9 Pumps

Selecting a pump is generally the last step in an irrigation system design. The pump is selected based on the required flow rate and pressure requirements of the irrigation system. The most common pumps used in sprinkler and drip irrigation systems are centrifugal pumps. Pump selection is generally a process of looking through pump catalogs and selecting the pump with highest efficiency at the required flow rate and pressure. The process also includes motor power selection, calculation of the net positive suction head, and possibly trimming the impeller in order to fine-tune the pump to the irrigation system requirements. The affinity laws govern the relationships between impeller diameter, motor frequency (RPM), flow rate, and pressure. The flow rate and pressure relationship for a given impeller diameter is called the pump curve. Pump curves and irrigation system curves can be mathematically combined in order to find the operating pressure and flow rate of the system. There are several possible sources of energy for pumps. The costs of three energy sources (solar, diesel, and electric) are compared in an example. Finally, the chapter covers basic principles of chemigation injection system design. Designing the pump station correctly is an essential final step in the provision of a reliable source of water.

Pump Types

Pumps add pressure energy and/or kinetic energy (velocity). The ratio of the kinetic to pressure energy varies widely between pumps. Axial flow pumps (water passes straight through a propeller or auger) primarily add kinetic energy to water: a propeller increases the velocity of water and adds a small amount of pressure. The water stream does not change direction as it passes through the pump. Propeller pumps are used for low pressure applications, such as pumping water from a canal for surface irrigation. At the other extreme, centrifugal pumps (Fig. 9.1) primarily add pressure energy to water, although it first has the form of kinetic energy within the pump. In centrifugal pumps, water enters into the center of the pump, is thrown by vanes in the perpendicular direction (90° change of direction), and water also exits the pump in a perpendicular direction to the inflow. Centrifugal pumps increase pressure by increasing water velocity in pump vanes (kinetic energy), and then convert kinetic energy to pressure energy as the water velocity slows in the volute (casing). Unlike the axial flow pump, the water exits the centrifugal pump in the perpendicular direction to the inflow direction. A typical application of a centrifugal pump is pressurizing water from a surface water body (pond or canal) for use in a pressurized irrigation system (sprinkler or drip). Centrifugal pumps can be driven by an electric motor (Fig. 9.2) or an engine. Submersible and Line shaft turbines are used to pump water from wells. Line shaft turbines can also be driven by an engine (Fig. 9.3) or motor.

Well pumps are Francis impeller or mixed flow pumps. These pumps throw off water at intermediate angles, between 0° (axial) and 90° (centrifugal). Francis impellers add comparable magnitudes of pressure and kinetic energy to water. The advantage of these pumps is that they can be stacked in wells (Fig. 9.2), one on top of the other. Each of the pumps (called bowls) adds some pressure to the water. Turbine pumps with many bowls can add hundreds of meters of pressure to water and remove water from very deep aquifers. Tens of bowls may be stacked for deep well applications.

Specific speed (ratio of discharge * RPM to pressure) is a measure of the ratio of kinetic to pressure energy. It is calculated to determine the best impeller type and RPM for a given application. Centrifugal, Francis impeller, mixed flow, and propeller flow pumps have specific speeds of 500, 2000, 5,000, and 10,000, respectively. Specific speed is not generally considered by irrigation engineers because the flow and pressure characteristics of available pumps are listed in pump catalogs; however, the number may be useful when developing a new application.

P. Waller, M. Yitayew, *Irrigation and Drainage Engineering*, DOI 10.1007/978-3-319-05699-9_9

$$N_s = 0.2108N\left(\frac{Q^{0.5}}{H^{0.75}}\right) \qquad (9.1)$$

where

N_s = specific speed, dimensionless,
N = revolutionary speed of the pump, rev/min,
Q = pump discharge, L/min,
H = discharge pressure, m.

In-class Exercise 9.1 The revolutionary speed of electric pumps is generally slightly less than one of the divisors of 3600; for example, typical pump rpm's are 875, 1750, and 3500. Why are most pumps manufactured with only these revolutionary speeds? Wouldn't it provide more flexibility if the pump could be adjusted to any speed?

Of the four impeller types, Frances impeller and mixed flow impellers, used in deep well turbines, have the highest potential efficiency (in the range of 90 %). Large centrifugal pumps have efficiencies approaching 85 %. The lowest efficiency pumps are axial flow pumps with maximum efficiencies in the lower 80s. Smaller pumps (<30 HP) generally have lower efficiency than large pumps, in the range of 70 %. A caveat to these efficiencies is that pumps are not efficient outside of their design pressure and flow rate. For example, a centrifugal pump is not efficient for low lift applications.

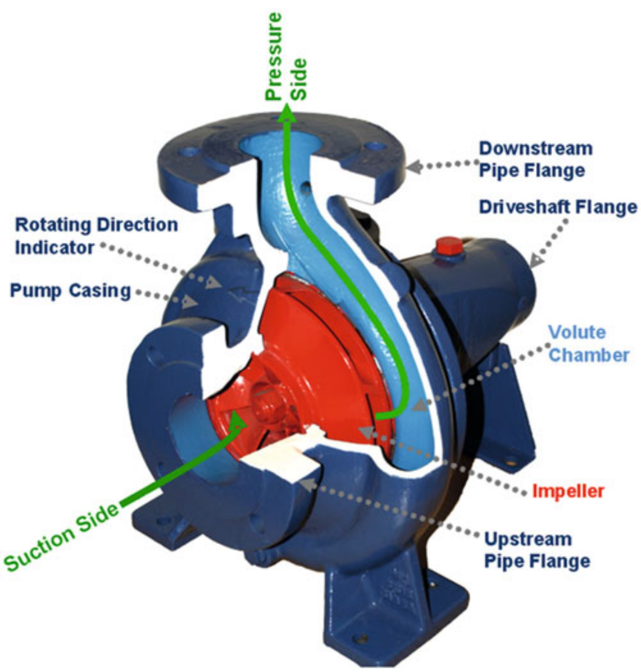


Fig. 9.1 Centrifugal pump (Wikipedia)

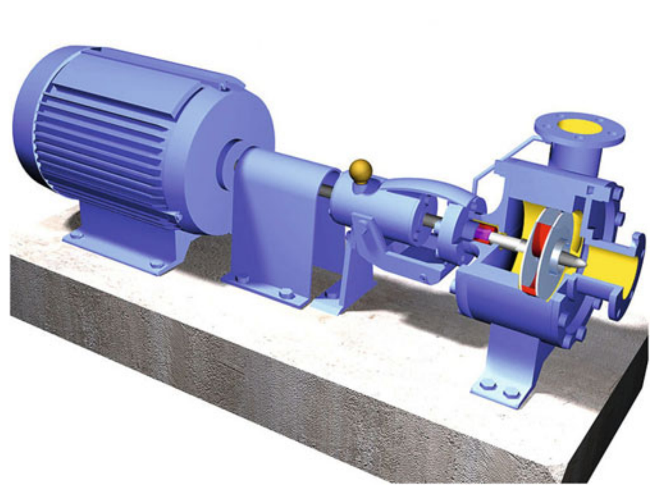


Fig. 9.2 Centrifugal pump and motor (Wikipedia)

Fig. 9.3 Deep well turbine connected to engine (Credit NRCS)

Pump Performance Curves

Pump performance curves are developed by controlling flow with a throttling valve and pressure gauge and evaluating the discharge pressure over a range of flow rates. Pump companies publish pump performance curves, also called pump characteristic curves or head-capacity curves, with this information. The head/capacity curve (Fig. 9.4) is a graph of flow rate vs. pressure. The total dynamic head is the pressure added by the pump at a given flow rate. The pump total dynamic head at zero flow rate is called the maximum shutoff pressure. Many pump curves are available at http://www.wcc.nrcs.usda.gov/ftpref/wntsc/Pump%20 Curves/

In-class Exercise 9.2 Think about the relationships between power, flow rate, and head. Based on those relationships, what would the shape of the head/capacity curve look like if efficiency was constant over a range of flow rates?

Each centrifugal pump model comes with a range of impellers and motors. For example, the impeller diameters for the pump in Fig. 9.4 range from 4.75 in to 5.9375 in. Total dynamic head can be adjusted for centrifugal pumps by adjusting the diameter of the impeller.

In-class Exercise 9.3 The irrigation system requirement is 600 gpm and 160 ft head. Select the impeller that results in this flow rate in Fig. 9.5.

Pumps designed to operate at high pressure should not be operated with no backpressure. If pumps are operated at a higher flow rate than the recommended operating range, then the motor is spinning too fast, which pushes high amperage through the windings of the motor, and the motor overheats. Head-capacity curves show the limit of the allowable operating range by ending the pump curve; for example, the 5.625 impeller in Fig. 9.4 should not be pumped at less than 6 m (15 ft) back pressure.

In Fig. 9.5, a head-capacity curve is plotted for each of the standard impeller diameters for the model B4JPBH pump. For example, the 11 3/8″ diameter impeller has a TDH of 145 ft at 200 gpm. Efficiency curves in Fig. 9.5 show that the 11 3/8″ impeller is most efficient at 740 gpm where the 11 3/8″ curve crosses the 80 % efficiency line.

For an extra fee, the pump company will trim a standard impeller to match the flow and pressure requirements of an irrigation system. The relationship between impeller diameter and flow, head, and power can be calculated with affinity laws. The discharge is directly proportional to the impeller diameter (Fig. 9.2), the flow rate is proportional to the square

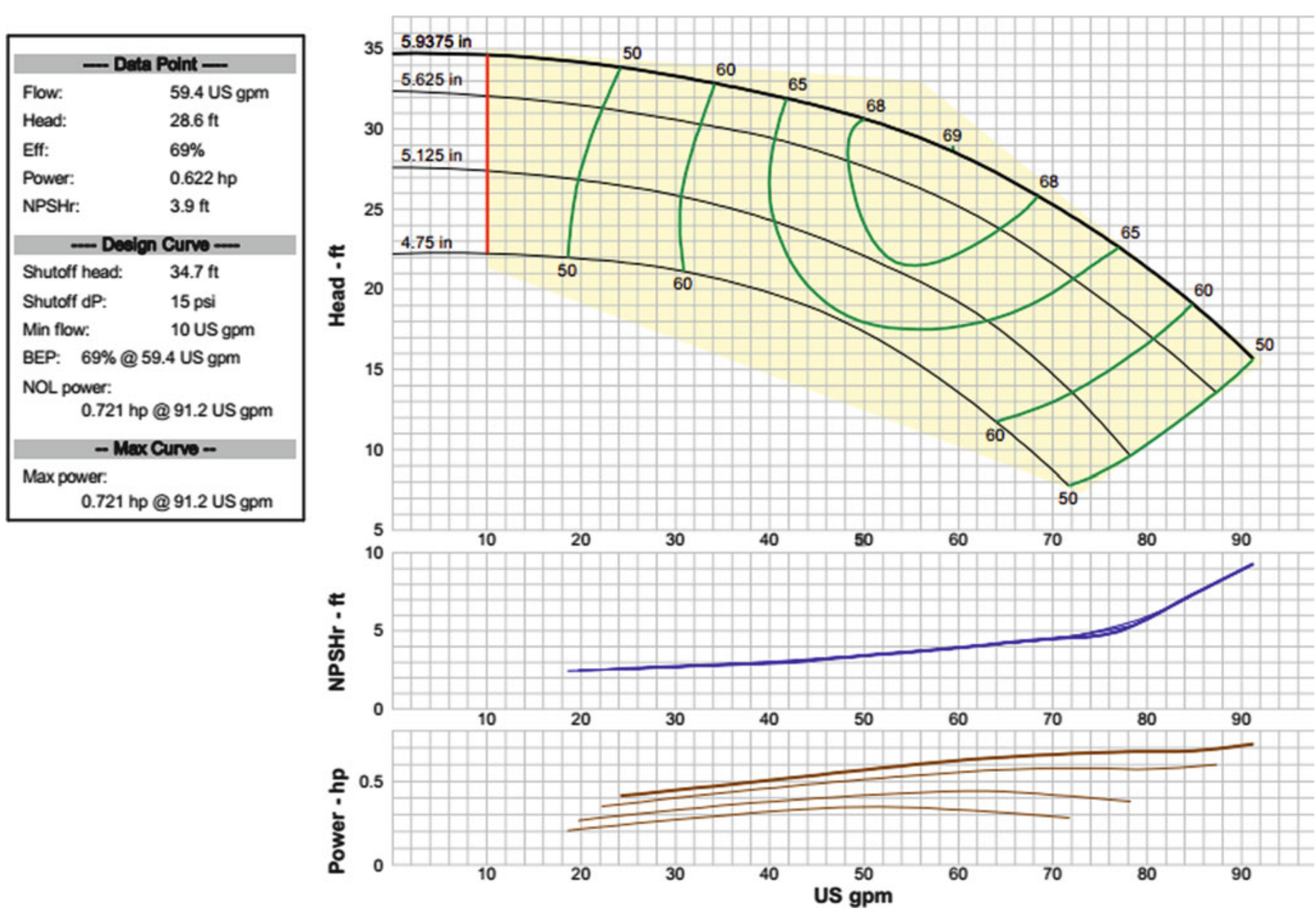


Fig. 9.4 Pump head-capacity curve for Goulds pump (Credit NRCS)

