SIMULATION AND EXPERIMENTAL VERIFICATION OF VERTICAL GROUND-COUPLED HEAT PUMP SYSTEMS

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PREFACE

This work investigates the performance characteristics of vertical ground-couplings that are used as a scurce and sink for water-to-air heat pumps. Two methods of predicting heat transfer capabilities of various coupling designs are compared with the results of an experimental system. Recommendations are made based upon the degree of accuracy that is desired. Procedures are provided for system design and simulation. Heat pump capacity and power requirements can be determined with computer algorithms or by hand calculations.

I wish to express my deepest gratitude to Dr. Jerald Parker for his unselfish commitment to me. He has provided direction, insight and an example of a kind and generous educator. I wish to also thank Dr. James Bose for his encouragement, advice and assistance with the experimental system. I also appreciate the help given by the other members of my committee, Drs. David Lilley, Afshin Ghajar and Jan Wagner.

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My wife, Penny, and sons, Kris and Kevin, have given me the desire to undertake this work. Their love and support is deeply appreciated and treasured. I thank them for their patience and understanding with of my lack of presence in these past few months.

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NOMENCLATURE

A	-	Area
ACH	-	Air change per hour
α	-	Thermal diffusivity $(k/\rho c_p)$
β	-	Coefficient of thermal expansion
C	-	Correction factor
c_p	-	Specific heat
CFM	-	Cubic feet per minute
COP	-	Coefficient of performance
D	-	Diameter
D_h	-	Hydraulic diameter
DD	-	Degree-day
dt, Δt	-	Time increment
E	-	Energy
EER	-	Energy efficiency ratio
FDE	-	Finite difference equation
Fo	-	Fourier number $(\alpha t/r^2)$
g	-	Gravitational constant
GPM	-	Gallons per minute
Gr	-	Grashof number $(g\beta\rho^2 \ell^3 \Delta t/\mu^2)$
h	-	Heat transfer coefficient
j	-	J-Factor $(h\ Pr^{2/3}[\mu_w/\mu_b]^{.14}/c_p\rho V)$
k	-	Thermal conductivity
L,1	-	Length

LMTD	-	Log-mean temperature difference
M	-	Node number in vertical directioin
m	-	Mass flow rate
MTD	-	Mean temperature difference
μ	-	Absolute viscosity
N	-	Node number in radial direction
Nt	-	Number of tubes
Nu	-	Nusselt number (hD/k)
Q	-	Volumetric flow rate
q	-	Heat transfer rate
P	-	Power
p	-	Dimensionless radius (r/r_o)
PB	-	Polybutylene
PE	-	Polyethylene
PVC	-	Poly vinyl chloride
Pr	-	Prandtl number $(\mu c_p/k)$
ϕ	-	Percent moisture
ρ	-	Density
R	-	Thermal resistance $(\Delta T/q)$
r	-	Radius
Rf	-	Run fraction
SDR	-	Standard dimension ratio $(D_o/2[D_o - D_i])$
Se	-	Expansion factor
SIDR	-	Standard inside dimension ratio $(D_i/2[D_o - D_i])$
SPF	-	Seasonal performance factor
T	-	Temperature
t	-	Time

TDMA	-	Tridiagonal matrix algorithm
TWB	-	Air wet bulb temperature
θ	-	Angular dimension in cylindrical coordinate
U	-	Heat transmission coefficient
V	-	Velocity
VB	-	Variable base
X	-	Line source equation factor ($r/w\sqrt{\alpha t}$)
x	-	U-tube separation distance
z	-	Axial dimension in cylindrical coordinates, Fo in Reference 39

Subscripts

a	-	Annular, average
ai	-	Inside annular
ao	-	Outside annular
b	-	Bulk water value
bl	-	Boundary layer
dl	-	Design load
eq	-	Equivalent
f	-	Final
ff	-	Far field
g	-	Ground
gc	-	Ground coupling, Ground-coupled
hp	-	Heat pump
i	-	In, Inner, Indoor
if	-	Infiltration
p	-	Pipe
o	-	Out, Outer, Outdoor

r	-	Rejected
sc	-	Short circuit
w	-	Water

Superscripts

$\overline{}$	-	Average quantity
$'$	-	Quantity at time $t + \Delta t$

CHAPTER I

INTRODUCTION

1.1 Basics of Closed Loop Ground-Coupled Heat Pumps

Rising energy costs have led to a consumer interest in water source heat pumps for residential use. By utilizing a heat pump, both heating and cooling can be provided by a single piece of equipment. Electric utilities have a particular interest in heat pumps since they show promise of leveling their load throughout the year. Water source heat pumps offer an advantage over the more common air-to-air types, in that heat is rejected or absorbed through water, a generally more desirable heat transfer medium. Installation of these units is greatly simplified if an open source of water is available, such as a well water. These types of water sources are often not present in the proper location and quantity required. In many instances savings are nullified because of the energy consumed by well pumps.

An alternative is closed loop ground-coupled heat pumps (CLGCHP). In these systems, water is circulated between a heat pump and a piping system buried in the ground. The water rejects or absorbs heat from the ground. Ground temperatures vary less than local air temperatures. A recently completed study at Oklahoma State University has shown that by using the ground for the heat pump source or sink, both high COP's and EER's can be realized (1). A significant finding was the fact that resistance heat was unnecessary even on the coldest days of the 1982-1983 winter.

Closed loop ground-couplings are typically divided into two types, horizontal and vertical. Horizontal types are installed with a trenching machine and in some cases with a backhoe. Burial depths are typically four feet below the surface. Better performance is possible with greater depths and with multiple pipes installed at different locations in the same trench.

This thesis will be limited to study of vertical ground-coupled heat pump systems. In many applications horizontal systems may be a better alternative when economics, local climate and geological conditions are considered. However, in many other cases vertical systems are a better option.

Figure 1 is a schematic of a typical water source/sink heat pump and two vertical ground-couplings used in previous installations. The heat pump unit, which includes the compressor, reversing valve, fan, and two heat exchangers, is located indoors. Water pumps and any additional flow meters and valves are also normally indoors. Indoor noise levels are comparable to those air source heat pumps or furnace type heating and cooling equipment. Water or a water and antifreeze mixture is pumped through the refrigerant-to-water heat exchanger and then into the ground-coupling where it absorbs or rejects heat before returning to the heat pump.

Vertical ground-couplings offer two primary advantages over horizontal types. First, the amount of ground area required is much less, and this is a significant factor in areas of increasing population density. Secondly, the properties of ground in contact with a vertical

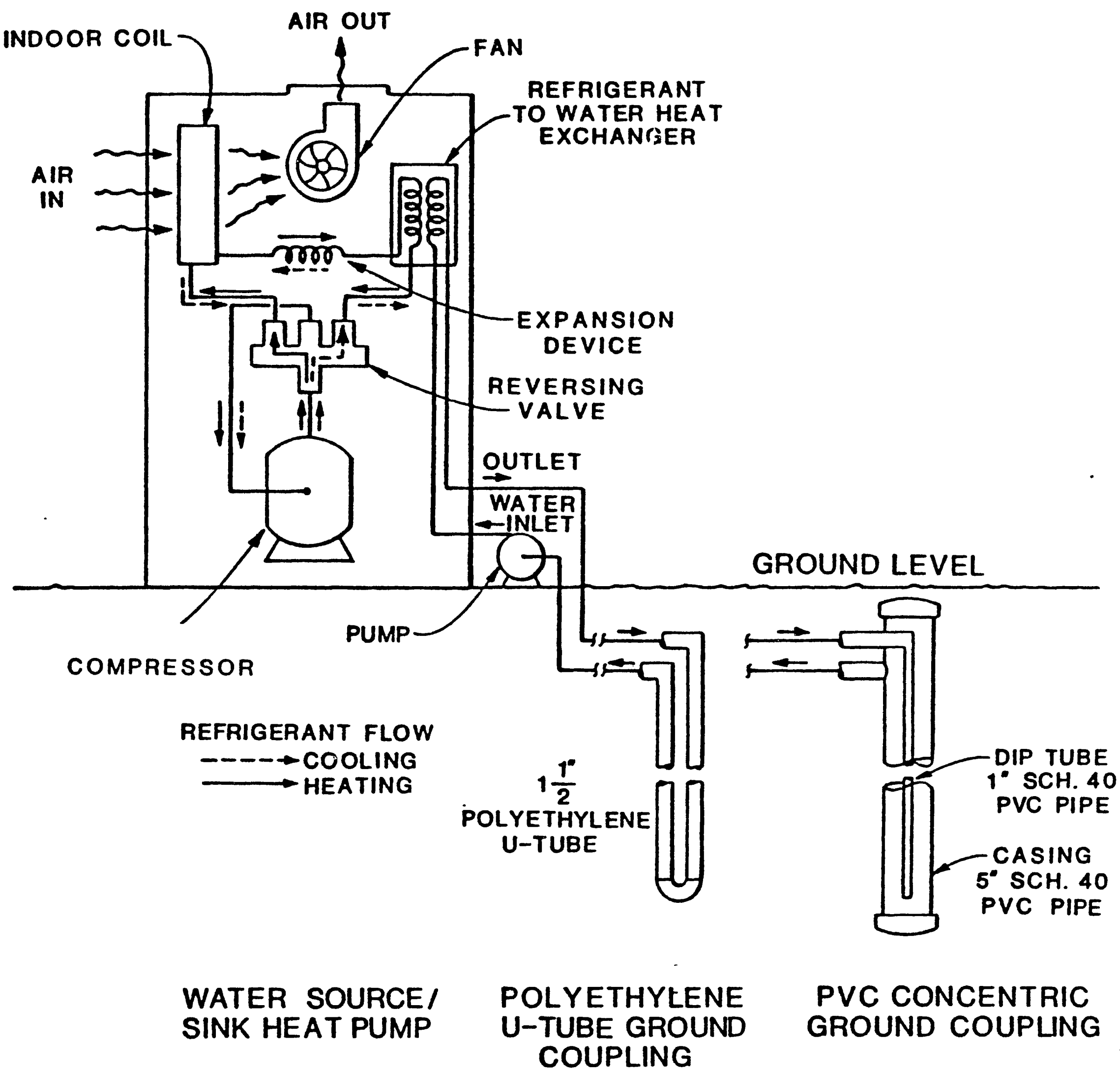


Figure 1. Vertical Closed Loop Ground-Coupled Heat Pump System

ground-coupling are usually more advantageous to good heat transfer. Ground temperatures at depths of greater than 20 feet have negligible annual variation, while these temperatures may vary 10^{o} to $20^{o}F$ above and below the average yearly air temperature at depth of four feet (2). The effective ground thermal conductivity is usually higher because of the generally greater moisture content of soils surrounding vertical ground-couplings. Therefore, the potential exists in vertical installations of warmer water temperatures in the winter and cooler temperatures in the summer than those possible in horizontal systems.

1.2 Heat Pump Fundamentals

The Coefficient of Performance (COP) of a heat pump is primarily a function of the temperature difference between the source and sink. The smaller this difference the higher the COP. Properly installed CLGCHP's have a much greater potential for minimizing this difference than air coupled systems. The effects of this can be seen when examining basic thermodynamics and heat pump manufacturer's performance data.

The heat pump process can be idealized in a Carnot cycle. The coefficient of performance (COP) for a Carnot heat pump is

$$COP = \frac{T_H}{T_H - T_C} \text{ (Heating)}, \quad COP = \frac{T_C}{T_H - T_C} \text{ (Cooling)}, \tag{1.1}$$

where T must be absolute temperature (Kelvin or Rankine). From these equations it can be seen that by reducing the difference between the high temperature T_H (condenser in cooling, evaporator in heating) and the low temperature T_C (evaporator in cooling, condenser in heating) the COP is increased.

Figure 2 also illustrates this point on a pressure-enthalpy chart. This chart represents the cooling cycle of a heat pump using two different condenser pressures. Path 1-2 represents condensation (heat rejection), 2-3 expansion, 3-4 evaporation (desired cooling), and 4-1 compression (work input). A lower condenser temperature results in a lower condenser pressure (represented by line 1A-2A) and a lower compressor discharge pressure and therefore the work input from 4-1A is less than 4-1B, yet the desired cooling effect (3-4) remains the same. In the heating mode path 1-2 is unchanged but the 3-4 isobar would have a higher value for warmer water and the value of work required would again be less.

These thermodynamic principles point to the value of minimizing the difference in temperature between the evaporator and condenser. Figure 3 is a diagram further illustrating this point by comparing the efficiencies of three different heat pumps to the temperature of the incoming water. The figure shows that heating performance (COP) increases with warmer water and cooling performance (EER) increases with cooler water temperature.

1.3 Economics of Installations

The variation of installed cost of CLGCHP systems is quite large compared to other types of residential and commercial heating ventilation and air-conditioning (HVAC) systems. Total equipment cost of a CLGCHP is comparable to or less than an air-coupled system of standard efficiency or a refrigeration cooling/fossil fuel heating system. However, installation cost vary significantly with local ground conditions, type of installation equipment used, availability of

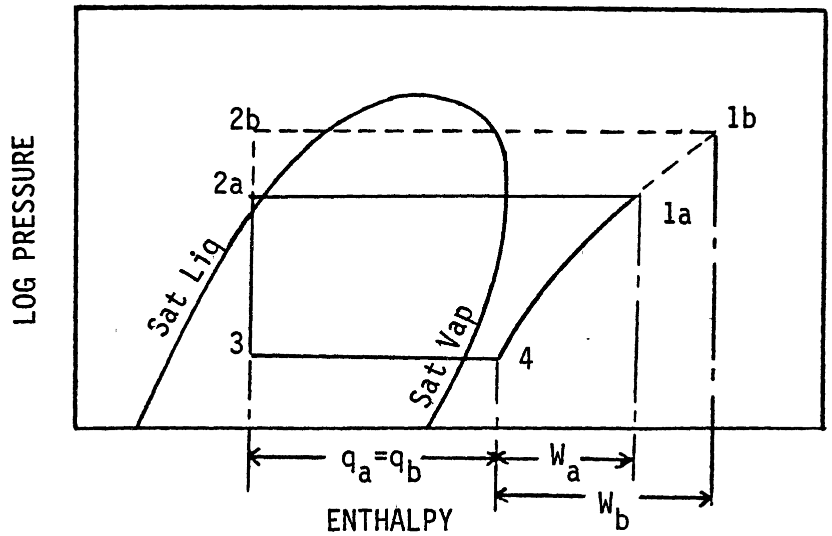


Figure 2. Pressure-Enthalpy Diagram for Heat Pump

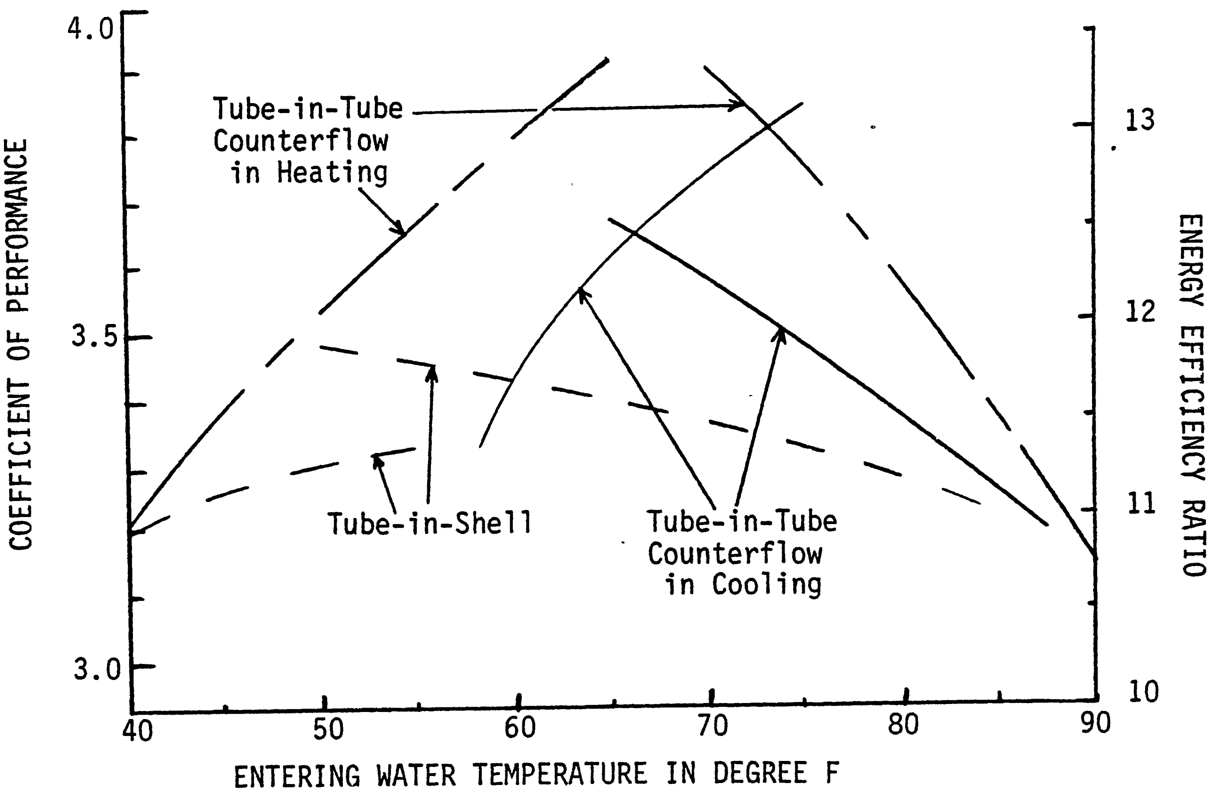


Figure 3. Water-to-Air Heat Pump Performance (30) (31)

coupling materials and qualified personnel. An installer in the Waco, Texas, area charges \$3.00/ft. of vertical bore using a small trailer mounted drilling rig and 3/4" U-tube ground-couplings. However, this type of drilling is neither available or adequate in many locations. In many areas much larger equipment is needed and costs may exceed \$10.00/ft. In central Oklahoma, installed ground-coupling costs are typically \$5.00/ft. (including pipe). This would therefore increase the installation cost \$750.00 per nominal ton. Air-coupled systems of course do not have this additional cost.

The calculation of economic payback is therefore critical to successful marketing of CLGCHP systems. However, this calculation is complicated due to the variation in performance of different types of vertical ground-couplings in different locations. As is the case in almost all equipment installations, long term energy savings can be increased by increased quality and cost of equipment.

The cost of plastic pipe is a relatively small portion of the overall cost of a ground-coupling. In order to significantly improve payback, research must be directed at installing ground-couplings that reduce the cost of required bore. In addition to design optimization, some type of seasonal energy consumption prediction procedure must be used in order to properly calculate economic payback of CLGCHP systems. Actual installations have proven to be a viable option in many heating and cooling applications in Oklahoma. Initial costs have limited the market acceptability of these high efficiency systems. In order to further reduce the costs of these systems, additional information is needed concerning the nature of heat transfer to and from water circulating through buried plastic pipes.

1.4 Present Status of Design Procedures

In the 1983 Transactions of the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE), the results of a comprehensive study by Ball, R. Fischer and Hodgett (3) are summarized.

> "Experience with vertical coils systems is much less than for horizontal systems. The only well known well-instrumented systems are in Sweden . . . the vertical ground coil system in a clay soil has a SPF similar to those of optimal horizontal systems but require half the surface area."

Although the report was heavily weighted toward horizontal CLGCHP systems, the authors concluded:

> ". . . no publicly available design guidelines exist at the present time . . . major uncertainties exist . . . for systems with substantial cooling operation, because of inability . . . to deal with moisture migration and soil recession."

The authors do not indicate if these shortcomings effect vertical ground-couplings significantly.

ASHRAE has recognized both the potential of CLGCHP systems and the necessity of comprehensive design guidelines. Therefore the society has commissioned a project to assemble and publish design methodologies. This report, Data Design Manual for Closed-Loop Ground-Coupled Heat Pump Systems (4), will cover the many complexities of design and installation. It recognizes the continuing need for additional experimentation and development of design methods in this relatively new field in HVAC.

As indicated in the above quote, vertical installations are less common than horizontal ones. Many questions remain unanswered concerning optimum arrangements of vertical CLCSHP systems. A variety of ground-coupling designs are used and general trends concerning efficient installation are known. However, direct comparisons cannot be accurately made between systems because of the large number of variables that cannot easily be controlled or measured. These variables include ground thermal properties, ground moisture content, local ground water movement, heat pump operational characteristics and system design and installation procedures. The literature survey of Chapter II discusses specifically the present status of research and installation method of vertical CLGCHP systems.

1.5 Plan of Attack

The optimization of vertical CLGCHP systems requires the development of a valid model that predicts operation during peak heating and cooling loads and estimates seasonal energy consumption. A literature search of material related to this topic indicates there are two primary areas that need further investigation. These are:

1. Relative performance of various vertical ground-coupling designs.
2. Inside heat transfer coefficients in ground-couplings for forced, free and mixed convection.

Therefore, two experimental systems were designed and installed to investigate these areas.

Chapter II deals with the relative performance of six various ground-coupling designs. This chapter includes a literature survey, a description of the experimental system and the results of a one year experimental test. Chapter III reports on the test concerned with heat transfer coefficients in a vertical ground-coupling. It also includes a literature survey, system description and experimental results. These two installations provide necessary information for model development and validation.

Model development will be in several stages ranging from complex to simplified. Computer simulation utilizing finite difference equations (FDE's) are developed and explained in Chapter IV. The first simulation utilizes an explicit formulation (forward time step) and applies only to the 2-dimensional (vertical and radial) heat flow patterns of concentric ground-couplings. This formulation requires considerable computer time since time steps for smaller pipes are less than 5 seconds, and the simulation must be performed over several months before peak conditions are reached. Conversion to the 3-dimensional heat flow variation (vertical, radial and circumferential) encountered in U-tube ground-couplings would require much more computation time. Therefore a series of simplifications are made to reduce computer time. These simplifications permit the use of a one-dimensional implicit formulation (backward time step) that allows the use of much larger time steps and models both concentric and U-tube arrangements.

Although considerable simplification of the FDE's are possible with small losses in accuracy, the scheme is still somewhat complex and requires considerable computer time. Chapter V outlines a simulation model utilizing the Kelvin Line Source Theory proposed by Ingersoll (5), and developed by others (2) (6) (7). Although this scheme does not have the flexibility and instantaneous accuracy of FDE's it reduces the computation time significantly. This method has been primarily used as a design tool for predicting water temperatures and heat pump performance at peak conditions. It can be modified and used to predict heat pump performance at less than peak conditions. Therefore it can be used as a tool for calculating energy consumption.

The performance of standard air-coupled heat pump systems are primarily dependent upon outdoor air conditions. Capacity and efficiency can be calculated by knowing the outdoor temperature. Similar performance calculations of ground-coupled heat pump system are dependent on a great many more variables. These include soil temperature, soil moisture, groundwater movement, soil density, pipe thermal characteristics, pipe size, pipe length, water flow rate and amount of time the system has operated before the performance calculation is made. This last variable requires that some type of building load simulation be performed in conjunction with a simulation of the ground-coupling in order to calculate the water temperature entering the heat pump.

The building load simulation used in this paper utilizes the monthly degree-day method for heating and cooling. Current procedures of utilizing this method do not yield a high degree of accuracy, especially in the cooling mode. However, efforts are being undertaken to reduce error with this method due to improper accounting for internal loads, daily range, thermal mass, infiltration and insulating methods. Although bin and transfer function methods yield higher accuracy, the varible base degree-day method is sufficient for general applications.

1.6 Limitations

The simplifications utilized to reduce computation in this model require that limitations be placed on its range of application. These are:

1) Ground couplings are of small bore (less than six inches).
2) Freezing of soil is not considered, but can be implemented using methods of references (2) and (4).
3) Less than 80% of the total ground-coupling is located in soils whose undisturbed temperature varies $\pm$ 3^{O}F or is located in an aquifer with water movement greater than 20 ft/year.
4) Limited interference from adjacent ground-couplings (separation distance greater than 20 feet).
5) System is used for both heating and cooling.
6) If moisture migration is significant, performance must be adjusted utilizing methods of reference (4).
7) No separation between ground-coupling and soil.
8) Maximum of two U-tubes per bore.

CHAPTER II

EXPERIMENTAL GROUND-COUPLING SYSTEM

Experimental testing of vertical CLGCHP systems has been more limited than testing of horizontal systems. The work of Ball, Fischer and Hodgett (3) which was published in 1983, indicated there were no well instrumented systems operating in the U.S. While this may be somewhat of an overstatement, there indeed remains several aspects of vertical systems that need experimental verification. Many analytical and numerical design methodologies exist and several have been validated by experiment for local conditions. The following literature survey summarizes experiments directly related to vertical CLGCHP systems. It provides necessary background to the experimental system used in this project.

2.1 Literature Survey

There was considerable research concerning the use of the ground as a source or sink for heat pumps in the period of 1948-1953. At this time the popularity of gas and oil as a heating source, resulted in declining interest in the use of heat pumps. The price increases of fossils fuels experienced in the mid-1970's rekindled interest in CLGCHP systems. Experimental and analytical investigations resumed in 1977 and 1978. This literature survey is limited to experimental investigations with sufficient instrumentation to validate models.

Dr. James Bose and coworkers (2) in the School of Technology at Oklahoma State University have studied actual installations of ground-couplings beginning in 1978. One of the several ground-coupling devices observed was a 5 inch diameter PVC vertical pipe similar to the system shown in Figure 1. The system included a heat pump that typically rejected 34,000 Btu/hr (cooling mode), 232 feet of 5 in. sch. 40 PVC sealed outer casing and a dip tube of 1 1/4 inch PVC pipe. Thermocouples were located in the soil around the pipe, in the inlet and outlet water and inside the outer pipe at various depths. Test were conducted with water injected in the top of the well and also with injection at the bottom. Results were plotted in the form of water temperature vs. continuous run time. The ground temperatures below 20 feet at the site were 62^{O}F. Significant results were:

1. After one hour of continuous running, water returned to the heat pump at 77^{O}F. The temperatures were 97^{O}F after four hours and 103^{O}F after 7 hours.
2. After one hour continuous running, the well had a "U-value" of 9.0 Btu/hr^{O}F-ft per linear foot of pipe, 4.1 Btu/hr-^{O}F-ft after four hours, 3.0 Btu/hr-^{O}F-ft after 7 hours and a minimum value of 1.7 Btu/hr-^{O}F-ft at peak periods in August.
3. When the water return temperature reached 105^{O}F and the system was shut down, it took 12 hours for the water temperature to return to 76^{O}F, and four days to reach 69^{O}F.

For the same system in the heating mode (earth acting as a heat source) and coupled with a 210 ft^2 solar assist (using the coupling and ground around it as heat storage) results were:

1. A minimum recorded return water temperature of 38^{O}F. Thus, the possibility of freezing exists in the ground-coupling.
2. A long term steady state U-value of 1.7 Btu/hr-F^{O}-ft and a value of 3.4 for 50 per cent cycle.
3. An increase of 20% in U-values when an insulated dip tube was used (2).

Test results of a similar system in Beaumont, Texas, agree well with the OSU data (8). A water source heat pump (41,000 Btu/hr cooling, 59,000 Btu/hr heating) was coupled with a 360 foot 5 in. sch. 40 PVC well. The system worked extremely well in the heating mode, because of the lighter load and higher earth temperature (69^{O}F). However, in periods of continuous operation (6 or more hours) in the cooling mode the return water temperature reached 105^{O}F. At this temperature the heat pump yielded a relatively poor EER of 7.23. However, the value increased to 9.0 during periods of 50% cycle time.

Another test on this type of system was conducted at Louisiana State University (9). A 504 ft. 2 1/2 in. steel pipe ground-coupling and a 265 ft. 1 1/4 in. U-tube polyethylene pipe coupling were tested. Researchers made tests on these systems for 48 hours using continuous operation, 30 minutes on - 30 minutes off, and 15 minutes on - 45 minutes off. Results are shown in Table I for the heating mode. Ground temperature is 21^{O}C (69.8^{O}).

A study conducted by Oklahoma State University to determine the performance of these systems in more realistic cyclic operation, collected data in Perkins, Oklahoma, on three homes of identical

TABLE I

RESULTS OF GROUND COUPLING STUDY AT LOUISIANA STATE UNIVERSITY (9)

Run Time Percent	2 1/2" Concentric Steel Pipe	1 1/4" Poly Eth. U-tube	2 1/2" Concentric PVC (Calculated)
100	2.81	2.0	2.07
50	4.34	2.72	2.8
25	6.87	6.7	3.68
"U-values" in Btu/hr-^{o}F-ft			

construction (1). One home had an air-to-air heat pump, the other two CLGCHP systems. One was solar assisted. Sixty-three data points were collected every fifteen minutes. The ground-couplings for the water source heat pumps were originally 250 ft. 5 in. PVC casings, but in March of 1982 these were replaced with 250 foot, 1 1/2 in. sch. 40 high density polyethylene U-tube couplings.

The west house (water source heat pump <u>without</u> a solar assist) is of particular interest when examining the performance of vertical ground-couplings because of the absence of other variables present in the solar assisted system. Actual heat pump efficiencies are difficult to calculate because the experiment was designed primarily to determine how effective a heat pump can be in reducing overall demand as well as consumption. Data were taken in "lumps". A hot water heat recovery unit (desuperheater) was installed on the compressor discharge line. This reduces overall home consumption especially in the cooling mode, but complicates efficiency calculations. Therefore, the temperature of the water entering the heat pump can be considered the primary variable that determines well and heat pump performance.

Figure 4 is a plot of temperature over a 24-hour period for four days of extremely high cooling load. Figure 4 a) and c) reflect temperatures for the 5 inch PVC well on August 3, 1981, and August 5, 1981. These days were selected to determine the performance of the well over a relatively long period (3 days) of high loads. Figure 4 b) and d) are similar days for the 1 1/2 inch polyethlene U-tube. Indoor and outdoor temperatures are plotted continuously. However, the well inlet and outlet temperatures are plotted only when the heat pump is on (other values do not reflect the temperature of the circulating water).

Examination of these figures indicate the following:

1. The polyethlene U-tube returns cooler water to the heat pump even though the load on it is larger than on the PVC coupling. (Notice the increased load is due to the lower thermostat setting in the summer of 1982).
2. The return water temperature does not increase noticeably if the on time is 30 minutes or less. In many cases it actually decreases over a 30 to 45 minute on time.
3. The return water temperature begins increasing noticeably if the on time is longer than 45 to 75 minutes.

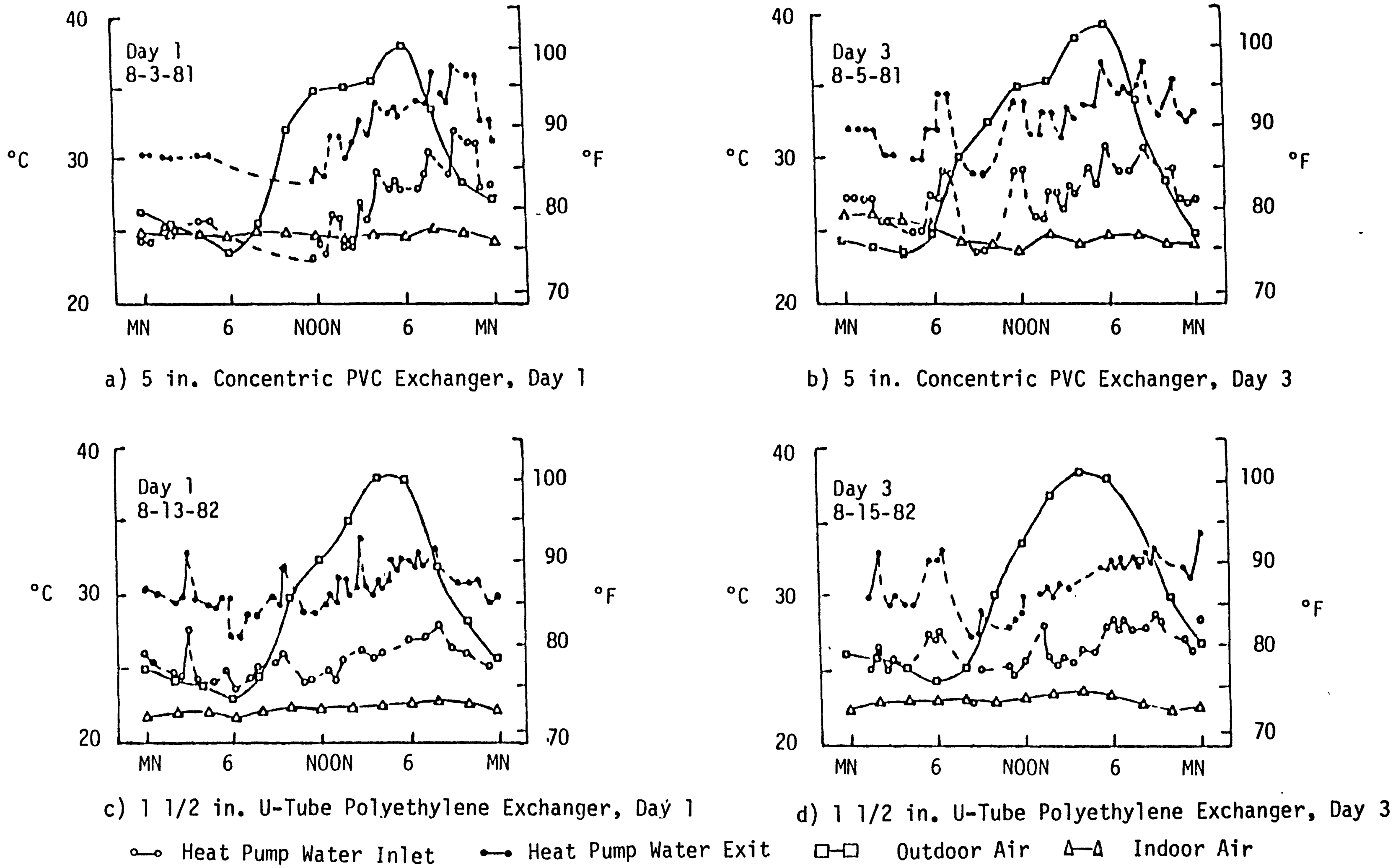


Figure 4. Comparison of Water Temperatures of Two Ground Heat Exchangers during Periods of High Cooling Load (1)

4. Return water temperatures sometimes exceeds outdoor temperature in the late afternoon.
5. Return water temperatures are below 85^{o}F with the polyethylene U-tube arrangement, therefore, performance at least equal to ARI ratings can be expected even during periods of highest loads. This assumes systems are properly sized and installed.

Figure 5 is a plot of water temperature in and out of the ground-coupling for January 11, 1982, the coldest day of the 1981-1982 winter in Perkins. The minimum coupling inlet temperature was 4.8^{o}C (40.6^{o}F) for the PVC system. The polyethlene U-tube has an even higher minimum temperature under similar conditions. It can also be concluded that the addition of antifreeze to the water is unnecessary in this climate for similarly designed systems. The heat pump provided all the heat needed for this extreme case and no auxiliary heat was used.

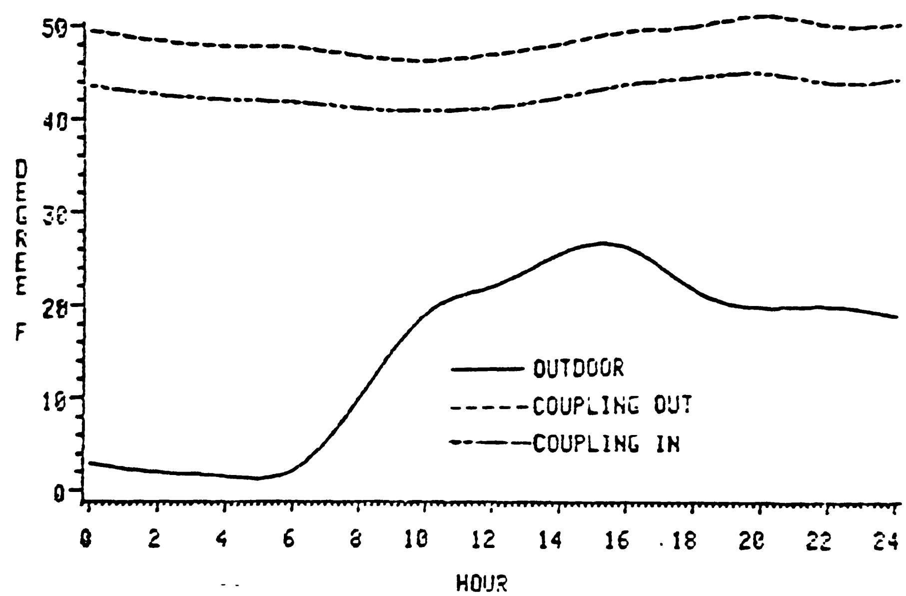


Figure 5. Temperatures on Coldest Day of 1981-2 Winter

Mei and Fischer (10) conducted test on a concentric PVC ground-coupling in order to develop and validate a FDE simulation of a vertical CLGCHP. The apparatus consisted of a 155 ft. coupling with a 5 inch schedule 40 casing and a 1 inch PVC inner pipe, similar to the one shown in Figure 1. The coupling was set in an 8-in. well casing and backfilled to establish good thermal contact.

Tests were conducted primarily to validate the computer model and do not reflect the gradual injection water temperature change associated with CLGCHP systems. Water at 107°F was injected at a rate of 5.0 gpm continuously and cyclically. No outlet temperatures were reported, however, water and outside shell temperatures were recorded after water had passed through 100 ft. of the coupling (55 ft.). Outside pipe wall temperatures were also recorded at this location.

A six-hour test was performed. After one hour the water temperature was 99°F and the outside wall temperature was 75°F. At six hours the water temperature was 102°F and the wall temperature 78°F. The hot water injection was halted at this point for six hours. The water temperatures dropped to 77°F and the wall temperature to 72°F.

Another test was performed by injecting 40°F water at the same flow rate continuously for 12 hours. The 100 ft. water temperatures were 44°F at 1 hour and 43°F at 12 hours. Pipe temperatures were 59°F at 1 hour and 54°F at 12 hours.

Hot and cold water tests were also performed for 30 min. on - 30 min. off cyclic test. Hourly water temperature fluctuations were about 7°F for hot water and 5°F for the cold water at the 100 ft. point. Average temperatures were 101°F after 12 hours of on-off hot water injection and 41°F for cold water injection. All test reported in this reference were relatively short term.

G. Rosenblad (11) utilized shallow vertical couplings in a heat pump test in Utby, Sweden. The couplings were round PVC pipes 10-meters in length and divided by a partition for the up and down flowing brine solution. Thirty seven couplings were arranged in a triangle pattern 2 meters apart and flow was split into 3 parallel paths. The location has little cooling requirement, so heat from collectors (wind convectors) was injected into the coupling during the summer. The focus of this experiment was to determine the usefulness of the ground for long term (seasonal) heat storage. A air-to-air heat pump was used when temperatures were above $1^{o}C$ to conserve the heat in storage.

The average ground temperature was dropped 3 to $5^{o}C$ below the average undisturbed ground temperature at a distance of 1 meter from the couplings during winter use. The heat addition during the summer months raised this temperature 1.3 to $2.0^{o}C$ above the normal undisturbed temperature. The resulting COP values of this complex systems was about 3.0.

These studies indicate the vertical ground-coupling is indeed an attractive option for water source heat pumps. Many actual installations have proved successful. However, continued study is warranted in order to further increase performance by designing coupling installations and heat pumps specifically for this application in different environments.

2.2 Experimental System

The literature available lacks a comprehensive study of factors that effect heat transfer in and near vertical ground-couplings over an extended period of time. The large temperature difference encountered across the pipe wall of PVC couplings indicates this design can be improved. Insignificant temperature change would occur with metal couplings, but increased cost and corrosion problems may exclude them from consideration. Additionally, the use of U-tube designs seem to perform better than concentric. However direct comparison of concentric and U-tube couplings using the same pipe material has not been attempted. There has been little experimental treatment of thermal short circuiting between the upward and downward flow streams in non-concentric coupling designs.

The experimental systems shown in Figures 6, 7, and 8 is the result of a design procedure that investigates the areas mentioned above and adds to the existing body of experimental data concerning vertical CLGCHP systems. Although the experiment does not exhaust the possibilities of ground-coupling designs, most small bore (less than 6 inch diameter) coupling performance can be estimated from results. Larger bore couplings are not considered primarily because of current pipe and installation costs. The experimental system is designed to study the effects of varying geometric arrangement (concentric, U-tube, multiple U-tube), pipe diameter, pipe wall thermal resistance, water flow rate, and heat pump on/off cycle duration. Figure 6 shows the design of the six 100 foot vertical ground-couplings. Thermocouple locations are shown along with normal flow arrangement. Figure 7 shows the plan of the coupling layout along with the water supply and return piping. Figure 8 is a schematic of the test room equipment.

