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Heat Transfer Analysis of Heat Pump Ground-Sourcing Using Large Boreholes and Concentric Flow

Kent Udell[1] and Kevin Maher[2],
[1]University of Utah
Department of Mechanical Engineering, Salt Lake City, USA
Kent.Udell@utah.edu; Kevin.Maher @aexp-tech.com
[2]Applied Exponential Technologies, LLC

Abstract **-** Optimal performance of ground source heat pumps requires that pumped fluid temperatures are at a maximum for heating applications, and at a minimum for cooling applications. Concentric flow, where a fluid is pumped into either the center tube or the annulus, is seen as an approach that brings ground-sourcing heat transfer closer to the ideal of the exiting fluid temperature equaling the background soil temperature. For varying operation cycles of real systems, representative analyses need to account for both variations in vertical and radial temperatures during periods when the fluid is being pumped, heated or cooled, or at rest between cycles. The model presented in this paper represents such an analysis. A coupled transient model was developed to predict soil temperature fields and borehole fluid temperature transients, both during active pumping and inactive thermal relaxation "rest" periods when the heat pump is off, and the well bore fluid exchanges heat with the adjacent media of variable temperature outside of the well bore. The wellbore heat transfer model is verified through comparison to field data, allowing deeper understanding of observed temperature responses to various periods of heating and rest. The model was applied to 100 hours of operation with two different schedules: "on" for a period of time equal to one wellbore fluid flush, and "off" for a period twice that, and the other of 20 minutes on and 40 minutes off. Four different wellbore diameters were chosen for comparison: 5 cm (2"), 10 cm (4"), 15 cm (6") and 20 cm (8"). The results are represented through calculations of overall compressor energy use for both heating and cooling operation. It was found that compressor energy use decreases with wellbore diameter with 25% more energy used for heat pump operation for a 2" diameter wellbore than a 6" or 8" wellbore. Further, small wellbores produce greater temperature variations and thus are more likely to experience subfreezing temperatures during heat pump operation.

Keywords**:** Ground Source Heat Exchanger, Concentric Tubing, Heat Transfer Effectiveness, Optimum Wellbore Diameter, Compressor Energy Savings

1. Introduction

As we move towards sustainable energy sources such as wind and solar to meet our energy needs, the conversion of building heating and cooling systems to heat pumps is an obvious needed step, as a large fraction of our fossil fuel energy use is for residential and commercial heating. Heat pumps provide both heating and cooling functions, taking advantage of ground temperatures that are normally greater than air temperatures to reduce the compressor energy to heat buildings. Further, since ground temperatures are lower than the temperature in a building to be cooled, ground-sourcing dramatically reduce cooling power needs in the summer months when the air temperature is much higher than the ground temperature. Ground source heat pumps use less energy than air source heat pumps when the ambient temperatures are greater than the ground temperatures in summer or higher than the ambient temperatures in the winter.

While above ground heat exchangers have known performance specifications, transients in the borehole heat exchangers are less well understood. The ideal performance of a borehole heat exchanger would be the injection of water at any temperature and for it to return at the far-field media temperature. However, either fluid to media heat transfer resistances or adjacent media radial temperature gradients, prohibit that level of effectiveness as the temperatures in the adjacent media changes with time. The effectiveness also changes with the transients of cycling times of the above-ground equipment driving the heat transfer and pumping rates. Thus, the simulation of the overall energy transport, during both periods of operation and rest, becomes complex, and has become a topic of research in recent years.

A concentric borehole configuration has better performance that a U-Tube configuration, Fang, *et al*, [7], and allows options of greater wellbore diameters. Thus, this study is restricted to concentric configurations. Studies of the performance of concentric borehole heat exchanger are found in the literature. Zanchini, *et al* [1] and [2], studied the performance of small concentric wellbores. Rees and He [3] developed a three-dimensional model of heat transfer near a concentric wellbore.

Experimental work has also been conducted, Acuna, et al, [4], Beier, et al, [5] and [6], using fiber cable for downhole temperature measurements.

In addition to the theoretical work of Beier, et al, [6], Fang, et al, [8], developed a model for the simulation of deep boreholes. Li, et al [9], constructed a comprehensive model using commercial Multiphysics software to describe the coupled heat transfer in the borehole and adjacent geologic media.

The Fang, et al, [7], model provided a theoretical basis for the temperature distribution in a concentric wellbore with fluid injection into the center pipe. While the results are only valid for steady-state flow and heat transfer with a constant media wall temperature, the basic relationships between flow rates, thermal resistances, analytical expressions of annulus and pipe fluid temperature distributions, and effectiveness were introduced.