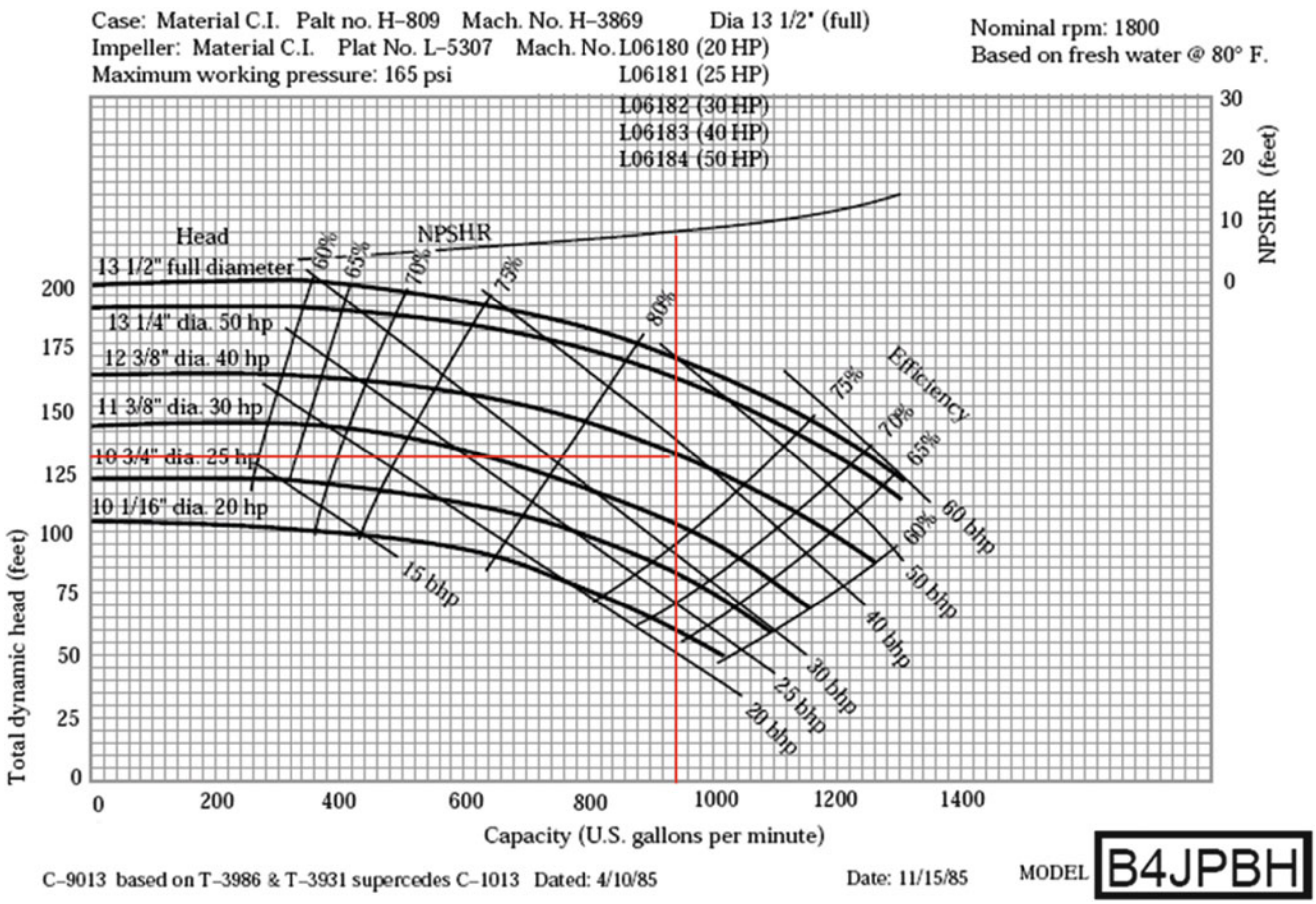


Fig. 9.5 Berkeley pump centrifugal pump performance curves (From NRCS)

of the impeller diameter, and the power requirement is proportional to the cube of impeller diameter.

$$\frac{Q_1}{Q_2} = \frac{D_{\mathrm{Im}-1}}{D_{\mathrm{Im}=2}} \quad \frac{H_1}{H_2} = \left(\frac{D_{\mathrm{Im}-1}}{D_{\mathrm{Im}=2}}\right)^2 \quad \frac{P_1}{P_2} = \left(\frac{D_{\mathrm{Im}-1}}{D_{\mathrm{Im}=2}}\right)^3 \tag{9.2}$$

where

Q = Pump discharge, GPM or L/min,
H = Total dynamic head, ft, PSI, or m,
D_{im} = Impeller diameter, in or cm,
P = Power, HP or kW.

Example 9.1 Verify that the relationships in Eq. 9.2 are correct by comparing the discharge and head output from the 11 3/8″ and 12 3/8″ impellers in Fig. 9.5 at the optimal efficiency (80 %): flow rate and pressure are 740 GPM and 122 ft head with the 11 3/8″ impeller, and are 800 GPM and 145 ft head with the 12 3/8″ impeller.

Ratio of impeller diameters is

$$12.375/11.375 = 1.08$$

Calculate TDH and flow for the 12 3/8″ impeller based on flow and pressure with the 11 3/8″

$$H_2 = \left(\frac{D_{\mathrm{Im}-2}}{D_{\mathrm{Im}=1}}\right)^2 H_1 = 1.08^2 * 122 = 144 \ ft$$
$$Q_2 = \left(\frac{D_{\mathrm{Im}-2}}{D_{\mathrm{Im}=1}}\right) Q_1 = 1.08 * 740 = 800 \ gpm$$

These values correspond with the reported values (Fig. 9.5) for the 12 3/8" impeller.

Example 9.2 What trimmed impeller diameter (Fig. 9.5) would be required if the customer needed a TDH of 133 ft at a flow rate of approximately 760 gpm?

$$D_{\mathrm{Im}-2} = D_{\mathrm{Im}-1}\sqrt{\frac{H_2}{H_1}} = 11.375\sqrt{\frac{133}{122}} = 11.876$$

The trim of the impeller refers to the outer radius of the impeller (R in Fig. 9.6). The manufacturer leaves everything else the same in a given pump model but just trims the radius of the already machined impeller. The volute also remains the same. Thus, the distance between the volute casing and

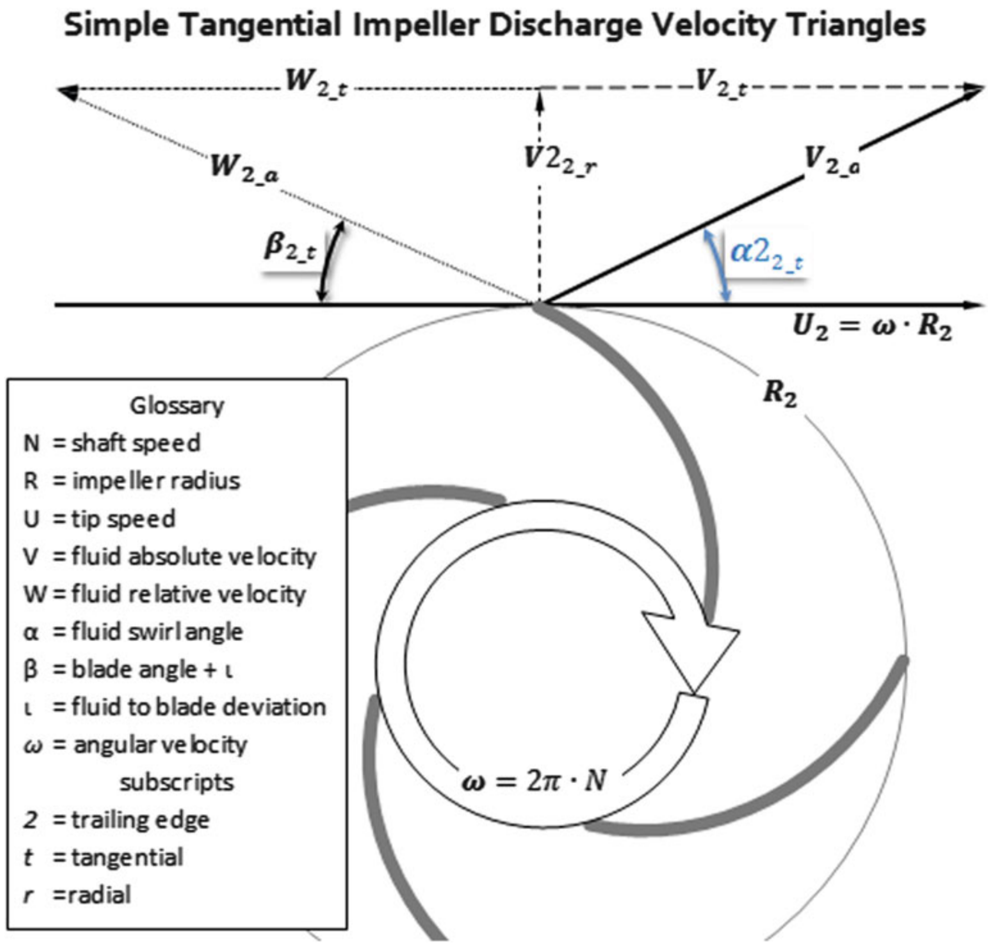


Fig. 9.6 Impeller characteristics (Credit Wikipedia, Koronowski)

the impeller increases when the impeller is trimmed. Because of the shape of the impeller, the discharge velocity angle decreases with the radius, and high efficiency is maintained. Note that kinetic energy is converted to pressure energy in the volute.

Motor brake horsepower (BHP) curves are also plotted in Fig. 9.5. In order to protect the pump motor, the pump must never be operated at a point where the impeller curve (water power) exceeds the BHP curve by even a small amount. This also causes excessive current to travel through the windings in the motor, and this excess current heats and burns up the motor. Thus, the selected motor BHP must always exceed the WHP at the range of flow rates and pressures expected for the pump/irrigation system. This concept is associated with the low backpressure rule because the water horsepower (WHP) curve exceeds the BHP curve at low backpressure. Note that the BHP curves in Fig. 9.5 are straight lines while the pump curves are curved, which causes the BHP requirement to exceed the WHP curve at some point.

