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GEOTHERMAL DOWNHOLE HEAT EXCHANGER (DHE) PERFORMANCE ANALYSIS

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ABSTRACT

A typical downhole heat exchanger (DHE) consists of a single loop of tubing placed in a geothermal well. Energy is extracted from the geothermal fluid in the well by transfer of heat through the tube wall to clean water, the working fluid, circulating in the loop.

The performance of a DHE can be improved by placing a "promoter", a single length of tubing which is open or slotted at both ends, in the well to promote circulation of the geothermal fluid within the wellbore itself. In this study, two configurations involving such a "promoter" are considered: in the first, the DHE tubing loop carrying the clean water is placed inside the promoter tube, while in the second configuration the loop is outside, i.e. beside, the promoter. In the present context, the first configuration can include the case where a slotted liner hangs loosely in the wellbore and fluid can move between it and the well face.

Here, modelling of such DHE/promoter systems is canied out by using mass and heat balances, resulting in a set of linear ordinary differential equations which describe the steady-state flow. With the assumption that the circulation of the clean water is maintained at a fixed constant rate, and that the geothermal fluid is circulating in the wellbore by natural convection (under a balance of buoyancy and frictional forces), the equations are solved by a combination of analytical and numerical methods.

Material parameters and dimensions are varied to investigate their effect on the power output of the DHE. One major theoretical result is that a much higher power output is obtained if the loop is placed beside the promoter tube, rather than inside it.

INTRODUCTION

Geothermal resources have attracted much attention as an alternative power source in current energy crises. However, concern about the long term effects of draw-off on geothermal reservoirs has prompted interest in more efficient methods for extracting geothermal energy. The downhole heat exchanger (DHE) is one method that is being investigated.

Being a heat exchanger, the DHE has an output that depends strongly on the heat transfer area, which in this case is restricted by well size. Consequently a DHE system is inherently restricted to the lower end of the energy output spectrum. Nevertheless a significant and useful contribution to energy demand is possible. For example, a 70–120 m deep well with a resource temperature of 90–100°C can be expected to have a useful output of 0.8 MW per well (Freeston & Pan, 1985). The 400 wells at Rotorua, for example, could make a significant contribution to local energy needs without endangering the geothermal reserves, if they were all converted to DHEs.

The downhole heat exchanger is an economically attractive option; once the heat exchanger is installed it is cheap to maintain, because clean water is circulated through it thereby minimising deposition and corrosion on the inside of the tubing. The DHE is also environmentally appealing because the heat transfer takes place within the well rather than by removing the geothermal fluid. As a result there is no decline in groundwater level or likely subsidence effect, nor any atmospheric, thermal or chemical pollution due to waste disposal.

A typical DHE installation is shown schematically in Figure 1. It consists of a wellbore generally from 15 to 36 cm in diameter, a casing sealed to some depth, a promoter tube consisting of a pipe with diameter approximately half that of the well positioned beneath the water level, and an unfinned u-shaped heat exchanger tube. Clean cold water is circulated by forced convection (i.e. is pumped) through the u-tube and is heated during its passage. The presence of a promoter has been shown to increase the efficiency of utilisation of a DHE by creating a pathway for thermal convection to occur in the well with hot water from the reservoir flowing up to the top of the wellbore (Allis & James, 1979).

In this study, two configurations are considered: Configuration 1 with the u-tube inside the promoter [see Figures 1 and 2(a)] and Configuration 2 with the u-tube outside, or beside, the promoter [see Figure 3(a)]. The direction of flow of the natural convection within the wellbore is determined by the position of the u-tube (Allis & James, 1979); the heat extraction by the u-tube causes the fluid surrounding it to be cooler and therefore to move downward

The permeability at the base of the well also has an effect on the DHE output (Allis, 1981). Low permeability wells have very little cross-flow so that only a small fraction of the convective flow within the wellbore is replaced by fresh reservoir fluid and most of the wellbore fluid recirculates. For high permeability wells, there is constant replacement of the cooler wellbore fluid with hot reservoir fluid.

Culver & Reistad (1978) presented a computer-based study of the characteristics of a typical DHE installation in Oregon, USA, using a network analysis to model the heat and fluid flow paths with various resistances. The theoretical model was developed further by Freeston & Pan (1983).

In this work, heat and mass transfer equations similar to those used by Culver & Reistad (1978) are derived and are solved directly, subject to suitable boundary conditions, to yield the temperature profiles in the fluid flows, the natural convective flowrate in the well and the power output of the DHE.

The temperature outside the casing of the well (i.e. in the reservoir) is assumed to be a time-independent function of depth; in the examples taken here, this is a linear function between the temperatures T_B at the bottom and T_T at the top of the well casing. This assumption supposes that heat is not being "mined" from the reservoir. A more complete model should perhaps consider heat and fluid flow in the surrounding reservoir, but would complicate the simpler investigation being made here.

The current model is used to investigate the effects of varying the geometry and material parameters on the output of the DHE. A standard well structure, whose properties are listed in Table 1, is used as a basis for comparison. Five parameters of this basic structure are varied individually: the promoter tube diameter, the thermal conductivity of the promoter tube, the thermal conductivity of the u-tube, the clean water flowrate and the fraction of fluid within the well-bore being replaced at the bottom by fresh reservoir fluid crossflow (this fraction, a, is called the mixing ratio). The effects of such variations are described below in the results section, and ways of optimising the configuration are discussed.

