Modelling and evaluation of a cistern-based ground-coupled heat pump: its implications in reducing energy in a commercial swine operation

James D. Leary

Iowa State University

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Modelling and evaluation of a cistern-based ground-coupled heat pump; its implications in reducing energy in a commercial swine operation

Leary, James D., Ph.D.
Iowa State University, 1994
Modelling and evaluation of a cistern-based ground-coupled heat pump: its implications in reducing energy in a commercial swine operation

by

James D. Leary

A Dissertation Submitted to the
Graduate Faculty in Partial Fulfillment of the
Requirements for the Degree of
DOCTOR OF PHILOSOPHY

Department: Agricultural and Biosystems Engineering
Major: Agricultural Engineering

Approved:
Signature was redacted for privacy.
In Charge of Major Work
Signature was redacted for privacy.
For the Major Department
Signature was redacted for privacy.
For the Graduate College

Iowa State University
Ames, Iowa
1994
To my wife

*Marcia*

and son

*Rick*
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NOMENCLATURE

\[ A_g = \text{Total exterior surface area of piping in the ground, } \text{ft}^2 \]
\[ A_w = \text{Total exterior surface area of piping in the well water storage tank, } \text{ft}^2 \]
\[ \beta = \text{Soil thermal diffusivity, } \text{ft}^2/\text{day} \]
\[ c = \text{Specific heat of water, Btu/lb-\text{F}} \]
\[ D_i = \text{Internal pipe diameter, ft} \]
\[ D_o = \text{Outside pipe diameter, ft} \]
\[ \varepsilon = \text{Coil effectiveness, dimensionless} \]
\[ g = \text{Gravitational constant, } \text{ft}/\text{s} \]
\[ h_{ci} = \text{Convective heat transfer coefficient inside piping, Btu/h ft}^2\text{F} \]
\[ h_{co} = \text{Convective heat transfer coefficient outside piping, Btu/h ft}^2\text{F} \]
\[ k = \text{Thermal conductance, Btu/h ft F} \]
\[ \dot{m}_o = \text{Mass flow rate of water through the heat exchanger coil, lb/h} \]
\[ \dot{m}_t = \text{Mass flow rate of well water through the well water storage tank, lb/h} \]
\[ m_r = \text{Mass of water in the tank, lb} \]
\[ \mu = \text{Fluid viscosity, lb}_{\text{m}}/\text{ft-h} \]
\[ \text{Nu}_D = \text{Nusselt number based on pipe diameter, dimensionless} \]
\[ P = \text{Power used by the heat pump, Btu/h} \]
\[ P_r = \text{Prandtl number} \]
\[ \dot{q}_c = \text{Cooling capacity, Btu/h} \]
\[ \dot{q}_{\text{cond}} = \text{Heat rate from heat pump condenser, Btu/h} \]
\[ \dot{q}_{\text{evap}} = \text{Heat rate to the heat pump, Btu/h} \]
\[ \dot{q}_h = \text{Heating capacity, Btu/h} \]
\[ \dot{q}_{\text{hpi}} = \text{Heat rate from the indoor side of the heat pump, Btu/h} \]
\( \dot{q}_{\text{hpo}} \) = Heat rate from the outdoor side of the heat pump. Btu/h
\( \dot{q}_{\text{hx}} \) = Heat rate from the heat exchanger coil in tank. Btu/h
\( Ra_D \) = Rayleigh number based on outside pipe diameter. dimensionless
\( Re_D \) = Reynolds number. dimensionless
\( R_g \) = Total radial resistance of piping in the ground. h-ft²-F/Btu
\( R_w \) = Total radial resistance of piping in the tank. h-ft²-F/Btu
\( \rho \) = Density of water. lb/m³/ft³
\( t \) = Time step. h
\( T_i \) = Inside fluid temperature. F
\( T_m \) = Mean earth temperature. F
\( T_o \) = Outside fluid temperature. F
\( T_s \) = Surface temperature. F
\( T_{\infty} \) = Fluid temperature at a distance. F
\( T_t \) = Temperature of the water in the well water storage tank. F
\( T_2 \) = Entrance temperature to the heat exchanger. F
\( T_3 \) = Exit temperature from the heat exchanger. F
\( T_4 \) = Exit temperature from the outdoor side of the heat pump. F
\( T_5 \) = Initial input temperature of source water to the heat pump. F
\( T_{5n} \) = Calculated temperature of source water to the heat pump. F
\( T_6 \) = Entrance temperature to the indoor side of the heat pump. F
\( T_7 \) = Exit temperature from the indoor side of the heat pump. F
\( T_8 \) = Indoor room temperature. F
\( T_{gw} \) = Temperature of the well water entering the storage tank. F
\( T_{g1} \) = Temperature of the ground around the piping. F
\( T_{g2} \) = Temperature of the ground around the tank. F
\[ \Sigma UAT_{g_2} = \text{Sum of the overall heat transfer coefficient-area products times ground temperatures at a given depth around the tank, Btu/h-ft}^2 \]

