CHAPTER 9
WELL PUMPS

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9.1 PUMPING GEOTHERMAL FLUIDS

9.1.1 Introduction

Pumping is often necessary in order to bring geothermal fluid to the surface. For direct-use applications, there are primarily two types of production well pumps; (a) lineshaft turbine pumps and (b) submersible pumps - the difference being the location of the driver. In a lineshaft pump, the driver, usually a vertical shaft electric motor, is mounted above the wellhead and drives the pump, which may be located as much as 2,000 ft below the ground surface, by means of a lineshaft. In a submersible pump, the driver (a long, small diameter electric motor) is usually located below the pump itself and drives the pump through a relatively short shaft with a seal section to protect the motor from the well fluid.

Lineshaft pumps have two definite limitations: (a) they must be installed in relatively straight wells and (b) they are economically limited to settings of <=2000 ft. For direct heat applications, the economic setting depth limit is probably closer to 800 ft. (Refer to Chapter 6, 6.3.5 Plumbness and Alignment.) A general comparison of lineshaft and submersible pumps appears below in Table 9.1.

Table 9.1 Comparison of Lineshaft and Submersible Pumps

Lineshaft	Submersible
Pump stage efficiencies of 68 to 78%. Lower head/stage and flow/unit diameter. Higher motor efficiency. Little loss in power cable. Mechanical losses in shaft bearings.	Pump stage efficiencies of 68 to 78%. Generally, higher flow/unit diameter. Lower motor efficiency--operates in oil at elevated temperature. Higher losses in power cable. Cable at least partially submerged and attached to hot tubing.
Motor, thrust bearing and seal accessible at surface.	Motor, thrust bearings, seal, and power cable in well--less accessible.
Usually lower speed (1,750 rpm or less). Usually lower wear rate.	Usually higher speeds (3,600 rpm). Usually higher wear rate.
Higher temperature capability, up to 400°F+.	Lower temperature capability but sufficient for most direct heat and some binary power applications, assuming the use of special high-temperature motors.
Shallower settings, 2,000 ft maximum.	Deeper settings. Up to 12,000 ft in oil wells.
Longer installation and pump pull time.	Less installation and pump pull time.
Well must be relatively straight or oversized to accommodate stiff pump and column.	Can be installed in crooked wells up to 4 degrees deviation per 100 ft. Up to 75 degrees off vertical. If it can be cased, it can be pumped.
Impeller position must be adjusted at initial startup.	Impeller position set.
Generally lower purchase price at direct use temperatures and depths	Generally higher purchase price at direct use temperatures and depths.

In some installations, selection of a pump type will be dictated by setting depth, well size, well deviation, or temperature. If not restricted by these, the engineer or developer should select a pump based on lowest life cycle costs, including important factors such as expected life, repair costs, availability of parts, and downtime costs. Power consumption costs and wire-to-water efficiency, although certainly worth evaluating, may not be nearly as important as others factors, such as those above. For most direct heat applications, the lineshaft pump has been the preferred selection.

There are many factors that can affect the relative efficiencies of lineshaft versus submersible pumps: i.e. temperature, power cable length, specific design of impeller and bowl, column length and friction losses. The wire-to-water efficiency in the particular application is the important factor. The bowl efficiency of a pump with extra lateral will be less than for standard lateral (discussed in the subsection on Relative Elongation) and clearances. The bowl efficiency of a submersible will be higher than a lineshaft of similar design because extra lateral is not required in the submersible. Motor efficiency generally favors the lineshaft design.

9.1.2 Lineshaft Turbine Pump

To understand the potential problems and solutions in lineshaft pumping, it is necessary to understand how the pumps are constructed. Figure 9.1 shows a typical lineshaft turbine pump with an enclosed oil-lubricated shaft. Enclosed shaft water lubricated pumps are also manufactured. The discharge head supports the column and shaft enclosing tube which, in turn, supports the multi-stage pump bowls and intake arrangement. The column is usually in 20 ft sections with either screwed or flanged connections. A shaft enclosing tube support "spider" is provided at intervals along the column. The enclosing tube is usually in 5 ft sections with a shaft bearing at each joint, although high speed pumps may have closer spacing. The lineshaft sections are the same length as the corresponding column. The enclosing tube is connected at the top of the bowl assembly to the discharge bowl where lubricating oil outlet ports are located. At the surface, it is connected to the discharge head with a tube tensioning assembly. The enclosing tube is tensioned after installation to help maintain bearing alignment. The enclosing tube provides a waterproof enclosure for the shaft and a path for gravity feed or pressure lubrication.

In an enclosed lineshaft oil lubricated pump, only the shaft bearings are oil lubricated. The pump shaft bearings (in the bowls between each impeller) are water lubricated. The oil is discharged into the well fluid outside the pump through the pump discharge case.

A recent report (Culver, 1991) evaluated the environmental situation with respect to exposure of groundwater to pump lubricating oil. In summary, the report found that standard turbine pump oils are not FDA approved for incidental contact with food (H-1 rated). Despite this, these oils are approved for use in municipal water wells in at least two states. The primary difference between H-1 approved lubricants and non-approved products is in the 1% of the content which constitutes lubricant enhancing additives. So called "white mineral oils," with H-1 approval are available in viscosities suitable for use in turbine pump applications. The report concluded that in the event of regulatory problems with the standard oils, these H-1 approved products should be an acceptable substitute.

Open lineshaft pumps have seen limited success in geothermal applications. Most successful applications have been characterized by very high static water levels or flowing artesian conditions. Because the bearings are lubricated by geothermal hot water, bearings tend to heat and wear faster. Many of the more common bearing materials are subject to corrosion or de-alloying by geothermal water and special bearing materials increase costs. If an open lineshaft design is used, the shaft should be of stainless steel to resist corrosion, again at a higher cost. As a result of the added costs for special materials and, likely shorter service life, the enclosed shaft design is preferred except for very clean, relatively cool (<140ºF) fluid.

The pump impellers are connected to the shaft by a collet or collet and key with locking screws. The shaft and all rotating parts are supported by the thrust bearings of the hollow shaft motor or a separate thrust bearing assembly. There is an impeller adjusting nut at the top in hollow shaft motor assemblies, or a coupling with adjusting nut for solid shaft driver arrangements.

When a vertical turbine pump stops, water flowing back down the column causes the pump to back spin. Because the pump is acting as the driver, there is very little danger of unscrewing shafting, but if the pump is started during back spin, it is likely to break shafting, loosen collets, or damage the motor. This could occur during momentary power failure or when a control signals a pump to start before the column fully drains. Foot valves, non-reversing ratchets, time delay switches, and rotation sensing switches can prevent this. Of these, non-reversing ratchet and time delays are the most common.

There are some advantages in allowing back spin. The free back spin indicates that nothing is dragging or binding and gives an indication of bearing conditions. It also permits the pump to be started with low load, reducing shock loads on shafting and bearings. A non-reversing ratchet also permits the column to drain, but it takes more time because the water flows backward through the bowls and impellers that are not rotating.

Foot valves prevent back spin and keep the column full of water, reducing the entrance of air and associated corrosion and scaling. They are, however, difficult to maintain in good condition because of scaling and corrosion

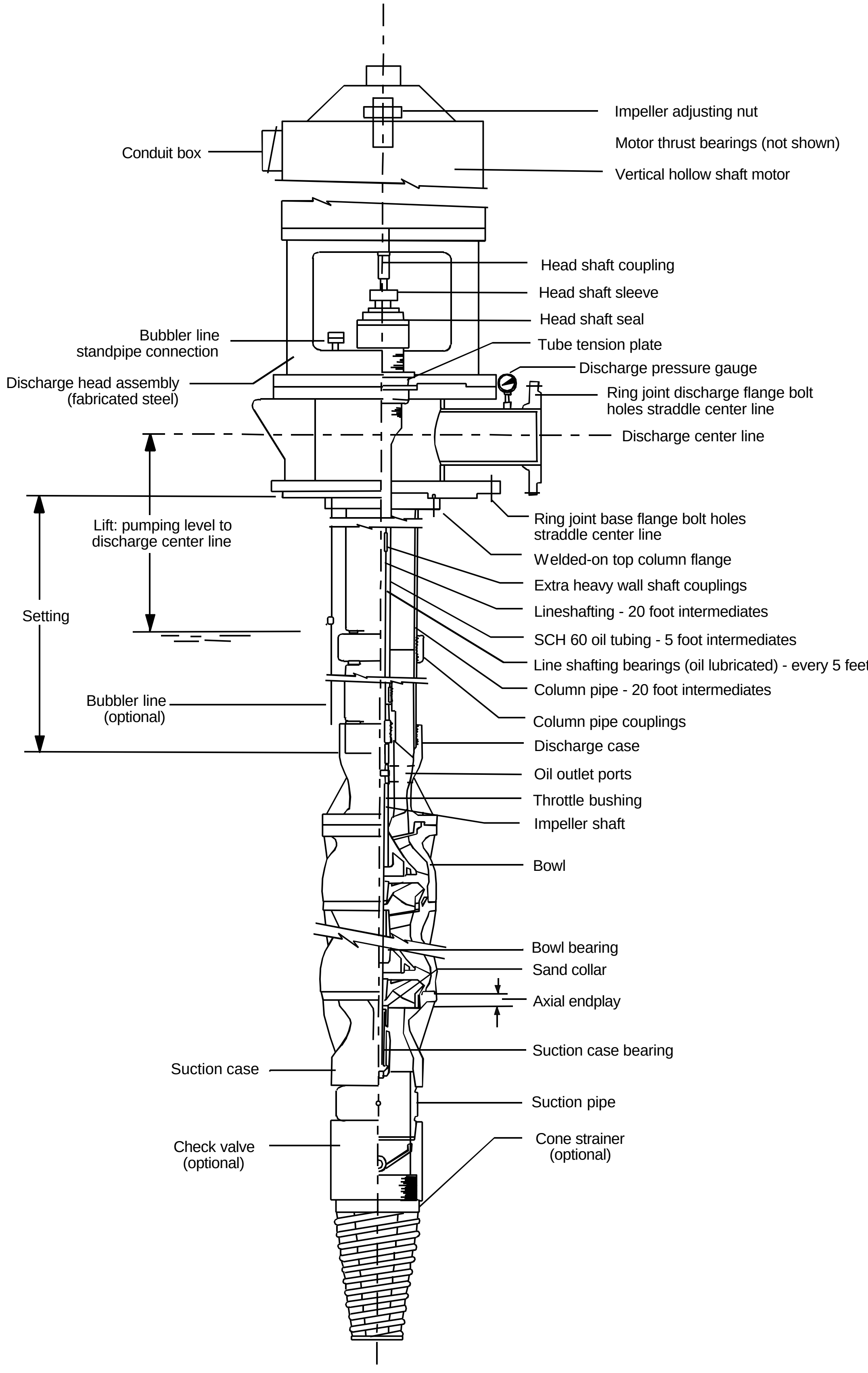


Figure 9.1 Typical lineshaft turbine pump with an enclosed oil-lubricated shaft.

properties of many geothermal fluids. Also, the pump always starts under a high load condition. Foot valves are recommended only for pumping levels <50 ft and when exclusion of air is mandatory.

Relative Elongation and Axial End Play or Lateral

A vertical turbine pump can be thought of as two concentric systems. The outer system consists of the column, impeller housings (bowls) and shaft enclosing tube. The inner system consists of the shaft and impellers. Forces resulting from dead weight, hydraulic thrust and thermal expansion result in different changes in length of these two systems. If not adequately allowed for in the design and operation of the pump, interference can occur resulting in damage to the pump.

Due to the difference in pressure between the entrance and exit of each stage of the pump, a downward force is developed known as hydraulic down thrust. This force can be calculated by multiplying the total head by the thrust factor K, and by the specific gravity of the fluid pumped. The thrust factor depends primarily on the impeller design and diameter and is determined by the pump manufacturer. Note that the deeper the setting and the greater the loads, the greater the resulting shaft stretch. The thrust factor K, is stated in pounds of down thrust per foot of head.

The bowls and column also impose a deadweight load resulting in column stretch. The relative deadweight stretch is compensated for by lifting the impellers free of the bowls before initial startup, as noted above.

The bowls change the direction of water flow passing through them, resulting in dynamic forces and a downward load related to hydraulic thrust. The column and shaft enclosing tube, however, have a much larger cross-sectional area than the shaft, so they stretch less. The difference is relative elongation.

Relative elongation can be calculated if the setting depth, specific gravity, total dynamic head, volume of water supported, lineshaft diameter, column and tube diameter and wall thickness, and impeller thrust coefficient are all known. In cold water pumps, axial end play or lateral, which is the total axial movement of the impellers within the bowl from topping out to bottoming out, must be greater than relative elongation plus allowances for safety.

In geothermal applications, an additional consideration is thermal expansion. Because of their differences in thickness, material and mass, the column, shaft enclosing tube, and shaft will all expand at different rates and reach thermal equilibrium at different times after initial startup. Additionally, the shaft in an enclosed lineshaft pump is somewhat thermally isolated from the water in the column by the space between the shaft and the inside diameter of the tube. Once thermal equilibrium is reached, thermal expansion has no direct affect on relative shaft elongation, but it must be compensated for as it occurs, either by adjusting the impellers or by allowing extra lateral. Obviously, in a system that cycles on and off, it must be allowed for in extra lateral.

