To the Graduate Council:

I am submitting herewith a thesis written by Francis Conlin entitled "Analysis of a Ground-Coupled Heat Pump During a Heating Season." I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Engineering Science.

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ANALYSIS OF A GROUND-COUPLED HEAT PUMP DURING A HEATING SEASON

A Thesis

Presented For The

Master Of Science

Degree

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ABSTRACT

A ground-coupled heat pump, installed in an 170 square meter residential building in Knox County Tennessee, has been examined both experimentally and analytically. The ground coupling was realized with a 210 meter coil buried 1.2 meters beneath the soil, through which circulated a secondary heat transfer fluid. The house, the ground and the power consuming devices were instrumented, and hourly readings of temperatures, heat flows, power consumption, and weather data were recorded.

The experiment was carried out during the 1982-1983 heating season begining in November and lasting 26 weeks. The overall heating seasonal performance factor was 2.57. No backup heat was required throughout the duration of the experiment.

An analysis with the TRNSYS and GROCS computer programs was also undertaken. These programs were specifically modified to model the heating system, and the ground heat exchanger. The model prediction of overall seasonal performance factor was within 5% of the measured values. The effect of varying several parameters such as ground coil length, soil thermal properties, and the heat pump performance curve, was examined. The model was found to be relatively insensitive to small variations in thermal

properties of the soil. The maximum seasonal performance factor for the Knoxville area was predicted to be 2.85 using the maximum estimated thermal conductivity, twice the original coil length, and the observed heat pump performance curve for the system.

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CHAPTER I

INTRODUCTION

Residential heating is responsible for a large fraction of the total energy consumed in many homes today. The heat pump, because of its ability to transfer more energy as heat than the amount of energy required for its operation, provides one of the more popular methods for heating. The ability of the heat pump to serve as a cooling unit increases its popularity even more.

Heat Pump Classifications

Heat pump systems may be classified by the type of heat source employed. Most systems currently use air as a heat source and use a forced convection refrigerant-to-air coil to provide the heat transfer from the outside air to the heat pump. Other types of heat sources include bodies of water which use a heat exchanger designed for natural convection heat transfer, and the ground which uses a heat exchanger designed for conduction heat transfer (1).

The heat pump employing forced convection refrigerant-to-air heat exchanger coils both inside and outside the conditioned space, often refered to as an air-to-air heat pump, is by far the most common heat pump on the market today. However, this popularity is due largely the ease of manufacturing and of installation of the air-to-air system, as several alternative types of heat pump systems demonstrate better performance.

Ground-Coupled Heat Pumps

The ground-coupled heat pump is a classification of heat pumps with its environmental heat exchanger located in the soil, and it is one of the heat pump options with several performance benefits over the air-to-air system. The ground-coupled heat pump is not subject to poor performance during periods of low ambient air temperatures, as is the air-to-air heat pump, and thus minimizes the use of resistance backup heat. In addition to decreasing the overall system performance, resistance backup heat required by air-to-air heat pumps can add significantly to a utility's peak load requirements. Also, there is no need for a performance damaging defrost cycle for the earth coupled heat pump as there is in the air-to-air system. Studies have shown that the inherent problems with the air-to-air heat pump result in an average overall heating season performance factor of 1.5 for the East Tennessee region (2). A properly designed residential ground-coupled heat pump system, however, can expect to have a heating season performance factor of near 3.0 (3). (The heating seasonal performance factor is defined as the amount of energy delivered as heat divided by the amount of energy that costs for the entire heating season.)

One reason the ground-coupled heat pump has not been as warmly recieved as the air-to-air heat pump is because of the additional expense involved in the installation of the ground coil. A second reason is the slow advance of design criteria for earth coils that would enable a dependable application. However, rising heating costs and increased design capabilities serve to make the ground-coupled heat pump increasingly attractive.

Several designs of heat exchanger coils are availiable for the ground-coil of the ground-coupled heat pump. These include both open-loop and closed-loop systems. The open-loop system is for use below the water table, and in this design, ground water is pumped through the heat pump heat exchanger and is then returned to the ground. The closed-loop system has a more general application and is usually a long coil buried beneath the soil in any variety of heat exchanging designs. The geometry of the ground coils is refered to as horizontal if the coils are parallel to the ground surface and as vertical if the coils are perpendicular to the surface.

Study Objective

This study concerns a ground-coupled heat pump employing a closed-loop horizontal earth coil which was installed at the Tennessee Energy Conservation in Housing (TECH) facility located on Alcoa Highway in Knox County

Tennessee. The TECH facility is a joint research venture operated by the Tennessee Valley Authority, the Department of Energy, and the University of Tennessee. The design of the heat pump system was provided by Battelle-Columbus Laboratories and specified a 3.0 centimeter inside diameter polybutylene coil 210 meters in length buried to a depth of 1.2 meters (4). This heat pump system was installed in TECH house I and was used to provide heat for the 1982-1983 heating season.

The first objective of this study is to examine experimentally the performance of a horizontal ground-coupled heat pump in the East Tennessee region. The second objective is to make analytical predictions using an available simulation model and to compare these predictions with the experimental results. Parametric studies have been performed with the program GROCS to explore the forcasting ability and sensitivities of the modeling to particular variables, and to predict the maximum performance obtainable within the range of these variables.

CHAPTER II

LITERATURE REVIEW

Initial Heat Pumps 1850-1945

The idea of the heat pump has been around since 1852 when William Thomson (Later to become Lord Kelvin) proposed an open cycle air-to-air "heat multiplier" which could be used for both the heating and cooling of buildings (5). In the years to follow, much progress was made with the refrigerating application of this idea, which met an established need. Interest in the heating application, however, was slight due to low cost heat generators already available. It is commonly accepted that the first heating application of the heat pump occured when Haldane constructed an experimental heat pump in his home in Scotland in the mid 1920's (6). Haldane analyzed data from a selection of refrigerating plants to obtain background for his design. His first heat pump used ammonia as a working fluid and used a combination of air and municipal water as a heat source. The heat was distributed by low temperature hot water radiators, and the system had a reported efficiency of 200 to 300%. Haldane's success inspired many of the design studies and demonstrations which were to follow.

Over the next 20 years several large output heat pumps were built. Larger plants were more attractive than smaller plants because their cost related to the heat output is lower, and also a greater opportunity exists to improve the performance (7). A major development was made in heat pump technology when in the early 40's the first of the modern halocarbon refrigerants, R-12, was made available.

The Swiss made an early alliance with the heat pump, producing 10 large plants from 1938-1945 yeilding outputs from 58 to 7000 kilowatts (8). A review of many important early heat pump applications in the United Kingdom can be found by Heap (9), and the principal early applications in the United States are examined by Kemler (10). These early heat pumps span the period from 1922 to 1945, and use water and air as heat sources.

Ground-Coupled Heat Pumps

By the mid 1940's air and water source heat pumps had been demonstrated as valuable additions to heating technology. The ground source heat pump, however, took longer to develop. Sporn (11) points out in 1947 the potential of using the earth as a heat source, but he also notes the lack of data available describing the conduction of heat to and from the soil. The data that was available came from water, gas, and oil lines and from electrical conduits that were buried beneath the ground. This information was somewhat useful in describing a ground coil as a condenser, but virtually no data was available that could be used to describe the ground coil as an evaporator.

Ingersol (12) was the pioneer of much of the analytical work on earth heat exchangers. His application of the constant heat rate line source solution to the unsteady conduction equation provided a basis for future study and experimentation using earth as a heat source. Hadley (13) Guernsy (14) and Smith (15) augmented the knowledge in this field by devising design theory for heating ground coils and encouraged the experimentation with earth source heat pumps. The disscussion of both the Guernsy and Smith papers reflect valuable knowledge gained from numerous ground-coupled heat pump applications, especially from 112 heat pumps installed in the Portland, Oregon area between 1945 and 1949. Many of these Oregon heat pumps failed due to improper coil length or due to damage by freezing. Ingersol later analyzes a number of ground coil configurations including a flat plate, a star, a horizontal pipe grid, a vertical pipe grid, and a spherical cavity (16). He concludes that the isolated pipe is superior to the other configurations as an earth heat exchanger.

More complex applications of ground-coupled heat pumps began to appear in the period after 1945. Baker (17) investigated a ground-coupled heat pump using both the earth as a heat sink and a 4000 liter tank of water for thermal storage, and time delay switching which allowed the circulation pumps to operate longer than the compressor. A near constant monthly heating energy ratio of 3 was reported for the 1950-1951 winter. Heat flows, as well as run times and power consumptions, were recorded.

The results of earth-coupled heat pumps from 28 institutions were correlated and information about their investigations were tabulated by the Edison Electric Institute (18). This survey cites investigations from 1945-1953 and includes coil data, heat pump equipment, test durations, heat flows, soil temperatures, and a breif summary of results and conclusions from those studies. Although much progress was shown, the AEIC-EEI heat pump committee concluded that "even with this considerable amount of information, there is still no apparently workable design equation which can be used by the individual contractor."

During this time period, data regarding earth coupled heat pumps began to multiply. With this wealth of data grew much analyitical examination of soil properties, especially those properties having to do with the influence of moisture on the thermal conductivity of soils and with the phenomenon of moisture migration in a porus medium, such as soil, towards a cold object (19,20,21,22,23). Thermal conductiviy is a function of soil type, density and moisture content.

Therefore, the effect of heat addition in the soil, and other factors which influence moisture have particular consequences for the thermal conductivity of the soil and thus the performance of the ground-coupled heat pump. Soil thermal conductivity was shown to change 4 to 10 times depending on the moisture content and type of soil (19). The effects of freezing around the coil during the heating operation and the benificial capture of latent heat is also reviewed (10, 14, 17). The latent heat can provide more than 80% of the available heat in cooling soil from 10C to 1C (14).

Vestal (24) in 1957 performed an extensive research program exploring the earth as both a heat sink and heat source and presented an extensive experimental description. Vestal completed 57 tests at his buried coil installation. The coils were examined both as condensers and as evaporators. The tests operated from 5 to 90 days duration, and a variety of conditions were explored. Records are available on runtimes, fluid temperatures, Reynolds numbers, soil thermal conductivities, heat flows and volume of frozen soil for each test. Vestal applied a dimensional analysis to correlate these data and to establish a design criteria for pipe length. Penrod (25) also contributed to the ground-source heat pump science during this time period. Several papers by Penrod disscussing design procedures and analysis of earth heat pumps coupled with solar heat collectors and related studies were produced at the Kentucky Engineering Experiment Station. Equations are published for determining the size or capacity of principal system components.

Current Studies

Due to the high availability of other low cost heat generators in the 1950's and 1960's, much of the work with ground-coupled heat pumps was abandoned. However, recent interest in this subject is increasing due to the rising cost of energy. Several advancements in technology by the mid 1970's aided the resurgent interest in ground-coupled heat pumps. The use of plastic pipes and the use of motorized trenchers and drilling machines for burying the pipe have simplified and economized the installation of the ground coil. The use of the computer has vastly extended the analyitic ability from graphical and empirical design procedures. Numerical techniques for ground heat transfer calculations are available for ground coil design. Design methodologies from thirty-six United States' and European countries are reviewed by Battelle (3). These vary in complexity from rules of thumb to multi-dimensional transient finite element and finite difference models, some of which include moisture migration and freezing considerations.

A significant contribution both to the science and to the application of earth-coupled heat pumps in the United States is being made by the Oklahoma State University Heat Pump Research Laboratory. A 1982 report summarizes the work being done at that facility (26). Investigations with 6 different experimental earth coils and two full scale solar systems have provided much data to support earth-coupled applications in that area. Data collected includes heat flows, and both fluid and soil temperatures. Different pipe materials, antifreeze solutions and corrosion inhibitors have been tested for many arrangements of horizontal and vertical ground coils. In addition, several brands of water source heat pumps have been investigated and performances have been reported from 2.6 to 3.75. Another study in Stillwater which is associated with the University, is supported by Oklahoma Gas and Electric company and the Electric Power Research Institute. This study compares three identical residences each with different heating systems including water source heat pumps employing a vertical ground coil alone and in combination with a solar system. The work at Oklahoma State University enjoys the support of several local manufacturers of products and services for ground-coupled heat pump application, and the

University has developed a series of design briefings, service training courses, and installation seminars which support the local demand for the product.

Brookhaven National Laboratory has been involved in ground-coupled heat pump research since 1979. A paper by Metz outlines the basic ground-coupled research hardware which includes 4 buried tanks, 6 earth coils, and 6 sealed vertical wells (27). One coil is used to heat a small, well insulated house with a design heat rate of 3.4 KW (at -12C). In another report Metz summarizes the performance of these systems (28). Data for this research venture is reported as monthly averages for heat flows, fluid and ground temperatures, runtimes and ambient conditions. A performance of 2.3 is reported for the laboratory's location in Upton New York. Much of Brookhaven National Laboratory's experimental work has been used to validate a finite difference computer model of ground heat transfer. This model uses constant soil properties and a comparatively small number of large nodal volumes. Comparison of average long term brine temperatures show fairly good agreement with computed values from ground-coupling experiments.

A symposium on earth-coupled heat pumps held in Sweden in 1979 attracted participants from 9 countries as far away as Australia (29). The proceedings from this gathering suggests an optimistic future for the ground-coupled heat pump, and reports a variety of findings from these countries.

A survey by Ball (3), reveals that there exist today over 6000 heating only horizontal ground-coupled heat pump installations in Sweden alone, and unknown but significant numbers in other European countries. In addition, it is estimated that there are over 1000 ground-coupled installations active in the United States. An abrieviated list of current international ground-coupled heat pump studies and their performance characteristics is tabulated by Battelle (3). This table shows a seasonal performance factor ranging from 2.0 to 3.4 for 12 ground-coupled research laboratories.

Inspite of the progress of ground-coupled heat pump research and applications, further studies need to continue if a reliable and general design procedure is to develop. Concurrent research and application of ground coil heat exchangers will provide information to enable researchers to achieve this goal.

CHAPTER III

EXPERIMENTAL METHODS

Facilities

TECH house I was chosen among the houses of the TECH facility to incorporate the ground-coupled heat pump system. Design data and previous investigations with TECH house I show the design heating load to be 5.32 KW calculated at -10C outside and 20C inside (30). A Tetco water-to-water heat pump was installed for use in the study. Even though the unit was designed for heating only, space cooling for summer application is provided by the use of three-way valves which exchange the flow through the ground coil with the flow through the air side coil as seen in Figure 1.

Ground Coil

The horizontal coil geometry was determined largely by the availability of suitable ground surface area. Ideal conditions are a homogeneous soil in a flat unshaded area which would catch near-by drainage and that is of sufficient size to accommodate a coil 210 meters in length placed 1.2 meters below the surface while spaced no closer than 2 meters (4). The best approximation of these ideal conditions at the TECH site was determined and the coil layout was established as shown in Figure 2. The ground was excavated with a motorized chain trencher to an approximate

