Design of the Glenwood Springs
Downhole Heat Exchanger

MASTER

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Idaho Operations Office • Idaho National Engineering Laboratory

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FOOTNOTES

Peter Chiu

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ABSTRACT

A heat exchanger has been designed to obtain 250,000 Btu/hr from a 20-in.-diameter geothermal well at various brine temperatures. The system consists of a 10-in.-diameter plastic pipe to promote convective flow in the well and a 4-in.-diameter, Schedule 40 steel U-tube containing distilled water to extract the energy. Subject to the validity of the major assumptions, the required lengths of one leg of the U-tube at various brine temperatures are 34 ft (150°F), 42 ft (140°F), 54 ft (130°F), and 75 ft (120°F) for a mean working fluid temperature of 90°F.

FOREWORD

The Geothermal Technical Assistance Program was developed under the premise that the majority of groups or individuals with available geothermal resources do not have the experience or manpower necessary to do a preliminary engineering and economic feasibility evaluation for geothermal energy projects. In order to disseminate technical information and to facilitate expanded use of geothermal energy resources, assistance was provided through FY-1981 in a consulting format on a first-come, staff-and-funds-available basis. Technical assistance can relate to conceptualization; engineering; economics; water chemistry implications for environmental, disposal, and material selection considerations; and planning and development strategies. This report is one of a series adapted from consultation provided to requesters either through in-house efforts or through limited efforts subcontracted to local engineering firms. The Geothermal Technical Assistance (GTA) reports in this series, which are listed below, will be available for purchase early in 1982 by those with interest in specific geothermal applications from the U.S. National Technical Information Service:

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8.	*EGG-2137	<u>Geothermal Source Potential and Utilization for Methane Generation and Alcohol Production (subcontractor report)</u>
9.	*EGG-2138	<u>Geothermal Source Potential and Utilization for Alcohol Production (subcontractor report)</u>
10.	*EGG-2139	<u>Potential Geothermal Energy Applications for Idaho Elks Rehabilitation Hospital (subcontractor report)</u>
11.	*EGG-2144	<u>Technical Assistance Report on a Geothermal Heating Utility for Lemmon, South Dakota (subcontractor report)</u>
12.	*EGG-2145	<u>Economic Analysis for Utilization of Geothermal Energy by North Dakota Concrete Products Company (subcontractor report)</u>
13.	*EGG-2146	<u>Geothermal Feasibility Analysis II for Polo School District No. 29-2, South Dakota (subcontractor report)</u>
14.	*EGG-2147	<u>Preliminary Feasibility Study of Heating and Cooling Alternatives for Nebraska Western College, Scottsbluff, Nebraska (subcontractor report)</u>
15.	*EGG-2148	<u>Inventory of Thermal Springs and Wells Within a One-Mile Radius of Yucca Lodge, Truth or Consequences, New Mexico (subcontractor report)</u>

<u>GTA Report Number</u>	<u>EG&G Report Number</u>	<u>Title</u>
16.	EGG-2149	<u>Utilization of Geothermal Energy, Feasibility Study--Ojo Caliente Mineral Springs Company, Ojo Caliente, New Mexico (subcontractor report)</u>
17.	*EGG-2150	<u>Geothermal Heated Office Building at Glenwood Springs, Colorado (subcontractor report)</u>
18.	EGG-2151	<u>Final Report--Dickinson Geothermal Study, Dickinson, North Dakota (subcontractor report)</u>
19.	EGG-2152	CANCELLED
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28.	EGG-GTH-5804	<u>An Overview of Engineering and Agricultural Design Considerations of the Raft River Soil-Warming and Heat-Dissipation Experiment</u>
29.	EGG-GTH-5812	<u>Design of the Glenwood Springs Downhole Heat Exchanger</u>

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DESIGN OF THE GLENWOOD SPRINGS DOWNHOLE HEAT EXCHANGER

INTRODUCTION

This report describes the design of a downhole heat exchanger (DHX), with the parameters given in Figure 1, to use the thermal energy of brine in a geothermal well for space heating. A later redesign with new parameters is discussed in Appendix A.

