PERFORMANCE OF A GROUND-COUPLED HEAT PUMP WITH MULTIPLE DISSIMILAR U-TUBE COILS IN SERIES

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ABSTRACT

A ground–coil heat pump (GCHP) system with six shallow–well U–tubes of unequal depth was installed at the test site in Knoxville, TN. The system was operated for one cooling and one heating season. During that time, the cooling seasonal performance factor (SPF) was 1.75 and the heating SPF was 2.85. An annual performance factor (APF) of 2.39 was achieved, compared to 2.10–2.17 for an air source heat pump (ASHP) in an identical house. Hourly integrated peak demands for the GCHP system were 3.3 kW for winter and 3.8 kW for summer. Peaks for the ASHP were 9.1 kW and 3.7 kW, respectively.

The GCHP system was also subjected to two weeks of continuous tests and five days of cyclic tests. An equivalent radius approach was adopted to analyze the experimental results. It was found that the equivalent radii for the wells were not equal. However, the average equivalent radius for the six wells was found to be about 1.3 times the coil radius, which is very close to the multiplier, the square root of 2, used by others.

INTRODUCTION

Ground–coupled heat pumps (GCHP) have long been known for their potential for energy conservation (Budenholzer 1947; Kelmer 1947; Coogan 1949; Guernsey et al. 1949; Vestal 1949; Ingersoll et al. 1950; Smith 1950; Vestal and Fluker 1957; Kalman 1980; Metz 1983). Because of different geological conditions and limitations at different sites, different ground–coil heat exchanger geometries may be used. Generally, horizontal coils are used where large lots are available. Vertical tube–in–tube or U–tube type ground heat exchangers, configured as one deep well or several shallow wells, are used for smaller lots with high water tables.

Horizontal ground coils have been the subject of many previous studies, the results of which were based on rule–of–thumb type analyses. However, several detailed mathematical modeling studies, including soil moisture phase change (Mei and Emerson 1985) and/or the influence of ground seasonal temperature variation (Mei 1985), have been done so that future design work could be calculated more realistically. Vertical tube–in–tube type ground heat exchangers were first introduced by Ambrose (1946). Several field tests have

been conducted to analyze the exchanger experimentally (Bose and Parker 1983; Bose et al. 1980, 1982). An analytical study of such heat exchangers, which uses a detailed mathematical model to describe their performance, has been completed (Mei and Fischer 1983, 1984). The model was validated by the laboratory experimental results and was incorporated into a computer code.

Field experiments have been done for U–tube type ground coils (Bose and Parker 1983; Bose et al. 1982). However, no detailed analysis of the test results has yet been performed due to the difficulty of modeling two tubes with different fluid temperatures that are very close to each other.

For this study, a U–tube shallow well GCHP system was installed at the test site (Figure 1). The system was tested for seasonal performance over one cooling and one heating season. A five–day continuous test and a three–day cyclic operation test were also performed. An empirical approach (to find the equivalent diameter of the U–tube) was adopted in analyzing the ground heat exchanger performance.

TEST APPARATUS AND PROCEDURE

Between October 1982 and June 1983, the GCHP, illustrated schematically in Figure 2, was installed in House 2. The mechanical system consisted of a water–to–water heat pump with a water–to–air fan coil. A water–to–water system was used for ease in measuring cooling load. As a result, the system had two circulating pumps (fan coil loop and ground loop) rather than one, as is used in more typical installations.

The ground coil was composed of six vertical U–tube heat exchangers. Six 6–in (152–mm) wells were drilled at depths of 53, 60, 70, 80, 90, and 100 ft (16, 18, 21, 24, 27 and 30 m). The location of the wells is shown in Figure 3. Each well was iron cased down to bedrock. The distance between two adjacent wells was 12 ft (3.7 m). Core samples taken to a depth of 300 ft (91 m) revealed that the geological formation at the test site was clay on top of limestone. However, the underground limestone formed a 60° slope, which meant that the soil thickness varied from 30 to 70 ft (9 to 21 m) throughout the well field. As a result, both the 100–ft (30–m) well and the 53–ft (16–m) well had about the same length of coil below bedrock. The water table was found at a depth of 30 ft (9 m). Wells 3, 4, and 5 are on the edge of a slope of about 30° that drops 6 ft (1.8 m) down.

