STUDY OF SOLAR-ASSISTED GROUND SOURCE HEAT PUMP SYSTEMS

By

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Figure		Page

CHAPTER I

INTRODUCTION

Recent rapid increases in the cost of energy has promoted intensive efforts to discover ways to reduce energy consumption. In addition, the necessity of attaining energy independence has been emphasized by the turbulence and political instability in the oil rich parts of the world that supply the United States with about half of its oil. Since the demand for energy is continuously increasing, the solution to the above problem is to develop means to use the energy more efficiently.

More than one-fourth of the energy consumed in the United States is used for residential and commercial space heating, cooling, and water heating (1). A heat pump is a device which can make a significant contribution to energy conservation. This is mainly because of two reasons. First, the heat pump has an ability to deliver more energy (in the form of heat) than it takes to operate it. Second, it can perform the dual functions of heating and cooling the house with the same equipment.

In the conventional air source heat pump, which is the most commonly used heat pump today, the air acts as both heat source and sink, depending upon whether the heat pump is operating in heating or cooling mode. When the heat pump is operating in the heating mode, its capacity decreases as ambient temperature decreases. But at the same time the heating demand of the house increases as ambient temperature decreases. At some outdoor temperature the heating capacity of the heat pump is equal to the heating load of the space being heated. This is called the balance point of the system. At ambient temperatures below the balance point the heat pump has inadequate capacity to heat the building and the electric resistance heat is usually used as a back-up to assist the heat pump. If this occurred in many buildings simultaneously it could put heavy peak loads on the utilities. This could increase the demand factor* considerably. Figure 1 shows a plot of the outdoor temperatures in a cold week in February 1981. Figure 2 shows the plot of the total energy consumption of a house in Oklahoma in the same week. This house has a conventional air source heat pump system. Figure 3 shows the system demand on Oklahoma Gas & Electric Company in the same week. The term system demand used in this study is calculated over a short duration of time between 30 to 60 minutes. It won't be possible to get an accurate estimate of system demand which is calculated over a longer duration of time because the maximum power consumption in different houses in the community doesn't occur at the same time. If the system demand is plotted as shown in Figure 3 then the highest value of system demand in the graph is termed as the peak demand. These graphs clearly indicate the relationship between the system demand and outdoor temperature as discussed above.

In the summer, when the heat pump is in the cooling mode, its efficiency decreases as the outdoor temperature increases. Figure 4 shows a plot of the outdoor temperatures during the hot week of August 1981. Figure 5 shows a plot of the total energy consumption of a house

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*Demand factor is the ratio of the maximum demand over a specified time period to the total connected load.

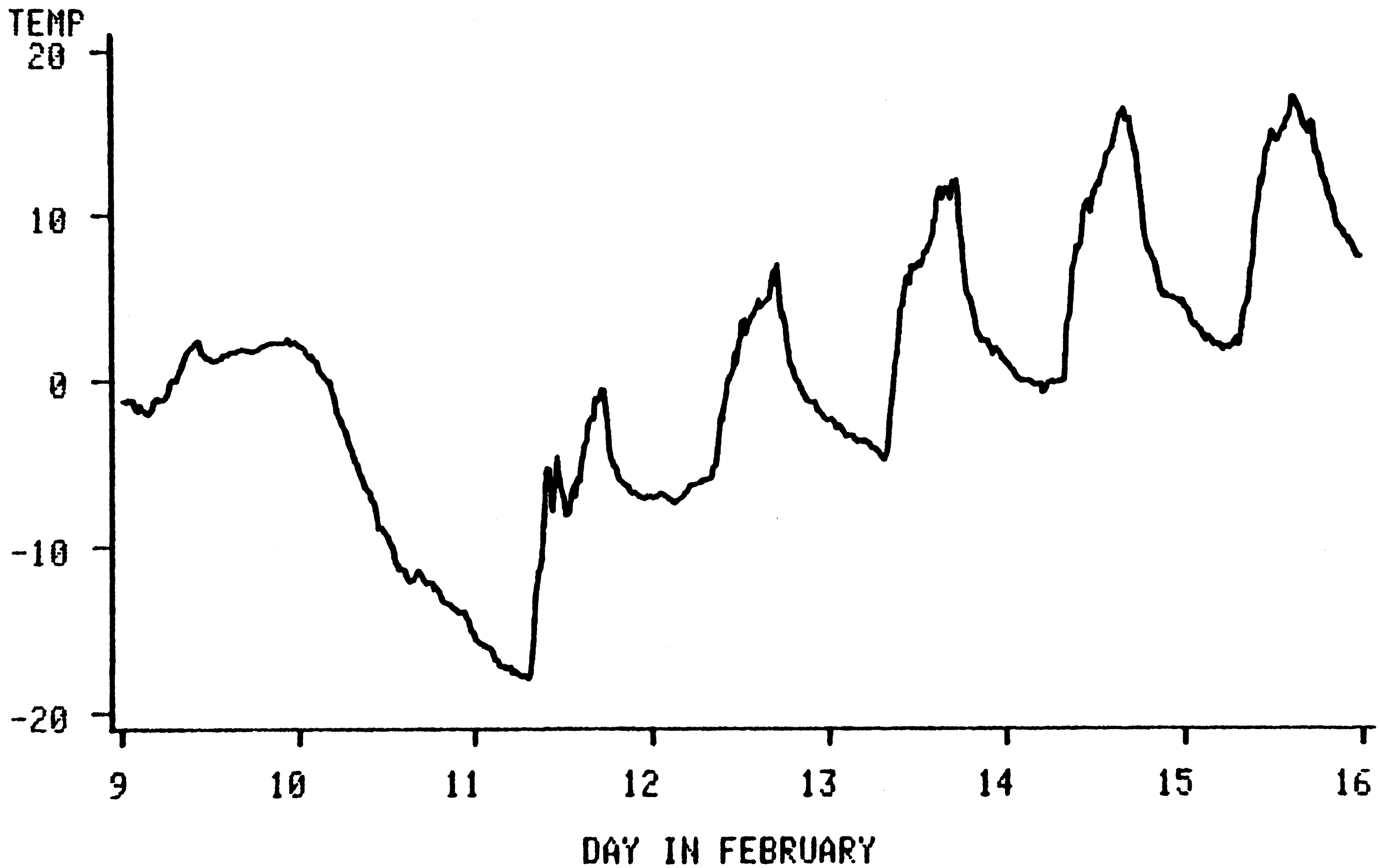

Figure 1. Plot of Outdoor Temperatures for a Week in February 1981

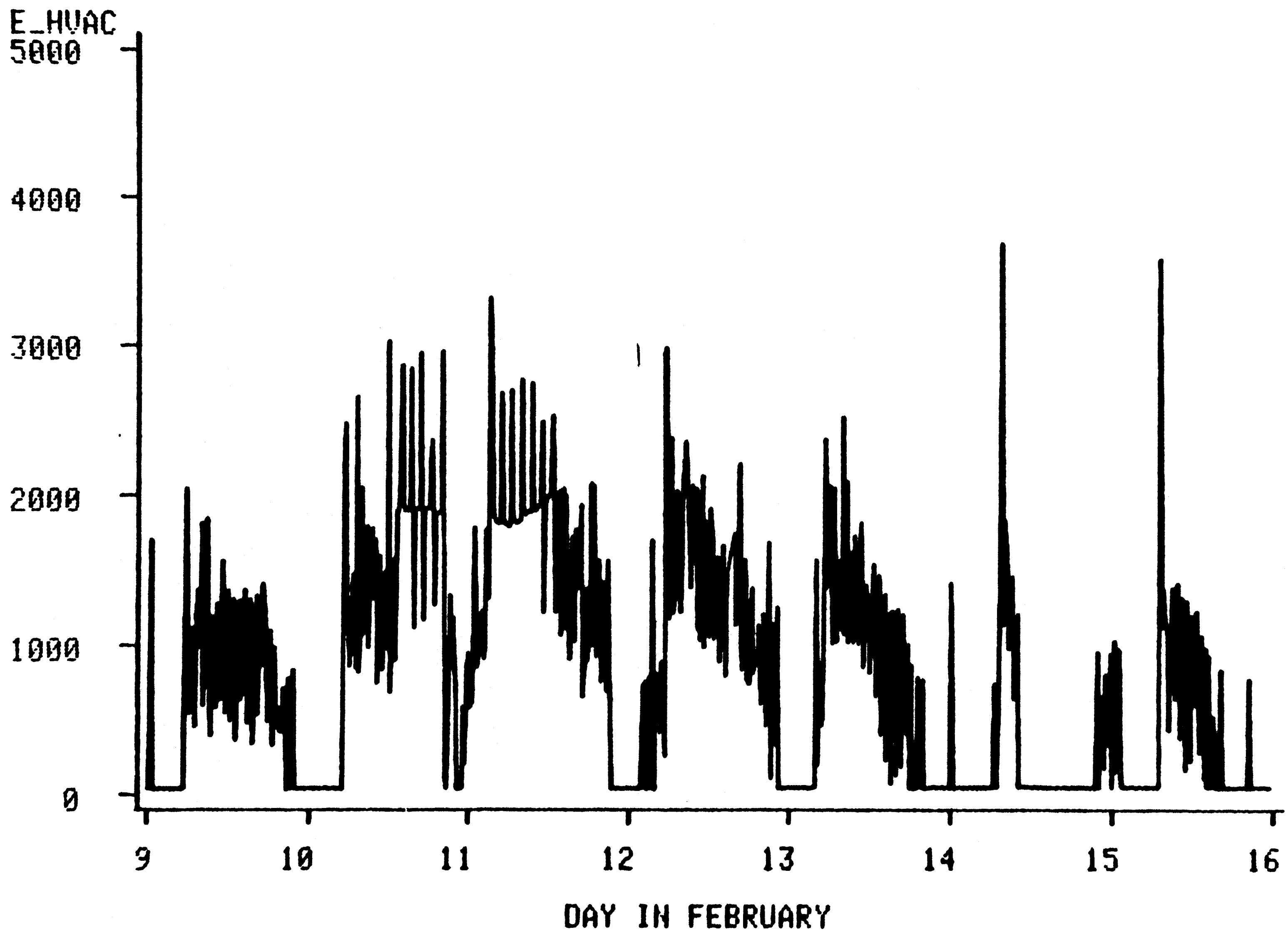

Figure 2. Energy Consumption of a House With a Conventional Air Source Heat Pump for a Week in February 1981

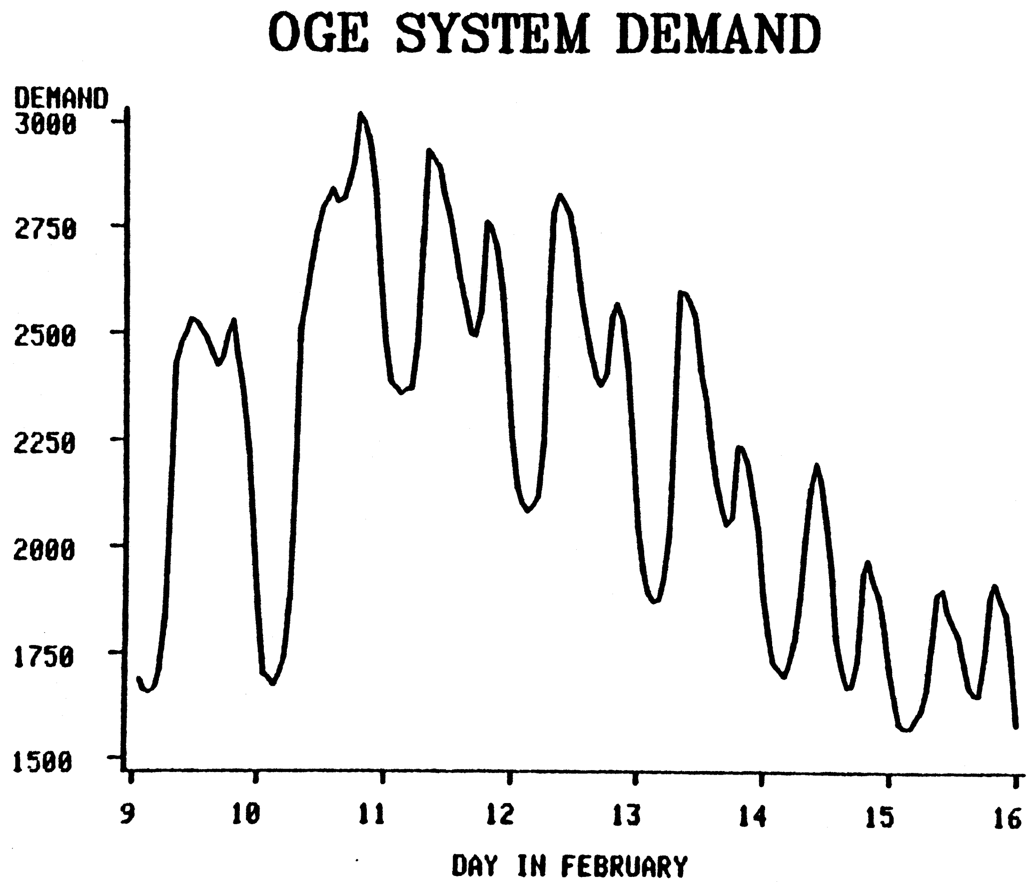

Figure 3. Utility Power Demand for a Week in February 1981

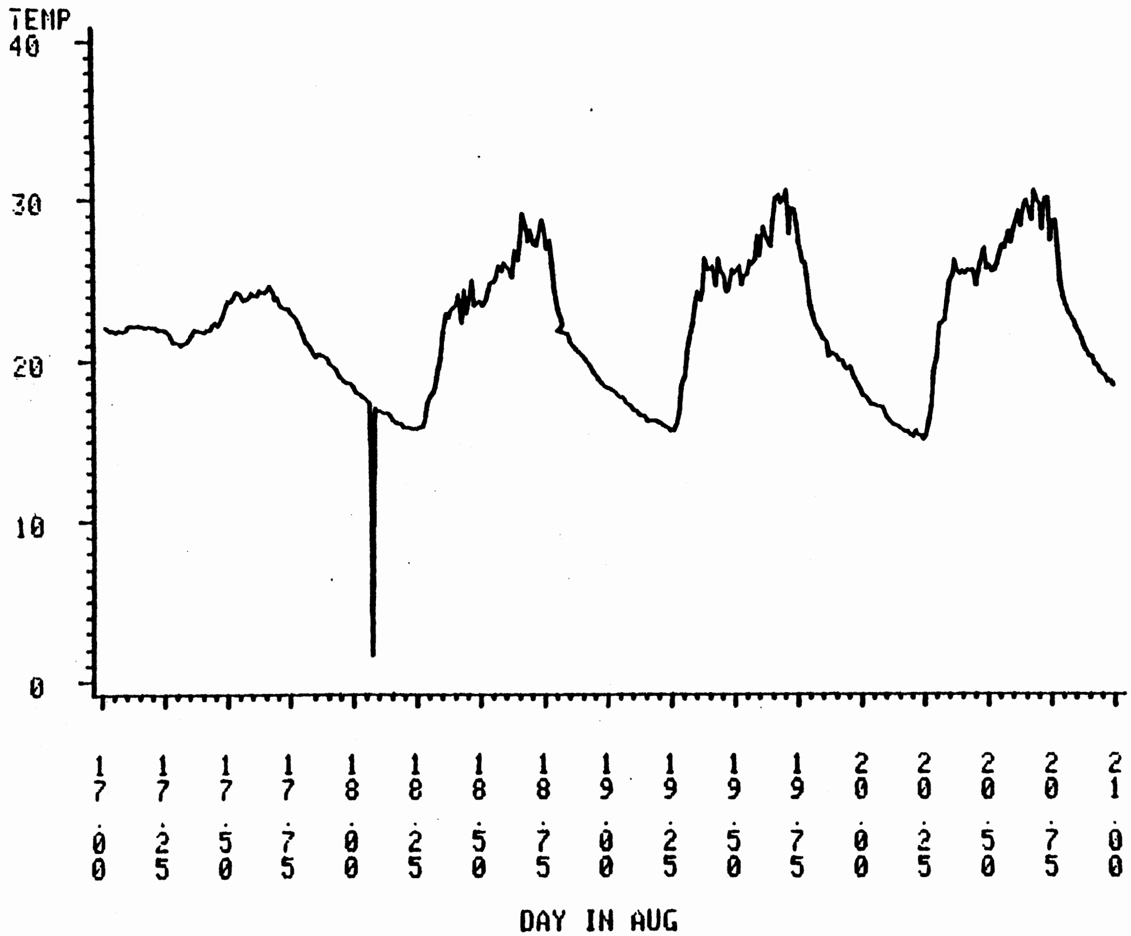

Figure 4. Plot of Outdoor Temperatures for Four Days in August 1981

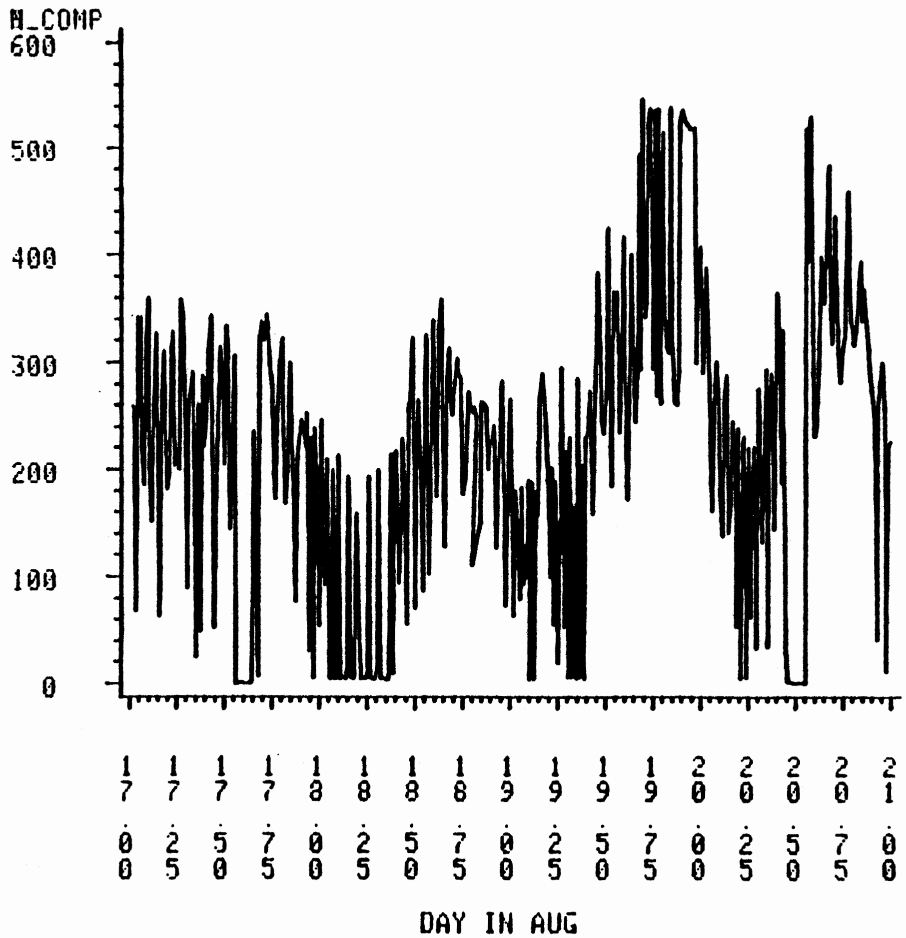

Figure 5. Energy Consumption of a House for Four Days in August 1981

in Oklahoma for the same week. The house has a conventional air source heat pump system.

The climatic conditions of a particular place contribute strongly as to whether the peak loads are higher in summer or winter. The utilities will need larger power production capacity to handle these loads which may force them to build more plants. As these heavy peak loads occur for a relatively short duration of time, this increased reserve power production capacity will remain idle for most of the time in the year. This is expensive for the utility company as a lot of capital is involved in this investment.

As has already been stated, at outdoor temperatures below the balance point, electric resistance backup heat is added to the house. Since resistance heat has a coefficient of performance* of unity the total coefficient of performance of the entire heating system is reduced. So, even if a heat pump operates at high coefficient of performance above the balance point, when the system requires a large amount of backup heat, the system efficiency throughout the season will be low. This indicates that the performance of a heat pump system depends not only on the coefficient of performance of the heat pump but also on the capacity of the heat pump system used and the seasonal temperature distribution to which the building is exposed. The ratio of the total seasonal demand to the energy required is known as the seasonal performance factor (SPF) which is the best measure of the heat pump system performance.

Several solutions have been proposed to improve the seasonal performance factor of the heat pump and to reduce the peak demand of the system. One of them is to use the earth as a source/sink. The ground

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*Coefficient of performance is defined as the ratio of the heating or cooling effect desired to the work input.

temperature, at sufficient depth, is nearly constant throughout the year. At depths from 30 to 60 ft. the seasonal variation in ground temperature is not more than 1^{o}F. For a major part of the United States, the ground temperature for these depths is in the range of 60 to 64^{o}F (2). So the ground temperature is higher than the ambient temperature in winter, and lower in summer. This indicates that the ground has a potential to act as a heat source/sink for the heat pump and will improve the performance (both SPF and demand) of the heat pump system.

Solar energy might be used to further improve the efficiency of ground source heat pump systems. In the heating mode the solar heat might be added to the well water loop and in turn to the evaporator in the heat pump system, resulting in an improvement in its efficiency. In the cooling mode, the solar collectors might be used to reject heat on the cool summer nights.

Both experimental and theoretical work is needed to determine the effect of including both ground and solar sources in the heat pump systems. The Electric Power Research Institute (EPRI) and Oklahoma Gas & Electric Company (OG&E) have sponsored a project whose objective involves the comparison of three different heat pump systems presently installed in three nearly identical houses in Perkins, Oklahoma. The three heat pump systems are a conventional air source, a ground source/sink, and a solar-assisted ground source/sink. The three systems are being compared in terms of

1. C.O.P. and hence the total energy consumption (SPF)
2. Peak demand on the utility
3. Reliability and maintenance
4. Overall life cycle costs.

It is also thought that inclusion of a storage tank will help greatly in reducing the peak demand of one or more of these systems, both in summer and winter. The main objective of this study was the development and application of a computer subprogram to simulate a storage tank to be included in the solar-assisted ground source/sink heat pump system.

The simulation procedures and the discussion of results are presented in detail in later chapters.

CHAPTER II

OBJECTIVE

This study has mainly two objectives:

1. The development and application of a computer subprogram to simulate a storage tank to be included in the solar-assisted ground source heat pump system. Hopefully a storage tank could make a significant contribution in reducing the peak demand on the system, which is one of the important objectives of this project.

2. The development of a computer subroutine to simulate the vertical well which serves as a link to exchange the heat between the heat pump and the ground. Because of leakage problems in the original well, a new well based on a different design, was installed in March 1982. This new design is much cheaper, simpler, easy to install and maintain than the previous system.

CHAPTER III

DESCRIPTION OF THE SYSTEMS

The heat pump systems being compared in this project are installed in three identical houses in Perkins, Oklahoma. These houses are facing North, and they are built on adjacent lots. Due to their location along an East-West road, the houses have been designated as the East, the West, and the middle house. The East house is equipped with the air-to-air heat pump, the West with a ground source/sink heat pump, and the middle with a solar-assisted ground source/sink heat pump. All of the houses have been instrumented for direct measurement of heat pump performance data. The weather monitoring sensors have been located on a 25-foot pole behind the middle house.

An air-to-air heat pump is installed in the East house. This system is considered the base line of comparison. The 1.5-ton unit is a Carrier heat pump.

The West house has a water source heat pump. The schematic of the system is shown in Figure 6. This system uses a Commandaire water source heat pump with a 250-foot well (originally a 5-inch pvc pipe cased and sealed) and using ordinary water at 15 psig. The water is pressurized to prevent the air buildup in the system. Two 1-inch pvc pipes are used to circulate the exchange fluid through the water refrigerant heat exchanger. In the original version water was taken from the top of the well to go through the heat pump and returned to the bottom

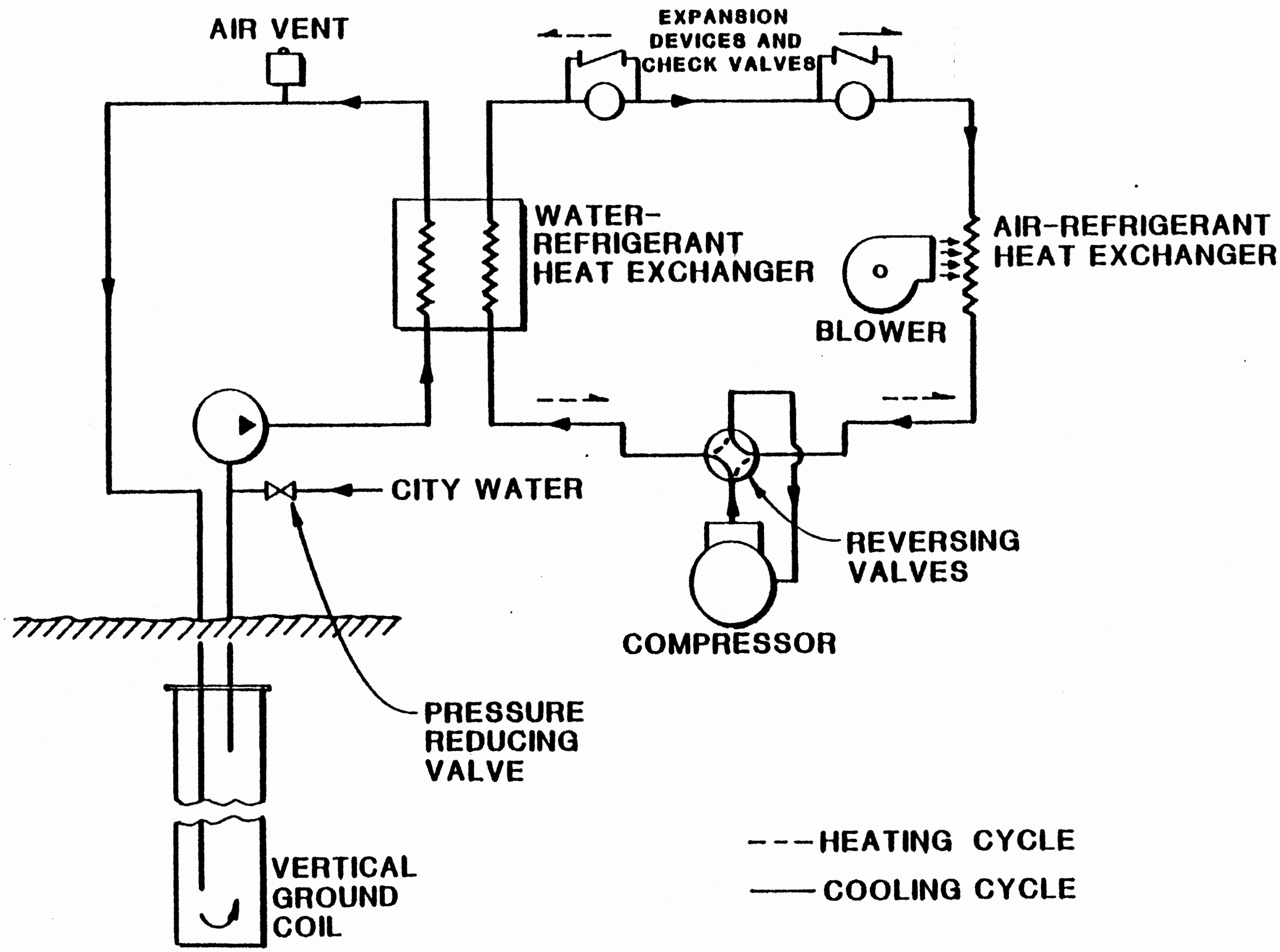

Figure 6. Schematic of the Ground Source/Sink Heat Pump System

of the well. As the water rises to the top, heat is transferred to or from the surrounding earth.