The designs shown in Figure 6 are variations of couplings that have worked well in actual installations and ones that appear to be

NOTES: 1. •-DENOTES THERMOCOUPLE LOCATIONS
2. ALL EARTH COUPLINGS 100 FT. IN LENGTH
3. P.E. - POLYETHYLENE, P.B. - POLYBUTALENE

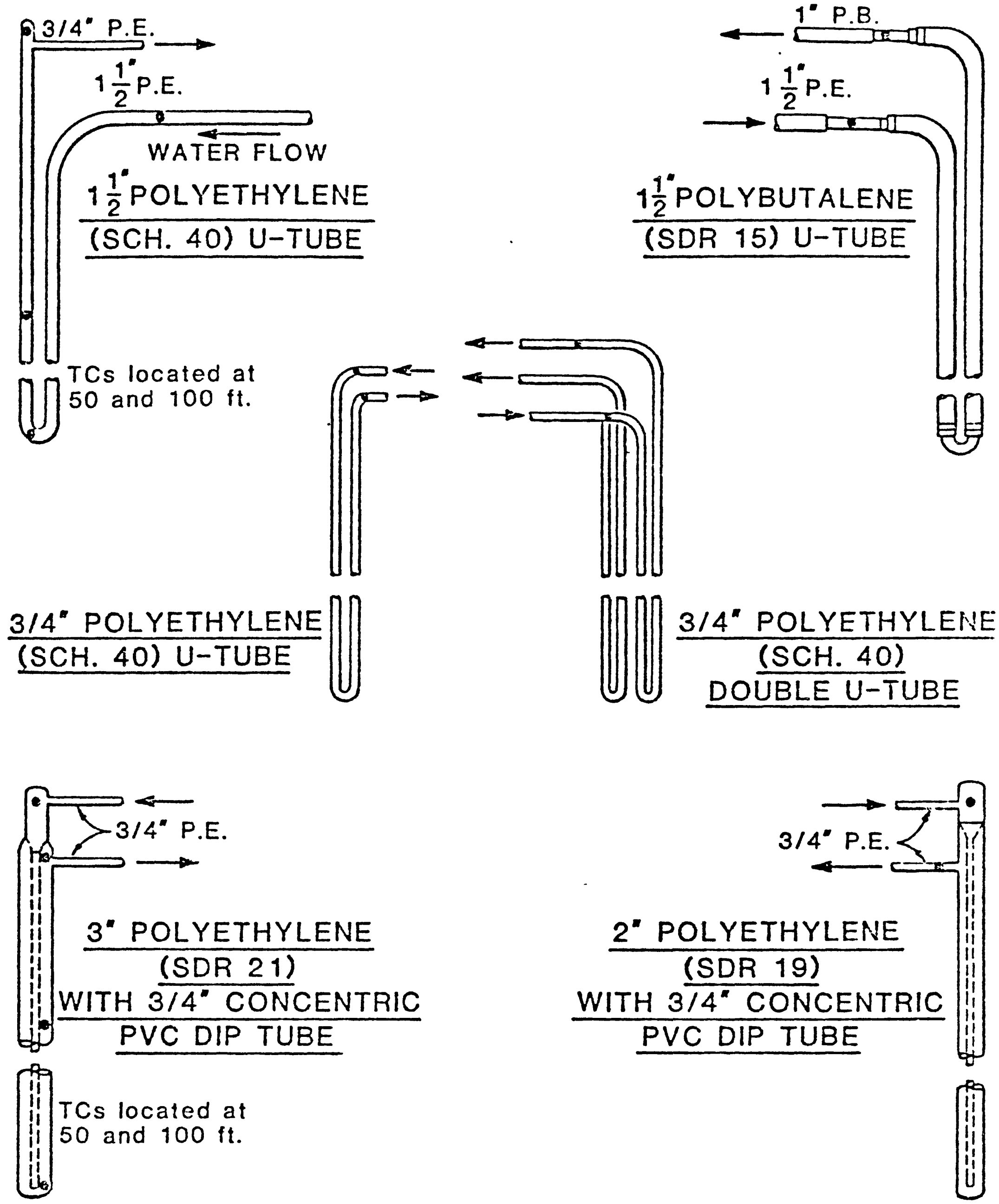


Figure 6. Parallel Ground-Coupling Designs

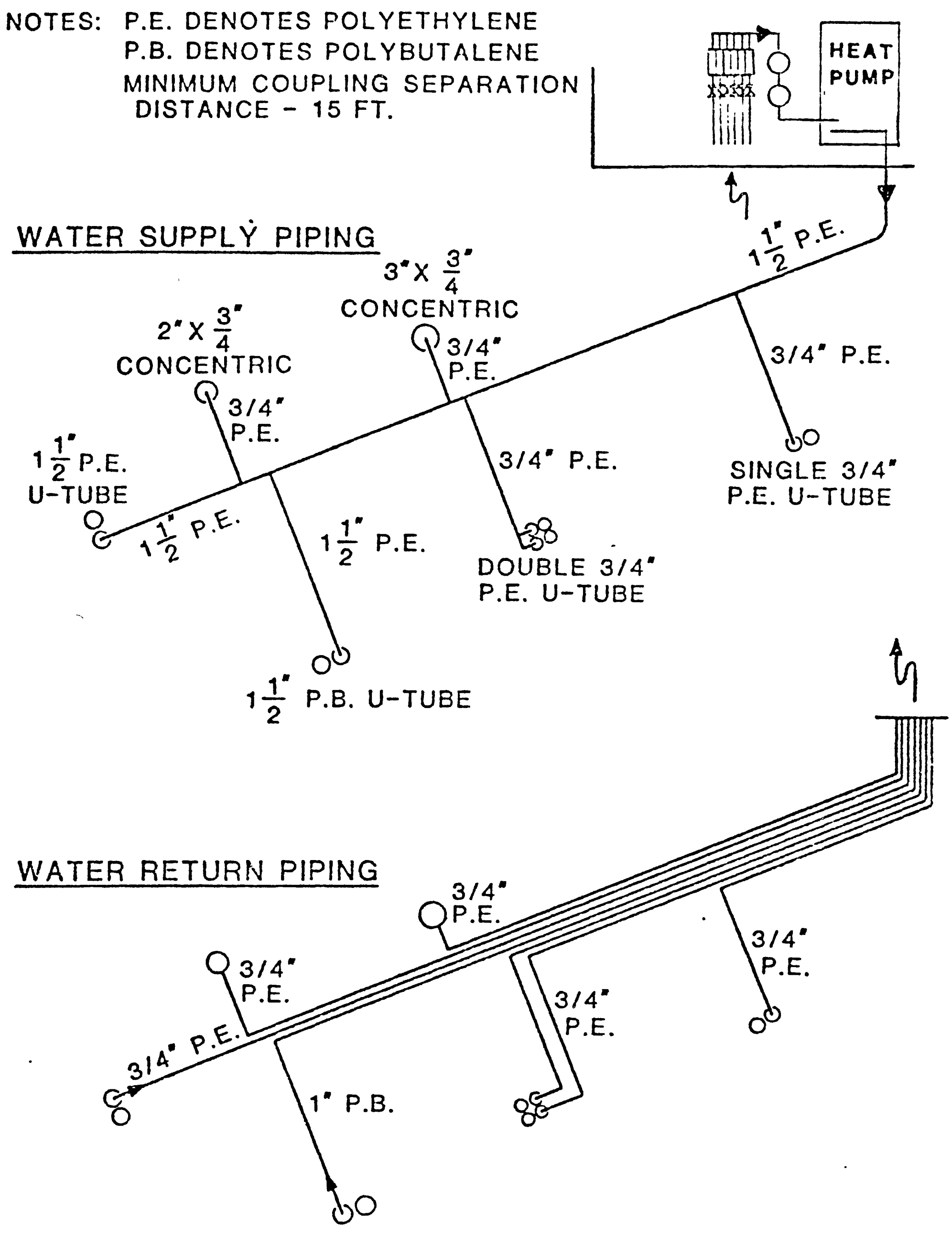


Figure 7. Ground-Coupling Layout Piping Diagram

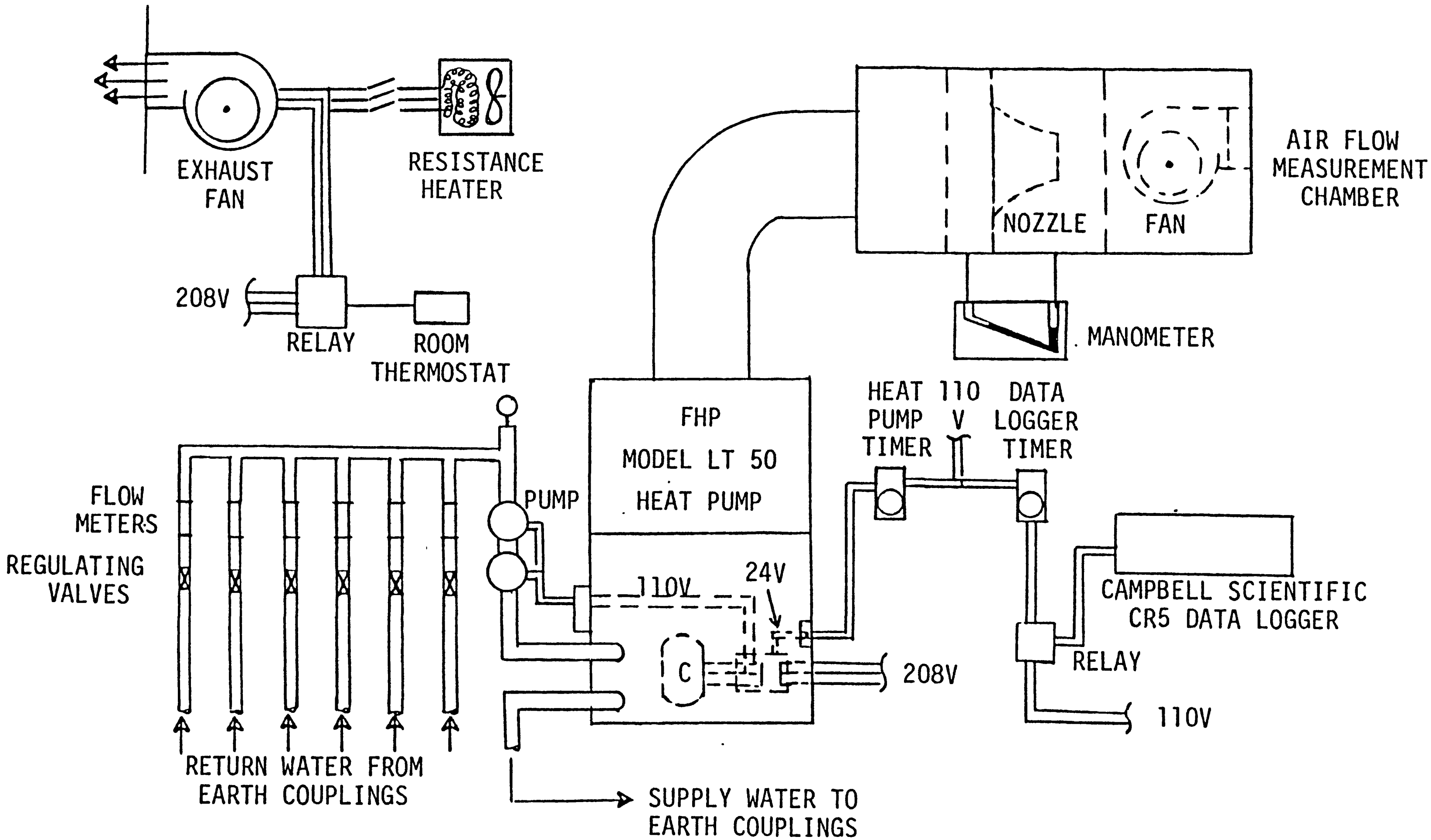


Figure 8. Heat Pump Test Room Schematic

effective based on preliminary computer simulation. The predominate material is high density polyethylene (ASTM 3408) which has a relatively high thermal conductivity for plastic (0.226 Btu/hr-ft-F). Polybutylene (k = 0.13 Btu/hr-ft-F) is used in one coupling to examine the trade-off between a material of lower thermal conductivity, higher strength and therefore thinner pipe wall. Schedule 80 PVC is used as the annular injection tube in the concentric designs because its thick wall and low thermal conductivity increase resistance to thermal short circuiting.

In addition to giving a comparison in terms of pipe material, the 1 1/2 inch-polyethylene coupling provides a means of determining the effects of increased diameter when its performance is compared with that of the 3/4 inch polyethylene U-tube. A double 3/4 inch U-tube shows the effects of increasing pipe wall thermal contact area with the ground when compared to a single coupling of the same size.

Water exits the heat pump into a common header and branches off into the six 100 foot ground-couplings, as shown in Figure 7. Water returns to the heat pump through individual pipes. Figure 8 shows the test room layout, and the location of valves and visual flowmeters. The water from each loop passes through a valve and flowmeter before mixing into a common header at the pump suction. Two small circulating pumps operating in series discharge into the heat pump. Pump power input ranges from 300 watts at 7.5 GPM to 375 watts at 12.0 GPM.

The heat pump used in the experiment is an FHP model LT50, with a nominal capacity of 48,000 Btuh. Manufacturer's data state that this unit is capable of operating with entering water temperatures as low as 40^{o}F. Performance data are included in the Appendix. Actual unit output and power consumption agree well with published data.

Temperatures were measured with type T thermocouples and recorded on a Campbell Scientific CR5 Data Logger. In addition to the positions shown in Figure 6, thermocouples were located to measure the temperatures of the water entering and leaving the heat pump, the air entering and leaving the heat pump and the outdoor temperature. Wet bulb temperatures were periodically measured with a sling psychrometer.

Air flow rates were originally measured with the chamber and nozzle arrangement shown in Figure 8 in accordance with ASHRAE Standard 37-78. However, the heat pump was normally operated without the chamber because of the excessive pressure drop across the nozzle and corresponding reduction in flow rate. Only ninety percent of rated flow could be obtained with a 1-1/2 horsepower fan in the heating mode.

The chamber was replaced with a circular duct at the beginning of the summer test. Flow was measured with a pitot tube and the fan differential pressure was simultaneously recorded. The resulting curve of flow vs. pressure served as the basis of flow measurement for the remainder of the test. The operation of the heat pump was controlled by a timer rather than a thermostat. This timer was capable of cycling the unit in 15 minute increments and was programmed on a 24-hour basis. Typical cycles tested during the winter were continuous (100%), 45 minutes on - 15 minutes off (75%), 30 on - 15 off (67%), 30 on - 30 off (50%), 15 on - 15 off (50%), and 15 on - 45 off (25%). The unit was normally operated 75% at night and 50% during the day when testing is not being performed. This level of operation was maintained for 2 1/2 months for an equivalent run fraction of approximately 60% from mid-January until April 4. The cooling mode test began June 3 with an equivalent run fraction of 25% for three weeks. This run fraction was accomplished by a 50% fraction in the afternoon, 25% during the morning and evenings and off at night. The daily run fraction was increased to 50% for eight weeks, and 62% for the following two weeks. At this point tests varying flow rates and the number of loops in operation were performed.

The water pumps were wired to the compressor contacts on the heat pump and therefore operate when the unit was running. However, they could be operated independently of the heat pump for testing and calibration purposes.

The data logger is capable of being operated synchronously with the heat pump in order to give temperature samplings at known times during on-off cycles. It can also operate independently and is activated by a timer, to limit the collection of unnecessary data.

In order to control temperature in the test room during winter tests, an exhaust fan with a 1500 CFM capacity was controlled by a room thermostat. Outside air was induced into the building when room temperature exceeded typical operating levels. This scheme allowed full load testing to be done when outdoor conditions were only moderately cool. Therefore, peak heating loads could be simulated into April. However, when the outdoor temperature rose above $55^{\circ}F$, full load operation could not be maintained with test room temperature below $80^{\circ}F$.

This scheme did not work as well in the cooling mode because induction of warm air was accompanied by large amounts of water vapor. This often resulted in latent loads approaching the magnitude of sensible loads. This situation was partially remedied by a controlled dumping of heat pump outlet air in addition to a thermostatically controlled addition of resistive heat to the test room. However, the problem remained during rainy periods. Since the months of July and August were extremely dry, the overall effects upon the test were minimal.

2.3 Winter Test Results

Peak heat transfer rates ofground-coupled heat pumps are lower in the heating mode than in the cooling mode of operation. The amount of heat absorbed by the coupling device is the unit capacity less the input to the compressor and pumps. However, it cannot be assumed that ground-couplings sized to the cooling load are sufficiently sized for heating operation in Oklahoma, a region in which the annual cooling load often exceeds the heating load. The heating mode of operation is critical because of the possibility of freezing in the evaporator and because of the sharp drop in capacity of heat pumps not specifically designed for operation with entering water temperatures below 50°F. The FHP Model LT50 is designed for low water temperatures. No antifreeze solution was used in this experiment. The unit shut down when the water leaving the heat exchanger dropped below 36°F.

The necessity of antifreeze solutions in vertical ground-couplings in Oklahoma is debatable. It is necessary to examine actual heat absorption rates and local ground conditions before making a decision. However, it appears that in installations with a climate and ground properties similar to test site, antifreeze is unnecessary provided that ground couplings similar to those used in this test (not including the single 3/4 U-tube or 2 inch concentric) are installed with 150 feet of hole per nominal ton capacity and water flow rates of at least 3 GPM per ton. This assumes that the system will be sized to the load and the monthly run fraction will not exceed 75%. Additional lengths are needed if a single 3/4 inch U-tube or a 2 inch concentric coupling are used. Determining the lengths will be discussed in Chapter V.

Caution must be used in applying the temperatures recorded in these tests directly to installations. Figure 9 appears to indicate the performance of 5 of these loops is almost identical. However, since the loops are returned to a common pipe, the ones having a larger heat transfer capacity compensate for the ones of lower capacity. This means that if six single 3/4 inch loops were installed on a FHP LT50 the resulting water temperature curve would be lower than the one shown in Figure 9. This would result in lower unit capacity and efficiency. Conversely, if six double 3/4 inch tubes were installed, the resulting curve would be higher than the one shown in Figure 9 for the double U-tube. The temperatures shown in Figures 9, 11, 13, 14, 15 and 16 are adjusted. This is necessary since the water entering the single 3/4 inch U-tube is normally 0.4 to 0.2^{o}F colder than the water entering the 1 1/2 inch U-tubes and the 2 inch concentric (see Figure 7). Therefore, all inlet temperatues are adjusted to agree with the inlet temperature of the 3 inch concentric and the double 3/4 inch U-tube. The outlet temperatures were accordingly adjusted so the temperature difference on each loop remained unchanged. Additionally, the temperatures shown in Figures 10, 11 and 14 are average temperatures over the period in which the heat pump was on. Off time temperatures were not normally taken during these tests.

Examination of Figure 9 shows the results of a continuous run performed March 17-18, after the system had been operated at over 60%

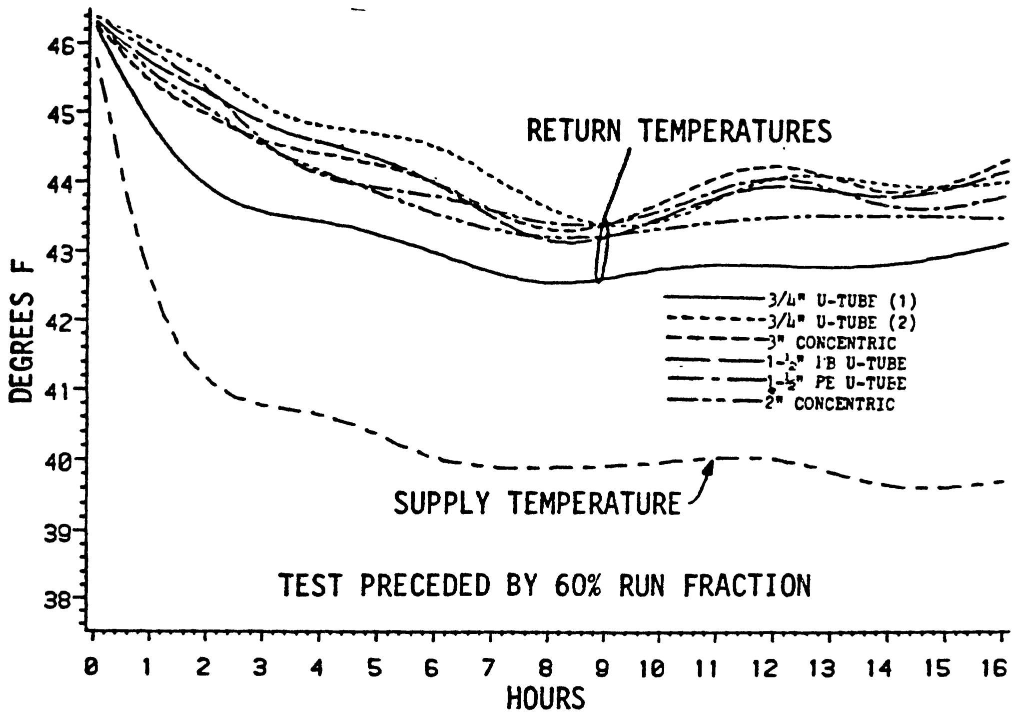


Figure 9. Water Temperatures during Continuous Operation

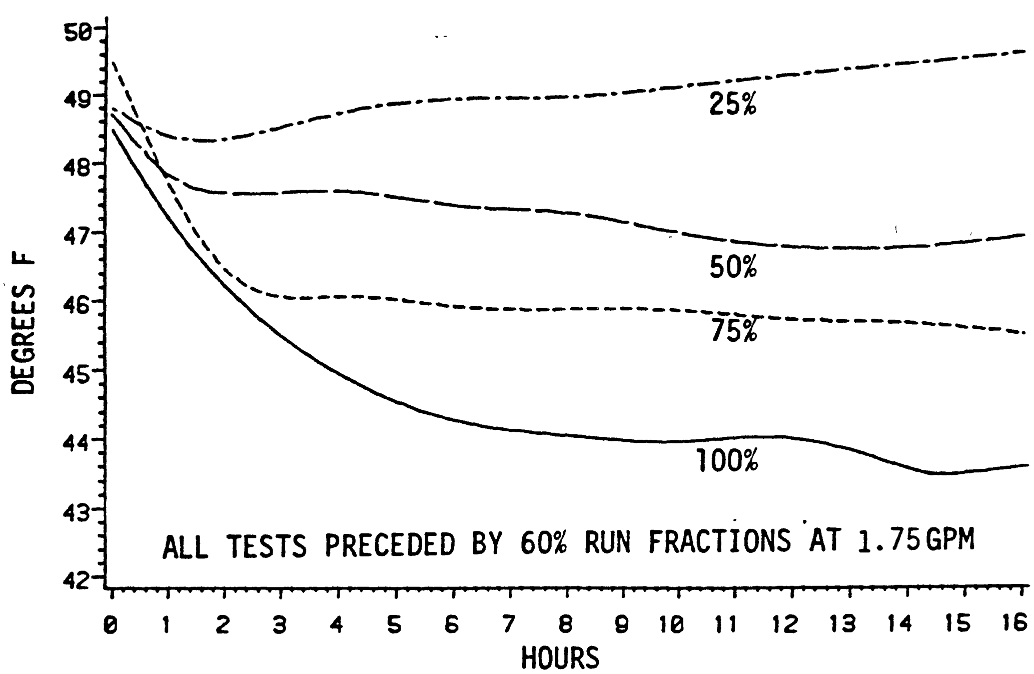


Figure 10. Return Water Temperatures for Different Run Fractions

run time for two months. This test would be equivalent to the load experienced by a properly sized unit operating in Stillwater with an $13^{o}F$ outdoor temperature for 16 hours. The test was preceded by an eight hour 75% run, which included a 15 minute off period immediately before the unit was turned on. The performance results are typical. The double 3/4 inch polyethylene U-tube normally returns water 0.4 to $0.6^{o}F$ warmer than the 3 inch concentric coupling, except for periods when the larger thermal mass of the concentric reduces fluctuations (notice hours 9 to 13). The polybutylene and polyethylene 1 1/2 inch U-tubes also typically return water 0.4 to $0.6^{o}F$ cooler than the double 3/4 inch U-tube. The 2-inch concentric arrangement matches the performance of the 1 1/2 inch U-tubes for continuous runs of 4 hours or less, but temperatures are 0.2 to $0.4^{o}F$ lower for most other runs. The single 3/4 inch polyethylene coupling typically returns water 1.0 to $1.2^{o}F$ cooler than the double 3/4 inch. This translates into approximately a 25% reduction in heat transfer capacity.

Figure 10 is a comparison of the return water temperatures in the 3/4 inch double U-tube for various run fractions. In all four tests, the flow rate is 1.75 GPM and all are preceded by at least two days of run fractions of 60 to 65%. Notice that the run fractions less than 60% seek an average thermal equilibrium temperature above $46^{o}F$ and those greater than 60% seek a lower temperature.

Figure 11 is a simulation of a typical run fraction that would be encountered on a cold night in Oklahoma (6 hours 50%, 6 hours 75%, 4 hours 50%). The relative performances of the ground couplings are similar to the continuous run. The 3/4 inch double U-tube returns the warmest water, next are the 1-1/2 inch U-tubes and the 3 inch

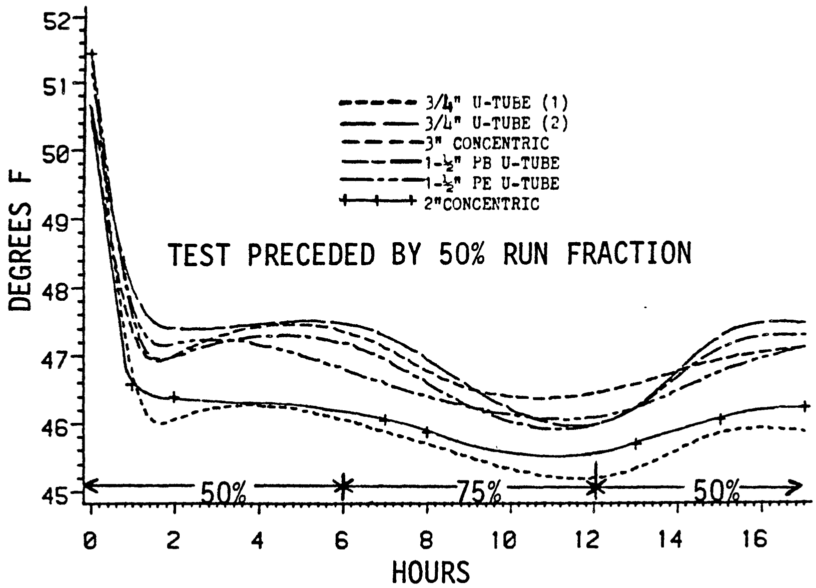


Figure 11. Return Water Temperatures for Varying Run Fractions

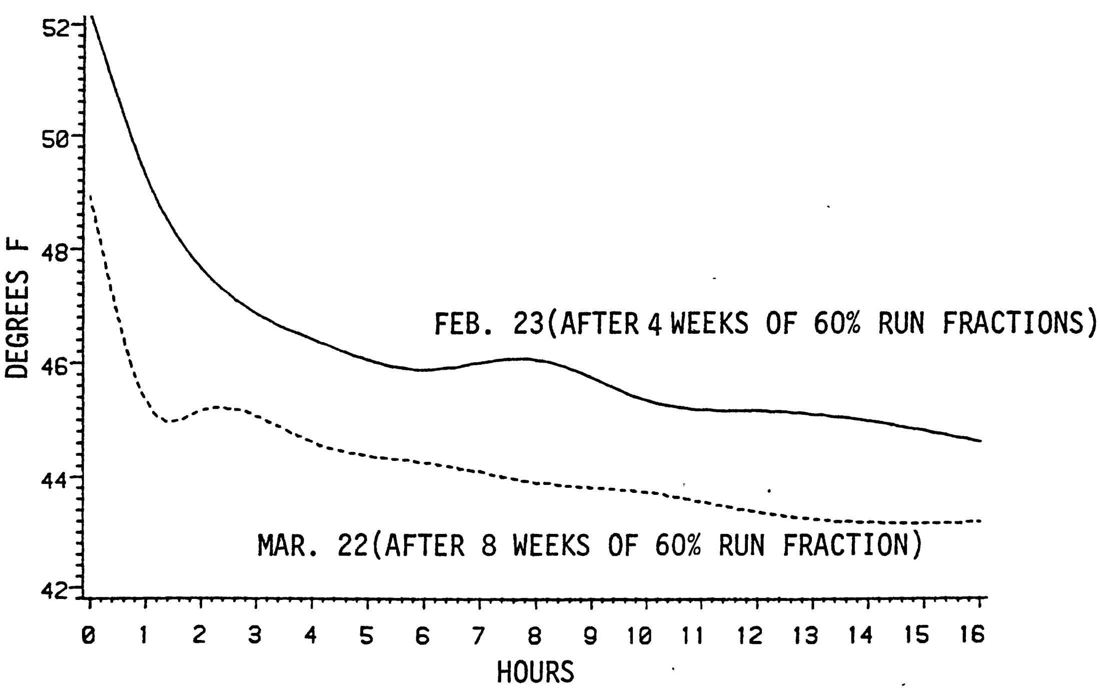


Figure 12. Return Water Temperatures during Continuous Run at Four and Eight Weeks

concentric, the 2 inch concentric and the 3/4 inch U-tube returns the coldest water. This test indicates that it takes 6 hours of 75% run fraction to drop the average temperature 1^{O}F in the 4 loops of larger heat transfer capacity. It takes 3 hours of 50% run fraction to regain this 1^{O}F. The tendency for decreasing temperatures associated with periodic large run fractions is damped out by the large thermal mass of the earth.

Figure 12 shows the considerable effects of a "thermal history". The upper curve is the return water temperature of the 1-1/2 inch polyethylene U-tube after 4 weeks of 60% run fraction. The lower curve is an identical test performed after 4 more weeks of 60% run fraction. A test conducted two weeks later showed little difference from this second test. Performance of ground coupled heat pump systems are strong functions of run fractions and the number of weeks or months the system has been in operation.

Figure 13 is a comparison of the return water temperatures for operation using only 4 loops. The 1-1/2 inch polybutylene and the 2 inch concentric loops were closed. The flow rate in each loop was increased to 2.5 GPM. The flow rate for the six-loop test was 1.75 GPM. The drop in temperature is not dramatic; however, note that the run fraction is 75%. A continuous run was attempted on the four loop system. During this test the water temperature dropped to the point at which the heat pump shut down. This indicates that in periods of high run fractions earth coupling systems similar to those tested installed at 100 ft. per nominal ton should have antifreeze protection in Oklahoma.

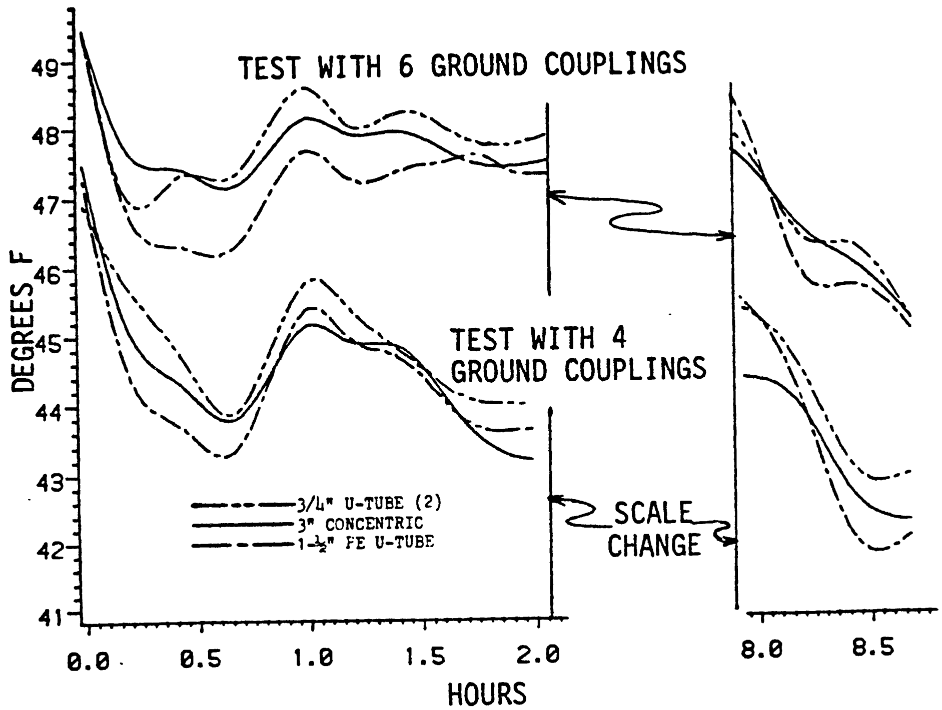


Figure 13. Return Water Temperatures for Six and Four Loop Operation

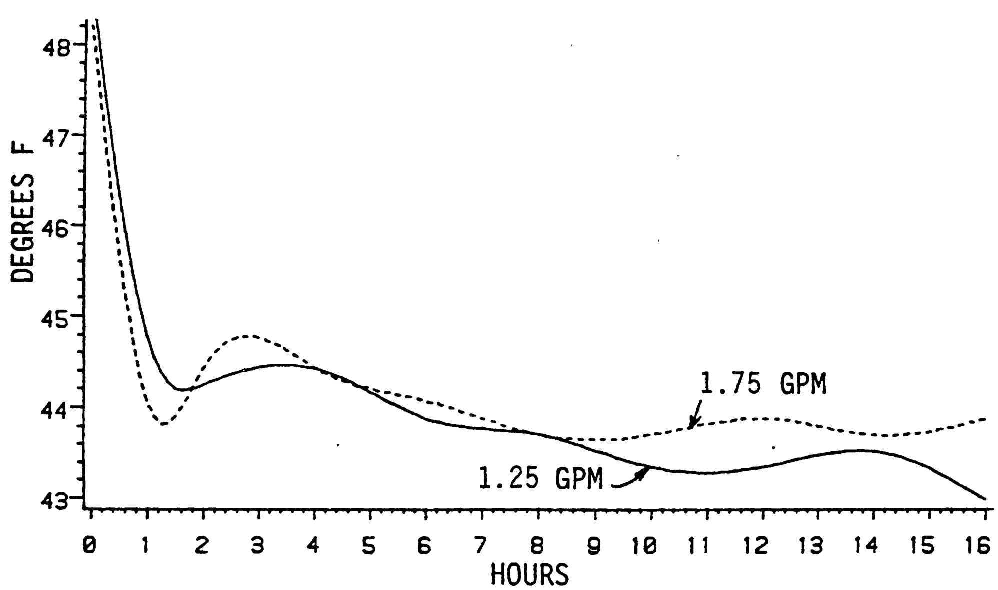


Figure 14. Return Water Temperatures for Different Flow Rates

Figure 14 shows the effects of varying flow rate. Identical tests of a 75% run fraction were performed on the system on two different days for flow rates of 1.25 GPM/Loop (1.875 GPM/ton) and 1.75 GPM/Loop (2.625 GPM/Ton). Figure 14 is a comparison of the resulting return water temperatures of the 1-1/2 inch polyethylene U-tube. Again the difference is apparently insignificant for the 75% run fraction. A 100% run fraction was not used because the heat pump would not operate continuously at 1.25 GPM/Loop because of low water temperature. The difference in performance would be greater if the system had been operated longer at 1.25 GPM/Loop before the test was taken. Only one day of 1.25 GPM/Loop operation occurred before the test was made. It is apparent that at a 75% run fraction, this was not enough time for the system to stabilize for a good comparison. Notice that from 0-8 hours the temperatures are almost equal. From 8-16 hours the water return temperatures during the 1.75 GPM test are significantly warmer. Additionally, unit capacity and heat pump outlet temperatures are reduced at lower flow rates.

Figure 15 is included to show instantaneous return water temperatures over a short period immediately after startup. It was initially theorized that the large thermal mass of water in the 3 inch concentric ground coupling would significantly improve return water temperatures for a 15 to 30 minute period following start-up. This effect appears to be minimal (0.5^{o}F from 18 to 26 minutes) and is almost negated by the lower temperatures during the first five minutes of operation. The thermal mass appears to have primarily only a damping effect on the temperature. Notice the large swing of the single 3/4 inch U-tube. The improved performance of the 3 inch concentric after start-up is much more significant when the system is off for longer

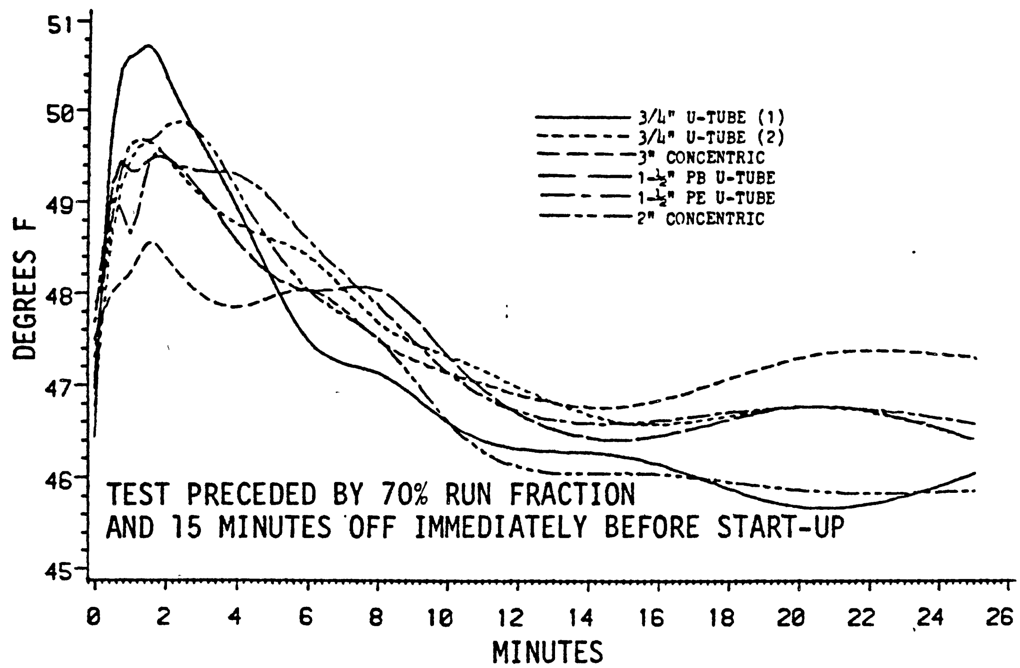


Figure 15. Return Water Temperatures at Start of On-Cycle

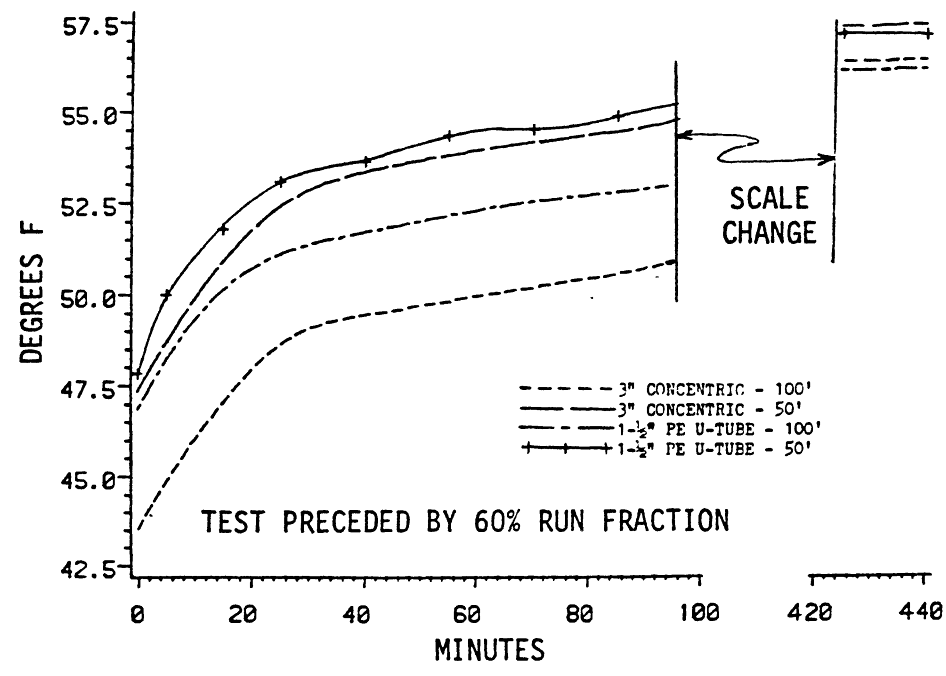


Figure 16. Thermal Recovery of U-tube and Concentric Couplings

periods. The unit was off for only 15 minutes before this test. The lower temperatures during the first 30 seconds of the test are the result of thermocouples being located in the couplings only a few feet below the surface, where the ground temperature is lower than average loop temperature.

Figure 16 shows the water temperature at different depths in the three inch concentric and 1 1/2 inch polyethylene U-tube loops when the unit was off for an eight hour period. Although the water temperature recovery for the first 25 minutes is good (due to natural convection heat transfer), ground temperature recovery is much slower (conduction is the basic mode of heat transfer). It is primarily the ground temperature that dictates the return water temperature after the unit is started up. This can be verified by returning to Figure 15 and observing that after eight minutes of operation all temperatures are significantly reduced compared to start-up values.

These tests serve as a tool in determining design precautions that must be considering when installing systems in Oklahoma, particularly in sizing backup heat and the necessity of antifreeze precautions. They also serve as a verification of models intended for use in this and other climates. However, the cooling mode test is a more valuable tool in determining the relative heat transfer capabilities of the different ground-coupling because of the 50 to 80 per cent increase in the heat transfer rate.

2.4 Summer Test Results

The amount of time the ground-coupling system was operated in the summer test exceeds the run fractions that would be experienced by a properly sized system during normal summers. However, run fractions greater than those simulated during this test could be experienced if the system is not properly designed for both heating and cooling seasons. Details of proper sizing procedures will be discussed in Chapter V and in the Appendix.

The cooling mode test began June 8, 1984. Two days of 50% run fraction were followed by three weeks of 25%, eight weeks of 50% and two weeks of 62%. The run fractions at night were approximately 25% less than the daily values, the afternoon values were 25% greater and the morning and evening run fractions were approximately equal to the daily run fraction. Table II shows a typical scheme for attaining a 50% run fraction.

At the end of the 62% run fraction test the system was returned to 50% for several days. Tests performed at this time were a four-loop test at 50%, several recovery tests and a return to 25% run fraction in October. The precautions and adjustments mentioned before the winter test description also apply to the summer test.

Figure 17 compares the performance of the six ground-couplings during the peak cooling load at the end of the fifth week of the 50% run fraction test. The top curve is the normalized temperature entering the couplings. All six return water temperatures are also normalized as described in the winter test. The double 3/4 inch and 1 1/2 inch polyethylene return approximately the same temperature water. The 1-1/2 inch polybutylene and 3 inch concentric return water 1.0 to 1.5^{o}F warmer, the 2 inch concentric 1.5 to 2.0^{o}F and the single 3/4 inch is 2.0 to 2.5^{o}F warmer. The 3 inch concentric and the double 3/4 inch U-tube have greater damping capacity than the other couplings.

TABLE II

TYPICAL 50% DAILY RUN FRACTION

Time	Cycle in Minutes ON	OFF	% Run Fraction
MN-6:00 AM	15	45	25
6:00 AM - Noon	30	30	50
Noon - 6:00 PM	45	15	75
6:00 PM - MN	30	30	50

Figure 18 is a 24-hour plot of the return water temperatures after 4 weeks of 50% run fraction. Over this period the temperatures of the two 1-1/2 inch couplings, the double 3/4-inch and the 3 inch concentric are approximately the same. However, the average temperature of the double 3/4 and concentric are about 1.0^{o}F less during the critical afternoon period, while the recovery of the larger U-tubes is more rapid during times of decreasing run fraction. Again the 2 inch concentric water is about 1.0^{o}F warmer while the single 3/4 inch U-tube is 1.5 to 2.5^{o}F warmer than the couplings of larger capacity.