THE MATHEMATICAL MODEL

A section of the first configuration to be studied here is shown schematically in Figure 2(a). Here the u-tube is placed inside the promoter tube. It is assumed that the bulk flow in each region is fully developed and that heat and force balances can be made using standard correlations. The mass flow in the u-tube loop is a constant, m_h, while the resulting natural convective mass flow in the promoter and annulus, m, moves up the annulus between the casing and promoter and downwards around the u-tube. The four flows, viz.

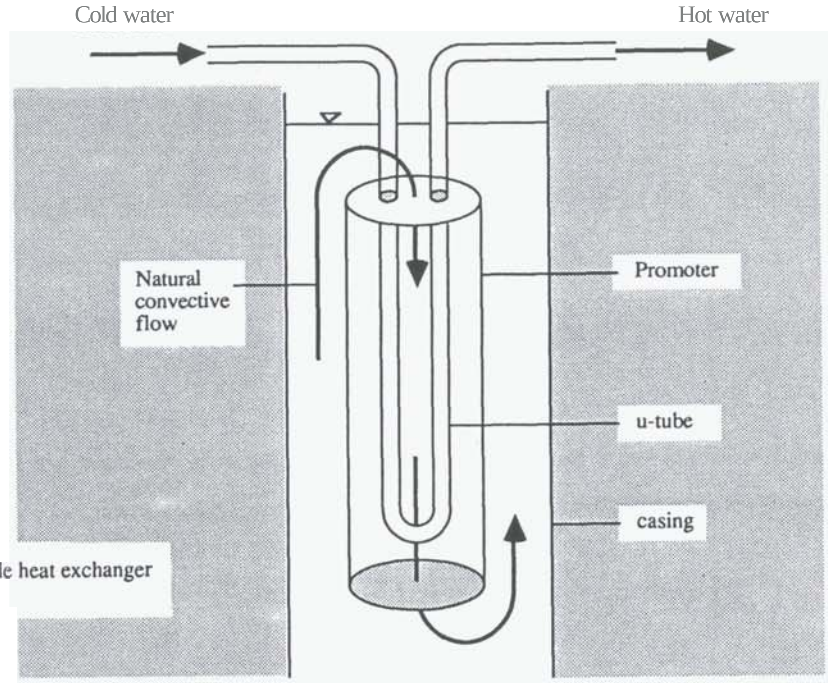


Figure 1. Schematic diagram of a typical downhole heat exchanger (Configuration 1).

Standard well structure

Casing:	outside diameter	d_o	0.224 m
	thickness	t_o	0.015 m
	thermal conductivity	k_o	54 W/m K
promoter	outside diameter	d_i	0.124 m
	thickness	t_i	0.008 m
	thermal conductivity	k_i	54 W/m K
u-tube:	outside diameter	d_1, d_2	0.050 m
	thickness	t_1, t_2	0.003 m
	thermal conductivity	k_1, k_2	54 W/m K
Well depth		L	200 m
Reservoir temperature:	top	T_T	13°C
	bottom	T_B	180°C
u-tube:	inlet temperature	T_{in}	17°C
	mass flowrate	m_h	1 kg/s
Mixing ratio		a	1.0
Pipe roughnesses		ε	0 m
Fouling factors		$f_{\beta\gamma}$	0 m^2 K/W

Fluid properties: constant (at $T \approx 100°C$)

Specific heat	c_p	4.219×10^3 J/kg K
Density	ρ	957.8 kg/m^3
Dynamic viscosity	μ	279×10^{-6} kg/m s
Thermal conductivity	k_f	680×10^{-3} W/m K

fluid properties: temperature-dependent

Specific heat	c_p	$4.179 \times 10^3 + (T-40)^2/90$ J/kg K
Density	ρ	$1000.0 - 4.22 \times 10^{-3}\, T^2$ kg/m^3
Dynamic viscosity	μ	$279 \times 10^{-4}/T$ kg/m s
Thermal conductivity	k_f	$0.688 - 6.2 \times 10^{-6}(T-135)^2$ W/m K

Table 1. Standard well structure parameters and fluid properties.

up the annulus, down the inside of the promoter, down the inlet leg of the u-tube and up the outlet leg, are treated separately but are connected by mass and heat flow continuity conditions. The temperatures, T_o, T_i, T_1 and T_2 respectively, of the fluid in each of the flows depend on the distance, x, from the bottom of the well, which is of length L.

Referring to Figure 2(a), the rate at which heat is being gained by the fluid in the annulus between x and $x+\Delta x$ is

$$(mu_o)_{x+\Delta x} - (mu_o)_x = m\Delta u_o = mc_p\Delta T_o = mc_p\frac{dT_o}{dx}\Delta x + O(\Delta x^2) \quad (1)$$

where u is the specific internal energy and c_p is the specific heat of the fluid. The heat flux into the fluid flowing in the annulus, using the standard heat flow equation ($Q = UA\Delta T$), is given by

$$U_o\pi d_o\Delta x(T_w - T_o) + U_i\pi d_i\Delta x(T_i - T_o) \quad (2)$$

where U_o is the overall coefficient for heat transfer between the reservoir [temperature $T_w(x)$] and the annulus flow, and U_i is the overall heat transfer coefficient between the annulus and the inside of the promoter. The diameters d_o and d_i are those of the outside of the casing and promoter pipes respectively (the overall heat transfer coefficients are values per unit outside area of the tubes).

