PERFORMANCE OF A VERTICAL HEAT PUMP GROUND-COUPLING DEVICE

By

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1977

Submitted to the Faculty of the
Graduate College of the
Oklahoma State University
in partial fulfillment of
the requirements for
the Degree of
MASTER OF SCIENCE
December, 1983

PERFORMANCE OF A VERTICAL HEAT PUMP

GROUND-COUPLING DEVICE

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ACKNOWLEDGMENTS

I wish to thank most of all Dr. Jerald D. Parker for his patience, guidance and understanding throughout my studies here at Oklahoma State University. I would also like to express my sincere appreciation to Drs. John A. Wiebelt, Faye C. McQuiston and Afshin J. Ghajar for their special contribution to my education.

I would like to thank Rick Kroecker, Steve Kavanaugh and Gary Ferrell for their moral support and encouragement. I am also grateful to my colleagues at the University Computer Center for their understanding and help along the way.

My entire family, and in particular my husband, Steve, have my deepest appreciation for their loving support and understanding.

TABLE OF CONTENTS

Chapter		Page
I.	INTRODUCTION	1
II.	OBJECTIVE	8
	System Model	8
	System Performance	9
III.	LITERATURE REVIEW	10
IV.	SYSTEM DESCRIPTION	17
V.	GROUND COIL SIMULATION MODEL	21
	Model Assumptions	21
	Model Background	22
	Simulation Procedure	32
VI.	SIMULATION RESULTS	39
VII.	CONCLUSIONS	48
BIBLIOGRAPHY		50
APPENDIXES		53
	APPENDIX A - LISTING OF COMPUTER PROGRAM	54
	APPENDIX B - FLOW CHART OF PROGRAM TUBE	65
	APPENDIX C - SIMULATION PRINTOUTS	75

LIST OF TABLES

Table		Page
I.	Heat Pump Rated Capacity	17
II.	Tube and Soil Specifications	20

LIST OF FIGURES

Figure		Page
1.	Typical Heat Pump-House Characteristics	3
2.	Temperature-Entropy Diagram	6
3.	Original and Modified Ground-coupled Heat Pump systems	19
4.	Tube Increment for Analysis	24
5.	Schematic of U-tube Heat Exchange	27
6.	Diagram for Equivalent Distance	28
7.	Grid System Definition.	30
8.	Start-up Period of Cyclic Operation	40
9.	Variation of Cycle Ratios	42
10.	Variation of Cycle On time.	43
11.	Variation of Distance Between Tube Legs	44
12.	Variation of Tube Depth	47

LIST OF SYMBOLS

A	- Tube interval surface area (sq ft)
Ag	- Tube interval surface area that exchanges heat with the ground (sq ft)
A	- Tube interval surface area that exchanges heat with opposite leg of tube (sq ft)
$\mathcal{L}$	- Thermal diffusivity of the soil (sq ft/hr)
COP	- Coefficient of performance
C	- Specific heat (Btu/lbm- F)
D	- Tube diameter (ft)
EADB	- Entering air dry-bulb temperature ($^{\circ}$F)
EAWB	- Entering air wet-bulb temperature ($^{\circ}$F)
EWT	- Entering water temperature ($^{\circ}$F)
FA	- Face area of the control volume in the ground (sq ft)
g	- Gravitational acceleration (sq ft/sec)
h	- Average heat transfer coefficient (Btu/hr-sq ft-$^{\circ}$F)
j	- Enthalpy (Btu/lbm)
k	- Thermal conductivity of the soil (Btu/ft-hr-$^{\circ}$F)
P	- Pressure (Psi)
•	- Density of the fluid (lbm/cubic ft)
q	- Heat transfer rate (Btu/hr)
Qwell	- Heat transfer rate to the well from the heat pump (Btu/hr)

Qhouse	-	Heat extraction rate of the house (Btu/hr)
R	-	Radius of the tube (ft)
r	-	Distance in the radial direction
T	-	Temperature (^{o}F)
Tav	-	Average fluid temperature from previous time period
Tavg	-	Average fluid temperature this time period
Twadj	-	Temperature of the wall adjacent to the current interval of study
Twall	-	Temperature of the wall of the current interval of study
Δ t	-	Time step of study (hr)
V	-	Velocity (ft/hr)
Vol	-	Volume of the current interval of study (cubic ft)
w	-	Fluid mass flow rate (lbm/hr)
$\dot{W}$	-	Rate of work (Btu/hr)
Wcomp	-	Compressor work (Btu/hr)
Z	-	Distance in the vertical direction (ft)

PROGRAM VARIABLES

CPF	-	Specific heat of fluid (Btu/lbm-^{o}F)
DA	-	Average distance apart between the tube legs (ft)
DTIME	-	Incremental time step of study (hrs)
DIV	-	Number of incremental time steps
FLRATE	-	Mass flow rate of the fluid (lbm/hr)
H	-	Average heat transfer coefficient (Btu/hr-sq ft-^{o}F)
Q	-	Total average rate of heat transfer (Btu/hr)
Q1	-	Rate of heat exchange with the ground (Btu/hr)

Q2	- Rate of heat exchange with the other tube leg (Btu/hr)
RHO	- Density of fluid (lbm/cubic ft)
TAV	- Average fluid temperature from previous time period
TAVG	- Average fluid temperature this time period
TF	- Fluid temperature at interval inlet
TFIN	- Fluid temperature at well inlet
TFOUT	- Fluid temperature at well outlet
TN	- New fluid temperature at interval outlet
TWALL	- Temperature of the wall of the current interval of study
VOL	- Volume of the current interval of study (cubic ft)

CHAPTER I

INTRODUCTION

About one third of the total energy consumption in the United States is residential and commercial building energy usage. Seventy percent of this amount is for indoor space conditioning and domestic hot water production (1). In the past, these energy needs have been met in most part by oil and gas. In Oklahoma during the year 1965, approximately seventy-eight percent of the energy consumed by commercial buildings was supplied by gas and oil while electricity accounted for about twenty percent. During the same year, residential usage was supplied by seventy-seven percent natural gas and seventeen percent electricity. However recent years have seen a change in this trend and projections indicate that by 1990 electricity will supply fifty-two percent of the residential energy needs and fifty-six percent of the commercial (2). This trend is in part a result of increasing fuel prices and spot shortages. Consequently, conservation efforts have become an important concern to both consumers and the utilities. These efforts have been in the form of reducing consumer demand and increasing the efficiency or performance of energy consuming devices. The heat pump has the potential of aiding conservation. During the heating operation, it has the ability to deliver more energy in the form of heat than it requires for operation. Also, the same equipment can be used for both heating and cooling.

The air source heat pump is the most commonly used heat pump. In 1981, one out of every four new single family houses was equipped with air source, electric heat pumps (3). Here the air acts as both source and sink. During the heating season, as the outside temperature decreases, the heating demand of the house increases while the capacity of the heat pump diminishes. The balance point is the condition where the heat pump's heating capacity is equal to the space heating load. These trends and a balance point are depicted in Figure 1. For efficient operation, the balance point of a heat pump system is designed so that it can supply all the heating needs most of the time. During extreme weather conditions such as continuous days of unusually low temperatures, additional heating will be needed. This is most frequently supplied by resistance heating. This would then create an undesirable load situation for the electicity suppliers. Utilities experience a peak demand in either the winter or summer depending on the local conditions. This peak demand dictates the necessary capacity of the utility. If a utility experiences its peak in the winter, a large increase in the number of conventional air source heat pumps could contribute to an even greater peak and the need for more

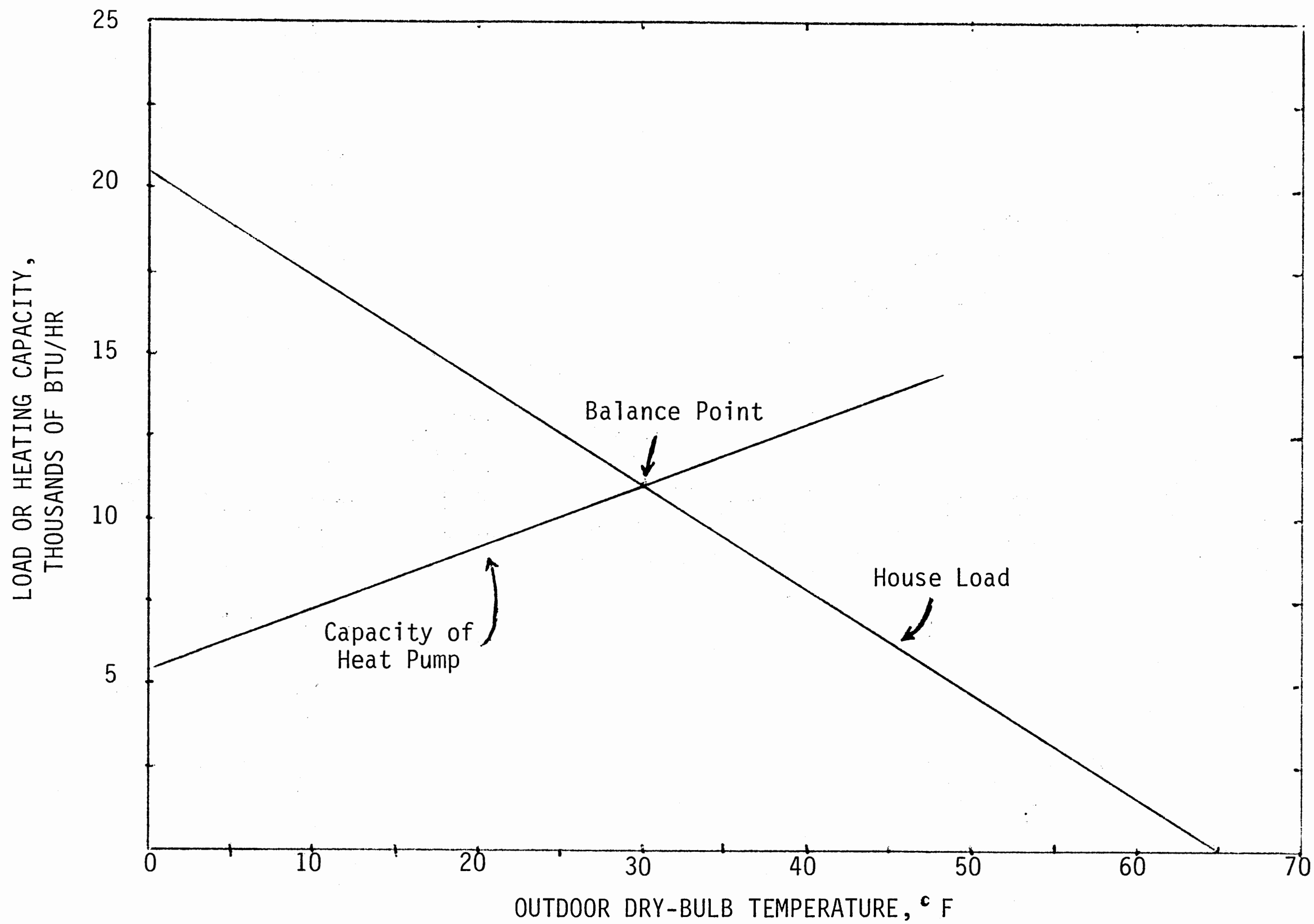

Figure 1. Typical Heat Pump-House Characteristics

capacity. Expansion of facilities is costly and an increase in the price of electricity would result. The ground coil heat pump could be a part of the solution to this problem by maintaining or even reducing this peak.

Utilizing the earth's stored energy with a ground-coupled heat pump is not a new idea. The technology was developed in the early 1900's and involves the transfer of heat to or from the earth by piping buried beneath the frost line.

In this system, the ground acts as a heat source or sink for the heat pump much as the air does for a conventional air-coupled system. In the 1940's and 1950's significant efforts were made in the United States, Britain and Germany to develop systems using horizontal ground pipes or coils (4). They were used where large surface areas were available such as schools and houses on large lots. Recent research efforts have been on the vertical ground coil systems since they require less area and so are more generally applicable. Ground temperatures at greater depths are also less susceptible to seasonal variation which has a desirable effect on the heat pump performance (see discussion below).

Oklahoma State University has been actively involved in this area of research. Projects supported by Oklahoma Gas & Electric (OG&E) and Electric Power Research Institute (EPRI) include the installation and performance study of three different heat pump systems including an air-to-air, a ground-source/sink and a solar assisted ground-source/sink system. Results have shown that the air source heat pump requires backup resistance heat regularly during the colder months whereas the water source heat pump handles the load on its own (5).

The heat exchange with the earth is an integral part of the ground-source/sink system as the source/sink temperature plays a major role in the heat pump performance. The importance of the source/sink temperature can be demonstrated with the aid of the temperature-entropy diagram of Figure 2. By definition, the coefficient of performance, COP, of the heat pump is given by:

$$COP(h) = Q(h) / W \qquad \text{for heating}$$

$$COP(c) = Q(c) / W \qquad \text{for cooling}$$

Referring to Figure 2, the COP can be re-written in terms of the areas A1 and A2 to yield:

$$COP(h) = (A1 + A2) / A1 \qquad \text{for heating}$$

$$COP(c) = A2 / A1 \qquad \text{for cooling}$$

It then becomes obvious that during the heating operation, decreasing A1 by increasing T(c), the source temperature, will result in an improved COP. Likewise, during the cooling mode, decreasing A1 by decreasing T(h), the sink temperature, will have the same desirable effect on the COP.

Ground temperatures tend to be more consistently moderate than air temperatures. Collins (6) has indicated that at depths of thirty to sixty feet the ground temperature varies seasonally by only one degree Fahrenheit

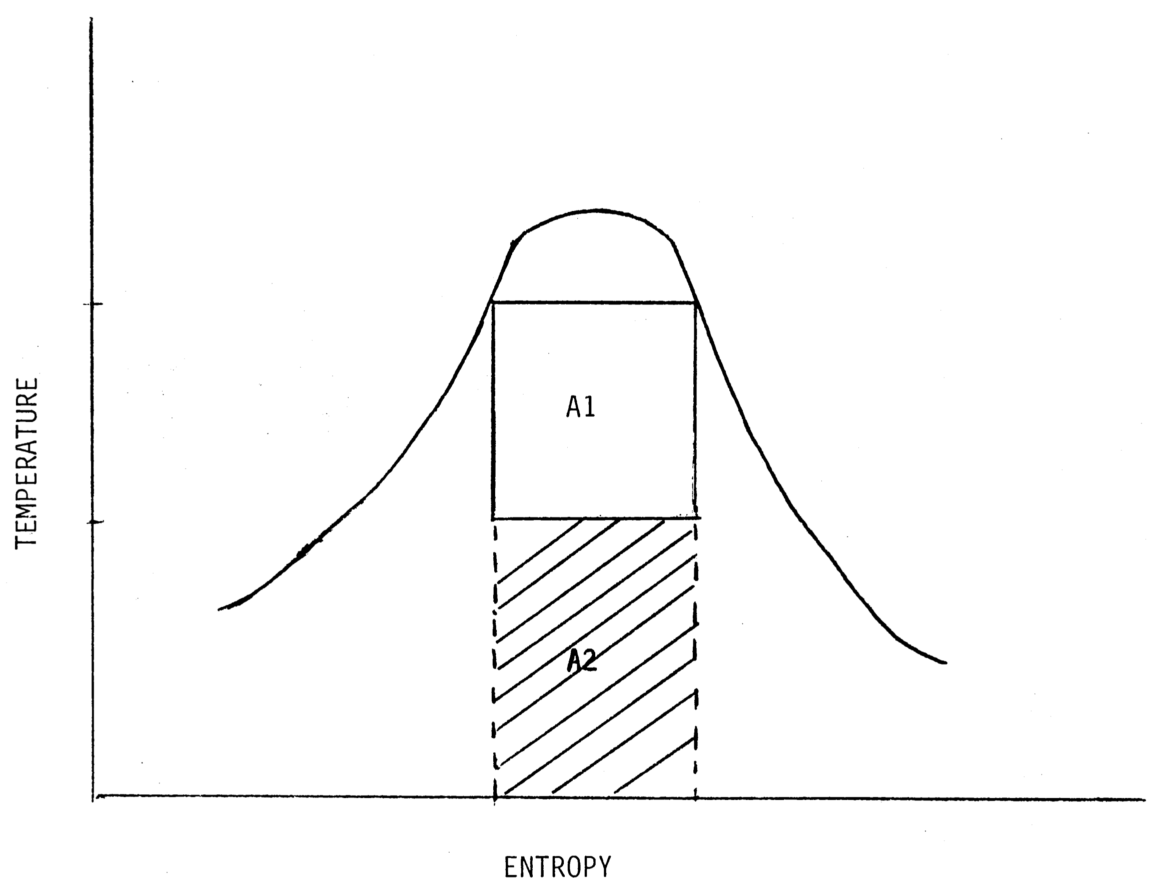

Figure 2. Temperature_entropy Diagram

and that over most of the continental United States at these depths, the ground temperature is at least sixty degrees Fahrenheit.

The ground-coupling device considered in this study is a high density polyethylene U-tube. This type of device has been used in projects at Oklahoma State University (5). More details on the tube configuration are included in the SYSTEM DESCRIPTION.

CHAPTER II

OBJECTIVE

The purpose of this study was to develop a system model describing the heat exchange and resulting water temperature variation within the vertical heat pump ground-coupling device. This model served as the basis for a computer program which was then used to investigate the effects of various parameters on the system's performance.

System Model

The system model should represent realistically the thermal characteristics and fluid flow involved in the transfer of heat between the water in the tubing and the ground. Important considerations are:

1. The ground's capacity for heat transfer and storage,
2. The fluid movement through the tube during the circulation period,
3. The thermal short circuiting between the legs of the U-tube.

Reasonable assumptions were made to prevent the resulting computer program from becoming unnecessarily detailed and costly. The results of the study may support the assumptions made or indicate areas that require further attention.

System Performance

The list of parameters which could be varied in a sensitivity analysis is almost endless. Several factors were identified as having the greatest potential for affecting system performance. These include:

1. Ratio of time that the heat pump is on and circulating water to the time of a complete cycle,
2. The distance between the downcomer and the upcomer in the U-tube installation,
3. Tube radius,
4. Ratio of tube length to capacity of heat input or extraction.

The program to simulate the system was written in general terms to allow the investigator to vary the above parameters, as well as others, with a minimum of effort.

CHAPTER III

LITERATURE REVIEW

Early investigations into the heat transfer between the earth and a heat pump ground-coupling device included an elaboration of the Kelvin heat source theory by Ingersoll and Plass (7). This theory treats the tube as an infinitely long source or sink of heat in an infinite medium. All heat flow is considered radial due to the long tube length. The resulting analytical expression allows calculations of temperatures within the medium after specified time intervals. Because the theory is based on a line source, significant error is introduced if the pipe diameter is too large or the time period is less than a few days. Ingersoll and Plass consider an average constant heat transfer rate over a time period of months and with these conditions they obtain reasonable results. They also found the effect of two pipes in the same trench is to decrease efficiency below that of two isolated pipes but for short-period high-capacity operation, the effect is probably negligible.

Coogan (8) conducted an experimental investigation into earth heat absorption rates. Although the actual application was to obtain specific information on direct expansion of a vapor within a buried tube, some insight can be gained into cases where liquid is circulated. His measurements show the variation in the ground's temperature distribution is limited to a small radius surrounding the tube and are in reasonable agreement with the line sink theory under steady state conditions.