Figure 19 shows a similar test performed after 4 more weeks of 50% and 2 weeks of 62% run fraction. The most obvious result is about an 8^{o}F increase in average return temperatures. The temperatures of the water for the five couplings of larger capacity are much closer during the peak load period. The single 3/4 inch coupling is about 1.0^{o}F warmer. The 1-1/2 inch polyethlene coupling has the best 24-hour

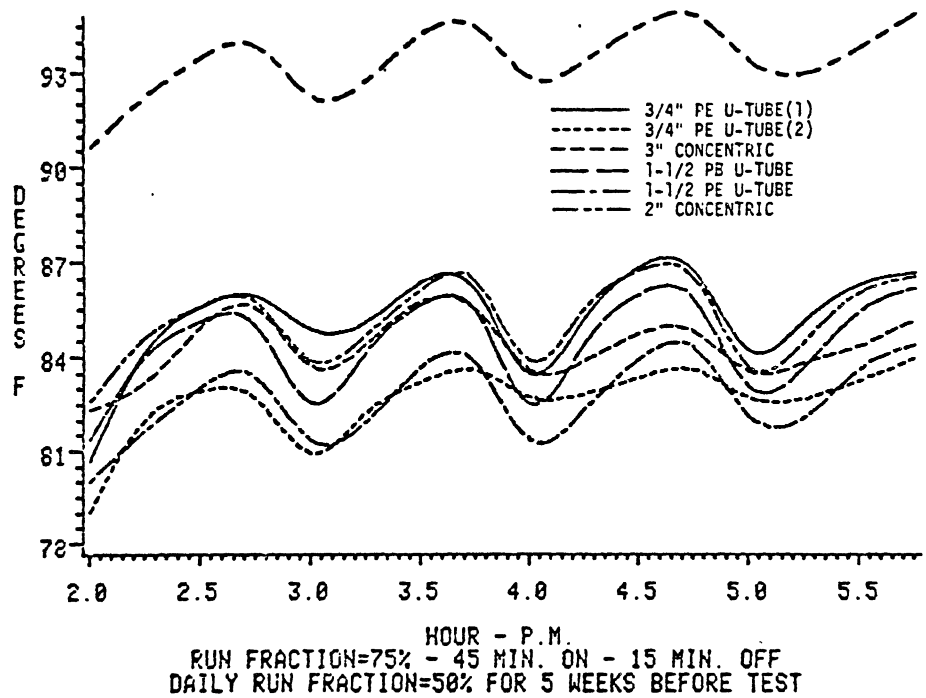


Figure 17. Afternoon Water Temperatures

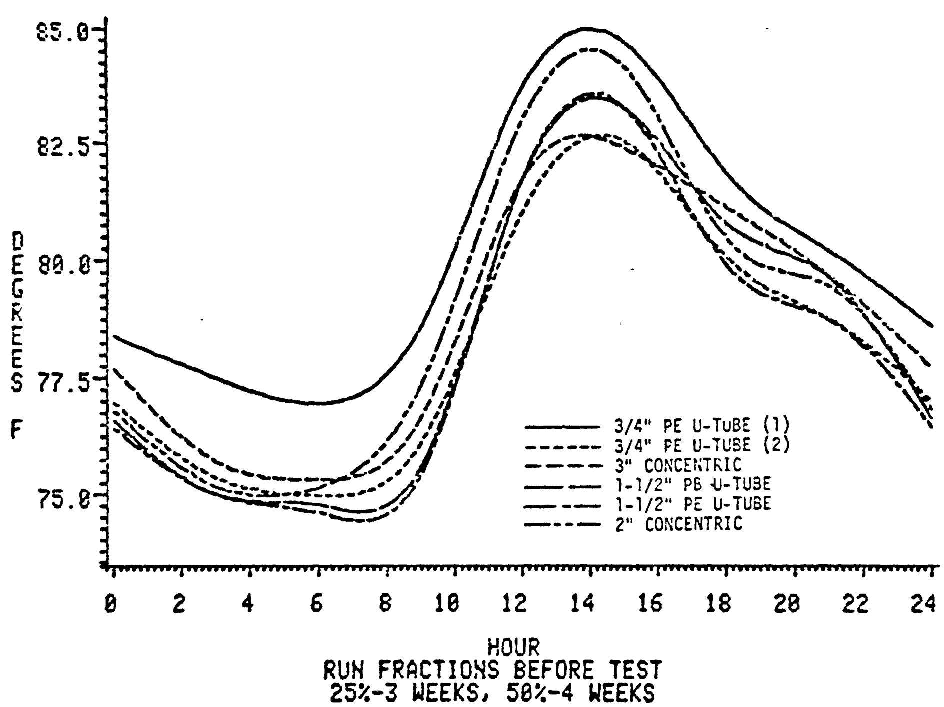


Figure 18. Daily Return Temperatures during 50% Run Fraction

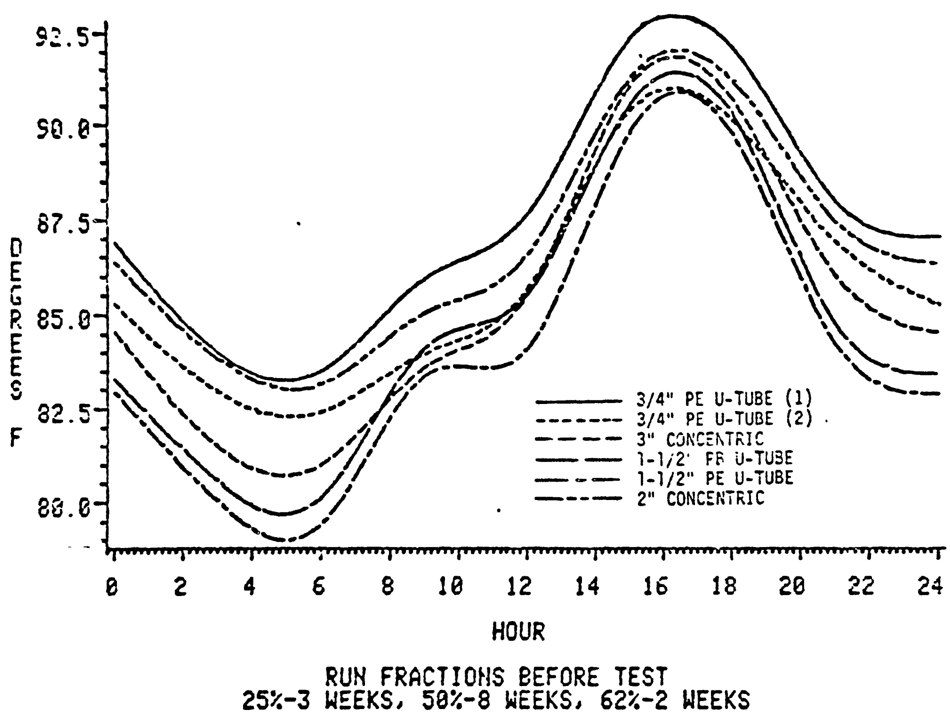


Figure 19. Daily Return Water Temperatures During 62% Run Fraction

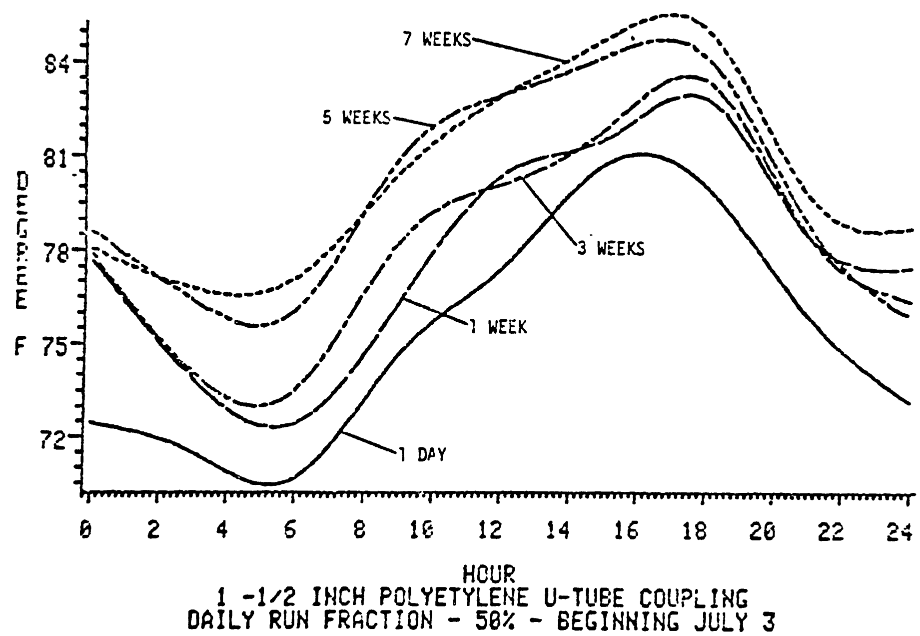


Figure 20. Daily Return Water Temperatures at One, Three, Five, and Seven Weeks of 50% Run Fraction

performance and a slightly lower temperature during peak periods. The polybutylene coupling returns water about 1.0^{o}F warmer during off peaks times. The double 3/4 inch returns water about 0.5^{o}F warmer than the 1-1/2 inch polyethylene during the peak and 1.5 to 2.0^{o}F warmer during off peak. On peak temperatures of the 3 inch and 2 inch concentric are about 1.0^{o}F warmer than the 1-1/2 inch polyethylene and 1.5 to 3.0^{o}F warmer during off peak. The single 3/4 inch polyethylene is almost always 3.0 to 4.0^{o}F warmer than the 1-1/2 inch.

Figure 20 shows the increase in daily temperature values of the 1-1/2 polyethylene U-tube during the 50% run fraction test. The lower curve is the daily profile on the first day of the test while the upper curve is is after seven weeks of 50% run fraction. Fifty percent corresponds to a run fraction experienced during a day slightly below design load.

Figures 21 and 22 show the temperatures in two of the couplings at various depths during a 45 minute run that was preceded by a 15 minute off period. Notice that in the 1-1/2 inch polyethylene approximately two-thirds of the temperature drop occurs from the entrance to the bottom (100'), another 18 to 20% from the bottom to the mid-point in the riser and only about 12 to 15% from the mid-point to the outlet. This phenomenon is caused by some short circuiting but primarily is a result of the water in the downcoming tube being at a high temperature.

In the concentric tube the water entering the coupling is normally about 1.0^{o}F warmer than the water at the bottom. This is the amount of "short circuiting" that occurs in the loop and it may be calculated directly. About two-thirds of the heat transfer occurs in the lower

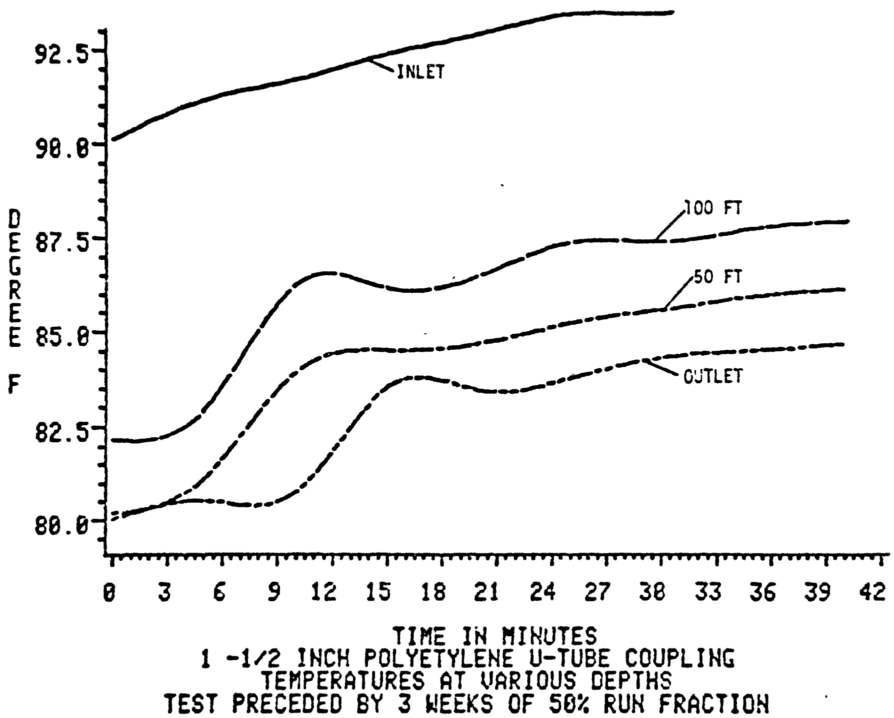


Figure 21. U-tube Water Temperatures at Various Locations

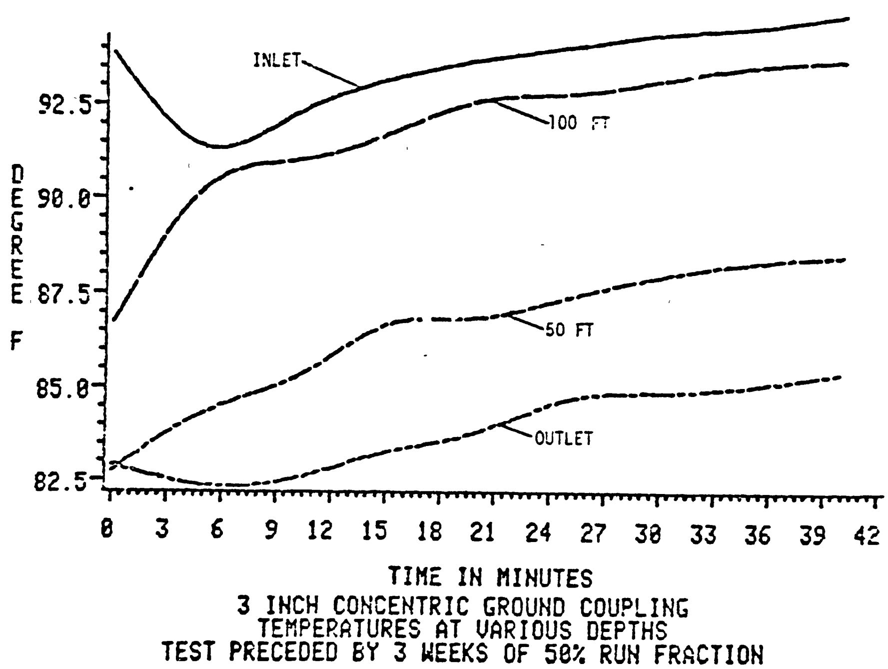


Figure 22. Concentric Tube Water Temperatures at Various Locations

half of the coupling because of the higher temperature and reduced "short circuiting". The temperature of the inlet and outlet water at the start of these tests can be ignored since they reflect primarily the shallow earth temperature near the top of each coupling. Notice also the slight jump in temperature in each temperature plot. This results from the relatively warm water in the return and supply headers, which is surrounded by the warm shallow soil at startup, being further warmed by the heat pumps before entering the coupling. It takes water approximately 15 minutes to pass through the heat pump, the header and up and down the 1-1/2 inch U-tube. This cycle requires 24 minutes for the 3 inch concentric.

Figure 23 shows the results of two-day test using only four loops during a 50% daily run fraction. The 3 inch concentric and the polybutylene loops were closed. Temperatures increased approximately $7^{o}F$ in the first day and about 0.5 to $1.0^{o}F$ during the second day. Figure 24 shows the increase in the inlet and outlet temperatures of the 1-1/2 inch polyethylene coupling. The temperatures were taken from 5:00 to 5:40 p.m. on consecutive days before and after the two loops were closed. The flow rate was increased from 1.75 GPM/coupling to 2.5 GPM/coupling. Notice the reduced temperature difference between the inlet and outlet streams that results from the reduced heat pump capacity at the higher inlet temperatures.

Figure 25 shows temperatures taken during an extended off period at the 100 and 50 ft depths in the 1-1/2 inch polyethylene U-tube and 3 inch concentric couplings. The temperature falls faster in the U-tube for the first 15 minutes because of its smaller thermal mass of water. The higher temperature of the concentric represent the higher operating

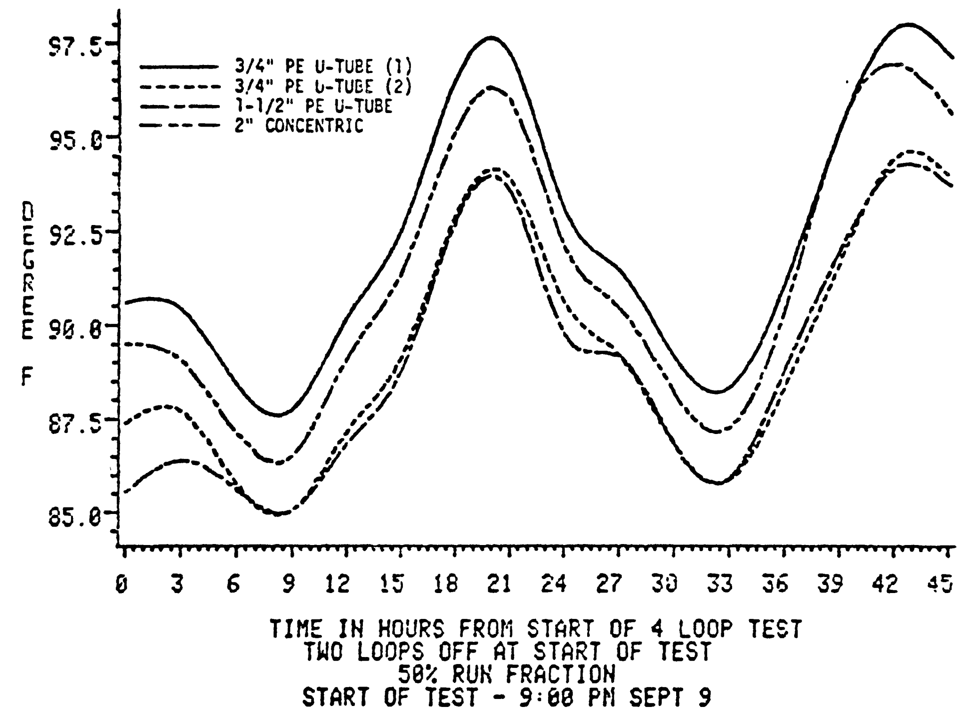


Figure 23. Return Water Temperatures for Four Loop Operation

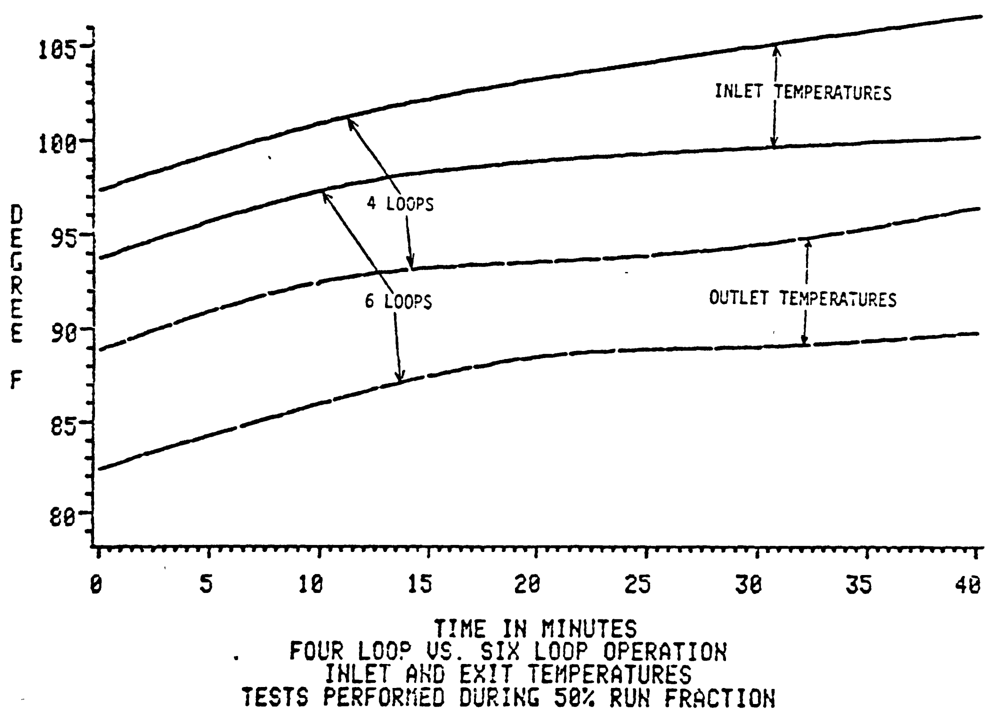


Figure 24. Water Temperatures for Six and Four Loop Operation

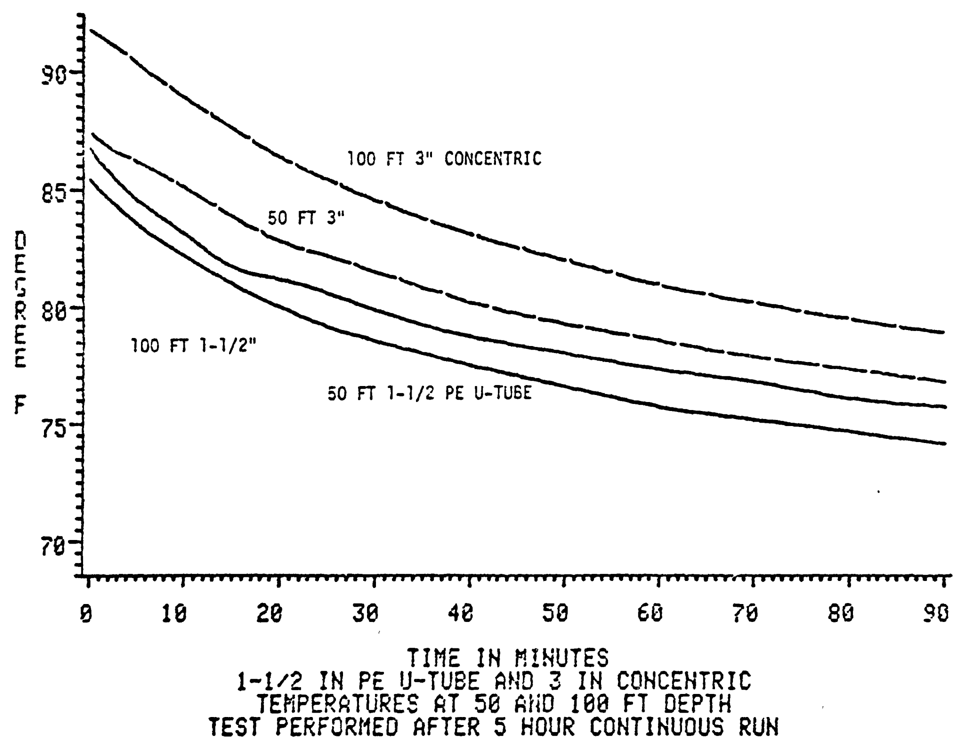


Figure 25. Recovery Water Temperatures

temperature of this coupling. Table III shows the results of another test performed at a later date when the heat pump was turned off for one week.

TABLE III

TEMPERATURE FOR ONE WEEK RECOVERY

Coupling	Depth in Feet	September 26 Temperatures (^{o}F)	October 3 Temperatures (^{o}F)
U-Tube	100	82.2	66.2
U-Tube	50	78.7	66.1
3" Conc.	100	86.1	67.6
3" Conc.	50	81.7	67.4

Table III indicates that long term heat storage using vertical tubes in the ground is not feasible in this type of soil. Table IV shows the rise in temperature in bore holes located seven feet from the 1-1/2 inch polyethlene U-tube and one at a distance approximately 100 feet from the test site. The test was performed after the 62% run fraction test.

This table shows the limited amount of significant heat diffusion outside a seven foot radius of earth from ground-couplings buried in clayey soils.

TABLE IV

INCREASE IN EARTH TEMPERATURE NEAR 1-1/2 INCH POLYETHYLENE COUPLING

Depth (Feet)	Temperature in Undisturbed Bore Hole (^{O}F)	Temperatures in Hole Located 7 Ft. From Coupling (^{O}F)
20	63.7	64.8
30	61.6	63.1

2.5 Conclusions and Recommendations

Many recommendations can be drawn from the results of the tests performed on the parallel ground-couplings system. The results shown indicate primarily the relative heat transfer capabilities of the six designs in a prolonged test. These characteristics are very important in designing a system but others must also be considered. These include cost, ease of installation, availability of material, reliability and pressure or head loss. Some designs can be eliminated (such as the 5 inch PVC concentric arrangement shown in Figure 1) because they are costly, difficult to install, unreliable and perform poorly. Many others however have cost-performance or performance-pressure loss trade-offs. Therefore comments based on this test will be made concerning the above mentioned characteristics of each of the six ground couplings. A caution must be made concerning the cost of materials. They are highly dependent upon shipping cost and the number of price mark-ups occurring between the manufacturer and customer. These factors coupled with the relative infancy of the systems in the marketplace result in a large variation in cost. Installations cost also vary widely as alluded to in Section 1.3.

The 1-1/2 inch polyethylene and the 3/4 inch double U-tubes have about the same performance during peak loads. The 1-1/2 coupling has only slightly lower heat transfer capability at most other times. In terms of pressure loss it and the polybutylene coupling have the smallest values. It can be installed in much deeper loops with a much larger flow rate without significant increases in pressure loss and the accompanying increase in pumping requirements.

The couplings are relatively easy to install with the proper equipment. The polyethylene used in this test is high density ASTM 3408 and 3406. This pipe should be fused thermally and this requires an additional equipment investment. Butt fusion equipment was used in this installation, but the less expensive and less bulky socket fusion equipment can also be used. However, the 1-1/2 polyethylene U-tube that is butt fused requires a minimum 5 inch bore hole if couplings greater than 100 feet are to be installed (a 4.5 inch bore hole was used for this test). Therefore, a 1-1/2 inch socket fused coupling would require an even larger bore hole because of the larger width of the socket U-bend. The Schedule 40 polyethylene pipe used in this test is very stiff and resistant to crimping. It can be stuffed to great depths (a 375 foot loop was recently installed near the test site) if bore hole size is sufficient. Caution should be taken against using ASTM 2306 polyethylene which is also classified as high density. This pipe can either be clamped or thermally fused. It does not have the strength and crimping resistance necessary in some installations. Cost savings are minimal.

The double 3/4 inch polyethylene coupling had the best thermal performance except during periods of highest loading, which included the afternoons of the 62% run fraction and the eighth week of the 50% run fraction. This pipe is much easier to handle during installation, it is also fused easily and is very easy to stuff and does not require a very large bore hole. The problems of "coiling" down the bore did not occur in the 100 ft. installation. This problem may limit the depth to which this couplings can be installed. If a parallel flow arrangement is used, the pressure drop is 3 to 4 times as large as the 1-1/2 inch loop and if a series flow is used the drop will be roughly 30 times as large. At the time of installation pipe cost of the double 3/4 inch loop was 25% lower than the 1-1/2 inch.

The 1-1/2 inch polybutylene U-tube performance was almost equal to the 1-1/2 inch polyethylene U-tube during periods of moderate loads (winter test and early period of 50% summer run fraction). However, its recovery is slower, which results in poorer performance during light loads, and performance dropped off slightly during heavy loads. This pipe is either clamped or socket fused. Clamping with all stainless steel clamps eliminates the necessity of fushion equipment but reduces reliability. At the time of installation the socket U-tube fittings available were subject to cracking during installation. However, tougher, more flexible socket fittings are now available. The thinner wall polybutylene does not have the crimping resistance of 3408 or 3406 polyethylene. More care must be taken during installation. A five inch bore hole was required for the 100 foot loop. The two legs were installed so that the curvature or "memory" of the coiled pipes opposed each other at the U-tube fitting. The resulting coupling was much straighter than a U-tube not installed in this manner. Pressure drop is equal to the polyethylene and larger bore holes are required for loops of equal depths. Cost could not be compared at the time of installation because of differences in methods of distribution to local pipe dealers.

The 3 inch concentric pipe had performance roughly equal to the polybutylene U-tube. The difference was primarily due to the damping of temperature variations of the 3 inch coupling. The large amount of water in this coupling and resulting weight made installation by hand difficult. A larger fusion machine is necessary. Although 3 inch coils are available, straight joints of 10, 20 or 38 feet length are recommended. The straight pipe is much easier to stuff and the butt fusion joints provide an approximately 1/4 inch ridge inside the pipe which increases the relatively low internal heat transfer coefficient in larger pipes. In some areas U-tubes are difficult to stuff through boggy or sandy layers of soil because of the collapse or near collapse of bore hole walls. The straight pipe of concentric couplings may offer an advantage over U-tubes in these situations. The pressure drop in a concentric pipe with a 3/4 inch schedule 80 U-tube is 6 to 8 times as large as the 1-1/2 inch U-tubes. Cost is 15% higher than the 1-1/2 inch polyethylene U-tube. The performance of this coupling was enhanced by the ridges in this pipe at 10 foot intervals which improved the relatively low heat transfer coefficient.

The 2 inch concentric has thermal capabilities about 10% lower than the 3 inch, and reduced camping capacity. Installation of this loop is the simplest of the six couplings because of its stiffness and small diameter. Pressure loss is 8 to 10 times as large as the 1-1/2 inch U-tubes and cost was equal to the polyethylene at the time of installation. However, thin wall (SDR 19) 2 inch ASTM 3408 polyethylene pipe is not widely available.

The single 3/4 inch polyethylene U-tube is the least expensive, it is easily installed at depths up to 175 feet, requires a small bore (3 inch or even less in some soils) and is widely available. Its reduced heat transfer capabilities are apparent in the results of these tests. Pressure drop is approximately 16 times as large as in the 1-1/2 inch U-tubes. Therefore series flow arrangements are virtually excluded.

The results of these tests cannot be universally applied, because of the wide variation of local variables. It is therefore necessary to develop procedures that in some way account for these variations. The next step is to design a computer program that simulates the performance of all six couplings for the variables characteristic of the tests performed locally. This simulation can then be used to predict performance for ground-coupling variables not reproducable at the test site. The simulation can also be used to check simplified design procedures.

CHAPTER III

HEAT TRANSFER COEFFICIENTS IN VERTICAL GROUND-COUPLINGS

3.1 Significance of Heat Transfer Coefficients

The primary resistance to heat flow in properly installed ground-couplings is caused by the low thermal conductivity of the pipe wall and earth. However, in some cases the thermal resistance of the boundary layer (film) becomes significant. This occurs when water or brine flow is laminar or in the early transition stage. In this regime water flows smoothly and does not mix. Significant temperature change is experienced between the bulk water temperature and the inside pipe wall. Temperature differences across the layer may be several degrees at heat flow rates characteristic of vertical ground-couplings. Additionally, the resistance to heat flow at the boundary layer is always significant while heat is being transferred when forced flow is stopped (natural or free convection). For calculation purposes the heat transfer capability of boundary layers are often expressed as heat transfer coefficients.

Boundary layer heat transfer coefficients characteristic of vertical ground-couplings are not easily calculated since flow regimes are often transition, mixed (laminar and natural) or natural. General equations for coefficients in the transition regime are not easily developed and often heat transfer text avoid listing equations.

Additionally, the general equations for natural or mixed convection do not fall within the range of application for ground-coupling because of the very large L/D ratios encountered.

Fortunately, the low thermal conductance of the plastic pipe normally predominates in the calculation of the overall conductance from the bulk water to the outside pipe wall. Therefore large errors in calculating boundary layer heat transfer coefficients result in small errors in overall conductance. This is particularly true in transition flows. For example, the equivalent heat transfer coefficient from the water to the outside pipe wall is

$$h_{eq} = \left(\frac{r_o}{r_i h_i} + \frac{r_o}{k_p} \ln \frac{r_o}{r_i} \right)^{-1} \tag{3.1}$$

If transition flow occurs in a 1-1/2 inch schedule 40 PE pipe and an error of 50% is made in determining the film coefficient of 100 Btu/hr-ft-F. The actual coefficient is

$$h_{eq} = \left[\frac{0.0792}{(0.0671)(100)} + \frac{.0792}{0.226} \ln \frac{0.0792}{0.0671} \right]^{-1} = 14.31 \text{ Btu/hr-ft}^2\text{-F}$$

If the value is erroneously determined to be 150, using Equation 3.1 yields

$$h_{eq} = 15.15 \text{ Btu/hr-ft}^2\text{-F}$$

Therefore a 50% error in h_i resulted in a 5.9% error in calculation of the overall heat transfer coefficient. Therefore small errors in film heat transfer coefficient determination do not have a significant effect on overall results. However, significant error can result if these values are neglected especially in laminar, free or mixed convection regimes.

3.2 Literature Survey

The vertical ground-coupling provides an interesting combination of problems when solving for heat transfer coefficients. Classical methods of solution use either uniform wall temperature or uniform heat flux as boundary conditions. However, the vertical ground-coupling has neither of the above and is highly transient. Metais and Eckert (12) have summarized the work done concerning heat transfer regimes in flow through vertical tubes. They have devised a graphical presentation of flow regimes with Reynolds number (Re) being ordinate and the abscissa is the product of the diameter/length ratio, Grashof (Gr_D) and Prandtl (Pr) numbers. However, the diagram is valid for

$$10^{-2} \leq PrD/L \leq 1.0.$$

Many ground-couplings fall outside this range in the cooling mode. (For water at 90^oF in a 100 ft. 1-1/2 inch U-tube, $PrD/L = 0.81 \times 10^{-2}$). Most couplings are close enough to the lower limit to warrant consideration of this work.

Two flow inducing forces determine the magnitude of the fluid velocity. The first is pressure gradient or forced flow and the second is body or natural forces that are a result of density gradients in fluids. The authors state that although work has been done to determine which regimes predominate, other parameters complicate the calculation of heat transfer coefficients. Body forces may either oppose or support pressure gradient forces.

An earlier work by Colburn and Hougen (13) develops an equation for the heat transfer coefficient as

$$h = 0.082\ k\ \left(\frac{\beta g \rho^2 \Delta T}{\mu^2} \right)^{1/3} \tag{3.2}$$

where ΔT is measured across the boundary layer.

They report that experimental results for air yields

$$h = 0.115k\ \left(\frac{\beta g \rho^2 \Delta T}{\mu^2} \right)^{1/3}. \tag{3.3}$$

The authors also give a transitional velocity. If the mean fluid velocity is above this value forced convection predominates. Below this value natural convection predominates. This velocity is given by

$$V_t = 19.8\ (\beta g \rho^2 \mu \Delta T)^{1/3} \tag{3.4}$$

Hartnett and Welsh (14) conducted an experiment for natural convection in a vertical tube. The heat rates studied were much higher than those encountered in ground-couplings. Results are presented on a Log-Log plot of Nusselt numbers (Nu) versus GrPr. This generalized plot is for circular tubes with Prandtl number greater than 0.1.

Martin and Cohen (15) conducted an experiment and presented results in a plot similar to that of Hartnett and Welsh. They also presented the following equations.

$$Nu = \frac{Gr_r Pr\ r}{892\ L} \text{ when } 10^3 < Gr_r Pr < 10^{5.2} \tag{3.5}$$

$$Nu = 0.426\ \left(\frac{Gr_r\ Pr\ r}{L} \right)^{0.28}\ 10^{5.2} < Gr_r Pr < 10^{6.15} \tag{3.6}$$

Note that the length dimension (L^3 or D^3) in determining Gr is replaced by r^3 to determine Gr_r.

Brown and Gaurvin (16) summarized much of the previous work concerning combined free and forced convection in vertical tubes. They also conducted several experiments to verify results. They dealt with both laminar and turbulent mixed convection. Because of the modest temperature differences occuring in ground-couplings, this discussion can be restricted to laminar flow. The authors approach the solution in two parts. The first is when bouyancy forces and forced flow aid each other (i.e. hot water at bottom of tube and forced flow from bottom to top) and the second is when they oppose each other. Results for laminar aiding flow are plotted graphically in terms of Nu vs. Gr_D/Re_D. They also suggest the equation

$$Nu = 0.931\ (Gr_D/Re_D)^{0.389} \tag{3.7}$$

The authors also suggest that the boundary of equal free and forced convection can be represented by

$$Re = 9.2\ Gr_D^{0.417}\ Pr^{-0.108} \tag{3.8}$$

Values $\pm$ 100% of this line are considered to be mixed convection. The equation suggested for pure free laminar convection is

$$Nu_L = 0.59 \ (Gr_L Pr)^{1/4} \tag{3.9}$$

In the discussion of combined convection in which buoyancy forces oppose forced flow, the authors state that flow is unstable and transition from laminar to turbulent occurs at very low values of (GrPr). This region is similar to the transition region of forced flow in that coefficients are difficult to predict.

> "If buoyancy forces are larger then the heat transfer rate will be higher than predicted from forced flow but there is no satisfactory equation at present for predicing their actual values".

If forced flow is in the fully turbulent regime heat transfer coefficients can be assumed to be infinitely large when calculating the equivalent heat transfer coefficients in plastic pipe for Equation 3.1. The first term of the right side of the equation can be neglected. This can not be done if the thermal conductivity of the pipe is large as is the case with metal pipe. The coefficient can then be predicted by the Dittus-Boelter Equation (17).

$$Nu_D = 0.023 \ Re_D^{0.8} Pr^n \tag{3.10}$$

when n is 0.4 for heating and 0.3 for cooling.

The figure produced by Metais and Eckert indicates that in transition forced flow regimes the effects of natural convective forces are negligible. The focus in these cases is to determine the forced convection heat transfer coefficient. Equations in the laminar region (Re < 2300) and turbulent region (Re > 10,000) are readily available. In the transition region (2300 < Re < 10,000), equations are not consistent. Text often include something similar to the following (18).

> "A word of caution is appropriate cocerning the transition from laminar to turbulent flow. The region is defined by approximately 2000 < Re_D < 10,000. Prediction of heat transfer and friction coefficients is uncertain during transition".

There appears to be no classical equation for heat transfer coefficient in this region of forced flow. Sieder and Tate (19) conducted an early extensive test of heat transfer characteristics in the laminar and transition regions. The summary of their testing in the transition region was a set of curves with which they were able to predict coefficients with a $\pm$ 20% accuracy for a wide variety of fluids. Kreith (20) has presented this graph in a clarified form of j vs. Re_D.

In the experiment described above Sieder and Tate (18) also developed the classical equation for coefficients in the laminar forced flow regime.

$$Nu = 1.86\ (Re\ PrD/L)^{1/3}(\mu_b/\mu_s)^{0.14} \qquad (3.11)$$

Extensive work is reported by Kays and Crawford (21) concerning heat transfer in laminar flow for a variety of conditions. Methodologies for calculation of coefficients for many situations are suggested, including flow in concentric tubes. However, neither Equation 3.11 or the methods of Kays and Crawford have provisions for calculating coefficients when buoyancy forces are of the magnitude of those encountered in vertical ground-couplings. Therefore, considerable underprediction will result if these forces are not considered.

In laminar flow entrance effects are often significant in calculating average overall heat transfer coefficients. Heaton, Reynolds and Kays (22) present a method of calculating Nussult numbers in the entrance region of tubes where the values are higher than those in fully developed flow region down the pipe. Results are in tabular form and Nusselt number can be interpolated by entering the value of Prandtl number and a nondimensional distance from the entrance defined as

$$x^+ = \frac{2(x/D_h)}{RePr} \tag{3.12}$$

These effects can be significant especially in concentric ground-couplings in which laminar flow is often encountered.

3.3 Heat Transfer Coefficient Experimental System

The experiments discussed in the previous section did not include tests of conditions similar to those present in vertical ground-couplings. The combination of low heat transfer rate per unit surface area, large L/D ratios and relatively small vertical density gradients are not considered. Most equations and graphs include the above combination within their range of applicability. However, the results present a discrepancy. In the case of free convection of a 100 foot 1-

1/2 inch P.E. U-tube at $90^{o}F$ rejecting heat at a rate of 2000 Btuh (a typical value for one tube), the free convection coefficient is calculated to be 13.4 Btu/hr-ft^2-F using Equation 3.9, 17.5 using Equation 3.2 and 23.2 using 3.3.

The use of finite difference equations over a longer period of time makes reasonably accurate prediction of coefficients in the laminar, mixed and free convection regimes necessary. Coefficients in the transition and turbulent regime can be inaccurate $\pm$ 25% without effecting overall FDE accuracy. Although it is possible to design ground-couplings so that laminar flow does not occur, free convection coefficients must always be predicted (unless the water pump runs continuously) regardless of forced convection regimes. Mei (23) has reported that using values of heat transfer coefficients calculated by using laminar forced convection equations substantially underpredicts heat transfer in his FDE formulation in both vertical and horizontal ground-couplings (14).

In order to more accurately predict heat transfer coefficients for the FDE formulation described in Chapter IV and the simplified method of Chapter V, it was necessary to design and construct the experimental system shown in Figure 26. This system allows the experimental determination of equivalent heat transfer coefficients in transition, laminar-mixed and free convection regimes. Since the coupling casing is steel, temperature differences across the wall are small compared to boundary layer differences and can be easily accounted for in order to arrive at boundary layer coefficients.

The ground-coupling is a 2 inch schedule 40 galvanized steel pipe with a 1/2 inch schedule 40 PVC inner tube. Total coupling length is 41

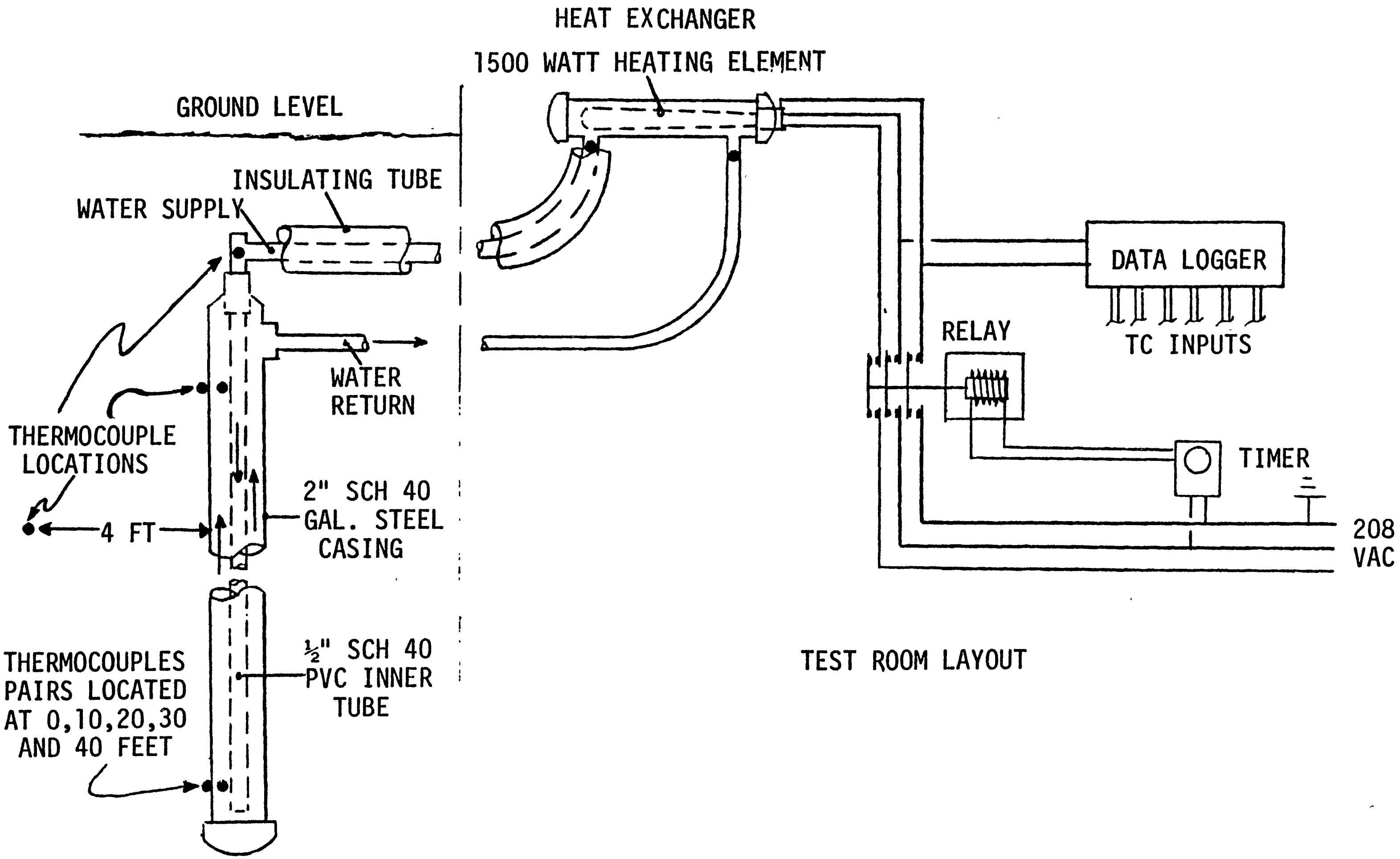


Figure 26. Heat Transfer Coefficient Experiment

feet. Water is normally injected down the dip tube and up the annular region. Calibrated thermocouples are located in the annular region and on the outer pipe wall at 0.5, 10.5, 20.5, 30.5, and 40.5 feet above the bottom of the inner tube. Heat for the test was supplied by a 1500 watt resistance element. The system is capable of being cycled by a timer identical to the one used for the heat pump system. A data logger is likewise turned on and off by the timer so that the time of cycle can be determined. A 90 watt pump is used to provide water flow rates up to 2.4 GPM. The water supply line to the coupling is placed inside a larger tube so that the buried line does not reject significant amounts of heat before entering the test section.