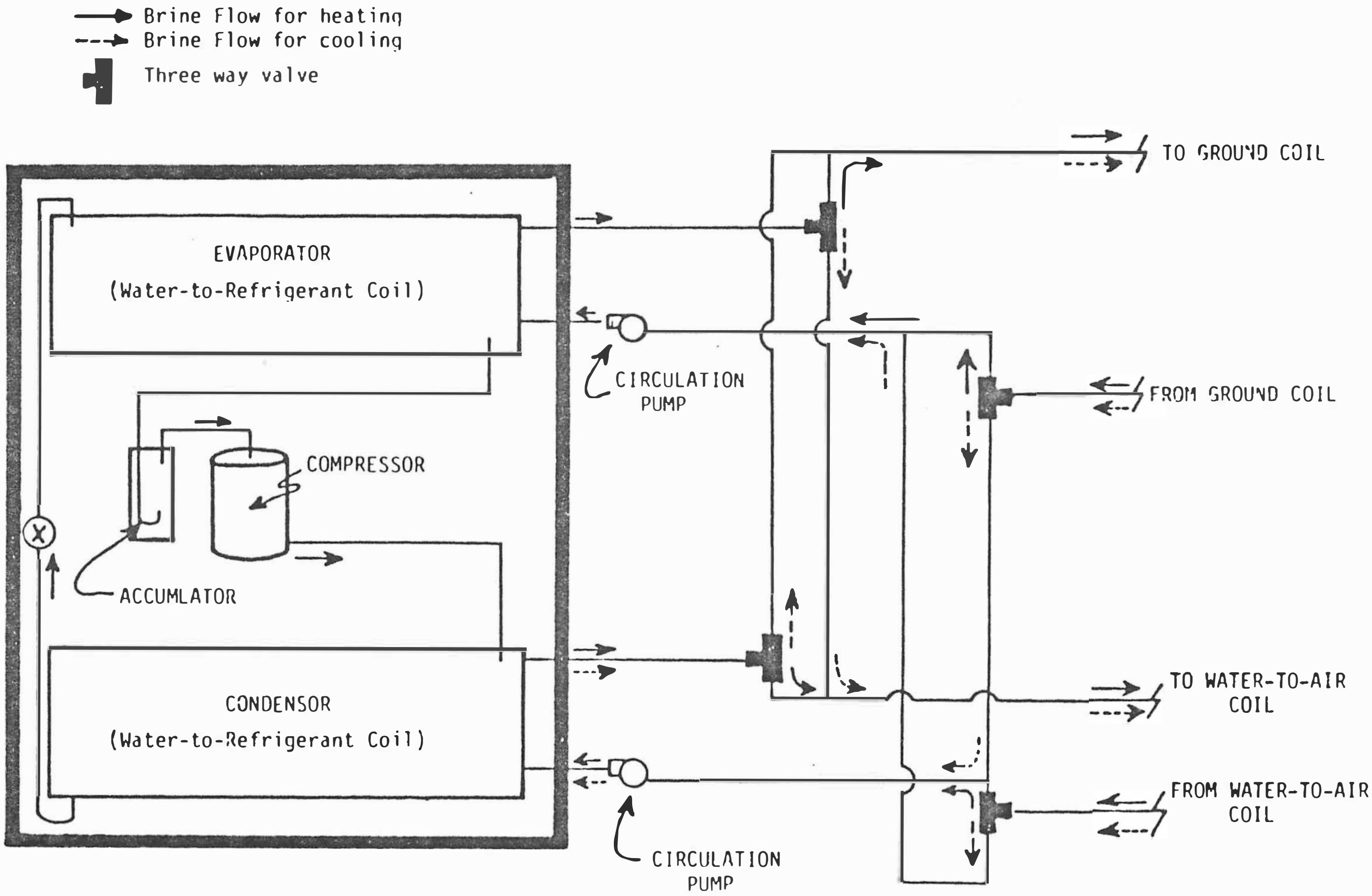


Figure 1. Schematic of Ground-Coupled Heat Pump

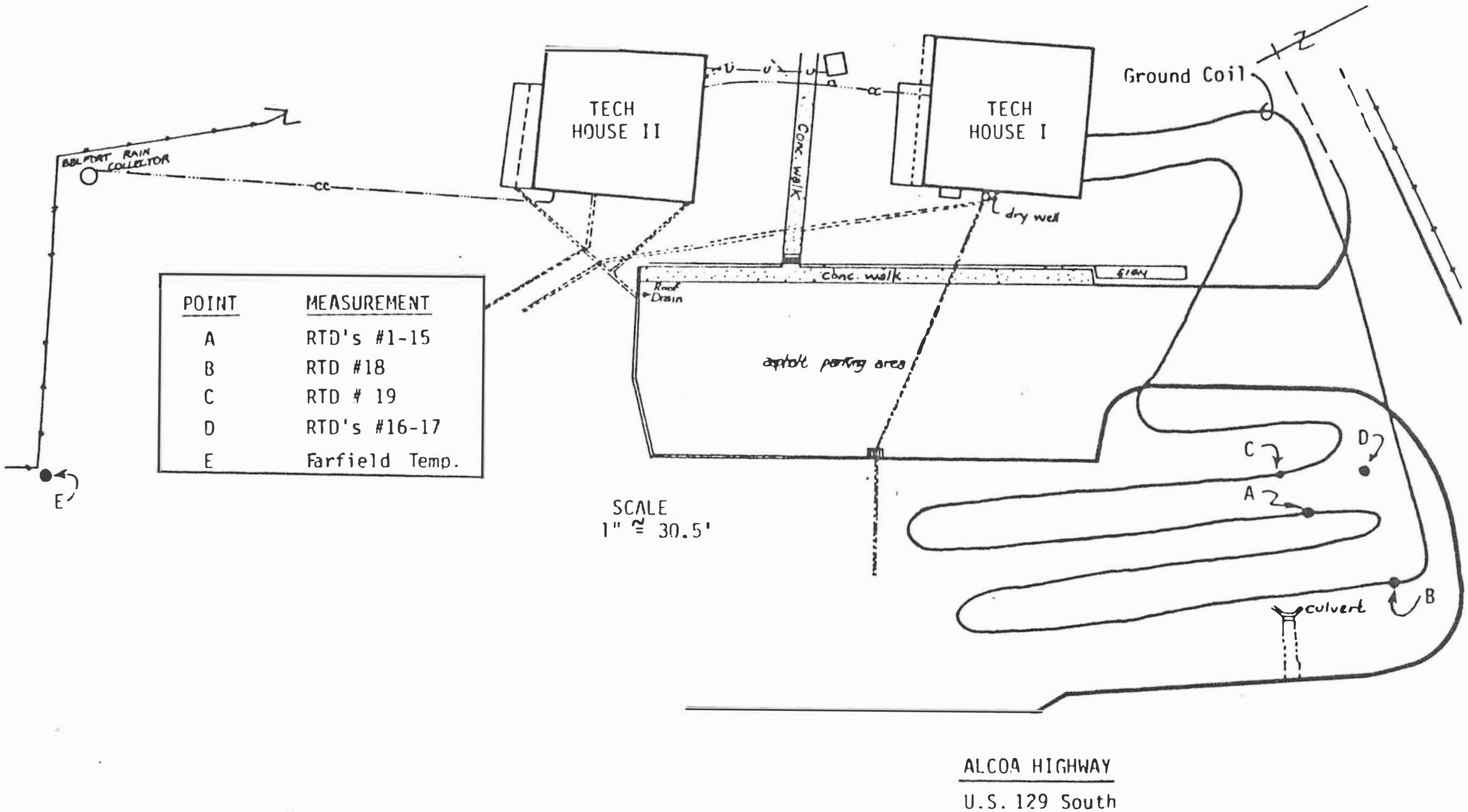


Figure 2. Plan View of Horizontal Ground Coil Facility

depth of 1.2 meters and 206 meters of nominal 3.0 centimeter diameter thick-walled polybutylene pipe was installed. The trench was then backfilled with the original material and tamped. The coil was filled with a 20% by weight methanol/water solution which is circulated throughout the loop with a Gorman-Rupp pump. The pump produces a flow rate of 36 liters per minute through the ground loop.

Instrumentation

The data recorded for the ground-coupled heat pump experiment include temperature, heat flows and power consumption measurements. Most of these data are recorded on magnetic tape with a Hewlet Packard data acquisition system (DAS) for each hour. The temperatures are recorded as instantaneous values, and all other measured quantities are recorded as sums over each hour.

The temperature at nineteen locations in the soil in the vicinity of the pipe were measured with a platinum resistance temperature device (RTD). Fourteen RTD's were located at approximately 50% of the pipe length and were secured in place by attaching them to a support. Six RTD's were located in a horizontal plane with the pipe, and up to a distance of one meter from the pipe. Eight of these were located in a vertical plane with the pipe from 0.75 meters below the pipe to 0.6 meters above the pipe. Figure 3 shows the profiles of the temperature sensors at 50% of the pipe

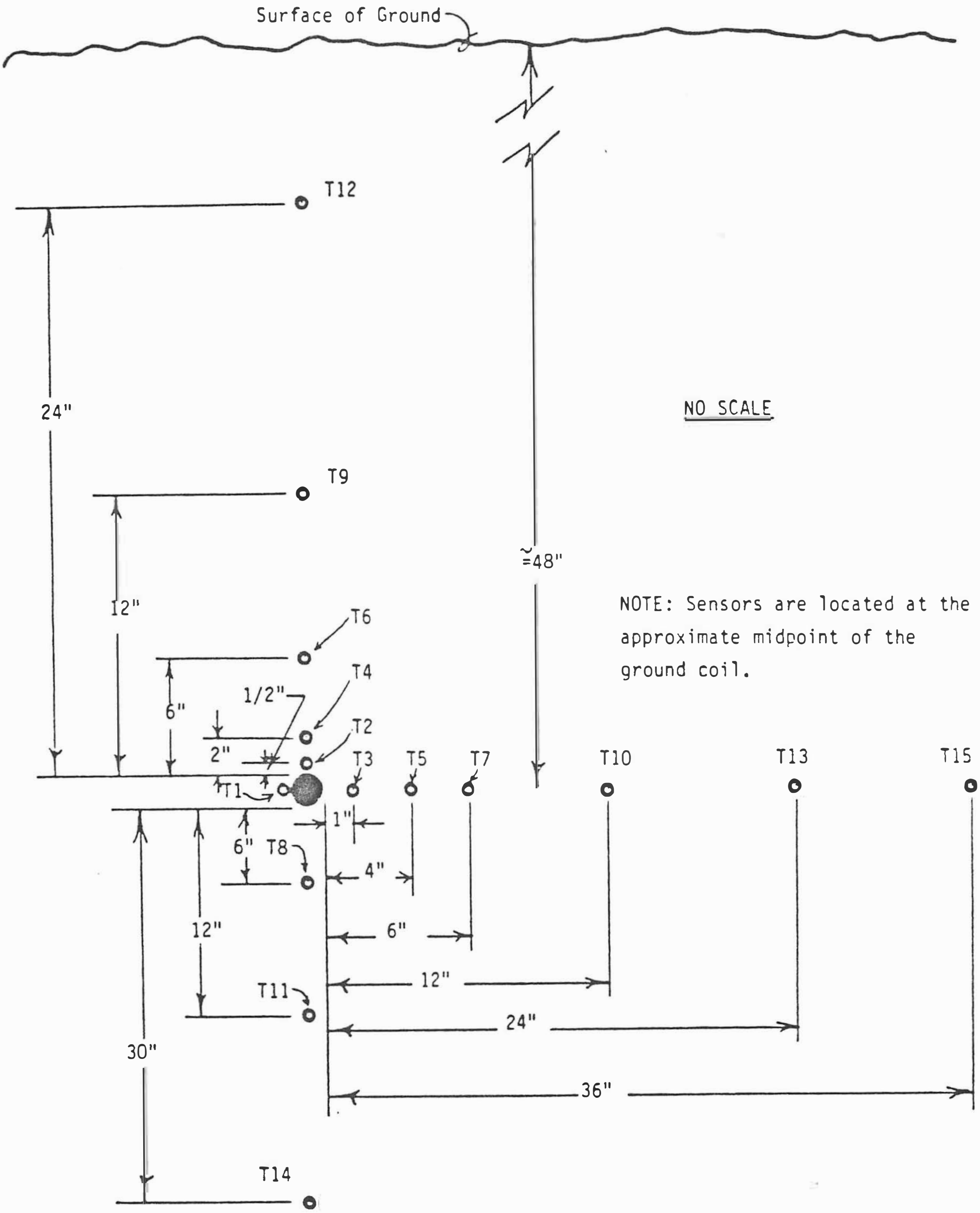


Figure 3. Location of Ground Temperature Probes

length. Three RTD's were secured to the pipe wall and insulated at different locations along the pipe length. One was located at 25% from the entrance of the supply line, one was located at 50%, and the final RTD was located at 75% of the pipe length from the entrance. Some of the temperature sensors within the soil at the 50% length were relocated during the first month of the heating season (November 24, 1982). This resulted in some discontinuities in the experimental description of the soil temperatures, but is of little other consequence.

The final two RTD's in the ground were placed in a hole drilled near the pipe field. One was buried at 3 meters from the surface and the other was buried at 5 centimeters from the surface. In addition the conditioned space wet bulb and dry bulb temperatures were measured via solid state temperature transducers.

A flow meter and two RTD's comprise the components for measuring the heat flow from the ground and to the conditioned space. The temperature difference from the supply and return of each coil is electronically multiplied by the flow and the specific heat constant and is then sent to the DAS to be summed and recorded.

Power consumption is measured by four standard utility power meters. Measured seperately are the power consumption of the heat pump compressor, the air blower, the circulation pumps and the total house power. Each of these standard meters is equipped with optical sensors which count the revolutions of the meter's eddy disc, and this value is sent to the DAS where it is converted and stored.

Gypsum soil moisture sensor blocks were located in the soil near the 50% pipe length area, however, these proved to be inadequate as described in the experimental observation section. Also after the experiment had begun it was determined that a far field temperature measurement was desirable. Ten thermocouples were buried at 0.3 meter increments at a location far removed from the pipe field. The soil moisuture and the far field temperatures were measured manually on a weekly basis.

Data from other instrumentation, although not installed particularly for this experiment, was available for the TECH house I and the site in general including weather and insolation data. For a more detailed desrciption of the instrumentation see Interim Report #2 (31).

Preliminary Operation

Preliminary operation of the heat pump was initiated in the summer of 1982 with the heat pump operating in the cooling mode. Many of the system's initial bugs were worked out during this time which provided for the smooth winter operation. However during this period it was realized that the air side coil was not well matched with the heat pump specifications. During some operating conditions, the heat pump was prone to build up a high refrigerant pressure and to force a protective shutdown. A better matched air coil was installed February 14 1982, - well into the heating season. This resulted in a discontinuous experiment. Since both the heat pump and the air side coil are considered as one unit in the experimental and in the analytical observations, this study essentially deals with two different heat pumps, each having different performance characteristics. The first heat pump system operated from November 1 - February 13, and the second system operated from February 14 - April 30. This is important to note here and it will be refered to in the observation sections when appropriate.

Chapter IV

EXPERIMENTAL OBSERVATIONS

The experiment operated throughout the 1982-1983 winter heating season (November 1 - April 30). During these 26 weeks, the heating system performed with no major difficulties and generated extensive amounts of data. To reduce the number of data points and to enable a manageable presentation, daily and weekly averages of data are often used in the following observations to describe particular phenomenon.

Soil Temperatures

One factor influencing the performance of the ground-coupled heat pump system is the thermal response of the soil to heat extraction by the coil. The daily average ground temperatures for different horizontal distances from the buried pipe spanning the heating season are shown on Figure 4. As expected, the temperature fluctuations of the soil are less prominent as distance from the pipe increases. In addition the temperatures increase with increasing distance from the pipe. Also of interest is the horizontal profiles from the ground temperature data. The profiles shown in Figure 5 represent the temperature profiles for two points in time separated by five days. Note that most of the significant temperature difference from the far field

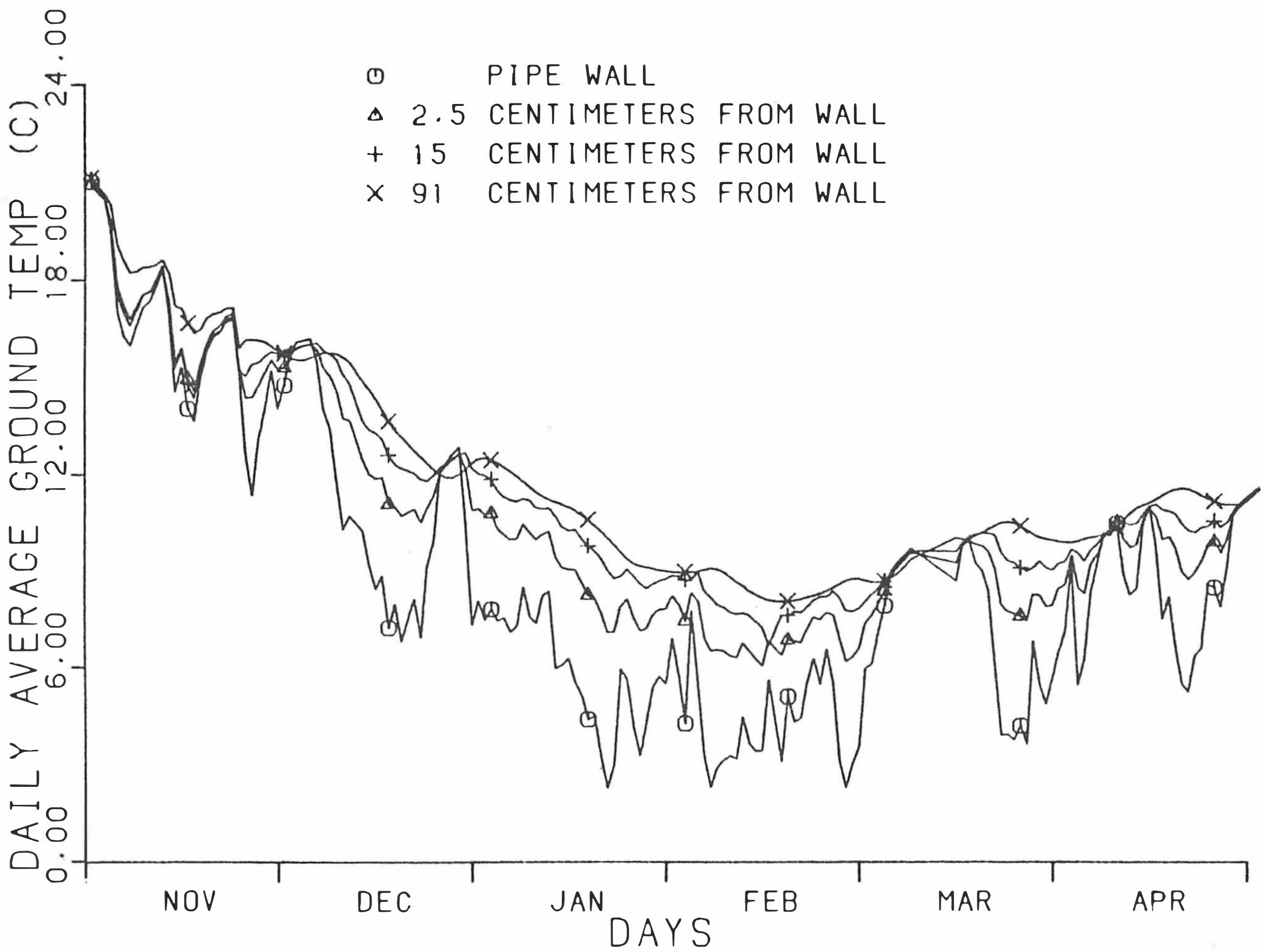


Figure 4. Average Horizontal Ground Temperatures

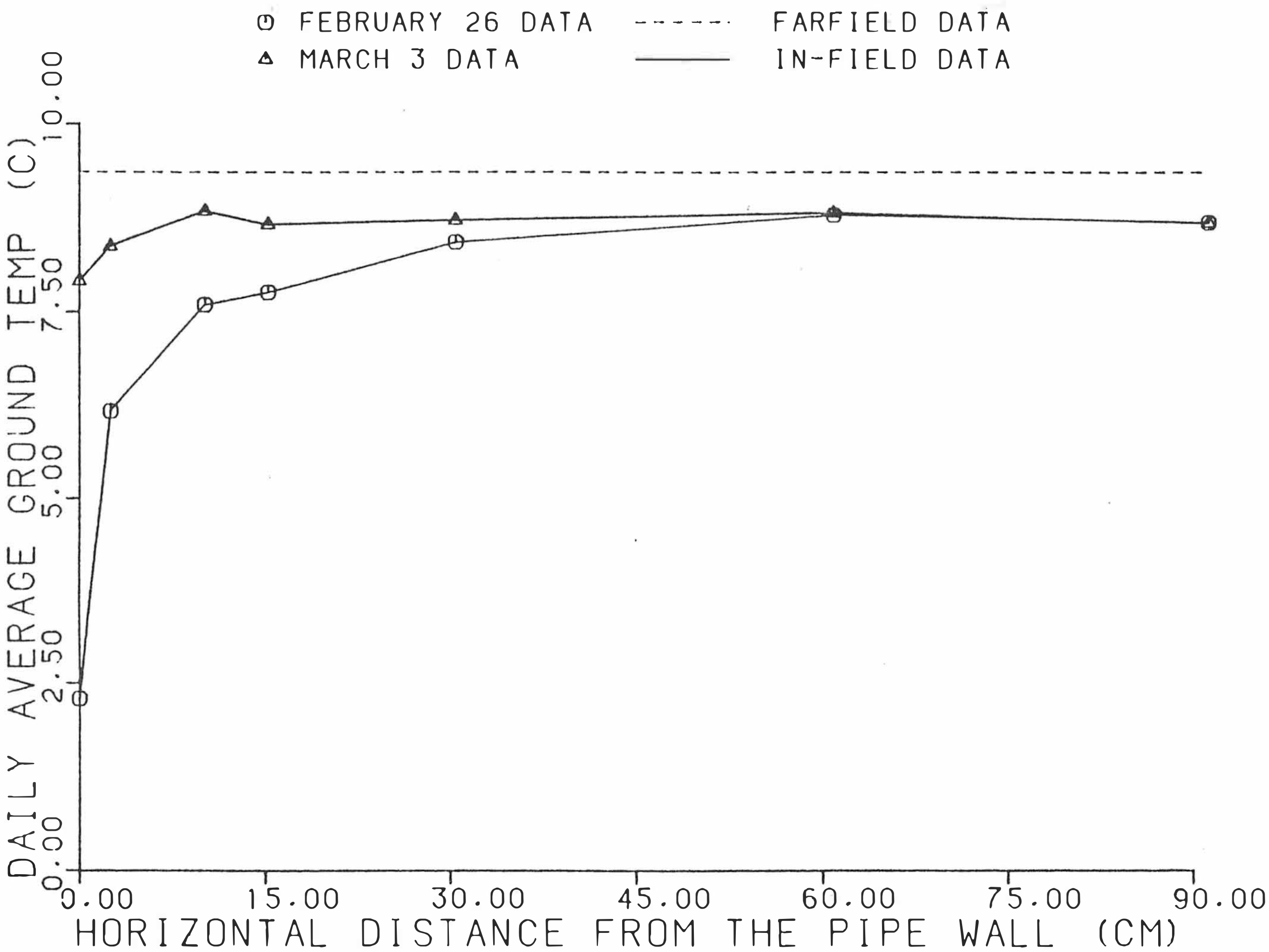


Figure 5. Horizontal Temperature Profiles

occurs within the first 0.15 meters from the pipe coil. Also note that in a relatively short time period, when the heating demand is low, that the soil temperature near the coil rises relatively rapidly to a point close to the far field temperature.

As distance from the influence of the ground coil is increased in the downward direction, the pattern displayed by the plot of soil temperature versus time resembles more closely the characteristic sine wave pattern describing far field earth temperature as a function of time of year and depth. The average daily horizontal soil temperatures are illustrated in Figure 6. The temperature histories within 0.15 meters of the pipe are left out of this Figure to enhance clarity, however, they are represented in the vertical temperature profile illustrated in Figure 7. This Figure demonstrates the vertical temperature profile for the same two days shown on Figure 5. The period of high heat extraction from the soil is illustrated by the sudden dive in temperature at the point near the location of the pipe. During a period of low heat extraction from the only 5 days later shows that the soil temperatures have significantly recovered, and the curve does not display a large temperature difference near the pipe coil. Also shown on this Figure, for reference, are the corresponding far field temperatures measured during the season.

