Aquifer seasonal thermal energy storage (ASTES) is considered in conjunction with residential size water source heat pumps and solar collectors. The system investigated consists of a set of solar collectors, one confined aquifer which has two wells (injection and production), and numerous heat pumps attached by a distribution network. A computer model of the system is developed and yearly simulations are made to determine the influences of several parameters on overall system performance. Energy consumed by pumping equipment to inject and withdraw fluid from the aquifer and circulate fluid through a recharge system is included in the overall system performance.

Parameters investigated include aquifer permeability and dimensions, safety factor, number of residences, storage temperature and water temperature drop. Of these parameters, the water temperature drop was identified as an important factor. It is a major predictor of the fluid flow rate which in turn affects heat pump design and aquifer performance.

Ranges of the water temperature drop where the overall system performance is high enough to constitute energy feasibility and too low for energy feasibility to exist are discussed for several different values of the above parameters. In general, any parameter change that decreases the pumping power without decreasing the heat pump evaporator temperature
will improve the overall system performance and expand the region of energy feasibility.

These results indicate that in designing heat pumps for use with ASTES, the temperature drop across the evaporator should be large compared to a value of \( \sim 4 \) C which is the present norm for water source heat pumps.

An economic analysis is also performed to determine if an ASTES system can compete with conventional heating systems. A present worth analysis is used to obtain an accurate comparison of the different systems. The analysis indicates that the ASTES system can easily compete with electric resistance furnaces, but only under limited conditions is it able to compete with natural gas furnaces. It is totally unable to compete with air source heat pumps. If solar collectors are replaced with an alternative recharge system, such as waste thermal energy, the ASTES system becomes substantially cheaper than electric resistance furnaces and very competitive with natural gas furnaces, but only under limited conditions is it able to compete with air source heat pumps.
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Date Thesis is presented: January 22, 1981

Typed by Sandy Orr for William R. McMahon
ACKNOWLEDGEMENTS

There are several people who helped to bring about the completion of this thesis and are due recognition and a special thanks.

I would especially like to thank Dr. Gordon M. Reistad for the many hours of input and guidance he has so generously given and the research funds he conveyed which kept my family afloat.

I would like to express my gratitude to my wife, Pam, for the editing and her continual support. I am forever indebted to my parents, John and Shannon, for their years of encouragement, which enabled me to advance to this point.

I am grateful to my fellow graduate students for countless hours of discussion which are so important in any learning process. Special thanks to my office mates, Masao Fukuda and Nozar Jafarey, for their advice and concern.

I thank Dr. James R. Welty for providing an office the last term of my work and Battelle Northwest Laboratories for funding this research.
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ANALYSIS OF HEAT PUMP COUPLED AQUIFER
SEASONAL THERMAL ENERGY STORAGE SYSTEMS

I. INTRODUCTION

In a period of decreasing fossil fuel reserves and increasing energy demands it is essential to use energy efficiently and at the same time develop renewable energy sources. The sun supplies large quantities of renewable energy, but the utilization of it is hampered because of its low energy density and intermittent character. Heat pumps are capable of utilizing this low grade energy thereby making electricity a more efficient means of space heating. Unfortunately, large heat loads usually exist when energy is least available. Storage has been incorporated to circumvent the sun's intermittent character, however cost and size limitations have only permitted small storage volumes to be constructed economically.

In areas of temperate climates where there is an appreciable amount of solar energy in the summer and also a substantial heating load in the winter, seasonal storage systems appear to have considerable potential. Porous underground formations in which water can be stored and transmitted (aquifers) have been proposed as a means of long term, low cost, thermal energy storage. Seasonal storage in confined aquifers has additional advantages in that they are bound above and below by impermeable clay layers. The relative low heat conductivity of the clay layers makes confined aquifers very large, naturally occurring, storage vessels. These storage systems will be referred to as aquifer seasonal thermal energy storage.
This thesis considers aquifer seasonal thermal energy storage in conjunction with solar collectors and residential size water source heat pumps. The system being investigated is illustrated in Figure 1.1. It contains a set of solar collectors, one confined aquifer which has two wells (injection and production), and numerous heat pumps attached by a distribution network. The system can be subdivided into five subsystems; aquifer, collectors, distribution network, load, and heat pumps. Each subsystem is discussed and its performance specified.

A computer model of the system was developed and yearly simulations were made to determine the influence of several variables on overall system performance. Variables investigated include aquifer permeability and dimensions, safety factors, number of residences, storage temperature \( T_{\text{STOR}} \), and water temperature drop \( \Delta T_W \). The system's overall performance (COPH) includes energy consumed by pumping equipment to inject and withdraw fluid from the aquifer and circulate fluid through the collectors. The primary design variables whose influence on overall system performance was investigated were \( T_{\text{STOR}} \) and \( \Delta T_W \). Ranges of \( \Delta T_W \) and \( T_{\text{STOR}} \) where these systems exhibit optimum performance and potential energy feasibility are discussed.

Other authors have investigated seasonal storage in aquifers and its use in conjunction with solar systems and/or heat pumps, but their emphasis has been on describing a specific system and not on the influence of design variables.

An economic analysis was also performed to determine if seasonal thermal energy storage systems can compete with conventional systems. To be an economically viable alternative to conventional systems, reduced annual energy cost, as a result of increased system performance must
Figure 1.1. Schematic of the aquifer seasonal thermal energy storage (ASTES) system as modeled.
offset higher initial cost. A present worth analysis was used to obtain an accurate comparison of the different systems.
II. AQUIFER OVERVIEW

The aquifer is the major subsystem to be considered which is not readily characterized by a refined, widely accepted model. Thus, before considering the overall system under investigation, aquifer thermal processes will be discussed along with analytical and numerical modeling of them. As mentioned before, an aquifer is a porous underground formation in which water can be stored and transmitted. The aquifers being considered in this study are assumed to be confined in that they are bound above and below by an impermeable clay layer (Figure 2.1). It is also assumed that the natural flow through the aquifer is neglectable.

If the aquifer is initially at some constant temperature and hot water is caused to flow through the porous medium, a temperature front will begin to move through the confined region similar to "slug flow" [25,27]. The rapid transfer of energy from the fluid to the rock, due to small rock size, is responsible for the formation of a temperature front. The velocity at which the temperature front moves through the confined region \(v_{\text{thermal}}\) is a function of the rocks' density \(\rho_R\) and heat capacity \(C_R\), the water's density \(\rho_W\) and heat capacity \(C_W\),
fluid velocity \( (v_f) \), and aquifer porosity \( (\phi) \), defined as the ratio of void space to the total volume of the mass. It is given by the following equation:

\[
\frac{\rho_w C_w \phi}{\rho_w C_w \phi + \rho_r C_r (1-\phi)} v_f.
\] (2.1)

Typical aquifers with a porosity of 0.20, as assumed in this thesis, have a ratio of \( v_{\text{thermal}} \) to \( v_f \) of 0.292. The temperature front travels over three times slower than the fluid front because most of the aquifer is rock (80%) and before the temperature front can advance, the rock must reach the injected temperature. As a result, most of the energy is stored in the rock with only a limited portion stored in the fluid.

In actual aquifers, the temperature front is somewhat diffused due to finite heat transfer processes and losses to the confining layers. The plausible progression of a temperature front moving radially outward through a confined aquifer from a wellbore is shown in Figure 2.2. Isotherms are shown as a function of the radial distance from the wellbore hole center line for a constant injection temperature of 1.0 and original aquifer temperature of 0.0. The time elapsed from the onset of injection (injection time) increases going from Figure 2.2(a) to 2.2(c).

The temperature front in Figure 2.2(a) is quite sharp and vertical. As injection time increases the temperature front broadens and tilts (Figures 2.2(b) and (c)). Heat transfer processes are not instantaneous thus finite temperature gradients exist at the temperature front. As the front moves outward the confining layers are warmed. Since temperature gradients are largest at the front, much of the heat transferred to the confining layers does so as the front passes. These effects broaden

\[------------------------
1 Aquifer parameters and values assigned to them can be found in Chapter 4.
------------------------\]
Figure 2.2. Hypothetical progress of a temperature front through a confined aquifer. Isotherms are shown as a function of the radial distance from the wellbore hole centerline for a constant injection temperature of 1.0 and original aquifer temperature of 0.0.
the temperature front as it travels outward from the wellbore. While energy is transferred from the fluid to the rock (at the front), buoyancy effects induce the cooling fluid to drift downward and the warm fluid to drift upward. This causes the temperature front to tilt. If injection continues long enough, eventually it becomes impossible to distinguish a front and the aquifer temperature will decrease gradually as "r" increases.

If the flow direction within the aquifer is reversed by producing (withdrawing) hot water from a well that was previously used for injection, then the temperature front begins to progress back toward the wellbore. Thermal losses and buoyancy effects still exist and consequently the temperature front continues to diffuse and tilt. Since most of the injected energy is stored in the rock, it must be retrieved from the rock during production.

The first water removed during production will be at the injected water temperature. As more water is produced its temperature will decrease and if production continues long enough, the water temperature will eventually reach the original aquifer temperature. In most practical situations production is halted before the aquifer reaches its original temperature. For seasonal energy storage there will be one injection and one production period per year (injection-production cycle). Some energy will be left in the aquifer, upon completion of production, when the same fluid volume is produced as was injected. This will result in less losses for the next injection-production cycle with more of the injected energy recovered. After several injection-production cycles have been completed a steady-state will be reached in the aquifer and further cycles will recover essentially the same amount of energy.
2.1 ANALYTIC SOLUTIONS

Analytical models of simple aquifers have been developed to predict aquifer performance. One such model was developed by Gringarten and Sauty [12] to investigate heat extraction from aquifers with uniform regional flow. The model consists of a horizontal confined aquifer of thickness $b$ with two wells (injection and production), separated by a distance $D$ as shown in Figure 2.3. The flow net shown is that of a recharging-discharging pair of wells in an aquifer having no areal flow. Assumptions regarding the model are listed below:

1) The total injection volume flow rate, $Q$, is equal to the total production volume flow rate and is assumed constant.

2) Initially, the water and rock in the aquifer and the cap rock and bed rock are at the same temperature, $T_\infty$.

3) At time $t=0$ the temperature of the water, at the point of injection, is set equal to $T_{\text{inj}}$ and is maintained constant thereafter.

4) Thermal equilibrium between the water and rock in the aquifer takes place instantaneously so that anywhere in the aquifer the rock has the same temperature as the surrounding fluid.

5) In the aquifer, the thermal conductivity is negligible in the horizontal direction.

6) The aquifer is assumed to be thin enough that the temperature is always uniform in the vertical direction.
Figure 2.3. Flow net formed by a recharging-discharging pair of wells in a hypothetical aquifer having no areal flow, from [19].
7) In the cap rock and bed rock the effect of the horizontal thermal conductivity is negligible, and the vertical thermal conductivity \(k_R\) is finite. The temperature at infinity remains constant and equal to the initial temperature.

8) The cap rock temperature at the interface of the cap rock and aquifer is assumed to be equal to the aquifer temperature at the interface.

9) The product of density and heat capacity for both the water and the rock, the cap rock vertical thermal conductivity, and the porosity, are all assumed constant.

In analyzing the aquifer each stream channel is treated independently. The temperature along a stream channel, bounded by the streamlines \(\psi\) and \(\psi+d\psi\), can now be represented by the function \(T(S,t)\) in the aquifer and the function \(T(z,S,t)\) in the cap rock where \(S\) is the stream channel area from the injection well (Figure 2.4). The differential equation governing the water temperature within a stream channel is obtained by doing an energy balance on an elemental section along the stream channel (Figure 2.5). Similar derivations for one-dimensional mass flow have been done by Lauwerier [16] and Arpaci [2]. The resulting equation is:

\[
\frac{b}{2} \rho_A C_A \frac{\partial T(S,t)}{\partial t} + \frac{m}{2} \rho_W C_W \frac{\partial T(S,t)}{\partial S} = k_R \frac{\partial T(z,S,t)}{\partial z} \bigg|_{z=0}
\]  

(2.2)

where \(m\) = volume flow rate within stream channel

\[
\rho_A C_A = \phi \rho_W C_W + (1-\phi) \rho_mC_m
\]

\(\rho_mC_m\) = heat capacity of aquifer matrix.
Figure 2.4. Stream channel used in the analysis of a confined aquifer.
Figure 2.5. Heat transfer energy balance on an elemental section along the stream channel.
The cap rock temperature is governed by the heat conduction equation which is as follows:

\[
\frac{\partial^2 T(z,S,t)}{\partial z^2} = \frac{\sigma_R C_R}{k_R} \frac{\delta T(z,S,t)}{\delta t}.
\]  

(2.3)

Equation 2.2 becomes a boundary condition for Equation 2.3 along with

\[
T(\infty,S,t) = T_\infty
\]

(2.4)

\[
T(0,0,t) = T_{\text{inj}}
\]

and the initial condition

\[
T(z,S,0) = T_\infty.
\]

(2.5)

Making the following variable substitutions:

\[
n = \frac{z}{2b}, \quad \zeta = \frac{k_R S}{b \rho W C_W}, \quad \tau = \frac{k_R t}{b^2 \rho A C_A}, \quad T - T_\infty = \theta,
\]

Equation 2.3, the boundary conditions, and the initial condition become:

\[
a \frac{\delta^2 \theta}{\delta n^2} = \frac{\delta \theta}{\delta \tau}
\]

(2.6)

\[
\begin{aligned}
\theta(n,\zeta,0) &= 0 \\
\theta(\infty,\zeta,\tau) &= 0 \\
\frac{\delta \theta}{\delta \tau} + m \frac{\delta \theta}{\delta \zeta} &= \frac{\delta T|_{n=0}}{\delta n} \\
\theta(0,0,\tau) &= T_{\text{inj}} - T_\infty = \theta_0
\end{aligned}
\]

(2.7)

where \( a = \rho_A C_A / 4 \rho_R C_R \).

