# Geothermal Recourses Council Annual Convention <br> August, 2004 Indian Wells, CA <br> Down Hole Geothermal Pump Seminar Monitoring, Evaluating and Optimizing Pump Selections by Jack A. Frost 




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# Monitoring, Evaluating and Optimizing Down Hole Geothermal Pump Performance by Jack A. Frost, Frost Consulting 


#### Abstract

Geothermal downhole pumps are typically monitored at least twice per day in the industry. Evaluation of this data can determine wear patterns and the economics of when a given pump should be pulled for service and /or pump (bowl assembly) replacement. In many cases, however, evaluation of this data plus that of simple field testing of a given pump in a given well may allow for the pump and well to be optimized. That is, the installation of a pump of increased flow, and subsequently increased revenue, or simply a more efficient pump can be selected for the characteristics of its given well.

\section*{Preface}


Before any meaningful analysis, calculations or determinations can be made regarding the field performance of a pump, the following details are of concern and must be addressed:

- Pump and well at thermal equilibrium
- Calibration of all instrumentation

1) Pressure gauges --- pump surface discharge pressure, nitrogen / bubbler pressure gauge, annulus pressure gauge.
2) Flow meter --- generally, the greatest single cause of erroneous readings; a calibrated (and maintained) Venturi or orifice plate should only be considered.
3) Motor control center (MCC) read-outs of amperes and voltage

- Flooded lubestring with properly selected oil (Unocal 76Steaval D150 recommended) and supply (3-5 gallons per day)
- Impellers properly spaced and hot checked
- Bubble tube pressure (as well as surface discharge pressure) well above brine flash point pressure as tested by a gas bomb.


## Data Collection ... See Figure 1

Collect the following data (simultaneously) as a "snap-shot" while the pump is in operation.

- Surface discharge pressure, Pd (psig)
- Bubbler (nitrogen) pressure, Pb (psig)

NOTES:

1) Record reading after freshly purging line first
2) Plugging of bubble tube is obvious. A leak in the bubble line reacts quite differently. A leak above the water level and the pressure will settle out to that of the annulus pressure. A leak below the water level will quickly settle down after purging and then bleed off slowly to another value.
3) The depth of the bubble tube installed, $\mathbf{L b}$ (ft), must be a known value. Generally, strapped-on continuous tubing terminates just above the pump bowls. That is, Lb is the same as the pump setting depth.

- Annulus pressure, Pa (psig)
- Flow meter reading, GPM (US gallons per minute)
- Temperature (degrees F)
- Amperes on all 3 legs


## NOTE:

Average out the values of all three legs unless one leg illustrates imbalance. If one leg is imbalanced, disregard its value and average the other two legs.

- Voltmeter read out NOTE:
If no voltmeter is available, assume, typically, 4160 volts.
- Note any miscellaneous mechanical concerns or other oddities observed during operation (examples)

1) Sand, scale, etc. present in brine production
2) High motor winding or (thrust) bearing temperatures
3) Noise and / or vibration (usually can be attributed to motor)
4) Shaft run-out
5) If lube oil pressure equals pump discharge pressure, a leak in the lubestring is present.
6) If annulus pressure equals pump discharge pressure, a leak in the (typically 9 5/8 API) pipe is present above the water level.

