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Irrigation Pumps - Preserving Performance

Derrel L. Martin

University of Nebraska-Lincoln

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IRRIGATION PUMPS - PRESERVING PERFORMANCE

D.L. Martin

Professor, Irrigation and Water Resources Engineering
Department of Biological Systems Engineering
University of Nebraska-Lincoln, Lincoln, Nebraska
Voice: 402-472-1586 Fax: 402-472-6338
email: derrel.martin@unl.edu

Introduction

Approximately 120,000 irrigation pumps functioned on farms to pump groundwater in the three-state area according to the USDA Farm and Ranch Irrigation Survey in 2013 (USDA, 2013)—see Table 1. The average pumping capacity varies from 600 gallons per minute (gpm) in Kansas to 775 gpm for Colorado. The average operating pressure was 30 pounds/square inch (psi) in Kansas and about 40 psi for Colorado and Nebraska. The average horsepower varies from 78 to 129 across the region while the average hours of operation varies from about 680 to almost 1400 hours.

Table 1. Characteristics of on-farm pumps for groundwater in the three states (from USDA, 2013).

State	Number of Pumps	Average Pumping Capacity, gallons per minute	Average Operating Pressure, pounds/sq. inch	Average Engine Size, hp	Average Hours of Operation
Colorado	14,062	775	41	78	1152
Kansas	24,137	605	30	129	1397
Nebraska	83,018	763	40	93	681
Total	121,217				

Of course, pumps require energy to lift, convey and pressurized water for irrigation systems. The forms of energy used in each state varies considerably (Figure 1). Electricity is dominant in Colorado and represents about 56% of the pumps in Nebraska. Natural gas leads in Kansas at about 41% with electricity at about 35%. If all systems used diesel engines, and the pumps operated at the average efficiency found in earlier tests, then the annual cost of pumping would be approximately two billion dollars if diesel fuel cost \$2.50/gallon. If all systems were electrically powered, and electricity cost \$0.10/kWh, then the aggregate cost would be approximately \$1.2 billion annually. Obviously, this represents a very large use of energy and significant production costs for the three states. Tests have shown that the average performance of pumping plants is from 75–80% of what a good pumping plant should use; thus energy costs are 20–35% higher than necessary. Many wells were much lower than the performance of a good pumping plant.

The goal of this paper is to review how pumps work relative to what is required to achieve and maintain high pumping efficiency to minimize energy requirements and pumping costs. A paper was presented at the conference last year that describes a method to monitor and ultimately test pumping plant performance (Martin, et al. 2017). The papers form a series on how pumps work and how to monitor to ensure they work efficiently.

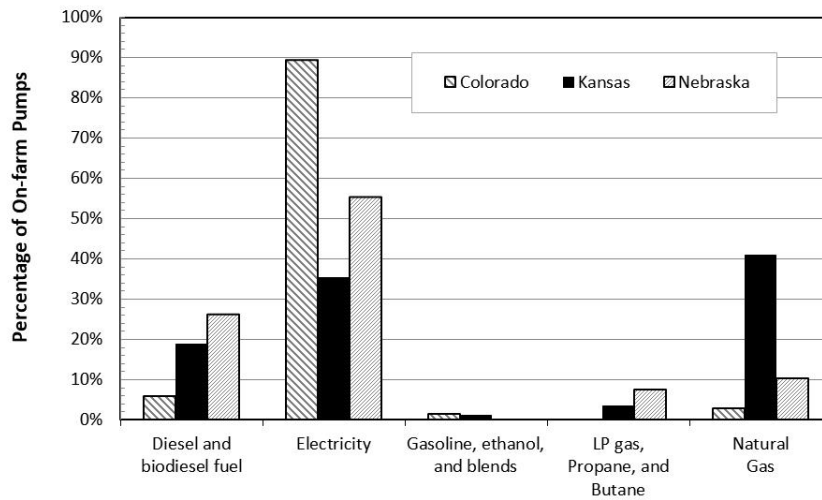


Figure 1. Energy sources used for irrigation in the Northern HighPlains.

Pump Types and Components

Most irrigation systems require a pump to lift water from the source and develop the pressure required to distribute water through an irrigation system. Pumps convert mechanical energy from the power source into hydraulic energy. The types of pumps most often used to convey water for irrigation include horizontal centrifugal and vertical turbine pumps. Horizontal centrifugal pumps commonly pump water from an open source or add pressure for enclosed systems. These pumps usually have a horizontal shaft and can be coupled to an electric motor or driven by an engine. Typical installations are shown in Figure 2 where water is pumped from a pond, canal or stream, and for pressurizing water in pipelines for sprinkler systems.

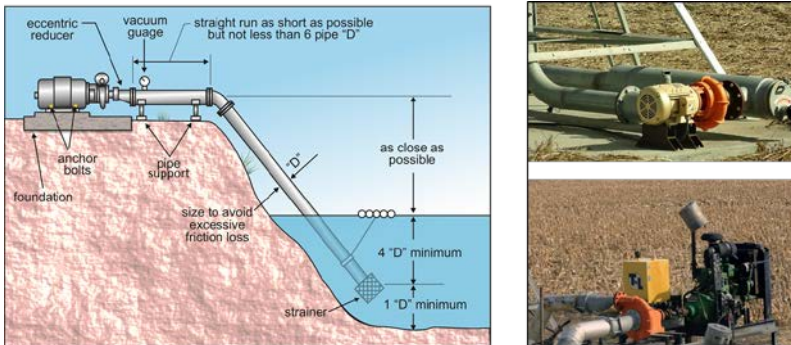


Figure 2. Examples of centrifugal pumps for lifting from a canal and/or boosting pressure for center pivots.

An example of a deep well turbine pump is included in Figure 3. Note that the pump for turbine systems is actually at the bottom of the column, not at the soil surface. Vertical turbine pumps often supply water from a well. Water enters the eye of the impeller and through centrifugal force, water pushed outward and upward by the vanes of the impeller. This process develops head needed for an irrigation system. Lifting water and delivering it at a desired pressure may require staging the impellers and bowls—two stages are shown in Figure 3. With staging, water flows from one impeller into another until the desired head is achieved.

Pumps that are powered by electrical motors usually operate at the same speed as the motor unless v-belt drives are used. Pumps powered by engines can operate over a wide range of speeds.

All centrifugal and turbine pumps consist of one or more impellers that rotate at the speed of the line shaft. The Impellers contain curved vanes that force water from the outer edge of the impeller due to centrifugal force. Impellers generally have a shroud on the back of the impeller, and many impellers also have a shroud on the front of the impeller. Impellers with only one shroud at the back are called semi-open impellers while impellers with a full or partial shroud on the front are closed or semi-enclosed impellers (Figure 4).

Water enters impellers through the eye of the impeller—the center of the front of the impeller. Water is channeled away from the impeller by the bowl that contains smooth waterways that efficiently transport water to the pump discharge.

Pumps are only one component of a groundwater pumping plant. The characteristics of the well, gravel pack, well screen and casing all affect performance (Figure 5). All components of the pumping plant must be matched to the aquifer characteristics and the water requirements of the proposed crops. The power supplied by a

motor or engine must be matched to the power requirements for lifting, conveying and pressurizing water. Pumping efficiency decreases substantially when components are not properly matched. Pumping cost increases when pumping efficiency decreases. This is significant because pump and well components are often in place for decades so inefficiency costs for a very long time.

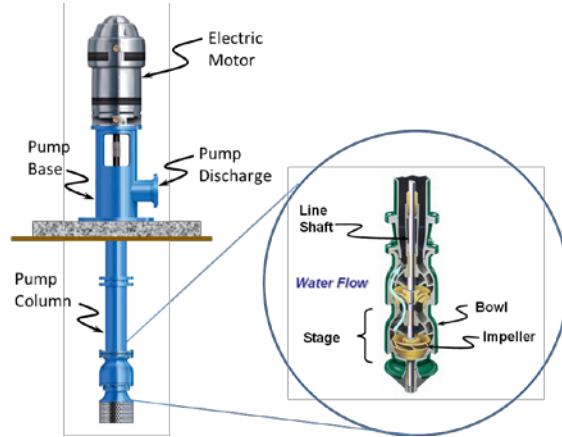


Figure 3. Example of deep well turbine pump with cutaway of pump components.