Axial end play or lateral is accommodated through the vertical seal between the impeller and the bowl (shown in Figure 9.2). This is a kind of extended skirt on the bottom of the impeller and matching bore in the lower end of each bowl. These areas may have wear rings on the bowls, impeller or both. Standard cold water axial end play typically varies from 3/16 in. in a 4-in. diameter pump to 1-3/8 in. in a 30-in. diameter high head/stage pump. Corresponding maximum axial end play using standard castings is 1/4 to 1-3/4 in. This is obtained by additional machining of the bowls. Thermal expansion alone for a 400-ft static water level, 200ºF well could be 4-3/4 in., which far surpasses the maximum axial end play for standard pumps. This illustrates why standard pumps are sometimes unsuitable for geothermal service, especially in a cycling situation. Failure to consider this has led to premature wear of impellers, bowls and bearings, broken lineshafts, and burned out electric motors. Proper end play and lineshaft sizing requires experience and understanding of relative shaft stretch, and knowledge of the range of operation on the head versus flow curves.

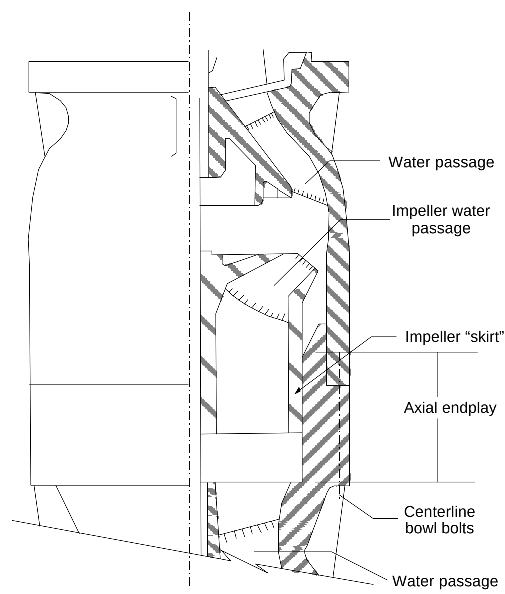


Figure 9.2 Cross-sectional of a pump bowl (Johnston Pump Company).

The dynamic elongation of the shaft is independent of the dynamic elongation of the column pipe. Relative shaft stretch, denoting the position of the impellers in the bowls, is the shaft elongation minus the column elongation. The axial end play must be safely greater than the relative shaft stretch.

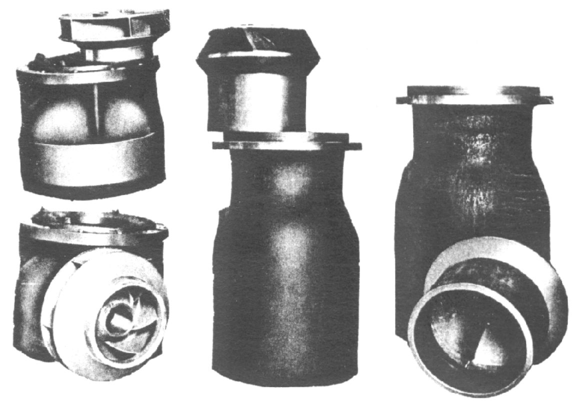


Figure 9.3 Standard and extra lateral bowls and impellers (Johnston Pump Company).

The special design of the geothermal bowls and impellers is shown in the illustration. The taller bowl and impeller are for the geothermal pump in contrast with the shorter bowl and impeller used in a standard industrial pump of comparable size. The geothermal bowls and impellers are designed to accommodate the extra impeller lateral adjustment made necessary by different rates of expansion and relative shaft stretch.

At this point, it may be instructive to illustrate the calculation of relative shaft stretch and thermal expansion. In order to do this, it is necessary to go through a preliminary pump selection. Although this is best left to the pump designer, it is important for the purchaser or engineer to understand the problems so the pump designer can be provided with the correct design parameters.

Assume a space and domestic water heating application requires 700 gal/min (gpm) peak flow. Surface system piping, valves, heat exchangers, and disposal lines require 60 lb/in.^{2} psi) at the wellhead at 700 gpm flow. The static water level is 350 ft. Test pumping indicates drawdown will be 50 ft at 700 gpm and the discharge temperature is 200ºF. For the most part, flows will vary between 400 and 600 gpm during winter and, with domestic hot water storage the pump can be shut off from time to time during spring and fall, and for extended periods during summers. Minimum flows will be 100 gpm, controlled by a throttle valve and, at that rate, wellhead pressure required will be 15 psi. The system is at an elevation of 5000 ft above sea level.

The pump curves for a pump that might satisfy the requirements are shown in Figure 9.4. The curve is somewhat steep, but the efficiency (77 to 83%) is good over the usual operating range of 400 to 600 gpm.

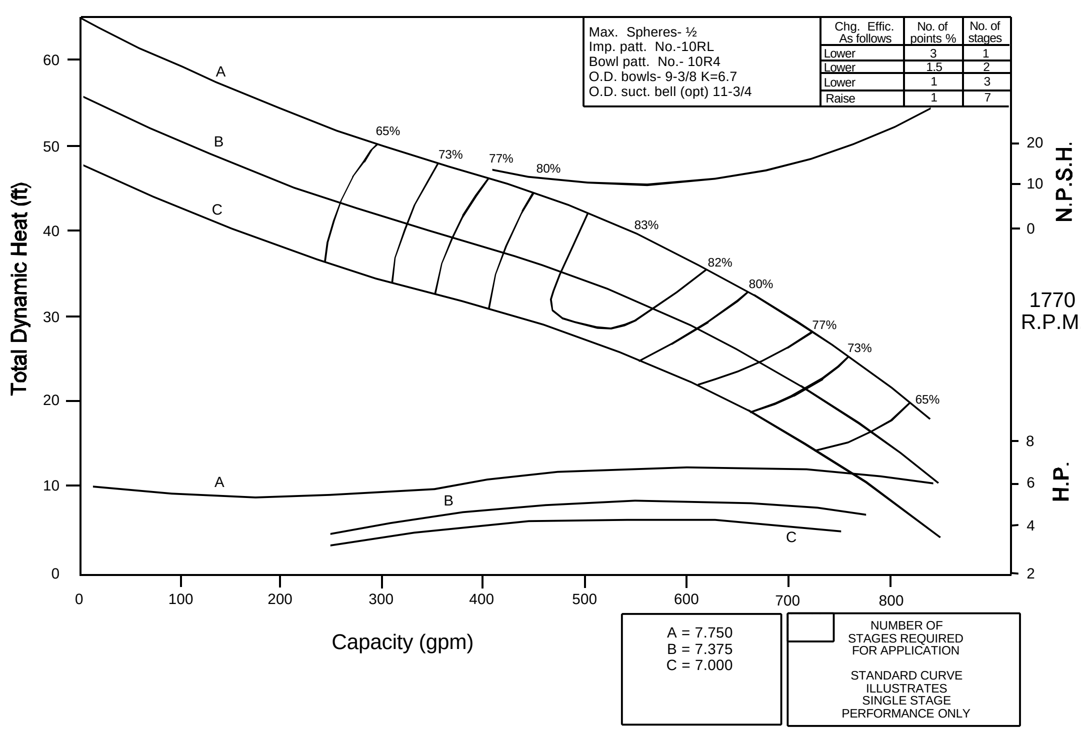


Figure 9.4 Pump curves (Aurora Pump).

The pump curves provide the flow (all of which goes through each stage) and the total dynamic head per stage for three different impeller diameters. Also shown are power requirements per stage, information about the bowls and impeller including the thrust factor K mentioned earlier, efficiency changes for number of bowls and net positive suction head (NPSH). Some manufacturers also provide weight of impellers on the pump curves. In this case, this is found elsewhere. The first stage weighs 38 lb and each additional stage is 19 lb.

Before relative elongation can be calculated, it is necessary to determine the setting depth which, in turn, requires consideration of the NPSH. The net positive suction head available (NPSHA) is the total suction head in feet of liquid absolute at the first stage impeller eye less the vapor pressure of the liquid in feet absolute. Net positive suction head required (NPSHR) is a function of pump design and determines the minimum distance the first stage impeller must be below the pumping level to prevent cavitation. The NPSHA must equal or exceed NPSHR as shown on the pump curve.

Because most pump curves are based on tests at 58°F, the engineer should calculate the NPSH available under operating conditions or, in this case, select a setting to provide the required NPSH considering water temperature, local elevation and intake piping losses. The available NPSH is given by:

$$NPSH = \frac{Ha}{SG} + Hl - Hf - \frac{Hvp}{SG}$$

where

Ha = absolute pressure on surface of liquid, usually atmospheric pressure, in feet of water

Hl = level of liquid above or below the first stage impeller eye in feet (positive if above impeller, negative if below)

Hf = friction loss in intake piping and screen in feet

Hvp = absolute vapor pressure of liquid at pumping temperature in feet of water.

SG = fluid specific gravity

Rearranging to determine the minimum depth below pumping level gives:

$$Hl = NPSHR - \frac{Ha}{SG} + Hf + \frac{Hvp}{SG}$$

The friction head loss is normally very small in geothermal pumps because the tail pipe is short and screens or strainers are often omitted. Reasonable values range from 0.3 to 3 ft of water, but could be more if a long tail pipe and fine screen was installed.

From the pump curves chosen for the example, the NPSH required at 700 gpm is 13.8 ft. Assuming that friction loss in the intake is 1 ft, the minimum submergence below the pumping level is:

$$Hl = 13.8 ft - \frac{28.2}{0.963} ft + 1 + \frac{25.6}{0.963} ft$$

Hl = 12.1 ft.

Additional submergence should be allowed for: (a) long term drawdown (see Chapter 7), (b) non-condensable gases such as carbon dioxide and hydrogen sulfide, which may increase the vapor pressure, and (c) a safety factor that will depend on how much is known about the well (i.e. gases, drawdown).

The total dynamic head required is equal to the lift head plus surface head requirement plus friction losses in the column and surface discharge head assembly. Discharge head assembly losses are normally small, 0.3 to 3 ft, and will be neglected here. Column friction head losses are small and depend on the size of the annulus between the inside of the column and the outside of the oil tube (which, in turn, somewhat depends on the shaft size), and the number and design of the tube support spiders.

The lift head is equal to the static level plus drawdown. For example:

Lift head = (350 ft + 50 ft)	= 400.0 ft
Surface head = 60 psi x 2.31 ft/psi	= <u>139.0</u> ft
Total head required	539.0 ft

From the pump curve, at 700 gpm, using the 7.750 in. diameter impeller, each stage will produce 29 ft head and require 6.5 hp. To satisfy the requirement of 539 ft, require:

$$No.\ stages = \frac{539 ft}{29 ft/stage} = 18.6 stages$$

The horsepower requirement is:

1. Pump horsepower required:

   hp = 19 stages x 6.5 hp/stage x 0.963
   hp = 118.9 hp.

2. Thrust bearing horsepower = (0.0075 hp/100 rpm/1000 lb)(1770 rpm x 6700 lb) = 0.89 hp

3. Column shaft and bearing horsepower required:

   (1.18 hp/100 ft) x 460 ft* = 5.4 hp

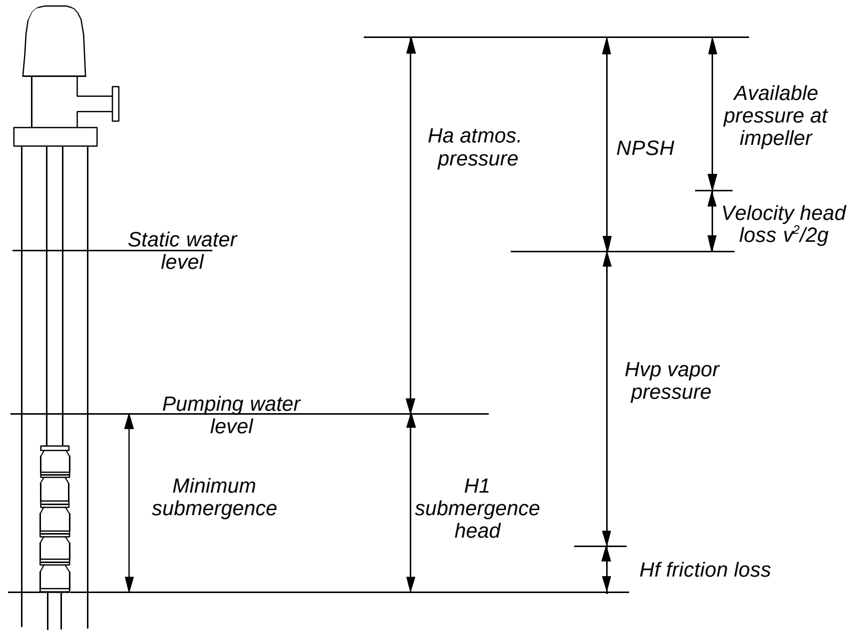


Figure 9.5 Relationship of pressures for vertical turbine pump.

4. Motor brake horsepower required:

   11.8.9 + 0.9 + 5.4 = 125.2 hp.

The lineshaft size is based on combined torsion and axial stresses. Because most pump manufacturers provide shaft sizing tables [or charts based on revolutions per minute (rpm), horsepower required at the pump and axial load] it is only necessary to calculate axial load to enter their tables. These tables or charts are based on AWWA standards.

Axial shaft load or total downthrust is equal to the hydraulic downthrust, plus the weight of the shaft and rotating pump parts. Each impeller pattern has a specific hydraulic thrust factor (usually designated as K), expressed in pounds of thrust per foot of total dynamic head. The K factor varies with specific gravity. The K factor for the impeller selected is 6.7 lb/ft. With some experience, it is possible to estimate the size of shafting, calculate the weight and enter the tables to check the size estimated. Based on the total head and K factor, it can be estimated that a 1-1/2 in. shaft weighing 6.01 lb/ft will be required.

Total downthrust equals:

Head ft x 6.7 lb/ft x SG = 539.4 x 6.7 x 0.963		= 3,478 lb
Wt/ft x 460 ft* = 6.01 x 460		= 2,765 lb
Impellers & rotating parts,	1st stage	= 38 lb
Add'l stages @ 19 lb ea = 18 x 19		= 342 lb
TOTAL		6,623 lb

* 460 ft = 400 ft pumping level + 60 ft allowance for submergence and future drawdown.

