Feasibility Assessment of Underground Cooling for Geothermal Power Cycles

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Geothermal heat sources offer significant potential for electricity generation in Australia by Hot Fractured Rock, HFR, technology and other approaches. However, all of the sites of greatest geothermal potential are situated remote from any significant surface water source, which poses a significant challenge to the method by which the working fluid in the power cycle will be cooled. The need for cooling in any thermal power cycle has driven many conventional power plants to be located close to a river, lake, or ocean to provide an environmental heat sink (Department of Environment, 2001). Where this is not realistic, cooling towers are almost invariably used to provide greater cooling than is possible by air-cooling alone, utilising the evaporation of water, much like a large evaporative air cooler. However, the water consumption due to evaporation and fouling losses, even for a conventional power station with comparatively high efficiency, is typically 1363 L/MW.h per day (Ricketts et al., 2006), and will be greater for typical geothermal plants owing to their low thermodynamic efficiency.

Indeed, the recent increase in demand for air-cooled condensers is a direct result of water shortages at potential plant sites and increasing government legislation limiting the use of water in the wet cooling systems (Ricketts et al., 2006). As noted above, the regions in which geothermal heat is the most viable are arid or semi-arid, with no readily available source of surface water for cooling. While underground water is present, notably from the Great Artesian Basin, it is unlikely that environmental regulators will allow it to be utilised because of the very large water consumption required by any large thermal power plant. Therefore, alternatives to conventional water-based cooling methods are expected to be required for these power plants.

The present commercially available alternative method of heat dissipation is through the use of conventional fin-and-tube air cooled heat exchangers. Fins on the surface of the heat exchanger are typically used to improve the heat transfer by increasing convection and radiation away from the surface. Such systems are used in a number of geothermal plants, e.g. in the Mokai plant in New Zealand. However, in the areas of interest, such as the Cooper Basin, daily ambient air temperatures can reach 45ºC in the summer, in which case the minimum temperature of the working fluid temperature must be in the range 48-50ºC, owing to the fact that the effectiveness of a heat exchanger is always less than 100%. Such high condenser temperatures result in a low efficiency of the power cycle. For example, Langman et al (2008) estimate that the output from a geothermal plant at a typical site in South Australia would drop by 40% as the ambient air temperature is increased from 15 to 45ºC. Furthermore, the peak demand, and hence peak prices, in the national grid are greatest during the very time when the output is lowest. Such a scenario could significantly influence the economic viability of a plant. Fans may be employed to force air over the fins to improve the performance of the heat exchanger, but consume large amounts of energy and hence reduce the net amount of electricity produced by the plant.

A potential solution to the problem of cooling the working fluid is to install a heat exchanger underground where temperatures are lower and more stable. Heat in the working fluid may then be rejected to the soil and in turn dissipated to the atmosphere. For this reason, the aim of the present investigation is to assess the feasibility of underground cooling for such a geothermal plant.

Keywords: geothermal, power cycle, condenser, underground cooling.

Approach and Methodology

The heat transfer processes within an underground heat exchanger are time dependent and three dimensional. Analytical solutions are unable to account for such variations in soil and atmospheric conditions. Hence we opted for a modelling strategy that accounts for temperature variation and heat flow over 24-hour period, repeated for many days. In this approach we are able to account for the radiation in the day and night and the temperature variation of soil and the air over an extended period.

In order to simplify the problem while maintaining the core issue intact, the following assumptions were made:

- the soil is homogeneous;
- there are no phase changes in water, and no latent heat effects;
- the soil emissivity, absorptivity and reflectivity are constant;
- convection over the surface is a function of wind speed, which is fixed;
- the pipe depth is fixed, and

- the effects of the pipe wall on conduction are negligible.

A block of soil with sides of approximately equal length, shown in Figure 1, is analysed. A pipe representing the heat exchanger is buried underground at a certain depth, ***l***, and the working fluid of the power cycle is passed through the pipe at a temperature $\boldsymbol{T_i}$. Since the soil is at a lower temperature than the pipe, heat energy is lost to the soil, such that the fluid exits the pipe at some lower temperature $\boldsymbol{T_o}$. It should be noted that the boundaries of the block of soil are assumed to be semi-infinite, and thus heat may be lost through the faces of the block with negligible resistance.