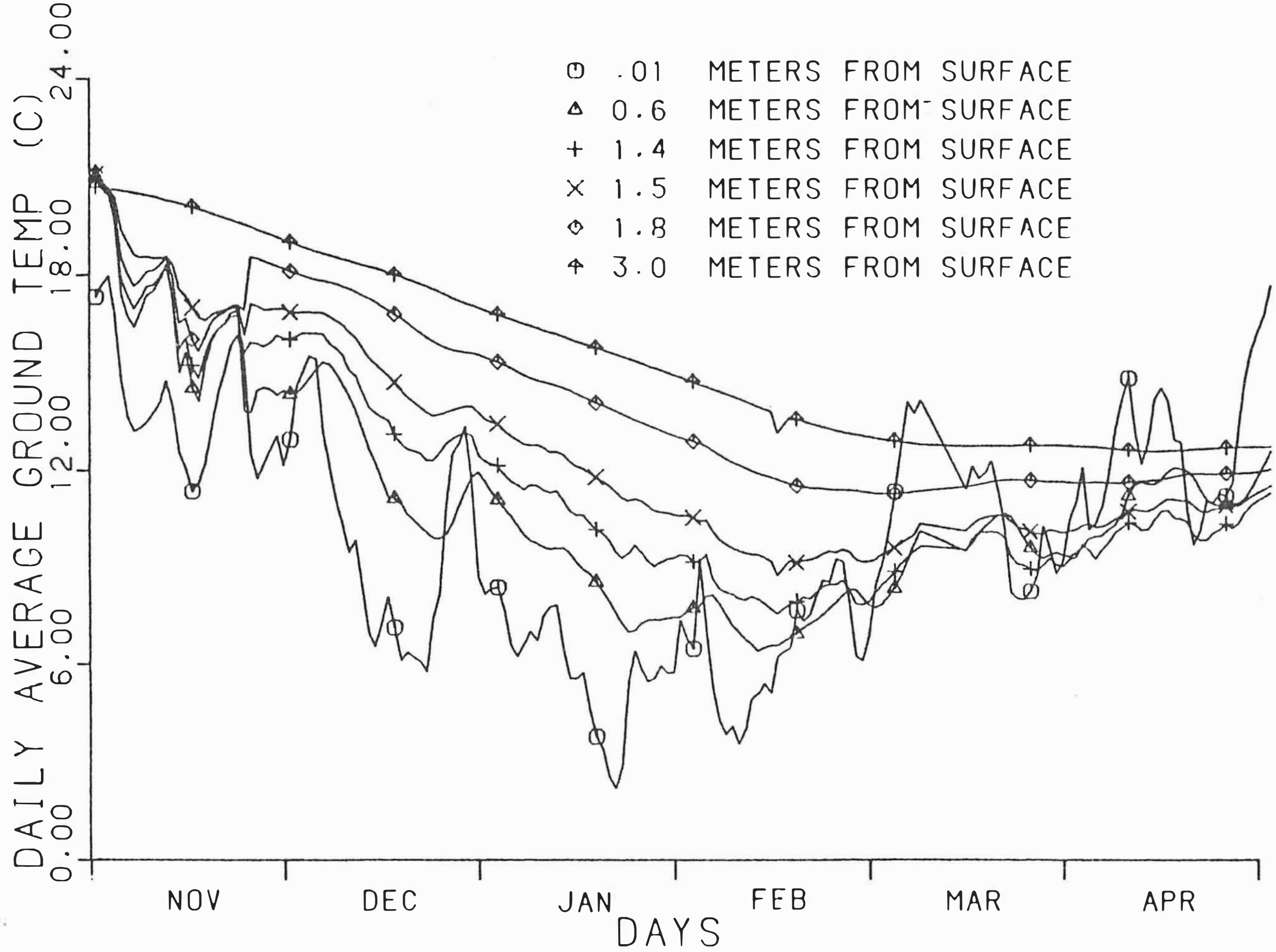


Figure 6. Average Vertical Ground Temperatures

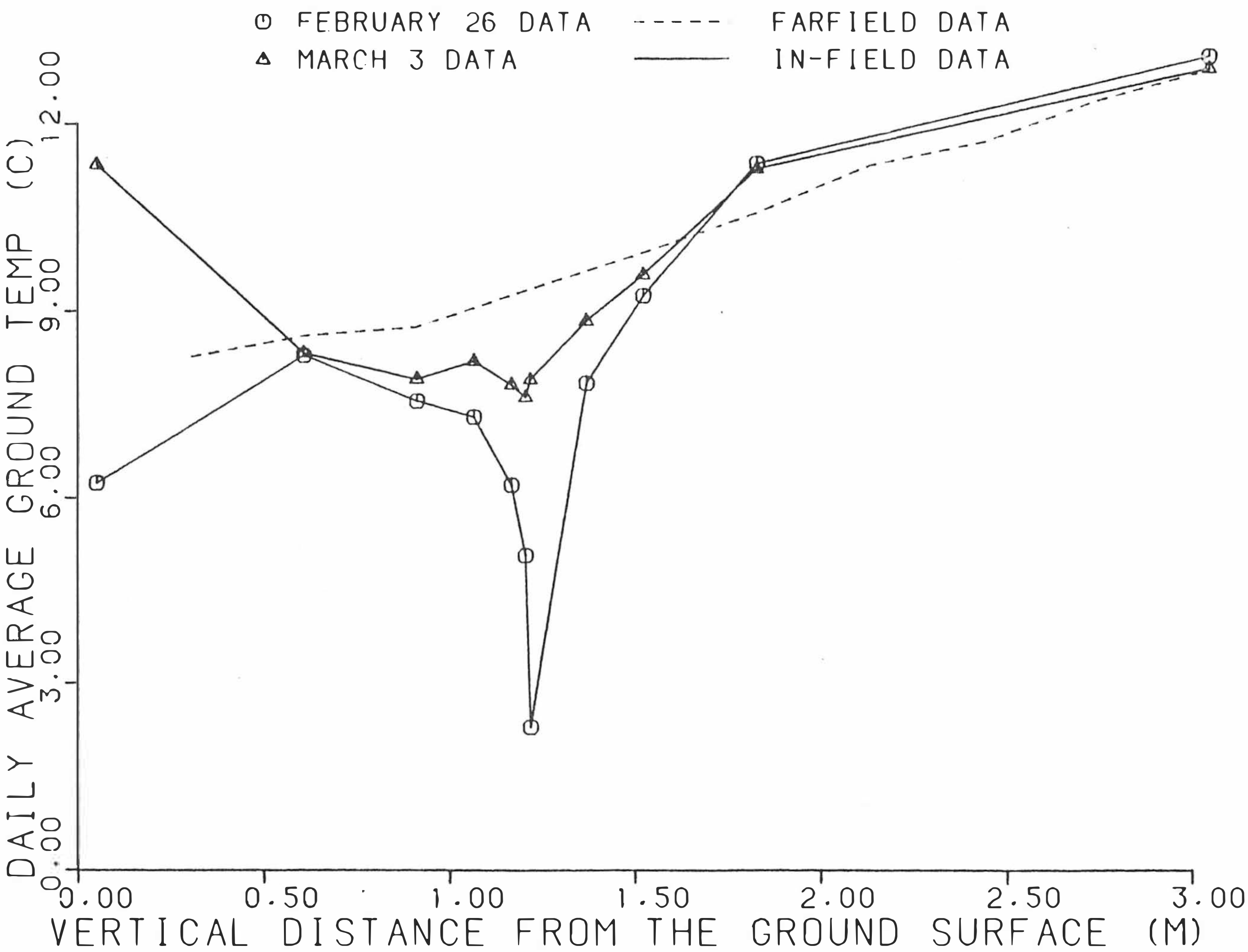


Figure 7. Vertical Temperature Profiles

Figure 8 shows temperature data collected from the earth at depths of 1.2, 1.8, and 3.0 meters within the pipe field, and 0.9, 0.6, and 2.0 meters away from the pipe respectively. Also superimposed on this Figure are data from the far field temperature site recorded at the same depth. There is good agreement between the far field and near field temperatures, and this Figure reiterates that the ground heat exchanger has little measurable influence on the soil at these distances from the coil.

Source Temperatures

Figure 9 shows the daily average temperatures for ambient air, the 'far field' soil temperature at a depth of 1.2 meters and the inlet brine entering the heat pump from the ground. This Figure demonstrates how consistent the soil is as a heat source when compared to the air. It is seen that the ambient air temperature varies from -6 C to 20 C, and that the ground varies only from 9 C to 21 C. Also, when the heating demand is greatest, the air is at its lowest advantage as a heat source. However, the ground temperature is much more slowly influenced by the daily weather patterns and serves as a more consistant and higer temperature source.

The inlet brine temperature is shown also on this Figure because it has the most direct influence on the heat pump performance. The brine temperature follows the pattern

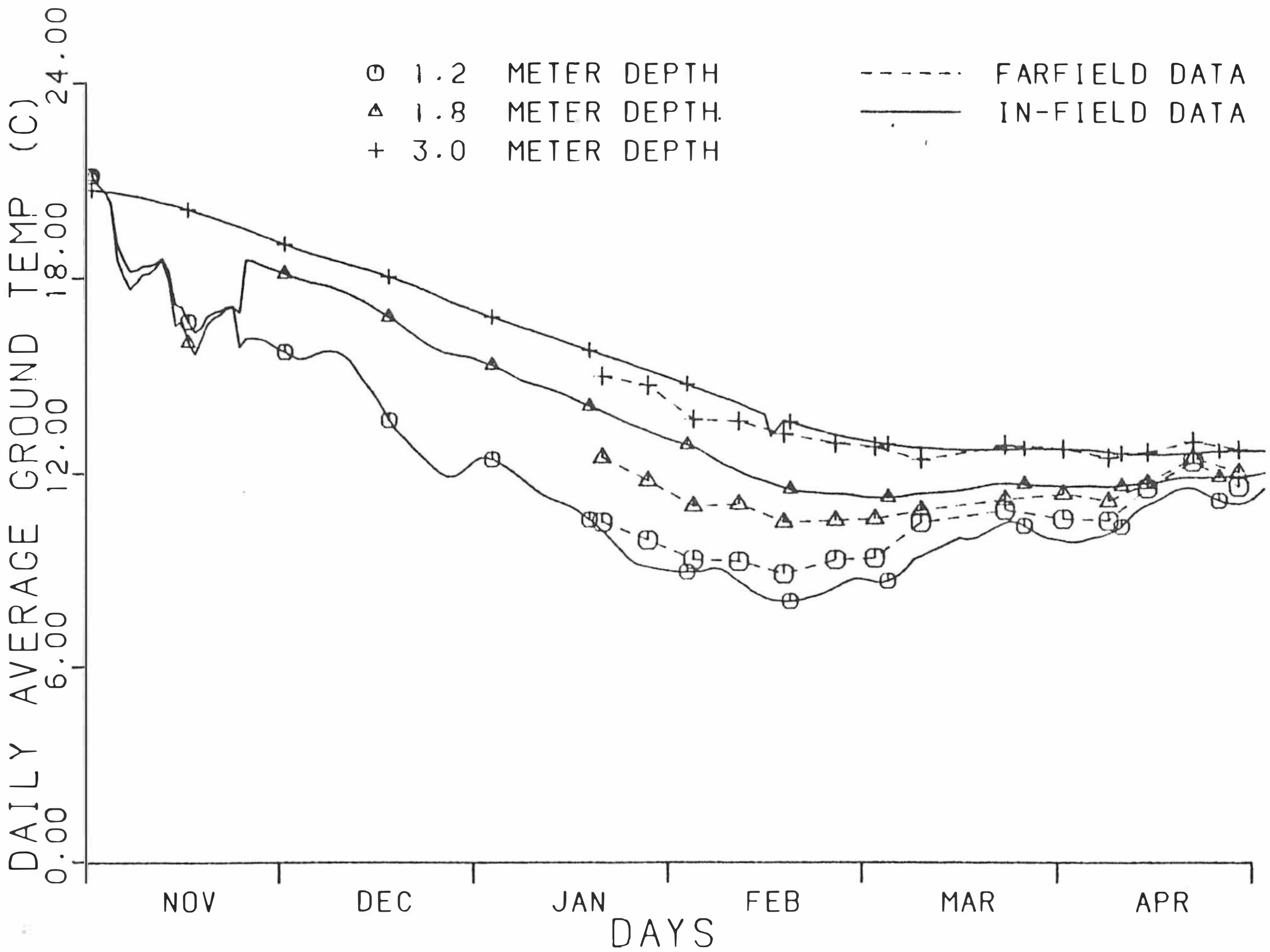


Figure 8. Far-Field and In-Field Temperature Comparison

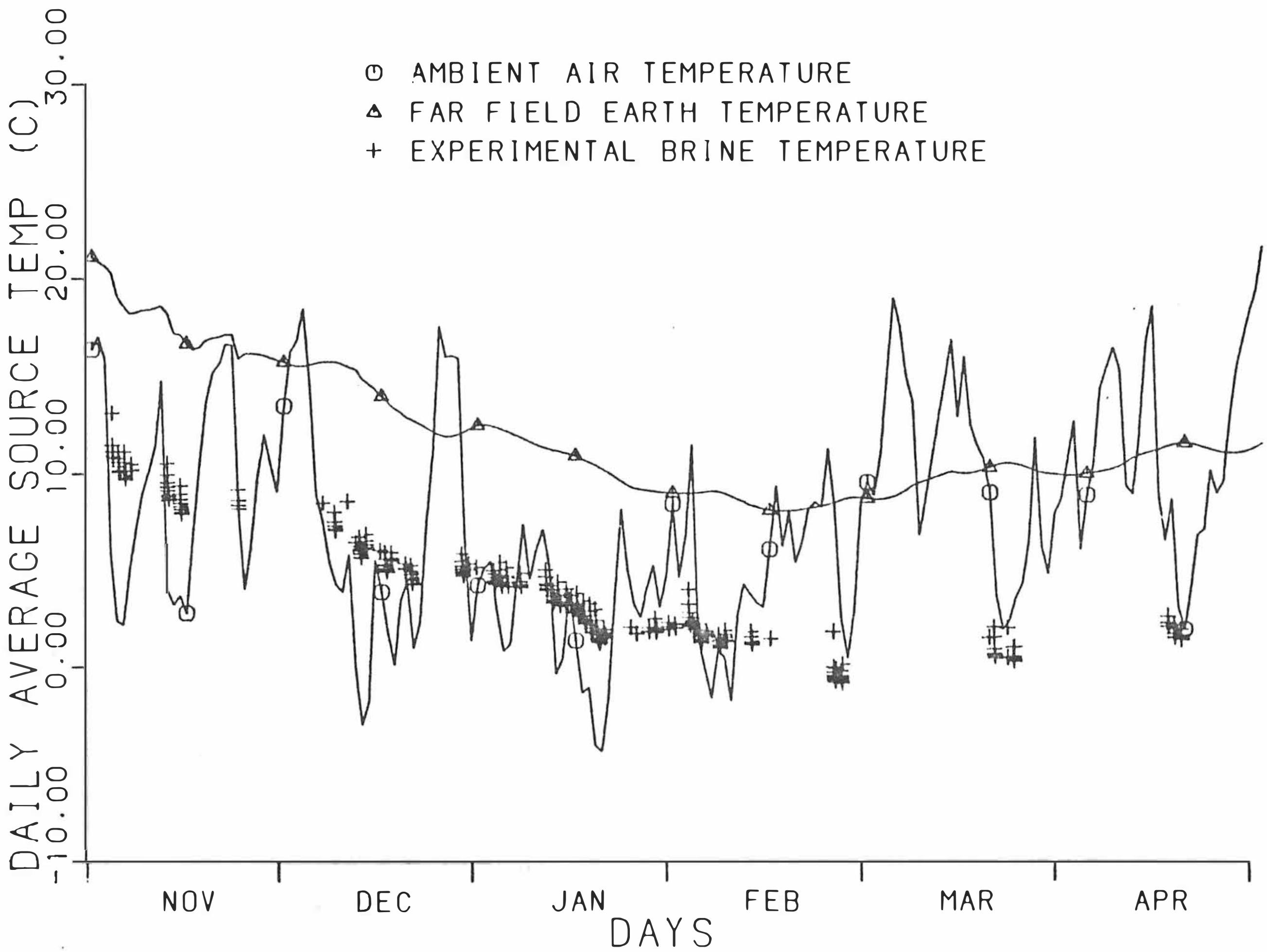


Figure 9. Source Temperature Comparisons

set by the far field soil temperature, decreasing until mid-February, and then beginning its upward trend. Because of an oversight, the RTD measuring inlet brine temperature was located such that, when the unit was off, heat from the house would conduct into the pipe and influence the temperature measurement. Therefore, only representative temperatures, which were recorded after the heat pump had been operating a significant period of the previous hour, are illustrated.

Soil Moisture

In May 1982 gypsum moisture sensing blocks were installed in the ground around the coil. In theory, the moisture in the gypsum block reaches equilibrium with any surrounding soil and the resistance of a current passed through the block is proportional to the percent moisture within the block. This instrument, however, proved inadequate. During November 1982 the gypsum blocks were excavated and examined. Several of the blocks had dissolved and were decreased in volume up to 30%. Further investigation with the manufacturer revealed that the gypsum blocks had been designed for agricultural purposes for use in less saturated soils, and had a life expectancy of only one growing season. When the sensors were excavated, the soil was relatively moist. Water had to be removed from the excavated area as it drained continually from visible voids in the backfilled soil. This condition is believed to have aided in the deterioration of the moisture blocks.

<u>Experimental Thermal Conductivity</u>

Because a reliable value for the thermal conductivity of the soil was difficult to secure, an experimental examination of this property was undertaken. Based on temperature observations, the soil within the first 0.10 meters from the pipe could be considered relatively near steady state if the heat pump has been operating for a large fraction of the previous hour. A program was written to screen the appropriate data from the entire winter. Assuming steady state conditions, a symetric temperature profile, and a constant thermal conductivity value, the one dimensional condution equation for radial heat flow through a composite cylinder was used to calculate the thermal conductivity of the pipe and soil system. The average fluid temperature in the pipe and the temperature at 0.10 meters from the pipe were used as boundary conditions. Using the known thermal properties of the pipe the value for the soil was then calclated.

From 242 hours of data a soil thermal conductivity of 4.1 KJ/C/M/HR was calculated. Using one standard deviation of the data, the probable range of soil thermal conductivity is between 3.0 and 5.6 KJ/C/M/HR. The monthly averages of the soil thermal conductivity calculated by this method ranged from 3.1 KJ/C/M/HR in December to 5.9 KJ/C/M/HR in March, thus demonstrating that the thermal conductivity is not constant throughout the winter.

A thermal conductivity value calculated by the method of Lunardini (32), and using information gathered from a soil test of the ground-coupled site predicted a thermal conductivity value of 4.6 KJ/M/HR for the disturbed soil and a value of 5.9 KJ/M/HR for the undisturbed soil. An example of the calculation proceedures is found in Appendix A. The thermal conductivity prediction for the disturbed soil lies within the values calculated experimentally and gives credibility to the method of Lunardini. These values were used as input values for the modeling.

Freezing

Past experiments have noted the occurance of freezing around the ground coil and its effect on performance. The migration of moisture in a porus medium towards a cold object enhances the availability of water near the coil. The latent heat gained by the freezing process would increase the effective thermal conductivity of the soil. Freezing was not observed during this season. A more severe winter or a shorter pipe coil length could have caused freezing to occur.

Heat Pump Performance

The monthly and seasonal totals of the three electric power consuming components of the system, the heat flow values and the number of degree days are presented in Table 1. The ground-coupled heat pump was capable of providing heat to the conditioned space throughout the heating season without the need for auxiliary electric resistance heat. The performance factor is defined as the total useful heat provided to the conditioned space divided by the total purchased power required to deliver that heat. The heating performance factor is then calculated as the sum of the purchased electric power and the ground coil heat flow divided by the purchased electric power for the entire season. This value is also included on Table 1. Note that because of the larger inside air coil installed in mid-February, the performance factor has an irregular increase at that time. In general, the performance factor decreases slowly as the season progresses and the ground temperature experiences its seasonal decline. Then in February, when the ground temperature begins to increase, the performance improves as seen in Figure 10.