The transient models described in the literature are either mathematically complex or computationally burdensome, limiting the general applicability of the models or sacrificing simulation efficiency. The objective of this study is to identify and quantify improvements that can be found using large volume boreholes cycled with periods dictated by hourly heating or cooling energy needs. To meet this objective, a model that accounts for transient vertical borehole fluid temperature distributions, transient adjacent media radial temperature gradients at multiple depths, and transient conductive heating of stagnant fluid in the well bore at times when the heat pump is not operational, was developed. The model is validated against field data to ensure its accuracy, then applied to 100 hours of operation to compare relative compressor energy usages using two different operational schedules and for four different wellbore sizes.

2. Heat Transfer Model

Two modes of operation need to be considered: active injection of water into the annular region of wellbore or into the pipe over a period of time related to periodic heating or cooling needs, and passive times between periods of active heat pump operation. In either case, transient heat transfer occurs between the geologic media adjacent to the wall and the water in the wellbore, changing temperature profiles in both the fluid and the adjacent media over time.

To construct a model that captures the fundamental transport physics, while accurately predicting, location-specific temperatures in the wellbore fluids and surrounding media, the following assumptions are made.

1. Vertical heat transfer in the surrounding media is negligible due to meter scale distances between adjacent layers and small temperature differences.
2. Diffusion and dispersion in the annulus and pipe are neglected.
3. The thermal properties of the inside and outside pipes, and surrounding media, are independent of temperature.
4. The flowrate is constant when the pump is operational.

The geometry of the wellbore is shown in Figure 1.

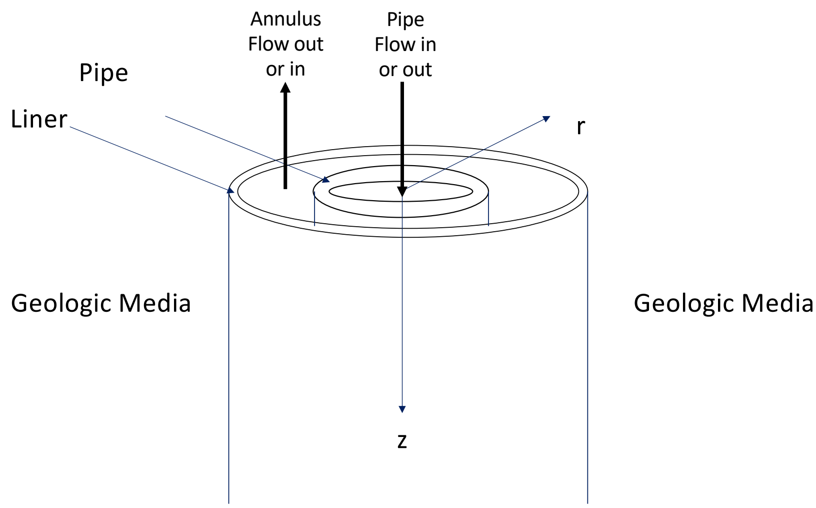


Figure 1. Configuration of the concentric wellbore configuration. The radial dimension is measured from the center of the pipe and z extends to the bottom of the wellbore. A liner (or large diameter pipe) is adjacent to the geologic media, with grout, if any, having the same thermal properties of the geologic media.

During operation, heat is transferred between annular fluid and the wall and the annular fluid and the center tube. Following Fang, *et al* [7], energy balances on the annular fluid and pipe fluid can be written as:

$$\dot{m}c\frac{dT_a}{dz} = \frac{(T_w(z,t) - T_a(z,t))}{R_1} + \frac{(T_p(z,t) - T_a(z,t))}{R_2} \tag{1}$$

and

$$\dot{m}c\frac{dT_p}{dz} = \frac{(T_p(z,t) - T_a(z,t))}{R_2} \tag{2}$$

where $T_w(z,t)$ is the wall temperature, $T_a(z,t)$ is the mean annulus temperature, $T_p(z,t)$ is the mean pipe fluid temperature.

Relationships for the resistances in the annular region (R_1) and the center tube (R_2) are shown given below.

$$R_1 = \frac{1}{2\pi r_{1i} h_1} \tag{3}$$

$$R_2 = \frac{1}{2\pi r_{2i} h_1} + \frac{1}{2\pi k_{p2}} ln\left(\frac{r_{2o}}{r_{2i}}\right) + \frac{1}{2\pi r_{2o} h_2} \tag{4}$$

where k_{p2} is thermal conductivity of the inner pipe; r_{2i} and r_{2o} are the inner and outer radii of the center pipe.

