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Final Report: 1984–85 Annual Performance Testing and Analysis of Two Horizontal Coil Ground-Coupled Heat Pump Systems

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Report Prepared by

THE UNIVERSITY OF TENNESSEE
Energy, Environment, and Resources Center
327 South Stadium Hall
Knoxville, Tennessee 37996-0710

under
Subcontract 7685 PA X92

for

OAK RIDGE NATIONAL LABORATORY
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MARTIN MARIETTA ENERGY SYSTEMS, INC.
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Date of Issue—April 1987

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ABSTRACT

Performance of two different ground coupled heat pump (GCHP) installations at the TECH site at the University of Tennessee is reported for the 1984-85 heating season and the 1985 cooling season. One system, located in TECH House I (a single family residence with design heating and cooling loads of 30,000 and 22,000 Btu/hr respectively), is a hydronic heating only unit (field modified for cooling operation) which has been operating continuously since 1982. The second system is a water-air unit located in TECH House V (a single family residence with design heating and cooling loads of 24,000 and 16,000 Btu/hr respectively). It was designed and installed by a private sector participant in November 1984 and has been evaluated over one calender year.

Seasonal performance factors of the TECH House I system were found to be 2.23 and 1.51 for heating and cooling, respectively. The corresponding seasonal performance values of the new GCHP system in TECH House V were found to be 2.84 and 2.59. Previous seasonal performance factors of the TECH House I system were found to be 2.6 (heating) and 1.1 - 1.3 (cooling). The values reported for the 1984-85 winter are different primarily due to the use of sand as a backfill material and fewer cooling degree days during the 1985 summer. System V performance is superior to system I due to an increased length of ground coil and a heat pump unit having better cooling performance characteristics. A comparison of the seasonal peak load required by the GCHP system in TECH House I to that required by a standard air-air heat pump in TECH House III (a house with design heating and cooling loads comparable to TECH House I) showed a reduction of 8 kW. Thus, the GCHP system should show a significant savings should the utilities institute residential peak demand pricing.

Executive Summary

As part of the DOE/ORNL Building Equipment Research Program, the University of Tennessee, in cooperation with GEOsystems, Inc., and Charles Machine Works, Inc., installed and tested two ground-coupled heat pump (GCHP) systems in Knoxville, Tennessee.

In 1982, the University of Tennessee installed a horizontal coil GCHP system in House I (30,000 Btu/h design heating load) at the Tennessee Energy Conservation in Housing (TECH) site; the University has conducted tests on the system for 3 years. The GCHP system consisted of a 675-ft ground coil with a nominal 3-ton (heating) water-to-water heat pump. Results of these tests (given in previous reports in this series) showed heating seasonal performance factors (SPF) of 2.6 and 2.59 and a cooling SPF of 1.1 for the first cooling season. Prior to the second cooling season, the ground coil was reinstalled with sand backfill to eliminate voids around the pipe. With this modification, the cooling SPF was 1.3, a result due partly to the sand and partly to a milder cooling season. Reasons for the poor cooling SPFs were (1) soil drying, (2) coil undersizing, and (3) poor cooling performance of the heat pump.

The objective of this study was to evaluate the use of water injection during the summer to maintain saturated soil conditions and to compare performance of the existing GCHP system with a system designed according to current industry standards for a high cooling load climate.

GEOsystems, Inc., and Charles Machine Works, Inc., designed and installed in TECH House 5 (24,000 Btu/h design heating load) a second GCHP system. This system consisted of a 1135-ft coil in a two-coil-per-trench arrangement with a nominal 2.5 ton (heating) water-to-air heat pump. Performance of this system and that of the TECH House I system were compared over the 1984-85 test year. The effect of water injection on the House I system performance was evaluated during the latter part of the 1985 cooling season.

Heating and cooling SPFs for the House I GCHP were 2.23 and 1.51, respectively, for the 1984-85 test year. Corresponding values for the House 5 GCHP were 2.84 and 2.59. Neither system required any backup electric heating. Poor heating performance for the House I system relative to previously demonstrated SPFs of 2.6 apparently was caused by failure of the sand backfill to resaturate following the 1984 cooling season. Performance of the industry-designed GCHP in House 5 exceeded that of the House 1 system for two primary reasons: (1) the ground coil was less heavily loaded, and (2) a heat pump unit with much better cooling performance was used.

Use of water injection in the latter half of the 1985 cooling season resulted in only a 4 percent improvement in House 1 system cooling performance (SPF of 1.54 with water vs. 1.48 without). Evaluation of hourly data showed rapid soil drying adjacent to the pipe and a corresponding reduction of the soil thermal conductivity at that location. Soil thermal conductivity actually dropped to the value recorded prior to the water injection within a few days. A comparison was made of winter peak demand between the House I GCHP and an

air-source heat pump (ASHP) in a similar house. Peak demands were 3.5 kW and 11.5 kW for the GCHP and ASHP, respectively.

A simple payback analysis was conducted to compare the economics of the House 5 GCHP with a standard ASHP previously tested at the TECH Site (heating and cooling SPFs of 1.56 and 1.64, respectively). At current Knoxville electricity prices, the simple payback period of the GCHP system was about 15 years. When a peak load pricing scenario currently used for small commercial customers was assumed, payback was about 7 years. Other factors could reduce the payback period even more. These include: (1) operation in a colder climate where the GCHP should enjoy a greater performance advantage and (2) operation in a region with higher electricity costs (The Knoxville, Tennessee area is well below the national average).

General conclusions obtained from the study include:

1) A well-designed GCHP offers a significant performance advantage over a standard air-air system.
2) Under conditions tested, sand appears to be a poor choice for backfill.
3) Moisture injection improves performance for only a short period of time.
4) Under present electrical pricing policies of the Knoxville Utilities Board, the GCHP is not economically competitive with the lower performance air-air system.

Chapter 1. Introduction

The University of Tennessee Energy, Environment, and Resources Center (UT-EERC) has been contracted by the Oak Ridge National Laboratory (ORNL) for the procurement, installation, maintenance and instrumentation of a horizontal ground coil heat pump (GCHP) system. This system is located in TECH House I at the Tennessee Energy Conservation in Housing (TECH) Facility and has been operated continuously since September, 1982.

The seasonal performance factor (SPF) during the summer of 1983 was measured and reported to be only 1.1.[1] Although GCHP cooling seasonal performance data appear to be scant[2], several members of this industry expressed concern over the low SPF. Specifically, they believed that the cooling SPF could have been much higher "near or exceeding 3.0" had the system been designed properly.[3]

Consequently, members of the private sector (GEOsystems, Inc., and the Charles Machine Works, Inc.) were invited to design and install a GCHP system in another building (TECH House V) on the TECH site. GEOsystems sized and supplied the heat pump and the Charles Machine Works Company sized, supplied, and installed the ground coil. The system was installed in November, 1984 and evaluated by the UT-EERC during the following heating and cooling seasons.

This report deals with the operation and performance of the GCHP systems located in TECH Houses I and V during the 1984-85 heating season (January 3, 1985 - March 31, 1985) and the 1985 cooling season

(June 6, 1985 - September 31, 1985). Data were recorded hourly with few interruptions throughout both seasons and are presented and discussed herein.

Chapter 2. System Description

2.1 TECH House I

Figure 2.1 is a schematic of the GCHP system located in TECH House I as it operated from September, 1982 to the present time. The heat pump is a TETCO® Model HEWTW-042-C hydronic heating only heat pump. Space cooling is accomplished by re-directing the water-methanol brine solution with three way valves as shown in Figure 2.1.

According to the Directory of Certified Applied Air Conditioning Products[4], water source heat pumps should be rated at a water source temperature of 70°F for heating and 85°F for cooling. The TETCO heat pump has been field tested at these conditions and found to deliver 36,900 Btu/hr in the heating mode and 25,400 Btu/hr in the cooling mode. TECH House I design loads for heating and cooling respectively are 30,000 Btu/hr (at 13°F ambient dry-bulb temperature) and 22,000 Btu/hr (at 95°F ambient dry-bulb temperature).

The ground coil is high density polyethylene nominal 1-1/4 inch diameter pipe with butt-fusion joints and is depicted in plan view in Figure 2.2. The installed length of ground coil is 675 feet and is buried 4 feet deep with no run of pipe closer than 6 feet to any other section of the ground coil. The flow rate in the ground coil was measured to be 7.8 and 8.9 GPM for the heating and cooling modes, respectively. A complete description of the GCHP system located in TECH House I is given in Appendix B and in Reference [5].

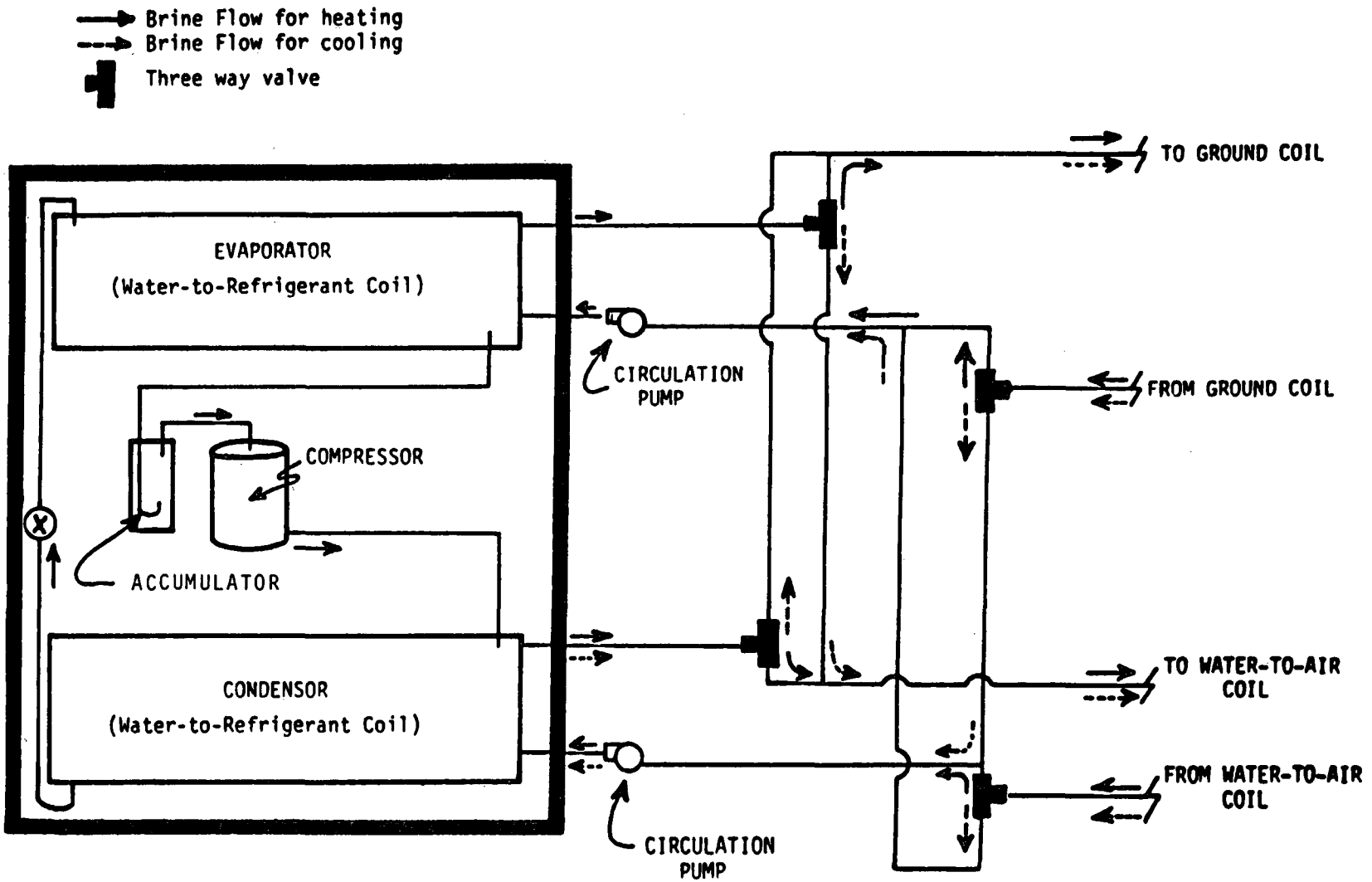

Fig. 2.1. Schematic of TECH House I ground coupled heat pump (September 1982 to present).