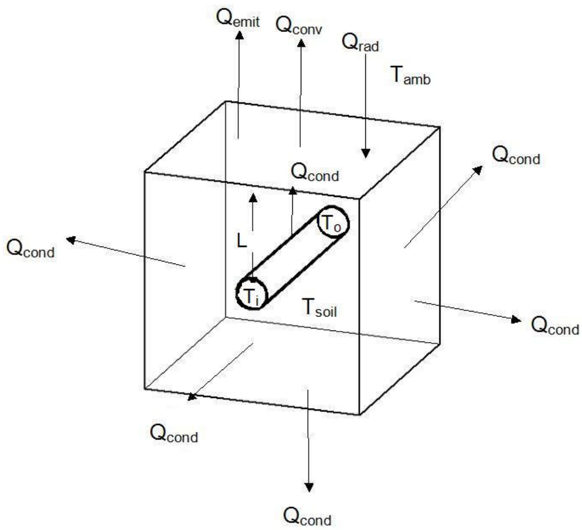

Figure 1: Schematic diagram of a pipe buried in soil

The pipe depth ***l*** is an important parameter and will have a crucial effect on the behaviour of the model. It is known that fluctuations in temperature over a period of time decrease with soil depth, depending on the properties of the soil (Pavelka et al. 2006). The measured change in temperature over the period of one day for various soil depths is presented in Figure 2. Thus, at a depth of just 30cm the temperature fluctuation for soil over a day can be small (typically 1 to 2ºC).

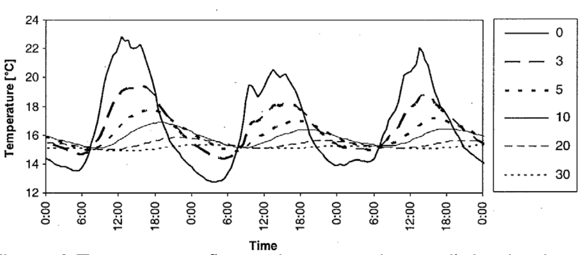

Figure 2 Temperature fluctuations at various soil depths, in cm, over 24 hours period [Paveleka et al., 2006]

For the present comparative assessment, it is sufficient to assume a typical temperature for the working fluid entering the condenser. We have chosen this to be 100ºC. This is typical of the temperature at which the geoliquid is expected to be returned underground from Enhanced Geothermal Systems (EGS) under conditions in central Australia (Langman et al, 2008). While being somewhat higher than expected condenser temperatures, for the present comparative purposes it is sufficient to ensure that both the geothermal and air-cooled temperatures are based on the same reference condition. The heat being lost to the soil through conduction is termed $\mathbf{Q_{cond}}$ in Figure 1 and depends on a number of parameters, including the conductivity of the soil. The conductivity of soil changes as the moisture content of the soil increases or decreases and is largest when the soil is wet. However, as rain is rare in the arid regions being considered, the soil conductivity is assumed to be that of dry soil, and is assumed to be constant. Note that this assumption is conservative, since the presence of moisture will increase the thermal conductivity. One report suggests that thermal conductivity values for sandy loams range from 0.54W/mK to 1.94W/mK (Abu-Hamdeh and Reeder, 2000). Exact data for the geomorphology of the Cooper Basin region is difficult to obtain, as reports suggest that the region contains both wetlands and desert, which have widely varying soil properties (Burdon, 2006). To allow for a conservative solution for conduction then, it was decided to use a value of 0.75W/mK for the thermal conductivity of all models, which is typical of sandy loam soils.

The heat transferred to the surface by radiation from the Sun during the day is termed $\mathbf{Q_{rad}}$, shown in Figure 1. Solar radiation is made up of two components, namely direct radiation and indirect radiation. For the purposes of the present model, the direct and indirect radiation are grouped into a single quantity. Measured data for the daily and seasonal variation in $\mathbf{Q_{rad}}$, are readily available.

The infra-red radiation from the soil is shown in Figure 1 as $\mathbf{Q_{emit}}$. It is assumed that the soil is a gray body and will emit radiation as a function of the emittance of the soil and the temperature difference between the soil and the air. The emittance of the soil is assumed to be constant and equal to 0.75 (Mills, 1999). Radiant losses are assumed, conservatively, to be to the air at ambient temperature. Measured data for the daily and seasonal variation in ambient temperature are readily available.

Heat is also dissipated through convection from the surface, termed $\mathbf{Q_{conv}}$ in Figure 1. This depends on the wind speed of the air flowing over the surface and the ambient air temperature. The average daily wind speeds for the month of January were obtained from Energy Efficiency and Renewable Energy (2008) and were averaged to provide a monthly average value.