Entering the table (Figure 9.6) at 6700 lb and 125 hp, the value is above the allowable limit for 1-3/16 in. shaft and below 1-1/2 in., so the initial estimate was correct.

Using additional tables and charts from the manufacturer's engineering data, it can be determined that:

1. 1-1/2 in. shaft requires 2-1/2 in. oil tube.

2. Column friction loss with 2-1/2 in. tube and 8 in. column = 2 ft/100 ft or 9.2 ft.

3. Shaft elongation under the calculated load = (0.0825 in./100 ft) (460 ft) = 0.380 in.

4. Column elongation under the calculated load = (0.0155 in./100 ft) (460 ft) = 0.071 in.

If tables are not provided, shaft and column elongation can be calculated by:

$$e = \frac{LW}{EA}$$

where

e = elongation in inches (in.)
L = length in inches (in.)
W = total load or down thrust in pounds (lb)
E = modulus of elasticity = 29×10^6 psi
A = gross cross-sectional area of shaft or column in in^2 (including oil tube if applicable).

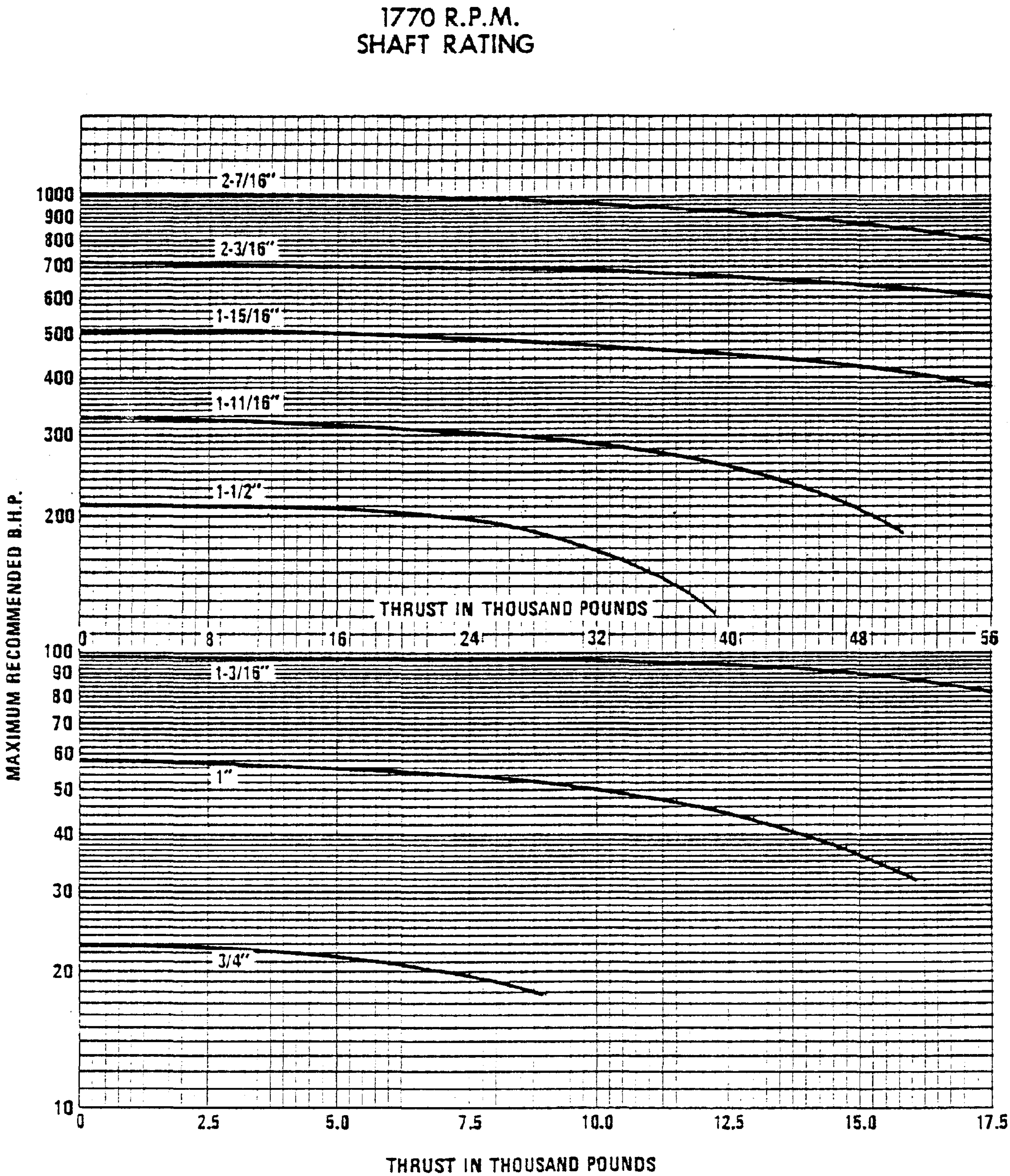


Figure 9.6 Maximum recommended bhp versus thrust (Aurora Pump).

Checking the shaft size (Figure 9.6), we find 1-1/2 in. is large enough to carry the increased load.

Relative shaft elongation is the result of shaft elongation minus column elongation:

Relative shaft elongation
= 0.380 in. - 0.071 in. = 0.309 in.

An additional allowance must be made for machining and assembly tolerances. Pump manufacturers usually recommend 0.010 in./stage, so the total lateral allowance, not including thermal expansion, is:

0.309 in. + (19 stages x 0.010 in.) = 0.499 in.

That portion of the column and oil tube above the static water level will thermally expand much faster than the shaft that is enclosed in, and somewhat insulated by, the oil tube. Thermal expansion of steel is 6.3×10^{-6} in./in. °F. If the average temperature of the air above static water level is 80°F, the Δt is 120 °F.

The thermal expansion is:

$(6.3 \times 10^{-6}$ in./in./°F) (350 ft x 12 in./ft)(120°F) = 3.175 in.

This means our lateral must be increased to:

0.499 in. + 3.175 in. = 3.674 in.

This assumes that all of the column expansion occurs before any of the shaft expansion. This would constitute a worst case situation. In reality, the lateral requirement necessary to accommodate actual net thermal expansion difference would be less than this value.

Figure 9.7 shows the sequence of (1) installing, (2) raising the impellers, (3) initial start, (4) column hot and (5) thermal equilibrium.

To this point, all major items have been considered, except one that was left until last for emphasis. Consider the consequences of closing the throttle valve to reduce the flow to 100 gpm at low load conditions. As stated earlier, the surface head required is 15 psi at 100 gpm.

Referring to the pump curve (Figure 9.4), as the throttle valve closes, the pump moves up its head versus capacity curve until, at 100 gpm, the total head is 59 ft/stage. At that point,total down thrust is:

59 ft/stage x 19 stages x 6.7 lb/ft	= 7,511 lb
shaft wt	= 2,765 lb
rotating parts wt	= 399 lb
TOTAL	10,675 lb

Going back to the tables to check the shaft and column elongation at the increased load we have shaft elongation:

(0.176 in./100 ft)(460 ft) = 0.810 in.

Column elongation:

(0.033 in./100 ft)(460 ft) = 0.152 in.

Relative elongation:

(0.810 in. - 0.152 in.) = 0.658 in.

This means that 0.349 in. (0.658 - 0.309 in.) must be added to the previous lateral to allow for increased relative shaft elongation under minimum load conditions. The total lateral then becomes:

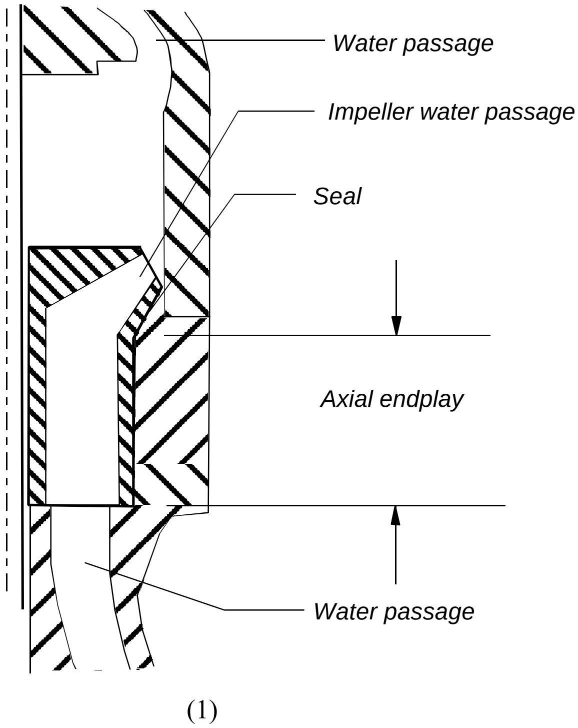
3.674 in. + 0.349 in. = 4.023 in.

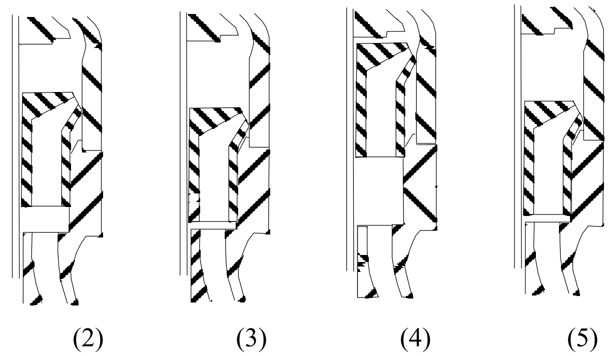
This emphasizes the fact that the system engineer must provide the pump manufacturer with the whole range of operating conditions (preferably a flow versus total dynamic head curve for the system) including well conditions similar to that shown in Figure 9.8. If this is accomplished before well completion, the pump manufacturer is not restricted by casing size and can suggest the most cost effective and efficient pump for the system. Savings in the pump and pumping costs can then be compared to possible increased well costs.

In addition to the affects on the pump design, operation at throttled flow also affects the driver thrust bearing design. In the above example, the thrust load was increased to 161% of the thrust at full flow. If the bearing design does not allow for operation at throttled flow, premature failures will occur.

For the application cited, a larger pump with a flatter curve and higher head per stage would be a more economical selection. For the same flow rate, the larger diameter impeller would have a lower K factor and lower specific speed. This combined with a larger shaft would reduce relative shaft elongation and result in more efficient operation over the required flow range.

There is no real temperature break point for lineshaft pumps. For many applications up to 140°F, standard pumps, perhaps with machining up to maximum axial end play, will operate satisfactorily, particularly where the pump is operated continuously. For intermittent operation, thermal expansion and relative shaft elongation should be carefully checked.





1. Install - impeller resting on bowl.
2. Raise impeller to allow for relative stretch.
3. Operating cold.
4. Operating column hot.
5. Operating column and shaft hot.

Figure 9.7 Pump installation sequence.

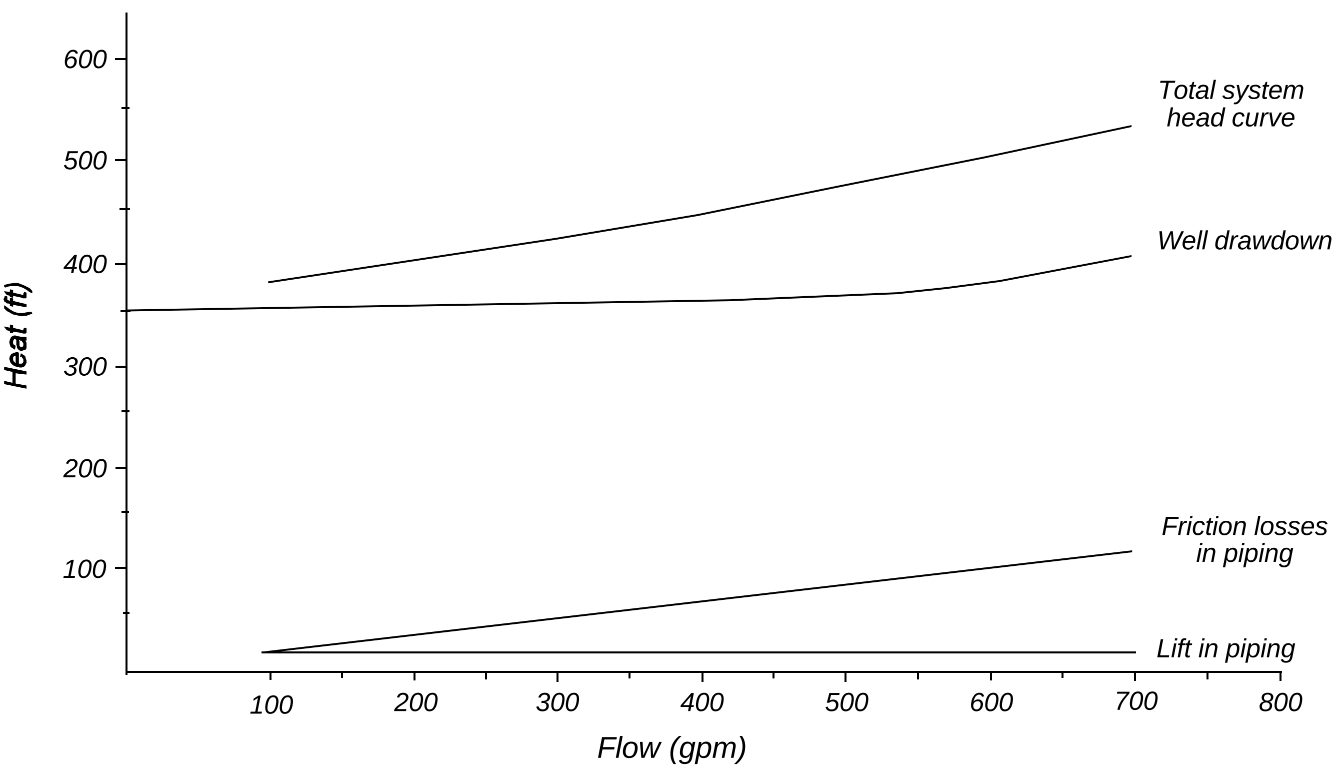


Figure 9.8 Well operating conditions.