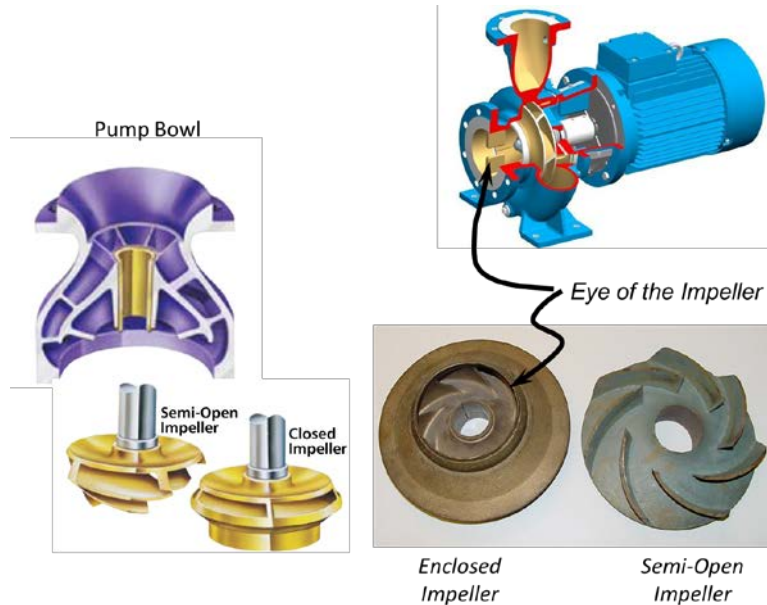


Figure 4. Impellers and bowls for centrifugal pumps.

Pump Information

Information about the well design and pump characteristics is available from manufacturers and the nameplate on the pump discharge head. The discharge head is the base on which the electric motor or right angle gearhead sits. A nameplate should be available that gives information about the make, impeller size and number of stages. There may also be information about the well size and depth. Landowners in Nebraska must register irrigation wells. Information from the registration process can provide data about the specific well and pump. Registration information can be accessed at the website sponsored by the Nebraska Department of Natural Resources.

Pump Curves

Pump curves—also known as characteristic or performance curves—describe the operating characteristics of pumps (Figure 6).

Manufacturers provide curves for their pumps. Curves, which are essential for design and analysis of pumping plants, provide information on flow capacity, total head developed, efficiency of the bowls, and the horsepower required to operate the pump. The pump curve in Figure 6 is for a single stage of a 12-inch pump operating at a 1760 rpm with an impeller diameter of 9.02 inches. Pump capacity (Q) in gallons per minute (gpm) is given on the bottom scale (x-axis) while the total dynamic head (TDH) in feet is given along the vertical scale (y-axis). A pump with a specific diameter of impeller operating at a selected speed produces a certain amount of head for a given flow. The thicker curve in Figure 6 is the head-capacity curve relating the flow to head. As the head increases the flow decreases; likewise, the smaller the head the larger the flow.

A pump curve also gives the efficiency of the bowls (E_p) and the brake horsepower (bhp) required to produce flow at the corresponding head. There are many formats for graphs used to describe pump performance. An example in Figure 6 shows the efficiency in the upper part of the graph and the bhp in the lower part. Pump manufacturers also provide information about the amount of head needed at the pump inlet to avoid cavitation in the pump. This process is described later.

The brake horsepower can be computed from the flow, head and pump efficiency. The brake horsepower is equal to the water horsepower (whp) divided by the pump efficiency (E_p expressed as a decimal fraction, *i.e.* 0.95 for 95% efficiency):

$$whp = \frac{Q \times TDH}{3960} \text{ and } bhp = \frac{whp}{E_p} \quad (1)$$

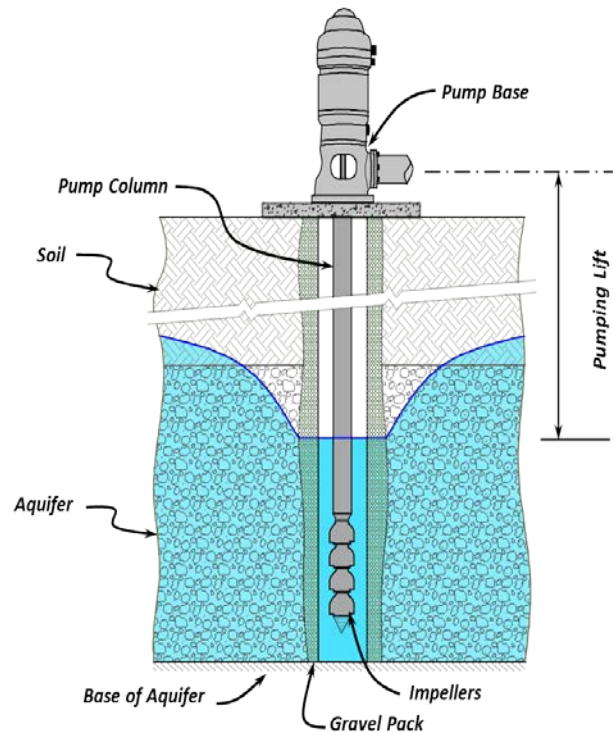


Figure 5. Components of a groundwater well.

Reading a pump curve is straightforward. Suppose the total dynamic head, pump efficiency, and horsepower are needed when the pump in Figure 6 produces 700 gpm. Start at 700 gpm for the flow. Move vertically upward to the head-capacity curve. Move to the left to read the total dynamic head the pump will produce; which is about 60 feet for one stage. Continuing upward and to the right axis shows that the bowl efficiency will be about 83%. The brake horsepower requirement for 700 gpm at 60 feet of head and an efficiency of 83% is about 13 horsepower per stage. The required net positive suction head for 700 gpm is about 8 feet. Note that data in Figure 6 represents performance for only one stage.

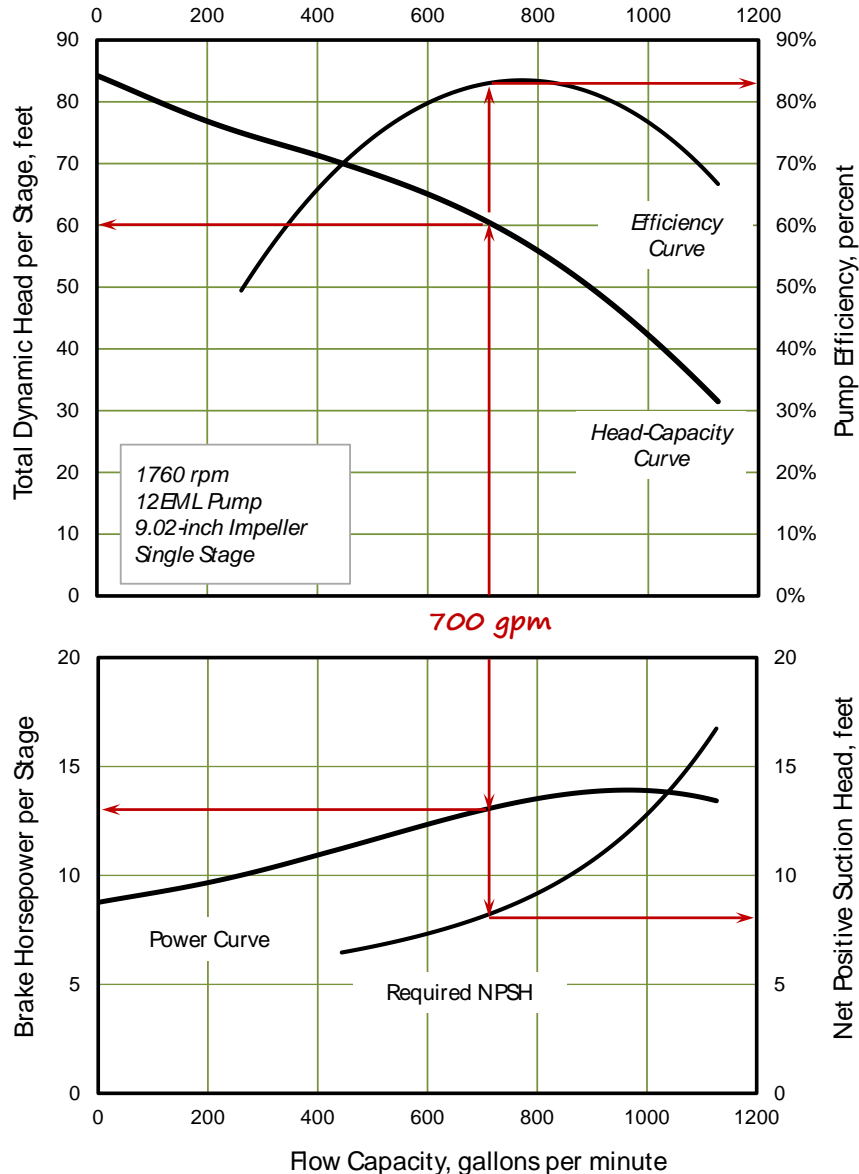


Figure 6. An example of a pump performance curve for a single stage.