\[ \Sigma UA = \text{Sum of the overall heat transfer coefficient-area products, Btu/h-ft}^2 \]

\[ \nu = \text{Kinematic viscosity, ft}^2/\text{h} \]

\[ \dot{V}_i = \text{Volumetric flow rate of water through indoor side of heat pump} \]

\[ \dot{V}_o = \text{Volumetric flow rate of water through outdoor side of heat pump} \]

\[ \dot{V}_t = \text{Volumetric flow rate of well water, gpm} \]
ABSTRACT

This research details the analysis of a ground-coupled heat pump system which utilizes a 20,000 gallon well-water storage tank as a source of heating and cooling. It addresses the feasibility of such a system and discusses the system in terms of energy efficiency--first law conservation analysis. The potential use of this type of system for the control of sensitive thermal environments involving swine is also evaluated.

The goal of this research was to investigate ISU's SNMRC heat pump system. Specific objectives were:

1. To evaluate the system in terms of heat exchanger and heat pump efficiencies.
2. To develop a mathematical model which simulates the heat pump system, accounting for energy gains/losses throughout, and which is capable of predicting energy availability based upon various heat exchanger coil lengths, cistern sizes, heat pump sizes and well water flow rates.
3. To evaluate the system in terms of its ability to provide energy.
4. To investigate the potential of the heat pump system for control of sensitive thermal environments for swine.

Investigations of this system were conducted both experimentally, and numerically through a mathematical model. Experimentally, the tank heat exchanger efficiencies were determined to average 0.71 during the heating mode and 0.99 during the cooling mode. From the mathematical model, the efficiencies were 0.72 and 1.00 for the heating and cooling modes, respectively. The model predicted heat pump coefficients of performance from 3.5 to 4.8 for heating and 2.4 to 4.0 for cooling. Results indicated that appropriately sized heat pump, heat exchanger coil and cistern were a source of supplemental heating, are energy conservative and offer control of sensitive thermal environments for specific applications involving swine.
1 INTRODUCTION

1.1 Background

Heat pumps have gained widespread acceptance in the United States in both rural and urban settings. Single units often provide both residential heating and cooling thereby replacing the separate units of a furnace and an air conditioner. Some heat pumps (air-source) utilize outdoor air as a means to moderate energy use. Higher efficiencies are realized when the indoor-to-outdoor temperature is at a minimum. Such systems, however, may have limited usefulness in an Iowa climate when low temperatures of winter reduce the heat pump efficiency to that of electric resistance heat.

A more efficient option for northern climates such as Iowa's, is the ground-coupled heat pump where consistent, deep ground temperatures of approximately 52 F (Kusuda and Achenbach, 1965) moderate energy use both summer and winter. This efficient use of energy is especially of interest to the commercial swine industry which must provide appropriate thermal environmental conditions for optimal growth and production rates. Costly rural electric energy or propane are often the energy of choice for heating purposes. And cooling, when necessary, most often utilizes electricity. Heat pump technology offers both while utilizing energy efficiently.