The one-dimensional solutions are directly applicable, giving the following result [2,16]:
\[ \frac{\partial}{\partial \tau} (n, \zeta, \tau^*) = \begin{cases} 
0 & \tau < \zeta/m \\
\text{erfc} \left[ \frac{n + \zeta/m}{2(\alpha \tau^*)^{1/2}} \right] & \tau \geq \zeta/m 
\end{cases} \tag{2.8} \]

where \( \tau^* = \tau - \zeta/m \).

When only investigating the aquifer temperature, \( n = 0 \), and upon rearranging and substituting the original variables, Equation 2.8 becomes:

\[
\frac{T - T_{inj}}{T_{inj} - T_\infty} = \begin{cases} 
0 & t < \frac{S_{bp} A C A}{\rho_W C_w m} \\
\text{erfc} \left[ \frac{(\rho_w C_w)^2}{k R C_R} \left( \frac{m^2}{2} - \frac{S_{bp} A C A}{\rho_W C_w m} \right) \right] & t \geq \frac{S_{bp} A C A}{\rho_W C_w m} 
\end{cases} \tag{2.9} \]

The maximum area \( (S_{\text{max}}) \) of a flow channel in which the temperature front has influenced is defined as follows:

\[
S_{\text{max}} = \frac{t_{inj} \rho_w C_w m}{S_{bp} A^C A} \tag{2.10} \]

where all variables are as previously defined with \( t_{inj} \) being the injection time in seconds.

Substituting \( S_{\text{max}} \) into Equation 2.9 and rearranging gives:

\[
\frac{T - T_{inj}}{T_{inj} - T_\infty} = \begin{cases} 
0 & \frac{S}{S_{\text{max}}} > 1 \\
\text{erfc} \left[ \frac{b^2 C^2 A^2}{k R C_R t_{inj}} \left( \frac{S_{\text{max}}}{S} - \frac{S_{\text{max}}}{S_{\text{max}}} \right)^{-1} \right] & \frac{S}{S_{\text{max}}} \leq 1 
\end{cases} \tag{2.11} \]

The product of density and heat capacity of the aquifer and rock, and the thermal conductivity of the rock can be generalized for aquifers,
hence from Equation 2.11, the nondimensional aquifer temperature is only a function of the nondimensional flow channel area \((S/\text{S}_{\text{max}})\) and \(b^2/t_{\text{inj}}\). Notice that flow rate is not a variable in Equation 2.11. The flow rate along with injection time and aquifer thickness are required to size the aquifer, in that they determine \(S_{\text{max}}\). In Figure 2.6 nondimensional aquifer temperature is plotted as a function of nondimensional \(S\) for various \(b^2/t_{\text{inj}}\) values.

Aquifer temperature equals the injection temperature at the well \((S/\text{S}_{\text{max}}=0)\), then decreases slowly until \(S/\text{S}_{\text{max}}=1\) at which point it decreases quickly to zero. If \(b\) is held constant, \(b^2/t_{\text{inj}}\) varies inversely with \(t_{\text{inj}}\) thus at the beginning of injection \(b^2/t_{\text{inj}}\) will be large and the aquifer temperature profile is similar to the step function:

\[
\frac{T-T_{\infty}}{T_{\infty}-T_{\text{inj}}} = \begin{cases} 
0 & S/\text{S}_{\text{max}} > 1 \\
1 & S/\text{S}_{\text{max}} < 1 
\end{cases}
\]  

(2.12)

As injection continues \((b^2/t_{\text{inj}}\) decreases) the temperature profile in the aquifer smoothes out, dropping more rapidly at small \(S/\text{S}_{\text{max}}\) and less rapidly at \(S/\text{S}_{\text{max}}=1\).

The energy contained within the aquifer is proportional to the area below the temperature curve and the energy contained in the cap rock is proportional to the area above the temperature curve. Aquifer losses will be minimum as the temperature profile approaches a step-function. From Figure 2.6 it is seen that losses are minimized as \(b^2/t_{\text{inj}}\) increases, but the relative extent of the losses change little compared to the magnitude of change in \(b^2/t_{\text{inj}}\). At \(b^2/t_{\text{inj}}=0.001\) and \(S/\text{S}_{\text{max}}=0.8\)
Figure 2.6. Nondimensional aquifer temperature as predicted by Equation 2.11 is plotted as a function of nondimensional flow channel area for various \( \frac{b^2}{t_{\text{inj}}} \) values.
a one fold decrease in $b^2/t_{inj}$ only changes the aquifer temperature by 8.5 percent.

Establishing conditions during injection where losses are minimized also establishes minimum loss conditions during production. Since the production temperature is a major concern in this study the next step is to determine absolute values for the production temperature. An analytical solution which determines production temperatures for an injection-production cycle does not exist. Furthermore an analytical solution is not likely because of the necessity of using results from the injection solution as the initial conditions for the production solution. Numerical methods are required to pursue the problem further.

2.2 NUMERICAL SOLUTIONS

This section discusses numerical evaluation of the aquifer in two main ways. First, a finite-difference approximation to the analytical model just discussed is made. This formulation, with flow rate set constant, is compared with the analytical solution to determine its accuracy, and then used to evaluate results which can not be obtained with the analytical solution. Next, the status of numerical solutions as reported in the literature is presented and representative results discussed.

2.2.1 Solution of Analytical Equations

The numerical solution of Equation 2.3 subject to the boundary and initial conditions given by Equations 2.2, 2.4 and 2.5 was formulated using an explicit finite-difference method. The equations written in finite-difference form are as follows:
\[
\begin{align*}
\theta_{i,j}^t &= \frac{\kappa_R \Delta t}{\rho_R C_R} \left( \frac{\theta_{i-1,j} + 2\theta_{i,j} + \theta_{i+1,j}}{(\Delta z)^2} \right) + \theta_{i,j}^t \\
\theta_{i,j}^t &= \frac{2\Delta t}{b \rho_A C_A} \left[ k_R \frac{(\theta_{i,j} - \theta_{i-1,j})}{\Delta z} - \frac{\rho_A C_{a,m}}{2} \frac{(\theta_{i,j} - \theta_{i,j-1})}{\Delta S} \right] + \theta_{i,j}^t \quad \text{if } i = 1
\end{align*}
\]

\[
\begin{align*}
\theta_{i,j}^t &= 0 \quad i = N \\
\theta_{i,j}^t &= \theta_0 \quad i = 1, j = 1 \\
\theta_{i,j}^t &= 0 \quad \text{all } i \& j \text{ (initial condition).}
\end{align*}
\]

The z-plane is represented by \( i \) where \( N \) is the total number of nodes and \( j \) represents the S-plane where \( M \) is the total number of nodes. The superscript on \( \theta \) refers to the newly calculated node temperatures. Convergence of Equation 2.13 requires that \( 0 < \lambda < 0.5 \) where \( \lambda = \Delta t k_R / \rho_R C_R (\Delta z)^2 \).

A computer program was developed which solves the above equations for successive time steps. When a given injection time is reached nodal values are printed which correspond to the aquifer temperature \( \theta_{i,j}, i = 1, \text{all } j \). Figure 2.7 shows these results for several combinations of \( \Delta z, \Delta t, \text{and } \Delta S \). Table 2.1 shows values assigned to the variables for each curve.

All three curves very closely approximate the analytical solution at small \( S/S_{\text{max}} \). As \( S/S_{\text{max}} \) increases, the numerical results lie above the analytical solution with smaller \( \Delta z \) resulting in smaller differences. At \( S/S_{\text{max}} = 1.0 \), the nondimensional temperature sharply decreases to zero. The smaller \( \Delta S \), the sharper the drop-off with the analytical solution giving almost a vertical drop-off. The finite-difference approximation tends to diffuse the sharp temperature front predicted by the analytical
TABLE 2.1. VARIABLE VALUES USED IN THE FINE-DIFFERENCE APPROXIMATION OF NONDIMENSIONAL AQUIFER TEMPERATURES AND NONDIMENSIONAL PRODUCTION TEMPERATURES.

<table>
<thead>
<tr>
<th>Curve 1</th>
<th>Curve 2</th>
<th>Curve 3</th>
<th>Curve 4</th>
<th>Curve 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta t$ (s)</td>
<td>900</td>
<td>900</td>
<td>1800</td>
<td>28800</td>
</tr>
<tr>
<td>$\Delta z$ (m)</td>
<td>0.2</td>
<td>0.5</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>$\Delta S$ (m^2)</td>
<td>2.5</td>
<td>5.6779</td>
<td>11.356</td>
<td>13.5</td>
</tr>
<tr>
<td>$N$</td>
<td>50</td>
<td>20</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>$M$</td>
<td>200</td>
<td>90</td>
<td>50</td>
<td>200</td>
</tr>
<tr>
<td>$b$ (m)</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>60</td>
</tr>
<tr>
<td>$Q$ (m^3/s)</td>
<td>0.01262</td>
<td>0.01262</td>
<td>0.01262</td>
<td>0.01161</td>
</tr>
<tr>
<td>$t_{inj}$ (s)</td>
<td>216000</td>
<td>216000</td>
<td>216000</td>
<td>7776000</td>
</tr>
<tr>
<td>$S_{max}/\Delta S$</td>
<td>159</td>
<td>70</td>
<td>35</td>
<td>163</td>
</tr>
</tbody>
</table>

solution; the extent being dependent on $\Delta S$. One would see similar effects if conduction was included in the flow direction.

Figure 2.8 illustrates nondimensional aquifer temperature as a function of nondimensional $S$ for three aquifers with the same $b^2/t_{inj}$ ratio, but different thicknesses and injection times. Again the values assigned to the variables for each curve are listed in Table 2.1. The results of all three curves are very similar and closely approximate the analytical solution. Better agreement with the analytical solution is obtained as $S_{max}/\Delta S$ increases.

Confident that the numerical model predicts aquifer temperature profiles, the next step is calculating production temperatures after injection under specified conditions. In the computer program this simply
Figure 2.7. The finite-difference approximation of nondimensional aquifer temperature is plotted as a function of nondimensional flow channel area for various combinations of $\Delta t$, $\Delta z$, and $\Delta S$. Variable assignments are those shown in Table 2.1 for curves 1, 2, and 3. The ratio $b^2/t_{inj}$ is held constant at 4.63x10^{-4} m^2/s for all four curves.
Figure 2.8. The finite-difference approximation of nondimensional aquifer temperature is plotted as a function of nondimensional flow channel area for three aquifers with the same $b^2/t_{inj}$ ratio ($4.63 \times 10^{-4}$ m$^2$/s), but different thicknesses and injection times. Variable assignments are those shown in Table 2.1 for curves 1, 4, and 5.
means a reversal of flow direction with the production temperature equal to the aquifer temperature at the wellbore \( (\theta_1, 1) \).

Two production simulations were carried out with variables equal to those of curves four and five (Table 2.1). All variable values were the same for injection and production. Nondimensional production temperature as a function of nondimensional time \( (t/t_{inj}) \) for the simulations are shown in Figure 2.9. Curve four has a sharper temperature front at the end of injection (Figure 2.8) which indicates that less losses have occurred and consequently higher production temperatures are achieved. Relatively speaking, both curves give comparable production results, in fact the results are identical until \( t/t_{inj} = 0.7 \) at which time curve five drops below curve four. As will be shown later, these results are substantially higher than results predicted by more involved models and field experiments.

Also investigated were the effects of having a variable flow rate. A comparison was done using curve five variable assignments. In the variable case the flow rate was held constant at 0.03483 m\(^3\)/s for eight hours a day, but was held at zero the remaining 16 hours a day for the entire simulation. Thus, the average flow rates are identical in both the constant and variable flow cases.

Figure 2.10 shows the results of each case. The variable flow case shows a significant increase in aquifer temperatures. Many of the differences between the two curves can be attributed to characteristics of the finite-difference approximation. Because \( Q \) is effectively three times larger in the variable flow case, \( S_{\text{max}}/\Delta S \) will also be three times larger and as stated before, the larger \( S_{\text{max}}/\Delta S \) is the closer the numerical simulation approximates the analytical solution. In comparing the variable
Figure 2.9. The finite-difference approximation of nondimensional production temperature is plotted as a function of nondimensional production time for two aquifers with the same $b^2/t_{inj}$ ratio ($4.63 \times 10^{-4}$ m$^2$/s), but different thicknesses and injection times. Variable assignments are those shown in Table 2.1 for curves 4 and 5.
Figure 2.10 The finite-difference approximation of nondimensional aquifer temperature is plotted as a function of nondimensional flow channel area for constant and variable injection flow rates. Variable assignments are those shown in Table 2.1 for curve 5.
flow rate results with the analytical solution it is seen that the differences are substantially reduced. One feature indicates that the variable flow case does improve aquifer performance. The variable flow rate curve lies completely above the analytical solution. In all other simulation results the temperature curves dip below, then cross, and finally end up above the analytical solution. These results indicate that decreased losses due to rapid injection (large $b^2/t_{inj}$) are more important than losses incurred while fluid sits idly in the aquifer. In actual aquifers this is not necessarily true because of the losses due to buoyancy effects which are not accounted for in the above simple simulations.

2.2.2 Literature Numerical Solutions and Results

To completely model water flowing through a porous medium, the equations for conservation of mass, momentum, and energy must be satisfied. Several authors have developed numerical solutions to these equations. Tsang [30] summarized current theoretical and modeling studies of energy storage in aquifers (Table 2.2). Many of these models are currently under development and only limited modeling results have been reported. Tsang has published several papers on the model developed at Lawrence Berkeley Laboratory, therefore I will concentrate the discussion on his results.