One can also start out knowing the head requirement and determine the gpm output, efficiency, and bhp. For example, using the pump in Figure 6, suppose the head requirement is 50 feet, then the flow will be about 900 gpm, the efficiency will be about 83% and the brake horsepower requirement will be 14 feet per stage.

Impeller Speed and Trim

Vertical turbine and centrifugal pumps are based on centrifugal force which means operating characteristics of a pump vary when the speed of rotation and/or the impeller diameter. This allows pumps to operate over a range of conditions while maintaining good efficiency. Instead of making a single pump performance chart for each size of impeller or speed, manufacturers often place several pump curves in a single graph. This gives better view of how pumps perform for different speeds or impeller sizes. The performance of the pump in Figure 6 is shown in the upper portion of Figure 7 for three diameters of impellers. The efficiency is often shown as a series of lines

superimposed over the head-discharge curve. The hatched curves show lines of equal bowl efficiency. The number of stages of impellers and bowls affects the efficiency as well. The insert in Figure 7 shows that the efficiency can be increased when multiple stages are used. For example, if the 8.89-inch impeller was used and the discharge was 700 gpm, then the efficiency for a single stage is about 80%. If four stages of the pump were needed the efficiency would increase to 83%. The performance curve of the same pump with an impeller diameter of 8.89 inches but operated at three speeds is shown in the lower portion of Figure 7.

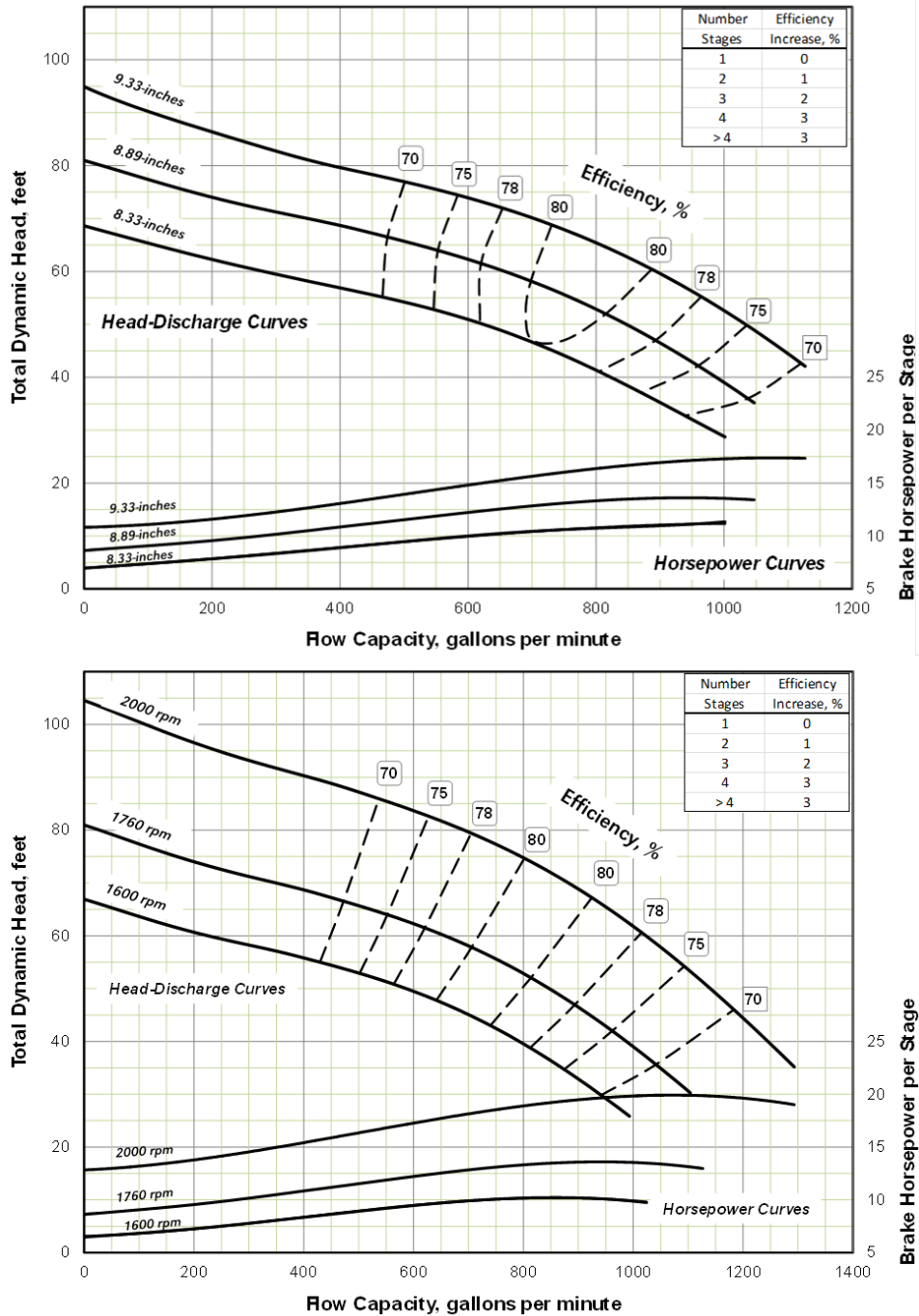


Figure 7. Pump curves for multiple impeller diameters or one size impeller operated at multiple speeds.

Pump Staging

A single stage pump often does not produce enough head for the lift and/or discharge pressure required for an irrigation system. Vertical turbine pumps can be installed with stages (i.e., bowl and impeller) in series to increase the head for a specific flow. This is known as staging. Water passes through each stage where pressure is added to the water for each stage. Thus, the head- capacity and horsepower curves for a single stage can be added to give the performance for a series of stages as shown in Figure 8. For example, a single stage produces about 60 feet of head when the flow is 650 gpm for a single stage. If two stages were installed in series the head would double to approximately 120 feet at 650 gpm. Three stages would triple the head to 180 feet and the horsepower would also triple. Four stages would quadruple the head to 240 feet and so on for more stages. In this manner, pump companies can use one pump model, and add stages to fit a range of pumping applications. The insert in Figure 8 illustrates that the efficiency usually improves when multiple stages are used compared to a single stage.

