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DESIGN AND SELECTION OF DEEP WELL PUMPS FOR GEOTHERMAL WELLS

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ABSTRACT

Deep well pumps are necessary in order to get geothermal fluids to the surface for all purposes. Well testing is required for optimum pump design and selection. Well testing involves collection of data and analysis to determine the well loss coefficient. A systematic calculation procedure is presented. The engineering equations used in detailed pump calculation are identified and explained in a case study. The study shows how seasonal and gradual variations of the water table influence the design of the pumps and selection of the optimum discharge. The proper selection of deep well pumps is important for trouble free operation and optimum production from the well. The relative movement of impellers is a critical parameter that can be calculated; the results are used to make sure that the adjustment of impeller to the bowl housing is in the best position. The proper procedures for installing and pulling a deep well pump are given in detail.

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1. INTRODUCTION

The author of this report was awarded a United Nations University (UNU) fellowship to attend the 1991 Geothermal Training Programme held at the National Energy Authority (Orkustofnun) of Iceland, which lasted from April 23 to October 23, 1991. The training programme started with an introductory lecture course for five weeks. The topics of the lectures related to geothermal development i.e. geothermal energy in the world, planning of geothermal projects, geological exploration, borehole geology and alteration mineralogy, hydrology and hydrogeology, geophysical exploration, chemistry of thermal fluids, environment and process utilization, borehole geophysics (logging), well testing, reservoir engineering, drilling technology, well design, low and high temperature utilization and computer applications.

During the second part of the training programme, UNU fellows went on a field excursion from July 10 to July 17, 1991. The main geothermal fields of Iceland were visited, both low and high temperature fields. During the excursion, the fellows received lectures and seminars in the respective areas on geological and geophysical exploration, utilization and the stages of development in each geothermal project.

In the last part of the specialized training programme, the author received training in deep well pump design and attended special lectures in reservoir engineering, well testing, and geochemistry for three weeks. This report will describe the design and selection of deep well pumps and the interpretation of well test data.

Deep well pumps are used to pump geothermal fluids to the surface and onward to the user. As most low temperature geothermal reservoirs are non-artesian, the wells will not produce without pumping. In artesian flow reservoirs, pumps may be installed to increase the flowrate, and in cases where the temperature is above boiling or in wells with a high gas concentration, the pumps may be used to pressurize the water so that it will not boil nor release the gas. The latter case is to be found at Fang in Northern Thailand where two artesian wells are producing flow water for a 300 kW binary turbine. The water temperature is 130°C and the flowrate approximately 22 l/s from the 6" diameter 150 m deep wells. The well fluid (steam and water) is piped to a separator at 1 bar pressure where the steam is vented to the atmosphere, but the hot water is pumped to the power plant. These wells suffer from calcite scaling in surface pipes and the top 10 m of the well. The scales have to be removed by a mechanical cleaning device suspended on a cable every 2 weeks. This procedure takes approximately two hours, and the well output declines over the production period down to approximately 15 l/s. One method that would improve the operation of the plant would be to install a deep well pump. The benefits would be:

1. Greater generating capacity and no reduction in output due to well scaling.
2. Increased production from each well, by lowering of water level.
3. Higher wellhead temperature.
4. No loss of steam to the atmosphere; Better energy recovery.
5. Scaling of calcite will be reduced, but the introduction of a scale inhibitor into the well below the pump to protect the pump will have to be investigated.

In this study, design data for a geothermal well in Iceland with a similar temperature was used, as the required pump test data is not available for the Fang well. The calculation procedure developed can, however, be used once the required data has been collected.

2. GEOTHERMAL WELL CHARACTERISTICS

2.1 General

The main objective of this study is to evaluate the feasibility of using deep well pumps to overcome the calcite scaling problem in low and medium temperature geothermal wells. Evaluation of the site's specific conditions is an important part of designing such a pump system. Several tests and measurements have to be made to establish the temperature of the well, the yield-drawdown relationship and the partial pressure of carbon dioxide. This information is used to make the design calculations and in the selection of the proper pump system. This chapter will describe how the well and reservoir characteristics are obtained and interpreted.

2.2 Well testing

Before a downhole pump can be selected for pumping fluid from a geothermal well, we must have an idea of the well and reservoir characteristics. Well testing methods have been used for decades to evaluate geothermal wells and reservoirs (Kjaran and Eliasson, 1983). There are several methods of testing such as pumping tests, injection tests, and recovery tests. Well tests will give information on the hydrological condition of the well and reservoir system. Future prediction on well discharge and drawdown in the reservoir can also be made.

During the well test, water is either discharged from or injected into the well. This will cause pressure changes, which are either monitored in the production well itself or in an observation well.

The parameters that are generally monitored or estimated during well testing are

- t : time, since well test started (s);
- Q_t : discharge from or injection into the well being tested (l/s or kg/s);
- H : water level in the well (m);
- r_w : radius of discharge well (m);
- r_1 : distance between the discharge well and the observation well (m);
- μ : dynamic viscosity of the reservoir fluid (Pa.s);
- ρ : density of reservoir fluid (kg/m^3).

The main parameters obtained in analyzing well test data are

$$\text{Transmissivity} = kh/\mu \ (\text{m}^3/\text{Pas})$$
$$\text{Storativity} = \phi ch \ (\text{m/Pa})$$

where

- k : permeability of the reservoir (m^2)
- ϕ : porosity of the reservoir rocks
- c : compressibility (Pa^{-1})
- h : reservoir thickness (m)

The initial water level, laminar drawdown coefficient and turbulent drawdown coefficient, which are necessary for deep well pump design are determined by use of Jacob's and Rorabaugh's method or from Theis solution to the diffusivity equation describing fluid flow in a porous media.

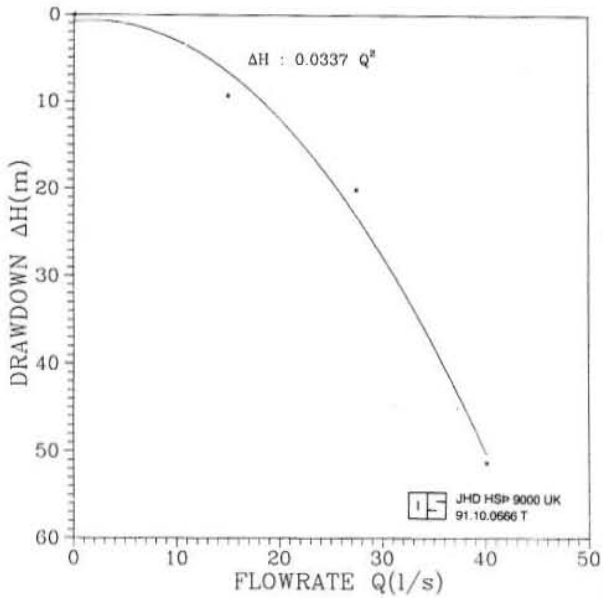

FIGURE 1: Step-drawdown test, (best fit by polynomial)

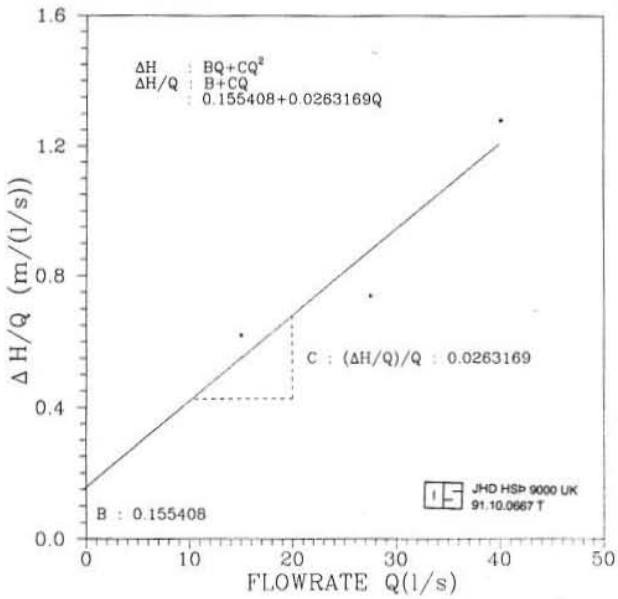

FIGURE 2: Step-drawdown test, $\Delta H/Q$ versus Q

2.2.1 Analytical model used in this study

The water level in a pumping well can be described by the following equation:

$$H = A + BQ + CQ^2 \qquad (1)$$

where

H	: water level (m)
A	: initial water level (m)
B	: laminar drawdown coefficient (m/(l/s))
C	: turbulent drawdown coefficient (m/(l/s)^2)
Q	: well discharge (l/s)

When drawdown is mostly controlled by well losses (Figure 1), the above equation can be simplified to

$$\Delta H = CQ^2 \qquad (2)$$

where

ΔH	: well drawdown (difference between initial and measured water level) (m)

Equation 1 can be simplified for graphical solution often referred to as Jacob's and Rorabaugh's method, by dividing it by Q:

$$\frac{\Delta H}{Q} = B + CQ \qquad (3)$$

This approach is useful for interpreting a step-test when the well is pumped at several different rates and the corresponding drawdown measured. The ratio $\Delta H/Q$ can then be plotted against Q which should yield in a straight line. The coefficients B and C can be determined, as shown in Figure 2. The linear part of Equation 1 corresponds to the laminar drawdown or the drawdown which occurs in the reservoir due to the well discharge. By idealizing the reservoir and boundary conditions, an equation can be derived which describes the flow of fluid in the reservoir. The resulting equation, known as the diffusivity equation, is obtained by defining and combining Darcy's law, the law of conservation of mass and an equation of state. An approximating solution to the diffusivity equation for an infinite acting reservoir is sometimes called the Theis solution. It's logarithmic approximation can be written as (Todd, 1980):

$$\Delta H = \frac{2.30Q}{4\pi T}\log\frac{2.25Tt}{r^2S} = m\log\frac{2.25Tt}{r^2S} \tag{4}$$

Plotting the drawdown, ΔH, as a function of the logarithm of time, should yield a straight line of slope m. This line can be used to determine the transmissivity of the reservoir.

If we read the change in water level (ΔH_{10}) during one log-cycle from such a graph, then we have (Todd, 1980)

$$\Delta H_{10} = \frac{2.30Q}{4\pi T} = m \tag{5}$$

which can be solved for the transmissivity coefficient T.

Furthermore, if we extrapolate the semilog straight line to the time t_o where $\Delta H = 0$, Equation 4 can be rearranged to give the following equation, allowing the storativity coefficient (S) to be calculated:

$$S = \frac{2.25Tt_o}{r^2} \tag{6}$$

This approach is generally called the Cooper - Jacob method of solution (Todd, 1980).

As mentioned before, Equation 4 corresponds to the laminar drawdown for transient case in Equation 1. Turbulent drawdown is then equal to total drawdown minus the laminar drawdown.

2.2.2 Pumping tests

For a pump test to work correctly and efficiently several criteria need to be fulfilled. The main four are to assure the entry of geothermal fluid into the well and facilitate the operation of pumping equipment, the collection of data from within the well and the measurement of pumped discharge.

The pumping tests are normally carried out to assess the hydraulic behaviour of a well and so determine its usefulness as a source of water, predict its performance under different pumping regimes, select the most suitable pumping plant for long term use and give some estimate of pumping costs. Also, to determine the hydraulic properties of the aquifer or aquifers which yield water to the well. These properties include the transmissivity and related hydraulic conductivities, storage coefficient, laminar pressure drop and turbulent pressure drop, type, and distance of any hydraulic boundaries. Furthermore, it may be necessary to determine the effect of pumping upon neighbouring wells. The pumping tests may be performed by the use of several methods, either with downhole pumping or with an air-lift.

The water level changes in the well and observation wells are measured as a function of time. Each step of the pumping test should be of similar duration. Furthermore, the flow rate and the wellhead temperature are measured. The main phases of a pumping test are shown in Figure 3. Pumping equipment and instruments required to collect data are shown in Figure 4.

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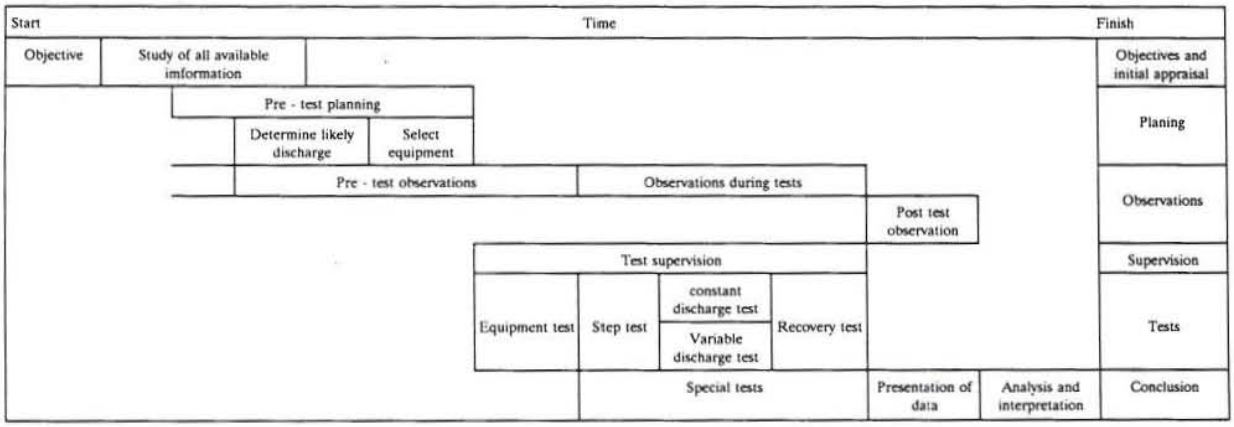

FIGURE 3: Typical pumping test procedure (British Standards Institution, 1983)

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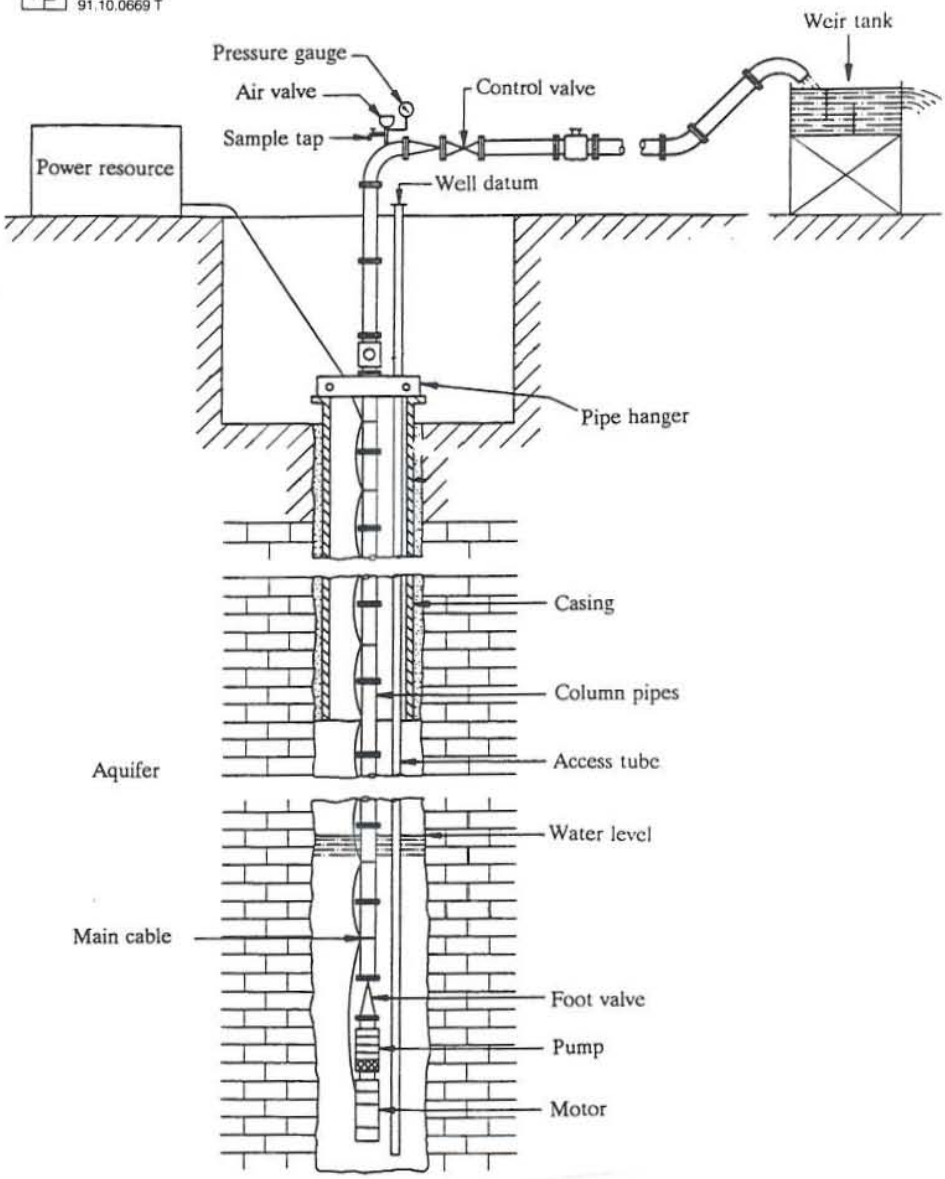

FIGURE 4: Typical arrangement of equipment for a well test (British Standards Institution, 1983)

2.2.3 Air-lift tests

An air-lift test is one pumping test method. The principle of air-lift testing is to use air for lifting the water to the surface by decreasing its density in the well. Compressed air is injected through the air line and through the perforated foot piece on its lower end (Figure 5).