The program CCC (Conduction, Convection, and Compaction) is used by Tsang [29] to investigate aquifer thermal energy storage. CCC employs the integrated finite-difference method and is a fully three-dimensional model incorporating the effects of complex geometry, regional flow, temperature-dependent fluid properties, gravity, and land subsidence or
<table>
<thead>
<tr>
<th>Research Institute</th>
<th>Project</th>
</tr>
</thead>
<tbody>
<tr>
<td>Technical University of Denmark, Denmark (Qvale, 1978)</td>
<td>One- and two-dimensional finite element models; study of using compensation wells for countering regional flow</td>
</tr>
<tr>
<td>Lund University, Sweden (Hellstrom, 1978; Claesson and others, 1978)</td>
<td>Two-dimensional, doublet, semi-analytic model; two-dimensional finite difference program developed to study storage in eskers</td>
</tr>
<tr>
<td>University of Neuchâtel, Switzerland (Mathey, 1977; Mathey and Menjos, 1978)</td>
<td>Two- and three-dimensional finite element models</td>
</tr>
<tr>
<td>Instiüt de Production d'Energie de l'Ecole Polytechnique Fédérale de Lausanne, Switzerland (Joos, 1978)</td>
<td>Three-dimensional finite element model; laboratory experiments on free convection in porous media</td>
</tr>
<tr>
<td>École des Mines de Paris, France (de Marsily, 1978)</td>
<td>Two-dimensional, radial, finite difference model; two- and three-dimensional finite element models</td>
</tr>
<tr>
<td>Bureau des Recherches Géologiques et Minières (BRGM), France (Gringarten and others, 1977; Sauty and others, 1979)</td>
<td>Layered two-dimensional finite difference model; modeling of the Bonnaud experiment; dispersion modeling studies</td>
</tr>
<tr>
<td>University of Yamagata, Japan (Yokoyama and others, 1978)</td>
<td>Finite difference method using a complex potential function</td>
</tr>
<tr>
<td>United States Geological Survey, United States (Papadopulos and Larson, 1978)</td>
<td>Intercomp model, based on a finite-difference scheme, used to model the Auburn (1976) experiment</td>
</tr>
<tr>
<td>Lawrence Berkeley Laboratory, United States (Tsang and others, 1979)</td>
<td>Three-dimensional integrated finite-difference model for conduction, convection, and consolidation; extensive generic studies; modeling of the Auburn (1978) experiment</td>
</tr>
<tr>
<td>University of Texas, United States (Collins and others, 1978)</td>
<td>Model to study steam injection into permeable earth strata (two-phase program)</td>
</tr>
</tbody>
</table>
uplift. Results obtained from CCC in simulating annual storage cycles are shown in Figures 2.11, 2.12, and 2.13. Tsang's results have been nondimensionalized for comparison purposes. All the specific variable assignments can be found in Tsang's papers [28,29].

Nondimensional production temperature as a function of nondimensional production volume is shown in Figure 2.11 for the first and fourth cycles of successive annual cycles. Each cycle of 360 days is composed of four equal periods: i) 90 days injection, ii) 90 days idle storage, iii) 90 days production, and iv) 90 days idle storage. The flow rate is kept constant at $10^6$ kg/day for both injection and production. Nondimensional production volume can also be represented as nondimensional production time ($t_{\text{prod}}/t_{\text{inj}}$) since $V=Qt$ and $Q$ is constant.

Figure 2.12 is actual production temperatures obtained from the Auburn experiment [29]. In the first cycle 54,784 m$^3$ were injected at 55 C, then after a storage period of 51 days, 55,345 m$^3$ were produced over a 41 day period. The second cycle started 18 days later. In the second cycle 58,010 m$^3$ were injected at the same 55 C temperature over a 64 day period. Production started 63 days later and lasted 84 days. During this period 100,100 m$^3$ was pumped.

There are several important features present in Figures 2.11 and 2.12 which are typical for aquifers under consideration in this thesis. Initially fluid is produced at the injection temperature. By the end of production it drops to a value substantially above the aquifer temperature. The specific shape of the curve depends on the extent of losses. Generally, less losses will yield a curve which is flatter at the beginning of production.

Both Figures 2.11 and 2.12 describe production from a single well.
Figure 2.11. Nondimensional production temperature versus nondimensional production volume for the first and fourth cycles of a simulated aquifer, from [28].

\[
\frac{T - T_\infty}{T_\text{inj} - T_\infty}
\]

\(T_\infty = T_{\text{AQUA}} = 20^\circ\text{C}\)

\(T_\text{inj} = 120^\circ\text{C}\)
Figure 2.12. Nondimensional production temperature versus nondimensional production volume for the first and second cycle test of the Auburn Aquifer, from [29].
Figure 2.13. Nondimensional production temperature versus nondimensional production volume for a two-well system and the corresponding single-well case, from [28].
The difference between a single and double well system is minimal if in
the doublet case the wells are adequately separated. In a comparison
by Tsang [28] this was shown to be the case. In Figure 2.13 the non-
dimensional production temperature as a function of nondimensional pro-
duction volume for both a singlet and doublet is shown and indeed the
difference is small.

In comparing Figures 2.9 and 2.13, in which aquifer parameters are
identical, the numerical solution of the analytical formulation predicts
higher aquifer performance than Tsang's numerical model. This is not
surprising because many phenomena were neglected in the analytical model,
namely buoyancy effects and temperature differences within the aquifer.
Figure 2.14 shows temperature profiles in the aquifer after 90 days injec-
tion for the doublet case of Figure 2.13. Tilting and diffusion of the
temperature front is readily visible. Also shown in Figure 2.14 are the
temperature profiles predicted by the analytical model. The temperature
front is sharper and vertical in the analytical case.

During production, water is taken equally from all levels of the
aquifer so any tilting or deformation of the temperature front will tend
to lower the production temperature. The variability introduced in the
results predicted by Tsang's model due to tilting and deformation of the
temperature front is much larger than the variability introduced due to
variations in b and t_{inj} as predicted by the analytical model.

In conclusion, the analytical model can predict trends with speci-
fied injection and production parameters, but the variability which
exists in actual aquifers are greater than these trends, therefore to
realistically predict aquifer performance a very involved model, such as
that employed by Tsang, must be used.
Figure 2.14. Calculated isotherms for a two-well system after 90 days of injection; plane and cross section views. Solid lines represent Tsang's [28] results and dashed lines represent isotherms as predicted by the Analytical Solution (Equation 2.11).
III. SYSTEM MODEL

The system being investigated is illustrated in Figure 1.1. It contains a set of solar collectors, one aquifer which has two wells (injection and production), and numerous heat pumps attached by a distribution network. The distance between wells (aquifer size), and collector area are determined by the number of heat pump units and local weather conditions. The system can be subdivided into five subsystems; aquifer, collectors, distribution network, load, and heat pumps. Each subsystem model is discussed and its performance specified.

3.1 AQUIFER

The aquifer is modeled as a confined horizontal porous medium of a specified permeability and porosity, of a certain height, and of sufficient areal extent that the supply and return wells can be located far enough apart to allow the annual heating load to be met from the storage.

As discussed in Chapter II, to accurately model an aquifer a very involved numerical model is required. A model like this requires extensive computer time which was beyond the means of this study. Since the principal objective of this study was aimed at subsystem interaction and not on aquifer modeling, it was decided to use a relatively simple aquifer model. This should provide information essential for subsystem interaction and minimize computer time. Tsang's results were ideal in this regard because they represented a wide spread of aquifer performance and although the aquifers' dimensions and temperatures are not those used in this study, many parameters are similar.

Curves from Figures 2.11 and 2.12 were selected as representative of
aquifers of interest to seasonal thermal energy storage. The least square fit of Figure 2.12 will be called curve A, which is representative of aquifer A. Curves B and C will be that of cycle 1 and cycle 4 of Figure 2.11 which are representative of aquifer B and C, respectively. The equations used to describe these curves are as follows:

Curve A: 
\[ \frac{T-T_\infty}{T_\text{inj}-T_\infty} = 0.9857 - 0.62404 \frac{V_{\text{prod}}}{V_{\text{inj}}} \]  
(3.1)

Curve B: 
\[ \frac{T-T_\infty}{T_\text{inj}-T_\infty} = 1.0 - 0.47927 \frac{V_{\text{prod}}}{V_{\text{inj}}} \]  
(3.2)

Curve C: 
\[ \frac{T-T_\infty}{T_\text{inj}-T_\infty} = 1.0 - 0.26577 \frac{V_{\text{prod}}}{V_{\text{inj}}} \]  
(3.3)

A measure of aquifer performance is contained in the recovery factor (RF) defined as the energy produced divided by the energy injected and is equal to the area under the production curve. The area above the production curve is equal to energy losses. The recovery factor of aquifers A, B, and C are 0.694, 0.816, and 0.908, respectively.

In order to calculate pumping power the aquifer model also needs to predict the head loss across the aquifer. The head loss across an aquifer can be determined from Darcy's law [27].

\[ Q = A \frac{k \rho g}{\mu} \frac{dh}{dl} \]  
(3.4)

The variables involved are viscosity, \( \mu \), density, \( \rho \), gravitational acceleration, \( g \), volume flow rate, \( Q \), area, \( A \), specific (or intrinsic) permeability, \( k \), and hydraulic gradient \( dh/dl \).\(^2\) The dimensions of \( k \) are

\(^2\) Darcy's law is sometimes written with a negative sign to indicate that the flow is in the direction of decreasing head. It is also written \( Q = K A \frac{dh}{dl} \), where \( K \) is the coefficient of permeability (hydraulic conductivity). A good discussion of Darcy's law in which these variations are addressed exists in Todd [27] chapter 3.
$L^2$, or area. The validity is limited to laminar flow which includes all areas except that immediately adjacent to the wellbore. Applying Darcy's law to radial flow, the equation becomes

$$Q = 2\pi r \frac{b k \rho g}{u} \frac{dh}{dr}$$

where $b$ is the aquifer thickness. Rearranging and integrating for boundary conditions of $r=r_w$, $h=h_w$ at the well and $r=r$, $h=h$ at a distance $r$ from the well yields,

$$h - h_w = \frac{Q u}{2 \pi b k \rho g} \ln \frac{r}{r_w}$$

In the case of a doublet (injection and production wells), a good approximation of the head loss is twice the sum of the head differential from a well to a point half-way between the wells [24]. Thus the head loss becomes

$$\Delta h = \frac{Q u}{\pi b k \rho g} \ln \frac{D}{2 r_w}$$

where $D$ is the distance between wells. The pumping power is equal to $\rho g Q \Delta h$ and $\Delta h$ is proportional to $Q$, therefore pumping power is proportional to the square of the flow rate.

The amount of storage contained between the wellbore pair was taken to be $1.05 D^2 \cdot b$ as evaluated by Schaetzle [25]. The storage is less than $\pi D^2 \cdot b$ because the thermal front does not move out radially around the injection well, but rather distorts toward the production well (see Figure 2.14). The 1.05 value represents a potential flow solution for an aquifer with no temperature stratification such that thermal breakthrough to the production well does not occur.
3.2 HEAT PUMP

Projected performance of "heat only" water source heat pumps designed for geothermal heating with source temperatures between 15 and 50°C were used in this study. Reistad and Means [21] calculated heat pump COPH's in this category and their results are shown in Figure 3.1. The COPH only includes compressor and fan power; omitted was the pumping power to circulate water through the evaporator. Table 3.1 lists the specifications for the heat pump units. The mathematical equation used to represent the COPH is as follows:

\[
\text{COPH} = \frac{580}{580 - T_{\text{evap}}} (1.1953 - 0.001498 T_{\text{evap}}).
\]  

(3.8)

The evaporator temperature \( T_{\text{evap}} \) in degrees Rankine is equal to

\[
T_{\text{evap}} = T_W - \Delta T_{\text{app}} - \Delta T_W
\]

(3.9)

where \( T_W \) is the source water temperature, \( \Delta T_{\text{app}} \) is the approach temperature, and \( \Delta T_W \) is the temperature drop across the evaporator.

In addition to COPH, heat pump capacity and source fluid flow rate must also be considered. The flow rate necessary to bring about a given temperature drop across the evaporator is

\[
Q = \frac{\dot{E}_{\text{evap}}}{C_W \Delta T_W}
\]

(3.10)

where \( \dot{E}_{\text{evap}} \) represents the rate at which energy is absorbed by refrigerant and is a function of the evaporator temperature. Heat pump capacity is assumed to be large enough to meet design load conditions. Specific assumptions used in calculating COPH, capacity, and flow rate can be found in the report by Reistad and Means [21].
Figure 3.1. Performance of water-source heat pumps, with the specifications listed in Table 3.1 (components resized at each new condition), as a function of the evaporator temperature.
### TABLE 3.1 SPECIFICATIONS FOR HEAT PUMP DESIGNED AT VARIOUS RESOURCE TEMPERATURES, FROM [21].

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating Output, $\dot{Q}_H$, 10.55 kW (36,000 Btu/hr)</td>
<td></td>
</tr>
<tr>
<td>Condenser Temperature 48.9°C (120°F)*</td>
<td></td>
</tr>
<tr>
<td>Adiabatic Compression Efficiency of Compressor-Motor Combination, $\eta = 53%$**</td>
<td></td>
</tr>
<tr>
<td>Refrigerant R-22</td>
<td></td>
</tr>
<tr>
<td>Fan Power $\dot{W}_F = 0.0421 \dot{Q}_H$*** ($\dot{W}_F$ and $\dot{Q}_H$ in the same units)</td>
<td></td>
</tr>
</tbody>
</table>

* The condenser temperature is specified as a constant because the heating capacity of the heat pump is specified as a constant.