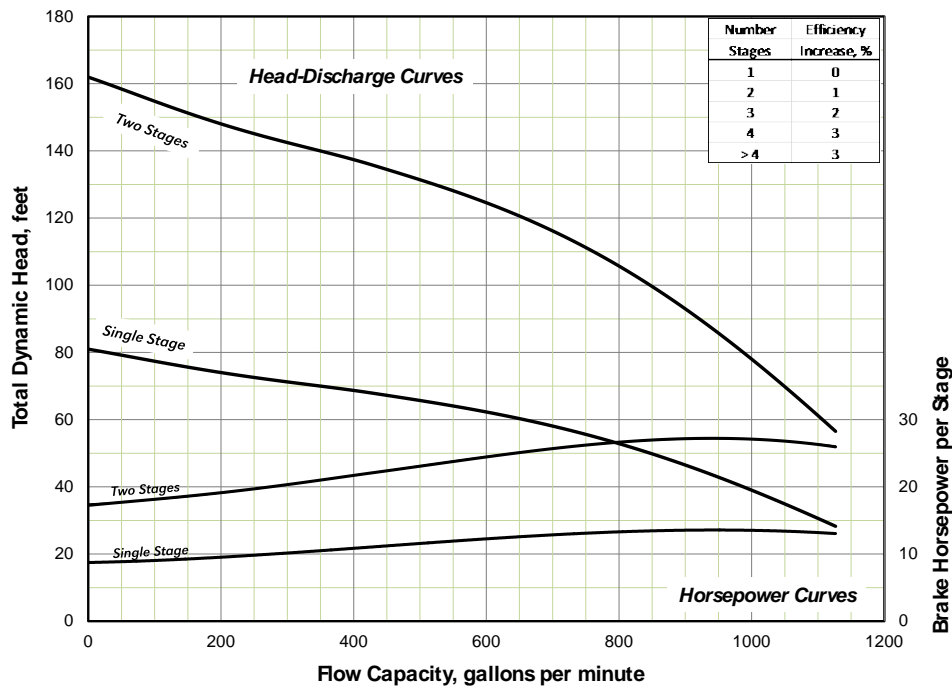


Figure 8. Pump performance curves for one and two pump stages in series.

Affinity Laws

Pump manufacturers often publish head-capacity curves for a full diameter impeller operating at the 4-pole electric motor speed of 1760 or 1770 rpm. They usually publish head-capacity curves for other speeds or diameter sizes. Pumping conditions may require an impeller diameter or speed that is not included on published pump curves. For example, if an engine is used to power the pump the speed may vary from speeds that occur with electric motors. There are laws, known as the Affinity Laws, which allow us to derive head- capacity and horsepower curves for speeds and diameters different from published curves. Many companies now offer software tools to create performance

curves for selected pumping conditions. Those tools are preferable to the affinity laws presented here for detailed analysis. However, affinity laws can be used to understand how pumps perform and to develop quick solutions for changes in rotation speed or impeller diameter.

One application of the Affinity Laws involves the rotational speed of a pump. The ratio of the flow for the new speed (Q_2) to the flow for the initial speed (Q_1) equals the ratio of the final (RPM_2) and initial rotational speed (RPM_1):

$$Q_2 = Q_1 \times \left(\frac{RPM_2}{RPM_1} \right) \quad (2)$$

The ratio of the final head to the initial head is proportional to the square of the ratio of the final and initial rotational speed:

$$H_2 = H_1 \times \left(\frac{RPM_2}{RPM_1} \right)^2 \quad (3)$$

The ratio of the final brake horsepower to the initial brake horsepower is proportional to the cube of the ratio of the final to the initial rotational speed:

$$bhp_2 = bhp_1 \times \left(\frac{RPM_2}{RPM_1} \right)^3 \quad (4)$$

The Affinity Laws holds true for most types of pumps used in irrigation including centrifugal, angle flow, mixed flow and propeller pumps. As a general rule the laws should not be applied for very large changes in speed.

To illustrate the approach consider Figure 9. This is the curve for a pump operated at 1760 rpm, which is the published speed. We desire a curve for a speed of 1900 rpm. Since all three values are based on the ratio of the final speed to the initial speed, the first step is to find this ratio, $1900 \text{ rpm} / 1760 \text{ rpm} = 1.08$. Select points along the original curve for analysis. It is usually good to pick points at the intersection of the efficiency lines.

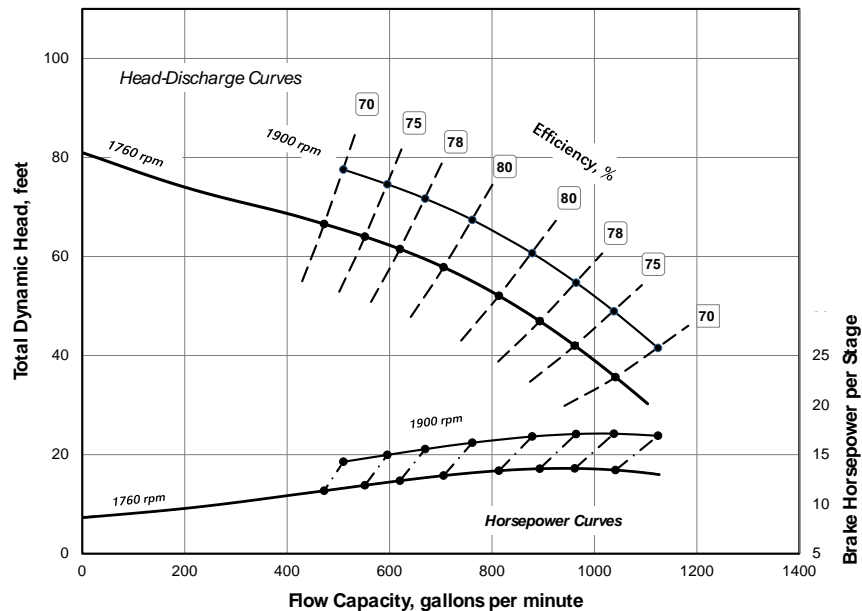


Figure 9. Example of affinity laws for pump speed changes.

Eight points on the 1760 rpm curve were selected in Figure 9. The first point is at a flow of 472 gpm and a head of 66.6 feet. This point translates to a flow of about 510 gpm (i.e., 1.08 x 472 gpm) and a head of 77.6 feet (i.e., 66.6 ft x 1.082) for 1900 rpm. The remaining seven points are translated in a similar fashion to produce the new curve for a speed of 1900 rpm. The efficiency for the points on the 1760 rpm curve translate to the 1900 rpm curve as well. Thus, the efficiency of each point on the 1760 will have the same efficiency on the 1900 rpm curve as it had on the 1760 rpm curve. A similar process is used to develop the new brake horsepower curve for 1900 rpm. Notice that the new head-capacity curve is parallel to the original curve but the new bhp curve is not parallel to the original. It is therefore necessary to plot several points to derive a new bhp curve.

The second form of the Affinity Law involves changes in the diameter of the impeller. This version of the Affinity Law is used to determine the change in performance when the diameter of the impeller is changed from the published size. The diameter of a full sized impeller can be machined down to a smaller diameter to provide the desired head-capacity curve when specific applications require such accuracy.

The effect of the diameter is similar to the effect of pump speed. The discharge for a new impeller diameter is linearly related to the ratio of the final (D_2) and initial (D_1) impeller diameters. The head is proportional to the diameter ratio squared and the brake horsepower is proportional to the cube of the diameter ratio. The relationships for the effect of diameter are expressed as:

$$\begin{aligned}
 Q_2 &= Q_1 \times \left(\frac{D_2}{D_1} \right) \\
 H_2 &= H_1 \times \left(\frac{D_2}{D_1} \right)^2 \\
 bhp_2 &= bhp_1 \times \left(\frac{D_2}{D_1} \right)^3
 \end{aligned}
 \tag{5}$$

The diameter-based version of the Affinity Law is applied similar to the speed-based version.

For example, if a full impeller as shown in Figure 7 is 9.33 inches and if the impeller was machined down to a diameter of 8.4 inches, then the diameter ratio would be 0.9. In this case the flow would be reduced to 0.9 times the original flow, the head would be 0.81 times the original head and the brake horsepower would be 0.73 times the original horsepower. The efficiency would be the same as the original point. New curves can be developed for the custom-made impeller similar to the process for the impact of changing impeller speed.

The diameter-based Affinity Law should not be used for trims larger than 20%. Changes in the nature of the impeller are usually too significant for such large reductions of diameter. As always, it is strongly advised to coordinate pump changes with a pump supplier to ensure that proposed changes are advisable.

Matching Pumps and Systems

Pump curves describe the amount of head that the pump will develop at specific flow rates. The previous sections describe the performance of a pumping system that has multiple stages operated at varying speeds and with specified diameters of impellers. The output from the pump has to

match the head requirements for the irrigation system. The head required for varying flows within the irrigation system is referred to as the system curve.