Air-lift has been found to produce the best results when the submergence ratio of the air line is about 60 percent. Both air pressure and air volume are important in initiating and maintaining an air-lift pumping operation. A useful rule of thumb for determining the proper compressor capacity for air-lift pumping is to provide about 0.007 m^3/s (3/4 cfm) of air for each 1 l/s (1 gpm) of water at the anticipated pumping rate. To calculate pumping submergence, the length of air line below the water level is divided by the total length of air line. Good results can be obtained by a skilful operator while pumping with a pumping submergence as low as 30% The air line should not be placed all the way to the bottom of the well when pumping begins, unless required for proper submergence, because the air must then overcome an unnecessarily high pressure head. If steady flow must be maintained, the air volume requirements will usually be greater than those given in Figure 6. For deep wells with low static water level, the actual volume of air required may be two or three times the volume shown in Figure 6 to maintain steady flow. Table 1 gives recommended sizes of educator pipe and air line for air-lift pumping.

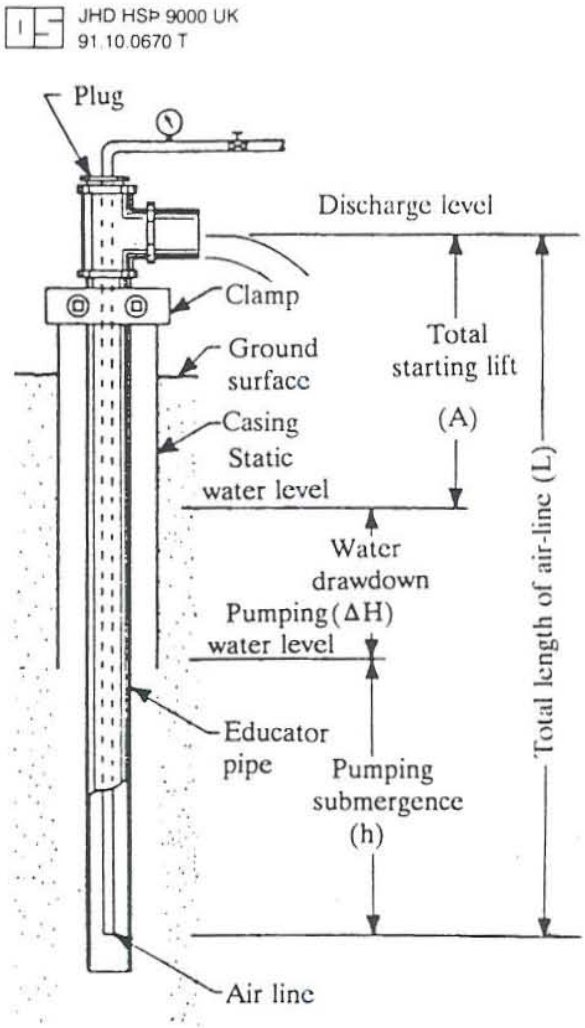

FIGURE 5: Common terms applied to air-lift pumping (Driscoll, 1986)

TABLE 1: Recommended pipe sizes for air-lift pumping (Driscoll, 1986)

Pumping Rate		Size of well casing if educator pipe is used		Size of educator pipe (or casing if educator is not used)		Minimum side of air line	
gpm	l/s	in	mm	in	mm	in	mm
30 - 60	2 - 4	4	102 or more	2	51	0.50	3
60 - 80	4 - 5	5	127 or more	3	76	1	25
80 - 100	5 - 6	6	152 or more	3.50	89	1	25
100 - 150	6 - 10	6	152 or more	4	102	1.25	32
150 - 250	10 - 16	8	203 or more	5	127	1.50	38
250 - 400	16 - 25	8	203 or more	6	152	2	51
400 - 700	25 - 44	10	254 or more	8	203	3	64
700 - 1,000	44 - 63	12	305 or more	10	254	3	64
1,000 - 1,500	63 - 95	16	406 or more	12	305	4	102

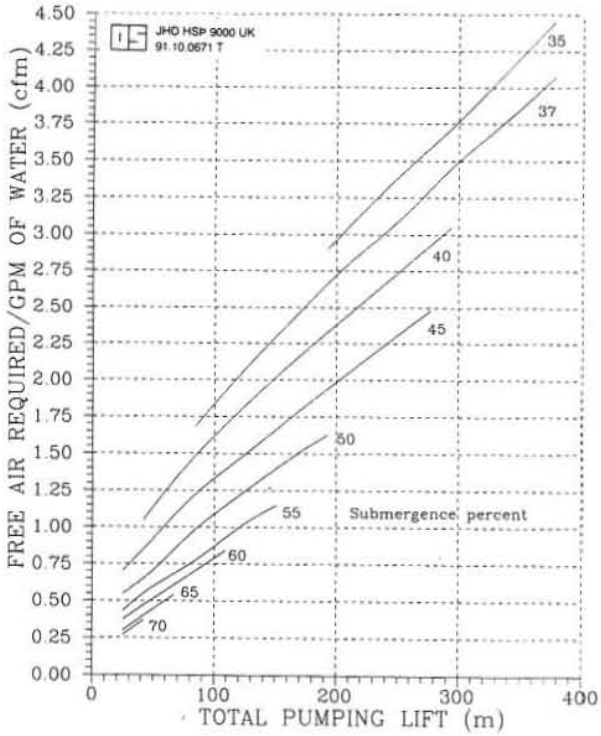

FIGURE 6: Air required for pumping (Driscoll, 1986)

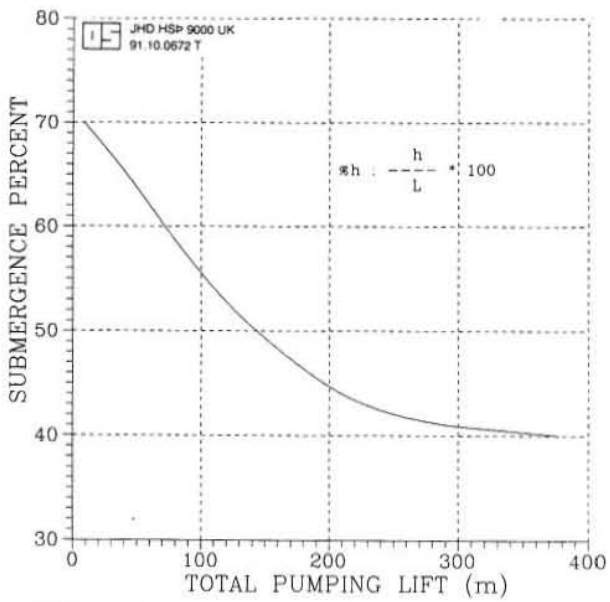

FIGURE 7: Approximate percent pumping submergence for optimum air-lift efficiency (Driscoll, 1986)

To illustrate an air-lift pumping design, one can look at the following example. A geothermal well is completed to a depth of 500 m with 254 mm (10 in) diameter and 284 m deep casing. This casing has been selected to minimize friction losses and provide adequate clearance for the pump bowls. Static water level is at 26 m. It is estimated that the specific capacity will be about 0.6 l/s/m (3 gpm/ft) for the anticipated pumping rate. It is decided to air-lift pump at a rate of 40 l/s (634 gpm), 20 percent above the design rate of 32 l/s (500 gpm). Thus, the water level during pumping will be at about 93 m (305 ft) depth. From this information, it is possible to select the proper size equipment for the air-lift and to estimate the potential efficiency of the system as shown in the following steps:

1. Determine the diameter of the educator pipe (if required) and the air line. Table 1 shows that a 203 mm (8 in) educator and a 64 mm (3 in) air line should be used for the 254 mm (10 in) casing. If no educator pipe is used, Table 1 recommends the same size of air line to be used.

2. Determine the lengths of the educator pipe and the air line. When educator pipe is used, the air line will always stay inside that pipe. From Figure 7 for the total pumping lift 93 m (305 ft), the optimum submergence is found to be 52%.

$$\%h = \frac{h}{L} 100 \qquad (7)$$

where

h : pumping submergence (m)
L : total length of air line (m)
or $L = h + \Delta H + A$
ΔH : well drawdown (m)
A : static water level (m)

then

$$52 = (L - 93)\, 100 / L \quad ==> \quad L = 194 \text{ m}$$

Which leads to selecting setting depth of educator below the end of air line at about 195 m.

3. Determine air volume requirements. For a total pumping lift of 93 m (305 ft), Figure 6 shows that 0.007 m^3/s (1 cfm) is required to pump 1 l/s (1 gpm). Thus, 0.3 m^3/s (634 cfm) are required to pump 40 l/s (634 gpm).

4. Determine whether the compressor has sufficient pressure to initiate flow in the air line.

   Minimum pressure requirement, P_{min}, (bar)

$$P_{min} = \frac{L-A}{10.2} \tag{8}$$

   where 1 bar corresponds approximately to 10.2 m of water column. Then

$$P_{min} = (194 - 26) / 10.2 = 16.5 \text{ bar}$$

   So an air pressure of at least 16.5 bars (239 psi) will be needed to start the air-lift for the starting submergence selected. As drawdown develops, the pressure requirement drops substantially because the head acting on the air line decreases. In general, air-lift pumping will be most efficient when the static water lever is high, the casing diameter is relatively small, and the pumping depth is not excessive in relation to the pressure capability of the compressor.

During air-lift pumping tests, flow rate, water level or drawdown and temperature are measured. A weir is usually used for measuring the flow rate. Water lever or drawdown in the well can be measured by a pressure gauge at the wellhead connected to a capillary tube, which is installed inside the casing well below the water level or at similar depth as the educator pipe. The water level can be calculated by

$$H = L - P\rho g \tag{9}$$

where

H	: water level (m)
L	: capillary tube length (m)
P	: reading pressure (bar)
ρ	: density of water (kg/m^3)
g	: acceleration of gravity (m/s^2)

2.2.4 Injection tests

Geothermal wells are commonly tested at the end of completion by injection tests. The principle is the same as for pumping tests, but the direction of water flow is reversed. The drilling rig pumps are generally used for injection tests. The water is injected through the drill pipes and the pressure (or water level) measured. The flow rate is measured by counting the strokes of the pumps. The water level will increase up to the condition of the reservoir during the test. The data is plotted and analyzed in the same manner as the pumping test in sections 4.2.1 and 4.2.2. The purpose of this operation is to determine the permeability, storativity, laminar and turbulent drawdown for the well.

2.3 Downhole pumps for geothermal wells

Basically, there are two types of deep well pumps most commonly used in geothermal wells, i.e. lineshaft turbine pumps and submersible pumps, the difference being the location of the driver (Rafferty and Culver, 1989). For a lineshaft pump, the driver is usually an electrical motor, mounted above the wellhead and drives the pump through a long shaft. In a submersible pump, the driver (a long, small diameter electric motor) is usually located below the pump itself. The pump is coupled to the driver through a relatively short shaft with a seal section to protect the motor from the well fluid.

Vertical lineshaft turbine pumps in deep well settings have two definite limitations. They must be installed in relatively straight wells and the economical setting limit is not more than 240 m. In geothermal wells, the temperature and chemical content of water are the main problem in selecting the proper material of component parts. The lineshaft material generally used is carbon steel 1045 (AISI C-1045). In some cases the lineshaft must be changed due to the condition of geothermal water. The bowl unit material generally used is cast iron, but special materials can be ordered. The impeller materials generally used are bronze (B 584-838) and cast iron. The most critical part of the pump for geothermal applications is the shaft bearings which are made of teflon (PTFE 1491 N).

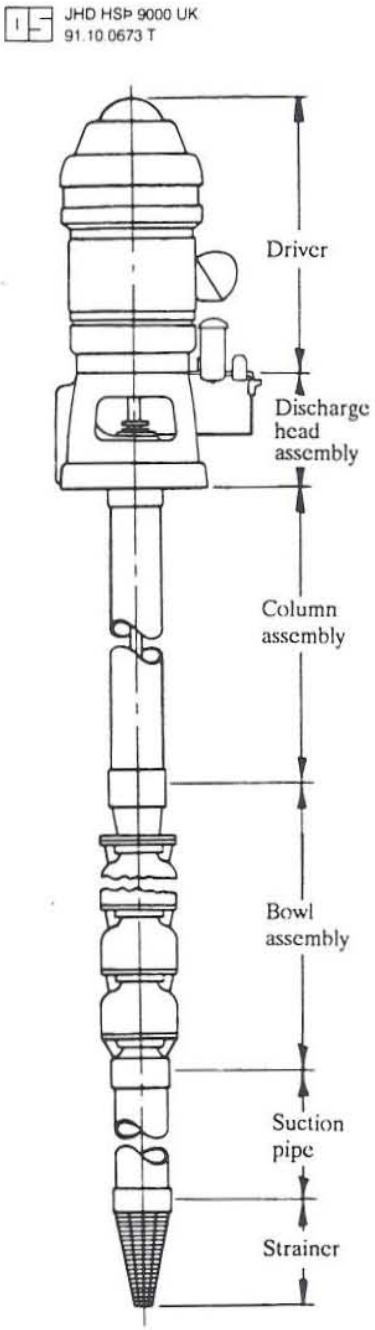

FIGURE 8: Basic components - enclosed lineshaft pump (Peabody Floway, 1988)

3. DEEP WELL LINESHAFT PUMPS

3.1 General

Pumping is required to produce water from geothermal wells and to deliver it to the end user. For certain geothermal fluids, such as those with high carbon dioxide content, corrosion and scaling can be reduced by maintaining the water under pressure.

3.2 Selection of a deep well pump

Selection of a pump type (Figure 8) is dictated by the setting depth, well design, well deviation, temperature, type and size of driver, based on lowest life cycle costs, repair costs, and availability of spare parts.

The lineshaft vertical turbine pumps have two basic methods for lubrication. One is the enclosed lineshaft construction where oil or filtered water as a lubricant is supplied to the lineshaft bearings through the enclosing tube. This report is written to cover this latter type of construction. The second method is open lineshaft construction which uses the pumped fluid directly for lubrication. The lineshaft and bearings are open to the pumped fluid in this case.

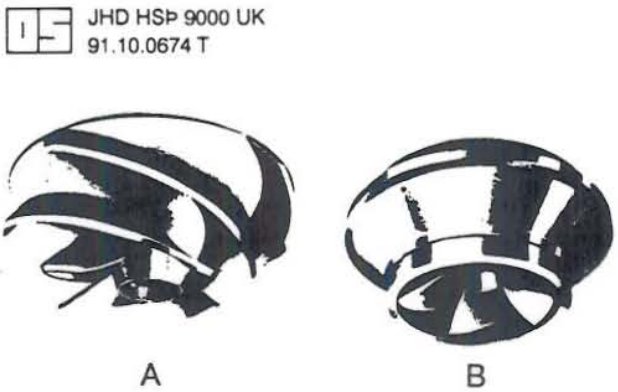

FIGURE 9: A) Semi-open impeller and B) enclosed type impeller (Peabody Floway, 1988)

Two types of impellers are available "semi-open" and "enclosed" (Figure 9). The arrangement of the bowl assembly is identical with either semi-open or enclosed type requiring only that the intermediate bowls and suction bell be designed to provide sealing surfaces for the type impeller used. For geothermal pumps enclosed impellers are used. The pump impellers are connected to the bowl shaft by tapered lock collets or collet and key with locking screw. The shaft and all rotating parts are supported by the thrust bearings of the hollow shaft motor or separate thrust bearing assembly. There is an impeller adjusting nut at the top of the hollow shaft motor assembly, or a three or four piece coupling with adjusting nut for solid shaft driver arrangements.