Equating (1) and (2) and letting $\Delta x \to 0$ gives

$$mc_p\frac{dT_o}{dx} = U_o\pi d_o(T_w - T_o) + U_i\pi d_i(T_i - T_o) \quad (3)$$

Similar heat flux balances for the other flows give

$$mc_p\frac{dT_i}{dx} = U_i\pi d_i(T_i - T_o) + U_1\pi d_1(T_i - T_1) + U_2\pi d_2(T_i - T_2) \quad (4)$$

$$m_hc_p\frac{dT_1}{dx} = U_1\pi d_1(T_1 - T_i) \quad (5)$$

$$m_hc_p\frac{dT_2}{dx} = U_2\pi d_2(T_i - T_2) \quad (6)$$

where the parameters have been described above and the subscripts 1 and 2 refer to the inlet and outlet legs of the u-tube.

Equations (3) – (6) form a system of four first-order ordinary differential equations which are to be solved subject to the boundary conditions:

$$T_o(0) = \alpha T_w(0) + (1-\alpha)T_i(0) \quad (7)$$

$$T_i(L) = T_o(L) \quad (8)$$

$$T_1(L) = T_{in} \quad (9)$$

$$T_2(0) = T_1(0) \quad (10)$$

These describe, respectively, the mixing at the base of the well, the continuity of temperature as the well fluid passes over the top of the promoter tube, the temperature T_{in} of the clean water flowing into the u-tube, and continuity of temperature at the bottom of the u-tube. The bottom and top of each of the well, promoter tube and u-tube are at $x = 0$ and $x = L$ respectively. The reservoir temperature is taken to be a known smooth function of depth [i.e. $T_w = T_w(x)$], and is a linear function in the examples below.

(The corresponding equations and boundary conditions for Configuration 2, shown schematically in Figure 3(a), are given in the Appendix.)

The overall heat transfer coefficients U are calculated as a combination of heat transfer coefficients of the boundary layers in the flows, the conduction of the pipes, and "fouling factors", and are given in values per unit outside area of each tube. The well casing is assumed to be cemented to the reservoir matrix. The coefficients are given by:

$$\frac{1}{U_o} = \frac{d_o \ln[d_o/(d_o-2t_o)]}{2k_o} + \frac{d_o}{d_o-2t_o}\left[\frac{1}{h_o} + f_{oi}\right]$$

$$\frac{1}{U_i} = f_{io} + \frac{1}{h_o} + \frac{d_i \ln[d_i/(d_i-2t_i)]}{2k_i} + \frac{d_i}{d_i-2t_i}\left[\frac{1}{h_i} + f_{ii}\right]$$

$$\frac{1}{U_1} = f_{1o} + \frac{1}{h_i} + \frac{d_1 \ln[d_1/(d_1-2t_1)]}{2k_1} + \frac{d_1}{d_1-2t_1}\left[\frac{1}{h_1} + f_{1i}\right]$$

$$\frac{1}{U_2} = f_{2o} + \frac{1}{h_i} + \frac{d_2 \ln[d_2/(d_2-2t_2)]}{2k_2} + \frac{d_2}{d_2-2t_2}\left[\frac{1}{h_2} + f_{2i}\right]$$

$$(11)$$

where the $f_{\beta\gamma}$ are fouling factors (β = pipe identification, $\gamma = o$ or i to indicate outside or inside surface) and t_β and k_β are respectively the thickness and thermal conductivity of pipe β. The convective heat transfer coefficients h_β are calculated using the Dittus-Boelter correlation for forced convection in pipes:

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.4}$$

and

$$Nu = \frac{hD_e}{k_f}, \quad (12)$$

where Nu, Re and Pr are respectively the Nusselt, Reynolds and Prandtl numbers for the flow, D_e is the effective hydraulic diameter and k_f is the thermal conductivity of the fluid.

The Reynolds number is calculated using the standard formula

$$Re = \frac{\rho \bar{v} D_e}{\mu}$$

where $\bar{v}$ is the average velocity (volume flux per unit area). For example, in the annulus, $\bar{v} = \frac{m}{\rho A_o}$ and

$$Re_o = \frac{mD_o}{\mu\pi[(d_o-2t_o)^2 - d_i^2]/4} \quad (13)$$

The effective diameters for each of the pipe flows are calculated using the formula "four times flow area divided by wetted perimeter", to give

$$D_o = d_o - 2t_o - d_i$$

$$D_i = \frac{(d_i-2t_i)^2 - d_1^2 - d_2^2}{d_i - 2t_i + d_1 + d_2}$$

$$D_1 = d_1 - 2t_1$$

$$D_2 = d_2 - 2t_2 \quad (14)$$

where d_β and t_β are the outside diameter and the thickness of pipe β.

So far, the mass flowrate, m, of the fluid in the well is not fixed. This is calculated as a function of the temperature profiles, as described next.

NATURAL CONVECTION IN THE WELL AND PROMOTER TUBE

Because of the cooling effect of heat extracted by the u-tube DHE, the wellbore fluid near the u-tube inside the promoter is cooler and therefore heavier than fluid in the annulus outside the promoter tube. In the steady state, the difference in the buoyancy of the two columns of wellbore fluid produces a steady convective flow, treated here for heat transfer purposes as forced convection, where the buoyancy forces are balanced by the flow resistance, or net pressure drop, in the pipework.