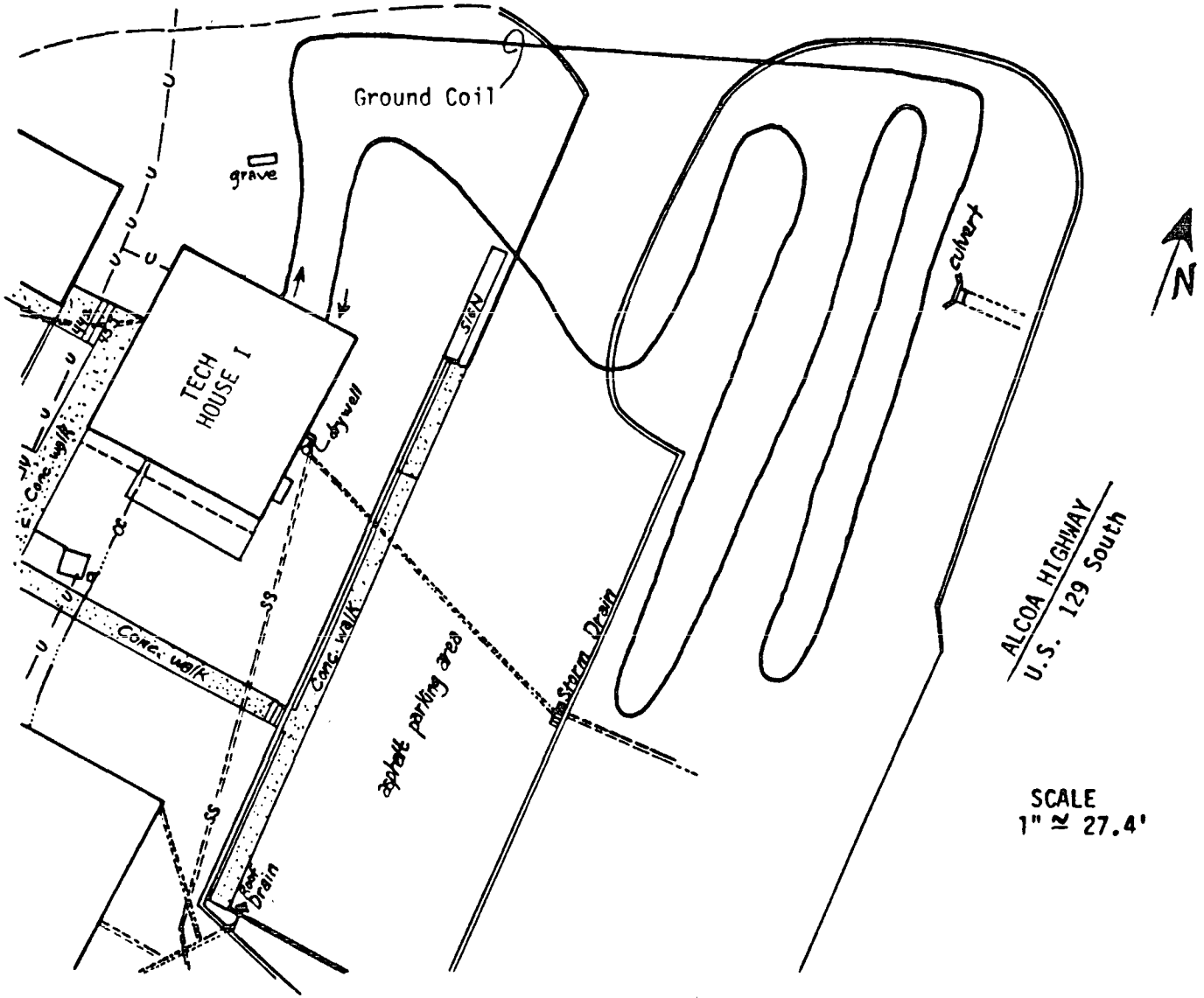

Fig. 2.2. Plan view of TECH House I horizontal ground coil.

2.2 TECH House V

The GCHP system curently in use in TECH House V was designed and installed by GEOsystems, Inc. and the Charles Machine Works, Inc. This system has operated from its installation in November, 1984 to the present time.

The heat pump is a GEOsystem ® Model SWPR-261 water-to-air heat pump with a rated heating capacity of 30,000 Btu/hr (at 70°F entering water temperature) and cooling capacity of 24,000 Btu/hr (at 85°F entering water temperature). The flow rate in the ground coil was measured to be 6.7 gallons/minute. Design heating and cooling loads for TECH House V are 24,000 Btu/hr (at 13°F ambient dry-bulb temperature) and 16,000 Btu/hr (at 95°F ambient dry-bulb temperature), respectively.

A plan view of the ground coil layout is depicted in Figure 2.3. The ground coil is polybutylene nominal 1-1/2 inch diameter pipe with socket-fusion joints and is buried at depths of 4 feet and 6 feet. Initially the required pipe length was calculated to be 850 feet by the installing contractors. However, upon examination of the soil property data, the installed pipe length was increased (by the installer) to 1,135 feet. Appendix B lists the characteristics of the ground coil used for the TECH House V GCHP system.

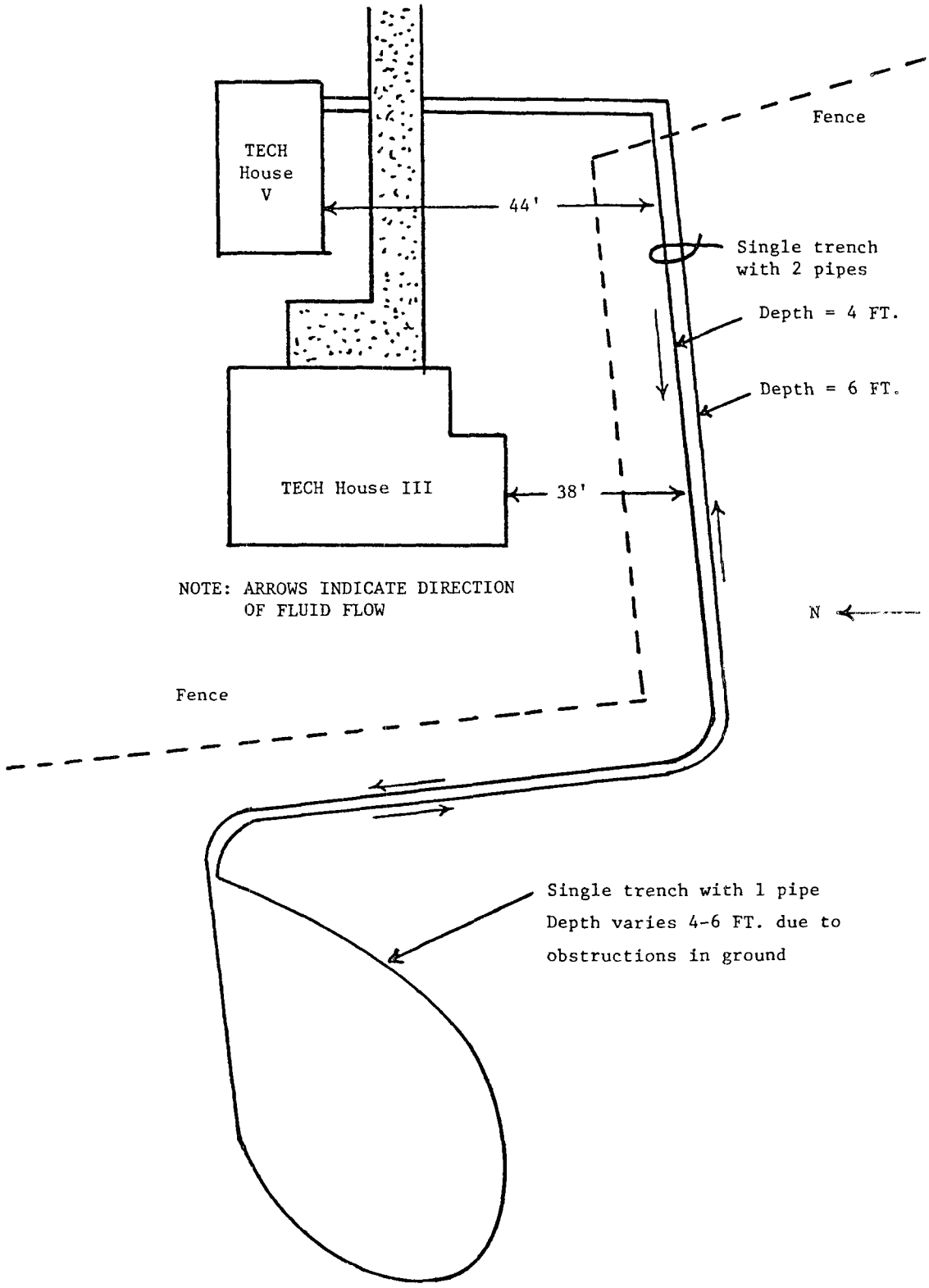

Fig. 2.3. Plan view of TECH House V ground coil.

Chapter 3. Instrumentation

Table C.1 in Appendix C is a listing of the measured parameters of the GCHP system located in TECH House I, and Table C.2 in Appendix C presents a listing of the measured parameters of the GCHP system located in TECH House V.

Platinum resistance temperature devices (RTDs) are used to measure soil and fluid temperatures. Conditioned space dry and wet bulb temperature measurements are also made with RTDs.

Ceramic soil moisture sensing devices are utilized to measure the moisture content of the soil in the vicinity of the ground coils. These devices were calibrated by UT-EERC personnel using soil samples taken from the site. The electrical resistance of these devices changes with moisture content of the surrounding soil. A more complete description of the soil moisture measurements is presented in Appendix C of Reference [6].

Individual watt-hour meters are used in both TECH Houses I and V to measure the electrical power consumption of the heat pump compressor, supply air blower, circulation pump(s), and total house power. The measurement of electrical power consumption is accomplished by counting the revolutions of each eddy disk by optical sensors. The output of each optical sensor is digitized and then sent to a summing circuit.

A flow meter and two RTDs form the basis of each measured water-side heat flow. The ground coil heat flows in TECH Houses I and

V are measured in this manner. The temperature difference measured by the two RTDs is electronically converted to a frequency. That frequency is then electronically multiplied by the frequency output of the in-stream flow meter. The product of the flow rate and the temperature difference (which is directly proportional to the heat flow) is then sent to a summing circuit.

A similar procedure is used to measure the air-side heat flow in TECH House V. Two RTDs and a rotating vane anemometer form the basis of this heat flow measurement. The temperature difference between the two RTDs results in an electrical potential difference which is then electronically converted to a frequency. Therefore, the product of flow rate, measured as a frequency by the in-stream vane anemoneter, and temperature difference is digitized and sent to a summing circuit.

The latent cooling load is measured by recording the total mass of condensate removed from each conditioned space. The condensate flows by gravity through a pipe to a tipping bucket rain gauge. Each tip of the bucket sends a 10 volt digital signal to a summing circuit.

The digital heat flows and electric power consumptions are accumulated by counters located in the data acquisition system (DAS). Each hour on the hour the counters are summed, and then zeroed, and the analog signals are scanned. These data are recorded on magnetic tape and printed on paper. The recorded data then contain the hourly heat flows, electrical energy consumptions, and specified soil and brine temperatures for both systems as well as complete on-site meteorological data.

Chapter 4. Steady State Tests

As was discussed in a previous report in this series[6], one of the most important factors affecting the performance of a GCHP system is how well the ground coil and the soil interact to promote heat transfer.

During the period between the heating and cooling seasons, steady state tests were conducted in both the heating and cooling modes of operation for the GCHP system located in TECH House V. The purpose of these tests was to determine the length of time required for the ground coil and soil to reach a steady state mode of heat conduction. A similar series of steady state tests were previously performed on the GCHP system in TECH House I with results presented in Reference [6].

The tests were performed by locking the GCHP system on in a particular mode of operation, heating or cooling. The system was then allowed to run continuously while controlling inside temperature with electric space heaters and/or opening windows, and data were recorded periodically by the data acquisition system.

Upon evaluation of the test data, it appears that the system requires approximately 7 hours of continuous operation in the heating mode and approximately 8 hours of continuous operation in the cooling mode for the ground coil and the soil to reach an approximate steady state mode of heat conduction. Table 4.1 presents the performance factors (PF) for the steady state tests performed on the GCHP system in TECH House V.

Table 4.1. Steady State Performance Tests

Mode of Operation	PF*	Farfield Temperature (F°)
Heating	3.05	68
Cooling	2.25	66

*Performance after 9-10 hours of continuous operation
Heating PF=(GRND + TSP)/TSP
Cooling PF=(GRND - TSP)/TSP
where, GRND is the ground coil heat flow.
TSP is the total system purchased electric power.

Presented in Figures 4.1 and 4.2 are PF, the total system power (TSP), and the ground coil heat flow plotted with respect to time during the heating and cooling steady state tests respectively. As can be seen in Figures 4.1 and 4.2, the performance of the GCHP system in both modes of operation decreases with time but tends to reach a stable value after approximately 7 to 8 hours of continuous operation. The performance factor decreases only a small amount during heating mode testing, indicating the adequacy of the heat transfer area and soil conditions to supply the required heat flow to the coil on virtually a continuous basis. However, the cooling mode results show a significant decrease in ground heat flow as a function of time resulting in a corresponding decrease in the performance factor. This behavior is likely due to soil drying adjacent to the pipe and also the fact that the summer ground heat transfer is almost double the value for winter operation.