Table 1. Experimental Results For The Heat Pump

Month	Total System Power (KWH)	Air Coil (KWH)	Ground Coil (KWH)	SPF	Degree Days F
Nov 82	545	1054	1005	2.84	457
Dec 82	843	1871	1350	2.60	581
Jan 83	1337	2389	1856	2.39	845
Feb 83	953	1806	1360	2.43	661
Mar 83	467	1039	821	2.76	390
Apr 83	462	1072	861	2.86	390
Total	4606	9233	7253	2.57	3324

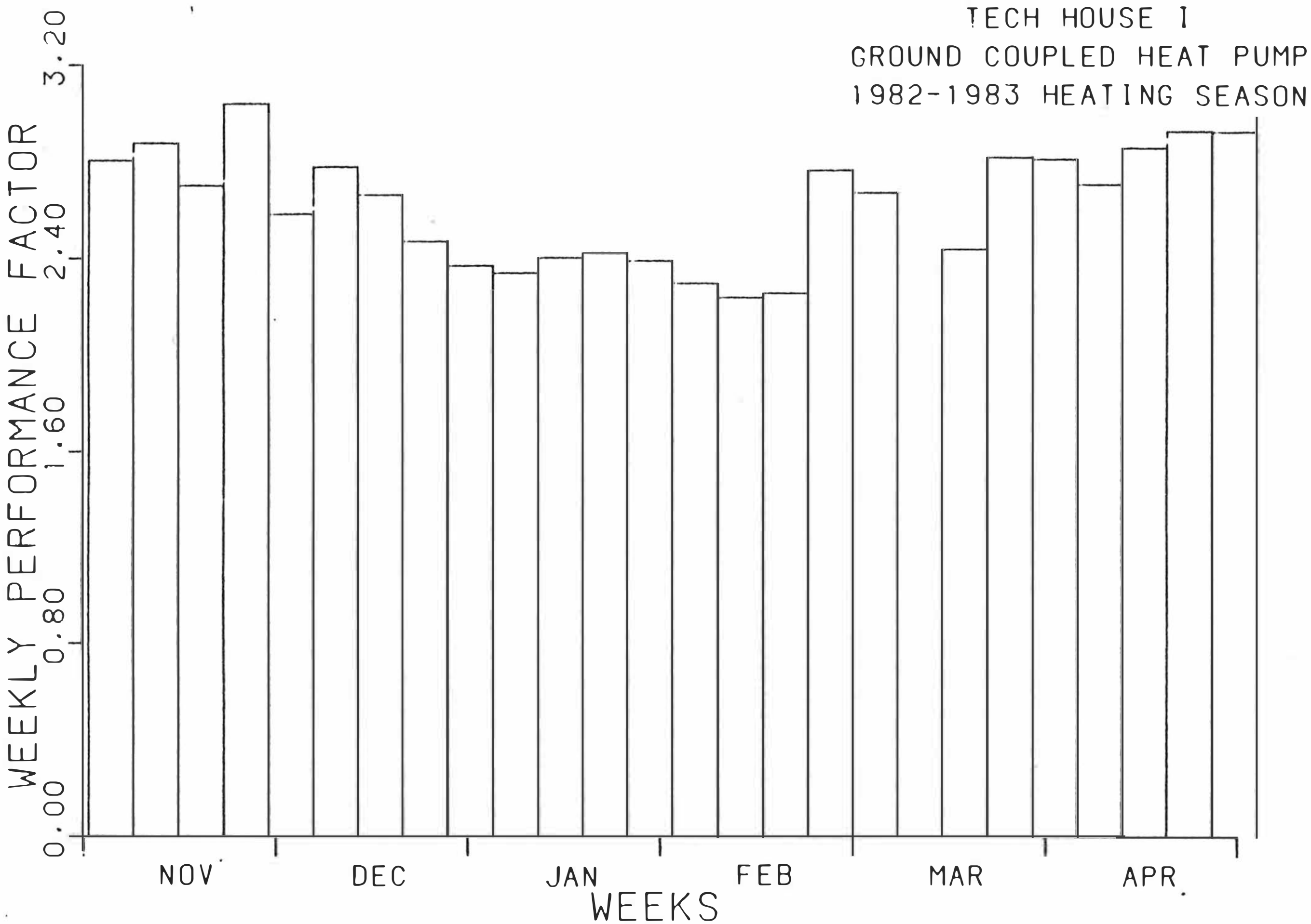


Figure 10. Weekly Performance Values

CHAPTER V

ANALYITICAL METHODS

Several items must be considered in order to perform an analyitical study of the ground-coupled heat pump as there are many factors which influence the system performance. The heat pump performance curve must be determined, the soil properties, including a constantly varying temperature, must be determined, the load to the heat pump must be obtained and finally a model which can connect all of these factors must be applied to the system before an analysis can be made. Once assembled, the model can be validated, and finally, parametric studies can be performed to predict the results of changes in the experimental system.

Heat Pump Performance Curve

Because the modeling of the heat pump itself is not the objective of this study, and since no suitable performance curve is available for the heat pump unit, a determination of the heat pump performance under operating conditions was extracted from the recorded experimental data. Extensive data was collected throughout the heating season which totaled 4368 hours; a summary of this data is found in Appendix B. In order to describe the performance of the heat pump, the data for each hour was inspected to insure that basic requirements were met before these data were to be included in the performance analysis. The first requirement was that the air temperature moving across the air side coil should be constant, however this was found to be too restrictive. Relaxing this restriction to include a small range of temperature about the mean air temperature, allowed sufficient data to be selected. The inside air temperature was relatively constant throughout the season and averaged at 21.0 C. Using a range of air temperatures from 20.25 C to 21.75 C provided sufficient data for the analysis. The other major requirement for selection was operating time. The system should be operating a sufficient length of time so that large transient effects would not be incorporated into the performance curve. In addition, an oversight in the sensor locations allowed heat from the house to conduct into the ground coil where sensors measuring inlet and outlet brine temperatures were located, thus significantly elevating the recorded temperatures when the system was inoperative. Enforcing a minimum runtime criteria necessary to consider this data as valid reduced the error obtained by this oversight.

Because two different inside air coils were in use during the study period, different criteria were needed for each of the resulting systems. Restricting the inlet brine temperature data as valid only for runtimes which were in excess of 75% for the November - February 13 system was found to yeild sufficient data for the performance curve. However, the restriction of the February 14 - April system had to be relaxed to include data for all hours which had a runtime in excess of 50%.

In order to describe the heat pump performance, the heat taken from the ground, the heat delivered to the space, and the work entering the heat pump were compared with the fluid temperature entering the heat pump for each of the hours that satisfied the selection criteria. A linear fit was assumed and a least squares fit was performed on the data. For illustration this data is shown as a performance factor in Figure 11 along with representative data points. This heat pump performance curve is used with the analyitical model to describe the heat pump.

Soil Properties

The soil properties of analytical interest included temperature, heat capacity and the thermal conductivity of the soil. A geological analysis of the site was performed at two locations in the field. One sample representing the disturbed soil was taken from the trench containing the ground coil. The second sample representing the undisturbed soil was taken from within the coil field but outside the trench. Thermal conductivity in particular is a function of density and moisture content of the soil, and these properties of the disturbed and undisturbed soils are quite

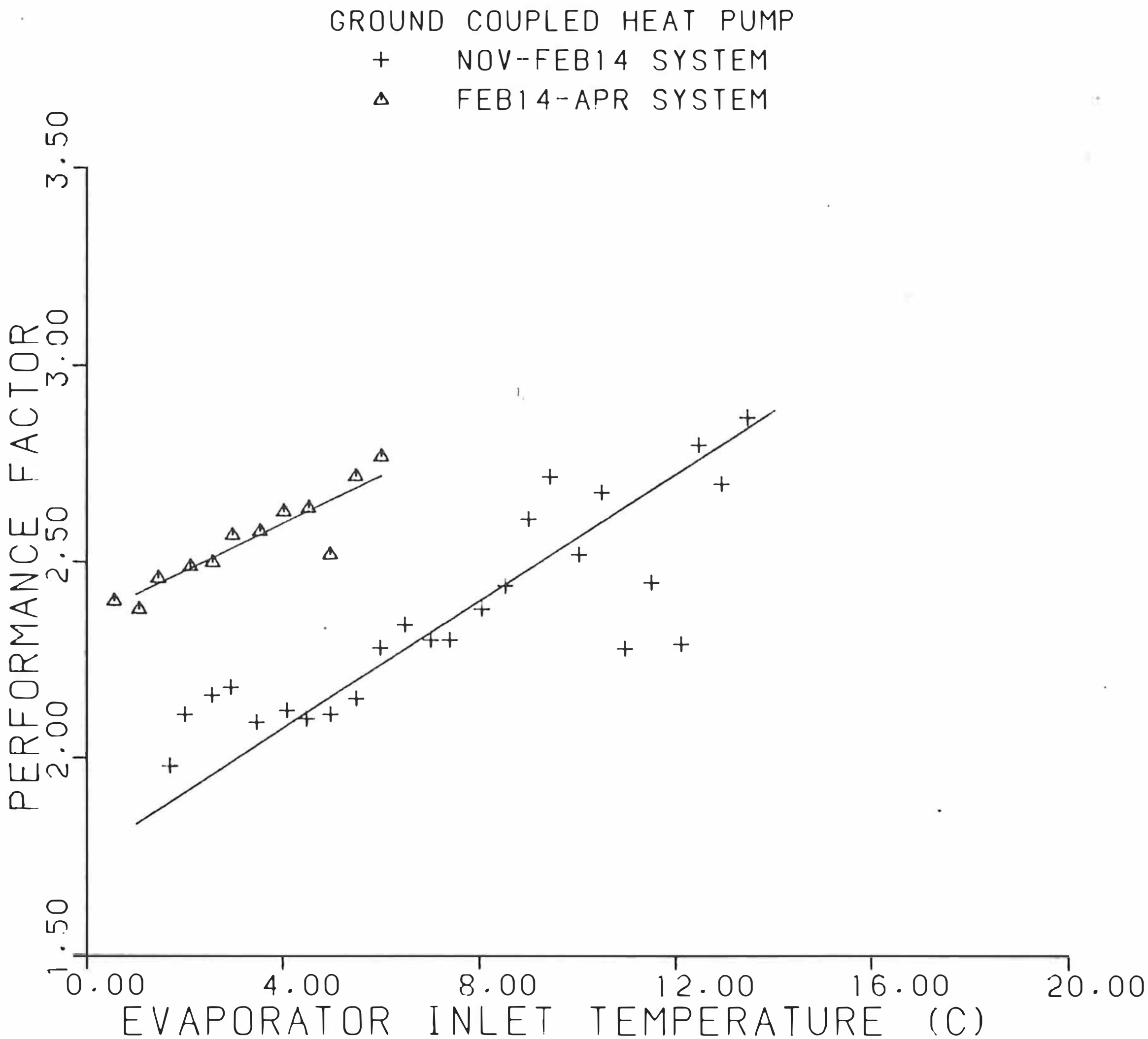


Figure 11. Experimental Performance Curve

different. Table 2 compares the soil properties for the disturbed and undisturbed soils. Although the volume of the trench is small compared to the volume of the entire field, it was a concern that this difference might be significant because by necessity, the pipe coil lies always within disturbed soil. Assuming that the soil was near saturation throughout the winter season, and using the geological analysis the values for conductivity and heat capacity were determined by the method described by Lunardini (32).

Table 2. Tech Site Soil Properties

Soil Type	Disturbed Soil	Undisturbed Soil
Natural Moisture	24%	17%
Wet Density g/cc	1.57	2.00
Thermal Conductivity At Saturation KJ/M/ C/HR	4.60	5.86
Volumetric Heat Capacity KJ/M*3/ C	2930	2673

The soil temperature is another important property affecting the heat transfer. There was no continuous monitoring of the soil temperature outside the pipe field, therefore, a model describing the far field temperature was required. The earth temperature oscillation near the surface has been discussed by Carslaw and Jaeger (33), and an equation describing the temperature follows:

$$T(x) = AET - BO \; e^{-\frac{\pi}{DP}x} \cos\left(\frac{2\pi\Theta}{P} - \frac{\pi}{DP}x - PO\right) \qquad (5\text{-}1)$$

Where:

$T(x)$ = the monthly average earth temperature at depth x

Θ = the time coordinate in hours beginning January 1

P = the period of the temperature cycle = 8766 hours

AET = the annual average earth temperature

D = the soil thermal diffusivity

BO = the earth temperature amplitude at the surface

PO = the surface temperature phase angle (radians)

Kusuda has applied this equation using available data on earth temperature with a discussion of its applicability and its use as a predictive tool (34). He reports that in most cases the long term average ambient air temperature values are close to the average temperatures of the soil. Initially, long term mean ambient air values were applied to the earth temperature model, however, a poor match was found between the model and the limited far field earth temperature data that had been collected for that season. Values for the annual average earth temperature and the earth surface temperature amplitude were adjusted to force a better fit between the observed earth temperatures and the model. Figure 12 illustrates the good correlation between the earth temperature model and the actual data. The values used for the annual average earth temperature was 15 C, for the earth temperature amplitude was 10 C, for the surface temperature phase angle was 0.3, and for the soil thermal diffusivity was 1.9 x 10-3 M*2/hour. Deviation between the predicted and the experimental values are due to several factors including irregular fluctuations of weather influence, non-homogeneity of soil, and varying topology and soil properties. It is noted by Kusuda that many factors are involved which do not allow exact prediction of the earth temperature. The earth temperature predictions are statistical in nature and some deviation from the mean is to be expected. The use of this method for the far field prediction is justified by the lack of continuous far field data and the analyitical goal which is to test the predictive ability of the model.

TRNSYS

Few simple methods are presently available for the analysis of complicated energy systems. An available and widely used computer simulation model of energy systems is the Transient Simulation Program (TRNSYS), which was developed by Klein et al. at the University of Wisconsin

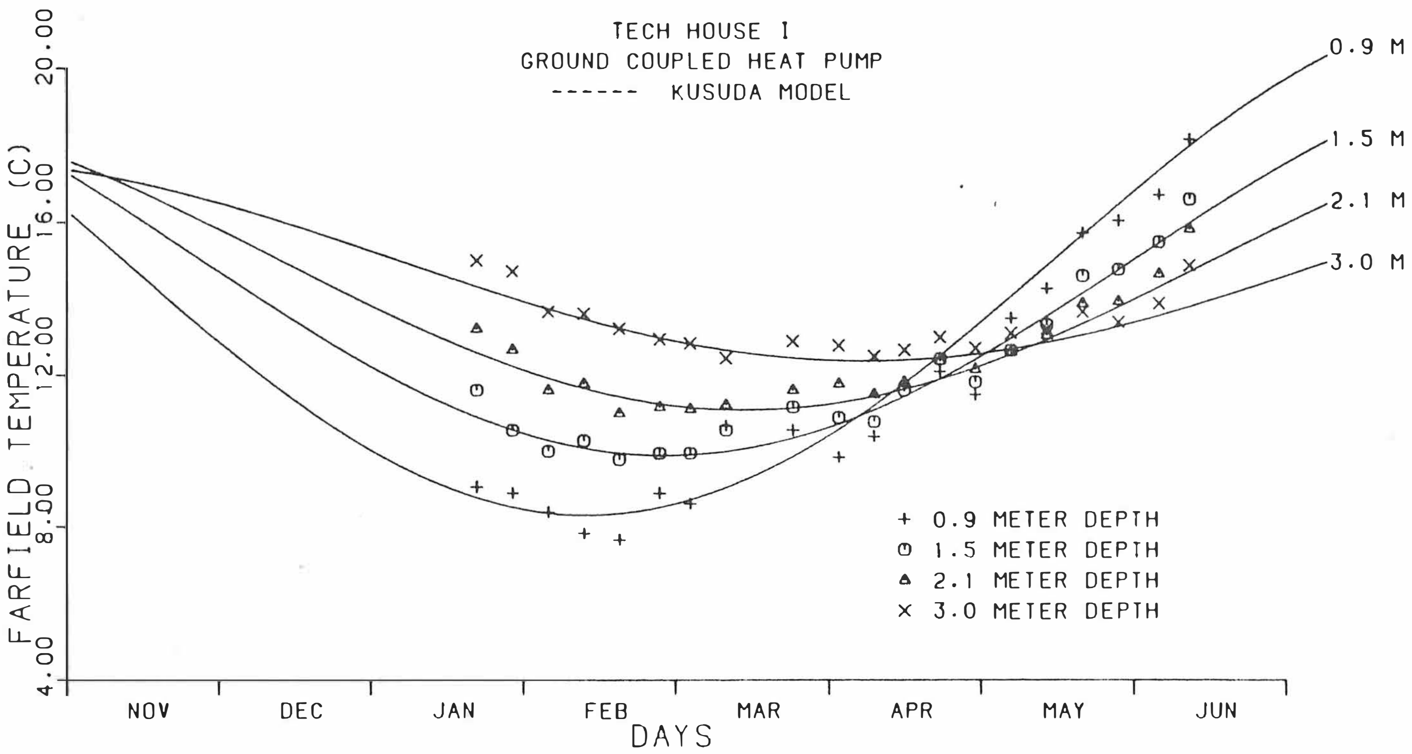


Figure 12. Farfield Ground Temperature Model

(35,36). TRNSYS defines an energy system as a set of independent but interconnected components, and simulates the performance of each physical component of the system by modeling each in a detailed Fortran subroutine. These subroutines employ a quasi-steady state approximation for those components with negligible thermal storage, and one dimensional transient conduction models for thermal storage devices at specified time steps. Using simple interconnection of the system components along with the input data, the program solves systems of differential and algebraic equations which represent the energy system. This modular simulation technique enables an experimenter to construct a model of a particular system to be studied by reducing a large problem into a number of smaller problems. Although TRNSYS itself does not have a subroutine which models ground coupling, an additional model provides this function.

GROCS

Three computer programs supplied by Battelle Columbus Laboratories were considered for the modeling of the ground coupling (37). All of the programs are based on constant thermal properties for the soil and neglect freezing effects and moisture migration. Only the program, Ground Coulped Systems (GROCS), demonstrated experimental validation in the literature (39,40). One version of GROCS available was immeadiately compatible with TRNSYS, and this was selected for use in this study.

GROCS is a TRNSYS compatible model of ground coupling developed by the solar technology group at Brookhaven National Laboratory (38). This program solves finite difference equations which represent the heat flow in the earth in the vicinity of a heat source. GROCS can be coupled to TRNSYS as an additional subroutine and is reputed to be useful in simulating any type of closed loop ground-coupled heat exchanger or storage device having the configuration of a buried tank, multiple shallow wells, a single deep well or shallow horizontal coils.

GROCS solves the finite difference equations of heat flow over a system of "blocks" of earth. Each block is a volume of earth whose size and shape are determined by the type of modeling being done. The block configuration suggested by Fisher (4) was adapted and used in the initial analysis of the buried coil, and in addition, two other ground configurations were investigated. Diagrams of these block configurations are seen in Appendix C.

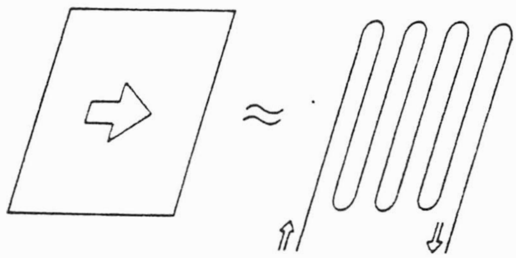
GROCS uses two kinds of blocks called "rigged blocks" and "free blocks." The rigged blocks surround the free blocks and provide the necessary spatial boundary conditions. The temperatures of the rigged blocks are determined at each time step using predictions derived from the Kusuda model for average ground temperatures for each month of the year at a number of different depths spanning the portion of ground to be simulated. The free blocks make up the bulk of the system of earth blocks and surround the ground coupling device. At each time step, the free blocks have their temperatures determined by their thermal interaction with each other, with the rigged blocks and with any heat inputs placed in them which simulate the effect of a heating system load.

