Comparison of a Thermoelectric Cooling System and a Groundwater Heat Pump*

V. C. MEI and F. C. CHEN
Research Staff
Energy Division
Oak Ridge National Laboratory
Oak Ridge, Tennessee

B. MATHIPRAKASAM
Principal Engineer
Midwest Research Institute
Kansas City, Missouri

ABSTRACT

The performance of a groundwater thermoelectric (TE) heat pump system, based on today's state of the art TE materials, was calculated and compared with that of a conventional groundwater heat pump under the same water inlet temperature and flow rate. It was found that the TE system was quite competitive for cooling, particularly for groundwater temperatures below 18°C (64°F). The TE system performed poorly for heating mode operation. A cooling coefficient of performance (COP) of 6.4 could be realized by a properly designed TE system at a groundwater temperature of 13°C (55°F), compared with a COP of 4.35 for a conventional heat pump. For heating mode operation at the same water temperature, the TE system achieved a COP of 1.82, while the conventional heat pump performed at a COP of 3.72. TE systems for many buildings, where part of the areas require year-round cooling, can be a good alternative if groundwater of sufficiently low temperature is available.

INTRODUCTION

Recent concern about possible depletion of the ozone layer by the release of certain CFC refrigerants [1] has renewed interest in alternative refrigeration systems. Thermoelectric (TE) refrigeration systems have always been considered for special applications only, where their durability and light weight outweigh their low thermal efficiency. However, the thermal efficiency of a TE system can be greatly improved if the TE modules' hot to cold side temperature differential can be kept small. One way to achieve that it to use low temperature groundwater as a cooling medium. Groundwater has long been considered an excellent heat transfer medium for conventional heat pump application [2,3]. The performance of a water-coupled TE cooling system has been studied before [4,5], and with very promising results. The combination of a TE system with low-temperature groundwater should have a much higher COP than air cooled TE systems.

In this study, the calculated performance of a groundwater-coupled TE heat pump, based on today's state of the art TE materials, was compared with the laboratory tested performance of a conventional groundwater heat pump system under the same water inlet temperature and flow rates and the same inlet air and outlet air temperatures. A cross-flow TE heat exchanger model by Mathiprakasam [6] was modified and used to calculate the TE system's performance.

The COP for the two systems are compared for both heating and cooling mode operation. The water inlet and outlet temperature differentials are also compared. Through these comparisons, it is found that the TE system, under certain design criterion, can outperform the conventional water source heat pump system (WSHP) in cooling mode operation.

MATHEMATICAL MODEL

A cross-flow TE unit, as shown in Fig. 1, is considered for the analysis. The model is based on the energy

ORNL-DWG 82-7964R

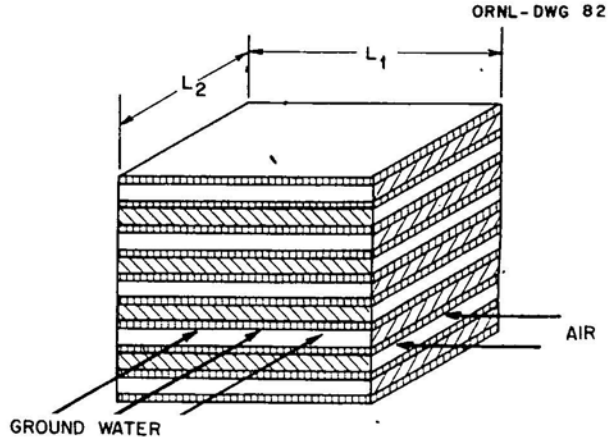

Fig. 1. Schematic of cross-flow TE unit.

balance with the heat transfer medium, air and water. The heating mode and cooling capacities are calculated by TE module equations [7]. The model has two parts, heating and cooling mode operation. They are described in detail below.