Table 9.2 Production Well Pump Materials Successfully Used at Oregon Institute of Technology

Shaft	Stainless steel	AISI 416
Column	Carbon steel	ASTM A53 Grade A
Shaft enclosure	Carbon steel	ASTM A53 Grade A
Bearings	Leaded red bronze	83% Cu, 5% Sn, 7% Pb, 5% Zn
Bowls	Cast iron	ASTM A48 Class 35
Impellers	Leaded red bronze	83% Cu, 5% Sn, 7% Pb, 5% Zn
Keys	Stainless steel	AISI 416
Fasteners	Stainless steel	AISI 303
Collet	Carbon steel	ASTM A108 Grade B113

A regular maintenance schedule is highly recommended. This includes lubrication of motor bearings (and thrust bearing if separate) and pump packing glands at specified intervals. Oil for shaft lubrication is usually gravity flow with a valve and sight glass to check the required number of drops per minute. This should be checked daily. Turbine oil 68 is the normally recommended lubricant.

Pump manufacturers can suggest a reasonable inspection frequency. It is usually more economic to pull a pump, inspect it and repair or replace parts as needed in the off season than to wait until it fails.

Materials

Since most geothermal systems for which accurate materials information is available have been operating for less than 10 years, the "history" of materials selection is quite short. The Oregon Institute of Technology system (operating since 1963) which has extensive experience with pump types and materials has settled on the materials outlined in Table 9.2. Pumps of this construction have operated in the system for as long as 17 years without major overhaul. With three production wells, the system has accumulated over 75 pump years of experience.

Table 9.3 presents the fluid chemistry for the system on which these materials have been successful.

Table 9.3 Fluid Chemistry

Key Species	mg/l
pH	8.6
Chloride, Cl	51
Sulfate, SO_4	330
Bicarbonate, HCO_3	20
Carbonate, CO_3	15
Hydrogen Sulfide, H_2S	1.5
Ammonium ion, NH_4	1.3
Oxygen, O_2	0 - 0.02
TDS	795
Silica, Si	48
Sodium, Na	205
Calcium, Ca	26
Nitrate, NO_3	4.9
Potassium, K	4.3
Fluorine, F	1.5
Iron, Fe	0.3
Temperature	190°F

Lineshaft Pump Pricing Factors

The major items that should be addressed in the preliminary pricing for lineshaft pumps include (a) motor, (b) wellhead equipment (discharge head, base plate, tension nut assembly), (c) column, and (d) bowl assembly.

Motors used for lineshaft pump drives are "L" frame, vertical, hollow shaft, induction motors, in either TEFC or weather protected configurations. As with most new motor installations, the incremental cost of a high efficiency type motor is generally justified. Costs shown in the procedure assume the use of high thrust, high efficiency 1800 rpm, hollow shaft motors.

Wellhead equipment includes the items listed above. The discharge head serves as a fluid discharge connection and pedestal for the driver. The base plate, located just below the discharge head and bolted to it, supports the weight of the bowl assembly and column. For purposes of this estimate, it has been assumed that these items would be of standard cast iron construction.

The tension nut assembly houses the upper shaft bearing and provides a means of adjusting the tension on the shaft enclosing tube for bearing alignment. In addition, it contains the connections for the external lubricating oil supply for the shaft bearings.

The pump column (assuming the use of an enclosed oil-lubricated design) is frequently the single largest cost of the entire pumping system, particularly for settings greater than about 300 ft. This portion of the system includes the column pipe through which the water flows, the pump driving shaft and bearings, and the shaft enclosing tube. Different size columns are available to accommodate various flow rates. Pricing is based on the use of threaded enclosed lineshaft column with 5 ft bearing spacing. This bearing spacing is acceptable for 1,750 rpm operation.

The bowl assembly includes the pump bowls, impellers, shaft, intake/strainer, and discharge sections. A number of construction alternatives are available for pump construction. The option employed for this cost calculation, and the one most frequently specified for geothermal systems is the all iron construction, with the following exceptions: stainless steel fasteners, bowl shaft and impeller keys, and special bowl bearings selected for compatibility with the water that is to be pumped.

A major cost consideration for lineshaft pumps is the allowance for larger than normal lateral in the pump section. A moderate adjustment for increased lateral can be accomplished for standard bowls by machining. This machining involves a 10% cost increase over the standard bowl assembly price. For lateral beyond that described above, extra-lateral bowls must be used. These optional bowls often involve a cost penalty of 50% over standard bowl assemblies.

Estimating Lineshaft Pump Prices

This method is intended to provide a preliminary budget estimate of a lineshaft pump system. For unusual applications or for a more specific cost estimate, manufacturers should be consulted. Costs are based on prices as of 1997.

1. Bowl Assembly

   a. Select pump from table that most closely matches flow requirement.

Table 9.4 Bowl Assembly (Pump) Costs

Flow (gpm)	Head/ Stage	Cost $
60 - 160	12	1600 + 290X
161 - 400	16	2150 + 415X
401 - 1000	20	2850 + 660X
1001 - 1500	22	3150 + 700X

   b. Divide the total dynamic head (TDH) required by head/stage from the table. Round off to the nearest whole number. Substitute the resulting value for X, in the cost formula from the table to determine basic bowl assembly cost.

   c. Adjust for lateral.

   For temperatures in excess of 150ºF and settings in excess of 200 ft, add 50% to final bowl cost.

2. Wellhead (discharge head, base plate, tension nut assembly) and column (column, enclosing, tube, shaft, bearings).

   Select column based on flow requirement. Length required is pump setting depth.

Table 9.5 Well Head and Column Costs

Maximum Flow (gpm)	Size (in.)	
125	4	2800 + 860X
300	5	2800 + 950X
500	6	2800 + 1050X
1300	8	2800 + 1200X

   Divide the pump setting depth by 20 and round off to the next larger whole number. Substitute the result for “X” in the above cost formulas.

3. Motor

   Calculate the required driver horsepower (hp) as:

   $$hp = gpm \cdot TDH \cdot 0.000337$$

where

gpm = gal/min
TDH = total dynamic head

Select the closest larger size motor from Table 9.6.

Table 9.6 Motor Costs

Size (hp)	Cost ($)	Size (hp)	Cost ($)
10	1600	50	3600
15	1325	60	4150
20	2075	75	5750
25	2160	100	6800
30	2500	125	8800
40	3075		

Example Cost Calculation

Required flow is 350 gpm, the fluid is 120°F, there is a 150 ft setting depth, and 320 ft TDH.

a. Bowl assembly cost

- From Table 9.4, 16 ft/stage
- Determine bowl cost:

$320 \div 16$ ft/stage = 20 stages
$2150 + (415 \cdot 20$ stages$)$ = \$10,450

b. Column and wellhead

$150 \div 20 = 7.5$ (use 8)

- Cost from Table 9.5 for 6 in. is:

$2800 + (8 \cdot 1050)$ = \$11,200

c. Motor

- Determine motor hp by:

hp = 350 gpm · 320 TDH · 0.000337
hp = 37.7, say 40 hp

- Select cost from Table 9.6:

Select 40 hp = \$ 3,075

d. Subtotal = \$24,725

e. Total Cost

- Assume a 25% discount from the list price
- Total Cost = (0.75)(\$24,725) = \$18,544

9.1.3 Submersible Pumps

A submersible pump is one in which the driver, or electric motor, is located in the well below the surface of the fluid being pumped and is usually below the pump itself. Submersible pumps, therefore, do not have the problems related to relative shaft elongation that lineshaft pumps do. Submersible pumps can be separated into low temperature or standard pumps and high temperature pumps. The temperature limit is set primarily by the allowable temperature of the motor.

Low-Temperature Submersibles

Almost without exception, standard submersible pump motors are warranted to 90°F or below. The allowable temperature is limited by the motor winding insulation and the heat dissipation available. Many standard submersible pump motors can be operated at 120 to 130°F if proper allowances are made.

There are three basic types of submersible pump motors: wet winding, oil filled, and hermetically sealed.

In the wet winding motor, the motor is filled with water. Water proofing is achieved by special insulation on the stator winding wire, usually plastic, and because the wire and its insulation are bulkier, the motors are larger for a given rating. The motor is carefully filled at the surface to ensure there are no air bubbles and a filter installed in the fill port to ensure that the motor operates in clean water. Some brands are pre-filled and have an expansion diaphragm to allow for expansion and contraction of the filling solution and motor. Rotating seals and a sand slinger at the upper end prevent free circulation of well fluid in and out of the motor and reduce seal and spline wear by abrasive particles. Bearings are water lubricated.

Oil filled motors are prefilled with a dielectric oil. A rotating shaft seal (with sand slinger) is utilized to keep the oil in and water out. Because water has a higher density than oil, the motors have an oil reservoir with expansion bladder at the bottom. Any water that leaks through the seal in time migrates to the bottom of the reservoir. However, if the seal leaks there is probably always a small amount of water mixed with the oil surrounding the windings. Bearings are oil lubricated giving them higher capacity.

Hermetically sealed motors have the winding encased in a welded can, usually stainless steel. The windings may be similar to a surface motor with air inside the can but usually are embedded in a thermo-setting resin to provide better heat dissipation and reduce the possibility of water leaking in. The rest of the motor is similar to the wet type described above with the bearings water lubricated.

All small submersible motors have a thrust bearing at the lower end to carry pump downthrust and a small thrust bearing at the upper end to carry the momentary upthrust during pump startup. Some larger motors intended primarily for deep settings have a separate seal section providing for sealing and expansion. The seal section is located between the motor and the pump and contains the main thrust bearings.

All submersible pump motors depend on the flow of fluid past the motor for cooling. This is usually in the range of 0.25 to 0.5 ft/s velocity. If the motor is installed in a large casing or well bore, or if the production zone is above the motor, this velocity may not be realized. In that case, the pump must be fitted with a flow inducer sleeve (Figure 9.9) or other method of increasing fluid velocity to provide cooling. The flow inducer sleeve also provides a means of increasing the ambient operating temperature. If the velocity past the motor is increased to 5 to 6 ft/s, the allowable ambient water temperature may be increased from the standard 85 to 90°F, to 120°F. The actual amount of allowable increase depends on the motor type and insulation. The manufacturer should also be consulted. Because the flow inducer restricts the inlet flow path, it is

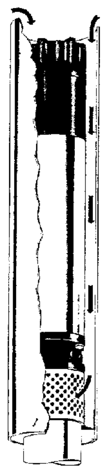


Figure 9.9 Flow inducer sleeve (Franklin Electric).

A flow inducer sleeve is a tube over the motor, closed off above the pump intake and extended to the bottom of the motor or lower. The sleeve material is corrosion resistant metal or heavy plastic.

necessary to check to be sure sufficient net positive suction head (NPSH) for the pump is maintained.

Another method of increasing allowable ambient fluid temperature is to decrease the service factor, requiring an increase in motor horsepower. For instance, if the ambient temperature requires a decrease in service factor from standard 1.2 down to 0.6, a pump that normally would require a 3 hp motor now requires a 5 hp motor. The 5 hp motor would be the same diameter but longer.

By using a combination of high flow velocity and reduced service factor, some motor manufacturers will authorize, but not necessarily warrant, operation at 130°F. So far as is known at this writing, there are no standard submersible motor manufacturer's that will authorize operation at ambient temperatures above 130°F, although several pump installers have reported 3- to 5-year life at 135 to 140°F. These installations were neither authorized nor warranted.

At least one motor manufacturer builds an "oil stripper" motor in 2 through 15 hp in a 4 in. frame. These motors are rated for continuous duty in 160°F ambient temperatures. The motor is basically an oil-filled type with an improved seal arrangement, with increased oil reservoir capacity and larger diaphragm to accommodate greater thermal expansion. Because the bearings are oil lubricated, the allowable thrust load is increased from 900 lb in a similar water lube motor to 1500 lb.

Many small submersible motor manufacturers install thermostatic protection in the motor windings rather than, or in addition to, having the overload protection at the surface. This effectively blocks use of the motor in higher ambient temperatures even if a flow inducer is installed and the service factor reduced.

At the temperatures and depths standard submersibles are likely to be employed, it is unlikely that electrical cable voltage drop would be a significant factor. At temperatures above 100°F, the manufacturer should be consulted concerning the allowable current at the pumped fluid temperature.

Most pump and motor manufacturers conform to NEMA standards for shafts and bolt patterns. It is possible, therefore, to match one manufacturer's pump to another manufacturer's motor. Many off the shelf pumping systems utilize one manufacturer's pump and another's motor.

When connecting one manufacturer's motor to another's pump, the pump should have sufficient lateral to allow for shaft deflection. As the load increases, the shaft will shorten elastically. Pump manufacturers usually state the thrust factor and motor manufacturers can provide elastic deformation data or, if shaft size and lengths are known, it can be calculated.

High-Temperature Submersibles

High-temperature submersible pumps were developed for deep settings in oil fields. They are almost universally rated in barrels per day (bpd) rather than gallons per minute (gpm = bpd/34.3). Pump curves (Figure 9.10) are arranged differently but show the same information as lineshaft pump curves. For elevated temperatures in both geothermal and oil fields, better elastomers for seals, higher temperature insulating materials for cable, and improved oils and bearings have been developed. Satisfactory operation has been attained in oil wells up to 290°F. Figure 9.11 shows a submersible installation. The gas separator shown is primarily used in oil field production. The function of the separator is to remove free gas from the fluid before it enters the pump where it would expand in the low-pressure suction area, possibly cause cavitation, and prevent proper pump operation.