The column pipes assembly and discharge head are selected for suitable size, length and type for assembly arrangements. The pump size recommended by FLOWAY vertical turbine pumps (Peabody Floway, 1988) for different flowrates are shown in Table 2 (see also Figure 10).

TABLE 2: Recommended flow ranges for vertical turbine pumps (Peabody Floway, 1988)

Pump size	Column size	Flow ranges recommended
6" and 8"	6" Column pipe 2" Enclosing tube 1 3/16" Lineshaft	14 - 40 l/s
10"	8" Column pipe 2 1/2" Enclosing tube 1 11/16" Lineshaft	50 - 75 l/s
12"	10" Column pipe 2 1/2" Enclosing tube 1 11/16" Lineshaft	85 - 110 l/s

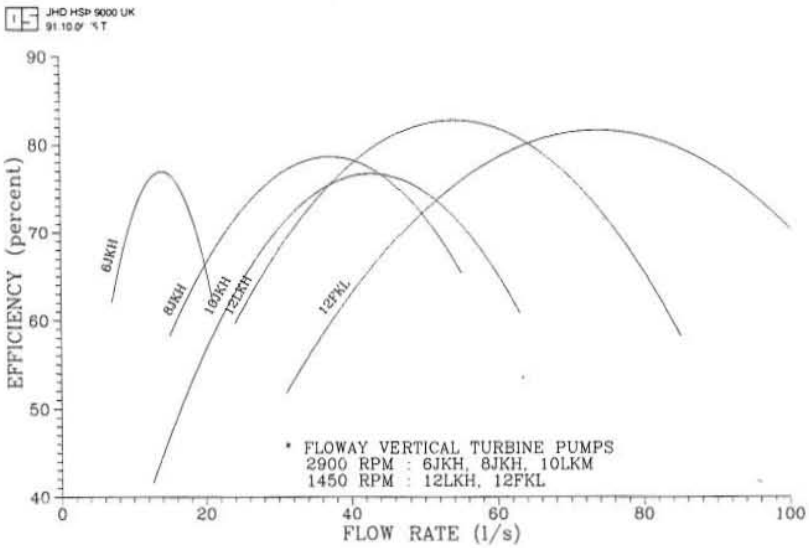

FIGURE 10: Multi pump performance curves

3.2.1 Borehole

The borehole data that must be known are

C_1 : well drawdown coefficient (m/(l/s)^2)
t : temperature of water (°C)
K_V : total static water level (m)

3.2.2 Pump performance curve

The equation of the pump curve (pump pressure versus output) is

$$P_D = z_o + z_1 Q + z_2 Q^2 + z_3 Q^3 \qquad (10)$$

This information is specific to each pump and in order to simplify the solution of the equations, the performance curve is approximated with two straight lines (see Figure 11 and Table 3):

$$P_D = (Y_{2i} - Y_{1i} Q) Z \qquad (11)$$

where

i = 1 : if $Q < Q_B$

i = 2 : if $Q \geq Q_B$

Q_B : discharge at point of intersection of the two lines (l/s)

P_D : total dynamic pressure head (m)

Z : number of impellers (stage)

This approximation is fairly accurate around Q_B

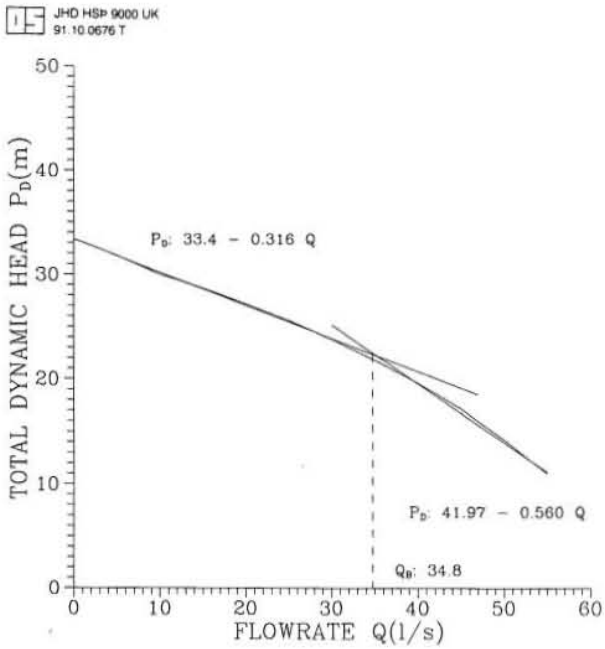

FIGURE 11: 8JKH 2900 rpm pump performance curve (Floway, 1970)

TABLE 3: Coefficient of pump performance curves

PUMP TYPE	Coefficient of pump performance curves				
	Y_{11}	Y_{21}	Y_{12}	Y_{22}	Q_B
FLOWAY 8JKH 3000 rpm	0.316	33.40	0.560	41.97	36.0
FLOWAY 6JKH 3000 rpm	0.240	17.72	0.763	22.89	9.8
FLOWAY 12DKH 1500 rpm	0.139	20.55	0.206	24.78	62.0
FLOWAY 12FKH 1500 rpm	0.066	18.76	0.136	25.83	99.0
FLOWAY 10HKH 1500 rpm	0.040	10.04	0.204	18.36	45.0
FLOWAY 10LKM 3000 rpm	0.330	59.60	1.179	99.90	49.0
PEERLESS 8HXB	0.212	27.04	0.528	42.56	50.0
F.MORSE 8HC 3000 rpm	0.286	31.20	0.546	41.55	39.5
F.MORSE 6HC 3000 rpm	0.181	13.97	0.613	18.66	11.0

3.2.3 Pump pressure break down

$$P_D = P_H + (K_V + K_N)\rho_t \cdot 10^{-3} + P_F + P_d \qquad (12)$$

where

P_H : discharge head pressure (m)

K_V : total static water level (m)

K_N : well drawdown (m)

P_F : pressure loss in column pipe (m)

P_d : dynamic pressure loss in column pipe (velocity head) (m)

ρ_t : density of water at pumping temperature (kg/m^2)

$$K_N = C_1 Q^2 \tag{13}$$

$$P_F = C_2 Q^2 (L/100) \tag{14}$$

$$Pd = \frac{V^2}{2g} = \frac{0.229 Q^2}{(DDI^2 - DSU^2)} \tag{15}$$

where L : length of pump setting (m)
Q : well discharge (l/s)
C_1 : well drawdown coefficient (m/(l/s)^2)
C_2 : pressure loss in column pipe coefficient (m/(l/s)^2/100 m of column)
V : velocity of water in column pipe (m/s)
g : acceleration of gravity (m/s^2)
DDI : inside diameter of column pipe (in)
DSU : outside diameter of enclosing tube (in)

Equations 12 to 15 are solved together in order to calculate the only unknown parameter Q:

$$Q = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \tag{16}$$

where

$$a = C_1 \rho_1 \cdot 10^{-3} + C_2 \cdot (L/100) + \frac{0.229}{(DDI^2 - DSU^2)^2}$$

$$b = Y_{1i} Z$$

$$c = P_H + K_V \rho_t \cdot 10^{-3} - Y_{2i} Z$$

3.2.4 The minimum submergence of bowl unit

In order to prevent cavitation, the minimum submergence is calculated by:

$$h_{\min} = \frac{(P_a - P_u) 10^5}{\rho_t g} + \frac{NPSHR}{(\rho_t / \rho_{20})} \tag{17}$$

where h_{min} : minimum submergence of bowl unit (m)
P_a : vapour pressure (bar)
P_u : atmospheric pressure (bar)
ρ_t : density of fluid at temperature T °C (kg/m^3)
ρ_{20} : density of water at 20°C (kg/m^3)
$NPSHR$: net positive suction head required (m)
g : acceleration of gravity (m/s^2)

3.2.5 The minimum length required of column

The minimum length of the pump column in the well is:

$$L_{min} = K_{V1} + K_{V2} + K_N + h_{min} \tag{18}$$

where K_{V1} : length above sea level to well head (m)
K_{V2} : length below sea level to static water level (m)
K_V $= K_{V1} + K_{V2}$

3.3 Adjustment of the impellers

A deep well turbine pump is characterized by its long shaft, enclosing tube and column pipe hanging in the well (Figure 12). To ensure trouble free running of this system the net elongation of shaft and column should not exceed the end play (axial clearance) provided for the impellers inside the bowl unit assembly. Following calculations of the impeller movement in the bowl unit assume that the whole pump unit is kept in thermal equilibrium and that the enclosing tube does not share load with the column pipe. The impellers must be adjusted in such a way that during the operation, no rubbing will occur either on the top or bottom. As already selected then, these calculations are for an enclosed line system.

3.3.1 Elongation of shaft due to hydraulic thrust

The hydraulic thrust on the shaft is calculated by

$$T_A = 14.6KP_D \tag{19}$$

where T_A : hydraulic thrust on shaft (N)
K : thrust constant (lb/ft)
P_D : total dynamic pressure head (m)

The pump shaft will elongate because of the hydraulic thrust on it:

$$E_A = 9.541 \cdot 10^{-6} \frac{T_A L}{D_o^2} \tag{20}$$

where E_A : elongation of shaft due to hydraulic thrust (mm)
L : length of pump shaft (assumed the same length of pump setting) (m)
D_o : diameter of pump shaft (in)

3.3.2 Elongation of the column due to hydraulic thrust

The total hydraulic thrust on column is equal to the thrust due to total pressure head (P_D) on the column system minus hydraulic thrust on the shaft (reaction force):

$$T_R = P_D A - T_A \tag{21}$$

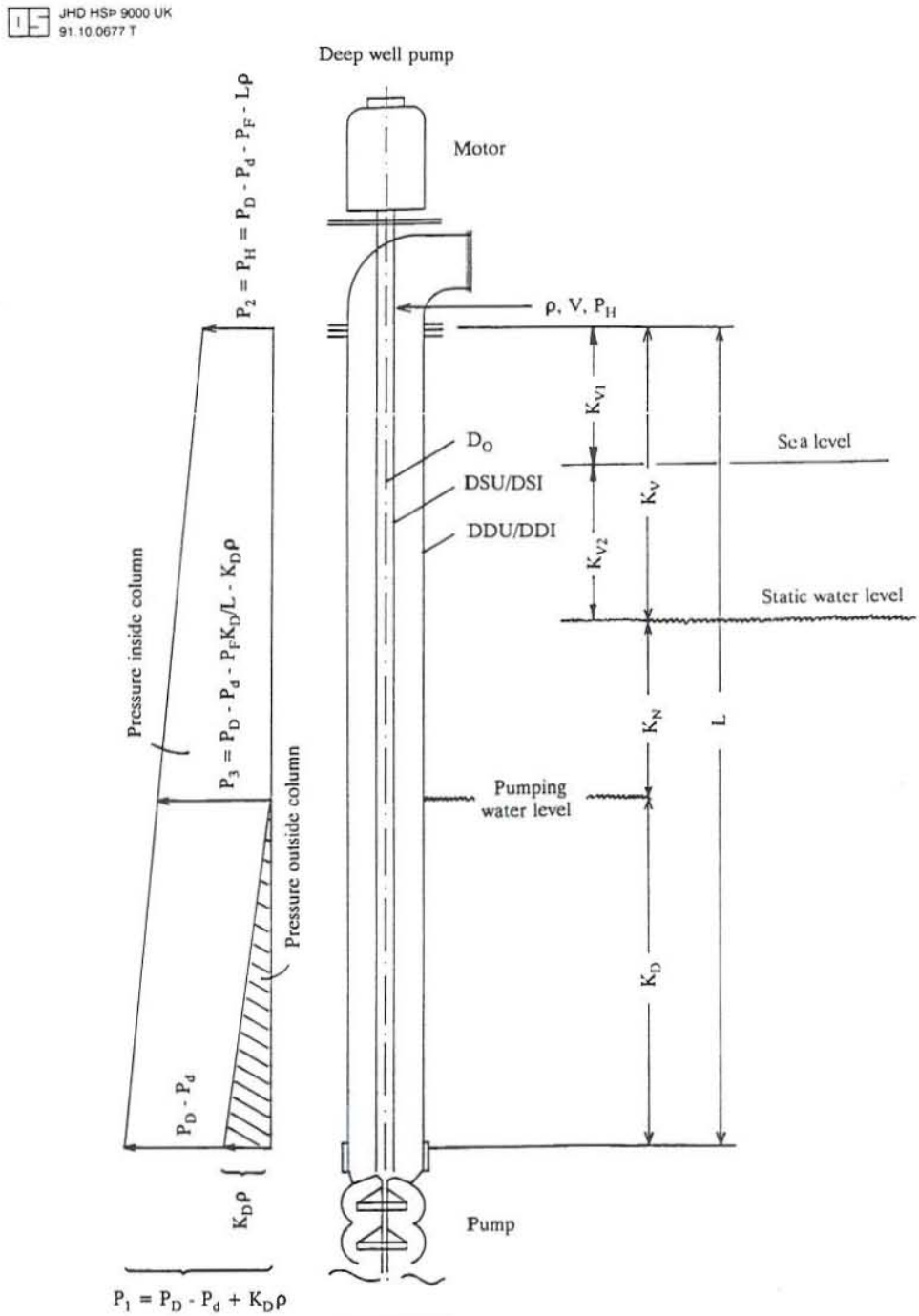

FIGURE 12: Diagram of definitions

or

$$T_R = 4.97(DDI^2 - DSU^2)P_D - T_A \tag{22}$$

where

T_R	: total hydraulic thrust on column pipe (N)
A	: effective cross-section area of column pipe (in^2)
DDI	: inside diameter of column pipe (in)
DSU	: outside diameter of enclosing tube (in)

The column pipe will be elongated because of the hydraulic thrust on column (T_R):

$$E_{R1} = 9.54 \cdot 10^{-6} \frac{T_R L}{DDU^2 - DDI^2} \tag{23}$$

where

E_{R1} : elongation of column (mm)
DDU : outside diameter of column pipe (in)

3.3.3 Column shortening due to internal pressure

When the pump is in operation, the column pipe is under internal pressure by the pumped fluid, being greatest at the lowest point. This pressure shortens the column pipe:

$$E_{R2} = \frac{1.37 \cdot 10^{-5} (2P_D - L\rho_t \cdot 10^{-3}) L}{(DDU/DDI)^2 - 1} \tag{24}$$

where E_{R2} : Shortening of column pipe (mm)
L : length of pump setting (m)

Net column elongation is:

$$E_R = E_{R1} - E_{R2} \tag{25}$$

where E_R : net column elongation (mm).

Net relative movement of impellers:

$$E_E = E_A - E_R = E_A - E_{R1} + E_{R2} \tag{26}$$

where E_E : net relative movement of impellers (mm)

E_E must not exceed the bowl unit axial clearance by some safety margin in order to guarantee problem free operation.

The total weight of pump shaft and impellers can be calculated by:

$$T_S = 41.4 D_o^2 L + T_I \tag{27}$$

where T_S : weight of pump shaft (N)
T_I : weight of impellers (N)

Total thrust load on motor: Total thrust on shaft, that is carried by the motor thrust bearing, is:

$$T_T = T_A + T_S \tag{28}$$

where T_T : total thrust on shaft (N)
T_A : hydraulic thrust on pump shaft (N)
T_S : weight of pump shaft and impellers (N)

3.3.4 Required size of the motor

The high thrust vertical induction motors of the hollow-shaft type are usually used to drive the deep well pumps (in some cases solid shaft motors are used). The power required size of the motor can be calculated by

$$A_M = A_D + A_L + A_A = A_D + A_F \tag{29}$$

$$A_D = 1.315 \cdot 10^{-5} \frac{P_D \rho_t Q}{\eta_D} \tag{30}$$

$$A_L = nT_T \cdot 1.684 \cdot 10^{-8} \tag{31}$$

$$A_A = fL \cdot 10^{-2} \tag{32}$$

where

A_M	: power required by motor (BHP)
A_D	: power required of pump (BHP)
A_L	: power loss in motor thrust bearings (HP)
A_A	: power loss of pump shaft (HP)
ρ_t	: density of water at T °C (kg/m^3)
Q	: well discharge (l/s)
η_D	: efficiency of pump (%)
n	: rotation of pump (rpm)
f	: coefficient of friction loss in lineshaft bearing (HP/L = 100 m)

The power consumption of motor is defined by the equation below

$$A_N = \frac{A_M}{\eta_M} \tag{33}$$

and kilowatt input to motor by

$$= 0.746 \cdot A_N = \sqrt{3} \cdot 10^{-3} \cdot EIPf \tag{34}$$

where

A_N	: power consumption of motor (HP)
η_M	: efficiency of motor (%)
E	: voltage per leg applied to motor
I	: amperes per leg applied to motor
Pf	: power factor of motor
$\sqrt{3}$	: factor for 3-phase motor (this reduces to 1 for single phase motors)

3.4 Material selection

In this section the subject is material selection for the deep well pump unit. The main components and their sizes are highlighted.