Consider the system illustrated in Figure 10. The pump must develop enough head to lift water from the pumping level in the well to the pump base, lift water from the pump base to the elevation of the center pivot lateral, and overcome pressure loss due to friction of water flow in pipes and fittings along the pipeline. The pump must also develop the pressure required to operate the sprinklers on the pivot lateral and to overcome the pressure loss in the pivot lateral and lift water to the highest elevation at the end of the pivot lateral. Some factors such as the elevation from the pump to the pivot inlet, or the depth of water below the pump base to the static water level are constant. Those values do not change with the amount of flow through the system. Other factors such as the friction loss and the pressure required to operate the pivot depend on the rate of flow through the system.

The head-capacity curves for pumps with 3, 4 and 5 stages are shown in Figure 11. The head required for specific flows is referred to the system curve and is illustrated for the example system in Figure 10.

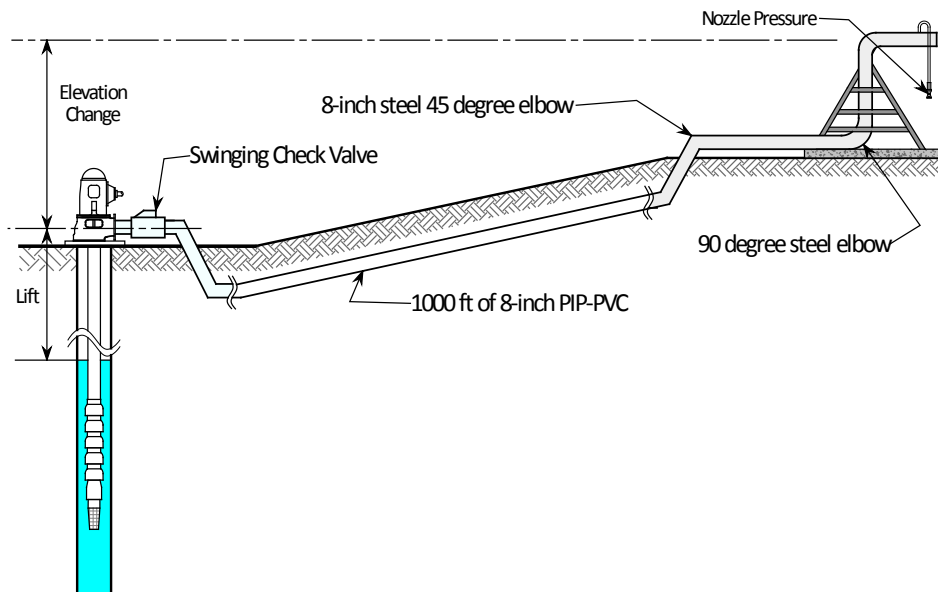


Figure 10. System layout for a typical center pivot field.

The actual operating point for the system occurs where the output head from the pump matches the head required for the system. The points where the heads are equal are often referred to as match points as shown in Figure 11. The operating point for the combined pump, pipe and pivot system depends on the number of stages of impellers installed with the pump. If only three stages were included then the flow would be about 655 gpm with a total dynamic head of 180 feet of head. If four stages were used, the flow and head increase to about 780 gpm and 215 feet. Five stages provide about 860 gpm and 245 feet of head. The most desirable operating point is the one closest to the design flow rate for the center pivot. The pump efficiency for the selected match point should also be near the peak efficiency so that operation is economical. In this case, all three match points have efficiencies above 80%, which is near the maximum value. The brake horsepower for the match point with four stages is about 51 horsepower.

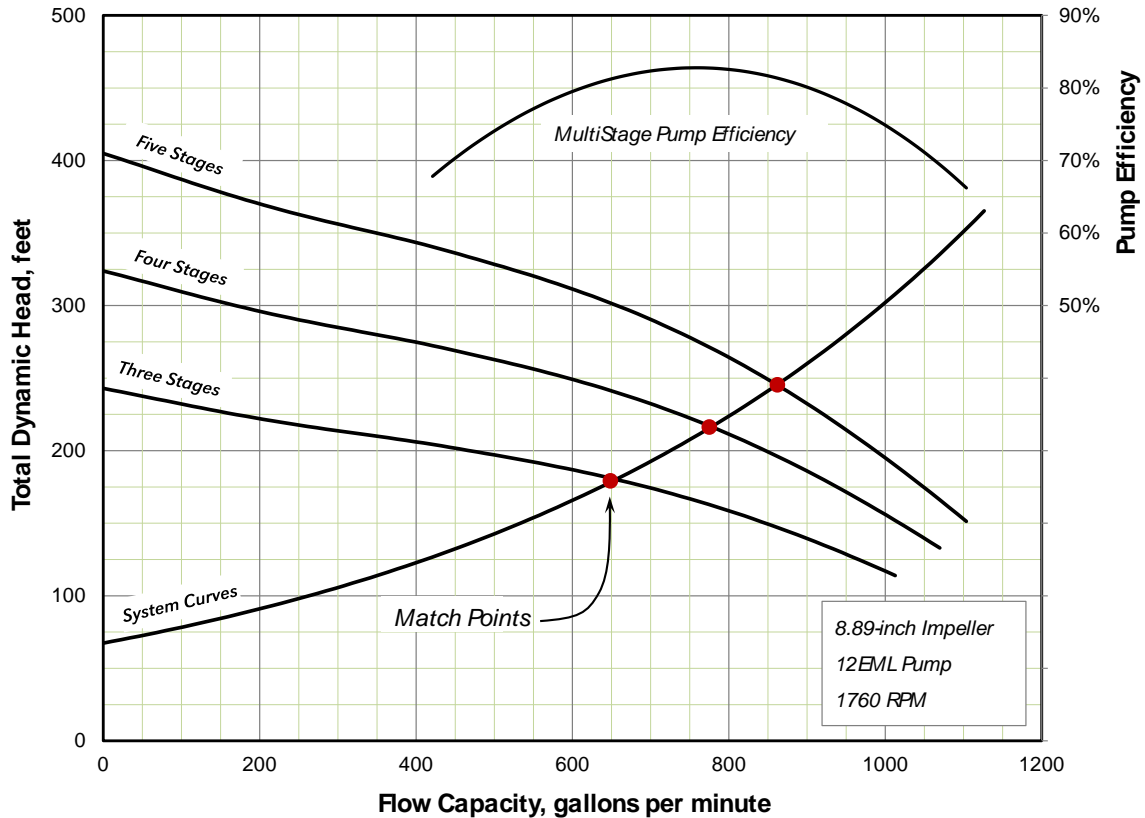


Figure 11. Characteristic curves used to determine the flow and head where the pump and the system match.

Irrigation systems only work efficiently when motors or engines, drive systems, pumps, and water distribution systems are properly matched. Any change in the system usually requires a change in some or all of the units. For example, switching from surface to sprinkler irrigation often requires modification of the pump and/or drive system, and may require engine changes. As a rule, changes in one component require evaluation of other components in the system to be sure they still match. Mismatches can materially increase pumping costs.

The bowls, line shaft, column, and base of a pumping plant must all be matched for efficient operation. The bowls should have the correct head characteristics or develop the needed pressure for the desired flow. The column and pump head must not offer too much resistance to flow and the line shaft should be the right size so impellers will operate properly.

PUMP DRIVES

There are three general methods of supplying power to a pump: direct drive, v-belt drive and right angle gearheads connected to an engine. Direct drives are mostly used with electric motors but occasionally with engines as shown in Figure 2. An examples of a direct drive for an electric motor is shown in Figure 3. Direct drives fix the speed of the pump to that of the power unit, so the speed ratio is 1:1. Since electric motors often operate at speeds of 1760 or 1770 rpm, or at multiples of that speed such as about 3500 rpm, pump manufacturers often publish pump curves for these pump speeds. This allows direct application of pump curve information. The brake power requirement from the curves provides the power output needed from an electric motor.