Figure 7 shows the schematic of the solar-assisted ground source/ sink heat pump system in the middle house. The heat pump unit and the well are the same as those described for the previous system. The addition of the solar loop is meant for improving the efficiency of the heat pump system during both the heating and cooling modes. The solar loop consists of the solar panels, the heat exchanger, the circulation pump and the accumulator. There is an array of five collector panels. The panels are bare steel plates (7 ft x 4 ft) with 0.5-inch copper tubing spaced four inches center to center. The panels have non-selective black coating. The array is mounted vertically against the south wall of the house. The solar loop uses a 50% ethylene glycol solution as the heat transfer fluid.

When the heat pump is in the heating mode, the collected solar heat will be transferred to the well water loop and in turn to the refrigerant, hopefully resulting in improvement of the efficiency of heat pump cycle. When the heat pump is in the cooling mode, it is rejecting heat to the well water loop. The solar loop was intended to be helpful in dissipating heat from the well water loop on cool summer nights. This was hoped to result in more efficient heat pump operation.

The solar loop controller starts the solar loop pump when the temperature of the collector panels is 20^{o} higher in winter, or lower in summer, than the temperature of water entering the ethylene glycol-water exchanger. The controller also starts the well water pump if it is not operating at that time. When the temperature difference decreases to $5^{o}F$, the pumps are switched off. The well water pump will continue to run if the compressor is on.

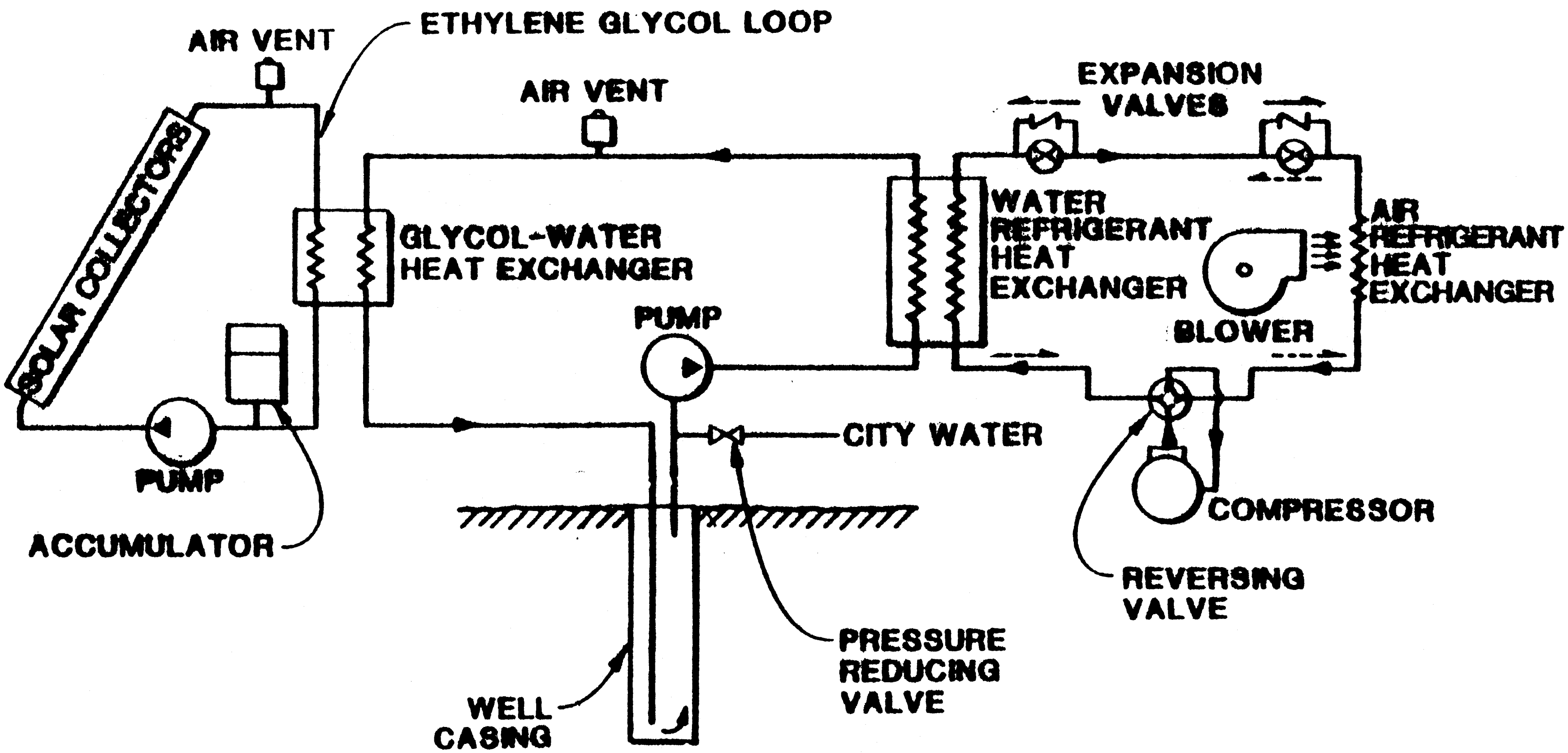

Figure 7. Schematic of the Solar-Assisted Ground Source/Sink Heat Pump System

All three houses have a domestic water preheater. When the compressor is running, the water preheater circulates the water through a coiled tubing heat exchanger and collects heat from hot gases exiting the compressor. This device operates when the compressor is running; the heat pump may be in a heating or cooling mode.

CHAPTER IV

APPLICATION OF THE STORAGE TANK

Need for Thermal Storage

In ground source heat pump systems the well serves as the link between the source/sink (the earth), and the heat pump. In very cold weather when the heating load for the house is large, the heat pump is "on" most of the time. So the temperature of the water in the earth water heat exchanger loop almost decreases continuously. This in turn reduces the capacity of the heat pump to supply heat to the house. This fact is clearly indicated by Figure 12. A stage comes when the heat pump capacity is insufficient to meet the heating load on the house, and the resistance heat comes on. This will put a large demand on the utility company during a peak period. Increase in the peak demand is a major consideration for a utility company because it forces them to increase their capacity for power production. If total energy sales do not increase at the same rate, a poor load factor results.

It was thought that demand for a particular house could be reduced by adding a storage tank in the heat pump system. Energy can be stored in it at off peak hours. Stored energy can be used effectively when the demand on the system is maximum. This will reduce the peak demand on the system.

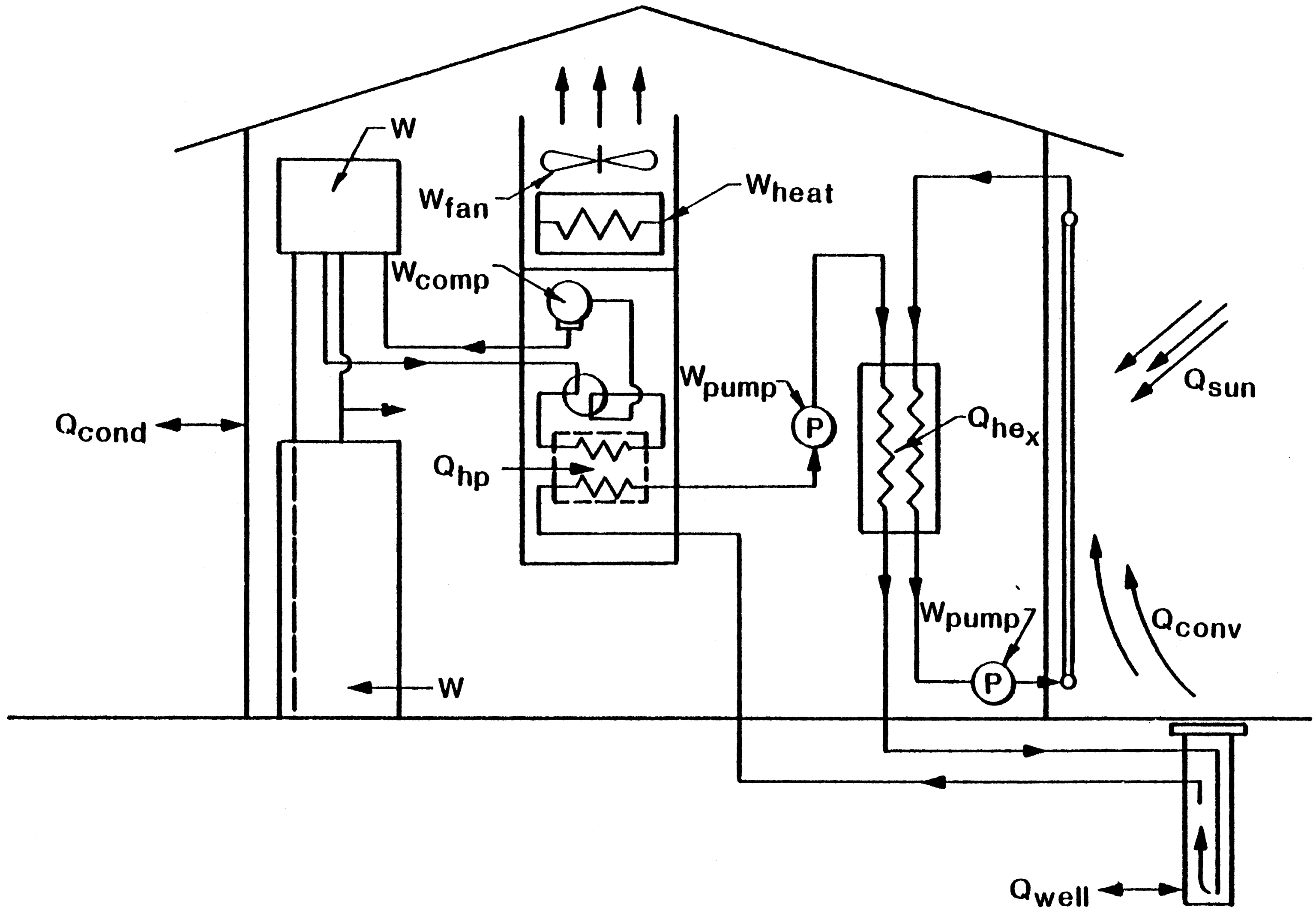

Figure 8. Energy Gains and Losses in the System

Storage Tank Simulation

The common method of storing hot water is in an insulated upright cylindrical tank. These water tanks may operate with significant degrees of thermal stratification resulting from the temperature of water at the top of the tank being higher than that at the bottom. Thermal stratification is desirable for mainly two reasons: First, comparatively hotter water is drawn from the top of the tank and is supplied to the load. Second, comparatively cooler water is drawn from the bottom of the tank and is circulated through the solar collectors to collect heat. This promotes more efficient operation of the solar collectors.

The basic heat and mass transfer relations governing the behavior of a storage tank with thermal stratification are complicated. However, the tank can be modelled by making some simplifying assumptions. The tank is assumed as being divided into N fully mixed isothermal nodes. An energy balance is written for each node of the tank. Finally there will be a set of N simultaneous differential equations that can be solved for the temperatures of the N nodes as functions of time (5, 9).

Before writing an energy balance, it is necessary to make assumptions about how the entering water stream is distributed to various nodes. Actually, the mixing of flow streams will depend on the size and shape of the storage tank, location of inlets and outlets, and flow rates of entering and leaving streams. If the fluid flows into and out of the storage tank at low velocities, then it can be assumed that the entering water flow finds its way down inside the tank to a particular node where its density nearly matches with that of the water in that particular node.

Figure 9 shows a hot water storage tank with thermal stratification. It is divided into N fully mixed nodes. The energy balance on a segment

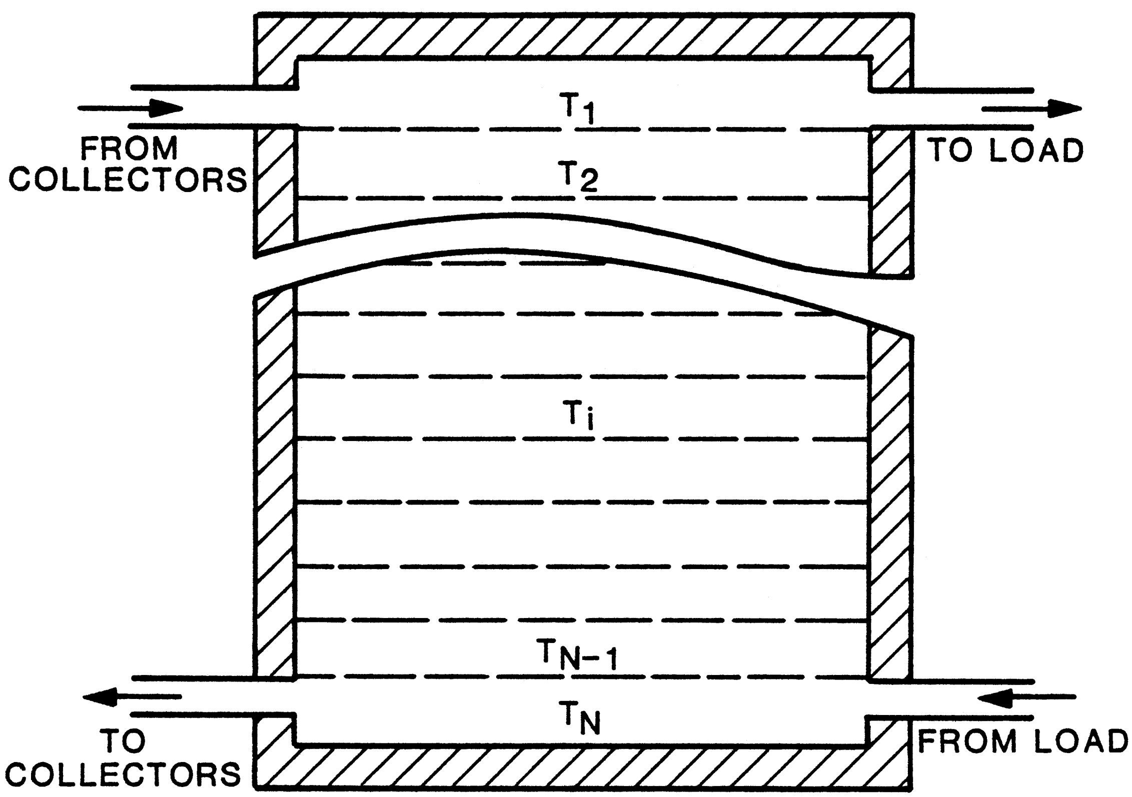

Figure 9. Schematic of the Hot Water Storage Tank With Thermal Stratification

i is expressed as follows:

$$\frac{M}{N} Cp \frac{dT_c}{dt} = \alpha_i m_c \; Cp \; (T_c - T_i) + B_i \, m_L C_p (T_L - T_i)$$

$$+ V_i \, m_c Cp (T_{i-1} - T_i)$$

$$- \delta_i \; m_L \; C_p (T_i - T_{i+1})$$

$$+ \frac{UA}{N} (T_{amb} - T_i) \tag{1}$$

where

$$\alpha_i = \begin{cases} 1 \text{ if } T_{i-1} > T_c > T_i \\ 0 \text{ otherwise} \end{cases}$$

$$\beta_i = \begin{cases} 1 \text{ if } T_i > T_L > T_i + 1 \\ 0 \text{ otherwise} \end{cases}$$

$$\gamma_i = \sum_{j=1}^{i-1} \alpha_j$$

$$\delta_i = \sum_{j=i+1}^{N} \beta_j$$

where:

M = mass of fluid in storage tank (lbm);

Cp = specific heat of fluid in the tank $\left(\frac{BTU}{lbm \; F}\right)$;

T = temperature of fluid in the tank (F);

t = time (hr);

m_c = mass flow rate of hot fluid from collector $\left(\frac{lbm}{hr}\right)$;

T_c = temperature of hot fluid entering the tank;

m_L = mass flow rate of the fluid drawn from the tank to supply the load $\left(\frac{lbm}{hr}\right)$;

T_L = temperature of the fluid coming back from the load to the tank (F);

U = overall heat transfer coefficient for the tank surface (BTU/hr-ft^2-F);

A = surface area of the tank (ft^2); and

T_{amb} = ambient temperature (F).

The net flow between nodes can be either up or down, depending upon the magnitudes of the collector and load flow rates and the values of the two control functions α and β at any particular time.

This set of N simultaneous differential equations can be solved by using Runge-Kutta method with the help of a computer.

However, simulations of the long term performance of residential solar space and water heating systems with fully mixed and stratified liquid storage have indicated that the improvement in system performance due to thermal stratification is small (5).

So the computer subroutine to be included in the simulation program is based on the conservative assumption that the tank is fully mixed, i.e., isothermal.

An energy balance can be written as shown in Equation (2).

$$MCp \frac{dT}{dt} = m_c C_p (T_c - T) + m_L \; Cp(T_L - T) - UA(T - T_{amb}) \qquad (2)$$

The tank temperature can be obtained by integration of its time derivative.

The overall heat loss coefficient was taken from (14).

$$U = 0.2 \frac{BTU}{hr\text{-}ft^2\text{-}{}^oF}$$

The storage tank is cylindrical in shape with the diameter equal to its length.

Application of Thermal Storage

An analysis was made of the effect of adding a storage tank in the solar-assisted ground source/sink heat pump system in the middle house.

The storage tank is added in parallel to the well, as shown in Figure 10.

Controls are set such that energy is stored in the tank when

1. heat pump is off, and
2. $(T_c - T_{Tank}) > 20\ ^oF$

where

T_c = collector temperature; and

T_{Tank} = temperature of water in the storage tank.

The basic objective is to include the storage tank in the circuit when the heat pump capacity is insufficient to handle the heating load on the house and thus prevent the use of resistance heat.

Two options were considered when setting controls to include the storage tank in the circuit.

Option I - when temperature of the water going to the well goes down to 34^oF the storage tank is included in the circuit. Storage tank water is in the range of 55^o to 60^oF. The storage tank takes care of the system until the storage temperature drops down to 40^oF. As the ground is warmer as compared to the well water, the well picks up heat from the ground in the meantime. The well is then again included in the circuit, and the storage tank is valved off and isolated from the circuit unless some additional energy is added to it from solar collectors.

Option II - when the temperature of water going to the well goes down to 38^oF, the well is cut off from the circuit and the storage tank is included in the circuit. Now comparatively hot water from the

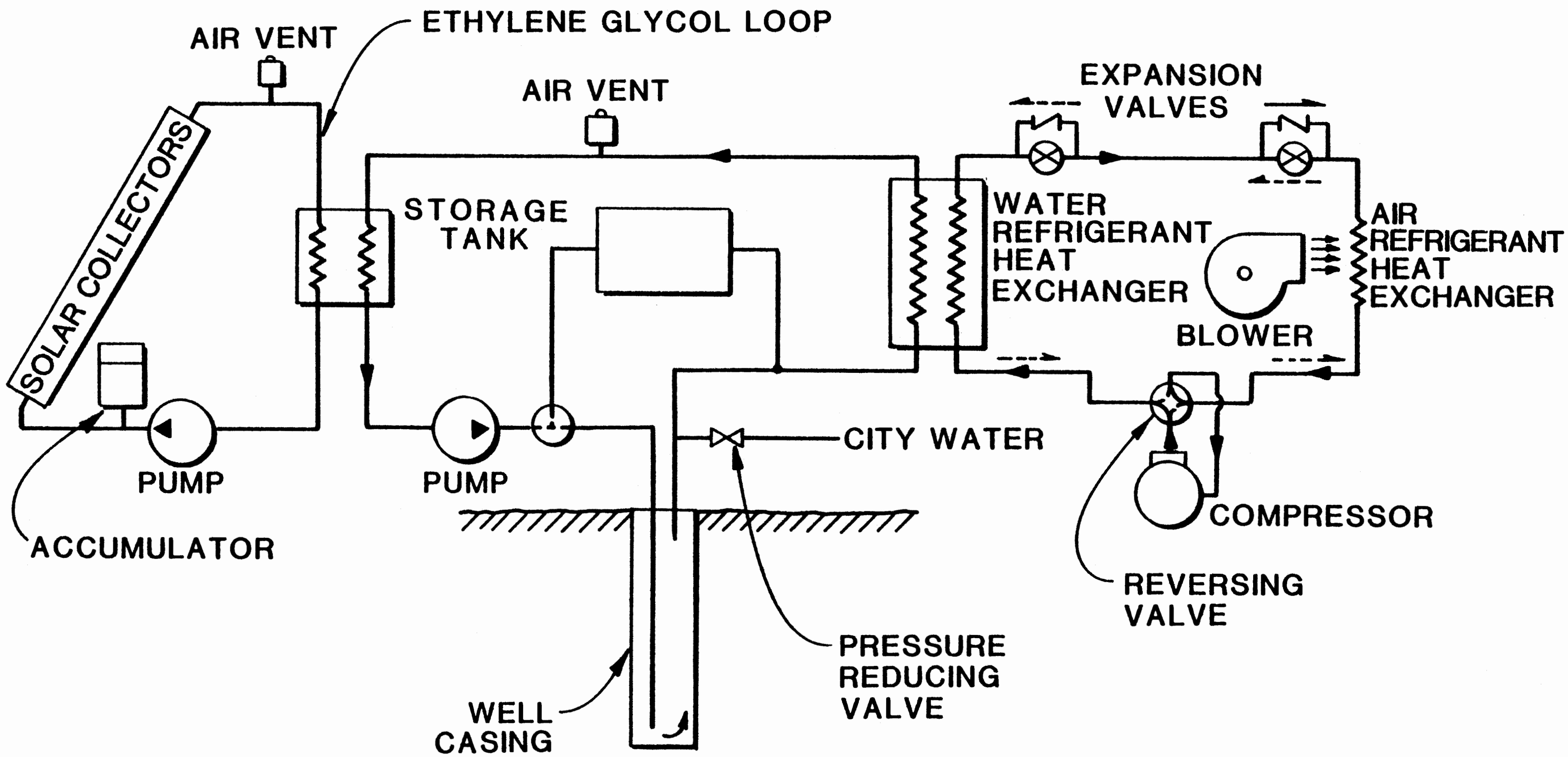

Figure 10. Schematic of the Solar-Assisted Ground Source/Sink Heat Pump System With a Storage Tank

storage tank is circulated in the loop. The well collects heat from the ground in the meantime. When the well water temperature goes to 45^{o}F, it is included in the circuit. So we switch back and forth between well and storage tank all the time.

The simulation program was run for artificially imposed cold day conditions for two days because the actual recorded temperatures were not low enough for the well water to freeze.

TABLE I

COMPARISON OF TWO DIFFERENT OPTIONS OF CONTROL SETTINGS TO INCLUDE THE STORAGE TANK IN THE HEAT PUMP SYSTEM

	Option I	Option II
Total compressor and fan KWH consumption	50.12	44.76
Total resistance heat (KWH)	5.36	0.0
Heat pump average (COP)	1.89	2.09

The control strategy stated in Option II appears to be better than Option I. This result can be predicted by plotting "well water temperature rise" against "time" as shown in Figure 11. The slope of the curve decreases with time continuously. So it is better to let the well recover in short spells of time, keeping the temperature rise to 4 to 6^{o}F. This way we can make maximum use of the well and collect maximum energy from the ground.

The simulation program was run for a cold week in February 1981

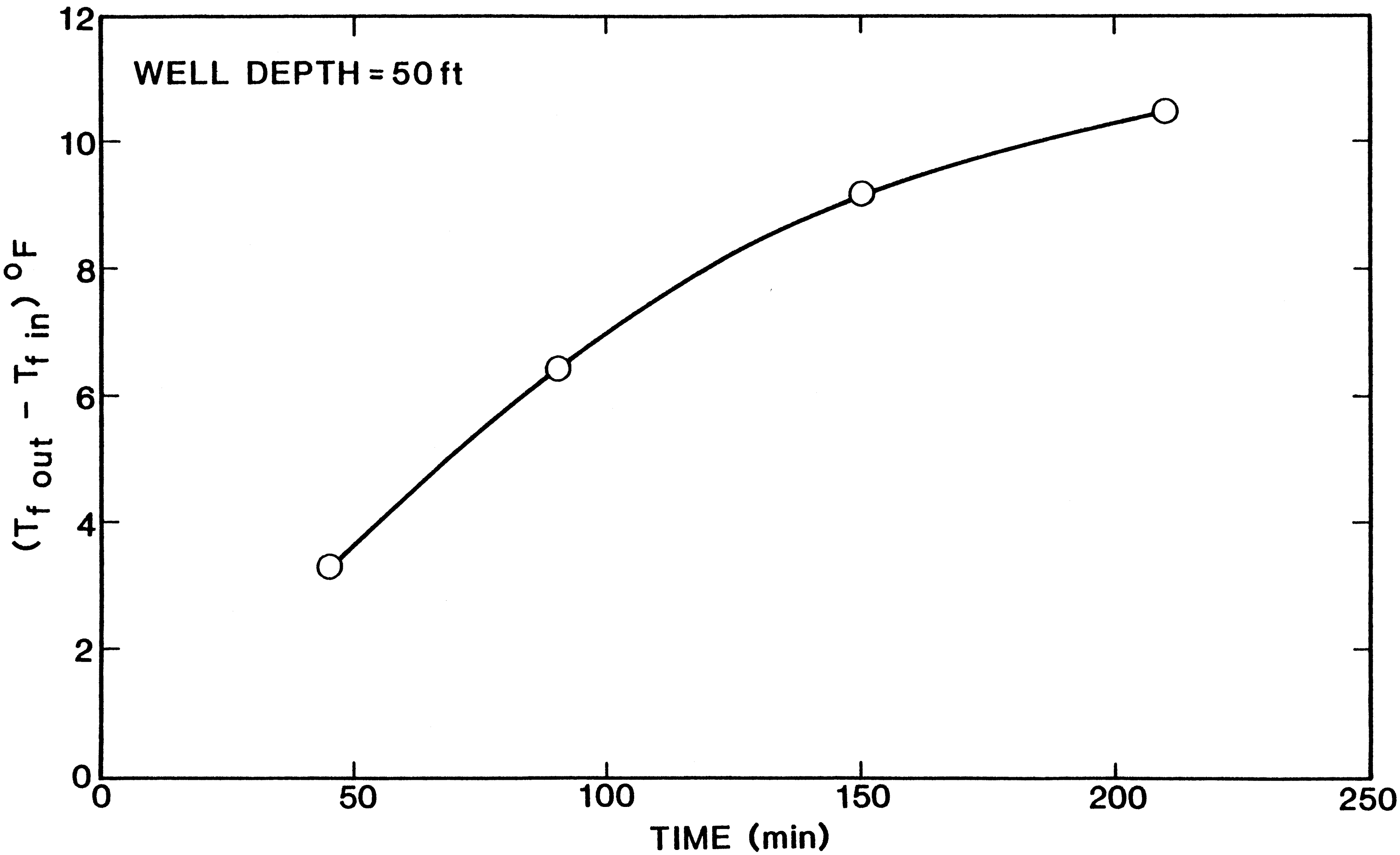

Figure 11. Well Water Temperature Rise as a Function of Time for Zero Heat Input

(February 9-16). Measured weather data, which was stored on a magnetic tape, was transferred to a TSO file and the necessary inputs for the simulation program were read from it. The results show that one sunny day raises the temperature of water in the storage tank of 1000 gallon capacity of approximately 5°F. Since bare collectors are being used, a lot of heat is lost by convection.