Several years later within the petroleum industry, these findings were utilized when Moss and White (9) applied the line source concept to an injection well to evaluate the heat transfer to or from the surrounding medium to the injection water. Here, the transient nature of the problem was considered and the well was broken into increments. Given the inlet water temperature and an assumed outlet temperature, an energy balance equating the heat necessary to raise the water temperature to that transferred to the surrounding medium was performed. After several interations, the correct outlet temperature was found and the process repeated throughout the tube section. The well casing temperature was assumed equal to the bulk fluid temperature. The applications were again limited to continuous time periods of operation, a constraint of the line source assumption. However, the error introduced by the finite radius appeared to be negligible.

Ramey (10) presented a generalized development that included the method of Moss and White as a special case. His approach included consideration of the thermal resistance of various components of a wellbore and he suggested the inclusion of a resistance term for any materials with comparatively low conductivity. Also, the inclusion of a time function allows any approximation method of the earth's heat transfer rates to be incorporated. For time periods greater than one week, the line source method is suggested. For shorter periods a convection boundary condition at the cylinder is recommended. Assuming radial heat transfer from the wellbore and that the heat flow in the immediate vicinity of the wellbore is rapid compared to that in the surrounding medium and thus can be adequately represented by steady state allows an energy balance on incremental tube sections. The heat lost by the liquid, equal to that transferred to the casing, is defined by the conduction heat rate to the surrounding ground. This energy balance approach is the basis for the algorithm used in this study.

In the early 1960's, finite-difference methods became increasingly popular with their application aided by the increasingly accessible digital computer. Dusinberre (11) presented solutions to various heat transfer problems including steady state and transient conditons in one, two, three and multi-dimensional configurations. He also pointed out possible instabilities resulting from the interval size of study being too large. Schenck (12) was among the first to then demonstrate the application of FORTRAN to some of these finite-difference solutions.

In the latter part of the 1970s, interest in the ground-coupled heat pump was rekindled. This more recent research and development has emphasized analytical computer techniques, some of which are based on the earlier theories, and the use of plastic piping. The majority of models have been for horizontal coil configurations. An overview of some of the computer design programs currently in use has been presented by Battelle (13). Complexity ranges from rule of thumb to steady-state and then transient conditions. The transient models are classified according to the number of dimensions treated in the analysis and the particular methodology. These methodologies include analytical, lumped parameter, finite difference and finite element techniques. Results of three of these models applied to a horizontal ground coil were compared by Fischer (14). These included one-dimensional analytical and lumped parameter models and a two-dimensional finite difference model.

The analytical program GSHP developed by Kalman (15) is based on the line-source theory and the integral is solved by a polynomial fit to tabular solutions. The American Heliothermal Corporation's lumped parameter model, AHGRND, treats the fluid and piping material as separate isothermal masses. The ground is divided into concentric volumes with constant far-field temperatures. The two-dimensional finite difference model, GROCS, is a product of the Brookhaven National Laboratory and uses a relatively small number of nodes. Temperatures of the tube fluid and the surrounding soil temperatures are given for time steps of one hour or less. The above programs were run for the first day of each month throughout a year and the resulting fluid temperatures were in good agreement. The GROCS program neglected the thermal resistance of the polybutylene tubing which contributed to a higher minimum and a lower maximum fluid temperature. Kalman's GSHP program had the advantage of considerably shorter execution time than did the other two programs.

Kanchanalai (16) investigated the application of finite-difference solution techniques to the conduction heat transfer to and from a heat pump ground-coupling device. He found that the fully implicit method with non-uniform grid spacing as described by Lilly and Croft (17) to be the most acceptable. A single tube with a given constant boundary condition was considered with concentric control volumes making one dimensional cylindrical coordinates most appropriate. The implicit method when applied to transient problems such as this one avoids stability problems associated with the explicit method, thus giving greater freedom in the choice of grid size and time increments. Relaxation of these parameter values helps maintain reasonable computer execution time but care must be taken since the grid size and time increment do affect the accuracy of the results. The tridiagonal matrix algorithm (TDMA), using gaussian elimination, was taken from reference (17) and combined with the non-uniform grid and cylindrical coordinates defined for the ground coil problem. Results obtained from this method compared quite well with the exact solution. This algorithm recommended by Kanchanalai is implemented in this study as well as in the studies described below.

A model of the 5 inch PVC annular tube configuration used in the Perkins project was presented by Al-Juwayhel (18). His methodology was based on an energy balance of a tube increment much like that suggested by Dusinberre (11). However, his energy balance was independent of the heat pump cycling. Heat transfer in the surrounding ground was computed using the method of Kanachanalai. A similar approach was taken by Joshi (19) who produced a simplified model of a U-tube device by assuming a single tube of equivalent diameter for a section actually comprised of two separate tubes. These works have provided a valuable foundation for this study.

Many of the models give reasonable results for continual and steady operation. However, test results published by Baxter, Abbatiello, and Minturn (20) indicate that the effects of ON/OFF cycling and frost accumulation can degrade heating performance of an air-coupled heat pump during the spring months by up to forty percent. Various studies have been made of these cyclic effects on the performance of the air-coupled heat pump. Testing by Miller and Jaster (21) show an overprediction of cyclic losses using the Department of Energy's algorithm. However, test results of Fagan (22) indicate that the same algorithm underestimates these cycling effects on the annual performance. Both studies propose the application of a degradation coefficient for the heat pump to correct for these losses.

Some effects of cycling on ground-coupled heat pumps have been considered by Braud (23), Bose (24) and others who have performed field tests of vertical ground-coupling devices. Resulting algorithms for determination of tube length as a function of other system parameters are suggested with consideration of an average ON/OFF cycle ratio. Effects of cycling on heat transfer coefficients are also investigated.

Many of the models available for the ground-coupling device consider long term results with average cycle ratios. However, simulations to determine energy use under normal operating conditions where an earth-coupled heat pump cycles at fractions of an hour may require a model that can predict tube outlet temperatures for these shorter cycle periods.

CHAPTER IV

SYSTEM DESCRIPTION

As part of the comparative study at Oklahoma State University of three different heat pump systems, a water source unit coupled to a ground coil was installed in a home in Perkins, Oklahoma. A more detailed descriptions of these homes can be found in reference (16). The rated capacity of this ground-coupled system are as shown below in Table I.

TABLE I

HEAT PUMP RATED CAPACITY

Model	Commandaire SWP-150
Heating Capacity	22,900
(Btu/hr)	70 deg F (EADB), and
	60 deg F (EWT) with 4 gpm flow
Cooling Capacity	19,500
(Btu/hr)	67 deg F (EAWB), and
	80 deg F (EWT) with 4 gpm flow

The original ground-coupling device used for study at the Perkins, Oklahoma site was a five inch Polyvinyl Chloride (PVC) casing fitted into a two hundred and fifty foot deep hole in the earth. Due to improper installation by the contractor, a major leak developed. Because leakage had been reported at other installations using PVC pipe, the device was replaced by a U-tube of 1-1/2 inch IPS Schedule 40 8600 Driscopipe made of high density polyethylene. This configuration has performed without trouble and has the following advantages:

1. Easier installation,
2. Less maintanence,
3. Reduced cost, and
4. Increased thermal conductivity.

This tubing fit into the existing hole with a diameter of roughly five inches. With the outside diameter of 1.90 inches, the maximum distance separating the two tube sections would be about one inch. However, at some points the tubes are in near contact and a reasonable average distance apart is 0.60 inches. These characteristics are summarized in Table II. Figure 3 is a schematic of the ground-coupled heat pump system which shows the original and current configurations for the ground-coupling device.

Soil in the Stillwater area is classified as Renfrow clay. Various properties of the soil were measured and are also shown in Table II. More detail on the soil and other project specifications are included in a report by Parker and Bose (25).

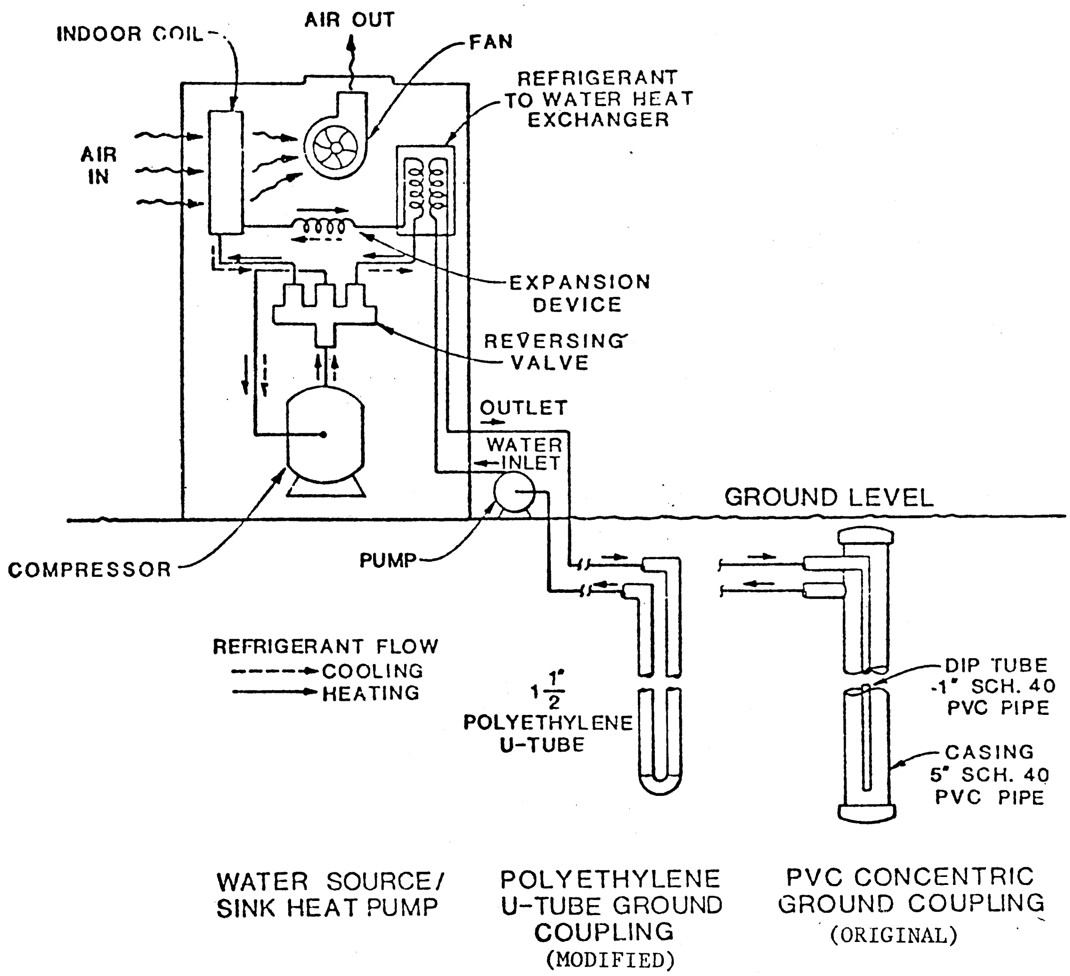

Figure 3. Original and Modified Ground-Coupled Heat Pump Systems

TABLE II

TUBE AND SOIL SPECIFICATIONS

Configuration	U-tube, 1-1/2 inch Schedule 40 1.54 inch I.D. 1.90 inch O.D. 250 feet of depth 0.60 inches average distance between tube sections
Material	High density polyethylene PE 3408 with conductance 0.226 Btu/(hr-ft-°F)
Soil	Renfrow clay Thermal conductivity: Dry .... 0.56 Btu/(hr-ft-°F) 15% Moisture .. 1.60 Btu/(hr-ft-°F) Volume-Heat Capacity: 40 Btu/(cubic ft - °F) Temperature at 65 ft: 62 °F

CHAPTER V

GROUND COIL SIMULATION MODEL

A mathematical model representing operation of the ground-coupling device was used to study the effects of various parameters on performance. This model was governed by the established laws and methods of heat transfer applied with carefully considered assumptions.

Model Assumptions

The complex transient nature of the heat transfer behavior of the ground coil system demands that a series of reasonable assumptions be made. The major assumptions made during the development of the model were as follows:

1. There was no heat transfer by radiation.
2. Only one-dimensional heat transfer by conduction in the radial direction was considered.
3. The thermal capacity of the tube wall was neglected to allow the heat transfer at the wall to approach steady state.
4. The tube was long enough that any end effects could be neglected.
5. There was perfect contact between the earth and tube.

6. The tube sections were separated by a constant average distance (see Table II) for each case considered.
7. Properties of the surrounding soil were constant at average values (see Table II).
8. Fluid properties were evaluated at an average bulk fluid temperature.
9. The fluid temperature across any tube cross-section was constant due to small tube radii.
10. The earth temperature at a large radial distance (r>30 ft) remained constant at the far-field value. This assumption was supported by experimental evidence as shown by Coogan (8).
11. All fluid and ground temperatures were initially at the far-field temperature.
12. When the heat pump was operating, the fluid flow rate was constant and the flow fully developed. The heat transfer between the heat pump and tube fluid was constant and instantaneous.
13. When the heat pump was off, the fluid was at rest and there was no heat transfer between the heat pump and tube fluid.

Model Background

The major components of the model are those that describe:

1. Heat transfer between the tube and the ground,

2. Heat transfer between the adjacent tube sections,

3. Conduction in the earth.

The determination of the heat transfer rates between the tube fluid and ground begins with the solution to the overall energy equation as applied to the tube increment shown in Figure 4. Following the assumption of Ramey (9) and others that the heat transfer within the tubing is steady state, the energy equation can be written as:

$$j_1 + gZ_1 + V_1^2/2 = j_2 + gZ_2 + V_2^2/2 + \dot{q}/w + \dot{W}/w \tag{1}$$

Since there is no shaft work and the flow rate of the incompressible liquid is constant, the above reduces to:

$$j_1 - j_2 = g(Z_2 - Z_1) + \dot{q}/w \tag{2}$$

From the definition of enthalpy for an incompressible liquid:

$$j_1 - j_2 = C(T_1 - T_2) + (P_1 - P_2)/\rho \tag{3}$$

By neglecting the friction due to viscous dissipation, the last term of equation (3) is equal to the change in fluid head giving:

$$j_1 - j_2 = C(T_1 - T_2) + g\Delta Z \tag{4}$$

Comparing equations (2) and (4), note that as the fluid flows down the tube, the increase in enthalpy due to increased pressure is approximately equal to the loss of potential energy. The same rationale applies to the

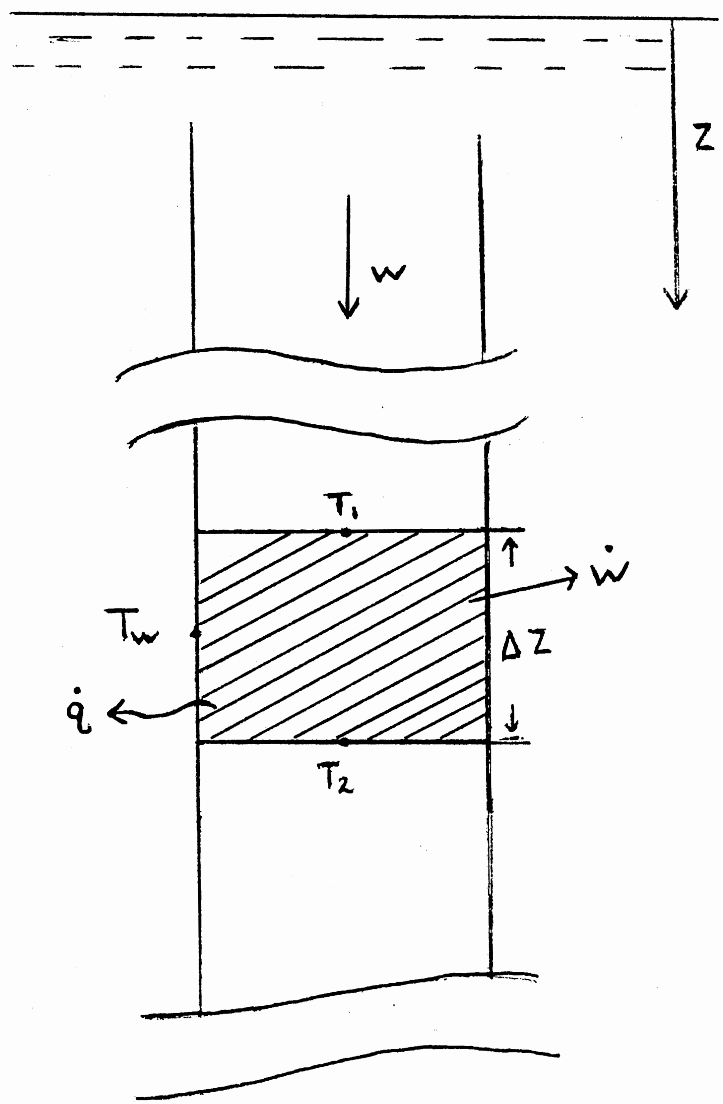

Figure 4. Tube Increment for Analysis

fluid flowing up the tube but in the opposite sense. Combining equations (2) and (4) yields:

$$\dot{q} = wC(T_1 - T_2) \tag{5}$$

An energy balance taken across the tube increment of length ΔZ shows that the rate of heat loss by the liquid as it flows through the control volume must equal the rate of heat transferred to the wall by convection. The convected energy is given by Newton's Law of Cooling:

$$\dot{q} = wC(T_1 - T_2) = hA(Tavg - Twall) \tag{6}$$

where $Tavg = (T_1 + T_2) / 2$

When the fluid is at rest, the rate of heat given up to the wall is that which decreases the temperature of the fluid of the control mass from its value at the beginning of the time period under study to the end of the same:

$$\dot{q} = \rho \ C \ Vol \ \frac{(Tavg - Tav)}{\Delta t} \tag{7}$$

where Δt is the change in time.

The extent to which the heat transfer between the downcomer and riser affects the outlet temperature is one of the points of interest of this investigation. Therefore, the tube interval surface area is divided into two portions; one surface area that exchanges heat with an infinite medium (the earth) which maintains a fixed temperature at a large radial distance, and another that exchanges heat with the adjacent tube section whose temperature varies with time. Figure 5 is a conceptual diagram of this idea.

To solve for transient conduction in a solid, suitable initial and boundary conditions must be known. During heat transfer between the tube sections, the boundary condition at the adjacent tube wall is not known and so steady state conduction is assumed. The rate of heat transfer can thus be approximated by Fourier's Law as:

$$q = -kA \left.\frac{\partial T}{\partial r}\right)_{r=R} = \frac{-kA_T\,(Tw - Twadj)}{\Delta x} \qquad (8)$$

where

	k	is the soil thermal conductivity,
	A_T	is the tube surface area facing the adjacent tube section,
	Tw,Twadj	are the tube wall temperatures across from one another,
	Δx	is an equivalent distance for heat transfer.