This buoyancy driving force is given by

$$\Delta p = \int_0^L \rho_i g\,dx - \int_0^L \rho_o g\,dx = \int_0^L (\rho_i - \rho_o) g\,dx \quad (15)$$

where ρ is the (temperature-dependent) density of the fluid.

The rate at which the pressure drops along a pipe of effective diameter D_e is given by the standard correlation

$$\left|\frac{dp}{dx}\right| = \lambda \frac{\rho \bar{v}^2}{2D_e} \quad (16)$$

where λ is the friction factor and $\bar{v} = \frac{m}{\rho A}$, where A is the cross-sectional area of the flow. Substitution gives

$$\left|\frac{dp}{dx}\right| = \frac{\lambda}{2}\frac{m^2}{\rho D_e A^2} \quad (17)$$

The friction factor coefficient λ is a function of Reynolds number, Re, and relative roughness, ε/d, of the pipe. It can be calculated explicitly using the Churchill equation, as follows:

$$\lambda = 8[(8/Re)^{12} + (A+B)^{-3/2}]^{1/12} \quad (18)$$

where

$$A = \{2.457 \ln[(7/Re)^{0.9} + 0.27\varepsilon/d]\}^{16} \text{ and } B = (37530/Re)^{16}.$$

From (17), the total pressure drop in the pipework is

$$\Delta p = \frac{m^2}{2D_oA_o^2}\int_0^L \left[\frac{\lambda}{\rho}\right]_o dx + \frac{m^2}{2D_iA_i^2}\int_0^L \left[\frac{\lambda}{\rho}\right]_i dx \quad (19)$$

Equating (15) and (19) gives an expression from which the wellbore mass flowrate, m, can be Calculated:

$$m^2 = \frac{2g \int_0^L (\rho_i - \rho_o)dx}{\frac{1}{D_oA_o^2}\int_0^L \left[\frac{\lambda}{\rho}\right]_o dx + \frac{1}{D_iA_i^2}\int_0^L \left[\frac{\lambda}{\rho}\right]_i dx} \tag{20}$$

It should be noted that it is assumed that the major part of the frictional pressure drop occurs between $x = 0$ and $x = L$, and that there is a negligible contribution from the bend in the flow at $x = L$.

POWER OUTPUT

The power output of the heat exchanger is given by

$$P = m_h u_2(L) - m_h u_1(L)$$

$$= m_h \left[\int_{x=L}^{x=0} du_1 + \int_{x=0}^{x=L} du_2 \right]$$

$$= m_h \left[- \int_{x=0}^{x=L} c_p \frac{dT_1}{dx} dx + \int_{x=0}^{x=L} c_p \frac{dT_2}{dx} dx \right]$$

Substitution from *(5)* and *(6)* gives

$$P = \int_0^L U_1 \pi d_1 (T_i - T_1)\, dx + \int_0^L U_2 \pi d_2 (T_i - T_2)\, dx \tag{21}$$

In the case where c_p is considered to be constant (i.e. independent of temperature T) then the expression for **P** is simplified to

$$P = m_h c_p [T_2(L) - T_1(L)]$$

$$= m_h c_p \Delta T \tag{22}$$

where $\Delta T = T_2(L) - T_{in}$ is the difference between the inlet and outlet temperatures of the water flowing in the u-tube.

METHODS OF SOLUTION

If it is assumed that the parameters c_p, ρ, μ and k_f are independent of temperature for the purpose of calculating the Reynolds and Prandtl numbers Re and Pr and the heat transfer coefficients h_β, then the overall heat transfer coefficients U_β are independent of temperature. This reduces the equations (3) – (6) to a system of four first-order ordinary differential equations with constant coefficients. The system can be solved analytically. If the reservoir temperature $T_w(x)$ is a linear function

$$T_w = T_B - (T_B - T_T)\frac{x}{L}$$

then the general solution has the form

$$T_\beta = \sum_{k=1}^{4} T_{\beta k} e^{\lambda_k x} + T_{\beta 5} x + T_{\beta 6} \tag{23}$$

where $\beta = o, i, 1, 2$. The four constants λ_k, $k = 1, 2, 3, 4$ are the roots of the quartic polynomial equation derived as the auxiliary equation for the complementary function. The coefficients $T_{\beta k}$ are found by substituting (23) into equations (3) – *(6)* and equating coefficients (this gives **24** equations, **4** of which are redundant) and into the boundary conditions (7) – (10) (this gives **4** equations).

The resulting set of 20 + **4** = **24** equations in the **24** unknowns $T_{\beta k}$ can be solved using any suitable algorithm (in this case a NAg FORTRAN routine was used). The expressions (23) can then be used to calculate the temperature profiles in the four fluid flows. An expression describing the variation of density with temperature in the wellbore fluid (see Table **1**) is used to calculate $\rho_o(x) = \rho[T_o(x)]$ and $\rho_i = \rho[T_i(x)]$. A numerical evaluation of the integrals in (20) gives a value of m, the mass flow in the wellbore. This value is used to recalculate the coefficients in equations (3) – (6) and the solution process is repeated until convergence takes place. The power output can then be calculated using (22).