Stored energy can be utilized to prevent the freezing of water in the well on a very cold night. The chosen week of February, for which the simulation program was run, was one of the coldest weeks in the year. But still the outdoor temperature was always above the balance point* so the well water did not freeze and resistance heat never came on. In order to observe the usefulness of the storage tank, the simulation program was run at the balance point conditions.

Figure 12 is used to find the balance point for the houses with ground coupling. Graph I shows how the heating loads (BTU/hr) on the houses vary with the outdoor temperature. CHLSYM (8), the transient heating-cooling load simulation program was used to predict load requirements of the house. The program reads fifteen minutes' interval data from the weather tapes. The information regarding the structural geometry of the house is stored on the separate magnetic discs, which serves as another input to the program.

Graph II is a plot of the heat pump heating capacity (BTU/hr) vs. the entering water temperature, which in this case is the temperature of water coming out of the well. The steady state performance of the heat pump units is described by polynomials obtained from least-square

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*Balance point is the outdoor temperature at which the heating capacity of a heat pump is equal to the heating load of the space being heated.

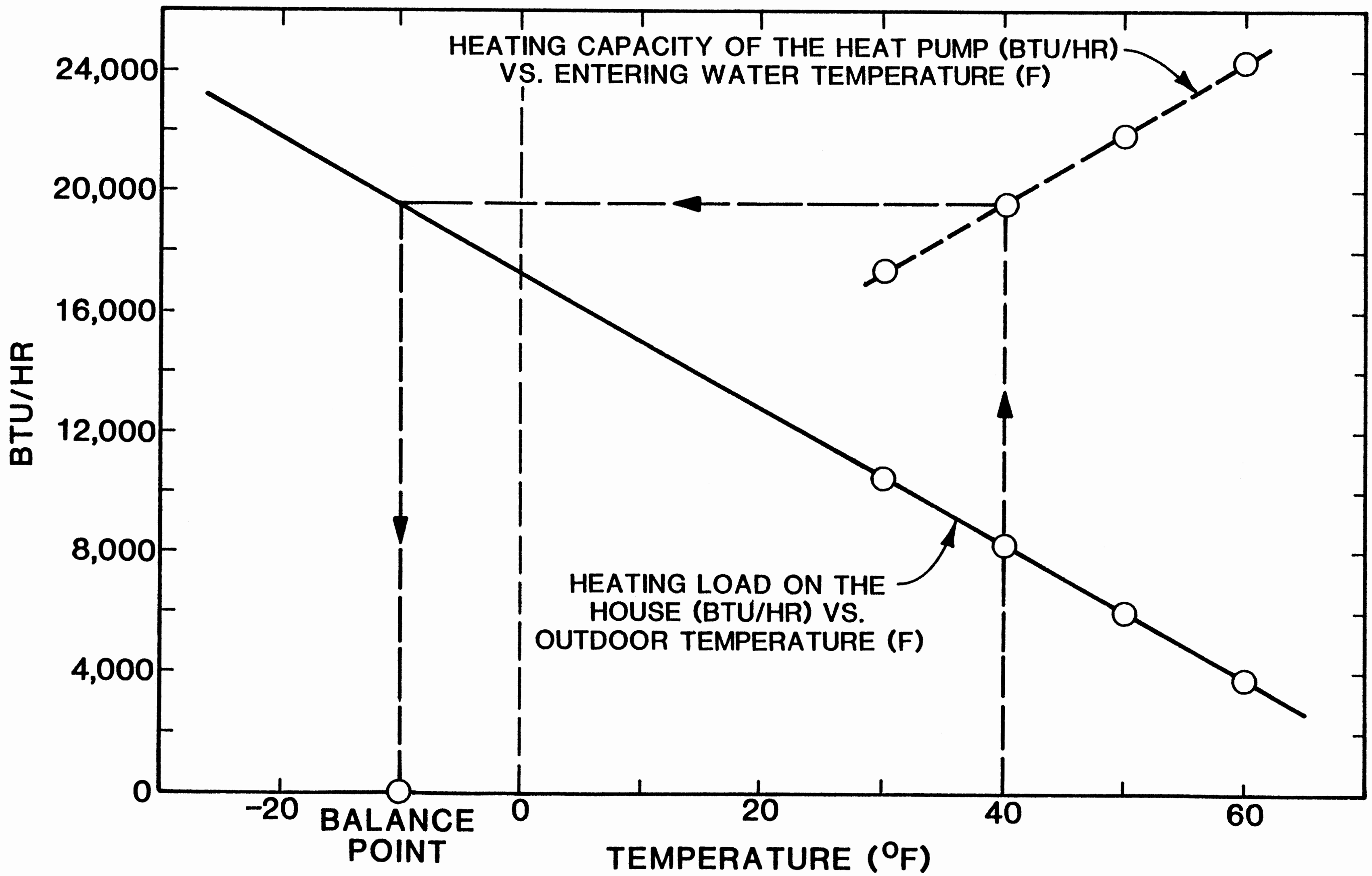

Figure 12. Determination of the Balance Point of the System

curve fit of the manufacturer's performance data (6). The two graphs are plotted on the same sheet with the same scale on both the axes. If the temperature of water coming out of the well is 40^{o}F, the corresponding heating capacity of the heat pump is 19600 $\frac{BTU}{hr}$.

If the heating load on the house is 19600 $\frac{BTU}{hr}$, then the corresponding outdoor temperature as read from the graph is (-10^{o})F, which is the balance point of the system.

The simulation program was run for the middle house under these artificially imposed cold day conditions. The controls were set as described in Option II. Comparison of results is as shown in Table II.

TABLE II

EFFECT OF STORAGE TANK ON THE PERFORMANCE OF HEAT PUMP SYSTEM IN WINTER

	With Storage Tank	Without Storage Tank
Total compressor and fan consumption (KWH)	44.76	56.71
Total resistance heat (KWH)	0.0	26.72
Heat pump average coefficient of performance	2.09	1.62

The temperature of water in the storage tank dropped by 18^{o}F. So the storage tank took care of the system for this exceptionally cold day. The resistance heat never came on. Also, the well water did not freeze. Three to four sunny days are enough to store sufficient energy in the tank, which can be utilized on a very cold day. If any other kind of cheap fuel is available at a particular place, it can be used to add more energy to the tank.

The simulation program was run for the third and fourth weeks in November 1981, and the first week of December 1981. For these sunny days, when the outdoor temperature was not very low, the heat pump was off for most of the time and a lot of energy was stored in the tank. The storage tank temperature reached above 80°F, which was not desirable because the collectors will operate at lower efficiency under these conditions. Since it is unlikely that the temperature of water entering the well will go down to 35°F under these conditions, the controller settings must be changed to make use of the stored energy in the tank.

The same principle was used to reduce the peak demand in summer. The idea was to cool the water in the storage tank by circulating it through the solar collectors on the cool summer nights. The simulation program was run for a hot week in August 1981, and the results obtained were as shown in Table III.

TABLE III

EFFECT OF STORAGE TANK ON THE PERFORMANCE OF THE HEAT PUMP SYSTEM IN SUMMER

	With Storage Tank	Without Storage Tank
Total compressor and fan KWH consumption	38.85	39.11
Total heat pump operating time (hr)	16.54	17.12
Heat pump average coefficient of performance	2.16	2.13

Simulation program results for other days in summer also indicate that the storage tank is more useful in reducing the peak demand in winter than in summer.

Circuit Modifications

With the present design of the heat pump system in the middle house, it is not possible to make use of the energy collected by solar collectors on summer afternoons. This energy can be utilized by modifying the circuit, as shown in Figure 13.

The energy collected by solar collectors is exchanged with water in the domestic water preheater tank through a heat exchanger. The water preheater circulates the water through another heat exchanger and collects heat from the hot gases exiting the compressor.

When the simulation program was run for sunny days in winter, it was observed that when both the heat pump and solar loop pump are on the temperature of water going to the well is normally 66^{o} to $70^{o}F$, and the temperature of water coming out of the well is 62^{o} to $64^{o}F$. The heat lost to the ground is expected to be recovered later when the heat pump is on and solar loop pump is off.

An attempt was made to modify the circuit and utilize this energy more effectively to improve the overall efficiency of the system. The circuit is modified, as shown in Figure 14. A fraction of the flow is bypassed to the well. The other fraction collects heat from the fluid in the solar loop, and then both the fractions are mixed together. The best results are obtained when 80 percent of the flow is bypassed to the well. The simulation program results for cold days in February 1981 show a saving of 0.75 KWH per day. The energy saving is increased when the collector area is increased. Doubling the collector area in the

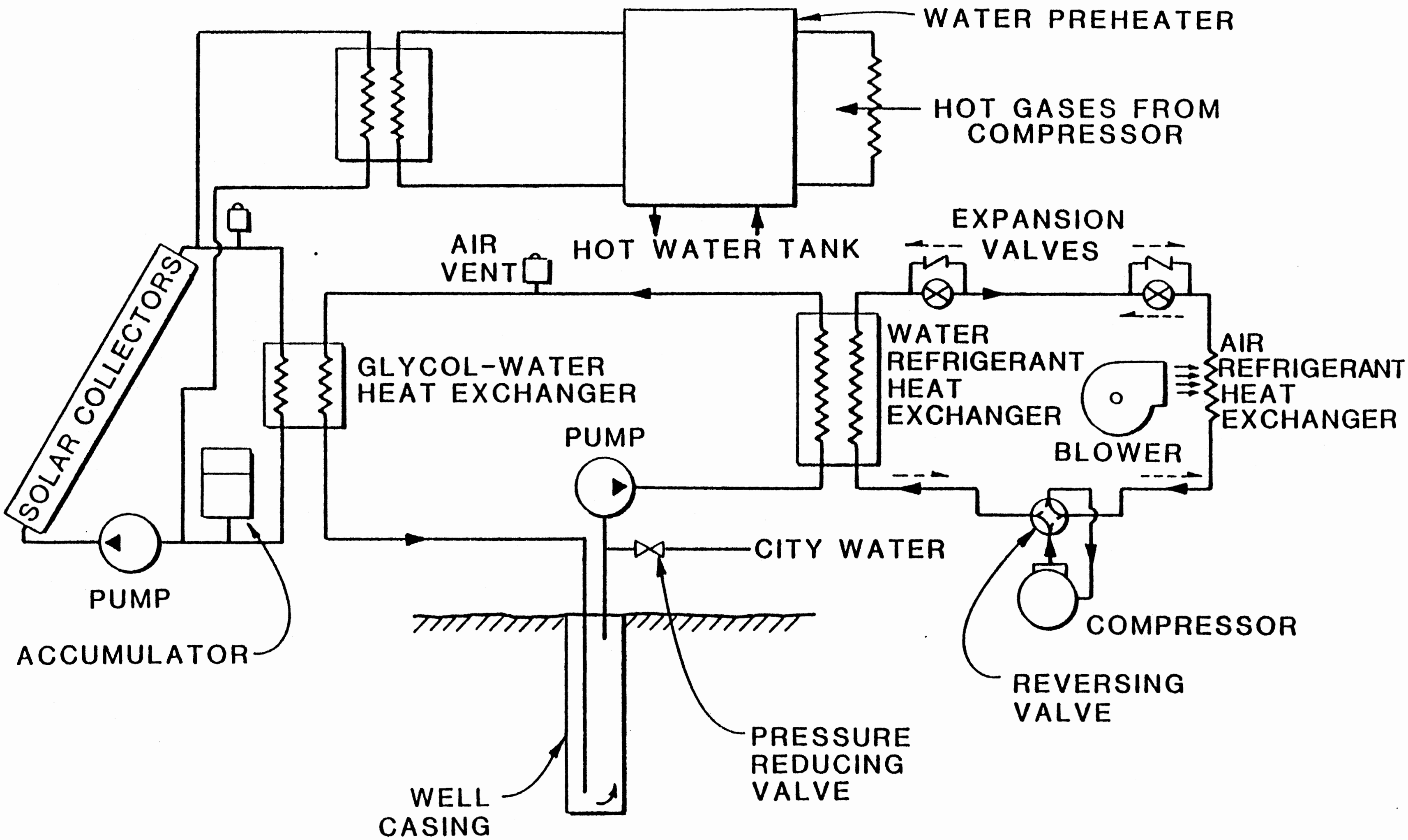

Figure 13. Suggested Circuit Modification for Heat Pump Operation in Summer

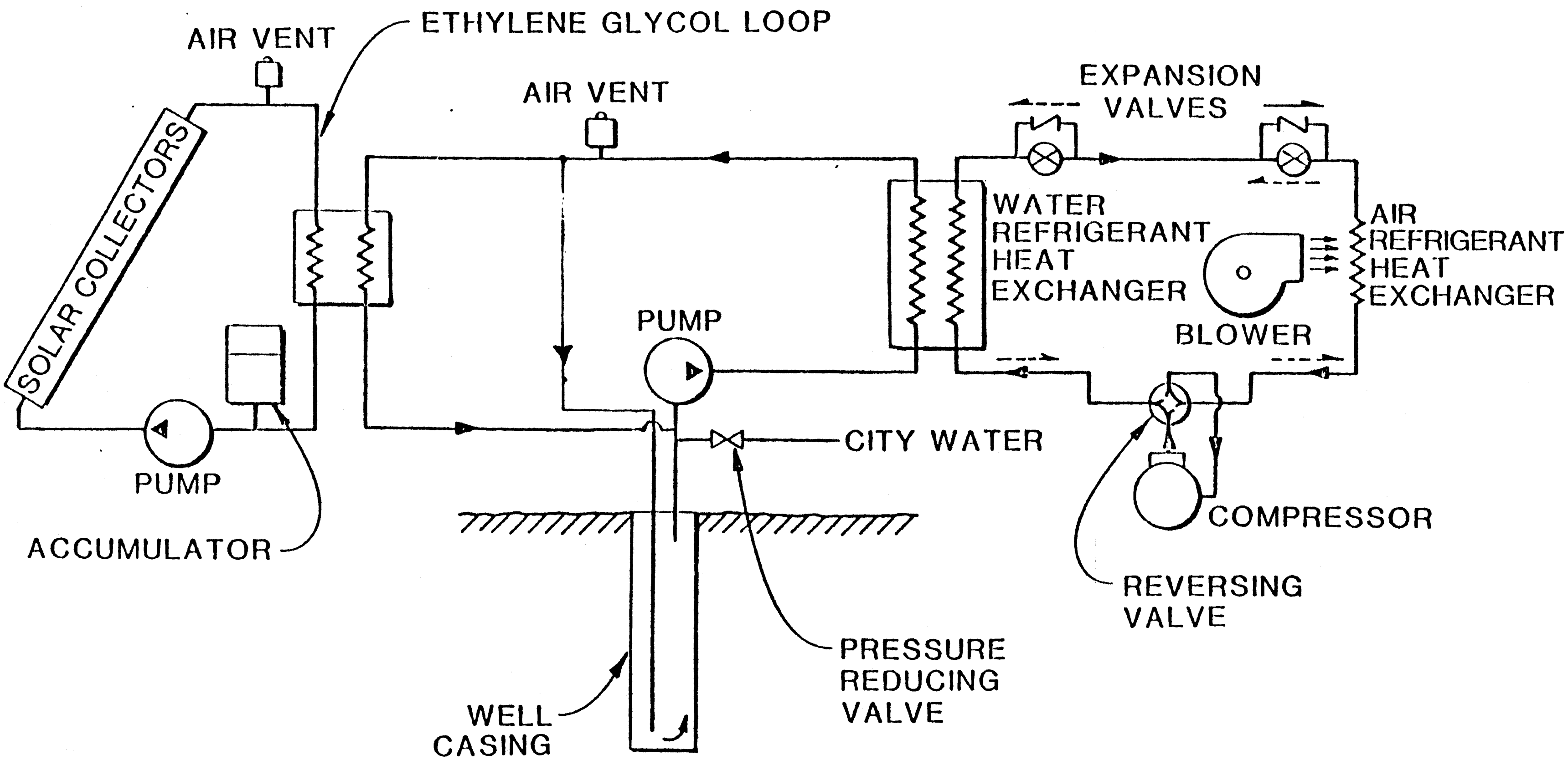

Figure 14. Suggested Circuit Modification of Heat Pump Operation in Winter

above case resulted in energy saving of 1.4 KWH per day.

The simulation program runs for November 1981 show an energy saving of 1.8 KWH per day. This is due to the fact that when outdoor temperature is not very low (never goes below 32°F), the heat pump is not on continuously, but it goes on and off frequently. When the heat pump is off, the well gets a chance to recover and in turn collects more energy from the ground.

In general, this circuit modification seems to improve the efficiency of the system; however, experimental observations are necessary to support these findings.

Additional Storage Subsystem Requirements

The application of the storage tank is to reduce the peak demand in a heat pump system as discussed above. These results are based on the computer simulation of all components in the system. In practice, when the tank is to be installed in the house, a lot of other requirements must be satisfied. The designed function of the storage subsystem will have an influence in deciding the shape, size, location, interface requirements, insulation requirements, etc. Some of the important factors are discussed briefly here.

1. Choice of material - Most of the conventional construction materials like steel, aluminum, wood, concrete, and various other plastics have potential in hot water solar storage systems. A variety of insulating materials and anticorrosion coatings are also useful for these tanks. Durability, weight, and cost are the major considerations in the selection of proper material for the tank. A lot of factors affect the durability of storage tanks, the major ones being corrosion, heat and chemical deterioration, and insulation degradation due to

water and vapor infiltration. A detailed discussion about various tank materials is given in reference (7).

2. Compatibility with structure - The storage system must be compatible with the other elements of the system connected to it. It should have ready facility for the attachment and integration of input and output pipe connections, heat exchangers, valves, and control system components. As far as possible, the system should cause the least interference with the normal functions of residents and should not damage the aesthetic value of the house.

3. Safety and health - The system operation should be safe for the residents of the house. An adequate number of valves should be fitted to avoid any cross connections with the domestic water supply and consequent contamination.

4. Maintainability - The system must be readily repairable with minimum skills and minimum downtime. System components must be readily accessible for adjustment, repairs, etc. System components must be readily replaceable at the end of their service life.

5. Flexibility - The system design should promote maximum flexibility in terms of future changes in capacity, adding insulation, adding function, etc.

6. Capacity - It is difficult to come up with some standard rules to decide the capacity of the storage tank for a particular house at a particular location. For the heat pump system in the middle house, a storage tank size 1000 to 1200 gallons seems to be an optimum size. For smaller tank sizes, the change in the storage tank water temperature will be larger for the same heat addition or removal. For a tank of 600 gallon capacity, the temperature change in one day for the balance point conditions in the middle house would be approximately 30°F.

This means that the tank water temperature should be above 70°F, which will cause more heat losses from the tank. Secondly, if the storage water temperature is large, it will promote inefficient operation of solar collectors and consequently the amount of heat added from the solar loop will be less.

CHAPTER V

SIMULATION OF VERTICAL EARTH WATER HEAT EXCHANGER

Initially, the design of vertical earth water heat exchanger was as shown in Figure 15. Water is taken from the top of the well to go through the heat pump and is returned to the bottom of the well in a $1\frac{1}{2}$-inch pvc pipe. As the water rises to the top, heat is transferred to or from the surrounding earth. This configuration has given successful experimental results, but because of the leakage problem in the well cap at the upper end of the well, a new well was installed in March 1982 based on a different design, as shown in Figure 16. The new design is a simple "U" tube of high density polyethylene PE 3408. This new design is much cheaper, simpler, easier to install and maintain than the older pvc system.

Simulation Procedure

The literature search revealed that relatively few techniques have been developed to efficiently give the relationship of well water and soil temperatures as functions of time and position. Comparatively more efforts have been spent on analytical solutions. Numerical methods based on finite difference approximations offer a powerful technique for the solution of these heat transfer problems. The technique used is fairly straightforward and generally applicable, and gives reasonably accurate answers to the problems which are often awkward and difficult to solve analytically (12). Wise (10) presented an analytical solution

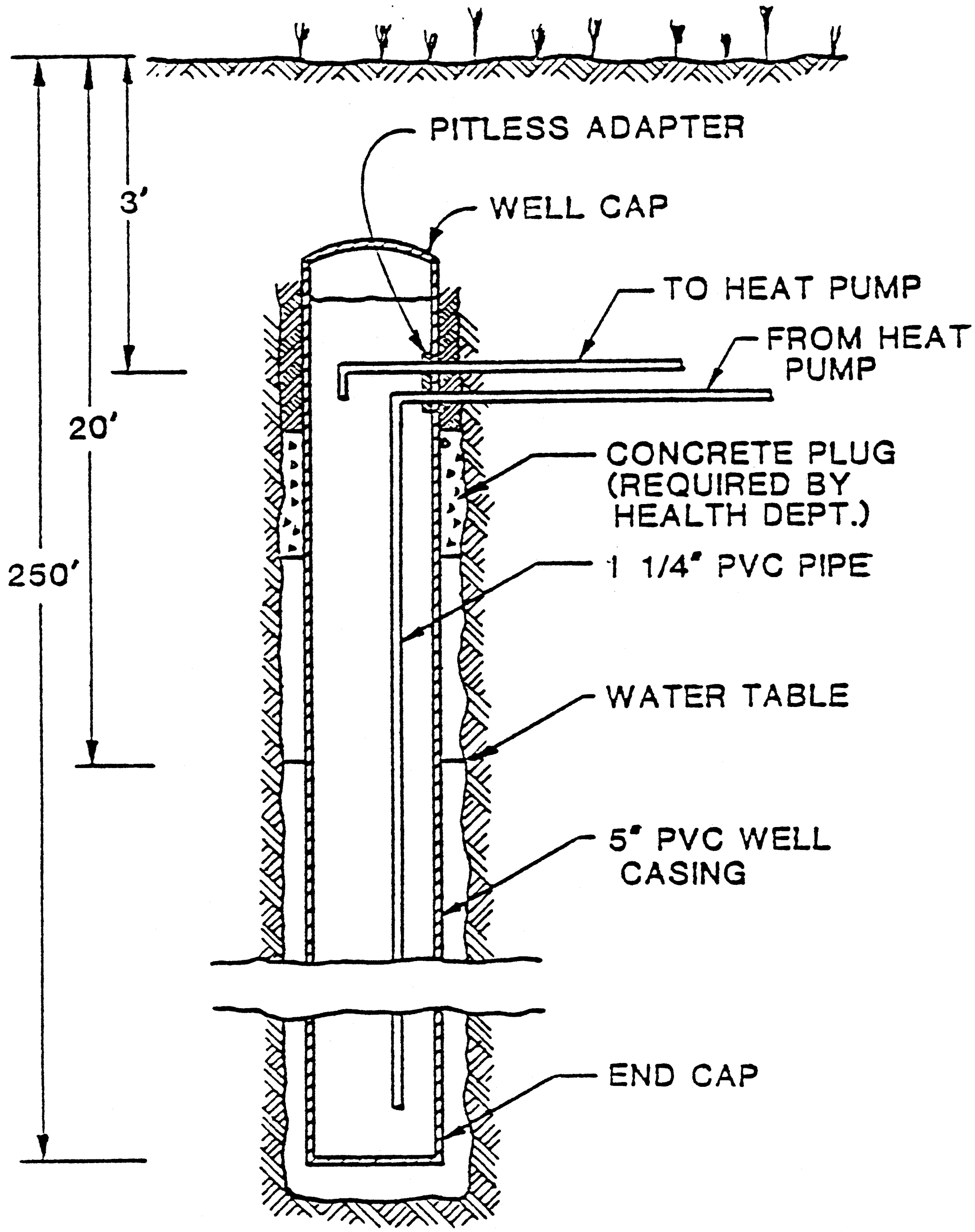

Figure 15. Schematic of the Vertical Earth Water Heat Exchanger System Based on the Old Design

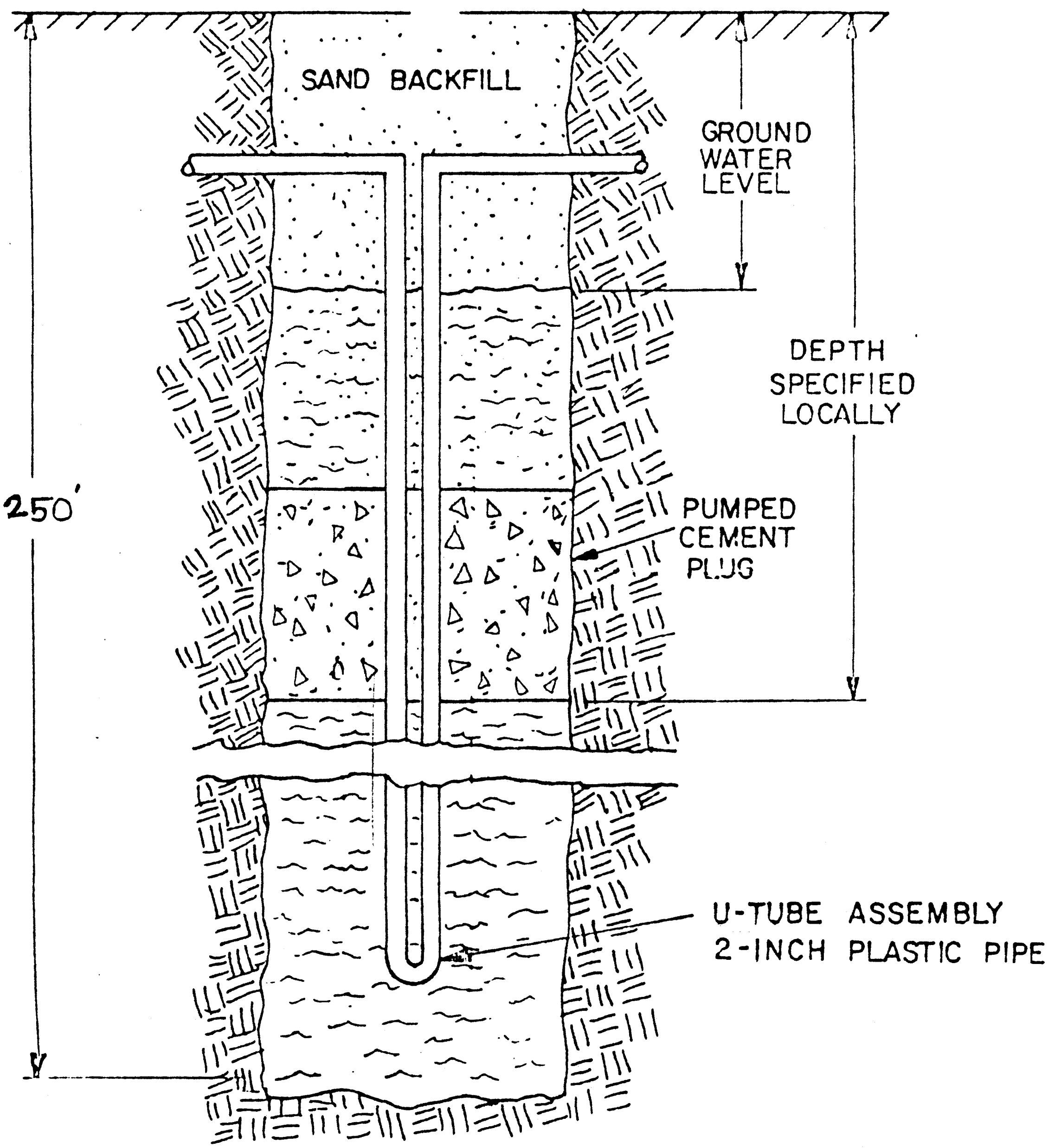

Figure 16. Schematic of U Tube

to the problem of vertical ground coil system with the step function heat input, but his results are not valid for cycle times less than ten hours; whereas in practice, the heat pump cycle time may be less than thirty minutes.