The equivalent distance is an average distance apart of points on the two tube surfaces. The variables shown in Figure 6 allow this distance to by found by:

$$d = R\cos(\theta / 2)$$

$$MAXDA = 2\,(R-d) + DA$$

$$\Delta x = \frac{MAXDA + DA}{2} \qquad (9)$$

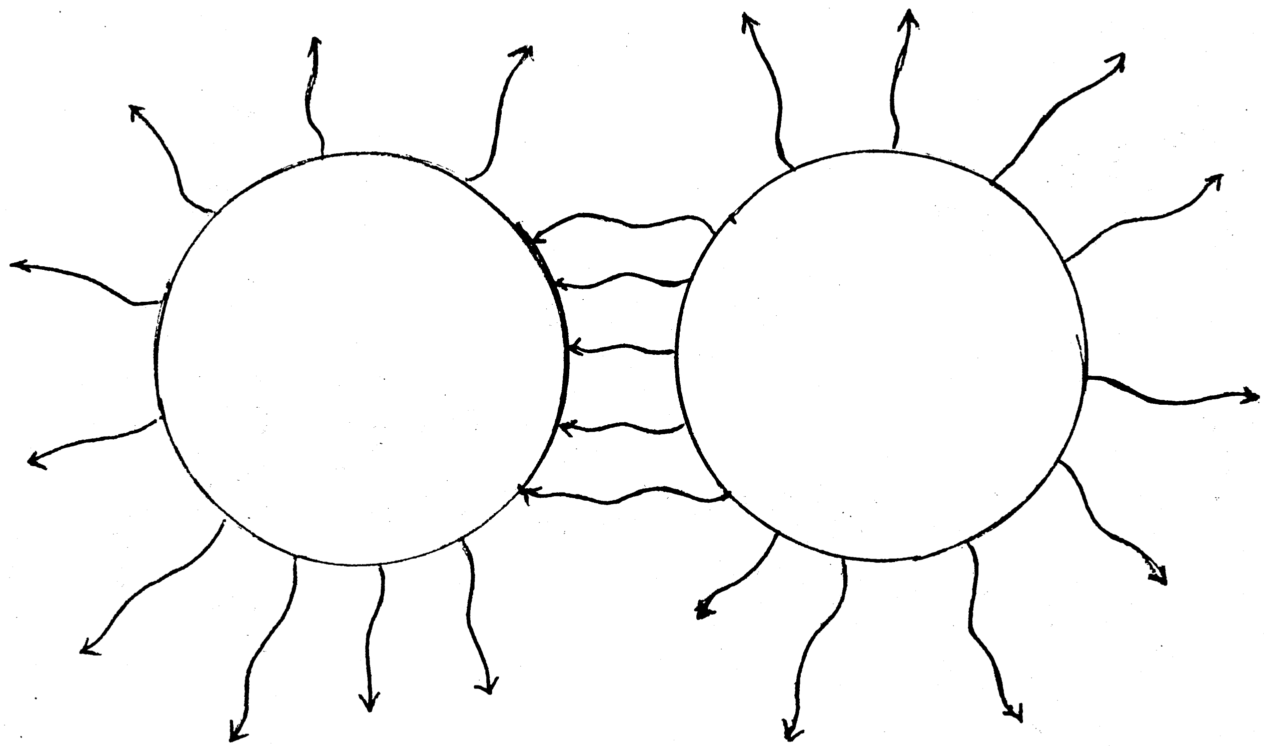

Figure 5. Schematic of U-Tube Heat Exchange

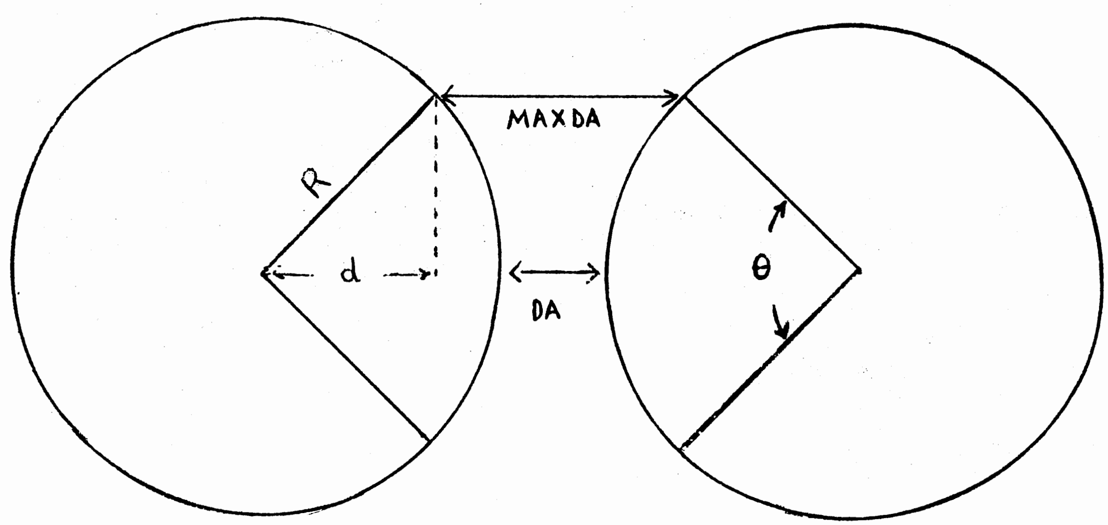

Figure 6. Diagram for Equivalent Distance

The transient conduction through the surrounding earth with constant properties and far-field temperature can be described by the following partial differential equation:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \tag{10}$$

with boundary conditions:

$$T(\infty, t) = T_i$$

$$-\left.\frac{\partial T}{\partial r}\right)_{r=R} = \frac{\dot{q}}{kA}$$

and initial conditions:

$$T(r,0) = T_i$$

Kanchanalai (16) defined a grid system and applied the fully implicit finite difference method to the solution of the above equation. A schematic of the control volume considered is shown in Figure 7 for an aribitrary section. An energy balance applied to this control volume yields the following:

Rate of conduction from I-1 to I

$$= kFA(I)\left[\frac{T(I-1) - T(I)}{DR(I)}\right] \tag{11}$$

Rate of conduction from I+1 to I

$$= kFA(I+1)\left[\frac{T(I+1) - T(I)}{DR(I+1)}\right] \tag{12}$$

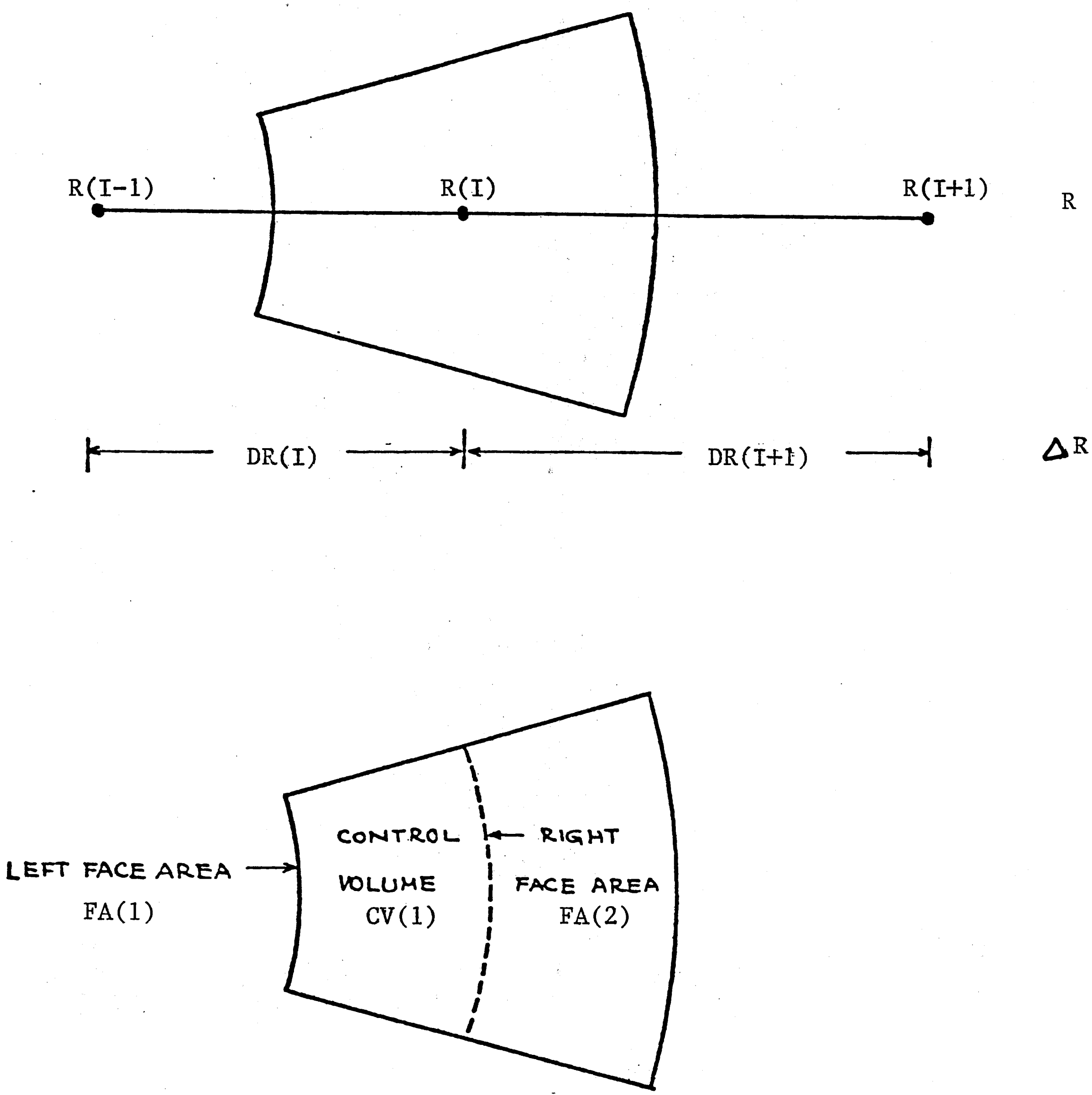

Figure 7. Grid System Definition

Rate of heat storage in ground

$$= \rho\, C\, (CV(I)) \left[\frac{TP(I) - T(I)}{\Delta t} \right] \tag{13}$$

The sum of equations (11) and (12) must equal (13):

$$kFA(I) \left[\frac{T(I-1) - T(I)}{DR(I)} \right] + kFA(I+1) \left[\frac{T(I+1) - T(I)}{DR(I+1)} \right]$$

$$= \rho\, C\, (CV(I)) \left[\frac{TP(I) - T(I)}{\Delta t} \right] \tag{14}$$

The boundary condition at the tube wall is approximated by:

$$\dot{q} = \frac{-kAg\ (Tw - Tg)}{DR(2)} \tag{15}$$

where Ag is the tube surface area facing the ground,

Tg is the temperature of the ground,

DR(2) is the distance to the ground point.

The interested reader is referred to reference 16 for more detail.

The rates of heat transfer from the fluid to the wall and then from the wall into the ground must then be equal with the given assumptions. The following equations result from this fact:

HEAT PUMP CYCLE ON:

Heat lost by the fluid = $wC(T_1 - T_2)$ (16)

Heat convected to the wall

= Heat conducted away from the wall

$$h(Tavg - Tw) = k(Tw - T) \tag{17}$$

where T is either Tg or Twadj .

HEAT PUMP CYCLE OFF:

Heat lost by fluid = Heat conducted away from wall

$$\rho \, C \, Vol \, \frac{(Tavg - Tav)}{\Delta t} = kA \, (Tw - Twadj) \frac{A_T}{A} + kAg(Tw - Tg)\frac{Ag}{A} \tag{18}$$

where A is the total surface area,

A_T is the surface area facing the adjacent tube, and

Ag is the surface area facing the earth.

Simulation Procedure

The ON/OFF cycling of the heat pump plays a major role in the method of heat transfer and an accurate description of the system's response to this cycling is of interest. The relationship between the fluid flow rate, tube radius, tube length, and amount of time the water is circulated must be preserved. Because of the energy balance's dependency on cycling, the fluid must either flow completely through a tube increment or be completely at rest within it. Therefore, when the heat pump cycles off, the fluid must stop at the end of a tube increment. The tube could theoretically be divided into enough tube intervals so that the fluid would meet the above requirement. For most applications, ten tube intervals was found to be sufficiently accurate. The on cycle period is adjusted so the water travels a distance at most five percent from the original stopping point. The cycle time is then adjusted to maintain the desired ratio of on time to cycle time. If it happens that the fluid travels one half or quarter tube length during an on time, the number of tube increments will be set at two or four respectively. After the number of intervals is set, the interval size can be subdivided by the input value of DIV. Although this may increase the accuracy of the results, it also significantly increases the execution time of the computer program. Tests showed that increasing the subdivision from one to two presented only a 0.7 percent difference in results. Care was taken to minimize the number of tube intervals.

The basic procedure involved in the simulation of the U-tube is outline below:

1. Define the heat transfer rate to the well fluid (+) or from the well fluid (-) based on the heat pump's mode of operaton:

   COOLING: Qwell = Qhouse + Wcomp

   HEATING: Qwell = -(Qhouse - Wcomp)

   where Qhouse is the heat extraction rate of the house and
   Wcomp is the compressor work

2. Approximate the heat transfer coefficient using the Dittus-Boelter equation or the Nusselt number based on the fluid temperature and flow characteristics at the tube inlet:

   For fully developed laminar flow:

   $$\frac{hD}{k} = 4.36$$

   For fully developed turbulent flow:

   $$\frac{hD}{k} = 0.023\ Re^{0.8}\ Pr^{n}$$

   where n = 0.4 Tw>Tavg (heating)
   n = 0.3 Tw<Tavg (cooling).

   For the simulations performed in this analysis, the flow was turbulent with a Reynolds number near 8200.

3. Define the grid system and all associated parameters.
4. Determine the average distance between tubes for heat transfer calculation (see Figure 6).
5. Calculate the velocity of the fluid and the fraction of tube length it travels beyond the last completed

loop.

6. Define the number of tube intervals and determine the number of tube intervals the fluid travels during the circulation period.

7. Define the differential time of study for the current state of operation (ON/OFF) and approximate the total rate of heat transfer (Q). Estimate the distribution of this heat rate between that to the earth and that to the adjacent tube section:

   To ground : Q1 = Q * (1.0 - Portion)

   To tube : Q2 = Q * Portion

   where Portion is that part of the tube surface that exchanges heat with the adjacent tube.

8. With this approximation of heat transfer to the ground, Q1, determine the new temperature distribution in the surrounding earth using the TDMA method.

9. If the heat pump is on, use the approximated total heat transfer rate, Q, to calculate the new interval outlet temperature:

   TN = TF - Q / (FLRATE * CPF)

   where TF is the inlet temperature to the tube interval,

   FLRATE is the fluid mass flow rate,

   CPF is the constant specific heat of the fluid.

The bulk fluid temperature can then be found:

TAVG = (TF + TN) / 2

This temperature along with the estimated heat transfer rates Q1 and Q2 allow determination of the tube wall temperatures:

TWALL(J) = TAVG - Q(J) / H * AREA(J)

where Q(J) is either Q1 or Q2,

H is the average heat transfer coefficient,

AREA(J) is the tube surface area of either the portion that sees the earth or the portion that sees the adjacent tube section.

If the heat pump is off, the total heat transfer rate is used to calculate the new bulk fluid temperature for the tube interval:

TAVG = TAV - (Q * DTIME) / (RHO * CPF * VOL)

where TAV is the average fluid temperature at the beginning of the time period,

DTIME is the time period of study for the off time,

RHO is the density of the fluid,

VOL is the volume of fluid in the tube interval.

Decrement the outlet temperature of this section to reflect this lowered average temperature. Assume that the tube wall temperature approaches the bulk fluid temperature.

10. Using the temperature gradient defined by the average wall temperature and that of the adjacent tube or the ground temperature at a designated node, calculate the conduction heat transfer rate from the interval of study.
11. Compare the heat transfer rates from the fluid to the tube wall, which were originally estimations, to the resulting rates of heat conduction from the tube wall. If these are equal, or within five percent of one another, the rates and resulting temperatures are acceptable. If not, adjust the estimations and repeat steps 8 through 11.
12. Repeat steps 8 through 11 for each tube interval to determine the new tube outlet temperature.
13. Increment the time step and, if the fluid is moving, the fluid position as well.
14. If at the end of an on cycle or complete cycle time, print the current tube outlet temperature.
15. If the current cycle is on and it is not the end of the on time, shift the exit temperature from section I to the inlet temperature for section I+1. The new tube inlet temperature is given by:

    TFIN = TFOUT + QWELL / (FLRATE * CPF)

    If it is the end of an on time, the fluid stops and there is no heat exchange with the heat pump.

    If the current cycle is off and it is not the end of the off cycle, the fluid remains at rest with no heat exchange with the heat pump. If it is the end of the off cycle, shift the temperature from exit of section I to the inlet temperature for section I+1 and compute as shown above.

16. Recalculate the heat transfer coefficient based on a new average bulk fluid temperature.
17. Increment the total time step and repeat steps 7 through 16 until the desired simulation time has elapsed.

The computer program that performs the above procedure is listed in Appendix A. The symbolic flow chart of the program logic is found in Appendix B.

CHAPTER VI

SIMULATION RESULTS

A number of simulation runs were made to verify the desired cyclic behavior of the system, to consider the effect of those factors identified in the OBJECTIVE section, and to compare results with existing design methods. Figure 8 shows tube outlet temperatures during the first hour of operation with a heat pump cycle of five minutes on and five minutes off. The simulation results which are the basis for this plot, and some of the following plots, are located in Appendix C. The fluid temperature throughout the tube is initially 62.5 degrees Fahrenheit and the tube inlet temperature is 68 degrees Fahrenheit. With a circulation period of five minutes, the water travels one half the tube length. Therefore, the slight temperature rise at the end of the first five minutes, designated by point A, is due solely to heat exchange between the downcomer and riser. The first off period, point B, creates a very slight temperature increase due to the same effect. The next circulation period brings the warm water to the tube outlet, point C, but at a lowered temperature than its original input value. The next off period has a slight cooling effect shown at point D. Then the water that has actually

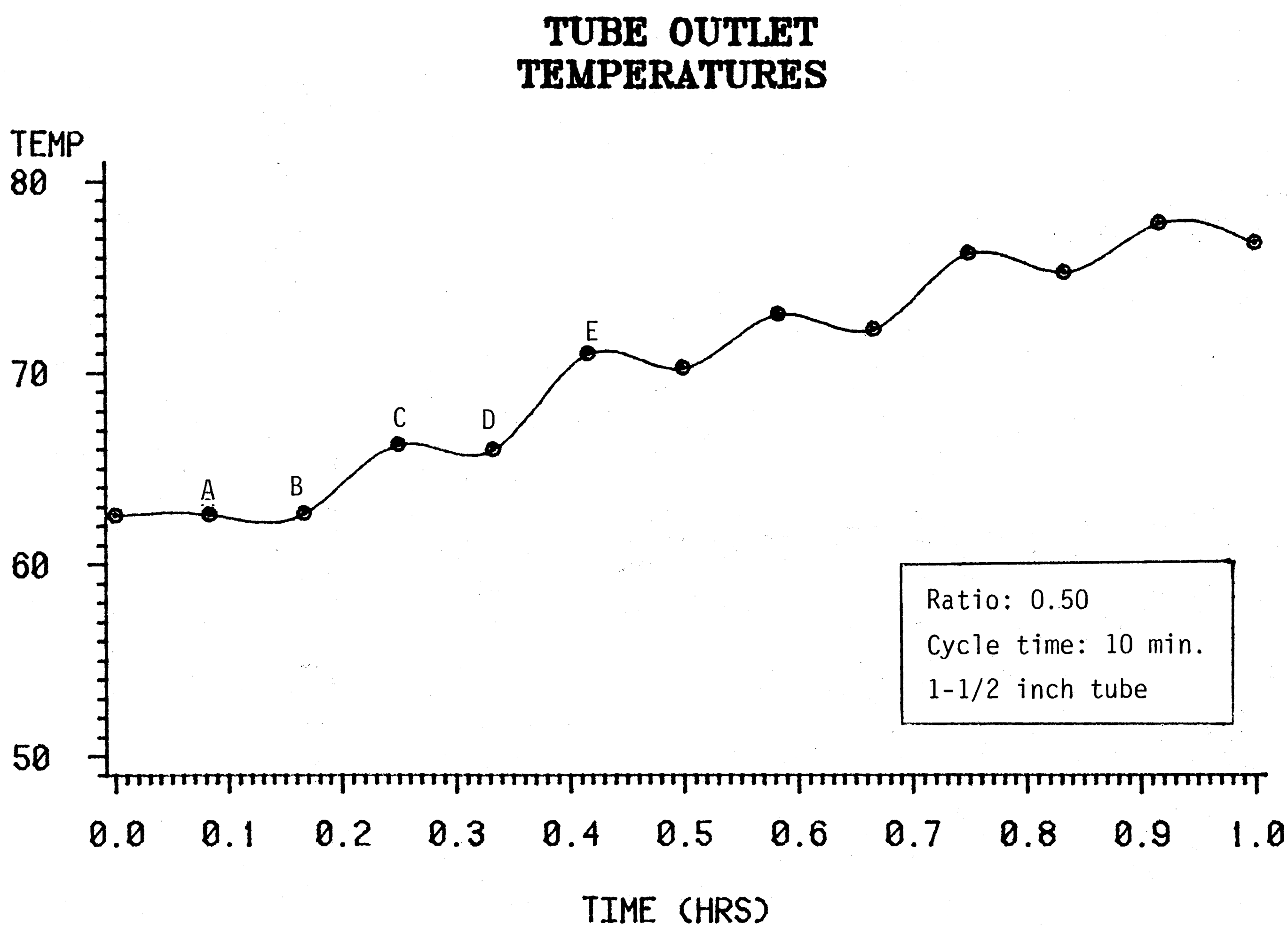

Figure 8. Start-up Period of Cyclic Operation

absorbed heat from the heat pump makes its way to the tube outlet and the sharpest temperature rise is observed at point E. From this time on, the tube outlet temperature follows a rise and fall pattern as the maximum outlet temperature steadily increases.