Heating mode: (1) water side

$$-M_w C_w L_1 \frac{\partial T_w}{\partial X} = H_w (T_w - T_c) \quad (1)$$

$$-M_w C_w L_1 \frac{\partial T_w}{\partial X} = n\left[\alpha T_c I - \frac{0.5 I^2 \rho}{\nu} - k\nu (T_h - T_c)\right] \quad (2)$$

where

$$k\nu = k_p A_p / l_p + k_n A_n / l_n$$

$$\rho/\nu = \rho_p l_p / A_p + \rho_n l_n / A_n$$

(2) air side

$$M_a C_a L_2 \frac{\partial T_a}{\partial Y} = H_a (T_h - T_a) \quad (3)$$

$$M_a C_a L_2 \frac{\partial T_a}{\partial Y} = n\left[\alpha T_h I + \frac{0.5 I^2 \rho}{\nu} - k\nu (T_h - T_c)\right] \quad (4)$$

(3) for each TE module

$$COP = \frac{M_a C_a (T_{a_o} - T_{a_i})}{P} \quad (5)$$

Cooling mode: (1) water side

$$M_w C_w L_1 \frac{\partial T_w}{\partial X} = H_w (T_h - T_w) \quad (6)$$

$$M_w C_w L_1 \frac{\partial T_w}{\partial X} = n\left[\alpha T_h I + \frac{0.5 I^2 \rho}{\nu} - k\nu (T_h - T_c)\right] \quad (7)$$

(2) air side

$$-M_a C_a L_2 \frac{\partial T_a}{\partial Y} = H_a (T_a - T_c) \quad (8)$$

$$-M_a L_2 \frac{\partial h_a}{\partial Y} = n\left[\alpha T_c I - \frac{0.5 I^2 \rho}{\nu} - k\nu (T_h - T_c)\right] \quad (9)$$

$$-M_a L_2 \frac{\partial W_a}{\partial Y} = J_a (W_a - W_c) \quad (10)$$

(3) for each TE module

$$COP = \frac{M_a (h_{a_i} - h_{a_o})}{P} \quad (11)$$

The two sets of equations, 1 through 5 and 6 through 11, represent the heating and cooling operations. The problem involves heat generation in the process. This problem is solved numerically with some modifications of the computer code developed by Mathiprakasam [6].

STATE-OF-THE-ART TE MATERIALS

For the past twenty years, there has been no major breakthrough in TE material development. The figure of merit of available TE modules today is around $2.5 \times 10^{-3}/°K$. Horst and Williams [8] using existing TE material under careful control of some factors, such as sample preparation, material growth, and doping, demonstrated experimentally that a figure of merit of $3.1 \times 10^{-3}/°K$ could be achieved. While such performance is available, the current technology does not reflect it. The following state-of-the-art TE material properties for each couple are, therefore, assumed.

α (Seebeck coefficient) = 0.00044 V/°K
k (thermal conductivity) = 0.030 W/cm°K
ρ (electrical resistivity) = 2.10×10^{-3} ohm-cm

TE SYSTEM OPERATING CONDITIONS AND PARAMETERS

Table 1 presents the WSHP operating conditions [3]. The TE system operating conditions are assumed to be the same. Two parameters, P_1 and P_2, are used for TE system's parametric study. P_1 and P_2 represent the air side and groundwater number of transfer unit (NTU). In this study, the groundwater and air mass flow rates are kept constant. P_1 and P_2 actually represent H_a or H_w. Since heat transfer area is a design factor, P_1 and P_2 can then be considered as design parameters. For example, with different heat sink devices attached to the TE modules, the heat transfer area can vary over a wide range. With P_1 and P_2 as the parameters, a total of 108 runs was made for heating and cooling mode operation.

RESULTS AND DISCUSSION

Figure 2 shows the comparison of COPs of the WSHP [3] and the TE system in heating mode operation. The COPs of the WSHP are much higher than those of the TE system. The parameters P_1 and P_2 do not have much effect on the system COPs. At 13°C groundwater temperature, the heating COP for the TE system with $P_1 = 1.5$, and $P_2 = 1.5$, is only 1.72. For WSHP, the COP is 3.72. This indicates that groundwater temperature is not high enough to be a good heat source for TE systems. Table 2 presents the rest of the computed TE system performance.