V-belt drives or right angle gearheads (see Figures 12 for examples) allow for varying pump speeds relative to the engine or motor speed. These drives are usually categorized by the drive ratio, which is the speed of the power unit relative to the speed of the pump. A ratio of 11:10 means that the power unit operates at a higher speed than the pump. The ratio depends on the diameter of the drive gear or pulley to the diameter of the driven gear or pulley. Pulley size ratios for a v-belt drive are given in the same manner. Motor speed is given as the first number in the ratio. However, with v-belt drives, the ratio refers to the pitch diameter (effective diameter of a loaded pulley). The careful selection of the gear or pulley ratio will give the drive speed required to get the recommended pump speed. In addition, the shaft for v-belt drives should be carefully aligned so angularity in either the horizontal or the vertical direction does not exceed five degrees. Then the power loss through the drive will not exceed five percent.

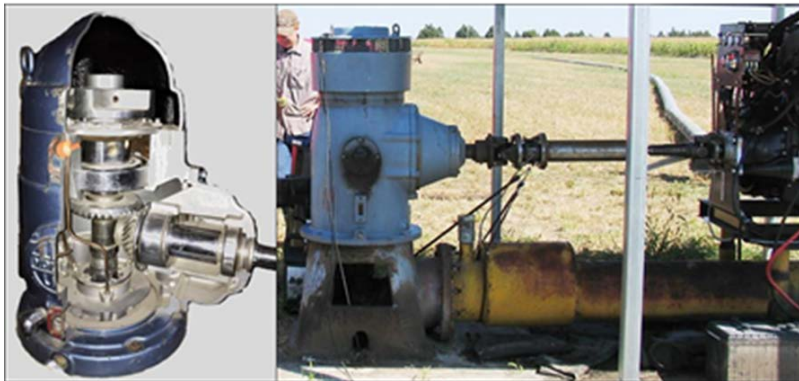
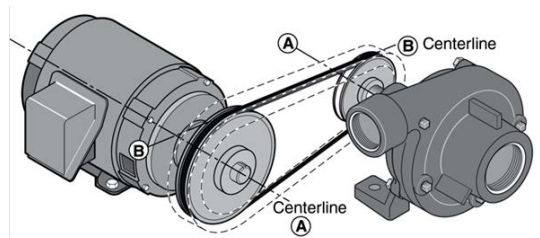


Figure 12. Examples of V-belt and right angle drives for irrigation pumps.

Gearheads and v-belt drives lose some mechanical energy in transferring power from the motor or engine to the pump. The loss of energy is represented by the drive efficiency (E_d). The drive efficiency is the percent of the brake horsepower provided to the pump relative to the power output of the motor or engine. Conversely, the power output from a motor or engine (ehp) must satisfy the brake horsepower requirement and the loss of energy in the drive system:

$$ehp = \frac{bhp}{E_d} \quad (6)$$

The drive efficiency for direct drives is usually taken as 100% while the drive efficiency of right-angle gearhead drives is usually 95%. The drive efficiency should be used as a decimal fractional above (*i.e.*, 0.95 for 95%). The drive efficiency of v-belt systems is more variable depending on the configuration of pulleys and idlers. The efficiency for good systems should be above 90% for well-maintained v-belt applications.

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Power Units

Electric Motors

The nameplate power output of an electric motor should be closely matched to power requirement of the pump when a direct connect drive is used. If other pump drives are used, then the drive loss should be considered. There is no advantage to oversizing an electric motor as the original investment is higher and no operating savings occur, also, standby charges may be greater.

Engines

The power unit on an irrigation pumping plant must supply power to lift water, build pressure, overcome power losses in pumps and drives while operating under the temperature and elevation conditions at the field location.

Engine manufacturers usually publish horsepower curves when engines need to supply full power intermittently or when subjected to a constant load (Figure 13). Since irrigation is a continuous load, the curve that is of interest is the continuous horsepower curve. If only the intermittent horsepower curve is available for the engine, the continuous horsepower can be estimated by multiplying the intermittent horsepower by 0.85.

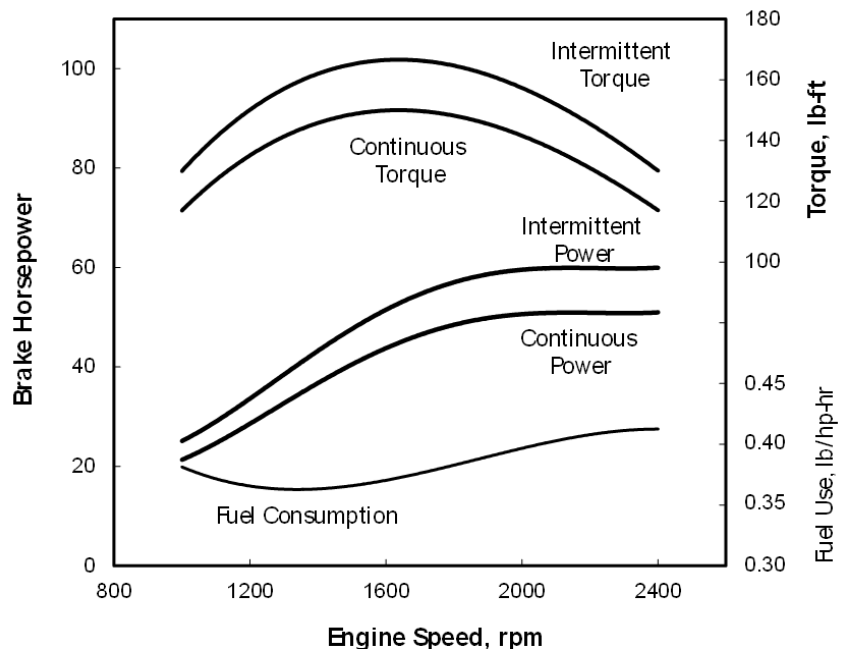


Figure 13. Engine performance curves.

Performance curves also provide the torque for intermittent and continuous loading. The specific fuel consumption rate is also provided on the performance curves. The consumption is the mass of fuel consumed per horsepower per hour of operation. Smaller values for the specific fuel consumption result in more economical engine operation.

The engine performance curve is needed to ensure that the engine will produce enough power to meet the pump and drive system needs at the specific engine and pump speed. If more power is needed, the gear ratio can be changed to operate the engine at a higher speed to gain more power while operating the pump at the designed speed. Matching the pump and the power source is an essential step in developing an efficient pumping plant.

An internal combustion engine may have accessories, such as fans and water-cooling coils that consume some of the power produced by the engine. Thus, less power is available to pump water than the continuous brake horsepower rating of an engine without accessories. Therefore, take care in reading engine curves and specifications. Engine manufacturers may not use the same accessories during engine tests; therefore, it is necessary to determine what accessories were used during the test of a specific engine. Know what accessories were on the engine when it was tested and what affect other accessories not included at the time of the test will have on usable power. The elevation and temperature for the test at the installation location may also be important. Some engines need to be derated for high elevations or hot operating environments. Other engines may not require any adjustment of test results for a wide range of environments. Power requirements for generators used to power a center pivot should also be considered.

An engine dealer should be consulted regarding specific information for available engine models that match your requirements (i.e. needed bhp at desired rpm). Look at the engine curves and determine which models have the best fuel economy at the needed horsepower. These engines will be well suited to your pump. When a fuel curve is not available with the engine's performance curve, then determine which model produces the highest torque at the desired rpm.

Matching Engines to Pumps

An example will illustrate the process of matching engine output to pumping needs. From the farm field make-up and cropping system, and water supply, the decision was made to use a low pressure, electric drive center pivot irrigation system that requires 40-psi pressure at the pump. The needed information to select a pump impeller and bowl assembly include:

Pumping rate	950 gpm
Pumping water level to surface	39.5 ft
Converting psi to feet of head	92.4 ft (40 psi x 2.31)
Total head in feet =	132 ft

- 1) From the manufacturer's curves, an impeller is selected that will deliver 950 gpm, at 66 ft of head, and at 1760 rpm with the highest possible efficiency. The selected pump model delivers 950 gpm at a respectable 81% efficiency and produces 66 ft of head per stage.
- 2) To calculate the number of stages needed simply take total head (132 feet) and divide by the head produced by each stage (66 ft/stage). In this case, the pump will require 2 bowls or stages.
- 3) Water horsepower is calculated as follows:

$$whp = \frac{950 \text{ gpm} \times 132 \text{ ft}}{3960} = 32 \text{ whp}$$

- 4) Determining the size of power plant needed:

Because of the location relative to electric lines, the decision was made to use an internal combustion engine. In this example, a turbocharged diesel engine with a cooling fan, charging alternator and power generator for the pivot will be used.