The simulation of a serpentine array of buried pipe presents a special problem for the GROCS program. This problem results because the typical pipe diameter is small enough to require very small block sizes, and straightforward modeling of the pipe would require inordinate amounts of time. For this reason the pipe field is modeled as a thin sheet of water flowing in the plane of the pipe field as discussed by Andrews (38).

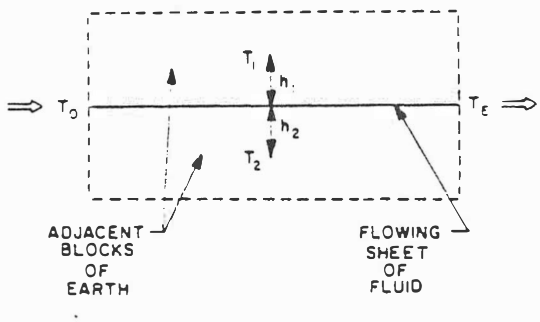
The heat flow from a sheet will not be the same as that from a set of pipes. A direct application of the flowing sheet approximation does not take into account the radius of the pipes or the distance between them. However, it is possible to take these factors into account by reducing the effective heat transfer area between the assumed plane sheet of water and the adjacent free blocks of earth, thus making the effective heat transfer area of the block less than the actual geometrical surface area (38). A cross section of the "flowing sheet" geometry is shown in Figure 13. The approach to making the correction for the difference between a flowing sheet and an array of pipes is to consider the thermal resistance of a half-block of earth with a constant temperature boundary on one side and a planar array of pipes on the other. This resistance is compared with that for the same half-block of earth with the same constant temperature boundary, but now with the sheet of fluid on the other side in place of the pipe array. In general, the resistance in the case of the plane sheet is the lesser of the two. To correct for this difference, the heat transfer area from the sheet to the adjacent block is reduced by a factor equal to the ratio of these resistances, so that the resistance for the flowing sheet case, with the reduced heat transfer area is the same for the actual pipe array.

For pipes embedded in a slab of earth equidistant from the two plane boundaries both at the same temperatures, as illustrated in Figure 14, the thermal resistance for a single pipe in one direction is given by:

$$R_{pipe} = \frac{1}{\pi L k} \ell n \left[\frac{s}{\pi R} \sinh \frac{\pi h}{s} \right] \qquad (5\text{-}2)$$



Flowing-sheet approximation to pipe field



Geometry of flowing sheet approximation

Figure 13. Flowing Sheet Approximation

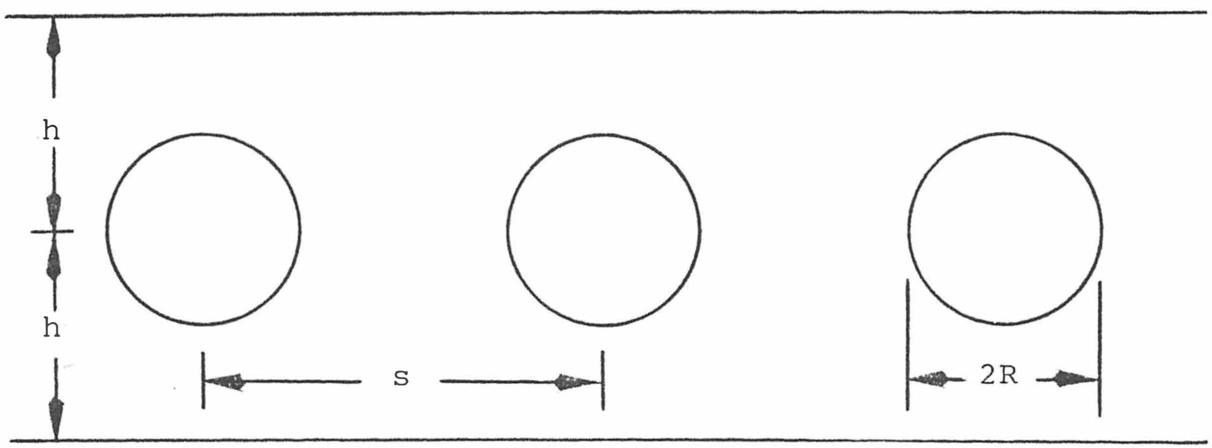


Figure 14. Pipe Field Embedded Between Two Boundaries

This resistance is compared with that of a rectangular slab of the same thickness (h), area (SL), and conductivity (k):

$$R_{slab} = \frac{h}{sLk} \tag{5-3}$$

The factor n by which the heat transfer area between the plane sheet of fluid and the adjaecent blocks must be reduced is then equal to equation (5-2) divided by equation (5-3).

$$\eta = \frac{\pi h}{s} \quad \frac{1}{\ell n\left[\frac{s}{\pi R} \sinh \frac{\pi h}{s}\right]} \tag{5-4}$$

This factor is multiplied with the area of the adjacent block which interfaces the pipe coils, making the effective area smaller than the actual block area, and thus correcting for the flowing sheet approximation.

Concern for the validity of the flowing sheet approximation and that the geometry of the adjacent blocks could play a significant role in the predictive ability of the model prompted an investigation concerning the block configuration design.

In order to guarantee convergence, it can be rigorously shown that for the heat flow finite difference equations on an evenly spaced mesh in one-dimension the constraint

$$\frac{k}{\rho c} \cdot \frac{\Delta t}{(\Delta x)^2} \leq \frac{1}{2} \tag{5-5}$$

is required. It has been found satisfactory for GROCS to use the smallest block width in the above equation to satisfy the convergence criteria. Using one hour input intervals, the thermal conductivity value k=4.60 KJ/M/C/Hr and Cp=2930 KJ/M*3/C the minimum block width must exceed 5.6 centimeters. The flowing sheet approximation imposes another resriction on the block width. This approximation assumes a constant temperature at the boundary of the adjacent block. As the width of the adjacent block is decreased, this assumption becomes less likely. The formula for the correction factor in equation (5-5), suggests this by approaching infinity as the adjacent block thickness (h) approaches the radius of the pipe coil (R). No criteria is specified to aid in the selection of an adjacent block width, however, a plot of the equation shows that it behaves nicely when $h/R > 5$, for the pipe spacing and pipe radius previously specified. Comparing the predicted performance of the model using the range $2 < h/R < 14$ showed that the results are stable when $h/R > 3$. Since the pipe's outer radius was 2.13 centimeters this implies that h must be greater than 6.4 centimeters. This is more restrictive than the guaranteed convergence criterion and this limit was adhered to in the analysis.

TRNSYS Component Alterations

A simple arrangement for TRNSYS and GROCS was incorporated for the analysis. The modeling was arranged such that only the heat pump and the ground with the coil were simulated by the model. TRNSYS has a much larger capacity than this, however, it was intented to explore the ground and ground coil as directly as possible with the available model, and not to model the entire house and heating system. The heating load used to drive the analyitical evaluation of the heat pump is the actual experimental load measured and delivered to the house each hour by the air side coil. Using the experimental hourly load, the validation procedure is much more direct than generating a load by other means.

The TRNSYS subroutine designed to model the heat pump was not readily adapted to the experiment. The heat pump model was set up to operate from several sources including a water storage tank which stored heat from a solar collector. With little difficulty the model accepted the fluid from the ground coil into its storage tank and thus used the ground circulation fluid as its heat supply. The heat pump model is not usually used with such little support from other TRNSYS subroutines. In more complex arrangements the heat pump model would have depended on a multistage thermostat model to regulate its operation. The house model would then store heat energy by raising or lowering its temperature depending on whether the heat pump had met its load demand. The arrangement of TRNSYS and GROCS could not be easily modified to incorporate storing the balance of the heat load, and therfore an alternative had to be provided. If the experimental load input was greater than the analytical heat pump could provide at its modeled inlet temperature, then the additional load was carried over to the next time increment. This could be carried on indefinitely until all of the load is provided for. This arrangement insured that the parametric studies would provide an analysis based on the same total load.

An additional alteration in the heat pump subroutine was needed to account for the difference in the heat pump performance with the installation of the new air side coil in mid-February. A timer was added to the heat pump subroutine which caused the model to use different heat pump performance data at the appropriate time. The alterations to TRNSYS are discussed in further detail in Appendix D.

One criticism of the GROCS model has been the assumption of a uniform thermal conductivity for the entire coil field which does not vary with time. This assumption is known to be incorrect especially throughout the summer months. To partially remedy this situation an option was created in the programing to allow different thermal conductivity to be input for each block. The adjacent block was assigned a thermal conductivity which was a volumetric average of the thermal conductivity of the disturbed and undisturbed soils based on the volumes of the actual coil trench and the volume of the adjacent blocks. The thermal conductivity determined for the undisturbed soil was used for the free blocks.

In the TRNSYS-GROCS configuration, algebraic equations are solved by iterative methods subject to user specified accuracy criteria, and differential equations are solved by a first-order predictor-corrector method. A simplified block diagram of the program is shown in Figure 15.

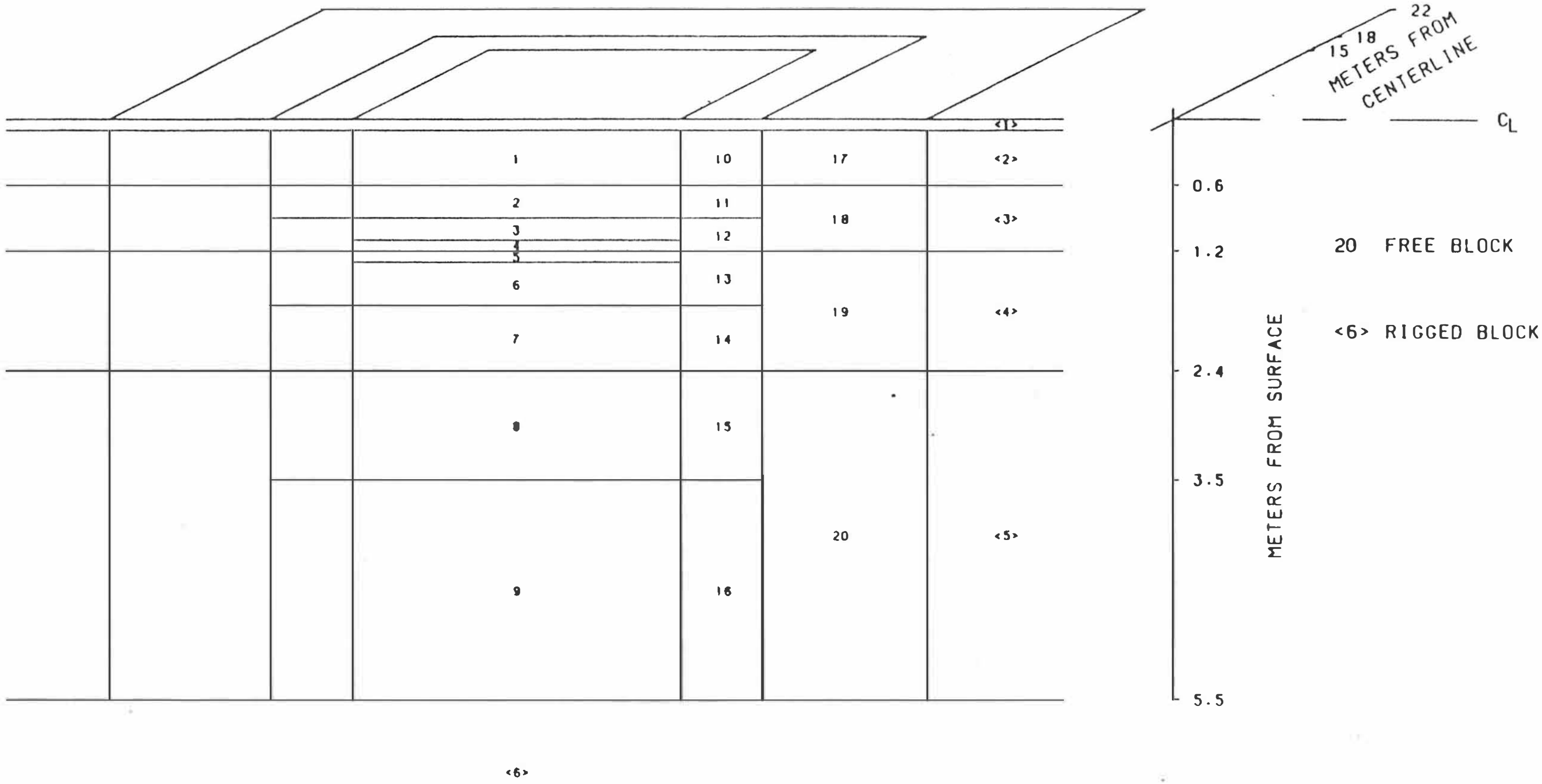


Figure 15. TRNSYS-GROCS Block Model

CHAPTER VI

ANALYTICAL OBSERVATIONS

Objectives

The objective in the analytical investigation was two-fold. The first objective was to validate the model by examining its forecasting ability. Validation studies of GROCS have been performed for buried tanks and serpentine coils using weekly average heat transfer input data (39,40). Limited material is available regarding the validation of GROCS when utilizing a transient input. The second objective was to examine particular parameters to see how they influence the predictability of the model. This examination demonstrates the sensitivity of the model to particular variables and indicates which input parameters are most critical.

Validation

A model can be validated if it is found to be a predictive tool. Many of the parameters to be used in the GROCS model cannot be uniquely determined with a high degree of confidence, and this possibly limits the reliability of the model. Some of the parameters of the model change with their environment such as thermal conductivity which changes with the local moisture content. Other parameters show an irregular pattern such as the ground temperature for a specific season. The approach to validation is to determine these parameters as accurately as possible, to execute the model, and to compare the predictions of the model with the experimental data.

Figure 16 compares the analytical and the experimental values for heat transfer from the ground to the earth coil. GROCS assumes an average fluid temperature to calculate the heat flow from the coil to the ground, and this assumption seems justified by the good agreement shown. Deviation between the experimental and analytical ground heat transfer values are observed at ground heat transfer values above 100 KJ/HR. The predictions of GROCS have been shown to be sensitive to the thermal conductivity of the soil during large heat withdrawls using average heat inputs (41), and the same seems to hold when using transient heat inputs. Figure 17 compares the experimental and analytical values of the work involved in the compressor unit of the heat pump. In general the analytical values are slightly higher than the experimental values, however the correlation is otherwise good.

Two block temperature history's and their corresponding experimentally recorded temperatures are shown on Figure 18. The two experimental temperatures were recorded at 5 centimeters above the pipe which corresponds to the center of the upper adjacent block and at 30 centimeters below the