The pump section of a submersible is similar to a lineshaft in that it is a multi-stage centrifugal. Pump rpm is usually 3,475, which is higher than most lineshafts. Impellers are usually of the balanced or floating type to offset hydraulic thrust, because space for thrust bearings is limited. Figure 9.12 shows a typical pump section with its balancing ring and balancing holes.

The seal section between the pump and motor provides for equalization of well fluid and internal motor pressure, allows for expansion and contraction of dielectric motor oil, provides a seal between the well fluid and motor oil and houses the thrust bearings. Separation of the well fluid and motor oil is accomplished by two or more mechanical shaft seals, elastomer expansion chamber and backup labyrinth.

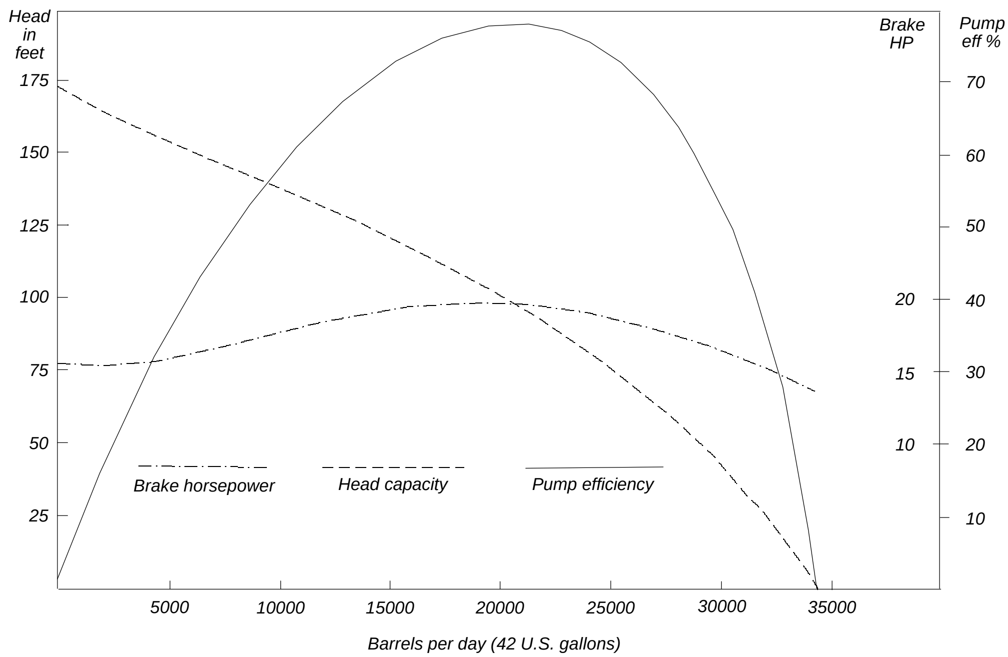


Figure 9.10 One-stage performance curve (Centrilift Hughes).

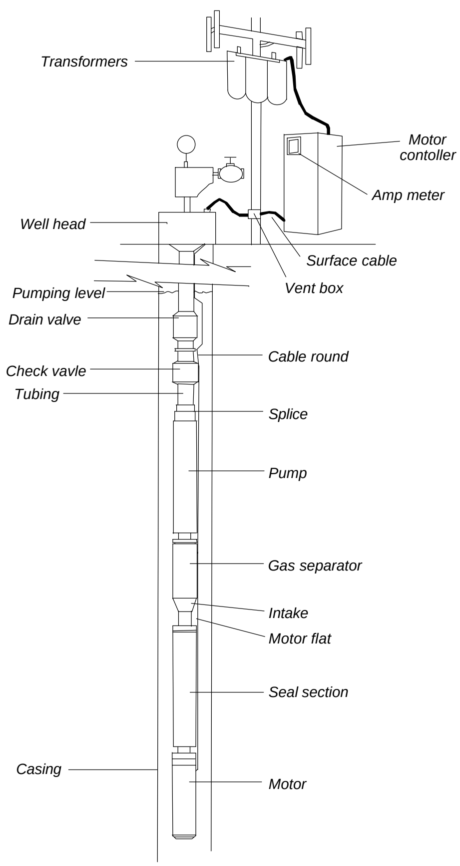


Figure 9.11 Submersible pump installation (Centrilift Hughes).

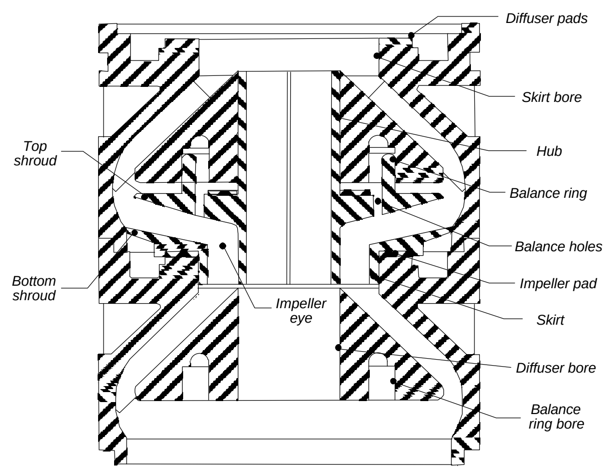


Figure 9.12 Submersible pump typical stage non-memclature (Centrilift Hughes).

Impellers are designed for balancing at peak efficiency. Operation at higher than design capacity results in upthrust, and lower than design capacity results in downthrust. Bearings are usually of the multiple tilting pad type; there are two, one for upthrust and one for downthrust.

Motors used in high-temperature submersibles are oil-filled, two-pole, three-phase, squirrel cage, induction type. Design voltages range from 230 to 5000 V.

In deep setting applications, motors are run at high voltages in order to reduce current flow. Voltages often are not the common values used in aboveground motors. In deep settings, there can be significant voltage drops in the downhole power cable. Submersibles, therefore, can require special above ground equipment, transformers and controllers, which are supplied by the manufacturers to match existing conditions.

Motors are built in 3-1/2 in. to 7-1/2 in. outside diameters to fit inside standard American Petroleum Institute (API) casing sizes. Rotors are generally 12 to 24 in. long and hp is increased by adding rotors. Single-motor lengths may reach 30 ft producing 400 hp and tandem motors 90 ft producing 750 hp. Motors have bearings designed to carry the rotor loads but not any additional pump loads.

Motor cooling is critical, and at least 1 ft/s flow past the motor is recommended. Flow inducer sleeves can increase flow velocity as described above for standard submersibles, and centralizers are often used to ensure even flows completely around the motors. Centralizers are required in deviated wells.

The cable providing electrical connection between the pump and surface is an important part of a submersible system. The cable is connected to the motor at a waterproof pothead that is usually a plug in type. Waterproof integrity is essential, and special EPDM elastomers are used for sealing. Pothead leaks were a continuing source of trouble in early submersibles for geothermal use, but the new designs have somewhat alleviated the problems. A flat motor lead extension cable is usually installed from the pothead to above the pumps. A cable guard is installed over the cable along the seal and pump sections to prevent mechanical damage during installation and removal. Either round or flat cable is spliced above the pump and run to the surface through the wellhead and to a junction box. Cable is available for several operating temperatures. Up to 180 to 200°F polypropylene insulation with nitrile jacket is used. At temperatures above 200°F, insulation and jacket are EPDM. Various configurations with or without tape and braid and lead sheathing are available for temperatures up to 450°F. Most cable has an interlocking armor of galvanized steel or monel. Galvanized steel will have a very short life in most geothermal fluids. Monel metals generally have longer expected life depending on the alloy and amount of hydrogen sulfide (H_2S) present.

Because all the submersible equipment is in the well, there is no maintenance that can be performed except scheduled pulling and inspection. Large submersibles may be equipped with recording ammeters that can help determine causes of failures and give an indication of pump and well performance. Pump wear, for instance, is indicated by decreasing motor output and current draw. Excessive current in one or more legs might indicate motor or cable problems. If recording ammeters are installed, they should be checked regularly and the records analyzed.

Submersible Pump Pricing Factors

The major components to be included in the estimate for a submersible pump include: (a) pump, (b) motor, (c) protector (seal), (d) cable, (e) wellhead junction box, (f) switchboard (controller), and (g) transformer.

Performance data for pumps are generally published in flow units of barrels per day (bpd). These units can be converted to gallons per minute (gpm) by dividing by 34.3. In the smaller sizes, pumps are priced by the housing, rather than by the stage. Each housing is capable of accommodating a specific number of stages. Each housing has a specific sales price regardless of the actual stage requirement.

Table 9.7 Stage/Housing Arrangements

Housing	Stages
10	10
20	22
30	34
40	47
50	59

In larger pumps, pricing is on a per stage basis. In addition to the pump itself, bolt-on intake and discharge sections are also sometimes required. The intake section includes a screen and components for maintaining proper flow profile into the pump section and coupling to the seal section. The discharge sections allow the pump to be coupled to the tubing. Some pumps include integral intake and discharge.

Downhole motors are available in a number of different voltages. In general, sizes of 60 hp and below are available in standard 440 V. Above this size, higher operating voltage is generally employed. The voltage value may range from 750 to 2200 V. As a result, a surface transformer would be required to adjust the available voltage to the required motor voltage. For unusually deep pump settings, the surface voltage must also be adjusted to allow for losses in the downhole power cable.

The protector section is generally available in several different materials and configurations depending upon pressure and temperature in the pumping zone. The prices included in the estimating method are for the basic unit.

The type of downhole cable required is a function of current flow, temperature and space availability. Two basic profiles are available: flat and round. Of these round cable is generally less expensive for a given conductor size. Standard conductors are available in Numbers 1, 2, 4, and 6 AWG with insulating ratings of 3 and 5 kV.

The cable is three conductor and generally includes a layer of individual conductor insulation, a second layer of conductor jacket covering all three conductors, and an outer armor covering the jacket. Standard materials for the insulation, jacket, and armor are polypropylene, nitrile and galvanized steel. EPDM is employed in the insulation and jacket of high-temperature cable.

The preliminary pricing scheme outlined is based on the use of Numbers 2 or 4 flat cable with a 205ºF temperature limit (standard materials as above). In most cases, this cable will be of adequate size to achieve an acceptable voltage drop (20 to 30 V/1000 ft).

The function of the wellhead is to support the pump and column, and provide a means of sealing the top of the well casing.

A high voltage junction box is required on all downhole pump installations. The junction box provides for a connection between the line from the motor control center and the line from the well pump motor. In addition, it provides a vent for any gases that might migrate up the cable from the well.

The switchboard, or controller, contains a starter, controls, and monitoring equipment for the downhole motor. Manufacturers offer a wide range of sophistication in terms of monitoring and protection devices, both electromechanical and electronic. The pricing method assumes the use of a basic switchboard with electromechanical protection devices.

Estimating Submersible Pump System Prices

This method is intended to provide a preliminary budget cost estimate for a submersible pump system. For unusual applications or for a more specific price quote, manufacturers should be consulted. Costs are based on prices as of 1997.

1. Pump

   a. From Table 9.8, select the maximum flow rate close to the required flow.

   b. Divide the TDH required by the head per stage from Column 2.

   c. Divide the number of stages required by the stage divider (Column 3) and subtract one. Round off to the nearest whole number.

   d. Substitute the value found in step c for X in the pump cost formula.

Table 9.8 Submersible Pump Cost Formula

Maximum Flow (gpm)	Head/ Stage (ft)	Stage Divider	Cost ($)
108	40	10	3150 + (760X)
248	29	4	3550 + (350X)
478	33	4	4800 + (1000X)
568	55	3	6800 + (1500X)
947	75	1	9400 + (850X)

2. Seal Section

   add $3,250

3. Motor

   $$hp = gpm \cdot TDH \cdot 0.000337$$

   where

   gpm = gallons per min
   TDH = total dynamic head in ft

   Select closest larger value from Table 9.9.

Table 9.9 Submersible Motor Costs (High-Temperature Waters)

hp	Cost ($)
20	8,000
30	9,300
40	10,300
50	12,300
60	14,300
70	15,700
80	17,600

4. Cable

   Cable size will depend upon motor hp, depth, and operating voltage. Size 4 wire should suffice to 40 hp @ 460 V.

Table 9.10 Cable Costs

Size	Rating V)	Type	Cost ($/lf)
4	3 K	Flat galvanized armor	4.80
2	3 K	Flat galvanized armor	7.30

5. Wellhead

   Wellhead costs are shown in Table 9.11.

Table 9.11 Wellhead Costs

Casing OD (in.)	Cost ($)
5-1/2	800
7	800
8-5/8	1200
9-5/8	1750
10-3/4	1800

6. Switchboard and High Voltage Junction Box

   Prices given in Table 9.12 are for a basic board with electro-mechanical protection.

Table 9.12 Switchboard and Junction Box Costs

hp	Cost ($)
25	3100
50	3500
100	4250

Example Cost Calculation

Required flow is 240 gpm at 450 ft TDH, the pump is set at 350 ft, and 7 in. casing.

a. Pump Cost

- Select pump from Table 9.8 and determine number of stages:

  (450 ft)/(29 ft/stage) = 15.5 stages

- For stage divider of 4, X = (15.5/4) -1 = 2.875

  (Use X = 3)

- From pump cost formula, Table 9.10:

  3550 + (850 x 3) = $ 6,100

b. Seal Section

- Add = $ 3,250

c. Motor Cost

- Determine hp by

  hp = 240 gpm · 450 TDH · 0.000337
  hp = 36.4 (use 40 hp)

- Select motor cost from Table 9.9:

  Select 50 hp = $10,300

d. Cable Cost

- Select cable cost from Table 9.10:

  (350 ft + 100 ft)($4.80/ft) = $ 2,160

e. Wellhead Cost

- Select wellhead cost from Table 9.11 for 7 in. casing = $ 800

f. Switchboard Cost

- Select switchboard and junction box from Table 9.12 for 40 hp motor = $ 3,500

g. Subtotal $26,110

h. Total Cost (assumes 25% discount from kit price)

- Total Cost = (0.75)($26,110) = $19,580

9.2 VARIABLE-SPEED DRIVES FOR GEOTHERMAL APPLICATION

9.2.1 Introduction

Energy costs associated with the operation of production well pumps constitute a large expense for many geothermal systems. In direct use systems, particularly those serving predominantly space heating loads, there is a wide variation in flow requirements. As a result, an efficient means of controlling flow should be an integral part of these systems.