Pipes
Specification of material: Seamless black steel tube according to DIN 2448/1629/page 3, mat. st 35, mill test certificate to DIN 50.049/2.2, plain ends, in exact lengths of 3100 mm ± 5 mm tolerance (Table 4).

TABLE 4: Sizes of column, coupling and enclosing tube

COLUMN PIPES		
Nominal dia. (in)	Outside dia. (mm)	Thickness (mm)
5	139.7	6.3
6	168.3	7.1
8	219.1	8.0
10	273.0	10.0
COUPLING		
5	159.0	12.5
6	193.7	17.5
8	244.5	17.5
10	298.5	17.5
ENCLOSING TUBE		
2	60.3	6.3
2 1/2	76.1	8.0

Capillary tubes
Specification of material: Welded black steel tubes according to DIN 2441, st 33 DIN 1626, with threads and muffs. Pipes outside protected against rust in fixed lengths of 3100 mm ± 5 mm, bundled. Sizes 1/4" or 3/8" black.

Teflon bearings for lineshaft
Specification of material: Teflon type PTFE 1491 N. See Table 5.

TABLE 5: Size of bearing stock material required for fabrication

SIZE OF SHAFT (in)	OD/ID (mm)	LENGTH (mm)
1 3/16 shaft with sleeve	45/35	130
1 3/16 shaft without sleeve	45/28	200
1 11/16 shaft with sleeve	58/47	170
1 11/16 shaft without sleeve	60/40	200

Sleeves of shaft
Specification of stock material: Seamless steel tubing of AISI 304 stainless steel, 4-5 m.

SIZE OF SHAFT (in)	SIZE OF SLEEVES OD/ID (mm)
1 3/16	40/25
1 11/16	56/40

Lineshaft pump
Specification of material: Shaft materials of deep well pumps are in lengths 3100 mm, bright precision, cold drawn, stress relieved ground and polished bars of steel C 45 (AISI C1045). Maximum diameter tolerances + 0-0.06 mm (h 9 according to DIN 7157). Shaft straightness must be within 0.08 mm total indicator reading and end sawn straight.

Shaft size stock material: 1 3/16 in (30.16 mm), 1 11/16 in (42.86 mm), 35 mm, 45 mm (Tables 6 and 7, Figures 13 and 14).

TABLE 6: Calculated weight of components

SIZE	THICKNESS (mm)	WEIGHT (kg/m)	LENGTH (m/unit)	WEIGHT (kg/unit)
Column 10" (273.0 mm)	10.0	64.9	3.0226	196.17
Column 8" (219.1 mm)	8.0	41.5	3.0226	125.44
Column 6" (168.3 (mm)	7.1	28.3	3.0226	85.54
Enclosing tube 2 1/2"	7.1	12.1	1.5208	18.40
Enclosing tube 2"	6.3	8.42	1.5208	12.81
Enclosing tube 2"	5.6	7.53	1.5208	11.45
Shaft 1 3/16"	(2 * 0.31) + (5.617 * 3.042)			17.71
Shaft 1 11/16"	(2 * 0.55) + (11.34 * 3.042)			35.60

TABLE 7: Total weight of components in kg/unit (3.042 m)

ITEMS	WEIGHT OF COMPONENT		
	10" 2 1/2" 1 11/16"	8" 2 1/2" 1 11/16"	6" 2" 1 3/16"
Shaft	35.60	35.60	17.71
Shaft coupling	0.79	0.79	0.45
Enclosing tube x 2	36.80	36.80	25.62
Bearing x 2	2.09	2.09	1.32
Centralizer	3.01	1.30	0.98
Column	196.17	125.44	85.54
Column coupling	15.0	13.5	6.1
Total weight	289.5	215.52	137.7

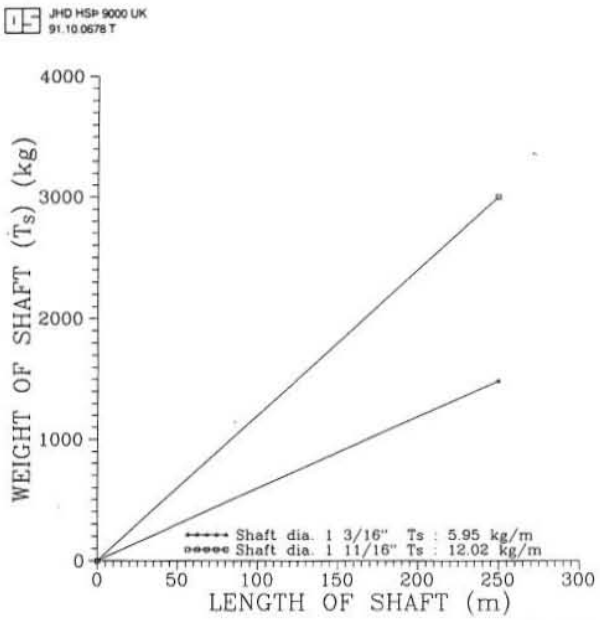

FIGURE 13: Weight of shaft with length

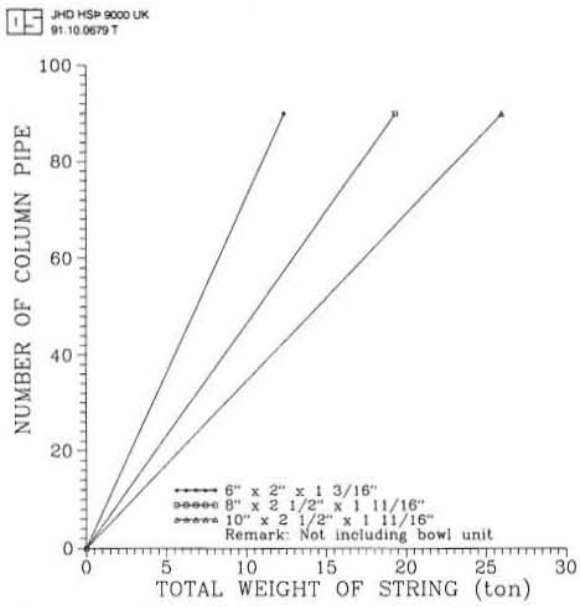

FIGURE 14: Total weight of deep well string

3.5 Maintenance

A periodic inspection is recommended as the best means of preventing breakdown and keeping maintenance cost to a minimum. A change in noise level, amplitude of vibration, or performance can be an indication of impending trouble. Variances from initial performance will indicate changing system conditions or wear or impending breakdown of the unit.

A periodic once-a-week inspection is suggested for all units. During this inspection, the pump and driver should be checked for performance and change in noise or vibration level, loose bolts or piping, dirt and corrosion.

Impeller re-adjustment after thermal equilibrium is reached if properly set at initial installation of pump. After extended operation under abrasive conditions, the sealing faces will wear, causing a reduction in performance.

Pump lubrication is very critical and requires careful monitoring. Adjustment of packing gland is necessary to prevent excessive leakage. The driver lubrication will require periodic attention. Refer to the driver instruction manual for recommendations.

3.6 Repairs

Before pulling the bowl unit from the well, the installation and operation manual should be reviewed for causes of reduced performance not related to the condition of the bowl assembly. Some of the possible causes of reduced performance could be:

- Incorrect lateral adjustment. (This is especially critical with units utilizing semi-open impellers)
- Well conditions, such as gas or air in the water.
- Change in operating conditions such as increased pumping fluid level or increased discharge head requirements.
- Slow motor speed due to overload, low voltage or low frequency.
- Incorrect direction of rotation.
- Strainer clogged or suction "sanded in".
- Inadequate submergence of pump suction.

Disassembly of the unit is basically the reverse of the assembly procedure. Handle all parts with care to prevent bending of shafts and damage to machined surfaces.

Check shaft for straightness; shaft must be straight within **0.005" (0.127 mm)** total indicator reading. Figure 15 indicates the recommended method for checking shaft straightness. If the shaft is not straight it must be straightened or replaced. If the deflection is gradual over the considerable length, the shaft can usually be straightened by supporting the crooked section on two blocks straddling and applying pressure to the high side to deflect the shaft to the opposite direction. If the shaft has a sharp crook (dog-leg), it is recommended that the

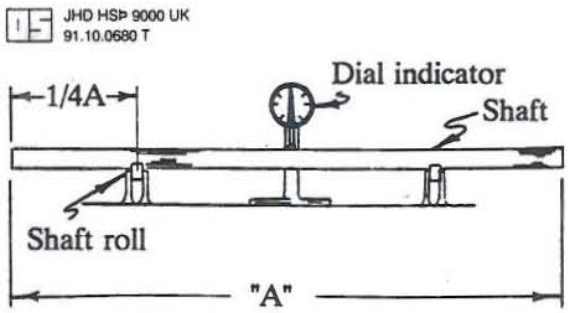

FIGURE 15: Checking shaft for straightness (Peabody Floway)

shaft be replaced since the shaft will not always remain straight even if satisfactorily straightened.

After disassembly, all components should be thoroughly cleaned and examined for physical defects, wear, corrosion and damage. The bowl assembly becomes worn when operated a long time. The nominal and maximum allowable of parts are shown in Table 8.

Parts showing signs of damage, cracks or excessive wear should be replaced. When repairing a pump that has been in service for several years, the physical condition or straightness of all parts such as cap screws, bowls, threads, etc., must be carefully checked to be sure these parts can continue to perform their function without failure.

TABLE 8: Deep well pump material nominal size and maximum allowable wear

TYPE	MATERIAL	NOMINAL (mm)	MAX. ALLOWABLE WEAR (mm)
Shaft 1 11/16"	Shaft sleeve OD	48.00	-
	Bearing ID	48.35	48.78
	Shaft OD	42.86	-
	Tube adaptor bearing ID	43.00	43.43
	Stuffing box bearing ID	43.10	43.53
Shaft 1 3/16"	Shaft sleeve OD	35.00	-
	Bearing ID	35.35	35.66
	Shaft OD	30.16	-
	Tube adaptor bearing ID	30.30	30.61
	Stuffing box bearing ID	30.40	30.71
Bowl units from Floway	6JKH 1"		
	Shaft OD	24.50	-
	Bearing ID	24.70	25.01
	Impeller skirt OD	-	- 0.31
	8JKH 1 3/16"		
	Shaft OD	30.16	-
	Bearing ID	30.36	30.67
	Impeller skirt OD	-	- 0.31
	12DKH 1 11/16"		
	Shaft OD	42.86	-
	Bearing ID	43.07	43.50
	Impeller skirt OD	-	- 0.33
	12FKH 1 11/16"		
	Shaft OD	42.86	-
	Bearing ID	43.07	43.50
	Impeller skirt OD	-	- 0.41

3.7 Engineering information

This section will give information about limitations of mechanical and electrical equipment for deep well pump assemblies (Tables 9, 10, 11 and 12).

Standard bowl assembles are generally designed for non-corrosive, non-abrasive service over a

wide temperature range (Table 9). Published performance curves are based on a maximum fluid viscosity of 50 SSU. The maximum pump speed is also limited by lineshaft RPM limitations, critical speed, and/or submergence (NPSH) available to prevent cavitation.

TABLE 9: Limitations of standard bowl assemblies (Peabody Floway, 1988)

SIZE & TYPE	SHAFT SIZE (IN.)①	MAX. STGS.②	MAX. PRESSURE P.S.I.③	STD. AXIAL④ CLEARANCE	MAX. AXIAL CLEARANCE (by mach.)④⑤	THRUST FACTOR (K)⑥	IMPELLER WEIGHT (LBS.)	EYE AREA (IN^2)	SPECIFIC SPEED (N_s)	$WR^2$⑦ (WET @ 1.0 S.G.)	MAX. SPHERE (IN.)⑧
4HOLL	3/4	32	500	...	...	1.6	0.9	1.4	2526	0.008	1/8
4HOL	3/4	32	500	...	...	2.4	0.9	2.0	2750	0.009	3/16
4HOH	3/4	32	500	...	...	2.2	0.9	2.4	3139	0.008	5/16
6LKM	1	32	506	1/4	7/16	2.1	2.4	1.3	2155	0.046	3/16
6JOLL	1	33	489	...	...	3.6	1.9	2.7	2355	0.047	1/4
6JOL	1	33	489	...	...	3.6	1.9	3.9	2350	0.051	3/8
6JOH	1	33	489	...	...	3.6	1.9	4.9	2710	0.046	1/2
6JKL	1	33	489	3/8	1/2	3.6	2.9	6.5	2680	0.059	1/2
6JKM	1	33	489	3/8	1/2	3.6	2.9	6.5	2800	0.057	1/2
6JKH	1	33	489	3/8	1/2	3.6	3.0	6.5	2740	0.055	1/2
8LKL	1 3/16	35	429	1/2	5/8	4.2	4.5	4.1	2510	0.132	3/8
8LKM	1 3/16	35	429	7/16	9/16	4.2	4.5	4.8	2370	0.132	1/2
8JOL	1 3/16	29	414	...	...	4.4	3.8	7.65	2690	0.108	3/4
8JOH	1 3/16	29	414	...	...	5.5	3.4	7.65	2550	0.102	11/16
8JKL	1 3/16	29	414	9/16	3/4	3.8	5.2	11.8	2580	0.210	3/4
8JKM	1 3/16	29	414	9/16	3/4	4.7	5.4	11.8	2660	0.210	3/4
8JKH	1 3/16	29	414	9/16	3/4	4.7	5.5	11.8	2700	0.215	11/16
8FKH	1 3/16	31	310	3/8	1/2	5.5	5.7	11.2	4030	0.178	1/2
8XKL	1 3/16	19	575/1000	7/16	...	2.4	4.3	2.9	1500	0.129	3/8
8XKH	1 3/16	19	575/1000	7/16	...	2.4	4.2	2.9	1510	0.126	3/8
8XKN	1 3/16	...	...	7/16	...	7.2	6.1	10.5	1853	0.133	3/8
10LKL	1 1/2	19	488	1/2	5/8	5.2	9.0	10.1	1690	0.675	1/2
10LKM	1 1/2	19	488	1/2	5/8	5.2	9.0	10.1	1715	0.692	1/2
10DOL	1 1/2	24	383	...	...	10.4	6.0	18.2	3140	0.381	1
10DOM	1 1/2	24	383	...	...	10.4	6.0	18.2	2980	0.381	7/8
10DOH	1 1/2	24	383	...	...	10.4	6.2	18.2	2970	0.376	7/8
10DKL	1 1/2	24	383	9/16	3/4	8.3	10.8	18.2	3150	0.598	1
10DKM	1 1/2	24	383	9/16	3/4	8.3	10.8	18.2	3110	0.605	7/8
10DKH	1 1/2	24	383	9/16	3/4	8.3	10.9	18.2	3160	0.613	7/8
10BKL	1 1/2	29	410	7/16	5/8	10.0	10.75	20.9	3850	0.675	1
10BKM	1 1/2	29	410	7/16	5/8	10.0	9.75	20.9	4260	0.609	1
10BKH	1 1/2	29	410	7/16	5/8	10.0	10.75	20.9	4260	0.675	1
10FKH	1 11/16	24	333	9/16	3/4	13.3	9.0	28.8	5800	0.711	1
10XKL	1 1/2	14	500/1000	1/2	...	4.6	8.1	7.9	1630	0.409	1/2
10XKH	1 1/2	14	500/1000	1/2	...	4.6	8.7	7.9	1560	0.439	1/2
10XKN	1 1/2	1	NA	1/2	...	8.8	8.8	17.3	1650	0.379	7/16
11XKL	1 11/16	10	750/1500	3/4	...	5.7	11.5	9.5	1595	0.709	1/2
11XKH	1 11/16	...	750/1500	3/4	...	5.7	11.7	9.5	1545	0.721	1/2
11XKN	1 11/16	1	NA	3/4	...	10.3	10.6	20.9	2025	0.580	1/2
12LKL	1 11/16	28	405	5/8	7/8	8.0	19.9	15.0	2680	1.788	3/4
12LKM	1 11/16	28	405	1/2	3/4	8.0	16.9	17.1	2280	1.594	7/8
12LKH	1 11/16	28	405	1/2	3/4	6.0	17.1	17.1	2370	1.537	7/8
12DOL	1 11/16	25	352	...	...	15.0	11.0	26.4	2900	0.711	1 1/4
12DOM	1 11/16	25	352	...	...	15.0	11.5	26.4	2860	0.734	1 1/8
12DOH	1 11/16	25	352	...	...	15.0	12.0	26.4	2845	0.739	1
12DKL	1 11/16	25	352	5/8	15/16	12.0	16.5	26.4	3180	1.015	1 1/4
12DKM	1 11/16	25	352	5/8	15/16	12.0	15.0	26.4	3110	0.993	1 1/8
12DKH	1 11/16	25	352	5/8	15/16	12.0	15.0	26.4	3050	0.997	1
12FKL	1 11/16	25	267	1/2	1	15.5	23.6	32.2	3370	2.164	1
12FKH	1 11/16	25	267	1/2	1	13.5	18.6	32.2	3770	2.178	1
13XKL	1 15/16	6	750/1500	7/8	...	8.1	19.3	13.7	1590	1.687	5/8
13XKH	1 15/16	6	750/1500	7/8	...	8.1	19.8	13.7	1540	1.730	5/8
13XKN	1 15/16	1	NA	7/8	...	16.6	23.0	33.7	1610	2.059	5/8

The horsepower loss due to mechanical friction of shaft rotation in the column (A_A in Equation 32) may be determined from Table 10 below. For an enclosed lineshaft with a flooded tube or the enclosed lineshaft using water injection lubrication, use two times the factor values listed in the table.