A direct numerical solution is necessary if the fluid parameters are allowed to vary with temperature. (The correlations for such variations are shown in Table 1). The numerical procedure used here was a straightforward second-order finite-difference scheme, with a direct solution for the grid-point values (again via a NAg routine) and iteration to the final solution. All parameters were allowed to vary with temperature at each grid point, and an updated value of m used at each pass. After convergence, the power output was calculated using (21).

As mentioned in the Introduction, a standard well structure whose properties are listed in Table **1** is used as a basis. Five parameters were varied, one at a time: d_i, k_i, k_1 $(= k_2)$, m_h and *a.* The values for constant fluid properties used in the "analytical" solution method, and the correlations for temperature dependent fluid properties are. also shown in Table **1.** The pipes were all assumed smooth (pipe roughnesses $\varepsilon = 0$) and clean (fouling factors $f_{\beta\gamma} = 0$).

RESULTS

For the standard well structure, comparisons were made between the case where the fluid properties c_p, ρ, μ, and k_f are assumed constant and where they *are* taken as functions of temperature (see Table 1). For Configuration 2 (u-tube beside the promoter) there was little difference between the respective induced well mass flowrates m, the temperature differences ΔT between inlet and outlet of the u-tube and the power ouputs **P** for the two cases (see Table 2). However, for Configuration **1** (u-tube inside the promoter) it can be seen that there was a significant difference. To ensure that there was no mistake or numerical error, an alternative numerical procedure was used to derive each set of results. Calculations were also made for another well structure (the standard well structure parameters were varied by taking $k_i = 2.0$ W/m **K**, $m_h = 2.0$ kg/s, $a = 0.6$); a similar difference in results was found (see Table *3).* The reason for the different results is not clear, although it may be that the temperature variations with depth in Configuration 2 are more pronounced than in Configuration 1 where the temperature profiles have near constant gradients [see Figures 2(b) and 3(b)].

The temperature profiles on the well wall (i.e. in the reservoir) and inside the well, promoter and legs of the u-tube for the standard well structure, together with graphs showing the effects of separately varying the parameters d_i, k_i, k_1 $(= k_2)$, m_h and *a,* are shown for Configuration **1** in Figure 2 and for Configuration 2 in Figure 3. The calculations have been made using the constant property assumption for the fluids.

Configuration 1

Figure 2(b) shows that the cold fluid entering the u-tube gains heat rapidly as it flows to the bottom; as this fluid rises again in the upward leg, it reaches a point where it is hotter than the convecting fluid in the promoter. This is marked by a change in gradient of T_2 from positive to negative as the fluid now starts to lose heat to its colder surroundings. Nearer the top of the well, the difference between the temperatures of the flows is less and the gradient of T_2 decreases in magnitude.

Similarly, the hot well fluid entering at the bottom loses heat, as it rises, to the flow in the promoter and to the well wall. The fluid inside the promoter tube gains heat from the rising well fluid and the rising u-tube flow while losing heat to the descending u-tube fluid.

The parameter d_i is restricted in its variation for Configuration **1** by the geometry of the well structure; the promoter tube has to contain the two u-tube legs while also being contained in the well casing. Variation of d_i [see Figure 2(c)] shows a maximum for the power output **P** when $d_i \approx 0.145$ m, which corresponds to the promoter tube occupying approximately half the cross-sectional area of the well and also coincides with a maximum for the wellbore flow m. This also agrees with results obtained by Allis **(1981)** who suggested that the maximum velocity in the convection cell occurs at this ratio. The greater circulation allows the colder convective fluid to be constantly replaced by hot fluid; this improves the rate of heat transfer.

Config.	Fluidprops	m (kg/s)	ΔT (°C)	P (kW)
1	constant	4.3	11.4	48.2
	variable	3.6	16.5	68.7
2	constant	12.6	50.3	212
	variable	12.1	50.9	213

Table 2. Comparison of calculated results for constant and variable fluid properties (standard well structure).

Config.	Fluid props	m (kg/s)	ΔT (°C)	P (kW)
1	constant	4.6	14.5	122
	variable	3.9	20.7	173
2	constant	13.9	59.1	498
	variable	13.8	60.5	506

Table 3. Comparison of calculated results for constant and variable fluid properties (variation of standard well structure to $k_i = 2.0$ W/m K, $m_h = 2.0$ kg/s, $\alpha = 0.6$).

Variation of the thermal conductivity of the promoter, shown in Figure 2(d), and the u-tube, shown in Figure 2(e), indicate that it is better to use relatively low-conductivity materials. This is clear for the promoter tube: the well fluid is transported with less heat loss to the top of the well so that the u-tube is presented with a hotter fluid. Also, because the natural convective flow is driven by density variation, a large difference in temperature between the fluid inside the promoter and that in the annulus causes a higher flow velocity for the hot water circulating in the well.

For a highly conductive u-tube pipe, the fluid gains heat quickly as it flows downward; however, as the fluid returns up the outlet leg, it starts to lose heat to its surroundings after a short distance. In a conductive pipe, the heat loss is large, whereas in a less conductive pipe, although the initial heat gain is not as large, the fluid is better able to retain its heat. A possible better solution to the problem, not investigated in detail here, would be to make the two legs of the u-tube from different materials. The inlet leg of the u-tube should be highly conductive to maximise heat intake, while the outlet leg should be a poor conductor, or insulated, to minimise heat loss from the ascending fluid.