## Pump Terminology- See Figures 1, 2, \& 3

- TDH (ft) - Total Dynamic Head of complete pump over its setting depth at a given flow; also referred to as "Pump Head"; the "Bowl Head" (ft) of the pump at a given flow less the friction losses attributed to water moving up within the pipe and lubestring.
- Total BHP - Total Brake Horsepower of the complete pump at a given flow; also referred to as "Pump BHP"; "Bowl BHP" plus lineshaft mechanical friction loss (hp) plus motor thrust bearing losses (hp). (The typical 2 3/16 dia. lineshaft at 1785 rpm has losses of 4.66 hp per 100 ft . pump setting while motor thrust bearing losses are typically only $3.5-5.0 \mathrm{hp}$ ).
- Specific Gravity - Specific Gravity ( $\mathbf{S p} \mathbf{G r}$ ) is a unitless value of density where water is arbitrarily assigned 1.0 Specific Gravity at 70 F and at sea level. (Example via Steam Table: Water @ 340F has a Specific Gravity of 0.8964). For purposes of calculations, assume that the Specific Gravity of the brine is that of values from that of Steam Table
- Static Water Level (ft) - Vertical distance to water level under static conditions; may also be expressed in psig; an artesian well would, therefore, be expressed in a negative value.
- Dynamic Water Level (ft) - Vertical distance to water level at a given flow rate; may also be expressed in psig (see "Lift").
- Drawdown (ft) - Dynamic water level minus Static Water level; often, however, expressed in psig.
- Productivity Index (psig) - The increase in flow for a given increase in drawdown, typically expressed in psi per GPM but may also be expressed in ft per GPM; commonly denoted as PI.
- Submergence (ft) - Vertical distance from (typically dynamic) water level to end of bubble tube; actual submergence to pump suction is greater by the length of the bowl assembly ("pump") with the bubble tube depth same as pump setting depth.
- Lift (ft) - Vertical distance from (typically dynamic) water level to surface.
- System-Head Curve (ft) - "A pump will always operate at the point at which it intersects the System-Head curve" ... this is true whether the pump is worn, mismatched to the well or if the pump is throttled; In somewhat simplistic terms, the System-Head curve can be drawn / determined by the pump surface discharge pressure (converted to feet of head) plus that of the Dynamic Water Level at variously increasing flow rates




## Equations

$$
\mathrm{psi}=\frac{(\mathrm{Head})(\mathrm{Sp} \mathrm{Gr})}{2.31} \quad \text { Head }(\mathrm{ft})=\frac{(\mathrm{psi})(2.31)}{\mathrm{Sp} \mathrm{Gr}}
$$

$$
\mathrm{TDH}^{*}(\mathrm{ft})=\frac{(\mathrm{Pd}-\mathrm{Pb})(2.31)}{\mathrm{Sb} \mathrm{Gr}}+\mathrm{Lb}
$$

Assume Specific Gravity same as Steam Table values

$$
\operatorname{Lift}(\mathrm{ft})=\mathrm{Lb}-\frac{(\mathrm{Pb}-\mathrm{Pa})(2.31)}{\mathrm{Sp} \mathrm{Gr}}
$$

$$
\text { Submergence }(\mathrm{ft})=\frac{(\mathrm{Pb}-\mathrm{Pa})(2.31)}{\mathrm{Sp} \mathrm{Gr}}
$$

Total BHP** $=\underline{1.73(\mathrm{amps})(\mathrm{volts})(\mathrm{Me})(\mathrm{Mpf})}$ 745.7

|  | Typical Downhole Geothermal (Lineshaft)Pump Motor |  |  |
| :---: | :---: | :---: | :---: |
|  | Typical (new) 800 HP 4160 V Motor <br> Full Load Amps (100\%) = 103 1785 Full Load RPM |  |  |
|  | Load | Motor Efficiency Me (\%) | Motor Power Factor Mpf (\%) |
| See Terminology |  |  |  |
| * also referred to as "Pump Head" <br> ** also referred to as "Pump BHP" | 115 | 91.1 | 87.5 |
|  | 100 | 92.4 | 87.5 |
|  | 75 | 92.6 | 86.0 |
|  | 50 | 92.6 | 80 |
|  | Rebuilt, old and /or rewound motors ... deduct 1-2 points |  |  |
|  | "Thumbnail" Calculation" |  |  |
|  | $\text { Total Pump BHP }=\frac{(\text { field amp average })(800)}{102 ~ f 1}$ |  |  |