TABLE 10: Mechanical friction in BHP per 100 m of lineshafts (Peabody Floway, 1988)

SHAFT SIZE (in)	RPM OF SHAFT								
	3460	2900	1760	1450	1160	960	860	720	690
3/4	0.60	0.51	0.31	0.26	0.20	0.17			
1	1.05	0.87	0.53	0.44	0.35	0.29	0.26	0.25	
1 1/4	1.60	1.33	0.79	0.64	0.52	0.44	0.39	0.34	
1 1/2	2.20	1.90	1.14	0.96	0.74	0.63	0.56	0.47	0.44
1 11/16	2.81	2.36	1.43	1.18	0.94	0.78	0.70	0.59	0.56
1 15/16			1.83	1.51	1.21	1.00	0.89	0.75	0.72
2 1/4			2.40	2.00	1.55	1.35	1.20	1.00	0.96
2 7/16			2.90	2.39	1.91	1.58	1.42	1.19	1.14
2 11/16			3.34	2.75	2.20	1.82	1.63	1.38	1.31
2 15/16				3.50	2.77	2.28	2.00	1.74	1.66
3 3/16				4.04	3.28	2.72	2.46	2.08	1.92
3 7/16					3.75	3.08	2.83	2.45	2.30
3 11/16					4.10	3.46	3.00	2.62	2.47
3 15/16					4.80	3.89	3.48	2.94	2.77

The shaft horsepower ratings are shown in Table 11.

TABLE 11: AISI C-1045 carbon steel shaft horsepower ratings (Peabody Floway, 1988)

RPM	SHAFT DIAMETER (inches)														
	3/4	1	1 3/16	1 1/4	1 1/2	1 11/16	1 15/16	2 1/4	2 7/16	2 11/16	2 15/16	3 3/16	3 7/16	3 11/16	3 15/16
3500	55	137	222	264	475	706	1094	168							
2900	46	113	184	219	393	585	906	1394							
1770	28	69	112	134	240	357	553	851	932	1280					
1450	23	57	92	109	197	293	454	697	764	1048	1389				
1180	19	46	75	89	160	238	369	567	622	853	1131	1467	1859	2315	2843
980	15	38	62	74	134	198	308	471	516	709	939	1218	1543	1923	2361
880	14	34	56	66	120	178	276	423	464	636	843	1093	1386	1727	2120
730		29	46	56	99	146	228	351	385	528	699	907	1150	1432	1758
705		28	45	54	96	142	220	340	371	510	676	875	1110	1383	1699
590			38	45	80	118	184	284	311	427	565	732	929	1158	1421
505			33	39	68	102	158	243	266	365	484	627	795	990	1217
Max.thrust (lbs)	3000	4000	5000	5000	10000	10000	15000	20000	20000	20000	25000	25000	30000	35000	40000

Column friction loss is loss due to fluid flowing between the column surface and the enclosing tube. This column friction loss is used for calculation in Equation 14 as shown in Table 12.

TABLE 12: Friction loss (m) per 100 m of column (Peabody Floway, 1988)
(Open or enclosed lineshaft design)

	COLUMN AND SHAFT SIZE (Inches)																								
	2½	3	4			5			6				8				10				12				
GPM	¾	¾	¾	1	1¼	¾	1	1¼	¾	1	1¼	1½	1	1¼	1½ 1 11/16	1 15/16	1¼	1½ 1 11/16	1 15/16	2¼	1¼	1½ 1 11/16	1 15/16	2¼	2 7/16 2 11/16
10	1.2																								
15	2.0																								
20	2.8	1.0																							
25	3.5	1.4																							
30	4.2	1.9																							
40	5.4	3.1		.6	1.2																				
50	6.6	4.5	.7	.9	1.7																				
60	9.0	6.1	.9	1.2	2.3																				
70		8.5	1.2	1.6	2.9																				
80		9.9	1.5	1.9	3.7																				
90		12.0	1.8	2.4	4.4																				
100		14.0	2.2	2.8	5.3	.6	.7	1.0																	
125			3.2	4.2	7.8	.8	.9	1.4																	
150			4.4	5.8		1.2	1.3	1.9																	
175			5.8	7.5		1.6	1.7	2.5																	
200			7.3			2.0	2.2	3.1	.8	.7	1.0	1.4													
225						2.9	2.8	3.9	1.0	.9	1.2	1.7													
250						3.1	3.3	4.7	1.1	1.1	1.4	2.0													
275						3.7	3.9	5.6	1.3	1.3	1.7	2.4													
300						4.3	4.6	6.4	1.6	1.5	2.0	2.8													
325						5.0	5.4	7.4	1.8	1.7	2.3	3.2													
350						5.8	6.2	8.4	2.0	2.0	2.6	3.6													
375						6.6	7.0		2.3	2.2	2.9	4.1													
400						7.4	7.9		2.9	2.5	3.3	4.6		.6	.7	1.0									
450						9.3			3.5	3.1	4.1	5.7		.8	.9	1.3									
500									4.2	3.7	5.0	6.9		1.0	1.1	1.5									
550									4.9	4.4	5.8		1.1	1.2	1.3	1.8									
600									5.7	5.2	6.8		1.3	1.4	1.5	2.1									
650									6.6	6.0			1.5	1.6	1.8	2.5									
700									7.5				1.7	1.9	2.0	2.8									
750													2.0	2.1	2.3	3.2									
800													2.2	2.4	2.6	3.6	.6	.7	.8	1.0					
850													2.5	2.7	2.9	4.0	.7	.8	.9	1.1					
900													2.8	3.0	3.2	4.5	.8	.8	1.0	1.2					
950													3.1	3.3	3.6	4.9	.9	.9	1.1	1.3					
1000													3.4	3.6	3.9	5.4	1.0	1.0	1.2	1.4	.4	.4	.4	.5	
1200													4.8	5.1	5.6	7.6	1.3	1.4	1.7	2.0	.6	.6	.6	.7	
1400													6.4	6.8	7.4	10.0	1.8	1.9	2.2	2.7	.7	.8	.8	1.0	1.2
1600													8.3	8.8	9.5		2.3	2.4	2.8	3.4	.9	1.0	1.1	1.2	1.4
1800													10.3	11.0	11.9		2.9	3.0	3.5	4.3	1.2	1.2	1.3	1.5	1.8
2000													12.6				3.5	3.7	4.3	5.2	1.4	1.5	1.6	1.8	2.1
2200													15.1				4.2	4.4	5.1	6.1	1.7	1.8	1.9	2.1	2.5
2400																	4.9	5.2	6.0	7.2	2.0	2.1	2.3	2.5	3.0
2600																	5.7	6.1	7.0	8.2	2.3	2.5	2.6	2.9	3.5
2800																	6.6	7.0	8.0	9.6	2.7	2.8	3.0	3.3	4.0
3000																	7.5	7.9	9.1	10.0	3.1	3.2	3.4	3.8	4.5
3200																					3.5	3.6	3.9	4.3	5.1
3400																					3.9	4.1	4.3	4.8	5.7
3600																					4.3	4.5	4.8	5.3	6.4
3800																					4.8	5.0	5.3	5.9	7.1
4000																					5.3	5.5	5.9	6.4	7.8
4200																					5.8	6.0	6.4	7.1	8.8
4400																					6.3	6.6	7.0	7.7	9.9
4600																					6.8	7.2	7.6	8.4	
4800																					7.4	7.8	8.3	9.0	

4. PUMP DESIGN AND SELECTION

UTILIZATION OF THE WELL TEST-1 WITH THE TEMPERATURE 130°C

KNOWN DATA:

1. Well data.

Depth	: 928 m
Casing diameter	: 230 m
Casing length	: 284 m
Elevation (K_{V1}) (m.a.s.l.)	: 26 m
Water temperature	: 130 °C
Density of water at 20°C (ρ_{20})	: 998 kg/m^3
Density of water at 130°C (ρ_t)	: 935 kg/m^3
Wellhead pressure (P_H)	: 44 m (4.4 bar)
Vapour pressure (P_a)	: 2.7 bar
Minimum atmospheric pressure (P_u)	: 0.96 bar
Acceleration of gravity (g)	: 9.81 m/s^2

Drawdown characteristics of the well were measured by air-lift pumping. The temperature of the water during the test was 66°C and the data from well testing is as follows:

Flow rate	Drawdown
15.0 l/s	: 9.3 m
27.5 l/s	: 20.1 m
40.1 l/s	: 51.1 m

2. Geothermal reservoir data. The water table in the reservoir is estimated to vary between 60-100 m b.s.l. (K_{V2}) during the operation period.

3. The need for the water is approximately 40 l/s for our process.

CALCULATE THE FOLLOWING AND DETERMINE:

I - Well drawdown coefficient (C_1)
II - Suitable pump to approximately pump at 40 l/s. Draw the curve for Q as a function of K_{V2}. The variation as a function of time influences the selection of the pump by seasonal variation and gradually decreasing water table.
III - Adjustment of the impellers after thermal equilibrium is reached.
IV - Suitable motor size for the pump.
V - Verify that the equipment selected has not reached its limitation.
VI - Minimum lifting capacity of the mobile crane.

Figures 16-17 show the calculation sequence used in this report. The equation number used in each step is shown in brackets (see Section 3).

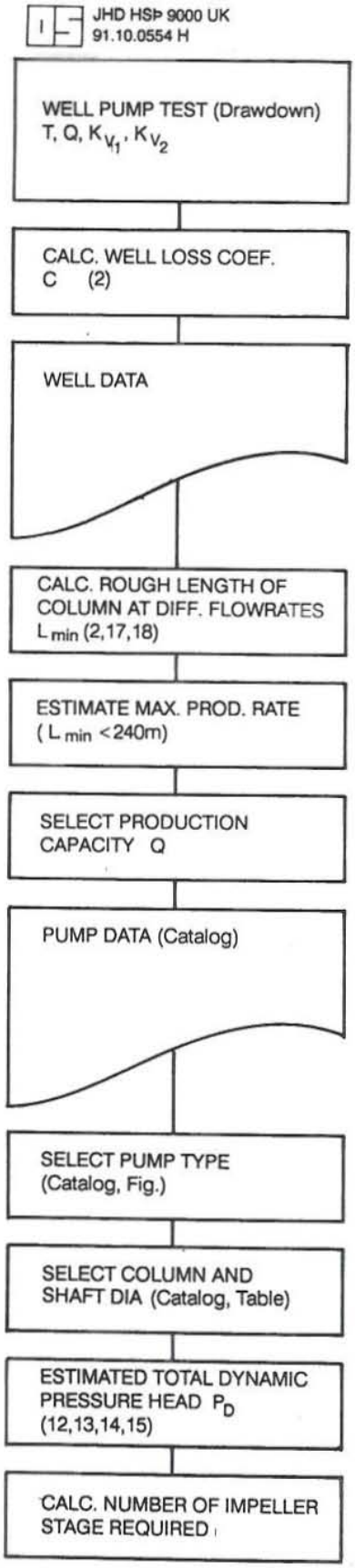

FIGURE 16: Preliminary pump calculation

SOLUTION:

I - Well drawdown coefficient (C_1)

Air-lift pumping data is plotted. Flow rate versus drawdown and best fit curve are calculated. The data was provided by the built-in function of the program "GRAPHER" (Golden Software) as shown in Figure 1. Equation 2 for the drawdown curve is

$$K_N = C_1 Q^2$$

where

$K_N = \Delta H$ = well drawdown (m)
C_1 = well drawdown coefficient (m/(l/s)^2)

giving

$K_N = 0.0337\ Q^2$
$C_1 = 0.0337$ m/(l/s)^2

Rough calculation of the maximum pumping rate by using deep well pump:
From Equations 11 and 18, where we assume the lowest water level in the reservoir (K_{V2}=100 m):

At pumping rate 40 l/s:

Elevation (K_{V1})	26 m
Water table (K_{V2}) maximum depth	100 m
Well drawdown (K_N)	54 m
Minimum submergence (h_{min})	28 m
Total depth required (L_{min})	208 m
Selected length setting bowl unit (add 5% for safety)	219 m

At pumping rate 60 l/s:

Elevation (K_{V1})	26 m
Water table (K_{V2}) maximum depth	100 m
Well drawdown (K_N)	122 m
Minimum submergence (h_{min})	28 m
Total depth required (L_{min})	276 m
Selected length setting bowl unit (add 5% for safety)	289 m

At pumping rate 70 l/s:

Elevation (K_{V1})	26 m
Water table (K_{V2}) maximum depth	100 m
Well drawdown (K_N)	165 m
Minimum submergence (h_{min})	28 m
Total depth required (L_{min})	319 m
Selected length setting bowl unit (add 5% for safety)	335 m

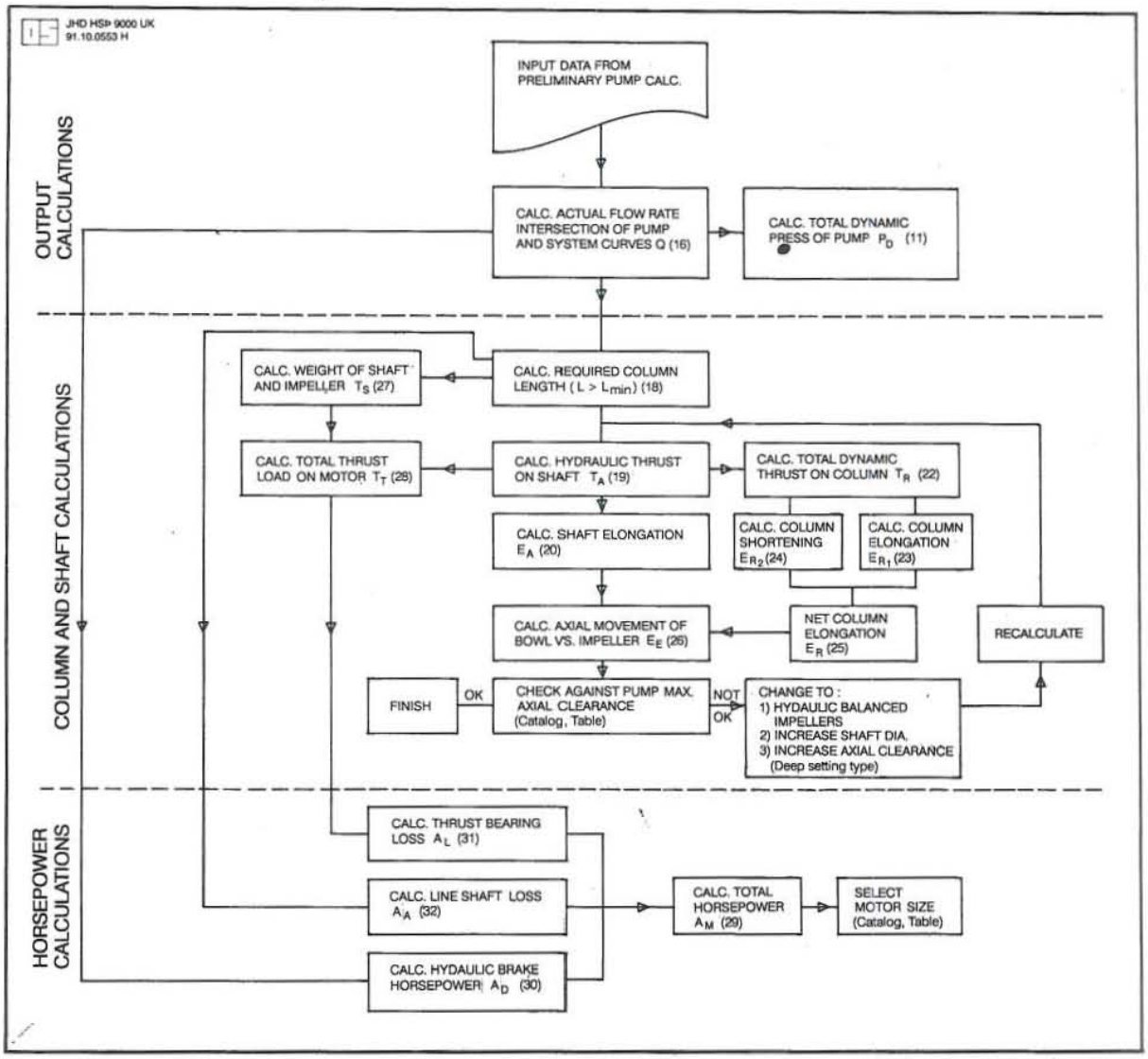

FIGURE 17: Detailed pump calculations

Rough calculation of pump length:
The water table in the reservoir is variable over the operating time between 60-100 m. There are two types of variations, seasonal variations and gradual decrease (Figures 18 and 19). These can influence the selection of pump and number of stages. As we are going to pump 40 l/s, the minimum pump length is 208 m. For safety, the length of bowl unit is selected as 219 m (72 column pipes).