These results indicate a need for short runtimes in cooling mode operation to achieve highest performance. These short runtimes can be

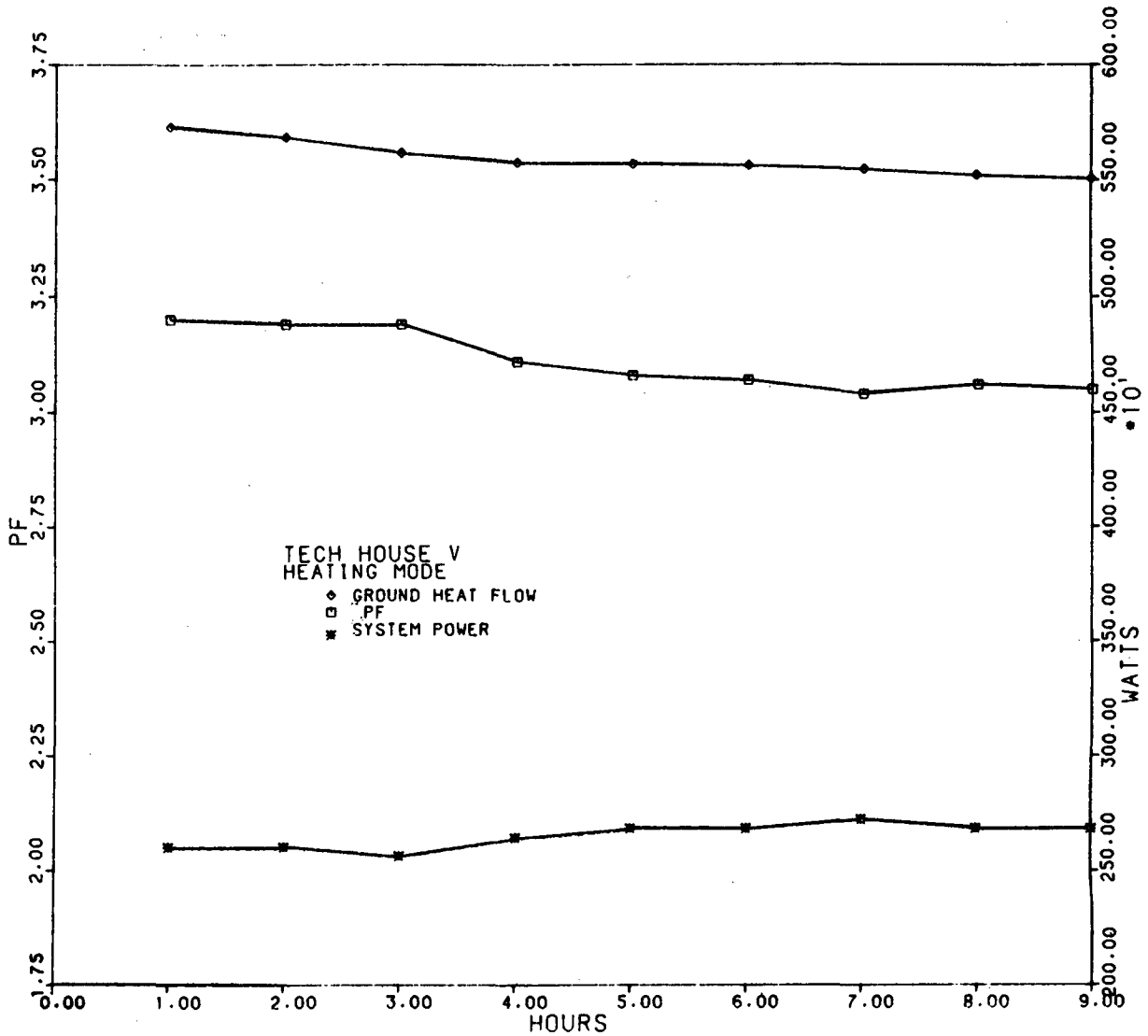

Fig. 4.1. System parameter variation with time (heating mode steady state test).

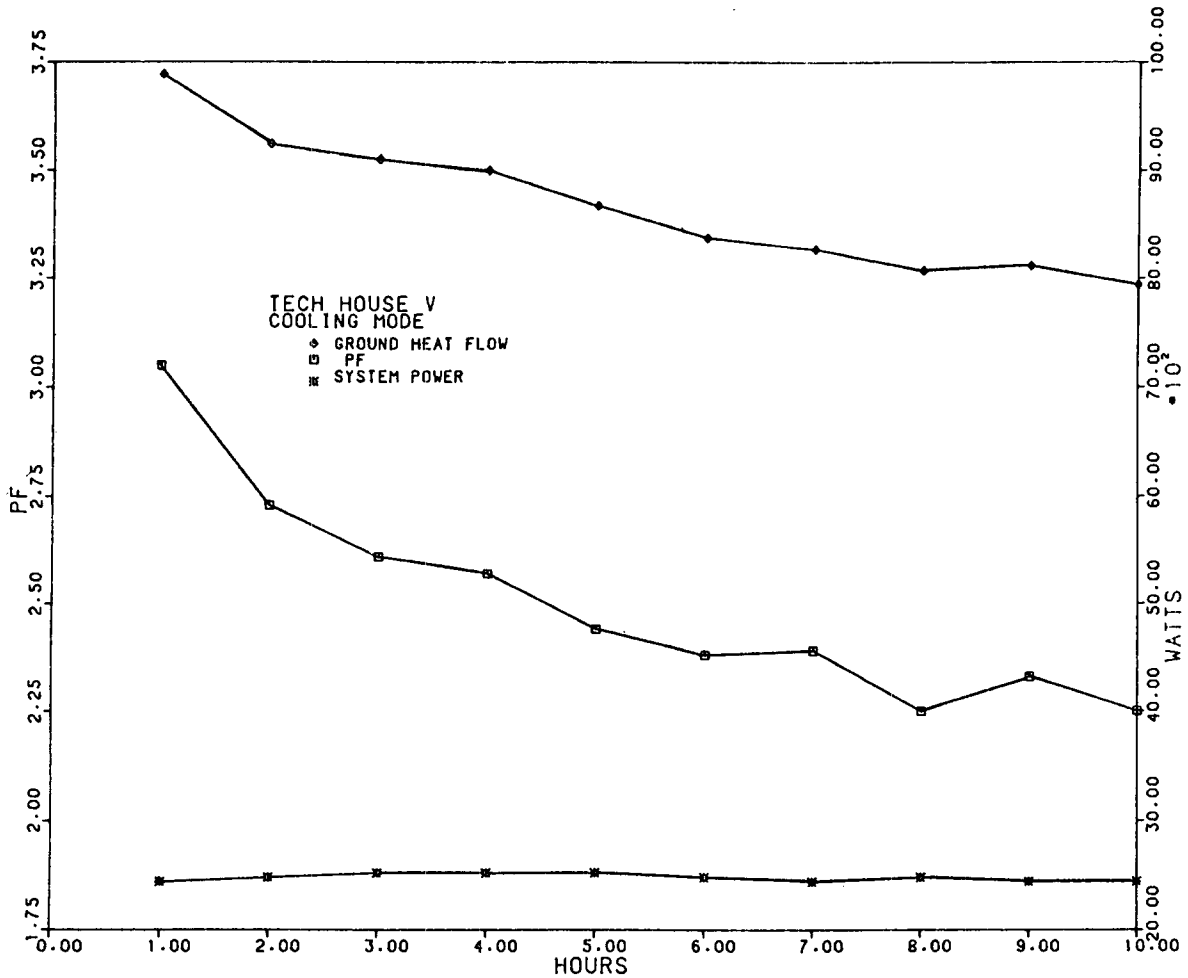

Fig. 4.2. System parameter variation with time (cooling mode steady state test).

produced by oversizing both the ground coil and heat pump unit taking care, however, to allow sufficient runtime for dehumidification of inside air.

In addition to the steady state tests performed on the GCHP system in TECH House V, a series of "city water" tests were performed on both GCHP systems in TECH House I and V. These tests were performed in an attempt to learn the maximum performance factor which can be achieved by these particular GCHP systems. The use of city water in the test approximates an infinitely long ground coil heat exchanger with a farfield ground temperature of 68°F.

In these tests, city water (at 68°F) was run through the heat pump's condenser, or evaporator depending on the particular mode of operation. Table 4.2 presents the results of these tests for both GCHP systems.

Table 4.2. Constant Supply Temperature Performance Tests

Mode of Operation	TECH House I GCHP System	Tech House V GCHP System
Heating	2.93	3.11
Cooling	2.07	2.79

The performance factors presented in Table 4.2 represent the upper limit of performance for the heat pumps currently installed in TECH Houses I and V assuming a farfield ground temperature of 68°F. This temperature normally occurs at the 4 ft. depth during months between

seasons (April and October). It appears that the two heat pump units have near equal performance in the heating mode while the TECH House V system performs significantly better in the cooling mode than the TECH House I system. This difference in performance is apparently due to the fact that the TECH House V heat pump was designed for both heating and cooling operation, while the TECH House I heat pump was originally designed for only heating operation and was later field modified for cooling operation.

Chapter 5. Seasonal Performance Data

5.1. Heat Flows, Power Consumptions, Meteorological Conditions

Specific temperatures, heat flows, electric power consumptions and meteorological data pertinent to the performance of the GCHP systems located in TECH Houses I and V were recorded hourly during the 1984-85 heating season (1/3/85 - 3/31/85) and the 1985 cooling season (6/3/85 - 9/30/85).

Presented in Tables 5.1 and 5.2 are the monthly and seasonal totals of the electric power consumptions of the GCHP systems and the heat flows for TECH Houses I and V respectively for the 1984-85 heating season.

Presented in Tables 5.3 and 5.4 are the monthly and seasonal totals of the electric power consumptions and the heat flows for TECH Houses I and V respectively for the 1985 cooling season.

Table 5.1. TECH House I GCHP System Measured Electric Power Consumptions and Heat Flows (1984-85 Heating Season)

Month	Electric Power (kWh)				Heat Flows (kWh)
	Blower	Pumps	Comp.	Total	GRND
Jan. 3-31	264	167	1215	1646	2098
Feb.	199	133	919	1251	1423
Mar.	103	67	528	698	906
TOTAL	566	367	2662	3595	4427

GRND: Heat extracted from earth via ground coil.

Table 5.2. TECH House V GCHP System Measured Electric Power Consumptions and Heat Flows (1984-85 Heating Season)

Month	TSP (kWh)	Heat Flows (kWh)	
		GRND	ASHF
Jan. 3-31	773	1409	1945
Feb.	570	1076	1548
Mar.	250	454	582
TOTAL	1593	2939	4075

TSP: Total system power (electric power consumed by compressor, blower, and circulation pump).
ASHF: Air side heat flow (heat delivered to conditioned space).

Table 5.3. TECH House I GCHP System Measured Electric Power Consumptions and Heat Flows (1985 Cooling Season)

Month	Electric Power (kWh)				Heat Flows (kWh)
	Blower	Pumps	Comp.	Total	GRND
Jun. 3-30	77	49	501	627	1586
Jul.	89	56	603	748	1809
Aug.	122	77	803	1002	2565
Sept.	68	43	458	569	1440
TOTAL	356	225	2365	2946	7400

GRND: Heat rejected to earth via ground coil.

Table 5.4. TECH House V GCHP System Measured Electric Power Consumptions and Heat Flows (1985 Cooling Season)

Month	TSP (kWh)	Heat Flows (kWh) GRND	Heat Flows (kWh) ASHF	Latent Load (kWh)
Jun. 3-30	284	1037	594	315
Jul.	261	954	540	247
Aug.	497	1752	986	398
Sept.	399	1425	816	241
TOTAL	1441	5168	2936	1201

Presented in Table 5.5 are the heating and cooling degree days measured at the TECH Facility along with the corresponding long-term average heating and cooling degree days recorded by the U.S. Weather Service at McGhee Tyson airport in Knoxville, TN. Both heating and cooling degree days are measured from a base of 65°F.

Table 5.5. Heating and Cooling Degree Days

1984-85 Heating Season			1985 Cooling Season		
Month	Measured °F Days	Average °F Days	Month	Measured °F Days	Average °F Days
Jan. 3-31	1032	831	Jun. 3-30	175	283
Feb.	792	658	July	188	391
Mar.	514	483	Aug.	264	372
———	———	———	Sept.	138	209
TOTAL	2338	1972	TOTAL	765	1255

5.2 Seasonal Performance Factor

The seasonal performance factor (SPF) defines the overall performance of a heat pump system. The SPF is calculated by dividing the total useful heat provided to or removed from the conditioned space by the total purchased electric power required by the heat pump system. Two methods of calculating the SPF are represented.

SPF_i is the performance factor for the case in which the heat pump package and all other power consuming elements (circulation pump(s) and blower) are located within the conditioned space. Both GCHP systems are currently installed in this manner.

The heating SPF_i for both GCHP systems can be calculated as,

$$SPF_i = \frac{GRND + TSP}{TSP} \qquad (5.1)$$

where: GRND is the ground coil heat flow.
TSP is the total system purchased electric power.

