{H_1}{H_2}\right)^{1/2} = \frac{N_1}{N_2}$$

$$\frac{Q_1}{Q_2} = \left(\frac{H_1}{H_2}\right)^{1/2}$$

$$Q_1 = Q_2\left(\frac{H_1}{H_2}\right)^{1/2}$$

$$Q_1 = \frac{2900}{\sqrt{20}} H_1^{1/2} = 648.5\sqrt{H_1}$$

Since Q_1 and H_1 are unknowns in the previous equation, an additional equation is required to evaluate Q_1 and H_1. The pump curve provides the required relationship. The following table summarizes an iterative procedure for evaluating Q_1 and H_1:

Trial H_1 (m)	Computed Q_1 (l/min)	Q_1 from Pump Curve (l/lim)
21	2972	2150
15	2511	3100
19	2827	2500
17.5	2712	2712 ← solution

$$N_2 = N_1\left(\frac{Q_2}{Q_1}\right) = (1800)\left(\frac{2900}{2712}\right) = 1925 \text{ rpm}$$

Pump curve for $N_2 = 1925$ rpm.

1800 rpm		1925 rpm	
Q_1 (l/min)	H_1 (m)	Q_2 (l/min)	H_2 (m)
0	29.0	0	33.2
1000	26.5	1069	30.3
2000	22.0	2139	25.2
3000	15.8	3208	18.1
4000	8.0	4278	9.2

The operating point can also be altered by changing system head loss. This is accomplished with different pipe and fitting sizes in the design phase or by throttling (valve adjustment) after the system is in place. Increasing head loss results in steeper system head curves. Figure 4.14 shows how the operating point was shifted from point 1 to 2 by increasing head loss.

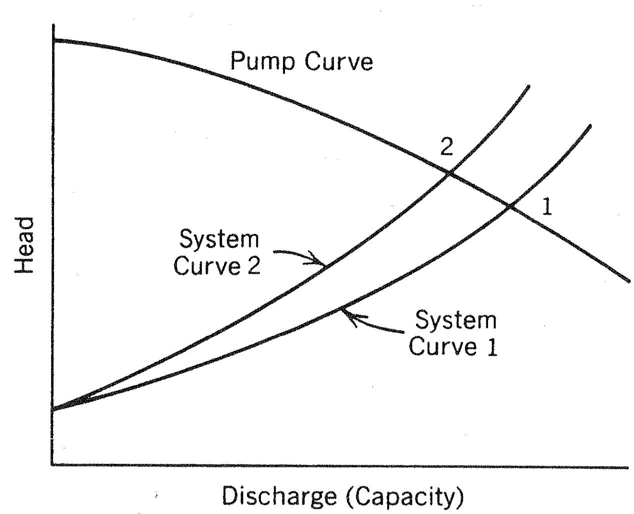


Figure 4.14 A shift in operating point resulting from a change in system head loss. There is more head loss associated with system curve 2 than system curve 1.

4.4 Pump Selection

Pump selection is the process of choosing the most suitable pump for a particular irrigation system. It involves specifying the performance requirements of the irrigation system, selecting the required pump type, and identifying alternate pumps that meet the requirements of the irrigation system. Normally, the most suitable pump is chosen from these pumps on the basis of economics.

4.4.1 Performance Requirements

The discharge and head requirements of the irrigation system must be known to select the most suitable pump(s). In systems where discharge and/or head requirements vary (depending on the number and identity of the fields being irrigated, for example), the range of discharges and/or heads required by the irrigation system must be determined. Discharge and head requirements are computed with Eqs. 3.6 and 4.18, respectively.

4.4.2 Pump Type

Horizontal volute and vertical diffuser (turbine) pumps are the main pump types used with farm irrigation systems. Because horizontal volute pumps are usually less expensive and cost less to install (than vertical pumps) they are normally used whenever possible. Vertical turbine pumps, which can be positioned below the water surface, are used in deep wells or with surface sources where it is not practical (economical) to position horizontal volute pumps so that their $NPSH_r$ needs are provided. Vertical turbine pumps are also used to eliminate the need for priming horizontal pumps.