3.4 Results

The experimental system was operated at a 75% run fraction for two weeks so that typical heat transfer rates could be simulated. Tests were conducted for flow rates of 2.0, 1.5, 1.0 and 0 (free convection) for water temperatures between 85 and 100^{o}F. All flowing tests were for buoyancy forces aiding forced flow. Tests for buoyancy forces opposing forced flow with dip tube injection require cooling the water and a properly sized system was not available. Table V gives a summary of three of the tests conducted on the system for three different flow regimes.

The first test was conducted at a flow rate of 2.0 GPM and at an average water temperature of 90^{o}F. Temperature differences, shown in Figure 27, are measured from the bulk water to the outside pipe wall. Average heat flow was calculated by measuring the water temperature difference from one vertical location to the next and applying the equation.

$$q = \dot{m} c_p (T_{N-1} - T_N) \tag{3.13}$$

This average heat loss was used in the equation

$$T_i = T_o + \frac{q \, \ell n \, (D_o/D_i)}{2\pi k_s \Delta L} \tag{3.14}$$

to find the inside wall temperature. The value of h was then calculated from

$$h = q/\pi D_i \Delta L (T_b - T_i) \; . \tag{3.15}$$

TABLE V

HEAT TRANSFER COEFFICIENTS IN STEEL GROUND-COUPLING

Reynolds No.	Distance from Entrance Feet	Experimental Heat Transfer Coefficient Btu/hr-ft^2-F	Theoretical Value Btu/hr-ft^2-F
3100	0.5	202	
"	10	104	
"	30	55	
"	Average	102	90 (19)
1230	0.5	109	
"	10	56	
"	30	35	26.5 (16)
"	Average	56	24.6 (21), 14.0 (19)
Natural	Average	24.2	19.1 (13), 26.8 (13), 29.6 (15)

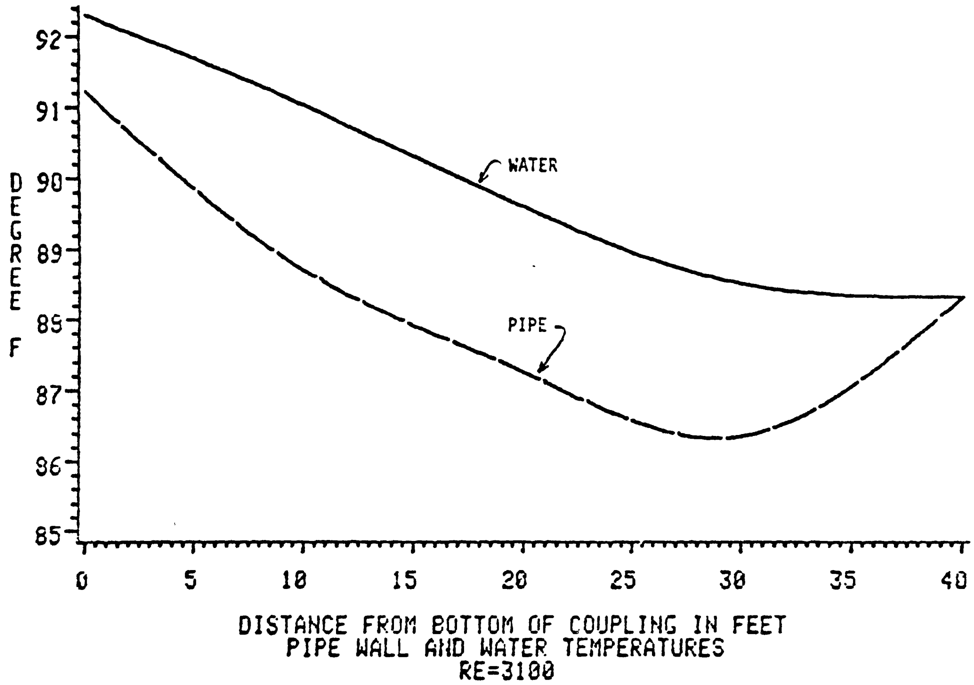


Figure 27. Differential Temperatures for Transition Regime

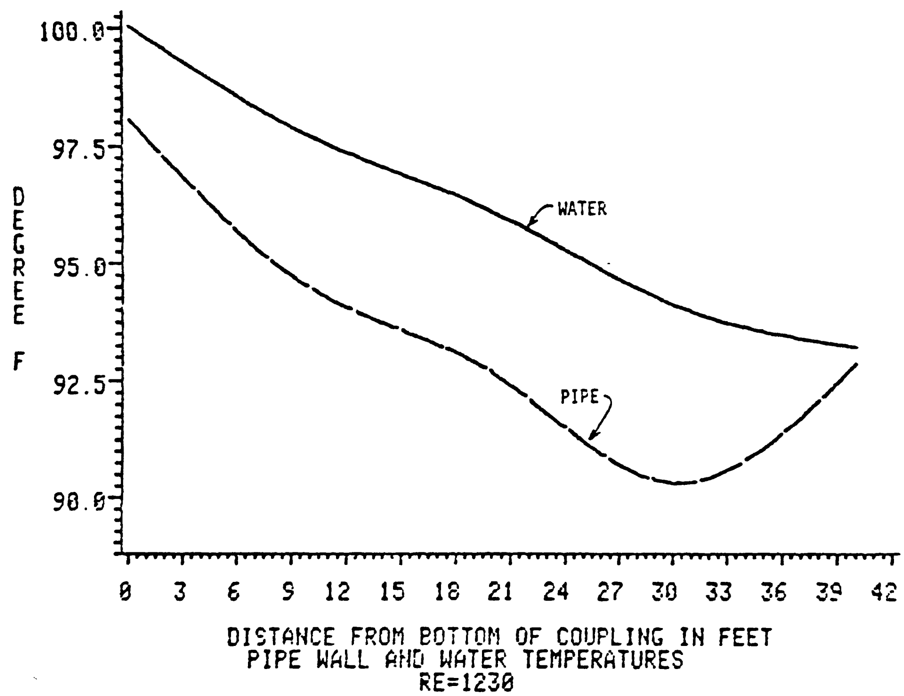


Figure 28. Differential Temperatures for Mixed Convection

The heat transfer coefficients near the entrance are substantially higher as expected. The average values shown are a graphical average of the local values. The theoretical value shown is calculated by the computer program HCAL, which uses an exponential curve fit equation of the chart published by Krieth, for the transition regime. The equations are for 2200 < Re < 7000,

$$j = (0.0044 - 0.0012\, e^{\frac{2200 - Re}{1300}}) - (1.4 - \frac{Re}{5000})(\frac{L/D - 50}{8.7 \times 10^{10}})^{1/3} \qquad (3.16)$$

and for 7000 < Re < 10,000,

$$j = -6.67 \times 10^{-8}\, Re + 0.00486 \qquad (3.17)$$

The average heat transfer coefficient h is derived from the non-dimensionial value j by the equation

$$h = \frac{j c_p \rho V}{Pr^{2/3} (\frac{\mu_w}{\mu_b})^{0.14}} . \qquad (3.18)$$

The experimental value of 102 Btu/hr-ft^2-F agrees well with the calculated value of 90, for transition flow regimes. Although this error of about 12% seems large, the resulting error of equivalent heat transfer coefficient is small for plastic pipe.

A similar test was conducted in water at 95°F with the flow rate reduced to 1.0 GPM. Temperatures are shown in Figure 28 and values for the heat transfer coefficient were calculated in the same manner. However, the experimental results shown for average coefficients differ radically from the value calculated using the methods described by Kays

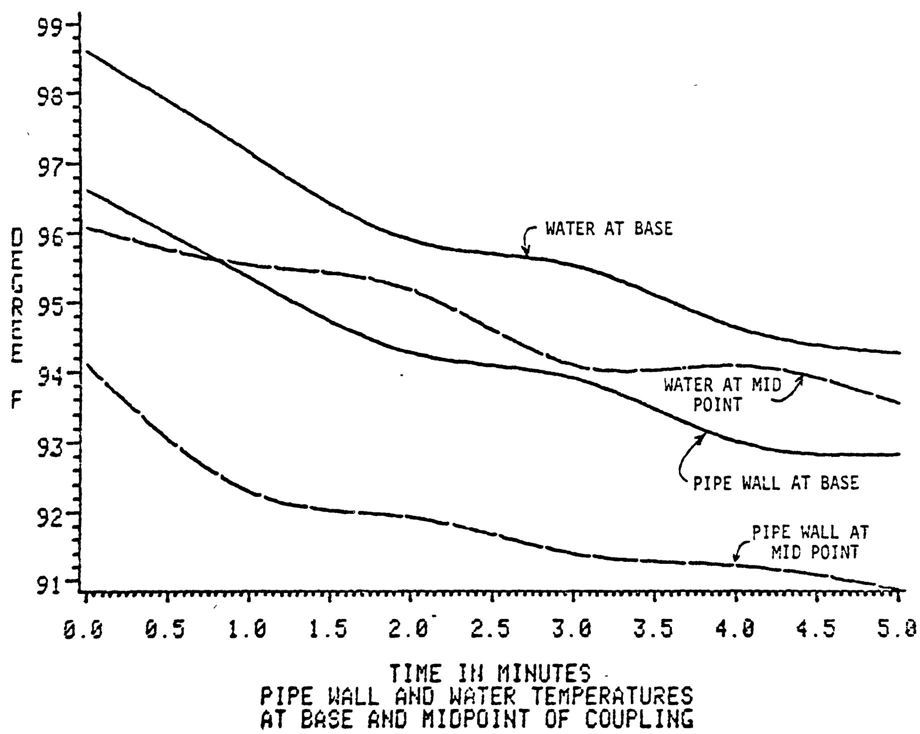


Figure 29. Differential Temperatures for Natural Convection

and Crawford (24.6 Btu/hr-ft^2-F) and Equation 3.11 (14.0). These two values do not consider the flow to be mixed convection. A more reasonable agreement is shown for the value calculated 30 feet from the entrance. The experimental value is 35 Btu/hr-ft^2-F and at this distance entrance effects are assumed to be negligible. A value of 26.5 Btu/hr-ft^2-F is calculated using Equation 3.7, which is an equation for local value during mixed convection.

A third test determined free convection coefficients for a five-minute period immediately after forced water flow was stopped. Although temperatures were taken at all levels only the values for the 40 and 20 ft. levels are shown in Figure 29. Only an average coefficient for the entire coupling can be calculated since the temperature at a particular level is influenced by not only loss through the pipe wall, but also by temperature changes due to buoyancy effects. Notice how quickly the temperature at the 40 ft. level (bottom) decreases compared to the 20 ft. level. This higher rate is due in large part to the water of lower density rising and being replaced by cooler denser water. Therefore only the total coupling heat transfer rate could be calculated using

$$q = \rho\pi r_j^2 L c_p (T_5 - T_o). \tag{3.19}$$

The heat transfer coefficients were then calculated using Equations 3.14 and 3.15. The experimental value of 24.2 Btu/hr-ft^2-F is in fairly good agreement with the experimentally determined Equation 3.3 (26.8) and somewhat higher than the value calculated using Equation 3.2 (19.1).

3.6 Conclusions

The results of this test are far from universally conclusive and additional work is warranted. However, the values obtained here can be applied to vertical ground-couplings with acceptable accuracy. The major uncertainity is the method for calculating heat transfer coefficients in the laminar mixed flow regime. This is especially true near the entrance of the tube and downstream from flow disturbances such as the ridges caused by butt fusion joints in polyethylene pipe. This would tend to increase coefficients. The tests result in the following recommendations concerning the calculation of boundary layer heat transfer coefficients.

1. Equations 3.16 and 3.17 yield sufficient accuracy for transition forced flow in vertical platic ground-couplings.
2. Equation 3.2 underpredicts and Equation 3.3 overpredicts the value of free convection coefficients during the off periods occurring in ground-couplings.
3. Heat transfer coefficients in ground-couplings can best be calculated using mixed convection equations when the Reynolds number due to forced convection is below 2500.
4. Entrance effects on average coefficients for both mixed (laminar and natural) and forced transition regimes are substantial.

0.112500
1.740740

CHAPTER IV

GROUND-COUPLING SIMULATION USING FINITE DIFFERENCE EQUATIONS

4.1 Basics of Finite Difference Equations

The two dimensional equation for temperature variation in cylindrical coordinates (24) is the basis for simulation of vertical ground-couplings.

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial Z^2} + \frac{q}{K} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \tag{4.1}$$

Several simplifications can be made to this partial differential equation (PDE), but the variation of the boundary conditions encountered in ground-couplings makes exact analytical solutions impractical. Finite difference equations are a powerful tool for the solution of PDEs. Solutions are obtained by placing a finite number of points in some sort of grid pattern within the conducting medium. The equations for the temperatures at these points or nodes are described by FDEs that are arrived at by direct replacement of the PDE or by an energy balance performed on the solid bodies surrounding each finite point. For example, if points 1, 2, and 3 in the r-direction are separated by a distance Δr then,

$$\left(\frac{\partial T}{\partial r} \right)_2 = \frac{T_3 - T_1}{\Delta r} \tag{4.2}$$

and if the derivatives are also estimated at r = 1.5 and 2.5,

$$\frac{\partial}{\partial r}\left(\frac{\partial T}{\partial r}\right)_2 = \frac{\left(\frac{\partial T}{\partial r}\right)_{1.5} - \left(\frac{\partial T}{\partial r}\right)_{2.5}}{\Delta r} = \frac{\frac{T_1 - T_2}{\Delta r} - \frac{T_2 - T_3}{\Delta r}}{\Delta r}$$

$$\left(\frac{\partial^2 T}{\partial r^2}\right)_2 = \frac{T_1 - 2T_2 + T_3}{\Delta r^2} \tag{4.3}$$

This form of PDE replacement is known as central difference.

In ground-couplings heat is transferred by convection from the water in plastic pipes to the conducting medium of the pipe and ground. In such cases boundary conditions must be placed on the equations for the nodes at the interface of the form

$$k\,\frac{\partial T}{\partial r} = -h\,(T_B - T_W). \tag{4.4}$$

This also can be accomplished by direct PDE replacement or by energy balance methods to obtain a FDE.

Simplifications of Equation 4.1 are possible and will be presented later in this chapter. However, the right hand term of this equation can not be eliminated primarily because of the relatively small value of the thermal diffusivity of the ground. The rapidly changing boundary conditions also contribute to the unlikely occurrence of ground-couplings approaching steady state heat transfer during normal operating conditions. The result is FDE solutions are always transient.

Two basic schemes and combinations of these schemes are possible when solving transient FDEs. Explicit formulations utilize forward time difference in that the temperature of a node at a future time increment is expressed in terms of the surrounding node temperature at the beginning of the time step. The temperature of all the nodes are calculated before going to the next time increment. If Equation 4.1 is reduced by assuming there is no internal heat source ($q = 0$) and no variation with respect to angle ($\frac{\partial^2 T}{\partial \theta^2} = 0$) or axial direction ($\frac{\partial^2 T}{\partial z^2} = 0$) we have

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \, . \tag{4.5}$$

The PDE for Equation 4.5 becomes the FDE for node N (25)

$$\frac{T_{N-1} - 2T_N + T_{N+1}}{(\Delta r)^2} + \frac{T_{N+1} - T_{N-1}}{2r\Delta r} = \frac{T_N' - T_N}{\alpha\Delta t} \, , \tag{4.6}$$

where N is increasing radially outward. Solving for the unknown temperature we have

$$T_N' = \frac{\alpha\Delta t}{\Delta r}\left[\frac{1}{\Delta r} + \frac{1}{2r}\right] T_{N+1} + \left[1 - \frac{2\alpha\Delta t}{\Delta r^2}\right] T_N + \frac{\alpha\Delta t}{\Delta r}\left[\frac{1}{\Delta r} - \frac{1}{2r}\right] T_{N-1} . \tag{4.7}$$

The primary limitation on explicit formulations of this type is that the coefficient of the temperature at node N must be greater than or equal to zero for stability of solution. Therefore the Fourier number (Fo) has the restriction

$$Fo \equiv \frac{\alpha\Delta t}{\Delta r^2} \leq 0.5 \, . \tag{4.8}$$

The result is that time step size must be limited in Equation 4.7 once a value of Δr is selected in order to insure a valid solution.

Restrictions on Fo also result if two or three dimensional FDEs are used. An additional restriction occurs when the FDE is applied at a node at which a convective boundary condition applies. When equations such as (4.4) are applied to FDEs, the coefficient for the boundary node FDE takes the form,

$$\frac{A\Delta r^2}{\alpha\Delta t} - B - C\,\frac{h\Delta r}{K}$$

where A, B, and C are constants depending on the physical arrangement and number of dimensions of heat flow at the boundary. Again this quantity must be positive for FDE stability. The result is that when values of h are large, such as those encountered in water flow, the time step size is again restricted.

This restriction on time step can be avoided by the use of implicit formulations that incorporate backward time steps in the FDE's. Applying this method Equation 4.5 becomes (25)

$$\frac{T_{N-1}' - 2T_N' + T_{N+1}'}{(\Delta r)^2} + \frac{T_{N+1}' - T_{N-1}'}{2r\Delta r} = \frac{T_N' - T_N}{\alpha\Delta t}\,. \tag{4.9}$$

This yields

$$T_N = F_o\,[1 + \frac{\Delta r}{2R}\,]\,T_{N+1}' - F_o[1 - \frac{\Delta r}{2r}\,]\,T_{N-1}' + [2F_o + 1]T_N'\,. \tag{4.10}$$

The coefficient of the temperature in the last term of the above equation can therefore never become negative and stability is maintained regardless of time step size. Similar reasoning can be applied to the restriction concerning the boundary condition.

The use of implicit formulation is restricted in other ways. Since none of the updated temperature are known, simultaneous calculations must be performed for all temperatures of the grid. The FDEs for each node in a one dimensional equation like 4.8 can be arranged in a tridiagonal matrix and solved by methods described in (25). Accuracy is lost when rounding errors are made in the simultaneous calculations. This can occur if the grid size or time increments are excessive. A second restriction on the implicit method is that the above scheme can only be directly applied to one dimensional problems. If transient heat flow is two dimensional, explicit methods must be alternated with implicit formulations to avoid the time increment restriction.

As previously mentioned the use of FDEs for ground-coupling simulations would require vast amounts of computer time if simplifications are not made. Time steps would be restricted by high values of heat transfer coefficients in turbulent flow and radial node distances would be restricted by thin pipe walls. The following progression of simplifications are applied. The domain of a vertical concentric ground-coupling can be described as a two dimensional (radial, axial), transient problem in an infinite medium, no internal heat generation, a convective boundary condition at the inside pipe radius and a material interface at the outer pipe wall. The initial simplifications are:

1. Perfect thermal contact at the outer pipe wall.
2. FDEs are not applied to the inside tube and heat transfer from the inner tube to the water in the annulus is

accounted for by energy balance methods.

3. The coupling can be divided up into vertical increments (axial) and ground conduction in this direction can be neglected since gradients are small compared to those in the radial direction.
4. Annulus water temperature variation in the vertical direction can be calculated by energy balances on each vertical increment. That is, the energy (temperature) contained in increment N is the energy of increment N-1 plus the heat transferred through the inner and outer pipe walls.
5. Heat transfer does not vary circumferencially and radial nodes are placed at the inner and outer pipe wall and at equal 0.25 inch increments in the ground.
6. Heat does not diffuse outside a 12 foot radius of ground, therefore the node at $r = 12$ feet is held at a constant temperature.

A computer program utilizing explicit FDEs and the above assumptions was formulated and results were obtained, however energy balances were not initially used. Water in these simulations was held at a constant temperature and results are presented in terms of heat transfer for a vertical increment one foot in length.

Additional simplifications were made and results were compared with the previous simulation. The simplifications are as follows:

1. The heat storage effects of the pipe wall are neglected and the FDE at the inner pipe wall node is eliminated by the use of an equivalent heat transfer coefficient

2. The uniform radial grid is replaced by a grid with an expansion factor (Se) of 1.1 that is

$$r_N - r_{N-1} = 1.1(r_{N+1} - r_N) \; . \tag{4.11}$$

3. The expansion factor was increased to 1.25 and 1.5.

At this point the difference between the results of the initial formulation and the program with the above simplifications were small and calculation time was significantly reduced. Therefore energy balances were incorporated into a simulation with 20 vertical increments five feet in height. In order to arrive at the final forms of simulation the following steps were taken.

1. An implicit formulation replaced the explicit formulation.
2. Vertical nodes were reduced to 10 and then to 5.
3. The time step, which formally had been dictated by the average water velocity divided by the time increment for energy balance purposes, was increased to a value of 5 minutes.
4. Vertical increments were reduced to one and the FDEs were performed using average loop water temperatures.
5. Heat transfer of vertical U-tubes were implemented using equivalent diameters, heat transfer coefficients and equations for "short circuit" heat transfer.

4.2. Literature Survey

The procedures for formulating FDEs utilized by Croft and Lilley (25) are the primary reference for this work. The authors have developed FDEs for the general heat conduction PDE in cylindrical polar coordinates. Also presented are methods for implementing convective boundary conditions. They have presented a computer program to solve the tridiagonal matrix algorithm (TDMA) that results from one-dimensional implicit FDEs.

Kanchanalai (26) applied the techniques outlined in (25) directly to a vertical ground-coupling similar to the PVC design of Figure 1. The model is well developed and begins with analytical solutions and proceeds into one dimensional FDE development. The author uses both a uniform and non-uniform grid as well as an explicit and fully implicit formulations. Results are presented for both constant heat input as well as step input. The boundary condition at the pipe soil interface was considered to be a constant heat input. The temperature difference from the bulk water to the outside pipe wall is assumed to be zero.

The results of this method are presented for a soil thermal conductivity of 0.8 Btu/hr-ft-F, various run fraction, various run times and various coupling depths for a heat input of 72 Btu/hr-ft. Although calculated temperatures do not agree well with those recorded in actual installations (1) because of the low value of thermal conductivity used, the formulation yields good results even over long periods of time (125 days). The error due to the low value of conductivity is partially offset by assuming no temperature difference across the pipe wall and boundary layer.

Mei and Fischer (10) have developed a more elaborate set of FDEs to simulate a ground-coupling identical to the one used by Kanchanalai. The authors wrote FDEs for the PDEs describing the heat flow at the following locations; the water to the dip tube, within the dip tube, the dip tube to the water in the annulus, the annulus water to the outer tube, within the outer tube, and from the outer tube to the ground. The time step utilized was 0.15 seconds due to the exactness of the explicit formulation. Mei and Fischer utilized convective boundary conditions at all fluid to pipe interfaces and calculated water temperatures by energy balances on vertical nodes. Water to the coupling is injected at a constant temperature for both heating and cooling. This is done continuously and in 30 minute on-off cycles. When the water flow is stopped, heat transfer from the water is assumed to be by conduction. Results of the simulation are compared with an actual experiment of relatively short duration. The simulation matches the experiment with the exception of temperatures immediately after start-up and during off periods.

Hopkins (27) used the formulation of Kanchanalai as the basis for a one dimensional simulation of a vertical 1-1/2 polyethylene U-tube. This model includes elaborate energy balances to calculate exit fluid temperatures. Included are provisions for "short circuit" heat transfer between the tubes and the simulation is linked to the performance of a heat pump that is cycled. However, the resulting water temperatures for the cooling mode operation does not match actual installations (1).

A survey of FDE formulations related to ground-couplings is listed by Ball, Fischer and Hodgett (3). Of the nine works listed, only the work of Mei and Fischer (10) and a discontinued European model are applicable to vertical isolated coils.

4.3 Preliminary Considerations to Model Development

Finite difference equations coupled with a digital computer are a powerful way to solve the PDEs characteristic of conduction heat transfer problems. The formulations are however useless if input parameters are not evaluated by sound engineering procedures. In the case of ground-couplings it is essential to properly model the heat transfer rate of the heat pump and make the necessary corrections to this rate as conditions change. It is also necessary to input representative values for the thermal properties of the ground. A good model is also capable of making adjustment to actual physical phenomena that may cause the system to vary from the ideal case. Examples of this include water movement due to thermal or hydraulic effects. Although all variations from ideal can not be accounted for, the principle ones will be discussed before model development continues.

4.3.1 Heat Pump Performance

Actual water to air heat pump performance is a function of among other things; water inlet temperature and flow rate, air inlet temperature (dry bulb in heating, wet bulb in cooling) and flow rate, input voltages and manufacturer's quality control, which is outside the scope of this work. Manufacturers typically supply data concerning thermal performance and unit power consumption over a wide range of operating conditions. Al-Juwayhel (28) utilized a polynomial curve fit to adjust performance values for varying water flow rate, entering water temperature and entering air temperature. The result is one equation with a constant and first and degree terms for each variable. Application of this equation yields inconsistent results, especially for varying water flow. Ravindran (29) improved this equation by providing polynomial curve fits for the air and water temperatures and an exponential term for water flow rate.

Manufacturers present performance by the use of either tables (30) or curves (31). In using curves the corrections must be made for each variable independently. The methods of (28) and (29) correct all variables simultaneously. This procedure is more accurate if the performance curve for one variable changes radically when a second variable is changed. Heat pumps do not exhibit this characteristic within normal operating ranges as suggested by the independent correction used by some manufacturers.

Heat pump performance can be accurately predicted by independent correction. The method employed here will be to correct capacity and power consumption with a polynomial curve fit for inlet water temperature. The three remaining primary variables are considered in the form of correction factors. Curve fits are made for these correction factors by dividing the dependent variable by its rated value over the range of independent variables possible. The following values were arrived at for the cooling performance of a FHP LT50 by using a computer program entitled LSCF.

$$q = 61921.0 - 223.9\ T_w - 0.253\ T_w^2\ \text{(Btuh)} \tag{4.12}$$

$$P = 2247.3 + 12.29\ T_w + 0.05\ T_w^2\ \text{(Watts)} \tag{4.13}$$

Dimensionless correction factors to capacity for water flow rate, entering air wet bulb temperature and air flow rate are as follows:

$$CFC = 0.971 + 0.00366\ GPM \tag{4.14}$$

$$CWBC = -0.513 + .0314\ TWB - 0.00013\ TWB^2 \tag{4.15}$$

$$CAC = 0.98 \tag{4.16}$$

These factors for the power consumption are as follows:

$$CFP = 1.211 - 0.0393\ GPM + 0.0016 GPM^2 \tag{4.17}$$

$$CWBP = 0.410 + 0.012\ TWB - 0.000048\ TWB^2 \tag{4.18}$$

$$CAP = 0.99 \tag{4.19}$$

The corrections for low input voltage are made by reducing capacity and performance by a fixed amount according to values published by the American Refrigeration Institute (32). The voltage often dropped below recommended values and additional corrections were made according to actual experimental results.

Equations 4.12 through 4.19 consider the effects of all energy inputs to the unit (fan power, compressor heat losses) except for the pumping power. The heat added by the pump is advantageous in the winter but must be rejected by the ground-coupling in the summer. Winter variation of pump input power was greatest (refer to Section 2.2). Summer input was fairly constant at 375 watts.

4.3.2 Heat Transfer Coefficients

The results of Chapter III are applied to the FDE development. The forced convection heat transfer coefficients are calculated in a separate computer program named HCAL and are input to the ground-coupling simulation. All flows encountered in this project were either in the transition or mixed convection regimes. For flows in the transition regime, Equation 3.16 was applied directly as an average coefficient. Values in the laminar forced flow regime are considered to be in the mixed convection regime. Since no equations are available for the entrance region of concentric tubes, experimental results were applied directly to the simulation. In the 3 inch ground-coupling, the butt fusion ridges on the inside of the outer coupling at 10 foot spacing, prevent fully developed flow. This increases average heat transfer coefficients to values near those encountered in the entrance region of the heat transfer coefficient experiment. A value of 40 Btu/hr-ft^2-F was used for a 1.75 GPM flow rate at 90°F.

During off periods, the calculation of natural convection coefficients were made by a rounded average of Equations 3.2 and 3.3.

$$h = 0.10\ k_w \left(\frac{\beta g \rho^2 \Delta T}{\mu^2} \right)^{1/3} \qquad (4.20)$$

This equation is particularly convenient to use because all the properties of water can be easily input. The only remaining input is the value for ΔT which is measured across the boundary layer. The value used in this program is the film temperature differential of the previous time step.

4.3.3 Ground Thermal Properties

The most uncertain variable in almost all ground-coupling simulations is the thermal conductivity of the ground (k_g). The values of density (ρ_g) and specific heat (c_{pg}), which combined with k_g give the thermal diffusivity (α_g), also effect thermal performance. The diffusivity is defined by

$$\alpha_g = \frac{k_g}{c_{pg}\rho_g} . \tag{4.21}$$

The value of k_g and α_g for dry soil is relatively low and vertical ground-coupling should not be installed in dry soil. However, moisture in soils improves effective thermal conductivities significantly by providing a heat transfer, and in some cases mass transfer, medium that is a substantial improvement over dry voids in the grain structures (33). Soils need not be saturated in order for this improvement to occur. Therefore, the thermal conductivity of soils above the water table are increased significantly. Two zones occur above the water table that have high moisture contents (34). The capillary zone is immediately above the water table and typical moisture content ranges from 100 to 50% saturation. Above this zone is the pellicular and gravitational water zone which consist of water that is held in place by hygroscopic forces and water which is moving downward. Percent saturation in this zone for fine grained soils typically varies from 50% to 30%. This zone may extend up to the ground level in some cases.

In addition to moisture content, dry density and a general classification of soil type must be known in order to estimate thermal conductivity. Kersten (35) developed a set of equations that predict soil conductivity from dry density and moisture content. The equations for clay is

$$k_g = (0.9 \log \phi - 0.2)10^{0.01\rho_g} \qquad (4.22)$$

If the soil is sandy use

$$k_g = (0.7 \log \phi + 0.4)10^{0.01\rho_g}. \qquad (4.23)$$

where ϕ is percent moisture of total weight and ρ_g is the dry density. Bose (2) has presented these equations in graphical form.

A detailed description of the determination of thermal properties of soils is given by Salmone, Kovacs and Wechsler (33). The difficulty in determining these properties arises primarily in sampling. Best estimate of the soil type at the test site is a granular cohesive soil with a dry density of 105 lb/ft^3. The soil can be considered saturated since 90 to 95% of the ground-couplings are below the water table. Applying Equation 4.22 and 4.23 the range for k_g is between 1.04 to 1.4 Btu/hr-ft-F. A graphical plot appearing in Salmone (33) suggests a thermal resistivity of between 40 and 50 W/cm-^{o}C for a saturated soil at this weight. This converts to a thermal conductivity between 1.15 to 1.44 Btu/hr-ft-F. Measurements at the test site indicate slightly higher values (2). Several values within the range of 1.0 to 1.4 will be implemented into the program for verification. The problem of moisture migration is not significant due to the high water table at the site. A more detailed method of determining thermal properties of soils is contained in (4).

4.3.4 Ground Water Movement

Significant groundwater movement can transfer heat to and from the ground-coupling at a much faster rate than possible with pure conduction. This would generally assist the performance of the system but could cause considerable underpredictions of pure conduction models. Although there appears to be no significant movement at the test site, this possibility must be considered.

Typical groundwater flow velocities range up to 5 ft/year in clays and 5 ft/day in course sands and gravels. These velocities can be determined from the equation (34)

$$V = \frac{K_s}{7.48} \frac{dh}{dL} \, . \tag{4.24}$$

Velocity in the equation is in ft/day, $\frac{dh}{dL}$ is the dimensionless slope of the water table, and K_s is defined as the laboratory coefficient of permeability. Typical values are given in Table VI (36).

The materials described in Table VI usually occur in layers through which the ground-coupling is placed. Therefore water velocities around coupling vary with height. A coupling could be placed in 90 feet of impervious clay, with a 10 foot layer of fine sand. The simulation must then be corrected for the improved heat transfer at this 10 foot section, while the remaining 90 feet is considered to be pure conduction.

The test site is located near the top of a hill. The maximum height that could be obtained is approximately 30 feet to the top which is at a distance of 300 feet. The soil has been described as a silty or sandy clay. This would indicate a permeability of 10^{-3} gal/day-ft^2, therefore application of Equation 4.24 yields

$$V = \frac{10^{-3}}{7.48}\,\frac{30}{300}\,365 = 0.05 \text{ ft/yr} .$$

TABLE VI

SOIL PERMEABILITIES (34)

Material	Permeability (Gal/day-ft^2)
Clay	10^{-5} - 10^{-3}
Sandy Clay	10^{-4} - 10^{-2}
Sandy clay loam	10^{-2} - 1
Very fine sand	1 - 10^{2}
Medium sand	10^{2} - 10^{3}
Course sand	10^{2} - 10^{4}
Gravel	10^{2} - 10^{4}

Although there may be more permeable layers at the test site, drillers have not found any significant strata other than clay and soft rock above 100 feet. The results of the test shown in Table IV indicate that water movement near the coupling is small. The values shown agree well with the values resulting from simulations that assume pure conduction.

4.4 Development of Finite Difference Equations

As outlined in Section 4.1 the initial set of FDEs to describe the ground-coupling is an explicit formulation for a single vertical node. Figure 30 can be used to describe the development of the equations. However, the value for Se is 1.0 in this development (uniform grid size). An energy balance on node 1 is performed as follows.

$$q_{w-1} = hr_i \Delta\theta\Delta z(T_w - T_1) \tag{4.25}$$

$$q_{1-2} = k_p(r_i + \frac{\Delta r_p}{2}) \Delta\theta\Delta z \frac{T_2 - T_1}{\Delta r_p} \tag{4.26}$$

$$q_v = \rho_p c_{pp}(r_i + \frac{\Delta r_p}{4})\Delta\theta\Delta z \frac{\Delta r_p}{2} (\frac{T_1' - T_1}{\Delta t}) \tag{4.27}$$

When Equations 4.25, 4.26 and 4.27 are combined the FDE for node 1 is

t + Δt

$$T_1' = [\frac{h}{k_p \Delta r_p} (T_w - T_1) + \frac{T_2 - T_1}{\Delta r_p^2} + \frac{T_2 - T_1}{r_i \Delta r_p}] \frac{k_p \Delta t}{\rho_p c_{pp} A} + T_1 \tag{4.28}$$

where $A = (r_i + \frac{\Delta r_p}{4}) \frac{\Delta r_p}{2}$.

If similar energy balances are performed on node 2 the resulting FDE is:

$$T_2' = [k_p(r_i + \frac{\Delta r_p}{2}) \frac{T_1 - T_2}{\Delta r_p} + k_g (r_o + \frac{\Delta r_g}{2})(\frac{T_3 - T_2}{\Delta rg})] \frac{\Delta t}{B} + T_N \tag{4.29}$$

where $B = [\rho_p c_{pp}(r_i + \frac{3\Delta r_p}{4}) \frac{\Delta r_p}{2} + \rho_g c_{pg}(r_o + \frac{\Delta r_g}{4}) \frac{\Delta r_g}{2}]$

The FDEs for nodes 3 to the far field are:

$$T_N' = \left[\frac{T_{N-1} - 2T_N + T_{N+1}}{\Delta r_g^2} + \frac{T_{N+1} - T_{N-1}}{2r_N \Delta r} \right] \alpha \Delta t + T_N \tag{4.30}$$

These FDE formulations are the basis to the computer program CX1. This program is for a single vertical node. The water temperature is held constant and no energy balance is performed on the water.

The next simplification involves replacing Equations 4.28 and (4.29) to reduce computation time and allow a larger time step in the explicit formulation. This is accomplished by the use of Equation 3.1 which neglects the thermal storage capacity of the pipe wall. An intermediate program was developed with a uniform grid size. This program required numerous applications of Equation 4.30 in order to calculate node temperatures to the experimentally determined far field radius, where ground temperatures are not effected by the coupling.

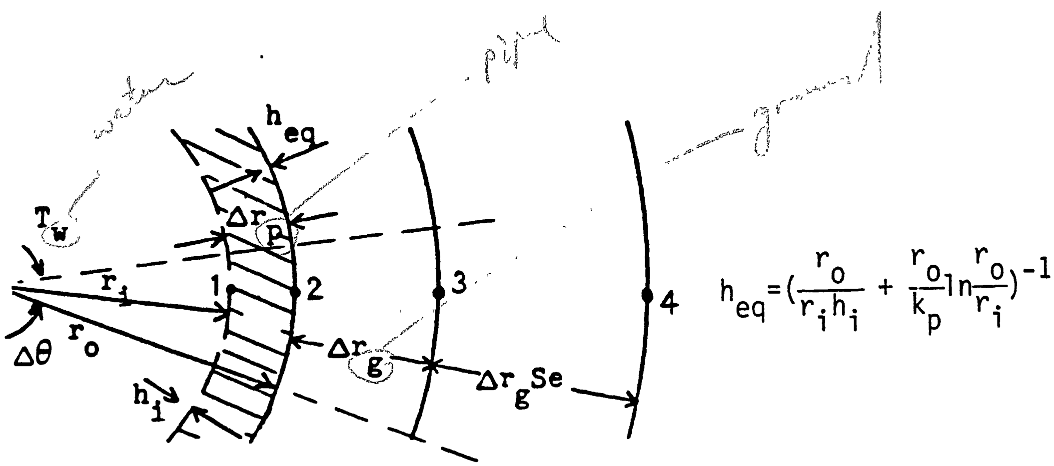
A reduction in computations can be made if grid size is allowed to expand with each successive calculation of Equation 4.30. Since there are no abrupt changes in grid size, accuracy can be maintained. The arrangement of the resulting grid is also shown in Figure 30. Notice that node 1 is no longer necessary.

An energy balance on node 2 of Figure 30 is as follows.

$$q_{w-2} = h_{eq}\, r_o \Delta\theta \Delta z (T_w - T_2) \tag{4.31}$$

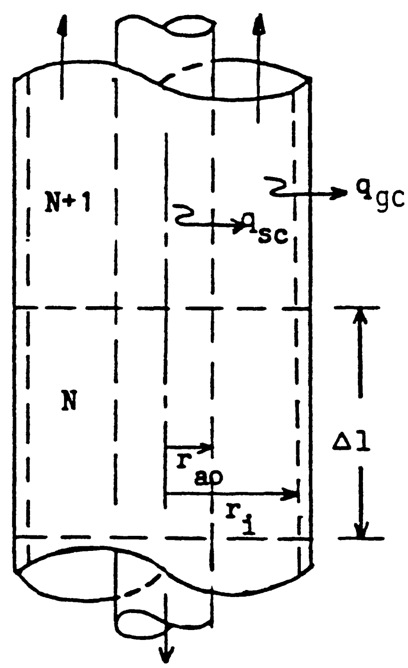
$$q_{2-3} = k_g \left(r_o + \frac{\Delta r_g}{2}\right) \Delta\theta \Delta z \frac{T_3 - T_2}{\Delta r_g} \tag{4.32}$$

$$q_v = \rho_g c_{pg} \left(r_o + \frac{\Delta r_g}{4}\right) \Delta\theta \Delta z \frac{\Delta r_g}{2} (T_2' - T_2) \tag{4.33}$$



RADIAL DIRECTION

Figure 30. Radial Grids Used in Finite Difference Equation Development



VERTICAL DIRECTION

Figure 31. Vertical Grids of Concentric Coupling

The resulting FDE for node 2 is:

$$T_2' = \frac{r_o h_{eq}}{X1}(T_w - T_2) + \frac{X1}{X2}(T_3 - T_2) + T_2 \qquad (4.34)$$

where $X1 = k_g(r_o + \frac{\Delta r_g}{2})/\Delta r_g$

$$X2 = \rho_g c_{pg} \Delta r_g (r_o + \frac{\Delta r_g}{4})/2\Delta t$$

Utilizing methods described in (25) the FDE describing the equation for the ground nodes can be replaced by:

$$\frac{2}{\Delta r_g^2}\left[\frac{T_{N+1}}{Se(Se+1)} - \frac{T_N}{Se} + \frac{T_{N-1}}{Se+1}\right] + \frac{1}{\Delta r_g r_N}\left[\frac{T_{N+1}}{Se(Se+1)} - \frac{(1-Se)T_N}{Se} - \frac{SeT_{N-1}}{Se+1}\right] = \frac{T_N' - T_N}{\alpha_g \Delta t} \qquad (4.35)$$

This leads to the equation:

$$T_N' = \frac{\alpha_g \Delta t}{Se(Se+1)\Delta r_g}\left[\left(\frac{2}{\Delta r_g} + \frac{1}{r_N}\right) T_{N+1} - \left(\frac{2(Se+1)}{\Delta r_g} + \frac{1 - Se^2}{r_N}\right)T_N + \left(\frac{2Se}{\Delta r_g} - \frac{Se^2}{r_N}\right)T_{N-1}\right] + T_N \qquad (4.36)$$

The program utilizing the equivalent heat transfer coefficient with a uniform grid is CX2. The program utilizing the equivalent heat transfer coefficient and the FDEs (Equations 4.34 and 4.36) for the non-uniform grid is CONEX. This program allows variable value for Se.

The simplifications employed to this point permitted time steps in excess of one minute when the values for a 3 inch schedule 40 PE pipe are input. The expansion factor was increased to a value of 1.5. With this ratio and a 0.25 inch grid width nearest the pipe, a 16 foot cylinder of ground can be covered with 15 radial nodes. Computation time was reduced enough to permit the simulation of the ground-couplings using energy balances to determine temperature variations in the vertical water nodes.

Figure 31 describes the energy balance procedure utilized. Water enters the first vertical node at the bottom (M=1) through the dip tube. The set of FDEs used in CONEX are applied to this node (radially). In the cooling mode, heat is transferred to the ground according to the equation

$$q = h_{eq} \, 2\pi r_o \Delta L (T_2 - T_w) \; . \qquad (4.37)$$

The heat is transferred through the annular tube by the equation

$$q_{sc} = \left(\frac{r_{ao}}{k_a} \ell n \frac{r_{ao}}{r_{ai}} + \frac{1}{h_{ao}} \right)^{-1} 2\pi r_{ao} \Delta L (T_a - T_w) \; . \qquad (4.38)$$

Note that the heat transfer coefficient inside the annular tube is neglected because of its large value.