It has been suggested that the convection-promoting (c-p) pipe be nonconducting, with a diameter of half the well-bore, and that the DHX pipe have a diameter of one quarter the well bore.¹ Therefore, the design will use a 10-in. (ID) plastic tube ($t_w = 0.125$ in.) for the c-p pipe and 4-in. Schedule 40 steel piping for the U-tube (ID = 4.026 in.; OD = 4.500 in.).

HEAT TRANSFER COEFFICIENTS

Given that the C_p of water can be considered constant at $1.00 \frac{\text{Btu}}{\text{lb}_m \cdot ^\circ\text{F}}$ over the temperature range of 60 to 150°F,

$$\dot{m}_{wf} = \frac{\dot{q}}{C_p \Delta T_{wf}}$$

$$\dot{m}_{wf} = 6.9444 \text{ lb}_m/\text{sec}$$

Other pertinent properties of water at various temperatures are given in Table 1.²

TABLE 1. THERMODYNAMIC PROPERTIES OF WATER

Temperature (°F)	Viscosity (lbm/ft-sec)	Density (lbm/ft ³)	Thermal Conductivity (Btu/sec-ft-°F)
150	2.90×10^{-4}	61.30	1.064×10^{-4}
140	3.24×10^{-4}	61.46	1.053×10^{-4}
130	3.57×10^{-4}	61.62	1.043×10^{-4}
120	3.91×10^{-4}	61.78	1.032×10^{-4}
90	5.18×10^{-4}	62.15	0.996×10^{-4}

Well bore with casing (ID)—20 in.
 Well temperature (T_h)—150°, 140°, 130°, & 120°F
 DHX configuration—single U-tube of Schedule 40 steel
 Heat load (q)—250,000 Btu/hr (69.4444 Btu/sec)
 Working fluid—distilled water
 ΔT_{wf} —10°F

INEL 2 0755

Figure 1. Schematic design.

Since the energy is to be used for space heating, the lowest temperature allowable for the working fluid is room temperature plus approach temperature. Using 85° for T_c , $T_u = 95^\circ\text{F}$. With a mean temperature of 90°F and a diameter of 4.026 in., the Reynolds and Prandtl numbers (see Appendix B) for the working fluid are

$$\text{Re} = 5.09 \times 10^4$$

$$\text{Pr} = 5.20$$

Using the Dittus-Boelter correlation (see Appendix C), the inside heat transfer coefficient is

$$h_i = 0.077 \text{ Btu/sec-ft}^2\text{-}^\circ\text{F}$$

The thermal conductivity of mild steel (1% C) is $0.00689 \text{ Btu/sec-ft-}^\circ\text{F}$, used in determining the thermal resistance of a cylinder by the equation²

$$R_c = [\ln(r_o/r_i)]/(2\pi kL)$$

Fouling resistances for distilled water (inside) and hard water above 125°F (outside) are³

$$R_i = 1.8 \text{ sec-ft}^2\text{-}^\circ\text{F/Btu}$$

$$R_o = 18.0 \text{ sec-ft}^2\text{-}^\circ\text{F/Btu}$$

To determine the outside heat transfer coefficient, an energy balance of convective flow in the well is initially performed. The control volume used consists of the well to the depth L , excluding the U-tube internal flow but including the c-p pipe.

$$\dot{m}H_{T_h} = \dot{m}H_{T_f} + \dot{q} + \text{frictional loss rate}$$

$$\dot{m} \left(H_{T_h} - H_{T_f} \right) = \dot{q} + \dot{m} \times \text{head losses}$$

The head losses that actually exist involve the entrance to the c-p pipe, flow through the c-p pipe, the exit of the c-p pipe, the directional change of flow, the entrance to the DHX shell (shell: that part of the control volume outside the c-p solid cylinder), flow through the shell, flow around the U-tube bend, and the exit of the shell.

Assuming that $\frac{L}{D} \gg 50$, then entrance and exit effects can be neglected.² Since flow around the U-tube bend is expected to occur near the shell exit, this source of head loss will also be neglected. The system is then modeled as shown in Figure 2, where Section I accounts for c-p pipe flow and directional change, and Section II accounts for shell flow.