The cooling SPF_i for both GCHP systems can be calculated as,

$$SPF_i = \frac{GRND - TSP}{TSP} \qquad (5.2)$$

SPF_o is the performance factor for the case in which the heat pump package and all other power consuming elements are located outside the conditioned space (i.e., in a garage). For the TECH House I system, the heating SPF_o can be calculated as,

$$SPF_o = \frac{WTAC + BLWR}{TSP} \qquad (5.3)$$

where: WTAC is the water-to-air coil heat flow.
BLWR is the electric power required by the blower.
TSP is the total system purchased electric power.

The cooling SPF_0 for the TECH House I system can then be calculated as,

$$SPF_0 = \frac{WTAC - BLWR}{TSP} \qquad (5.4)$$

For the TECH House V system, the heating SPF_0 can be calculated as,

$$SPF_0 = \frac{ASHF}{TSP} \qquad (5.5)$$

Where ASHF is the air side sensible heat flow including blower energy.

The cooling SPF_0 for the TECH House V system can be calculated as,

$$SPF_0 = \frac{ASHF + COND}{TSP} \qquad (5.6)$$

Where: COND is the latent load measured in TECH House V.

Tables 5.6 and 5.7 show a comparison of the monthly and seasonal values for the performance of the GCHP systems in TECH Houses I and V for the 1984-85 heating season and the 1985 cooling season. Due to technical problems encountered with the measurement of the water-to-air coil heat flow, SPF_0 for the GCHP system in TECH House I is not reported. Based on the performance during previous heating and cooling seasons[6], however, it is estimated that the overall heating SPF_0 would have been 1.87 and the overall cooling SPF_0 would have been 1.84.

Table 5.6 1984-85 Heating Season Performance Comparison.

Month	TECH House I	TECH House V	
	SPF_i	SPF_i	SPF_o
Jan. 3-31	2.27	2.82	2.52
Feb.	2.14	2.89	2.72
Mar.	2.30	2.82	2.33
TOTAL	2.23	2.84	2.56

Table 5.7 1985 Cooling Season Performance Comparison

Month	TECH House I	TECH House V	
	SPF_i	SPF_i	SPF_o
Jun. 6-30	1.53	2.65	3.20
Jul.	1.42	2.66	3.02
Aug.	1.56	2.53	2.78
Sept.	1.53	2.57	2.65
TOTAL	1.51	2.59	2.87

5.3 Soil Moisture Measurements

5.3.1 TECH House I

Soil moisture and its relationship to the performance of the GCHP system in TECH House I has been established in previous reports in this series[1,6]. With indigenous clay as a backfill material, saturation conditions have prevailed during winter operation, resulting in a high performance factor. Summer operation has produced

progressive soil drying resulting in a corresponding drop in performance.

The soil moisture for both the TECH House I and V ground coil fields was measured at weekly intervals throughout the reporting period. Figure 5.1 presents soil moisture versus time in both fields. These particular values represent the soil moisture near the mid-point of each ground coil. As was reported in Reference [6], the sand backfill around the TECH House I ground coil was observed to only partially rehydrate during the Autumn after having become dry during the 1984 cooling season (the first summer of operation with the sand backfill). Figure 5.1 shows that this soil never became fully saturated (reaching a maximum level of only 60% saturation) during the 1984-85 heating season as it did in previous heating seasons when the backfill material was indigenous clay. The lower value of the overall SPF recorded during the 1984-85 heating season than during the 1983-84 season may be partially explained by the failure of the sand backfill to fully rehydrate to a saturated moisture level.

At the beginning of the 1985 cooling season (early June), the soil near the TECH House I ground coil became completely dry and remained so until mid-August. At that time moisture was artificially introduced into the TECH House I ground coil field through the soaker pipe which was installed during the ground coil reinstallation prior to the 1984 cooling season[6]. A total of 21,000 gallons of water was injected at three different times spaced about one week apart. A resulting increase in soil moisture can be observed in Figure 5.1.

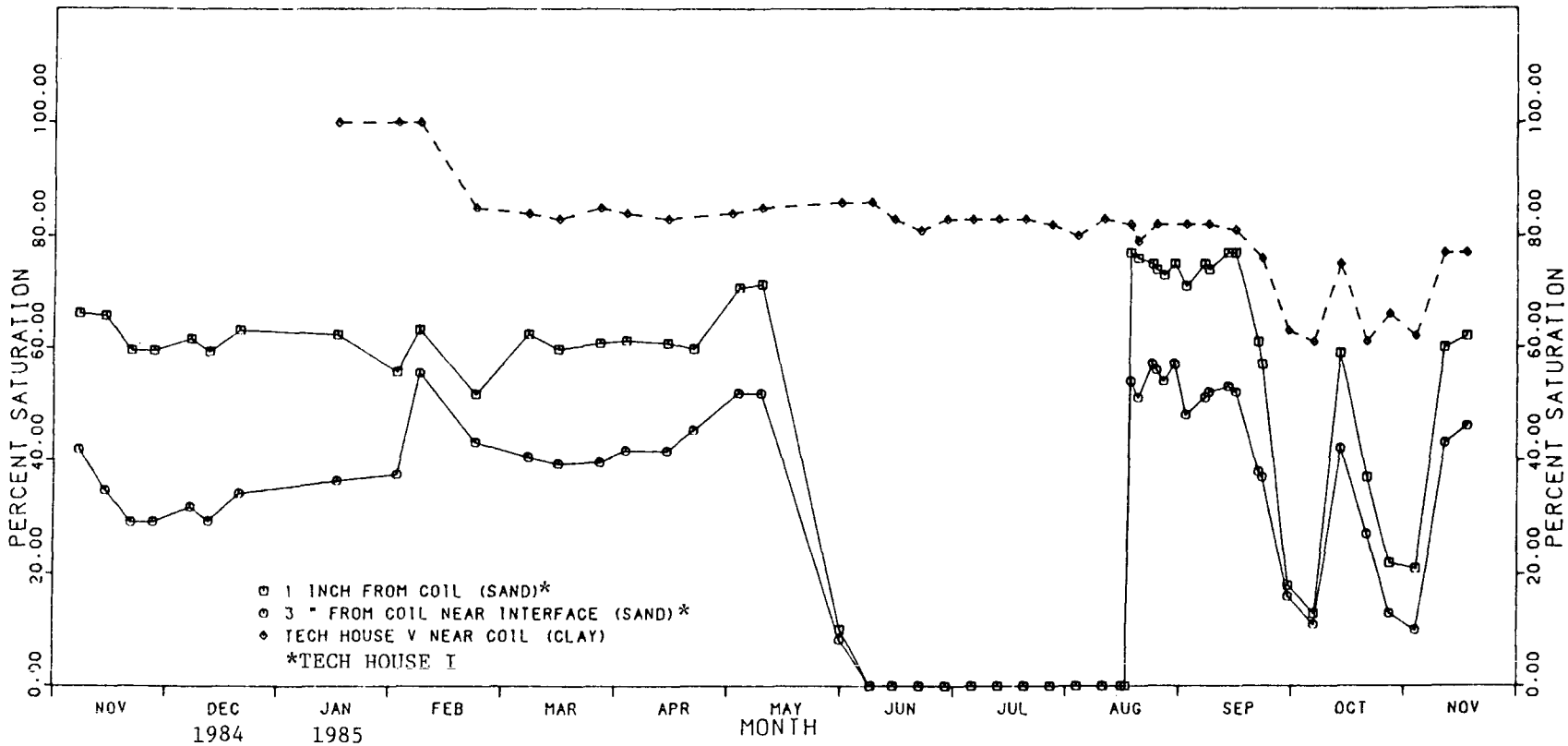

Fig. 5.1. Soil moisture plot.

Due to the design of the soaker pipe, most of the water was admitted to the soil near the beginning and end of the coil field and was allowed to move through the soil to the mid-region of the ground coil. After addition of the water, the soil moisture level at the midpoint was increased to a value of only about 80 percent saturation. Following the time of the soil moisture addition, the performance of the TECH House I system was observed to increase by only 4 percent over the average performance prior to that time.

This surprisingly small increase can be explained with the aid of Figure 5.2 which shows the system performance factor and the calculated soil thermal conductivity* adjacent to the pipe wall for several days just before and after the first two periods of water injection. At the first water injection period, the soil thermal conductivity is seen to jump from a value of 0.63 W/m-°C to 1.5 W/m-°C and then return gradually to the value prior to injection over only a two-day period. The performance factor is seen to closely track the calculated soil thermal conductivity as has been reported previously in this work[1].

The region over which thermal conductivity was measured was within 1/2 inch of the pipe wall. However, moisture probes are located no closer than 1 inch from the pipe and, as noted in Figure 5.1, showed almost constant moisture content for four consecutive weeks. In this case, performance is seen to be significantly affected by the moisture migration very near the pipe surface, which is not indicated by any of the moisture probes.

*Based on ground and pipe wall temperatures and ground heat flows for the time period considered. See Reference [6] for an explanation of how the calculation was performed. The paucity of the data points is a result of a requirement of 4 consecutive hours of constant operation to obtain near steady state heat transfer in the soil as required by the Calculation Procedure.

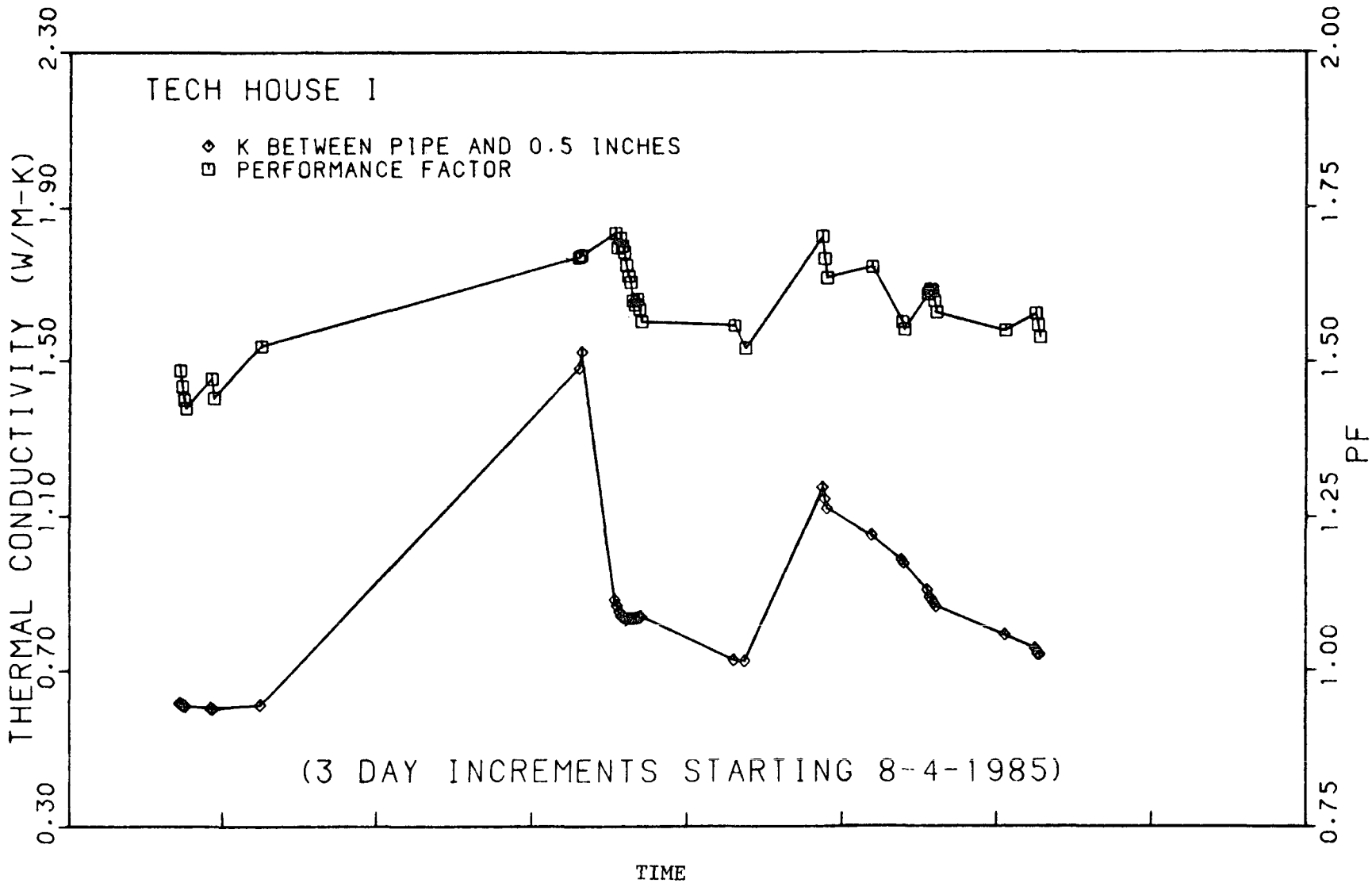

Fig. 5.2. Thermal conductivity and performance factor comparison.