The first factor considered was the ratio of cycle on time to the total cycle time. Simulations were run for twelve hours with cycle on times of five, ten and twenty minutes. With the on time unchanged, the ratio was given values of 0.25, 0.50, and 0.75. Results of these simulations are summarized in Figure 9. As expected, the larger the ratio of on time to cycle time, the higher the tube outlet temperature for a given on time. This increase in temperature is about 12 °F from a ratio of 0.25 to 0.50 and about 9 °F from 0.50 to 0.75. At a constant ratio, the length of the cycle on time has minimal effect and this effect diminishes as the on time gets longer. Note that for a ratio of 0.75 the difference in outlet temperature between an on time of ten and twenty minutes is about 1 °F. This trend is shown clearly in Figure 10. Runs were made for cycle on times of 5, 10 and 20 minutes each with a cycle ratio of 0.50. Again, it can be seen that the difference between a 10 and 20 minute on time at a constant ratio is about 1 °F. As the on time decreases, the difference is somewhat more but still relatively small.

The effect of the distance between the legs of the U-tube is shown in Figure 11. The outlet temperature was

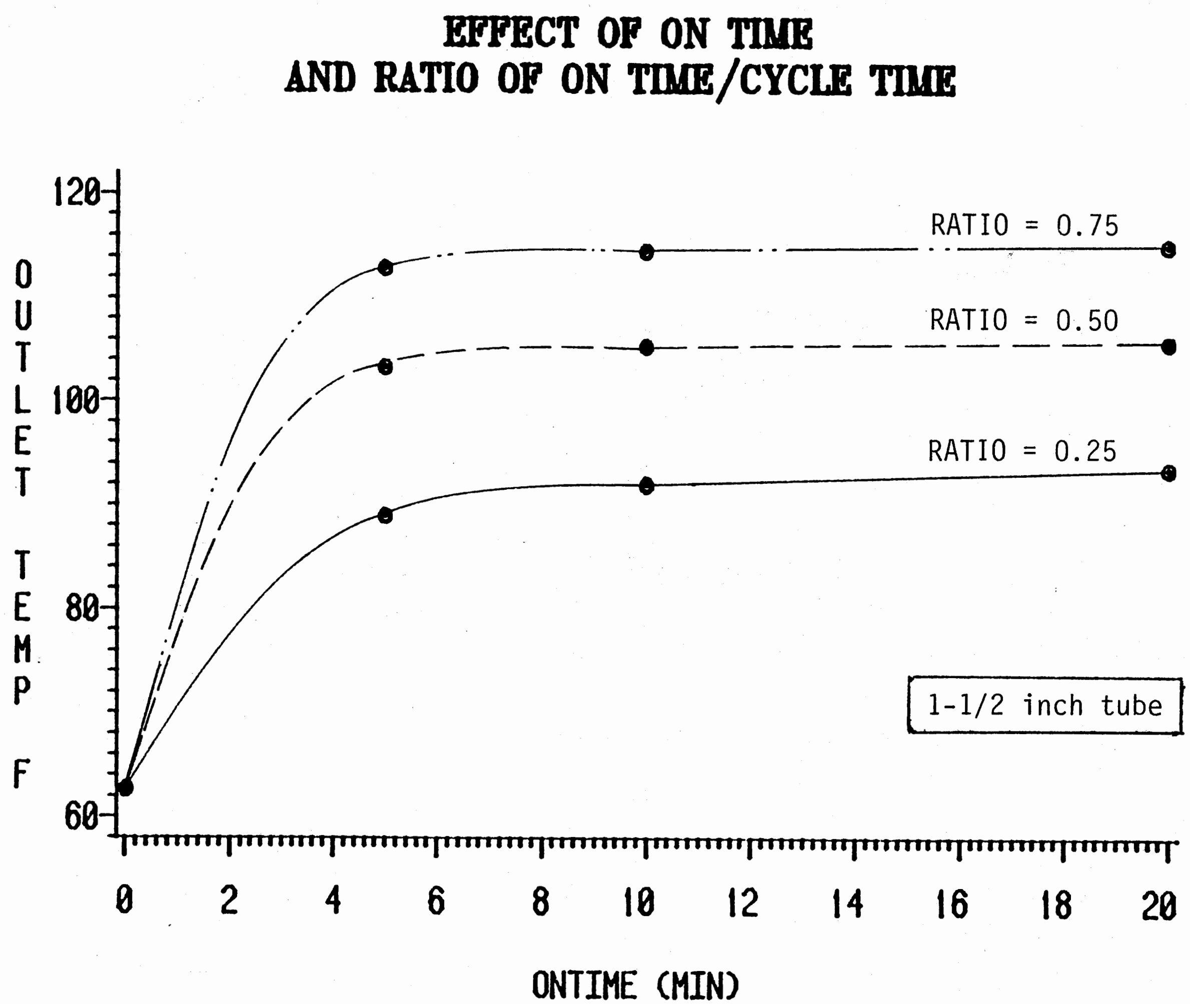

Figure 9. Variation of Cycle Ratio

EFFECT OF ONTIME

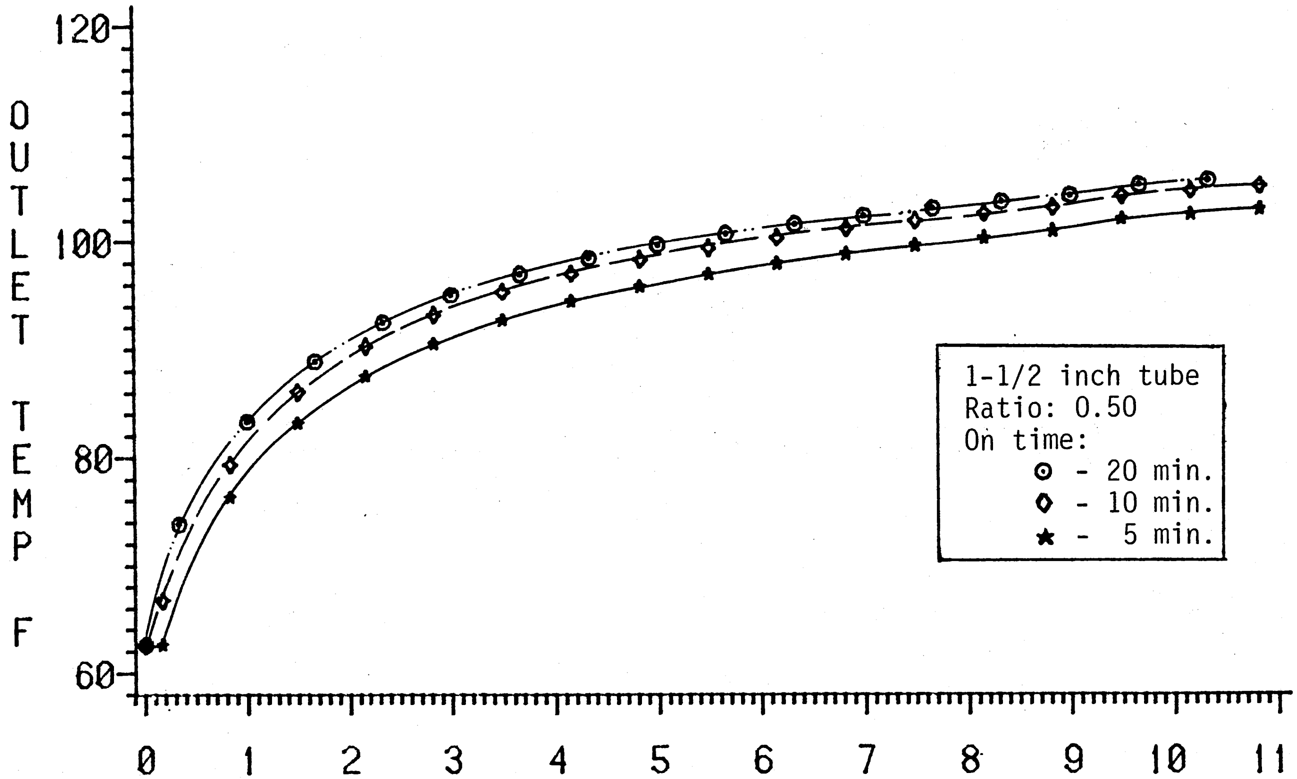

Figure 10. Variation of Cycle On Time

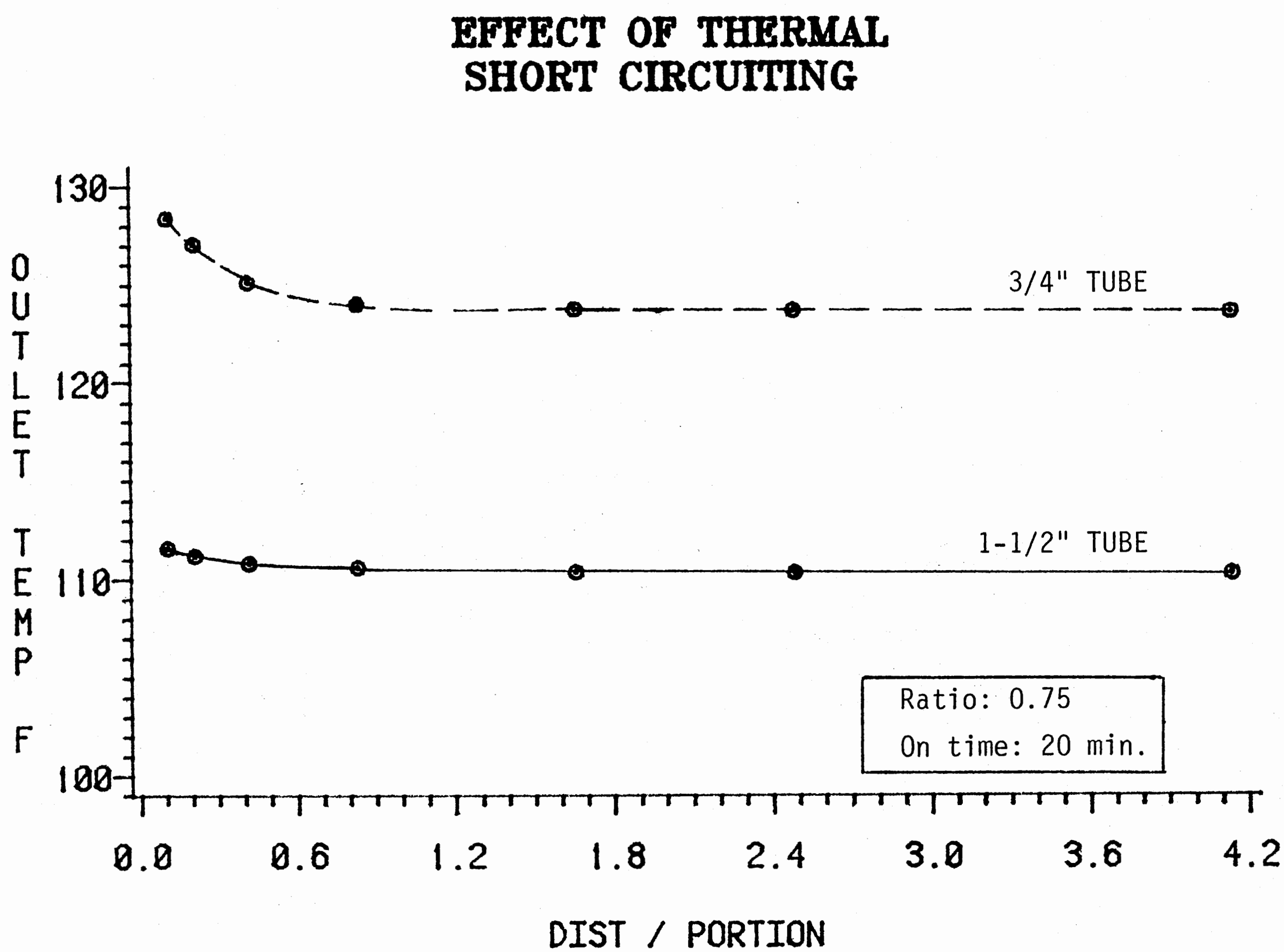

Figure 11. Variation of Distance Between Tube Legs

plotted against the dimensionless quantity defined below:

DIST / PORTION

where DIST is the average distance between the downcomer and riser.

PORTION is equal to θR . These quantities are described by Figure 6.

The relationship between these parameters is intuitively an inverse one. Increasing the distance apart has the same effect as decreasing that portion of the face area that exchanges heat with the other leg of the U-tube. Figure 11 indicates that for the 1 1/2 inch tube and a ratio greater than 1.5, the heat transfer between the tube legs is negligible. If θ is 180 degrees then spacing the tubes at least 3 1/2 inches apart would minimize any thermal short circuiting. Soil conductivity values were given in Table II and it should be noted that an average value of 0.82 Btu/(hr-ft-°F) was used for all simulations discussed here.

Figure 11 also illustrates the effect of the smaller tube radius of 3/4 inches. Short circuiting is no longer apparent after a ratio of about 2.0. Again for θ of 180 degrees, a tube spacing of 2 1/2 inches would be sufficient to neglect the heat transfer. The outlet temperature for this smaller tube is about 14 degrees higher where thermal short circuiting is negligible. However, this tube shows a greater sensitivity at ratios less than 0.5.

The last set of simulations were run at various tube lengths. The tube outlet temperatures are plotted in Figure 12 as a function of depth, which is one half the tube length, per ton of heat extraction. To maintain a fluid temperature under 120 degrees Fahrenheit with a 1 1/2 inch tube, a U-tube with a minimum depth of 132 feet is required. If the 3/4 inch tube is used, the minimum depth increases to about 165 feet. Temperature differences between the two tube sizes varies from about 15 °F at depth/ton ratios near 100 down to about 10 °F at ratios closer to 200. This difference reduces even further as the depth/ton ratio increases.

The ideal verification for the computer model would be to compare simulation results to measured temperatures under similar conditions. Future plans are for such measurements to be made but a comparison at this point in time is not possible. Bose (24) has developed an earth coil design procedure from experimental data together with the computer model GROCS III. The reader is referred to reference (24) for details. Calculations using this procedure with parameters defined to approximate those of this study result in a tube depth of 198 feet per ton for a 33 degree temperature rise. From Figure 12, a depth of 212 feet per ton is needed to keep the water under 95 °F when it is initially at 62 °F. Thus the program results appear to be in reasonable agreement with the design procedure.

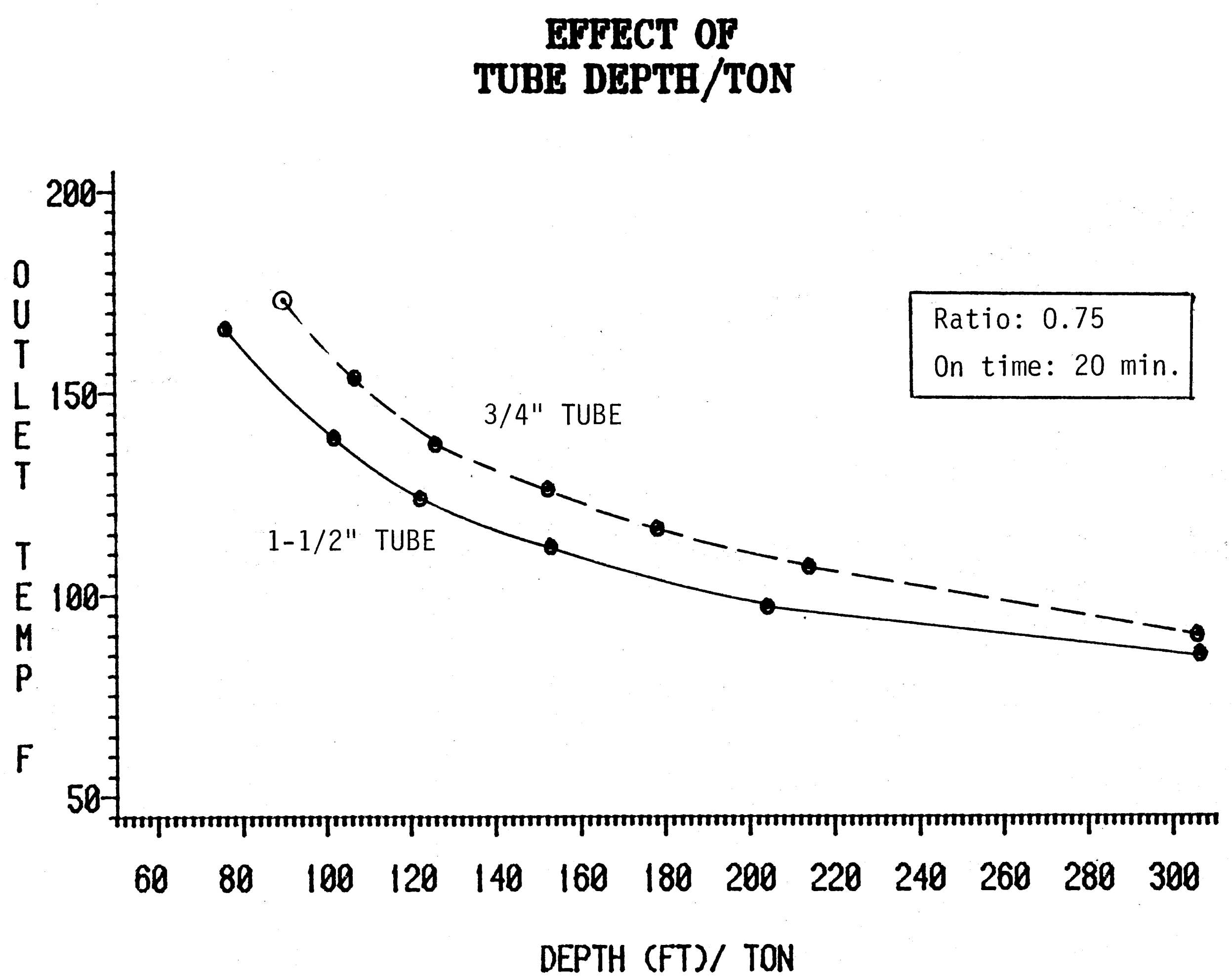

Figure 12. Variation of Tube Depth

CHAPTER VII

CONCLUSIONS

From the results presented above, the ratio of time that the heat pump is on to the total cycle time is significant when considering the tube outlet temperature's effect on the coefficient of performance of the heat pump. However, the actual length of the on time has less effect especially as the time of operation proceeds. These results give support then to those models that assume an average cycle ratio over a relatively long period of time.