Section I — c-p pipe + 180° bend
Section II — DHX shell

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Figure 2. Convective flow model.

The energy balance then becomes

$$\dot{m} c_p (T_h - T_f) = \dot{q} + 0.0012841 \frac{\text{Btu}}{\text{ft-lb}_f} \times \dot{m} (h_L^I + h_L^{II})$$

where

$$h_L = \text{head loss} = 2 f_F \left(\frac{L}{D} \right) \left(\frac{v_{\text{avg}}}{g} \right)^2$$

To calculate head loss through the shell, an equivalent diameter (D_{eq}) must be determined via the hydraulic radius (R_h).

$$R_h = S/Z = \text{cross-sectional area/wetted perimeter}$$

$$S = \frac{\pi}{4} [20^2 - 10.25^2 - 2(4.5^2)] \times \frac{1}{144} = 0.442 \pi \text{ ft}^2$$

$$Z = \pi [20 + 10.25 + 2(4.5)] \times \frac{1}{12} = 3.271 \pi \text{ ft}$$

$$D_{\text{eq}} = 4 R_h = 4 S/Z = 0.54021 \text{ ft}$$

An iterative method must now be used to determine $\dot{m}$, $(T_h - T_f)$, and L such that the system will maintain flow. Assuming that the thermodynamic properties of brine can be considered the same as water, and that $T_h - T_f = 2^\circ\text{F}$, then $\dot{m}_{\text{min}} = 34.722 \text{ lb}_m/\text{sec}$. Using an allowance of 1% in flow rate to account for head losses, $\dot{m} = 35.0 \text{ lb}_m/\text{sec}$. Relative roughness in the c-p is considered negligible, while roughness in the shell is weighted according to fractional wetted perimeter. Using $E = 0.01 \text{ ft}$ for concrete casing and $E = 0.00015 \text{ ft}$ for commercial steel, $(D_{\text{eq}}/E_{\text{avg}}) = 105.3$.

Reynolds numbers, average velocities, and Fanning friction factors are then determined. The resulting values are given in Table 2.

TABLE 2. CONVECTIVE FLOW DATA

Temperature (°F)	Re _I	Re _{II}	v_{avg}^I (ft/sec)	v_{avg}^{II} (ft/sec)	$f_F^{I^a}$	$f_F^{II^a}$
150	1.844×10^5	4.697×10^4	1.047	0.411	0.0040	0.0098
140	1.651×10^5	4.204×10^4	1.044	0.410	0.0040	0.0098
130	1.498×10^5	3.816×10^4	1.041	0.409	0.0041	0.0099
120	1.368×10^5	3.484×10^4	1.039	0.408	0.0041	0.0099

a. Fanning friction factors from Reference 2, p. 209.

To determine the head loss in Section I of the system, an equivalent length $L_{eq} = 75D = 62.5$ ft is used for the 180-degree bend.² In Section II, D_{eq} is used with v_{avg} based on $S (= R_h Z)$.

$$h_L^I = 2 \times 0.004 [(L + 62.5 \text{ ft})/0.8333 \text{ ft}][1.043 \text{ ft/sec}^2/32.174 \text{ ft/sec}^2]$$

$$= 0.000325L + 0.0203 \text{ ft}$$

$$h_L^{II} = 2 \times 0.01 (L/0.54021 \text{ ft})[(0.41 \text{ ft/sec})^2/32.174 \text{ ft/sec}^2]$$

$$= 0.000193L$$

$$h_L^{ttl} = h_L^I + h_L^{II} = 0.000518L + 0.0203 \text{ ft}$$

Rearranging the energy balance with the assumed m and $T_h - T_f$ and equating it with h_L^{ttl}

$$h_L \text{ (available)} = \frac{c_p (T_h - T_f) - (\dot{q}/\dot{m})}{0.0012841 \text{ Btu/ft-lb}_f}$$

$$= 12.4 \text{ ft}$$

$$= 0.000518L + 0.0203 \text{ ft}$$

yields $L = 2.4 \times 10^4 \text{ ft}$, indicating that any given temperature drop should easily maintain the required convective flow rate for q , or that near-isothermal conditions prevail in the DHX shell.