Because most systems utilize centrifugal lineshaft-driven or submersible well pumps, there are three methods available for controlling flow:

1. Throttling pump output
2. Varying the speed of the pump
3. Intermittent pump operation with storage tank.

Throttling the output of any fluid handling device is simply dissipating energy through the addition of friction. This is an inherently inefficient approach to flow control.

Intermittent pump operation can impose serious shock loads in the pumping system, particularly at bearings and impeller connections. This has, in several projects, led to pump failures. Storage tanks can serve as a point of entrance for oxygen, thus aggravating corrosion problems. The results of these combined effects has been unreasonably high maintenance costs.

Use of variable speed drives can significantly increase pump life. A slow speed pump will outlive a faster pump with identical installations and pump construction. The wear rate is proportional to somewhere between the square and cube of the speed ratio; as a result, a pump rotating twice as fast will wear at four to eight times the rate (Frost, 1988).

A review of the response of a basic pumping system suggests that pump speed control is a much more energy efficient approach to controlling flow rate. In a closed piping loop, flow varies directly with pump speed, pressure drop with the square of the pump speed and horsepower requirement with the cube of the pump speed.

One must realize that the above relationships are based upon a situation in which the pump head is composed entirely of friction head. In a geothermal system, much of the pump head may be composed of static head. Static head is, of course, independent of flow. As a result, for a pump operating against a 100% static head, the system response is one in which flow is directly related to speed, pressure drop is in-dependent of speed and horsepower varies directly with speed.

The savings to be achieved through speed control of a centrifugal fluid handling device under a 100% static head situation are then significantly less than the savings achieved in a 100% friction head situation over the same speed range. In addition, there is a limit imposed by a large static head upon the minimum pump speed. This minimum speed is a function of the ability of the pump to develop sufficient head to move the water out of the well.

Geothermal systems will fall somewhere between these two extremes (100% static head and 100% friction head) depending upon static level, drawdown and surface head requirements.

If the control strategy is based upon a constant wellhead pressure, the system very nearly approaches the 100% static head situation. In general, large surface pressure requirements (which vary with flow) relative to static head requirements tend to make speed controls more cost effective.

Most geothermal applications involve the use of a squirrel cage induction motor. The results in two basic approaches to pump speed control:

1. Motor oriented control
   a. Multi-speed motor
   b. Variable frequency drive (AC).

2. Shaft oriented control through the use of a fluid coupling.

The choice among the above techniques should consider: capital cost, duty cycle, hp, speed/torque relationship, efficiency, and maintenance requirement.

9.2.2 Motor Oriented Control

Multi-Speed Motors

Multi-speed motors in the integral hp (greater than one horsepower) size are available in constant hp, constant torque, and variable torque configurations. For most geothermal applications with large static head, the constant torque type would be the most appropriate. There are three basic varieties of multi-speed motors: one winding two speed, two winding two speed, and two winding four speed.

The one winding two-speed motor offers a 2:1 speed reduction ratio, such as 1750/850 rpm. The two winding two-speed motor offers a somewhat greater choice of speeds in that it is not limited to the 2:1 ratio. The two winding four speed motor has the greatest range of speed adjustment. This is typified by a common configuration such as 1,750 rpm/1,150 rpm/850 rpm/575 rpm (Andreas, 1982). The constant torque arrangement would be most applicable to geothermal applications. Under this configur-ation, hp varies directly as the speed. Obviously, the multi- speed motor offers a stepped adjustment of output. For systems with infinitely varying requirements (such as space heating), throttling would be required to adjust flow in between the available speeds. This, of course, would decrease the potential savings available from this type of speed control. Moreover, the sudden changes in speed that would result from multi-speed operation could impose additional mechanical constraints on pump shaft and bearing design.

The multi-speed motor approach offers relatively low cost, and simplicity, in comparison to other drives. The only costs incurred are those of incremental motor cost (multi-speed over single-speed) and speed switching equipment. Motor efficiency is less than equivalently sized single-speed motors.

Cost for multi-speed motors is a function of the number of speeds, number of windings, and speed/torque configuration and as a result, it is not possible to characterize costs in general terms.

Adjustable Frequency Control

To understand an AC variable frequency control, one must first review some basics of induction motor operation. The speed of an induction motor is a function of the number of poles and the frequency of the applied power supply according to the following relationship:

$$N_s = 120 f/p$$

where

N_s = synchronous speed
120 = constant
f = frequency in Hz (hertz)
p = number of poles in the rotor.

In reality, there is a slight "slip" in the actual motor speed compared to synchronous speed. This slip amounts to 2 to 6% at full load, depending upon motor design.

As suggested by the above relationship, motor speed can be adjusted by controlling the frequency of the power supply. This frequency adjustment must be carried out at a constant relationship to voltage or at constant volts per Hz. This is necessary due to the method by which a motor produces torque. Torque is produced by a magnetic flux that is directly proportional to voltage applied, and inversely proportional to the frequency (Baumiester, 1978). Therefore, as motor speed is reduced by frequency adjustment, voltage must also be reduced to avoid unreasonable motor losses and magnetic saturation (Andreas, 1982).

Frequency and voltage adjustment are accomplished by drives referred to as inverters. Several designs are in use for the inverter including:

- Pulse width modulation (PWM),
- Variable-voltage inverter (VVI), and
- Current source inverter (CSI).

Of these, the PWM type is the most common in the range of motor sizes commonly found on direct-use well pumps. CSI and VVI inverters are found most often on large (>100) horsepower applications.

The term inverter is somewhat misleading in that an electronic adjustable speed drive unit actually contains more than an invertor. All systems include at least three basic components. Th rectifier serves to convert the incoming AC power to DC. The DC is then fed to the inverter section for conversion back to variable frequency AC. The inverter accomplishes the conversion task using either transistors or thyristors. These electronic devices switch the DC input (from the rectifier) on and off to provide a controllable AC output (EPRI, 1987). The third major component in the drive unit, the controls, regulates the activity of the inverter switching such that the motor operates at the speed required.

The PWM drive employs a voltage source inverter that produces positive and negative voltage pulses of different widths (EPRI, 1987). As indicated in Figure 9.13, motor current is a system using a PWM drive in close approximation to that in a non-adjustable speed drive application.

The heating effect of the frequency controller upon the motor can be compounded by operation at constant torque. Under a constant torque load, as speed is reduced, motor current remains fairly constant because of the load. As a result, motor losses and heating are also constant. However, the self-cooling produced by the motor fan is reduced by the lower speed. This raises motor winding temperatures. At full-load, temperatures reach 212°F at 75% speed, 320°F at 35% and 428°F at 20% speed (Andreas, 1982). Although most manufacturers state that these controllers can be used with standard induction motors, it would be wise to employ motors with high-temperature insulation characteristics or those specifically designed for variable-speed applications.

Efficiency of adjustable frequency drives is generally quoted by the manufacturers at 95%. This value applies only to the base frequency, which is usually 60 Hz (Andreas, 1982). Figure 9.14 shows efficiency at other operating points. The plot is based only on the efficiency of the frequency controller. Sinusoidal motor performance at the same torque must be considered to arrive at drive efficiency. For example, at 50% speed and 75% torque, a value of approximately 88% is read from the diagram. In order to obtain drive efficiency, this value would be multiplied by the motor efficiency at that point (75% torque). Assuming a 100 hp motor, this value might be 90%. As a result, drive efficiency would be 0.90 x 0.88 = 0.792. Using the same pump figures from the example in the wound rotor section (50 hydraulic horsepower requirement, 60% pump efficiency), a total electrical input of 50/(0.792 x 0.60) = 105 hp, or 78.5 kW results.

When selecting a variable frequency drive, users should be careful to specify a high power factor. All types show relatively high (95%) power factor at full load; however, some drop off considerably as motor load diminishes. Generally, drives employing a diode bridge- type converter will maintain a consistently high power

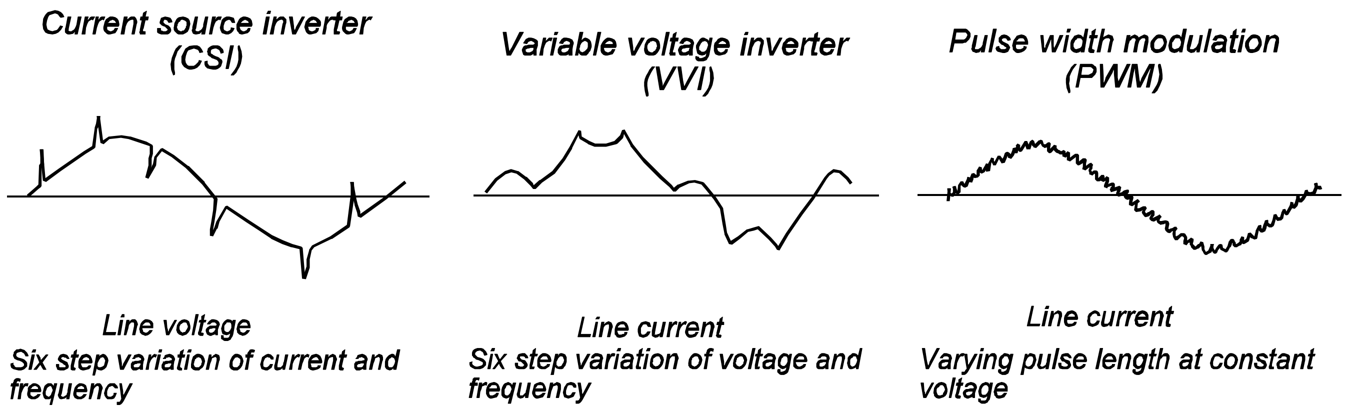


Figure 9.13 Generalized waveforms for adjustable frequency drives (Bell and Gossett, 1988).

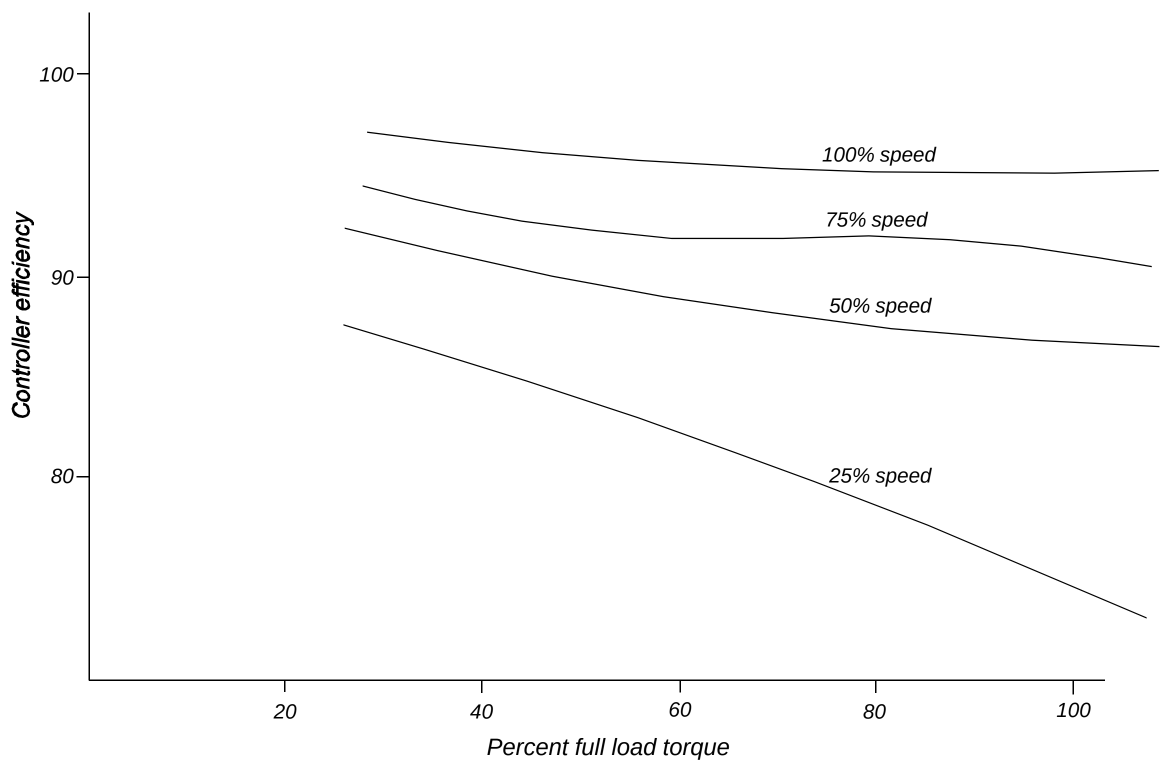


Figure 9.14 Adjustable frequency drive performance (Andreas, 1982).

factor regardless of load or speed. The diode bridge connection is most commonly found in pulse width modulation (PWM) drives. Recent advances have brought the cost of constant (high) power factor drives down to the point where they are competitive with variable power factor drives.