Schedule 40 polyethylene pipe [1.5 in (38 mm) nominal diameter] was used for the ground coils. The physical properties of the soil (Fischer and Newman 1981), limestone, and pipe (Lange 1979) are listed in Table 1.

A piece of expanded rubber insulation was put between all the U–tubes except at Well 5 (see Figure 2). It was hoped that the insulation would reduce energy short circuiting. All wells were backfilled with cement except Well 5, which was backfilled with sand. Calibrated thermocouples were used to measure the fluid inlet and exit temperatures of each well.

The ground coil was connected so that each well could be tested separately. All six wells were connected in series for the field experiments.

The water–to–water heat pump (nominal 2.5–ton cooling, 3.5–ton heating) was connected to the ground coil using a water–methanol (20% by weight) mixture as the fluid. The two pumps each circulated about 9 gpm (2.04 m^3/h) through both the ground coil and the fan coil when the heat pump was operating. Each pump drew about 140 W when operating.

A minicomputer–based data acquisition system (Baxter 1981; McGraw and Miller 1983) normally recorded on tape all temperatures, power consumption, and heating/cooling energy delivery readings on an

hourly basis. However, for continuous and 30–minute on/off cycle tests, the temperatures were printed out every 10 minutes.

The tests were divided into two parts. The first part involved testing the overall performance of the heat pump system for one year, during which the heat pump was controlled by the house thermostat. The second part consisted of ground–coil continuous and cyclic testing to estimate the performance of the coil. After the system test was completed, the wells were allowed to rest for one month so that the ground could return to its natural temperature state. Then the ground–coil continuous operation test was performed for almost five days, followed by a week–long rest period and a five–day test of the 30–minute on/off cyclic operation.

FIELD EXPERIMENTAL RESULTS

The system was operated successfully from June 1983 through April 1984 with less than ten days downtime. Table 2 provides a detailed summary of the system performance for the 1983 cooling season and the 1983–84 heating season. The table indicates that the system as installed achieved a cooling SPF of 1.75 and a heating SPF of 2.85. For both the heating and cooling seasons, an APF of 2.39 is obtained. A high–efficiency ASHP located in House 3 by comparison had cooling SPFs of 2.30 and 2.41 and heating SPFs of 1.91 to 1.98 (Baxter and Moyers 1985). The ASHP's annual performance factors ranged from 2.10 to 2.17.

The cooling performance of the GCHP system was relatively poor compared to its heating performance. This was due primarily to the fact that the heat pump unit was not designed for air conditioning operation. An identical unit was tested with the condenser water entering temperature constant at 72 F (22°C) to establish the maximum cooling SPF given an infinitely long ground coil. An SPF of 2.00 was obtained from this test (Johnson et al. 1985). The measured system SPF was 1.75 indicating that, for this system, little potential exists for improving cooling performance by lengthening the ground coil.

Because of the elimination of the additional liquid–to–air heat exchanger and circulation pump, both cooling and heating SPFs would have been higher had a water–to–air unit been used instead of a water–to–water type unit. Excluding the energy use of the fan coil loop circulating pump results in an increase of 4.2% in the APF. A similar increase would be attained by eliminating the compressor crankcase heater.

Table 3 gives a comparison of the measured monthly heating and cooling energy deliveries for Houses 2 and 3 (House 3 is a mirror image twin of House 2) from June 1983 through April 1984. The monthly energy deliveries for the two houses agree reasonably well except for the period from January to March 1984. The much higher heating deliveries for House 2 over that period were traced to an open access way to the attic. This plus the fact that the GCHP system did not become operational until June 27, 1983 biased the APFs reported in Table 2, making the energy totals for heating higher than they would normally be for the Knoxville, TN, area. Because monthly energy deliveries for the two houses have been shown to be within ±10% of one another normally (Baxter and McGraw 1984), loads from House 3 were used to estimate an APF for the GCHP system for a more typical balance between heating and cooling seasons.

Table 4 illustrates the seasonal and annual performance of the House 2 GCHP system using energy deliveries from House 3 for May 31 through June 27, 1983, and for January through March 1984. Energy consumption for the GCHP system during the winter period was based on the measured SPFs from Table 2, and consumption for the May through June 1983 period was based on the measured cooling SPF for the first two weeks of the June 27 through July 24, 1983, period. The APF obtained using the given data was 2.32 compared with the 2.39 measured for the actual case.