## Evaluation of Field Performance

Again, calibration of field instrumentation, particularly that of pressure gauges and an accurate flow meter, cannot be overstated.
Calculate TDH (Pump Head) and Total BHP (Pump BHP) from collected "snap-shot"field data. Assume the flow meter as accurate. Compare the calculated TDH with that of the pump performance curve Pump Head at the flow meter reading. Compare the calculated Total BHP to that of the pump performance curve Pump BHP at the flow meter reading.
Assuming all instrumentation is accurate:, no flashing is occurring, proper oil lubrication, and impellers are spaced properly:

- Tolerances: Typically, $3 \%$ of head at flow (up to 2000 GPM) with the pump operating within +/- $25 \%$ of peak efficiency flow rate; greater tolerances are to be expected beyond 2000 GPM or with "right-hand side" operation. Total BHP typically is within $5 \%$ with same noted limitations as head.
- Other causes for discrepancies:
a) Sand, scale or other particles that may "drag" or increase horsepower
b) Plugging of slotted liner strainer and / or suction
c) Plugging or erosion at flow meter
d) Leak in (9 5/8 API) pipe .... not common
e) Motor speed less then $1785 \mathrm{rpm} . .$. not common and should be voltage apparent
f) Mechanical wear of the pump itself .... See Figure 3


## Optimizing Pump Selection

- Obviously, for example, a pump is mismatched to a well if:
a) A 1000 GPM pump is operating at 1500 GPM ... past the most efficient flow rate ("right-hand side operation). In such a case, the properly selected pump would be simply a pump with its most efficient point near or at 1500 GPM with a TDH the same as the 1000 GPM pump operating at 1500 GPM.
b) A pump is throttled to prevent flashing / cavitation. In such a case, the properly selected pump would be that of a pump with its most efficient flow rate at or near the throttled GPM and its TDH calculated from the Lift at the throttled GPM but with its discharge pressure $(\mathrm{Pd})$ reduced to what is required if the pump were not throttled
- Other reasons for optimization:
a) If a given well has good temperature and productivity, can a higher flow pump (and/or increased setting depth) be selected?
b) If a given well is relatively cool and is lowering the overall plant inlet temperature, can a lower flow pump be selected?
- Considerations which may limit pump selection: Motor HP and thrust bearing ratings, lineshaft combined stress (HP limitations), relative shaft stress versus lateral and depth to liner hanger.
- In the case of a hotter well where increasing the flow is desirable, a Multiple Point Field Test is necessary. The following example illustrates how such an optimal pump selection may be made. See Figure 4.

a) Given data

| Current Pump / Well Operation |  |  |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Existing 12 in. 27 Stage Low Flow Pump Model set at 1400 ft / / 800 HP-4160 Volt Motor |  |  |  |  |  |  |  |  |
| $\begin{gathered} \hline \mathrm{Pd} \\ (\mathrm{psig}) \\ \text { wide } \\ \text { open } \end{gathered}$ | $\begin{gathered} \mathrm{Pb} \\ \text { (psig) } \end{gathered}$ | $\begin{gathered} \mathrm{Pa} \\ (\mathrm{psig}) \end{gathered}$ | Lb pump setting depth (ft) | GPM | $\begin{gathered} \begin{array}{c} \text { Temp } \\ \text { (F) } \end{array} \\ \hline \text { Sp Gr } \end{gathered}$ | Flash Point via Gas Bomb Test $(p s i g)$ | Depth (minus KB) to Liner Hanger | Average Ampere (BHP) |
| 195 | 224 | 95 | 1400 | 1025 | 350 | 178 | 1700 | 59.6 |

b) From the above data, the following has been calculated and the pump is found to be operating per its pump performance curve within 1\% of both TDH and BHP at 1200 GPM. Lift (ft) and Submergence (psi and ft ) are also calculated for reference.