The numerical technique as given in (11) and (6) has been modified to simulate the new design of vertical earth water heat exchanger. The well is divided into a finite number of increments, as shown in Figure 17. The far field soil temperatures for different well increments were decided according to the suggestions of Collins (13). According to the Collins map, the annual average ground temperature is 62°F in Oklahoma for depths from 30 to 60 ft. with an increase of 1°F for every 64 ft. further increase in depth.

For each tube increment, an energy balance equation is written and heat exchange between the tube and soil as well as between two adjacent tube increments is computed. This computation leads to the determination of fluid temperature at the end of a tube increment. This process is continued until the fluid temperature at the end of the tube is determined.

Five heat transfer terms must be considered while writing an energy balance for a tube increment. These are the heat transfer by conduction between two adjacent tube increments, heat transfer by convection between fluid and well inner surface, enthalpy change in the fluid passing through this particular well increment, the heat stored in the fluid increment because of temperature difference in a definite time step, and the heat transfer by conduction to the ground.

The term for heat flow by conduction between two adjacent tube increments is computed as shown in Figure 18. The left-hand side and

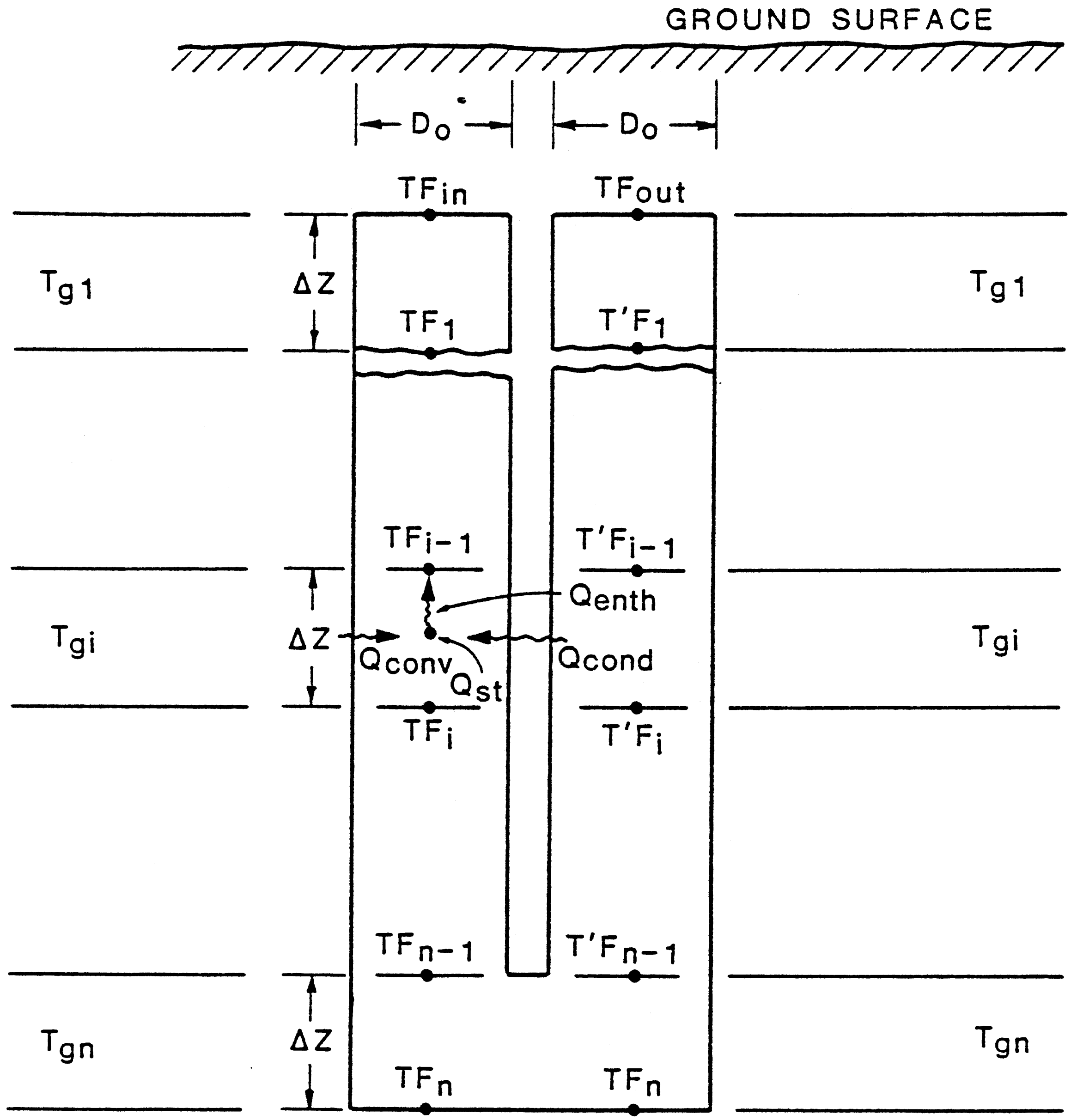

Figure 17. Schematic of "U" Tube for Computational Purposes

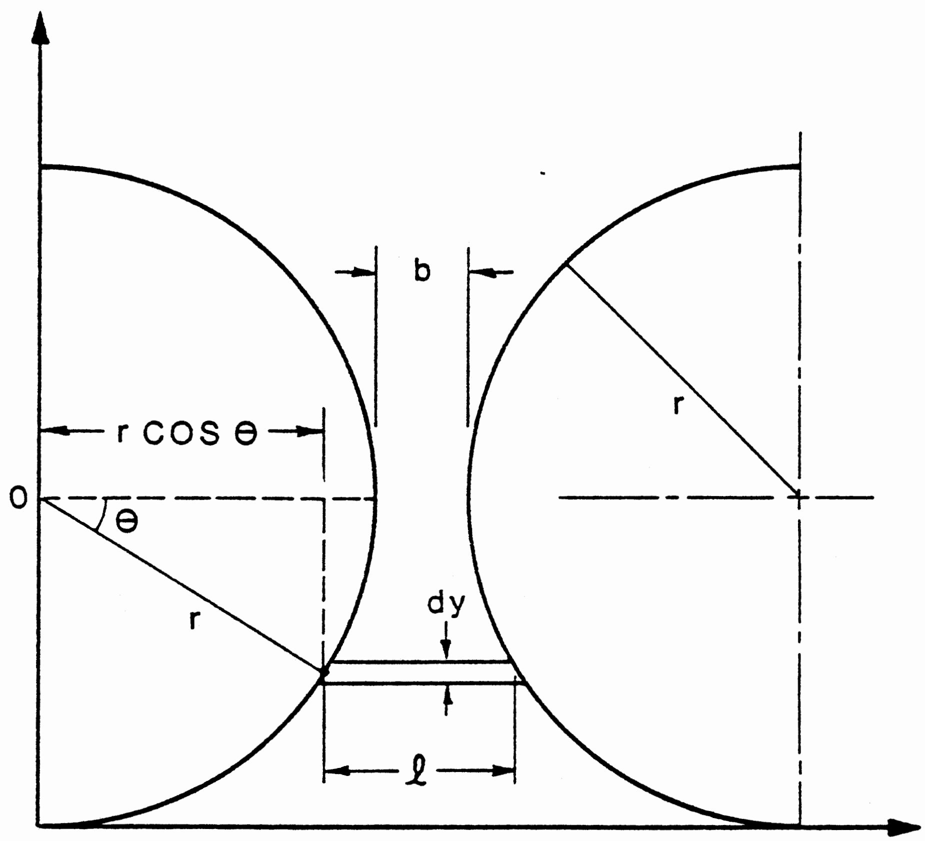

Figure 18. Section of 'U" Tube to Find the Conduction Heat Transfer Between Adjacent Tube Sections

right-hand side tube increments are assumed to be at uniform temperatures of T_{avg_i} and T'_{avg_i}, respectively.

The heat transfer rate through an element of length ℓ, width dy and height ΔZ is given by

$$dq = - KA \frac{dT}{dx} = - K(dy\Delta Z) \frac{\Delta T}{\ell}$$

Using cylindrical polar coordinates:

$$dy = r \cos d\Theta$$

$$\ell = r(1-\cos\Theta) + b$$

$$dq = \frac{- K(dy\ \Delta Z)\left(T_{avg_i} - T'_{avg_i}\right)}{r\ [1-(\cos\Theta)] + b}$$

$$Q_{cond} = 2 \int_0^{\Pi/2} \frac{- K(\gamma\cos\Theta)\ \Delta Z \left(T_{avg_i} - T'_{avg_i}\right) d\Theta}{(1 - \cos\Theta) + b}$$

$$Q_{cond} = -2 \times \Delta Z \times K\left(T_{avg_i} - T'_{avg_i}\right) \int_0^{\Pi/2} \frac{r \cos\Theta\ d\Theta}{r(1 - \cos\Theta) + b}$$

$$Q_{cond} = \frac{-K\left(T_{avg_i} - T'_{avg_i}\right)}{\cosh^{-1} \frac{b^2}{2r^2}} 2\Pi\Delta Z$$

$$Q_{cond} = - K\left(T_{avg_n} - T'_{avg_n}\right) \frac{2\Pi\Delta Z}{\cosh^{-1} \frac{b^2}{2r^2}}$$

The heat flow by convection is given by

$$Q_{conv} = h\ A_n \left(T_{avg_n} - T_{w_n}\right)$$

where

h = convection heat transfer coefficient $\left(\frac{BTU}{hr\text{-}ft^2\text{-}{}^oF}\right)$;

An = surface area of ΔZ_n increment (ft^2);

Tw_n = well surface temp for the n^{th} increment (^{o}F); and

T_{avg_n} = mean average of inlet and outlet temps

$= \left(Tf_n + Tf_{n+1}\right)/2 \quad (^{o}F)$.

The heat flow due to enthalpy change in the fluid passing through an increment is given

$$Q_{enth} = mCp\left(Tf_n - Tf_{n+1}\right)$$

where

mCp = fluid capacity rate$\left(\frac{BTU}{hr - F}\right)$

The heat stored in the fluid increment over a time step is

$$Q_{st} = \frac{MCp\left(T_{avg_n} - T'_{avg_n}\right)}{\Delta\Theta}$$

where

M = mass of fluid increment (lbm);

$Tavg'_n$ = mean average temp. in the fluid increment a time step later (F); and

$\Delta\Theta$ = time step (Hr)

Energy balance on a fluid increment can then be written as

$$Q_{st} = Q_{Enth} - Q_{conv} + Q_{cond}$$

Substituting all of the terms as calculated before in the above equation, we get

$$T_{f_{n+1}} = A\ Tavg_n + B\ Tw_n - C\ T_{fn} \quad C\ T_{fn} + D$$

where

$$A = \frac{2}{1 + \left[\frac{2m}{M} + \frac{hAn}{MCp}\right]} \Delta\Theta - \frac{2\Pi L \ K}{\cosh^{-1}\left[\frac{b^2}{2r^2}\right]}$$

$$B = \left[\frac{2h \ An \ \Delta\Theta}{MCp}\right]$$

$$C = -\left[\frac{2m}{M} + \frac{hAn}{MCp}\right]^{-1} \Delta\Theta$$

$$D = \frac{2\Pi L \quad K}{\cosh^{-1}\left[\frac{b^2}{2r^2}\right]} \bar{T}_{avg_n} .$$

The temperature distribution in the soil is governed by the one-dimensional heat conduction equation.

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{-} \frac{\partial T}{\partial r} = \frac{1}{\alpha} \frac{\partial T}{\partial \Theta}$$

With the initial condition

$$T(r,\Theta) = T_i \qquad \text{for all } r > r_o$$

and boundary conditions

$$T(a,\Theta) = T_i \qquad \text{for all } \Theta > \Theta_o$$

$$\left[\frac{\partial T}{\partial r}\right]_{r=r_o} = \frac{Q}{\pi D \Delta Z K} \qquad \text{for all } \Theta > \Theta_o$$

Finite difference techniques were used to solve for the radial temperatures in the ground. An implicit method with nonuniform grid, as given by Kahchanalai (11) was used to solve the problem.

The iterative loop to compute the temperatures at various tube increments can be summarized as follows:

1. Guess Q, the heat conducted to the ground and the temperatures at all the nodes in the R.H.S. tube. TF_{in}, the temperature of fluid entering the tube is known. So start with TF_{in} and calculate the

temperatures at the end of successive tube increments. Finally, TF_n the temperature at the end of the tube is computed.

2. Temperatures at all the nodes in the L.H.S. tube are now known. Start with TF_n and follow similar steps to get the temperature of fluid going out of the well.

3. Compare the temperatures at all nodes in the R.H.S. tube with the assumed ones. If they don't agree within certain limits, then change the initial guess Q and repeat those steps until convergence is established.

The method described above is expensive so far as computer time is concerned (since this subroutine is called twice in every fifteen minutes of time interval). Experimental observations show that the temperature differences between the water going in and coming out from a 250 ft. deep well is approximately 4 to 5^{o}F. So a simplifying assumption was made. That is, the average temperature of any two adjacent tube increments throughout the length of the well is constant.

The computational strategy can be summarized as follows:

1. The temperature of fluid going to the well TF_{in} is known. Assume TF_{out} the temperature of fluid coming out of "U" tube.

2. Calculate the equivalent diameter of a single pipe. Since the pipe diameters are small, the cross-sectional areas were just added to get cross-sectional area and hence the diameter of a single pipe.

3. Initially, this pipe was at a uniform temperature of TF_{in}. Net heat exchange between the fluid and the ground is

$$Q = mC_p \ (TF_{out} - TF_{in})$$

where

mC_p = well fluid capacity rate$\left(\frac{BTU}{hr-^{o}F}\right)$.

Now for this heat flow rate, the temperature of the whole pipe is calculated after a definite time step (heat pump on time, in this case).

Compare this temperature with the average temperature of pipe

$$T_{avg} = \frac{TF_{in} + TF_{out}}{2}$$

Repeat these steps until convergence is established.

4. Calculate

$$TF_{out} = 2 * T_{avg} - TF_{in}$$

Model Validation

Figure 19 shows the plot of measured and simulated well outlet temperatures for May 26, 1982 for the West house. Actual recorded data stored on the magnetic discs have been used. Well outlet temperature is an important parameter because it determines the compressor KWH consumption. The simulated values show a close agreement with the measured ones.

Figure 20 shows a plot of the temperature difference of water going to and coming out of the well for May 25, 1982. This particular plot and the simulation results for few other days in the summer indicate that the temperature difference in water as predicted by the computer simulation is on an average five percent higher than the observed values. This is expected since the cross-sectional area of a single pipe is computed by just adding the cross-sectional areas of individual pipes. More experimental data for summer as well as winter operations is required for comparison with simulated results so that an empirical relation to calculate the equivalent diameter of a single pipe can be established.

The simulation program for the vertical earth water heat exchanger

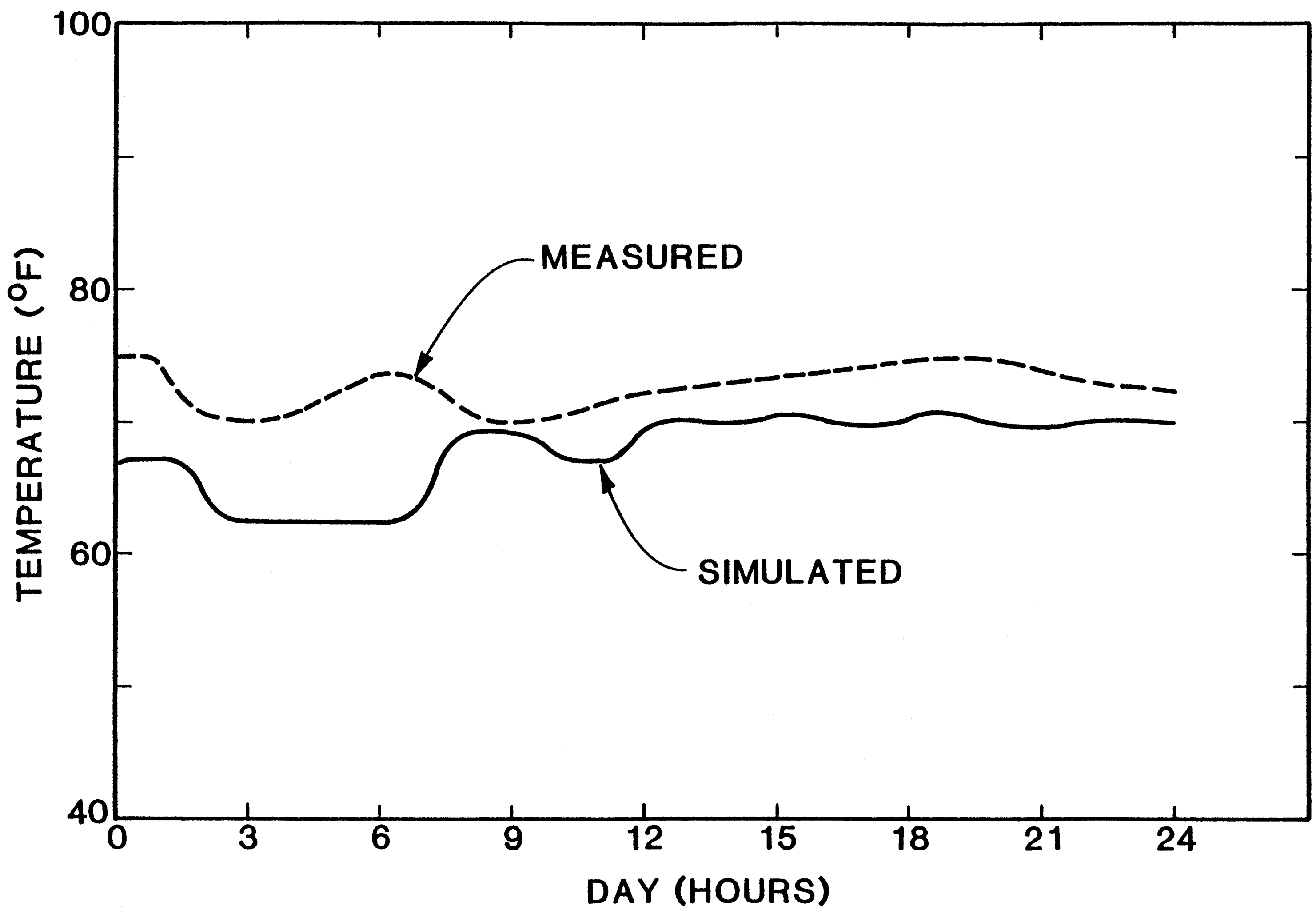

Figure 19. Plot of Measured and Simulated Well Water Temperatures for May 26, 1982

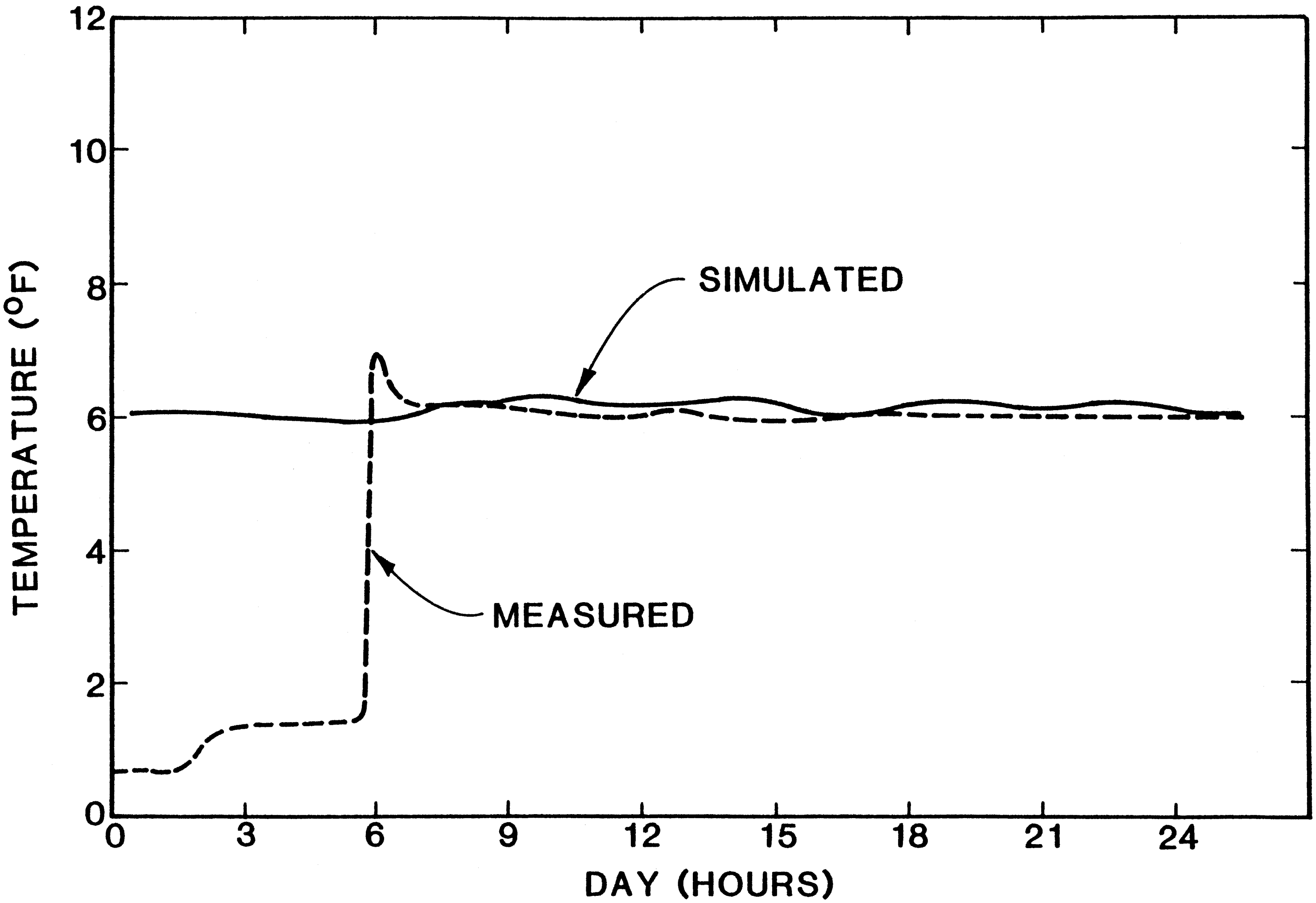

Figure 20. Plot of the Temperature Difference of Water Going to and Coming out of the Well for May 25, 1982

can be used to determine the effects of varying different design parameters on the performance of the heat pump system. The program can also be used to determine the various design parameters concerning the geometry of the well. An analytical approach is given here to determine the well depth for the ground/source sink heat pump system installed in the West house.

Figure 21 shows a plot of the well water temperatures after 3000 hours of heat pump operation during summer for various well depths. The following assumptions have been made:

1. Heating or cooling season consists of approximately 3000 hours.

2. The heat pump cycle time is 30 minutes. The heat pump operates at its full capacity of 19,600 BTU/hr for half of the cycle time and is off for the rest of the cycle time.

Actual well radius of 2.5 in. and the average values of thermal conductivity and thermal diffusivity are taken.

Figure 22 shows a similar plot for the same heat pump system during winter operations. It can be concluded from the graphs that a well depth of 200 ft. is required to prevent freezing ($32^{o}F$) of water in winter operations under the assumed conditions.