The water temperature in the next node is calculated by adding Equations 4.37 and 4.38 to obtain the net heat loss and applying the equation

$$T'_{M+1} = \frac{q_{NET}}{\rho_w c_{pw} Q} + T_M \; . \qquad (4.39)$$

This procedure is repeated for all vertical nodes. When there is no flow the update equation becomes

$$T_N' = \frac{q_{NET}\Delta t}{\rho_w c_{pw} r_i^2 \Delta L} + T_N \quad . \tag{4.40}$$

Equation 4.37 is used to calculate q_{NET} in this case and the value of h_{eq} must be determined using the natural convection value of h_i. In the heating mode the formulation is identical but q_{NET} is a heat gain.

In order for energy balances performed in this manner to be valid, a "lump" of water must move exactly one vertical node during each time step. Therefore time steps for a particular concentric ground-coupling are dictated by the choice of average water velocity and node length according to the equation

$$\Delta t = \frac{\pi \Delta L (r_i^2 - r_{ao}^2)}{Q} = \frac{\Delta L}{V} \quad . \tag{4.41}$$

After the energy balance has been performed on all of the vertical nodes the water temperature is used as an input value to the heat pump performance. Heat transfer to the water by the heat pump is adjusted according to the methods of Section 4.3.1. The water entering the ground-coupling for the next time step is calculated by the equation

$$T_{w,1}' = T_{w,M} + \frac{q_{hp}}{\rho_w c_{pw} Q} \quad . \tag{4.42}$$

The value of q_{hp} is of course not calculated when flow is stopped (Q=0).

These formulations are combined into the most basic simulation of ground-couplings used in this project. Additional details are included in the Appendix in the computer program CVHE.

The program CVHE has time step limitations imposed by stability characteristics of explicit formulations. Computation time can not be decreased by reducing the number of vertical nodes, because Equation 4.41 indicates that if ΔL is increased for given velocity, the value of Δt must also increase. It is necessary to convert to an implicit formulation in order to increase the time step or decrease the number of vertical nodes.

The formulation of FDEs using forward time step is similar to the explicit formulation. Equation 4.34 is now

$$T_2 = - \frac{r_o h_{eq}}{X1} (T'_w - T'_2) - \frac{X1}{X2} (T'_3 - T'_2) + T'_2 \ . \tag{4.43}$$

Equation 4.36 is

$$T_N = \frac{\alpha_g \Delta t}{Se(Se+1)\Delta r_g} [-(\frac{2}{\Delta r_g} + \frac{1}{r_N})T'_{N+1} + (\frac{2(Se+1)}{\Delta r_g} + \frac{1 - Se^2}{r_N})T'_N$$
$$(\frac{2Se}{\Delta r_g} - \frac{Se^2}{r_N})T'_{N-1}] + T'_N \tag{4.44}$$

These equations remove time step limitations due to stability. However, considerable rounding error may result because Equation 4.43 must be solved simultaneously with Equation 4.44 being applied to every ground node that needs to be updated.

A TDMA is used to solve this equation for T'_w. The resulting matrix is diagonally arranged with the T'_N term being on the diagonal, the T'_{N-1} term before the diagonal, the T'_{N+1} term after the diagonal and the constant coefficient is T_N, except on the first and last rows. The constant coefficient on the first row is T_2 less the T'_w term. The constant coefficient for the last row is T_N less the T'_{N+1} term, which is constant. A computer algorithm to solve this matrix is provided by (25). A sample matrix is shown in the Appendix.

The computer program CVHI is identical to CVHE with the exception of the implicit formulation and the TDMA. With CVHI it is possible to reduce the number of vertical nodes and correspondingly increase the time step. The time step is still dictated by Equation 4.41. This proves to be inconvenient since the time step is the dependent variable and is therefore usually an odd value.

To overcome this problem and to further reduce computation time, another simplification is made by reducing the number of vertical nodes to one. This will result in some inaccuracy since the temperature profile of the earth is non-uniform near the surface. The heat transfer rate in concentric coupling also varies with depth. It is possible to adjust the program to compensate for these non-uniformities.

An additional compensation must be made to the water temperature entering the heat pump immediately after start-up. Water temperatures are significantly reduced during off periods due to natural convection in the cooling mode. CHVE and CHVI store these reduced values in the vertical nodes and the account for additional heat transfer as these "lumps" of water proceed through the coupling. A single node program utilizes an average temperature, and a gradual change in water temperature is not possible unless provisions are made.

The program CHI is a single vertical node implicit formulation. However, a vertical water temperature profile is calculated and maintained after each start up so that water temperatures entering the heat pump gradually increase. Once the water has made one complete loop through the coupling, a single average water temperature is used in the FDE formulation. A slight adjustment can be made so that the far field temperature reflects the average value of the entire depth not just the average far field value below 20 feet.

The final program developed is the conversion of CHI to account for U-tube ground-couplings. The transient three-dimensional heat flow of vertical U-tubes is complex. Numerical solutions will not be attempted here because of this complexity and in the interest of computer time. An alternative is to derive values equivalent to those encountered in concentric couplings. These values include an equivalent diameter, heat transfer coefficient, thermal mass and short circuit heat transfer. The approach will be to use basic steady state heat transfer principles between the pipes and experimental results. Figure 32 shows the physical arrangement of the development.

Bose (2) suggest that the equivalent diameter is

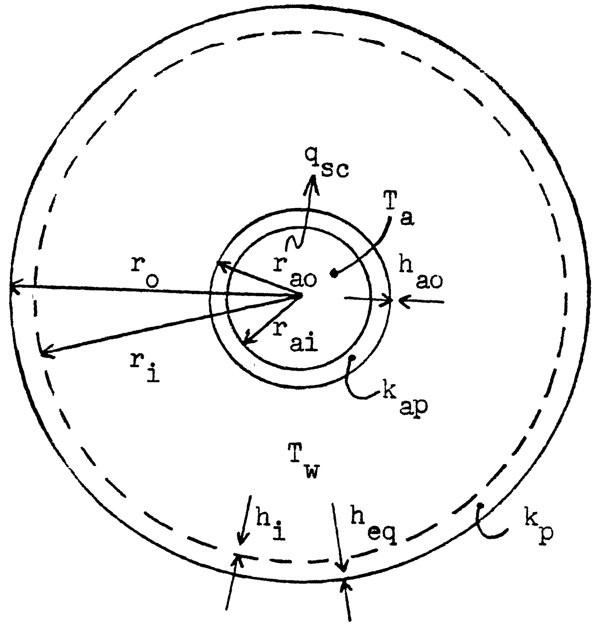
$$D_{eq} = \sqrt{Nt}\ D_o\ . \tag{4.45}$$

He also suggests an equivalent resistance per unit length can be calculated by

$$R_{eq} = \frac{1}{2\pi k_p(Nt)}\ \ln\frac{r_o}{r_i} = \frac{q_{gc}}{\Delta T_p}\ . \tag{4.46}$$

In terms of equivalent heat transfer coefficient this becomes:

$$h_{eq} = Nt\ \left(\frac{r_o}{k_p}\ \ln\frac{r_o}{r_i} + \frac{r_o}{r_i h_i}\right)^{-1} \tag{4.47}$$



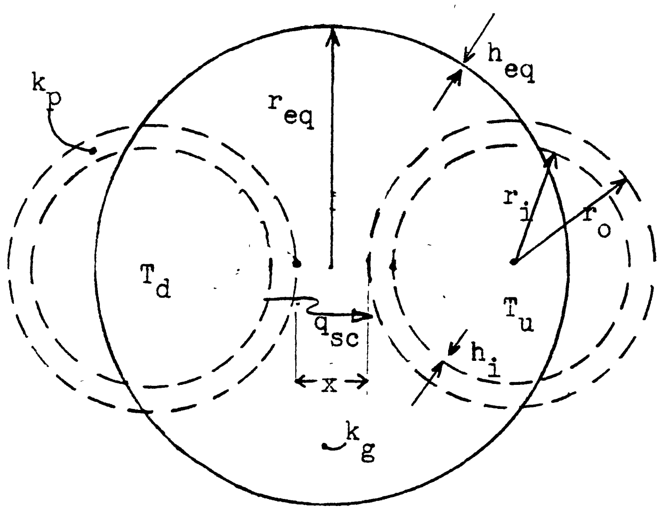
$$r_{eq} = r_o$$

$$h_{eq} = \frac{1}{\frac{r_o}{r_i h_i} + \frac{r_o}{k_p} \ln \frac{r_o}{r_i}}$$

$$q_{sc} = \frac{2\pi r_{ao} l \, (T_w - T_a)}{\frac{1}{h_o} + \frac{r_{ao}}{k} \ln \frac{r_{ao}}{r_{ai}}}$$

NOTE: $\frac{r_{ao}}{r_{ai} h_{ai}}$ is neglible

CONCENTRIC ARRANGEMENT



$$r_{eq} = f(r,x)$$

$$h_{eq} = f(C_f r_o (\frac{1}{r_i h_i} + \frac{\ln r_o/r_i}{k_p}))$$

$$q_{sc} = f(x, k_g, T_{wd} - T_{wu})$$

U-TUBE ARRANGEMENT

Figure 32. Equivalent Dimensions of U-tube Ground-Couplings

Short circuit heat transfer is neglected and thermal mass is unnecessary when using the line source equation.

A small adjustment is made to Equation 4.45 to account for U-tubes separated by a distance x.

$$D_{eq} = \sqrt{Nt}\, D_o + x \tag{4.48}$$

Equation 4.47 overpredicts the heat transfer coefficient. This equation assumes uniform parallel heat flow from the pipes. However, the areas of the pipes facing each other are largely ineffective conductors of heat. Experimental results and simulations indicate that a conversion factor should be applied to Equation 4.47.

$$h_{eq} = C_{eq} Nt \left(\frac{r_o}{k_p} \ln \frac{r_o}{r_i} + \frac{r_o}{r_i h_i} \right)^{-1} \tag{4.49}$$

where $C_{eq} = 0.85$ when $Nt = 2$ (Single U-tube)

$0.6 \leq C_{eq} \leq 0.7$ when $Nt = 4$ (Double U-tube)

For this case $A = 2\pi r_o$.

Thermal short circuiting in plastic U-tubes is not as great as in concentric couplings without insulated dip tubes. It is much more difficult to calculate because of three dimensional heat flow pattern. Hopkins (27) suggests the use of an average thickness of soil between the tube. She assumes short circuiting occurs only between the inside tube quandrants facing each other. She also neglects the thermal resistance of the tube wall, which leads to an overprediction of short circuiting.

An alternative is to calculate the resistance of the five terms between the up and down water streams. These include two film resistances, two pipe wall resistances and the soil resistance. The film and wall resistances will be calculated for only three-eighths of the tube walls facing each other in order to agree with experimental results.

$$R_F = 4/3\pi r_i h_i L \tag{4.50}$$

$$R_{pw} = \frac{4 \;\ell n\, r_o/r_i}{3 \;\pi k_p L} \tag{4.51}$$

The heat transfer between cylinders of equal size buried in an infinite medium is (37)

$$q = \frac{\pi k_g L \Delta T}{\cosh^{-1}[\frac{x+2r_o}{2r_o}]} \; . \tag{4.52}$$

The soil resistance is therefore

$$R_s = \frac{\cosh^{-1}[\frac{(x+2r_o)}{2r_o}]}{\pi k_g L} \; . \tag{4.53}$$

This equation agrees with the values of resistance calculated by the methods of Hopkins (27). The total thermal resistance to short circuiting can be estimated by

$$R_{sc} = 2R_F + 2R_{pw} + R_s \;. \tag{4.54}$$

For concentric couplings this value is

$$R_{sc} = \left[2\pi r_{ao} L \left(\frac{1}{h_{ao}} + \frac{r_{ao} \ln \frac{r_{ao}}{r_{ai}}}{k_{ap}} + \frac{r_{ao}}{r_{ai}\, h_{ai}}\right)^{-1}\right]^{-1} \tag{4.55}$$

Another value that is necessary to calculate is an equivalent thermal mass. This value must be input in order to simulate water temperature recovery during off periods. If the thermal mass for small U-tubes (3/4 inch) is assumed to be the thermal mass of the water, temperatures recover too rapidly since there is very little water in these couplings. Therefore, necessary accuracy is achieved if the thermal mass is assumed to be the thermal mass of water in a circular tube of diameter D_{eq}.

Equations 4.48, 4.49 and 4.54 are applied to the vertical node implicit formulation used in CHI to form the simulation UTI for U-tube ground-couplings. The thermal mass assumption described in the previous paragraph is also applied. The formulations can be used for single or double U-tubes.

4.5 Results of Finite Difference Equations Simulation

The results of primary concern are the simulation of the entire ground-coupling system. Care has been taken in the development of the set of FDEs describing the radial temperature distribution for a single vertical node. CX1, CX2 and CONEX all simulate a single vertical node with a constant water temperature. Values are in close agreement for

CX1, CX2 and CONEX (with values of Se up to 1.5). They all give an average heat transfer rate of 1.35 $\pm$ 0.05 Btu/min-ft for a 3 inch SDR 21 pipe during the first hour of operation for a water temperature of 85^{o}F in soil with a thermal conductivity of 1.2 Btu/hr-ft-F. This rate increases to 1.78 with 95^{o}F water. These values are good for comparison purposes, but are somewhat meaningless unless a complete simulation is developed.

The initial simulation is CVHE as described in the previous section. It uses CONEX with Se = 1.5 as the basic radial FDE equations. The heat pump used in the experiment was normally operated at 10.2 GPM or 1.70 PM per coupling. If Equation 4.41 is applied, the largest increment that could be used is five feet because of the time step limitation. The time step used is 1.07045 minutes. For each time step Equation 4.34 is applied once and Equation 4.36 is applied 14 times for each vertical node. The result is that for a simulation of one day, these two equations must be recalculated over 400,000 times.

The program CVHI was developed into order to increase the time step and therefore reduce the number of vertical nodes. When the time step is doubled the vertical number of nodes is halved and the number of computations is reduced by a factor of four. The limitation on time step is simulation accuracy. Figure 33 is a plot of average coupling water temperature during at two-week simulation of the 3 inch concentric coupling for two weeks of 50% run fraction. Shown on the figure are the results using CVHE, CVHI with 10 vertical nodes and a constant 62^{o}F far field temperature and CVHI with 10 nodes and 70^{o}F and 64^{o}F far field temperatures for the upper two vertical nodes. Figure 34 is a plot of the hourly variation at the end of the simulation.

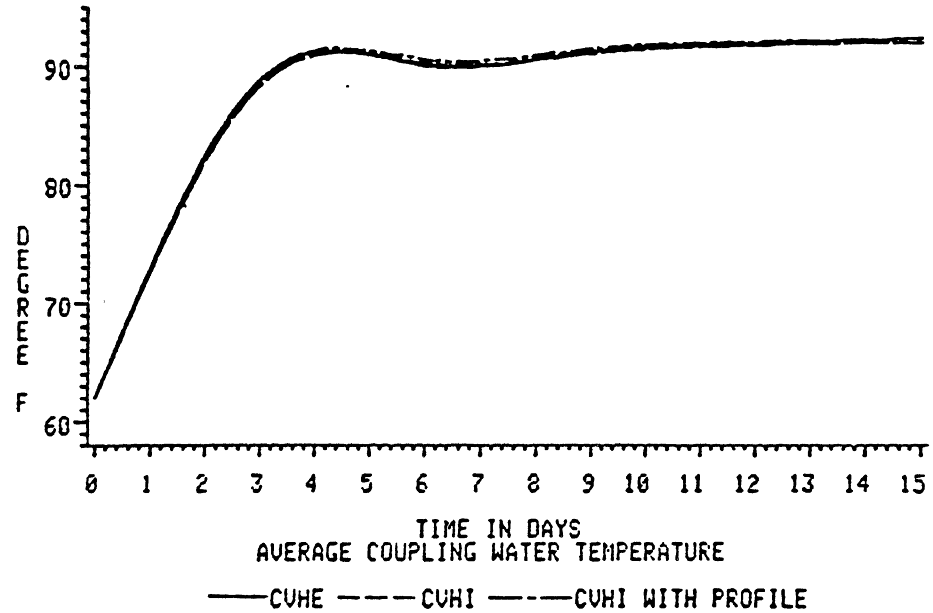


Figure 33. Two Week Comparison of Explicit and Implicit Formulation

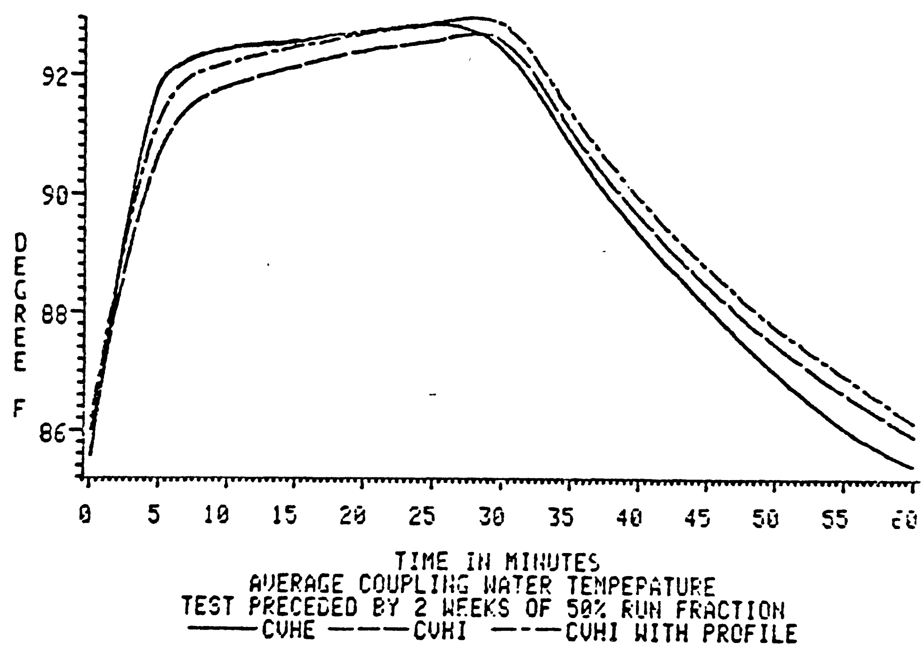


Figure 34. One Hour Comparison of Explicit and Implicit Formulation

The results show that accuracy does not suffer when the time step is increased to a value as large as 5 minutes (4 vertical nodes). Increments larger than this are not necessary because they exceed the time of on-off cycles in periods of very light or very heavy loads. These results also indicate that accuracy in a 100 foot ground-coupling is not appreciably effected by assuming a constant far field temperature instead of one that is dependent on distance below the ground surface.

These results indicate the possibility of simulation with a single vertical node and a five minute time step using average water and far field temperatures. The simulation CHI is for concentric couplings and UTI is for U-tube designs. At this point it is possible to compare simulation result with the experimental results. Comparison for the best couplings (double 3/4 inch and 1-1/2 inch PE U-tube) and the worst (2 inch concentric and 3/4 inch U-tube) are excluded. This is necessary since their performance is either reduced or increased because of the common water supply. Therefore the simulation is performed on the 1-1/2 inch PB U-tube and the 3 inch concentric since their performance is more indicative of actual performance if all six couplings were of their identical design.

Figure 35 is the average daily coupling temperature for the simulation and the experiment for the 1-1/2 PB U-tube during the 13 week operation. Figure 36 is the same plot for the 3 inch concentric coupling. Results show good agreement except for the 62% run fraction period. At this time a low voltage problem occurred at the test site and heat pump performance could not be accurately predicted. Figure 37 is a comparison of the daily simulated and experimental variation of temperature.

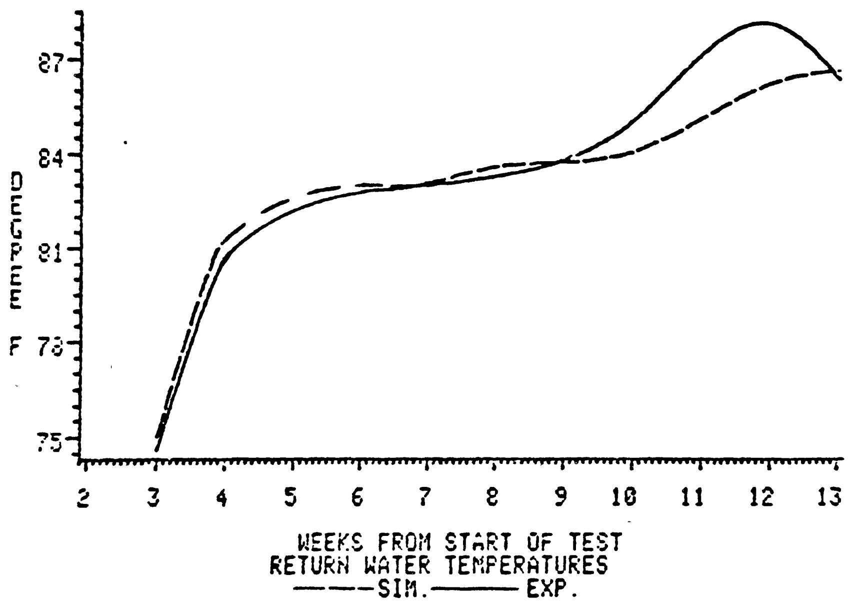


Figure 35. Long Term Comparison of Experimental and Simulated Water Temperature of U-tube

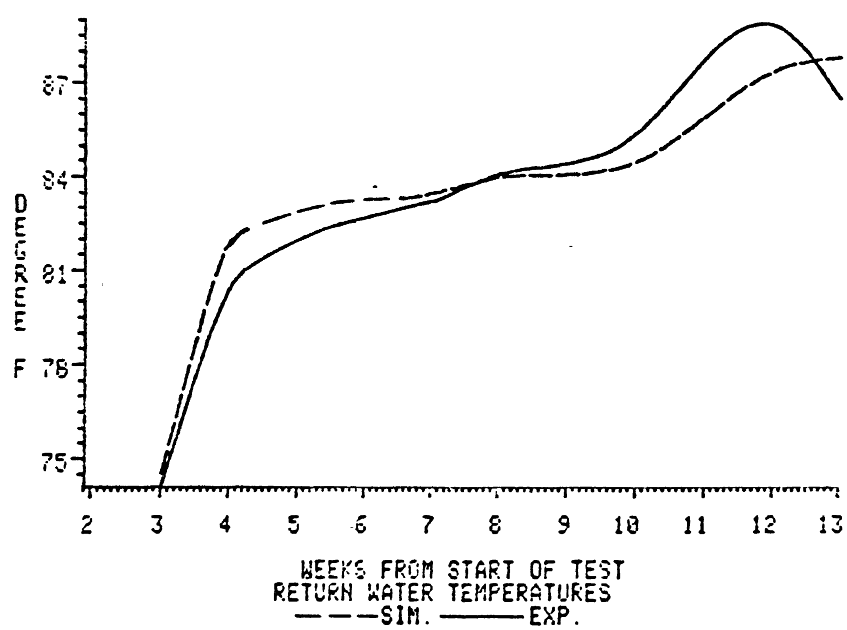


Figure 36. Long Term Comparison of Experimental and Simulated Water Temperatures of Concentric Tube

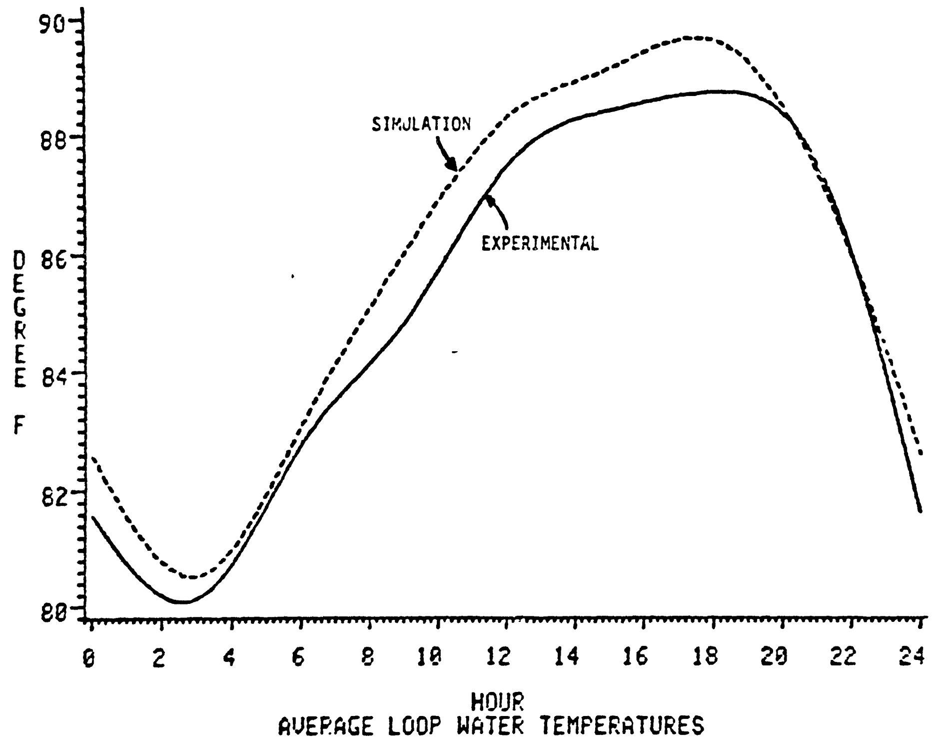


Figure 37. Daily Comparison of Experimental and Simulated Water Temperatures of U-tube

Simulations were performed as if independent systems were installed using six loops of identical design to each of these tested in the experiment. Figure 38 is the results of these six simulations. Figure 39 is included to show the effect of decreases in thermal conductivity or water flow rate.

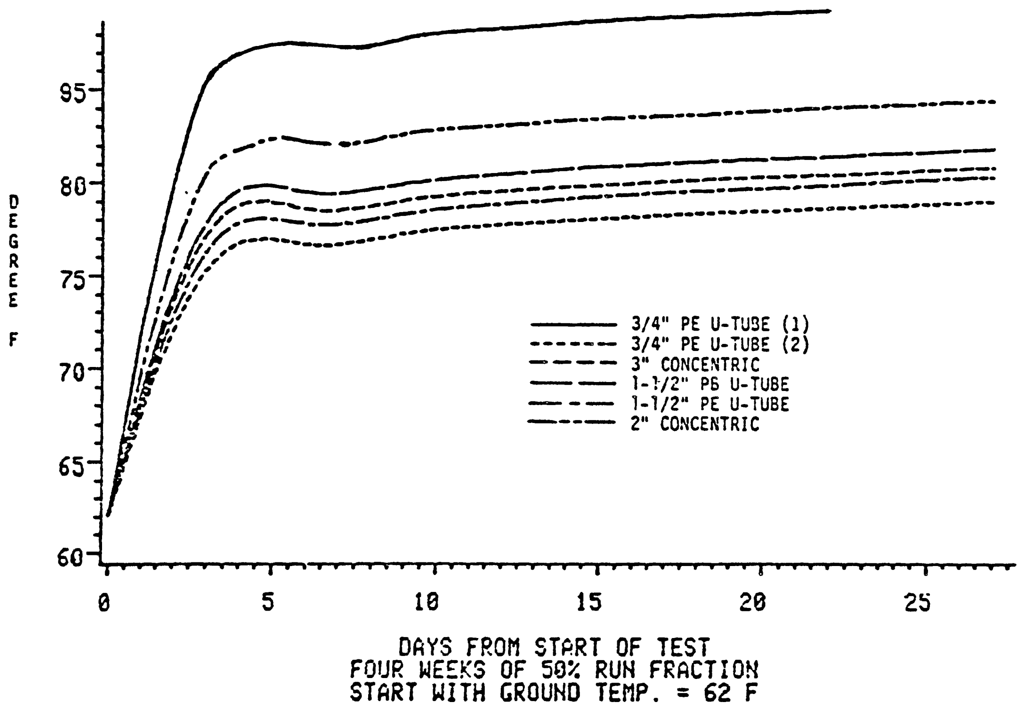


Figure 38. Simulated Comparison of Ground Coupling Designs

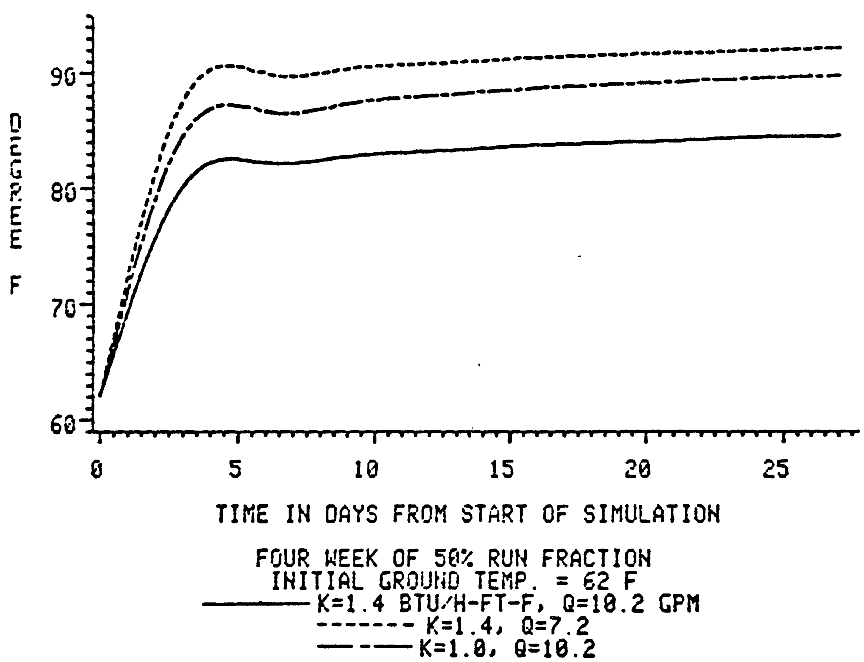


Figure 39. Simulated Comparison of Water Flow and Ground Conductivity

CHAPTER V

VERTICAL GROUND-COUPLING SIMULATION WITH LINE SOURCE EQUATION

5.1 Overview of Numerical and Analytical Methods

Simulation of ground-coupled heat pump systems using numerical methods, such as finite difference or finite element solutions, offer a high degree of accuracy and flexibility when properly implemented. A wide variety of physical variables and operating conditions can be accounted for using these powerful techniques. The accuracy and flexibility of these methods necessitate that variables also be input with accuracy. Many of these variables can be determined. However, many can not be properly calculated or controlled. For example, the performance of the double 3/4 inch U-tube used in this experiment is dependent not only on tube separation distance, but also on location of tubes in relation to each other. One of the down flowing tubes could be surrounded by the two up flowing tubes. This would reduce the capacity of the up flowing tube and increase short circuiting. Unless elaborate measures are taken during installation the arrangement can not be determined.

There are a great many other variables that are difficult to find including forced and free convection heat transfer coefficients, thermal properties of the soil and local groundwater movement. If the numerical method is to be used as part of a simulation it must be linked to a simulation of the building thermal load to determine heat pump operation patterns. This can add to a greater error since the accuracy of cooling/heating simulations are also dependent on input variables that are difficult to predict. These variables include among other things the weather, thermal integrity of the structure and internal loads.

These factors may lead to an overconfidence in the results of a numerical simulation based on its power, not its accuracy, which is highly dependent on the quality of input assumptions. A primary example may be the prediction that air source heat pumps have a greater annual power savings than vertical ground-coupled heat pumps in Houston, Texas (37). This is based on the assumption of a relatively poor soil thermal conductivity. A more realistic result of the annual power consumption will be arrived at if the locations significant groundwater movement is considered in the ground-coupling simulation.

It is apparent that most firms installing vertical ground-coupled heat pumps do not have sufficient facilities and resources to properly test local conditions and apply numerical method simulations. Current procedure is to install a system based on "rules of thumb" and make adjustments to subsequent installations based on the performance of the initial one.

An intermediate design procedure between "rules of thumb" and numerical methods is to utilize the Kelvin Line Source Theory as applied to heat pumps by L. R. Ingersoll, Zobel and A. C. Ingersoll (39). This method has been developed by Kalman (7) and also by Bose (3) to account for current installation methods. Although this method does not have the flexibility and accuracy possible with numerical methods, it requires much less computation time and adjustments can be made to account for many complications so that accuracy approaches that of numerical methods.

5.2 Literature Review of Analytical Methods

Ingersoll, Zobel and Ingersoll (39) applied the line source equation to the solution of temperatures near pipes buried in the ground. When the equation presented in their text is converted into symbols consistent with this work, we have

$$\Delta T_g = T_{ff} - T_r = \frac{q_{gc}/L}{2\pi k_g} \int_{r/2\sqrt{\alpha_g t}}^{\infty} \frac{e^{-\beta^2}}{\beta}\, d\beta = \frac{q_{gc}/L}{2\pi k_g}\; I(X) \qquad (5.1)$$

where q_{gc} is the rate of heat rejected or absorbed by the ground-coupling, β is in this case a variable of integration and $X = r/2\sqrt{\alpha_g t}$ The results of the integration for I(X) are given for a range of X characteristic of many ground-couplings in Figure 40. Additional values can be obtained from a table utilized by Ingersoll, Zobel and Ingersoll.

The authors apply the equation primarily to evaluate the temperature at the outer pipe wall. This is accomplished by setting r in the term for X equal to the outside pipe radius. Temperatures at any point in the ground can be evaluated with Equation 5.1. Error does result for large pipe and for small values of time. The authors suggest that noticeable error occurs using line source equation when

$$\frac{\alpha_g t}{r^2} < 20 \qquad (5.2)$$

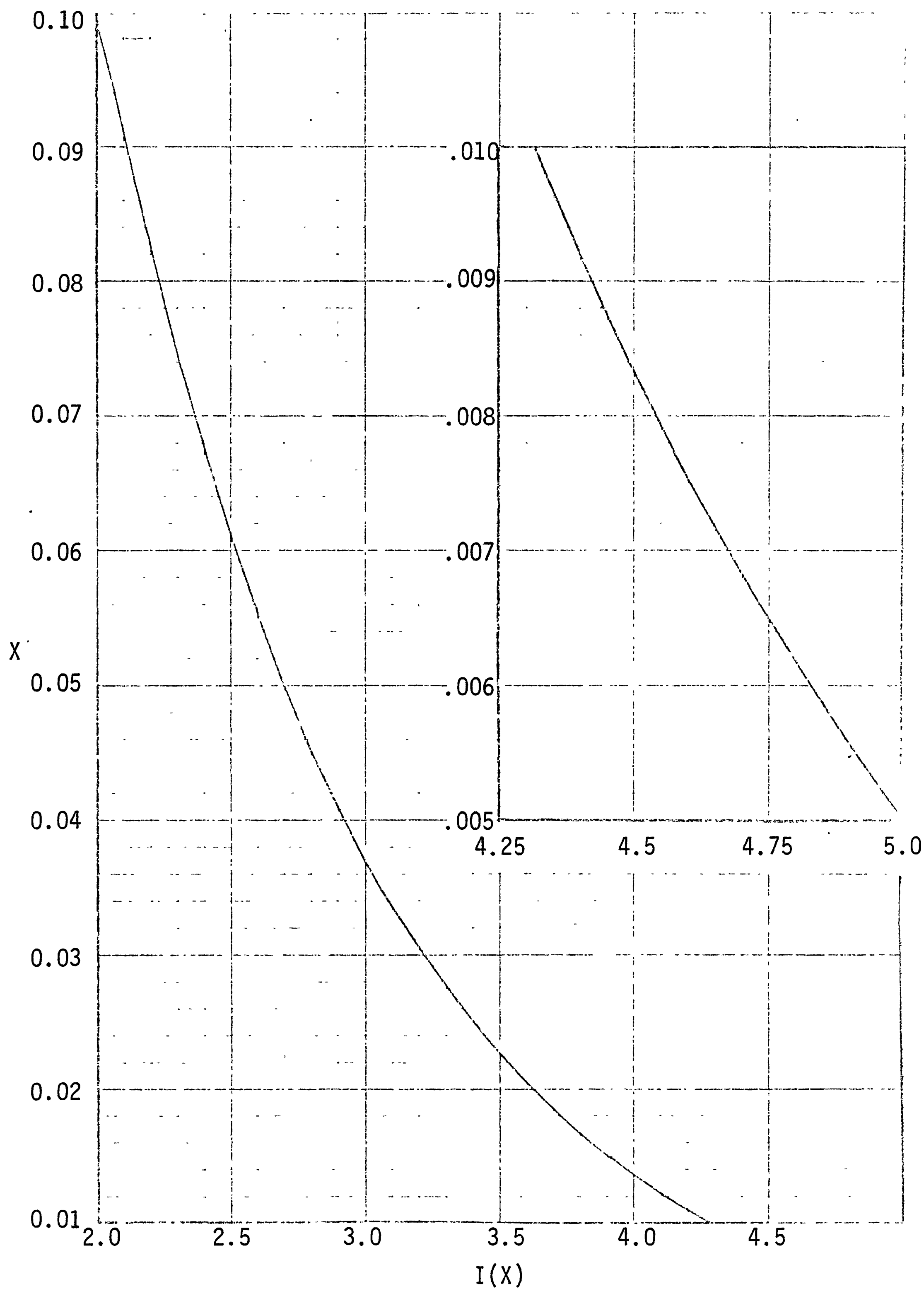


Figure 40. Graphical Results of Line Source Intergral

For smaller values the work of Jaeger (40) is suggested as a better method. Jaeger proposes the use of the equation for cylindrical sources

$$\Delta T_g = \frac{q_{gc}}{k_g} G(z,p) \tag{5.3}$$

where

$$G(z,p) = \frac{1}{\pi^2} \int_0^\infty \frac{e^{-\beta^2 z} - 1}{J_1^2(\beta) + Y_1^2(\beta)} [J_0(p\beta)Y_1(\beta) - J_1(\beta)\, Y_0(p\beta)] \frac{d\beta}{\beta^2} ,$$

$$z = \frac{\alpha_g t}{r^2} \qquad \text{and} \qquad p = \frac{r}{r_o} \text{ (z is also } F_o) .$$

The values of $G(z,p)$ of most interest are when $p = 1$ (outside pipe wall). Values of $G(z,1)$ can be obtained from Figure 41. Additional values can be found in References (39) and (40).

Equations 5.1 and 5.3 are derived for constant heat transfer rates (q_{sc}). However, an average value for ΔT_g at any location can be determined by inputting an average value of heat transfer over a limited time period. For soils this time period is well within the hourly and daily fluctuations of heat pumps. Methods will be described later to calculate values other than average ones.

The authors of (39) provide many methods of applying Equation 5.1 to ground-coupled heat pumps, including the complexities involved when horizontal or multiple pipe couplings are used. Isolated vertical pipes generally have fewer complexities but several methods of application of the line source equation are necessary for it to be a useful design tool.

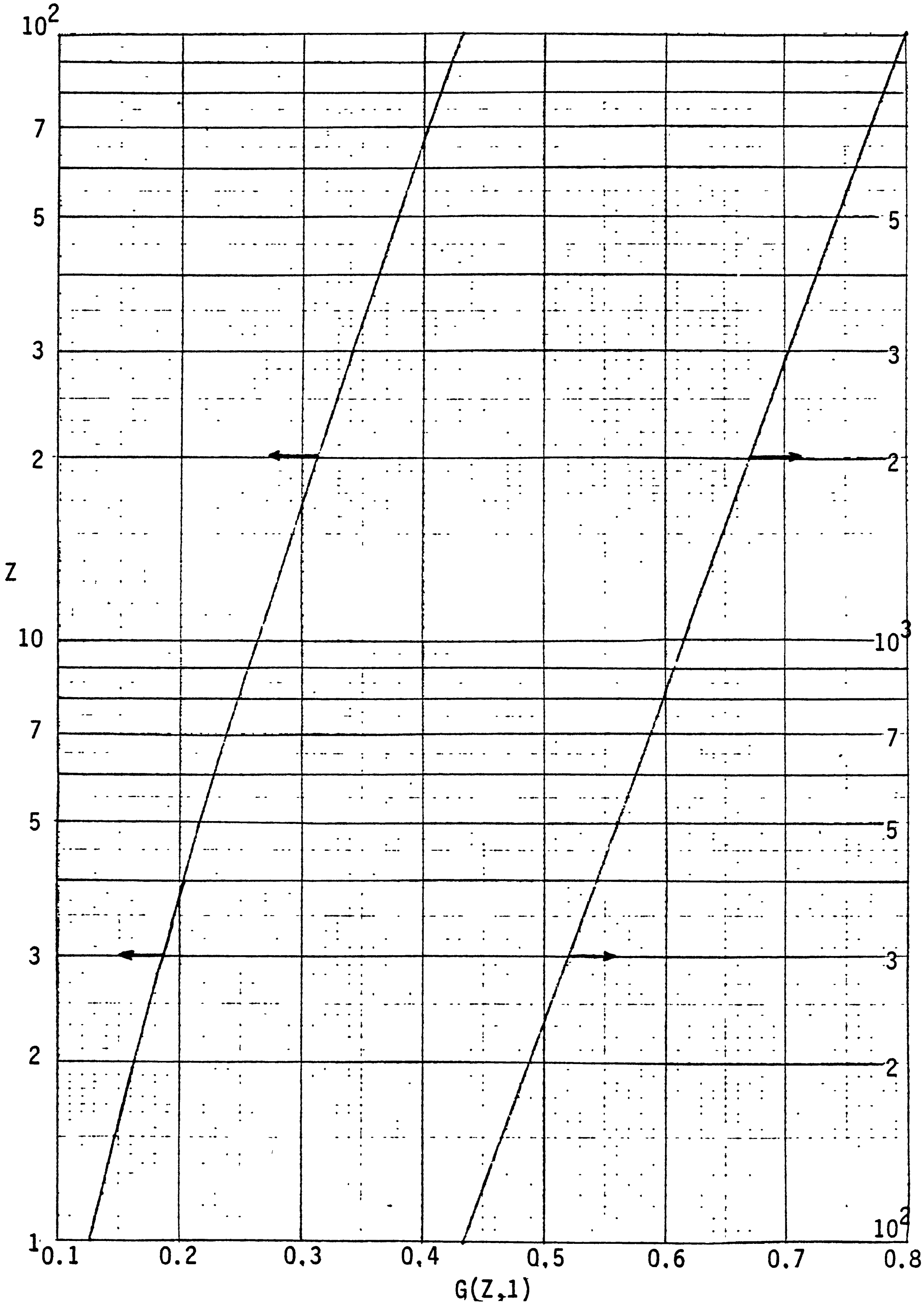


Figure 41. Graphical Results of Cylindrical Source Equation

The primary adjustment is for the variable heat absorption rate characteristic of heat pump operation. The variation may be in terms of month, day or hours in actual cases. The authors apply the equation to monthly variations, but shorter variations are possible if the relation of 5.2 is not true. The line source equation is broken up for variable rates from t = 0 to t_f by

$$\Delta T_g = \frac{1}{2\pi k_g} \{ \frac{g_{gc1}}{L} [I(X)_{t_f - t_o} - I(X)_{t_f - t_1}] + \frac{q_{gc2}}{L} [I(X)_{t_f - t_1} - I(X)_{t_f - t_2}] + \dots \frac{q_{gcn}}{L} [I(X)_{t_f - t_{f-1}}]\} \quad (5.4)$$

$t \to 0$
$X \to \infty$
$I(X) \to 0$

where $X_{t_f - t_n} = \frac{r}{2\sqrt{\alpha_g (t_f - t_n)}}$ and $q_{gcn} = q_{gc}$ at $t = t_n$.