The elevation and temperature at the well site are:

Elevation above sea level =	1000 ft
Temperature, maximum intake =	100°F.

Elevation and temperature affect naturally aspirated engines, but the performance of turbocharged engines is not usually affected by elevation and temperature until the elevation is greater than 7,000 feet. Some engines can be used without adjustment up to 10,000 feet.

Adjustments are needed for the accessories installed on the engine:

Accessories, cooling fan 5%;	100% - 5% = .95
Charging alternator 1%;	100% - 1% = .99

The engine also needs to be large enough to overcome the friction loss of the gearhead and the losses due to the pump efficiency:

Drive efficiency, for the gearhead is (95%):	0.95
Pump efficiency, (81%)	0.81

The engine must also drive a 10 kVa 3-phase generator to supply power to the drive motors on the center pivot. Generators are generally about 85% efficient, therefore:

$$\frac{10 \text{ kVa}}{0.85} \times \frac{\text{hp}}{0.746 \text{ kVa}} = 15.8 \text{ hp for generator}$$

A 15% reserve is usually added to the pumping requirement to provide for changes due to wear or water level changes—so we need to divide by 0.85.

The continuous horsepower requirement is:

$$bhp = \frac{32 \text{ whp}}{0.95 \times 0.99 \times 0.95 \times 0.81 \times 0.85} + 15.8 = 52 + 15.8 = 68 \text{ hp}$$

- 5) Since the engine needs to run at 1760 rpm to produce 68 horsepower and the pump also needs to run at 1760 rpm, a gearhead with a 1:1 ratio is needed.

To ensure maximum pumping plant efficiency, the pump and engine both must operate at 1760 rpm, but the pivot generator has to run at 2000 rpm to produce the necessary 480 volts. (Remember to operate the system at the proper rpm for the pump and engine, not by the just volt meter on the generator).

For the generator to operate properly, calculate the ratio between the engine speed and required generator speed: $\frac{1760 \text{ rpm}}{2000 \text{ rpm}} = \frac{0.88}{1}$

If the pulley on the engine is 8 inches in diameter, multiply 8 inches by .88 for the size of the alternator pulley. In this case, the alternator pulley should be 7" in diameter.

The matched components of the pumping plant are now complete. A 12" pump with 2 bowls will supply 950 gpm of water to an electric drive center pivot sprinkler system at 40 psi. The system will

be powered continuously by a 68 horsepower turbocharged diesel engine operating at 1760 rpm. A gearhead with a 1:1 gear ratio will run the pump at 1760 rpm. The pulleys to drive the alternator are 8" on the engine and 7" on the alternator. If the diesel engine will not provide 68 hp at 1760 then the gear ratio should be changed to increase the speed of the engine to a point that it will provide 68 hp.

Matching an Electric Motor to an Irrigation Pump

The amount of water and total head and other conditions for the pump and drive are the same as for the diesel engine example. This includes steps 1-3. In step 3 we found 32 whp for 950 gpm and head of 132 feet. Because of a location near an electric transmission line, a 3-phase power line to the pump site is economical. Therefore, an electric motor is selected for the power unit (like system in Figure 79).

To determine the correct size of the electric motor, information about the whp, operating temperature, drive efficiency, and pump efficiency are needed.

- whp output = 32 whp
- Temperature of the well site, maximum 110° F is acceptable for an electric motor
- Drive efficiency, direct drive no loss for the drive-direct coupled = 1.00
- Pump efficiency = 0.81



Figure 14. Electric motor for pumping irrigation water.

Most electric motors have a service factor rating printed on the nameplate. The service factor for large three-phase electric motors is often about 1.15. This allows an overload of 15% above nameplate horsepower provided the motor is used in an environment conducive to adequate cooling (e.g. not dusty or enclosed in a non-ventilated well house). For this example the next motor size smaller than a 39.5 hp is 30 hp. To see if a 30 hp motor could be used multiply by the service factor $30 \times 1.15 = 34.5$ hp. The required horsepower is 39.5 hp is greater than the allowable overload so the next larger motor size would be required. This would be a 40 hp motor. As a word of caution, some motor enclosures have smaller service factors so one must be cautious about overloading motors.

Cavitation

Cavitation results due to the formation of vapor bubbles in a liquid— often because water is subjected to rapid changes in velocity or drops in water pressure below atmospheric pressure. Subjecting vapor bubbles to higher pressure at a downstream location causes the bubbles to implode generating micro jets that produce shock waves. Within a centrifugal pump, the flow area at the eye of the impeller is smaller than either the flow area of the pump suction piping or the flow area through the impeller. The velocity of water entering the impeller increases because of the

smaller flow area which then results in a pressure decrease. The greater the pump flow rate, the greater the pressure drop between the pump suction and the eye of the impeller. If the pressure drop is large enough, or if the temperature is high enough, the pressure drop may cause the water to flash to vapor when the local pressure falls below the saturation vapor pressure of water. Vapor bubbles formed by the pressure drop at the eye of the impellers are swept along the impeller vanes by the flow of the fluid. When the bubbles enter a region where the local pressure is greater than the saturation vapor pressure, the vapor bubbles abruptly collapse. This process of the formation and subsequent collapse of vapor bubbles in a pump is cavitation.

Cavitation in a centrifugal pump significantly effects pump performance. Cavitation degrades pump performance, resulting in a fluctuating flow rate and discharge pressure. Cavitation can also be destructive to pump components. The shock resulting from implosion of the vapor bubbles can create small pits on the leading edge of the impeller. Pits may be microscopic in size, but the cumulative effect over a period of hours or days can damage an impeller. Cavitation can also cause excessive pump vibration, which could damage pump bearings, wearing rings, and seals.

Cavitation can be avoided by maintaining adequate absolute pressure on the suction side of the pump. Water at the surface of a pit, well or channel is at the atmospheric pressure. A pump like shown in Figure 15 is above the water level in the pond; thus, the water in the suction pipe and the suction side of the pump is below atmospheric pressure (i.e., there is a vacuum on the suction side of the pump). The severity of the vacuum depends on the friction loss in the pipe components on the suction side of the pump and the distance that water is lifted from the pond or channel. If the absolute pressure (i.e., the atmospheric pressure minus the vacuum) drops below the pressure where water vapor forms (i.e., the saturation vapor pressure) then cavitation may occur.

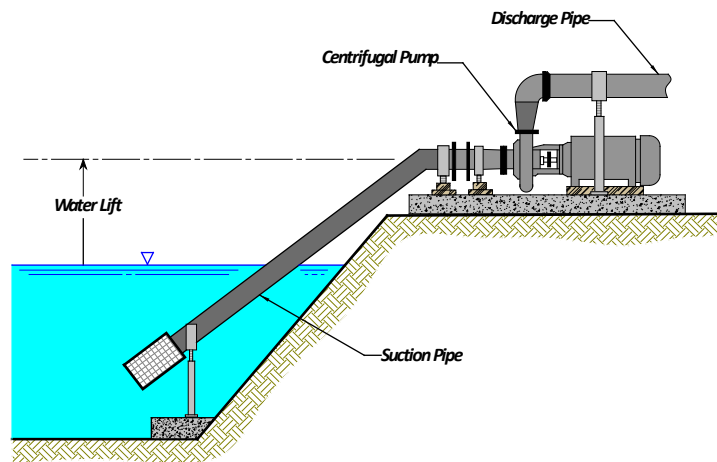


Figure 15. Typical centrifugal pumping plant that lifts water from a canal or reservoir.