5.3.2 TECH House V

Figure 5.1 shows that during the month of January the soil in the TECH House V ground coil field was fully saturated. The moisture level then decreased to approximately 80% saturation in mid-February and remained at that level for the remainder of the heating season. For the 1985 cooling season, the moisture level of the soil near the TECH House V ground coil remained almost constant at 80% saturation until late September when the cooling season was essentially over.

5.4 Performance Variation with System Runtime

As was reported in a previous study[6], the performance of a GCHP system is partially dependent on the system runtime, or, whether the system is operating more in a transient or steady state mode of operation.

Figures 5.3 and 5.4 show the system performance-runtime relationship for the GCHP systems in TECH House I and V during the 1985 cooling season. As the system runtime increases, thus approaching the steady state mode of operation, the value for the performance factor is normally found to decrease. As the system operates more in the transient region mode (decreasing runtime), the value for the performance factor usually increases. Data for August-September, 1985 in Figure 5.3 are influenced somewhat by the addition of soil moisture during that period. This effect appears to be relatively small as discussed in Section 5.3.1. Heating season data are not available for the performance-runtime relationship due to difficulties with data collection and, consequently, for this period are omitted.

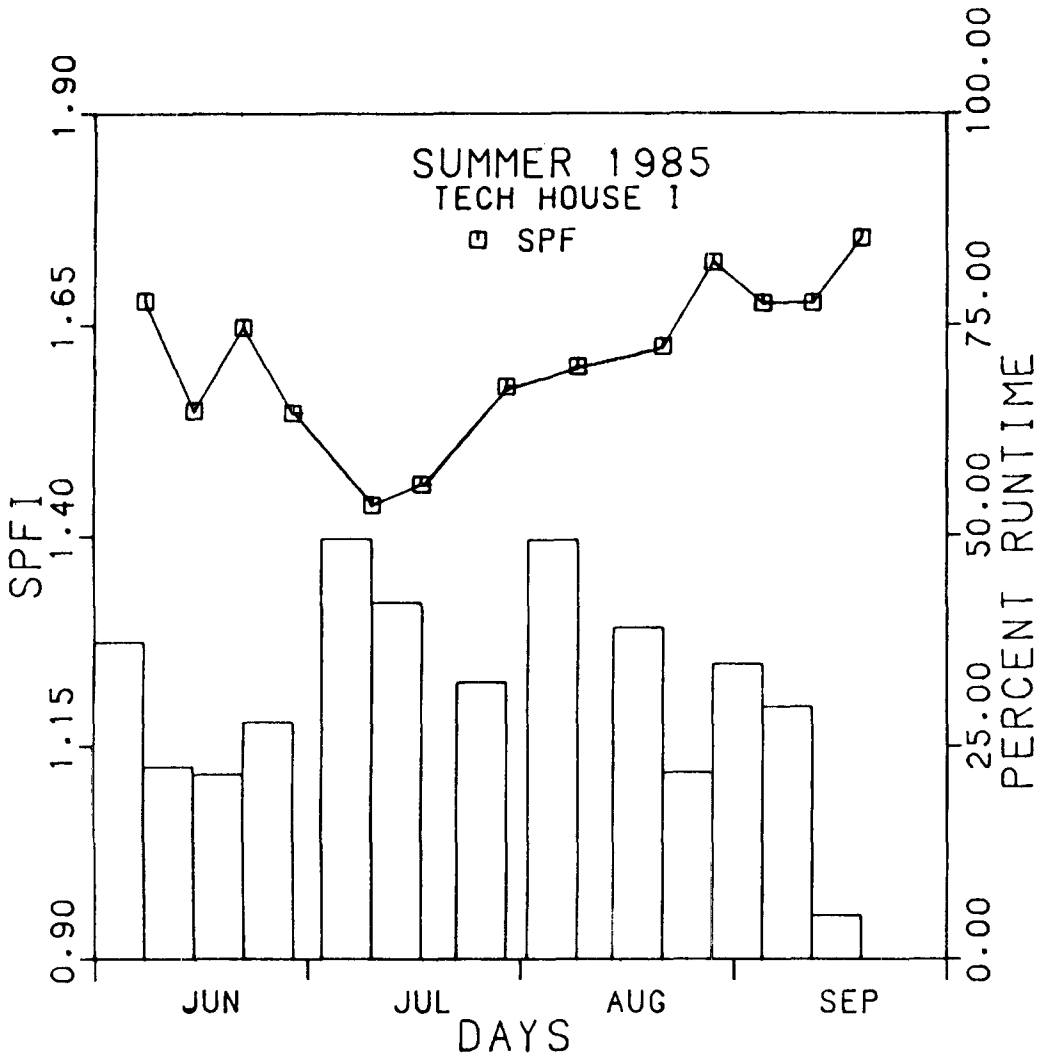

Fig. 5.3. TECH House I performance-runtime relationship (1985 cooling season).

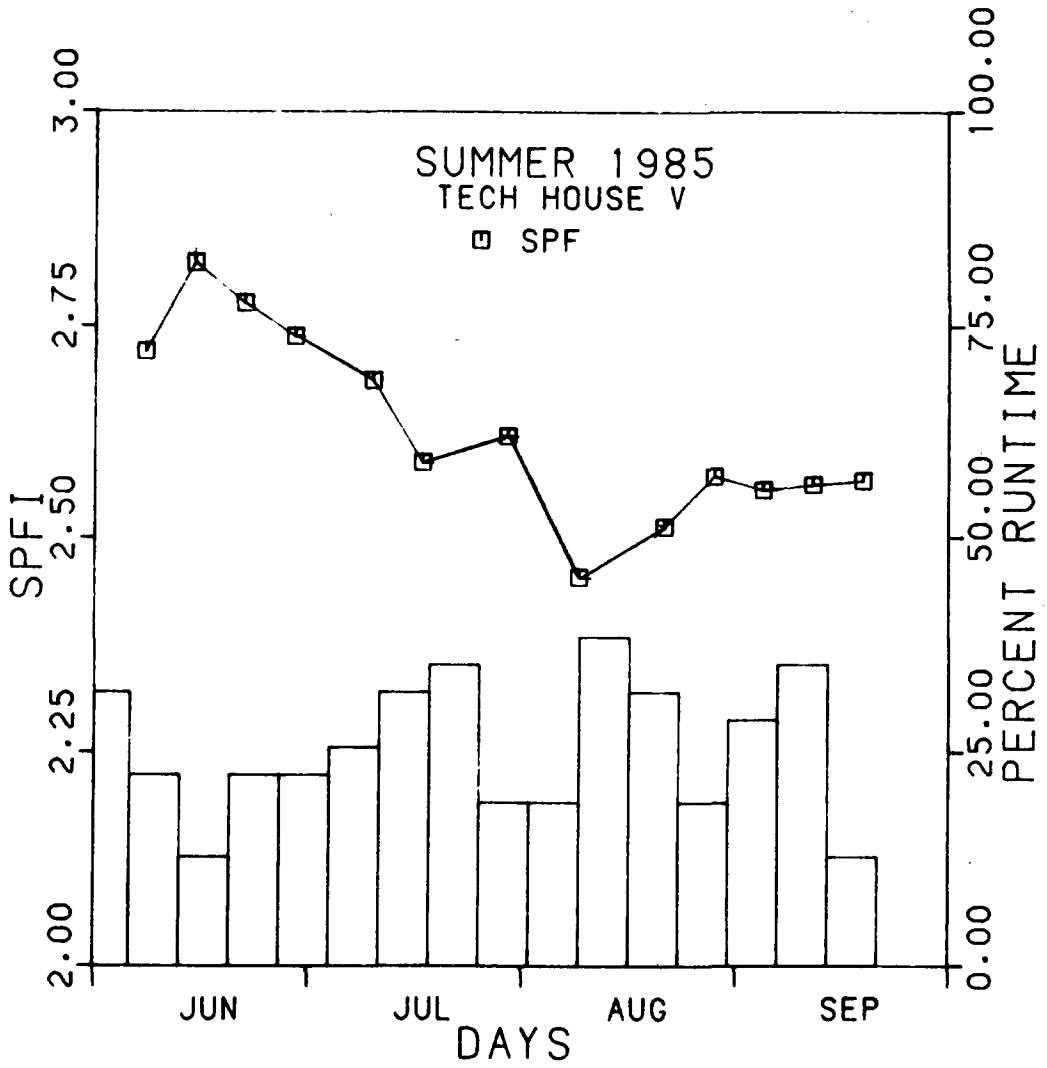

Fig. 5.4. TECH House V performance-runtime relationship (1985 cooling season).

Table 5.8 shows a comparison of the seasonal average runtime plus average runtime for temperature bins containing both summer and winter design temperatures for both GCHP systems during the 1984-85 heating season and the 1985 cooling season.

Table 5.8. GCHP System Runtime Comparison

Time Period	TECH I		TECH V	
	Average Seasonal Runtime	Average* Severe Weather Runtime	Average Seasonal Runtime	Average* Severe Weather Runtime
1984-85 Heating Season	56%	97%	30%	72%
1985 Cooling Season	32%	78%	23%	56%

*Below ambient temperature of 20°F in winter and above 85°F in summer.

One can observe that the GCHP system in TECH House V operated significantly less than the GCHP system in TECH House I. The seasonal average runtime for the TECH House V system was found to be 46 percent lower than the seasonal average runtime of the TECH House I system during the 1984-85 heating season and 28 percent lower during the 1985 cooling season.

Runtime is determined primarily by system capacity. The system in TECH House V consisted of a heat pump unit with approximately 50 percent excess cooling capacity compared to the house design cooling load. By contrast, the heat pump unit in TECH House I had an excess cooling capacity over the design house cooling load of only 15 percent. A second factor affecting runtime is the ground coil capacity. While

there are no universally accepted rules to fix the size of a ground coil relative to the house design load, it is noted that the length of ground coil installed in TECH House V is approximately 2.3 times greater per ton of house design cooling load than the ground coil in the TECH House I system.

5.4.1 Soil Comparison with Other Installations

The performance of a ground coupled heat exchanger is largely determined by the physical characteristics of the soil in which it is placed. Therefore, during the month of July, 1983, a sample of soil was taken from the vicinity of the TECH House I coil field at the TECH site and sent to materials testing facility for analysis. The results of this analysis are presented in Table 5.9. Also shown in Table 5.9, are the available soil characteristics of some other testing facilities which have ground coil installations.[7, 8, 9]

Table 5.9 Soil Sample Data

Soil Property	TEST FACILITY LOCATION			
	TN	NY	OK	SD
Dry Density (lb/ft^3)	103.0	83.0	105.0	108.2
Natural Moisture (%)	17.0	18.0	--	14.0
Liquid Limit (%)	34.0	35.6	--	46.0
Plastic Limit	21	27.2	--	24.0
Plasticity Index (%)	13	8.4	--	22.0
Soil Texture (%)				
Clay and Silt	87	42	--	--
Sand	13	47	--	--
Gravel	0	11	--	--
Thermal Conductivity (dry) (Btu/hr-ft-°F)	0.32*	--	0.56	--

*In close proximity to the ground coil as reported in Reference 1.

The 1982 ASHRAE Applications Handbook categorizes four general types of soil. ASHRAE suggests that soils can be generally classified as either heavy damp, heavy dry, light damp or light dry. In order to classify soils into the various categories, ASHRAE suggests that four specific soil properties be known. The soil properties being thermal conductivity, thermal diffusivity, density, and specific heat. Based solely upon the data provided in Table 5.9 above, it is not possible to accurately predict the soil type at each test location. However, using the values given for the dry density and the natural moisture content of each site, the thermal conductivity for each location can be estimated using Figure A6 in Reference 8. Once the thermal conductivity has been determined, the thermal diffusivity for each location can then be estimated. Since the values for specific heat suggested in the ASHRAE classification of soils do not vary a great deal, a constant value of 0.23 Btu/lb-°F was assumed for each soil-type prediction.

Using the method described above, the general soil types for each test facility are presented in Table 5.10. It should be pointed out, however, that the value predicted for the thermal conductivity of the Tennessee soil (estimated from Figure A6 of Reference 8) was substantially higher than the value for the thermal conductivity determined from actual experimental data.

Table 5.10. Test Facility Soil Classification

Test Facility Location	Soil Classification
Tennessee	heavy-damp
New York	heavy-damp or heavy-dry
Oklahoma	heavy-dry
South Dakota	heavy-damp

Table 5.11 presents the pipe length per nominal ton of cooling and per kW of heating for the GCHP installations at the various testing facilities. The installations presented in this particular table are horizontal coil, single-pass heat exchangers.