Figure 2(f) shows that the calculated heat output is almost proportional to the u-tube flowrate. This cannot be realistic from a practical point of view; the reason for the high values arises from the assumption that the well wall can be modelled as being at constant temperature. High heat flowrates from the u-tube would certainly cause a temperature drop in a real well wall; the model needs adjusting to account for this.

The mixing ratio α models the cross flow at the bottom of the well. Here α varies from 0, where there is no replenishment of wellbore fluid from the reservoir, to 1, where the downflowing fluid in the promoter tube passes out to the reservoir and is replaced by fresh fluid at the reservoir temperature T_B. Figure 2(g) shows the effect of varying α on the power output P for three different promoter conductivities, 5, 54 (standard) and 200 W/m K. The results indicate that the effects of mixing are important only when the promoter conductivity is relatively small.

Configuration 2

Comments for this configuration are generally the same as given above. The main difference is that the heat output when the promoter is beside the u-tube is about four times that when the u-tube is inside the promoter, all other parameters being the same.

The function of the promoter tube in Configuration 2, shown in Figure 3(a), is to transport hot fluid from the bottom of the well to the top from where it descends past the u-tube legs and past the well wall. Temperature T_i is consequently higher than for Configuration 1 [see Figure 3(b)] and has the effect of significantly boosting the u-tube temperatures and providing a higher power output from the well.

The promoter diameter d_i remains constrained at its maximum, but can have a small radius. The variation of power output P with d_i, shown in Figure 3(c), has two turning points, with a maximum at $d_i \approx 0.11$, which is about half of the well diameter. This also corresponds to a maximum value of m, and agrees with the results of Allis (1981). The cross-sectional flow areas in the annulus and inside the promoter are approximately equal at this ratio. Allis suggested that for promoters with larger diameters, heat flow is mainly through convection and not conduction.

The smaller values of d_i again give larger P values, but the corresponding flowrates m are not large; the greater power output presumably occurs because of the thermal "inertia" of the larger volume of fluid surrounding the u-tube legs.

The variation of P with k_i, k_l, m_h and α, shown in Figures 3(d – g), are similar to those for Configuration 1, except that the graph of P vs k_l has a minimum near $k_l = 10$ W/m K.

SUMMARY and CONCLUSIONS

The downhole heat exchanger (DHE) is an important alternative method of tapping geothermal energy. This investigation has attempted to model the DHE using steady-state heat transfer equations, solved by a combination of analytical and numerical methods.

A standard well with typical dimensions and materials has been examined. Two different configurations have been studied: Configuration 1 has the u-tube placed inside the promoter, while the u-tube is placed beside the promoter tube in Configuration 2. The effect on the heat output of the DHE of separately varying five of the well structure parameters has been investigated.

The following conclusions can be drawn:

It is much more efficient to have the u-tube beside the promoter than inside it. Even for a considerable variation from the optimum structure, the heat transfer resulting from Configuration 2 is much greater than from Configuration 1. The simple procedure of placing a tube into a well beside the DHE u-tube should cause a greatly enhanced power output.

The promoter in any case should ideally be made of a material with low conductivity, to decrease heat loss from the ascending well fluid.

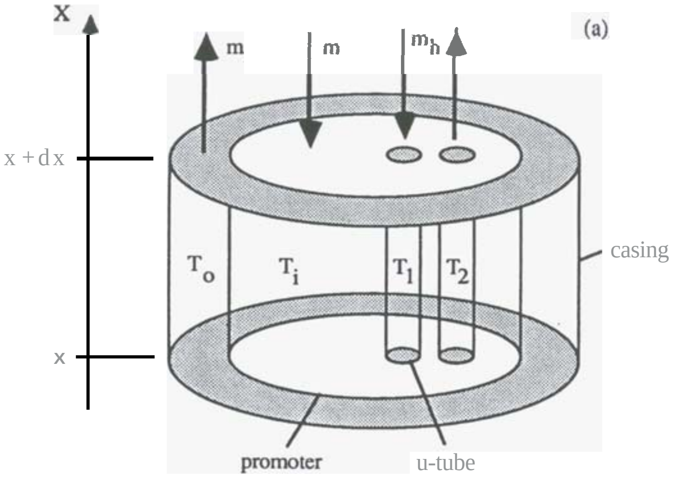
The promoter tube has the greatest effect if it is about 0.5 times the well diameter for Configuration 1 and about 0.7 times the well diameter for Configuration 2.

The u-tube may perform more efficiently if the outlet leg was insulated or made of a material with low conductivity; this would ensure that the heat gained by the descending working fluid was not lost during ascent.

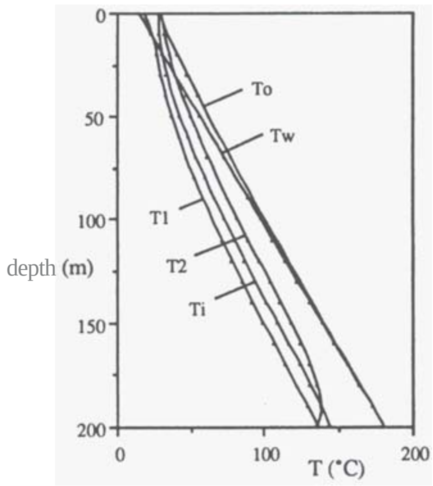
The amount of mixing, or replacement of fluid at the base of the well, has little effect on the power output unless the promoter has low conductivity; even in that case the effect is apparently not greater than about 25% across the full range of α.