The convective coefficients, h_1 and h_2, are the heat transfer coefficients relating the heat transfer from the fluid to the wall of the annulus and inner pipe, respectively. Local convective heat transfer coefficients between each fluid (annulus & center tube) and the corresponding walls, were determined using correlations for laminar or turbulent flow in either region.

Equations *(1)* and *(2)* are not transient relationships. To transform those equations to forms conducive to transient numerical simulations, the fluid flow in the wellbore is modeled as a series of thermal waves being flushed through the wellbore: essentially a Lagrangian transformation of Equations 1 and 2 through recognition that dT/dx can be expressed as *dT/(Vdt)*, where V is the velocity of the fluid in either domain. By formulating the numerical solution from a wave viewpoint, we can incorporate the transient response by relating the time, Δt, it takes a specified volume of water in the annulus to travel a distance, Δz, thus setting the time step, Δt, from pumping rates, the depth of the borehole, L, the number of vertical elements for simulation, $N\ (=L/\Delta z)$, and geometry. Once the annulus and tube average fluid vertical temperature distribution is updated, implicit numerical solutions are employed to determine temperature distributions for N vertical elements, subjected to convective heat transfer from the annulus water to the inside wall of the outer pipe to numerically determine the transient response of the wall temperature and geologic media over that period, Δt.

Heat transfer between the fluid in the annulus and the surrounding geologic media is described using implicit finite difference solutions to the energy equation in the solid domain during pumping, and in the geologic media and stationary fluids during rest periods. The finite difference formulation in the geologic media is identical for either case, but the coupling between the wellbore fluid energy transfer rate and the wall temperature requires expansion of the grid to include fluids and pipe in the wellbore.

To couple the heat transfer between the fluid and the surrounding media and heat transfer, a 2-D, transient numerical solution was developed. The solution is constructed to calculate the radial temperature distribution in the surrounding media and fluids at multiple depths. Heat conduction in the vertical direction was considered negligible due to the low thermal conductivity, distance between vertical elements and small temperature differences. A far-field boundary condition is set by including enough nodes such that the far-field has no change in temperature over the time of simulations. This corresponded to a radius of 1.8 meters for the calculations presented herein.

To transition between flowing and rest portions of the cycle, the mean values of annulus and pipe fluid temperatures need to be recast as temperatures that vary with radial location. The reference temperatures for the fluids in the annulus and pipe are the outer annulus wall temperature, the temperatures of the inner and outer surfaces of the inside pipe, which are related to the mean temperatures of the fluids in each region.

The annulus temperature distributions in the pipe and anulus are specified according to the following quadratic form.

$$T_i(r) = a_i + b_i r + c_i r^2 \qquad (5)$$

The coefficients a_i, b_i and c_i are determined by the conditions that the temperatures match the outside wall and outside wall of the inner pipe, and that the mean annulus temperatures match the last values from the wellbore calculations. The transition from rest to flow conditions requires the numeric integration of the temperatures within each fluid region.

4. Field Studies

Field tests were performed on a coaxial ground heat exchanger (GHEX). The well bore was constructed with 5.5 m of 178 mm surface casing followed by 165 mm diameter bore drilled into competent rock to a depth of 91 meters. A membrane was then installed and placed in direct contact with the borehole wall using a proprietary process to create a sealed "vessel" capable of isolating the circulating fluid of the ground heat exchanger (GHEX) from groundwater. A 51 mm ID center pipe was also installed to depth within the larger wellbore to provide the second flow path.

A series of thermal response tests were performed using an integrated Thermal Response Test (TRT) unit which contained electric resistance heaters capable of generating 8,000+watts, a circulating pump capable of up to 7.6E-4 m^3/sec (10.5 GPM), sensors for the measurement of Entering Water Temperature (EWT), Leaving Water Temperature (LWT), a flow meter, a pressure sensor, a multi-channel/ multi-function data logger and controls. The TRT unit was powered with line power provided at the test site.

The first test was a thermal response experiment to determine the formation effective thermal conductivity. The thermal response test was followed by a period of heat pulses at intervals of 30 minutes of heating followed by 85 minutes of flow without heating. Flow remained continuous during the test. The first 75 hours represented a typical thermal response test. From that response, the thermal conductivity of the rock was determined.