** Representative value for motor-compressor units in newly designed heat pumps.

*** Obtained as an average value from three water-source heat pump units at rated conditions.
3.3 HEATING LOAD

The heating load was calculated from weather data for a suburban Portland, Oregon residence. The house was constructed according to standards in common use during the past five years [5]. Important features of the residence under investigation are listed in Table 3.2. Hourly weather information from 1963 is used to calculate the total heat load each hour for NB residential units and is as follows:

\[ HL = UA \times NB \times \frac{\text{FLOOR AREA}}{\text{UNIT}} \times (T_D - T_a) \times 1 \text{ HOUR} \]  
\[ \text{(BTU's or JOULES)} \]  

(3.11)

where \( UA = \) Overall loss coefficient based on floor area

\( T_D = \) Inside design temperature

\( T_a = \) Ambient temperature.

The heating season was taken from the beginning of November to the end of March. The remainder of the months are assumed to have little heat load, or the daily solar energy is enough to satisfy the space heating requirements. These assumptions can be validated with the aid of Table 3.3 which lists the monthly average daily radiation on a horizontal surface \( (\overline{H}) \), monthly average daily ambient temperature \( (T_a) \), and monthly total C-degree days \( (C-DD) \) for Portland, Oregon [3]. Both April and October have substantial C-degree days, however they also have appreciable solar radiation. A quick calculation\(^3\) shows that \( 1150/n_C \) and \( 1570/n_C \) square meters are needed to capture enough energy to provide the

\[ \text{COLLECTOR AREA (m}^2) = \frac{(2.35 \frac{W}{m^2 \cdot ^\circ C})(75 \text{ UNITS})(139 \frac{m^2}{\text{UNIT}})(C-DD)(3600 \frac{s}{HR})(24 \frac{HR}{DAY})}{\overline{H}(\frac{J}{m^2})} \]
### TABLE 3.2. CHARACTERISTICS OF RESIDENCE BEING CONSIDERED, FROM [5].

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Loss Coefficient</td>
<td>2.35 W/°C m² (0.39 Btu/hr °F ft²)</td>
</tr>
<tr>
<td>Average Floor Area Per Unit</td>
<td>139 m² (1500 ft²)</td>
</tr>
<tr>
<td>Residence Density</td>
<td>1000 Units/Km² (4/Acre)</td>
</tr>
<tr>
<td>Population Density</td>
<td>3500 people/Km² (14/Acre)</td>
</tr>
</tbody>
</table>

### TABLE 3.3. METEOROLOGICAL DATA FOR PORTLAND, OREGON, FROM [3].

<table>
<thead>
<tr>
<th></th>
<th>JAN</th>
<th>FEB</th>
<th>MAR</th>
<th>APR</th>
<th>MAY</th>
<th>JUNE</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\bar{H}$ MJ/m²</td>
<td>4.02</td>
<td>6.57</td>
<td>10.05</td>
<td>14.78</td>
<td>17.71</td>
<td>19.80</td>
</tr>
<tr>
<td>$T_a$ °C</td>
<td>3.0</td>
<td>6.0</td>
<td>8.0</td>
<td>10.0</td>
<td>14.0</td>
<td>17.0</td>
</tr>
<tr>
<td>$C-\text{DD C-DAY}$</td>
<td>463.</td>
<td>346.</td>
<td>332.</td>
<td>240.</td>
<td>147.</td>
<td>71.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>JULY</th>
<th>AUG</th>
<th>SEP</th>
<th>OCT</th>
<th>NOV</th>
<th>DEC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\bar{H}$ MJ/m²</td>
<td>23.15</td>
<td>18.59</td>
<td>14.36</td>
<td>8.41</td>
<td>4.86</td>
<td>3.47</td>
</tr>
<tr>
<td>$T_a$ °C</td>
<td>19.0</td>
<td>19.0</td>
<td>17.0</td>
<td>12.0</td>
<td>7.0</td>
<td>5.0</td>
</tr>
<tr>
<td>$C-\text{DD C-DAY}$</td>
<td>27.</td>
<td>31.</td>
<td>66.</td>
<td>193.</td>
<td>328.</td>
<td>418.</td>
</tr>
</tbody>
</table>
heating load in April and October respectively. If one assumes solar energy will be collected from May 1 to September 30, it is found that 1393/\eta_c square meters of collectors are required to provide enough energy to meet the total heat load from November 1 to March 31. At a collector area of 1393/\eta_c square meters there will be excessive energy collected in April which will offset the deficit in October. Thus, ignoring April and October in heat load calculations should not affect the results in that the energy collected and stored from May 1 to September 30 will be adequate to supply the heating load from November 1 to March 30.

3.4 COLLECTORS

The collectors are assumed to be of a low cost variety\(^4\). Both glazed and unglazed collectors were investigated. The collectors are modeled with a conventional efficiency curve (Figure 3.2). The equations used to describe these curves are as follows:

**UNGLAZED:** \[ \eta_c = 1.0 - 0.036533 \exp \left( \frac{22.00436 (T_i - T_a)}{I} \right) \]  

**GLAZED:** \[ \eta_c = 0.77 - 1.12 \frac{(T_i - T_a)}{I} \]

where \(T_i\) = Collector Inlet Temperature (F), 
\(T_a\) = Ambient Temperature (F), 
\(I\) = Incident Solar Radiation (BTU/HR/FT\(^2\)).

The unglazed collector efficiency drops quite rapidly at elevated temperatures due to radiant losses, but at low \((T_i - T_a)/I\) its efficiency is better than the glazed collector because of losses associated with the glazing.

\(^4\) SolaRoll heat absorbing mat was selected in this study because of its low cost, ease of installation, and durability.
Figure 3.2. University of Connecticut - Solar Energy Evaluation Center 30-day exposure test (HUD 4930.2/ASHRAE 93-77).
In modeling the collectors, the temperature rise across them is held constant by varying the flow rate. Operation of the collectors is restricted to sunlight hours, from May 1 to September 30, in which a net gain of energy is realized.

3.5 DISTRIBUTION NETWORK

In this thesis, there are no energy losses attributed to the distribution system. The only influence that is considered which could be attributed to the distribution system is that a constant head of ten meters of water is assumed to be required over and above the head loss through the aquifer for the flow of fluid in both the delivery and recharge systems.
Yearly simulations were performed with 1963 hourly weather information from Portland, Oregon. The year was divided into two periods, injection from May 1 to September 30 and withdrawal from November 1 to March 31. During injection, whenever the collectors realized a net energy gain, water was pumped from the aquifer's cold side, through the collectors and back into the aquifer's hot side. During withdrawal, water was pumped from the aquifer's hot side, through the heat pump units which subsequently satisfied the space heating load, and then back into the aquifer's cold side. In both cases the flow rate was adjusted such that a set temperature drop would result.

The computer program developed to carry out these functions is listed in Appendix B. The program consists of three sections; injection, production, and print-out. Water temperature drop ($\Delta T_W$) and storage temperature ($T_{STOR}$) are the primary parameters used in calculations. In every section evaporator temperature is calculated for all combinations of $\Delta T_W$ and $T_{STOR}$ and if it is greater than zero degrees, calculations are performed. A complete list of parameters and the values assigned to them are given in Table 4.1.

One variable in particular, the safety factor, needs to be clarified. The safety factor is defined as the volume of fluid injected divided by the volume of fluid that would have to be injected if the storage performed just as modeled. The reason for a safety factor is to account for any unusual flow situations and mask a number of system unknowns. Its function is to increase the amount of energy injected during the summer months. As the safety factor increases above one, the volume injected
<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>DESCRIPTION</th>
<th>VALUES ASSIGNED</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>AQUIFER</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(k) (mD)</td>
<td>Permeability</td>
<td>5000</td>
</tr>
<tr>
<td>(b) (m)</td>
<td>Thickness</td>
<td>30</td>
</tr>
<tr>
<td>SF</td>
<td>Design Safety Factor</td>
<td>1.0, 2.0*</td>
</tr>
<tr>
<td>(r_w) (m)</td>
<td>Radius of the Wellbore</td>
<td>0.076</td>
</tr>
<tr>
<td>RF</td>
<td>Recovery Factor</td>
<td>0.694, 0.816*, 0.908</td>
</tr>
<tr>
<td>(\rho_{wC_w}) (J/m(^3)C)</td>
<td>Heat Capacity Water</td>
<td>4167460</td>
</tr>
<tr>
<td>(\rho_{RCR}) (J/m(^3)C)</td>
<td>Heat Capacity Rock</td>
<td>2522000</td>
</tr>
<tr>
<td>(\phi)</td>
<td>Porosity</td>
<td>0.2</td>
</tr>
<tr>
<td>(K_{R}) (J/smC)</td>
<td>Thermal Conductivity</td>
<td>1.157</td>
</tr>
<tr>
<td><strong>HEAT PUMP</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(\Delta T_{app}) (C)</td>
<td>Approach Temperature</td>
<td>4.0</td>
</tr>
<tr>
<td><strong>LOAD</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>NB</td>
<td>Number Building Units Per Borehole Pair</td>
<td>75</td>
</tr>
<tr>
<td>UA (W/m(^2)C)</td>
<td>Overall Loss Coefficient Based On Floor Area</td>
<td>2.35</td>
</tr>
<tr>
<td>ANB (m(^2))</td>
<td>Floor Area Per Building Unit</td>
<td>139</td>
</tr>
<tr>
<td>C-DD (C-Day)</td>
<td>C Degree Days Per Heating Season</td>
<td>2303</td>
</tr>
<tr>
<td>(T_D) (C)</td>
<td>Inside Design Temperature</td>
<td>21.1</td>
</tr>
<tr>
<td><strong>DESIGN VARIABLES</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(\Delta T_w) (C)</td>
<td>Temperature Change of the Water Through Heat Pump, Collectors, or Aquifer</td>
<td>4, 8, 12, 16, 20, 24+</td>
</tr>
<tr>
<td>(T_{STOR}) (C)</td>
<td>Storage Temperature of the Water In the Aquifer</td>
<td>40, 35, 30, 25, 20+</td>
</tr>
</tbody>
</table>

* Base Case Values

+ No Base Case Values because these are primary variables
becomes proportionally larger, but the volume produced will only become slightly more due to the unchanging load and high production temperature. This effectively improves the aquifer's performance during the production period\textsuperscript{5}. To compensate this effect, the volume of water produced was multiplied by the safety factor. The production temperature was the only variable affected. Only the fluid necessary to provide the heating load was used to calculate pumping power.

During injection the program sums collected energy along with the square of collected energy. The former is used to calculate collector area and volume of water injected and both are needed to calculate pumping power. There are five summations in the production section, i) heating load, ii) energy to evaporator, iii) square of energy to evaporator, iv) energy used by heat pump, and v) volume of water produced. Energy to evaporator and its square are required to calculate pumping power. This together with summer pumping power, energy used by heat pump, and heat load go into calculating overall system COPH. The volume of water produced is a primary variable in calculating production temperature which determines heat pump COPH.

The base case print out is shown in Appendix B. Many parameters are printed as a function of $\Delta T_W$ and $T_{STOR}$, with COPH and collector area being the primary variables of interest in this study.

\textsuperscript{5} Production temperature is a function of volume produced divided by volume injected ($V_{prod}/V_{inj}$). With a safety factor of two, twice as much fluid will be injected as is required, thus the ratio $V_{prod}/V_{inj}$ will never exceed 0.5 and consequently the production temperature will be higher.
V. SIMULATION RESULTS AND DISCUSSION

Before going into the results of the system just described, it should be stated that part of this thesis is a continuation of work done in a report by Reistad and McMahon [22]. The system modeled in the previous report was much the same except in two regards; i) the aquifer production temperature was held constant at some constant specified value, and ii) the recharge system was not designated other than it is a constant flow system that operates for a specified summer recharge period (540 hrs and 2160 hrs). For the remainder of the thesis this model will be called the Constant Temperature Model (CTM) and the system modelling described in Chapter III will be called the Variable Temperature Model (VTM).

Results from the constant temperature model are available over a wide range of variables. Therefore if the two models can be correlated, the variable temperature model results can be extended. Only a limited number of yearly simulations of the VTM were run due to time and money restraints. However if the two models can be correlated, extra runs are not really needed.

In this Chapter, results from the variable temperature model will first be discussed, then the extension of these with results from the constant temperature model will be considered. Each discussion also indicates ranges of parameters where such systems have no energy feasibility, and ranges where the systems have the potential to be feasible, depending on economic conditions.

Figures 5.1 to 5.4 show base case results of system simulations for the three aquifers considered. Dashed lines represent the system performance when the energy requirement for summer recharge pumping is not
considered (designated here as "winter only" performance) and solid lines represent the performance of the overall system, summer pumping included. Circles, squares, and triangles refer to the aquifer production curve used in the simulation (\(\Delta =\) Aquifer A, RF = 0.694; \(\circ =\) Aquifer B, RF = 0.816; \(\sigma =\) Aquifer C, RF = 0.908).

There are several features which all these curves have in common. "Winter only" performance is greater than the overall system performance throughout the range of \(\Delta T_W\) considered. Overall system performance approaches "winter only" performance at high \(\Delta T_W\) with a larger spread existing as \(\Delta T_W\) decreases. Aquifer C gives the best performance with Aquifer B and Aquifer A falling below it. The dramatic drop-off of performance at low \(\Delta T_W\) is a result of pumping power being proportional to the square of the flow rate which in turn is inversely proportional to \(\Delta T_W\).