The assumption that the well wall has a fixed temperature profile for all mass flows produces what are probably unrealistic results when m_h, the u-tube flowrate, is large. Some modification of the model would be necessary to cope with such high flowrates.

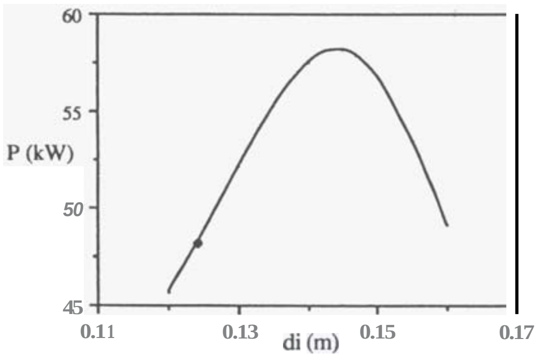
Configuration 1



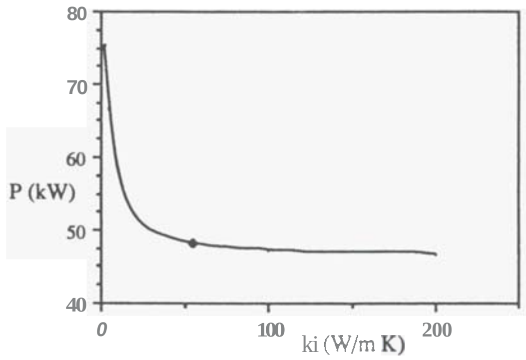
(b) Configuration 1: temperature profiles