This monthly average value was used to calculate the convective heat transfer coefficient.

The finite quadrilateral elements for the 2-dimensional longitudinal cross section model with a 50mm diameter pipe, 0.1m deep and 50m long is shown in Figure 3. Owing to constraints of the ANSYS package (ANSYS, 2007), to model an infinite boundary condition, it is necessary for the boundary of the problem to be circular as shown in the model below.

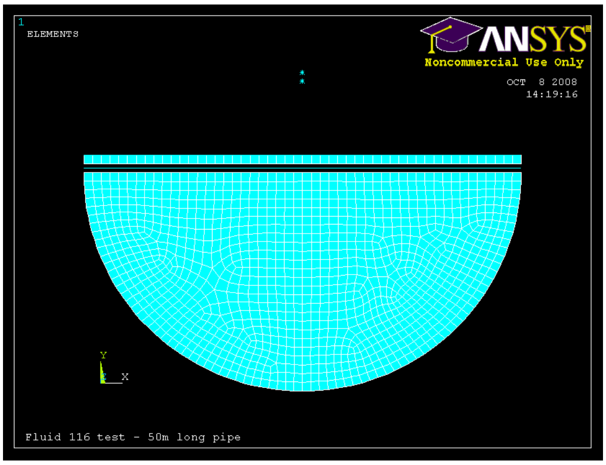

Figure 3 Finite element model of the 2-D Longitudinal Cross Section. Pipe diameter 0.05m, depth 0.1m and length 50m.

The thermal properties of the soil and the air, used in the model, are shown in Table 1.

Table 1: Soil and air thermal properties

Symbol	Soil	Air
k(W/mK)	0.75	0.0271
C(J/kgK)	1000	1900
ρ(kg/m^3)	1500	1.225
ε	0.77	-

The inlet temperature of the water was assumed to be 373K. The heat dissipated by the underground pipe system was set to 5MW. The mass flow rate of water in the 0.005m diameter pipe was assumed to be 16 kg/s. The heat transfer coefficient inside the pipe was calculated using the Dittus-Boelter correlation for turbulent flow inside a smooth pipe.

$$Nu_D = 0.023\,\mathrm{Re}_D^{0.8}\,\mathrm{Pr}^{0.4}$$

The pipe thermal conductivity is estimated at 0.41 W/mK. Thermal resistance between the soil and the pipe was assumes to be negligible. In practical systems this resistance can be minimized through proper compactness of the soil surrounding the pipes.

Weather data including the hourly average ambient temperature and an average hourly radiation for all days in the month of January were obtained from a weather station in the town of Oodnadatta, which is representative of summer conditions in the Far North East of the state of South Australia (EERE, 2008). The choice of January was due to it being in the middle of summer with the hottest daily temperatures. This corresponds to the worst loading conditions for the heat exchanger.

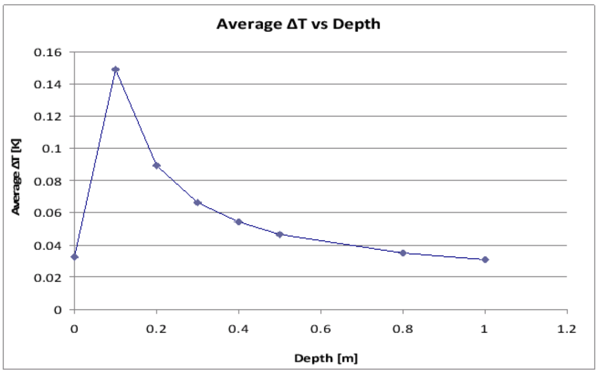

Figure 4: Water temperature drop plotted against the pipe depth after 20 days. Pipe diameter 0.05m, pipe length 50m.

The daily average wind speed for the month of January was used to calculate the heat convection coefficient to be 23.6W/m^{2}K.

Results and Discussions

Presented in Figure 4 is the temperature at the pipe exit plotted versus the pipe depth at end of a 20-day period. The 20-day period ensures that the temperature reaches pseudo-steady state conditions. It is clear from the figure that a depth of 10 cm is the most appropriate and provides the highest temperature loss for a fixed length of pipe.