Figure 3 presents the comparison of COPs of the two systems for cooling mode operation. The nonlinearity of the TE system's performance as a function of groundwater temperature is clearly shown. The effect of P_1 and P_2 is much stronger than in heating mode operation. For

Table 1. Operating conditions

1. Water inlet temperatures	21.1°C (70°F)	18.3°C (65°F)	15.6°C (60°F)	12.8°C (55°F)	10°C (50°F)	7.2°C (45°F)
2. Water flow rate	2044 kg/hr (4505 lb/hr)					
3. Air flow rate	2396 kg/hr (5280 lb/hr)					
4. Inlet air temperature (cooling mode)	25.6°C (78°F)					
5. Inlet air temperature (heating mode)	21.1°C (70°F)					
6. Inlet air relative humidity (cooling mode)	65%					

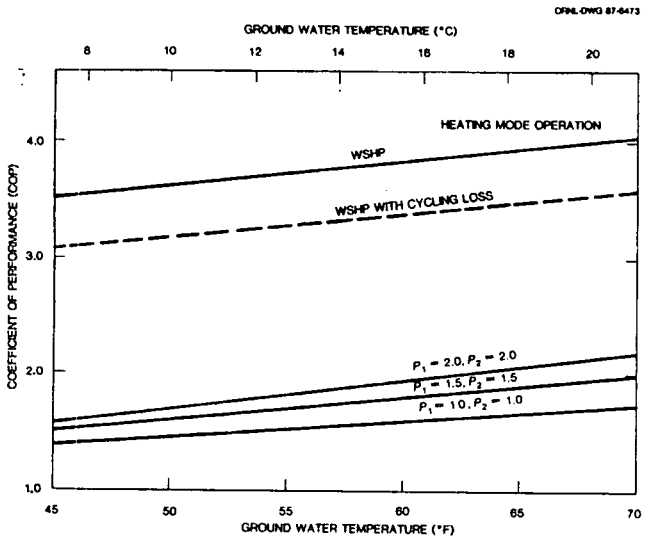

Fig. 2. Comparison of heating mode performance of WSHP and TE systems with P_1 and P_2 as parameters.

$P_1 = 1.5$ and $P_2 = 1.5$, the TE system outperforms the WSHP at groundwater temperature 18°C or lower, if the cyclic loss of the WSHP is included. For example, at 13°C water, with $P_1 = 1.5$ and $P_2 = 1.5$, the TE cooling system can achieve a COP of 6.50, versus 4.10 for WSHP. When the groundwater temperature becomes lower, the COPs of TE system increase exponentially because each TE module's performance is a nonlinear function of the TE module's hot and cold side temperature differential as shown in [7]. The groundwater temperature in the continental United States is ranging from 21°C in the south to around 7°C in the northern region [2]. Its temperature is low enough to be an excellent heat sink for TE system's cooling mode operation. Figure 3 also shows that the design of the TE heat exchanger becomes important. The amount of heat transfer area will greatly affect the TE system cooling performance. Table 3 lists the calculated TE system cooling performance.

Figure 4 shows the inlet and outlet water temperature differentials for WSHP and TE system. For heating mode, the TE system has a much smaller temperature differential than WSHP. The design factors, P_1 and P_2,

Table 2. Calculated TE System Heating Performance

Ti = 21.1 C, To = 37.8 C, RH = 65%

P1 = 1.0, P2 = 1.0

T_{w_i}	T_{w_o}	COP
21.1	19.1	1.73
18.3	16.5	1.65
15.6	13.8	1.57
12.8	11.2	1.50
10.0	8.6	1.44
7.2	5.9	1.38

P1 = 1.0, P2 = 1.5

T_{w_i}	T_{w_o}	COP
21.1	19.1	1.75
18.3	16.4	1.67
15.6	13.8	1.59
12.8	11.2	1.52
10.0	8.5	1.45
7.2	5.9	1.38

P1 = 1.0, P2 = 2.0

T_{w_i}	T_{w_o}	COP
21.1	19.1	1.76
18.3	16.4	1.67
15.6	13.8	1.59
12.8	11.2	1.52
10.0	8.5	1.45
7.2	5.9	1.39

P1 = 1.5, P2 = 1.0

T_{w_i}	T_{w_o}	COP
21.1	18.8	1.96
18.3	16.2	1.84
15.6	13.5	1.74
12.8	10.9	1.65
10.0	8.3	1.56
7.2	5.7	1.49

P1 = 1.5, P2 = 1.5

T_{w_i}	T_{w_o}	COP
21.1	18.7	1.99
18.3	16.1	1.87
15.6	13.5	1.76
12.8	10.9	1.67
10.0	8.3	1.58
7.2	5.6	1.50