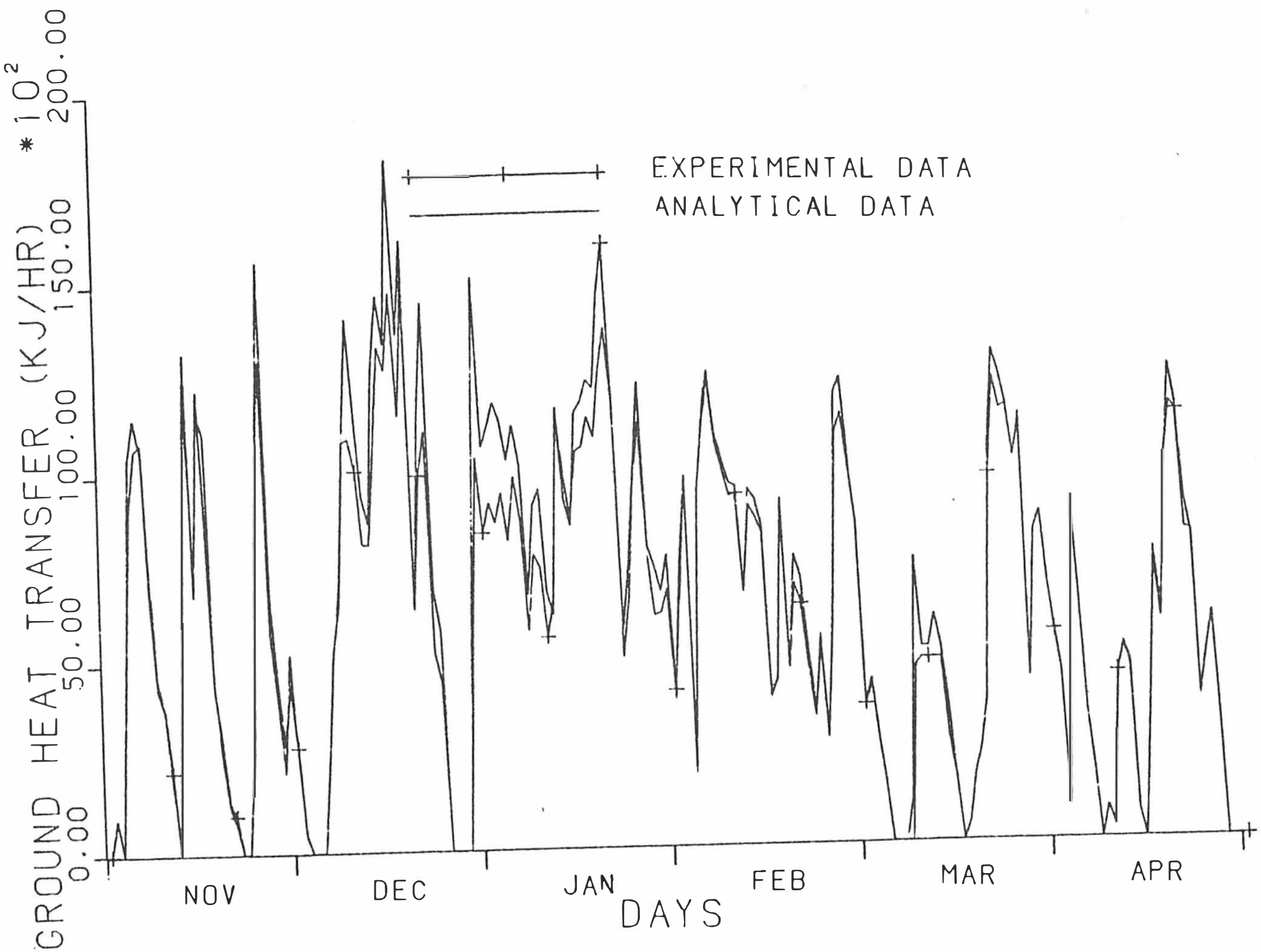


Figure 16. Ground Heat Transfer Prediction

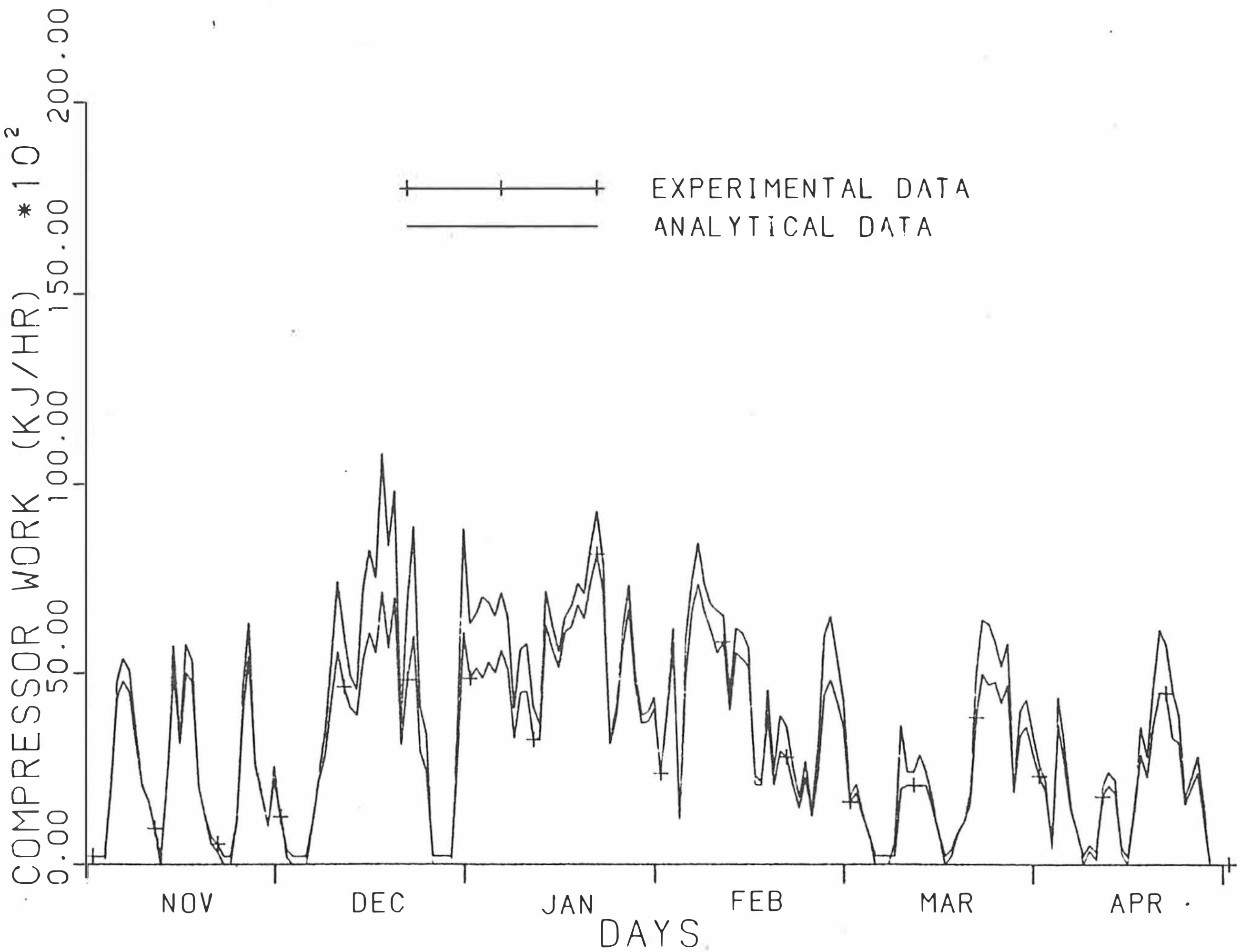


Figure 17. Compressor Work Prediction

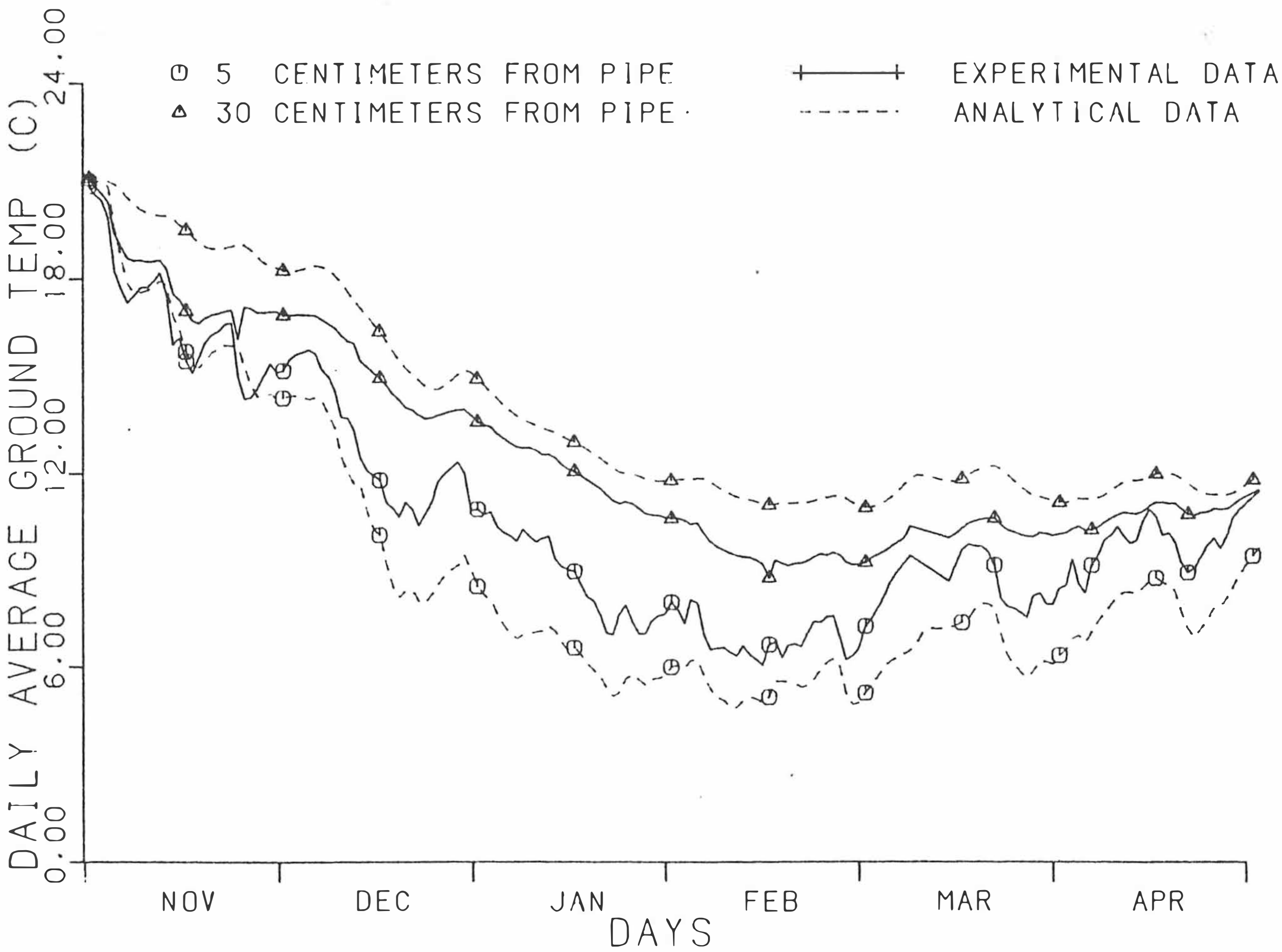


Figure 18. Ground Temperature Prediction

pipe which corresponds to the center of the free block in contact with the lower adjacent block. These two locations were chosen because the blocks centers' coresponded to the placement of a temperature probe. The root mean square temperature difference (Trms) for each location was calculated as a measure of the accuracy of the temperature prediction. The Trms for the point 5 centimeters from the pipe was found to be 1.9 C, and the Trms for the point 30 centimeters from the pipe was found to be 1.4 C.

Because of the error encountered by the inappropriate location of the brine temperature sensor, it is difficult to compare the analytical and experimental brine temperature on an average basis. Simple daily averages of the recorded temperatures are unrealisically high due to the conduction of heat from the house into the ground coil temperature sensor. A daily average of experimental brine temperatures taken only during periods of high runtime uses very limited data. Furthermore, this data represents the lower side of the mean because the brine temperature will drop considerably during large runtimes due to the relatively low thermal conductivity of the pipe and the adjacent soil. The average hourly run time for the heat pump was found to be 30%. A comparison of the analytical and experimental brine temperature values corresponding with runtimes from 15% to 45% is shown on Figure 19. This Figure illustrates that the

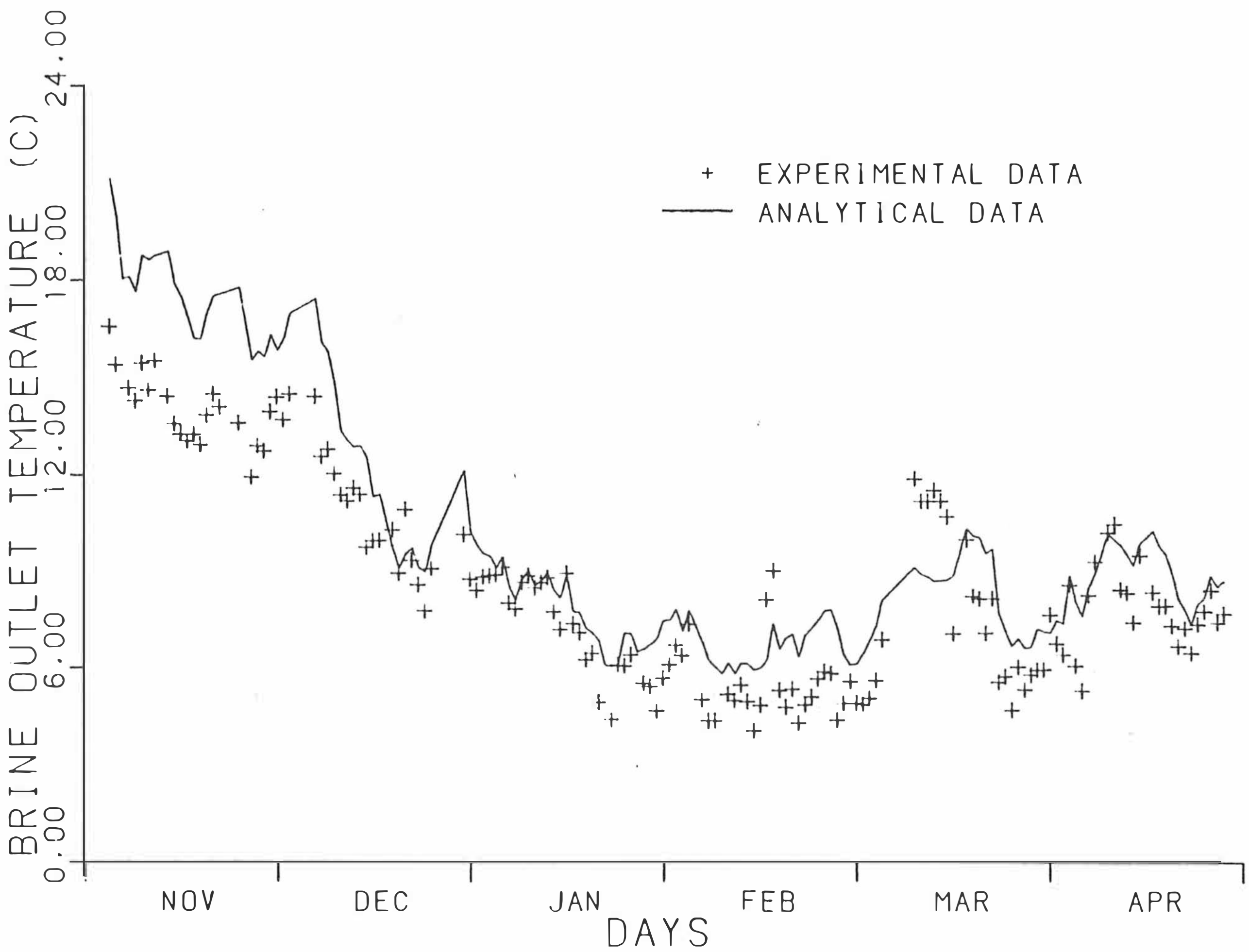


Figure 19. Brine Temperature Prediction

predicted brine temperature agrees well with the experimental temperature near the mean runtime. An examination of hourly data also demonstrated periods of excellent correlation particularly when there seemed to be much cycling.

The experimental ground-coupled heat pump delivered heat with an overall SPF of 2.57, while the model predicted a performance of 2.46. This is a good agreement especially when consideration is made for the questionable accuracy of several of the input parameters. These results show that a GROCS-TRNSYS modeling of a future prospective study could be a reliable predictor of the performance of a ground-coupled heat pump using a serpentine coil.

Parametric Studies

The sensitivity of the GROCS model is revealed by independently varying its input parameters. The input parameters were calculated and used in an initial execution of the model. The prediction for the performance factor was 2.46 while the experimental performance factor was 2.57. Since much of the input data for the modeling was derived from experimental data, a good performance forecast was expected. The simulation was run again while varying the input parameters for farfield temperature, thermal conductivity, coil length, block configuration and the heat pump performance relation.

Far Field Ground Temperatures

The far field ground temperature is a basic indicator of the earth's capacity as a heat source. However, very little information is available which describes local or even regional ground temperatures. For this reason, most applications of ground coupling rely on models to calculate the thermal boundary conditions for analysis. There is much room for judgement in dealing with ground temperature approximations. It is therefore prudent to know how sensitive an analysis is to a particular ground temperature prediction. The average temperature and the temperature amplidude parameters of the far field ground temperature model were varied, and the sensitivity of grocs to these variations was investigated.

Figure 20 demonstrates the variation of the GROCS prediction related to the change in the temperature boundary conditions. The performance of the heat pump is predicted to improve with an increase in average ground temperature and to decrease with an increase in the magnitude of the temperature amplitude. The values shown cover the range of ground temperature data reported by Kusuda for 29 ground temperature stations located throughout the United States. This Figure does not reflect how this house and heat pump system would perform in other areas of the country. The load would also change in areas where the ambient air temperature is different.

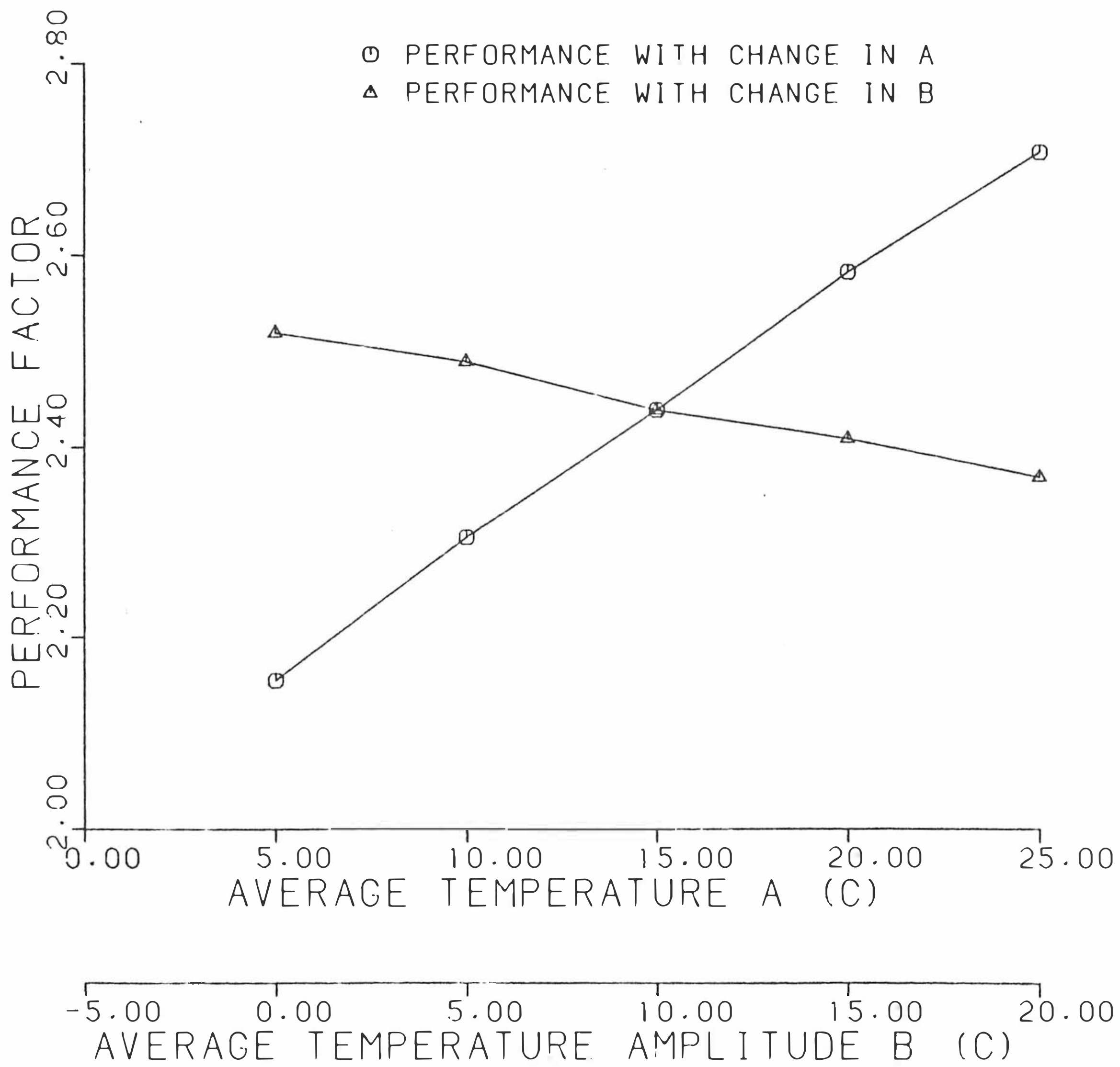


Figure 20. Grocs Prediction with Changing Thermal Boundaries

Kusuda's data shows that the long term average earth temperature is a complicated function depending on parameters such as latitude, elevation, slope and soil thermal properties to name a few. The average earth temperature is found to correlate strongly with the average air temperature. The temperature amplitude, however has a weaker correlation and care must be taken when assuming that the ambient air temperature values are close to the average earth temperature values as is often done.

Thermal Conductivity

Thermal conductivity of the soil, along with its related heat capacity are the most elusive of the GROCS input parameters to determine. The thermal conductivity of soil depends on the type of soil, the density and the moisture content. The density of a soil can be altered when a soil sample is collected for testing and it may vary throughout the site. In addition, soil may vary widely in texture in a small area; and finally, the soil thermal conductivity can change drastically with changes in the soil moisture. These factors decrease the likelihood that a particular sample will represent the thermal properties of the soil field for analytical purposes, and make a sensitivity analysis on thermal conductivity of upmost importance.

The thermal conductivity and the related heat capacity of the soil were varied and the predicted performance of the system was found. The adjacent block was assigned soil properties calculated for the disturbed soil and the remaining free blocks were assigned properties calculated from the undisturbed soil. The thermal conductivity is varied for both the adjacent blocks and the free blocks first seperately and then together. The results are shown on Figure 21.

The value of the thermal conductivity of the adjacent blocks and the free blocks were calculated by the method of Lunardini to be 4.6 kJ/M-C-hr and 5.9 kJ/M-C-hr respectively. The first simulations were made by varying the free block thermal conductivity values while holding the adjacent block values constant. The performance improves almost linearly with changes in the thermal conductivity, however, the improvement is small. The performance improves by only 0.57%/unit k. The free blocks were held constant and the adjacent blocks were varied in a second test which showed similar results. The performance improves by 0.61%/unit k. The greatest increase in performance occurs when the thermal conductivity of both the adjacent and free blocks are elevated together. The performance increased by 1.0%/unit k when both blocks' thermal conductivities were improved simultaneously.