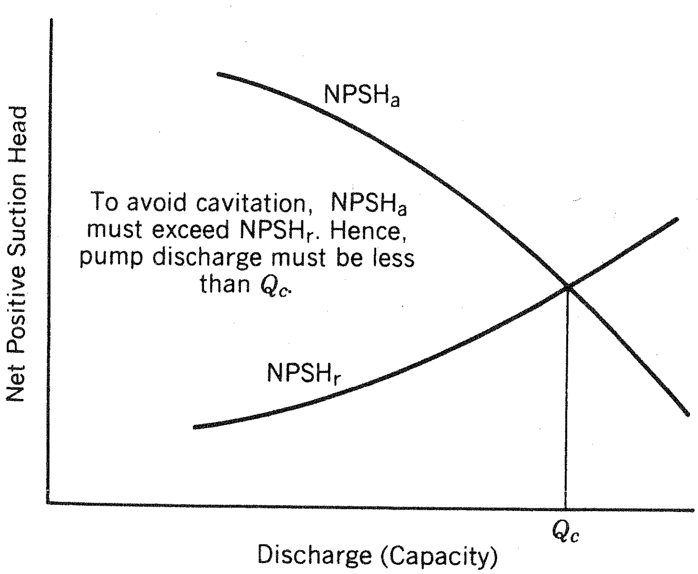


Figure 4.15 Relationship between pump discharge and required net positive suction head, $NPSH_r$, and available net positive suction head, $NPSH_a$. (The relationship between drawdown and discharge from the water source must be known to compute the $NPSH_a$–Q relationship.)

Available net positive suction head ($NPSH_a$) is often used to determine if a pump's $NPSH_r$ can be provided. A pump will operate without cavitation for all discharges where $NPSH_a \geq NPSH_r$ (see Figure 4.15). $NPSH_a$ is defined by the following equation

$$NPSH_a = BP - VP_w - (H_l)_s - (M_l)_s - VH_s - SL - DD \tag{4.21}$$

$$BP = K_1 - 1.17(10)^{-3}h + K_2h^2 \tag{4.21a}$$

where

BP = barometric pressure (m or ft of water);

VP_w = vapor pressure of water (see Appendix F) (m or ft of water);

$(H_l)_s$ = head loss due to pipe friction in the suction line (m, ft);

$(M_l)_s$ = minor losses in suction line (m, ft);

VH_s = velocity head in suction line (m, ft);

SL = suction-side lift (m, ft);

DD = drawdown (m, ft);

h = elevation above sea level (m, ft);

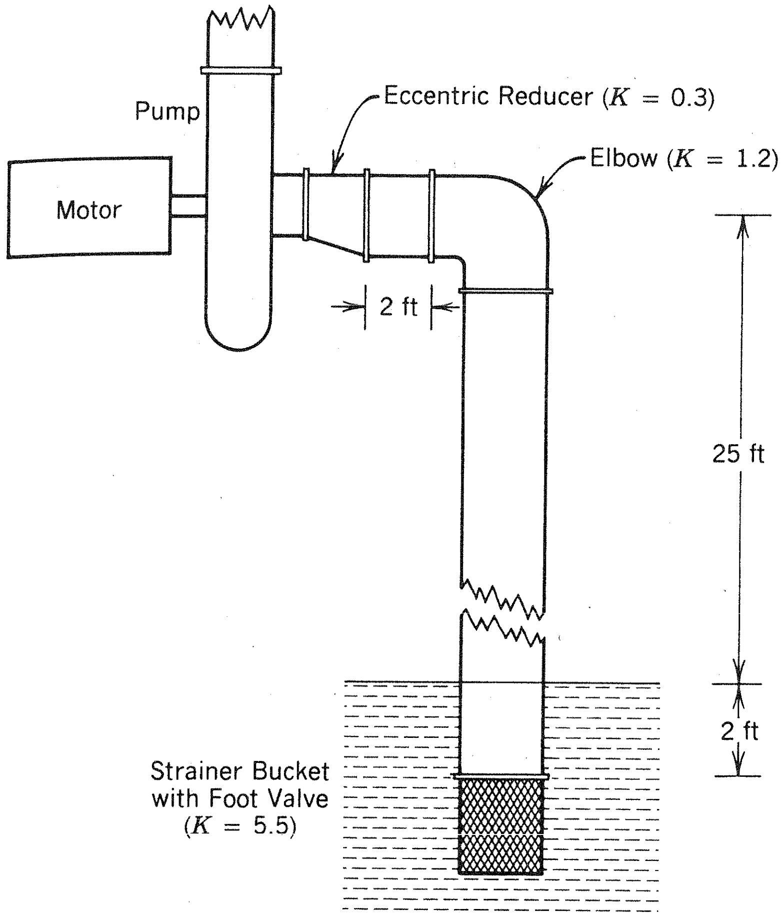
K_1, K_2 = unit constants ($K_1 = 10.33$ and $K_2 = 5.55(10)^{-8}$ for BP in m of water and h in m. $K_1 = 33.89$ and $K_2 = 1.69(10)^{-8}$ for BP in ft of water and h in ft).