Maintenance requirements are low for the variable frequency drive. The controller itself is constructed primarily of solid state components that require virtually no attention. However, the controller units sometimes have maximum and minimum temperature limits. As a result, they would have to be housed in some type of conditioned wellhead structure for protection. Both oil and air cooled units are available. For many geothermal applications, the oil-cooled unit (which is suitable for outdoor installation) would be useful in eliminating exposure of electrical components to hydrogen sulfide (H_2S) environments. The variable frequency type of speed control is the only one that would permit system operation in the event of controller failure because electrical supply to the motor could be routed around the inverter. List prices are shown in Table 9.13 according to hp requirements. These figures are for the inverter only and additional costs would be incurred for control interface, starter, manual or automatic bypass equipment.

Table 9.13 Adjustable Frequency Inverter Costs (1997)

hp	List Price ($)
10	2,400
20	4,600
50	7,200
75	8,200
100	10,700

Some specifications related issues which warrant consideration include (McFatter, 1994):

- Converter type - Most equipment includes a 6-pulse bridge as standard. Some manufacturers offer a 12-pulse design which can substantially reduce line harmonics. The economics of the higher priced inverter must be measured against the possible requirements for external components (reactors or transformers) which may be required to otherwise control harmonics.

- Carrier frequency - In high frequency applications where the motor is located a sufficient distance from the drive unit, excessive voltage peaks can occur in the motorwindings. This problem is more serious in applications using 575 V. Most direct-use pump motors operate at 460 V. Assuming the motors conform to NOMA Standard MG1 Section 31.40.42 (designed for 1600 V), this should not present a problem.

- Line loss ride through - The duration of the line loss event is the primary distinction between manufacturer. Values vary from a few cycles to a few seconds.

- Auto restart - Capability to restart after a power outage.

- Critical speeds - Allow the programming of the drive to avoid operation at speeds subject to resonance in the driven equipment.

- Overspeed capability - Provision for operation of the motor in excess of 60 Hz.

Fluid Coupling

The fluid coupling falls into a class of fluid drives known as hydrokinetic. It consists of input and output members that are mechanically independent. The impeller (input member) accelerates the oil, which then enters the runner (output member) where it is decelerated and the kinetic energy in the fluid is converted into shaft power (Andreas, 1982). As shown in Figure 9.15, the level of oil in the impeller/runner area is varied by a scoop tube to adjust speed output. Lost energy or inefficiency is dissipated as heat. This heat is rejected to an external heat exchanger. Speed reduction capabilities are 4:1 with a constant torque load and 5:1 with variable torque loads. Sizes range from 5 to several thousand hp.

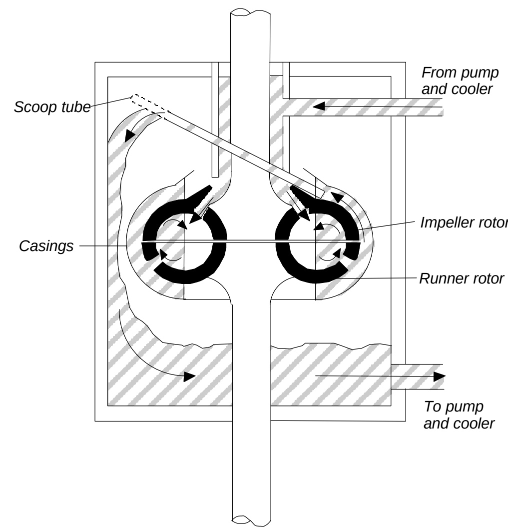


Figure 9.15 Fluid coupling.

Fluid coupling is a slip loss type device, in which efficiency is primarily a function of the “slip” or the difference in the input and output shaft speeds. Losses amounting to approximately 1.5% of unit rating are experienced because of parasitic losses for oil cooling and circulating (Andreas,1982). Figure 9.16 illustrates typical unit efficiency.

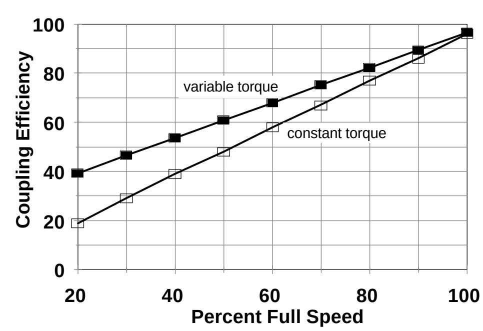


Figure 9.16 Fluid coupling performance.

Although there appears to be little difference between the drive efficiency in constant and variable torque applications, losses under constant torque conditions are substantial. As mentioned earlier in this chapter, direct-use projects often result in conditions which closely approach the constant torque application. This occurs when the well is characterized by higher specific capacity (low drawdown) considerations or constant surface pressure. This results in a constant head (regardless of the flow) and from the drive standpoint, a constant torque.

According to Andreas (1982), the torque required by the load is equal to the torque input from the electric motor. Accordingly, the horsepower at the output of the fluid coupling is:

Output horsepower = Te x rpm_o /5252
Input horsepower = Te x rpm_i /5252

where:

Te = load torque
rpm_o = output rpm
rpm_i = input rpm

This can be rearranged to calculate the loss (converted to heat) in the fluid coupling:

Coupling loss = $Te(rpm_i - rpm_o)/5252$

In a constant torque application, by definition, the torque remains unchanged. This means that input horsepower remains the same while output horsepower is reduced. This results in larger losses and poor efficiency. Figure 9.17 illustrates the difference in fluid coupling performances in constant and variable torque mode through the use of an example. For a pump with a design (full speed) flow of 450 gpm @ 406 ft head, the input and output power consumption is illustrated for both a 100% friction head (variable torque) and a 100% static head (constant torque) application. It is apparent that under constant torque operation, the input power to the drive remains constant regardless of the output horsepower requirement. This results in increasing loss as the speed is reduced. In the variable torque example, the difference between input and output power is much smaller. Fluid coupling use in constant torque applications must, therefore, be justified by other than energy considerations.

Maintenance of the fluid coupling itself is relatively low, with most service requirements related to the external heat exchanger and circulating pump. Costs for fluid couplings are shown in Table 9.14.

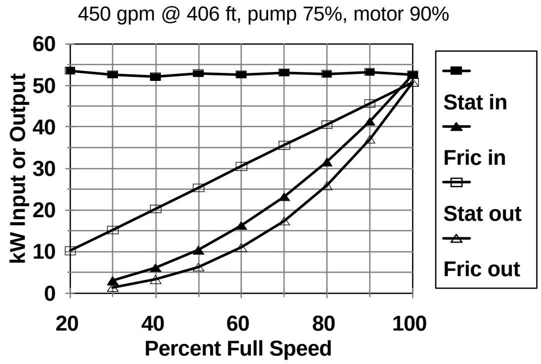


Figure 9.17 Fluid coupling example.

Table 9.14 Fluid Coupling Costs

hp	List Price ($)
10	10,600
25	11,500
50	12,300
125	14,100
200	15,900

System Example

The following table outlines key characteristics for a hypothetical direct-use production well pump. The head imposed by the system is characteristically dominated by static head (lift). Figure 9.18 presents a comparison of the pump performance curve, the system head requirement curve and the head available from a pump whose speed is controlled to meet the system flow requirement. This speed control could be accomplished by either a variable frequency drive or a fluid coupling.

Static water level	150 ft
Specific capacity	8 gpm/ft
Peak production rate	450 gpm
Wellhead pressure	45 psi peak, 15 psi minimum
Duty cycle	As shown in Table 9.15
Motor efficiency	Assumed at 0.92

It is apparent that the system head requirement would necessitate throttling of the pump, even if variable speed was used. As a result, accurate control of the system is a choice of simple throttling or a combination of throttling and speed control.

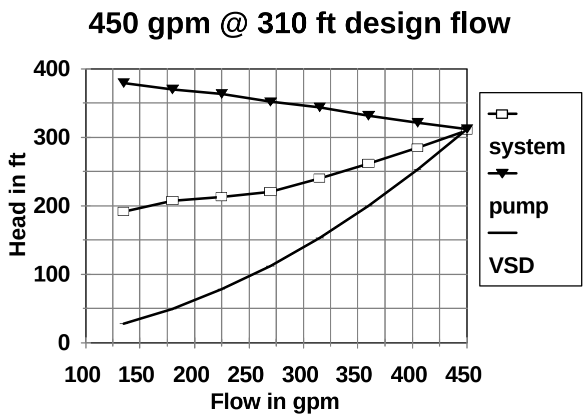


Figure 9.18 Pump example - head.

Table 9.15 Example Well Duty Cycle

Flow (gpm)	Hours/ Year
450	25
405	78
360	232
315	476
270	675
225	1,132
180	1,270
135	486

Figure 9.19 presents a comparison of the power input requirement to the pump motor over the range of system flow rates. Integrating these power requirements over the opera-ting hours (Table 9. 15) permits the calculation of the annual energy consumption for the pump as shown in Table 9.16.

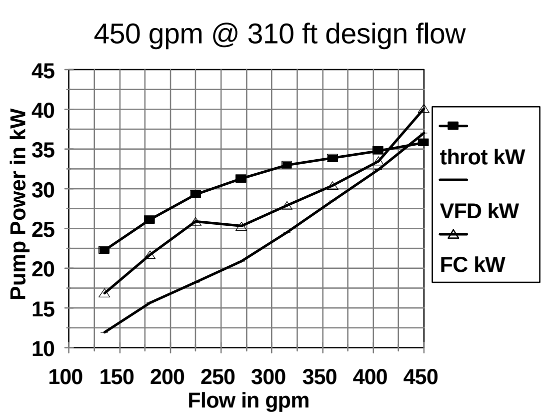


Figure 9.19 Pump example - power requirement.

Table 9.16 Results of Energy Use Calculations

Flow Control Method	Energy Use in kWh/Year
Throttle	125,700
Fluid coupling	106,300
Adjustable frequency	82,200

Conclusion

Among the various drive technologies available, the choice is a function of a host of project specific parameters. The information presented here, along with pump and well in-formation from your project, should permit an accurate analy-sis to be carried out. The results of this analysis can then be employed in the decision process. Table 9.17 summarizes the various characteristics of the speed control techniques outlined herein.

9.2.3 Adjustable-Speed Drive Application

Calculating Speed Requirements

In order to use the variable speed drive, it is necessary to understand the affects of speed variation on the pump characteristics. Within limits, centrifugal pump affinity laws state that:

1. Flow rate is proportional pump rpm
2. Pump head is proportional (pump rpm)^2
3. Brake hp is proportional (pump rpm)^3.

Using these proportionalities, a pump curve for any speed within the useful limits can be generated. For instance, from the pump curve used earlier (Figure 9.4), at 700 gpm the head is 29 ft/stage, efficiency is 78% and brake hp 6.5 hp/stage when the pump is run at 1770 rpm. To find the characteristics at a reduced speed of 1622 rpm [which is equivalent to 55 Hz operation with a variable frequency drive (VFD)], the following relationships apply:

$$Flow = \frac{1622\,rpm}{1770\,rpm} 700\,gpm = 641.5\,gpm$$

$$Head = \left(\frac{1622\,rpm}{1770\,rpm}\right)^2 (29\,ft/stage)$$

= 24.4 ft/stage

$$hp = \left(\frac{1622\,rpm}{1770\,rpm}\right)^3 (6.5\,hp/stage)$$

= 5.0 hp/stage

Table 9.17 Summary of Speed Control Techniques

Method	Efficiency	Capital Cost	Maint. Required	Over Speed Capacity	Effect on Motor Life[e]	Turn Down	Auto Control	Size Range
Adjustable[a] Frequency (AC)	High	Moderate	Low	Y	Lowers	Inf.	Y	Franctional to several hundred
Fluid[b] Coupling	Moderate	High	Moderate	N	None	4:1	Y	5 - 10,000 hp
Multi-speed[c] Motors	Moderate	Low	Low	N	None	2:1	Y	Fractional to several hundred
Throttling[d]	Very low	Low	Low	N	None	No limit	Y	No limit

a. Allows motor operation in failure mode. Should use high-temperature rise motors. Minimum ambient temperature 50°F.
b. Poor efficiency at low output speeds.
c. Poor efficienty at low output speeds.
d. Stopped output speed in 2 or 4 increments, must throttle in between, possible problems with shaft and bearings.
e. Refers to older motors--depends on application.