Centrifugal pumps should always be started with a closed throttling valve (gate valve), and then the pipeline should be filled slowly in order to avoid overheating the motor. Pumps should not be run against a completely closed valve for more than 30 seconds. Pumps are water cooled, and if no water is moving through the pump, then the pump will overheat.

Example 9.3 Select the appropriate motor (BHP) in Fig. 9.5 if the expected flow rate and head are 800 GPM and 145 ft head, respectively.

The 12 3/8″ impeller head-capacity curve matches the specified requirements. The 40 HP motor exceeds the 12 3/8″ impeller curve (Fig. 9.5) at the operating point. If it is expected that the irrigation system will never exceed 1,000 GPM (see Fig. 9.3), then the 40 HP motor is sufficient. However, if there is some reason that the pump will exceed 1,000 GPM for a process such as flushing the irrigation system, then the 40 HP motor would not be large enough.

According to Fig. 9.5, the pump operates at 80 % efficiency at the operating point. Although the plotted head-capacity curve and BHP curve are normally used to determine the required motor HP, the pump BHP requirement can also be calculated with Eq. 2.21 as follows:

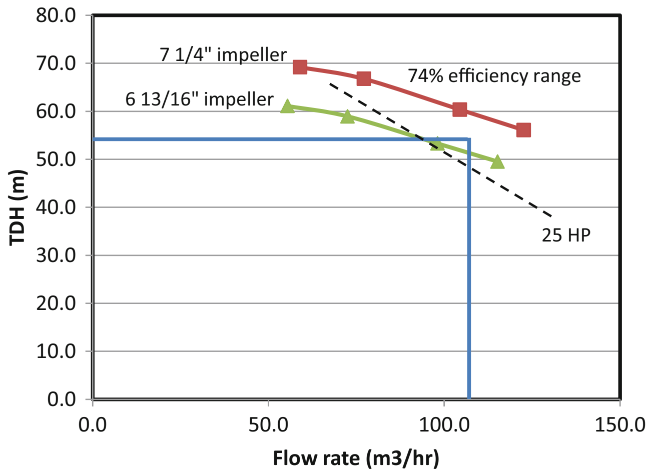


Fig. 9.7 Goulds Model 16BZ with 7¼″ and 6 13/16″impeller curves (units converted to metric) and required operating point for the irrigation system in Example 9.4

$$BHP = HP = \frac{GPM*ft}{3,960*Eff} = \frac{800*145}{3,960*0.8} = 37\ \ HP$$

Select the 40 HP motor because the power requirements are rounded up to the next motor size.

Example 9.4 An irrigation system design requires 107 m^3/hr, and TDH = 49 m. In addition, there is 2 m head loss in pump fittings and valves and 3 m pressure is added to account for degradation of the pump over time. Thus, the required pump discharge pressure is 49 + 2 + 3 = 54 m. Find the operating point.

The pump is a Goulds Model 16BZ 3 × 4 – 8). Only the 7 ¼″ and 6 13/16″ impeller curves are shown (units converted to metric). The numbers, 3 × 4 – 8, refer to the discharge and inlet diameters, 3″ and 4″, respectively, followed by the maximum impeller diameter (8″).

The blue lines in Fig. 9.7 show the required flow rate and pressure. The 25 HP motor line is the dashed line. The pump curve calculations for this example are in the *Pump curves* worksheet.

The selection of the impeller diameter causes a bit of a dilemma because the design point is just slightly higher than one of the standard impeller diameters, (6 13/16″). Remember that 3 m head was added as a safety factor. One could cut the safety factor, but this should be explained to the farmer.

Do you want to purchase a trimmed impeller with a safety factor or do you want to spend less money on a standard impeller and have no safety factor? The smaller impeller will use less energy and be less expensive, but when the pump degrades over time, there might be inadequate pressure for the design flow rate; thus, sprinkler flow rates will decrease.

If the farmer selects the standard 6 13/16″ impeller, then the pump operates at a slightly higher pressure than 25 HP motor curve. Because it is never acceptable to select a HP curve that is below the pump curve at the operating point, select a 30 HP motor. Selecting the 30 HP motor does not mean that 4 HP will be wasted. The 30 HP motor will only use the fraction of HP required (approximately 26 HP) to run the pump. If the farmer decides to use a trimmed impeller on the 16BZ pump, then affinity laws can be used to select the correct impeller size.

$$D_{\mathrm{Im}-2} = D_{\mathrm{Im}-1}\sqrt{\frac{H_2}{H_1}} = 6.8125\sqrt{\frac{53}{50.5}} = 7''$$

The calculation of the 7″ impeller flow rate and head are shown in the *Pump curves* worksheet.

Finding the Intersection of the System Curve and Pump Curve

A system curve describes the pressure-flow relationship for the irrigation system: higher pressure generally results in higher flow rate. For many irrigation systems, the flow is proportional to the square root of pressure because many irrigation components are orifices or dissipate energy with turbulent flow. A typical irrigation system has a flow rate that is approximately proportional to the square root of pump pressure (x = 0.5).

$$Q_{\mathrm{system}}\left(\mathrm{m}^3/\mathrm{h}\right) = C_{\mathrm{system}}\mathrm{TDH}^x \tag{9.3}$$

where

TDH = pump pressure, m,
Q_{system} = irrigation system flow rate, m^3/hr,
x = system flow exponent,
C_{system} = system coefficient.

If an equation for the pump head-capacity curve is found by regression from the manufacturer's pump curve, then the head-capacity equation and the system equation can be solved simultaneously in order to find the system operating point.

Example 9.5 The 16BZ pump (Fig. 9.7) with a 7¼ ″ impeller is used to run a sprinkler system with the following system curve:

$$Q_{\mathrm{system}}\left(\mathrm{m}^3/\mathrm{h}\right) = 14.175\left(\mathrm{H}_{\mathrm{system}}\right)^{0.531}$$

Find the operating point of the irrigation system. This is the intersection of the pump curve and system curve.

Assume that there is a 4 m pressure loss in pump fittings and filters. This problem is solved with the *System and pump curves* worksheet. Adjust the system curve to account for pump fitting and filter losses ($H_{system} = TDH - 4$ m).

$$Q_{system}\left(m^3/h\right) = 14.175(TDH - 4)^{0.531}$$

The 7¼″ impeller pump-capacity curve and the system curve are plotted in Fig. 9.8 (metric units). The operating point is at the intersection of the two curves. In order to calculate the operating point, a second order polynomial can be calculated for the pump head-capacity curve with the 7¼″ impeller with Excel Trendline (Fig. 9.8). Next substitute the head-capacity equation for TDH in the system equation.

$$TDH = -0.00111Q^2 - 0.00926Q + 73.74$$
$$Q = 14.175\left(-0.00111Q^2 - 0.00926Q + 73.74 - 4\right)^{0.531}$$
$$\text{After iteration}, Q = 118\,m^3/hr, TDH = 58\ m.$$

If a farm has multiple center pivots, then some or all pivots may run at the same time. Multiple, pumps can be connected to a central manifold. Each of the pumps in the manifold is equal to the flow rate of one center pivot; however, all of the pumps must be sized for the worst case (pivot with highest pressure requirement).

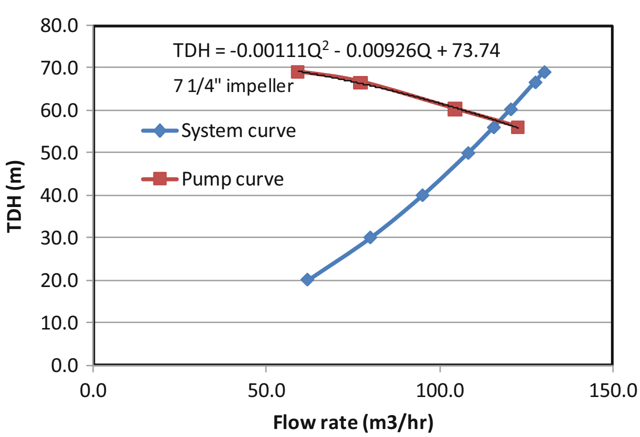


Fig. 9.8 System curve and pump curve for Example 9.5

Example 9.6 Calculate the TDH required for four parallel pumps in a pump manifold that supply the four center pivots shown in Fig. 9.9. A filter is located at the pump station with a 2 m pressure loss. Pump fittings losses are 3 m. Supply 2 m extra pressure as a safety factor. Pivot pipe loss is 2 m. Minimum sprinkler pressure is 12 m and sprinklers are 1 m above the ground surface.

Construct the following table to find the pressure requirement at each pivot inlet. Calculate the pressure required at each pivot inlet by summing all pressure losses and elevation gains (Table 9.1).

All pumps must supply 50 m TDH, the worst case. Note that the system is designed for pivot 2, not for the farthest pivots, 3 and 4, because pivot 2 is the worst case with respect to pressure. Pivots 3 and 4 are down a hill and gain pressure by elevation change. Pressure regulating valves may be needed at pivots 1, 3, and 4 in order to dissipate extra energy. It is possible that a lower pressure pump could be used to supply water when only pivot 1 is running. A second alternative would be to install booster pumps at the three pivots with higher pressure requirements and run the pumps at the main pump station at 30 m TDH.

In-class Exercise 9.4 Redo Example 9.6 except assume that pivot 2 is at 100 m elevation and pivots 3 and 4 are at 120 m elevation. Select the pump operating pressure. If each pivot requires 100 L/sec, then how many pumps should be installed and what should their flow rate be?

Variable Speed Pump Controllers

As shown in the previous examples, if a farm has irrigation zones that operate at different pressures, then using a single-speed pump will result in wasted energy. Unlike pivot farms where all pivots may operate at the same time, orchard and turf (permanent sprinklers) irrigation systems only operate one zone at a time. However, the problem with variable pump requirements is the same. Orchards and turf systems

Fig. 9.9 Maximum pivot elevations (E) and pipeline pressure (h_f) losses

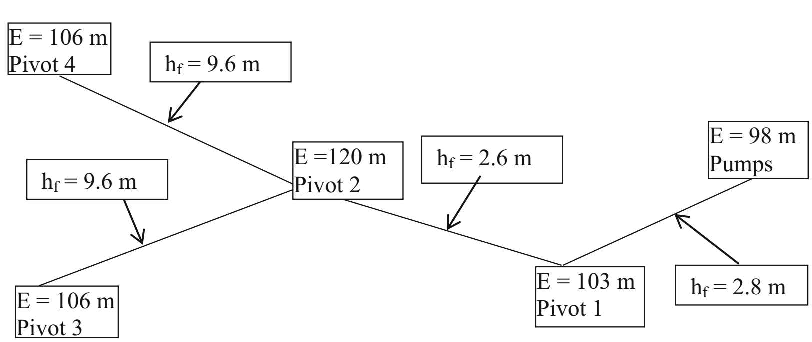


Table 9.1 Pump pressure requirements

Component (All hf in meters)	Pivot 1	Pivot 2	Pivot 3	Pivot 4
Sprinkler pressure (+ elevation) required	13	13	13	13
Pressure loss in pivot pipeline	2	2	2	2
Elevation difference (Max pivot – pump Elev)	5	22	8	8
Cum. pressure loss in mainline prior to pivot	2.8	5.4	15	15
Screen filter	2	2	2	2
Pump fittings losses	3	3	3	3
Safety factor	2	2	2	2
Total pressure requirement (TDH)	29.8	49.4	45	45

are often laid out on hills, which may have zones with dramatically different pressure requirements. If a pump supplies two or more zones with different pressure and flow requirements, then the farmer has four options. First, pressure can be reduced to the lower pressure zone with a pressure regulator. Second, the size of the upper zone can be reduced so that it runs at a different point on the pump curve with more pressure and less flow. Third a booster pump can be installed to increase pressure to the upper zone. Fourth a variable speed pump controller can vary the frequency of AC current that is delivered to the pump motor, which then changes the rotational velocity and discharge characteristics of the pump.