P1 = 1.5, P2 = 2.0

T_{w_i}	T_{w_o}	COP
21.1	18.7	2.00
18.3	16.1	1.88
15.6	13.5	1.77
12.8	10.9	1.68
10.0	8.2	1.59
7.2	5.6	1.51

P1 = 2.0, P2 = 1.0

T_{w_i}	T_{w_o}	COP
21.1	18.6	2.09
18.3	16.0	1.96
15.6	13.4	1.84
12.8	10.8	1.74
10.0	8.1	1.64
7.2	5.5	1.55

P1 = 2.0, P2 = 1.5

T_{w_i}	T_{w_o}	COP
21.1	18.6	2.16
18.3	15.9	2.01
15.6	13.3	1.88
12.8	10.7	1.77
10.0	8.1	1.66
7.2	5.5	1.57

P1 = 2.0, P2 = 2.0

T_{w_i}	T_{w_o}	COP
21.1	18.5	2.18
18.3	15.9	2.03
15.6	13.3	1.90
12.8	10.7	1.78
10.0	8.1	1.67
7.2	5.5	1.58

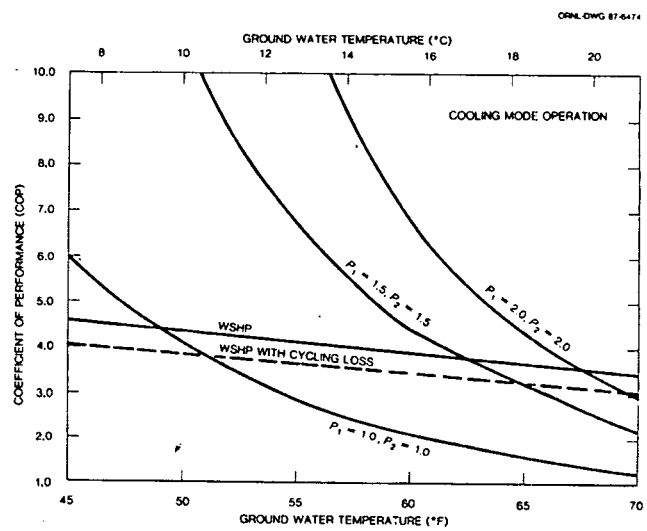

Fig. 3. Comparison of cooling mode performance of WSHP and TE systems with P_1 and P_2 as parameters.

have practically no effect on it. For cooling mode, for $P_1 = 1$ and $P_2 = 1$, when the groundwater temperature is 14°C or higher, the TE unit actually dissipates more heat than the WSHP, and yet has lower COPs. At higher groundwater temperature, the TE system's cooling performance decreases. Since the cooling capacity is assumed constant, more power is needed to achieve the same capacity output, thus dissipating more heat to the water. Again, the effect of the design factors P_1 and P_2 of the TE heat exchanger is apparent.

Table 3. Calculated TE System Cooling Performance

Ti = 25.6 C, To = 17.2 C, RH = 65% (Inlet)

P1 = 1.0, P2 = 1.0			P1 = 1.0, P2 = 1.5			P1 = 1.0, P2 = 2.0		
T_{w_i}	T_{w_o}	COP	T_{w_i}	T_{w_o}	COP	T_{w_i}	T_{w_o}	COP
21.1	29.1	1.20	21.1	28.3	1.50	21.1	28.1	1.64
18.3	25.4	1.60	18.3	24.9	1.98	18.3	24.7	2.18
15.6	21.9	2.14	15.6	21.5	2.68	15.6	21.3	2.97
12.8	18.6	2.92	12.8	18.3	3.73	12.8	18.1	4.19
10.0	15.4	4.10	10.0	15.1	5.41	10.0	15.0	6.20
7.2	12.3	6.00	7.2	12.1	8.32	7.2	12.0	9.84