The effect of thermal short circuiting is significant and is dependent on the tube radius. As the surface area available for exchanging heat with the ground is decreased, the heat exchange between the tube legs plays a greater role. Since the resistance of the tube wall was neglected in this analysis, the rate of heat transfer between the tube legs may actually be less significant than is shown here. In the cooling mode, this decrease in thermal short circuiting would result in slightly lower outlet temperatures. Including the insulating effect of the wall would decrease heat transfer between the tube fluid and the outside wall resulting in an increase in outlet temperatures. Therefore, these temperature changes would tend to offset one another and reduce the overall effect of introducing the tube wall resistance. Sample calculations using the method by Bose (24) described above for a typical U-tube configuration, such as that used in this study, indicate that neglecting the wall resistance during cooling would yield a temperature increase of about 28 degrees Fahrenheit when it otherwise would have been 33 degrees. However, this does not include the effect of an increase in thermal short circuiting which could reduce this to a 3 or 4 degree drop. Although the number of degrees involved is small, they account for about ten percent of the total temperature difference. Modification of the program to account for the tube wall resistance could be made. Changes to the program logic that facilitates solution convergence may then be necessary.

Tube length is an important factor in the performance of the ground-coupling device. Benefits of increased tube length diminish after a tube depth of about 200 feet per ton. Of course a cost-benefit analysis would be needed to determine the optimal tube length. This analysis should include the various tube dimensions since the performance of the smaller tube does approach that of the larger tube at sufficient tube lengths. At some given well depth, the decreased performance of the heat pump as a result of a higher outlet temperature from a smaller tube radius would be offset by a savings in tube and installation costs.

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APPENDIXES

APPENDIX A

LISTING OF COMPUTER PROGRAM

```
//COMPILE   JOB (?????,000-00-0000),'SOL',TIME=(0,10),CLASS=A,
// NOTIFY=*,MSGCLASS=X
/*PASSWORD ????
//  EXEC FORTVCL,REGION.FORT=1000K
//FORT.SYSIN DD *
C
C
C  ******************************************************************
C  ****                                                          ****
C  ****   VERTICAL EARTH-WATER HEAT EXCHANGER                    ****
C  ****                                                          ****
C  ******************************************************************
C
C
C
C
C       PURPOSE:
C             TO COMPUTE THE TUBE EXITING FLUID TEMPERATURE BY
C             APPLICATION OF AN ENERGY BALANCE ON SUCCESSIVE TUBE
C             INCREMENTS.  HEAT TRANSFER BETWEEN THE TUBE AND THE
C             GROUND AS WELL AS THE TUBE AND OPPOSITE TUBE SECTION
C             ARE FOUND BY ITERATIVE TECHNIQUES. GROUND TEMPERATURE
C             GRADIENT MAKES USE OF FINITE DIFFERENCE METHODS.
C
C       NOTE: THE PHYSICAL PROPERTIES OF THE SOIL ARE ASSUMED CONSTANT
C             IN THIS PROGRAM.
C
C
C          DESCRIPTION OF INPUT/OUTPUT PARAMETERS:
C          INPUT-
C              HOURS - HOURS OF SIMULATION
C              RATIO - FRACTION OF THE HEAT PUMP CYCLE THE HEAT PUMP
C                      IS ON AND THE WATER CIRCULATING.
C              ONTIME- PERIOD OF TIME THE HEAT PUMP IS OPERATING. (HR)
C              CYTIME- CYCLE TIME  (HR)
C              DIV   - NUMBER BY WHICH THE TIMES (ON AND OFF) ARE TO BE
C                      DIVIDED
C              MODE  - EITHER COOLING (1) OR HEATING (2)
C              TFIN  - TEMPERATURE OF THE FLUID ENTERING THE WELL. (F)
C              CAP   - CAPACITY OF HEAT PUMP (BTU/HR)
C              COP   - COEFFICIENT OF PERFORMANCE OF HEAT PUMP
C              KPRINT- EXTENT OF INFORMATION TO BE PRINTED.
C                      MINIMUM - 0     MAXIMUM - 1
C              IPLOT - OUTLET TEMPERATURES STORED FOR PLOTTING
C                      NO - 0      YES - 1
C              PORTON- THAT PORTION OF THE TUBE THAT EXCHANGES HEAT
C                      WITH THE ADJACENT TUBE INCREMENT
C              DA    - THE DISTANCE BETWEEN TWO ADJACENT TUBE INCREMENTS
C
C
C
C         OUTPUT-
C               TFOUT - TEMPERATURE OF THE FLUID EXITING THE WELL. (F)
C
C
      DIMENSION R(33),DR(50),RL(50),FA(50),CV(50),A(50),TAVG(50),TAV(50)
      DIMENSION B(50),C(50,50),CC(50,50),D(50),TF(50),TN(50),TGROND(50)
      DIMENSION TWALL(2,50),OLDTW(2,50)
      DIMENSION INT(3)
      NAMELIST/INPUT/CYTIME,ONTIME,RATIO,HOURS,DIV
      NAMELIST/INPUT2/MODE,TFIN,CAP,COP,KPRINT,IPLOT
      NAMELIST/INPUT3/PORTON,DA,DEPTH,RO
      DATA PHI,N,RRL,EPSR/3.14159,32,30.0,1.1/
C
C         WELL PARAMETERS
C
C         RO  -  WELL RADIUS  (FT)
C         ALPHA  -  THERMAL DIFFUSSIVITY OF SOIL  (SQ.FT/HR)
C         DEPTH  -  WELL DEPTH  (FT)
C         COND  -  THERMAL CONDUCTIVITY OF SOIL  (BTU/HR-FT-F)
C         CPF  -  SPECIFIC HEAT OF WELL FLUID  (BTU/LB-F)
C
```

```
C
      DATA ALPHA,COND,CPF/0.0290,0.82,
     &                        1.0/,RHO/62.0/
C
C     DEFINE MAXIMUM NUMBER OF TUBE INTERVALS(MINT) AND POSSIBLE
C     NUMBER OF INTERVALS (2,4,10)
C
      DATA MINT/10/,INT/2,4,10/
C
C
C
C
C     PROGRAM MAIN VARIABLES
C
C     TF(I)     -  TEMPERATURE OF THE FLUID INTO THE ITH INCREMENT
C                  DURING THIS TIME STEP
C     TN(I)     -  TEMPERATURE OF THE FLUID OUT OF THE ITH INCREMENT
C                  DURING THIS TIME STEP ( WILL BECOME THE ENTERING
C                  TEMPERATURE TO THE INCREMENT DURING THE NEXT TIME)
C     TAVG(I)   -  AVERAGE TEMPERATURE OF THE FLUID THIS TIME STEP
C     TAV(I)    -  AVERAGE TEMPERATURE OF THE FLUID PREVIOUS TIME STEP
C     C(I,K)    -  SOIL TEMPERATURE IN THE ITH INCREMENT AND THE KTH
C                  RADIAL SECTION SURROUNDING WELL - THIS TIME STEP
C     CC(I,K)   -  SAME AS C(I,K) BUT FOR PREVIOUS TIME STEP
C     Q         -  RATE OF HEAT TRANSFER TO/FROM THE WELL WATER
C     QON       -  AS ABOVE BUT FOR PREVIOUS ONTIME
C     QOFF      -  AS ABOVE BUT FOR PREVIOUS OFF TIME
C     Q1        -  RATE OF HEAT TRANSFER TO THE GROUND
C     Q2        -  RATE OF HEAT TRANSFER TO THE ADJACENT TUBE
C     QCONV     -  RATE OF HEAT TRANSFER TO/FROM THE WELL WALL
C     QWELL     -  RATE OF HEAT TRANSFER FROM THE HEAT PUMP TO THE
C                  WELL WATER
C
C
C
C  READ NUMBER OF CASES TO BE PROCESSED AND FLOW RATE TO BE USED
C
      READ(5,1000) NCASES,GGPM
C
C  START OF PROCESSING FOR EACH CASE
C
      DO 500 ICASE=1,NCASES
C
C  INITIALIZE VALUES
      TIME=0.0
      KOUNT1=0
      READ(5,INPUT)
      READ(5,INPUT2)
      READ(5,INPUT3)
C   CHECK FOR HEATING OR COOLING MODE - IF NOT STOP
      IF(MODE.EQ.1.OR.MODE.EQ.2) GO TO 10
      WRITE(6,1005)
      GO TO 500
C
C   WRITE OUT INPUT PARAMETERS FOR VERIFICATION
C
   10 WRITE(6,1010) CYTIME,ONTIME,RATIO,HOURS,DIV,TFIN,CAP,COP,GGPM
      IF(MODE.EQ.1) WRITE(6,1015)
      IF(MODE.EQ.1 .AND. TFIN.LT.66.0) WRITE(6,1020)
      IF(MODE.EQ.2) WRITE(6,1025)
      IF(MODE.EQ.2 .AND. TFIN.GT.54.0) WRITE(6,1030)
C
C     DEFINE TUBE LENGTH TO BE TWICE TUBE DEPTH
C
      TLEN = 2.0 * DEPTH
      FLRATE=497.3*GGPM
      GFLUX=FLRATE/(PHI*RO*RO)
C
C     CALCULATE THE HEAT TRANSFER TO/FROM THE WELL BASED ON
C     THAT FROM THE HOUSE AND THE COP OF THE COMPRESSOR.
C
C     QWELL(+) - IN     QWELL(-) - OUT
```

```
C
      IDIV = DIV
      QHOUSE = CAP
      WCOMP = QHOUSE/COP
      IF(MODE.EQ.1) QWELL = QHOUSE + WCOMP
      IF(MODE.EQ.2) QWELL = -(QHOUSE - WCOMP)
C     USE QWELL AS FIRST APPROXIMATION OF Q (HEAT RATE FROM THE WATER)
      QON = QWELL/10.
      QOFF = .10*QON
      QTOTAL=0.0
C
C     DETERMINE THE HEAT TRANSFER COEFFICIENT
C
      WMU=8.3574-.18457*TFIN+.2332E-02*TFIN**2.-.17931E-04*TFIN**3.0
     &    +.81845E-07*TFIN**4.-.20274E-09*TFIN**5.+.20919E-12*TFIN**6.
      WPR=27.51876-.65809*TFIN+.85657E-02*TFIN**2.-.66433E-04*TFIN**3.
     &    +.30315E-06*TFIN**4.-.74791E-09*TFIN**5.+.7675E-12*TFIN**6.
      WK=WMU*CPF/WPR
      RED=2.0*GFLUX*RO/WMU
      IF(RED.LE.2000.0) H=4.364*WK/(2.0*RO)
      IF(RED.GT.2000.0.AND.QWELL.GT.0.0) H=0.023*(RED**0.8)*(WPR**0.3)
     &                                   *WK/(2.0*RO)
      IF(RED.GT.2000.0.AND.QWELL.LE.0.0) H=0.023*(RED**0.8)*(WPR**0.4)
     &                                   *WK/(2.0*RO)
C
      IF(KPRINT.EQ.1) WRITE(6,1035) H
C
C        SET THE VALUE OF RADIUS AND RADIUS INTERVALS.
C        DOMAIN SIZE=30.0 FT. FROM WELL SURFACE.
C
      NP1=N+1
      DRAPP=RRL/N
      R(1)=0.0
      DO 50 I=2,NP1
      R(I)=R(I-1)+DRAPP
      DRAPP=DRAPP*EPSR
   50 CONTINUE
      FACTOR=RRL/R(NP1)
      DO 60 I=2,NP1
      R(I)=R(I)*FACTOR
      DR(I)=R(I)-R(I-1)
   60 CONTINUE
      IF(KPRINT.EQ.0) GO TO 70
C
C        PRINT THE VALUES OF NON-UNIFORM GRID
C
      WRITE(6,1037)
      WRITE(6,1038) (I,DR(I),I=2,NP1)
   70 DO 80 I=1,NP1
      R(I)=R(I)+RO
   80 CONTINUE
      IF(KPRINT.EQ.0) GO TO 90
C
C        PRINT THE VALUES OF RADIUS
C
      WRITE(6,1040)
      WRITE(6,1042) (I,R(I),I=1,NP1)
      WRITE(6,1045)
C
C        CALCULATE RADIUS AND FACE AREA OF LEFT FACE OF THE CONTROL
C        VOLUME
C
C
   90 RL(1)=R(1)
      DO 100 I=2,NP1
      RL(I)=(R(I)+R(I-1))/2.0
      FA(I)=2.0*PHI*RL(I)
  100 CONTINUE
C     DETERMINE THE FACE AREA THAT SEES THE OTHER TUBE
C     NOTE THAT THE AVERAGE DISTANCE BETWEEN TUBES IS 'DA'
C
C        BUILD UP CONTROL VOLUME
```

```
C
      CV(1)=PHI*(RL(2)*RL(2)-RL(1)*RL(1))
      DO 103 I=2,N
      CV(I)=PHI*(RL(I+1)*RL(I+1)-RL(I)*RL(I))
  103 CONTINUE
C
C     DIVIDE THE CONTROL VOLUME INTO THAT PORTION THAT "SEES"
C     THE GROUND AND THAT PORTION THAT "SEES" THE OTHER TUBE
      DO 105 I = 1,N
      CV(I) = CV(I) * (1.0-PORTON)
      FA(I) = FA(I) * (1.0-PORTON)
  105 CONTINUE
C
C     DETERMINE THE AVERAGE DISTANCE APART BETWEEN THE TWO
C     ADJACENT TUBE SECTIONS.
C
      FAT = 6.28318*RO * PORTON
      THETA = 6.28318*PORTON
      XDA = 2*(RO-(RO*COS(THETA/2.0))) + DA
      DA = (XDA+DA)/2.0
C
C     DIVIDE THE TUBE INTO A NUMBER OF INTERVALS(NINT) FOR
C     STUDY SUCH THAT TEN INTERVALS WILL BE APPROXIMATELY
C     THE DISTANCE TRAVELLED BY THE WATER DURING ONE ON CYCLE.
C     USE A MINIMUM OF TEN INTERVALS.
C
C
C     FIND THE VELOCITY OF THE FLUID AND THUS THE DISTANCE
C     THE FLUID WILL TRAVEL DURING A GIVEN ONTIME.
C     UNITS: V (FT/HR) DIST (FT)
C
      V = 9.6638 * GGPM / (PHI*RO*RO)
      DIST = V * ONTIME
C
C     DETERMINE THE NUMBER OF TIMES THE FLUID WILL COMPLETE
C     THE TUBE LOOP DURING THE ON TIME.
C
      K = DIST/TLEN
C
C     NOW DETERMINE HOW FAR THE FLUID GOES BEYOND THE
C     FINAL COMPLETE LOOP.
C
      DELTAZ = DIST - (TLEN * K)
C
C     IF DELTAZ IS LESS THEN FIVE PERCENT OF THE TUBE LENGTH
C     CALL IT AN EVEN NUMBER OF LOOPS COMPLETED AND USE
C     TWO TUBE INTERVALS FOR STUDY.
C
      IF(DELTAZ .GT. 0.05*TLEN) GO TO 115
      K1=0
      FRACT=0.0
      NINT = 2
      GO TO 122
C
C     FORCE THIS DISTANCE TO BE THE NEAREST 1/NINT  OF
C     TUBE LENGTH. FIVE PERCENT ERROR IS ALLOWABLE.
C
  115 DO 118 J=1,3
      NINT = INT(J)
      DO 117 K1=1,NINT
      XK1 = K1
      FRACT = XK1 / NINT
      IF(ABS(FRACT*TLEN-DELTAZ).LE.0.05*DELTAZ) GO TO 122
  117 CONTINUE
  118 CONTINUE
C
C     FIVE PERCENT ERROR WAS NOT MET SO MUST NOW
C     FORCE TUBE FLOW TO CLOSEST MINT OF TUBE LENGTH
C
      XINT = MINT
      TINC = TLEN * 0.5 /XINT
      DO 119 K1 = 1,MINT
```

```
      K2 = K1-1
      THI = TLEN * K2 / XINT + TINC
      IF(DELTAZ .LT. THI) GO TO 120
  119 CONTINUE
      K = K + 1
      K2 = 0
C
C     FLUID MUST TRAVEL A MINIMUM OF  1/MINT
C
  120 IF(K2.EQ.O.AND.K.NE.O) K2 = 0
      IF(K2.EQ.O.AND.K.EQ.O) K2 = 1
      FRACT = K2 / XINT
      K1 = K2
      NINT = XINT
C
C     WRITE NUMBER OF INTERVALS
C
  122 WRITE(6,1048) NINT
C
C     CALCULATE THE ADJUSTED DISTANCE TRAVELLED BY THE FLUID
C     DISTANCE = COMPLETE LOOPS + PARTIAL LOOPS
C     WRITE OUT THE NUMBER OF TUBE INTERVALS THAT THE FLUID
C     PASSES THROUGH DURING AN ON TIME (NDIV)
C
      DIST = K * TLEN + FRACT*TLEN
      NDIV = K*NINT + K1
      WRITE(6,1049) NDIV
C
C     CALCULATE THE ADJUSTED ON TIME AND CYCLE TIME ACCORDINGLY
C
      ONTIME = DIST / V
      CYTIME = ONTIME/RATIO
      WRITE(6,1050) ONTIME,CYTIME,RATIO
      WRITE(6,1052)
C
C     DETERMINE THE TUBE INTERVAL SIZE FOR STUDY (DELTAZ),
C     THE TOTAL NUMBER OF TUBE INTERVALS AND THE NUMBER PASSED
C     DURING AN ONTIME AFTER SUBDIVIDED BY THE INPUT VALUE DIV
C
      DELTAZ = TLEN / NINT
      DELTAZ = DELTAZ/DIV
      NINT = NINT*IDIV
      NDIV = NDIV*IDIV
C
C        INITIAL TEMPERATURE DISTRIBUTION
C
  130 DO 135 J=1,NINT
      TGROND(J)=62.5
      TAV(J)=TGROND(J)
      TWALL(1,J) = TGROND(J)
      TWALL(2,J) = TGROND(J)
      OLDTW(1,J) = TWALL(1,J)
      OLDTW(2,J) = TWALL(2,J)
      DO 135 I=1,NP1
  135 CC(J,I)=TGROND(J)
C
C     DETERMINE AREA AND VOLUME OF TUBE INCREMENT FOR STUDY
C
      AREA=2.0*PHI*RO*DELTAZ
      VOLUME=PHI*RO*RO*DELTAZ
C
C     SEPERATE AREAS & VOLUMES INTO GROUND AND TUBE FACING
C
      AREAG = AREA * (1.0-PORTON)
      VOL = VOLUME * (1.0-PORTON)
      AREAT = AREA * PORTON
      VOLT = VOL * PORTON
C
C     INITIALIZE TUBE INCREMENT TEMPERATURES
C
      NINT2=NINT+1
      DO 140 I=2,NINT2
```

```
      TF(I)=TGROND(I)
  140 CONTINUE
C
C     SET UP TIME INTERVALS
C
  150 OFTIME = (CYTIME-ONTIME)/DIV
      ONTIME   = ONTIME/NDIV
      INC = 1
C
  200 TF(1)=TFIN
C
C         SET UP CALCULATIONS ACCORDING TO HEAT PUMP CYCLING
C         KOUNT1 = 1   - HEAT PUMP IS TO BE TURNED OFF
C         KOUNT1 = 0   - HEAT PUMP IS TO BE TURNED ON
C         USE Q FROM LAST TIME STEP AS AN APPROXIMATION TO
C         THAT FOR CURRENT TIME STEP
C
      IF(KOUNT1.EQ.1) GO TO 210
      DTIME=ONTIME
      MSET=1
      Q=QON
      GO TO 215
  210 MSET=0
      DTIME=OFTIME
      Q=QOFF
C
C     USE PORTION AS A FIRST APPROXIMATION OF HEAT RATES
C     TO GROUND AND ADJACENT TUBE SECTION
C
  215 Q1 = Q * (1.0-PORTON)
      Q2 = Q * PORTON
C
C     BEGIN SUCCESSIVE ENERGY BALANCE FOR A TUBE SECTION
C
      DO 350 KK=1,NINT
      KCHECK=0
C
C     DETERMINE TEMPERATURE DISTRIBUTION THROUGH GROUND
C     BASED ON THIS GUESS OF HEAT TRANSFER TO THE GROUND
C
  220 DO 225 I=1,NP1
  225 C(KK,I)=CC(KK,I)
      A(1)=-(ALPHA*DTIME*FA(2)/(CV(1)*DR(2)))
      B(1)=0.0
      C(KK,1)=C(KK,1)+(Q1*ALPHA*DTIME)/(DELTAZ*COND*CV(1))
      D(1)=1.0-A(1)
      DO 230 I=2,N
      B(I)=-(ALPHA*DTIME*FA(I)/(CV(I)*DR(I)))
      A(I)=-(ALPHA*DTIME*FA(I+1)/(CV(I)*DR(I+1)))
      D(I)=1.0-B(I)-A(I)
  230 CONTINUE
      C(KK,N)=C(KK,N)-A(N)*C(KK,N+1)
      A(N)=0.0
C
C         TRIDIAGONAL SYSTEM GAUSS ELIMINATION
C
C         COMPUTE THE NEW MATRIX.  SOLUTION WILL BE STORED IN C ARRAY
C
      DO 240 I=2,N
      RR=B(I)/D(I-1)
      D(I)=D(I)-RR*A(I-1)
      C(KK,I)=C(KK,I)-RR*C(KK,I-1)
  240 CONTINUE
C
C         BACK SUBSTITUTION
C
      C(KK,N)=C(KK,N)/D(N)
      DO 245 I=2,N
      J=N-I+1
      C(KK,J)=(C(KK,J)-A(J)*C(KK,J+1))/D(J)
C
C     GROUND TEMPERATURE AT INFINITY IS CONSTANT
```

```
C
      IF(C(KK,N).LT.TGROND(KK)) C(KK,N)=TGROND(KK)
  245 CONTINUE
C
C
C     ENERGY BALANCE FOR HEAT PUMP ON (MSET=1)
C
      IF (MSET.EQ.0) GO TO 250
      TN(KK+1) = TF(KK) -  Q /(FLRATE*CPF)
      IF (KPRINT.EQ.1) WRITE(6,1075) TN(KK+1)
      TAVG(KK)=(TF(KK)+TN(KK+1))/2.0
      TWALL(1,KK) =  TAVG(KK) - Q1/(H*AREAG)
      IF (PORTON.LT.0.0001) TWALL(2,KK) = TWALL(1,KK)
      IF (PORTON.GE.0.0001) TWALL(2,KK) =  TAVG(KK) - Q2/(H*AREAT)
      IF (KPRINT.EQ.1) WRITE(6,1080) TWALL(1,KK),TWALL(2,KK)
      GO TO 260
C
C    ENERGY BALANCE FOR HEAT PUMP OFF
C
 250  TAVG(KK) = TAV(KK) - (Q*DTIME)/(RHO*VOLUME*CPF)
      IF(KPRINT.EQ.1) WRITE(6,1090) TAV(KK),TAVG(KK),TF(KK)
C
C    NOTE THAT DURING THE OFF TIME, TIN = TAVE = TOUT
C    ASSUME THAT TWALL = TAVG
C
      TWALL(1,KK) = TAVG(KK)
      TWALL(2,KK) = TAVG(KK)
      DIFF = (TAV(KK) - TAVG(KK))/2.
      TN(KK+1) = TN(KK+1) - DIFF
      IF(KPRINT.EQ.1) WRITE(6,1092) TN(KK+1)
 260  P1 = (TWALL(1,KK) + OLDTW(1,KK))/2.0
      P2 = (C(KK,1)+CC(KK,1))/2.0
      QCOND1= COND*FA(2)*DELTAZ*((P1 - P2)/DR(2))
C
C     DETERMINE THE NUMBER OF FLUID INCREMENT THAT IS DIRECTLY
C     ACROSS FROM THE CURRENT INCREMENT AND USE ITS WALL TEMPERATURE
C     FOR THE TEMPERATURE GRADIENT BETWEEN ADJACENT TUBES
C
      LL = (NINT+1) - KK
      IF(KPRINT.EQ.1) WRITE(6,1095) KK,LL
      P1 = (TWALL(2,KK) + OLDTW(2,KK))/2.0
      P2 = (TWALL(2,LL) + OLDTW(2,LL))/2.0
      IF (PORTON.LT.0.0001) QCOND2 = 0.0
      IF (PORTON.GE.0.0001) QCOND2 = COND*FAT*DELTAZ*((P1 - P2)/DA)
      Q = (1.0-PORTON)*Q1 + PORTON*Q2
      TFOUT = TN(KK+1)
      IF(KPRINT.EQ.1) WRITE(6,1100) Q1,QCOND1,Q2,QCOND2
C
C     TEST FOR CONVERGENCE OF HEAT TRANSFER TO GROUND, Q1
C     RATES MUST BE WITHIN 5 PERCENT FOR CONVERGENCE
C
      KEEP = 0
      TEST = ABS(0.05*Q1)
      IF (TEST.LT.0.1) TEST = 0.1
      IF(ABS(Q1-QCOND1).GT.TEST) GO TO 270
      KEEP = 1
      GO TO 280
  270 IF(QCOND1.GT.Q1) Q1=Q1+(QCOND1-Q1)/2.0
      IF(Q1.GT.QCOND1) Q1=Q1-(Q1-QCOND1)/2.0
C
C     TEST FOR CONVERGENCE OF HEAT TRANSFER TO OTHER TUBE, Q2
C
  280 TEST = ABS(0.05*Q2)
      IF (TEST.LT.0.1) TEST = 0.1
      IF(ABS(Q2-QCOND2).LE.TEST.AND.KEEP.EQ.1) GO TO 290
      IF(QCOND2.GT.Q2) Q2=Q2+(QCOND2-Q2)/2.0
      IF(Q2.GT.QCOND2) Q2=Q2-(Q2-QCOND2)/2.0
      KCHECK=KCHECK+1
      IF(KCHECK.GE.20.0) WRITE(6,1102)
      IF(KCHECK.GE.20.0) STOP
      Q = (1.0-PORTON)*Q1 + PORTON*Q2
      IF (MSET.EQ.0) TN(KK+1) = TN(KK+1) + DIFF
```