Pump discharge, pressure, and power relationships for variable speed pumps are calculated with the following affinity laws, which are similar to Eq. 9.2 but vary RPM rather than diameter:

$$\frac{Q_1}{Q_2} = \frac{RPM_1}{RPM_2} \quad \frac{H_1}{H_2} = \left(\frac{RPM_1}{RPM_2}\right)^2 \quad \frac{P_1}{P_2} = \left(\frac{RPM_1}{RPM_2}\right)^3 \qquad (9.4)$$

where

RPM = rotational speed, revolutions per minute.

Variable speed pump controllers can save significant energy if an irrigation pump is sized for a zone with one flow rate and pressure but must also service a smaller zone. They can increase or decrease flow with constant pressure, increase or decrease pressure with constant flow, or match the pressure and flow rate to the system curve requirements.

Example 9.7 A farmer decides to operate his irrigation system at half of the original pressure and use low-pressure sprinkler nozzles (hexagon-shaped orifice). A variable speed pump controller is used to vary the flow rate of the 16BZ pump (Fig. 9.7) with the 7¼″ impeller. The revolutionary speed is lowered from 3500 to 2500 RPM. The system curve is Q_{system} (m^3/h) = 14.175 $(H_{system})^{0.531}$. There is 5 m head loss in the pump fittings and filters. Find the operating point TDH and flow rate. This problem is solved in the *Variable speed pumps* worksheet. Assume that efficiency remains the same at the new RPM, which is probably not a good assumption.

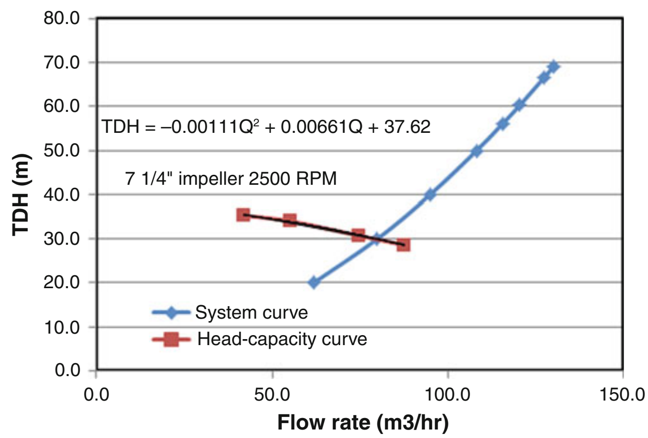


Fig. 9.10 Head-capacity curve for Model 16BZ pump at 2500 RPM for Example 9.7

The pump head-capacity curve can be adjusted downward by calculating the adjusted flow rate at four points with Eq. 9.4. For example, at one point on the curve, the flow rate is 260 gpm and 227 ft.

$$Q_2 = \frac{RPM_2}{RPM_1} Q_1 = \frac{2,500}{3,500} 260 = 186 \ \ gpm$$
$$H_2 = \left(\frac{RPM_2}{RPM_1}\right)^2 H_1 = \left(\frac{2,500}{3,500}\right)^2 227 \ ft = 116 \ ft$$

The adjusted pump curve, its equation, and the system curve (in metric units) are plotted in Fig. 9.10.

$$\text{TDH} = -0.00111\text{Q}^2 - 0.00661\text{Q} + 37.62$$
$$\text{Q} = 14.175 \left(-0.00111\text{Q}^2 - 0.00661\text{Q} + 37.62 - 5\right)^{0.531}$$
$$\text{After iteration, Q} = 79 \text{ m}^3/\text{hr}, \text{TDH} = 30 \text{ m}.$$

The power requirement is 1/3 of the original. Power is calculated with Eq. 2.20

$$\text{Power(kW)} = (\text{Q(m}^3/\sec)*\text{H})/(0.102*\text{Eff}) = (118/3600*58)/(0.102*0.74) = 25\text{ kW}$$
$$\text{Power(kW)} = (\text{Q(m}^3/\sec)*\text{H})/(0.102*\text{Eff}) = (79/3600*30)/(0.102*0.74) = 8.7\text{ kW}$$

The energy savings would not be as great since the lower pressure version would need to operate for a longer time in order to supply the same amount of water.

Centrifugal Pump Installation

Centrifugal pumps should be mounted and leveled on a concrete pad. Whenever possible, install a centrifugal pump at an elevation below the water surface. Otherwise, pumps must be primed. Even though it is desirable to have the pump intake below the water surface, pumps should not be placed in a hole because the pump will overheat.

If the pump must be installed above the static water level, then the pump should be installed as shown in Fig. 9.11. The suction must be placed at a depth in the water that is not going to cause a vortex when water is sucked into the pipe. The rule of thumb is typically that the intake should be 30 cm below the water surface; however, equations are available that calculate minimum elevation below the water surface as a function of flow rate. If there is a vortex, a plastic sheet can be floated on the water surface or a vortex breaker can be placed near the pump suction inlet.

The goals in pump suction design are to avoid excessive turbulence before water enters the pump, which reduces efficiency, and to avoid cavitation, which is caused by low pressure in the suction line, an air vapor leak in the suction line, or an air pocket at a high point in the suction line. The flow velocity in the pump suction is typically 1.5 m/s and up to 6 m/s in the pump inlet. Thus, a larger diameter pipe than the inlet nozzle delivers water from the pond to the inlet nozzle. Excessive turbulence can be avoided by using a

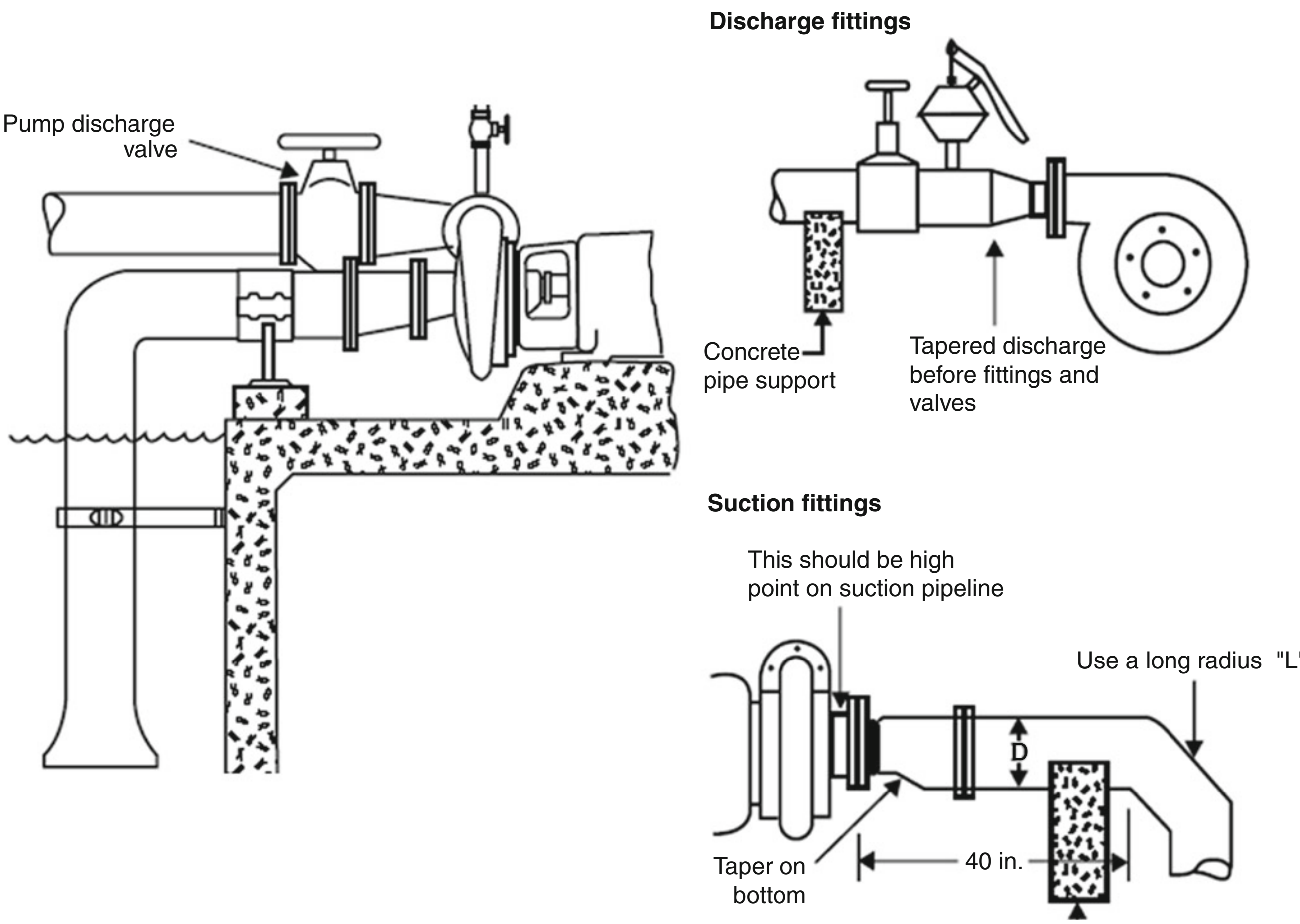


Fig. 9.11 Centrifugal pump fittings and primer pump (*upper right*) (Credit NRCS). National Irrigation Guide, Part 652, chapter 12. See other pump installation diagrams at this site. http://www.irrigationtoolbox.com/NEH/Part652_NationalIrrigationGuide/ch12.pdf

sweep 90 (Long radius) or two mitered 45 degree ells. This is followed by a horizontal pipe leading into the pump that is at least four pipe diameters in length. Pipe diameter is reduced with an eccentric (Fig. 9.11), which is a cone cut at an angle. The purpose of the eccentric is to maintain the top of the pipe at one elevation so that gas pockets are not trapped prior to the reducing cone, which would then be entrained into the flow, and then cause cavitation into the pump. Although use of an eccentric is a standard design practice on horizontal inlet pipes, is recommended by pump manufacturers, and should therefore be followed, some engineers disagree with this practice and think that it causes excess turbulence and does not reduce cavitation. On the discharge side of the pump, an expansion cone (Fig. 9.11) is placed just after the outlet. A sudden expansion right after the pump causes turbulence and loss of energy. A gate valve is placed after the cone.

For pumps that are not self-priming and are located above the water level, a primer pump can be connected to the discharge side of the pump (Fig. 9.11). If the pump is powered by an internal combustion engine, then the exhaust from the engine can be connected to a Venturi valve in order to create a priming pump. The Venturi creates a vacuum and thus sucks water into the pump.

The pump suction inlet generally includes a foot valve and a screen. A foot valve (check valve) can keep water in the pump suction after shut off so that the pump does not need to be primed before each use (Fig. 9.12). If there is debris in the water source, then screens are installed at the pump inlet (Fig. 9.12). For large debris loads, large cylindrical screens with an internal rotating sprinkler to keep the screen free of debris may be needed. Alternatively, water can be screened before or after the pump with a stainless steel screen.

In order to safeguard pumps, pump switches should be installed on the discharge side of pumps. These switches measure the discharge pressure from the pump and shut down the pump if there is no pressure.

When a centrifugal pump is higher than the water source, then the water in the suction pipe will be under negative pressure (less than atmospheric). If the pressure is too negative, then there is a danger that water will enter the gas phase and form bubbles in the water. The pump increases the water velocity inside the pump and thus lowers the water pressure even further. The bubbles implode when they hit the impeller. This causes pitting and eventually destroys the impeller. Each pump has a specified net positive suction head required ($NPSH_R$) which is the minimum absolute pressure (not gage pressure) in the suction pipe at the pump inlet that will not result in bubble formation and cavitation in the pump. The $NPSH_R$ increases with pump flow rate; for example, the $NPSH_R$ curve is shown on the B4JPBH pump curve diagram (Fig. 9.5) with $NPSH_R$ increasing with pump flow rate.