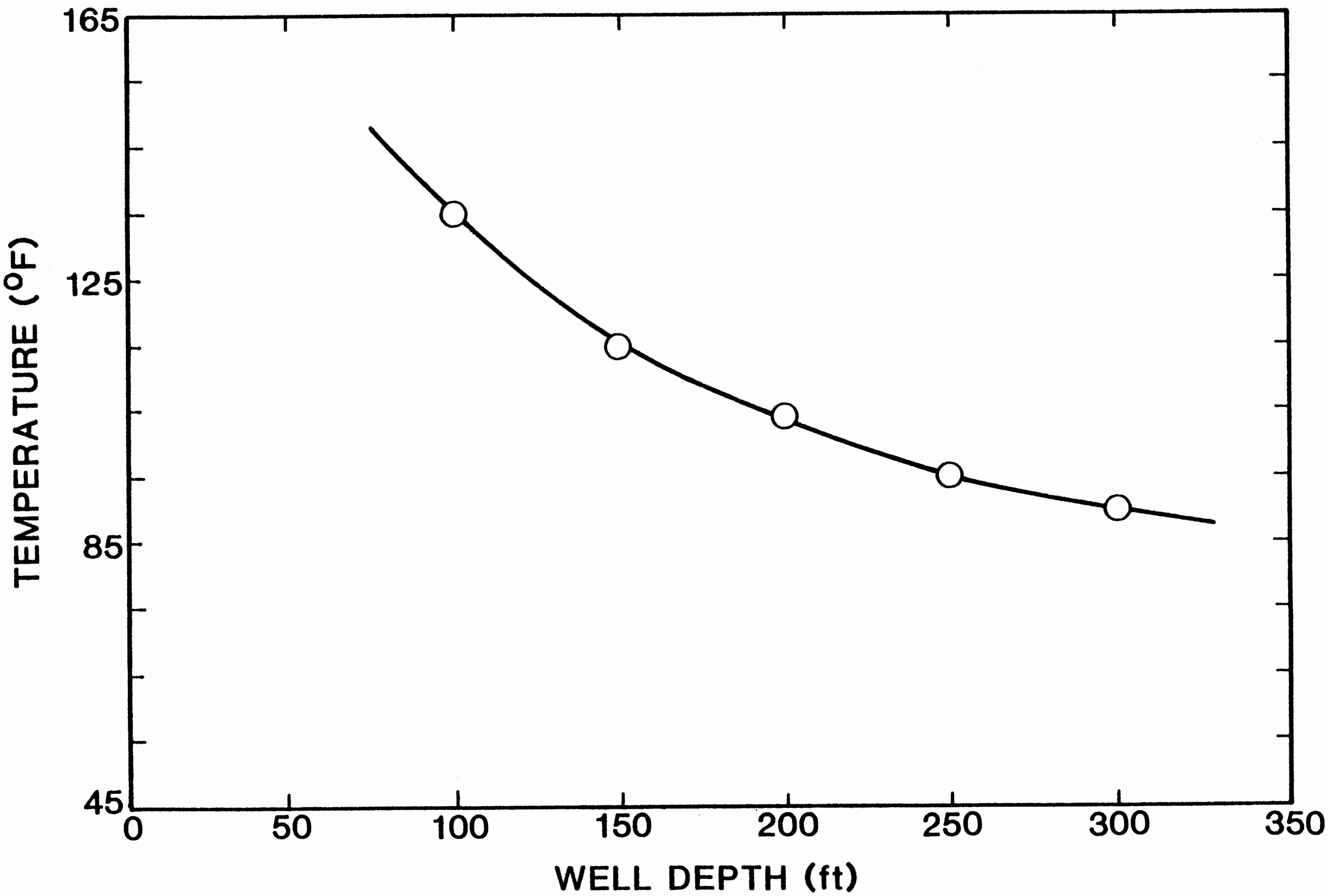

Figure 21. Plot of the Well Water Temperatures After 3000 Hours of Summer Operation

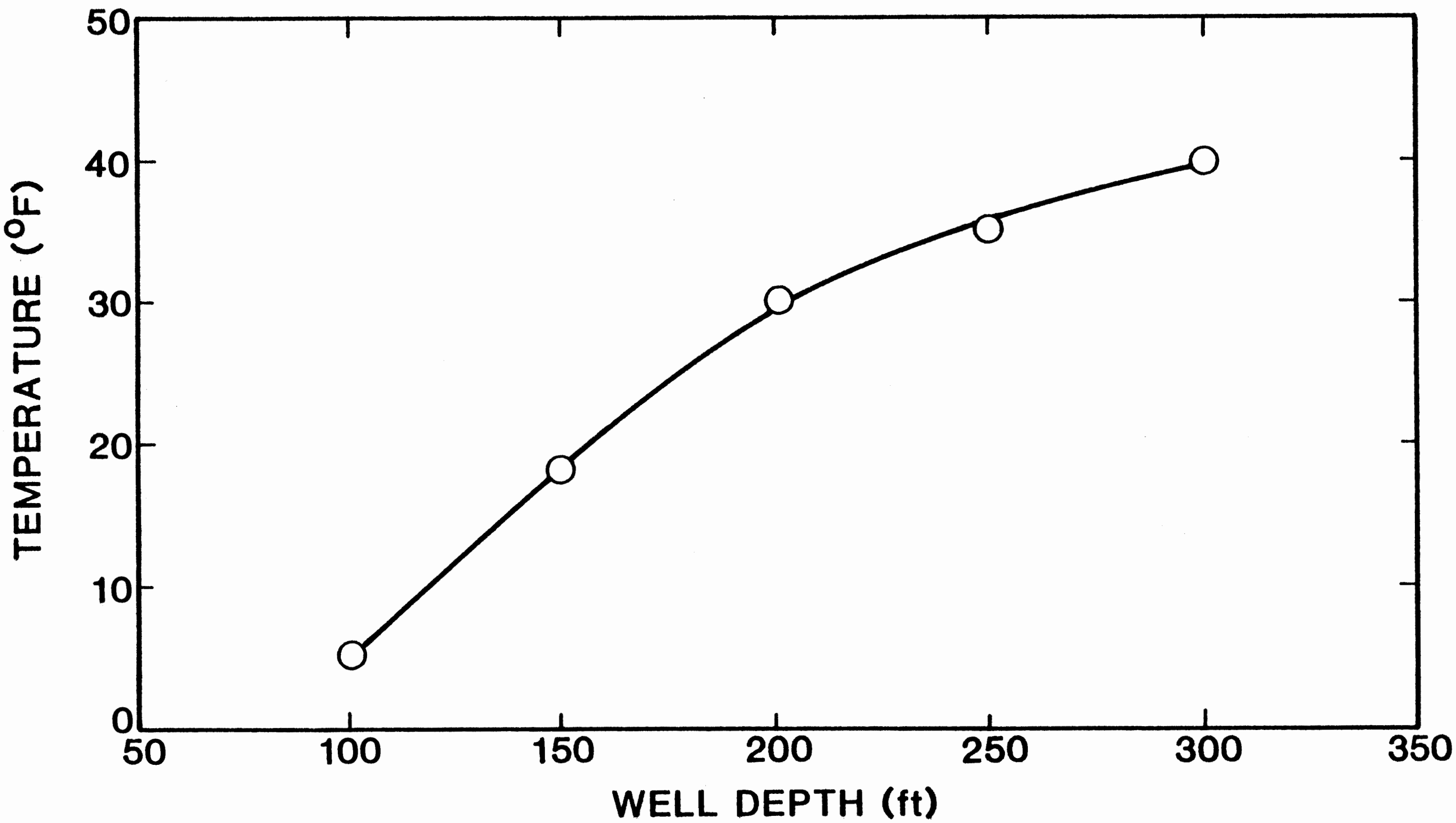

Figure 22. Plot of the Well Water Temperatures After 3000 Hours of Winter Operation

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

One of the main objectives of this study was the development and application of a computer subprogram to simulate a storage tank to be included in the solar-assisted ground source heat pump system. The simulation results show that the storage tank helps greatly in preventing the freezing of water in the well water loop, and prevents the electric resistance backup heat from coming on, on a very cold day. Results given in Table II indicate the significant contribution made by the storage tank in reducing the peak demand on the system in winter.

Simulation results also indicate that the storage tank is more helpful in reducing peak demand in winter than in summer.

Experimental and economic analyses are required to support the circuit modifications suggested in Figures 13 and 14.

Finite difference approximation techniques for the modelling of vertical earth water heat exchangers are demonstrated.

The new well was installed in March, 1982 based on a different design. This new design is cheaper, simpler, easy to install and maintain.

As of this date, a few days of edited data for the West house and middle house are available to run the simulation program. A lot of data are necessary to make a comparison of simulated and actually recorded well water temperatures so that an empirical relation to determine the equivalent diameter of a single pipe as described in Chapter V can be established.

It is suggested that other horizontal and vertical configurations of earth water heat exchangers should be studied to come up with the cheapest and most efficient configuration.

Use of antifreeze solution in the well water loop can be beneficial, however, necessary care should be taken to avoid any cross connections with the domestic water supply and consequent contamination.

The simulation results about heat dissipation by the solar collectors, when the heat pump is in the cooling mode, are not very encouraging. So it may be a better idea to use glass-covered collectors which will increase the heat gain in winter significantly. However, the present location of the solar collectors will have to be changed so that the collectors will not be damaged by children who might be playing in the backyard.

Economic study should be performed to optimize the various system components.

Multiple wells of smaller depth have shown no significant advantages over a single well of the same total depth in energy saving and increasing the capacity of the heat pump system. The decision depends on

1. drilling cost

2. drilling problems

if, for instance, there should be rocks after a certain depth in the soil.

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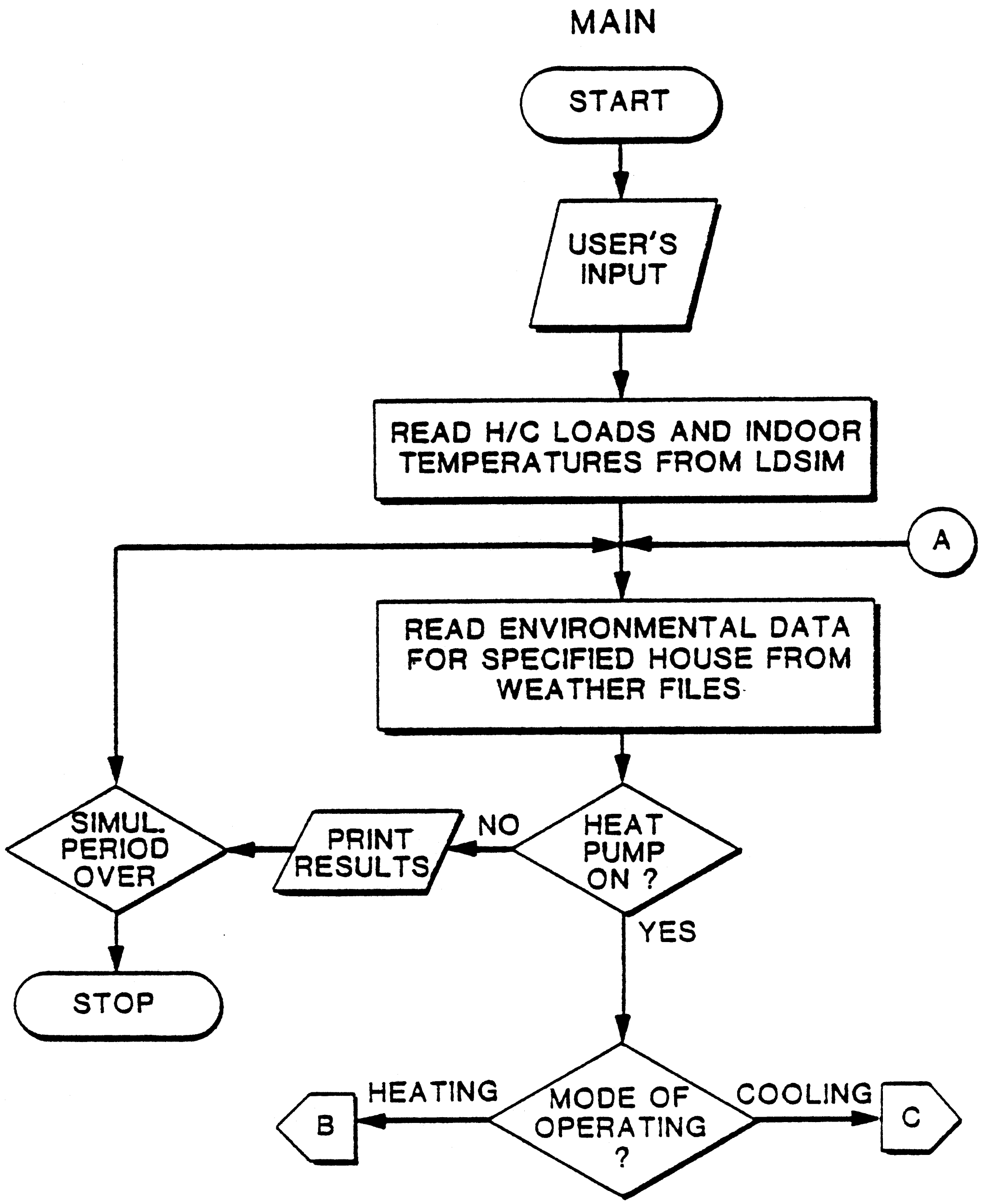
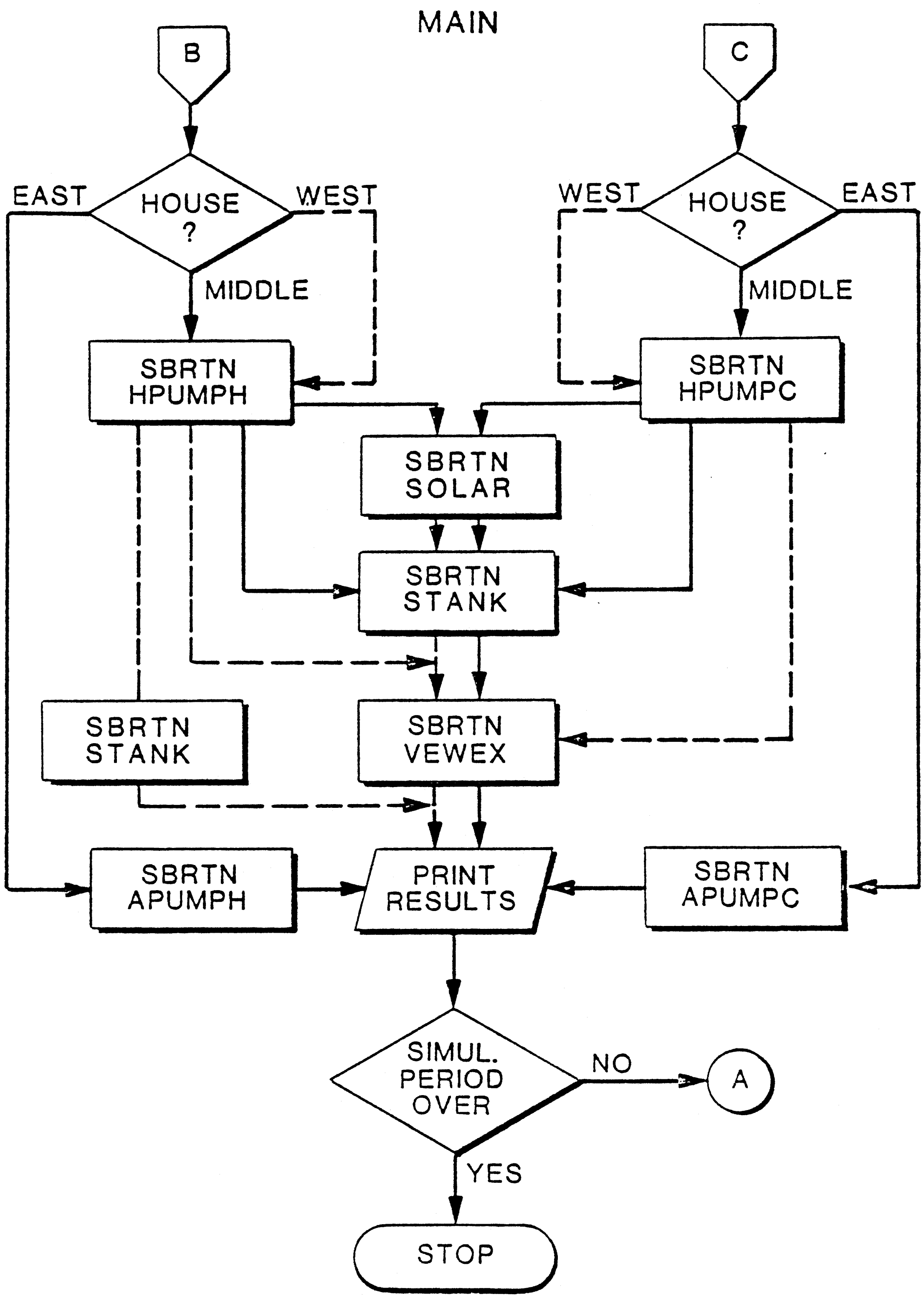
APPENDIX

FLOW CHARTS AND COMPUTER PROGRAM LISTINGS

Flow Charts

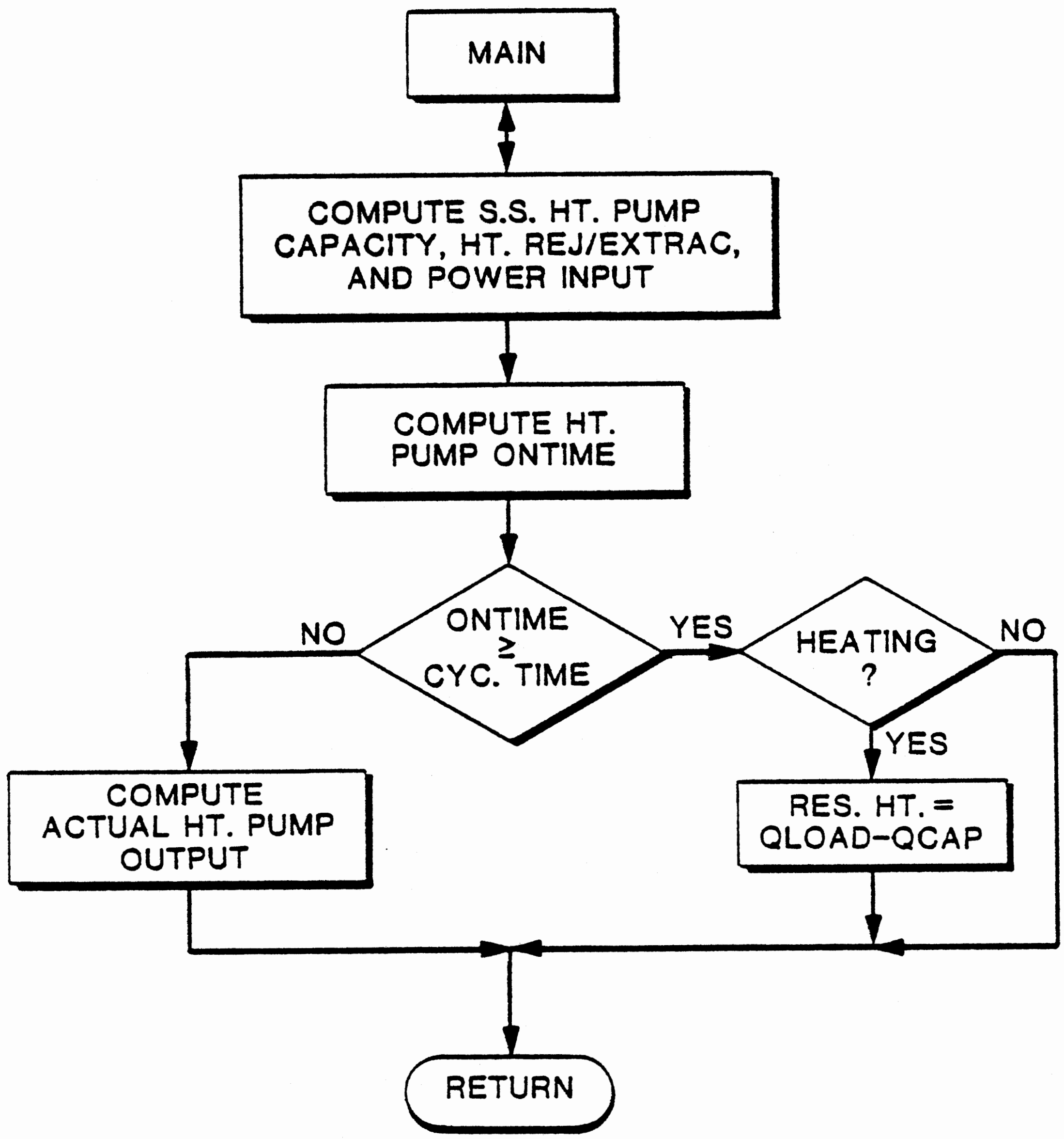
Main program with storage tank included; storage tank is modeled in a subroutine STANK. The well based on a new design is modeled in a subroutine TUBE.

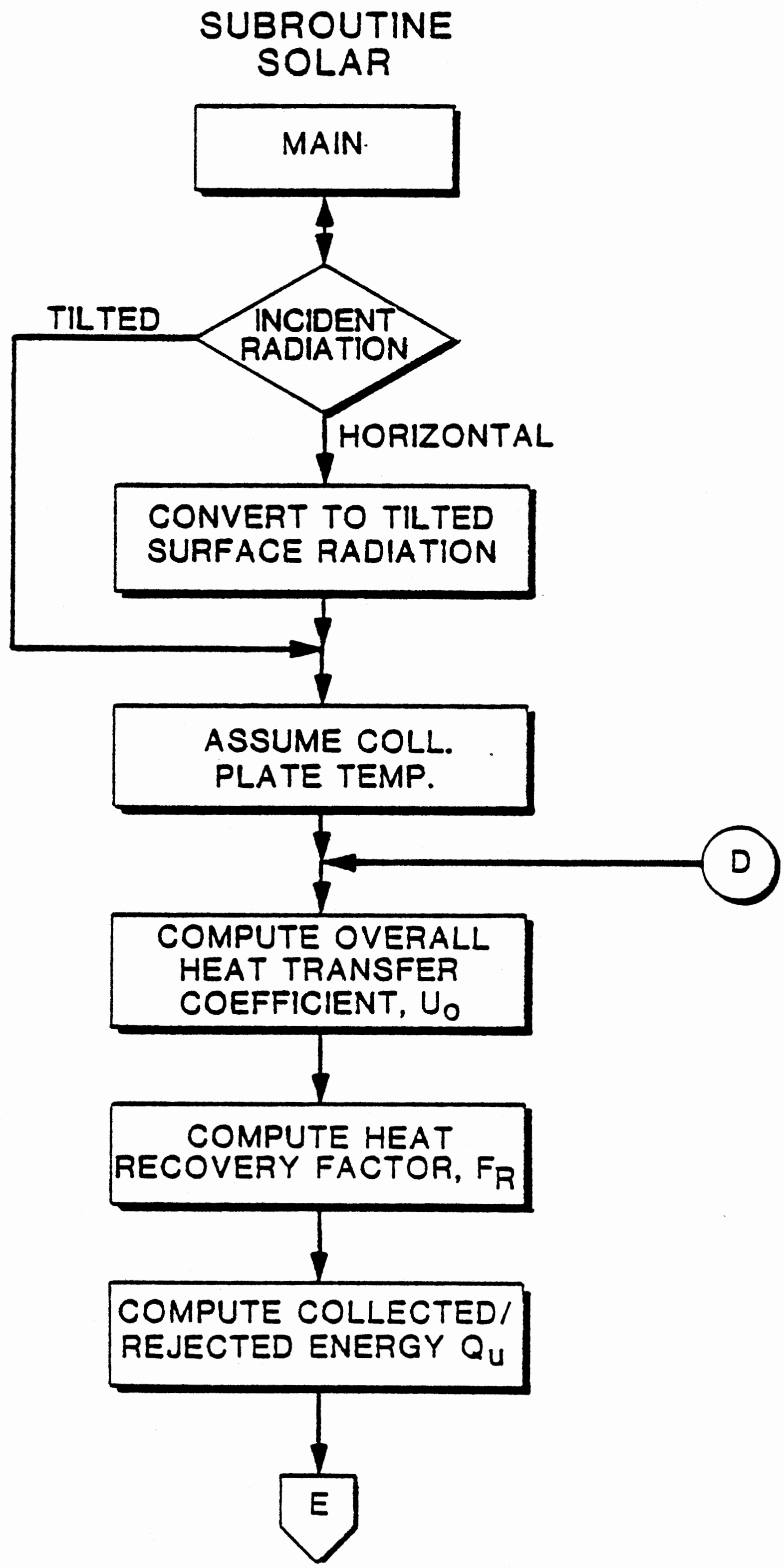

MAIN
START
USER'S INPUT
READ H/C LOADS AND INDOOR TEMPERATURES FROM LDSIM
A
READ ENVIRONMENTAL DATA FOR SPECIFIED HOUSE FROM WEATHER FILES
SIMUL. PERIOD OVER
PRINT RESULTS
NO
HEAT PUMP ON ?
YES
STOP
HEATING
B
MODE OF OPERATING ?
COOLING
C


MAIN
B
C
HOUSE ?
EAST
WEST
MIDDLE
HOUSE ?
WEST
EAST
MIDDLE
SBRTN HPUMPH
SBRTN HPUMPC
SBRTN SOLAR
SBRTN STANK
SBRTN STANK
SBRTN VEWEX
SBRTN APUMPH
PRINT RESULTS
SBRTN APUMPC
SIMUL. PERIOD OVER
NO
A
YES
STOP