II - Selection of a suitable pump

Pumping rate 40 l/s and setting bowl unit at depth 219 m have been selected within the maximum practical length of the deep well pumps. From Peabody Floway multi pump performance curves (see Figure 10) we select 8JKH pump type for this application.

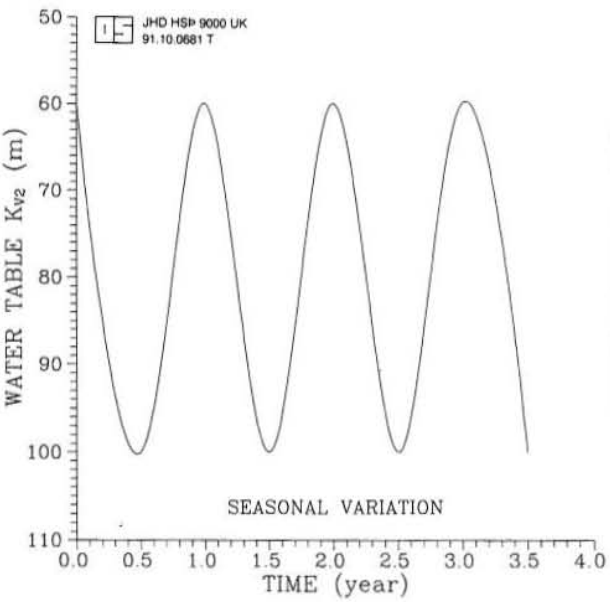

FIGURE 18: Seasonal variation of water table

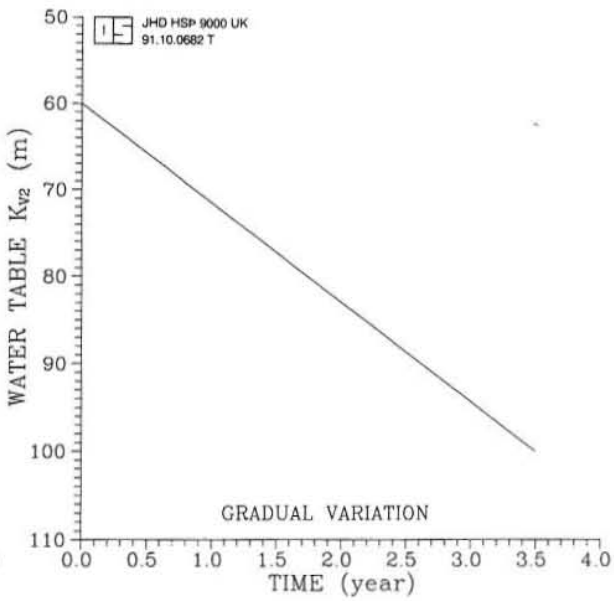

FIGURE 19: Gradual variation of water table

Pump data:
From Peabody Floway vertical turbine pumps, turbine data handbook (2nd ed. 1988)

8JKH, 2900 rpm		
Thrust constant (K)	4.7	lb/ft
Impeller weight	5.5	lb
WR^2	0.215	lb/ft^2
Shaft diameter (D_o)	1 3/16	in
Maximum sphere	11/16	in
Eye area	11.8	in^2
Maximum head (std. const.)	957	ft (292 m)
Maximum number of stages (std. const.)	29	stg
Axial clearance (std.)	9/16	in (14.29 mm)
Axial clearance (max.)	3/4	in (19.00 mm)
Bowl diameter	7 3/4	in
NPSHR (at approximate 30-40 l/s)	8	m

A pump performance curve of 8JKH 2900 rpm with two straight lines is used to approximate the characteristic of the curve (see Figure 11 and Table 3).

Selection of column and shaft from Section 3.4:

Column x enclosing tube x lineshaft 6" x 2" x 1 3/16"
(DDU = 6.626", DDI = 6.067", DSU = 2.374", DSI = 1.878")

Rough calculation of total dynamic pressure head required:
In order to find out how many impellers (stages) we need for the 8JKH pump, one has to estimate the total dynamic pressure head required to pump 40 l/s from the well.

$$P_D = P_H + (K_V + K_N)\,\rho_t 10^{-3} + P_F + P_D \qquad (12)$$

P_H = 44 m
K_V = K_{V1} + K_{V2}
K_VA = 26 + 60 = 86 m
K_VB = 26 + 80 = 106 m
K_VC = 26 + 100 = 126 m

where
- A : in case water table K_{V2} 60 m b.s.l.
- B : in case water table K_{V2} 80 m b.s.l.
- C : in case water table K_{V2} 100 m b.s.l.

<u>NOTE: Each step in the following calculation is made for the above three cases with different water tables in the reservoir. These cases are designated, for example for P_D, as P_DA, P_DB, P_DC etc.</u>

$$K_N = C_1 Q^2 \qquad (13)$$

K_N = 0.0337 * 40^2 = 54.24 m

$$P_F = C_2 Q^2 L/100 \qquad (14)$$

From Table 12, 6" column friction loss, 1 3/16" shaft diameter at 40 l/s (Section 3.7):

P_F = 7.5 m / 100 m of column

or for column length 219 m

P_F = 7.5 * 219 / 100 = 16.4 m

C_2 = 16.4 / [40^2 * (219 / 100)] = 0.0047 m/(l/s)^2/100m

$$P_D = 0.229\,Q^2/(DDI^2 - DSU^2)^2 \qquad (15)$$

P_D = 0.229 * 40^2 / $(6.067)^2$ - $(2.374)^2)^2$ = 0.377 m

then
P_DA = 44 + (86 + 54.24)935 * 10^{-3} + 16.4 + 0.377 = **191.90 m**
P_DB = 44 + (106 + 54.24)935 * 10^{-3} + 16.4 + 0.377 = **210.60 m**
P_DC = 44 + (126 + 54.24)935 * 10^{-3} + 16.4 + 0.377 = **229.30 m**

Calculation of the required number of impellers:
From 8JKH performance curve at flow rate 40 l/s, the total dynamic head is 19.6 m per impeller stage.

Number of impellers (Z):
A = 191.90/19.6 = 9.79 ≈ **10 stages**
B = 210.60/19.6 = 10.74≈ **11 stages**
C = 229.30/19.6 = 11.70≈ **12 stages**

Detailed calculations:

We have already selected the 8JKH pump and its length to be 219 m. The selection of impellers required for the pump depends on how the water table in the reservoir (K_V) varies with time. If it decreases gradually, over for example a ten year period, then it is economical, especially if the price of electricity is high, to start with 10 impellers. When the water table nears 80 m.b.s.l. (case B) then one more impeller is added to the pump etc. The size of the motor is, instead, based on the maximum number of impellers to be used. If the water table has a seasonal variation of 60-100 m.b.s.l. and our process needs 40 l/s all the time, then we must select 12 impellers for the pump so it can pump 40 l/s at the lowest level (case C). The following detailed calculation are shown for case A where K_{V2} = 60 m. The same calculations were made for case B where K_{V2} = 80 m and C where K_{V2} = 100 m. The results of these calculations are shown in Table 13.

Calculation of actual flow rate:
For the selected pump the actual flowrate is calculated by Equation 16

$$Q = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \tag{16}$$

where the coefficients *a, b* and *c* are

$$a = C_1 \rho_1 \cdot 10^{-3} + C_2 \cdot (L/100) + \frac{0.229}{(DDI^2 - DSU^2)^2}$$

$$b = Y_{1i} Z$$

$$c = P_H + K_V \rho_t \cdot 10^{-3} - Y_{2i} Z$$

Hence,

$$a = 0.0337 * 935 * 10^{-3} + 0.0047 * (219/100) + 0.229 / ((6.067)^2 - (2.374)^2)^2$$
$$= 0.042$$

The equation of straight line from performance curve where $Q > Q_B$ is

$$P_D = Y_{2i} - Y_{1i} Q$$

Hence,

$$P_D = 41.97 - 0.56Q$$

where $Y_{2i} = 41.97$
$Y_{1i} = 0.56$

This gives $b = 0.56 * 12$ $= 6.72$

and $cA = 44 + (86 * 935 * 10^{-3}) - (41.97 * 12)$ $= - 379.23$

Hence, QA = **44.2 l/s**

Calculation of total dynamic pressure head of pump (Equation 11):

$$P_D = (Y_{2i} - Y_{1i}Q)Z \qquad (11)$$

P_DA = (41.97 - 0.56 * 44.2)12 = **206.6 m**

A graph illustrating the flow rate change due to water table variation in the reservoir is shown in Figure 20.

Calculation of minimum submergence of pump:
The minimum submergence is calculated with the help of Equation 17.

$$h_{min} = \frac{(P_a - P_u)10^5}{\rho_t g} + \frac{NPSHR}{(\rho_t/\rho_{20})} \qquad (17)$$

h_{min} = (2.7-0.96) 10^5 / (935 * 9.81)
+ 8 / (935/998) = **27.50 m**

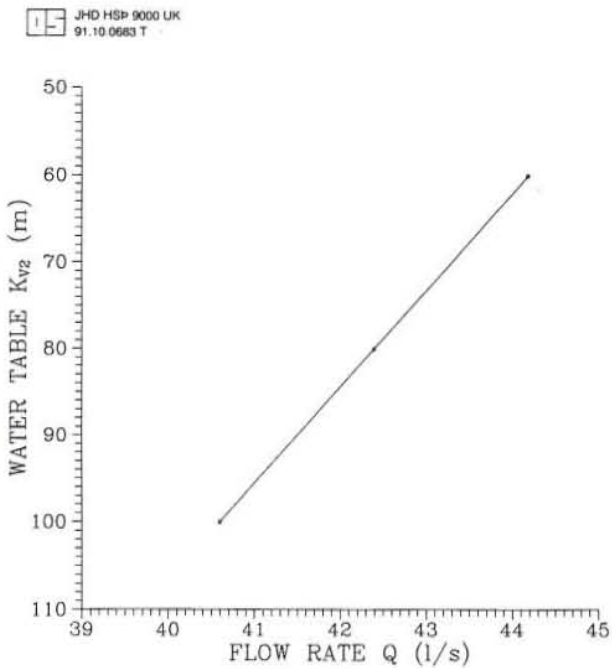

FIGURE 20: Plot showing flow rate versus water table

Calculation of minimum column length:
For setting bowl unit in the well is given by Equation 18.

$$L_{min} = K_V + K_N + h_{min} \qquad (18)$$

$L_{min}A$ = 86 + 0.0337 * $(44.2)^2$ + 27.5 = **179.3 m**

Determine the actual length of column pipes:
$L_{min}C$ (Table 13) shows the maximum depth of the three cases. If we select 209 m and add 5 % column pipe then the length of column pipes will be 219 m. The column unit is 3.042 m/unit. To calculate the number of column pipes, 219 m are divided by 3.042 m, which gives 72 units.

The actual column pipe's length is **219 m** for setting bowl unit.

III - Decide the adjustment of the impellers after thermal equilibrium is reached

Calculate the thrust on shaft, column, elongation of shaft and column. Then we will get the relative movement of impeller. If we use a standard 8JKH pump with 19 mm axial clearance and a thrust constant K = 4.7 lb/f, we find from Equation 26 that the net elongation of the impellers are between 19-22 mm (case A-C). Therefore, we cannot use the standard pump in our application.

The problem can be solved by:

- Increasing shaft diameter (D_o) to decrease elongation and relative movement of impeller.
- Decrease thrust constant (K) by using hydraulic thrust balanced impellers.
- Use a special deep well setting bowl unit with greater lateral clearance. The bowl unit cost is higher than normal bowl unit.

In our case, we select the alternative using balanced impellers, 6 out of 12. This means 6 hydraulic thrust balanced impellers and 6 normal impellers. The new thrust constant (K) is 4.7 * 6/12 = 2.35 lb/f.

Calculation of hydraulic thrust on shaft (Equation 19):

$$T_A = 14.6KP_D \tag{19}$$

T_AA = 14.6 * 2.35 * 206.6 **= 7,137.0 N**

Calculation of shaft elongation (Equation 20):

$$E_A = 9.541 \cdot 10^{-6} \frac{T_A L}{D_o^2} \tag{20}$$

E_AA = 9.541 * 10^{-6} * 7,137.0 * 219 / $(1.187)^2$ **= 10.6 mm**

Calculation of total hydraulic thrust on column (Equation 22):

$$T_R = 4.97(DDI^2 - DSU^2)P_D - T_A \tag{22}$$

T_RA = 4.97 $((6.067)^2$ - $(2.374)^2)$ 206.6 - 7,137.0 **= 24,871.1 N**

Calculation of column elongation due to hydraulic load (Equation 23):

$$E_{R1} = 9.54 \cdot 10^{-6} \frac{T_R L}{DDU^2 - DDI^2} \tag{23}$$

$E_{R1}A$ = 9.54 * 10^{-6} * 24,871.1 * 219 / $((6.626)^2$ - $(6.067)^2)$ **= 7.3 mm**

Calculation of column shortening due to inside pressure (Equation 24):

$$E_{R2} = \frac{1.37 \cdot 10^{-5}(2P_D - L\rho_t \cdot 10^{-3})L}{(DDU/DDI)^2 - 1} \tag{24}$$

$E_{R2}A$ = 1.37 * 10^{-5} (2 * 206.6 - 219 * 935 * 10^{-3}) 219 / $[(6.626/6.067)^2$ - 1] **= 3.2 mm**

Net column elongation (Equation 25):

$$E_R = E_{R1} - E_{R2} \tag{25}$$

E_RA = 7.3 - 3.2 **= 4.1 mm**

Relative movement of the impellers in the bowl unit (Equation 26):

$$E_E = E_A - E_R \tag{26}$$

E_EA = 10.6 - 4.1 **= 6.5 mm**

The relative movement of impellers must not exceed the bowl unit's axial clearance by some safety margin in order to guarantee problem-free operation. For 8JKH 2900 rpm the axial clearance is 19 mm, which is more than enough in this case. It is, therefore, easy to adjust its impellers, when thermal equilibrium is reached. We suggest adjustment of impellers **13.5 mm** up.