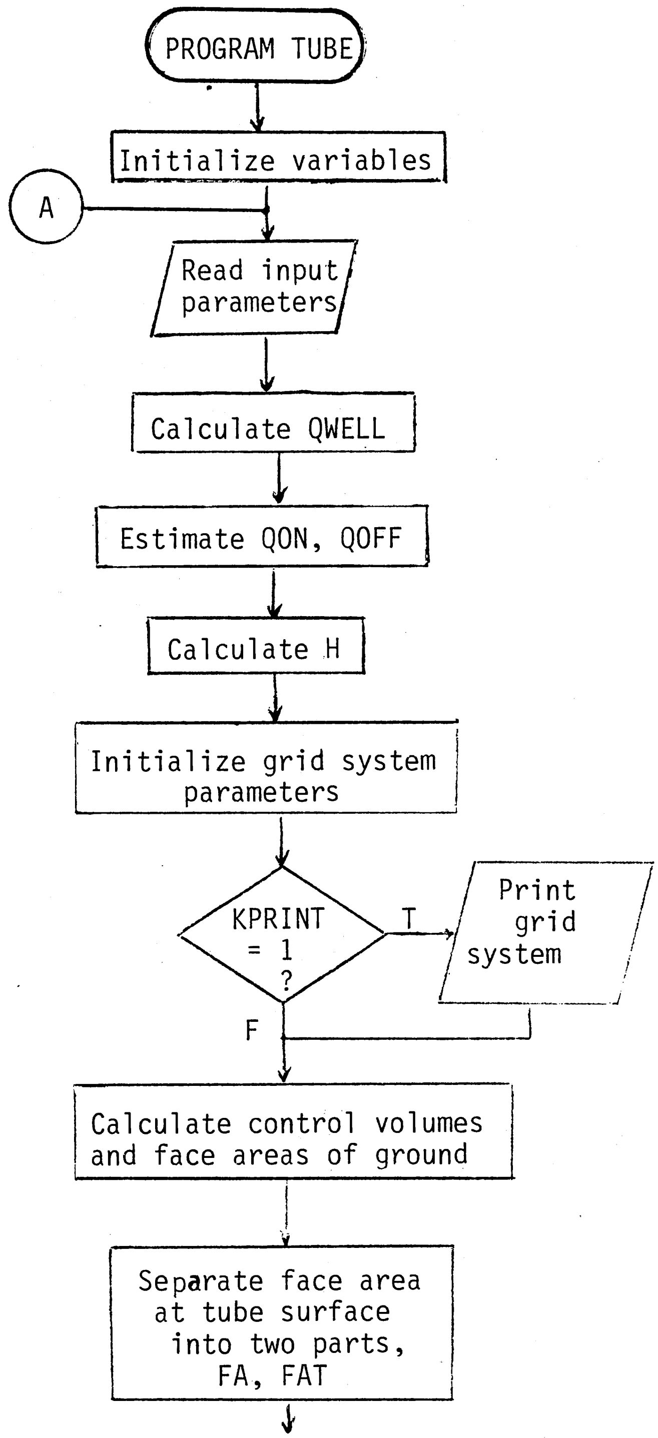
```
      GO TO 220
C
C     ENERGY BALANCE SATISFIED
C
  290 QDT=Q*DTIME
      QTOTAL=QTOTAL+QDT
C
C   SAVE THESE RATES FOR APPROXIMATION FOR NEXT TIME STEP
C
      IF(MSET.EQ.1.AND.KK.EQ.1) QON=Q
      IF(MSET.EQ.O.AND.KK.EQ.1) QOFF=Q
C   VALUES FOR THIS TUBE INCREMENT BECOME PAST VALUES
C
      TAV(KK)=TAVG(KK)
      DO 300 I=1,NP1
  300 CC(KK,I)=C(KK,I)
      HZ=KK*DELTAZ
C     ADJUST Q FOR BETTER GUESS OF NEXT TUBE INCREMENT
C
      Q1= Q1/10.0
      Q2= Q2/10.0
      Q = (1.0-PORTON)*Q1 + PORTON*Q2
      IF (TF(KK+1).LE.CC(KK+1,1)) Q1 = 0.0
      IF (TF(KK+1).LE.TWALL(2,LL-1)) Q2 = 0.0
C        PRINT THE RESULTS
C
      IF(KPRINT.EQ.0) GO TO 350
      WRITE(6,1104) HZ
      WRITE(6,1105) TFOUT,QDT
      WRITE(6,1106)
      WRITE(6,1107) (C(KK,I),I=1,32)
 350  CONTINUE
C
C     MAKE WALL TEMPERATURES PAST TEMPERATURES FOR ALL
C     TUBE INCREMENTS
C
      DO 360 KK = 1,NINT
      OLDTW(1,KK) = TWALL(1,KK)
 360  OLDTW(2,KK) = TWALL(2,KK)
C
C        PRINT THE RESULTS IF AT THE END OF EITHER THE
C        ON TIME OR CYCLE TIME
C
 370    IF ((MSET.EQ.1.AND.INC.LT.NDIV)
     1        .OR. (MSET.EQ.0.AND.INC.LT.IDIV)) GO TO 380
        IF(IPLOT.EQ.1) WRITE(3,1118) TIME,TFOUT
        IF (MSET.EQ.0.AND.RATIO.LT.1.0) WRITE(6,1120) TFOUT
        IF (MSET.EQ.1.AND.RATIO.LT.1.0) WRITE(6,1125) TFINCY,TFOUT
        IF(RATIO.GE.1.0) WRITE(6,1130) TFINCY,TFOUT,TFOUT
        IF(KPRINT.EQ.1) WRITE (6,1132) ITER , TFOUT , DIF, Q
        IF(KPRINT.EQ.1) WRITE(6,1134) QTOTAL
        QTOTAL = 0.0
C
C        RETURN TO FORWARD ANOTHER TIME STEP
C
380   KOUNT=KOUNT+1
C
C     DETERMINE WHETHER CYCLE SHOULD BE ON OR OFF
C
390   IF(RATIO.GE.1.0  .OR. (MSET.EQ.0 .AND. INC.EQ.IDIV)
     1                 .OR. (MSET.EQ.1 .AND. INC.LT.NDIV)) KOUNT1 = 0
      IF(RATIO.LE.0.0  .OR. (MSET.EQ.1 .AND. INC.EQ.NDIV)
     1                 .OR. (MSET.EQ.0 .AND. INC.LT.IDIV)) KOUNT1 = 1
C
C       COMPUTE A NEW TFIN FROM RELATION Q=M*CP*DELTA TEMP
C       MAKE EXIT TEMP FROM NODE I THE INLET TEMP TO NODE I+1
C
      IF(KOUNT1 .EQ. 1) GO TO 420
C     SAVE FIRST VALUE OF TFIN FOR CYCLE OUTPUT
      IF(INC .EQ. 1) TFINCY = TFIN
      TFIN = TFOUT + QWELL/(FLRATE*CPF)
      DO 400 KK = 1,NINT2
```

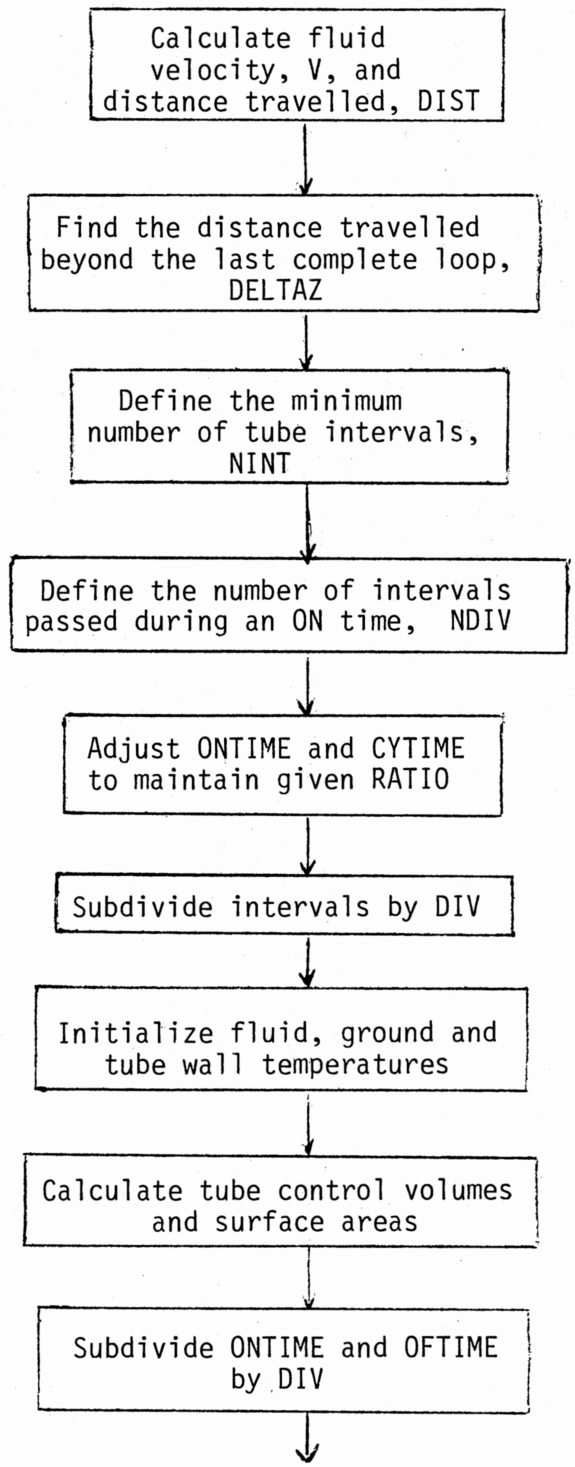
```
      TF(KK) = TN(KK)
 400  CONTINUE
C
C    DETERMINE THE HEAT TRANSFER COEFFICIENT
C
      TEMP = (TFIN + TFOUT) / 2.0
      WMU=8.3574-.18457*TEMP+.2332E-02*TEMP**2.-.17931E-04*TEMP**3.0
     & + .81845E-07*TEMP**4.-.20274E-09*TEMP**5.+.20919E-12*TEMP**6.
      WPR=27.51876-.65809*TEMP+.85657E-02*TEMP**2.-.66433E-04*TEMP**3.
     &     +.30315E-06*TEMP**4.-.74791E-09*TEMP**5.+.7675E-12*TEMP**6.
      WK=WMU*CPF/WPR
      RED=2.0*GFLUX*RO/WMU
      IF(RED.LE.2000.0) H=4.364*WK/(2.0*RO)
      IF(RED.GT.2000.0.AND.QWELL.GT.0.0) H=0.023*(RED**0.8)*(WPR**0.3)
     &                                    *WK/(2.0*RO)
      IF(RED.GT.2000.0.AND.QWELL.LE.0.0) H=0.023*(RED**0.8)*(WPR**0.4)
     &                                    *WK/(2.0*RO)
C
      IF(KPRINT.EQ.1) WRITE(6,1035) H
C
C     INCREMENT TIME AND THE NUMBER OF INCREMENTS THROUGH
C     WHICH THE WATER HAS PASSED
C
 420  IF(KPRINT.EQ.1) WRITE(6,1140)  TFIN
      TIME = TIME + DTIME
      IF(IPLOT.EQ.1) WRITE(4,1150) TIME,TFOUT
 1150 FORMAT(F12.6,F10.2)
      INC = INC + 1
      IF ((MSET.EQ.1.AND.INC.LE.NDIV)
     1       .OR. (MSET.EQ.0.AND.INC.LE.IDIV)) GO TO 200
      INC = 1
      IF (MSET.EQ.1.AND.RATIO.LT.1.0) GO TO 200
      IF ((HOURS - TIME) .GE. CYTIME) GO TO 200
 500  CONTINUE
      STOP
 1000 FORMAT(I2,1X,F5.1)
 1005 FORMAT(5X,'MODE MUST EQUAL 1 OR 2. CASE SKIPPED.')
 1010 FORMAT('1',5X,'INPUT VALUES ARE :', //,
     &           10X,'CYC TIME = ',F9.6,' HR    ',/,
     &           10X,'ON TIME  = ',F9.6,' HR    ',/,
     &           10X,'RATIO    = ',F9.2,/,
     &           10X,'HOURS    = ',F9.2,' HR    ',/,
     &           10X,'DIVISIONS= ',F9.2,/,
     &           10X,'TFIN     = ',F9.2,' DEG F',/,
     &           10X,'CAPACITY = ',F9.2,' BTU/HR',/,
     &           10X,'COP      = ',F9.3,
     &           10X,'FLOW RATE= ',F9.1,' GPM    ')
 1015 FORMAT(5X,'SIMULATION IS FOR THE COOLING MODE.')
 1020 FORMAT(5X,'WARNING: AN INLET TEMPERATURE LESS THAN 66 DEG F',
     1       /14X,'IS NOT FEASIBLE FOR COOLING.')
 1025 FORMAT(5X,'SIMULATION IS FOR THE HEATING MODE.')
 1030 FORMAT(5X,'WARNING: AN INLET TEMPERATURE GREATER THAN 54 DEG F',
     1       /14X,'IS NOT FEASIBLE FOR HEATING.')
 1035 FORMAT(5X,'HEAT TRANSFER COEFFICIENT IS ',F10.3)
 1037 FORMAT('1',//////,54X,'VALUES OF NON-UNIFORM GRID',//)
 1038 FORMAT(4(5X,8('DR(',I2,')=',F7.4,2X),///))
 1040 FORMAT(////,54X,'THE RADIUS VALUES',//)
 1042 FORMAT(5(5X,8('R(',I2,')=',F7.4,2X),///))
 1045 FORMAT('1')
 1048 FORMAT(5X,'USE ',I4,' TUBE INTERVALS.')
 1049 FORMAT(5X,'** WATER TRAVELS THROUGH ',I3,' TUBE INCREMENTS ',
     1       /,8X,'DURING AN ON-TIME PERIOD.')
 1050 FORMAT(5X,'ADJUSTED ON TIME = ',F10.5,
     1    /,5X,'ADJUSTED CYCLE TIME* = ',F10.5,
     2    //,5X,'* ADJUSTED TO MAINTAIN A RATIO OF ',F5.3,///)
 1052 FORMAT(///,5X,'TUBE WATER TEMPERATURES AT END OF: ',
     1        //,20X,'ON TIME',T47,'CYCLE TIME',/,
     2          14X,'TIN        TOUT          ',T50,'TOUT',//)
 1075 FORMAT(5X,'** TN(KK+1) = ',F10.3)
 1080 FORMAT(5X,'** TWALL GR AND TU = ',F10.3,5X,F10.3)
 1090 FORMAT(2X,'==> PAST TAVERAGE = ',F8.2,/,
     1       2X,'     PRESENT TAVE  = ',F8.2,/,
```