1.2 Research Objectives

A 1.5 ton (18,000 Btu/h) water-source heat pump system was installed at Iowa State University's Swine Nutrition and Management Research Center (SNMRC). The system utilized the center's cistern—an in-ground, 20,000 gallon well-water storage tank—as the ground-couple. The heat pump itself was located in the center's machine shop. Its use was strictly experimental since the primary source of heating for the machine shop, a gas-fired
furnace, provided heat for the adjacent feed mill as well. Because the machine shop is used with irregularity, cooling was not a necessity.

The primary link between the heat pump and the cistern was a custom designed heat exchanger constructed from 500 feet of one-inch polyethylene piping with a coil diameter of four feet. The piping was submerged in the well water storage tank, a 10-foot diameter, 37-foot long, ribbed fiberglass tank. Additional polyethylene piping and polyvinyl chloride (PVC) piping provided supply and return lines to the heat pump located 70 feet away.

The goal of this research was to investigate ISU’s SNMRC heat pump system. Specific objectives were:

1. To evaluate the system in terms of heat exchanger and heat pump efficiencies.
2. To develop a mathematical model which simulates the heat pump system, accounting for energy gains/losses throughout, and which is capable of predicting energy availability based upon various heat exchanger coil lengths, cistern sizes, heat pump sizes and well water flow rates.
3. To evaluate the system in terms of its ability to provide energy.
4. To investigate the potential of the heat pump system for control of sensitive thermal environments for swine.
2 LITERATURE REVIEW

2.1 Ground-Coupled Heat Pump Research

Ground-coupled heat pump systems are constructed to take advantage of the earth as a heat source or a heat sink. As such, the extent to which thermal contact is maintained ultimately determines the effectiveness of the ground couple.

Several configurations exist to create the heat exchange required of ground-coupled heat pump systems. Systems are considered either "open" or "closed". In an open system, water for the heat pump is introduced continually without recycling during operation. Typical open systems shown in Figure 2.1 are a) single well, b) dual well and c) single well/waste. Water quality and quantity in such systems is important. Proper measures must be taken to prevent corrosion and scaling of water heat exchangers in the heat pump since open systems have a continuous source of oxygen and minerals (Bose et al., 1985). Withdrawal and discharge of water may pose additional problems for the local aquifer.

Many of the problems evident in the open system are eliminated in the closed system (also referred to as "closed-loop"). In this type of system, a fixed amount of fluid, usually water, is recirculated through sealed piping. An appropriate length and configuration of piping provides the necessary heat exchange with the contact medium. Horizontal and vertical loops are shown in Figure 2.2 with series and parallel configurations for vertical loops detailed in Figure 2.3. One additional closed-loop configuration (not shown) involves submerging the piping in the lake.

Soil thermal conductivity, affected by moisture content of the soil is an important parameter to consider for earth-coupled closed loops. The vertical heat exchanger is installed deep enough to take advantage of both the constant temperature of the earth and the high thermal conductivity of saturated soil which lowers its resistance to heat transfer. The
Figure 2.1 Open system water-source heat pump arrangements utilizing well water.
Figure 2.2 Typical closed-loop systems
Figure 2.3 Vertical closed-loop ground-couple heat exchanger arrangements
horizontal heat exchanger which is installed within 6.5 feet of the ground's surface, on the other hand, is subject to seasonal variations in soil temperature and moisture. These variations result in the need for an increased length of heat exchanger for the horizontal loop in comparison to the vertical loop (Klimkowski et al., 1985). The lake loop system will result in lower winter fluid temperatures (northern climates), but the reduced installation costs may compensate for any minor reduction in performance.