| TDH <br> $(\mathrm{ft})$ | Total (Pump) BHP | Submergence <br> $(\mathrm{psi})$ | Submergence <br> $(\mathrm{ft})$ | Lift <br> $(\mathrm{ft})$ |
| :---: | :---: | :---: | :---: | :---: |
| 1325 | 463 | 129 | 332.5 | 1067.5 |

c) From the above data, it is readily apparent that this well could be drawn down further and subsequently produce greater flow ... 224 psig Pb (versus 178 psig Flash Point) at 1400 ft . Lb. How much more flow may be obtained requires a Multiple Point Test and establishing the Productivity Index (PI).The immediately apparent limitations are that of the existing motor HP and the depth to the liner hanger. (Again, even if the existing pump is worn, testing results are still valid to the limits that the pump can produce).

| Multiple Point Test to determine Productivity Index (PI) |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| Existing 12 in. 27 Stage Low Flow Pump Model set at 1400 ft . / 800 HP 4160 Volt Motor |  |  |  |  |  |  |
| Point | $\begin{aligned} & \mathrm{Lb} \\ & (\mathrm{ft}) \end{aligned}$ | Temp <br> (F) | GPM | $\begin{gathered} \mathrm{Pb} \\ \text { (psig) } \end{gathered}$ | $\begin{gathered} \mathrm{Pa} \\ \text { (psig) } \end{gathered}$ | NOTES |
| 1 | 1400 | 350 | 800 | 260 | 95 | Valve throttled - Pump on-line |
| 2 |  | 350 | 900 | 244 | 95 | Valve throttled - Pump on-line |
| 3 |  | 350 | 1025 | 224 | 95 | Valve wide open - Pump on-line / Operational Point |
| 4 |  | 350 | 1100 | 212 | 95 | Pump by-passed directly to injection well so as to lower discharge pressure / increase flow |
| 5 |  | 350 | 1200 | 196 | 95 | Pump by-passed directly to injection well so as to lower discharge pressure / increase flow |
| 6 |  | 350 | 1300 | 180 | 95 | Pump by-passed directly to injection well so as to lower discharge pressure / increase flow NOTE: Pb just above Flash Point. |

d) From the above test data, a Productivity Index (PI) of 16 psi per 100 GPM (or 0.16 psi / GPM) drawdown is established and is linear out to at least 1300 GPM. For purpose of calculations, the drawdown shall be assumed as linear beyond 1300 GPM. No testing beyond 1300 GPM is considered as the existing pump set at 1400 ft . would be subject to flashing.
e) Two additional concerns:
o An ample safety factor must be maintained above the Flash Point (178 psig as determined by a gas bomb test) and 205 psig has been selected. As the liner hanger depth (less KB) will allow a deeper set pump (and Lb), a new optimal setting depth of 1560 ft . is selected. Therefore, for optimal pump considerations, calculations of head (TDH) will need to be based upon 205 psig Pb at 1560 ft . Lb.
0 With an increase in flow into the line leading into the plant, a greater pump discharge pressure will be needed to overcome the additional pipeline losses. An additional 15 psi is determined to overcome such losses out to a total flow of 1600 GPM. Therefore, for optimal pump considerations, calculations of head (TDH) will need to also be based upon a pump discharge pressure (Pd) of 210 psig.
f) Calculations for Optimal Pump Sizing

$$
\mathrm{TDH}(\mathrm{ft})=\frac{(210 \mathrm{psig} \mathrm{Pd}-205 \mathrm{psig} \mathrm{~Pb})(2.31)}{0.896 \mathrm{Sp} \mathrm{Gr}}+1560 \mathrm{ft} \mathrm{Lb}=1573 \mathrm{ft} .
$$