Figures 4 and 5 show total daily heating/cooling energy delivery and energy consumption, respectively, for the GCHP system and the ASHP. As indicated in Table 3, the GCHP system delivered significantly more heating energy than did the ASHP, but its total energy consumption was somewhat lower, as shown in Figure 5.

The hourly integrated peak power draw of the GCHP in House 2 compared to that of the ASHP in House 3 is presented in Figure 6 and Table 5. Maximum heating season peak for the GCHP was only 3.3 kW compared to 9.1 kW for the ASHP. This was because the GCHP did not require use of electric resistance backup heating. Cooling season peaks were nearly the same for both systems.

FORMING A CORRELATION FOR FUTURE DESIGN

An effort was made to analyze the U–tube systems mathematically. When two tubes with finite diameters were within a very short distance of each other, the ordinary method of superimposing two solutions wouldn't work. To simplify this problem researchers assumed that a U–tube was equivalent to a single tube with a somewhat larger diameter. This approach, suggested by Claesson and Dunard (1984), assumes that

$$R_{eq} = \sqrt{2} \times R \ , \tag{1}$$

where R_{eq} is the equivalent radius and R is the outside radius of the tube. The same approach was used by Fischer and Stickford (1983), except they assumed that

$$R_{eq} = 1.84R \ . \tag{2}$$

A number of questions were asked with respect to the operating conditions related to the coil size. For instance, (1) With the equivalent diameter larger than the actual tube size, should one assume constant flow rate or constant velocity; (2) How would the convective heat transfer coefficient, which would be highly dependent on Reynolds number (Re), be calculated; (3) How should the backfill material, which might have different thermal properties than those of pipe or ground, be dealt with; and (4) How should one deal with the step change in thermal conductivity at the soil/rock interface.

In answering these questions, the flow rate was assumed to be constant, but the velocity was allowed to change. The convective heat transfer coefficient between the fluid and tube wall was calculated by using Re based on the actual tube radius.

The mathematical model for a single pipe buried in an infinite medium is described in detail by Mei and Emerson (1985). With some minor changes, the mathematical model could handle the backfill material as well as the change of ground thermal conductivity along the coil length. By assuming a correlation similar to Equations 1 and 2,

$$R_{eq} = C \cdot R \ , \tag{3}$$

we could easily determine the value of the constant C by comparing calculated fluid exit temperatures with experimental fluid exit temperatures at different values of C.

We determined statistically the C value for cooling mode operation. Using a series of computer simulations for a single pipe (Mei and Emerson 1985) with different C–values, we calculated the fluid exit temperatures and then compared them with measured ones. We found that each well had a different C–value and that the differences from well to well were in some cases quite large.

Table 6 shows the C–value for each well. It indicates that the first well performed much better than the others, as expected. However, Wells 3 and 4 did not perform as well as the others. Wells 3, 4, and 5 are located 5 ft (1.5 m) from a slope. Part of the coils of these wells could possibly have a much higher farfield temperature in one direction for about the first 10 ft (3.1 m) of depth. The house could also have affected

the farfield temperature for Wells 1, 2, and 3. Wells 2, 5, and 6 had fairly uniform performance even though Well 5 was backfilled with sand instead of cement.

Figure 7 is a comparison of calculated and measured steady–state fluid exit temperatures for Well 1 with $C = 1.622$. With the right choice of C, the model can match the test results very well. With the known C–values, a computer simulation of cyclic operation was performed. Figure 8 shows a comparison of calculated and measured fluid exit temperature for Well 1 on a 30–minute on/off cycling schedule. Note that the match is mostly good except at certain peaks. Because both inlet and exit fluid temperature measurements took place above the ground, the readings were highly influenced by ambient conditions once the fluid flow stopped. However, the figure does indicate the same trends for both sets of temperatures.

Figures 7 and 8 show that the equivalent diameter method for U–tube operation works well when the heat pump is in continuous operation. When the heat pump is in cyclic operation, the model is inadequate.

The average C–value in this study ($C_{ave} = 1.279$) is smaller than previously suggested values, $C = \sqrt{2}$ (Claesson and Dunard 1983) and $C = 1.84$ (Fischer and Stickford 1983). As a result, designs based on this study will be conservative.