The actual NPSH at the pump inlet is calculated as follows:

NPSH = Atmos. pressure – water vapor pres. – pump elev above water – friction loss in suction

Water vapor pressure at 30 °C is 0.43 m. Atmospheric pressure is 9.9 m.

Example 9.8 Calculate NPSH and determine whether a pump suction with the following design characteristics meets the $NPSH_R$ requirement at the design operating point in Fig. 9.5. The pump is 3 m above the surface of the pond, there is 0.5 m head loss in the suction pipe and foot valve prior to the pump, and the pump flow rate is 1,000 gpm. Atmospheric pressure is 101 kPa (9.9 m).

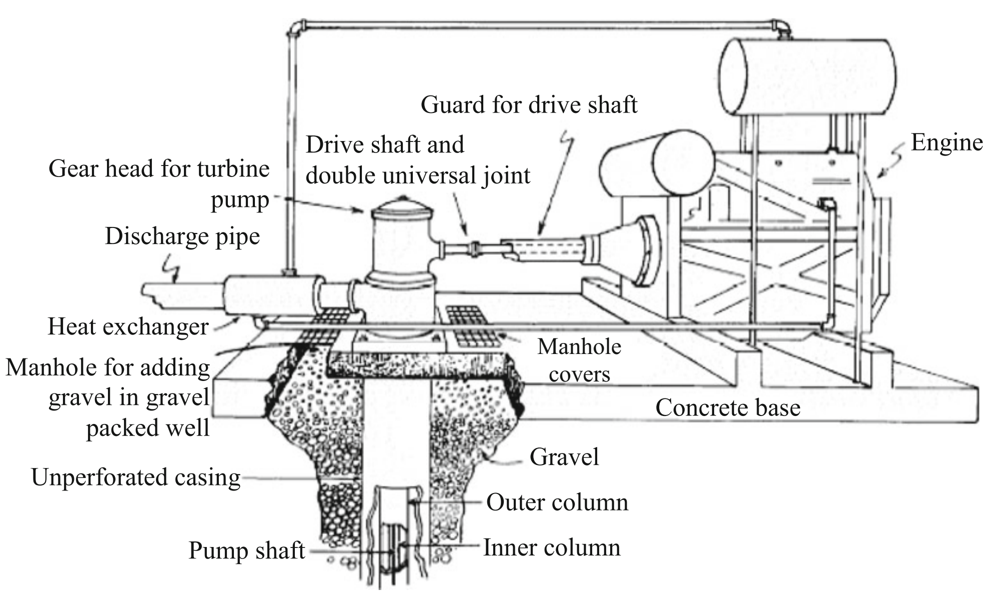


Fig. 9.12 Line shaft turbine pump (Credit NRCS)

The net positive suction head (NPSH) is calculated as follows:

$$\begin{aligned} \text{NPSH} &= \text{Atmospheric pressure} - \text{water vapor pressure} \\ &\quad - \text{pumpelevation} - \text{friction loss} \\ \text{NPSH} &= 9.9\text{ m} - 0.43\text{m} - 3\text{m} - 0.5\text{m} = 6\text{m}. \end{aligned}$$

The $NPSH_R$ requirement for 1,000 gpm flow rate in Fig. 9.5 is 8 ft (2.4 m). The design NPSH at 6.0 m is greater than the $NPSH_R$ requirement (2.4 m); thus, the pump installation will not cause cavitation and is acceptable.

Turbine Pumps (Well Pumps)

There are two primary types of irrigation well pumps: submersible pumps and line shaft turbines. Line shaft turbines can be driven by a diesel engine or an electric motor at the ground surface (Fig. 9.12). Alternatively, an electric motor can be mounted directly over the well. The pump hangs from a steel pipe called the outer column (Fig. 9.12). The shaft that delivers torque to the pump is contained inside the inner column. The pump shaft is kept in the center of the inner column by bearings. Line shaft turbines and submersible pumps have a series of bowls with their own impellers, and all impellers are connected to the same shaft.

Turbines can also be used to pump from shallow wells, reservoirs (pump vaults) near the ground surface, and can be used as booster pumps.

The pump shown in Fig. 9.13 is a submersible turbine. The motor hangs below the the pump, and power is delivered to the motor with a power cable in the well. The advantage of "submersibles" is that they do not need a long shaft in the well connecting the pump and motor.

Some deep well pumps are equipped with a "soft start" mechanism, which varies RPM with a variable speed controller. The soft start allows the pump to start slowly and then increases the revolutionary speed of the pump over time. Starting the pump slowly prevents damage to the pump and damage to the irrigation system. The variable speed controller only operates during start up and does not change the revolutionary speed of the pump during normal operation. The reason for this is that turbine pumps in wells use mixed flow impellers, which are generally designed to operate at just one rotational speed; their efficiency decreases if the speed changes. The reason for this is that the water must come off the impeller blades at the same angle as the vanes in the bowl in order to achieve maximum pump efficiency. If the speed changes, then the angle changes.

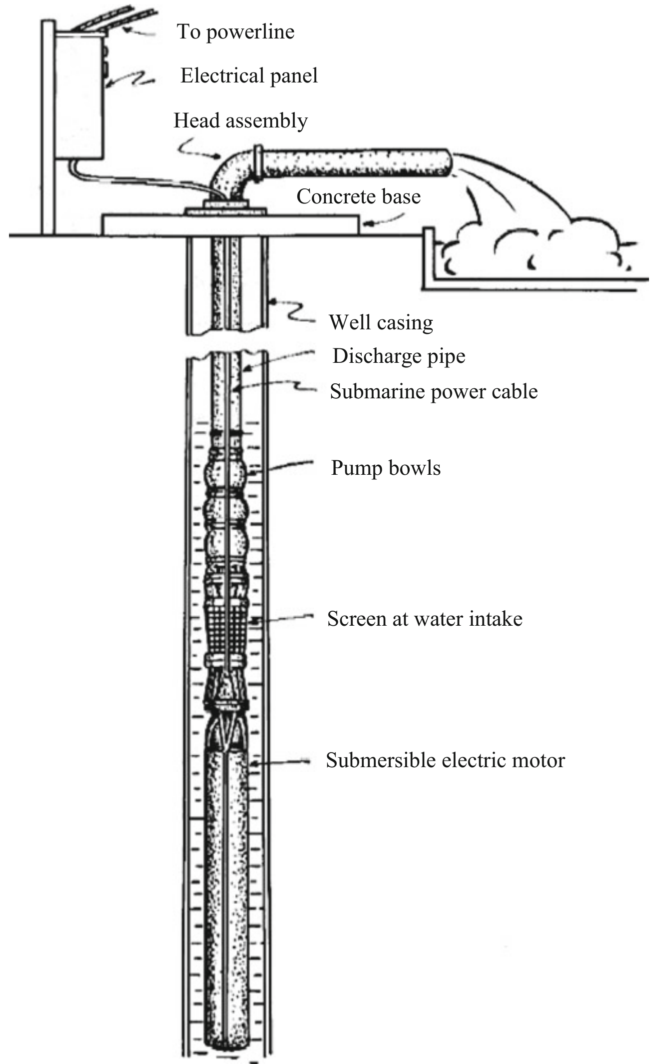


Fig. 9.13 Submersible pump (Credit: Natural Resource Conservation Service, National Engineering Handbook, Sec. 15)

Pump Station Power and Cost

Pump station power can be separated into water horsepower, brake horsepower, engine or motor power, and fuel source power. Energy losses are found in each step of the transfer of energy from the fuel source to the water. In addition, energy losses take place in pump station fittings, pipes, valves, and filters.

The water horsepower (not including efficiency) is the power actually delivered to the water and can be calculated for different units as follows.

$$\begin{aligned} &\text{WHP} = (\text{Qrg}) && && (9.5)\\ &\text{HP} = \text{gpm*ft}/3960 && (\text{U.S. units horsepower}) \\ &\text{mhp} = \text{L/min*m}/4634 && (\text{metric horsepower}) \\ &\text{kW} = \text{HP*0.746} && \text{mhp} = \text{HP}/1.01422 \qquad \text{kW} = \text{mhp*0.757} \end{aligned}$$

Table 9.2 Nebraska pumping plant performance criteria for fossil fuel powered pumps (After Huffman et al. (2013)). Listed values are power delivered to the pump, and not raw energy content of fuel

Energy source	b-kW-hr per unit	w-kW-hr per unit	Unit	Overall efficiency
Diesel	3.282	2.46	Liter	23
Gasoline	2.273	1.71	Liter	17
Liquid propane	1.813	1.36	Liter	18
Natural gas	2.166	1.62	Liter	17

where

$$\text{WHP} = \text{water horsepower, HP.}$$

The brake horsepower is the power delivered by the motor or engine to the pump.

$$\text{BHP} = \text{WHP}/\text{E}_\text{p} \tag{9.6}$$

where

BHP = brake horsepower (U.S. units),
E_p = efficiency of pump.

The power required by the pump/motor assembly is the BHP divided by the efficiency of the motor or engine. In the case of an electric pump, the pump and motor are generally sold as a unit and the efficiencies of the pump and motor are not separated. Thus, the HP requirement for electric pumps (Figs. 9.5, 9.6, and 9.7) refers to the power requirement of the motor, and not to the BHP supplied by the motor to the pump (Beard and Hill 2000).

Even though an electric motor is rated at a certain HP, this is not necessarily the power output of the motor during operation. The motor will only draw the amount of power required by the pump; for example, if a 40 HP motor was used to supply the energy for pump in Example 9.4, the motor would only draw 31 HP from the power grid. Thus, it is not necessarily a waste of energy to select a motor that is larger than required (neglecting possible changes in motor efficiency).

The selection between different types of pump power sources depends on the cost of energy, and energy prices vary regionally. For example, electricity may be expensive and natural gas inexpensive in one region so most irrigation systems will use natural gas engines. In another region, natural gas may be unavailable and most pumps will be powered by electricity. Even if the cost of a given form of energy is inexpensive, the cost of energy delivery may be high; for example, a new farm may be miles from the nearest connection to the electric power grid.

Internal combustion engines are inefficient converters of chemical energy in fuel to mechanical pumping energy. The Nebraska Pumping Plant Performance Criteria (Table 9.2) give the overall kW-hr delivered to the pump and associated efficiencies.

Example 9.9 A turbine well pump to be installed at a new farm requires 36 BHP and delivers 27 HP to the water (75 % efficiency). The pump will operate for 1,800 hours per year and deliver water for surface irrigation. Select between a diesel engine or an electric motor as the power source. Assume a 20 yr project life and 8 % ROR. Ignore inflation, but include replacement costs. For both pump systems, the cost of the pump is $3,000.

The cost of diesel delivered to the tank is $2.70/gallon. The diesel engine costs $15,000, and the diesel tank costs $3,000 installed. The tank has a service life of 20 years. Annual maintenance cost of pump and engine is $1,000/yr. The pump will need to be replaced after 16 years, and the diesel engine replaced after 14 years.

The cost of tying into the existing power grid and bringing an electric power line to the pump is $40,000, the cost of electrical energy is $0.09/kW-hr, and the cost of the electric motor is $3,000. The electric motor efficiency is 90 %. Annual maintenance cost is $400/yr. The electric motor will not need to be replaced, and the pump will be replaced after 16 years.

This problem is solved in the *Fuel and Pump costs* worksheet

Solution:
See Table 9.3

The present value of the electric pump is less so choose the electric system.