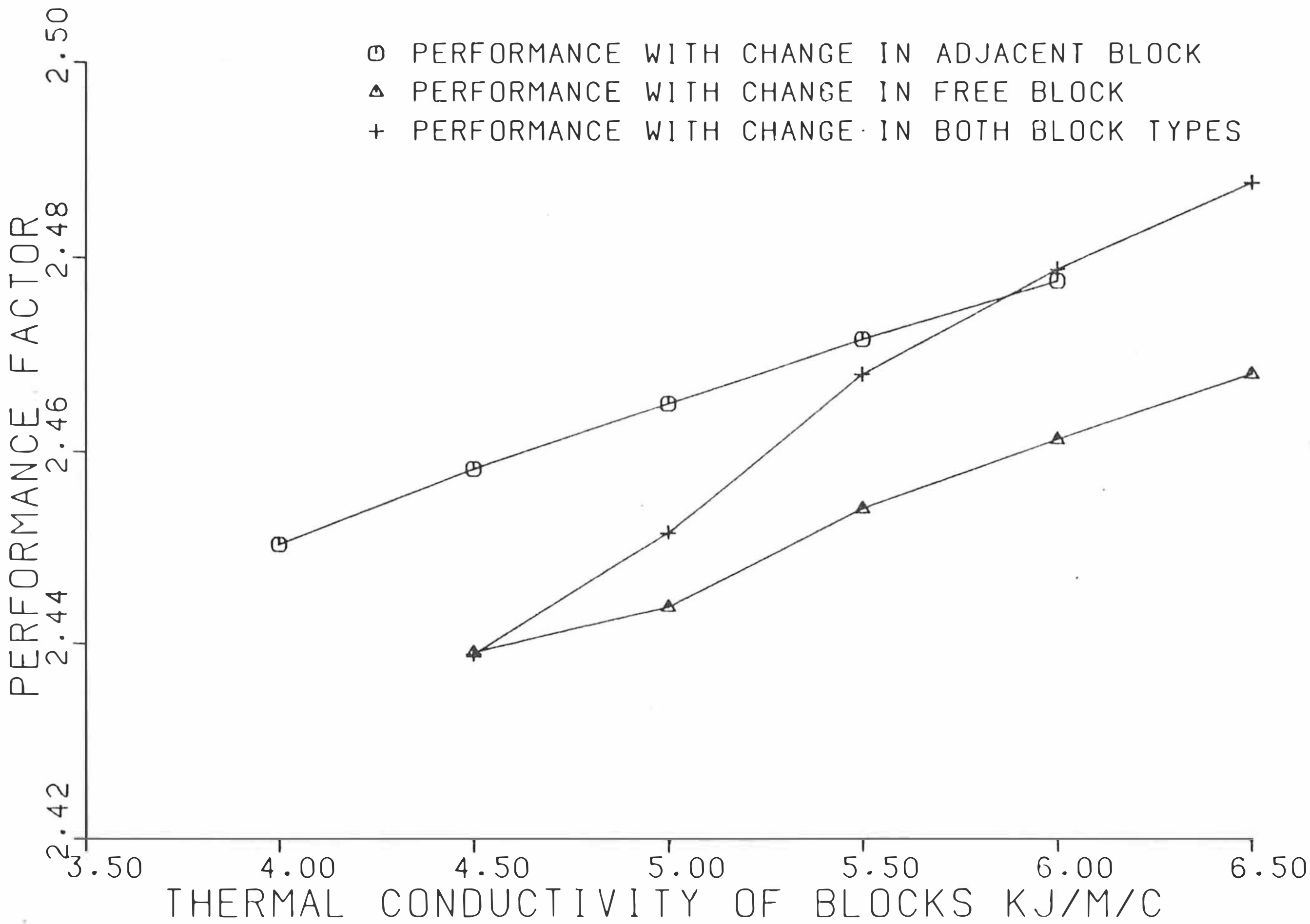


Figure 21. Grocs Prediction with Changing Soil Thermal Conductivity

Coil Length

Ultimately the most critical design element for the ground-coupled heat pump is the ground coil length. Increased coil length results in better performance potential, however, the rate of improvement diminishes with increasing pipe length. In addition, inefficiencies due to excessive cycling losses, pumping costs and the added excavation expense for a longer pipe require that the pipe length not be oversized. An undersized pipe, however, is also undesireable. An undersized pipe will have generally poor performance and may not be able to supply the needed heat and therfore require a backup heat supply.

Ground coils of different lengths were simulated by calculating block configurations based on Fisher's block diagram design for nine pipe lengths (4). The average seasonal heat rate per meter coil length versus performance curve is shown on Figure 22. This Figure demonstrates the diminishing return effect on performance with respect to the pipe length. For heating purposes, little improvement in performance can be expected for the TECH site ground-coupled heat pump with a longer ground coil. An improvement in performance of 8.9% is predicted for a heating coil twice the present length. Since the pumps consumed 8.8% of the total power to operate the exisitng heat pump, the additional power needed to circulate the fluid through the

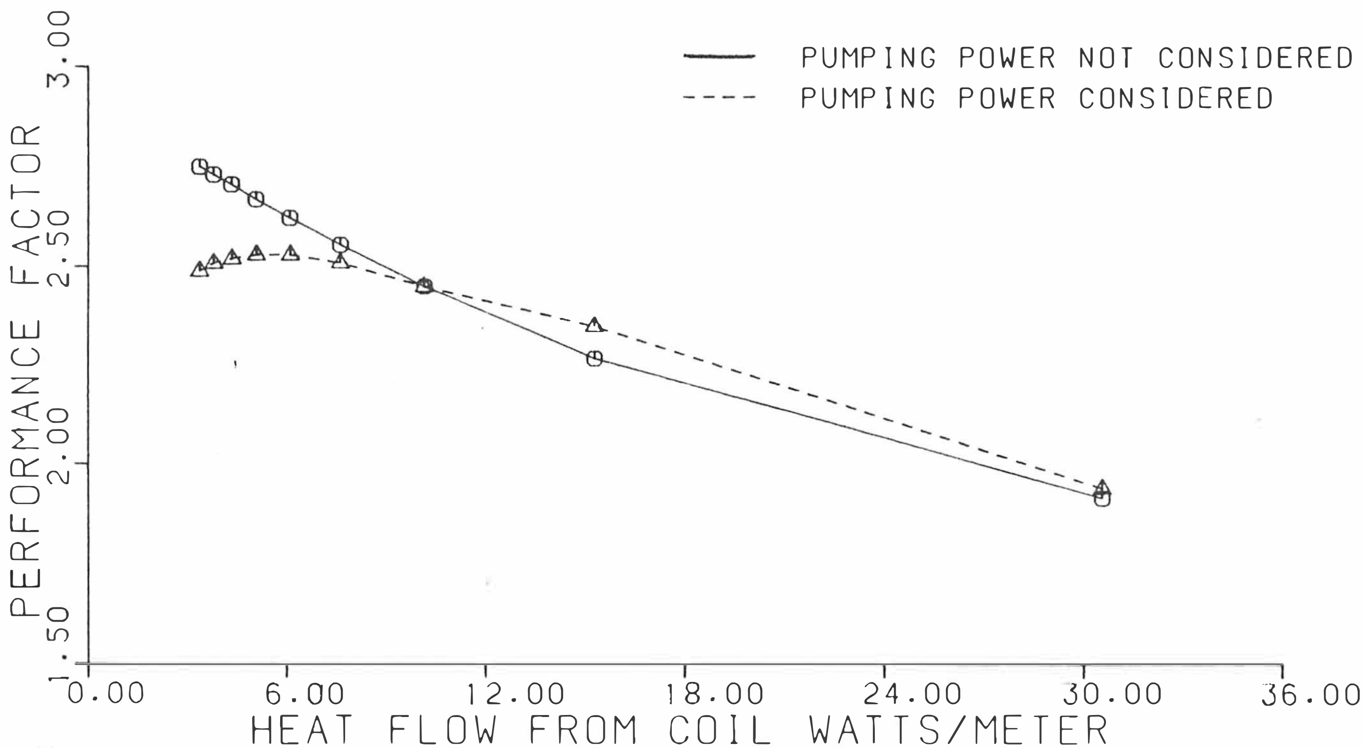


Figure 22. GROCS Prediction with Changing Heat Flux to the Ground

increased pipe length will diminish the predicted improvement in performance. If the assumption is made that the additional pumping power required would increase proportionally with the pipe length, the improvement is only 3% for the doubled pipe length. Figure 22 also shows that any further increase in pipe length will begin to decrease performance when the pumping power is taken into consideration.

Block Configuration

Three different block configuration designs for the ground of varying complexity were examined with the modeling. The simplest configuration was composed of 2 free blocks and had 4 interfaces between them. A more complicated configuration was examined which had 6 free blocks and 12 interfaces. Finally a block diagram which was modeled after Fisher's block diagram (4), was examined which had 20 free blocks and 42 interfaces. The model was executed with each configuration, and the seasonal performance predicted by the model was compared to the observed experimental performance. The results are discribed on Table 3. The Table shows a 5% difference between the predicted and the experimental performance with the simplest block design, and a 4% difference when using the block configuration with the maximum number of free blocks available. It is interesting that the intermediate block design provides the most accurate prediction. The predicted performances are all within 10% of the experimental performance.

Table 3. Results From Three Different Block Configurations

SIMULATION	6MO SPF	%DIFF
CONFIGURATION A	2.43	5.45
CONFIGURATION B	2.58	0.40
FISHER'S	2.46	4.28
Experimental	2.57	

These results indicate that an increase in complexity of the block configuration design does not always improve the forecasting ability of GROCS. The specific instructions for the design of the block configuration are arbitrary and leave much up to the discrimination of the investigator. In spite of the arbitrariness of the block diagram and the sensitivity of the model to the changes in the block diagram, none of the predicted performance factors for the different block designs are poor. Experience with the GROCS model and good engineering judgement can compensate for this limitation.

An attempt was made to examine the maximum possible performance of the heat pump system as predicted by GROCS. The experimentally derived thermal conductivity was not constant throughout the winter. It can be assumed that had the ground coil been irrigated that the thermal conductivity would have been constant at its maximum value. In addition, it would not be uncommon to have designed for twice the coil length in the ground to better accommodate for the summer cooling load. Comparing the extraploated performance curve of the TECH house heat pump with performance ratings from other water source heat pumps listed in the Air-Conditioning and Refrigeration Institute Directory shows that the TECH house heat pump performs as good or better for heating than other heat pumps at the rated conditions (42). A simulation was performed using twice the present coil length, the maximum monthly observed thermal conductivity, and using the performance curve derived for the heat pump and indoor heat exchanger system in place at the end of the season. A heating perfomance factor of 2.85 was predicted. Since the heat pump model underpredicted the performance of the experimental system by 5%, it is reasonable to predict a maximum performance of near 3.0 for this system with the above conditions.

Heat Pump Performance Curve

Because the experiment was subjected to a change of indoor coils in mid-season, and therefore to a change in its performance function, it is interesting to predict how the system would have performed if it had run the entire season with either air side coil. The analysis was performed using each of the heat pump performance curves described previously for the entire season. The results are shown on Table 4.

Table 4. Performance with Different Air Side Coils

	SPF	Ave Runtime
HP w/Old Coil	2.36	.300
HP w/New Coil	2.65	.275
HP w/Both	2.46	.290

The predicted improvement of the system performance with the replacement air side coil over the system with the original coil is 12%. This is a significant improvement in performance and illustrates the importance of a properly sized coil. A larger significance of this result is that the performance function of the heat pump can have a significant effect on performance when compared to the other parameters investigated.

CHAPTER VII

CONCLUSIONS

The ground-coupled heat pump system at TECH House I was found to provide adequate space heat for the 1982-83 winter. The experimental results obtained during the investigation are encouraging, and the system operated with an average winter performance factor of 2.57.

The modeling suggests that the heating system is not extremely sensitive to most of the input parameters. Little change in performance is predicted by changing the far-field soil temperatures. Little information is available which describes soil temperatures, however, common estimates should give sufficient results. Changing the thermal conductivity of the soil was found to influence the performance of the system only slightly. The thermal conductivity of the soil could be improved by irrigation. The experimental determination of the soil thermal conductivity demonstrated a change throughout the season which reached a maximum in March. The change in the thermal conductivity is attributed to the change in soil moisture. Irrigation by both feed water pipes and septic drains have been suggested in the literature. Increasing the coil length would increase the performance only slightly when taking into consideration the additional pumping power required. It is doubtful that the digging of an additional trench would prove economical due to the diminishing improvement of performance with increased coil length. Improvement in the heat pump performance curve was found to be the most significant factor in increasing the overall performance of the system.

The TRNSYS-GROCS modeling of the TECH house ground-coupled heat pump is found to be an adequate predictor of performance inspite of an arbitrary block configuration for modeling the ground and the need for the flowing sheet approximation. The claims by Metz that adequate accuracy can be obtained with GROCS seems justified by the results shown here. Modeling with different block configurations showed no significant difference in predicting performance for the system.

Discussion

No backup heat was used with the ground-coupled heat pump during the heating season. This has economic advantages for both the consumer and the local utility. Backup heat is usually provided by relatively expensive electrical resistance heaters. In addition, the auxiliary heat is usually required coincident with the utilities peak load time. Wide spread use of ground-coupled heat pumps could quell the demand for power during peak times - unlike the more common air-to-air heat pump systems which generally rely on resistance heating coincident with peak demand periods. A farsighted utility would investigate the use of economic incentives for a homeowner considering a ground-coupled heating system.

When assesing the potential of a consumer's interest in the ground-coupled heating system, the improvement in performance must be compared with the increase in initial capital cost over an air-to-air system. In any event, it is desireable to reduce capital cost expenditures when performance is only slightly impaired. Initial trenching costs could be lessened by placing two or more coils within the same trench. Since most of the temperature effect on the soil was felt within 15 centimeters distant from the pipe, the pipes should be separated vertically by at least 30 centimeters. The trenching costs could be at least halved by this method, and in addition the need for a large ground surface area is lessened.

The heat pump currently installed at the TECH site is rated by the manufacturer only between 4C and 16C fluid inlet temperature. Often the ground-coupled heat pump operated at lower inlet fluid temperatures, and in more northern parts of the country, such a system may operate with the inlet fluid temperature below freezing for a large part of the heating season. In addition, since this heat pump system is designed to serve both heating and cooling needs, one can expect that the entering water temperature in the summer will generally be much hotter than the high temperature rating. This creates a problem for a single stage compressor which connot operate at its optimum at both temperature extremes. Compressor flow modulation has been found benificial in improving performance in the air-to-air heat pumps, and the potential exists for this application with the ground-coupled heat pumps (43). Development of a water to air heat pump with a multistage compressor designed for ground-coupled applications could improve the overall efficiency of these systems.

Improvements in some of the instrumentation of the TECH house system would enhance the analysis. Most troublesome was the unfortunate location of the brine temperature probes within the heated space. These should have been placed within the buried section of the coil and several feet away from the house. Also the reading of instantaneous temperatures could be misleading. There was no indication if a temperature was measured while the heat pump was on. With an average runtime of about 30%, as established from actual and maximum potential pump power data, it is more likely that the system was off at any particular instant. A switch indicating whether the heat pump was on or off during a measurement would help to determine which temperature records were of more significant value.

The determination of the thermal properties of the soil also leaves room for improvement. Methods for determining soil thermal conductivity on site with minimal soil disturbance are available, and the use of these would be an improvement in determining soil thermal properties over using general data from soils related only by gross differences in grain size. It is concievable to design an experimental proceedure to determine the range of thermal properties and soil moistures and to apply these values to an empirical relation describing the change in these properties over the seasons. Accurate and durable soil moisture sensors should be installed before any relationship between soil moisture and thermal conductivities can be approximated.

Little guidance is available from the GROCS literature in suggesting specific types of block configurations or specific limitations regarding the block architecture. If the block adjacent to the heat source is too small, convergence may become difficult. In addition, if this block is small, the flowing sheet approximation becomes more unrealistic. The flowing sheet approximation assumes a constant temperature boundary at the interface of the adjacent block with the nearest free block. This requires that the block adjacent to the heat source have a minimum thickness before the approximation can be considered realistic. Further, the flowing sheet approximation can be used only for coil fields with limited coil geometries. There is no apparent justification for using this approximation to model single isolated pipes, two pipes within one trench or other arrangements of ground pipes such as vertical grids. Observation of the temperature profiles of the soil show that most of the significant temperature changes occur near the pipe, however, GROCS lumps each block as having a uniform temperature. Reliance on the flowing sheet approximation resricts the ability of the model in representing the actual physical phenomenon. The deviation from the physical approach to modeling could make future development of GROCS more difficult. It would be difficult in the GROCS model to include the effects of freezing around the pipe, or the effect of local drying of the pipe during the cooling season as often occurs. While GROCS is reportedly suited for ground-coupled tank analysis, its application with ground-coupled pipe coils is more tenuous. Furthermore, it is suggested by modeling different block configurations that there may be little advantage to including a detailed mesh of free blocks around the adjacent block pairs. The difference in the performance prediction from using a 2 free block mesh and the maximum 20 block mesh is only 5%. In the GROCS manual Andrews states that naturally occuring ground inhomogeneities limit the accuracy of any model to about 10%, and that it may be a waste to use models more accurate than this limit. Based on this opinion, perhaps a model dealing with only the soil near the ground coil would be more appropriate. Other models or alternatives for the flowing sheet approximation should be compared with GROCS in future studies.

Future studies should combine the analysis of both heating and cooling seasons. Although ground coils have been used extensively for heating only heat pumps in Scandinavian countries in Europe and to a lesser extent in the Northern United States, limited experience is available with regard to the cooling application of ground coil heat pumps. In addition, the TRNSYS model should be more completely used to model the entire house using only weather data as input to the model. This modeling approach would allow for regional studies.

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APPENDIXES

APPENDIX A

SOIL PROPERTIES

Property	Disturbed Soil	Undisturbed Soil
Natural Moisture	24%	17%
Void Ratio	1.12	0.536
Wet Density	98.09lbs/ft3	124.68lbs/ft3
Degree of Saturation	54.4%	78.04%
Grain Size	(15% Sand and 85% Clay and Silt)	

Sample Calculation Undisturbed Soil

1. Calculate Dry Density
   Density = 124.68lbs/ft3 at 17% moisture
   124.68lbs/ft3 X 0.83 = 103.48lbs/ft3 = Dry Density

2. Calculate Void Volume
   Void Ratio = 0.536
   Vs + Vv = 1 and Vv/Vs = 0.536
   Vv/(1-Vv) = 0.536 --- Vv = 0.349

3. Calculate Saturated Density
   103.48lbs/ft3 + 62.4lbs/ft3 X (0.349) = 125.26lbs/ft3

4. Saturated Moisture
   62.4lbs/ft3 X Vv / 125.26lbs/ft3 = 17.4%

APPENDIX B

Data Summary

Specific data pertinent to the performance of the ground-coupled heat pump were recorded hourly from November 1, 1982 through April 30, 1983. During these 26 weeks, data were obtained for 94.3% of the total 4368 hours. A summary of selected data is shown on Table A-1.

Interruptions in the data record were delt with depending on how the data was to be used. Erroneous data was typically displayed as an unrealistcally high value, and data was screened by assigning limits for the values for each type of data. Temperature values exceeding 75 C and heat flows exceeding 25,000 watts/hour were considered in error. When daily average data was of interest, the values for each data below its respective upper limit was summed and averaged. If more than half of the data for a value for a particular day exceeded this limit, then no value was taken as that day's average. When hourly input was required erroneous data was replaced by that value's hourly average for the entire season. Only one period experienced significant loss of data due to a malfunction in the Data Aquisition System. This week also experienced the mildest temperature of the heating season and little useful data was lost.