CONCLUSION

A GCHP with a water–to–water heat pump and six shallow–well U–tube vertical ground coils with a total well depth of 454 ft (138 m) was installed in a house at the test site. All wells except one were backfilled with cement to have better thermal conductivity in case the ground is dry. The field test results from the 1983 cooling season indicate that the system achieved a cooling SPF of 1.75 with the compressor located indoors, which is somewhat lower than that of a high–efficiency, split system ASHP (compressor outdoors) operating in an identical house at the site. Tests on an identical water–to–water heat pump show that the maximum cooling SPF attainable with an infinite ground coil was 2.00. However, this GCHP achieved a heating SPF of 2.85 for the 1983–84 heating season. Combining heating and cooling season test results, an APF of 2.39 was achieved. Excluding the fan coil loop pump energy consumption yields an APF of 2.49, approximating operation with a water–to–air heat pump. Judging from these figures, the shallow–well GCHP system works much better for winter operation, as expected. The system outperformed an ASHP system in an identical house on an annual basis.

Because of the difficulties involved in setting up a detailed mathematical model for U–tube operation, an equivalent–diameter approach was adopted. Along with experimental data, a computer code was applied for a single coil buried in an infinite medium (Mei and Emerson 1985). It was found that each well had a different equivalent diameter, with the first well having the best performance. This indicates that the $R_{eq} = CR$ approach does not interpret the heat transfer and coil fluid flow phenomena correctly. An average C value of 1.279 was obtained after almost five days of continuous operation. Wells 3, 4 and 5 were expected to have C values between 1.283 and 1.256 (between Wells 2 and 6). However, lower C values for these three wells were obtained. One possible reason for this discrepancy is that these wells are along the edge of a 30° slope with a vertical drop of about 6 ft (1.8 m). The performance of these three coils near the surface for those three wells was probably affected by the ambient air conditions. If we assume that the C value for Wells 3, 4 and 5 equals 1.256 (the C value of Well 6), then the average C value would be 1.35, which is much closer to the Claesson and Dunard (1983) value of $C = \sqrt{2}$ but is significantly different from values reported by Fischer and Stickford (1983).

Well 5 was backfilled with sand and had no insulation between U–tube legs. The results indicate that its performance was better than that of the wells backfilled with cement. The insulation probably did more harm than good, because while it prevented some energy short circuiting, it also insulated part of the coil so that total heat transfer area was reduced. However, the ground was not very dry during the test. If the

ground was exceptionally dry, the sand could act as a layer of insulation at least for the first 20 ft (6.1 m) of coil depth.

The multiple shallow–well U–tube GCHP performed very well in winter and reasonably well in summer. Its annual experimental performance at APF = 2.39 was better than that of a high–efficiency ASHP operating at the same site in an identical house (APF range: 2.10 to 2.17).

Hourly integrated peak demand for the GCHP was 3.3 kW for heating and 3.8 kW for cooling. Comparable values for the ASHP were 9.1 kW and 3.7 kW respectively.

There is no apparent difference in performance between wells backfilled with sand or cement at least in the relatively wet southeastern United States.

The correlation, $R_{eq} = CR$, is not a very good approach because of large scatter in values of C for different wells in the same area. The average value of C is close to but smaller than the value reported by Claesson and Dunard (1983), and it is significantly different from values reported by Fischer and Stickford (1983).

We realize that this empirical approach does not treat the heat transfer phenomena in detail. It appears unlikely that the $R_{eq} = C \cdot R$ approach can be used with a high degree of confidence. It is hoped that a more thorough mathematical model can be formed to describe the operation of the U–tube ground coil.

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TABLE 1

Thermal Property Measurement on Rock and High-Density Polyethylene Pipe Specimens

Specimen	Density, lb/ft^3 (cm^3)	Thermal conductivity, Btu/h·ft·F (W/m·°C)	Thermal diffusivity, ft^2/h (m^2/h)	Specific heat, Btu/lb·F (J/kg·°C)
Limestone[a]	167.3 (2.68)	1.733 (0.3)	4.17×10^{-2} (3.87×10^{-3})	0.255 (1.065×10^3)
High-density polyethylene tube[a]	58.6 (0.94)	0.266 (0.046)	0.87×10^{-2} (0.81×10^{-3})	0.52[b] (2.171×10^3)
Soil[c] (clay, 20% moisture content)	71.6 (1.15)	0.7 (0.121) (dry)	2.4×10^{-2} (2.22×10^{-3})	0.407 (1.700×10^3)
Concrete[d] (dry)	140 (2.24)	0.75 (0.433)	2.44×10^{-2} (2.27×10^{-3})	0.22 (0.919×10^3)

[a]Thermal properties of rock and high-density polyethylene were tested by the Physical Property Group, Material Science Section, Metals and Ceramics Division, Oak Ridge National Laboratory, Oak Ridge, TN.
[b]From Lange (1979).
[c]From Fischer and Newman (1981).
[d]From ASHRAE (1985).