There are two competing phenomena which determine system performance. The first is heat pump performance, which is highest at low \(\Delta T_W\) and decreases as \(\Delta T_W\) increases (Figure 3.2). This means energy consumption will be lowest at small \(\Delta T_W\) and increase as \(\Delta T_W\) increases. The second phenomena is pumping power, both summer and winter, which is highest at low \(\Delta T_W\) and decreases as \(\Delta T_W\) increases. In this case energy consumption is highest at small \(\Delta T_W\) and decreases as \(\Delta T_W\) increases. The reverse effects of these two phenomena produce a minimum where energy consumption is lowest; the position depends on the relative influence of each phenomenon. The system's performance (COPH) is maximum when energy consumption is a minimum, therefore, a value of \(\Delta T_W\) also exists for maximum system performance.
Figure 5.1. COPH as a function of water temperature change; base case variables with a storage temperature of 40°C.
Figure 5.2. COPH as a function of water temperature change; base case variables with a storage temperature of 35 C.
Figure 5.3. COPH as a function of water temperature change; base case variables with a storage temperature of 30°C.
Figure 5.4. COPH as a function of water temperature change; base case variables with a storage temperature of 25°C.
Only the effects of $\Delta T_w$ have been addressed, but system performance also depends, to a great extent, on the storage temperature. Figures 5.1 to 5.4 show system performance decreasing as $T_{STOR}$ decreases. This is as expected due to decreasing heat pump performance. Another effect upon decreasing $T_{STOR}$ is the sudden drop-off of performance at large $\Delta T_w$. This occurs because the evaporator temperature, which started the heating season above zero degrees (C), drops to zero (C) before the heating season is completed and a standby electric resistance heating system is used to supply the remaining load. By operating at variable $\Delta T_w$ in those instances where $T_{evap}$ would normally drop below zero (C), the sharp drop-off might be eliminated or reduced depending on the particular situation.

In evaluating energy feasibility on an absolute scale, a COPH greater than three guarantees energy feasibility. With more efficient power plants coming on line, in some instances a value less than three will represent energy feasibility. For this discussion a COPH of three is taken as the dividing line.

Figure 5.1 shows energy feasibility for all aquifers at $\Delta T_w$ above ten degrees (C). Those aquifers with recovery factors of 0.816 and 0.908 have energy feasibility at $\Delta T_w$ above eight and seven degrees (C) respectively. Maximum performance falls at $\Delta T_w=12$ C. As $T_{STOR}$ decreases, energy feasibility is restricted to a range of $\Delta T_w$; the COPH drops below three at both low and high $\Delta T_w$. Table 5.1 presents the ranges of energy feasibility for Figures 5.1 through 5.4. Two trends are prominent, i) range of feasibility increases as aquifer recovery factor increases, and ii) range of feasibility increases as $T_{STOR}$ increases.
TABLE 5.1. RANGES OF WATER TEMPERATURE CHANGE WHERE ENERGY FEASIBILITY EXISTS AS PREDICTED BY THE VARIABLE TEMPERATURE MODEL WITH VARIABLES EQUAL TO THEIR BASE CASE VALUES.

<table>
<thead>
<tr>
<th>$T_{STOR}$ (C)</th>
<th>AQUIFER A</th>
<th>AQUIFER B</th>
<th>AQUIFER C</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>&gt;9.5</td>
<td>&gt;7.8</td>
<td>&gt;6.8</td>
</tr>
<tr>
<td>35</td>
<td>9.5 - 21.1</td>
<td>7.8 - 22.0</td>
<td>&gt;6.8</td>
</tr>
<tr>
<td>30</td>
<td>10.6 - 16.5</td>
<td>8.1 - 17.6</td>
<td>7.2 - 20.8</td>
</tr>
<tr>
<td>25</td>
<td>-</td>
<td>9.2 - 13.0</td>
<td>8.0 - 16.2</td>
</tr>
</tbody>
</table>

A major source of variability in overall system performance is the value assigned to the safety factor. Figure 5.5 shows system performance with parameters those of the base case except the safety factor which is one. Only Aquifer B with RF=0.816 is shown. The other aquifers give results slightly above and below this case. Figure 5.5 shows a significant increase in system performance over the base case. Maximum performance (COPH) at $T_{STOR}$=40 C increases from 3.9 to 4.6 while at $T_{STOR}$=30 C the increase is from 3.3 to 3.7. The maximum also moves to lower $\Delta T_W$ because the pumping power is reduced, thus it has less influence on overall performance. Energy feasibility can be realized over a much wider range with a safety factor of one. Feasibility is extended to $\Delta T_W$ as low as four degrees (C).

In all base case simulations unglazed collectors were modeled as the collection system. Figures 5.6 and 5.7 compare system performance when glazed or unglazed collectors are modeled. "Winter only" performance is identical in both cases because the same volume and temperature water is stored in the aquifer. Glazed collectors give better overall performance at low-to-mid $\Delta T_W$, but at large $\Delta T_W$ the two collectors give identical
Figure 5.5. COPH as a function of water temperature change; parameters are those of the base case (T_{STOR}=40 & 30°C) except for the safety factor which is 1.0.
Figure 5.6. COPH as a function of the water temperature change; parameters are those of the base case ($T_{STOR} = 40^\circ C$). Both glazed and unglazed collectors are shown.
Figure 5.7. COPH as a function of the water temperature change; parameters are those of the base case ($T_{STOR}=30$°C). Both glazed and unglazed collectors are shown.
results. The differences arise due to longer summer operating hours for glazed collectors. This reduces flow rates thus reducing pumping power. As $T_{\text{STOR}}$ decreases, the difference in operating hours between the two collectors decreases, thus the two cases give identical overall performance over a wider range of $\Delta T_w$. Unglazed collectors never exhibit better overall performance than glazed collectors because there is a minimum summer pumping power which can be obtained and is determined by maximum operating hours. Glazed collectors will operate for as many or more hours than unglazed collectors. Therefore, the latter will, at best, have the same number of operating hours and identical overall performance.

The difference which does exist between the two collectors is the area required to meet the system's energy demands. Figure 5.8 shows the collector area required for the base cases when either glazed or unglazed collectors are part of the system. It requires more unglazed collectors at low-to-mid $\Delta T_w$ and mid-to-high $T_{\text{STOR}}$. This is a direct result of collector performance (Figure 3.2). Figure 5.9 shows the collector area (unglazed) required with base case parameters except for the safety factor which is one. Collector area requirements are reduced by 50 percent when the safety factor value is changed from two to one.

Collector area has been included in the results because of its economic importance. An economic analysis of the system will be done in Chapter VI, but from the results shown it is apparent that unglazed collectors will be favored in most instances because there is little or no difference in system performance and unglazed collectors cost much less.

Another important system constraint is the pumping head required during the summer and winter operation. Figure 5.10 illustrates winter maximum and average head while Figures 5.11 and 5.12 show summer maximum
Figure 5.8. Glazed and unglazed collector area as a function of water temperature change with parameters equal to their base case values.
Figure 5.9. Unglazed collector area as a function of water temperature change with parameters equal to their base case values except the safety factor which is 1.0.
Figure 5.10. Winter maximum and average pumping head as a function of water temperature change with variables equal to their base case values.
Figure 5.11. Summer maximum and average pumping head as a function of water temperature change with variables equal to their base case values (T_{STOR} = 40°C).
Figure 5.12. Summer maximum and average pumping head as a function of water temperature change with variables equal to their base case values ($T_{STOR}$=30°C).
and average head for base case parameters. The winter head is independent of the aquifer production curve used in simulations, whereas the summer head is greatest for Aquifer A and least for Aquifer C. A small recovery factor (Aquifer A) results in large amounts of water being stored, thus higher flow rates and head. The winter head is much smaller than the summer head due to longer operating hours. Figures 5.11 and 5.12 show summer head becoming quite extreme as $\Delta T_W$ decreases to four degrees (C). Aquifers and well construction can withstand only limited amounts of head before destruction occurs. For initial evaluations, a head of 250 meters is deemed the point at which a major concern about the pumping head must be raised.

Results of the constant temperature model and a complete discussion of them has been reported by Reistad and McMahon [22]. Only specific results will be presented below.

Table 5.2 lists the values of parameters investigated along with the values used for the base case. Values used in the base case are identical for both the variable temperature model (VTM) and the constant temperature model (CTM) with the exception of $T_{\text{STOR}}$. No value was specified in VTM, but in CTM $T_{\text{STOR}}$ of 40 C was included with other base case variables. One other difference in the models is the heating season length. The CTM provides heating whenever the ambient temperature drops below 65 F. This results in heating loads 30 percent higher given the same number of units. In order to make a reasonable comparison, the total load met by a well pair must be comparable. Therefore, in initial comparisons, the number of units for the CTM was decreased from 75 to 50. The total load met by the CTM is then 5.201x10^{12} Joules and the VTM is 4.875x10^{12} Joules.

Figure 5.13 shows results of both the CTM and VTM for parameters
TABLE 5.2. VALUES OF PARAMETERS INVESTIGATED WITH THE CONSTANT TEMPERATURE MODEL, FROM [22].

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>VALUE OF PARAMETER BEING CONSIDERED</th>
</tr>
</thead>
<tbody>
<tr>
<td>k (D)</td>
<td>0.5, 5.0*, 50</td>
</tr>
<tr>
<td>b (m)</td>
<td>10, 30*, 100</td>
</tr>
<tr>
<td>SF</td>
<td>1, 2*</td>
</tr>
<tr>
<td>r_w (m)</td>
<td>0.0760*, 0.127</td>
</tr>
<tr>
<td>NB</td>
<td>25, 50 75*, 100, 125, 150</td>
</tr>
<tr>
<td>RF</td>
<td>0.4, 0.7*</td>
</tr>
<tr>
<td>T_STOR (C)</td>
<td>40*, 35, 30, 25, 20, 15, 10</td>
</tr>
<tr>
<td>ΔT_W (C)</td>
<td>4, 8, 12, 16, 20, 24, 28**</td>
</tr>
</tbody>
</table>

* Base case value.
+ Parameter descriptions can be found in Table 5.1.
++ No base case value because this is a primary variable.
Figure 5.13. COPH as a function of water temperature change for both the CTM and VTM; VTM parameters are those of the base case (TSTOR=40 °C) and CTM parameters are those of the base case except for the number of building units which is 50.
those of the base case, except in the CTM, where the number of building units is 50. Only Aquifer B of the VTM is plotted. As before, dashed lines indicate "winter only" performance and solid lines represent overall system performance. The two recharge periods considered in the CTM (2160 hrs and 540 hrs) are labeled accordingly. The CTM results bracket those of the VTM quite well. A better match would be seen if the CTM results were aligned such that the "winter only" performance curves coincided with each other. If this were done, it would be evident that at low ΔT_w, the VTM results are closest to the CTM curve with 540 hrs recharge, and at large ΔT_w, the VTM results are approximated by the CTM curve with 2160 hrs recharge. Figure 5.14 illustrates the summer operating hours as predicted by the VTM in which the parameter values are those of the base case. The summer operating hours are limited by recharge times of 2160 and 525 hrs. These recharge times are very comparable to those assumed in the CTM.

Figure 5.15 compares results of the CTM and VTM at a storage temperature of 30°C. At low-to-mid ΔT_w, the CTM again brackets the VTM, but at large ΔT_w, as discussed earlier, the VTM predicts much lower system performance. The results predicted by the CTM represent an upper limit of obtainable system performance. The difference between the two models at large ΔT_w would be minimized if, as suggested before, ΔT_w were allowed to vary (decrease) whenever T_{evap} drops below zero (°C).

Figure 5.16 shows results of both the CTM and VTM with parameters those of the base case except the safety factor which is one. Again the CTM results bracket those of the VTM. From the comparisons just discussed and others not mentioned, it appears that the CTM predicts a range which contains results predicted by the VTM.
Figure 5.14. Summer pumping hours as a function of water temperature change for the VTM; parameters are those of the base case values.
Figure 5.15. COPH as a function of water temperature change for both the CTM and VTM; VTM parameters are those of the base case ($T_{STOR} = 30$ °C) and CTM parameters are those of the base case except for the storage temperature and number of building units which are 30 °C and 50 respectively.
Figure 5.16. COPH as a function of water temperature change for both the CTM and VTM; parameters are those of the base case ($T_{\text{STOR}}=40^\circ \text{C}$) except for the safety factor which is 1.0.
With a correlation established between the two models, other results from the CTM will now be presented. The variables under investigation are permeability (k), aquifer thickness (b), well casing radius \( (r_w) \), and number building units (NB). The values of each variable investigated are listed in Table 5.2.

The effects of these variables are only manifested in altering the system's pumping power. It therefore, is instructive to look at the equation used to calculate pumping power.

\[
P = \rho gQ\Delta h = \frac{Q^2_d}{\pi bk} \ln \frac{D}{r_w}
\]

The pumping power is inversely proportional to permeability and aquifer thickness. Figure 5.17 illustrates the effects of decreasing the permeability to 500 mD. Pumping power becomes so excessive that energy feasibility does not exist over the entire \( \Delta T_w \) range. Increasing the permeability to 50 Darcy's affects system performance much the same as decreasing the safety factor to one. Figure 5.18 shows the effects of decreasing aquifer thickness to ten meters. The distance between wells \( (D) \) is also influenced by aquifer thickness in that as \( b \) decreases, \( D \) will increase; both of which lead to larger pumping power. At an aquifer thickness of ten meters energy feasibility is limited to large recharge periods and \( \Delta T_w \) above 11 C.