P1 = 1.5, P2 = 1.0			P1 = 1.5, P2 = 1.5			P1 = 1.5, P2 = 2.0		
T_{w_i}	T_{w_o}	COP	T_{w_i}	T_{w_o}	COP	T_{w_i}	T_{w_o}	COP
21.1	27.3	1.79	21.1	26.9	2.18	21.1	26.7	2.37
18.3	23.9	2.48	18.3	23.6	3.04	18.3	23.5	3.30
15.6	20.6	3.53	15.6	20.4	4.40	15.6	20.3	4.81
12.8	17.5	5.26	12.8	17.3	6.74	12.8	17.3	7.42
10.0	14.4	8.32	10.0	14.3	11.29	10.0	14.3	12.71
7.2	11.4	14.80	7.2	11.3	22.27	7.2	11.3	26.12

P1 = 2.0, P2 = 1.0			P1 = 2.0, P2 = 1.5			P1 = 2.0, P2 = 2.0		
T_{w_i}	T_{w_o}	COP	T_{w_i}	T_{w_o}	COP	T_{w_i}	T_{w_o}	COP
21.1	26.4	2.24	21.1	26.1	2.70	21.1	26.0	2.93
18.3	23.2	3.17	18.3	22.9	3.92	18.3	22.8	4.33
15.6	20.0	4.69	15.6	19.8	6.02	15.6	19.8	6.82
12.8	16.9	7.34	12.8	16.8	10.14	12.8	16.8	12.02
10.0	14.0	12.78	10.0	13.9	19.95*	10.0	13.8	25.75*
7.2	11.0	26.76*	7.2	11.0	55.24*	7.2	10.9	91.17*

*Poor energy balance

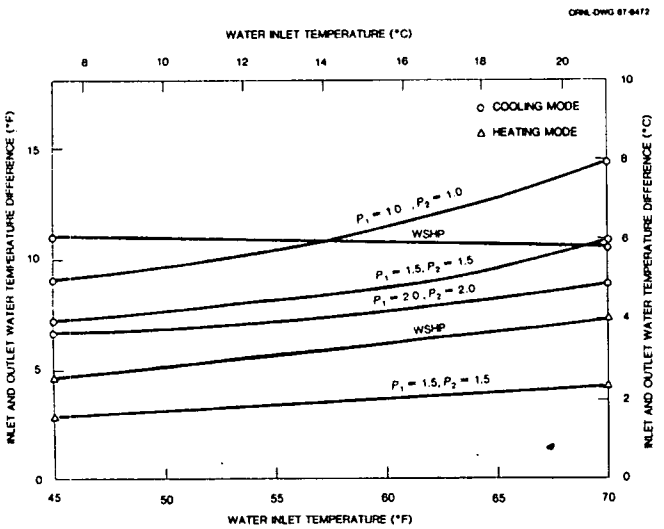

Fig. 4. Comparison of inlet and outlet water temperature differentials of WSHP and TE systems with P_1 and P_2 as parameters.

Figure 5 shows the COPs of the WSHP as a function of inlet water temperature with P_1 and P_2 varies from 0.5 to 0.97. The variation of P_2 is due to the change of water flow rate, from 1136 kg/h to 2953 kg/h. It could be that convective heat transfer coefficient did not increase as fast as the water mass flow rate, P_2 thus decreased when the flow rate was increased. However, it is clear that the performance of WSHP is not as sensitive to the P_2 parameters as the TE systems.

The above results indicate that TE system coupled with groundwater will not be able to compete with WSHP for heating purposes. For cooling operation, however, a properly designed TE system can easily outperform a WSHP, particularly in areas where the groundwater temperature is low. For buildings where some area might need cooling constantly, a groundwater-coupled TE cooling system may be a good alternative. With many known advantages of TE system, such as no CFC-refrigerant, no moving parts, ease of capacity control, no cycling loss, etc., a TE cooling system could be very attractive. At present, the cost of a TE cooling system is about three times that of a comparable vapor compression system [5]. If mass production of the TE modules can be achieved to replace the present day labor intensive method in production, the initial cost of TE cooling systems will be drastically reduced, which will make the TE cooling system more competitive with conventional systems.