TABLE 2

Test House 2 GCHP Performance

Period	Space conditioning energy delivery, MBtu (kWh)	Energy consumption, kWh			Seasonal performance factor[a]		
		Note *b*	Note *c*	Note *d*	Note *b*	Note *c*	Note *d*
June 27-July 29	7.16 (2,098.2)	1174.0	1126.6	1101.7	1.78	1.86	1.90
July 29-Aug. 29	7.49 (2,195.6)	1284.0	1231.1	1206.7	1.71	1.78	1.82
Aug. 29-Sept. 30	4.21 (1,232.5)	701.0	672.2	635.8	1.76	1.83	1.94
Sept. 30-Oct. 10	0.41 (121.5)	76.0	73.4	60.0	1.60	1.66	2.02
Summer totals	19.27 (5,647.8)	3235.0	3103.3	3004.2	1.75	1.82	1.88
Oct. 31-Nov. 28	4.11 (1,204.4)	389.6	376.3	350.9	3.09	3.20	3.43
Nov. 28-Jan. 3[e]	8.88 (2,602.3)	654.5	821.3	801.6	3.05	3.17	3.25
Jan. 3-Jan. 30	11.85 (3,473.8)	1242.7	1141.7	1181.5	2.80	2.91	2.94
Jan. 30-March 2	10.97 (3,214.6)	1157.0	1109.0	1092.4	2.78	2.90	2.94
March 2-April 2	7.28 (2,134.2)	771.2	740.4	719.4	2.77	2.88	2.97
April 2-April 26	1.31 (383.4)	150.1	145.9	122.9	2.55	2.63	3.12
Winter totals	44.40 (13,012.8)	4565.1	4384.6	4268.7	2.85	2.97	3.05
Annual totals	63.67 (18,660.6)	7800.1	7487.9	7272.9	2.39	2.49	2.57

[a]*Calculated for heat pump package located inside conditioned space.*
[b]As measured.
[c]Fan coil pump energy was excluded.
[d]Fan coil pump and crankcase heater energy was excluded.
[e]Excluding Dec. 24-30.

TABLE 3

Heating/Cooling Energy Delivery for Test Houses 2 and 3 for June 1983 through April 1984

Period	Energy delivery, MBtu (kWh)	
	House 2 GCHP	House 3 ASHP
May 31-June 27, 1983		2.82 (825.7)
June 27-July 29	7.16 (2098.2)	7.21 (2114.0)
July 29-Aug. 29	7.49 (2195.6)	7.98 (2338.5)
Aug. 29-Sept. 30	4.21 (1232.5)	3.95 (1158.0)
Sept. 30-Oct. 10	0.41 (121.5)	0.33 (96.0)
Oct. 31-Nov. 28	4.11 (1204.4)	3.69 (1081.6)
Nov. 28-Jan. 3, 1984[a]	8.88 (2602.3)	8.30 (2432.1)
Jan. 3-Jan. 30	11.85 (3473.8)	8.17 (2394.6)
Jan. 30-March 2	10.97 (3214.6)	7.76 (2273.4)
March 2-April 2	7.28 (2134.3)	5.65 (1656.6)
April 2-April 26	1.31 (383.4)	1.50 (439.6)

[a]Dec. 24-30 excluded.