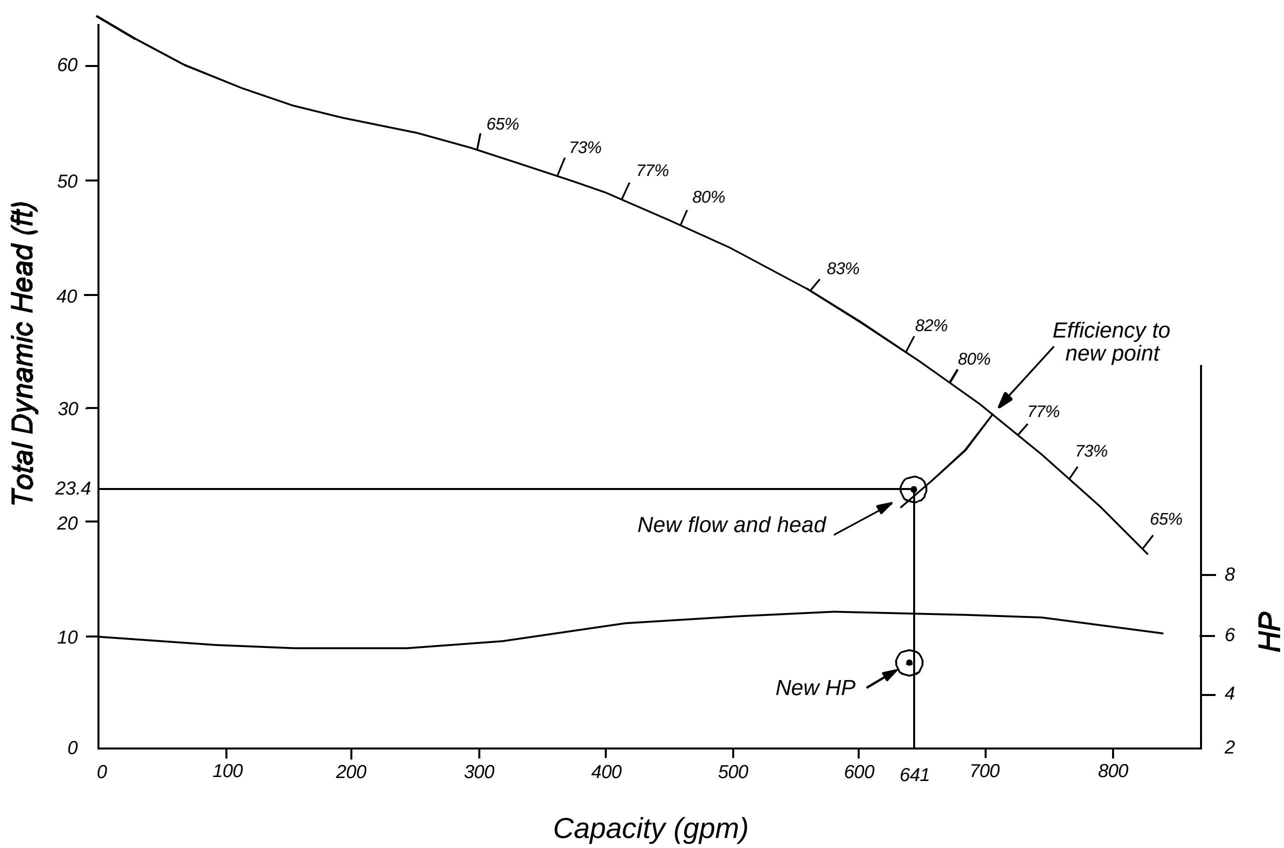


Figure 9.20 Single-stage pump curve (see Figure 9.4)(Aurora Pump).

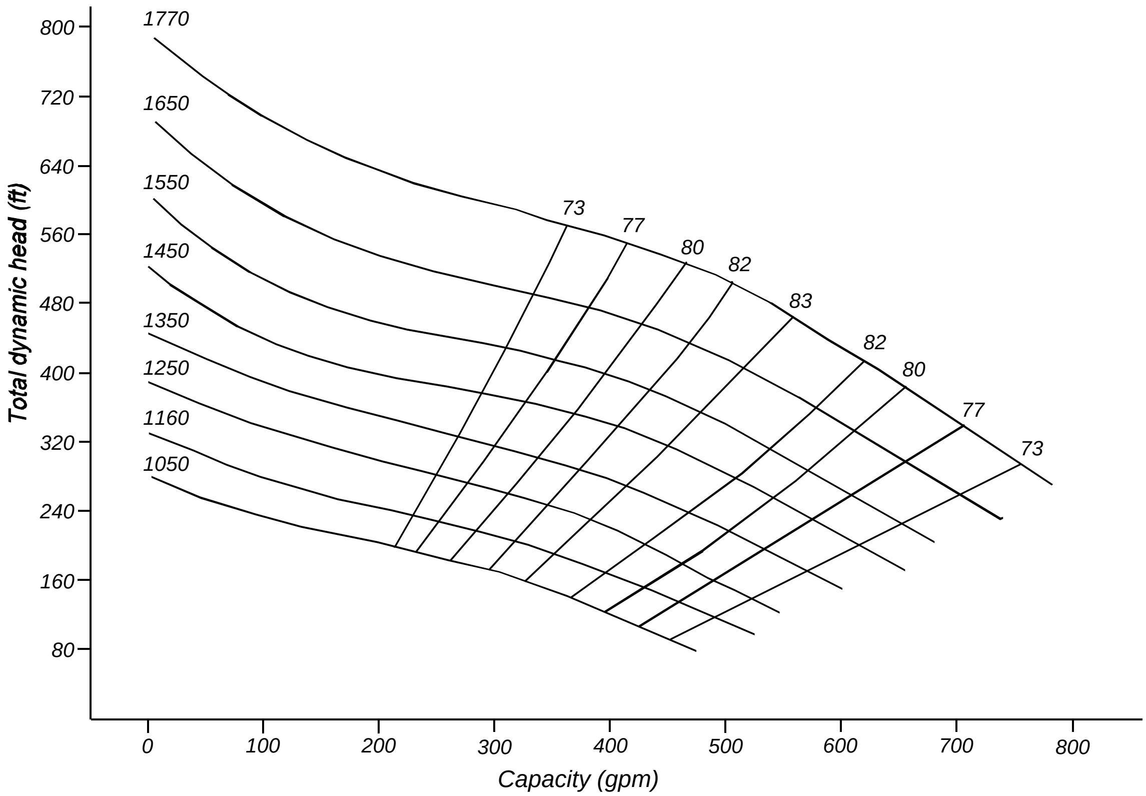


Figure 9.21 Pump characteristics for 12 stages.

Efficiency at the old head versus rate point is moved to the new head versus rate point. The new points are shown in Figure 9.20.

Using the proportionalities, a new pump curve can be generated for 1622 rpm or a family of curves for any desired rpm. This is what was done to generate the family of curves shown in Figure 9.21. Note that the curves are for a 12-stage pump. The total dynamic head is shown, rather than the head per stage.

Temperature Increase Considerations

To illustrate another point, refer to the previous example where there is a 350 ft static level and 50 ft drawdown of 0.963 specific gravity fluid. The lift head is 400 ft. The pump chosen has 19 stages, producing 700 gpm at 29 ft per stage, or 551 ft TDH, and requires 6.5 x 0.963= 6.26 hp per stage.

The zero flow speed, or the speed where that pump would just maintain fluid at the surface, can be found by:

$$New\,Speed = \left(\frac{New\,Head}{Old\,Head}\right)^{1/2} x\,Old\,Speed$$

$$New\,Speed = \left(\frac{400ft}{551ft}\right)^{1/2} x\,1770rpm = 1508rpm.$$

At that point, the pump would be dissipating brake horsepower speed (bhp) shown as:

$$New\,bhp = \left(\frac{new\,rpm}{old\,rpm}\right)^{3} x\,old\,bhp$$

New bhp - 73.56 bhp

(73.56 bhp) (42.42 Btu/min hp) = 3120 Btu/min.

Because the pump is already operating in hot fluid and there is no flow to carry away heat, the potential exists for excessive temperature to occur. This can lead to drastically premature pump failure.

The temperature rise at no flow can be approximated by:

Tr = (Cm x Ps)/(W x C)

where

Tr = temperature rise, ºF/min
Cm = 42.42 Btu/min/hp
Ps = shut off hp
W = weight of liquid in pump, lb
C = specific heat, Btu/lb ºF.

The volume of water in a pump stage is not usually given in pump specifications, but with an estimate of 3 gal/stage we will have 476 lb in the 19-stage pump. At no flow, the water level will soon reach static level of 350 ft, giving 160 ft of submergence (allowing 50 ft for the length of the bowl assembly), and an absolute pressure of 79.8 psi (at 0.963 specific gravity and 5000 ft elevation). This corresponds to a saturated steam temperature of 310ºF and signifies that water in the pump could start to boil in approximately 17 min.

Of course, some heat will be transferred to the water but, in any case, the pump will cavitate, clearances will change, collets, keys and set screws loosen and bearings overheat in a relatively short time.

Because variable speed drives are often used in space heating applications where there may be no flow requirements for substantial periods of time, it is important to maintain some minimum flow for cooling. The pump manufacturer should be consulted to determine what that minimum flow is. With constant speed pumps, an orifice or bypass control valve can be used with discharge back to the well or through the system. With variable speed drives, the lower speed limit must be set to attain the minimum flow. That flow can be diverted back to the well through an orifice or valve, or through the system. In any case, it will result in increased pumping costs, but should pay for itself in decreased maintenance. Discharge back to the well is usually preferred, because the resource is conserved.

9.2.4 Recirculation

Recirculation is a somewhat poorly understood phenomenon that occurs at decreasing flow rate in all centrifugal pumps. A portion of the flow reverses at the impeller suction or discharge, resulting in high shear velocities and vortices (Figure 9.22). Within the vortices, low pressure is created. When the pressure reaches the vapor pressure, bubbles are formed that subsequently collapse, resulting in cavitation. In addition to cavitation damage to the impellers, vanes, shrouds, and stationary vanes of the casing, the formation and collapse of bubbles cause high frequency vibrations and thrust reversals, resulting in bearing damage and shaft failures.

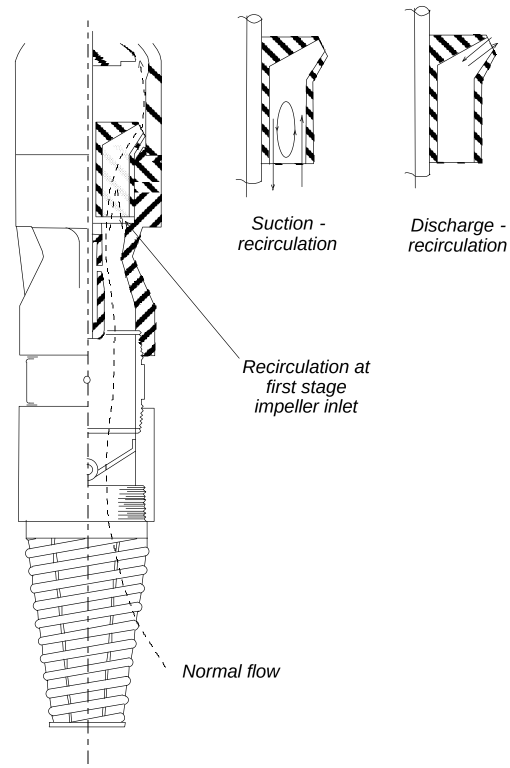


Figure 9.22 Recirculation in centrifugal pumps (Plant Engineering).

As far as is known, failures caused by recirculation have not been observed in geothermal lineshaft or submersible pumps. Apparently, the problem is more severe in larger pumps with high specific suction speeds and high efficiencies. If the ratio of impeller eye diameter to outside diameter (OD) is greater than 0.5, discharge recirculation occurs before suction recirculation. Recirculation usually occurs at 50 to 65% of rated capacity, depending on the eye to OD ratio. The onset of recirculation is indicated by a sudden increase of pressure fluctuations at the suction or discharge, which may reach 60 to 90 psi, resulting in vibration and noise. These indications could be noticed in a surface pump, but would be difficult to detect in a pump set deep in a well.

Recirculation cavitation damage possibly has been mistaken for net positive suction head (NPSH) cavitation damage. In cavitation caused by insufficient NPSH, cavitation occurs on the low-pressure side of the vanes. Both suction and discharge recirculation cavitation occurs on the high-pressure side of the vanes.

Although there have been no reported problems, the engineer or pump user should be aware of possible problems when high-efficiency pumps are throttled, as is often the case in space heating applications. Frasier provides equations for calculating the flow at both suction and discharge recirculation onset (Frasier, 1981). Flow is directly proportional to rpm, and flow control through the use of variable-speed drives would reduce the possibility of problems, unless high lift conditions result in esentially the same situation as throttling.

9.3 LESSONS LEARNED

Listed below are a number of factors relating to pumps that can lead to premature failure of pumps and other components. Many of these have been noted or alluded to elsewhere, but are restated here. Some seem obvious, but the obvious is often overlooked (Culver, 1994).

1. Pump suppliers/manufacturers should be provided with complete data on all foreseen operating condi-tions and complete chemical analyses. Standard potable water analysis is not adequate, because they do not test for important constituents, such as dissolved gases.

2. In general, continuous or nearly continuous operation of well pumps is preferred. Short cycle start/stop operation should be avoided. This is particularly true for open lineshaft pumps. When the column drains, bearings and the inside of the column are exposed to oxygen, leading to corrosion.

   Start/stop operations often necessitate a storage tank. This is often a source of air in-leakage. Parts per billion (ppb) of oxygen (O_2) in combination with ppb hydrogen sulfide (H_2S) can lead to early failure of copper and copper alloys, dezincification of brass and bronze and soldering alloys used in valves, fan coils, and piping.

   As noted in Chapter 8, almost without exception, geothermal fluid contains some H_2S. If a start/stop mode of operation is used, air is drawn into the system when fluid drains down the column after the pump stops. This can cause a greatly accelerated rate of pit-ting corrosion in carbon steels, formation of cuprous sulfide films, and crevice corrosion of copper, brass and bronze (except leaded brass and bronze), de- alloying of lead/tin solders and dissolution of silver solder.

   Start/stop operation imposes high shaft and coupling torque loads. It is believed this has led to early failure of lineshafts and lineshaft to motor couplings.

3. Records of pressure and flow versus rpm or power should be kept on a regular basis. Decreases in flow or pressure indicate something is wrong and is a por-tent of more drastic trouble that could occur later on.

4. Pumps should be pulled and inspected on a regular basis, based on experience or as recommended by the manufacturer.

5. Some minimum flow must be maintained in variable-speed applications. Relatively short periods of operation at shutoff will overheat pumps and motors.

6. Motors should be well ventilated. Although this seems obvious, several motors have been installed in below ground unventilated pits. With hot water piping in close proximity, the motor is near its upper operating temperature even when not in operation.

7. Packing glands should be well maintained. All above surface centrifugal pumps tend to in-leak air through packing glands, especially if starting at low suc-tion pressure. Air in-leakage leads to corrosion. Leaks around lineshaft packing lead to corrosion/ scaling of the shaft, making sealing progressively more difficult.

8. Enclosed lineshaft pumps require that lubricant (water or oil) be supplied before the pump is started. It has been observed that in installations where the lubricant flow started and stopped simultaneously with the pump motor, pumps failed prematurely.

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