Pump Station Friction Loss and Head Requirement

Steel pipe used in wells is sold in two classifications: gage and schedule. Gage refers to the thickness of the pipe wall (Table 9.4), which is sometimes reported in mils, where mils are thousandths of an inch. Schedule 40 steel pipe dimensions are the same as PVC (Table 8.2). Dimensions of other Schedules (10, 20, 30, and 80) are available in handbooks or on the Internet. Outside diameters are the same as PVC. Subtract 2x wall thickness from outside diameter to find the inside diameter.

The head requirement for a well pump is the sum of the following (Fig. 9.4): the difference between the discharge elevation and the dynamic (pumping) water level, friction

Table 9.3 Costs of pumping for Example 9.9

Diesel fuel cost	**Diesel fuel costs**
The cost of fuel $2.70/gallon	$0.713/liter
Energy density	3.282 b-kW-hr/liter
Power required 36 HP * 0.746	26.9 kW
Energy required 1,800 hours * 26.9	48,300 b-kW-hr
Volume of fuel 48,300 b-kW-hr/3.28 b-kW-hr/L	14,729 L/yr
Annual cost 14,729 L * $0.713/L	$10,507/yr
Present value PV(0.08, 20, 10,507)	$103,158
Diesel pump cost	
Initial installation of engine, pump, and tank	$21,000
Present value pump year 16 $3,000 $(1 + 0.08)^{-16}$	$876
Present value engine year 14 $15,000 $(1+ 0.08)^{-14}$ =	$5,106
Annual maintenance cost present value PV(0.08, 20, 1,000)	$9,818.
Total PV $103,158 + $21,000 + $876 + $5,106 + $9,818	**$139,959**
Electric power cost.	
Cost of electricity	$0.09/kW-hr
Power required 36 HP * 0.746	26.9 kW
Electric power required 26.9 kW/Eff = 26.9 / 0.9 (40 HP)	29.89 kW
Energy required. 1,800 * 29.89	53,800 kW-hr
Annual cost 53,800 kW-hr * $0.09/kW-hr	$4,842
Present value PV(0.08, 20, 4,842)	$47,462
Electric pump costs	
Initial installation of pump, motor and power line $6,000 + $40,000	$46,000
Present value pump year 16 $3,000 $(1 + 0.08)^{-16}$	$876
Annual maintenance cost present value PV(0.08, 20, 400)	$3,927
Total PV $46,000 + $47,462 + $876 + $3,927	**$98,265**

Table 9.4 Wall thickness for steel gage pipe

Gage	Wall thickness (inch)	Wall thickness (mm)
20	0.040	1.02
18	0.052	1.32
16	0.064	1.63
14	0.079	2.01
12	0.109	2.77
10	0.138	3.51
8	0.164	4.17
6	0.203	5.16
3	0.259	6.58

losses in the pipe and fittings, and velocity and/or pressure head at the discharge.

Example 9.10 Calculate the discharge head requirement and HP for the pump shown in Fig. 9.14. Flow rate 64.6 L/sec. Pipe is 8″ (200 mm) 12 gage steel. Use Hazen-Williams C = 120.

Outside diameter of 200 mm pipe is found in Table 8.2: 219.1 mm. The wall thickness of 12 gage pipe is 2.77 mm. Thus, the inside diameter of pipe is 219.1 – 2.77 * 2 = 213.6 mm.

The equivalent length of a 45° standard steel elbow for 200 mm pipe as 16 * 200 mm = 3.2 m The two 45° elbows + length of pipe = 100 m + 6.4 m = 106.4 m.

Pipe friction pressure loss:

$$h_f = 1.22*10^{10}*106.4\ m\left(\left(\frac{64.6}{120}\right)^{1.852}/213.6^{4.87}\right)$$
$$= 1.9\ m$$

The kinetic energy at the discharge:

$$V = \frac{Q}{A} = \frac{0.0646\ m^3/\sec}{(0.2136\ m/2)^2*\pi} = 1.80\ m/\sec \quad \frac{V^2}{2g} = \frac{1.80^2}{2*9.8}$$
$$= 0.33\ m$$

Energy losses are summed for Pump TDH. The difference between the pumping water level (water level is drawn down during pumping) and the discharge elevation is the elevation gain:

7 m + 18 m = 25 m.

Elevation gain	7 + 18 m
Loss in 45°elbows + pipe	1.9 m
Velocity head lost at discharge	0.33 m
Total dynamic head requirement (TDH)	27.2 m

$$\text{Power} = \mathrm{Q}(\mathrm{m}^3/\sec)*\mathrm{H})/(0.102*\mathrm{Eff})$$
$$= (64.6/1000*27.2)/(0.102*0.75) = 23\ \mathrm{kW}$$

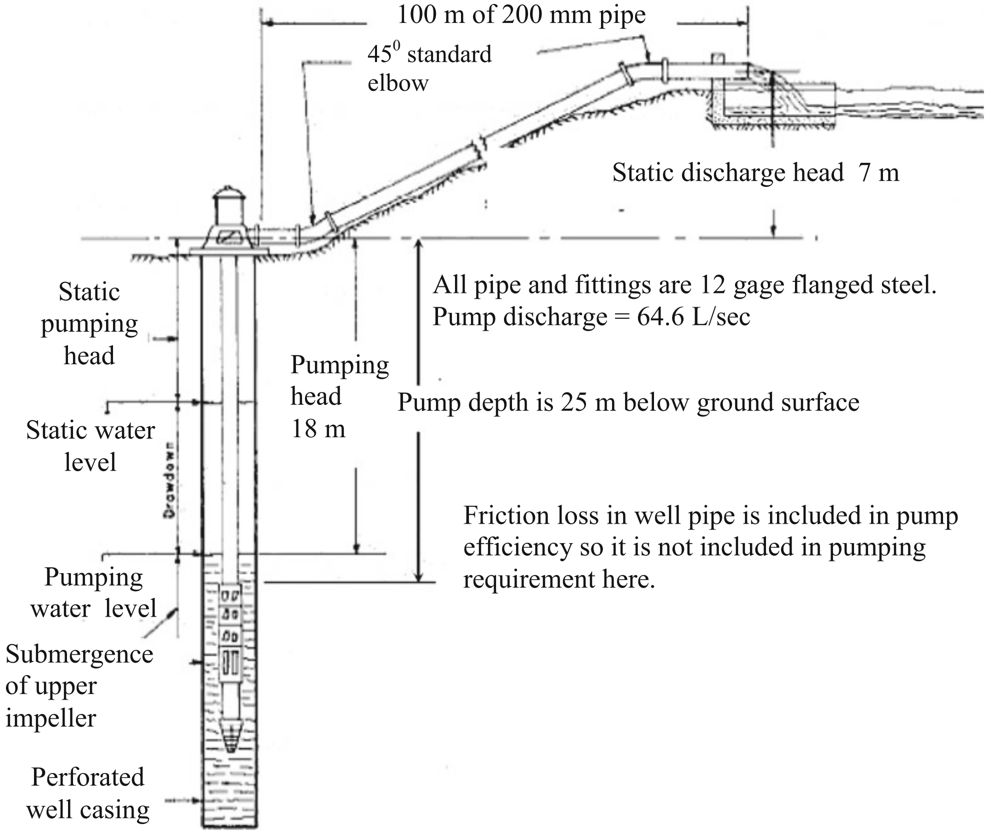
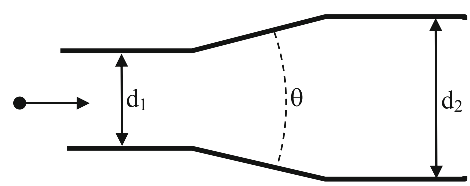


Fig. 9.14 Turbine pump pressure requirement calculation (Credit NRCS (converted to metric))

Fig. 9.15 Concentric cone expansion and reducer: equations for minor loss coefficients

$\theta < 45^0$ $$K = 2.6 \sin\frac{\theta}{2}\left(1 - \frac{d_1^2}{d_2^2}\right)^2$$

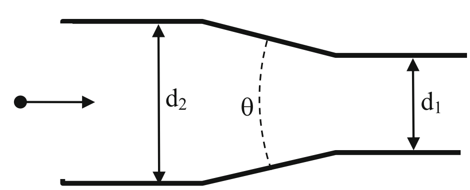
$\theta > 45^0$ $$K = \left(1 - \frac{d_1^2}{d_2^2}\right)^2 \sqrt{sin\frac{\theta}{2}}$$



Concentric cone reducer.

$\theta < 45^0$ $$K = 0.8 \sin\frac{\theta}{2}\left(1 - \frac{d_1^2}{d_2^2}\right)^2$$

$\theta > 45^0$ $$K = 0.5\left(1 - \frac{d_1^2}{d_2^2}\right)^2 \sqrt{sin\frac{\theta}{2}}$$



Centrifugal pump stations often use concentric cone expansion fittings and eccentric reducers to reduce friction loss as pipe size changes (Fig. 9.11). The minor loss coefficients are a function of inlet and discharge diameter and cone angle (Fig. 9.15). As with minor loss coefficients for a sudden expansion or sudden contraction, minor loss coefficients in these gradual transitions are multiplied by the higher velocity in the transition; thus, the minor loss coefficient is multiplied by the inlet velocity for an expansion cone, and by the discharge velocity for a reducing cone.

Example 9.11 A pump (Fig. 9.16) sucks water from a canal at a flow rate of 10 L/sec. The suction pipe is 4″. The suction section includes the following: 4″ fittings and pipe: basket strainer, foot valve, flanged steel sweep 90 with r/d = 4, and 2 m of straight pipe. An eccentric reducer is used to reduce pipe diameter from 4″ to the 0.2 m length 2″ diameter inlet nozzle pipe diameter (angle). Use the concentration cone equation to calculate eccentric K with angle θ = 44°. The pump discharge is located 1 m above the water surface of the supply canal. The discharge side of the pump is connected to

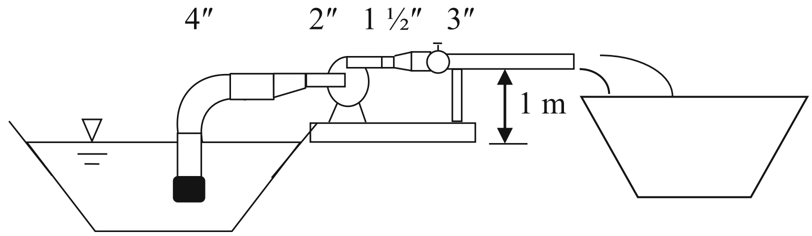


Fig. 9.16 Centrifugal pump station

a 4 cm length of 1½″ steel outlet pipe. A concentric cone with angle θ = 30° transitions from 1 ½″ to the 3″ steel discharge pipe. The 3″ discharge section includes a 3″ threaded gate valve and a 1.5 m length of steel pipe, which then has an open discharge that feeds water to a second canal. All pipe is 6 gage steel. Find the total friction loss in the pump station fittings, and the fitting with greatest head loss. Note: a centrifugal pump is not generally used for such a low head application as the one shown in Fig. 9.16, but this example demonstrates the calculation method.

The wall thickness of 6 gage pipe is 5.16 mm (Table 9.4), and outside diameters are found in Table 8.2. Inside diameters are calculated as follows:

Diam	OD (mm)	Wall thickness (mm)	ID (mm)
4	114.3	5.16	104.0
2	60.3	5.16	50.0
1½	48.3	5.16	38.0
3	88.9	5.16	78.6

The *Centrifugal pump fittings* worksheet (Fig. 9.17) calculates the head loss. The worksheet is divided into four sections because there are four pipe diameters represented in the fittings surrounding a centrifugal pump. The four sections are the suction, inlet, outlet, and discharge sections. There is also a mainline section, but this example does not have a mainline.