Table 5.11. Ground Coil Size Comparison

Test Facility	Pipe Length per Ton*	Pipe Length per Kw†
Tennessee		
- TECH House 1 coil	338 ft	63 ft
- TECH House 5 coil	568 ft	129 ft
New York	428 ft (avg.)	Data Unavailable
Oklahoma	Data Unavailable	112 ft
South Dakota	Data Unavailable	Data Unavailable

*Installed cooling capacity

†Installed heating capacity

5.5 Heat Pump Electric Demand Comparison

With the increasing use of heat pumps as residential heating and cooling systems, many electric utilities have become concerned with the loads that air-source heat pump systems impose on the power grid, particularly during the heating season. These peak loads result primarily from the use of electric resistance heat in addition to heat pump operation during periods when heat pump heating capacity is insufficient. The electric consumption of GCHP systems is essentially constant and does not significantly increase with decreasing ambient temperatures. Consequently, the use of GCHP systems may significantly reduce the peak utility electric demand.

Presented in Figure 5.5 are the power consumptions of the GCHP system in TECH House I and an air-source heat pump system in TECH House III recorded for a 36 hour period during the 1984-85 heating season. The air-source heat pump in question was configured to run in either

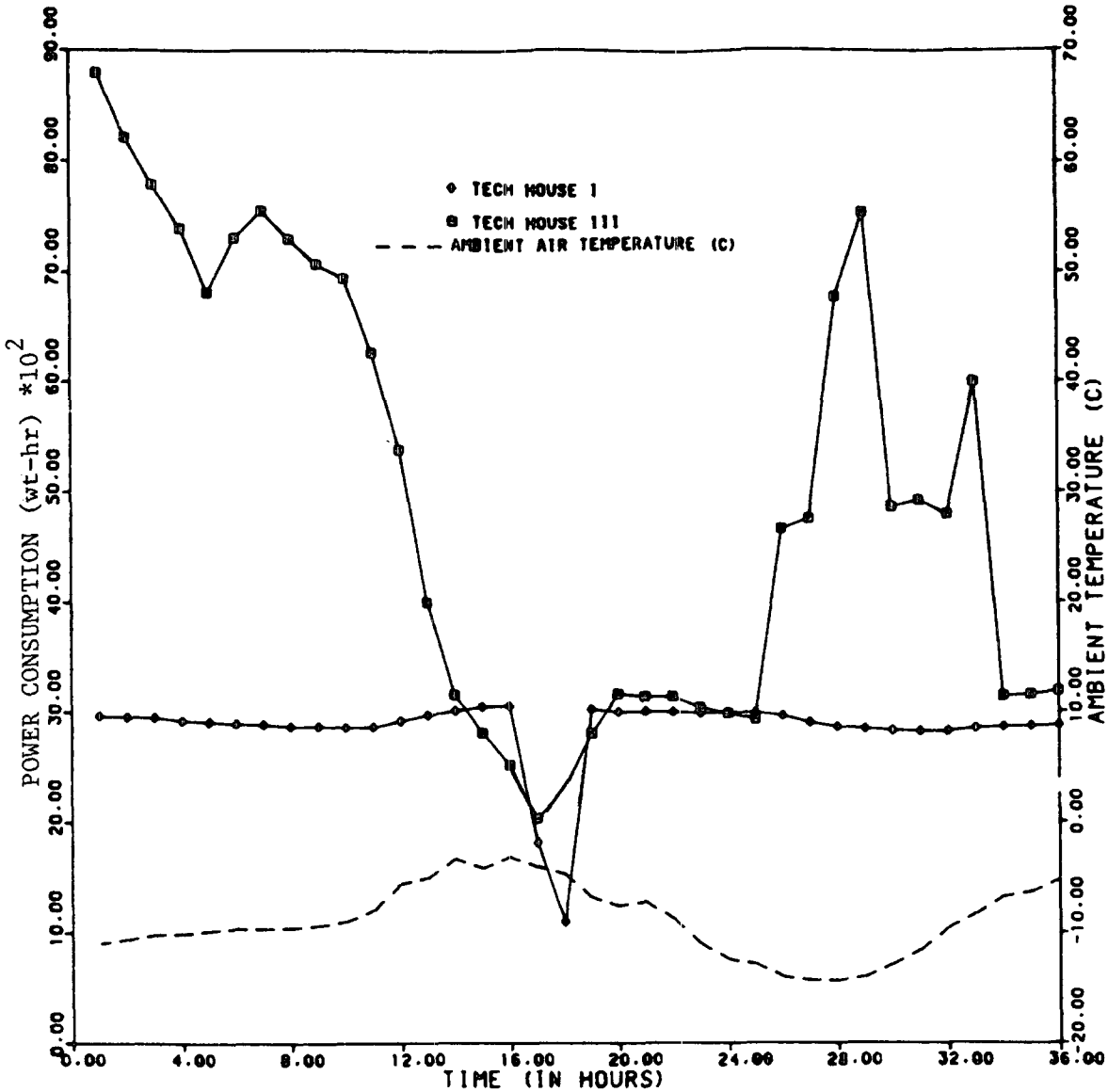

Fig. 5.5. Power consumption comparison.

two-speed or single-speed mode.[10] Comparisons made in this section use data from periods when the TECH House III system ran in the single speed mode. Since the orientation, floor plan, and construction of TECH House III are identical to that of TECH House I, the heating and cooling loads are comparable. The power consumption of the air-source heat pump system shown in Figure 5.5 reflects the need for auxiliary heat (auxiliary heat is generally provided by an electric resistance duct heater). As can be seen, the GCHP system draws fairly constant electrical load. The air-source heat pump system, however, requires approximately a 9 kW power input when the ambient temperature is low compared to a demand of about 3 kW for the GCHP system.

Figures 5.6 and 5.7 show the daily average power consumptions and heating loads for the GCHP system in TECH House I and for the air-source heat pump system in TECH House III. These figures depict electric power requirements during a typical day (obtained by averaging the power requirement for a given hour over the entire season). The peak electrical loads recorded during the 1984-85 heating season were about 3.5 kW and 11.5 kW for TECH Houses I and III, respectively, while Figures 5.6 and 5.7 show average peak power consumptions for both systems of about 2 kW. Thus, it appears that the averaged power consumption curve shown in Figure 5.7 greatly under estimates the range of power consumptions for an air-source system. This difference also gives an indication of the widely varying and somewhat unpredictable load that an air-source heat pump system has on the electric utility.

Figures 5.8 and 5.9 show the frequency of occurrence of various daily peak electrical demands for the heating and cooling seasons,

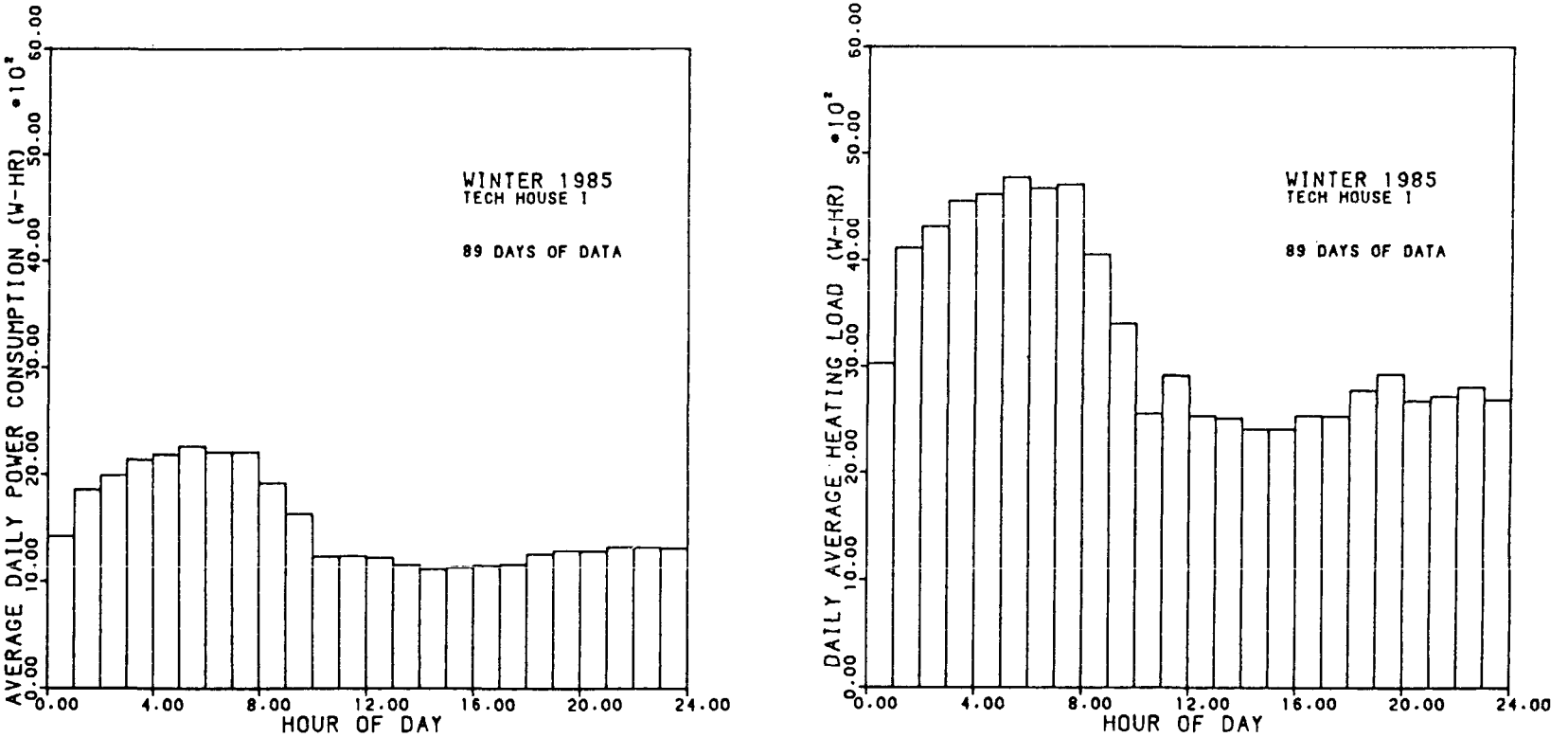

Fig. 5.6. Average daily power consumption and average daily heating load comparison of TECH House I GCHP system.

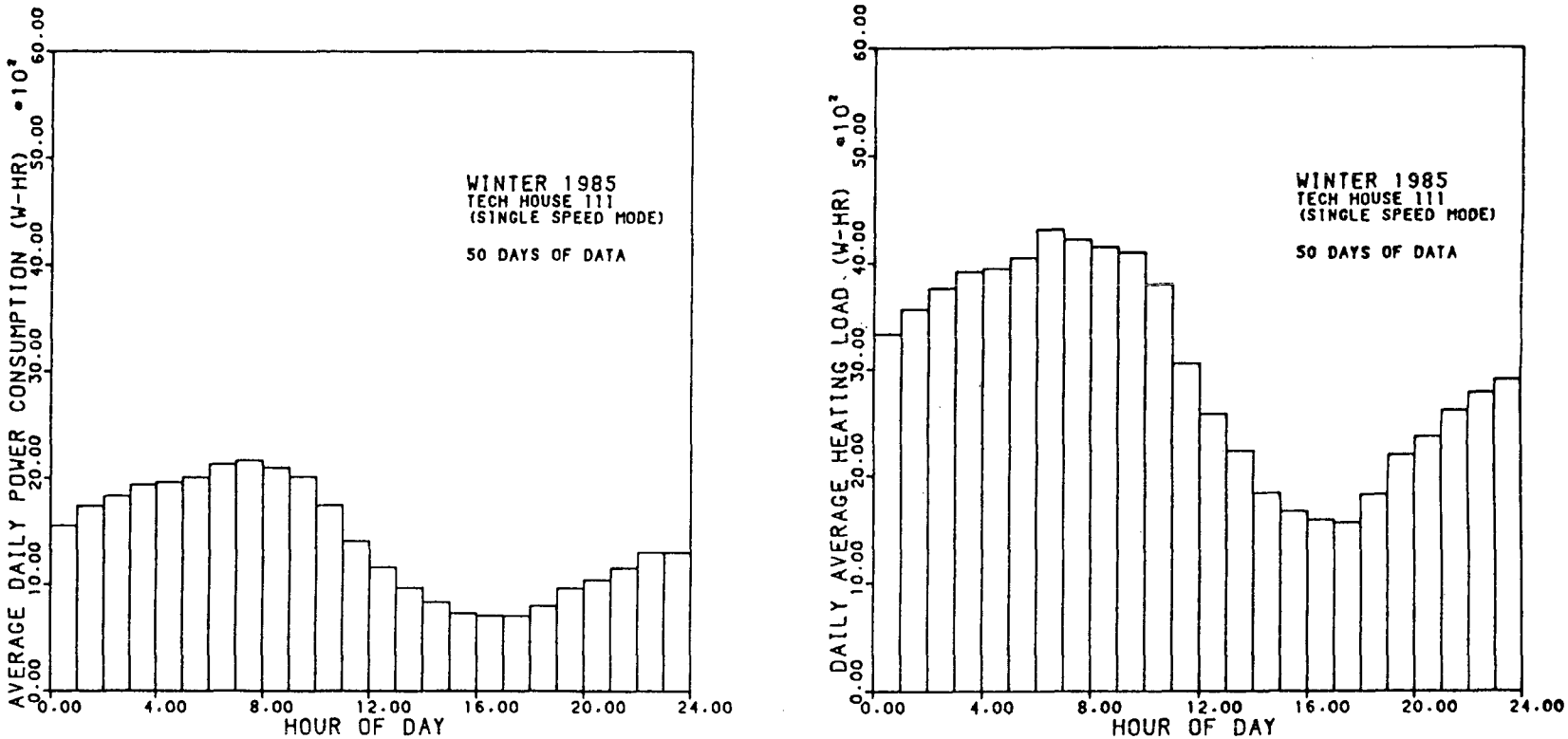

Fig. 5.7. Average daily power consumption and average daily heating load comparison of TECH House III air source heat pump system.