Net Positive Suction Head

Pump manufacturers test pumps and provide information on the amount of absolute pressure required to avoid cavitation within their pumps. The pressure head needed to avoid cavitation is called the required net positive suction head (NPSHR). The required NPSH increases with the pump discharge (capacity) see Figure 16. The Berkeley pump shown in Figure 16 requires a NPSH of 9.6 feet at a flow rate of 1000 gallons per minute to avoid cavitation.

Pump Size: 4 x 5 x 13 BH

Model: B4J_BH

Curve No. 9013

Type	CCMD	FM CPLG	FM BELT	SAE	Hydraulic	AC Engine
Model	B4JPBH	B4JRMBH	B4JRMBH	B4JQBH		

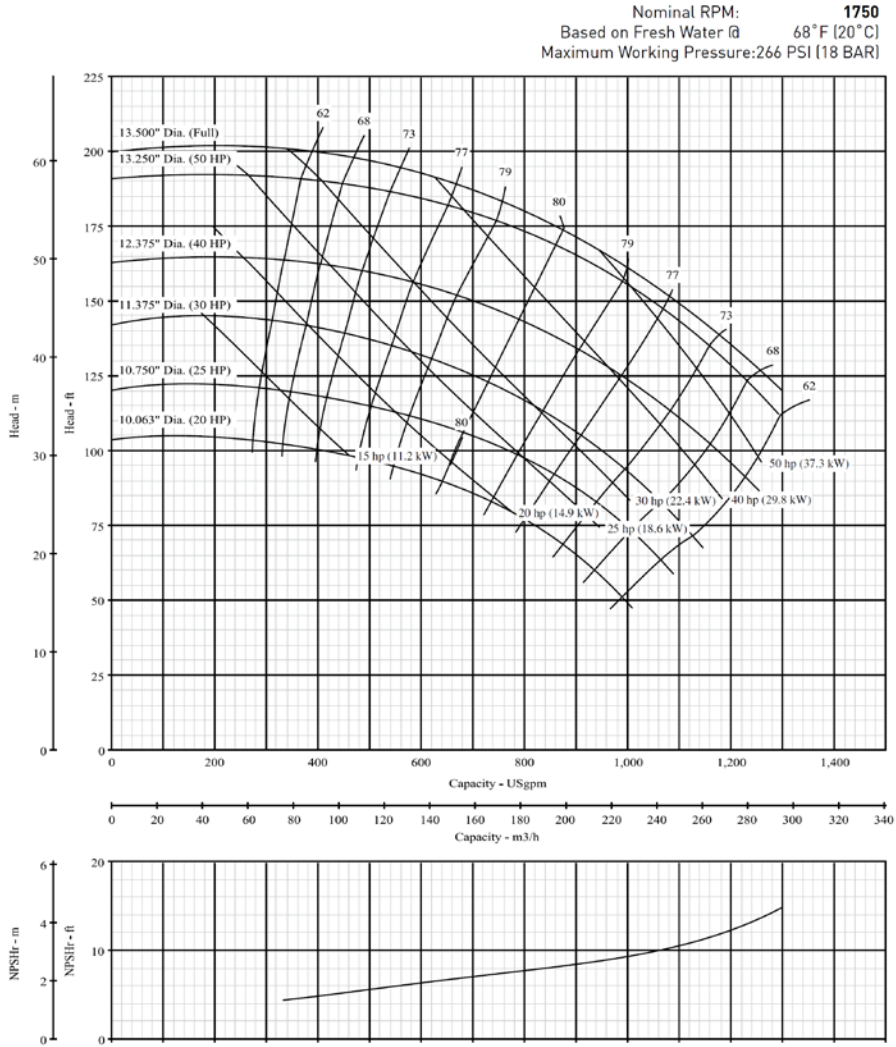


Figure 16. Example pump curve for a centrifugal pump showing net positive suction head (courtesy of Bekeley/Pentair Ltd.).

To avoid cavitation the absolute pressure available at the pump inlet should exceed the NPSH required for the pump. The amount of pressure available is often referred to as the net positive suction head available at the pump inlet (NPSH_A); thus, to avoid cavitation the NPSH_A should be greater than the NPSH_R. The NPSH_A is determined by the:

- Atmospheric pressure at the elevation of pump (P)
- Saturation vapor pressure at the water temperature (e_s)
- Friction loss in plumbing on the suction side of pump (F_L)
- Distance water must be lifted above the level in the pond or canal (L), and
- A safety factor (S_F) of two feet is often used to account for uncertainty.

The $NPSH_A$ is computed as:

$$NPSH_A = P - e_s - F_L - L - S_F \quad (7)$$

Frequently the challenge is to compute the maximum distance that water can be lifted above the open water source without the risk of cavitation. In this case, the required $NPSH_R$ is substituted for the $NPSH_A$ and the above equation is solved for the maximum lift as:

$$\begin{aligned} L_{max} &= P - e_s - S_F - F_L - NPSH_R \\ &= L_{pot} - F_L - NPSH_R \end{aligned} \quad (8)$$

$$L_{pot} = P - e_s - S_F$$

The potential theoretical lift (L_{pot}) depends on the altitude at the pumping site and the temperature of the water. Results in Table 2 list the potential theoretical lift for a range of elevations above sea level and water temperatures. The values in Table 2 include a safety factor of 2 feet. If the pump shown in Figure 16 were installed at a location 2000 feet above sea level and the water temperature was 70 degrees then the potential theoretical lift would be 30.7 feet. If the pump discharge is 1000 gpm then the maximum lift and the friction loss on the suction loss of the pump must be less than 21.1 feet since the $NPSH_R$ is 9.6 feet.

Friction loss in the pipe and fittings on the suction side of the pump must be determined carefully using the friction loss procedures described in the various handbooks.

Table 2. Potential theoretical lift (feet) as a function of elevation and water temperature.

Elevation Above Sea Level, feet	Water Temperature, F							
	45	50	55	60	65	70	75	80
0	33.6	33.5	33.4	33.3	33.2	33.1	32.9	32.7
500	33.0	32.9	32.8	32.7	32.6	32.5	32.3	32.1
1000	32.4	32.3	32.2	32.1	32.0	31.9	31.7	31.5
1500	31.8	31.7	31.6	31.5	31.4	31.3	31.1	31.0
2000	31.2	31.1	31.1	31.0	30.8	30.7	30.6	30.4
2500	30.6	30.6	30.5	30.4	30.3	30.1	30.0	29.8
3000	30.1	30.0	29.9	29.8	29.7	29.6	29.4	29.3
3500	29.5	29.5	29.4	29.3	29.2	29.0	28.9	28.7
4000	29.0	28.9	28.8	28.8	28.6	28.5	28.3	28.2
4500	28.5	28.4	28.3	28.2	28.1	28.0	27.8	27.6
5000	27.9	27.9	27.8	27.7	27.6	27.4	27.3	27.1
5500	27.4	27.4	27.3	27.2	27.1	26.9	26.8	26.6
6000	26.9	26.8	26.8	26.7	26.6	26.4	26.3	26.1
6500	26.4	26.3	26.3	26.2	26.1	25.9	25.8	25.6

Cavitation can also occur for deep well turbine pumps. The required NPSH is shown on pump curves for turbine pumps as in Figure 6. The inlet to the turbine pump must have at least the head shown

on the pump curve. Cavitation can occur when groundwater levels drop over time and the pumping level in the well drops to near the pump setting or when wells begin to “pump air”. Cavitation is more difficult to determine in deep wells than for surface pumps. Historic records of pumping levels will help indicate when problems may be emerging.

Summary

This paper describes how pumps must be matched to operating conditions and power units to provide the desired performance and efficiency. We have recently seen installations where one or more components are not properly matched causing higher than necessary pumping costs. Inspection and maintenance of systems and measurement of flow and pressure will go a long way toward monitoring if the proper conditions are being met.

References

Martin D. L., W. L. Kranz, S. Irmak, Daran R. Rudnick, C. Burr, S. Melvin. 2017. Pumping Plant Performance. Proceedings of the 2017 Central Plains Irrigation Conference. 167-187. Burlington, CO. Feb. 21-22, 2017.