Fitting minor losses can be found in Appendix A. In this example, the basket strainer is assigned a K value of 1.3, which corresponds with the point at which the basket strainer K value line crosses the 4 inch line, which is the nominal diameter of the suction pipe. The sweep 90° has a minor loss coefficient of 0.24, which is found in Table 8.10 for a 4 in diameter pipe with r/d = 4.

The eccentric minor loss coefficient is calculated as follows:

$$\theta < 45^\circ \quad K = 0.8\ \sin\left(\frac{44(\pi/180)}{2}\right)\left(1-\frac{0.05^2}{0.104^2}\right)^2 = 0.18$$

The expansion cone minor loss coefficient is calculated as follows:

$$\theta < 45^\circ \quad K = 2.6\ \sin\left(\frac{30(\pi/180)}{2}\right)\left(1-\frac{0.038^2}{0.0786^2}\right)^2 = 0.40$$

Use the larger velocity for the expansion cone minor loss calculation:

$$h_m = K\frac{V^2}{2g} = 0.4\frac{8.82^2}{2{*}9.8} = 1.6\ \ m$$

The discharge pipe losses include the gate valve minor loss (K = 0.09 for screwed gate valve), pipe friction loss, and loss of kinetic energy at the discharge point (K = 1).

There are no mainline losses since there is no mainline pipe. Thus, the length of the mainline pipe is set at zero, and sum of K values for the mainline is also set to zero.

Total head loss in all fittings is 2.58 m. The head loss in the expansion cone, 1.6 m, is greater than all other losses in the pump station combined. If a 10° angle cone expansion joint were used, which allows for a smoother transition and less turbulence, then the friction loss is reduced to 0.53 m or 1/3 of the head loss with a 30° angle; however, a 10° cone expansion is not a normal fitting.

Example 9.12 A pump (Fig. 9.18) sucks water from a canal and discharges to a reservoir 20 m above the canal. All suction, inlet, and outlet parameters are the same as in Example 9.11. The discharge section includes the same length of pipe but also includes a steel threaded 102 mm (4 in. 90°. The length of the mainline pipe is 500 m and the pipe classification is 102 mm (4 in. SCH 40 PVC with C = 140. The mainline pipe also includes two 102 mm (4 in. PVC 90° in the ground just below the pump station. The pipe discharges to the upper reservoir through a submerged open outlet. Calculate the pump TDH required. Compare the percent of TDH due to pump station losses to the mainline friction loss and the elevation gain.

The *Centrifugal pump fittings* worksheet was used to calculate the head requirement. Changes are made to the discharge and mainline sections (Fig. 9.19).

The 90° Elbow loss in the discharge pipe is added here as an equivalent length of pipe rather than a minor loss. The equivalent length to diameter ratio is 20–30 and the diameter is 0.1 m; thus, the equivalent length is 2.5 m. Thus, the equivalent discharge pipe length is 1.5 m + 2.5 m = 4.0 m. Because the 90° elbow was included as an equivalent pipe length, the K value for angle bends in Cell B38 remains as zero. There is also a 75 mm (3 in. to 102 mm (4 in. bushing

	A	B	C	D	E	F
1	Component			Head requirement		Input data in
2	Required pump TDH			3.58	m	white cells
3	Total friction loss			2.58	m	
4	Mainline friction loss			0.00	m	
5	Total pump station friction			2.58	m	
6	Elevation gain			1	m	
7	Flow rate	10	L/sec			
8	Steel pipe C	120		Head loss		
9	Suction pipe total			0.2023	m	
10	Suction pipe length	2	m	0.0370	m	Pipe friction
11	Suction pipe diameter	0.104	m			
12	Basket strainer K	1.3		0.0919	m	Fittings
13	Entrance coefficient K	0		0.0000	m	
14	Foot valve K	0.8		0.0565	m	
15	Sweep 90 K	0.24		0.0170	m	
16	Cross-sectional Area	0.0085	m3			
17	Flow velocity	1.18	m/sec			
18	Inlet pipe total			0.3652	m	
19	Inlet pipe length	0.2	m	0.1309	m	Pipe friction
20	Inlet pipe diameter	0.05	m			
21	Eccentric angle (degrees)	44				
22	Eccentric K	0.18		0.2343	m	Fittings
23	Cross-sectional Area	0.001963	m2			
24	Flow velocity	5.09	m/sec			
25	Outlet pipe total			1.6660	m	
26	Outlet pipe length	0.04	m	0.0996	m	Pipe friction
27	Outlet pipe diameter	0.038	m			
28	Cone angle	30				
29	Cone expansion K	0.40		1.5664	m	Fittings
30	Other fittings K	0.0		0.0000	m	
31	Cross-sectional Area	0.001134	m2			
32	Flow velocity	8.82	m/sec			
33	Discharge pipe total			0.3445	m	
34	Discharge pipe length	1.5	m	0.1085	m	Pipe friction
35	Discharge pipe diameter	0.0786	m			
36	Discharge valve K	0.09		0.0195	m	Fittings
37	Discharge loss K (kinetic)	1		0.2166		
38	Angle bends K (sum)	0		0.0000	m	
39	Cross-sectional Area	0.004852	m2			
40	Flow velocity	2.06	m/sec			
41	Mainline pipe total			0.0000		
42	Mainline pipe length	0	m	0.0000	m	Pipe friction
43	Mainline pipe diameter	0.1023	m			
44	Mainline pipe C	140				
45	Mainline pipe total K	0		0.0000	m	Fittings
46	Cross-sectional Area	0.008219	m2			
47	Flow velocity	1.22	m/sec			

Steel pipe diameter			
6 gage	OD	t	ID (mm)
4	114.3	5.16	104.0
2	60.3	5.16	50.0
1.5	48.3	5.16	38.0
3	88.9	5.16	78.6

PVC mainline diameter			
Sch 40	OD	t	ID (mm)
4	114.3	6.02	102.3

Fig. 9.17 *Centrifugal pump fittings* worksheet for Example 9.11

as the diameter increases from the discharge pipe to the mainline pipe. This is classified as a sudden enlargement in *Chapter 8 minor losses* worksheet. All of the kinetic energy is lost in a sudden enlargement so the K value is 1.0. This should be included in the discharge section since it is the kinetic energy in the 75 mm (3 in. pipe that is lost. As a result, there is a discharge K value equal to 1.0 in the discharge section.

In the mainline, the two 102 mm (4 in. PVC 90° elbows in the ground have an equivalent length of 3.5 m each (Table 8.12). As a result of these fittings and the length of pipe, the equivalent length of mainline pipe is 507 m in cell B42. There is also a loss of kinetic energy at the discharge to the upper canal so the K value for the mainline pipe is 1.

With the additional energy loss and elevation gain, the total dynamic head that must be supplied by the pump is

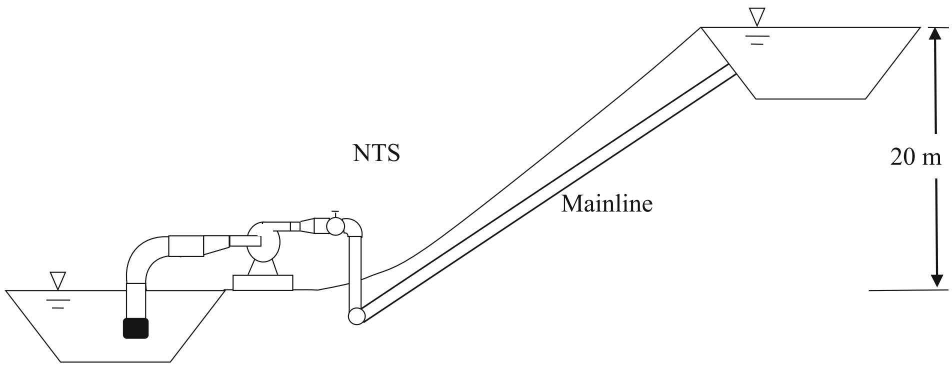


Fig. 9.18 Centrifugal pump lifting water 20 m from one canal to another

33	Discharge pipe total			0.5253	m	
34	Discharge pipe length	4	m	0.2892	m	Pipe friction
35	Discharge pipe diameter	0.0786	m			
36	Discharge valve K	0.09		0.0195	m	Fittings
37	Discharge loss K (kinetic)	1		0.2166	m	
38	Angle bends K (sum)	0		0.0000	m	
39	Cross-sectional Area	0.004852	m2			
40	Flow velocity	2.06	m/sec			
41	Mainline pipe total			7.8538	m	
42	Mainline pipe length	507	m	7.6372	m	Pipe friction
43	Mainline pipe diameter	0.1023	m			
44	Mainline pipe C	140				
45	Mainline pipe total K	1		0.2166	m	Fittings
46	Cross-sectional Area	0.008219	m2			
47	Flow velocity	1.22	m/sec			

Fig. 9.19 *Centrifugal pump fittings* worksheet for Example 9.12

30.61 m. Of that total 10.61 m is friction loss and 20 m is elevation. The pump station friction losses are 2.76 m. Thus, pump station friction losses are 9 % of the total dynamic head requirement.

Chemigation Systems

Many irrigation systems include the capability to inject chemicals, such as fertilizer, into the irrigation water. Chemigation systems include a chemical supply tank, injection system and safety devices (Fig. 9.20).

The three most common chemigation injection devices are piston pumps, diaphragm pumps, and Venturi injectors. Piston pumps and diaphragm pumps are classified as positive displacement pumps, whereas Venturi injectors rely on the Venturi pressure drop principle to draw the chemical from the tank into the irrigation pipeline. Positive-displacement pumps (Fig. 9.20) are recommended where precise control of chemical flow rate is required, because these pump flow rates remain stable over a range of irrigation pipeline pressures and flow rates.

In Venturi injector systems (Fig. 9.21), water is extracted from the main line and pressure is added with a centrifugal pump or by a pressure differential created in the mainline. The water velocity increases within the Venturi, and the pressure decreases below atmospheric pressure in the throat of the Venturi, causing the chemical to be sucked into the injector from the chemical reservoir. Because chemical is sucked into the irrigation system after the centrifugal pump, there is no contact between the chemical and the pump; thus, Venturi injection systems are less susceptible to corrosion than positive displacement pumps; however, Venturi injection flow rate is dependent upon chemical viscosity as well

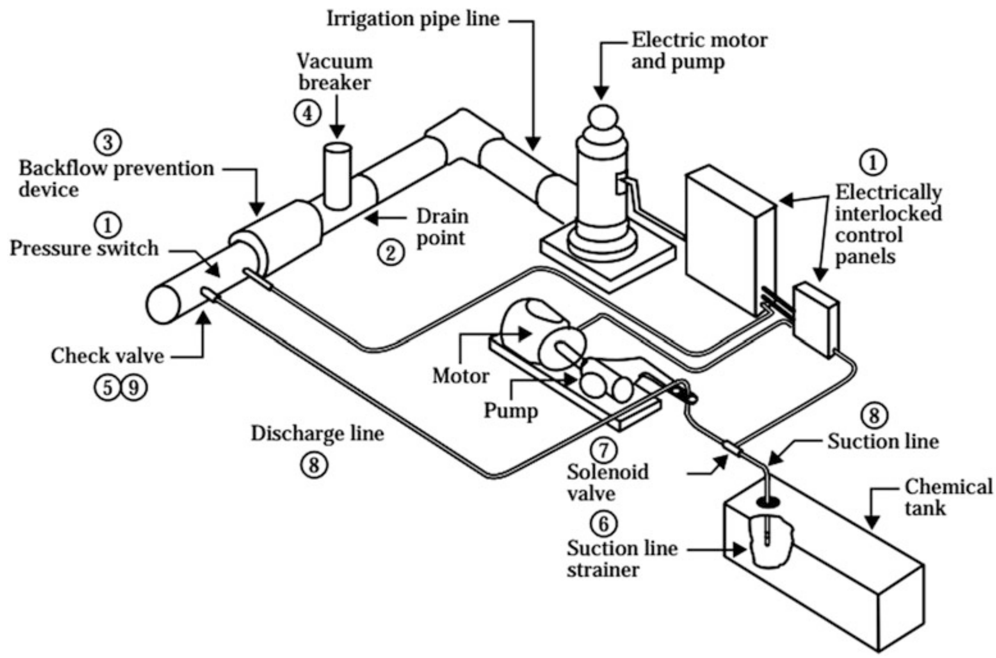


Fig. 9.20 Chemigation injection system (Credit NRCS 1997. NEH, Part 652, chapter 7)

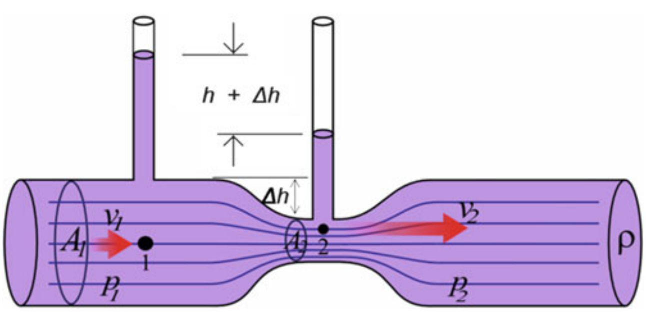


Fig. 9.21 Venturi injector (Wikipedia)

as irrigation pipeline pressure. Thus, the injection rate control is not precise. The change in Venturi injection flow rate can be in the range of 5–10 % for a temperature change of 20 °C for viscous fertilizers

In addition to flow rate range and precision, the primary selection criteria for chemigation systems are durability, accuracy, ease of operation and repair, service life, and susceptibility to corrosion.