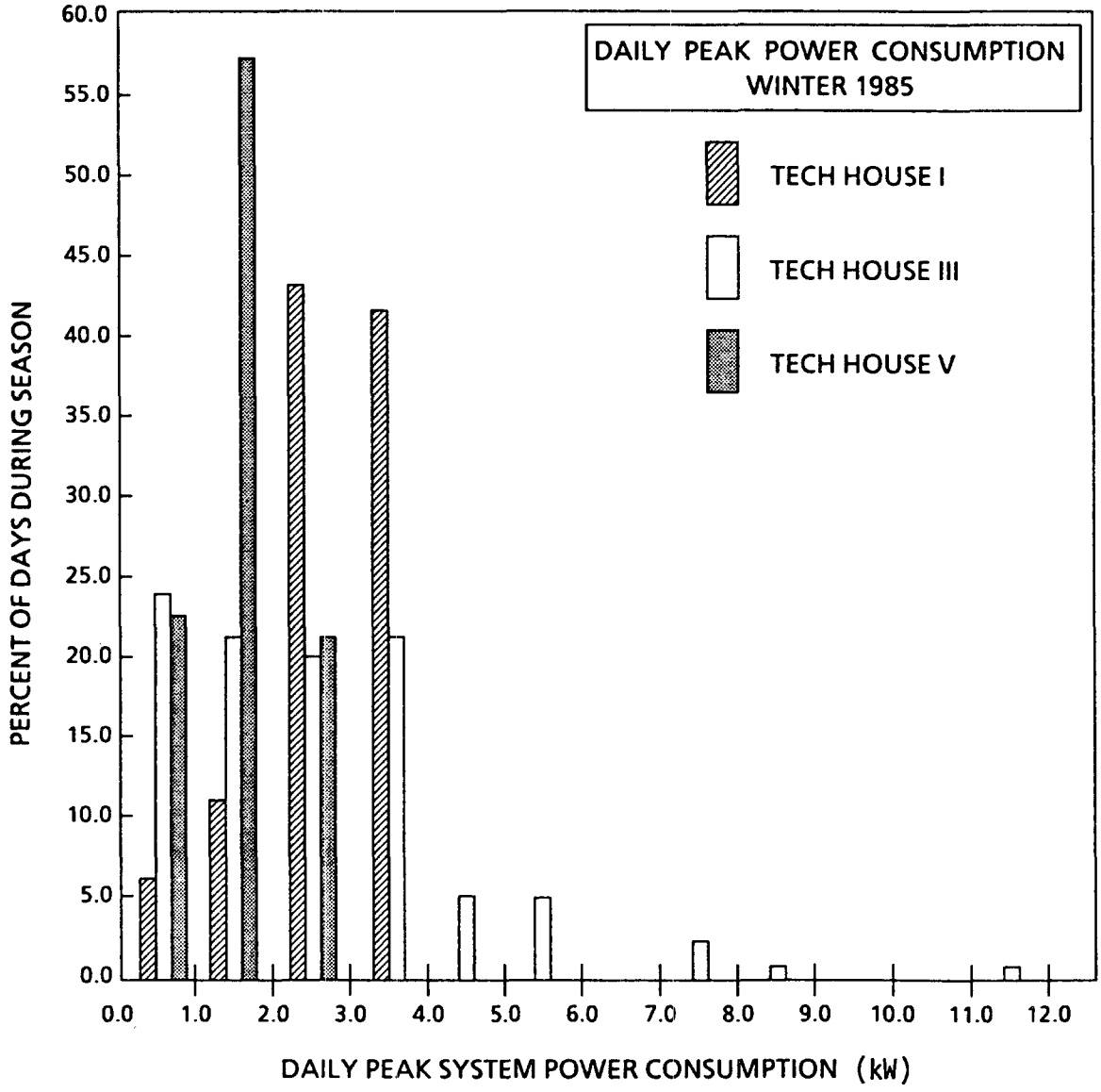

Fig. 5.8. Frequency of occurrence of various daily peak electrical demands during the 1984-85 heating season.

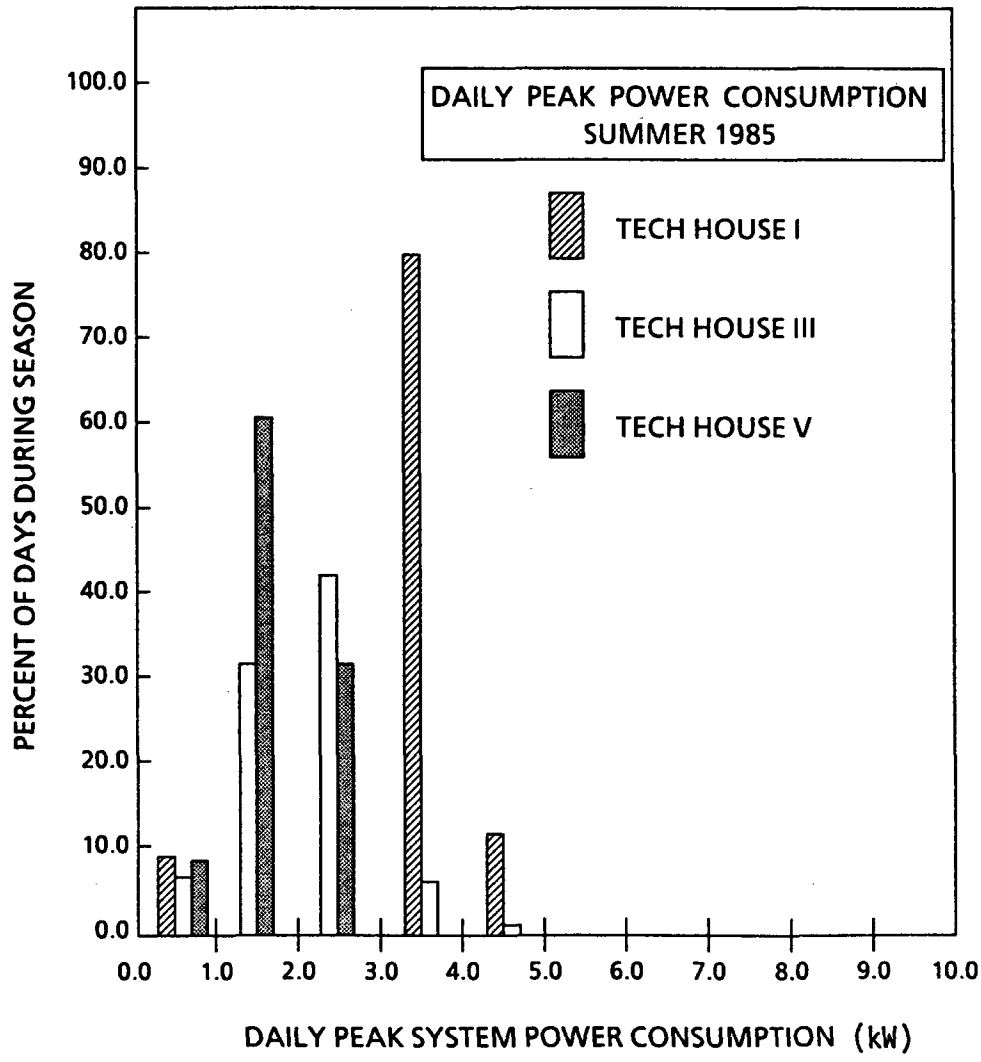

Fig. 5.9. Frequency of occurrence of various daily peak electrical demands during the 1985 cooling season.

respectively. Although data for TECH Houses I, III and V are shown, the only meaningful comparison is between TECH Houses I and III which are comparable in size, orientation and design. This comparison reveals a clear advantage of the TECH House I system in the heating mode with all daily peaks falling below 4 kW. The air-air system in TECH House III, however, required more than 4 kW for about 18% of the days of the season with a peak seasonal demand of about 11.5 kW. In the cooling mode of operation the air-air system in TECH House III shows somewhat lower peaks although it is noted that the TECH House I GCHP system is poorly designed for cooling operation.

5.6. TECH House I Heat Pump Performance Summary

The GCHP system located in TECH House I has been operated for three consecutive heating and cooling seasons. Table 5.12 presents a summary of the performance measured during these seasons.

The SPF for both the 1982-83 and 1983-84 heating seasons was about 2.6. However, during the 1984-85 heating season the SPF was measured to be only 2.23. This decrease in performance is primarily attributed to the fact that the sand backfill did not fully rehydrate to a saturated moisture level as had the indigenous clay backfill prior to both the 1982-83 and 1983-84 heating seasons. Another possible reason for the decrease in system performance was leakage in the three-way valves that control the mode of operation of the heat pump system. Several of the valves were found to be leaking slightly prior to the 1985 cooling season, however, tests indicated that the leakage had a negligible effect on the overall system performance. A more complete description of the leakage problem can be found in Appendix A.

The SPF for the 1983, 1984, and 1985 cooling seasons was measured to be 1.11, 1.35, and 1.51, respectively. The increase in performance during the 1984 and 1985 cooling seasons (operation with sand backfill) may be partially due to the ground coil having a better soil to pipe contact. However, a lower average system runtime (a result of lower ambient temperatures) also contributed significantly to the improvement in performance. The cooling degree days measured during the 1983 cooling season were approximately 7 percent higher than the long-term average, while the cooling degree days measured during the 1984 and 1985 cooling seasons were lower than the long term average by about 18 percent and 39 percent, respectively.

Table 5.12. Heating and Cooling Seasonal Performance Factors for TECH House I

Period	SPF_i	Measured °F Days	Average† °F Days	Avg. % Runtime
1982-83 heating season	2.60	3290	3380	33%
1983-84 heating season	2.59	3680	3199	47%
1984-85 heating season	2.23	2338*	1972*	56%
1983 cooling season	1.11	1339	1255	43%
1984 cooling season	1.35	947	1157	39%
1985 cooling season	1.51	765	1255	32%

*Includes only the months of January, February, and March.

† Seasonal differences in average degree days result from using slightly different periods of time for the heating or cooling season.

Thus, it appears that while the sand backfill may have helped to improve the cooling performance, it lowered the heating performance significantly due to a reduction in moisture migration toward the ground coil.

Chapter 6. Economic Analysis

To estimate the present economic feasibility of a ground coupled heat pump in the Knoxville, Tennessee area, a simple payback economic analysis was carried out. It should be noted that the analysis applies only to the specific conditions of these tests including equipment, building location, and climate. Results will vary, perhaps significantly, under other conditions and consequently, these results should not be extrapolated for other conditions or locations. The ground coupled heat pump system in TECH House V was compared to both a standard and a high efficiency air-to-air split system heat pump. The three heat pump systems are described below:

SYSTEM 1: Standard air-air system

Heating capacity: 21,000 Btu/h
Heating COP: 2.7 at 47 F
1.8 at 13 F
Cooling capacity: 24,000 Btu/h
Cooling SEER: 8.1

Cost:	Heat pump	$1450	
	Installation*	400	
	Total	$1850	(Provided by a local contractor)

SYSTEM 2: High Efficiency air-air system

Heating capacity: 23,000 Btu/h
Heating COP: 3.0 at 47 F
2.2 at 13 F
Cooling capacity: 23,400 Btu/h
Cooling SEER: 10.1

Cost:	Heat pump	$1950	
	Installation*	400	
	Total	$2350	(Provided by a local contractor)

SYSTEM 3: Ground Coupled Heat Pump

Heating capacity: 30,000 Btu/h
Heating COP: 3.7 at 70°F E.W.T.

Cooling Capacity: 24,000 Btu/h
Cooling EER: 9.8 at 85°F E.W.T.

Cost:			
	Heat pump	$2077	
	Water pump module	$ 604	
	Ground Coil	$1200	(including trenching and installation)
	Installation*	$ 310	(heat pump and water pump)
	Total	$4191	(Provided by installing contractor)

*For retrofit with ducts, wiring, and thermostat already in place.

The two particular air-air systems were selected because similar, although slightly larger systems from the same contractor have been tested at TECH House III[11] and the actual measured seasonal performance factors are comparable for comparison purposes with those from the ground coupled heat pump test. A comparison of seasonal performance factors of the three systems is included in Table 6.1. Since each test was carried out over a different season, degree days based on 65°F are also listed. It was assumed that air-air system seasonal performance factor would not vary significantly with degree days in the range over which the three units were tested.