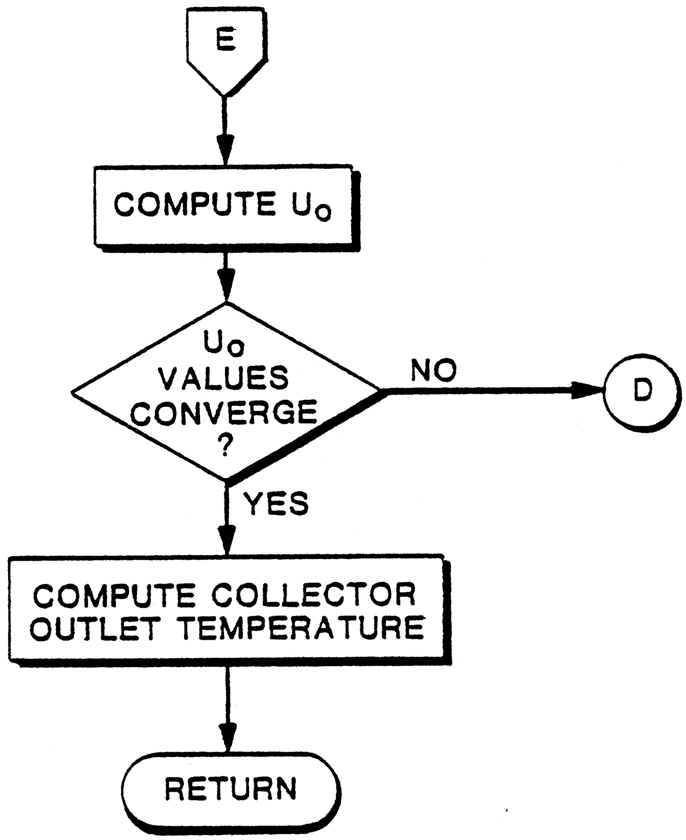
HEAT PUMP CAPACITY SUBROUTINES

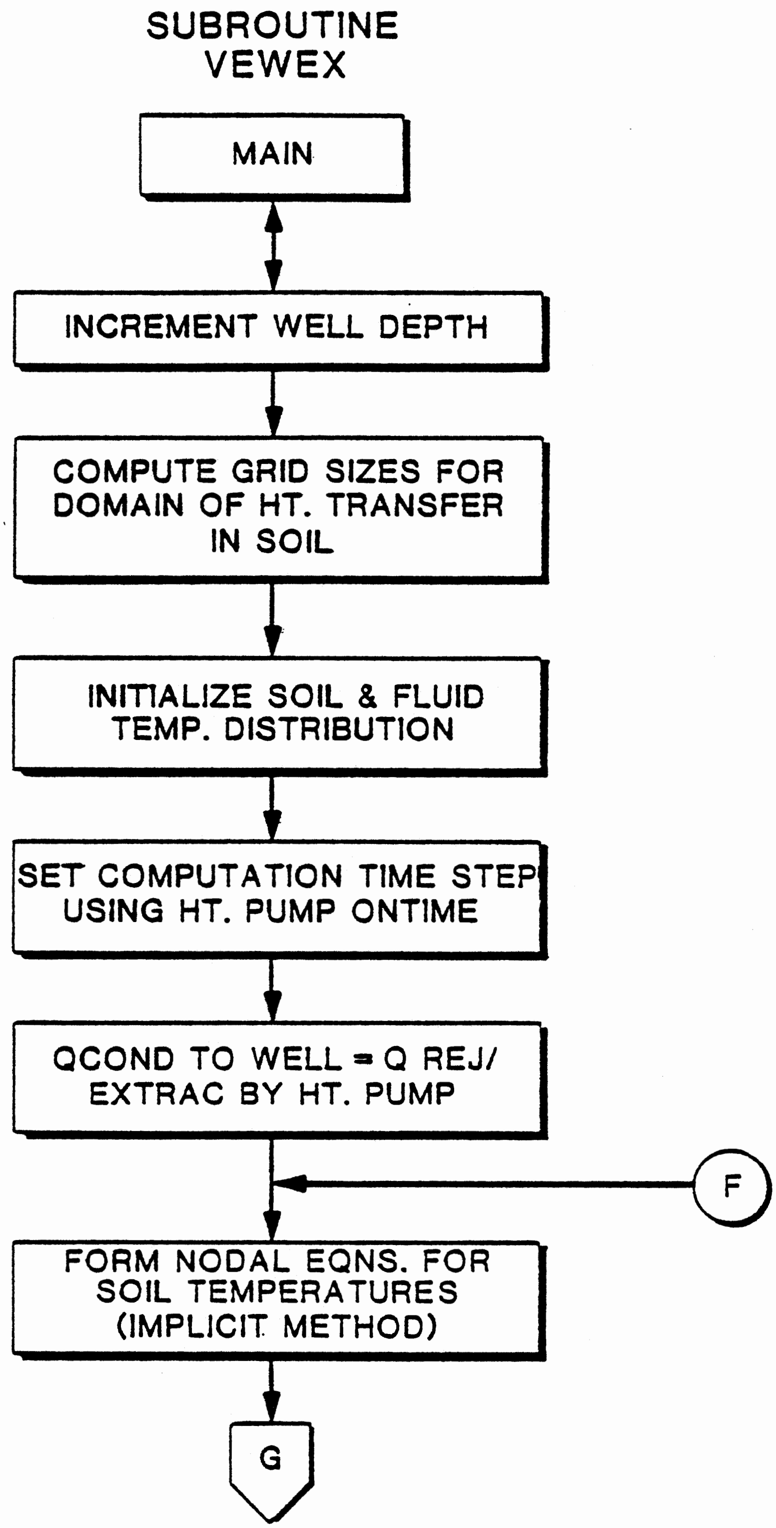
HPUMPC HPUMPH
APUMPC APUMPH

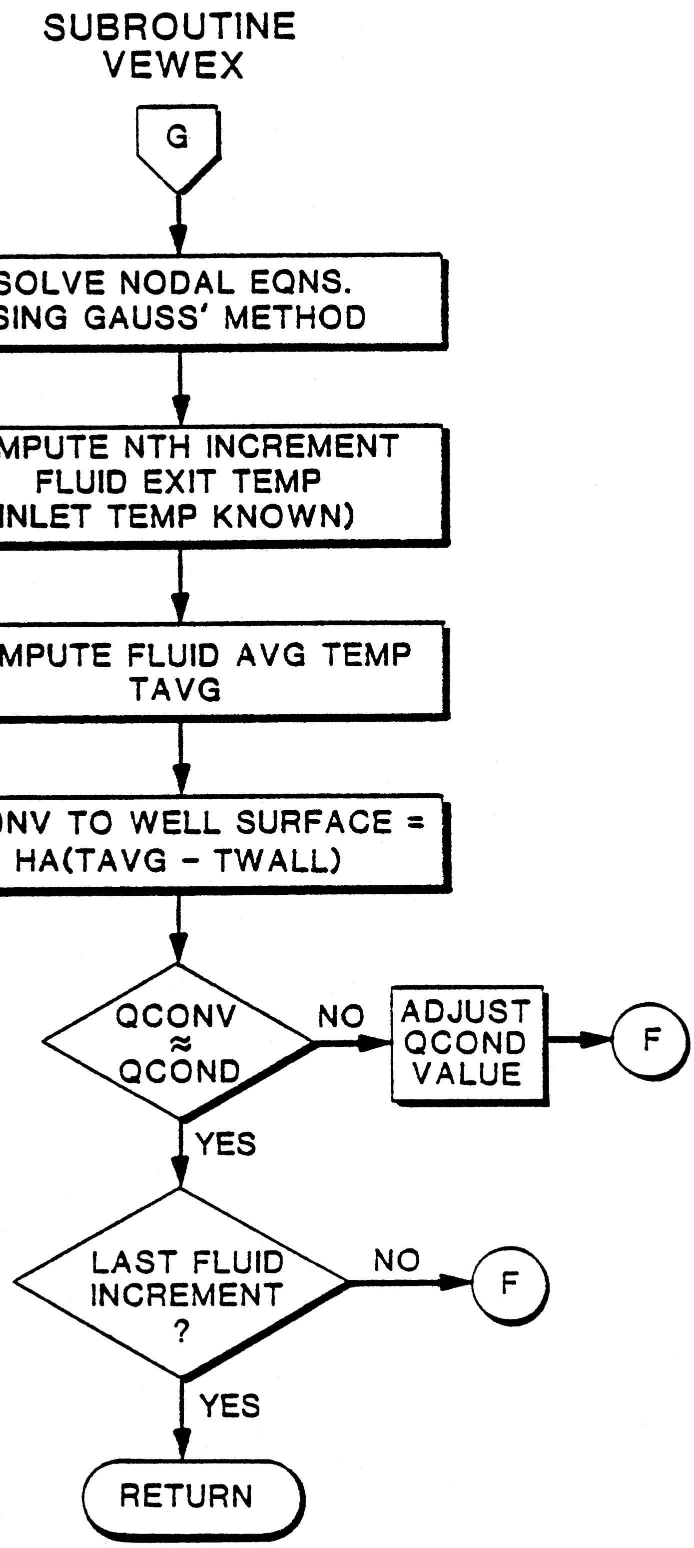


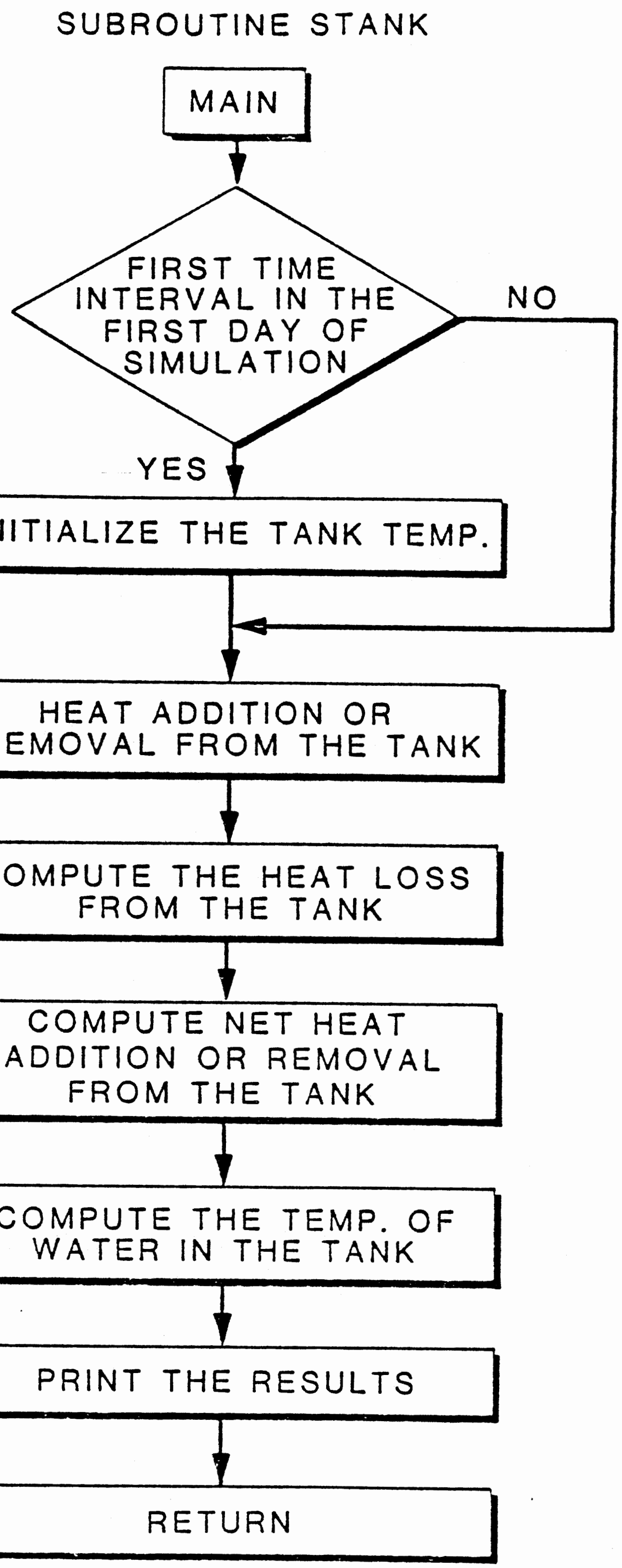

SUBROUTINE
SOLAR
MAIN
INCIDENT RADIATION
TILTED
HORIZONTAL
CONVERT TO TILTED SURFACE RADIATION
ASSUME COLL. PLATE TEMP.
D
COMPUTE OVERALL HEAT TRANSFER COEFFICIENT, U_o
COMPUTE HEAT RECOVERY FACTOR, F_R
COMPUTE COLLECTED/ REJECTED ENERGY Q_u
E

SUBROUTINE
SOLAR

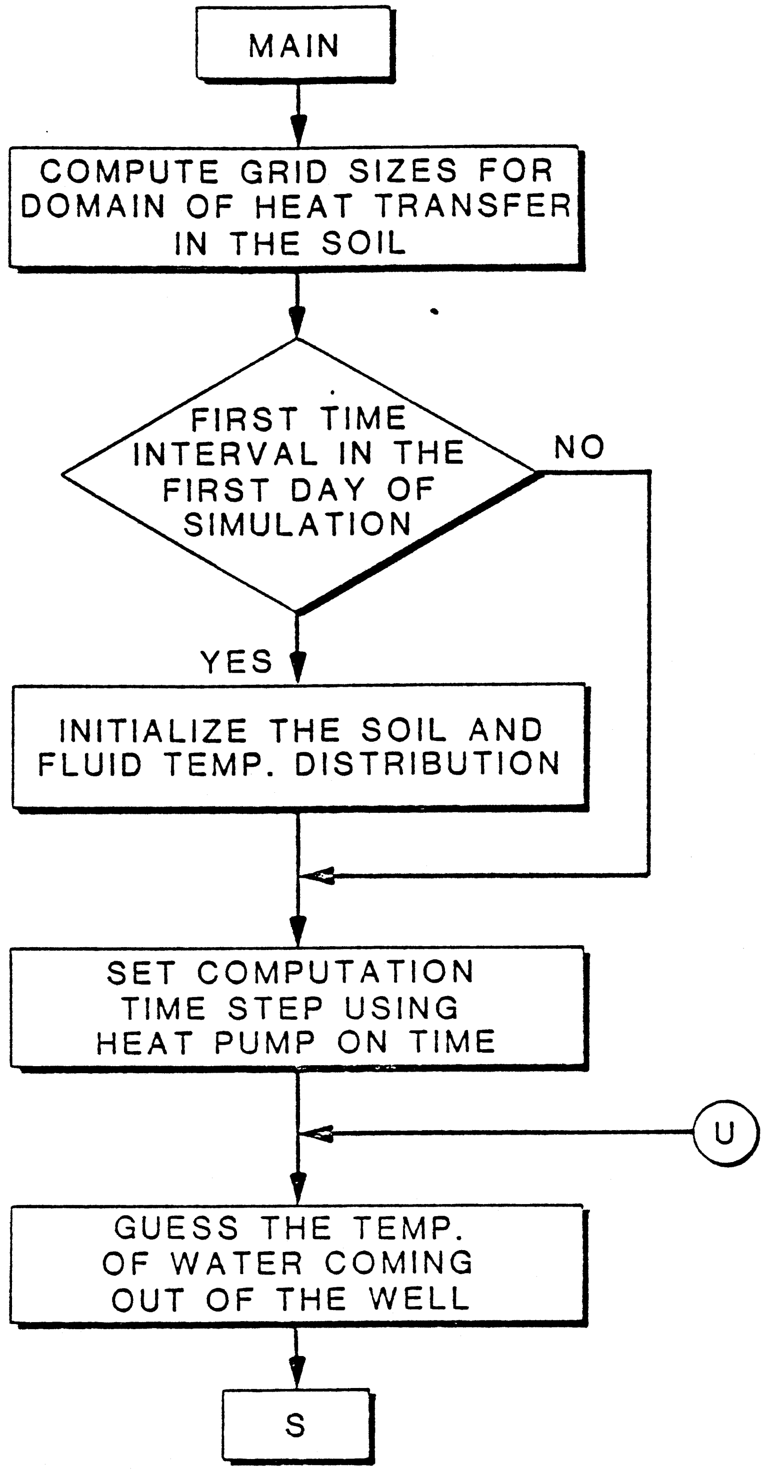



SUBROUTINE
VEWEX
MAIN
INCREMENT WELL DEPTH
COMPUTE GRID SIZES FOR
DOMAIN OF HT. TRANSFER
IN SOIL
INITIALIZE SOIL & FLUID
TEMP. DISTRIBUTION
SET COMPUTATION TIME STEP
USING HT. PUMP ONTIME
QCOND TO WELL = Q REJ/
EXTRAC BY HT. PUMP
F
FORM NODAL EQNS. FOR
SOIL TEMPERATURES
(IMPLICIT METHOD)
G

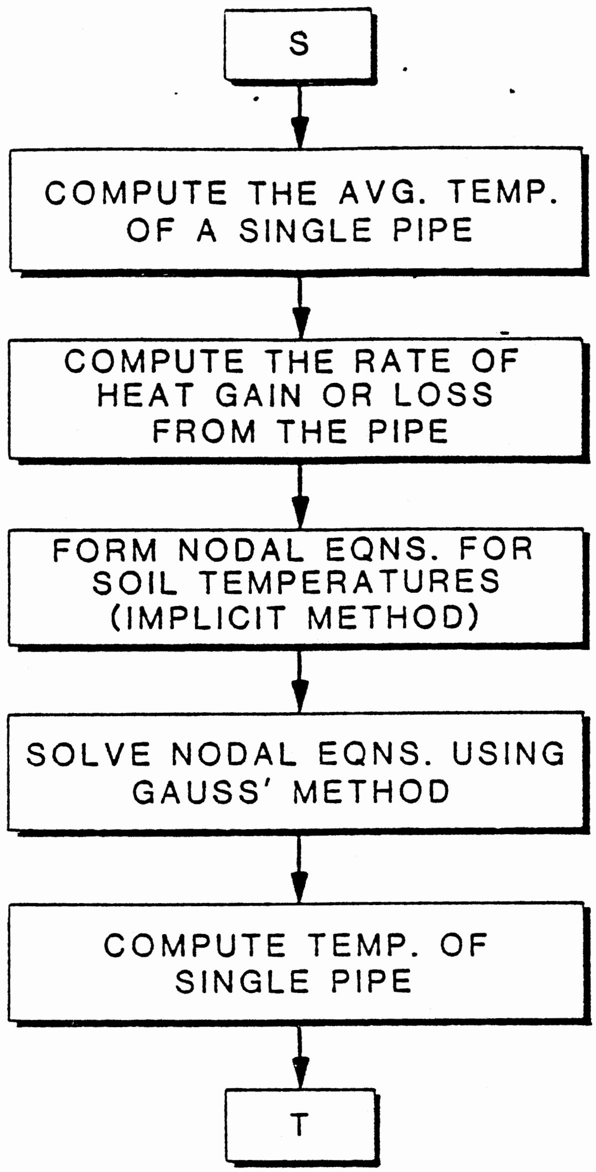

SUBROUTINE VEWEX
G
SOLVE NODAL EQNS. USING GAUSS' METHOD
COMPUTE NTH INCREMENT FLUID EXIT TEMP (INLET TEMP KNOWN)
COMPUTE FLUID AVG TEMP TAVG
QCONV TO WELL SURFACE = HA(TAVG - TWALL)
QCONV ≈ QCOND
NO
ADJUST QCOND VALUE
F
YES
LAST FLUID INCREMENT ?
NO
F
YES
RETURN


SUBROUTINE STANK
MAIN
FIRST TIME INTERVAL IN THE FIRST DAY OF SIMULATION
NO
YES
INITIALIZE THE TANK TEMP.
HEAT ADDITION OR REMOVAL FROM THE TANK
COMPUTE THE HEAT LOSS FROM THE TANK
COMPUTE NET HEAT ADDITION OR REMOVAL FROM THE TANK
COMPUTE THE TEMP. OF WATER IN THE TANK
PRINT THE RESULTS
RETURN