VP_w can be obtained from Appendix F. The following example illustrates the use of Eq. 4.21.

EXAMPLE 4.5 Determining $NPSH_a$

Given:

- $Q = 1000$ gpm
- 8-in-diameter, 25-ft-long aluminum suction line (Hazen-Williams $C = 135$)
- water temperature = 65°F
- water level remains constant during pumping
- elevation is 1000 ft above sea level
- the following diagram



Required:
$NPSH_a$

Solution:
use Eq. 4.21

$$BP = 33.89 - 1.17(10)^3(1000) + 1.69(10)^8(1000)^2$$
$$= 32.74 \text{ ft}$$

$$VP_w @ 65°F = 0.31 \qquad \text{(From Appendix F)}$$
$$= (0.31)(2.31 \text{ ft/psi}) = 0.72 \text{ ft}$$

$$(H_l)_s = \frac{((0.285)(135))^{-1.852}(1.0)(29)(1000)^{1.85}}{8^{4.87}} = 0.48 \text{ ft} \qquad (5.15)$$

$$(M_l)_s + VH_s = (0.3 + 1.2 + 5.5 + 1.0)\frac{Q^2}{385.9D^4} = (8.0)\frac{1000^2}{385.9(8)^4}$$
$$= 5.06 \text{ ft}$$

$$SL + DD = 25 + 0 = 25 \text{ ft}$$

$$NPSH_a = 32.74 - 0.72 - 0.48 - 5.06 - 25 = 1.49 \text{ ft}$$

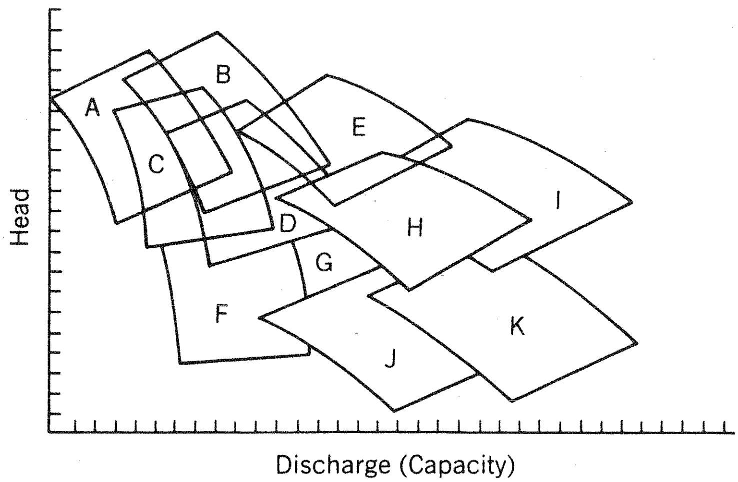


Figure 4.16 Typical diagram from a pump catalog for directing catalog users to pumps with the desired combination of head and capacity. The letters A through K denote different pump models.