For the second series of tests, additional temperature sensors were placed at the bottom of the hole TP300 (91m), and in the annulus at TP150(46m) and TP050(15m). In the first of these tests, an interval of 35 minutes of heating followed by 70 minutes of rest. Four minutes of pumping preceded each heating interval follow by a minute of pumping after the power to the heater was turned off. Otherwise, the GHEX was fully at rest. Data for EWT and LWT are only shown when active flow was occurring. The TP temperature sensors logged changes in the GHEX continuously.

5. Verification of Model Using Field Data

The model was tested against the sets of data from the field studies. The thermal properties of the of the fluid and site are listed in Table 1. The wellbore geometry and material properties used in the simulations of field tests are listed in Table 2. The first test, Test 1, combined a thermal response test with cyclic heating.

The thermal conductivity was measured from the thermal response testing. Values of the rock density and heat capacity were estimated from published values and chosen to best represent the thermal response observed in Test 1. Further, the value of the heat transfer coefficient was adjusted to fit the cyclic heating portion of Test 1. That value (120 W/m^2-K) is in the range of what would be expected in the flow regime (turbulent) in the annulus.

Table 1: List of fluid, rock and wellbore thermal properties used in simulations

Property	Units	Value
Formation conductivity	W/m-K	3.29
Formation diffusivity	m^2/s	1.24E-6
Water heat Capacity	J/kg-K	4180
Thermal Conductivity of Pipe	W/m-K	.12
Thermal conductivity of Liner	W/m-K	.19

Table 2. Wellbore Properties for Validation Studies.

Property	Units	Value
Liner Thickness	m	.003
Inside Diameter of Pipe	m	.058
Outside Diameter of Pipe	m	.073
Diameter of Wellbore	m	.15
Wellbore Membrane thickness	m	.003
Flow rate	m^3/s	6.86E-4
Heat Transfer Coefficient	W/m^2-K	120
Wellbore Depth	m	90

The model was first compared to the measured temperatures of the water entering and leaving the surface heater from Test 1. Details of the measured and predicted temperatures during the cyclic operation are shown in Figure 2.

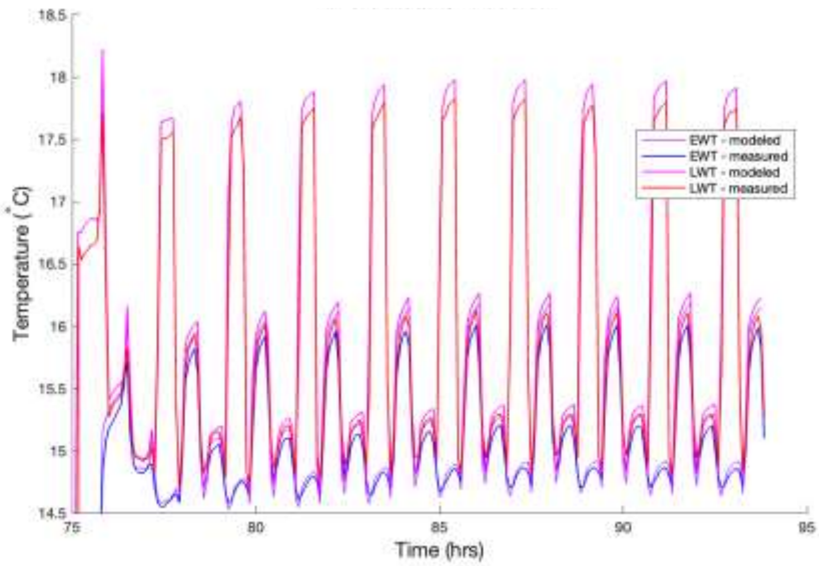


Figure 2. Detail of modeled heater inlet (EWT) and outlet (LWT) temperatures compared to data from cyclic heating (30 minutes of heating followed by 85 minutes heat off) from 75 hours to the end.

As shown in Figure 2, the simulations are in excellent agreement with the field data, with slight discrepancies within error of temperature measurements, heat rate or pumping rate. The correct prediction of the amplitude of the thermal pulse with each flush reflects a value of 120 W/m^2 as the heat transfer coefficient on the outside wall of the annulus.

Test 2 and Test 3 were run sequentially, with the heating and rest periods varied for each test. For these tests, the water was injected into the annulus. Further, downhole annulus temperature measurements were available for comparisons to model predictions. The surface temperature data and model predictions from Test 2 are shown in Figures 3.