Therefore, the problem of determining outside heat transfer coefficients can be solved by using Re_{II} (Table 2) and Prandtl numbers with the Dittus-Boelter correlation. These coefficients are listed in Table 3, along with Prandtl numbers.

TABLE 3. OUTSIDE HEAT TRANSFER COEFFICIENTS

Temperature (°F)	Prandtl Number	h_o (Btu/sec-ft ² -°F)
150	2.73	0.0335
140	3.08	0.0314
130	3.42	0.0297
120	3.79	0.0282

Overall heat transfer coefficients can finally be calculated from the outside surface area, using the equation

$$U_o = \left(\frac{A_o}{A_i h_i} + R_i + A_o R_c + R_o + \frac{1}{h_o} \right)^{-1}$$

CONCLUSION

The required length of a single leg is determined from

$$q = U_o A_o (F \Delta T_{lm})$$

where

$$\Delta T_{lm} = [(T_h - T_u) - (T_f - T_c)] / \ln \left(\frac{T_h - T_u}{T_f - T_c} \right)$$

$$F = 1.0 \text{ (no corrections needed)}$$

$$A_o = \pi \left(\frac{4.5}{12} \right) \text{ ft} \times 2L$$

Overall transfer coefficients, temperature differences, and required are lengths listed in Table 4.

TABLE 4. FINAL RESULTS

Temperature (°F)	U_o (Btu/sec-ft ² -°F)	ΔT_{lm} (°F)	L (ft)
150	0.0149	58.9	34
140	0.0145	48.9	42
130	0.0141	38.9	54
120	0.0137	28.8	75

It should be noted that the validity of the final results are based on the validity of the following assumptions:

- Flow through the c-p pipe is isothermal.
- There is no heat loss through the casing.
- There is no recycling (mixing of T_f and T_h).

- The properties of brine are the same as those of water.
- The working fluid temperature is satisfactory for the system's purpose.

Pilot plant-scale experiments involving various flow rates for the working fluid should be run in order to account for deviations.

REFERENCES

1. R. G. Allis and R. James, "A Natural-Convection Promoter for Geothermal Wells," Transactions, Geothermal Resources Council IV, p. 410, 1980.
2. J. R. Welty et al., Fundamentals of Momentum, Heat, and Mass Transfer, 2nd edition, John Wiley and Sons, 1976.
3. M. S. Peters and K. D. Timmerhaus, Plant Design and Economics for Chemical Engineers, 2nd edition, McGraw-Hill, 1968, p. 536.

APPENDIX A
DOWNHOLE HEAT EXCHANGER REDESIGN

APPENDIX A DOWNHOLE HEAT EXCHANGER REDESIGN

The Glenwood Springs DHX has been redesigned with the following new parameters:

- 10-in. (ID) well bore with casing
- 5-in. (ID) c-p pipe
- 0.125-in. c-p pipe wall thickness
- 2.5-in. Schedule 40 steel U-tube (2.469-in. ID; 2.875-in. OD)
- $10^{\circ}\text{F } T_h - T_u$
- 10°F approach temperature.

Given that $\dot{m}_{wf} = 6.944 \text{ lb}_m/\text{sec}$ (the same as in the original design), new data on inside heat transfer coefficients are given in Table A-1.

TABLE A-1. INSIDE HEAT TRANSFER COEFFICIENTS

Temperature ($^{\circ}\text{F}$)	$\mu \times 10^4$ (lbm/ft-sec)	$k \times 10^4$ (Btu/sec-ft- $^{\circ}\text{F}$)	Re $\times 10^{-5}$	Pr	h_i (Btu/sec-ft ^{2} - $^{\circ}\text{F}$)
135	3.40	1.048	1.26	3.24	0.226
125	3.74	1.038	1.15	3.60	0.216
115	4.08	1.027	1.05	3.97	0.208
105	4.41	1.016	0.974	4.34	0.200

Although convective flow should theoretically maintain near-isothermal conditions in the shell, actual mass flow in the shell is critical to achieving the required heat transfer. Therefore, the new DHX designs are based on minimum shell flow requirements subject to minimum area constraints.