Since $NPSH_r$ for most 1000-gpm-capacity horizontal volute pumps exceeds 1.49 ft, reducing the suction lift from 25 ft by repositioning the pump closer to the water surface should be considered. If this is not practical or too expensive, a vertical turbine pump should probably be considered.

4.4.3 Identifying Suitable Pumps

Manufacturer catalogs are consulted to identify pumps of the proper type that are capable of supplying the discharge and head requirements of the irrigation system. Most pump catalogs have tables or graphs similar to Figure 4.16 to direct catalog users to pumps with the desired capacity and head combinations. Characteristic curves for these pumps are examined to determine which of these pumps are suitable for the irrigation system.

4.4.4 Selecting the Most Suitable Pump or Combination of Pumps

Economics is often the primary criterion for selecting the most suitable pump or combination of pumps for a farm irrigation system. In such cases, a detailed analysis of the total annual ownership and operation costs is necessary. In other situations, minimizing energy use or first cost may be more important. Selection criteria must be specified before the most suitable pump or combination of pumps can be selected. The following example illustrates a selection process and the effect of different criteria on pump selection.

EXAMPLE 4.6 Selecting the Most Suitable Pump

Given:

- irrigation system in which the lift ranges between 100 and 130 ft
- system curves for each lift

- pump operates: $\frac{1}{4}$ of time at 100-ft lift
  $\frac{1}{4}$ of time at 130-ft lift
  $\frac{1}{2}$ of time at 115-ft lift
- crop requires 30 inches of water per year
- head–discharge and efficiency–discharge curves for pumps A and B
- pump A is a 3-stage, 7.5-in-diameter, 1760-rpm, vertical turbine pump that costs \$12,200 (cost includes 50-hp electric motor)
- pump *B* is a single-stage, 6.25-in-diameter, 3500-rpm vertical turbine pump that costs \$9,200 (cost includes 50-hp electric motor)

Required:

a. pump with lowest energy cost
b. pump with lowest annual fixed cost
c. pump with lowest total annual fixed and operating cost

Solution:

On the basis of the economic data summarized in the following table, pump A has the lowest energy costs and total annual fixed and operating costs. Pump B has the lowest total annual fixed costs.

Fixed Costs	**Pump A**	**Pump B**
Annual depreciation and interest	\$1555	\$1173
Annual taxes and insurance	244	184
Total Fixed Costs	\$1799	\$1357
Operating Costs		
Annual maintenance and repair	\$ 366	\$ 276
Energy	4176	4539
Total operating costs	\$4542	\$4539
Total annual fixed and operating costs	\$6341	\$5896

Annual Depreciation and Interest Costs (ADIC):

ADIC = CRF(PW of pump and motor) (Eq. 2.18)

CRF is computed with Eq. 2.17*a* using an interest rate of 12 percent and an analysis period of 25 yr

$$\text{CRF} = \frac{(0.12)(1 + 0.12)^{25}}{(1 + 0.12)^{25} - 1} = 0.13$$

Assume SV = 0 and a useful life of 25 yrs (from Table 2.4)
PW = IC when SV = 0 and AP = UL (Eq. 2.19)
ADIC = 0.13 (IC)
Annual taxes and insurances = 2 percent of initial cost (see Section 2.4.6a)
Maintenance and Repair = 3 percent of initial cost (see Section 2.4.6b)
Energy = (\$0.03/kWh)(kWh)

Kilowatt-hour calculations are summarized in the following table.

Pump	H (ft)	Q^a (gpm)	E_p^a	BP^b (kW)	t^c (hrs)	kWh^d	Total kWh
A	166	820	79	32.5	497	16121	
	158	940	81	34.5	867	29930	
	150	1000	81	34.9	407	14207	60,258
B	165	810	77	32.7	503	16440	
	158	940	79	35.4	867	30688	
	150	1000	80	35.3	407	14385	61,513

[a] Q and E_p were obtained from the pump characteristic curves.