The pumping power is only inversely proportional to the natural logarithm of \( r_w \). Consequently, its effects are quite minimal. The number of building units affects the flow rate and the distance between wells. Both increase as NB increases which in turn results in larger pumping power. Figure 5.19 shows system performance (COPH) as a function of NB for a fixed \( \Delta T_w \) of 12 C with the other parameters at their base
Figure 5.17. COPH as a function of the water temperature change; parameters are those of the base case except for the permeability which is 500 mD, from [22].
Figure 5.18. COPH as a function of the water temperature change for the CTM; parameters are those of the base case except for the aquifer height which is 10 m, from [22].
Figure 5.19. COPH as a function of the number of building units per borehole pair for a fixed water temperature change of 12°C, with the other parameters set at their base case values, from [22].
case values. As expected, system performance decreases with increasing
NB although there is little change in winter only performance. At a
small recharge period (540 hrs) energy feasibility is limited to less
than 75 units. It is interesting to note that although system perform-
ance improves as the number of units decrease, the opposite is probably
true of economic considerations.

Thus far only pumping power has been altered upon changing para-
meter values, in actuality these parameters also influence aquifer per-
formance. As mentioned before increasing permeability decreases pumping
power, but larger permeability also increases buoyancy effects. Thus,
the temperature front will tilt which in turn lowers aquifer performance.
Claesson [7] has investigated tilting in aquifers and his results show
tilting quite excessive (60° from vertical) over a permeability of about
3*10^{11} m^2 (30 Darcy's) for an aquifer 20 meters thick with a 90 day
storage period. To evaluate the overall effect of changing permeability,
gains in system performance which result from increasing permeability
will have to be weighed against system losses due to decreasing aquifer
performance. Comparing Figures 5.13 and 5.17 shows that the gains out-
weighed the losses when increasing permeability from 0.5 to 5.0 Darcy's.

In summing up the discussion of energy feasibility, it has been
shown that under certain instances aquifer seasonal thermal energy
storage (ASTES) is feasible from an energy consumption viewpoint. To
obtain this energy feasibility, heat pump design requires further evalua-
tion. The temperature drop across the evaporator ($\Delta T_w$) is typically
going to be large compared to a temperature drop of ~ 4 C which is pre-
sently the norm. However, the ultimate feasibility must also take econo-
mic considerations into account.
VI. ECONOMIC COMPARISON

In this chapter an economic comparison will be made between the aquifer seasonal thermal energy storage (ASTES) system previously discussed and three conventional heating systems: air source heat pump (ASHP); natural gas furnace; and electric resistance furnace. In order for the ASTES system to be an economically viable alternative to conventional systems, reduced annual energy cost, as a result of increased system performance, must off-set higher initial cost.

A present worth analysis was used to obtain an accurate comparison of the four systems. To arrive at the present worth of each system, future energy and maintenance costs are returned to their present value with discount rates, then added to the system's initial cost. Since present and future costs dramatically affect the comparison's final outcome, all assumptions and cost estimates pertinent to each system will be discussed.

6.1 ASSUMPTIONS AND COST ESTIMATES

The analysis period and the useful life of the heat pumps are assumed to be 15 years. This penalizes the gas and electric resistance furnaces because their useful lives are somewhat longer, but making allowances would have little effect since energy costs dominate in the analysis and future expenditures have decreasing importance the farther they are in the future. The overall economic inflation rate was projected at seven percent over the analysis period and the discount rate (cost of capital) was assumed to be eight percent. Current inflation is much higher, but it is better to use conservative estimates in determining
economic feasibility, then if actual inflation rates prove higher than those assumed, a system with large capital cost will have an even greater economic advantage.

Table 6.1 lists heating unit cost and performance of the four different systems under investigation. The three conventional heating units are assumed to be installed in the same residence as the water source heat pump of the ASTES system (Section 3.3). Cost and performance estimates were obtained from a local heating/cooling contractor and are representative of systems currently installed in local residences applicable to this study. Water source heat pumps with the projected performance specified in Section 3.2 do not exist, therefore it was decided to use cost of presently available water source heat pumps with the thought that if they were available the price would be competitive. Installation and ducting costs are assumed to be the same for all systems and are not included in the unit price. No seasonal coefficient of performance (COPH) is listed for the water source heat pump because it varies with many ASTES system parameters. Appropriate COPH's were determined from the results presented in Chapter V. Seasonal efficiencies and COPH's for the gas and electric resistance furnaces, and air source heat pump respectively are those suggested by Thomann and Fulton [26].

Maintenance costs were set at $25 per year for gas and electric resistance furnaces, and an additional ten dollars per year was added to maintain the heat pumps. The natural gas and ASTES system also had an additional monthly service charge. Northwest Natural Gas Co. currently charges $2.50 per month, which translates into a $30 annual cost. The ASTES system's service charge includes administrative and operational expenses. There is little information available on these costs.
TABLE 6.1. COST AND PERFORMANCE ESTIMATES OF FOUR DIFFERENT HEATING UNITS.

<table>
<thead>
<tr>
<th></th>
<th>NATURAL GAS FURNACE</th>
<th>ELECTRIC RESISTANCE FURNACE</th>
<th>AIR SOURCE HEAT PUMP</th>
<th>WATER SOURCE HEAT PUMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unit Cost ($)</td>
<td>500</td>
<td>450</td>
<td>2000</td>
<td>1500</td>
</tr>
<tr>
<td>Maintenance Cost per Yr ($)</td>
<td>25</td>
<td>25</td>
<td>35</td>
<td>35</td>
</tr>
<tr>
<td>Rated Capacity</td>
<td>85,000 Btu/hr</td>
<td>85,000 Btu/hr</td>
<td>3 ton</td>
<td>3 ton</td>
</tr>
<tr>
<td>Seasonal Efficiency or COPH</td>
<td>0.65</td>
<td>0.95</td>
<td>2.0</td>
<td>varies</td>
</tr>
</tbody>
</table>

TABLE 6.2. CURRENT NATURAL GAS AND ELECTRIC POWER COSTS AND THEIR ESTIMATED INFLATION RATE.

Natural Gas : Initial Cost .... 4.8719 $/GJ (0.51389 $/therm);
Inflation Rate .. 5.2% above the economic inflation rate through 1986;
1.5% above the economic inflation rate thereafter;

Electric Power : Initial Cost .... 10.867 $/GJ (3.912 ¢/kWh);
Inflation Rate .. 2.5% above the economic inflation rate through 1986;
1.58% above the economic inflation rate thereafter.
Therefore, on the basis that service costs will be somewhat larger than
natural gas's due to smaller overall system size, $40 per year was
assumed. Energy consumption associated with producing, recharging, and
distributing water is included in the seasonal COPH. No service charge
was included for the ASHP and electric resistance furnace systems because
the service charge for electricity can be attributed to lighting and
other household power needs.

Current gas and electricity rates (August 1980) were obtained from
Northwest Natural Gas and Portland General Electric respectively and are
shown in Table 6.2. Also shown are inflation rates as predicted by
Oregon Department of Energy. With an overall inflation rate of seven
percent, the inflation rate of natural gas becomes 12.2 percent through
1986, then drops to 8.5 percent thereafter. The inflation rate of elec-
tric power becomes 9.5 percent through 1986, then drops to 8.58 percent
thereafter.

In addition to the heating unit cost, the ASTES system's initial
capital outlay must also include the cost of wells, pumping equipment,
collectors and distribution system.

Well costs were estimated using the Geothermal Resources Council
Special Report No. 7 [1]. Drilling, casing, and miscellaneous cost are
combined to arrive at a total well cost. Well drilling costs to a depth
of 500 feet using cable or rotary rigs were assumed to be as follows:

- $1.10 per inch of diameter per foot of depth in "soft" rock;
- $2.75 per inch of diameter per foot of depth in "hard" rock.

Casing costs are estimated at $1.16 per inch diameter per foot of depth
and full depth casing is assumed for all wells. An additional 15 percent
is added to drilling and casing cost to account for miscellaneous cost (cement, drilling mud, standby time, etc.).

The inside case diameter is sized according to the requirements for various capacity pumps and is approximately as follows:

<table>
<thead>
<tr>
<th>FLOW RATE</th>
<th>CASING DIAMETER</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt;100 gpm</td>
<td>6 inch</td>
</tr>
<tr>
<td>&lt;500 gpm</td>
<td>9 &quot;</td>
</tr>
<tr>
<td>&lt;2000 gpm</td>
<td>12 &quot;</td>
</tr>
</tbody>
</table>

Assuming drilling will be done in half hard rock and half soft rock, the cost for a 250 foot well is shown for various casing diameters in Table 6.3.

A local pump service company was contacted to estimate the cost of installing a pump at various flow rates for a 200 foot setting and 160 feet drawdown. Their estimates are shown in Figure 6.1 along with a power curve least square fit.

**TABLE 6.3. WELL COSTS FOR A 250 FOOT WELL DRILLED IN HALF "HARD" AND HALF "SOFT" ROCK ARE LISTED FOR SEVERAL INSIDE CASING DIAMETERS.**

<table>
<thead>
<tr>
<th>CASING DIAMETER</th>
<th>6 inch</th>
<th>9 inch</th>
<th>12 inch</th>
</tr>
</thead>
<tbody>
<tr>
<td>drilling cost</td>
<td>$ 2888</td>
<td>$ 4331</td>
<td>$ 5775</td>
</tr>
<tr>
<td>casing cost</td>
<td>1733</td>
<td>2599</td>
<td>3465</td>
</tr>
<tr>
<td>miscellaneous cost</td>
<td>693</td>
<td>1040</td>
<td>1386</td>
</tr>
<tr>
<td>total cost</td>
<td>$ 5314</td>
<td>$ 7970</td>
<td>$10626</td>
</tr>
</tbody>
</table>

SolaRoll collectors, specified in Section 3.4 have been installed in swimming pool applications for a total cost of eight-to-ten dollars a square foot. This includes pumps, controls and any other accessories needed for an operational system. In this analysis the cost of installed collectors was assumed to be nine dollars per square foot.
Figure 6.1. Installed pump cost as a function of flow rate for a 200 foot setting and 160 feet drawdown.
The cost of a distribution system is mainly dependent on the density and number of buildings. A density of 1000 units per km$^2$ is assumed in this analysis (Table 3.2). Since there are only 75 residences associated with a pair of wells, several of these groups will have to be combined to obtain the desired number of residences. Davison, Harris, and Martin [10] estimated the cost of a 1000-residence distribution system, with a density of 907 units per km$^2$, at $2300 per residence (1975 dollars). Bloomster, Fassbender, and McDonald [4] estimated capital cost of district heating distribution systems for various population densities. With a population density of 3500 people/km$^2$, as assumed in this study (Table 3.2), they estimated capital cost at approximately $2500 per residence (1977 dollars). This assumes a two pipe system, 90 C fluid temperature with a 52 C temperature drop, and a total population of 12,000. In this study fluid temperature and temperature drop are much lower and total population smaller, therefore capital costs are going to be somewhat higher. In this analysis, an exact cost was not assumed for the distribution system. Rather, all other costs were calculated, then the amount which could be charged to the distribution system and still be competitive with conventional systems was determined. The objective of this short discussion is to establish some reasonable estimates of the distribution system's costs.

Two heating loads were considered in the economic analysis; the first was that specified in Section 3.3 in which 65 GJ of heating was required from November 1 through March 31 for each residence. The second heating load considered was calculated in the same manner except heating was provided whenever the temperature dropped below 18.3 C (65 F). This resulted in 87 GJ of heating from January 1 through December 31 for each
residence. The latter heat load is a more realistic estimate for the economic analysis.

6.2 PRESENT WORTH CALCULATIONS

After compiling each system's initial cost, the major task left is determining the present value of energy and maintenance cost throughout the life of each system. As stated before, future energy and maintenance costs are returned to their present value with discount rates. The formula for estimating future cost is as follows:

\[ FC = PC (1+i)^{n-1} \]  \hspace{1cm} (6.1)

where \( FC \) = Future cost

\( PC \) = Present cost

\( i \) = Inflation rate per year

\( n \) = Number of years into the future with inflation rate \( i \).

Equation 6.1 also applies to returning future costs to their present value. In this application "\( i \)" becomes the discount rate. If a uniform annual cost \( PC \) occurs throughout the life of a system, then the present value \( (PV) \) of all these future annual costs becomes

\[ PV = PC + PC \left( \frac{1+i}{1+d} \right) + \ldots + PC \left( \frac{1+i}{1+d} \right)^{n-1} \]  \hspace{1cm} (6.2)

where \( d \) is the discount rate. If the inflation rate changes within the analysis period, the present value of all future annual cost becomes

\[ PV = PC + PC \left( \frac{1+i}{1+d} \right) + \ldots + PC \left( \frac{1+i}{1+d} \right)^{n-1} + PC \left( \frac{1+i}{1+d} \right)^{n-1} \left( \frac{1+j}{1+d} \right) \]

\[ + \ldots + PC \left( \frac{1+i}{1+d} \right)^{n-1} \left( \frac{1+j}{1+d} \right)^{N} \]  \hspace{1cm} (6.3)
where $N$ is the number of years into the future beyond "n" with inflation rate $j$. In both Equations 6.2 and 6.3 PC can be factored out of each term in the summation leaving a summation with only the variables $i$, $j$, $d$, $n$, and $N$. This will be called the present value multiplying factor (PVMF). The present value of all future cost now becomes

$$PV = PC \times PVMF.$$  \hspace{1cm} (6.4)

Three PVMF's of interest in this analysis are shown in Table 6.4.