IV - Selection of suitable motor for the pump

Calculation of total weight of pump shaft and impellers (Equation 27):

$$T_S = 41.4 D_o^2 L + T_I \tag{27}$$

$T_S = 41.4 * (1.187)^2 * 219 + (5.5 * 12/0.225)$ **= 13,067.9 N**

Calculation of total thrust load on motor (Equation 28):

$$T_T = T_A + T_S \tag{28}$$

$T_T A = 7{,}137.0 + 13{,}067.9$ **= 20,205.0 N**

Required size of the motor (Equation 29 and 30-32):

$$A_M = A_D + A_L + A_A = A_D + A_F \tag{29}$$

Calculation of power required by pump (Eqution 30):
From the performance curve we get the efficiency of the pump:

$\eta_D A = 77\ \%$
$\eta_D B = 78\ \%$
$\eta_D C = 79\ \%$

$$A_D = 1.315 \cdot 10^{-5} \frac{P_D \rho_t Q}{\eta_D} \tag{30}$$

$A_D A = 1.315 * 10^{-5} * 206.6 * 935 * 44.2 / 0.77$ **= 145.8 BHP**

Calculation of thrust bearing power loss (Equation 31):

$$A_L = n T_T \cdot 1.684 \cdot 10^{-8} \tag{31}$$

$A_L A = 2{,}900 * 20{,}205.0 * 1.684 * 10^{-8}$ **= 1.0 HP**

Calculation of lineshaft power loss (Equation 32):
From mechanical friction in lineshafts, Table 10 gives $f = 7.97$ $HP/L = 100$ m. Inserting in Equation 32 gives:

$$A_A = fL \cdot 10^{-2} \tag{32}$$

$A_A = 7.97 * 219 * 10^{-2}$ **= 17.5 HP**

Total power required of motor:

$$A_M A = 145.8 + 1.0 + 17.5 \qquad = \mathbf{164.3\ BHP}$$

The suitable power of the motor for a pump assembly is at the highest load required in the three cases A, B and C, and according to Table 13 this is 164.7 BHP. This value is correlated with pump performance curves. The motors are generally produced in series, i.e. 150, 175, 200 BHP, therefore, we select the motor size **200 BHP** for this pump with an approximate 20% safety margin.

Why is the motor selected with a 20% safety margin?

- To account for inaccuracy in the calculations. If we select the motor to be 175 BHP, then the safety margin is only 6%. Normally one should have a margin of 15% or more.
- Because the pump can overload the motor when producing at high flow rate, for example if wellhead pressure (P_H) becomes 0 bar.
- In the future, it may be found necessary to increase the number of impellers (stages), because the well drawdown will probably increase.

TABLE 13: Summation of calculated parameters

CALCULATED PARAMETERS	VARIATION IN WATER TABLE K_{V2}		
	60 m	80 m	100 m
Water drawdown K_N (m)	65.8	60.6	55.5
Submergence length h_{min} (m)	27.5	27.5	27.5
Minimum length of column L_{min} (m)	179.3	194.1	209.0
Flow rate Q (l/s)	44.2	42.4	40.6
Hydraulic pressure head loss P_d (m)	0.41	0.38	0.35
Column pressure head loss P_F (m)	20.1	18.5	17.0
Total dynamic pressure head P_D (m)	206.6	218.7	230.8
Hydraulic thrust on shaft T_A (N)	7,137.0	7,555.0	7,973.0
Dynamic thrust on column T_R (N)	24,871.1	26,327.7	27,784.4
Weight of shaft and impellers T_S (N)	13,067.9	13,067.9	13,067.9
Total thrust on motor T_T (N)	20,205.0	20,623.0	21,041.0
Elongation of shaft E_A (mm)	10.6	11.2	11.8
Elongation of column due to dynamic load E_{R1} (mm)	8.2	7.8	8.2
Shortening of column due to pressure E_{R2} (mm)	3.2	3.6	4.0
Net column elongation E_R (mm)	4.1	4.2	4.2
Relative movement of impeller E_E (mm)	6.5	7.0	7.6
Efficiency of pump η_D (%)	77	78	79
Power required by pump A_D (BHP)	145.8	146.2	145.8
Power loss on thrust bearing A_L (HP)	1.0	1.0	1.0
Power loss of lineshaft A_A (HP)	17.5	17.5	17.5
Total power required of motor A_M (BHP)	164.3	164.7	164.3

V - Verify the limitation of the equipment

It is very important that the relative movement of impellers (E_E) does not exceed the maximum lateral clearance of the pump. If the relative movement of the impellers is larger than the clearance, one must apply non-standard measures and find out by recalculation (III) if it is suffificient. In our case, the maximum calculated E_E conclusion is **7.6 mm** (Table 13), but the

maximum axial clearance of the pump allows **19 mm**, so the 8JKH 2900 rpm with the assemblies above is a suitable pump for this well.

The shaft horsepower ratings are given in Table 11. For shaft diameter **1 3/16" at 2900 rpm**, maximum horsepower is **184 BHP** and maximum thrust **5,000 lbs (22,222 N)**. The calculations show that the total thrust load on the shaft, T_T, is **21,041 N** (Table 13) which is inside the maximum limit, but as the horsepower is 9% higher there is a reson to be careful.

The maximum thrust capacity of the **200 BHP** motor is provided to **7,000 lbs (31,100 N)** (General Electric Motor 3292C). This is sufficient, because the maximum thrust on shaft is **21,041 N.**

VI - Total weight of pump assembly

The maximum lifting capacity for the mobile crane is defined by Figure 14 and Table 7.

Total column length	= 219 m
Column length per unit	= 3.042 m
Number of column	= 72 unit
Weight of column 6" x 3.042 m	= 85.54 kg
Column coupling	= 6.1 kg
Centralizer	= 0.98 kg
Enclosing tube	= 25.62 kg
Bearing	= 1.32 kg
Shaft	= 17.71 kg
Shaft coupling	= 0.45 kg
Total weight per unit	= 137.7 kg
Total weight of string (not including bowl unit)	= **9,915 kg**

5. INSTALLATION OF GEOTHERMAL DEEP WELL LINESHAFT PUMPS

5.1 General

Proper installation is necessary to obtain maximum service life from the pump. To insure proper alignment three items are very important during installation:

1. All machined mating surfaces must be clean and free of burrs and nicks removed with a fine file or wire brush.

2. Exterior strain must not be transmitted to the pump discharge head. The most common cause of trouble in this respect is forcing the piping to mate with the pump. It is recommended that a flexible connector is installed in the piping adjacent to the pump.

3. All threads should be checked for damage and repaired if necessary. Clean all threads with wire brush and cleaning solvent, end of shafts must be cleaned and any burrs removed since alignment depends on the shaft ends butting squarely. Lubricate all screwed connections with a thread lubricant; a mixture of graphite grease is suitable for steel, an anti galling compound such as "Never-Seez" should be used on stainless and monel mating threads.

5.2 Installation

Preliminary:

a. Center lifting device over well opening and completion assembly.
b. Arrange two heavy timbers or I-beams across foundation to rest pipe clamps on.
c. Check shaft end play of bowl assembly by measuring distance shaft travels when pulled up then pushed all the way down. End play must be equal to or greater than total elongation of the unit indicated by Figure 21 and previous calculations.

Installation of bowl assembly:

a. Attach second pipe clamp near top of bowl assembly, lift bowl assembly and center over suction pipe.
b. Lower bowl assembly and screw onto suction pipe.
c. Lower bowl assembly until clamp rests on timbers, check enclosing tube and shaft projection (Figure 22).

Installing threaded column assembly:

a. Lineshafts are carefully straightened and placed inside the enclosing tube assembly for shipping. Each shaft should be checked by rotating several times to make sure it is not binding. Make sure each column pipe coupling is securely assembled to the column pipe.
b. Tie a rope to the lower end of the column pipe using a timber or clove hitch, then take a half hitch around the tube, and a reverse double half hitch around the lineshaft at the threads (Figure 23).
c. Attach pipe clamp near top of column pipe below coupling and raise the assembly into position over bowl assembly.
d. Slowly lower column section until lineshaft can be connected to the bowl shaft. Place a small amount of lubricant on the shaft threads and screw lineshaft into coupling. Threads are LEFT HAND. Make sure the shafts are butted together but do not use excess force in tightening. Wrenches should not be needed except for the final fraction of a turn to

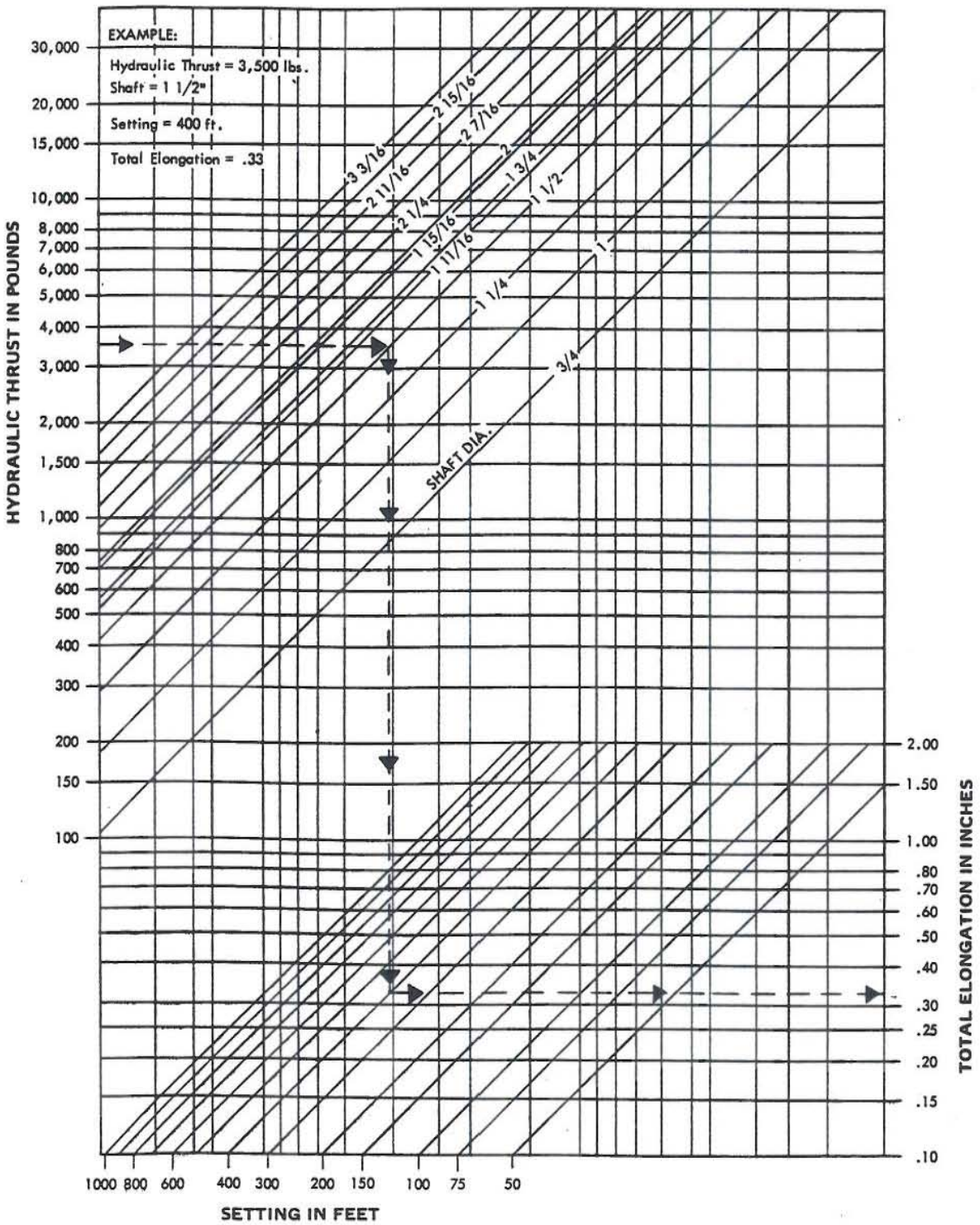

FIGURE 21: Shaft elongation (Floway, 1970)

make the joint tight. Do not allow filings to drop in to the bearings.

e. Lower the enclosing tube and column pipe until the enclosing tube can be connected to the tube adaptor bearing. Lubricate tube adaptor bearing threads and screw enclosing tube on. Threads are LEFT HAND. Tighten with pipe wrench or small chain tong.

f. Lower column pipe into column coupling after lubricating threads. Threads are RIGHT HAND. Tighten with pipe wrench.

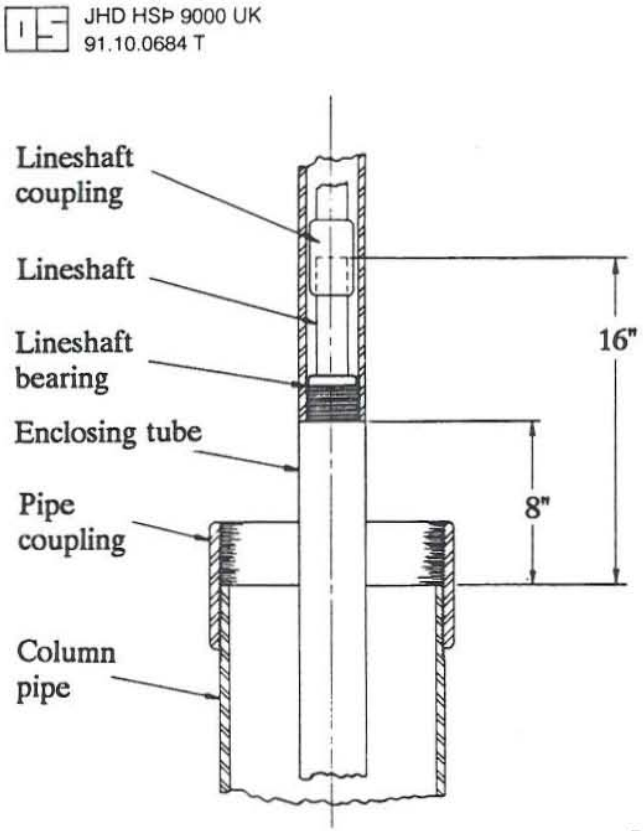

FIGURE 22: Standard enclosing tube and lineshaft projection (Peabody Floway)

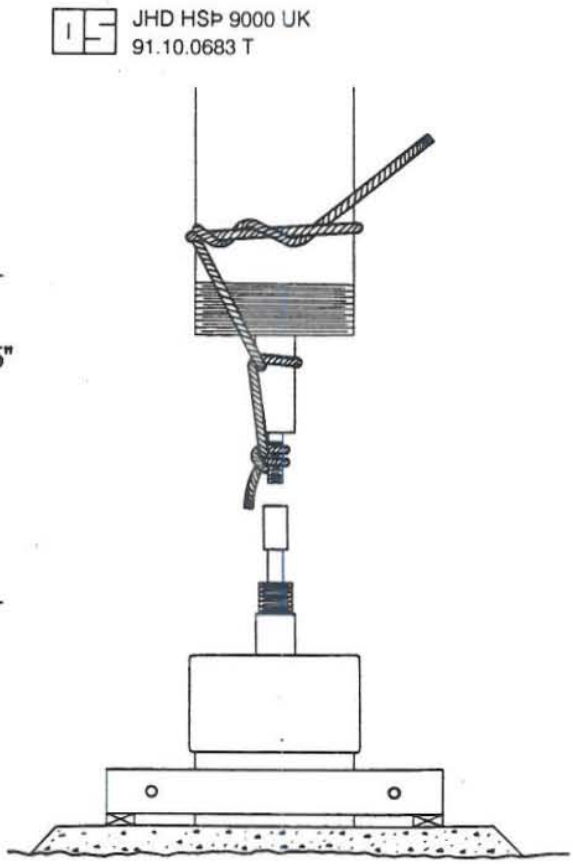

FIGURE 23: Installation of column assembly (Peabody Floway)

g. Lower assembly unit and clamp at top of column pipe. Lower assembly until it rests on timbers.

h. Install enclosing tube stabilizers at every column joint.

i. Repeat this procedure for the remaining sections except the top section with the following precautions. Keep all parts clean, check all threads, repairing any damage, do not bend lineshaft, make sure all joints butt and are tight, and check tube and shaft projection after each joint is installed.

Installing top column section:

a. The components of the top column section should be laid out for inspection and measuring. The top column section will consist of:
   - i. The top column pipe may be a standard 1.5-3 m length or special length required; this length of pipe will not have a coupling attached.
   - ii. The tube adjusting nipple assembly is a special length of enclosing tube with one end turned smooth on the outside and extra long threads on the inside of the same end.
   - iii. The top lineshaft is usually a special length, but is otherwise a standard shaft.
   - iv. The head shaft, extra long threads with key way on top end for connecting to driver.

b. Measure the shaft and enclosing tube projection before the top column section is installed as illustrated in Figure 22. The enclosing tube projection must be between 7 1/2 and 9", while the shaft projection must be between 14 and 18".