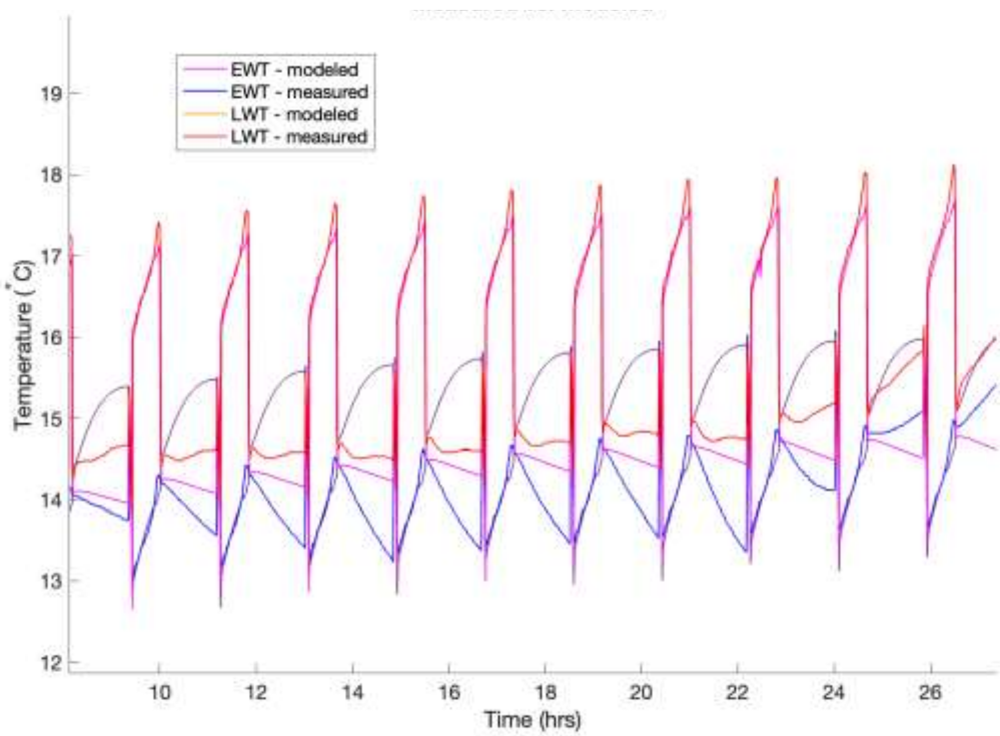


Figure 3. Detail of simulated and measured heater inlet (EWT) and outlet (LWT) temperature response during cyclic operation with a flowrate of 10.5 GPM, 8.8 kW heating for 35 minutes and resting for 70 minutes.

Excellent agreement between simulated temperatures and measured temperatures during periods of time of heating (35 min) is also seen in this figure. The transient EWT and LWT temperatures measured above ground between cycles are not comparable. However, the match between measured and modeled temperatures at the beginning of the heating cycle is a good indication that the wellbore model is correctly coupled to rock transients.

Comparisons between the downhole temperatures from Test 3 are compared to simulated temperatures in Figure 4.