The drop in temperature appears to be quite small for the 50 m length of the pipe amounting to 0.15K. However this amounts to 10 kW of heat which was dissipated away from the pipe and into the atmosphere. Worthy of note too, is that there was no attempt to enhance the pipe design which would have increased the amount of heat lost per length of pipe. Such enhancement can include fins, surface protection from sun radiation and other methods to increase heat loss during the night.

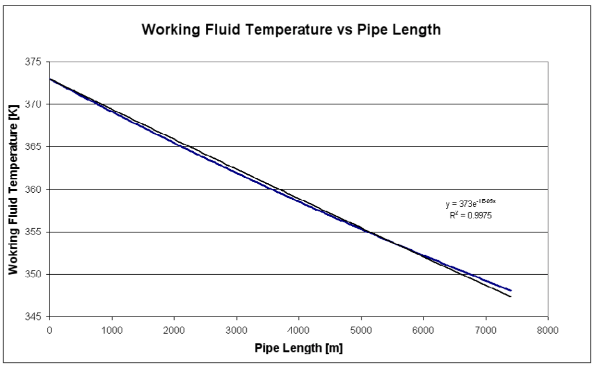

Figure 5: Calculated Temperature drop along a 50m length, repeated over a 7.4km length of pipe

The longitudinal model was also used to obtain an estimate of the overall pipe length to dissipate 5MW of heat. The 50m geometry, in Figure 3, was looped multiple times to calculate a temperature drop over 7.4km. In other words the exit temperature from the first 50m pipe after 20 days was used as inlet temperature to the second 50m pipe and the process was repeated 148 times to give the exit temperature at accumulated lengthy of 7400m. The exit temperature from each run is presented in Figure 5 for every 50m. A trend-line has been applied to this data and the equation of the trend-line is also shown on the graph.

Noteworthy is that the temperature difference between the pipe and the surrounding soil, which drives the potential for heat transfer, drops and hence the effectiveness of a meter length of pipe drops too. Hence extrapolating the curve generated above to longer pipes may not be accurate. Nonetheless, the curve fit is for almost half of the temperature reduction required (70ºC) and a reasonable estimate can be generated using this method. Thus, solving for length, and using a value of 300K (27ºC) for the pipe exit temperature, the length of pipe required is approximately 22km. To account for the issues discussed above a conservative estimate was deemed to be prudent and a factor of safety was introduced. Hence for the cost analysis, the overall length of pipe required for the underground cooling process was set to 25km.

The present preliminary cost estimate was undertaken for the above conditions without any specific reference to a relevant geothermal thermal cycle. That is, it does not consider the influence of the cooling cycle on the performance of the power plant. Nonetheless, these rough estimates, which considered worse weather conditions, have shown that in comparison to air cooled heat exchanger our approach is highly cost effective both from the initial investment and operating cost. It is anticipated that with further optimisation and accounting for year long weather conditions the above conclusions will hold.

Future work will focus on proving this concept experimentally and expanding the modelling work to a specific location accounting for local soil properties, likely thermal cycle and adopting standing heat transfer enhancing methods such as fins. This work will then give us further grounds to build a prototype to accurately quantify the benefits.

Summary

A preliminary assessment has been undertaken of the technical and economical feasibility of using soil to store energy during the day and dissipate it during the night. The assessment is based on conditions in the Copper Basin region of South Australia. Several approaches were used to calculate the rate of heat transfer and the viability of the concept, including analytical and an advanced computational technique, Finite Element Analyses, (FEA). It was estimated that the optimal depth to burry the heat exchanger pipes is about 0.1 m. This depth was arrived at through modelling the soil absorption of the heat dissipated from the pipe and the impact of the sun radiation, air temperature and wind speed on heat transfer from the soil to the atmosphere.

A two dimensional longitudinal model was used to estimate the length of pipe required to dissipate 5MW of heat by cooling a working fluid (Water) from 100ºC to 30ºC. The model revealed that an approximate length of 25km of 50mm diameter pipe is required to achieve the required heat transfer. The cost of dissipating heat through a water-based system at low pressure was estimated to be less than that of a conventional air cooled heat exchanger. Several other advantages are anticipated, such as avoiding the need for fans and a power output that is much less dependent on ambient temperature. However, a detailed assessment of its impact on the performance on the plant, or the economic feasibility is yet to be undertaken.

In addition, as with all models, a number of simplifications and approximations have been required, so that further model development and experimental validation are required to help better estimate the benefits. Further work is also required to optimise the design and consider its integration into specific geothermal cycles with a view to justifying a demonstration prototype.

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