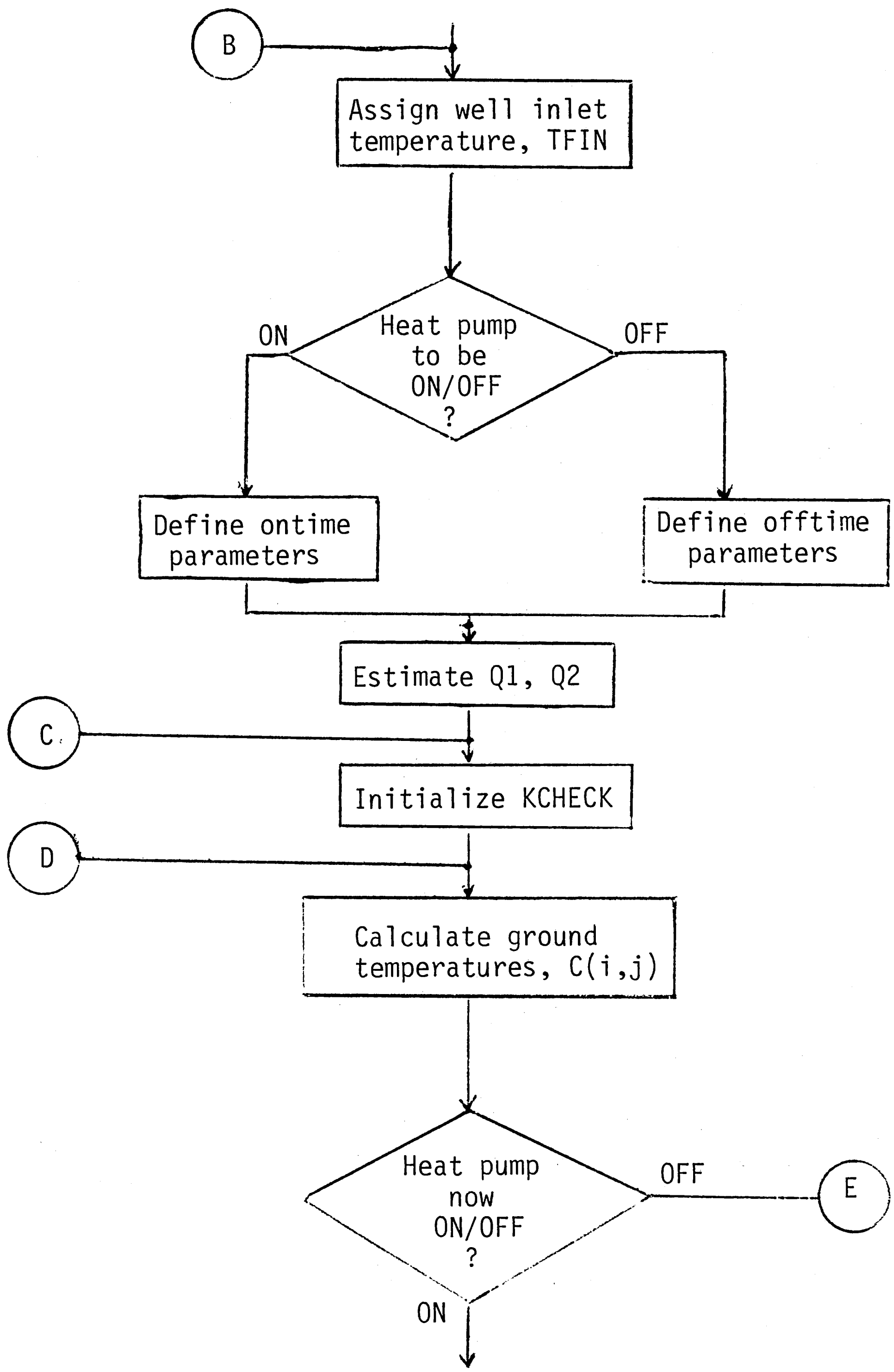
```
     2        2X,'    TF(KK),TIN     = ',F8.2)
 1092 FORMAT(5X,' TN(KK+1) =',F7.2)
 1095 FORMAT(5X,'CURRENT TUBE INCREMENT ',I3,' ACROSS FROM ',I3)
 1100 FORMAT(5X,'** Q1 = ',E12.5,'  QCOND1 = ',E12.5,
     1     /,5X,'** Q2 = ',E12.5,'  QCOND2 = ',E12.5)
 1102 FORMAT(//,10('*'),'TUBE DOES NOT CONVERGE AFTER 20 ITERATIONS')
 1104 FORMAT(' ',////,10X,'TEMPERATURE DISTRIBUTION',F6.1,'  FEET FROM'
     &        ,' THE BOTTOM OF THE WELL',3X,///)
 1105 FORMAT(5X,'THE WATER TEMPERATURE=',F8.2,10X,'THE HEAT CONDUCTED',
     &        ' TO OR FROM EARTH=',F10.0,'BTU.',/)
 1106 FORMAT(5X,'THE SOIL TEMPERATURES ARE,',/)
 1107 FORMAT(4(15X,8(F8.2,1X),/))
 1118 FORMAT(F12.6,F10.2)
 1120 FORMAT(1H+,45X,F10.3)
 1125 FORMAT(9X,2F10.3)
 1130 FORMAT(29X,F10.3,T45,F10.3)
 1132   FORMAT(5X,'CONVERGENCE AT :',
     &          5X,'ITER = ', I5 ,'  TFOUT = ',F10.5,' DIF=',F10.5,
     &        ' Q = ',E12.5)
 1134 FORMAT(55X,'THE TOTAL HEAT TO, OR FROM THE SOIL IS',F10.0,'BTU'/)
 1140 FORMAT(5X,'NEW TFIN = ',F6.2)
      END
//LKED.SYSLMOD DD DSN=U13075A.TUBE.LOAD,DISP=SHR,UNIT=3350,
//  VOL=SER=DASD25
//LKED.SYSIN DD *
 NAME TWOTUBE(R)
//
```

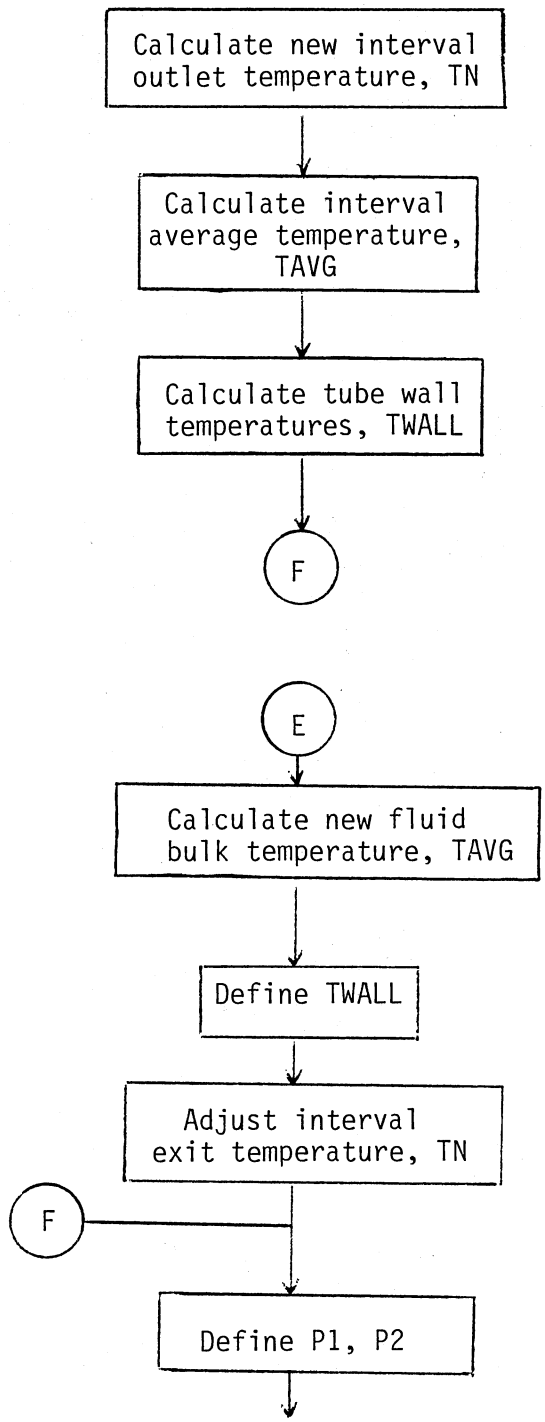
APPENDIX B

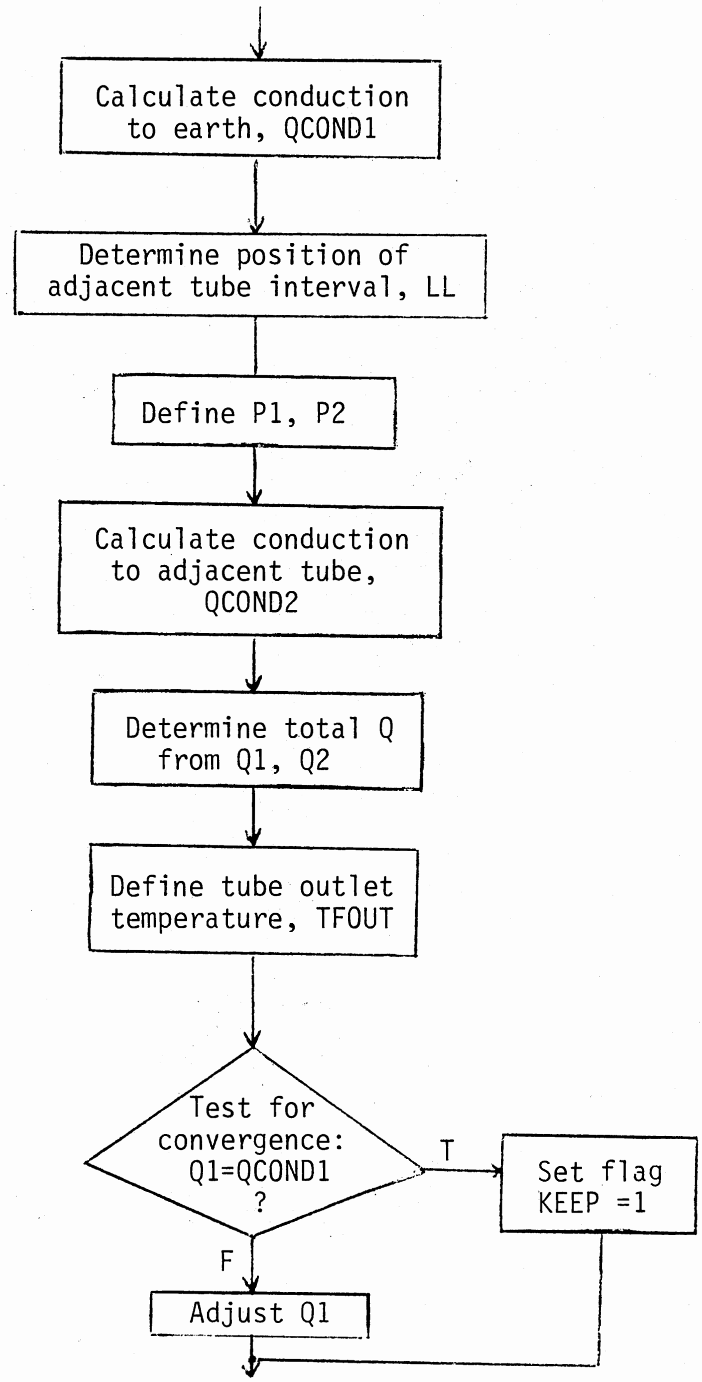
FLOW CHART OF PROGRAM TUBE


PROGRAM TUBE
Initialize variables
A
Read input parameters
Calculate QWELL
Estimate QON, QOFF
Calculate H
Initialize grid system parameters
KPRINT = 1 ?
T
Print grid system
F
Calculate control volumes and face areas of ground
Separate face area at tube surface into two parts, FA, FAT

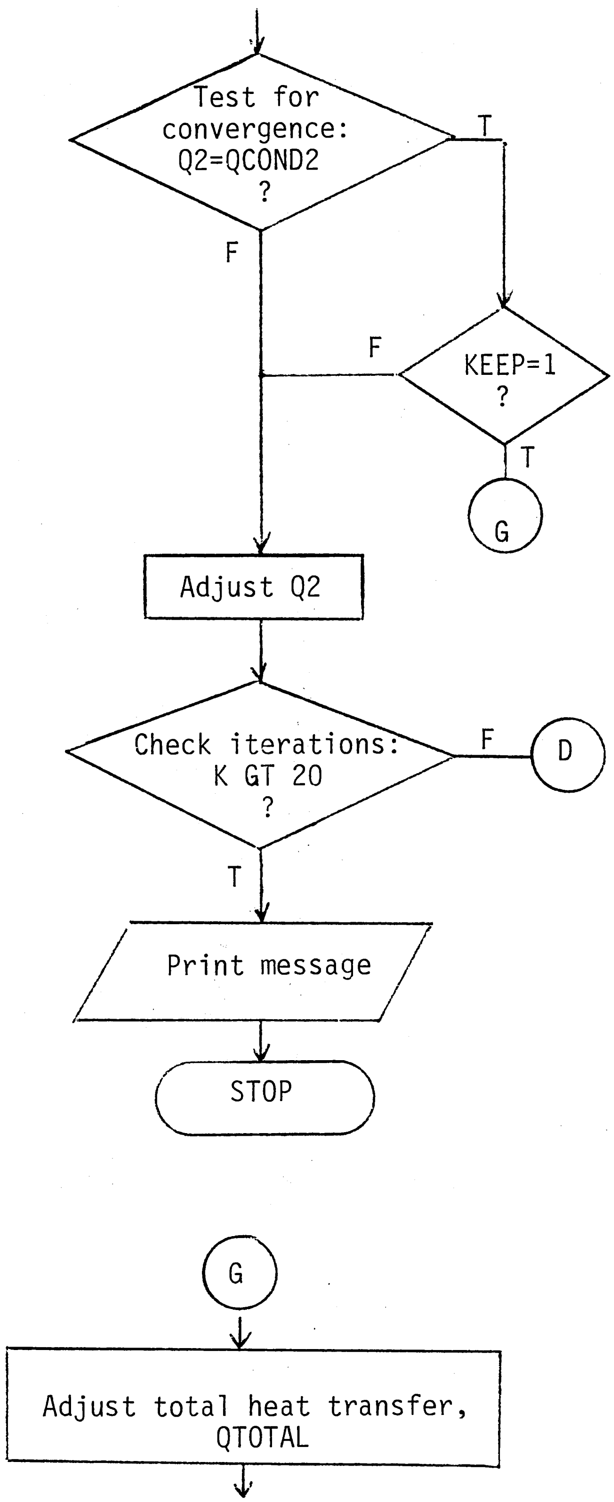

Calculate fluid velocity, V, and distance travelled, DIST
Find the distance travelled beyond the last complete loop, DELTAZ
Define the minimum number of tube intervals, NINT
Define the number of intervals passed during an ON time, NDIV
Adjust ONTIME and CYTIME to maintain given RATIO
Subdivide intervals by DIV
Initialize fluid, ground and tube wall temperatures
Calculate tube control volumes and surface areas
Subdivide ONTIME and OFTIME by DIV