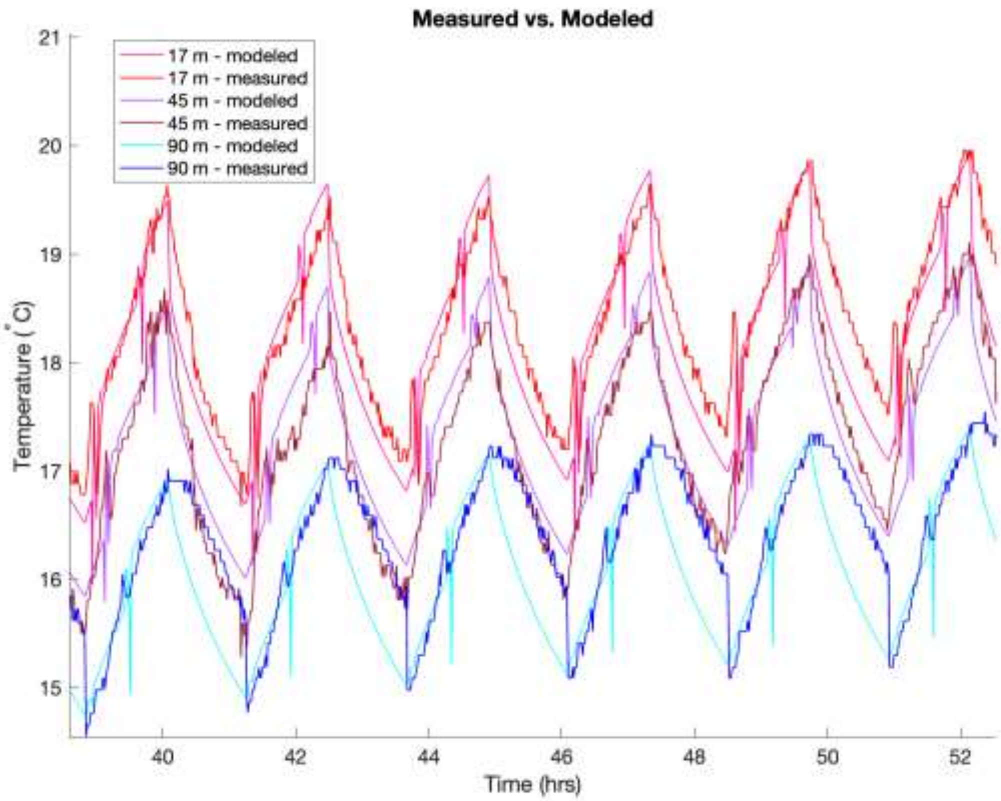


Figure 4. Detail of simulated and measured wellbore annulus temperatures, Test 3, annulus injection, cycling with 70 minutes on and 70 minutes off.

Again, very good to excellent agreement between simulated temperatures and measured temperatures during periods of time of heating (70 min), including a short-lived transient at the end of the first borehole volume transit time (38 min) due to the pump starting 4 minutes before heating, is also seen in this figure. The response to the four minutes of pumping before the heater is energized, showing in the simulations and data as "spikes" in temperatures, is apparent in both the simulations and data. What would be considered "noise" in the data is actually the response to 4 minutes of pumping before the heater was energized. The simulations during the rest period are in reasonable agreement with the data, with the intermediate temperatures generally tracking the decay times predicted by the model. The exception is at the bottom of the wellbore where the temperature measurements were consistently higher than those simulated during the rest period. It is conjectured that flow occurred the rest period due to buoyant forces allowing water in the pipe to displace annulus water at the bottom. Further, the placement of the intermediate temperature probes with respect to radial location in the annulus is not known, and as such, discrepancies in specific sensor thermal decay times would be expected.

6. Comparative Study of Different Wellbore Sizes and Different Cycle Schedules.

The objective of ground-sourcing a heat pump is to save energy needed to power the compressor. To relate wellbore thermal performance to energy savings, the heat flux needs to be related to the COPs. The correlation for the COP_{HP} from [10] is used in the calculations of compressor energy use in this work.

$$COP_{HP} = 3.6 + 0.0715T_{EWT} - 0.005T_{EWT}^2 \quad (6)$$

where T is the entering water temperature in °C.

For cooling, a correlation for the COP_R as a function of entering water temperature was derived from data for the specified equipment. Given the agreement of this equipment-specific heat pump COP_{HP} with the general correlation, a correlation of the cooling COP_R derived from this specific system's performance data is considered to be representative of cooling systems in general. That correlation is the following:

$$COP_R = 8.263 - 0.13T_{EWT} + 0.00018T_{EWT}^2 \tag{7}$$

From the definition of the COP_{HP}, the following relation for the work rate to transfer a specified heat rate, $\dot{q}$, from the subsurface for building heating is:

$$\dot{w} = \frac{\dot{q}}{(COP_{HP} - 1)} \tag{8}$$

Similarly, the work to the compressor during cooling for a given heat rate to the subsurface is:

$$\dot{w} = \frac{\dot{q}}{(COP_R + 1)} \tag{9}$$

Equations (31) and (32) were used to calculate the work for the compressor for each time step, and numerically integrated, as follows, to calculate the total work over the period of operation.

$$W_T = \sum_i^n \dot{w}\Delta t \tag{10}$$

While there are many studies that could be performed calculate the compressor energy in heating and cooling modes given different depths, flowrates, sizes, schedules, variations of heating and cooling loads, wellbore sizes, inside pipe material and dimensions, this study will focus on the variation of the wellbore diameter for 2 similar off/on schedules (1/3 time on, 2/3 time off), but different cycle times, for operation over a 100 hour period with an average heat flux to the media of 33.3 W/m. The flowrate is held constant for each simulation. Four different wellbore diameters were chosen for comparison: 5 cm (2"), 10 cm (4"), 15 cm (6") and 20 cm (8"). Specific geometric properties used in the simulations are listed in Table 3.