**TABLE 6.4. PRESENT VALUE MULTIPLYING FACTORS APPLICABLE TO GAS AND ELECTRIC HEATING SYSTEMS.**

<table>
<thead>
<tr>
<th></th>
<th>$i$</th>
<th>$j$</th>
<th>$d$</th>
<th>$n$</th>
<th>$N$</th>
<th>PVMF</th>
</tr>
</thead>
<tbody>
<tr>
<td>PVMF1</td>
<td>12.2</td>
<td>8.5</td>
<td>8.0</td>
<td>6</td>
<td>9</td>
<td>17.7612</td>
</tr>
<tr>
<td>PVMF2</td>
<td>9.5</td>
<td>8.58</td>
<td>8.0</td>
<td>6</td>
<td>9</td>
<td>16.1175</td>
</tr>
<tr>
<td>PVMF3</td>
<td>7.0</td>
<td>0.0</td>
<td>8.0</td>
<td>15</td>
<td>0</td>
<td>14.0657</td>
</tr>
</tbody>
</table>

PVMF1 and PVMF2 are substituted in Equation 6.4 to calculate the present value of future energy cost for natural gas and electric (electric resistance, ASHP, and ASTES) systems respectively. PVMF3 is used in Equation 6.4 to calculate the present value of future maintenance cost for all four systems.

The present worth of a system is the sum of all initial costs and the present value of all future costs. For the three conventional systems this can be expressed in equation form as follows:

$$\text{PRESENT WORTH} = \text{UNIT COST} + \frac{\text{PRESENT}^6 \text{MAINTENANCE} \times \text{PVMF3}}{\text{COST}} + \frac{\text{PRESENT} \text{ENERGY} \times \text{OR}}{\text{COST}} \text{PVMF2}.$$  \hspace{1cm} (6.5)

---

6 Any service charge is included with maintenance cost.

7 PVMF1 is used for the natural gas system and PVMF2 is used for both the electric resistance and ASHP system.
The ASTES system's present worth includes several additional initial costs and can be calculated knowing the seasonal COPH, collector area, and maximum summer flow rate for a given $T_{STOR}$ and $\Delta T_W$ combination. The maximum summer flow rate is used in calculating well and pump cost from Table 6.3 and Figure 6.1 respectively. COPH and collector area are needed to calculate energy and collector cost. In equation form, the present worth of the ASTES system is as follows:

\[
PW = \text{UNIT COST} + \text{COLLECTOR COST} + \text{WELL & PUMP COST} + \text{DISTRIBUTION COST} + \text{PRESENT MAINTENANCE COST} \times PVMF3 + \text{PRESENT ENERGY COST} \times PVMF2
\]  

(6.6)

Using Equation 6.6 in conjunction with Chapter V results, it must be remembered that well, pump, and collector cost have to be divided by NB to obtain the cost to a single residence.

6.3 RESULTS AND DISCUSSION

The three conventional systems' present worth as predicted by Equation 6.5 is shown in Table 6.5. The ASHP has the lowest present worth for both heat loads, with natural gas somewhat higher and electric resistance the highest.

<table>
<thead>
<tr>
<th>SYSTEM</th>
<th>PRESENT WORTH ($) WITH AN ANNUAL HEATING LOAD OF 6.500x10^{10} Joules</th>
<th>8.704x10^{10} Joules</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural Gas</td>
<td>9927</td>
<td>12861</td>
</tr>
<tr>
<td>Electric Resistance</td>
<td>12785</td>
<td>16849</td>
</tr>
<tr>
<td>Air Source Heat Pump</td>
<td>8185</td>
<td>10115</td>
</tr>
</tbody>
</table>

\footnote{Well and pump cost obtained from Table 6.3 and Figure 6.1 is multiplied by two because there are two wells per NB residences; each containing a pump.}
The ASTES system's present worth has been determined for all combinations of $T_{\text{STOR}}$ and $\Delta T_w$ for the base case variables and these same variables, with the exception of the safety factor which was set equal to one. For each combination of $T_{\text{STOR}}$ and $\Delta T_w$ the present worth was calculated for a heating load of 65 GJ and 87 GJ. The results of these calculations are shown in Figures 6.2 through 6.5. Solid lines represent the three conventional systems' present worth, dots represent the ASTES system's present worth minus distribution system cost, and dashes represent the ASTES system's present worth minus collector and distribution cost. Figures 6.6 and 6.7 show the individual costs which sum to give the ASTES system's present worth at $T_{\text{STOR}}$ equal to 35 C.

Distribution costs were not included in Figures 6.2 through 6.5 because they would put the ASTES system's present worth off scale. Estimates of distribution cost can vary extensively. Therefore, it is more informative to examine the amount that can be spent on a distribution system and still be economically competitive with conventional systems. Present worth is not plotted at low $\Delta T_w$ because design flow rates become extremely large ($\approx 2000$ gpm) and it is highly unlikely that this much water could be produced from an aquifer.

Figures 6.6 and 6.7 show that collector and energy costs dominate the ASTES system's present worth. Maintenance and unit cost influence the present worth somewhat at $\$1500$, but well and pump costs are relatively small, at the 250 foot depth range, and have little influence on the ASTES system's present worth.

It is seen from Figures 6.2 through 6.5 that the ASTES system's minimum present worth (minus distribution system cost) is almost independent of $T_{\text{STOR}}$. The only difference is the minimum occurs at higher $\Delta T_w$ with
Figure 6.2. The ASTES system's present worth as a function of water temperature change for a heating load of 65 GJ is compared with the present worth of three conventional systems; ASTES parameters are those of the base case.
Figure 6.3. The ASTES system's present worth as a function of water temperature change for a heating load of 65 GJ is compared with the present worth of three conventional systems; ASTES parameters are those of the base case except for the safety factor which is 1.0.
Figure 6.4. The ASTES system's present worth as a function of water temperature change for a heating load of 87 GJ is compared with the present worth of two conventional systems; ASTES parameters are those of the base case.
Figure 6.5. The ASTES system's present worth as a function of water temperature change for a heating load of 87 GJ is compared with the present worth of two conventional systems; ASTES parameters are those of the base case except for the safety factor which is 1.0.
Figure 6.6. A breakdown of the ASTES system's present worth as a function of water temperature change for heating loads of 65 GJ and 87 GJ; parameters are those of the base case with storage temperature equal to 35°C.
Figure 6.7. A breakdown of the ASTES system's present worth as a function of the water temperature change for heating loads of 65 GJ and 87 GJ, parameters are those of the base case except for the safety factor which is 1.0.
increasing $T_{\text{STOR}}$. The uniform and shifting minimum is a result of the opposing economic effect $\Delta T_W$ and $T_{\text{STOR}}$ have on energy and collector cost. From Chapter V results it is evident that energy cost decreases and collector cost increases with larger $T_{\text{STOR}}$. Because collector cost increases with larger $T_{\text{STOR}}$, its influence on the ASTES system's present worth increases. Collector cost decreases with increasing $\Delta T_W$ thus the minimum present worth tends to shift towards larger $\Delta T_W$ as $T_{\text{STOR}}$ increases. When both distribution and collector cost are subtracted from the ASTES system's present worth, energy costs dominate and conditions of maximum performance coincide with minimum present worth.

Also exhibited in the ASTES system's present worth is its rapid increase at large $\Delta T_W$. A sharp increase in energy consumption as a result of decreased system performance is responsible for this trend. As mentioned in Chapter V, by operating at variable $\Delta T_W$ the sharp drop-off in performance at elevated $\Delta T_W$ would be eliminated or reduced. Thus, under these conditions the sharp rise in present worth would also be eliminated or reduced.

The principal interest in the ASTES system's minimum present worth comes in comparing it to the present worth of conventional systems. Figure 6.2 shows all four systems' present worth with parameters of the ASTES system at their base case values and a heating load of 65 GJ. If the distribution system costs are approximately $2500-3000$ per residence then the ASTES system is not economically feasible even when compared with the electric resistance system. Figure 6.3 is the same case except the safety factor equals one. With these parameters the ASTES system is competitive with electric resistance if the distribution system costs are less than approximately $4250$ per residence. Natural gas and ASHP systems are
economically more attractive when a reasonable cost is assigned to the distribution system.

The ASTES system's present worth with parameters at their base case values and a heat load of 87 GJ is shown in Figure 6.4. At $16,849, the electric resistance system's present worth is too large to include. Again in this case the ASTES system can only compete with the electric resistance system. Figure 6.5 is the same case as Figure 6.4 except for the safety factor which equals one. With these parameters the ASTES system is competitive with natural gas if the distribution system costs are less than approximately $3250 per residence. The ASTES system is far more attractive than electric resistance, but the present worth is still not low enough to compete with an ASHP system.

Up to this point the ASTES system is unable to compete with an ASHP system. Since solar collectors are a major system cost, investigating an alternative to a solar recharge system could cause ASTES systems to be more competitive with ASHP systems. Recharging warm waste water from a power or industrial plant is a possible alternative to a solar recharge system at a minimal cost.

To examine this alternative, the ASTES system's present worth minus distribution and collector cost has also been included in Figures 6.2 through 6.5. With collector cost omitted, the ASTES system's present worth is quite dependent on $T_{STOR}$ (lower present worth as $T_{STOR}$ increases). At a heat load of 65 GJ, it is questionable whether the cost of a distribution and recharge system could remain low enough for the ASTES system to compete with the ASHP system. At a heat load of 87 GJ, the ASTES system is competitive with the ASHP system if distribution and recharge cost
are less than about $3000-3500. This is only true at $T_{STOR}$ equal to 35 to 40 C. At $T_{STOR}$ below 35 C, the economic feasibility of ASTES systems becomes questionable when compared to ASHP systems.
VII. CONCLUSION

It has been shown that several parameters affect the overall performance of the ASTES system described in Chapter III. Parameters investigated include aquifer permeability and dimensions, safety factor, number of residences, storage temperature, and temperature drop of the water. Of these parameters, $\Delta T_w$ was identified as an important factor because it is a major predictor of fluid flow rate which in turn affects heat pump design and aquifer performance.

Ranges of $\Delta T_w$ where the overall system performance is high enough to constitute energy feasibility or too low for energy feasibility to exist have been discussed for several different values of the above parameters. In general, any parameter change that decreases the pumping power without decreasing the heat pump evaporator temperature will improve the overall system performance and expand the region of energy feasibility. Establishing regions where energy feasibility exist allows further, more involved studies to avoid those regions where there is no possibility of energy feasibility.

In designing heat pumps for use with ASTES, the temperature drop across the evaporator ($\Delta T_w$) is typically going to have to be large compared to a temperature drop of $\sim 4\, \text{C}$ which is presently the norm for water source heat pumps.

An economic analysis was also performed to determine if an ASTES system could compete with conventional heating systems. The analysis indicates that the ASTES system described in Chapter III can easily compete with electric resistance furnaces, but only under limited conditions is it able to compete with natural gas furnaces. It is totally unable to
compete with air source heat pumps. If solar collectors are replaced with an alternative recharge system (waste thermal energy), the ASTES system becomes substantially cheaper than electric resistance furnaces and very competitive with natural gas furnaces, but only under limited conditions is it able to compete with air source heat pumps. In all these analyses, the cost assigned to the distribution system is a major factor in determining the economic feasibility of the ASTES system.

The performances and economic results presented in this thesis serve to identify important parameters and establish regions of these parameters where ASTES systems have performance and economic feasibility. Ultimately this information can aid in the design of such systems. While certain aquifer variables can be generalized, several are unique to a specific aquifer, therefore many of the conclusions drawn in this thesis must be qualitative and the results of modeling "typical" aquifers should be seen as illustrative rather than explicitly predictive. To accurately predict ASTES system performance, involved aquifer and collector models need to be incorporated into ASTES system model and yearly simulations made using detailed site specific aquifer and weather information. Additional work should be directed towards this end.
BIBLIOGRAPHY


### NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tr>
<td>a</td>
<td>Constant, $\rho A^C_A/4 \rho R^C_R$</td>
</tr>
<tr>
<td>A</td>
<td>Area</td>
</tr>
<tr>
<td>ANB</td>
<td>Floor area per building unit</td>
</tr>
<tr>
<td>ASHP</td>
<td>Air source heat pump</td>
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<tr>
<td>ASTES</td>
<td>Aquifer seasonal thermal energy storage</td>
</tr>
<tr>
<td>b</td>
<td>Aquifer thickness</td>
</tr>
<tr>
<td>C</td>
<td>Heat capacity</td>
</tr>
<tr>
<td>C-DD</td>
<td>C degree days per heating season</td>
</tr>
<tr>
<td>Ĉ-DD</td>
<td>Monthly total C degree days</td>
</tr>
<tr>
<td>COPH</td>
<td>Coefficient of performance in heating</td>
</tr>
<tr>
<td>CTM</td>
<td>Constant temperature model</td>
</tr>
<tr>
<td>d</td>
<td>Discount rate (cost of money)</td>
</tr>
<tr>
<td>D</td>
<td>Distance between borehole pairs (wells)</td>
</tr>
<tr>
<td>dw</td>
<td>Width of a stream channel</td>
</tr>
<tr>
<td>( \dot{E}_{evap} )</td>
<td>Rate at which energy is absorbed by refrigerant in the evaporator</td>
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<tr>
<td>FC</td>
<td>Future cost</td>
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<tr>
<td>g</td>
<td>Gravitational acceleration</td>
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<tr>
<td>h</td>
<td>Head</td>
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<tr>
<td>H</td>
<td>Monthly average daily radiation on a horizontal surface</td>
</tr>
<tr>
<td>HL</td>
<td>Heat load</td>
</tr>
<tr>
<td>i</td>
<td>Inflation rate per year</td>
</tr>
<tr>
<td>j</td>
<td>Inflation rate per year</td>
</tr>
<tr>
<td>k</td>
<td>Aquifer permeability (specific or intrinsic)</td>
</tr>
<tr>
<td>K</td>
<td>Coefficient of permeability (hydraulic conductivity)</td>
</tr>
<tr>
<td>( K^R )</td>
<td>Thermal conductivity of the cap rock</td>
</tr>
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</table>
\( l \)  Length dimension

\( m \)  Volume flow rate within a steam channel

\( M \)  Total number of nodes in the S-plane

\( n \)  Number of years into the future with inflation rate "i"

\( N \)  Total number of nodes in the z-plane, or
Number of years into the future beyond "n" with inflation
rate "j"

\( NB \)  Number building units per borehole pair.