[b] To obtain parametric pressure

$$^b\ BP = \frac{(100)(Q)(H)}{(E_p)(3960)}(0.746\ \text{kW/hp})$$

[c] To obtain duration

$$t = \frac{(452.5)(120\ \text{acres})(\text{inches applied})}{Q\ \text{in gpm}}$$

[d] To obtain Kilowatt-hours

kWh = (BP)(t)

Homework Problems

4.1 Water is being pumped from a reservoir into 25-cm-diameter pipe and conveyed to a sprinkle irrigation system. The system design capacity is 4000 l/min. Determine

a. the head that the pump supplies, and
b. the power that the pump imparts to the water (i.e., the water power)

when a pressure gauge located immediately downstream of the pump reads 600 kPa. The water surface of the reservoir is 5 m below the centerline of the pump.

4.2 Determine the specific speed of the pump in Problem 4.1 for a pump speed of 1750 rpm. Does the pump have a radial, mixed, or axial flow impeller?

4.3 Determine the brake power requirement of the pump in Problem 4.1 if it is 70 percent efficient.

4.4 Determine the demand power requirement of the pump in Problems 4.1 and 4.3 if the electric motor that powers it is 90 percent efficient. Demand power = brake power/motor efficiency.

4.5 Water is pumped from a reservoir into a 1000-m-long, 25-cm-diameter steel pipe and conveyed to an open canal. The water surface in the canal is 15 m above the water surface in the reservoir. Determine
a. the total head that the pump must supply,
b. the water power,
c. the brake power, and
d. the demand power

if 5000 l/min is being pumped, the pump speed is 3600 rpm, and the pump and motor efficiencies are 75 and 92 percent, respectively. There are three elbows ($k = 1.2$) in the pipeline and a strainer bucket with a foot valve ($k = 10$) on the pump inlet pipe (i.e., on the suction line).

4.6 A vertical turbine pump is being used to pump water from a 200-m-deep, 50-cm-diameter well that completely penetrates a 50-m-thick confined aquifer. The conductivity of the aquifer is 2.0 m/day and the static water level is 100 m below the ground surface. A pressure of 1000 kPa is required on the discharge side of the pump to operate a sprinkle system with a design capacity of 3000 l/min. Minor and pipe friction losses on the suction side of the pump total 20 m. Neglect velocity head. Determine
a. the total head that the pump must supply, and
b. the water power.

4.7 Use data from Figure 4.6 to develop
a. the head–discharge,
b. the brake power–discharge, and
c. the net positive suction head–discharge

curves for a 10.38-in-diameter impeller operating at 1200 rpm. The characteristic curves in Figure 4.6 are for a pump speed of 1800 rpm.

4.8 Use data from Figure 4.6 to develop
a. head–discharge and
b. brake power–discharge

curves for two identical pumps with 10.38-in-diameter impellers hooked in series.

4.9 Repeat Problem 4.8 for two identical pumps with 10.38-in-diameter impellers hooked in parallel.

*4.10 Develop a system curve for the conveyance system in Problem 4.5.

*4.11 Repeat Problem 4.10 for a 30-cm-diameter pipeline. How does a change in minor and/or pipe friction loss affect the system curve?

*4.12 Repeat Problem 4.10 using a difference in water surface elevations between the reservoir and the canal of 20 m. How does a change in lift affect the system curve?

* Indicates that a computer program will facilitate the solution of the problems so marked.

4.13 Use the system curve from Problem 4.10 to determine the operating point for the pump in Figure 4.6. The pump has a 10.38-in-diameter impeller and a speed of 1800 rpm.

4.14 Repeat Problem 4.13 using the system curves from
a. Problem 4.11, and
b. Problem 4.12.

4.15 Use data from Problem 4.13 to determine the speed at which the pump would have to be operated to obtain a discharge of 5000 l/min.