$t_f = t_n = 0$

Equation 5.3 can also be arranged in a similar manner.

In a later publication, Ingersoll, Adler, Platt and Ingersoll (41) suggest methods of accounting for ice formation and moisture migration near a ground-couplings, which are outside the scope of this work. The authors also consider the effect of groundwater movement. They suggest that with a groundwater velocity of 0.01 ft/hr, q_{gc}/L improves by 20% and by 79% for 0.1 ft/hr when compared to q_{gc}/L for zero velocity.

Penrod (42) utilizes the work of Ingersoll and Plass (43) as a basis for sizing horizontal ground pipes. This was applied to installations in three different locations. Resulting heating energy ratios for the systems were 3.1 to 3.3. The degree day method was used to estimate the heating demand. Various installations (44), (45) used the methods of Ingersoll, Plass, Zobel and Ingersoll before interest reawakened in the late 1970s.

The interest in energy efficiency, the common use of plastic pipe and the availability of low cost micro-computers lead to the next stage of development of the line source equation for heat pump applications. Kalman (7) utilizes the methods of (39) and develops procedures to account for the effects of the thermal resistance of plastic pipe and the boundary layer. In addition to these values Kalman corrects for heat pump performance variations due to water temperature. Also included in this thesis is an economic analysis.

An important development in a revised version of Kalman's thesis is the provision to calculate the outlet water temperature based on the log-mean temperature difference (LMTD). The original calculation utilized the line source equation to find an average water temperature.

$$T_w = T_{ff} + \Delta T_g + \Delta T_p + \Delta T_{b1} \tag{5.5}$$

water mean

Equation 3.13 was applied to the entire pipe length to obtain the total temperature difference of the inlet and outlet water.

water out water in

$$T_{wo} - T_{wi} = q_{gc}/mc_p \tag{5.6}$$

The outlet water temperature was then evaluated by

$$T_{wo} = T_w + \frac{T_{wo} - T_{wi}}{2} = T_w + \frac{q_{gc}}{2mc_p} \,. \tag{5.7}$$

This was revised according to the definition of log-mean temperature difference

$$LMTD = \frac{(T_{wi} - T_{ff}) - (T_{wo} - T_{ff})}{\ell n \frac{T_{wi} - T_{ff}}{T_{wo} - T_{ff}}} . \tag{5.8}$$

The resulting equations for water outlet temperatures were

$$T_{wo} = T_{ff} + \frac{\Delta T_w}{1 - e^{(\frac{\Delta T_w}{LMTD})}} \quad \text{(Cooling)} \tag{5.9}$$

$$T_{wo} = T_w + \frac{\Delta T_w}{1 - e^{(\frac{\Delta T_w}{T_w - T_{ff}})}} \quad \text{(Heating)} . \tag{5.10}$$

Bose (2) also utilizes the line source equation as the basis for ground-coupling design. Required lengths are calculated from the equations

$$L_c(\text{ft/ton}) = \frac{12{,}000(\frac{COP + 1}{COP})(\frac{R_p}{Nt} + R_g Rf)}{T_w - T_{ff}} \quad \text{(Cooling)} \tag{5.11}$$

$$L_H(\text{ft/ton}) = \frac{12{,}000\ (\frac{COP - 1}{COP})\ (R_p/Nt + R_g Rf)}{T_{ff} - T_w} \quad \text{(Heating)} . \tag{5.12}$$

R_g is found from a variation of Equation 5.1.

$$R_g = \frac{I(X)}{2\pi k_g} \tag{5.13}$$

Notice that the temperature difference term for the pipe ($q_{hp}R_p/N_t$) is not reduced by the run fraction term. This work also utilizes several of the adjustments to the line source equation as suggested by Ingersoll. Several graphs and other methods to simplify calculation procedure are included.

5.3 The Line Source Equation Applied to Vertical Ground Couplings

The work of Ingersoll, Bose, Kalman and others are primarily concerned with the application of the line source equation to horizontal ground-couplings. Vertical ground-couplings are less effected by complexities encountered in horizontal systems such as thermal conductivity and far-field temperature variations. One complication not considered in depth is short circuit heat transfer within a single line source. As an alternative to calculating the interference of two line sources in a single bore hole, a single source with short circuit losses will be considered.

Significant deterioration in performance of concentric couplings will be experienced if the dip tube has a large surface area and/or high thermal conductivity. The problem is more significant in parallel flow systems because of the larger temperature differences between the up and down flowing streams. Series arrangement are less effected. The correction for this can be performed in two ways. The first is to find the average loop water temperature (T_w), apply Equation 5.7 and correct for short circuiting by

$$\Delta T_{sc} = \frac{q_{sc}}{mc_p} . \qquad (5.14)$$

Combining Equations 4.55 and 5.6,

$$q_{sc} = \frac{T_{wi} - T_{wo}}{2R_{sc}} = \frac{-q_{gc}}{2mc_p R_{sc}} \qquad (5.15)$$

with 5.14 yields

$$\Delta T_{sc} = \frac{-q_{gc}}{2(mc_p)^2 R_{sc}} \; . \tag{5.16}$$

Equation 5.7 is then corrected for short circuiting by

$$T_{wo} = T_w + \frac{q_{gc}}{2mc_p} - \frac{q_{gc}}{2(mc_p)^2 \; R_{sc}} \tag{5.17}$$

which simplifies to

$$T_{wo} = T_w + \frac{q_{gc}}{2mc_p} \left(1 - \frac{1}{mc_p R_{sc}}\right) \; . \tag{5.18}$$

Recall that q_{gc} is positive for heating mode and negative for cooling.

A second method of calculating outlet water temperature is to correct for short circuiting in a general equation for heat transfer from an element of differential length and integrate this equation over the entire length of the coupling. Kalman (7) has suggested this method in the derivation of Equations 5.9 and 5.10. In heat exchanges it has been shown that when

$$\frac{\Delta T_{max}}{\Delta T_{min}} < 2.0 \tag{5.19}$$

LMTD can be replaced by mean temperature difference (MTD) with less than 1% error (46). For ground-couplings this translates to

$$\frac{\Delta T_{max}}{\Delta T_{min}} = \frac{T_{wi} - T_{ff}}{T_{wo} - T_{ff}} \text{ (Cooling)}, \; \frac{T_{ff} - T_{wi}}{T_{ff} - T_{wo}} \text{ (Heating)} \tag{5.20}$$

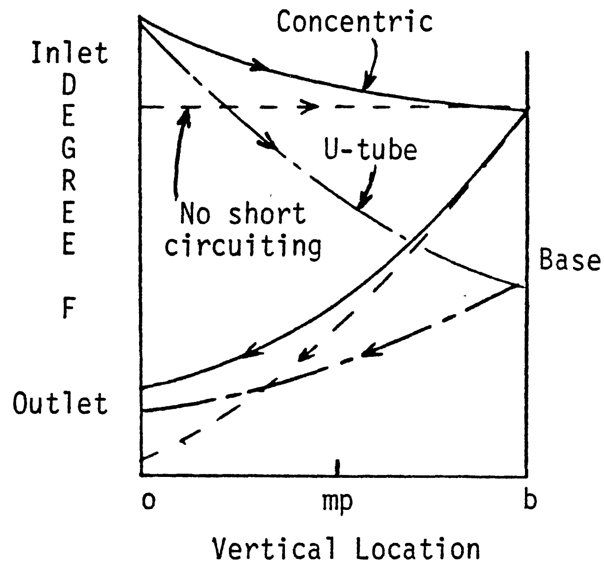


Figure 42. Vertical Water Temperature Variation in Ground-Couplings

However, it was noticed that when compared to experimental results and to the simulations CVHE and CVHI, in certain cases Equation 5.18 resulted in error. When flow rates were changed, T_{wo} remained fairly constant (see Figure 14) while the value of T_{wi} changed much more significantly. This discrepancy can be accounted for by referring to Figure 42. The short circuit heat loss represented by the temperature line from i(inlet) to b(bottom) causes small change in the slope of the temperature profile from b to mp (mid point). Since over 75% of the short circuiting occurs from mp to o (outlet), the slope in this portion of the profile is significantly decreased. The result of this is that the value of $T_w - T_{ff}$ tends to approach $T_{wo} - T_{ff}$, when short circuiting increases. Decreasing flow rates increase the value of $T_{wi} - T_{ff}$ and the error due to replacing LMTD with MTD is also compounded. Also shown in Figure 42, is the temperature profile for no short circuiting and for a U-tube.

An equivalent value for MTD is complicated by the fact that three temperatures (dip tube, annulus and far-field) and two U-values (dip tube and outer tube) must be included in the energy balance. This would result in equation much more complex than 5.9 and 5.10 for outlet water temperature.

An alternative is to utilize experimental results of mean temperatures to arrive at an equation for outlet water temperature. Figures 21 and 22 indicate that two-thirds of the heat transfer occurs in the downward flowing leg of the U-tube and lower half of the concentric. These values were consistent throughout the test except during start-up periods. An estimation of the mean temperature difference for the concentric coupling is

$$MTD = T_w - T_{ff} = \left(\frac{T_{wb} + T_{wmp}}{2} - T_{ff} + \frac{T_{wmp} + T_{wo}}{2} - T_{ff}\right)/2 \ , \quad (5.21)$$

where T_{wb} is the water at the base of the coupling. When simplified and written in terms of heat rejection we have

$$T_w = T_{wb} + 0.583 \frac{q_{gc}}{mc_p} \ . \quad (5.22)$$

When Equation 5.22 is combined with 5.6, 5.14, and 5.17 the result is

$$T_{wo} = \frac{0.42\ q_{gc}}{mc_p}\left(1 - \frac{1}{mc_p R_{sc}}\right) + T_w \tag{5.23}$$

Since this equation is empirical its range of application is limited. It is suggested as yielding better results when the coupling inlet-outlet water temperature difference ($|T_{wo} - T_{wi}|$) exceeds 10^oF and short circuiting is less than 10%. In all other cases Equation 5.18 is appropriate.

The temperature difference between the water and the ground can be found by rearranging Equation 4.49 to find an equivalent thermal resistance, thus

$$\Delta T_p + \Delta T_{bl} = q_{gc} Req = \frac{q_{gc}}{2\pi C_{eq} NtL}\left(\frac{1}{r_i h_i} + \frac{\ln(r_o/r_i)}{k_p}\right) . \tag{5.24}$$

It is sometimes necessary to calculate the minimum or maximum coupling temperature. This involves calculating daily swing utilizing Equation 5.4 and inputting a value of t = 6 hours for the last term. In most larger ground-couplings this would violate the condition of 5.2 and Equation 5.3 should be used. However, for values of $\frac{\alpha t}{r^2} = 10$ the error between the daily average value of T_w and the maximum value is less than 1.0^oF using Equation 5.4.

Another simplification that may be helpful in reducing computation involves an approximation so that Equation 5.1 can be utilized when Equation 5.4 is normally warranted. Typically heat pump operation patterns vary frequently. Line source theory of varying heat rates requires that Equation 5.4 have terms for each different rate. T_w is primarily dependent upon the heat rate during the hours and days immediately preceding evaluation and secondarily to the total amount of heat transferred to the ground over the season.

The single termed Equation 5.1 can be substituted for 5.4 if the heat rate used in 5.1 is the rate averaged over several days before T_w is evaluated and the total seasonal heat transfer is constant. This can be accomplished by finding q_{sc}/L for the week before evaluation. The total heat transfer is found by integrating the heat rate over the entire season (heating or cooling). An equivalent time for Equation 5.1 is then found by dividing this total heat transfer by the average heat rate.

For example suppose a three inch coupling rejects 20 Btu/h-ft for three weeks, then 30 Btu/h-ft for the next three weeks followed by three weeks of 40 Btu/h-ft. Application of Equation 5.4 results in a pipe wall to far field temperatures difference of 17.8^{o}F in soil of k = 1.4 Btu/h-ft-F and $\alpha = 0.027$ ft^2/hr. The average heat rate during the final week is of course 40 Btu/h-ft. The equivalent run time can be found from

$$t_{eq} = \frac{\int_o^{t_f} q_{gc}\,dt}{\bar{q}_{gc}} \, . \tag{5.25}$$

$$\int_o^{t_f} q_{gc}\,dt = (20 + 30 + 40)\,\frac{\text{Btu}}{\text{hr-ft}} \times \left(21 \text{ days} \times 24\,\frac{\text{hr}}{\text{day}}\right) = 45{,}360\,\frac{\text{Btu}}{\text{ft}} \, .$$

The equivalent run time is

$$t_{eq} = \frac{45{,}360}{40} = 1134 \text{ hr.} = 47.25 \text{ days}$$

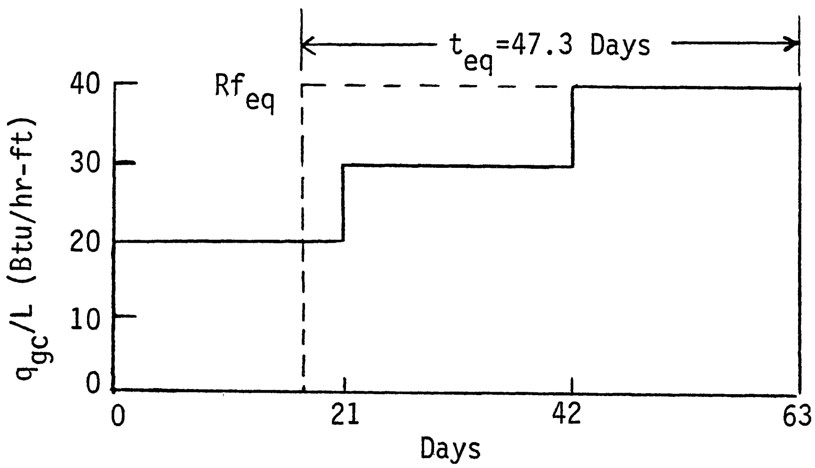


Figure 43. Determination of Equivalent Run Fraction and Time

Figure 43 shows this process graphically. Substituting q_{gc}/L = 40 Btu/hr-ft and t = 1134 hr into Equation 5.1 yeilds a temperature difference of 18.05^{o}F. The resulting error is 1.4%.

The literature search did not reveal a method of accounting for variations in the thermal conductivity of the soil in the axial direction. Ingersoll, Zobel and Ingersoll provide methods of accounting for variation at a given cross section. Hand computation would be difficult but an iterative procedure could be utilized with a microcomputer. The situation of variable thermal conductivity (and far-field temperature) in the axial direction is common to vertical couplings. In some situations the effects can not be neglected.

Solution in concentric couplings with the line source equation involves calculating inlet and outlet water temperatues of each section of ground-coupling that can be considered having constant thermal conductivity and far-field temperature. The outlet water temperature of each section is compared with the inlet temperature of the downstream sectin. If they do not match the values for q/L of each section are accordingly adjusted, water inlet and outlet temperatures are recalculated and the process is repeated until temperatures match.

As an example, a 150 ft 3 inch steel ground-coupling with a 1.25 inch insulated dip tube has a 50% run fraction at a 12,000 Btuh heat rejection rate for one month. Water flow rate is 1500 lb/hr (3 GPM) and soil properties are $k_g = 1.2$, $\alpha_g = 0.026$ and $T_{ff} = 60^oF$ for the lower 100 feet and $k = 0.9$, $\alpha = 0.02$ and $T_{ff} = 63^oF$ for the upper 50 feet. If a constant heat rejection rate is applied across the length of the coupling and Equations (5.1), (5.5), and (5.23) are applied to both sections, the water outlet temperature of the lower section is 77.7^oF and the inlet temperature to the upper section is 90.2^oF. The heat rejection rate of the upper section must be decreased and the lower increased until the two temperatures are equal. These two temperatures converge at a value of 80.7^oF when the lower section of the coupling rejects heat at a rate of 47.0 Btu/hr-ft and the upper section at 26.0 Btu/hr-ft. Coupling outlet temperature is 80.6^oF. The outlet temperature would be 77.1^oF if the entire coupling was in the soil of higher thermal conductivity and lower far field temperature.

This method does not apply directly to the U-tube designs since there are two flow streams in each section. In U-tubes the heat transfer rate varies much less with axial direction since the average temperature of the tubes is almost constant. Therefore the heat rejection rates for sections in soils of different thermal properties can be adjusted until the value for the average water temperature (T_w) is equal. The inlet and outlet temperatures for each section need not be calculated. This is only an estimate of the relative heat transfer capability of each section. Actual determination of inlet and outlet temperatures require finding the heat transfer rate of both tubes and transfer between them. This can not be accomplished with the line source equation alone.

The effect of water movement can also be accounted for in this way. Reference (39) indicates q_{gc}/L would increase 20% with a water movement of 0.01 ft/hr, which is in agreement with (40). When increases in q_{gc}/L are applied to sections of ground-coupling that are located in strata were water movement occurs, the effects are the same as increasing thermal conductivity. Therefore the methods for adjusting the line source for varying thermal properties is applicable. The solution would be somewhat more complex since the water temperature at two locations must be balanced instead of at one point as in the previous example.

An additional adjustment to the line source equation must be made for calculation of average water temperature while the unit is running. The values obtained for T_w using Equation 5.5 is the average when the unit is both off and on. The true average temperature must be adjusted. Figure 18 suggest the unweighted average water temperature is 79^{o}F and the instantaneous value can be approximated by

$$T = 79 - 4.0 \sin \frac{\pi t}{12}\ , \tag{5.26}$$

where t is in hours.

The run fraction at this time was 50% and can be estimated by

$$Rf = 0.5 - 0.25 \sin \frac{\pi t}{12} \tag{5.27}$$

If the time lag between Rf and T is neglected then

$$T_{AVG} = \frac{1}{\int_0^{24} R_f dt} \int_0^{24} TRfdt \tag{5.28}$$

Integraton and evaluaton leads to

$$T_{AVG} = 80.0^{o}F$$

A similar procedure was performed using the temperatures of Figure 19. The average daily temperatures (T_w) was 86.0^{o}F and the running weighted average (T_{wa}) was 87.2^{o}F. In both cases the weighted average was increased by about 12% of the daily water temperature swing. This suggests the equations

$$T_{wa} = T_w + 0.06\ (T_{w_{max}} - T_w)\ \text{(Cooling)} \tag{5.29}$$

$$T_{wa} = T_w + 0.06\ (T_{w_{min}} - T_w).\ \text{(Heating)} \tag{5.30}$$

The application of the line source equation to vertical ground-couplings is highly flexible in terms of the degree of accuracy. In many cases this may be determined by the method of computation (hand calculator or microcomputer). However, the primary limitation on accuracy of the line source, as well as one numerical methods, is the degree of accuracy that the thermal properties of the ground can be determined.

5.4 Heat Pump Air Side Load Calculation

As previously mentioned the performance of CLGCHP systems is not a direct function of outdoor air conditions. In addition to knowing the thermal properties of the soil and ground-coupling, the water temperature into the heat pump is a function of operating history during the preceding days and months. It is therefore necessary to estimate the building load to determine unit performance.

The most accurate way of determining building load is by utilizing a transfer function method such as DOE-2 or the OSU program BLSIM. These methods involve a significant amount of computer capability. Input, which includes weather tapes, usually requires considerable detail. A second alternative is the bin method, utilizing the standard fixed base temperature or a variable base. A third method is the use of degree-days or hours which can also be fixed base temperature with correction factors or variable base.

The accuracy of these methods increases with the degree of detail of the input. Fixed base degree-day methods for energy consumption do not yield good results even with regression analysis correction factors (47). The accuracy of the methods increases significantly with the use of a varible-base temperature (48). This method accounts for variations in solar gain, infiltration, internal heat generation, latent loads and window placement. The degree-day base temperature is accordingly adjusted. The method is particularly useful in determining monthly run fractions, since degree-days are often listed by month. Final selection of a method of energy calculation requirement must recognize that house-to-house variation in internal loads can not be accounted for even the most elaborate methods. These are often a result of occupant living habits.

Improvements in the variable-base degree-day method are currently being verified (49). A final report is scheduled to be published in June 1985. The present procedure is to follow the recommendations of (48) to determine the correct base temperature. The basic equation for the variable based degree day method is

$$E_{rq} = \frac{24\ q_{dl}\ DD\ (T_{VB})}{(T_{dl} - T_i)} \tag{5.31}$$

and this can be used on a monthly or annual basis. Notice that the term DD(VB) replaces the terms

$$C_f\ DD(65)/\eta V$$

in the standard degree-day formula with various correction factors. The design heat gain/loss load (dl) is normally found at the 97 1/2% value recommended by ASHRAE (40).

The primary difficulty in using this method is the evaluation of T_{VB}. Kusuda (47) defines this temperature in relation to the indoor design temperature

$$T_{VB} = T_i - \frac{\Sigma q_i}{\Sigma K_i} \tag{5.32}$$

Σq_i is the summation of internal heat generation and solar gains. The ΣK_i term includes the heat transmission-area terms (UA) for all envelope components and the infiltration term (1.08 x CFM). Heat gains are primarily from these components

$$\Sigma q_i = q_{i\ people} + q_{i\ appliances} + q_{i\ solar} \tag{5.33}$$

Reference (50) Chapter 26 lists values of adjusted total heat gain due to people for various activities. The total values should be multiplied by the average occupancy time, not 24 hrs/day, for energy calculations. An average value of 1200 Btuh is suggested by (50) for appliances. Any major appliance not typically located in homes should be accounted for by Tables 20 and 21 of (50). The solar gain is the most complex value to evaluate because it is considered at the design point for cooling and usually not considered at the heating load design point (night), unless significant thermal storage and passive design are present.

Tables 18A to 26A Chapter 27 of Reference (50) list values for half day totals of Solar Heat Gain Factors (Btu/ft^2) for various latitudes. In the heating season, the adjustment for heat gain through glass is made by summing the half day totals, accounting for shading coefficients and clearness index (K_T).

$$q_{i\ solar} = SC * SHGF * K_T * A_{windows} \tag{5.34}$$

Values for the clearness index indicate the relative amount of insolation for a given location and month to the total extraerrestrial value and are included in weather data appearing in Reference (51), or in similar publications dealing with solar energy utilization.

In the cooling mode solar gain is accounted for in the calculation of design load. If design load calculations are arrived at as a result of a short term peak solar gain (i.e.. one or two hours of gain through an unshaded west window), not considered in Equation 5.33, Equation 5.31 will overpredict energy consumption. More elaborate means of evaluating $q_{i\ solar}$ are needed in this case. This is also true with passive design or if buildings other than light weight are analyzed.

In evaluating K_i the average transmission coefficients for the roof, walls, floors, windows and doors are multiplied by their respective areas (except for slabs where perimeter is used). Details of evaluation are found in Chapters 23 and 25 of (50). These values are summed and added to the infiltration multiplied by 1.08, the conversion from CFM to Btu/h-^{o}F for air.

$$\Sigma K_i = UA_{roof} + UA_{walls} + UA_{floor} + UA_{doors} + UA_{windows} + 1.08\ Q_{if} \qquad (5.35)$$

The rate of infiltration can be estimated according to McQuiston (52).

$$Q_{if} = ACH * \text{Building Volume}/60 \qquad (5.36)$$

The air changes per hour (ACH) can be evaluated from

$$ACH = C_1 + C_2\ \overline{V}_w + C_3\ (T_i - T_o)\ . \qquad (5.37)$$

$\overline{V}_w$ is the average wind velocity in mph and a trial and error calculation is performed to match T_o with T_{VB}. Table VII gives the values of the constants in Equation 5.37 for houses that are tight (close fitting doors and windows, weather stripped, vapor barriers), medium (frame houses 10 years or older, average fitting doors and windows), and loose (poorly fitting doors and windows, more than 20 years old, average maintenance).

TABLE VII

CONSTANTS USED TO EVALUATE INFILTRATION RATE (52)

House Type	C_1	C_2	C_3
Tight	0.25	0.002	0.0085
Medium	0.25	0.004	0.0245
Loose	0.25	0.006	0.0525

The methods for evaluating T_{VB} will undoubtedly be improved upon by Reference (49) and should be implemented upon publication.

The heating and cooling load may be evaluated by a variety of means. If computer programs are not available the simplified method for residences appearing in Chapter 26 of (50) should be used for cooling. Other building types should use the longer methods of (50) or (52).

Heating loads can be calculated by the methods of Chapter 25 of (50). Values for degree-days for temperature bases other than 65^{o}F can be determined from References (53), (54), (55) and (56). Monthly variable base degree-day tables can also be generated by using bin data to determine the number of degree-days between 65^{o}F and T_{VB} and subtracting this amount from DD (65). An empirical equation is suggested by (58) of the form

$$DD(VB) = (T_{VB}/A)^{B} \tag{5.38}$$

Reference (56) list the values of the constants A and B for over 200 cities in the United States. Guntermann (57) presents a simplified variable method for annual calculations when degree-days referenced to bases other than 65^{o}F are not available. His method can be adopted to monthly calculations. Degree-days to different bases are also published by the National Climatic Center.

Before monthly run fractions can be determined, a heat pump that can meet the design heating and cooling load must be selected. Caution must be taken to determine both heating and cooling loads, since homes in identical locations may have peak loads during different seasons. Figure 44 shows the results of transfer funtion load calculations (53) applied to two 1500 ft^2 homes in Stillwater. One house is classified as tight according to (52). It has a R-35 ceiling, R-20 walls, simple passive orientation (windows on south wall, garage on west wall), storm doors and windows. The second house has medium infiltration, R-22 ceilings, R-13 walls, random window arrangement and storm doors. The results are obvious. If the heat pump in the medium house is sized for

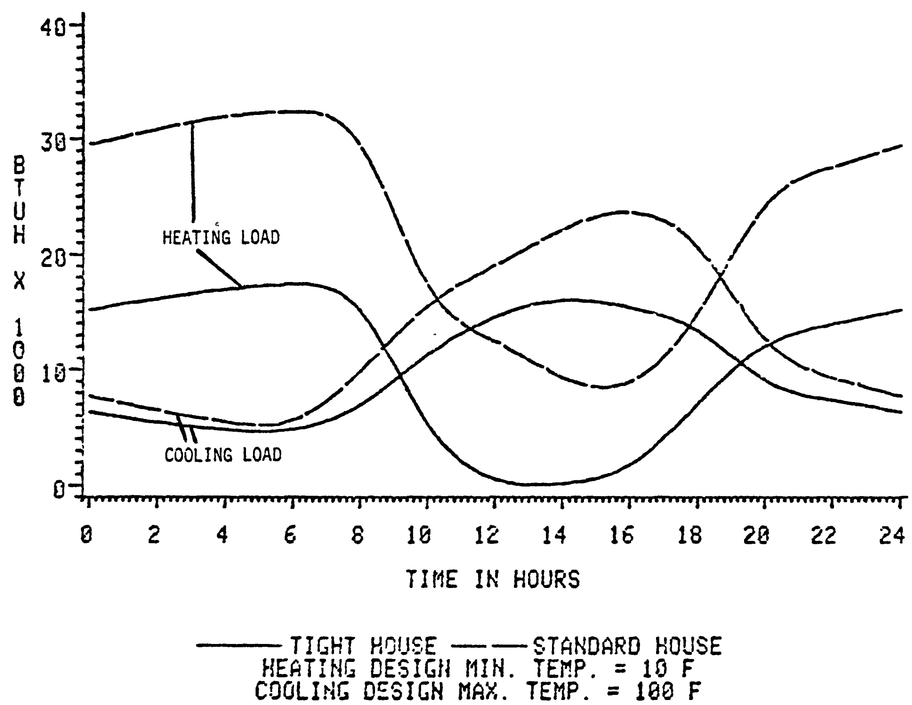


Figure 44. Daily Heating and Cooling Load Variation for Two Houses Near Design Conditions

cooling, substantial back-up heat is required for the design heating day.

Once the heat pump has been selected the monthly run fraction can be determined from

$$(Rf)_{mon} = \frac{(E_{rq})mon}{q_{hp} * DAYS * 24} \qquad (5.39)$$

This results in an iterative process since heat pump capacity is a function of coupling water temperature which is a function of monthly run fraction. Experience with the application of Equations 5.1, 5.4, 5.5, 5.14, and 5.7 or 5.23 leads to the ability to make "educated" first quesses for water temperature so that the number of iterations necessary is small.

5.5 Results

Figure 45 is a comparison of the results of using the line source Equation 5.4 for the experimental cooling mode test on the 1-1/2 inch polybutylene U-tube. Figure 46 compares similar results for the 3 inch concentric coupling. The figure also shows the values obtained using Equation 5.23 to calculate water outlet temperature. Figure 47 compares the maximum and minimum daily average water temperatues using the expanded form of Equation 5.3 with experimntal and FDE simulation values.

The line source equation slightly overpredicts the actual and simulated temperature distribution. The equation shows the greatest error during the off or light run fraction times, the least critical periods. Notice that as run fraction increases, error decreases. Table

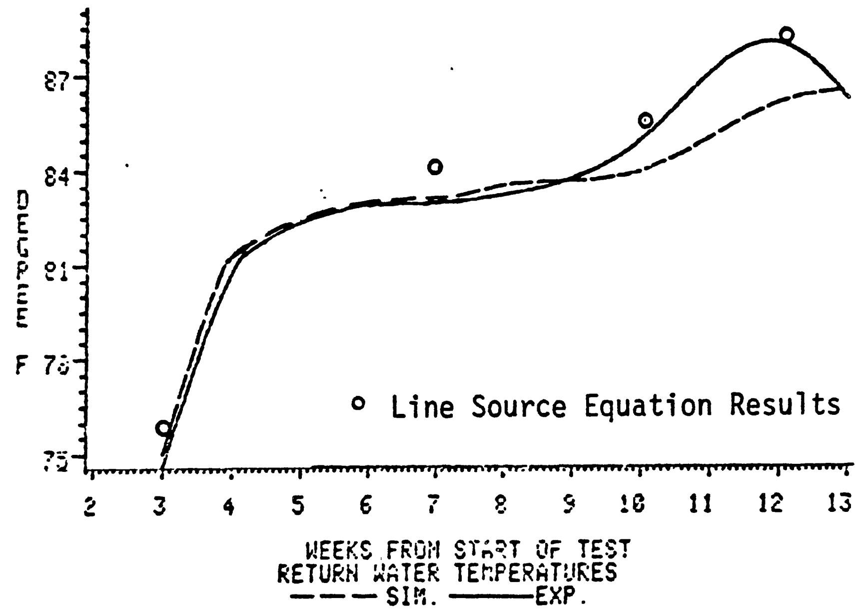


Figure 45. Experimental, Finite Difference and Line Source Water Temperature Results for Polybutylene U-Tube

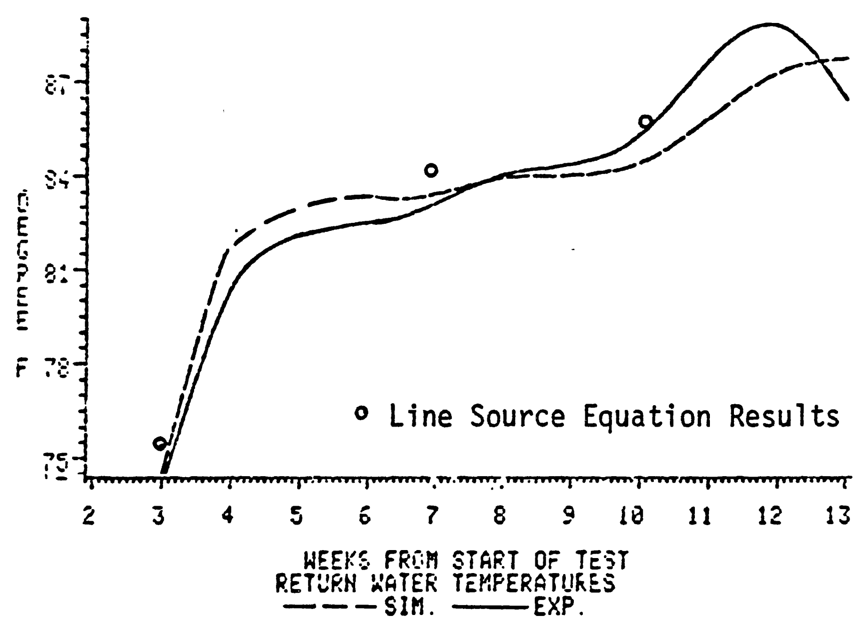


Figure 46. Experimental, Finite Difference and Line Source Water Temperature Results for Concentric Tube

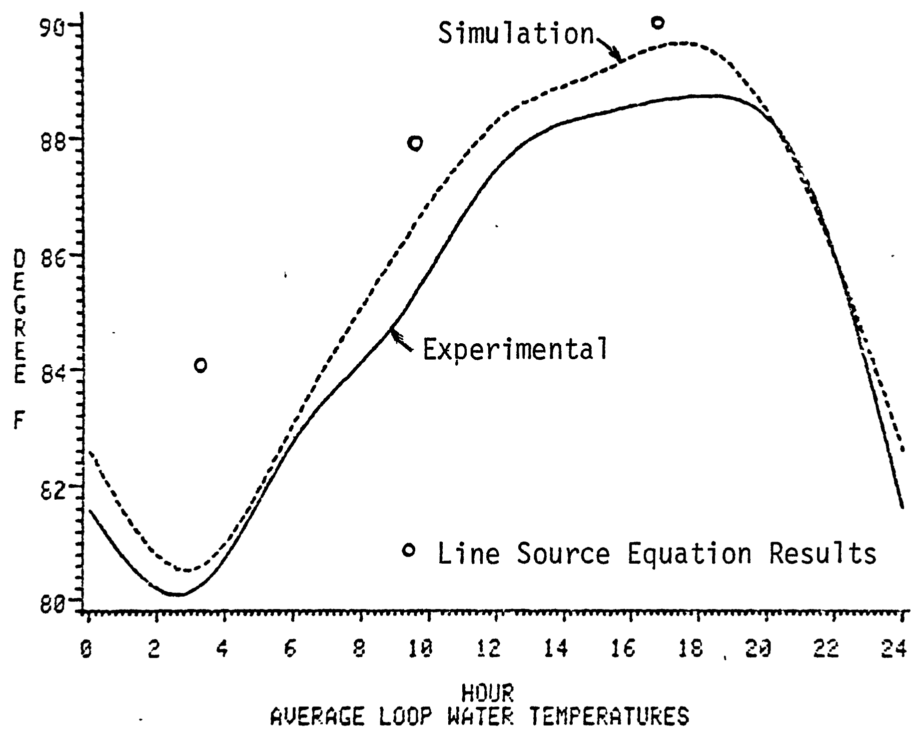


Figure 47. Maximum and Minimum Water Temperatures Results for Experiment and Line Source

VIII shows the results of a comparison between a simulation of the PB coupling using UTI and results calculated using Equation 5.4 Three conditions were simulated, a 92% run fraction a 50% run fraction and a 50% run fraction in which natural convection effects were suppressed during off periods. Notice that the value of the line source agrees more closely with the results of the simulation using the reduced natural convection. However all error with the line source is small when compared to those resulting from uncertainity in soil properly measurement. The appendix contains an example procedure for the design and simulation of a vertical CLGCHP system.

TABLE VIII

FOUR WEEK SIMULATION OF 1.5 INCH POLYBUTYLENE U-TUBE USING FINITE DIFFERENCE AND LINE SOURCE EQUATION

Test Simulated	Average Water Outlet Temperature (^{o}F)	
	FDE	Line Source
92% Run Fraction	95.9	96.7
50% Run Fraction	82.4	83.6
50% Run Fraction $h_{nat} = 1.0 \frac{Btu}{h\text{-}ft^2\text{-}F}$	83.8	--

CHAPTER VI

SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

6.1 Summary and Conclusions

Vertical closed loop ground-coupled heat pump systems are a viable means of heating and cooling. Manufacturers are beginning to devote more effort to water-to-air heat pump design and therefore unit efficiencies are rapidly increasing. The development of relatively low cost drilling equipment presently being marketed will continue to drive down the cost of installations. Plastic pipe, fittings and fusion equipment are being manufactured especially for ground-couplings.

This experiment has compared the performance of several vertical ground-coupling designs. The primary results are in the form of thermal performance. The decision as to which type coupling is optimum for a particular location must be based on many other variables mentioned in the conclusions of Chapter II. Generally, the large diameter and multiple U-tubes perform the best. This is a result of the smaller thermal resistances (large surface area, parallel heat transfer and large boundary layer coefficients), reduced "short circuit" heat transfer and smaller pressure losses (greater flow rates with less required pumping power).

Small diameter U-tubes have the poorest thermal performance and greatest pressure losses. However, they can be easily installed and could be the most economical in areas where drilling cost is small.

Concentric tube designs have medium thermal performance. Larger designs (greater than 2 inch) require that the inside heat transfer coefficient be enhanced for good performance. Small diameter designs have greater thermal resistance but are more easily installed. Care must be taken to minimize thermal "short circuiting" in concentric couplings.

Finite difference equations are a powerful tool in evaluating ground-coupling performance. They offer a high degree of flexibility and accuracy. They require significant computer time and input in order to achieve this accuracy. The line source equation is likewise a useful tool for design and simulation of ground-coupling performance. Accuracy is slightly reduced with this method but computation time is much less. Adjustments can be made to the calculation procedure to deal with added complexities or it can be simplified to the extent that results can be attained with a pocket calculator. The error appears to be primarily a result of improved heat transfer at the start of each on cycle. The heat transfer is improved at this time because of natural convection effects during off periods that are not accounted for with the line source equation. The thermal lag at the beginning of the cycle improves average water temperature. Therefore for longer cycles error is smaller.

System performance is linked to the amount of time that the unit has been previously operated. This amount can be determined by calculating the building heating or cooling requirement throughout the season and expressing this amount as a fraction of the total possible. The degree-day method is a simple but satisfactory way to estimate this fraction. Fixed based methods yeild significant error but estimates made with a variable base have proven to be much more reliable. A project is currently underway to validate and improve this method. Bin methods, either fixed or variable-base, offer the next step up in terms of accuracy. They require additional calculation. The most accurate way of predicting building energy requirements is by use of the transfer function method. It requires significant computer capabilities and time.

6.2 Recommendations

Section 2.5 offers specific recommendations concerning the six ground-coupling utilized in this project. Variations or novel coupling designs should be evaluated in terms of the following.

1. Cost of installation
2. Cost of material
3. Equivalent thermal resistance, the inverse of equivalent heat transfer coefficient multiplied by surface area, should be minimized. Increasing the surface area may decrease water velocity and therefore the inside film coefficient. The resistance of the pipe wall usually predominates in plastic couplings and should be the primary point of consideration.
4. "Short circuit" heat transfer resistance should be maximized without significantly increasing thermal resistance.
5. Pressure drop in coupling systems should be minimized. Pumping power for water only systems should not exceed 30 watts/GPM. Water control valves significantly degrade total system performance.

Additional precautions are outlined in References (2) and (4) for actual installations.

The use of the line source equation is the recommended method of designing and simulating vertical ground-couplings. The error resulting in the use of this method is small compared to typical errors in ground property determination. Although calculations are possible with a pocket calculator, the development of micro-computer software is recommended for design and simulation. Several iterative steps are necessary for proper results, especially if any of the complexities described to in the previous chapter are encountered. The recommended procedure is as follows.

1. Calculate heating and cooling load at ASHRAE 99% design conditions.
2. Select unit to meet conditions.
3. Calculate monthly run fractions from variable-base degree-day or bin method.
4. Size length of ground-coupling based on recommendations of previous chapters.
   a. Select minimum (heating) and maximum (cooling) acceptable water temperature based on unit capacity.
   b. As a first guess increase (heating) or decrease (cooling) this amount by one-half the expected daily water temperature range. First guess would be 3 to 4^{o}F for heating and 4 to 5^{o}F for cooling.

c. Rearrange Equations 5.6 and 5.7 to solve for T_w from T_{wi} and use the value found in b for T_{wi}.

d. Calculate values of h_{eq} and R_{sc}.

e. Rearrange Equations 5.5 and 5.4 or 5.1 with 5.25 to solve for length. The resulting equations will be similar in form to Equations 5.11 and 5.12. The value for q_{gc} should be for the maximum daily run fraction. Calculate t_{eq} based on run fractions of previous months if Equation 5.1 is used.

f. Check assumption b, using Equation 5.3.

g. Reiterate, replacing Equations 5.6 in c with Equation 5.18 or 5.23 until temperatures for T_{wo} agree with these for the assumed value of q_{gc}.

5. At this point the unit and coupling are properly sized. Energy requirements can be found by calculating power consumption from manufacturers performance data for monthly average water outlet temperature, T_{wo} (Equations 5.29 and 5.30) and multiplying by monthly operating hours (run fraction x total hours in month). The monthly values for T_{wo} must be found byiteration in order to find the proper value of q_{gc}. Pumping and other auxilary power is included.

6.3 Recommendations for Further Study

The primary areas in need of additional treatment are as follows.

1. Develop new or modify existing software to include results of this work using line source equation.
2. Develop simplified methods of ground property determination.
3. Conduct a controlled experiment and develop numerical methods to derive equations for equivalent heat transfer coefficients and thermal short circuit resistance in non-concentric ground-coupling designs.

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APPENDIX A

SAMPLE CALCULATIONS FOR SYSTEM DESIGN AND PERFORMANCE

Size a vertical CLGSHP for a 1500ft^2 home in Tulsa, Oklahoma with the following dimensions.

R-30 roof - Vented attic in summer

R-15 Frame walls

Slab on grade with 3/4" polystyrene perimeter insulation

Windows single-pane (W = 20ft^2, E = 20ft^2, S = 50ft^2), Blinds

Doors - solid with storm door - 20ft^2

Tight house, 5 occupants, Indoor temps. 68^o/75^o

Ground conditions k_g = 1.4 Btu/hr-ft-F, α_g = 0.026 ft^2/hr, T_{ff} = 62^{o}F

Use ASHRAE simplified cooling load calcaultion with infiltration calculated from Equation 5.35

97 1/2% Condition = 18,800 Btuh

99 Condition = 20,500 Btuh

Heating load

97 1/2% = 27,200 Btuh

99% = 29,600 Btuh

Calculate T_{VB}

$$T_{VB} = T_i - \frac{\Sigma q_i}{\Sigma k_i}$$

For cooling do not include solar gain, use average hours for occupants

$\Sigma q_i = q_{people} + q_{appliances} = 1070 + 1200 = 2270$ Btuh

$\Sigma K_i = UA_{roof} + UA_{walls} + UA_{win.} + UA_{doors} + 1.08\ Q_{if}$

$\Sigma K_i = \frac{1}{30} \times 1500 + \frac{1}{15}\ 2430 - 1.04 \times 90 + .38 \times 20 + 1.08\ (65) = 384$

Note: Q_{if} must be solved by trial and error.

$$T_{VB} = 75^{o} - \frac{2270}{384} = 69^{o}F$$

Using DD (65) and bin data to find DD (69) and applying equation 5.31 to find the monthly heating requirement.

	DD (65)	DD (69)	E_{rq} (10^6 Btu)
May	167	84	1.69
June	381	270	5.30
July	564	442	8.67
August	518	397	7.79
September	282	190	3.73
October	72	24	0.47

For heating - include solar gain using Table 22A, Chapter 27, ASHRAE Fundamentals (50)

Use an SC of 0.7 (half drawn blinds) and a clearness index for January of 0.51. Values will vary with month by use January as average.