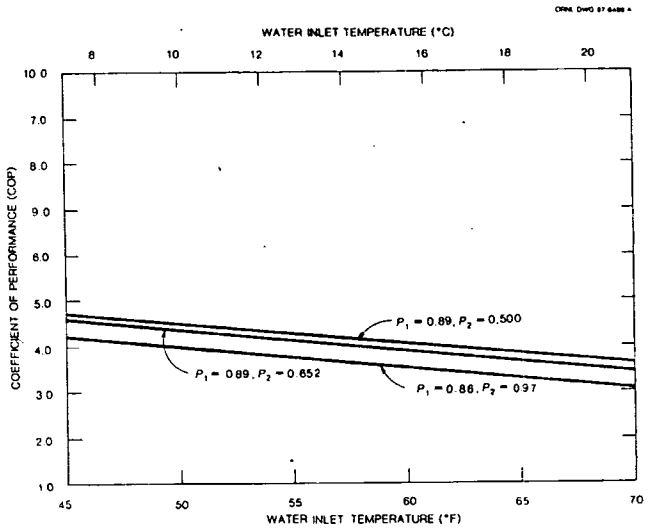

Fig. 5. Effect of P_2 on WSHP cooling mode performance.

CONCLUSION

The mathematical model of a groundwater-coupled thermoelectric cooling system, with state-of-the-art material, was solved numerically. The calculated TE system's performance was compared with experimental data from a water source heat pump under the same operating conditions. The comparison indicated that the TE system in heating mode operation cannot compete with WSHP. For cooling operation, a properly designed groundwater-coupled TE system can easily outperform the WSHP with COP of 6.4 or higher at 13°C water temperature. For building areas where year-round cooling is needed, TE systems can be a good alternative to conventional cooling systems.

NOMENCLATURE

A = TE element cross section area

C_a = specific heat of air

C_w = specific heat of water

H_a = product of air side heat transfer coefficient and area

H_w = product of water side heat transfer coefficient and area

h_a = enthalpy of air

I = electrical current flow

J_a = mass transfer rate

k = thermal conductivity of a TE couple

l = TE element length

L_1 = length in the direction of water

L_2 = length in the direction of air

M_a = air mass flow rate

M_w = water mass flow rate

n = total number of thermoelectric couples

P = power input to the TE system

P_1 = air side NTU, H_a/M_aC_a

P_2 = water side NTU, H_w/M_wC_w

RH = inlet air relative humidity

T_a = air temperature

T_c = thermoelectric module cold side temperature

T_h = thermoelectric module hot side temperature

T_w = water temperature

W_a = air moisture content

W_c = air moisture content at temperature T_c

X = space coordinate in the direction of water flow

Y = space coordinate in the direction of air flow

α = Seebeck effect of the thermoelectric couple

ν = area/length ratio of TE element

ρ = thermoelectric couple electric resistivity.

SUBSCRIPTS

i = inlet

o = outlet

n = TE module negative element

p = TE positive element

REFERENCE

1. "The Ozone Controversy and Its Relationship to Refrigeration and Air Conditioning," Air-Conditioning and Refrigeration Institute, June 1978.

2. "A Groundwater Heat Pump Anthology," National Water Well Association, 1979.

3. Mei, V. C., *Laboratory Test of a Residential Low-Temperature Water Source Heat Pump*, ORNL/CON-100, Oak Ridge National Laboratory, April 1984.

4. Mole, C. J., and Meenan, D. F., "Direct Transfer Thermoelectric Cooling," *ASHRAE Trans*, V. 70, pp. 130-138 (1964).

5. Buffet, J. P., and Stockholm, J. G., "Thermoelectric Air Conditioning with Water Heat Rejection," pp. 95-101 in *Proceedings of Fifth International Conference on Thermoelectric Energy Conversion, The University of Texas at Arlington, Arlington, Texas, March 14-16, 1984.*

6. Mathiprakasam, B., "Analytical Model for Predicting the Performance of Cross-Flow Thermoelectric Liquid Coolers," pp. 75-79 in *Proceedings of Fifth International Conference on Thermoelectric Energy Conversion, The University of Texas at Arlington, Arlington, Texas, March 14-16, 1984.*

7. ASHRAE 1981 Fundamental Handbook, pp. 1.27-1.29, ASHRAE, Atlanta, GA.

8. Horst, R. B., and Williams, L. R., "Preparation and Properties of High-Performance $(Bi,Sb)_2(Te,Se)_3$ Alloys," pp. 119-124 in *Proceedings of Fourth International Conference on Thermoelectric Energy Conversion, The University of Texas at Arlington, Arlington, Texas, March 10-12, 1982.*