Since all equations are linear, operation in the heat pump mode can be simply reconstructed by transforming T_{EWT} and T_{LWT} values by the following:

$$T_{EWT,HP} = T_\infty - (T_{EWT} - T_\infty) \tag{11}$$

$$T_{LWT,HP} = T_\infty - (T_{LWT} - T_\infty) \tag{12}$$

The $T_{EWT,HP}$ is the value used in the correlation for the COP_{HP} to calculate the energy used at a specific time.

Table 3: Geometries of Wellbore Heat Exchangers used for Comparative Studies

Property	(Units)	2”	4”	6”	8”
Membrane Thickness	m	.003	.003	.003	.003
Inside Diameter of Tube	m	.013	.030	.047	.060
Outside Diameter of Tube	m	.023	.044	.06	.083
Inside Diameter of Wellbore	m	.056	.102	.154	.203
Flow rate	m^3/s	6.86E-4	6.86E-4	6.86E-4	6.86E-4
Wellbore Depth	m	90	90	90	90
Heat rate	W/m	100	100	100	100
Transit Time	Minutes	3.14	18	43	70.2
Dt	Seconds	1.9	10.8	25.9	42.1

7. Results and Discussion

Temperatures at the inlet and outlet of the above-ground heat exchanger were computed at each time step to compute compressor energy use, and effectiveness. The temperatures for operation in refrigeration mode and heat pump mode are shown in Figure 5 for the two extremes: 2” and 8” run at a schedule of 20 minutes with heat transfer and pumping, and 40 minutes of rest.

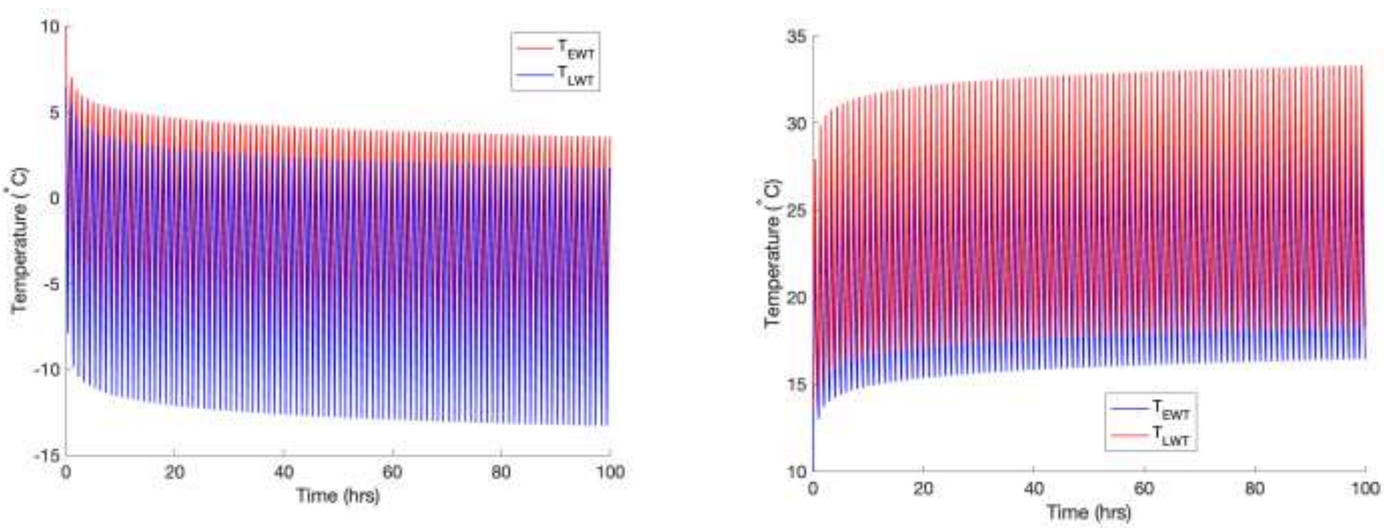
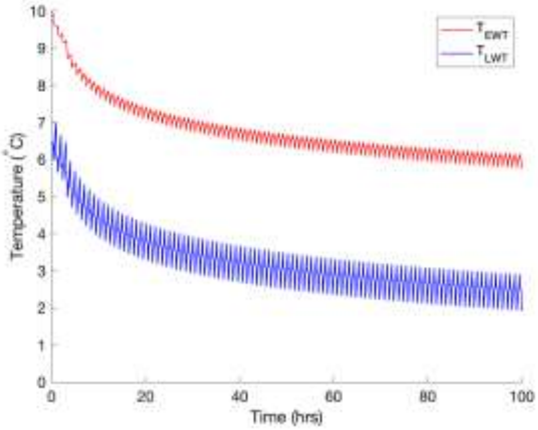


Figure 5. T_{EWT} and T_{LWT} for an 2” outside diameter borehole during heat pump operation, left, and refrigeration, right, of 20 minutes with heat transfer and pumping, and 40 minutes of rest.