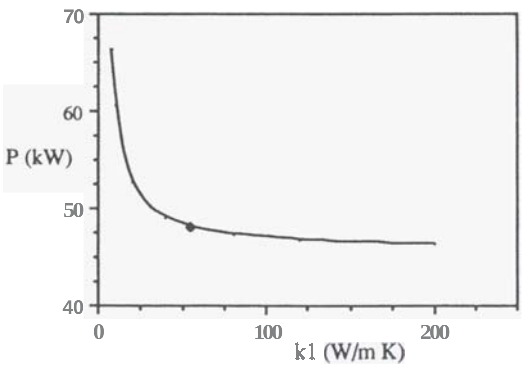
(c) Configuration 1: P vs di



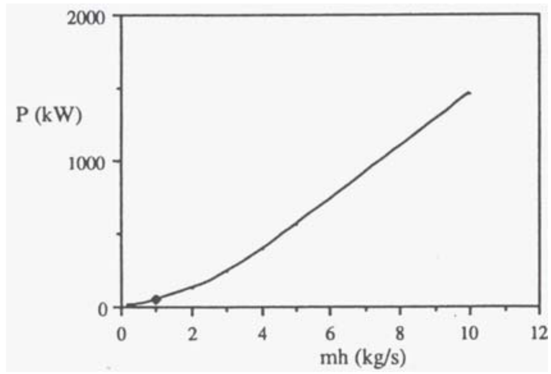
(d) Configuration 1: P vs ki



(e) Configuration 1: P vs k1



(f) Configuration 1: P vs mh



(g) Configuration 1: P vs alpha

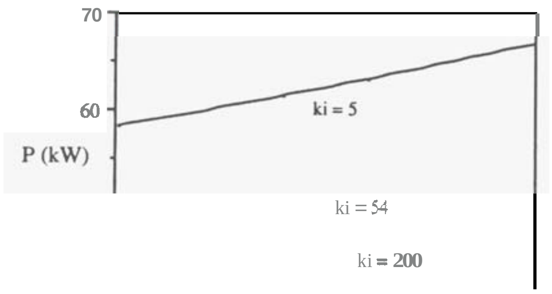
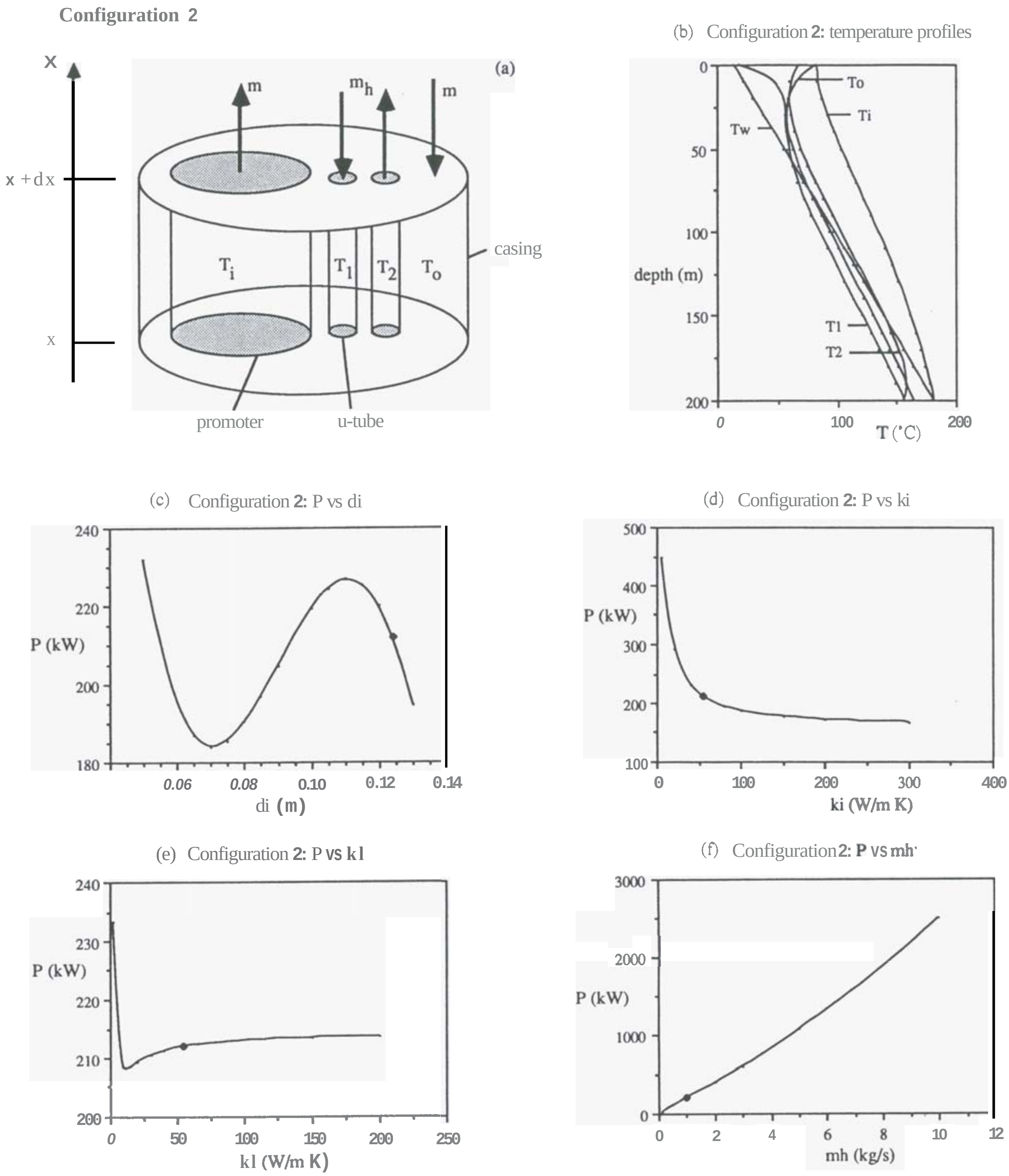


Figure 2. Results for Configuration 1: (a) schematic diagram of the tube configuration; (b) temperature profiles for T_w, T_o, T_i, T_1 and T_2; power output P as a function of (c) d_i; (d) k_i; (e) k_1 ($=k_2$); (f) m_h; (g) α. For each of (c) – (g), other parameters remain as in the standard well structure; the spot • marks the value for the standard well structure.



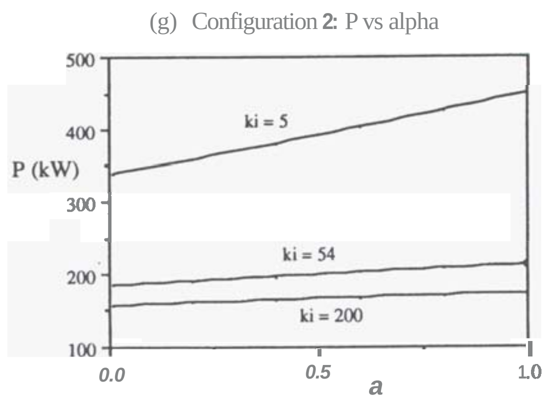


Figure 3. Results for Configuration 2: (a) schematic diagram of the tube configuration; (b) temperature profiles for T_w, T_o, T_i, T_1 and T_2; power output P as a function of (c) d_i; (d) k_i; (e) k_1 (=k_2); (f) m_h; (g) a. For each of (c) – (g), other parameters remain as in the standard well structure; the spot • marks the value for the standard well structure.

APPENDIX

Equations for Configuration 2 (u-tube beside promoter)

The set of equations resulting from heat flux balances for Configuration 2, where the promoter and u-tube stand side by side in the well [*see* Figure 3(a)] are

$$mc_p \frac{dT_o}{dx} = U_o \pi d_o (T_o - T_w) + U_i \pi d_i (T_o - T_i) + U_1 \pi d_1 (T_o - T_1) + U_2 \pi d_2 (T_o - T_2)$$

$$mc_p \frac{dT_i}{dx} = U_i \pi d_i (T_o - T_i)$$

$$m_h c_p \frac{dT_1}{dx} = U_1 \pi d_1 (T_1 - T_o)$$

$$m_h c_p \frac{dT_2}{dx} = U_2 \pi d_2 (T_o - T_2)$$

with boundary conditions

$$T_i(0) = \alpha T_w(0) + (1 - \alpha) T_o(0)$$

$$T_o(L) = T_i(L)$$

$$T_1(L) = T_{in}$$

$$T_2(0) = T_1(0)$$

Appropriate modifications are made to the overall heat transfer coefficients in (11), the effective diameters in (14) and the flow cross-sectional areas, due to the slight changes in geometry from Configuration 1.

NOTE

The general approach of this work is based on a study made by Sushila Nair for the Project in Engineering Science at the School of Engineering, University of Auckland, in 1986.

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