$q_{solar} = 2650$, $q_{app} = 1200$, $q_{people} = 1450$

$K_i = 414$ (Increase in Q_{if})

$$T_{VB} = 68 - \frac{5300}{414} = 55^{o}F$$

Repeating the method for cooling.

	DD (65)	DD (55)	E_{rg} (10^6 Btuh)
November	474	264	3.13
December	781	475	5.64
January	924	617	7.32
February	680	408	4.84
March	500	224	2.66
April	168	89	1.06

Select equipment - Use 1 1/2" P.E. U-tube

From Appendix B - We can use LT40 to meet design heating load without auxiliary or use LT30 with auxiliary heat and/or night set back.

Using the LT30 for these calculations, which is a nominal capacity of about 2-1/4 tons. As a first approximation use 150' bore/ton = 340'

Select coupling - 2 parallel 1-1/2 inch PE U-tubes

Pressure drop @ 3.5 GPM through an equivalent length of pipe (340' + fittings) = 450 = 0.7 ft-H_2O

Add - 3' for flowmeter and valves

Add - 18.7 for heat pump

$\Delta P = 22.4$ ft

Use Grundfos UP26-96 (205 watts)

Select lowest temperature desired = 38^oF

Add one half daily range, guess 4^oF = 42^oF

Add one-half ΔT_w = (guess 3^oF) = 45^oF

This is daily average value of T_{wo}

From manufacturer's data we can calculate ground coupling heat transfer

$q_{gc} = q_{hp} - 3.412\ (P_{hp} + P_{pump})$

T_{wo}	q_{hp}	q_{gc}
55	29,200	20,600
50	26,800	18,500
47.5	25,600	17,450
45	24,400	16,400

Using the simplified method of t_{eq} and guessing $\overline{T}_{wo} = 55^{o}F$ for November, $50^{o}F$ for December and 47.5 for January the run fractions are

$Rf_{Nov.} = 15\%$, $Rf_{Dec.} = 31\%$, $Rf_{Jan.} = 38\%$

@ $45^{o}F$ $q_{gc} = 16,400$, $t_{eq} = 70.7$ (Equation 5.25)

Calculate h_{eq}

For 1 1/2" PE and 3.50 GPM, V = 1985 ft/hr

Re = 4800, j = .0037, $h_c = 96$ Btu/hr-ft-ft

For 1 1/2" PE - $D_o = 1.9"$, $D_i = 1.61$, $k_p = .226$, $h_{eq} = 24.2$

For 1/4" separation distance, $D_{eq} = 2.94"$, $\rightarrow r_{eq} = 0.123$ ft

$$R_{sc} = 0.87 \frac{hr - {}^{o}F\text{-}ft}{Btu}/L$$

The simplest way to account for short circuiting is to calculate the temperature penalty and subtract this from the total. Using Eq. 5.16 for one loop assuming L = 170 ft.

$$\Delta T_{sc} = \frac{-q_{gc}}{2(mc)^2 R_{sc}} = \frac{-16,400/2}{2(1750)^2\ 0.87} = -0.3^{o}F$$

This gives a total

$$\Delta T_T = \Delta T_g + \Delta T_p + \Delta T_{sc}$$

For our case $\Delta T_g + \Delta T_p = 62 - 45 = 17^oF$

$$\Delta T_T = 16.7^oF$$

To size coupling assume seven days with a low temperature at the 97 1/2% ASHRAE condition (13^oF) and an average temperature of 18^oF.

$$HDH = (55 - 18)(24) = 888$$

$$E_{rq} = \frac{24,400\ (888)}{68 - 13} = 394,000 \text{ Btu (Auxiliary heat required).}$$

Total possible

$$E_T = (24,400)(24) = 585,600$$

$$Rf_{max} = 0.67$$

Combining equation 5.1 and the equation for ΔT_p and rearranging,

$$L_{req} = \frac{q_{gc}}{2\pi\Delta T_T} [\frac{1}{r_{eq}h_{eq}} + \frac{1}{k_g} (Rf_{Jan.} (I(X)_{t_f - 0} - I(X)_{t_f - t_{eq}}) + R_{f_{max}} I(X)_{t_f - t_{eq}})]$$

L_{req} = 350 ft or 2 - 175 ft couplings

Monthly Average Temperatures

November: Guess $T_{wo} = 50^{o}F$, q_{gc} = 18,500 Btuh or 9,250/Loop

Apply Eqn. 5.1

$$\Delta T_g = \frac{Rfq_{gc}}{2\pi k_g L} I(X) = \frac{(0.15)(9,250)}{2\pi(1.4)\ 175} I\left(\frac{.123}{2\sqrt{.026\ (30)\ 24}}\right) = 3.6^{o}F$$

$$\Delta T_p = \Delta T_{pw} + \Delta T_{b1} = q_{gc}\Delta R_{eq} = \frac{9,250}{2\pi(.85)(2)(175)}\left(\frac{1}{.0671\ (96)} + \frac{\ln(.0792/.0671)}{.226}\right)$$

$\Delta T_p = 4.4^{o}F$

$T_w = T_{ff} - \Delta T_p - \Delta T_g = 54.0^{o}F$

Using Eqn. 5.23 $T_{wo} = \frac{0.42\ (9250)}{1750}\left(1 - \frac{1}{1750\ (0.00732)}\right) = 56.0^{o}F$

Wrong guess! Try $T_{wo} = 55^{o}F$, q_{gc} = 10,300/Loop

Using same method $\underline{T_{wo} = 55.5^{o}F}$ Okay

December: Guess $T_{wo} = 52^{o}F$, q_{gc} = 19,400 Btuh, 9,700/Loop

Apply Eqn. 5.4

$$T_{wo} = \frac{1}{2\pi(1.4)175}\left\{0.15(10,300)\left[I\left(\frac{.123}{2\sqrt{.026\ (61)\ 24}}\right) - I\left(\frac{.123}{2\sqrt{.026\ (24)\ (61-30)}}\right)\right] + \right.$$

$$\left. 0.3\ (9700)\ I\left(\frac{.123}{2\sqrt{.026\ (24)\ (61-30)}}\right)\right\}$$

$T_{wo} = 7.9^{o}F$

$\Delta T_p = 4.6^{o}F$

$T_{wo} = 51.7^{o}F$ Guess Okay

January: Guess $T_{wo} = 50^{o}F$

Apply Eqn. 5.4 $\Delta T_g = 10.1^oF$

$\Delta T_p = 4.4^oF$ $T_w = 47.5^oF$

$T_{wo} = 49.7^o$ Guess okay

Repeat for February and March.

Since the values of T_{wo} do not agree with those values assumed on p. 152 the process should be repeated. Once these values are found, the monthly energy use can be found by multiplying R_f by hours by (P_{hp} + P_{pump}).

APPENDIX B

EQUIPMENT PERFORMANCE

actual performance data:

Model	Unit Water Flow			Cooling (1)				Heating (2)		
	Enter. Water Temp °F	G P.M	P.D. (4)	Total BTU/HR	Sensible Cap %(3)	Watts Input	EER	Total BTU/HR	Watts Input	C O.P
LT30	45°	3	4.6	31200	66	1940	16 1			
		5	10.4	31600	66	1750	18 1	23700	2110	3.3
		7	18.7	31900	65	1680	19 0	24400	2140	3.3
	50°	3	4.6	30400	67	2015	15.1			
		5	10.4	30800	66	1835	16.8	26100	2200	3.5
		7	18.7	31200	66	1790	17.4	26800	2230	3.5
	55°	3	4.6	29600	67	2090	14.2			
		5	10.4	30200	67	1930	15.6	28500	2290	3.6
		7	18 7	30500	67	1870	16.3	29200	2320	3.7
	60°	3	4 6	28800	68	2170	13.3			
		5	10.4	29500	68	2020	14.6	30800	2380	3.8
		7	18.7	29800	67	1960	15.2	31600	2415	3.8
	65°	3	4.6	28000	69	2240	12.5			
		5	10 4	28800	68	2100	13.7	33200	2470	3.9
		7	18.7	29000	68	2060	14.1	34000	2510	4 0
	70°	3	4.6	27200	69	2325	11.7			
		5	10.4	28000	69	2190	12.8	35500	2560	4 1
		7	18.7	28300	68	2155	13 1	36400	2600	4 1
LT40	45°	4	4.6	39400	69	2460	16.0			
		6.5	11.0	40500	68	2340	17.3	30500	2765	3.2
		9	18.5	41300	68	2285	18.1	31800	2790	3.3
	50°	4	4.6	38600	70	2560	15.1			
		6.5	11.0	39500	69	2410	16.4	33300	2870	3.4
		9	18.5	40500	68	2370	17.1	34800	2900	3.5
	55°	4	4.6	37800	71	2655	14.2	33800	2895	3 4
		6.5	11.0	39000	69	2515	15.5	36100	2965	3 6
		9	18.5	39700	69	2460	16.1	37700	3005	3 7
	60°	4	4 6	36900	72	2750	13 4	36200	2990	3 5
		6.5	11.0	38100	70	2600	14.6	38900	3065	3 7
		9	18.5	38800	69	2540	15.3	40600	3110	3.8
	65°	4	4.6	36100	73	2840	12 7	38500	3080	3.7
		6.5	11.0	37300	70	2680	13.9	41700	3165	3.9
		9	18 5	38000	70	2625	14.5	43500	3215	4.0
	70°	4	4 6	35300	74	2940	12.0	40900	3175	3.8
		6.5	11.0	36400	71	2765	13.2	44600	3270	4.0
		9	18.5	37200	70	2715	13.7	46500	3325	4 1
LT50	45°	5	2.0	50000	69	3050	16.4			
		8	11.3	50800	69	2870	17.7	35600	3240	3.2
		11	26.8	51900	68	2780	18 7	37300	3260	3 4
	50°	5	2 0	48800	70	3170	15.4			
		8	11.3	49500	69	2980	16 6	38600	3350	3 4
		11	26 8	50500	69	2895	17 4	40500	3385	3 5
	55°	5	2.0	47600	71	3280	14 5	39000	3420	3.3
		8	11.3	48200	70	3090	15.6	41700	3460	3.5
		11	26 8	49100	70	3010	16 3	43700	3510	3 6
	60°	5	2.0	46400	72	3400	13.6	42000	3520	3.5
		8	11.3	47000	71	3210	14.6	44900	3570	3.7
		11	26.8	47700	70	3120	15.3	47000	3630	3.8
	65°	5	2.0	45200	73	3510	12.9	45000	3620	3.6
		8	11.3	45700	72	3320	13 8	48000	3680	3 8
		11	26 8	46400	71	3230	14 4	50200	3760	3.9
	70°	5	2.0	44000	74	3630	12 1	48000	3720	3.8
		8	11.3	44500	73	3440	12 9	51100	3790	3.9
		11	26.8	45000	72	3350	13 4	53500	3890	4 0

CF= .970677+.0036[illegible] GPM   CF= 1.21112 − .03928 GPM + .001611 GPM²

(1) Cooling capacities stated, 80°D B . 67°W B entering air and at rated unit air flow

(2) Heating performance based on 70°F entering air and unit rated air flow

(3) Sensible capacities percentages stated. 80°/67° entering air and unit rated air flow

(4) Water pressure drop (P.D) stated in "Feet of Head"

Tinted areas signify operation at conditions not recommended

FHP

"LT"&"HLT" Heating Performance

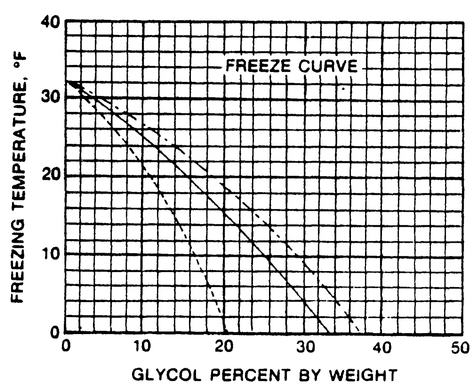
for Water & Glycol/Water entering temperatures below 45°F

LT & HLT MODEL	ENTERING WATER (1) TEMP, °F	FLOW RATE, GPM	HEATING PERFORMANCE (2)			SCFM
			BTUH	WATTS	COP	
30	35	7	19300	1950	2.9	950
	38		20600	2000	3.0	
	41		22400	2060	3.2	
	44		23500	2110	3.3	
40	35	9	25500	2580	2.9	1200
	38		27200	2640	3.0	
	41		28900	2700	3.1	
	44		30600	2770	3.2	
50	35	11	30000	3020	2.9	1500
	38		32000	3100	3.0	
	41		33900	3170	3.1	
	44		35800	3230	3.2	
60	35	13	39900	3830	3.1	1700
	38		42000	3930	3.1	
	41		44300	4010	3.2	
	44		46500	4100	3.3	
70	35	16	46500	4850	2.8	2000
	38		48000	4940	2.8	
	41		51000	5030	3.0	
	44		53600	5140	3.1	

NOTES:

(1) *Minimum entering temperature with plain fresh water is 40°F Below 40°F,* ratings are based on the *use* of an *Ethylene Glycol* in *water solution having a freezing point 20°F lower than* the *minimum expected entering temperature* (see chart below).

(2) Ratings are based on heat source fluid flow rate and indoor air SCFM as shown, with air entering the unit at 70°F.



ETHYENE GLYCOL ————————

PROPYLENE GLYCOL — - — - — -

CALCIUM CHLORIDE ------------

FHP

FHP Manufacturing Division
Leigh Products Inc.
601 N.W 65th Court,
Ft Lauderdale, FL 33309

Sept 12, 1983

FHP

"LT" Cooling Performance Ratings
E.W.T. 70°F thru 100°F

MODEL	EWT, °F (2)	FLOW RATE, GPM	COOLING (1) (3) BTU/HR.	WATTS	EER	SCFM
LT30	100	7	23,300	2560	9.1	950
	90		25,000	2430	10.3	
	80		26,700	2300	11.6	
	70		28,300	2155	13.1	
LT40	100	9	30,600	3220	9.5	1200
	90		32,800	3040	10.8	
	80		35,000	2870	12.2	
	70		37,200	2715	13.7	
LT50	100	11	37,000	3980	9.3	1500
	90		39,700	3750	10.6	
	80		42,400	3560	11.9	
	70		45,000	3350	13.4	
LT60	100	13	45,300	4820	9.4	1700
	90		48,500	4580	10.6	
	80		51,800	4320	12.0	
	70		55,000	4070	13.5	
LT70	100	16	54,300	6240	8.7	2000
	90		58,200	5880	9.9	
	80		62,200	5600	11.1	
	70		66,000	5280	12.5	

(1) Based on 80/67 entering air @ rated SCFM as shown.

(2) For earth coupled ground loops these ratings apply to the use of ethylene glycol, propylene glycol and calcium chloride brine solutions of a percentage in water so as to provide a freezing temperature of 15°F.

(3) **For heating** performance at entering water (anti-freeze) temperatures below 45°F, **see** companion **data sheet LT3070-2.**

(4) All above models are (UL) listed.

For LT 50 $\dot{q} = 61921 - 223.93\,T_W - 0.2534\,T_W^2$

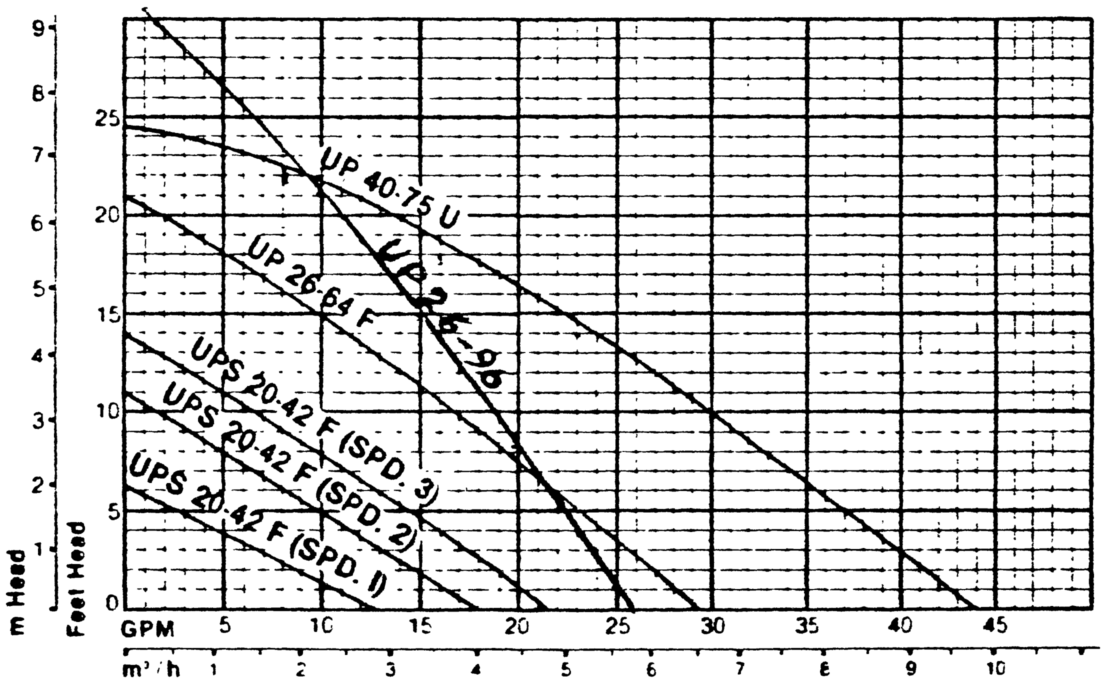
$P = 2247 + 12.29\,T_W + 0.05005\,T_W^2$

FHP

FHP Manufacturing Co.
601 N.W. 65th Court
Ft. Lauderdale, FL 33309
a CHARROW company

DS LT3070-1
June 13, 1984

Performance:



Applications:

Grundfos Domestic Circulators are single-stage, direct-drive, centrifugal pumps, designed primarily for closed system water applications. These pumps can be operated up to system pressures of 142 psi, with fluid temperatures of 230 °F and corresponding ambient temperatures of 68 °F (104 °F for UP 40-75 U). 50% by volume mixtures of water and ethylene or propylene glycol solutions may be pumped. Check with Grundfos for information regarding suitability of other fluids.

APPENDIX C

TRIDIAGONAL MATRIX ARRANGEMENT

TRIDIAGONAL MATRIX ARRANGEMENT

$$
\begin{array}{lr}
D_1T'_1 + A_1T'_2 & = C_1 \\
B_2T'_1 + D_2T'_2 + A_2T'_3 & = C_2 \\
\quad B_3T'_2 + D_3T'_3 + A_3T'_4 & = C_3 \\
\quad\quad \cdot & \cdot \\
\quad\quad\quad \cdot & \cdot \\
\quad\quad\quad\quad \cdot & \cdot \\
\quad\quad\quad\quad\quad \cdot & \cdot \\
\quad\quad\quad\quad\quad\quad \cdot & \cdot \\
B_{13}T'_{12} + D_{13}T'_{13} + A_{13}T'_{14} & = C_{13} \\
B_{14}T'_{13} + D_{14}T'_{14} & = C_{14}
\end{array}
$$

Let $X1 = \dfrac{2h_{eq}r_o\Delta t}{\rho_g(r + \frac{\Delta r}{4})\ \Delta r c_p}$ $\qquad X2 = \dfrac{2k_g(r + \frac{\Delta r}{2})\Delta t}{\rho_g\ (r + \frac{\Delta r}{4})\Delta r c_p}$

$$X3 = \frac{\alpha_g \Delta t^p}{Se(Se + 1)\Delta r}$$

$$C_1 = T_1 + X1\ T'_w,\ D_1 = (X1 + X2 + 1) \qquad A_1 = -X2$$

For N = 1 to 14

$$B_N = -X3\ Se\left(\frac{2}{\Delta r} - \frac{Se}{r}\right),\ D_N = \left[1 + \frac{2\ X3\ (Se + 1)}{\Delta r} + \frac{X3\ (1 - Se^2)}{r}\right]$$

For N = 1 to 13

$$A_N = -X3\ \left(\frac{2}{\Delta r} + \frac{1}{r}\right),\ C_N = T_N$$

For N = 14

$$C_{14} = T_{14} + X3\ \left(\frac{2}{\Delta r} + \frac{1}{r}\right)\ T_{ff}$$

APPENDIX D

COMPUTER CODES

UTI.FOR

```
C     HEAT TRANSFER AND TEMPERATURE DISTRIBUTION IN U-TUBE
C     GROUND COUPLINGS - IMPLICIT FORMULATION
      DIMENSION C(15),R(15),RI(15),A(15),B(15),D(15)
C    TEMPERATURE PROFILE AT BEGINNING OF SIMULATION
      TW=62.0
      TWO=62.0
      DATA C/15*62.0/
      TWA=(TWO+TW)/2.0
      WRITE(7,9)
    9 FORMAT(2X,'TIME(MIN)',2X,'IN TEMP',3X,'AVG TEMP',2X,'OUT
      NR=15
      NRM1=NR 1
C     PIPE DATA   O.D. & I.D. (INCHES),THERM. COND.,SPEC.HEAT,
C     DENSITY AND TOTAL LENGTH (FT.)
      DATA PID,POD,PKH,CPP,DNP,PL/1.61,1.824,0.13,0.4,60.0,100
C    CONVECTIVE HEAT TRANSFER COEFFICIENT (BTU/HR-SQ.FT.-F)
      HH=80.0
C    VALUE USED TO CALCULATE FREE CONVECTION COFF. IN WATER -
C    PROPERTY OF WATER IS THE GRASHOF NO. DIVIDED BY BOUNDARY
C    TEMPERATURE DIFFERENCE DIVIDED BY LENGTH CUBED - USE 70E6
C    COOLING MODE (85 F) AND 15E6 FOR HEATING (45 F)
      DATA FGR/70.E6/
C    TIME STEP IN SECONDS
      DT=300.0
C    DIAMETER OF BORE HOLE IN INCHES
      DB=4.5
C    NUMBER OF U TUBES (SINGLE   UT=1.0, DOUBLE - UT=2.0)
      UT=1.0
C    CALCULATION OF AVERAGE SEPERATION DISTANCE, INCHES
      SD=(0.5*DB)-POD
C    EQUIVALENT DIAMETER OF U TUBE
      DEQ=SQRT(2.0*UT)*POD+SD
C   EQUIVALENT RADIUS, FT.
      R(1)=DEQ/24.0
C     GROUND DATA
      DATA GKH,CPG,DNG,DR,SE/1.4,0.45,115.0,0.0208333,1.5/
      DRE=(POD-PID)/24.0
      RO=R(2)-DRP
C     NUMBER OF LOOPS (FOR PARALLEL SYSTEMS)
      CLS=6.0
C    CALCULATION OF SHORT CIRCUIT HEAT TRANSFER COEFFICIENT
      ASC=1.571*(POD/12.0)*PL
      ASCI=1.571*(PID/12.0)*PL
      RH=2.0/(HH*ASCI)
      RW=(POD/12.0)*ALOG(POD/PID)/(PKH*ASC)
      RS=(0.01708*POD+SD/12.0)/(GKH*ASC)
      RT=RH+RW+RS
      HSC=UT/RT
C    CORRECTION FACTOR FOR EQUIVALENT HEAT TRANSFER COEFF. DUE TO
C    NON UNIFORM HEAT FLOW VARIATION WITH CIRCUMFERENTIAL ANGLE
      CFH=0.7*UT+1.0
C    AVERAGE INDOOR AIR TEMPERATURES
      TWD=66.0
      TON=72.0
C    RUN FRACTIONS - ONL=TOTAL ITERATIONS PER CYCLE, KN=ON ITERATIONS
C    NIGHT, KM=ON ITERATIONS IN MORN., KA=ON ITERATIONS IN AFTERNOON,
C    KL=ON ITERATIONS IN EVENING - ALL NUMBERS ARE MULTIPLIED BY DT
C    TO GET AMOUNT OF TIME UNIT IS ON OR OFF
      DATA ONL,KN,KM,KA,KL/12,6,6,6,6/
C    NUMBER OF DO LOOP ITERATIONS FOR TIME STEP
      NDLI=8064
C    SET UP FOR PRNT OUT OF LAST ON-OFF CYCLE
      NPLC=NDLI ONE
C     PRINT FREQUENCY FOR HT.TRANS.RATES AND GROUND TEMP.DISTRIBUTION
      NPT=864
C     PRINT FREQUENCY OF WATER TEMPS. (MULTIPLY NFP BY DT FOR PRINT TI
      NPF=864
C    WATER DATA   DENSITY,SP. HEAT, TOTAL WATER FLOW, THERM. COND.
      DATA DNW,CPW,GPMT,WKH/62.4,1.0,10.2,0.36/
      GPM=GPMT/CLS
C    CONVERT GAL/MIN INTO FT**3/SEC
      VDOT=GPM/448.8
      PK=PKH/3600.0
      GK=GKH/3600.0
      WK=WKH/3600.0
      DTK=DTKH/3600.0
      ALP=PK/(DNP*CPP)
      ALG=GK/(DNG*CPG)
      ALW=WK/(DNW*CPW)
      AR=3.1416*PID/12.0
      ARI=3.1416*DTOD/12.0
      ARO=3.1416*POD/12.0
      AREQ=3.1416*(POD/12.0)*UT*2.0
      VOL=0.7854*(PID/12.0)**2.0*PL*2.0*UT
      LOVL=0.7854*(DEQ/12)**2.0*PL
      VEL=VDOT/(0.7854*UT*(PID/12.0)**2.0)
C    TIME REQ'D FOR WATER TO CIRCULATE THRU LOOP ONCE
      CT=2.0*PL/VEL
C    FIRST APPROXIMATION FOR FREE CONV. BOUNDARY LAYER TEMP. DIFFEREN
      DTBL=80.0/(HH*AREQ)
      IL=1
      WRITE(6,20)PID,POD,PKH,PL
   20 FORMAT(1X,'PIPE DATA: ID=',F8.3,' IN. OD=',F8.3,' IN K=',F8.3,
     &' BTU/HR-FT-F L=',F8.3,/)
      WRITE(6,21)DT
   21 FORMAT(1X,'TIME STEP =',F8.3,' SEC')
```

```
      TI=0.0
      TH=0.0
      JT=1
      TJT=DT
      TM=1.0
      ITD= 5
      IPT= 5
C     VALUES USED TO DETERMINE WATER TEMPS. AT START OF EACH ON CY
      NL=CT/DT
      REM=(CT-NL*DT)/DT
      IF(REM.LT.0.5)GO TO 45
      NL=NL+1
   45 NLP1=NL+1
C     FLOW CORRECTIONS FACTORS FOR HEAT PUMP CAPACITY & POWER*
      CFP=1.211-0.03928*GPMT+0.001611*GPMT**2.0
      CFC=0.9625+0.00366*GPMT
      CAC=0.98
      CAP=0.99
      CWBC=-0.5125+0.03141*TWB-0.0001321*TWB**2.0
      CWBP=0.4101+0.012*TWB-0.0000487*TWB**2.0
      DRG=DR
      RI(2)=R(2)*12.0
      DO 161 LR=2,NRM1
      DRG=DRG*SE
      R(LR+1)=R(LR)+DRG
      RI(LR+1)=R(LR+1)*12.0
  161 CONTINUE
C     BEGIN TIME STEP DO LOOP ITERATION
      DO 100 IT=1,NDLI
      ITD=ITD+1
      IPT=IPT+1
      IF(TM.EQ.0.0)IPT=1
      TI=TI+DT/60.0
      TH=TH+DT/3600.0
      IF(TH.GT.24.0)TH=0.0
      TM=TM+DT/60.0
      IF(TH.LE.6.0)ON=KA
      IF(TH.GT.6.0.AND.TH.LE.12.0)ON=KF
      IF(TH.GT.12.0.AND.TH.LE.18.0)ON=KN
      IF(TH.GT.18.0)ON=KM
      OFF=ONF-ON
      IF(JT.GT.ONF)GO TO 35
      IF(JT.GT.ON)GO TO 70
      GO TO 59
   35 JT=1
      TJT=DT
   59 H=HH/3600.0
      FSC=1.0
C     POWER INPUT TO WATER PUMP IN WATTS
      PP=375.0
C     CAPACITY(BTUH) & POWER(WATTS) CURVES FOR HEAT PUMP
      PC=2180.0+12.292*TWO+.05005*TWO**2.0
      PW=PP+PC*CFP*CAP*CWBP
      QREJ=21820.+581.*TWO+1.983*TWO**2.-0.05828*TWO**3.
      QTEC=QREJ*CFC*CWBC
      QCAP=QTEC-PW*3.412
      QHP=QTEC/(CLS*3600.0)
      EER=ABS(QCAP*CFC/PW)
      GO TO 60
C     FREE CONVECTION COEFFICIENT (BTU/S-SQ.FT.-F)
   70 HFH=0.082*WKH*(FGR*DTBL)**(1.0/3.0)
      H=HFH/3600.0
      QHP=0.0
      EER=0.0
      QTEC=0.0
      QCAP=0.0
      PW=0.0
      FSC=0.0
   60 RS1=POD/(PID*H)+R(2)*ALOG(POD/PID)/PK
      HFO=CFH/RS1
      X1=2.0*HFO*(RO+DRP)*DT/(DNG*(RO+.25*DR)*DR*CPG)
      X2=2.0*GK*(RO+0.5*DRP)*DT/(DNG*(RO+.25*DR)*CPG*DR**2.0)
      B(1)=X1+X2+1.0
      A(1)=-1.0*X2
      C(1)=C(1)+X1*TWA
      DRG=DR
      DO 120 L=2,NRM1
      X=ALG*DT/(SE*(SE+1.0)*DRG)
      B(L)=-1.0*X*SE*((2.0/DRG)-SE/R(L))
      D(L)=1.0+2.0*X*(SE+1.0)/DRG+X*(1-SE**2.0)/R(L)
      A(L)=-1.0*X*(2.0/DRG+1.0/R(L))
      DRG=DRG*SE
  120 CONTINUE
      C(NRM1)=C(NRM1)-A(NRM1)*C(NR)
      DO 1 I=2,NRM1
      P=B(I)/D(I-1)
      D(I)=D(I)-P*A(I-1)
    1 C(I)=C(I)-P*C(I-1)
C     BACK SUBSTITUTION
      C(NRM1)=C(NRM1)/D(NRM1)
      DO 2 I=2,NRM1
      J=NRM1-I+1
    2 C(J)=(C(J)-A(J)*C(J+1))/D(J)
      IF(IL.EQ.1) GO TO 105
      GO TO 80
  105 IL=IL+1
```

```
      CTM=CT/60.0
      WRITE(6,106)CTM
  106 FORMAT(1X,'LOOP CIRC. TIME=',F7.2,' MINUTES')
      HHEQ=HEQ*3600.0
      WRITE(6,112)DEQ,HHEQ
  112 FORMAT(1X,'EQ.DIA.=',F7.3,' IN.  EQ.HT.TR.COEF.=',F7.2,' BTU/
     @FT F',/)
      WRITE(6,115)
  115 FORMAT(1X,'DISTANCE FROM CENTER OF PIPE IN INCHES')
      WRITE(6,110)(RI(M),M=3,NRM1)
  110 FORMAT(3X,'WATER',2X,'POD',2X,13F7.2)
      WRITE(6,111)TWA,(C(I2),I2=1,NRM1)
  111 FORMAT(1X,15F7.2,/)
   80 QR=HEQ*AREQ*(C(1)-TWA)*3600.0*PL
      QSC=HSC*0.5*(TW-TWO)*FSC
      Q=QR+QSC
      QS=Q/3600.0
      DTHL=ABS(QS/(H*AREQ*PL))
C     IF HEAT PUMP IS RUNNING GO TO 81
      IF(JT.LE.ON)GO TO 81
C    WATER TEMP UPDATES WHEN UNIT IS OFF
      DTNC=(QS*DT)/(DNW*CPW*EQVL)
      TWA=TWA+DTNC
      TWO=TWO+DTNC
      TW=TW+DTNC
      GO TO 75
C    WATER TEMP UPDATE WHEN UNIT  IS  ON
   81 IF(NL.EQ.0)GO TO 95
      IF(JT.EQ.1)GO TO 94
      IF(JT.GT.NL)GO TO 95
      GO TO 97
C    UPDATE WHEN ON TIME < LOOP CIRCULATION TIME
   94 QHP=((DT-30.0)/DT)*QHP
      TAR=TWA
   97 TWO=TAR
      TW=TWO+QHP/(DNW*VDOT*CPW)
      TWA=((TAR*(NL-JT)+(TW*JT))/NL)+(QS/(DNW*VDOT*CPW))
      GO TO 75
C    UPDATE WHEN ON TIME > LOOP CIRCULATION TIME
   95 TW=TWO+QHP/(DNW*VDOT*CPW)
      TWO=TW+QS/(DNW*VDOT*CPW)
      IF(JT.EQ.NLP1)TWO=TAR
   96 TWA=(TW+TWO)/2.0
   75 IF(IPT.EQ.NPT)GO TO 51
      IF(IT.GT.NPLC)GO TO 51
      GO TO 300
   51 COP=EER/3.412
   65 FORMAT(1X,'TIME=',F10.2,' MIN. Q=',F10.2,' BTUH QSC=',F10.2,
     @ LOOP TIME=',F7.2,' MIN.',/,11F7.2)
      TJTM=TJT/60.0
      WRITE(6,65)IT,Q,QSC,TJTM,TWA,(C(II),II=1,10)
      WRITE(6,55)PW,QCAP,QTEC,EER,COP
   55 FORMAT(1X,' POWER =',F8.1,' W  QHP=',F10.1,' BTUH QTEC=
     @' BTUH  EER=',F7.2,'  COP=',F7.2,/)
  299 TM=0.0
  300 JT=JT+1
      TJT=TJT+DT
      IF(ITD.EQ.NPF)GO TO 98
      IF(IT.GT.NPLC)GO TO 98
      GO TO 100
   98 WRITE(9,99)IT,TW,TWA,TWO
   99 FORMAT(1X,4F10.2)
      ITD=0
  100 CONTINUE
      WRITE(8,101)TW,TWO,TWA,(C(NH),NH=1,NR)
  101 FORMAT(1X,2F7.2,/,1X,15F7.2)
      STOP
      END
```

TIME(MIN)	IN TEMP	AVG TEMP	OUT TEMP
4345.00	89.07	84.59	80.11
8665.00	90.30	85.84	81.38
12985.00	90.99	86.55	82.10
17305.00	91.47	87.03	82.60
21625.00	91.83	87.40	82.97
25945.00	92.12	87.70	83.28
30265.00	92.37	87.95	83.53
34585.00	92.57	88.16	83.75
38905.00	92.76	88.35	83.94
40265.00	90.26	82.46	81.27
40270.00	91.08	86.35	81.27
40275.00	91.08	86.18	81.27
40280.00	91.08	87.09	83.09
40285.00	92.81	88.40	83.99
40290.00	93.65	88.72	83.79
40295.00	91.95	87.02	82.09
40300.00	90.28	85.35	80.42
40305.00	88.99	84.06	79.13
40310.00	87.90	82.97	78.04
40315.00	86.99	82.06	77.13
40320.00	86.20	81.27	76.34

```
PIPE DATA: ID=   1.610 IN. OD=   1.824 IN K=   0.130 BTU/HR-FT-F L= 100.000

TIME STEP = 300.000 SEC
LOOP CIRC. TIME=  12.44 MINUTES
EQ.DIA.=  3.006 IN.  EQ.HT.TR.COEF.=  12.65 BTU/H-SQ FT-F

DISTANCE FROM CENTER OF PIPE IN INCHES
  WATER  POD     1.88    2.44    3.28    4.55    6.45    9.30   13.57   19.97   29.59   44.00   65.63   98.06
 62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00   62.00

TIME=   4345.00 MIN. Q=  -7620.83 BTUH QSC=    434.64 BTUH  LOOP TIME=  25.00 MIN.
 84.59  76.56  75.29  73.87  72.39  70.93  69.44  67.87  66.30  64.82  63.52
 POWER =  3697.8 W  QHP=   37939.3 BTUH QTEC=   50556.1 BTUH  EER=  10.26  COP=   3.01

TIME=   8665.00 MIN. Q=  -7589.11 BTUH QSC=    432.91 BTUH  LOOP TIME=  25.00 MIN.
 85.84  77.85  76.60  75.18  73.71  72.25  70.76  69.18  67.58  66.02  64.57
 POWER =  3722.8 W  QHP=   37579.8 BTUH QTEC=   50281.9 BTUH  EER=  10.09  COP=   2.96

TIME=  12985.00 MIN. Q=  -7567.91 BTUH QSC=    431.70 BTUH  LOOP TIME=  25.00 MIN.
 86.55  78.59  77.33  75.92  74.45  72.99  71.50  69.93  68.32  66.74  65.24
 POWER =  3736.9 W  QHP=   37359.7 BTUH QTEC=   50110.2 BTUH  EER=  10.00  COP=   2.93

TIME=  17305.00 MIN. Q=  -7551.95 BTUH QSC=    430.77 BTUH  LOOP TIME=  25.00 MIN.
 87.03  79.09  77.84  76.44  74.97  73.51  72.03  70.45  68.84  67.25  65.72
 POWER =  3746.8 W  QHP=   37199.9 BTUH QTEC=   49983.9 BTUH  EER=   9.93  COP=   2.91

TIME=  21625.00 MIN. Q=  -7539.12 BTUH QSC=    430.02 BTUH  LOOP TIME=  25.00 MIN.
 87.40  79.48  78.23  76.83  75.36  73.91  72.42  70.85  69.24  67.65  66.10
 POWER =  3754.3 W  QHP=   37074.1 BTUH QTEC=   49883.7 BTUH  EER=   9.87  COP=   2.89

TIME=  25945.00 MIN. Q=  -7528.35 BTUH QSC=    429.39 BTUH  LOOP TIME=  25.00 MIN.
 87.70  79.79  78.55  77.14  75.68  74.23  72.75  71.17  69.56  67.97  66.41
 POWER =  3760.3 W  QHP=   36970.3 BTUH QTEC=   49800.6 BTUH  EER=   9.83  COP=   2.88

TIME=  30265.00 MIN. Q=  -7519.10 BTUH QSC=    428.84 BTUH  LOOP TIME=  25.00 MIN.
 87.95  80.05  78.81  77.41  75.94  74.50  73.01  71.44  69.84  68.23  66.68
 POWER =  3765.4 W  QHP=   36881.8 BTUH QTEC=   49729.3 BTUH  EER=   9.79  COP=   2.87

TIME=  34585.00 MIN. Q=  -7510.95 BTUH QSC=    428.36 BTUH  LOOP TIME=  25.00 MIN.
 88.16  80.27  79.03  77.63  76.17  74.72  73.24  71.68  70.07  68.47  66.90
 POWER =  3769.7 W  QHP=   36804.8 BTUH QTEC=   49667.0 BTUH  EER=   9.76  COP=   2.86

TIME=  38905.00 MIN. Q=  -7503.70 BTUH QSC=    427.94 BTUH  LOOP TIME=  25.00 MIN.
 88.35  80.47  79.23  77.83  76.37  74.92  73.44  71.88  70.27  68.67  67.10
 POWER =  3773.5 W  QHP=   36736.5 BTUH QTEC=   49611.7 BTUH  EER=   9.73  COP=   2.85

TIME=  40265.00 MIN. Q=  -2815.80 BTUH QSC=    528.11 BTUH  LOOP TIME=   5.00 MIN.
 82.46  78.50  77.95  77.23  76.27  75.02  73.53  71.94  70.33  68.73  67.16
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```
POWER =  3643.4 W  QHP=   38591.6 BTUH QTEC=   51022.8 BTUH  EER=  10.59  COP=   3.10

TIME=  40270.00 MIN. Q=  -4024.22 BTUH QSC=    481.80 BTUH  LOOP TIME=  10.00 MIN.
 86.35  78.73  78.02  77.19  76.20  74.98  73.52  71.94  70.33  68.73  67.16
 POWER =  3738.7 W  QHP=   37331.0 BTUH QTEC=   50087.6 BTUH  EER=   9.98  COP=   2.93

TIME=  40275.00 MIN. Q=  -7362.97 BTUH QSC=    525.52 BTUH  LOOP TIME=  15.00 MIN.
 86.18  79.82  78.65  77.45  76.24  74.96  73.51  71.94  70.33  68.73  67.16
 POWER =  3738.7 W  QHP=   37331.0 BTUH QTEC=   50087.6 BTUH  EER=   9.98  COP=   2.93

TIME=  40280.00 MIN. Q=  -6795.77 BTUH QSC=    525.52 BTUH  LOOP TIME=  20.00 MIN.
 87.09  80.12  78.96  77.66  76.32  74.96  73.51  71.94  70.33  68.73  67.16
 POWER =  3738.7 W  QHP=   37331.0 BTUH QTEC=   50087.6 BTUH  EER=   9.98  COP=   2.93

TIME=  40285.00 MIN. Q=  -7501.54 BTUH QSC=    427.81 BTUH  LOOP TIME=  25.00 MIN.
 88.40  80.52  79.28  77.89  76.43  74.98  73.50  71.94  70.33  68.73  67.16
 POWER =  3774.6 W  QHP=   36716.3 BTUH QTEC=   49595.3 BTUH  EER=   9.73  COP=   2.85

TIME=  40290.00 MIN. Q=  -8388.89 BTUH QSC=    472.24 BTUH  LOOP TIME=  30.00 MIN.
 88.72  81.07  79.69  78.15  76.56  75.02  73.51  71.94  70.33  68.73  67.16
 POWER =  3792.4 W  QHP=   36383.2 BTUH QTEC=   49322.8 BTUH  EER=   9.59  COP=   2.81

TIME=  40295.00 MIN. Q=  -6280.51 BTUH QSC=      0.00 BTUH  LOOP TIME=  35.00 MIN.
 87.02  80.58  79.52  78.18  76.64  75.06  73.51  71.94  70.33  68.73  67.16
 POWER =     0.0 W  QHP=       0.0 BTUH QTEC=       0.0 BTUH  EER=   0.00  COP=   0.00

TIME=  40300.00 MIN. Q=  -6160.59 BTUH QSC=      0.00 BTUH  LOOP TIME=  40.00 MIN.
 85.35  80.43  79.42  78.16  76.68  75.09  73.52  71.94  70.33  68.73  67.16
 POWER =     0.0 W  QHP=       0.0 BTUH QTEC=       0.0 BTUH  EER=   0.00  COP=   0.00

TIME=  40305.00 MIN. Q=  -4770.18 BTUH QSC=      0.00 BTUH  LOOP TIME=  45.00 MIN.
 84.06  79.93  79.12  78.02  76.66  75.11  73.53  71.94  70.33  68.73  67.16
 POWER =     0.0 W  QHP=       0.0 BTUH QTEC=       0.0 BTUH  EER=   0.00  COP=   0.00

TIME=  40310.00 MIN. Q=  -4000.75 BTUH QSC=      0.00 BTUH  LOOP TIME=  50.00 MIN.
 82.97  79.47  78.78  77.84  76.60  75.12  73.53  71.94  70.33  68.73  67.16
 POWER =     0.0 W  QHP=       0.0 BTUH QTEC=       0.0 BTUH  EER=   0.00  COP=   0.00

TIME=  40315.00 MIN. Q=  -3377.76 BTUH QSC=      0.00 BTUH  LOOP TIME=  55.00 MIN.
 82.06  79.03  78.44  77.62  76.50  75.10  73.54  71.94  70.33  68.73  67.16
 POWER =     0.0 W  QHP=       0.0 BTUH QTEC=       0.0 BTUH  EER=   0.00  COP=   0.00

TIME=  40320.00 MIN. Q=  -2900.20 BTUH QSC=      0.00 BTUH  LOOP TIME=  60.00 MIN.
 81.27  78.62  78.12  77.39  76.38  75.07  73.54  71.94  70.33  68.73  67.16
 POWER =     0.0 W  QHP=       0.0 BTUH QTEC=       0.0 BTUH  EER=   0.00  COP=   0.00
```