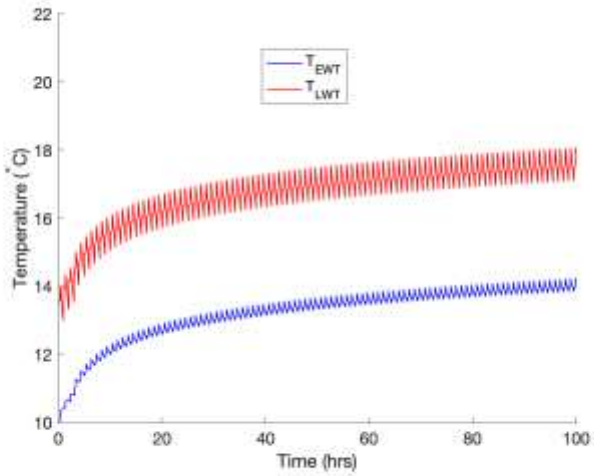


Figure 6. T_{EWT} and T_{LWT} for an 8" outside diameter borehole during heat pump operation, left, and refrigeration, right, of 20 minutes with heat transfer and pumping, and 40 minutes of rest.

Several observations can be made from considerations of the T_{EWT} and T_{LWT} histories shown in Figures 5 and 6. First, the temperature changes during cyclic operation are much more pronounced in the smaller diameter wellbore than the large wellbore. Second, the temperatures of the fluids in the 2" wellbore are significantly below the water freezing point while remaining above 2°C for the duration of the simulation. Simulations of temperature responses for the 4" and 6" cases fall between these two extremes, with the simulations of the 6" case being close to that of the 8" case.

The total calculated compressor energies for the 8 different cases are presented in Tables 4 for operation in the refrigeration mode where heat is added to the subsurface, and in Table 5 for operation as a heat pump where energy is removed from the subsurface.

Table 4. Refrigeration compressor energy use for 100 hours of operation for 4 different wellbore sizes and two different operational schedules.

Wellbore Size	2"	4"	6"	8"
1 Transit time on, 2 Transit times off	45.3	41.8	40.8	40.5
20 min on/40 min off	46.7	41.9	40.5	40.3

Table 5. Heat Pump compressor energy use for 100 hours of operation for 4 different wellbore sizes and two different operational schedules.

Wellbore Size	2"	4"	6"	8"
1 Transit time on, 2 Transit times off	118	105.2	101	100.1
20 min on/40 min off	125	105.4	100	99.6

As shown if Table 4, the energy use for the 6" and 8" cases are nearly the same for both heat pump and refrigeration operation. Energy use increases with decreasing wellbore size with increases in energy use of 12% to 25% for the 2" diameter wellbore compared to the 6" or 8" diameter wellbores.

The 2" diameter configuration was conceived as an approximation of the performance of a U-tube heat exchanger. As such, the limitations observed in the operation of the 2" case would generally apply. The first limitation is that the temperatures in the heat pump mode quickly drop below 0°C. While this is somewhat arbitrary given the specific heating rates, operation time and initial media temperatures of the simulations, the general conclusion is that smaller diameter wellbore are more likely to experience freezing temperatures than larger wellbores. The second limitation is that the effectiveness is low, driven by higher average wellbore wall temperatures for smaller diameters for the same average heat flux. The large variations in effectiveness to values below 0.2 would be expected for U-Tube configurations as well.

8. Conclusions

From this study, several conclusions can be drawn:

1. Wellbore thermal transients can be accurately simulated using a Lagrangian approach to the transport of energy in the wellbore fluids.
2. Coupling the wellbore heat transfer model with a transient rock temperature distribution model captures the short term and lon- term thermal dynamics of cyclic operation.
3. For similar operating conditions, smaller wellbores produce larger temperature extremes, lower effectivenesses, and would require larger compressor energy use (12-25%) for the specified heat transfer rate, heating or cooling mode, and operation time.

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