TABLE 4

Test House 2 GCHP Estimated Performance with Modified Load Profile

Period	Heating/cooling energy delivery, MBtu (kWh)	Energy consumption, kWh	Seasonal performance factor[a]
May 5-June 27, 1983	2.82 (825.7)	430.0	1.92
June 27-July 29	7.16 (2,098.2)	1174.0	1.78
July 29-Aug. 29	7.49 (2,195.6)	1284.0	1.71
Aug. 29-Sept. 30	4.21 (1,232.5)	701.0	1.76
Sept. 30-Oct. 10	0.41 (121.5)	76.0	1.60
Summer totals	22.09 (6,473.5)	3665.0	1.77
Oct. 31-Nov. 28	4.11 (1,204.4)	389.6	3.09
Nov. 28-Jan. 3, 1984[b]	8.88 (2,602.3)	854.5	3.05
Jan. 3-Jan. 30	8.17 (2,394.6)	855.2	2.80
Jan. 30-March 2	7.76 (2,273.4)	817.8	2.78
March 2-April 2	5.65 (1,656.6)	598.0	2.77
April 2-April 26	1.31 (383.4)	150.1	2.55
Winter totals	35.88 (10,514.7)	3665.2	2.87
Annual totals	57.97 (16,988.2)	7330.2	2.32

[a]Heat pump inside conditioned space.
[b]Dec. 24-30 excluded.

TABLE 5

Winter and Summer Peak Power Draws for the GCHP and the ASHP

	Hourly integrated peak, kW	
Season	ASHP	GCHP[a]
Heating	9.1	3.3
Cooling	3.7	3.8

[a]Includes two circulation pumps at about 140 W each.

TABLE 6

Calculated C-Value for Each Well

	C value	Depth below water table, ft	Depth below rock, ft
Well 1 100 ft (30.5 m)	1.622	70	30
Well 2 70 ft (21.3 m)	1.283	50	20
Well 3 53 ft (16.2 m)	1.000	23	23
Well 4 80 ft (24.4 m)	1.101	50	30
Well 5 60 ft (18.3 m)	1.229	30	0
Well 6 90 ft (27.4 m)	1.256	60	20
C_{ave}[a]	1.279		

[a]$C_{ave} = \Sigma(\text{C value} \times \text{feet})/\Sigma\text{feet}$.