CHI.FOR

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C      HEAT TRANSFER AND TEMPERATURE DISTRIBUTION IN CONCENTRIC
C      GROUND COUPLINGS - IMPLICIT FORMULATION
       DIMENSION C(15),R(15),RI(15),A(15),B(15),D(15),TS(20)
C     TEMPERATURE PROFILE AT BEGINNING OF SIMULATION
       OPEN(5,FILE='SW111D.DAT',STATUS='OLD')
       READ(5,131)TW,TWO,TWA,(C(NJ),NJ=1,15)
  131 FORMAT(1X,2F7.2,/,1X,16F7.2)
       WRITE(9,7)
    9 FORMAT(2X,'TIME(MIN)',2X,'IN TEMP',3X,'AVG TEMP',2X,'OUT T
       NR=15
       NRM1=NR-1
C      PIPE DATA - O.D. & I.D. (INCHES),THERM. COND.,SPEC.HEAT,
C      DENSITY AND TOTAL LENGTH (FT.)
       DATA PID,POD,PKH,CPP,DNP,PL/3.166,3.5,0.226,0.4,60.0,100.0,
C    CONVECTIVE HEAT TRANSFER COEFFICIENT (BTU/HR-SQ.FT.-F)
       HH=40.0
C     COEFFICIENCT USED TO CALCULATE GRASHOF NO. USED IN DETERMIN
C     OF FREE CONVECTION COEFFICIENTS - USE 70.C6 COOLING, 15C6 H
C     UNITS ARE 1/DEG F*CU.FT.
       DATA FGR//0.0L6/
C     TIME STEP IN SECONDS
       DT=300.0
C     DIP TUBE DATA - IN. & OUT.DIA IN INCHES, TH.  COND IN BTU/H
       DATA DTID,DTOD,DTKH/0.740,1.05,0.08/
C     EQUIVALENT DIAMETEROF U-TUBE
       DEQ=POD
C    EQUIVALENT RADIUS, FT.
       R(2)=DEQ/24.0
C      GROUND DATA
       DATA GKH,CPG,DNG,DR,SE/1.4,0.45,115.0,0.0208333,1.5/
       DRP=(POD-PID)/24.0
       RO=R(2) (POD-PID)/24.0
C      NUMBER OF LOOPS (FOR PARALLEL SYSTEMS)
       CLS=6.0
C     CALCULATION OF SHORT CIRCUIT HEAT TRANS. COEFFICIENT (BTU/H
       RDI=1.0/HH+(DTOD/24.0)*ALOG(DTOD/DTID)/DTKH
       ASC=3.1416*(DTOD/12.0)*PL
       HSC=ASC/RDT
C     AVERAGE INDOOR AIR TEMPERATURES
       TWR=66.0
       TDR=72.0
C     RUN FRACTIONS - ONE=TOTAL ITERATIONS PER CYCLE, KN=ON ITERA
C     NIGHT, KM=ON ITERATIONS IN MORN., KA=ON ITERATIONS IN AFTER
C     KE=ON ITERATIONS IN EVENING - ALL NUMBERS ARE MULTIPLIED BY
C     TO GET AMOUNT OF TIME UNIT IS ON OR OFF
       DATA ONE,KN,KM,KA,KE/12,4,8,12,6/
C     NUMBER OF DO LOOP ITERATIONS FOR TIME STEP
       NDLI=4032
C     SET UP FOR PRNT OUT OF LAST ON-OFF CYCLE
       NPLC=NDLI-ONE
C      PRINT FREQUENCY FOR HT.TRANS.RATES AND GROUND TEMP.DISTRIBUT
       NPT=864
C      PRINT FREQUENCY OF WATER TEMPS. (MULTIPLY NFP BY DT FOR PRIN
       NPF=864
C      WATER DATA
       DATA DNW,CPW,GPMT,WKH/62.4,1.0,10.2,0.36/
       GPM=GPMT/CLS
C      CONVERT GAL/MIN INTO FT**3/SEC
       VDOT=GPM/448.8
       PK=PKH/3600.0
       GK=GKH/3600.0
       WK=WKH/3600.0
       DTK=DTKH/3600.0
       ALP=PK/(DNP*CPP)
       ALG=GK/(DNG*CPG)
       ALW=WK/(DNW*CPW)
       AR=3.1416*PID/12.0
       ARI=3.1416*DTOD/12.0
       ARO=3.1416*POD/12.0
       AREQ=3.1416*DEQ/12.0
C     FIRST APPROXIAMATION FOR BOUND. LAYER TEMP. DIFF DUE TO FREE
       RDIFL=80.0/(HH*AREQ)
       VOL=0.7854*(PID/12.0)**2.0*PL
       VEL=VDOT/(0.7854*(PID**2.0-DTOD**2.0)/144.0)
       VELD=VDOT/(0.7854*(DTID/12.0)**2.0)
C     TIME  REQUIRED FOR WATER TO CIRCLATE THRU LOOP ONCE
       CT=PL/VEL+PL/VELD
       IL=1
       WRITE(6,20)PID,POD,PKH,PL
   20 FORMAT(1X,'PIPE DATA: ID=',F8.3,' IN. OD=',F8.3,' IN K=',F8.
     @' BTU/HR-FT-F L=',F8.3,/)
       WRITE(6,21)DT
   21 FORMAT(1X,'TIME STEP =',F8.3,' SEC')
       TT=0.0
       TH=0.0
       JI=1
       TJT=1.5*DT
       TM=1.0
       ITD=-8
       IPT=-8
C    VALUES USED TO CALCULATE WATER TEMPS. AT START OF ON CYCLES
       NL=CT/DT
       REM=(CT-NL*DT)/DT
       IF(REM.LT.0.5)GO TO 45
       NL=NL+1
   45 DO 88 LX=1,NL
```

```
      TS(LX)=TWA
   80 CONTINUE
      TS(NL+1)=TWA
C     FLOW CORRECTION FACTORS FOR HEAT PUMP CAPACITY & POWER
      CFP=1.211-0.03920*GPMT+0.001611*GPMT**2.0
      CFC=0.9625+0.00366*GPMT
      CAC=0.98
      CAP=0.99
      CWBC=-0.5125+0.03141*TWB-0.0001321*TWB**2.0
      CWBP=0.410+0.012*TWB-0.0000487*TWB**2.0
      DRG=DR
      R((2)=R(2)*12.0
      DO 161 LR=2,NRM1
      DRG=DRG*SE
      R(LR+1)=R(LR)+DRG
      RI(LR+1)=R(LR+1)*12.0
  161 CONTINUE
C     BEGIN TIME STEP DO LOOP ITERATION
      DO 100 IT=1,NDLI
      ITD=ITD+1
      IPT=IPT+1
      IF(TM.EQ.0.0)IPT=1
      TI=TI+DT/60.0
      TH=TH+DT/3600.0
      IF(TH.GT.24.0)TH=0.0
      TM=TM+DT/60.0
      IF(TH.LE.6.0)ON=KA
      IF(TH.GT.6.0.AND.TH.LE.12.0)ON=KE
      IF(TH.GT.12.0.AND.TH.LE.18.0)ON=KN
      IF(TH.GT.18.0)ON=KM
      OFF=ONF-ON
      IF(JT.GT.ONF)GO TO 35
      IF(JT.GT.ON)GO TO 70
      GO TO 59
   35 JT=1
      TJT=DT/2.0
   59 H=HH/3600.0
      FSC=1.0
C     POWER INPUT TO WATER PUMP IN WATTS
      PP=375.0
C     CAPACITY(BTUH) & POWER(WATTS) CURVES FOR HEAT PUMP
      PC=2100.0+12.292*TWO+.05005*TWO**2.0
      PW=PP+PC*CFP*CAP*CWBP
      QRCJ=21820.+501.*TWO+1.903*TWO**2.-0.05020*TWO**3.
      QTEC=QRCJ*CFC*CWBC
      QCAP=QTEC-PW*3.412
      QHP=QTEC/(CLS*3600.0)
      ECR=ABS(QCAP*CFC/PW)
      GO TO 60
C     FREE CONVECTION COEFFICIENT (BTU/H-SQ.FT.-F)
   70 HFH=30.0
      H=HFH/3600.0
      QHP=0.0
      ECR=0.0
      QTEC=0.0
      QCAP=0.0
      PW=0.0
      FSC=0.0
   60 RS1=POD/(PID*H)+R(2)*ALOG(POD/PID)/PK
      HEQ=1.0/RS1
      X1=2.0*HEQ*(RO+DRP)*DT/(DNG*(RO+.25*DR)*DR*CPG)
      X2=2.0*GK*(RO+0.5*DRP)*DT/(DNG*(RO+.25*DR)*CPG*DR**2.0)
      B(1)=X1+X2+1.0
      A(1)=-1.0*X2
      C(1)=C(1)+X1*TWA
      DRG=DR
      DO 120 L=2,NRM1
      X=ALG*DT/(SE*(SE+1.0)*DRG)
      B(L)= 1.0*X*SE*((2.0/DRG)-SE/R(L))
      B(L)=1.0+2.0*X*(SE+1.0)/DRG+X*(1-SE**2.0)/R(L)
      A(L)=-1.0*X*(2.0/DRG+1.0/R(L))
      DRG=DRG*SE
  120 CONTINUE
      C(NRM1)=C(NRM1)-A(NRM1)*C(NR)
      DO 1 I=2,NRM1
      P=B(I)/D(I-1)
      D(I)=D(I)-P*A(I-1)
    1 C(I)=C(I)-P*C(I-1)
C     BACK SUBSTITUTION
      C(NRM1)=C(NRM1)/D(NRM1)
      DO 2 I=2,NRM1
      J=NRM1-I+1
    2 C(J)=(C(J)-A(J)*C(J+1))/D(J)
      IF(IL.EQ.1) GO TO 105
      GO TO 80
  105 IL=IL+1
      CTM=CT/60.0
      WRITE(6,106)CTM
  106 FORMAT(1X,'LOOP CIRC. TIME=',F7.2,' MINUTES')
      HHEQ=HEQ*3600.0
      WRITE(6,112)DEQ,HHEQ
  112 FORMAT(1X,'EQ.DIA.=',F7.3,' IN.  EQ.HT.TR.COEF.=',F7.2,'
     *FT-F',/)
      WRITE(6,115)
  115 FORMAT(1X,'DISTANCE FROM CENTER OF PIPE IN INCHES')
      WRITE(6,110)(RI(M),M=3,NRM1)
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  110 FORMAT(3X,'WATER',2X,'POD',2X,13F7.2)
      WRITE(6,111)TWA,(C(I2),I2=1,NRM1)
  111 FORMAT(1X,15F7.2,/)
   80 QR=HEQ*AREQ*(C(1)-TWA)*3600.0*PL
      QSC=HSC*0.5*(TW-TWO)*FSC
      Q=QR+QSC
      QS=Q/3600.0
      DTRL=ABS(QS/(H*AREQ*PL))
      DTPW=ABS(C(1)-TWA)-DTRL
C     IF HEAT PUMP IS RUNNING GO TO 81
C    WATER TEMP UPDATE WHEN UNIT IS OFF
      IF(JT.LE.ON)GO TO 81
C    WATER TEMP UPDATED WHEN UNIT IS OFF
      DTNC=(QS*DT)/(DNW*CPW*VOL)
      TWA=TWA+DTNC
      TWO=TWO+DTNC
      TW=TW+DTNC
      TS(1)=TWO
      TS(NL+1)=TW
      DO 87 L1=2,NL
      TS(L1)=TS(1)+(TS(NL+1)-TS(1))*(L1-1)*DT/CT
   87 CONTINUE
      GO TO 75
C    WATER TEMP UPDATE WHEN UNIT  IS  ON
   81 IF(CT.GT.TJT)GO TO 85
      TWO=TW+QS/(DNW*VDOT*CPW)
      TW=TWO+QHP/(DNW*VDOT*CPW)
      TWA=(TW+TWO)/2.0
      GO TO 75
   85 DTQ=QS/(DNW*VDOT*CPW)
      TWO=((TS(JT)+TS(JT+1))/2.0)+DTQ
      DO 71 NQ=JT,NL
   71 TS(NQ)=TS(NQ)+DTQ
      TW=TWO+QHP/(DNW*VDOT*CPW)
      TWA=(TW+TWO)/2.0
   75 IF(IPT.EQ.NPT)GO TO 51
      IF(IT.GT.NPLC)GO TO 51
      GO TO 300
   51 COP=EER/3.412
   65 FORMAT(1X,'TIME=',F10.2,' MIN. Q=',F10.2,' BTUH QSC=',F10.2,' BTUH
     @ LOOP TIME =',F7.2,/,11F7.2)
       TJTM=TJT/60.0
      WRITE(6,65)TI,Q,QSC,TJTM,TWA,(C(I1),I1=1,10)
      WRITE(6,68)DTRL,DTPW
   68 FORMAT(1X,'DT FILM=',F7.2,' DEG. F  DT PIPE=',F7.2,' DEG. F')
      WRITE(6,55)PW,QCAP,QTEC,EER,COP
   55 FORMAT(1X,' POWER =',F8.1,' W  QHP=',F10.1,' BTUH QTEC=',F10.1,
     @' BTUH  EER=',F7.2,'  COP=',F7.2,/)
  299 TM=0.0
  300 JT=JT+1
      TJT=TJT+DT
      IF(ITD.EQ.NPF)GO TO 98
      IF(IT.GT.NPLC)GO TO 98
      GO TO 100
   98 WRITE(9,99)TI,TW,TWA,TWO
   99 FORMAT(1X,4F10.2)
      ITD=0
  100 CONTINUE
      WRITE(8,101)TW,TWO,TWA,(C(NH),NH=1,NR)
  101 FORMAT(1X,2F7.2,/,1X,16F7.2)
      STOP
      END
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CVHI.FOR

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C      HEAT TRANSFER AND TEMPERATURE DISTRIBUTION IN CONCENTRIC
C      GROUND COUPLINGS - IMPLICIT FORMULATION - MULTIPLE VERTICAL NON
       DIMENSION C(10,15),TW(10),TIW(10),R(15),RI(15),Q(10),QT(10),
      &POSC(10),A(10,15),B(10,15),D(10,15)
C      NUMBER OF VERTICAL INCREMENTS-MUST AGREE WITH DIMENSION T(IP,NR
C      TW(IP)
       DATA TW/10*62.0/
       DATA TIW/10*62.0/
       DATA C/150*62.0/
       WRITE(9,9)
     9 FORMAT(2X,'TIME(MIN)',2X,'IN TEMP',3X,'MID TEMP',2X,'OUT TEMP')
       NR=15
       NRM1=NR-1
       IP=5
       MP=IP/2+1
       IP1=IP-1
       DO 169 NM=1,NR
       C(IP,NM)=67.0
       C(IP1,NM)=62.0
   169 CONTINUE
C      PIPE DATA - O.D. & I.D. (INCHES),THERM. COND.,SPEC.HEAT,
C      DENSITY AND TOTAL LENGTH (FT.)
       DATA PID,POD,PKH,CPP,DNP,PL/3.166,3.5,0.226,0.4,60.0,100.0/
C      DIP TUBE DATA
       DATA DTID,DTOD,DTKH/0.748,1.05,0.08/
C      LENGTH OF PIPE INCREMENTS
       PIL=PL/IP
C      GROUND DATA
       DATA GKH,CPG,DNG,DR,SE/1.4,0.45,115.0,0.0208333,1.5/
C      NUMBER OF LOOPS (FOR PARALLEL SYSTEMS)
       CLS=6.0
C     AVERAGE INDOOR AIR TEMPERATURES
       TWB=65.0
       TDB=72.0
C     RUN FRACTIONS - ONF=TOTAL ITERATIONS PER CYCLE, KN=ON ITERATIONS
C     NIGHT, KM=ON ITERATIONS IN MORN., KA=ON ITERATIONS IN AFTERNOON,
C     KE=ON ITERATIONS IN EVENING
       DATA ONF,KN,KM,KA,KE/14,7,7,7,7/
C     NUMBER OF DO LOOP ITERATIONS FOR TIME STEP
       NDLI=4704
C     SET UP FOR PRNT OUT OF LAST ON-OFF CYCLE
       NPLC=NDLI-ONE
C      PRINT FREQUENCY FOR HT.TRANS.RATES AND GROUND TEMP.DISTRIBUTION
       NP1=1008
C      PRINT FREQUENCY OF WATER TEMPS. (MULTIPLY NPP BY DT FOR PRINT T
       NPP=336
C      WATER DATA
       DATA DNW,CPW,GPMT/62.4,1.0,10.2/
       GPM=GPMT/CLS
       WKH=0.3036+0.0007807*TW(1)-0.000001767*TW(1)**2.
C      CONVERT GAL/MIN INTO FT**3/SEC
       VDOT=GPM/448.8
C     CONVECTIVE HEAT TRANSFER COEFFICIENT
       HH=40.0
C     EQUIVALENT HT.TR.COEF. FOR DIP TUBE (BTU/H-SQ FT-F)
       RAP=1.0/HH+(DTOD/(24.0*DTKH))*ALOG(DTOD/DTID)
       HAP=1.0/RAP
    15 RO=PID/24.0
       DRP=(POD-PID)/24.0
       PK=PKH/3600.0
       GK=GKH/3600.0
       WK=WKH/3600.0
       DTK=DTKH/3600.0
       ALP=PK/(DNP*CPP)
       ALG=GK/(DNG*CPG)
       ALW=WK/(DNW*CPW)
       AR=3.1416*PID/12.0
       ARI=3.1416*DTOD/12.0
       ARO=3.1416*POD/12.0
       VOL=0.7854*(PID/12.0)**2.0*PIL
       VEL=VDOT/(0.7854*(PID**2.0-DTOD**2.0)/144.0)
C      CALCULATE TIME STEP TO EQUAL TIME REQUIRED FOR WATER TO
C      TRAVEL THRU ONE VERTICAL PIPE INCREMENT
       DT=PIL/VEL
       IL=1
       WRITE(6,20)PID,POD,PKH,PL,IP
    20 FORMAT(1X,'PIPE DATA: ID=',F8.3,' IN. OD=',F8.3,' IN K='
     @' BTU/HR-FT-F L=',F8.3,' FT INCREMENTS=',I3,/)
       WRITE(6,21)DT
    21 FORMAT(1X,'TIME STEP =',F8.3,' SEC')
       TI=0.0
       JT=0
       TM=1.0
       ITD=-5
       IPT= 5
C     FLOW CORRECTION FACTORS FOR HEAT PUMP CAPACITY & POWER
       CFP=1.211-0.03920*GPMT+0.001611*GPMT**2.0
       CFC=0.9625+0.00366*GPMT
       CAC=0.98
       CAP=0.99
       CWBC=-0.5125+0.03141*TWB-0.0001321*TWB**2.0
       CWBP=0.410+0.012*TWB-0.0000487*TWB**2.0
       DRG=DR
       R(2)=RO+DRP+DR
       RI(2)=R(2)*12.0
       DO 161 LR=2,NRM1
```

```
      DRG=DRG*SE
      R(LR+1)=R(LR)+DRG
      RI(LR+1)=R(LR+1)*12.0
  161 CONTINUE
      DO 100 IT=1,NDI
      ITD=ITD+1
      IPT=IPT+1
      IF(TM.EQ.0.0)IPT=1
      TI=TI+DT/60.0
      TH=TI/60.0
      TM=TM+DT/60.0
      IF(TH.LE.6.0)ON=KN
      IF(TH.GT.6.0.AND.TH.LE.12.0)ON=KM
      IF(TH.GT.12.0.AND.TH.LE.18.0)ON=KA
      IF(TH.GT.18.0)ON=KE
      OFF=ONE-ON
      IF(JT.GE.ONE)GO TO 35
      IF(JT.GT.ON)GO TO 70
      GO TO 59
   35 JT=0
   59 H=HH/3600.0
      HAP=1.0/RAP
C     PUMP POWER
      PI=375.0
C     CAPACITY & POWER CURVES FOR HEAT PUMP
      PC=2100.0+12.292*TW(IP)+.05005*TW(IP)**2.0
      PW=PI+PC*CFF*CAP*CWDP
      QREJ=21020.+501.*TW(IP)+1.983*TW(IP)**2.-0.05020*TW(IP
      QTEC=QREJ*CFC*CWRC
      QCAP=QTEC-PW*3.412
      QHP=QTEC/(CLS*3600.0)
      EER=ABS(QCAP*CFC/PW)
      TWL=TW(IP)+QHP/(DNW*VDOT*CPW)
      QDT=0.5*HAP*ARI*PL*(TW(IP)-TWL)/3600.0
      TW(1)=TWL+QDT/(DNW*VDOT*CPW)
      GO TO 60
C     FREE CONVECTION COEFFICIENT (BTU/S-SQ.FT.-F)
   70 H=0.012
      QHP=0.0
      CLR=0.0
      QTEC=0.0
      QCAP=0.0
      HAP=0.0
      PW=0.0
   60 HEQ=1.0/((POD/(H*PID))+(RO/PK)*ALOG(POD/PID))
      X1=2.0*HEQ*(RO+DRP)*DT/(DNG*(RO+.25*DR)*DR*CPG)
      X2=2.0*GK*(RO+0.5*DRP)*DT/(DNG*(RO+.25*DR)*CPG*DR**2.0
      DO 160 LV=1,IP
      D(LV,1)=X1+X2+1.0
      A(LV,1)=-1.0*X2
  160 CONTINUE
      DO 300 N=1,IP
      C(N,1)=C(N,1)+X1*TW(N)
      DRG=DR
      DO 120 L=2,NRM1
      X=ALG*DT/(SE*(SE+1.0)*DRG)
      B(N,L)=-1.0*X*SE*((2.0/DRG)-SE/R(L))
      D(N,L)=1.0+2.0*X*(SE+1.0)/DRG+X*(1-SE**2.0)/R(L)
      A(N,L)=-1.0*X*(2.0/DRG+1.0/R(L))
      DRG=DRG*SE
  120 CONTINUE
      C(N,NRM1)=C(N,NRM1)-A(N,NRM1)*C(N,NR)
      DO 1 I=2,NRM1
      F=B(N,I)/D(N,I-1)
      D(N,I)=D(N,I)-F*A(N,I-1)
    1 C(N,I)=C(N,I)-F*C(N,I-1)
C     BACK SUBSTITUTION
      C(N,NRM1)=C(N,NRM1)/D(N,NRM1)
      DO 2 I=2,NRM1
      J=NRM1-I+1
    2 C(N,J)=(C(N,J)-A(N,J)*C(N,J+1))/D(N,J)
      IF(IL.EQ.1) GO TO 105
      GO TO 80
  105 IL=IL+1
      HHEQ=HEQ*3600.0
      WRITE(6,112)POD,HHEQ
  112 FORMAT(1X,'EQ.DIA.=',F7.3,' IN.   EQ.HT.TR.COEF.=',F7.2,' BTU/H-SQ
     *FT-F',/)
      WRITE(6,115)
  115 FORMAT(1X,'DISTANCE FROM CENTER OF PIPE IN INCHES')
      WRITE(6,110)(RI(M),M=2,NRM1)
  110 FORMAT(2X,'WATER',3X,'PID',4X,'POD',2X,13F7.2)
      WRITE(6,111)TW(1),TIW(1),(C(1,I2),I2=1,NRM1)
  111 FORMAT(1X,16F7.3)
   80 QSC(N)=HAP*ARI*PIL*(TW(1)-TW(N))
      Q(N)=HEQ*ARO*(C(N,1)-TW(N))*3600.0*PIL-QSC(N)
      QS=Q(N)/3600.0
C     UPDATE EQUATION FOR INNER PIPE WALL
      TIW(N)=QS/(H*AR*PIL)+TW(N)
      QT(N)=QT(N)+QS*DT
C     IF HEAT PUMP IS RUNNING GO TO ST. 81
C     WATER TEMPERATURE UPDATE WHEN UNIT IS OFF
      IF(JT.LT.ON)GO TO 81
      TW(N)=TW(N)+(QS*DT)/(DNW*CPW*VOL)
      GO TO 75
C     WATER TEMPERATURE UPDATE WHEN UNIT IS ON
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```
   81 IF(N.EQ.IP) GO TO 75
      TW(N+1)=TW(N)+QS/(DNW*VDOT*CPW)
   75 IF(IP1.EQ.NPT)GO TO 50
      GO TO 300
   50 IF(N.EQ.1)GO TO 52
      IF(N.EQ.MP)GO TO 52
      IF(N.EQ.IP)GO TO 52
      GO TO 300
   51 COP=EER/3.412
      WRITE(6,55)PW,QCAP,QTEC,EER,COP
   55 FORMAT(1X,' POWER =',F8.1,' W  QHP=',F10.1,' BTUH QTEC=',F10.1,
     @' BTUH  EER=',F7.2,'  COP=',F7.2,/)
      GO TO 299
   52 WRITE(6,65)N,TI,Q(N),QSC(N),TW(N),(C(N,II),II=1,10)
   65 FORMAT(1X,'N=',I3,F10.2,' MIN,  Q=',F10.2,' BTU/HR-INC  QSC=',F10.
     @1,' BTU/HR-INC',/,11F7.3)
      IF(N.EQ.IP)GO TO 51
  299 TM=0.0
  300 CONTINUE
      JT=JT+1
      IF(ITB.EQ.NPF)GO TO 98
      IF(IT.GT.NPLC)GO TO 98
      GO TO 100
   98 WRITE(9,99)TI,TW(1),TW(MP),TW(IP)
   99 FORMAT(1X,4F10.2)
  150 CONTINUE
      ITB=0
  100 CONTINUE
      DO 102 NV=1,IP
      WRITE(8,101)TW(NV),TIW(NV),(C(NV,NH),NH=1,NR)
  101 FORMAT(1X,17F7.2)
  102 CONTINUE
      STOP
      END
```

TIME(MIN)	IN TEMP	MID TEMP	OUT TEMP
1460.09	90.94	85.72	82.00
2898.76	92.86	87.74	84.02
4337.45	93.94	88.87	85.17
5776.28	94.68	89.64	85.95
7215.11	95.23	90.23	86.55
8653.83	95.68	90.70	87.03
10092.33	96.05	91.09	87.42
11530.83	96.36	91.42	87.76
12969.33	96.64	91.71	88.06
14407.83	96.88	91.97	88.32
15846.33	97.09	92.20	88.55
17284.83	97.29	92.40	88.76
18723.33	97.47	92.59	88.96
20082.63	93.81	89.97	86.96
20084.77	95.69	91.14	87.75
20086.91	96.43	91.68	88.16
20089.05	96.80	91.98	88.42
20091.19	97.04	92.19	88.60
20093.33	97.21	92.35	88.74
20095.47	97.34	92.48	88.86
20097.61	97.45	92.53	88.96
20099.75	97.54	92.68	89.05
20101.89	97.62	92.76	89.13
20104.03	97.69	92.83	89.20
20106.17	97.76	92.89	89.26
20108.31	97.81	92.95	89.31
20110.46	97.87	93.01	89.37
20112.60	97.03	92.19	88.77
20114.74	96.21	91.48	88.33
20116.88	95.47	90.84	87.93
20119.02	94.79	90.26	87.56
20121.16	94.15	89.71	87.21
20123.30	93.57	89.21	86.89
20125.44	93.02	88.74	86.60
20127.58	92.51	88.29	86.32
20129.72	92.03	87.88	86.06
20131.86	91.58	87.49	85.81
20134.00	91.15	87.12	85.58
20136.14	90.75	86.78	85.36
20138.28	90.37	86.45	85.15
20140.42	90.01	86.14	84.96

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C     HEAT TRANSFER AND TEMPERATURE DISTRIBUTION IN CONCENTRIC
C     GROUND COUPLINGS
      DIMENSION T(20,16),TW(20),R(15),RI(15),Q(20),QT(20),QSC(20)
C     NUMBER OF VERTICAL INCREMENTS-MUST AGREE WITH DIMENSION T(IP,1
C     TW(IP)
      WRITE(9,9)
    9 FORMAT(2X,'TIME(MIN)',2X,'IN TEMP',3X,'MID TEMP',2X,'OUT TEMP'
      IP=20
      MP=IP/2+1
C     PIPE DATA - O.D. & I.D. (INCHES),THERM. COND.,SPEC.HEAT,
C     DENSITY AND TOTAL LENGTH (FT.)
      DATA PID,POD,PKH,CPP,DNP,PL/3.166,3.5,0.226,0.4,60.0,100.0/
C     DIP TUBE DATA
      DATA DTID,DTOD,DTKH/0.748,1.05,0.08/
C     LENGTH OF PIPE INCREMENTS
      PIL=PL/IP
C     GROUND DATA
      DATA GKH,CPG,DNG,DR,SC/1.4,0.45,100.0,0.0208333,1.5/
C     NUMBER OF LOOPS (FOR PARALLEL SYSTEMS)
      CLS=6.0
C     GROUND TEMPERATURES AT START UP
      DATA TW/20*62.0/
      DATA T/320*62.0/
C     CYCLE TIMES IN MINUTES
      DATA TON,TOFF/30.0,30.0/
      TCY=TON+TOFF
C     WATER DATA
      DATA DNW,CPW,GPMT/62.4,1.0,10.2/
      GPM=GPMT/CLS
      WKH=0.303610.0007807*TW(1)-0.000001767*TW(1)**2.
C     CONVERT GAL/MIN INTO FT**3/SEC
      VDOT=GPM/448.8
      HH=40.0
   15 RO=PID/24.0
      DRP=(POD-PID)/24.0
      PK=PKH/3600.0
      GK=GKH/3600.0
      WK=WKH/3600.0
      DTK=DTKH/3600.0
      ALP=PK/(DNP*CPP)
      ALG=GK/(DNG*CPG)
      ALW=WK/(DNW*CPW)
      AR=3.1416*PID/12.0
      ARI=3.1416*DTOD/12.0
      ARO=3.1416*POD/12.0
      VOL=0.7854*(PID/12.0)**2.0*PIL
      VEL=VDOT/(0.7854*(PID**2.0-DTOD**2.0)/144.0)
C     CALCULATE TIME STEP TO EQUAL TIME REQUIRED FOR WATER TO
C     TRAVEL THRU ONE VERTICAL PIPE INCREMENT
      DT=PIL/VEL
      II=1
      WRITE(6,20)PID,POD,PKH,PL,IP
   20 FORMAT(1X,'PIPE DATA: ID=',F8.3,' IN. OD=',F8.3,' IN K='
     @' BTU/HR-FT-F L=',F8.3,' FT INCREMENTS=',I3,/)
      WRITE(6,21)DT
   21 FORMAT(1X,'TIME STEP =',F8.3,' SEC')
      TT=0.0
      TJ=0.0
      TM=0.0
      ITD=-10
C    PRINT FREQ. FOR HT. RATES (MIN.) AND WATER TEMPS.(ITERATION
      PTM=4320.
      IPT=1344
      CFP=1.211-.03928*GPMT+.001611*GPMT**2.0
      CFC=0.9707+.00366*GPMT
      CWRC=0.985
      CWRP=0.99
      DO 100 I=1,18816
      ITD=ITD+1
      TT=TT+DT/60.0
      TM=TM+DT/60.0
      TMP=TM+DT/60.0
      IF(TJ.GT.TCY)GO TO 35
      IF(TJ.GT.TON)GO TO 70
      GO TO 59
   35 TJ=0.0
   59 H=HH/3600.0
      RAP=1.0/HH+(DTOD/(24.0*DTKH))*ALOG(DTOD/DTID)
      HAP=1.0/(RAP*3600.0)
      PC=2047.34+12.2918*TW(IP)+.05*TW(IP)**2.0
      PP=375.0
      PW=PP+PC
      QCAP=59921.0-223.93*TW(IP)+0.25336*TW(IP)**2.0
      QPW=3.412*PW*CWRP
      QREJ=21820.1561.*TW(IP)+1.983*TW(IP)**2.-0.0583*TW(IP)**3
      QTEC=CFC*CWRC*QREJ
      QHP=QTEC/(CLS*3600.0)
      TW(1)=TW(IP)+QHP/(DNW*VDOT*CPW)
      GO TO 60
   70 H=0.012
      QHP=0.0
      HAP=0.0
   60 DO 300 N=1,IP
C     UPDATE EQUATION FOR OUTER PIPE WALL
      HEQ=1.0/((POD/(H*PID))+(RO/PK)*ALOG(POD/PID))
      X1=HEQ*(RO+DRP)
```

```
      X2=GK*(RO+DRP+DR/2.0)/DR
      X3=DNG*CPG*(RO+DRP+DR/4.0)*DR/(2.0*DT)
      T(N,2)=(X1/X3)*(TW(N)-T(N,2))+(X2/X3)*(T(N,3)-T(N,2))+T(N,2)
C     UPDATE EQUATION FOR GROUND NODES
      R(3)=RO+DRP+DR
      DRG=DR
      DO 200 L=3,15
      A=T(N,L+1)/(SE*(SE+1.0))-T(N,L)/SE+T(N,L-1)/(SE+1.0)
      B=T(N,L+1)/(SE*(SE+1.0))-(1.0-SE)*T(N,L)/SE-SE*T(N,L-1)/(1.0+SE)
      C=DNG*CPG/(GK*DT)
      T(N,L)=2.0*A/(C*DRG**2.0)+B/(C*R(L)*DRG)+T(N,L)
      DRG=DRG*SE
      RI(L)=R(L)*12.0
      R(L+1)=R(L)+DRG
  200 CONTINUE
      IF(IL.EQ.1) GO TO 105
      GO TO 80
  105 IL=IL+1
      WRITE(6,115)
  115 FORMAT(1X,'DISTANCE FROM CENTER OF PIPE IN INCHES')
      WRITE(6,110)(RI(M),M=3,15)
  110 FORMAT(2X,'WATER',5X,'PID',4X,'POD',2X,13F7.2)
      WRITE(6,111)TW(1),(T(1,I2),I2=1,15)
  111 FORMAT(1X,16F7.3)
   80 QSC(N)=HAP*ARI*PIL*(TW(1)-TW(N))*3600.0
      Q(N)=HFD*AKO*(T(N,2)-TW(N))*3600.0*PIL-QSC(N)
      QS=Q(N)/3600.0
C     UPDATE EQUATION FOR INNER PIPE WALL
      T(N,1)=QS/(H*AK*PIL)+TW(N)
      QT(N)=QT(N)+QS*DT
C     IF HEAT PUMP IS RUNNING GO TO ST. 81
C     WATER TEMPERATURE UPDATE WHEN UNIT IS OFF
      IF(TJ.LT.TON)GO TO 81
      TW(N)=TW(N)+(QS*DT)/(DNW*CPW*VOL)
      GO TO 75
C     WATER TEMPERATURE UPDATE WHEN UNIT IS ON
   81 IF(N.EQ.IP) GO TO 75
      TW(N+1)=TW(N)+QS/(DNW*VDOT*CPW)
   75 IF(TMP.GT.PTH)GO TO 50
      GO TO 300
   50 IF(N.EQ.1)GO TO 51
      IF(N.EQ.MP)GO TO 51
      IF(N.EQ.IP)GO TO 51
      GO TO 300
   51 WRITE(6,55)N,PW,QCAP,QTEC
   55 FORMAT(1X,'N=',I3,' POWER =',F8.1,' W  Q=',F10.1,' BTUH QTEC=
     &',F10.1,' BTUH')
      WRITE(6,65)TI,Q(N),QT(N),TW(N),(T(N,II),II=1,10)
   65 FORMAT(1X,F10.2,' MIN.  Q=',F10.2,' BTU/HR-INC  Q
     &',/,11F7.3)
      TM=0.0
  300 CONTINUE
      TJ=TJ+DT/60.0
      IF(ITD.EQ.IPT)GO TO 98
      IF(I.GT.18760)GO TO 98
      GO TO 100
   98 WRITE(9,99)TI,TW(1),TW(MP),TW(IP)
   99 FORMAT(1X,4F10.2)
      ITD=0
  150 CONTINUE
  100 CONTINUE
      STOP
      END
```

LSCF.FOR

```
C     LEAST SQUARES CURVE FIT
      DIMENSION A(15,15),B(15,15),X(15),Y(15,15),W(15),C(15,15)
      OPEN(8,FILE='CFIT.DAT',STATUS='OLD')
      MM=0
      READ(8,100)LL
    5 IF(MM.GE.LL) GO TO 1000
   10 READ(8,200)N,M,L
      IF(MM.NE.0) GO TO 25
   15 WRITE(6,400)
      DO 20 I=1,N
      READ(8,300) X(I),W(I),(Y(I,J),J=1,L)
      DO 20 J=1,L
   20 WRITE(6,500) X(I),W(I),Y(I,J)
C     FORM COEFFICIENT MATRIX
   25 DO 30 I=1,N
   30 C(I,1)=1.0
      MP1=M+1
      DO 35 J=2,MP1
      DO 35 I=1,N
   35 C(I,J)=C(I,J-1)*X(I)
      DO 40 I=1,MP1
      DO 40 J=1,MP1
      A(I,J)=0.0
      DO 40 K=1,N
   40 A(I,J)=A(I,J)+C(K,I)*C(K,J)*W(K)
C     FORM CONSTANT MATRIX
      DO 44 J=1,L
      DO 44 I=1,MP1
      B(I,J)=0.0
      DO 44 K=1,N
   44 B(I,J)=B(I,J)+C(K,I)*Y(K,J)*W(K)
      CALL MATSUB(A,MP1,B,L,DET)
      WRITE(6,600)N
      WRITE(6,700)M
      DO 50 J=1,L
      DO 50 I=1,MP1
      II=I-1
   50 WRITE(6,800)II,B(I,J)
      MM=MM+1
      GO TO 5
  100 FORMAT(I4)
  200 FORMAT(3I4)
  300 FORMAT(6F10.5)
  400 FORMAT(14X,'X',20X,'WEIGHT',20X,'Y'/)
  500 FORMAT(8X,E12.5,12X,E12.5,13X,E12.5/)
  600 FORMAT(/,'NUMBER OF GIVEN DATA POINTS ='I2)
  700 FORMAT(/7X,'DEGREE OF POLYNOMIAL = ',I2,/)
  800 FORMAT(5X,I2,' DEGREE COEFFICIENT = ',E16.6,/)
 1000 STOP
      END
      SUBROUTINE MATSUB(A,N,B,M,DET)
C     SUBROUTINE FOR SIM. LIN. EQ. SOLUTION AND MATRIX INVERSION
      DIMENSION A(15,15),B(15,15),IPVOT(15),INDEX(15,2),PIVOT(15)
      COMMON IPVOT,INDEX,PIVOT
      EQUIVALENCE (IROW,JROW),(ICOL,JCOL)
C     INITIALIZATION
   57 DET=1.0
      DO 17 J=1,N
   17 IPVOT(J)=0
      DO 135 I=1,N
C     SEARCH FOR PIVOT ELEMENT
      T=0.0
      DO 9 J=1,N
      IF(IPVOT(J).EQ.1) GO TO 9
   13 DO 23 K=1,N
      IF (IPVOT(K)-1)43,23,81
   43 IF(ABS(T).GE.ABS(A(J,K))) GO TO 23
   83 IROW=J
      ICOL=K
      T=A(J,K)
   23 CONTINUE
    9 CONTINUE
      IPVOT(ICOL)=IPVOT(ICOL)+1
C     PUT PIVOT ELEMENTS ON DIAGONAL
      IF(IROW.EQ.ICOL) GO TO 109
   73 DET=-DET
      DO 12 L=1,N
      T=A(IROW,L)
      A(IROW,L)=A(ICOL,L)
   12 A(ICOL,L)=T
      IF(M.LE.0)GO TO 109
   33 DO 2 L=1,M
      T=B(IROW,L)
      B(IROW,L)=B(ICOL,L)
    2 B(ICOL,L)=T
  109 INDEX(I,1)=IROW
      INDEX(I,2)=ICOL
      PIVOT(I)=A(ICOL,ICOL)
      DET=DET*PIVOT(I)
C     DIVIDE PIVOT ROW BY PIVOT ELEMENT
      A(ICOL,ICOL)=1.0
      DO 205 L=1,N
  205 A(ICOL,L)=A(ICOL,L)/PIVOT(I)
      IF(M.LE.0)GO TO 347
   66 DO 52 L=1,M
   52 B(ICOL,L)=B(ICOL,L)/PIVOT(I)
```

```
C       REDUCE NON-PIVOT ROWS
    347 DO 135 LI=1,N
        IF(LI.EQ.ICOL) GO TO 135
     21 T=A(LI,ICOL)
        A(LI,ICOL)=0.0
        DO 89 L=1,N
     89 A(LI,L)=A(LI,L)-A(ICOL,L)*T
        IF(M.LE.0) GO TO 135
     10 DO 68 L=1,M
     68 B(LI,L)=B(LI,L)-B(ICOL,L)*T
    135 CONTINUE
C       INTERCHANGE COLUMNS
    222 DO 3 I=1,N
        L=N-I+1
        IF(INDEX(L,1).EQ.INDEX(L,2)) GO TO 3
     19 JROW=INDEX(L,1)
        JCOL=INDEX(L,2)
        DO 549 K=1,N
        T=A(K,JROW)
        A(K,JROW)=A(K,JCOL)
        A(K,JCOL)=T
    549 CONTINUE
      3 CONTINUE
     81 RETURN
        END
```

```
0002
000500020001
   75.0       1.0     52000.0
   80.0       1.0     51000.0
   85.0       1.0     50000.0
   90.0       1.0     47500.0
   95.0       1.0     45000.0
000500030001
```

```
UMBER OF GIVEN DATA POINTS = 6

      DEGREE OF POLYNOMIAL =  2

     0 DEGREE COEFFICIENT =      0.109000E+02

     1 DEGREE COEFFICIENT =     -0.425000E-01

     2 DEGREE COEFFICIENT =      0.892861E-04

UMBER OF GIVEN DATA POINTS = 6

      DEGREE OF POLYNOMIAL =  3

     0 DEGREE COEFFICIENT =     -0.109000E+02

     1 DEGREE COEFFICIENT =     -0.425018E-01

     2 DEGREE COEFFICIENT =      0.893157E-04

     3 DEGREE COEFFICIENT =     -0.145519E-09
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VITA 2

Stephen Paul Kavanaugh

Candidate for the Degree of

Doctor of Philosophy

Thesis: SIMULATION AND EXPERIMENTAL VERIFICATION OF VERTICAL GROUND-COUPLED HEAT PUMP SYSTEMS

Major Field: Mechanical Engineering

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