SUBROUTINE TUBE



SUBROUTINE TUBE



```
//U11864A  JOB (11864,000 00 0000),'SOL',TIME=(0,40),CLASS=A,MSGCLASS=X
/*PASSWORD ????
//   EXEC FORTGCG,PARM.FORT='NOSOURCE,TERM'
//FORT.SYSIN DD *
C
C
C
C    ********************************************************************
C    ********************************************************************
C    *                                                                  *
C    *                                                                  *
C    *                       ( H P S I M )                              *
C    *                                                                  *
C    *                                                                  *
C    *      THIS PROGRAM SIMULATES THE CYCLIC PERFORMANCE OF A SOLAR    *
C    *      ASSISTED-GROUND SOURCE/SINK AND AN AIR-TO-AIR HEAT PUMP     *
C    *      SYSTEMS.                                                    *
C    *      ENVIRONMENTAL DATA ARE READ FROM MONTHLY WEATHER FILES      *
C    *      COMPILED FOR THE PERKINS HOUSES.                            *
C    *                                                                  *
C    *                                                                  *
C    ********************************************************************
C    ********************************************************************
C
C
      COMMON FLOW
      COMMON /BLOCK/ GGPM,KPRINT
      DIMENSION QLOAD(24),QSLOP(24),QFORT(24)
      DIMENSION TROOM(24),TSLOP(24)
      NAMELIST/INPUT/MONTH,MDAY1,MDAY2,INDEX,KPRINT,IHOUSE,IDP
      DATA CPF,CPG,EXE,ALPH/1.0,0.84,0.63,0.95/,KTIM1/1/
      DATA HS1,HS2,HS22,HS3/'EAST','MIDD','LE','WEST'/
    1 FORMAT('1')
    2 FORMAT(/////,49X,'HEAT PUMP PERFORMANCE CALCULATION',/,57X,
     &'FOR THE ',A4,' HOUSE',/,50X,'FOR DAY',I5,'  OF MONTH',I5,/,
     &49X,34('*'),////)
    3 FORMAT(/////,49X,'HEAT PUMP PERFORMANCE CALCULATIONS',/,56X,
     &'FOR THE ',A4,A2,' HOUSE',/,51X,'FOR DAY',I5,'  OF MONTH',I5,/,
     &49X,34('*'),////)
    4 FORMAT(' ',10X,'TIME',5X,'OUTDOOR',5X,'INDOOR',7X,'HOUSE',7X,
     &'HT PUMP',6X,'HT PUMP',7X,'RES.',5X,'COMP. &',5X,'HEAT',/,19X,
     &'DB TEMP',5X,'DB TEMP',7X,'LOAD',8X,'ON TIME',6X,'CAPACITY',6X,
     &'HEAT',6X,'FANS',7X,'PUMP',/,10X,'(HR)',7X,'(F)',9X,'(F)',9X,
     &'(BTU)',8X,'(HR)',9X,'(BTU)',8X,'(KWH)',6X,'(KWH)',6X,'COP',/)
    5 FORMAT(5X,'TIME',3X,'OUT DB',4X,'IN DB',6X,'HOUSE',5X,'HT PUMP',4
     &X,'HT PUMP',7X,'RES.',4X,'COMP &',5X,'HEAT',5X,'SOLAR',7X,'GR COIL
     &',4X,'GR COIL',
     &/,13X,'TEMP',6X,'TEMP',6X,'LOAD',6X,'ON TIME',4X,'CAPACITY',4X,'
     &HEAT',5X,'FAN',7X,'PUMP',5X,'LOOP Q',6X,'IN TEMP',4X,'OUT TEMP',/,
     &5X,'(HR)',5X,'(F)',6X,'(F)',7X,'(BTU)',6X,'(HR)',7X,'(BTU)',8X,'(K
     &WH)',3X,'(KWH)',6X,'COP',5X,'(BTU/HR)',8X,'(F)',8X,'(F)',/)
   10 FORMAT(5X,I2,2(6X,I2),3(12X,I4),4X,I4,/,11X,I4,/////)
   11 FORMAT(/,41X,'HEAT PUMP AVERAGE COP =',14X,F10.2)
   12 FORMAT(////,38X,'TOTAL HEAT PUMP OPERATING TIME =',11X,F10.2,//,
     138X,'TOTAL COMPRESSOR & FAN(S) KWH CONSUMPTION =',F10.2,//,38X,
     3 'HEAT PUMP AVERAGE COEFF. OF PERFORMANCE =',2X,F10.2)
   15 FORMAT(4F20.0)
   20 FORMAT(5X,I2,6X,I2,6X,I2,3X,I5,28X,I4,/,3X,I4,
     &      16X,6(4X,I4),/,43X,I4,4X,I4,////)
  601 FORMAT('0',3X,F5.2,3X,F5.1,5X,F5.1,3X,E12.5,3X,F5.3,3X,E12.5,3X,
     &F5.2,3X,F5.2,5X,F5.3,2X,E12.5,2X,F5.1,6X,F5.1)
  602 FORMAT('0',10X,F5.2,5X,F5.1,7X,F5.1,4X,E12.5,5X,F5.3,4X,E12.5,
     &5X,F5.2,5X,F5.2,5X,F5.2)
  607 FORMAT(////,2(10X,72('*')/),10X,5('*'),62X,5('*'),/,10X,5('*'),
```

```
     &5X,'DATA FOR THE SPECIFIED SIMULATION PERIOD ARE MISSING',5X,5('*
     &'),/,10X,5('*'),62X,5('*'),/,2(10X,72('*')/))
  701 FORMAT(/////,2(10X,70('*')/),10X,5('*'),60X,5('*'),/,10X,5('*'),
     &5X,'OF THE SIMULATION PERIOD SPECIFIED.',20X,5('*'),/,10X,5('*'),
     &5X,I5,2X,'(15-MINUTE) DATA INTERVALS',22X,5('*'),/,10X,5('*'),5X,
     &'ARE MISSING FOR DAY',I5,2X,'OF MONTH',I5,15X,5('*'),/,10X,5('*')
     &,60X,5('*'),/,10X,5('*'),5X,'CHOOSE ANOTHER SIMULATION PERIOD',23
     &X,5('*'),/,10X,5('*'),60X,5('*'),/,2(10X,70('*')/))
  702 FORMAT('1')
  703 FORMAT(////,41X,'COMPRESSOR OPERATING TIME =',10X,F10.2,//,41X,
     1'COMPRESSOR & FAN(S) KWH CONSUMPTION =',F10.2)
  704 FORMAT(/,41X,'RESISTANCE HEAT (KWH) =',14X,F10.2,//,41X,
     &'HEAT PUMP AVERAGE COP =',14X,F10.2)
  707 FORMAT(////,38X,'TOTAL HEAT PUMP OPERATING TIME =',11X,F10.2,//,
     138X,'TOTAL COMPRESSOR & FAN(S) KWH CONSUMPTICN =',F10.2,//,38X,
     2'TOTAL RESISTANCE HEAT =',20X,F10.2,//,38X,
     3'HEAT PUMP AVERAGE COEFF. OF PERFORMANCE =',2X,F10.2)
  708 FORMAT(////,45X,39('*'),/,45X,'*',37X,'*',/,45X,'*',4X,
     1'TOTALS FOR DAY ',I2,' OF MONTH ',I2,4X,'*',/,45X,'*',37X,'*',/,
     245X,39('*'))
  709 FORMAT('1',15(/),34X,63('*'),/,34X,'*',61X,'*',/,34X,'*',4X,
     1'PERFORMANCE SUMMARY FOR THE ',I3,'-DAY SIMULATION PERIOD',4X,'*',
     2/,34X,'*',61X,'*',/,34X,63('*'))
      GGPM=5.5
      FLOW=12.0
      AMCW=497.3*CPF*GGPM
      AMCG=518.1*CPG*FLOW
      TMID=62.0
      AVGCOP=0.0
      SUM5=0.0
      SUM6=0.0
      SUM7=0.0
      SUM8=0.0
      KMODE=0.0
C
C          INDEX FOR THE HOUSE TO BE SIMULATED
C
C          IHOUSE=1          ( EAST HOUSE )
C          IHOUSE=2          ( MIDDLE HOUSE )
C          IHOUSE=3          ( WEST HOUSE )
C
C
      READ(5,INPUT)
      IF(MDAY1.EQ.24)GO TO 600
      KDAY=MDAY1-1
      DO 500 I=1,18000
      READ(1,10) MDAY,MHR,MIN,KTE,KTM,KTWW,KTW,KTWM
      MDAY=MDAY-20
      IF(MDAY.GT.MDAY1) GO TO 606
      IF(MDAY.EQ.KDAY.AND.MHR.EQ.23.AND.MIN.EQ.45) GO TO 600
  500 CONTINUE
  600 DO 3000 J=MDAY1,MDAY2
      NKOUNT=0
      SUM1=0.0
      SUM2=0.0
      SUM3=0.0
      SUM4=0.0
      WRITE(6,1)
      IF(IHOUSE.EQ.1) WRITE(6,2) HS1,J,MONTH
      IF(IHOUSE.EQ.2) WRITE(6,3) HS2,HS22,J,MONTH
      IF(IHOUSE.EQ.3) WRITE(6,2) HS3,J,MONTH
      IF(IHOUSE.EQ.1) WRITE(6,4)
      IF(IHOUSE.EQ.2.OR.IHOUSE.EQ.3) WRITE(6,5)
      NDUM=J
      READ(2,6) TSETD,TSETN,THRANG,ITIMD,ITIMN
    6 FORMAT(3F10.1,2I10)
```

```
      READ(2,7) (TROOM(I),I=1,24)
    7 FORMAT(4F20.1)
      READ(2,15) (QLOAD(I),I=1,24)
      DO 1200 KR=1,24
      QFORT(KR)=QLOAD(KR)/4.0
 1200 CONTINUE
      DO 1300 KS=1,23
      QSLOP(KS)=QFORT(KS+1)-QFORT(KS)
      TSLOP(KS)=TROOM(KS+1)-TROOM(KS)
 1300 CONTINUE
      QSLOP(24)=QFORT(24)-QFORT(1)
      TSLOP(24)=TROOM(24)-TROOM(1)
      DO 2900 I=1,24
      TOSUM=0.0
      HPQSUM=0.0
      RUNSUM=0.0
      SUMKWH=0.0
      SUMRES=0.0
      QSLSUM=0.0
      WTISUM=0.0
      WTOSUM=0.0
      TISET=TSETN
      IF(I.GT.ITIMD.AND.I.LT.ITIMN) TISET=TSETD
C
C         INPUT BUILDING LOAD AND INDOOR TEMPERATURES COMPUTED BY LCSIM
C
      DO 2000 JJ=1,4
      QBLDG=QFORT(I)+QSLOP(I)*(JJ-KTIM1)/4.0
      TDB=TROOM(I)+TSLOP(I)*(JJ-KTIM1)/4.0
      KON=0
      NKOUNT=NKOUNT+1
C
C         INPUT DATA FROM WEATHER FILES
C
      READ(1,20) MDAY,KHOUR,KMIN,KTOUT,KTWW,KTWM,KRHO,KRHE,
     &           KRHW,KRHM,KSRH,KSRV,KWD,KWV
       MDAY=MDAY-20
      HOUR=KHOUR+KMIN/60.0
      TWW=KTWW*0.18+32.0
      TWM=KTWM*0.18+32.0
      TAMB=KTOUT*0.18+32.0
      RHO=KRHO/100.0
      IF(RHO.GT.1.0) RHO=RHO/10.0
      IF(RHO.GT.1.0) RHO=1.0
      WV=KWV
      WDR=KWD
      IF(KSRH.LE.1) KSRH=0
      IF(KSRV.LE.1) KSRV=0
      IF(INDEX.EQ.1) SR=KSRH/2.79498
      IF(INDEX.EQ.2) SR=KSRV/2.17036
      IF(IHOUSE.GT.1) GO TO 401
      RHI=KRHE/100.0
  401 IF(IHOUSE.NE.2) GO TO 402
      RHI=KRHM/100.0
  402 IF(IHOUSE.LT.3) GO TO 403
      RHI=KRHW/100.0
  403 IF(RHI.GT.1.0) RHI=RHI/10.0
      IF(MDAY.GT.J.AND.NKOUNT.LT.96) GO TO 700
      IF(JJ.LE.1.AND.I.LE.1.AND.J.EQ.MDAY1) TCI=TDB
      IF(JJ.LE.1.AND.I.LE.1.AND.J.EQ.MDAY1.AND.IHOUSE.EQ.2) TFOUT=TWM
      IF(JJ.LE.1.AND.I.LE.1.AND.J.EQ.MDAY1.AND.IHOUSE.EQ.3) TFOUT=TWW
C
C          INDEX FOR HEAT PUMP MODE OF OPERATION
C
C          IMODE = 1     ( HEAT PUMP IS IN THE HEATING MODE )
C          IMODE = 2     ( HEAT PUMP IS IN THE COOLING MODE )
```

```
C          IMODE  =  3     ( HEAT PMUP IS OFF )
C
      RATIO=1.0
      ONTIME=0.0
      QCAP=0.0
      CKWH=0.0
      COP =0.0
      QSOL=0.0
      QRES=0.0
      IMODE=3
      IF(QBLDG.GT.0.0)IMODE=2
      IF(QBLDG.LT.0.0.OR.TAMB.LT.60.0) IMODE=1
      IF(QBLDG.GT.0.0.AND.TAMB.LT.60.0) IMODE=3
      IF(IMODE.EQ.1.AND.TDB.GE.(TISET+THRANG/2.0)) IMCDE=3
      IF(IMODE.EQ.2.AND.TDB.LE.(TISET-THRANG/2.0)) IMODE=3
      IF(ABS(QBLDG).LT.100.0) IMODE=3
      IF(IMODE.EQ.2) KMODE=1
      IF(IHOUSE.EQ.1) GO TO 507
      IF(IMODE.EQ.3) GO TO 504
      IF(IMODE.EQ.2) GO TO 501
      CALL HPUMPH (TFOUT,TDB,GGPM,QBLDG,QGROND,QCAP,CKWH,COP,ONTIME,
     &             RATIO,QRES)
      GO TO 502
  501 CALL XMOIST (TDB,TWB,RHI,2,14.696,HAIR,WSAT,WAIR,TWALL)
      CALL HPUMPC (TFOUT,TDB,TWB,GGPM,QBLDG,QGROND,QCAP,CKWH,CCP,ONTIME,
     &             RATIO)
  502 TFHP=TFOUT+QGROND/XMCW
  504 IF(IMODE.EQ.3) TFHP=TFOUT
      HC=2.2+WV*(0.32+0.001*WV)
      TSA=TAMB+ALPH*SR/HO
      DO 2500 ITIME=1,2
      IF(ITIME.GE.2)TFHP=TFOUT
      IF(ITIME.GE.2.AND.IMODE.LT.3.AND.ONTIME.GE.1.0)
     &   TFHP=TFOUT+QGROND/XMCW
C     IF(IHOUSE.EQ.3.OR.(IMODE.EQ.1.AND.SR.LE.0.0)) GO TO 511
      IF(KON.EQ.1) GO TO 512
C
C          CONTROL STRATEGY FOR TURNING THE SOLAR LOOP PUMP ON.
C
      IF(IMODE.EQ.1.AND.(TSA-TFHP).LT.20.0) GO TO 511
      IF(IMODE.EQ.3.AND.KMODE.EQ.0.AND.(TSA-TFHP).LT.20.0) GO TO 511
      IF(IMODE.EQ.2.AND.(TFHP TSA).LT.20.0) GO TO 511
      IF(IMODE.EQ.3.AND.KMODE.EQ.1.AND.(TFHP-TSA).LT.20.0) GO TO 511
  512 CALL SOLAR (TAMB,WDR,WV,SR,INDEX,HOUR,MONTH,NDOM,TCI,TCO,QU,TC,
     &           ITIME)
      QSOL=EXE*XMCW*(TCO-TFHP)
      QSOL1 = QSOL
      TCI=TCO-QSOL/XMCG
      TFIN=TFHP+QSOL/XMCW
C       WRITE(6,2711) TC,TFHP,TFIN,QSOL,TCO,TCI,QU
C2711    FORMAT(1X,3F6.2,E15.7,2X,2F6.2,E15.7)
C
C          CONTROL STRATEGY FOR TURNING THE SOLAR LOOP PUMP OFF.
C
      KON=1
      IF(IMODE.EQ.1.AND.(TC-TFHP).GE.5.0) GO TO 506
      IF(IMODE.EQ.3.AND.KMODE.EQ.0.AND.(TC-TFHP).GE.5.0) GO TO 506
      IF(IMODE.EQ.2.AND.(TFHP TC).GE.5.0) GO TO 506
      IF(IMODE.EQ.3.AND.KMODE.EQ.1.AND.(TFHP-TC).GE.5.0) GO TO 506
  511 TFIN=TFHP
      KON=0
      QSOL=0.0
      TCI=TDB
  506 QEXC=XMCW*(TFIN-TFOUT)
      IF(ITIME.LE.1) TWIN=TFIN
      CALL TUBE (TFIN,QEXC,RATIO,TMID,TFOUT,ONTIME)
```

```
      WTISUM=WTISUM+TFIN
      WTOSUM=WTOSUM+TFOUT
 2500 CONTINUE
      IF(IDP.LE.0) GO TO 1900
      IF(QSOL1.EQ.QSOL2) QSOL1 = 0.0
      QSOL2 = QSOL1
      WRITE(6,601) HOUR,TAMB,TDB,QBLDG,ONTIME,QCAP,QRES,CKWH,COP,QSOL1,
     6               TWIN,TFOUT
      GO TO 1900
  507 IF(IMODE.EQ.3) GO TO 609
      IF(IMODE.EQ.2) GO TO 508
      CALL APUMPH (TAMB,TDB,QBLDG,QCAP,CKWH,COP,ONTIME,QRES)
      GO TO 609
  508 CALL XMOIST (TDB,TWB,RHI,2,14.696,HAIR,WSAT,WAIR,TWALL)
      CALL APUMPC (TAMB,TDB,TWB,QBLDG,QCAP,CKWH,COP,ONTIME)
      IF(IDP.LE.0) GO TO 1900
  609 WRITE(6,602) HOUR,TAMB,TDB,QBLDG,ONTIME,QCAP,QRES,CKWH,COP
 1900 TOSUM=TOSUM+TAMB
      HPQSUM=HPQSUM+QCAP
      RUNSUM=RUNSUM+ONTIME
      SUMKWH=SUMKWH+CKWH
      SUMRES=SUMRES+QRES
      QSLSUM=QSLSUM+QSOL
 2000 CONTINUE
      IF(SUMKWH.LE.0.0) GO TO 2001
      AVGCOP=HPQSUM/(3412.15*SUMKWH)
 2001 TOAVG=TOSUM/4.0
      QSLAVG=QSLSUM/4.0
      WTIAVG=WTISUM/8.0
      WTOAVG=WTOSUM/8.0
      ZZ=I
      IF(IDP.GE.1) GO TO 2002
      WRITE(6,601)ZZ,TOAVG,TROOM(I),QLOAD(I),RUNSUM,HPQSUM,SUMRES,
     6SUMKWH,AVGCOP,QSLAVG,WTIAVG,WTOAVG
 2002 SUM1=SUM1+HPQSUM
      SUM2=SUM2+RUNSUM
      SUM3=SUM3+SUMRES
      SUM4=SUM4 + SUMKWH
 2900 CONTINUE
      IF(SUM4.LE.0.0) GO TO 2901
      DAVCOP=SUM1/(3412.15*SUM4)
 2901 WRITE(6,708) J,MONTH
      WRITE(6,703) SUM2,SUM4
      IF(KMODE.EQ.1) GO TO 2902
      WRITE(6,704) SUM3,DAVCOP
      GO TO 2903
 2902 WRITE(6,11) DAVCOP
 2903 SUM5=SUM5 +SUM1
      SUM6=SUM6+ SUM4
      SUM7=SUM7 +SUM3
      SUM8=SUM8+ SUM2
 3000 CONTINUE
      SAVCOP=SUM5/(3412.15*SUM6)
      IPRIOD=MDAY2-MDAY1+1
      WRITE(6,709) IPRIOD
      IF(KMODE.EQ.1) GO TO 3001
      WRITE(6,707) SUM8,SUM6,SUM7,SAVCOP
      GO TO 3002
 3001 WRITE(6,12) SUM8,SUM6,SAVCOP
 3002 WRITE(6,702)
      GO TO 99
  606 WRITE(6,607)
      GO TO 99
  700 MISS=97-NKOUNT
      WRITE(6,701) MISS,J,MONTH
   99 STOP
```

```
      END
C
C
C
C
C
C
C*********************************************************************
C****                                                            ****
C****     COOLING, WATER-TO-AIR HEAT PUMP: SUBROUTINE(HPUMPC)    ****
C****                                                            ****
C*********************************************************************
C
C
      SUBROUTINE HPUMPC (EWT,EDB,TWB,GPM,QBLDG,QREJ,QCAP,CKWH,COP,ONTIME
     &                   ,RATIO)
      DATA CFM/600.0/
      DATA CFST0,CFST1,CFST2,CFST3,CFST4,CFST5/8.8597,-0.23243,
     &     1.05873E-03,-3.724E-02,1.73322E-05,1.08562E-03/
      DATA CFC0,CFC1,CFC2,CFC3,CFC4/ 0.9313,1.18511E 02, 2.84E-05,
     &     3.08627E-08,-1.25126E-11/
      DATA CFS0,CFS1,CFS2,CFS3,CFS4/0.72047,-1.31175E-03,6.96749E-06,
     &     9.17738E 09,4.17127E 12/
      DATA CKW0,CKW1,CKW2,CKW3,CKW4/0.23951,4.54523E-03,-1.04679E-05,
     &     1.08445E-08,-4.17146E-12/
      DATA C0,C1,C2,C3,C4,C5,C6,C7/566.06,-73.96,-0.389,303.25,-0.3472,
     &     2542.8,-255.56,0.11636/
      DATA SC0,SC1,SC2,SC3,SC4,SC5,SC6,SC7/25582.0,-14.7,-0.278,208.84,
     &     5.903,737.24,-94.44,776.64/
      DATA CP0,CP1,CP2,CP3,CP4,CP5,CP5,CP7/-5.4522E-02,2.223E-03,
     &     8.333E-05,5.6769E-02,-2.0833E-04,-.45262,4.1667E-02,-6.84E-06/
      DATA HR0,HR1,HR2,HR3,HR4,HR5,HR6,HR7/1434.6,-58.31, 0.1667,489.8,
     &     1.04167,831.1,-100.0,0.10355/
C
C
      QLOAD=QBLDG
      CFCHR=1.0
      CFSC=1.0
      CFKW=1.0
      CFSCT=1.0
      X1=EWT
      X2=TWB
      X3=GPM
      X4=X1*X1
      X5=X2*X2
      X6=X3*X3
      X7=X1*X2*X3
      B=CFM/100.0
      SP=21.35 14.03667*B+3.49*B*B 0.37833*B*B*B+0.015*B*B*B*B
C
C        CORRECTION FACTORS FOR TOTAL COOLING CAPACITY, HEAT REJECTION
C        RATE, SENSIBLE COOLING CAPACITY AND COMPRESSOR POWER INPUT,
C        RESPECTIVELY, WHEN AIR VOLUME RATE NE. 600 CFM.
C
      IF(CFM.EQ.600.0) GO TO 101
      Z1=CFM
      Z2=Z1*Z1
      Z3=Z1*Z2
      Z4=Z2*Z2
      CFCHR=CFC0+CFC1*Z1+CFC2*Z2+CFC3*Z3+CFC4*Z4
      CFSC=CFS0+CFS1*Z1+CFS2*Z2+CFS3*Z3+CFS4*Z4
      CFKW=CKW0+CKW1*Z1+CKW2*Z2+CKW3*Z3+CKW4*Z4
  101 IF(EDB.EQ.80.0) GO TO 102
C
C        CORRECTION FACTOR FOR SENSIBLE COOLING CAPACITY WHEN THE
C        ENTERING DRY BULB TEMPERATURE IS NE. 80 F.
```

```
C
      Y1=EDB
      Y2=Y1*Y1
      CFSCT=CFST0+CFST1*X2+CFST2*X5+CFST3*Y1+CFST4*Y2+CFST5*X2*Y1
C
C        COMPUTE THE STEADY STATE COOLING CAPACITY (BTUH), SENSIBLE
C        COOLING CAPACITY (BTUH), HEAT REJECTION RATE (BTUH), AND
C        COMPRESSOR POWER INPUT, RESPECTIVELY.
C
  102 Q=CFCHR*(C0+C1*X1+C2*X4+C3*X2+C4*X5+C5*X3+C6*X6+C7*X7)
      QSCSS=CFSC*CFSCT*(SC0+SC1*X1+SC2*X4+SC3*X2+SC4*X5+SC5*X3+SC6*X6
     &                  +SC7*X7)
      QHRSS=CFCHR*(HR0+HR1*X1+HR2*X4+HR3*X2+HR4*X5+HR5*X3+HR6*X6+HR7*X7)
      CPOWER=CFKW*(CP0+CP1*X1+CP2*X4+CP3*X2+CP4*X5+CP5*X3+CP6*X6+CP7*X7)
      QPART=0.25*Q
      ONTIME=0.25
C
C        CHECK IF THE HEAT PUMP WILL CYCLE ON OFF
C
      RATIO=QLOAD/QPART
      ONTIME=0.25*RATIO
      IF(ONTIME.GT.0.25) ONTIME=0.25
      IF(ONTIME.LT.0.06) ONTIME=0.06
      TIMEC= ONTIME/ALOG(0.005)
      QCYCLE=Q*(ONTIME-TIMEC*(1.0-EXP(-ONTIME/TIMEC)))
      QPART=QCYCLE
      QREJ=QHRSS
      QCAP=QPART
      FP=9.395E-04*SP*CFM
      CKWH=(CPOWER+FP+0.075)*ONTIME
      COP=QPART/(3412.15*CKWH)
      RETURN
      END
C
C
C
C
C**************************************************************************
C****                                                                 ****
C****    HEATING, WATER-TO-AIR HEAT PUMP: SUBROUTINE(HPUMPH)          ****
C****                                                                 ****
C**************************************************************************
C
C
      SUBROUTINE HPUMPH (EWT,EDB,GPM,QBLDG,QEXT,QCAP,CKWH,COP,ONTIME,
     &                   RATIO,QRES)
      DATA CFM/600.0/
      DATA CFH0,CFH1,CFH2,CFH3,CFH4/2.48131,-1.08012E-02,2.79006E-05,
     &     3.06633E 08,1.25129E-11/
      DATA CHE0,CHE1,CHE2,CHE3,CHE4/3.20227,-1.70582E-02,4.53358E-05,
     &     5.0885E-08,2.08555E-11/
      DATA HKW0,HKW1,HKW2,HKW3,HKW4/3.24123, 1.17673E 02,2.64E-05,
     &     2.9195E-08,1.25121E-11/
      DATA H0,H1,H2,H3,H4,H5,H6,H7/1903.4,175.32,0.556,-135.8,0.556,
     &     5962.5, 544.4,1.6715E 02/
      DATA HE0,HE1,HE2,HE3,HE4,HE5,HE6,HE7/-2620.4,170.28,0.278,-117.5,
     &     0.278,6339.6,-622.2,-0.0625/
      DATA CI0,CI1,CI2,CI3,CI4,CI5,CI6,CI7/1.2965,5.99063E-03,4.444E-05
     &     ,-7.73159E-03,9.444E-05,-0.10627,0.02111,2.464E-05/
C
C
      QLOAD=-QBLDG
      CFHC=1.0
      CFHE=1.0
      CFKWH=1.0
      QRES=0.0
```

```
      X1=EWT
      X2=EDB
      X3=GPM
      X4=X1*X1
      X5=X2*X2
      X6=X3*X3
      X7=X1*X2*X3
      B=CFM/100.0
      SP=21.35 14.03667*B+3.49*B*B-0.37833*B*B*B+0.015*B*B*B*B
C
C        CORRECTION FACTORS FOR TOTAL HEATING CAPACITY, HEAT EXRACTICN
C        RATE AND COMPRESSOR POWER INPUT, RESPECTIVELY.
C
      IF(CFM.EQ.600.0) GO TO 101
      Z1=CFM
      Z2=Z1*Z1
      Z3=Z1*Z2
      Z4=Z2*Z2
      CFHC=CFH0+CFH1*Z1+CFH2*Z2+CFH3*Z3+CFH4*Z4
      CFHE=CHE0+CHE1*Z1+CHE2*Z2+CHE3*Z3+CHE4*Z4
      CFKWH=HKW0+HKW1*Z1+HKW2*Z2+HKW3*Z3+HKW4*Z4
C
C        COMPUTE THE STEADY STATE HEATING CAPACITY (BTUH), HEAT
C        EXTRACTION RATE (BTUH), AND COMPRESSOR POWER INPUT (KW),
C        RESPECTIVELY.
C
  101 Q=CFHC*(H0+H1*X1+H2*X4+H3*X2+H4*X5+H5*X3+H6*X6+H7*X7)
      QHESS=CFHE*(HE0+HE1*X1+HE2*X4+HE3*X2+HE4*X5+HE5*X3+HE6*X6+HE7*X7)
      CPOWER=CFKWH*(CI0+CI1*X1+CI2*X4+CI3*X2+CI4*X5+CI5*X3+CI6*X6
     &               +CI7*X7)
      QPART=0.25*Q
      ONTIME=0.25
C
C        CHECK IF THE HEAT PUMP WILL CYCLE ON-OFF
C
      RATIO=QLOAD/QPART
      IF(RATIO.GT.1.0) QRES=(QLOAD-QPART)/3412.15
      ONTIME=0.25*RATIO
      IF(ONTIME.GT.0.25) ONTIME=0.25
      IF(ONTIME.LT.0.06) ONTIME=0.06
      TIMEC= ONTIME/ALOG(0.005)
      QCYCLE=Q*(ONTIME TIMEC*(1.0-EXP(-ONTIME/TIMEC)))
      QPART=QCYCLE
      QEXT= QHESS
      QCAP=QPART
      FP=9.395E-04*SP*CFM
      CKWH=(CPOWER+FP+0.075)*ONTIME
      COP=QPART/(3412.15*(CKWH+QRES))
      RETURN
      END
C
C
C***********************************************************************
C****                                                              ****
C****      COOLING, AIR-TO-AIR HEAT PUMP: SUBROUTINE(APUMPC)       ****
C****                                                              ****
C***********************************************************************
C
C
      SUBROUTINE APUMPC (TAMB,TDB,TWB,QBLDG,QCAP,CKWH,COP,CNTIME)
      DATA CFM /575.0/
      DATA C0,C1,C2,C3,C4,C5,C6,C7/31793.8,36.78393,-0.7037,-528.723,
     &     4.93333, 13.24353,4.7619E 03,1.55021E 03/
      DATA U0,U1,U2,U3,U4,U5,U6,U7/2.33606,1.04283E-02,-2.381E-05,
     &     3.43148E-02,2.8E-04,-1.38345E-03,6.5E-07,1.5E-07/
      DATA S0,S1,S2,S3,S4,S5,S6,S7/-21778.0,75.67,-0.84127,1494.7,
```

```
     &    -15.867,-2.03211,-4.7619E-03,2.12277E-03/
C
C
      QLOAD=QBLDG
      X1=TAMB
      X2=TWB
      X3=CFM
      X4=X1*X1
      X5=X2*X2
      X6=X3*X3
      X7=X1*X2*X3
      ONTIME=0.25
C
C        CORRECTION FACTOR FOR THE SENSIBLE COOLING CAPACITY WHEN THE
C        ENTERING DRY BULB IS NE. 80.0 F.
C
      BF=0.0212+1.5354E-04*X3
      QCF=1.09*X3*(1.0-BF)*(TDB-80.0)
C
C        COMPUTE THE HEAT PUMP STEADY STATE TOTAL CAPACITY (BTUH),
C        SENSIBLE CAPACITY (BTUH), AND POWER INPUT (COMPRESSOR,INDOOR,
C        AND OUTDOOR FANS) (KW).
C
      Q=C0+C1*X1+C2*X4+C3*X2+C4*X5+C5*X3+C6*X6+C7*X7
      QS=QCF+S0+S1*X1+S2*X4+S3*X2+S4*X5+S5*X3+S6*X6+S7*X7