For minimum area, both legs of the U-tube must absorb heat. For minimum shell flow, $T_h - T_f$ should be maximized. With an approach temperature of 10°F , and given that $T_h - T_u = 10^\circ\text{F}$, heat transfer occurs in the return leg only if the temperature difference between the working fluid and the shell fluid is at least 10°F at all points. However, if the temperature difference is greater than 10°F , then shell flow can be reduced. The system is then initially modeled as two exchangers, where the incoming leg is represented by a parallel flow exchanger and the return leg is represented by a counterflow exchanger. The temperature at the U-tube bend (T_b) is 10°F less than T_f .

Using this model,

$$\dot{q}_o = \dot{q}_{inc} + \dot{q}_{ret}$$

$$U_o A_o (F \Delta T_{lm})_o = U_i A_i \Delta T_{lm}^i + U_r A_r \Delta T_{lm}^r$$

where

$$A_i = A_r = \frac{1}{2} A_o, U_i = U_r = U_o, \Delta T_{lm}^r = 10^\circ\text{F}.$$

Iterations of the equation yield the following results:

$$T_h - T_u = 10^\circ\text{F}$$

$$T_u - T_c = 10^\circ\text{F}$$

$$T_f - T_c = 16^\circ\text{F}$$

$$T_h - T_f = 4^\circ\text{F}$$

$$F \text{ (correction factor)} = 0.96.$$

Based on $T_h - T_f = 4^\circ\text{F}$, data for the shell are as follows:

$$\dot{m}_{sf} = 17.361 \text{ lb}_m/\text{sec}$$

$$D_{eq} = 4 \times (0.09706\pi \text{ ft}^2 / 1.75\pi \text{ ft}) = 0.22185 \text{ ft}$$

Data on outside heat transfer coefficients are given in Table A-2.

TABLE A-2. OUTSIDE HEAT TRANSFER COEFFICIENTS

Temperature (°F)	Re x 10 ⁻⁴	Pr	h_o (Btu/sec-ft ² -°F)
150	4.356	2.73	0.0767
140	3.899	3.08	0.0720
130	3.538	3.42	0.0681
120	3.231	3.79	0.0646

Overall heat transfer coefficients and the lengths of a single leg are given in Table A-3 for various well temperatures.

TABLE A-3. FINAL RESULTS

Well Temperature (°F)	U_o (Btu/sec-ft ² -°F)	Length (ft)
150	0.0246	153
140	0.0240	157
130	0.0234	161
120	0.0229	165

These lengths are considerably longer than those in the original design because

- Area/length is smaller
- Driving force is much smaller.

APPENDIX B
REYNOLDS AND PRANDTL NUMBERS

APPENDIX B
REYNOLDS AND PRANDTL NUMBERS

The Reynolds numbers for flow in a conduit is defined as

$$Re = Dv_{avg}\rho/\mu.$$

For a given mass flow rate in any cross-sectional shape,

$$A_{CS} = \text{cross-sectional area}$$

$$P_w = \text{wetted perimeter}$$

$$D = 4R_h = 4 A_{CS}/P_w$$

$$v_{avg} = \dot{m}/(\rho A_{CS})$$

$$Re = 4 \dot{m}/(P_w \mu) .$$

For the special case of a circular tube,

$$P_w = \pi D$$

$$Re = 4 \dot{m}/(\pi D \mu) .$$

The Prandtl number is defined as

$$Pr = \mu c_p/k .$$

APPENDIX C
DITTUS-BOELTER CORRELATION

APPENDIX C
DITTUS-BOELTER CORRELATION

This correlation is used to determine a transfer coefficient for heat exchange between a conduit surface and a fluid in turbulent flow. The correlation is

$$Nu_D = h D / k_{f1} = 0.023 Re_D^{0.8} Pr^n$$

where $n = 0.4$ if the fluid is being heated or $n = 0.3$ if the fluid is being cooled, and where fluid properties are evaluated at the arithmetic-mean bulk temperature.

Restrictions on its use are

- $Re_D > 10^4$
- $0.7 < Pr < 100$
- $L/D > 60.$