It is convenient to place a flow meter in the injection line in order to adjust the injection system to the proper flow rate.

Injection systems should always provide for complete mixing and uniform concentrations before the chemicals reach the field. Chemical will mix naturally with water in an irrigation pipeline due to turbulent eddies. However, a minimum length of pipe is required for complete mixing.

Proportional pumps vary the injection flow rate with water flow rate, providing a constant ratio of chemical to irrigation water. These are especially useful in hydroponic systems, where maintaining an injection ratio is the goal.

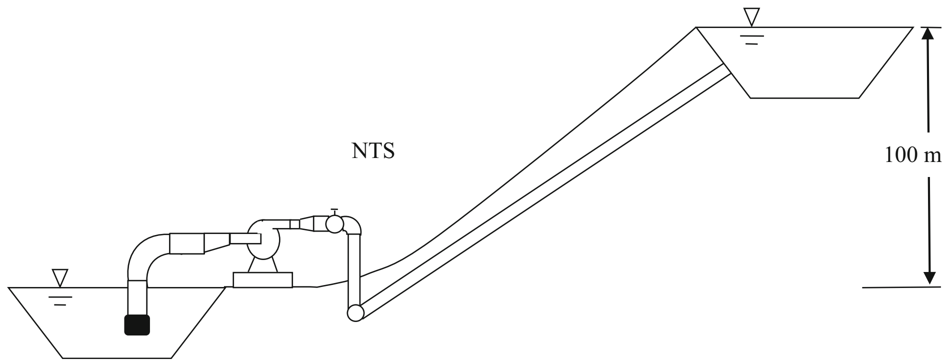
In-class Exercise 9.5 Venturi injectors are designed based on the principle that if water velocity increases, then pressure decreases, as shown by the Bernoulli equation. A narrow throat increases the velocity at the suction point. Concentric cones are used to gradually increase flow rate to the throat and decrease flow rate from the throat. Based on what you know about concentric cones, draw a Venturi injector geometry that has minimum head loss.

Questions

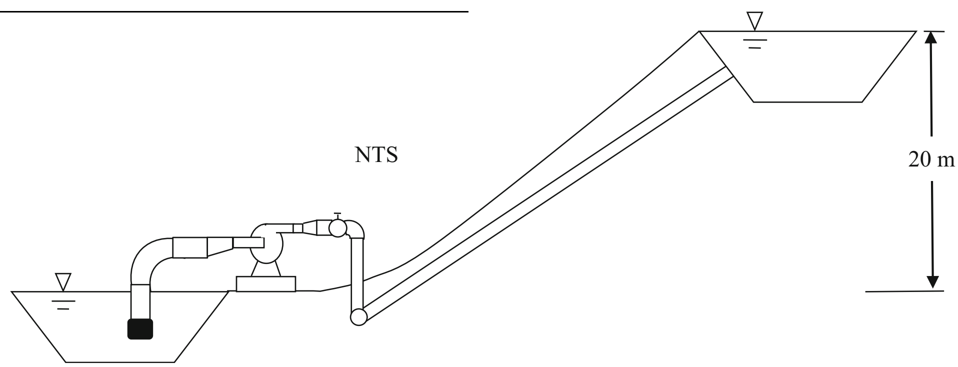
1. The revolutionary speed of electric pumps is slightly less than divisors of 3600. Typical pump rpm's are 875, 1750, and 3500. Why are most pumps manufactured with these revolutionary speeds?
2. What would be a typical TDH for a centrifugal pump with flow rate 1,000 m^3/hr based on the typical specific speed for a centrifugal pump? Recalculate for pump flow rates of 100 m^3/hr and 10 m^3/hr. What type of pump would be appropriate for a very high flow rate and very low head?
3. Using the equations for the relationships between power, flow rate, and head, describe the shape of the head/capacity curve if efficiency was constant over a range of flow rates?
4. Verify that the water horsepower generated by the 5.9375 impeller curve in Fig. 9.4 corresponds with the efficiency and brake horsepower (the curve below the head capacity curve). Calculate at the point of highest efficiency.
5. Describe the relationship between efficiency and flow rate in Fig. 9.4.
6. An irrigation system requires 600 gpm and 160 ft head. Select the best impeller for this application on the B4JPBH (Fig. 9.5) pump curve.
7. What is the maximum allowable flow rate of a B4JPBH pump (Fig. 9.5) with a 12 3/8″ impeller and a 40 HP motor? What is the maximum flow rate for the 50 HP motor with the same impeller?
8. An irrigation system requires TDH = 168 ft and Q = 600 gpm. Select an impeller diameter (trimmed if necessary) and select a motor HP with the B4JPBH pump.
9. The 16BZ pump (Fig. 9.7) with a 5¾″ impeller is used to run a sprinkler system. There is a 2 m pressure loss in pump fittings and filters. Find the operating point. Plot the two curves and verify that the calculated point is the correct point. The 5¾″ head-capacity curve and the irrigation system curve are:

$$\mathrm{TDH(m)} = -0.00170\mathrm{Q}^2 + 0.0743\mathrm{Q} + 43.76$$
$$\mathrm{Q_{system}}(\mathrm{m}^3/\mathrm{h}) = 14.175\left(\mathrm{H_{system}}\right)^{0.531}$$

10. In Example 9.6, change the elevation of pivot 2 to 100 m elevation and pivots 3 and 4 to 120 m elevation. Select the pump operating pressure. Each pivot requires 100 L/sec. Determine the number of pumps, flow rate, and TDH of the pump station. Discuss options to reduce energy.
11. A variable speed pump controller is used to vary the flow rate of the 16BZ pump with the 5 3/4″ impeller. The revolutionary speed is lowered from 3500 to 3000 RPM. The system curve is Q_{system} (m^3/h) = 14.175 $(H_{system})^{0.531}$. There is 3.5 m head loss in the pump fittings and filters. Find the operating point TDH and flow rate.
12. Imagine that a new technology was developed that enabled farmers to produce biodiesel from crop residue. The biodiesel production unit has a capital equipment cost of $50,000; a labor, maintenance, and energy cost of $0.30/L, and produces 15,000 L of biodiesel per year. Calculate whether this would be a less expensive alternative than the electric pump system in Example 9.9. Use the *Fuel and pump costs* worksheet in Chapter 9 Excel program.
13. Redo Example 9.9 with a solar powered pump. Based on the cost of materials and the service life and replacement cost of solar components, the solar panel array provides electrical energy at a cost of $0.08/kW-hr for the 20 year project life. The solar pump can only be used during daylight; thus a larger pump is required and a reservoir must be constructed for storage. Increased capital cost of hydraulic components is $50,000 and replacement and maintenance costs remain the same as Example 9.9. Recalculate if carbon credits for the system are worth $1,000/yr.
14. A pump sucks water from a canal and discharges to a reservoir 100 m above the canal. Pump station valves and fittings are the same as in Example 9.11 except that the pipe diameters are 6″, 3″, 2.5″, and 4″ instead of 4″, 2″, 1½″, and 3″. Two other changes are that the eccentric angle is 50° and the cone angle is 40°. Flow rate is 20 L/sec. All pump station pipe is 6 gage steel, and the mainline pipe is 4″ SCH 40 and is 500 m long. Assume an open discharge to the upper reservoir. Calculate the pump TDH required. Show calculations for the pressure loss in the eccentric reducer and the concentric cone. Calculate the percent of required TDH due to pump station losses, and the percent of total friction loss that is due to pump station losses. (Use worksheet)



15. Redo question 15, but discard the eccentric, and let the suction pipe be all 75 mm (3 in. pipe. Second, use a bushing on the discharge side (sudden expansion) rather than a cone expansion. Determine which change results in the greatest increase in head loss.
16. Use the 16BZ pump with 5 3/4″ impeller (M) to deliver water to the upper reservoir for the system shown below. Select pipe diameters equal to 6″, 4″, 3″, and 4″ for the four pump station pipe sections. Use 4″ Schedule 40 PVC for the mainline, which is 493 m long. Draw a system curve (develop with *Centrifugal pump fittings* worksheet by inputting different flow rates and corresponding TDH requirement) and pump head-capacity curve based on Fig. 9.6. Find an exponential equation for the system curve and equation for the head-capacity curve, and calculate the point of intersection (operating point) for the system



17. Venturi injectors are designed based on the principle that if water velocity increases, then pressure decreases as shown by the Bernoulli equation. A narrow throat increases the velocity at the suction point. Concentric cones are used to gradually increase flow rate to the throat and decrease flow rate from the throat. Based on what you know about concentric cones, draw a Venturi injector geometry that has minimum head loss.
18. Some people recommend creating the pressure differential across a Venturi by restricting mainline flow. It is a much better idea to have a separate centrifugal pump provide the pressure differential, as shownin this example. Mainline flow rate is 200 L/sec, and Venturi flow rate is 0.90 L/sec. Venturi injection time is 1,000 hours per year. The required pressure differential across the Venturi is 283 kPa. The cost of energy is $0.10/kW-hr. Calculate the energy cost per year for providing the required pressure differential across the Venturi by constricting the mainline flow with a valve. Calculate the energy cost of using a centrifugal pump in the bypass line to provide the pressure differential needed by the Venturi.
19. How could a Venturi be used within a pump to lift groundwater up to the surface in a well. (Hint: look up jet pumps).
20. Calculate throat pressure (gage pressure and absolute pressure) and discharge pressure in a Venturi injector that that has a 30 mm internal diameter at both ends and that has a flow rate of 0.9 L/sec. The length of the entire Venturi is 15 cm and the length of the throat is 2 cm; however, assume that the equivalent length of the throat is 10 cm due to flow entering the throat through the suction tube. The upstream pressure is 300 kPa. The

reducer cone angle (inlet side), θ, is 30°, and the expansion cone angle (discharge side) is 15°. Assume that Hazen-Williams C in the throat is 100. The inside diameter of the throat is 7 mm.

21. Redo question 20 but optimize the inlet and discharge angle in order to minimize pressure loss across the Venturi. Keep the same throat dimension and Venturi length. Derive an equation based on the geometry of the system that calculates discharge angle as a function of inlet angle.

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