      UKW=U0+U1*X1+U2*X4+U3*X2+U4*X5+U5*X3+U6*X6+U7*X7
      QPART=0.25*Q
C
C        CHECK IF THE HEAT PUMP WILL CYCLE ON-OFF
C
      RATIO=QLOAD/QPART
      IF(RATIO.LE.0.06) RATIO=0.06
      ONTIME=0.25*RATIO
      IF(ONTIME.GT.0.25) ONTIME=0.25
      TIMEC= ONTIME/ALOG(0.005)
      QCYCLE=Q*(ONTIME-TIMEC*(1.0-EXP(-ONTIME/TIMEC)))
      QPART=QCYCLE
      QCAP=QPART
      CKWH=UKW*ONTIME
      COP=QPART/(3412.15*UKW*ONTIME)
      RETURN
      END
C
C
C
C
C*************************************************************************
C****                                                                 ****
C****      HEATING, AIR-TO-AIR HEAT PUMP: SUBROUTINE(APUMPH)          ****
C****                                                                 ****
C*************************************************************************
C
C
      SUBROUTINE APUMPH (TAMB,TDB,QBLDG,QCAP,CKWH,COP,ONTIME,QRES)
      DATA CFM/575.0/
      DATA H0,H1,H2,H3,H4,H5,H6/5393.4,187.435,2.89421,-0.47104,
     &    2.04404E 02, 3.2059E 04,1.71E-06/
      DATA C0,C1,C2,C3,C4/1.59131,1.45083E-02,-1.11042E-03,3.365E-05,
     &     2.7E-07/
C
C
      QLOAD= QBLDG
      CFC=1.0
      CKW=1.0
      TCF=1.0
      TKW=1.0
```

```
      QRES=0.0
      X1=TAMB
      X2=X1*X1
      X3=X1*X2
      X4=X1*X3
      X5=X1*X4
      X6=X1*X5
      ONTIME=0.25
C
C        CORRECTION FACTOR FOR DIFFERENT VALUES OF CFM AND IDOOR DRY
C        BULB TEMPERATURE.
C
      CFC=0.77+0.0004*CFM
      CKW=0.885+0.0002*CFM
      TFC=1.28-0.004*TDB
      TKW=0.86+0.002*TDB
C
C        COMPUTE THE HEAT PUMP STEADY STATE PERFORMANCE AND POWER
C        INPUT (COMPRESSOR, INDOOR & OUTDOOR FANS).
C
      Q=CFC*TFC*(H0+H1*X1+H2*X2+H3*X3+H4*X4+H5*X5+H6*X6)
      UKW=CKW*TKW*(C0+C1*X1+C2*X2+C3*X3+C4*X4)
C
C        CHECK IF THE HEAT PUMP WILL CYCLE ON-OFF
C
      QPART=0.25*Q
      RATIO=QLOAD/QPART
      IF(RATIO.GT.1.0) QRES=(QLOAD-QPART)/3412.15
      ONTIME=0.25*RATIO
      IF(ONTIME.GT.0.25) ONTIME=0.25
      IF(ONTIME.LT.0.06) ONTIME=0.06
      TIMEC= ONTIME/ALOG(0.005)
      QCYCLE=Q*(ONTIME TIMEC*(1.0-EXP(-ONTIME/TIMEC)))
      QPART=QCYCLE
      QCAP=QPART
      COP=QPART/(3412.15*(UKW*ONTIME+QRES))
      CKWH=UKW*ONTIME
      RETURN
      END
C
C
C *****************************************************************************
C ********                                                           ********
C ********             S U B R O U T I N E   S O L A R               ********
C ********                                                           ********
C *****************************************************************************
C
C
      SUBROUTINE SOLAR (TAMB,WDR,WV,SR,INDEX,HOUR,MONTH,NDOM,TCI,TCO,QU,
     &                  TC,ITIME)
C
C
C     PURPOSE:
C           TO COMPUTE THE COLLECTOR EXIT TEMPERATURE AND THE RATE OF
C           HEAT EXCHANGE BETWEEN THE COLLECTOR AND THE AMBIENT FOR A
C           SOLAR ENERGY LOOP.  THE LOOP CONSISTS OF A BARE COLLECTOR
C           PANEL(S), A HEAT EXCHANGER AND A CIRCULATING PUMP.
C           HORIZONTAL OR TILTED SURFACE SOLAR RADIATION CAN BE INPUT
C           IF HORIZONTAL SOLAR RADIATION IS THE INPUT, THE SUBROUTINE
C           WILL CONVERT IT TO THAT INCIDENT ON A TILTED SURFACE.
C
C           DESCRIPTION OF PARAMETERS:
C           INPUT-
C                TAMB - DRY BULB TEMPERATURE OF THE AMBIENT (F)
C                WV   - WIND VELOCITY (MILE/HR)
C                WDR  - WIND DIRECTION (DEGREES)
```

```
C                 SR    - SOLAR RADIATION INCIDENT ON THE SURFACE
C                         (BTU/HR-FT**2)
C                 INDEX- INDEX IDENTIFYING THE TYPE OF SOLAR RADIATION
C                         INPUT
C                       = 1  (HORIZONTAL SURFACE SOLAR RADIATION)
C                       = 2  (TILTED SURFACE SOLAR RADIATION)
C                 HOUR    LOCAL STANDARD TIME (HOURS)
C                 MONTH- MONTH OF THE YEAR (E.G. 1,2,....,12)
C                 NDOM - NUMBER OF DAY THE OF MONTH (E.G. 1,2,....,30)
C                 TCI     COLLECTOR INLET TEMPERATURE (F)
C
C          OUTPUT-
C                 TCO     COLLECTOR OUTLET TEMPERATURE (F)
C                 QU    - RATE OF ENERGY EXCHANGE BETWEEN COLLECTOR AND
C                         AMBIENT (BTU/HR)
C                 TC    - COLLECTOR PLATE TEMPERATURE (F)
C
C
C
      COMMON FLOW
      DIMENSION CST(14),NDYM(12)
      DATA CST/0.057,0.058,0.06,0.071,0.097,0.121,0.134,0.136,0.122,
     &     0.092,0.073,0.063,0.057,0.058/
      DATA NDYM/0,31,59,90,120,151,181,212,243,273,304,334/
      DATA RAD,TZ,ACLONG,XLAT,REF,TILT/.01745,6.,97.,36.017,.2,90.0/
      DATA SIGMA,CHITE,AC,EP,W/0.1712E 08,7.0,140.0,0.20,.333/
      DATA DI,PI,CB,ALPHA,HFI/0.04542,3.14159,38.52,0.95,100.0/
      DATA EW,TN,TL,DO/0.4,60.0,7.0,0.05208/,CPG/0.84/
      DATA D0,D1,D2,D3,D4,D5,D6/8.63633E-02,-1.8859E-04,2.76608E-07,
     &3.05179E-09,-4.04333E-11,1.93653E-13,-3.32374E-16/
      DATA C0,C1,C2,C3,C4,C5,C6/1.31089E 02,2.63437E-05,2.03088E-08,
     & 9.33445E 10,1.07258E-11,-5.20563E-14,9.11518E-17/
      DATA V0,V1,V2,V3,V4,V5,V6/0.4558,1.79714E-03,2.42925E-06,
     & 3.32359E 08,3.50757E 10,-1.6353E 12,2.8E 15/
      DATA P0,P1,P2,P3,P4,P5,P6/0.72204,-2.01553E-04,-1.1113E-06,
     &5.87489E 08,-7.49793E-10,3.86345E-12,-7.03039E-15/
C
C
C
C
      IF(ITIME.EQ.1.AND.HOUR.LE.0.0) HOUR=24.0
      HT=SR
      HUR=HOUR
      G=FLOW*518.1/AC
      XMCG=518.1*CPG*FLOW
      IF(ITIME.EQ.2) HUR=HUR+0.125
      IF(INDEX.EQ.2) GO TO 77
C
C***  CONVERT HORIZONTAL SURFACE RADIATION TO THAT OF A TILTED SURFACE
C
      NM=MONTH
      NDY=NDYM(NM)+NDOM
      NM=NM+1
      NM1=NM
      DNDOM=21 NDOM
      IF(DNDOM.LT.0.0) NM=NM+1
      CSTT=(CST(NM)-CST(NM-1))/30.5
      IF(NDY.LT.100) EQT= 5.0 9.0*SIN((2.0*NDY-2.0)*RAD)
      IF(NDY.GE.100.AND.NDY.LE.242) EQT=-1.+5.0*SIN((NDY-100.)*RAD/.395)
      IF(NDY.GT.242) EQT=-2.5+18.6*SIN((NDY 242.)*RAD/0.685)
      C=CST(NM1)-(CSTT*DNDOM)
      D=23.45*SIN((NDY 80.0)*360.0*RAD/370.0)
      STIME=HUR+TZ-(ACLONG/15.0)+EQT/60.0
      HANGLE=(STIME-12.0)*15.0
      SINA=COS(XLAT*RAD)*COS(D*RAD)*COS(HANGLE*RAD)+SIN(XLAT*RAD)*
     &     SIN(D*RAD)
```

```
      HDN=HT/(SINA+C)
      COSTH=COS((XLAT-TILT)*RAD)*COS(D*RAD)*COS(HANGLE*RAD)+
     &       SIN((XLAT-TILT)*RAD)*SIN(D*RAD)
      HDRECT=HDN*COSTH
      HDIFUZ=0.5*HDN*(C*(1.0+COS(TILT*RAD))+REF*(C+SINA)*
     &       (1.0-COS(TILT*RAD)))
      HT=HDRECT+HDIFUZ
C
C        BEGIN CALCULATION FOR THE COLLECOR USEFUL HEAT GAIN/LCSS ANC
C        THE COLLECTOR OUTLET TEMPERATURE
C
   77 TC=200.0
      UCHK=0.0
      IF(WDR.LT.90.0.OR.WDR.GT.270.0.AND.WV.GT.100.0) WV=5.0
      HW=0.8+0.23*WV
      FS=0.5*(1.0+COS(TILT*RAD))
   88 TAVG=(TC+TAMB)/2.0
      TREF=TC+0.38*(TAMB-TC)
      X2=TAVG*TAVG
      X3=X2*TAVG
      X4=X3*TAVG
      X5=X4*TAVG
      X6=X5*TAVG
      TCR=TC+460.0
      TAMBK=((TAMB-32.0)/1.8)+273.0
      TSKYR=1.8*(0.0552*(TAMBK**1.5))
      TAMBR=TAMB+460.0
      HRPS=SIGMA*EP*FS*(TCR*TCR+TSKYR*TSKYR)*(TCR+TSKYR)
      HRPG=SIGMA*EP*FS*(TCR*TCR+TAMBR*TAMBR)*(TCR+TAMBR)
      HRPW=SIGMA*(TCR*TCR+TAMBR*TAMBR)*(TCR+TAMBR)/(1.0/EP+1.0/EW-1.0)
      ROTA=D0+D1*TAVG+D2*X2+D3*X3+D4*X4+D5*X5+D6*X6
      ROTR=D0+D1*TREF+D2*TREF**2.+D3*TREF**3.+D4*TREF**4+D5*TREF**5.
     &     +D6*TREF**6.0
      VISC=(V0+V1*TAVG+V2*X2+V3*X3+V4*X4+V5*X5+V6*X6)/3.6E03
      PRN=P0+P1*TAVG+P2*X2+P3*X3+P4*X4+P5*X5+P6*X6
      AIRK=C0+C1*TAVG+C2*X2+C3*X3+C4*X4+C5*X5+C6*X6
      BETA=ROTA*((1.0/ROTR-1.0/ROTA)/(TREF-TAVG))
      GRL=32.2*BETA*ABS(TC-TAMB)*(CHITE**3.)/(VISC*VISC)
      HNC=0.021*(AIRK/CHITE)*((PRN*GRL)**0.4)
      UT=HW+HRPS
      UB=HRPW+HNC
      UL=UT+UB
      IF(ABS(UL-UCHK).LE.0.001) GO TO 99
      UCHK=UL
      AM=SQRT(UL/0.00948)
      DUM=0.5*AM*(W-D0)
      FEFF=TANH(DUM)/DUM
      DUM1=UL*(D0+FEFF*(W-D0))
      F1=1.0/((W*UL)*(1.0/DUM1+1.0/CB+1.0/(PI*DI*HFI)))
      DUM2=G*CPG/UL
      FR=DUM2*(1.0-EXP(-F1/DUM2))
      QU=AC*FR*(HT*ALPHA-UL*(TCI-TAMB))
      TFM=TCI+(1.0-FR/F1)*QU/(AC*UL*FR)
      TC=TFM+QU*(1.0/(HFI*PI*DI*TN*TL))
      GO TO 88
   99 TCO=TCI+QU/XMCG
      RETURN
      END
C
C
C
C
C**********************************************************************
C****                                                              ****
C****        PSYCHROMETRIC PROPERTIES: SUBROUTINE(XMOIST)          ****
C****                                                              ****
```

```
C*************************************************************************
C
C
      SUBROUTINE XMOIST(TDB,TWB,RH,INDIC,PATM,HAIR,WSAT,WAIR,TWALL)
C
C        PURPOSE
C           TO DETERMINE THE ENTHALPY, SATURATION MOISTURE CONTENT,
C           AND ACTUAL MOISTURE CONTENT OF MOIST AIR, AND ALSO, THE
C           NECESSARY WALL TEMPERATURE TO INDUCE MOISTURE REMOVAL.
C           GIVEN DRY BULB TEMPERATURE AND EITHER WET BULB TEMPERATURE
C           OR RELATIVE HUMIDITY.
C           (NOTE : THIS PROGRAM ESSENTIALLY REPRODUCES PSYCHROMETRIC
C           CHART DATA)
C
C        DESCRIPTION OF PARAMETERS
C         INPUT
C           TDB     -  DRY BULB TEMPERATURE (F)
C           TWB     -  WET BULB TEMPERATURE (F)
C           RH         RELATIVE HUMIDITY
C           INDIC   -  INPUT INDICATOR
C                   =1, INPUTS ARE TDB, AND TWB
C                   =2, INPUTS ARE TDB, AND RH
C           PATM    -  ATMOSPHERIC PRESSURE (PSIA)
C
C         OUTPUT
C           HAIR    -  ENTHALPY OF MOIST AIR (BTU/LBM DRY AIR)
C           WSAT       SATURATION HUMIDITY (LBM WATER/LBM DRY AIR)
C                      CORRESPONDING TO THE EXISTING WET BULB TEMP.
C           WAIR    -  ACTUAL HUMIDITY (LBM WATER/LBM DRY AIR)
C                      CORRESPONDING TO THE GIVEN DRY BULB TEMP.,
C                      PRES., AND REL. HUMIDITY OR WET BULB TEMP.
C           TWALL      SATURATION OR DEW POINT TEMPERATURE (F)
C                      CORRESPONDING TO THE GIVEN TDB, PATM, AND TWB, OR
C
      K=0
      I=1
      IF(INDIC.NE.1)GO TO 30
      T=TWB
C
C        DETERMINING SATURATION PARTIAL PRESSURE 'PS' (PSIA)
C        OF WATER VAPOR AT THE GIVEN TEMPERATURE
   10 T1=273.16/(((T-32.0)/1.8)+273.16)
      A1= 8.29692*((1.0/T1)-1.0)
      A2=4.76955*(1.0-T1)
      A3=10.79586*(1.0-T1)+5.02808*ALOG10(T1)+1.50474E-04*(1.0-10.**A1)
     &+0.42873E-03*((10.0**A2)-1.0)-2.2196
      PS=(10.0**A3)*14.696
      W=1.004*18.01*PS/(28.967*(PATM-PS))
      IF(K.NE.0) GO TO 50
      IF(INDIC.EQ.2) GO TO 40
      IF(I.NE.1) GO TO 20
      I=2
      WSAT=W
      WAIR=WSAT 0.000235*(TDB T)
      HAIR=0.24*(TWB-32.0)+WSAT*(1060.9+0.444*TWB)
      P=PATM/(1.004*18.01/(28.967*WAIR)+1.0)
      T=TDB
      GO TO 10
C
C        FINDING THE CORRESPONDING RELATIVE HUMIDITY, GIVEN
C        THE WET BULB TEMPERATURE
C
   20 RH=P/PS
      GO TO 90
   30 T=TDB
      GO TO 10
```

```
   40 P=RH*PS
      WAIR=RH*W*(PATM-PS)/(PATM-P)
C
C        FINDING THE CORRESPONDING WET BULB TEMPERATURE, GIVEN
C        THE RELATIVE HUMIDITY
C
      DT= 10.0
   45 T=T+DT
      K=K+1
      IF(K.GT.30)GO TO 70
      GO TO 10
   50 WS=W-0.000236*(TDB-T)
      IF(ABS(WS-WAIR).LE.0.00005) GO TO 80
      IF(WS-WAIR) 60,80,65
   60 T=T-DT
      DT=DT/2.0
   65 CONTINUE
      GO TO 45
   70 WRITE(6,100)
   80 TWB=T
      WSAT=W
      HAIR=0.24*TDB+WAIR*(1060.9+0.444*TDB)
C
C        DETERMINING THE SATURATION OR DEW POINT TEMP. 'TWALL'
C        CORRESPONDING TO THE GIVEN PRESSURE, DRY BULB TEMPERATURE,
C        AND RELATIVE HUMIDITY OR WET BULB TEMP.
C
   90 IF(P.LE.0.0185) TWALL=(P-0.0185)/0.00077
      IF(P.GT.0.0185) TWALL=(P-0.0185)/0.00124
      IF(P.GT.0.0309) TWALL=(P-0.0113)/0.00196
      IF(P.GT.0.0505) TWALL=(P+0.0129)/0.00317
      IF(P.GT.0.0885) TWALL=(P+0.0441)/0.004145
      IF(P.GT.0.1217) TWALL=(P+0.10394)/0.005641
      IF(P.GT.0.17811) TWALL=(P+0.21284)/0.007819
      IF(P.GT.0.2563) TWALL=(P+0.3845)/0.01068
      IF(P.GT.0.3681) TWALL=(P+0.6435)/0.01438
      IF(P.GT.0.5069) TWALL=(P+1.0235)/0.01913
      IF(P.GT.0.6982) TWALL=(P+1.5608)/0.0251
  100 FORMAT(' ***** ITERATION IN XMOIST DOES NOT CONVERGE')
      RETURN
      END
C
C
C
C
C
C *********************************************************************
C ****                                                            ****
C ****   VERTICAL EARTH-WATER HEAT EXCHANGER: SUBROUTINE(TUBE)    ****
C ****                                                            ****
C *********************************************************************
C
C
        SUBROUTINE TUBE (TFIN,QEXC,RATIO,TMID,TFOUT,ONTIME)
C
C
C          PURPOSE:
C             TO COMPUTE THE WELL EXITING FLUID TEMPERATURE, THE HEAT
C             TRANSFER RATE BETWEEN THE FLUID AND THE SOIL, AND THE
C             TEMPERATURE PROFILE IN THE SOIL SURROUNDING THE WELL.
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               TFIN   - TEMPERATURE OF THE FLUID EXITING THE WELL. (F)
C               QEXC   - HEAT REJECTED (COOLING MODE) OR EXTRACTED (HEAT
C                        ING MODE) BY THE HEAT PUMP AFTER SOLAR LOOP
C                        CONTRIBUTION.  IN THIS SUBROUTINE, IT IS USED AS
C                        A FIRST ESTIMATE OF THE HEAT CONDUCTED TO OR
C                        FROM THE SOIL. (BTU/HR)
C               RATIO - FRACTION OF THE HEAT PUMP CYCLE THE HEAT PUMP
C                        IS ON.
C              ONTIME- PERIOD OF TIME THE HEAT PUMP IS OPERATING. (HR)
C
C
C        OUTPUT-
C               TMID - TEMPERATURE OF THE FLUID AT THE CENTER OF WELL.(F)
C               TFOUT - TEMPERATURE OF THE FLUID EXITING THE WELL. (F)
C
C
      COMMON /BLOCK/GGPM,KPRINT
      DIMENSION R(33),DR(30),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),TGROND(12)
      DATA KOUNT,KOUNT1,CYTIME/0,0,0.25/
      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 RO,ALPHA,DEPTH,COND,DELTAZ,CPF/0.10666,0.0290,250.,0.82
     &                 ,250.0,1.0/,RHO/62.0/
C
C
C
  199 FORMAT('1',//////,54X,'VALUES OF NON-UNIFORM GRID',//)
  201 FORMAT(5X,'THE WATER TEMPERATURE=',F8.2,10X,'THE HEAT CONDUCTED',
     &       ' TO OR FROM EARTH=',F10.0,'BTU.',/)
  202 FORMAT(5X,'THE SOIL TEMPERATURES ARE,',/)
  203 FORMAT(4(15X,8(F8.2,1X),/))
  204 FORMAT(55X,'THE TOTAL HEAT TO, OR FROM THE SOIL IS',F10.0,'BTU'/)
  300 FORMAT(' ',////,13X,'TEMPERATURE DISTRIBUTION',F6.1,'  FEET FROM'
     &        ,' THE BOTTOM OF THE WELL',3X,///)
  301 FORMAT(4(5X,8('DR(',I2,')=',F7.4,2X),///))
  302 FORMAT(////,54X,'THE RADIUS VALUES',//)
  303 FORMAT(5(5X,8('R(',I2,')=',F7.4,2X),///))
  304 FORMAT('1')
  305 FORMAT(//,10('*'),' TUBE DOES NOT CONVERGE AFTER 20 ITERATIONS')
C
C
      TF(1)=TFIN
      FLRATE=497.3*GGPM
      GFLUX=FLRATE/(PHI*RO*RO)
      IF(KOUNT.GE.1) GO TO 500
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 100 I=2,NP1
      R(I)=R(I-1)+DRAPP
      DRAPP=DRAPP*EPSR
```

```
  100 CONTINUE
      FACTOR=RRL/R(NP1)
      DO 101 I=2,NP1
      R(I)=R(I)*FACTOR
      DR(I)=R(I)-R(I-1)
  101 CONTINUE
      IF(KPRINT.EQ.0) GO TO 111
C
C        PRINT THE VALUES OF NON-UNIFORM GRID
C
      WRITE(6,199)
      WRITE(6,301) (I,DR(I),I=2,NP1)
  111 DO 122 I=1,NP1
      R(I)=R(I)+RO
  122 CONTINUE
      IF(KPRINT.EQ.0) GO TO 113
C
C        PRINT THE VALUES OF RADIUS
C
      WRITE(6,302)
      WRITE(6,303) (I,R(I),I=1,NP1)
      WRITE(6,304)
C
C        CALCULATE RADIUS AND FACE AREA OF LEFT FACE OF THE CONTROL
C        VOLUME
C
C
  113 RL(1)=R(1)
      DO 102 I=2,NP1
      RL(I)=(R(I)+R(I-1))/2.0
      FA(I)=2.0*PHI*RL(I)
  102 CONTINUE
C
C        BUILD UP CONTROL VOLUME
C
      CV(1)=PHI*(RL(2)*RL(2)-RL(1)*RL(1))
      DO 104 I=2,N
      CV(I)=PHI*(RL(I+1)*RL(I+1)-RL(I)*RL(I))
  104 CONTINUE
      K=(DEPTH/DELTAZ)
      NZ = K/2
      DDPTH=K
C
C        INITIAL TEMPERATURE DISTRIBUTION
C
      DO 106 J=1,K
      TGROND(J)=62.5
      TAV(J)=TGROND(J)
      DO 106 I=1,NP1
  106 CC(J,I)=TGROND(J)
      AREA=2.0*PHI*RO*DELTAZ
      VOL=PHI*RO*RO*DELTAZ
  500 QTOTAL=0.0
C
C        SET UP CALCULATIONS ACCORDING TO HEAT PUMP CYCLING
C
      IF(RATIO.GE.1.0) GO TO 991
      IF(KOUNT1.EQ.1) GO TO 990
      DTIME=ONTIME
      MSET=1.0
      GO TO 992
  990 MSET=0
      DTIME=CYTIME-ONTIME
      GO TO 992
  991 DTIME=0.125
C
```

```
C        BUILD UP COEFFICIENTS OF TDMA
C
  992 X=2.0*FLRATE*DTIME/(RHO*VOL)
      Y=H*AREA*DTIME/(RHO*VOL*CPF)
      XX=X Y 1.0
      YY=X+Y+1.0
C
C
C        GUESS WELL OUTLET TEMPERATURE
C
        TFOUT = TFIN - 6.0
       DO 51 ITER = 1 , 24
C
C        CALCULATE THE HEAT EXCHANGE BETWEEN THE WELL FLUID AND
C        THE SOIL
C
        Q =FLRATE * CPF * ( TFOUT - TFIN)
        IF(MSET .EQ. 0) Q = 0.0
        IF(MSET .EQ. 0) TFOUT = TFIN
      DO 37 KK=1,K
      KCHECK=0
  800 DO 109 I=1,NP1
  109 C(KK,I)=CC(KK,I)
      A(1)= (ALPHA*DTIME*FA(2)/(CV(1)*DR(2)))
      B(1)=0.0
        C(1,1) = TFIN
      C(KK,1)=C(KK,1)+(Q*ALPHA*DTIME)/(DELTAZ*CCND*CV(1))
      D(1)=1.0 A(1)
      DO 200 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)
  200 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 25 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)
   25 CONTINUE
C
C        BACK SUBSTITUTION
C
      C(KK,N)=C(KK,N)/D(N)
      DO 35 I=2,N
      J=N I+1
      C(KK,J)=(C(KK,J)-A(J)*C(KK,J+1))/D(J)
   35 CONTINUE
C
   37        CONTINUE
             TAVER = (TFIN + TFOUT ) / 2.0
             DIF = TAVER - C(1,1)
C
C            CHECK FOR CONVERGENCE
C
        IF (ABS(DIF) .LE. 1.0) GO TO 52
        TFOUT = TFOUT + 0.5
   51   CONTINUE
   52   TMID = C(1,1)* 2.0- TFIN
        TFOUT = TMID
11151C
```

```
C         PRINT THE RESULTS
C
        WRITE (6,53) ITER , TFOUT , DIF
  53    FORMAT (1X, 'ITER = ', I5 ,'TFOUT = ',F10.5,'DIF',F10.5)
C
C         RETURN TO FORWARD ANOTHER TIME STEP
C
      KOUNT=KOUNT+1
      IF(KPRINT.EQ.1) WRITE(5,204) QTOTAL
      IF(RATIO.GE.1.0) GO TO 40
      KOUNT1=1
      IF(MSET.EQ.0) KOUNT1=0
   40 RETURN
      END
C
          SUBROUTINE STANK (QSOLT,TTANK,SUMT,TAMB)
C
C         LIST OF MAIN VARIABLES
C
C         CAPA   -  CAPACITY OF THE TANK (GALLONS)
C         DTANK   -  DIAMETER OF TANK   (FT)
C         DLTANK   -  DENSITY OF FLUID IN THE TANK  (LB/CU.FT)
C         SLTANK   -  SPECIFIC HEAT OF FLUID IN THE TANK  ( BTU/LB-F)
C         QSOLT   -  RATE OF HEAT ADDITION TO THE TANK  ( BTU/HR)
C         TTANK      TEMPERATURE OF FLUID IN THE TANK  (F)
C
          CAPA = 1000.0
          DTANK = (CAPA*0.1337*4.0/3.142)**0.3333
          ALTANK = DTANK
          RTANK = 5.0
          DLTANK = 62.4
          SLTANK = 1.0
          ATANK = (3.142*0.5*DTANK**2) + (3.142*DTANK*ALTANK)
          VTANK = 3.142*0.25*ALTANK*(DTANK**2)
          AMTANK = DLTANK * VTANK
C
C         HEAT ADDITION TO THE TANK
C
          QSOLT = 0.125 * QSOLT
C
C         HEAT LOSS FROM THE TANK
C
          QLOSS = ATANK * (TTANK - TAMB) * 0.125 * RTANK
C
C         NET HEAT ADDITION TO THE TANK
C
          QNET = QSOLT - QLOSS
C
C         CALCULATE TEMPERATURE OF THE FLUID IN THE TANK
C
          TTANK = TTANK + QNET / (AMTANK * SLTANK)
          SUMT = SUMT + QSOLT
C
C         PRINT THE RESULTS
C
          WRITE (6,8000) QSOLT,QLOSS,QNET,TTANK
 8000     FORMAT (1X,'  HEAT ADDED (BTU)     = ',E15.7,/,1X,
     $               '  HEAT LOSS  (BTU)     = ',E15.7,/,1X,
     $               '  NET HEAT  ADDED (BTU)= ',E15.7,/,1X,
     $               '  TEMP OF TANK (F)     = ',F6.2)
//GO.SYSIN DD *
&INPUT MONTH=5,MDAY1=24,MDAY2=25,INDEX=2,KPRINT=0,IHOUSE=3,ICP=1,&END
//GO.FT01F001 DD DSN=U11864A.MAY82D24.JGSH.DATA,UNIT=3350,
// VOL=SER=SYSTSO,DISP=SHR,DCB=(RECFM=FB,LRECL=80,BLKSIZE=6160)
//GO.FT02F001 DD DSN=OSU.ACT11451.CHLOAD.DATA,UNIT=3350,
// VOL=SER=DASD60,DISP=SHR,DCB=(RECFM=FB,LRECL=80,BLKSIZE=6160)

//GO.SYSIN DD *
//
```

VITA

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