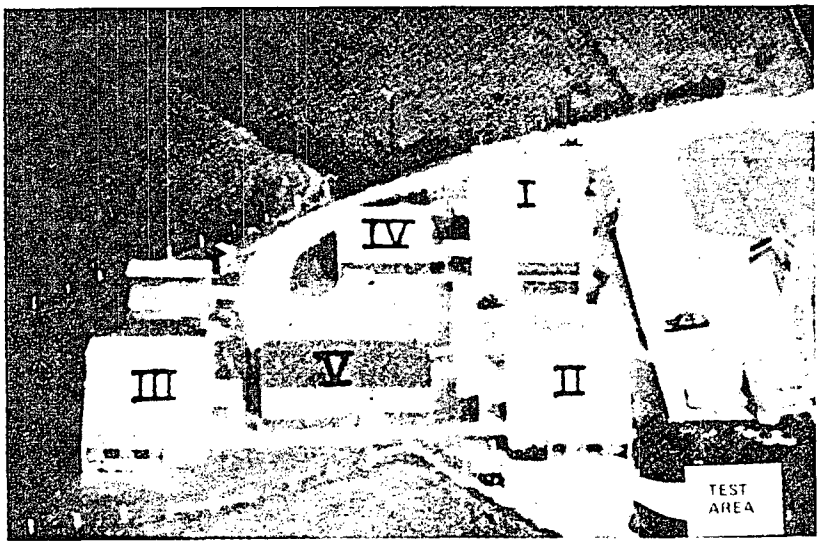

Figure 1. Test site, Knoxville, TN

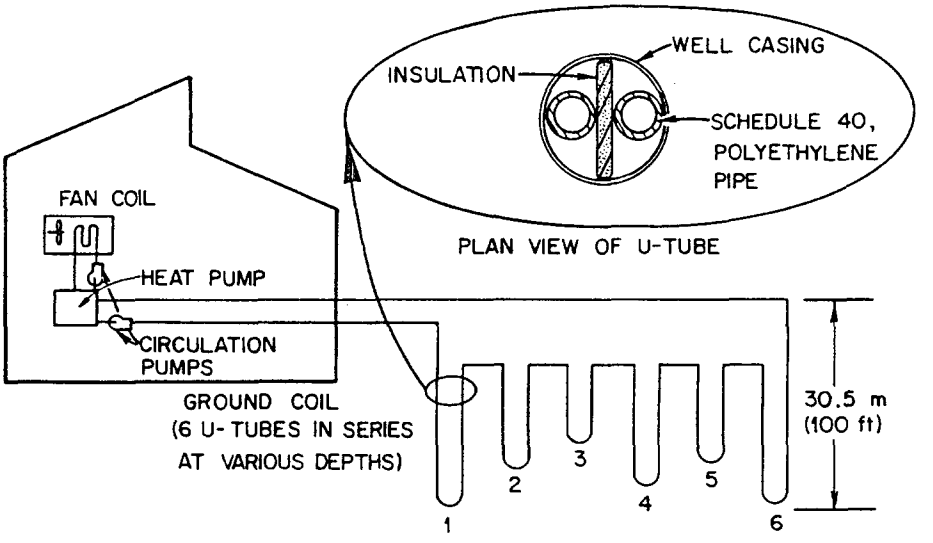

Figure 2. Schematic of U tube ground coil system

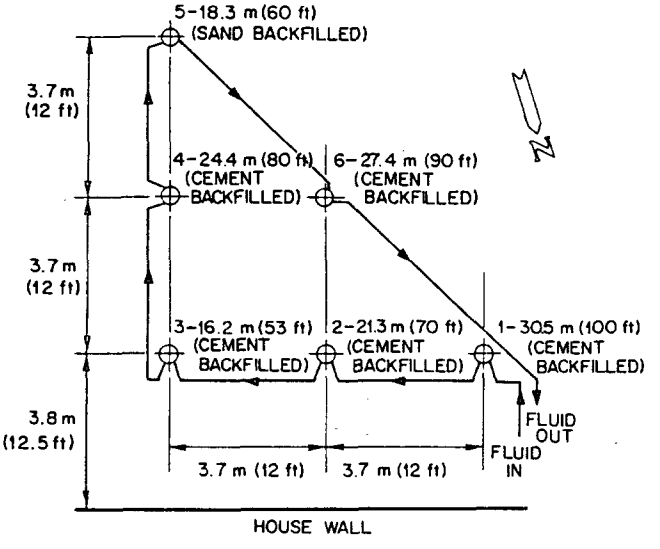

Figure 3. Schematic of coil route and well locations

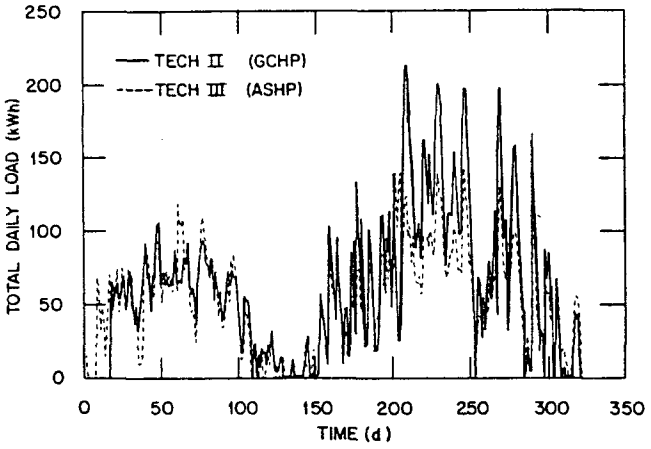

Figure 4. Daily heating/cooling load for Houses 2 and 3, June 1983-April 1984

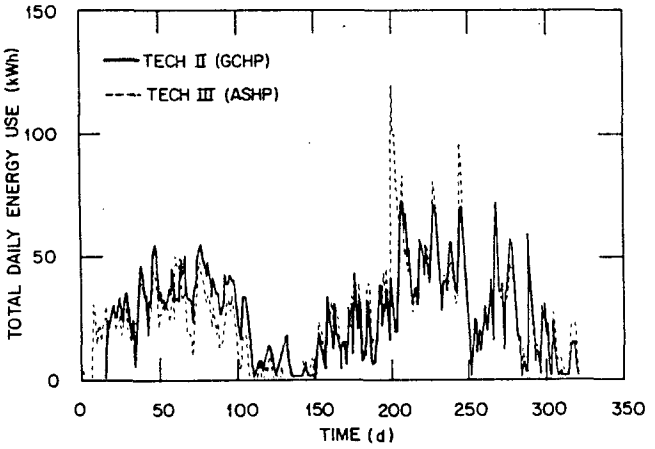

Figure 5. Daily energy use for heating and cooling for Houses 2 and 3, June 1983-April 1984