\( P \)  Pumping power

\( PC \)  Present cost

\( PV \)  Present value

\( PVMF \)  Present value multiplying factor

\( PW \)  Present worth

\( Q \)  Total volume flow rate through the system

\( r \)  Radius

\( RF \)  Recovery factor

\( S \)  Flow channel area

\( S_{\text{max}} \)  Maximum flow channel area which the temperature front
has influenced, \( t_{\text{inj}} \rho_w C_w \ m/b \rho_a C_a \)

\( SF \)  Safety factor

\( t \)  Time

\( T \)  Temperature

\( T_{\text{a}} \)  Monthly average daily ambient temperature

\( T_i \)  Collector inlet temperature

\( T_D \)  Inside design temperature

\( \Delta T_W \)  Water temperature change through heat pumps, collectors,
or aquifer

\( T_{\text{STOR}} \)  Storage temperature of the water in the aquifer
UA Overall loss coefficient based on floor area

\( v \) Velocity

\( v_{thermal} \) Velocity the temperature front moves through an aquifer

\( V \) Volume

\( VTM \) Variable temperature model

\( z \) Distance into the cap rock above or below aquifer

**Greek:**

\( \zeta \) Variable substitution, \( k_R S/b \rho_w C_w \)

\( \eta \) Variable substitution, \( z/2b \)

\( \eta_c \) Solar collector efficiency

\( \theta \) Variable substitution, \( T-T_\infty \)

\( \theta' \) Newly calculate node temperatures

\( \theta_\infty \) Variable substitution, \( T_{inj}-T_\infty \)

\( \mu \) Viscosity of water

\( \rho \) Density

\( \tau \) Variable substitution, \( k_R t/b^2 \rho_A C_A \)

\( \tau^* \) Variable substitution, \( \tau-\zeta/m \)

\( \phi \) Aquifer porosity

\( \psi \) stream lines

**Subscripts:**

\( a \) Ambient condition

\( \text{app} \) Approach

\( A \) Aquifer condition

\( \text{evap} \) Evaporator condition

\( f \) Fluid

\( i \) Node in the z-plane
inj  Injection condition
j    Node in the S-plane
m    Aquifer matrix
prod Production condition
R    Cap rock
W    Water
w    Well borehole
∞    infinity
APPENDIX B

LIST OF COMPUTER PROGRAM
PROGRAM AQUA (INPUT,OUTPUT,TAPE12=INPUT,TAPE10=OUTPUT,TAPE7,TAPE9)

C THIS PROGRAM SIMULATES AN ANNUAL ENERGY STORAGE SYSTEM WHICH INPUTS HOT WATER INTO AN AQUIFER DURING THE SUMMER (VIA LOW COST SOLAR COLLECTORS) FOR RETRIEVAL IN THE WINTER. A WEATHER TAPE IS USED TO DETERMINE THE AMOUNT OF ENERGY WHICH CAN BE COLLECTED. IN THE WINTER, THE HOT WATER IS PUT THROUGH A HIGH PERFORMANCE WATER SOURCE TO SUPPLY SPACE HEATING.

DIMENSION TSTOR(7),DWTP(10),TEHPM(7,10),T4EHPM(7,10),TLNOT(7,10)
DIMENSION TVOLW(7,10),TWHPM(7,10),SQCQLL(7,10),SDQCOL(7,10)
DIMENSION THOURS(7,10),TMXFL(7,10),THDRC(7,10),TMAXO(7,10)

HOUR=1.0
THLOAD=0.0

C SET SUMS TO ZERO
DO 7 I=1,5
DO 5 J=1,9
THOURC(I,J)=0.0
TLNOT(I,J)=0.0
TMAXO(I,J)=0.0
TMXFL(I,J)=0.0
THOURS(I,J)=0.0
TEHPM(I,J)=0.0
TSEHM(1,J)=0.0
TVOLW(I,J)=0.0
TWHPM(1,J)=0.0
SQCQ(1,J)=0.0
SDRCOL(I,J)=0.0
5 CONTINUE
7 CONTINUE

READ (9,9) IH,AAA,BBB,DTAPP,RWL,CLC,FTRA,FRUL

C LOOP TO SUM ENERGY COLLECTED
DO 20 K=1,IH
READ (9,6) SINSL,TOUT
IF (SINSL.LE.0.0) GO TO 20
DO 17 I=1,5
TSTOR(I)=4.5-5.*FLOAT(I)
DO 11 J=1,7
DWTP(J)=4.0*FLOAT(J)
TTEVAP=TSTOR(I)-DWTP(J)-DTAPF
C IF TEMPERATURE OF EVAPORATOR IS LESS THAN ZERO (C) BI-PASS CALCULATION
IF (TTEVAP.LE.0.0) GO TO 13
TATJ=(TSTOR(I)-DWTP(J))*1.84-32.
IF (CLC.EQ.1.0) GO TO 12
TITOI=(TATJ-TOUT)/SINSL
IF (TITOI.LE.15.) GO TO 11
IF (TITOU.EQ.9.0) GO TO 10
SQCQ(L)=SINSL-0.83633*SINSL*EXP(22.05436*TITOI))**3.154
10 CONTINUE
SQCQ(L)=(SINSL*.96347-.8039*(TOUT-TOUT))**3.154
16
CONTINUE
SCELL=SIML*FRTA-FRUL*(TAGUA-TOUT)*3.154
IF (SCOLL.LE.0.0) GO TO 11

CONTINUE
IF (SCOLL.LE.TMAXO(I,J)) GO TO 8
THAO(I,J)=SCOLL

CONTINUE
SCOLL=SCOLL+SCOLL
SCELL(I,J)=SCELL(I,J)+SCOLL
SCELL(I,J)=SCELL(I,J)+SCOLL
THOURC(I,J)=THOURC(I,J)+HOUR

CONTINUE
CONTINUE
CONTINUE
CONTINUE
READ (9,14) JH,TIND,UA,UNT,FUNT,DENSR,HCAPR,PORO
READ (9,21) CDAY,RF,SF,PERM,HAO,AMT

C LOOP TO SUM WINTER HEAT LOAD, WATER USE, PUMPING POWER
HCAP=4180.
DENSR=977.
vCEN=19.
FRAS=0.076
1.=3.1416
PERF=0.7

HCAU=PO0-DENS*HCAP+1.0-PORO)*DENSR*HCAP
DO 40 K=1,JH
READ (9,15) TOUT
IF (TOUT.GE.65.0) GO TO 40
HTLGAD=((TIND-TOUT)/1.8)*UOWNT*FUNT*3600.
THLOAD=THLOAD+HTLGAD
DO 37 I=1,5
DO 31 J=1,9
TTEVAP=TSTOR(I)-DWTP(J)-DTAPP
IF (TTEVAP.LE.0.0) GO TO 33
AFFVAP=(STOR(I)+DWTP(J)*RF-2.0)-DTAPP+273.)*1.8
ACOP=(530.0/530.-AFFVAP)*(1.1953-.001498*AFFVAP)
EHAT=1.0-1.0)/ACOP*CDAY*UA*UNr+FUN+3600.*24.
TWIN=EHAT*SF/(RF*HCAP*DENSR*DWTP(J))

C TEMPERATURE OF PRODUCTION WELL.
IF (AMT.EQ.1.0) GO TO 23
TWIN=(DWTP(J)*(1.0-AAA*((TVOLW(I,J)/TVINJ)**BBB))+ISTOR(I)
1-DWTP(J)+273.)*1.8
GO TO 25

CONTINUE
TWIN=(DWTP(J)*(AAA+BBB*TVOLW(I,J)/TVINJ)+TSTOR(I)
1-DWTP(J)+273.)*1.8
GO TO 25

C Named locations:

TEVAP=TWIN-1.3*(DTAPP+DWTP(J))
IF (TEVAP.LE.491.4) GO TO 28
COP=530.0/530.-TEVAP)*(1.1953-.001498*TEVAP)
SHPPM=THLOAD-(1.0-1.0/COPH)
SEHPM = EHPUM + EHPUM
VOLU = EHPUM + SF/(DENS * HCAP * DWT(I,J))
WHFU = HTLOAD / DWT(I,J)
TEHPM(I,J) = TEHPM(I,J) + SEHPM
TSEHPM(I,J) = TSEHPM(I,J) + SEHPM
TVOLU(I,J) = TVOLU(I,J) + VOLU
TUHPM(I,J) = TUHPM(I,J) + WHFU
TMAX = VOLU / 3600.
IF(TMAX.LE.TMAXFL(I,J)) GO TO 27
TMAXFL(I,J) = TMAX
27 CONTINUE
THOURS(I,J) = THOURS(I,J) + HOUR
GO TO 31
23 CONTINUE
TLNOT(I,J) = TLNOT(I,J) + HTLOAD
33 CONTINUE
37 CONTINUE
40 CONTINUE
C PRINT CONSTANTS USED IN CALCULATIONS.
WRITE (10,100) AAA, RF
WRITE (10,102) BBB, SF
WRITE (10,104) CDDAY, HA0
WRITE (10,106) FUNT, HCAP
WRITE (10,108) UNT, DENS
WRITE (10,110) RCAS, HCAPR
WRITE (10,112) TIND, DENSR
WRITE (10,114) DTAPP, PORO
WRITE (10,116) UA, PERM
WRITE (10,119) THLOAD
C LOOP TO PRINT DATA AND PERFORM ADDITIONAL CALCULATIONS.
DO 57 I = 1, 5
WRITE (10,120) TSTOR(I)
WRITE (10,122)
DO 47 J = 1, 9
TTEVAP = TSTOR(I) - DWT(I,J) - DTAPP
IF(TTEVAP.LE.0.0) GO TO 53
VISC = .0022416 - .00042666 * ALOG(TSTOR(I) - DWT(I,J) *.5)
AEVAP = (TSTOR(I) - DWT(I,J) * (RF - 2.0) - DTAPP + 273.)/1.8
AOOP = (580. / (580. - AEVAP)) * (1.1953 - .001498 * AEVAP)
EHEAT = (1.0 - 1.0 / AOOP) * CDDAY * UA * UNT * FUNT * 3600. * 24.
TAOUA = TSTOR(I) - DWT(I,J)
RWELL = SRT(EHEAT * SF / (DWT(I,J) * RF * RWL * HA0 * RHOACA))
COLA = EHEAT * SF / (RF * 3600. * SOCQT(I,J))
TMAXS = TMAXD(I,J) + COLA / (DENS * HCAP * DWT(I,J))
CNEN = 3600. * COLA / (HCAP * DENS * DWT(I,J))
TVINJ = CNEN * SOCQT(I,J)
TEHINJ = TVINJ + DENS * HCAP * DWT(I,J)
CAP = DENS * HCAP * DWT(I,J)
RPM = ALOG(RWELL / 2.0 / RCAS) * VISC / (PI * PERM * DENS + .9 * HA0)
110

122 FORMAT (///15X,13HTemp storage=,F8.3)
122 FORMAT (///33X,*volume volume summer winter winter total over*,
123X,*winter summer*,/10X,4HTemp,12X,*collt water water*
1 PUMP PUMP RUN ENERGY ALL ENERGY ENERGY winter max sum
1MER MAX*)
124 FORMAT (3X,*dwt aqua well area inj used energy ene*
1GY h pump used coph inj retriev hours flow hours flow=*
2)
125 FORMAT (11X,*winter winter winter summer
1 summer summer load not*,/10X,*avg flow max
2head avg head avg flow max head avg head*
3et by hp*)
126 FORMAT (2X,2F6.2,2X,8(E10.4X))
127 FORMAT (24X,7(E10.4X))
128 FORMAT (10X,7(E10.4X),/)
END
APPENDIX C

BASE CASE COMPUTER OUTPUT
<p>| | |</p>
<table>
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<tr>
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<tr>
<td><strong>AAA</strong></td>
<td>.479</td>
</tr>
<tr>
<td><strong>BBB</strong></td>
<td>1.602 BTU/FT**2 HR F</td>
</tr>
<tr>
<td><strong>C DEGREE DAYS</strong></td>
<td>2303.000 C DAY</td>
</tr>
<tr>
<td><strong>AREA PER UNIT</strong></td>
<td>139.000 M**2/UNIT</td>
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<tr>
<td><strong>NUMBER OF UNITS</strong></td>
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<tr>
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<td><strong>AQUIFER THICKNESS</strong></td>
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<tr>
<td><strong>DENSITY OF WATER</strong></td>
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<td><strong>RADIUS OF WELL CASING</strong></td>
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<td><strong>APPROACH TEMPERATURE</strong></td>
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<td><strong>UNIT UA</strong></td>
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<td><strong>HEAT CAPACITY ROCK</strong></td>
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<td><strong>DENSITY OF ROCK</strong></td>
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<td><strong>POROSITY</strong></td>
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<td><strong>PERMEABILITY</strong></td>
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**Total Heat Load** = .4875E+13
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<th>Temp</th>
<th>Collt</th>
<th>Well</th>
<th>Area</th>
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