Table 6.1. Seasonal Performance Factor Comparison

SYSTEM	HEATING		COOLING	
	SPF	°F Days	SPF	°F Days
Std, A-A	1.56	3457	1.64	1612
Hi Eff. A-A	1.99	3000	2.27	1820
GCHP	2.84	2338*	2.59	765

*includes only January, February, and March

The annual heating and cooling loads used in the economic analysis of each system are 7,800 kWh and 4,200 kWh respectively. These are estimated full season values for TECH House V and are higher than the loads reported elsewhere because of the fact that there were several hours during the season for which the data were not recorded. A constant electricity cost of $0.05/kWh was assumed, which is typical for the Knoxville area.

A comparison of the standard air-air heat pump with the ground coupled unit shows an annual energy savings of 3,193 kWh* at an increased initial cost of $2341 (i.e. $4191-$1850) which gives a simple payback of 15 years (i.e. $2341/(3193 kWh x $.05/kWh)). A similar comparison with the high efficiency air-air unit gives a payback period of 25 years.

Although the ground-coupled heat pump does not appear to be economically feasible for the installation reported herein, there are several factors that could alter, perhaps significantly, the reported economic viability of the ground-coupled system. Some of the more significant ones are discussed below:

1. Operation in a Colder Climate - the heating mode performance of an air-air heat pump is more sensitive to ambient temperature than is the ground coupled system. Consequently, operation in a colder climate should offer an additional performance advantage for the ground-coupled system.

$$*\frac{7800\ \text{KWh}}{1.56} - \frac{7800\ \text{KWh}}{2.84} + \frac{4200\ \text{KWh}}{1.64} - \frac{4200\ \text{KWh}}{2.59} = 3193\ \text{KWh}$$

2. Electricity Cost - The Knoxville, TN area enjoys electric rates below the national average. Increased electric rates should favor the ground coupled system.
3. Utility Demand Charge - Due to the popularity of the air-air heat pump, some utilities are considering a residential peak demand charge. Figure 5.8 shows that there could be up to an 8 kW reduction in winter peak load for the GCHP system over a typical air-air system. An estimated peak charge of $6/kW (currently the lowest rate for Knoxville commercial power customers[12]) applied over a 4 month heating season could produce additional savings of $192 per year. Institution of a demand charge based on conditions listed here and the exclusion of other factors could reduce the payback period of a GCHP system to about 7 years compared to the standard air-air system.
4. Oversized Ground Coupled Heat Pump - The cost of this unit was higher than necessary due to its being oversized. However, it would be necessary to evaluate the performance of a correctly sized unit before any conclusions regarding savings can be made. As discussed in Chapter 4, the increased runtime that would result from a smaller unit could produce a reduced performance factor.

Chapter 7. Conculsions and Recommendations

From hourly data obtained over the 1984-85 heating season and the 1985 cooling season, the performance of the GCHP systems in TECH Houses I and V is reported. Both GCHP systems performed well enough to meet all the required loads without a need for auxiliary backup heat during the winter months.

Seasonal performance factors of the GCHP system in TECH House I were found to be 2.23 and 1.51 for heating and cooling respectively, while corresponding values for the GCHP system in TECH House V were 2.84 and 2.59. There were three primary differences between the two GCHP systems that would account for the difference in performance:

1. A comparison of the heat pump units, exclusive of the ground coils, showed the unit in TECH House V to have a 6 percent higher performance factor in the heating mode and a 35 percent higher performance factor in the cooling mode than the unit in TECH House I. These results were based on steady state tests with once-through city water replacing the ground coil heat exchanger.
2. The ground coil in TECH House V is larger, 570 feet per ton of installed cooling capacity at 85°F entering water temperature compared to 340 feet per ton for the system in TECH House I. During the heating season, however, both units produced seasonal performance factors of about 90 percent of the ideal values (Table 4.2) (using previous seasons for TECH House I with clay backfill in the trench). For the cooling season, however, the TECH House V system produced a seasonal performance factor of 93 percent of the

ideal value, while the TECH House I system, over 3 seasons, produced an average seasonal performance factor of about 65 percent of the ideal value.

3. The ground coil of the TECH House I system is buried 4 feet deep and backfilled with sand while the TECH House V system is buried at depths of 4 feet and 6 feet and is backfilled with clay. Although the cooling performance in TECH House I has increased in 1984 and 1985 with the sand backfill over the level in 1983 (with clay backfill), it should also be noted that there is a significant decrease in degree-days corresponding to the performance increase. It is therefore not possible to determine for certain the role of the sand in improving summer performance. During the winter of 1984-85, however, it appears evident that the sand is responsible for the significant decrease in performance due to its inability to rehydrate to a saturated condition during the winter months as does the clay.

As a result of this study, the following general conclusions are offered.

1. A well-designed ground coupled heat pump system (with the ground coil sized for summer design conditions) offers a significant performance improvement over even a high efficiency air-air unit. However, it is not likely to become economically attractive in the Knoxville, TN area until the utilities institute some type of peak demand charges or other consumer incentives.
2. Under the conditions tested herein, sand does not appear to be an appropriate backfill material.

3. Attempts to improve summer performance of a GCHP system by introducing moisture into the ground were reasonably ineffective due to rapid drying of the soil adjacent to the pipe. Consequently, if water is to be added, it should be supplied on a continuous basis.

Additional research in two areas of GCHP technology could be fruitful, therefore the following recommendations are made:

1. Determine the overall savings to be obtained when waste heat from the heat pump is used to preheat domestic hot water during the cooling season.
2. Continue to operate one or both of these GCHP systems over several seasons leaving the ground coil undisturbed in order to determine the effect of soil and pipe "aging" in a region of the country where cooling loads are significant.

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Appendix A. Project Chronology November 1, 1984 - October 31, 1985

A discussion of the project history from the inception through October 31, 1985, is presented in detail in the preceding reports in this series, 1, 5, 6.

In preparation for the 1984-85 heating season, heat balances on the GCHP system in TECH House I indicated problems with the water side heat flows. Upon examination of the instrumentation, AC noise was detected on the brine RTDs which measure the ground coil inlet and exit temperatures. After working throughout the first half of the 1984-85 heating season in an attempt to correct the problem, ORNL personnel pinpointed the source of the A-C noise during the first week of January, 1985 and promptly corrected it. From this point on, the ground coil heat flow for TECH House I is believed to be accurate. However, heat balances on the TECH House I system after the noise problem was corrected indicated there still existed a problem with the water-to-air coil heat flow. UT-EERC personnel, in conjunction with ORNL personnel, worked throughout the remainder of the winter in an attempt to remedy the problem.

At the close of the 1984-85 heating season, ORNL personnel completely disassembled and rebuilt the heat-power multiplication unit and other various instrumentation items. During this time period, UT-EERC personnel thoroughly inspected the GCHP system in TECH House I. Upon inspection, two leaking three-way valves were discovered. The three-way valves, which control the mode of operation of the heat pump system, were allowing an unknown quantity of fluid in the water-to-air

coil loop and the ground coil loop to mix. UT-EERC personnel promptly repaired the GCHP system by installing two-way valves on each side of the three-way valves. In an attempt to quantify how much the leakage affected the system performance, the GCHP was locked on in the cooling mode and allowed to run for several hours. During this time period, the newly installed two-way valves were periodically opened and closed thus the leakage of the three-way valves could be controlled. Table A.1 presents the results of the performance test.

Table A.1. Performance Variation

Hour	PF_i	Two-way Valves
1	1.43	closed
2	1.40	open
3	1.39	open
4	1.39	closed

Results of the tests indicate that the leakage of the three-way valves had a negligible effect on the overall system performance for TECH House I.

The GCHP system in TECH House V was installed on November 12-13, 1984 by GEOsystems, Inc., and the Charles Machine Works, Inc. During the remainder of November and the month of December, 1984, the instrumentation for the GCHP system in TECH House V was installed and debugged. Air side and ground coil heat flows were calibrated on January 3, 1985 and the GCHP system in TECH House V began formal operation in the heating mode utilizing the ground coil for heat absorption.

With the exception of two minor malfunctions, the GCHP system in TECH House V operated throughout the months of January, February, and

March, 1985. During the early morning hours of February 13, 1985, the GCHP system in TECH House V shut off. A subsequent check of the data revealed low temperatures in the ground coil loop which caused the freeze-stat to shut the system down. On March 1, 1985, the GCHP system in TECH House V shut down due to an air lock in the ground coil line. The ground coil line was promptly purged of air and operation of the GCHP system continued. Operation of the GCHP systems in the heating mode was terminated on March 31, 1985.

Prior to the 1985 cooling season, both GCHP systems were calibrated and cooling mode operation began on June 6, 1985 utilizing the ground coil for heat rejection.

Operation of both GCHP systems continued throughout the 1985 cooling season without system failure. Periodic calibrations of the heat flow sensors were conducted during the cooling season and results indicated that the data constants had changed by insignificant amounts.

The soil near the TECH House I ground coil was observed to have become dried out during the month of August, 1985. On August 12, 1985, the ground coil field for TECH House I was saturated with approximately 21,000 gallons of water. The water was added to the coil field through the soaker pipe which was installed during the ground coil reinstallation prior to the 1984 cooling season. During the remainder of August and the month of September, 1985, the soil moisture was closely monitored and maintained at 80% saturation.

Operation of both GCHP systems in the cooling mode was concluded on September 30, 1985.

Appendix B. Ground Coil Characteristics

SYSTEM	MATERIAL	ID (inches)	OD (inches)	THERMAL CONDUCTIVITY (Btuh/ft-°f)	CIRCULATION FLUID
TECH HOUSE I	polyethylene	1.38	1.66	0.27	methanol-water (20% methanol by weight
TECH HOUSE V	polybutylene	1.61	1.82	0.13	water*

* For the Knoxville, TN area, GEOsystem design data recommends that the ground coil be charged with a propylene glycol-water solution (10% by volume). However, for this study, the ground coil was charged only with water for ease of property evaluation.

Appendix C. Instrumentation Listing

Table C.1. TECH House I GCHP System Measured Parameters

* * * ANALOG SIGNALS * * *

Probe Designation	Measurement
SA1, SA2	Soil temperatures near 4 foot and 2 foot pipes at the beginning of the ground coil.
FB1, FB2	Brine temperatures within 4 foot and 2 foot ground coils at 25% of coil length.
SB1-SB10	Soil temperatures at 25% of coil length.
FC1, FC2	Brine temperatures within 4 foot and 2 foot ground coils at 50% of coil length.
SC1-SC16	Soil temperatures at 50% of coil length.
FD1, FD2	Brine temperatures within 4 foot and 2 foot ground coils at 75% of coil length.
SD1-SD9	Soil temperatures at 75% of coil length.
SE1, SE2	Soil temperatures near 4 foot and 2 foot pipes at the end of the ground coil.
ST2, ST3 ST4, ST10	Far field temperatures at depths of 2, 3, 4, and 10 feet.

Table C.1 (Cont'd)

* * * DIGITAL SIGNALS * * *

GRND - Heat Flow into/from the Ground
WTAC - Water-to-air Coil
Cond - Pounds of Condensate
HTP - Heat Pump Compressor
BLWR - Supply Air Blower
Phtg - Circulation Pumps
SOLR - TECH House I Total Electric Power
RUNT - Runtime

* * * OTHER MEASUREMENTS * * *

$GRND_i$: Temperature of brine entering the ground coil
$GRND_o$: Temperature of brine exiting the ground coil
HDRY: Space dry bulb temperature
HWET: Space wet bulb temperature
FAN_i: Temperature of brine entering the water-to-air coil
FAN_o: Temperature of brine exiting the water-to-air coil
HEXG: Temperature of brine entering the brine-to-domestic hot water heat exchanger

* * * MOISTURE MEASUREMENTS * * *

Probe Designation	Measurement
33	Soil moisture sensors located near beginning of ground coil.
41 43	Soil moisture sensors located at 25% of coil length.
37 38 39 40 42	Soil moisture sensors located at 50% of coil length.
32	Soil moisture sensor located at 75% of coil length.
51 52	Soil moisture sensors located near end of ground coil.

Table C.2. TECH House V GCHP System Measured Parameters

* * * DIGITAL SIGNALS * * *

GRND - Heat flow into/from the ground
ASHF - Air side heat flow
COND - Pounds of condensate
TSP - Total purchased electric power consumed by GCHP system
TWPR - Total purchased electric power for house

* * * OTHER MEASURES * * *

$GRND_{in}$: Temperature of fluid entering the ground coil
$GRND_{out}$: Temperature of fluid exiting the ground coil
HDRY: Conditioned space dry bulb temperature
HWET: Conditioned space wet bulb temperature

* * * MOISTURE MEASUREMENT * * *

1 - Soil moisture sensor at depth of ground coil, located near midpoint of ground coil length.

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