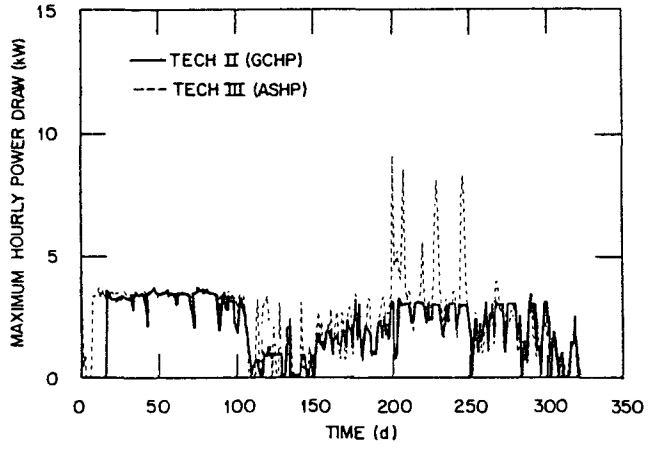

Figure 6. Peak hourly power draw for GCHP (House 2) and air-source heat pump (House 3), June 1983-April 1984

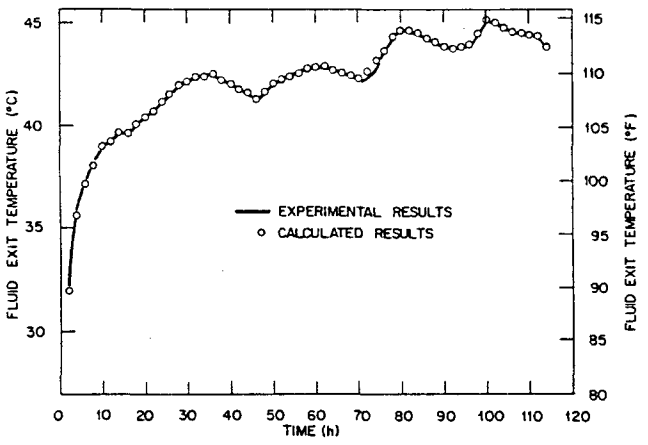

Figure 7. Fluid temperature simulation of continuous operation of 30.5-m (100-ft) well

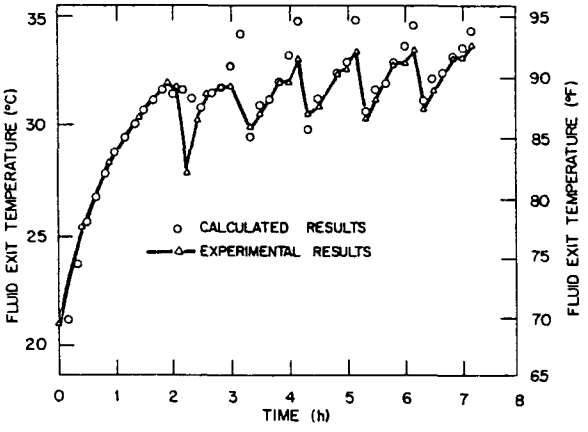

Figure 8. Fluid temperature simulation of cyclic operation of 30.5-m (100-ft) well

Discussion

W.E. MURPHY, Univ. of Kentucky, Lexington: Could you tell me why you chose to use these wells in a series arrangement, since that would make your analysis less useful and your modeling effort more difficult?

V.C. MEI and V.D. BAXTER: The design of the piping enabled us to run the six wells in series or in two parallel groups with each group composed of three wells of 230 ft (70 m) total well depth each. It was planned to run both cases. However, after analyzing the test results of wells in series, we found that the summer COP of 1.75 was very close to the limit of the heat pump peak performance (COP Of 1.9) measured by running the city water through the refrigerant-water heat exchanger. To run the wells in parallel is not going to improve the cooling COP much further.

Our test data did not make our modeling effort more difficult; except for some wells, the fluid temperature differentials did become smaller. Besides, we also worry that when the fluid flow rate is lowered, the flow could be in the laminar region for winter operation. The thermal film resistance could then be considerably higher than those in the turbulent, and transitional regions, which would reduce the coil-ground heat exchange rate.

W.S. FLEMING, W.S. Fleming & Assoc. Inc., Syracuse, NY: Did Oak Ridge consider utilizing a water/air heat pump system for the test, rather than the water/water heat pump?

MEI and BAXTER: Yes. We would if further tests are warranted.