Table A-1. Weekly Summary of Selected Data
Ground-Coupled Heat Pump Study
November 1982 - April 1983

Week Begining	Total Energy Consumed (MKJ)	Condenser Loop Heat Transfer (MKJ)	Evaporator Loop Heat Transfer (MKJ)	Weekly Performance	Operating Time	Degree Days F
NOV01	483	940	870	2.80	18.9%	114
NOV08	459	886	863	2.88	18.5%	124
NOV15	519	1001	884	2.70	21.0%	106
NOV22	456	894	931	3.04	18.2%	98
NOV29	159	266	251	2.59	5.2%	53
DEC06	665	1454	1183	2.78	27.7%	144
DEC13	1194	2726	1987	2.66	52.8%	195
DEC20	538	1210	792	2.47	23.1%	109
DEC27	734	1619	1007	2.37	31.6%	133
JAN03	1022	2067	1369	2.34	45.1%	182
JAN10	1106	1946	1552	2.40	49.6%	202
JAN17	1343	2255	1913	2.42	64.3%	221
JAN24	966	1592	1345	2.39	45.9%	175
JAN31	1157	1944	1504	2.30	60.0%	189
FEB07	1178	2003	1464	2.24	56.6%	203
FEB14	545	1167	884	2.62	27.3%	132
FEB21	672	1581	1187	2.77	34.3%	154
FEB28	297	603	497	2.67	13.6%	70
MAR07	***	***	***	1.00	0.0%	19
MAR14	171	320	247	2.44	5.8%	69
MAR21	927	2173	1686	2.82	46.0%	169
MAR28	575	1281	1042	2.81	27.0%	128
APR04	239	495	408	2.71	10.1%	68
APR11	346	801	643	2.86	15.7%	86
APR18	717	1747	1383	2.93	35.2%	155
APR25	114	267	220	2.93	5.5%	26
TOTAL	16582	33238	26112	2.57	29.0%	3324
AVERAGE	663	1329	1044	2.64	29.2%	128

APPENDIX C

GROCS Configurations

The program GROCS solves the heat flow finite difference equations over a system of "blocks" of earth. Each block is a volume of earth whose size and shape are determined by a hand drawn model (38). During the course of the study, 3 different hand drawn block configuration models were investigated. One of the models was adapted from the work of Fisher (4), and the other two models were of simpler design so that the effect of the block design could be examined. To model the pipe field, the pipes were assumed to be oriented in a serpentine array within a 9 meter by 30 meter area. This area was also the projected dimension of the adjacent blocks. The other free and the rigged blocks were built around the adjacent blocks. The blocks were shaped either as a rectangular parallelepiped or similar to a picture frame shape (as in Fisher's model). The following figures represent the 3 block models. In each figure the pipe field is located 1.2 meters below the surface, and the dimensions are proportional to the size of the adjacent blocks.

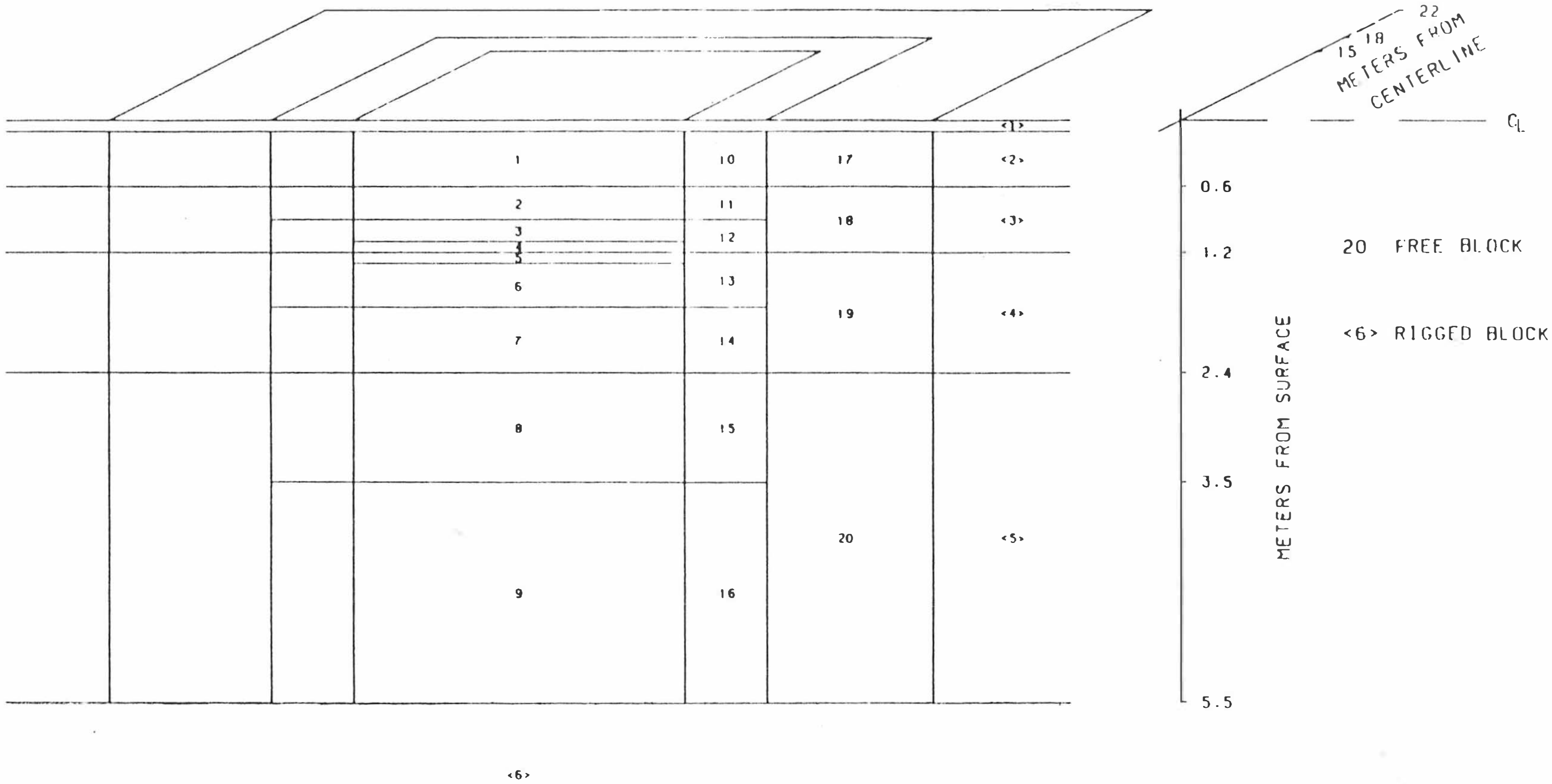
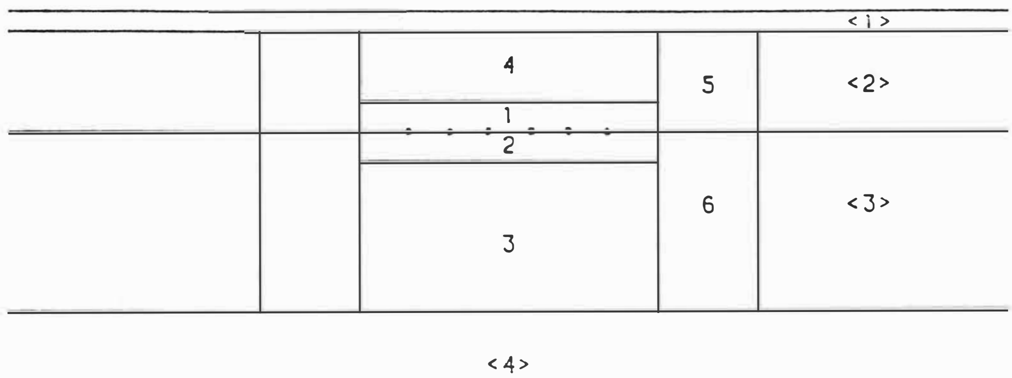
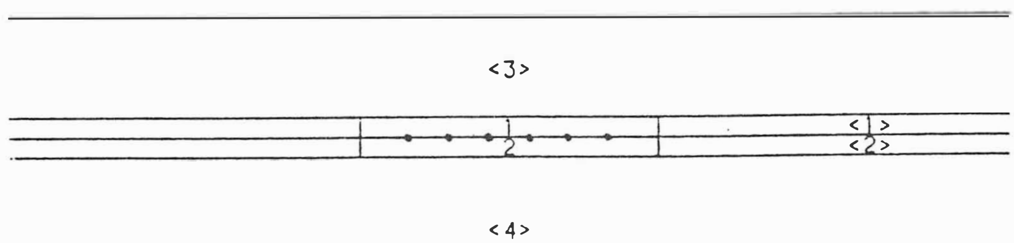


Figure C-1. Fisher's Earth Block Model.



BLOCK CONFIGURATION B



BLOCK CONFIGURATION A

Figure C-2. Earth Block Model's A and B.

APPENDIX D

TRNSYS ALTERATIONS

In order for TRNSYS to retain auxiliary heat and also for it to use two different heat pump curves for each of the two air coils described in Chapter 5, some minor alterations were made to the heat pump subroutine TYPE20.FOR. The program is listed in full. The corrections are found between the pairs of comment cards C(FC)****1 and C(FC)****2.

TYPE20.FOR

```
C.
SUBROUTINE TYPE20(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C  THIS ROUTINE MODELS A HEAT PUMP CAPABLE OF USING AIR OR
C  LIQUID SINKS AND SOURCES AND OPERATING IN A VARIETY OF
C  MODES. THE INTERNAL CONTROLLER WILL STICK AFTER 4
C  ITERATIONS AT A GIVEN TIMESTEP IF IT HAS NOT SETTLED ON
C  A MODE BY THAT TIME. THIS CAN BE CHANGED BY MODIFYING
C  THE VALUE OF NSTK IN THE DATA STATEMENT.
C THE FOLLOWING MODE CODE CONVENTION IS USED:
C   MODE=1 - HEAT WITH SOLAR SOURCE EVAPORATOR
C   MODE=2 - HEAT WITH OUTDOOR SOURCE EVAPORATOR
C   MODE=3 - COOL WITH SOLAR SINK CONDENSER
C   MODE=4 - COOL WITH OUTDOOR SINK CONDENSER
C   MODE=5 - DIRECT HEATING FROM SOLAR SOURCE
C   MODE=6 - AUXILIARY
C   MODE=7 - EVERYTHING OFF
CTHE FOLLOWING MODE COMBINATIONS MAY OCCUR TOGETHER:
C 2-5,2-5-6,1-6,2-6,5-6
 REAL MDOT1,MDOTO
 DIMENSION XIN(10),OUT(20),PAR(18),INFO(10),Y(3)
 DATA NSTK/4/,IUNIT/0/
  WAH=0.
  QA1=0.
  QR1=0.
  H1=0.
```

```
  H2=0.
  H3=0.
  QDH=0.
  QAC=0.
  QRC=0.
  WAC=0.
  QR2=0.
  QA2=0.
  QAUXH=0.
  QSHRTC=0.
 IF(INFO(7).GE.0) GO TO 10
 INFO(6)=16
 INFO(9)=1
 OUT(19)=0.
 IF(PAR(15).LT.0..OR.PAR(15).GT.1.)
*CALL TYPECK(4,INFO,0,0,0)
 CALL TYPECK(1,INFO,5,18,0)
10    IF (INFO(1) .EQ. IUNIT) GO TO 12
 IUNIT = INFO(1)
 CP1=PAR(1)
 MDOT1=PAR(2)
 TROOMH=PAR(3)
 TROOMC=PAR(4)
 TMIN1=PAR(5)
 TMIN2=PAR(6)
 NDATA1=PAR(7)
 NDATA2=PAR(8)
 NDATAC=PAR(9)
 ISAVE1=PAR(10)
 ISAVE2=PAR(11)
 ISAVEC=PAR(12)
 TCOOL=PAR(13)
 CAIR=PAR(14)
 EFF=PAR(15)
 TSET=PAR(16)
 ICOOL=PAR(17)
 THEAT=PAR(18)
12    TTANK=XIN(1)
C(FC)**********************1
 IF(INFO(7).EQ.0)QLOAD=XIN(4)-EXTRAQ
C(FC)**********************1
 TAMB=XIN(3)
 IGAM=XIN(5)
 MODE=IFIX(OUT(19)+.01)
 MDOTO=MDOT1
C  CHECK NUMBER OF ITERATIONS
 IF(INFO(7).LE.NSTK) GO TO 100
C    USE PREVIOUS MODE
 GO TO (310,400,520,520,110,600,700) ,MODE
C    FIND NEW MODE
100   IF(IGAM.EQ.0) GO TO 700
 IF (QLOAD.GE.0.) GO TO 500
```

```
      IF(TAMB.GT.THEAT) GO TO 700
      IF (TTANK.LE.TSET) GO TO 300
C  DIRECT HEATING FROM STORAGE TANK
110   MODE =5
      CHOT=MDOT1*CP1
      IF (CAIR.LE.CHOT) GO TO 120
      QMAX=CHOT*(TTANK-TROOMH)
      GO TO 140
120   QMAX=CAIR*(TTANK-TROOMH)
140   QTRAN=EFF*QMAX
      HRSON4=ABS(QLOAD/QTRAN)
      IF(HRSON4.LE.1) GO TO 155
      HRSON4=1.
      QAUXH=-(QLOAD+QTRAN)
155   QDH=QTRAN*HRSON4
      TOUT=TTANK-QDH/CHOT
C  CHECK AIR SOURCE HEAT PUMP AS AN AUXILIARY TO DIRECT HEAT.
C  DERATE HEAT PUMP PERFORMANCE BY DECREASING TAMB.
      TSRC=TAMB-QDH/CAIR
      IF(QAUXH.LE.0.0.OR.TAMB.LT.TMIN2) GO TO 800
      CALL DATA(ISAVE2,NDATA2,3,TSRC,Y,INFO)
      QA=Y(1)
      QR=Y(2)
      WA=Y(3)
      IF(QR.LE.0.) GO TO 800
      H2=QAUXH/QR
      IF(H2-1.) 159,159,158
158   H2=1.
      QAUXH=QAUXH-QR
      GO TO 160
159   QAUXH=0.
160   WAH=WA*H2
      QR2=QR*H2
      QA2=QA*H2
      GO TO 800
C  SYSTEM IS IN MODE 1,2,OR 6
300   IF(TTANK.LE.TMIN1.AND.TAMB.LE.TMIN2) GO TO 600
      IF(TTANK.LE.TAMB.AND.TAMB.GE.TMIN2) GO TO 400
      IF(TTANK.LE.TMIN1.AND.TTANK.GT.TAMB.AND.TAMB.GE.TMIN2)
     *GO TO 400
C  HEAT FROM SOLAR SOURCE EVAPORATOR
310   MODE=1
C(FC)********************************2
      IF(XIN(2).EQ.0.0)GO TO 700
      IF(TIME.GE.2520)GO TO 305
      CALL DATA(ISAVE1,NDATA1,3,TTANK,Y,INFO)
      GO TO 315
305   CALL DATA(ISAVE2,NDATA2,3,TTANK,Y,INFO)
315   CONTINUE
      QA=Y(1)
      QR=Y(2)
      WA=Y(3)
```

```
 IF(QR.LE.0.0) GO TO 600
 H1=-QLOAD/QR
 IF (H1.LE.1.) GO TO 360
 H1=1.
 QAUXH=-(QLOAD+QR)
360   WAH=WA*H1
 QA1=QA*H1
 QR1=QR*H1
 EXTRAQ=QAUXH
  TOUT=TTANK-(QA/(MDOT1*CP1))
 MDOTO=XIN(2)
C(FC)*****************************2
 GO TO 800
400   CONTINUE
C  HEAT FROM OUTDOOR SOURCE EVAPORATOR
 MODE=2
 CALL DATA(ISAVE2,NDATA2,3,TAMB,Y,INFO)
 QA=Y(1)
 QR=Y(2)
 WA=Y(3)
 IF(QR.LE.0.0) GO TO 600
 H2=-QLOAD/QR
 IF (H2.LE.1.) GO TO 460
 H2=1.
 QAUXH=-(QLOAD+QR)
460   WAH=WA*H2
 QR2=QR*H2
 QA2=QA*H2
 TOUT=TTANK
 MDOTO=0.
 GO TO 800
C  COOLING MODE
500   IF(TAMB.LE.TCOOL)GO TO 700
 MODE=3+ICOOL
520   TSRC = TAMB
 IF (ICOOL .EQ. 0) TSRC = TTANK
 CALL DATA(ISAVEC,NDATAC,3,TSRC,Y,INFO)
 QA=Y(1)
 QR=Y(2)
 WA=Y(3)
 IF(QA.LE.0.0) GO TO 600
 H3=QLOAD/QA
 IF (H3.LE.1.) GO TO 558
 H3=1.
 QSHRTC=QLOAD-QA
 GO TO 560
558   QSHRTC=0.0
560   WAC=WA*H3
 QAC=QA*H3
 QRC=QR*H3
 IF(ICOOL.GE.1) GO TO 595
 TOUT=TTANK+QRC/(MDOT1*CP1)
```

```
      GO TO 800
595   TOUT=TTANK
      MDOTO=0.
      GO TO 800
C  AUXILIARY ONLY OPERATION
600   MODE=6
      IF(QLOAD.GE.0.) GO TO 620
      QAUXH=-QLOAD
      GO TO 640
620   QSHRTC=QLOAD
640   MDOTO=0.
      TOUT=TTANK
      GO TO 800
C  HEAT PUMP NOT OPERATING
700   MODE=7
      TOUT=TTANK
      MDOTO=0.
800   OUT(1)=TOUT
      OUT(2)=MDOTO
      OUT(3)=WAH
      OUT(4)=QA1
      OUT(5)=QR1
      OUT(6)=H1
      OUT(7)=H2
      OUT(8)=H3
      OUT(9)=QDH
      OUT(10)=QAUXH
      OUT(11)=QSHRTC
      OUT(12)=QAC
      OUT(13)=QRC
      OUT(14)=WAC
      OUT(15)=QR2
      OUT(16)=QA2
      OUT(19)=MODE
      RETURN
      END
```

APPENDIX E

Experimental Error Analysis

Heat Flow Measurements

The method used to measure the heat flows in the ground-coupled heat pump system require that an error analysis be carried out to calculate the uncertainty in these measurements. As described in Chapter 3, the heat flow is measured by multiplying the mass flow rate, the temperature change and the specific heat of the fluid in the water to ground and the water to air coils. The flow meter produces a pulse signal whose frequency is proportional to the flow rate, and the temperature sensors produce a signal whose frequency is proportional to the temperature difference. The flow is then equal to the product of these frequencies and a calibrated constant (which includes the heat capacity and any conversion constants).

$$\underset{\sim}{Q} = K\ (Fm)\ (Ft)$$

The uncertainty of a measurement is expressed by Holman (43) as:

$$W_r = \left[\left(\frac{\partial R}{\partial X_1}\ Wx_1\right)^2 + \left(\frac{\partial R}{\partial X_2}\ Wx_2\right)^2 + \ldots + \left(\frac{\partial R}{Xn}\ Wx_n\right)^2\right]^{1/2}$$

Where Wr is the uncertainty of the equation:

$$R = R\ (x1, x2, x3, \ldots, xn)$$

The reported uncertainties of the independent variables for the heat flow measurements are:

$$Wk/K = \pm\ 7.4\%$$

$$WFm/Fm = \pm\ 0.5\%$$

$$WFt/Ft = \pm\ 1.0\%$$

Therefore the total uncertainty can be expressed as:

$$\frac{Wq}{Q} = \left[\left(\frac{Wk}{K}\right)^2 + \left(\frac{WFm}{Fm}\right)^2 + \left(\frac{WF\Delta t}{F\Delta t}\right)^2\right]^{\frac{1}{2}} = 7.5\%$$

It is seen that most of the uncertainty is due to the observed change in the calibration constant. The calibration constant was found to vary between each calibration throughout the investigation. An investigation into this matter could yield more accuracy in this measurement.

Ground Temperature Measurement

The manufacturer of the resistance temperature devices located in the ground reports an accuracy of ± 1.3% (44). The maximum error associated with the Data Acquisition System is 0.5%. Together these account for a maximum possible error of 1.8% for the ground temperatures.

Electrical Consumption Measurement

The watt-hour submeters had been calibrated to within 0.5%. The maximum error accociated with the Data Acquisition System is 0.5%. Together these account for a maximum possible error of 1.0% for the power consumption measurement (45).

VITA

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After working as an engineer for two years with a Knoxville based engineering company, he entered the Graduate School of the University of Tennessee, Knoxville in March 1982. He received a Master of Science degree in Engineering Science specializing in the thermal sciences in August 1984.