4.16 The pump in Figure 4.6 will be used to pump water from a very large reservoir into a sprinkle irrigation system with a design capacity of 5000 l/min. The suction side pipe is a 25-cm-diameter, 7-m-long aluminum pipe with a bucket strainer (without a foot valve), an elbow, and an eccentric reducer arranged as in Example 4.5. The temperature of the water in the reservoir is 20°C. The water surface in the reservoir is 500 m above mean sea level. Determine the maximum vertical distance that the pump can be located above the water surface of the reservoir.

***4.17** The pump in Problem 4.16 is being used to pump reservoir water through a 500-m-long, 25-cm-diameter aluminum pipe to a sprinkle irrigated field located 1 m above the pump. The minor loss coefficient, k, for the pipe network downstream of the pump is 15. The discharge of individual sprinklers is given by the following equation

$$Q = 2.31P^{0.5}$$

where

Q = sprinkler discharge in l/min
p = operating pressure in kPa.

Determine the operating point when
a. 100 sprinklers,
b. 150 sprinklers

are operating.

4.18 The sprinkle system in Problem 4.17 operates with 150 sprinklers approximately 65 percent of the time and 100 sprinklers the other 35 percent of the time. The pump operates 1000 hours per irrigation season. Which of the following pumps has
a. the lowest operating costs,
b. the lowest fixed costs, and
c. the lowest total annual costs.

Q (l/min)	Pump A Head	Pump A Efficiency	Pump B Head	Pump B Efficiency
0	51.0		37.5	
1000	48.5		39.5	
2000	44.3		40.2	70
3000	40.5	67	39.8	78
4000	36.3	78	36.2	81
5000	32.0	84	32.1	78
6000	27.4	85	23.5	62

The cost of pump A and B, including a 40-kW electric motor, are $5000 and $4350, respectively. Use an energy cost of $0.04 per kWh.

References

Anon. (1959). Irrigation pumping plants. Section 15 of the *National Engineering Handbook.* U.S. Department of Agriculture, Soil Conservation Service, 70 pp.

Anon. (1977). *Ground Water Manual.* U.S. Department of the Interior, Bureau of Reclamation, 480 pp.

Dornaus, W. L. (1976). Intakes and suction piping. In *Pump Handbook*, I. J. Karassik, W. G. Krutzsch, W. H. Fraser, and J. P. Messina (Eds.), McGraw-Hill, New York, pp. 11-1–11-19.

Hansen, V. E., I. O. Isrealsen, and G. Stringham (1979). *Irrigation Principles and Practices.* Wiley, New York. 417 pp.

Hicks, T. G., and T. W. Edwards (1971). *Pump Application Engineering.* McGraw-Hill, New York, 435 pp.

Karissik, I. J. (1971). *Centrifugal Pump Clinic.* McGraw-Hill, New York, 479 pp.

Karissik, I. J. (1976). Centrifugal pump construction. In *Pump Handbook*, I. J. Karissik, W. C. Krutzsch, W. H. Fraser, and J. P. Messina (Eds.), McGraw-Hill, New York, pp. 2-31–2-124.

Longenbaugh, R. A., and H. R. Duke (1980). Farm Pumps. In *Design and Operation of Farm Irrigation Systems*, M. E. Jensen (Ed.), ASAE Monograph 3, St. Joseph, MI pp. 347–391.

Pair, C. H., W. H. Hinz, K. R. Frost, R. E. Sneed, and T. J. Schiltz (1983). *Irrigation.* The Irrigation Association, Silver Spring, Md., 686 pp.

Schwab, G. O., R. K. Frevert, T. W. Edminster, and K. K. Barnes (1981). *Soil and Water Conservation Engineering.* Wiley, New York, 525 pp.

Simon, A. L. (1976). *Practical Hydraulics.* Wiley, New York, 306 pp.

Walker, R. (1972). *Pump Selection: A Consulting Engineer's Manual.* Ann Arbor Science Publishers, Ann Arbor, Mich., 118 pp.