Assuming linear drawdown out beyond 1300 GPM

$$
\begin{aligned}
& \frac{(1560 \mathrm{ft} \text { new Lb }-1400 \mathrm{ft} \text { old Lb) }(0.896 \mathrm{Sp} \mathrm{Gr})}{2.31}+224 \text { old } \mathrm{Pb}-205 \text { new } \mathrm{Pb} \\
& =81 \text { psi additional drawdown }
\end{aligned}
$$

$$
\text { which translates into } \frac{81 \mathrm{psi}}{0.16 \mathrm{psi} / \mathrm{GPM} \mathrm{PI}} \simeq 500 \text { additional GPM }
$$

$$
\text { or } 1025 \text { GPM + } 500 \text { GPM }=1500 \text { GPM anticipated }
$$

Therefore, our optimal pump selection sizing is 1500 GPM @ 1573 ft . TDH with 1560 ft . Setting (Lb)
g) The optimal pump selected for the new rating point is as follows with BHP and thrust within capabilities of existing motor.


| Optimal Pump Selection |  |  |  |  |  |  |  |  |
| :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: | :---: |
| 12 in. 29 Stage "Medium" Flow Pump Model set at 1560 ft. / 800 HP 4160 Volt Motor |  |  |  |  |  |  |  |  |
| $\begin{gathered} \mathrm{Pd} \\ (\mathrm{psig}) \\ \text { wide } \\ \text { open } \end{gathered}$ | $\begin{gathered} \mathrm{Pb} \\ (\mathrm{psig}) \end{gathered}$ | $\begin{gathered} \mathrm{Pa} \\ (\mathrm{psig}) \end{gathered}$ | Lb pump setting depth (ft) | $\begin{gathered} \mathrm{TDH} \\ (\mathrm{ft}) \end{gathered}$ | GPM | $\begin{aligned} & \text { Amps } \\ & \text { (BHP) } \end{aligned}$ | Temp <br> (F) <br> Sp Gr | Flash Point via Gas Bomb Test (psig) |
| 210 | 205 | 95 | 1560 | 1573 | 1500 | $\begin{aligned} & 101.5 \\ & (789) \end{aligned}$ | $\begin{gathered} 350 \\ \hline 0.896 \end{gathered}$ | 178 |

h) Each power plant needs to make its own evaluation as to the worth of revenue versus the cost and parasitic loads associated with such an optimization. With the example illustrated such considerations would include the following:

|  | Existing Pump | Optimal Pump |
| :---: | :---: | :---: |
| GPM | 1025 | 1500 |
| BHP load | 463 | 789 |
| Revenue produced <br> (annual) | $\operatorname{Re}$ | Ro |
| Parasitic cost* <br> (annual) | (Cost in \$ per kw-hr) $(3024469.7)=\mathrm{Pe}$ | (Cost in \$ per kw-hr) (5.154009.9) = Po |
| Net <br> (annual) | Re - Pe | Ro - Po |
| * Assumes continuous operation --- 8760 hrs per year |  |  |

- Submersible Pump versus Lineshaft Pump
a) Although the before mentioned calculations are based upon the industry standard lineshaft pumps, the same principles and calculations apply to that of a submersible pump.
b) Other comments regarding submersible versus lineshaft pumps are as follows:

1) Submersible pumps, being smaller in diameter and with "in the hole" physical size limitations of the motor, are less efficient then lineshaft pumps.
2) Consider pump longevity into that of optimal pump selection. Submersible pumps, again generally being of smaller diameters, operate at much greater speeds then that of ( 1785 rpm ) lineshaft pumps. An old pump axiom: Given two pumps both performing the same work but of two different speeds, the higher speed pump will wear out before the slower speed pump by the square to the cube of the ratio of their speeds. For example consider two pumps one operating at 1785 rdm and the other operating at 3550 rpm .

$\left[\frac{3500 \mathrm{rpm}}{1785 \mathrm{rpm}}\right]^{2-3} \simeq 3.8$ to $7.5 \quad$| That is, the higher speed pump will wear out, on |
| :--- |
| average, four to seven times faster then that of |
| the slower speed pump. |