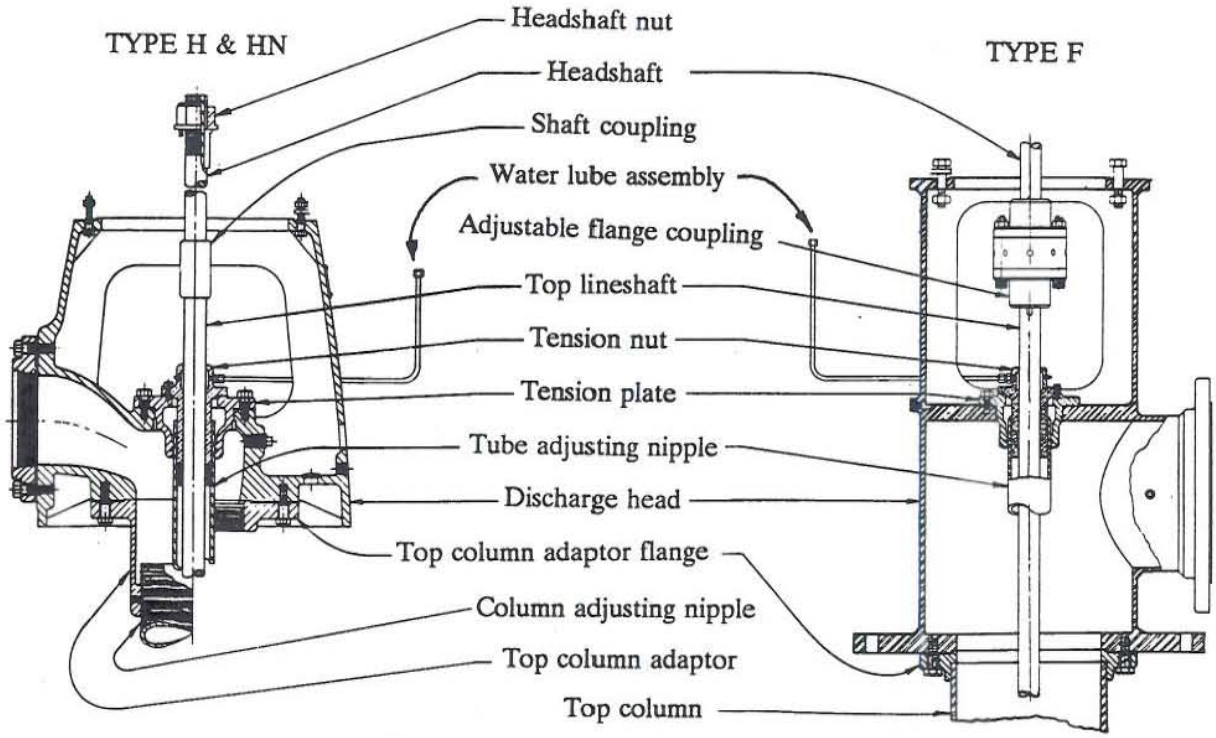

FIGURE 24: Discharge head assemblies (Peabody Floway, 1984)

c. If a separate top lineshaft and head shaft are used, determine where the shaft coupling will be in relation to the top of the tube adjusting nipple. If the shaft coupling is located below the top of the tube adjusting nipple, the headshaft should be connected before proceeding with assembly. If the shaft coupling will be above the top of the tube adjusting nipple, the headshaft should be set aside for installation after the discharge head is installed.

d. Inspect the turned section of the tube adjusting nipple for dirt, rust, nicks or scratches. Clean and polish with fine emery cloth as necessary to provide a smooth surface. Proceed with assembly by the same methods used for previous lengths and clean the "O-ring" groove.

Installing discharge head:

a. The discharge head assembly will consist of the following major parts (Figure 24):
   i. The discharge head may have either a flanged or threaded connection for attaching the column pipe to the bottom of the head.
   ii. A tension plate is used for centering and sealing the enclosing tube.
   iii. A tension nut used to support the enclosing tube, provides the top bearing.
   iv. The headshaft assembly connects the lineshaft to the driver.

b. A discharge head using screwed top column flanges are suggested for easier installation.

c. Lubricate tension nut threads, then lower into place and screw in tube adjusting nipple until tension nut comes down snug against tension plate. Continue to tighten tension nut until all of the enclosing tube is supported by the discharge head and is in slight tension. Proper tension is achieved by noting a gradual increase in the force needed to tighten the tension nut; when proper tension is reached, there will be a sudden increase in the

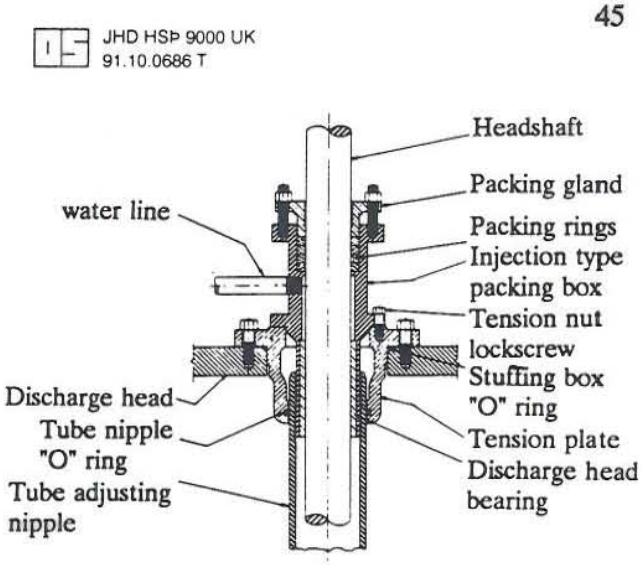

FIGURE 25: Water lubrication assembly (Peabody Floway)

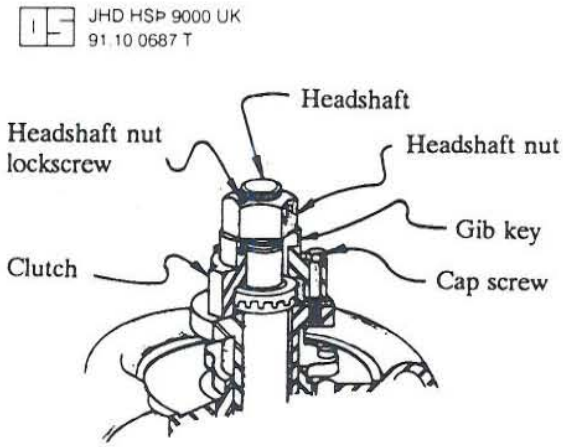

FIGURE 26: Hollow shaft driver clutch (Peabody Floway)

tightening force required. The approximate amount that the enclosing tube must be raised must be approximately = **0.03 (feet of column assembly)^2 10^{-3} (inches)**

Installing water lubrication systems:

The installation of units using water injection lubrication system (Figure 25) in geothermal wells use filtered fresh hot water for injection lube at **0.2-0.3 l/s (3-5 gpm) at 2.5-3.5 bar (40-50 psig)** for lubricating the lineshaft bearings. Note that the by-pass ports of bowl unit are open.

Installing hollow shaft driver:

The electric driver (Figure 26) should be checked for rotation before installation. The driver must be rotated counter-clockwise, when looking down at the top end of the motor. To change the direction of rotation on a three phase motor, interchange any two line leads. Raise and center the driver over the pump carefully until it sits on the discharge head and then tighten the mounting bolts. Check that the alignment of the shaft will center in the shaft clearance space at the top of the driver. If the shaft is not centered, place shims or leveling wedges between the discharge head and the foundation until the shaft is centered.

Impeller adjustment hollow shaft driver:

When using a hollow shaft driver impeller adjustment is accomplished at the top of the driver by the following procedure. The driver canopy will have to be removed before beginning. The headshaft nut screws down (Right hand threads), until impeller are just lifted off their seat and the shaft will rotate freely. When semi-open impellers are used, the correct determination of the point where the impellers just barely clear their seat is very important for proper adjustment. Adjust impellers as calculated before and according to shaft elongation graph. Lock headshaft nut with lockscrew inserted down through hole in headshaft nut and threaded into driver clutch.

Discharge piping:

The discharge piping must be installed so that stresses are not transmitted to the pump. A fixible connector installed next to the discharge head is highly recommended. Do not use the discharge head to support the piping and fittings, and support with blocks or saddles.

Capillary tube installation and use:

The capillary tube installed with the pump (Figure 27) is a simple and effective method of determining the depth of water in the well. Steel, stainless steel or plastic tubing can be use for the capillary tube. The lower end of tube is near the bottom of the pump. For reliable reading, the capillary tube should extend 7 m below lowest water level if possible. The upper end of the capillary tube is connected to a gauge and snifter valve. The exact vertical length of the capillary tube must be recorded (L). A tire pump or air compressor is used to expel all water from the tube; when this point is reached the gauge reading will remain constant. The maximum maintained pressure is equal to the height of the water above the end of the capillary tube (D). The water level (below the surface) is obtained by subtracting "D" from "L" ($WL = L - D$ or $WL = L - P\rho g$).

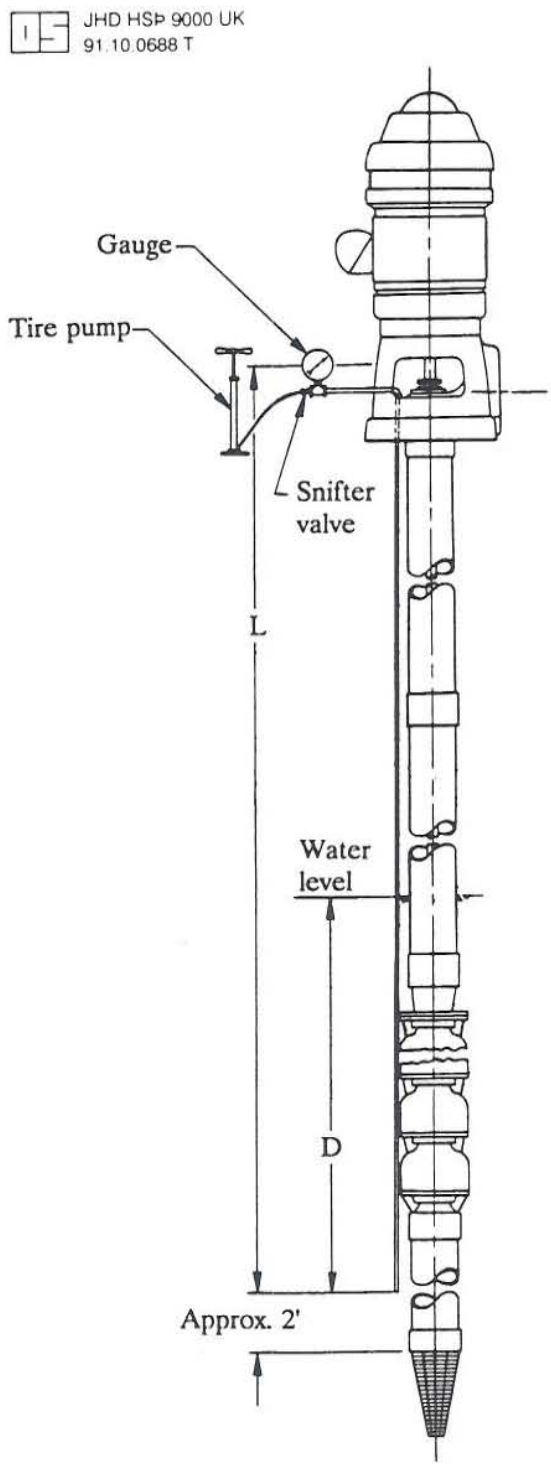

FIGURE 27: Air line installation and use (Peabody Floway)

6. CONCLUSIONS

1. The study describes three methods of pump testing a geothermal well, the procedure to be followed and the data collection required.

2. Interpretation of pump test results is described and a simplified analytical model used for pump selection.

3. A systematic procedure (flow diagram) for pump design and selection is presented.

4. Engineering equations used in detailed pump design are identified and explained, which will enable their use for other conditions than for the example used in this study.

5. The influence of variable water tables on pump output is analyzed and a pump selected accordingly.

6. The use of hydraulic balanced impellers is demonstrated for reducing shaft elongation.

7. Impeller adjustment calculations are made taking into account the different forces affecting shaft and column elongation.

8. The material selection for geothermal lineshaft pumps is described.

9. Proper procedures for installing and pulling a deep well pump are given. Critical items are identified such as shaft straightness and bearing clearances.

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NOMENCLATURE

A	= Initial water level	m
A_A	= Power loss in lineshaft	HP
A_D	= Power required of pump	BHP
A_L	= Power loss in motor thrust bearings	HP
A_M	= Power required by motor = $A_A + A_L + A_D$	BHP
A_N	= Power consumption of motor	HP
B	= Laminar drawdown coefficient	m/(l/s)
C	= Turbulent drawdown coefficient	$m/(l/s)^2$
C_1	= Well drawdown coefficient	$m/(l/s)^2$
C_2	= Coefficient of head loss in column	$m/(l/s)^2$
c	= Compressibility	Pa^{-1}
DDI	= Inner diameter of column	in
DDU	= Outer diameter of column	in
D_O	= Diameter of pump shaft	in
DSI	= Inner diameter of enclosing tube	in
DSU	= Outer diameter of enclosing tube	in
E	= Voltage per leg applied to motor	volt
E_A	= Elongation of shaft due to hydraulic thrust	mm
E_E	= Relative movement of impeller	mm
E_R	= Total elongation of column	mm
E_{R1}	= Elongation of column due to total pressure head	mm
E_{R2}	= Shortening of column due to inside pressure	mm
f	= Coefficient of mechanical friction in lineshaft bearings	HP/L=100 m
g	= Acceleration of gravity	m/s^2
H	= Water level in the well	m
ΔH	= Well drawdown	m
h	= Reservoir thickness	m
h	= Pumping submergence	m
h_{min}	= Minimum submergence of pump in water	m
I	= Amperes per leg applied to motor	ampere
K	= Thrust constant	lb/ft
K_D	= Portion of pump below water level	m
K_N	= Well drawdown	m
K_V	= Total static water level	m
K_{V1}	= Elevation above sea level to well head	m a.s.l.
K_{V2}	= Elevation below sea level to static water level	m b.s.l.
k	= Permeability of the reservoir	m^2
L	= Pump setting level = $K_V + K_N + K_D$	m
L_{min}	= Minimum length required of column	m
n	= Rotation of motor	rpm
$NPSHR$	= Net positive suction head required	m
P_{min}	= Pressure minimum required	bar
P_a	= Vaporize pressure of water	bar
P_D	= Total pressure head	m
P_d	= Dynamic pressure in column	m
P_F	= Pressure loss in column	m
Pf	= Power factor of motor	-
P_H	= Discharge head pressure	m
P_u	= Atmospheric pressure	bar

Q	= Well discharge	l/s
Q_B	= Bast point efficiency of discharge	l/s
Q_t	= Discharge from or injection into the well being tested	l/s
r_w	= Radius of discharge well	m
r_1	= Distance between the discharge and observation well	m
S	= Storativity	m/Pa
T	= Transmissivity	m^3/Pa.s
T_A	= Hydraulic thrust on pump shaft	N
T_I	= Weight of impeller	N
T_R	= Hydraulic thrust on column	N
T_S	= Weight of pump shaft	N
T_T	= Total thrust on pump shaft	N
t	= time, since well test started	sec
V	= Velocity of water in column	m/s
Y_{11}	= Coefficient of pump curve ($Q < Q_B$)	m/(l/s)
Y_{21}	= Coefficient of pump curve ($Q < Q_B$)	m
Y_{12}	= Coefficient of pump curve ($Q \geq Q_B$)	m/(l/s)
Y_{22}	= Coefficient of pump curve ($Q \geq Q_B$)	m
Z	= Number of pump impeller	stage

Greek letters:

η_M	= Efficiency of motor	%
η_D	= Efficiency of pump	%
μ	= Dynamic viscosity of the reservoir fluid	Pa.s
ρ	= Density of fluid	kg/m^3
ρ_t	= Density of fluid at T°C	kg/m^3
ϕ	= Porosity of the reservoir rocks	-

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