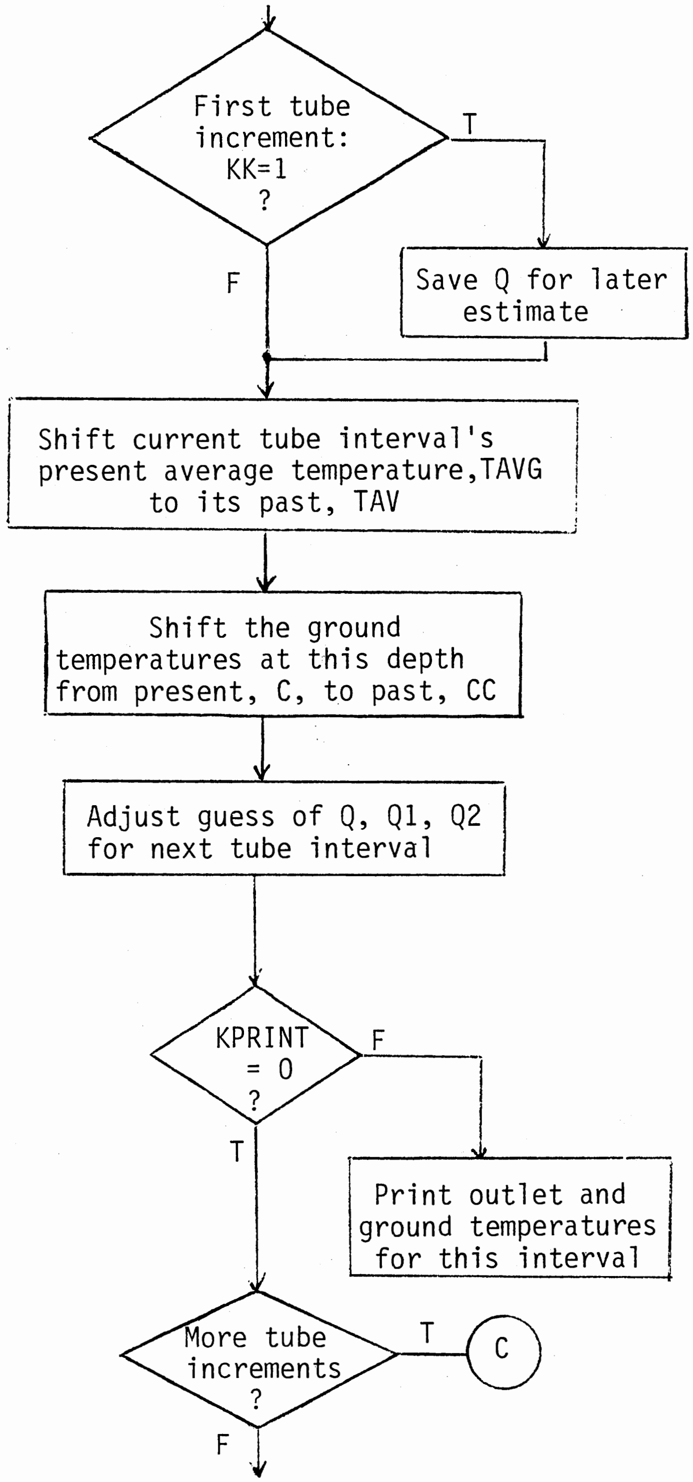

B
Assign well inlet temperature, TFIN
Heat pump to be ON/OFF ?
ON
OFF
Define ontime parameters
Define offtime parameters
Estimate Q1, Q2
C
Initialize KCHECK
D
Calculate ground temperatures, C(i,j)
Heat pump now ON/OFF ?
OFF
E
ON

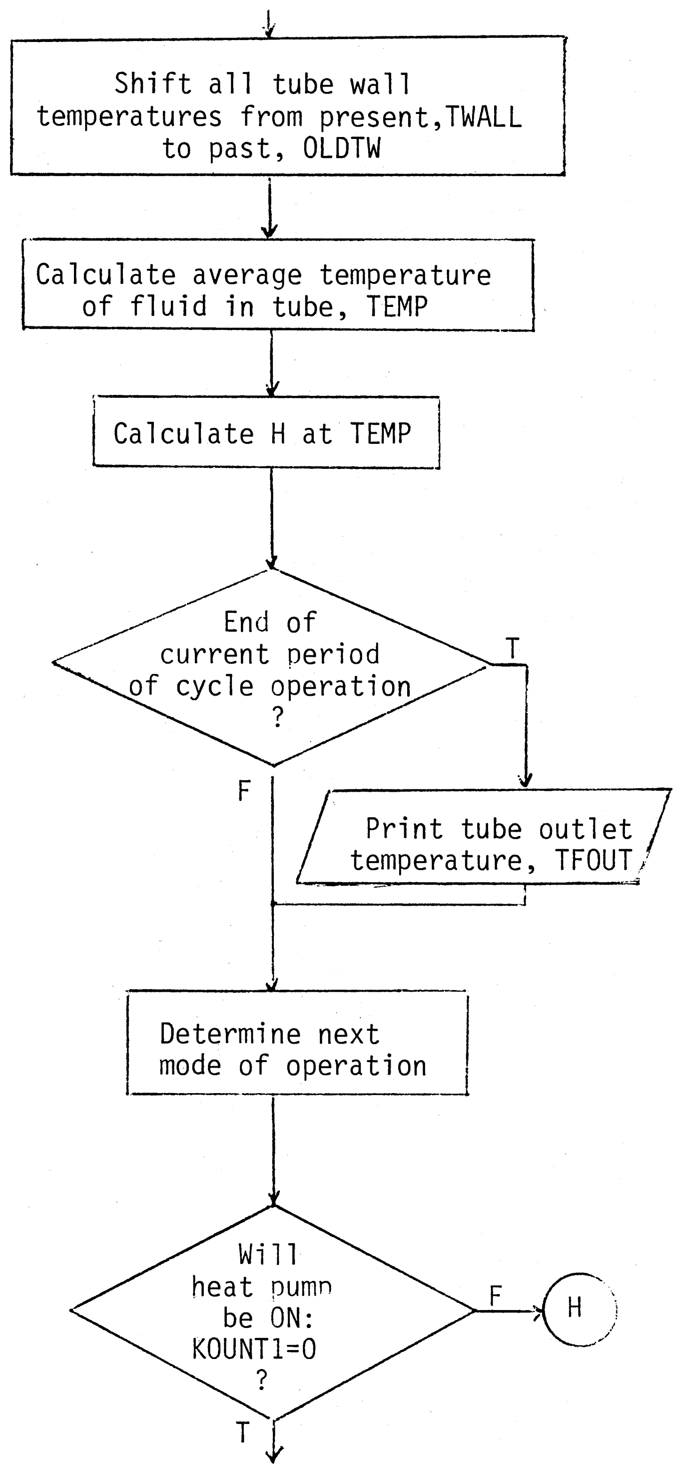

Calculate new interval outlet temperature, TN
Calculate interval average temperature, TAVG
Calculate tube wall temperatures, TWALL
F
E
Calculate new fluid bulk temperature, TAVG
Define TWALL
Adjust interval exit temperature, TN
F
Define P1, P2

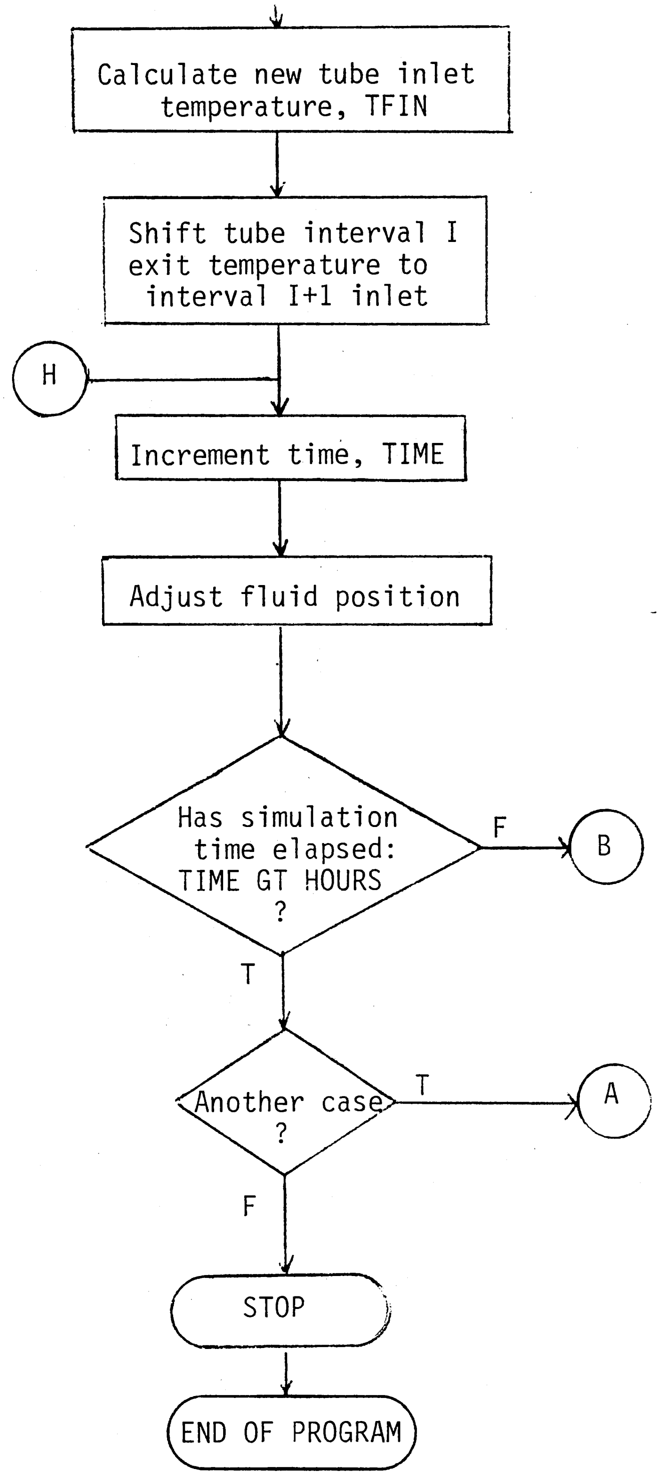


Calculate conduction to earth, QCOND1
Determine position of adjacent tube interval, LL
Define P1, P2
Calculate conduction to adjacent tube, QCOND2
Determine total Q from Q1, Q2
Define tube outlet temperature, TFOUT
Test for convergence: Q1=QCOND1 ?
T
Set flag KEEP =1
F
Adjust Q1


Test for convergence: Q2=QCOND2 ?
T
F
F
KEEP=1 ?
T
G
Adjust Q2
Check iterations: K GT 20 ?
F
D
T
Print message
STOP
G
Adjust total heat transfer, QTOTAL


First tube
increment:
KK=1
?
T
F
Save Q for later
estimate
Shift current tube interval's
present average temperature,TAVG
to its past, TAV
Shift the ground
temperatures at this depth
from present, C, to past, CC
Adjust guess of Q, Q1, Q2
for next tube interval
KPRINT
= 0
?
F
T
Print outlet and
ground temperatures
for this interval
More tube
increments
?
T
C
F


Shift all tube wall temperatures from present,TWALL to past, OLDTW
Calculate average temperature of fluid in tube, TEMP
Calculate H at TEMP
End of current period of cycle operation ?
T
F
Print tube outlet temperature, TFOUT
Determine next mode of operation
Will heat pump be ON: KOUNT1=0 ?
F
H
T


Calculate new tube inlet temperature, TFIN
Shift tube interval I exit temperature to interval I+1 inlet
H
Increment time, TIME
Adjust fluid position
Has simulation time elapsed: TIME GT HOURS ?
F
B
T
Another case ?
T
A
F
STOP
END OF PROGRAM

APPENDIX C

SIMULATION PRINTOUTS

```
 1 1/2 INCH POLYETHYLENE TUBE

INPUT VALUES ARE :

     CYC TIME  =  0.111100 HR
     ON TIME   =  0.083300 HR
     RATIO     =      0.75
     HOURS     =     12.00 HR
     DIST APT  =      0.05 FT
     DEPTH     =    250.00 FT
     PORTION   =      0.25
     DIVISIONS=       1.00
     TFIN      =     68.00 DEG F
     CAPACITY  =  19500.00 BTU/HR
     COP       =     3.000               FLOW RATE=        4.0 GPM
SIMULATION IS FOR THE COOLING MODE.
USE    2 TUBE INTERVALS.
** WATER TRAVELS THROUGH    1 TUBE INCREMENTS
   DURING AN ON-TIME PERIOD.
ADJUSTED ON TIME =    0.08367
ADJUSTED CYCLE TIME* =    0.11155

* ADJUSTED TO MAINTAIN A RATIO OF 0.750
```

TUBE WATER TEMPERATURES AT END OF:

ON TIME		CYCLE TIME
TIN	TOUT	TOUT
0.000	62.582	62.603
68.000	66.552	66.429
75.674	71.226	70.936
79.499	73.429	73.112
84.007	76.857	76.448
86.183	78.387	77.980
89.518	81.110	80.636
91.050	82.262	81.795
93.706	84.484	83.965
94.866	85.386	84.878
97.035	87.228	86.676
97.948	87.962	87.421
99.747	89.504	88.929
100.492	90.124	89.559
102.000	91.428	90.834
102.630	91.969	91.384
103.905	93.077	92.469
104.455	93.564	92.963

105.540	94.510	93.891
106.034	94.960	94.347
106.962	95.777	95.149
107.417	96.194	95.570
108.219	96.902	96.266
108.641	97.295	96.663
109.337	97.911	97.269
109.733	98.293	97.652
110.340	98.822	98.175
110.723	99.196	98.549
111.246	99.651	99.000
111.620	100.020	99.368
112.071	100.411	99.756
112.438	100.777	100.120
112.827	101.128	100.469
113.190	101.461	100.801
113.539	101.785	101.123
113.872	102.095	101.432
114.194	102.397	101.732
114.502	102.686	102.020
114.802	102.968	102.300
115.090	103.239	102.570
115.371	103.503	102.833
115.640	103.758	103.087
115.904	104.007	103.335
116.157	104.247	103.574
116.405	104.482	103.808
116.644	104.709	104.034
116.879	104.932	104.256
117.105	105.147	104.471
117.327	105.358	104.681
117.541	105.563	104.885
117.752	105.764	105.086
117.956	105.959	105.280
118.156	106.151	105.471
118.351	106.337	105.657
118.542	106.521	105.839
118.727	106.699	106.017
118.910	106.874	106.192
119.087	107.045	106.362
119.262	107.213	106.529
119.432	107.377	106.693
119.600	107.538	106.854
119.763	107.696	107.011
119.924	107.851	107.166
120.081	108.003	

		107.317
120.236	108.152	107.466
120.387	108.298	107.612
120.537	108.443	107.756
120.682	108.584	107.896
120.826	108.723	108.035
120.967	108.859	108.171
121.106	108.994	108.305
121.242	109.126	108.437
121.376	109.256	108.567
121.507	109.384	108.694
121.637	109.510	108.820
121.765	109.634	108.943
121.891	109.756	109.065
122.014	109.876	109.185
122.136	109.995	109.304
122.256	110.111	109.420
122.374	110.226	109.535
122.490	110.340	109.648
122.605	110.452	109.760
122.719	110.562	109.870
122.830	110.671	109.978
122.940	110.778	110.085
123.049	110.884	110.191
123.156	110.989	110.296
123.262	111.092	110.399
123.366	111.194	110.500
123.469	111.295	110.601
123.571	111.394	110.700
123.671	111.492	110.798
123.770	111.589	110.894
123.868	111.685	110.990
123.965	111.779	111.085
124.061	111.873	111.178
124.155	111.966	111.270
124.249	112.057	111.361
124.341	112.147	111.452
124.432	112.237	111.541
124.522	112.325	111.629
124.611	112.412	111.716
124.700	112.499	111.803
124.787	112.584	111.888
124.873	112.669	111.972
124.959	112.753	112.056

1 1/2 INCH POLYETHYLENE TUBE

INPUT VALUES ARE :

```
     CYC TIME =  0.166700 HR
     ON TIME  =  0.083300 HR
     RATIO    =      0.50
     HOURS    =     12.00 HR
     DIST APT =      0.05 FT
     DEPTH    =    250.00 FT
     PORTION  =      0.25
     DIVISIONS=      1.00
     TFIN     =     68.00 DEG F
     CAPACITY =  19500.00 BTU/HR
     COP      =     3.000                FLOW RATE=        4.0 GPM
SIMULATION IS FOR THE COOLING MODE.
USE    2 TUBE INTERVALS.
** WATER TRAVELS THROUGH   1 TUBE INCREMENTS
   DURING AN ON-TIME PERIOD.
ADJUSTED ON TIME =    0.08367
ADJUSTED CYCLE TIME* =    0.16733

* ADJUSTED TO MAINTAIN A RATIO OF 0.500
```

TUBE WATER TEMPERATURES AT END OF:

ON TIME TIN	ON TIME TOUT	CYCLE TIME TOUT
125.043	62.582	62.636
68.000	66.240	65.949
75.707	70.965	70.229
79.020	73.069	72.281
83.300	76.270	75.255
85.352	77.822	76.795
88.325	80.155	78.981
89.865	81.386	80.205
92.051	83.145	81.865
93.275	84.158	82.871
94.935	85.521	84.166
95.942	86.377	85.015
97.236	87.460	86.049
98.085	88.197	86.779
99.120	89.078	87.626
99.850	89.719	88.260
100.696	90.452	88.967
101.331	91.017	89.525

102.038	91.637	90.126
102.595	92.139	90.621
103.197	92.673	91.141
103.692	93.122	91.584
104.212	93.589	92.039
104.655	93.993	92.439
105.110	94.406	92.845
105.509	94.773	93.207
105.915	95.144	93.570
106.277	95.479	93.901
106.641	95.814	94.230
106.972	96.121	94.533
107.300	96.426	94.833
107.604	96.709	95.113
107.903	96.989	95.388
108.184	97.252	95.648
108.459	97.509	95.902
108.718	97.754	96.144
108.972	97.993	96.380
109.214	98.222	96.606
109.450	98.445	96.826
109.676	98.659	97.038
109.897	98.868	97.245
110.109	99.070	97.444
110.315	99.266	97.638
110.515	99.457	97.827
110.709	99.642	98.010
110.897	99.822	98.188
111.081	99.997	98.362
111.259	100.168	98.531
111.433	100.335	98.696
111.602	100.497	98.857
111.767	100.656	99.014
111.927	100.810	99.167
112.084	100.961	99.317
112.238	101.109	99.463
112.387	101.253	99.606
112.534	101.395	99.747
112.677	101.533	99.884
112.817	101.668	100.018
112.954	101.801	100.150
113.089	101.931	100.279
113.220	102.059	100.405
113.349	102.184	100.529
113.476	102.306	100.651
113.600	102.427	

		100.770
113.721	102.545	100.887
113.841	102.661	101.003
113.958	102.775	101.116
114.073	102.887	101.227
114.186	102.997	101.336
114.298	103.105	101.444
114.407	103.212	101.550

1 1/2 INCH POLYETHYLENE TUBE

INPUT VALUES ARE :

```
     CYC TIME =  0.333300 HR
     ON TIME  =  0.083300 HR
     RATIO    =      0.25
     HOURS    =     12.00 HR
     DIST APT =      0.05 FT
     DEPTH    =    250.00 FT
     PORTION  =      0.25
     DIVISIONS=      1.00
     TFIN     =     68.00 DEG F
     CAPACITY =  19500.00 BTU/HR
     COP      =     3.000               FLOW RATE=        4.0 GPM
SIMULATION IS FOR THE COOLING MODE.
USE    2 TUBE INTERVALS.
** WATER TRAVELS THROUGH    1 TUBE INCREMENTS
   DURING AN ON-TIME PERIOD.
ADJUSTED ON TIME =     0.08367
ADJUSTED CYCLE TIME* =     0.33466

* ADJUSTED TO MAINTAIN A RATIO OF 0.250
```

TUBE WATER TEMPERATURES AT END OF:

	ON TIME	CYCLE TIME
TIN	TOUT	TOUT
114.514	62.582	62.694
68.000	65.611	65.106
75.765	70.112	68.648
78.177	72.092	70.499
81.718	74.495	72.535
83.570	76.006	73.929
85.605	77.518	75.264
86.999	78.657	76.312
88.334	79.713	77.268
89.383	80.583	78.074
90.339	81.370	78.799
91.144	82.053	79.434
91.869	82.669	80.009
92.505	83.219	80.524
93.079	83.718	80.994
93.595	84.172	81.422
94.064	84.588	81.816
94.492	84.971	82.179

94.886	85.325	82.516
95.249	85.654	82.830
95.586	85.961	83.123
95.900	86.248	83.398
96.193	86.518	83.657
96.468	86.772	83.901
96.727	87.013	84.132
96.972	87.240	84.351
97.203	87.457	84.560
97.422	87.662	84.759
97.630	87.859	84.948
97.829	88.046	85.130
98.019	88.226	85.304
98.200	88.398	85.471
98.374	88.563	85.631
98.541	88.722	85.785
98.702	88.875	85.931

VITA 2

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Master of Science

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