The temperature of the ground and other soil thermal characteristics are extremely important factors when considering using the earth as a heat source or sink. Although exact ground temperature at a specific site can be obtained only by direct measurement, equations and statistical data (Mei, 1988; Bose et al., 1985; Akridge and Poulos, 1983; Kusuda and Achenbach, 1965) are available which allow reasonably accurate calculations of ground temperatures for various geographical locations, time of year, and for various depths and soil characteristics. Important soil characteristics are density, thermal conductivity, and specific heat. They combine to provide thermal diffusivity, a parameter which is an integral part of the ground temperatures reviewed. Thermal diffusivity varies with the thermal conductivity of the soil and soil moisture has a direct impact on thermal conductivity.

Van Wijk and De Vries (1963) demonstrated that when the moisture content of clay decreases from 40 to 20 percent, the thermal conductivity decreases by 26 percent. It is understandable, then, the importance of soil moisture content in utilizing the ground as a heat source/sink. Other parameters used within the ground temperature equations are annual surface temperature amplitude, mean annual ground temperature, day of minimum surface temperature (a phase constant) and, day of the year.

Piping length, diameter and material, fluid flow rates, and piping configuration are additional factors to consider for the heat exchange with the ground. A proportional relationship exists between the length of piping and the rate of heat transfer to/from the
ground. The pipe inner diameter (ID), wall thickness and thermal conductivity determine the pipe's resistance to heat transfer. Thin walls, large diameters and high thermal conductivities give lower thermal resistances and therefore higher heat transfer per unit length of pipe. On the other hand, fluid velocities--volumetric flow rate per cross-section area of pipe inside--influence the thermal resistance inversely. That is, higher fluid velocities result in lower resistances to heat transfer through the fluid layer at the wall (Incropera and DeWitt, 1985). In this case, smaller diameter piping for a given flow rate is favored.

Pressure drop in the piping is also a function of pipe ID and fluid flow rate but in combination with pipe length. In general, longer piping results in a greater pressure drop, however, an increase in ID for the same length of pipe will result in a lower pressure drop.

As can be seen, there are many variables which influence the design of heat exchangers for ground-coupled systems. In addition to the few combinations listed, the economics of construction and installation is yet another variable to be considered. While higher efficiencies may be designed into a system, they are usually associated with a higher cost. For example, in a vertical loop system, the cost of 0.75 inch pipe can be as low as $0.20 per foot of bore and as high as $1.00 per foot of bore for 1.5 inch (Kavanaugh, 1992). And, according to Trelease (1989), the length of pipe will be reduced less than 10 percent by using 1.5 inch pipe instead of 1.0 inch pipe.

2.2 Heat Pump Economics

Initial investment in ground-coupled heat pump technology can be considerable when compared to conventional types of heating and air conditioning. A life cycle analysis completed by Bierbaum (1986) indicated that the initial investment in a closed-loop ground-coupled heat pump system was approximately 40 percent more than the initial cost of an air-source heat pump system and 140 percent greater than the initial cost of an electric furnace.
and air-conditioner combined. However, payback was achieved after less than 7.5 years. Kavanaugh (1992) reported that as much as 50% of the system's first costs may be involved in the drilling/trenching necessary to establish a ground-couple. He further asserted that premiums, when compared to a base electric cooling/natural gas heating systems, were typically $500 to $800 per ton for horizontal systems and $600 to $1000 per ton for vertical systems but pointed out that simple payback occurs within five to eight years. Bose and Parker (1985) concurred with Kavanaugh (1992), stating that first costs will usually be dominated by the cost of the ground heat exchanger. The tradeoff, however, is due to the high thermodynamic efficiency of heat pumps. For ground-coupled heat pumps, the heating coefficient of performance (COP) ranges from approximately 2.5 to 3.2 (units of energy output for each unit of electric energy input) while the energy efficiency rating (EER) for cooling ranges from 10.6 to 15 Btuh/W (ARI, 1989). Such efficiencies make ground-coupled heat pumps attractive despite the high initial costs. In terms of heating only and for less efficient air-to-water heat pumps, Tassou et al. (1986), using an annualized life-cycle cost method of analysis, determined that the heat pump offers economic advantages over electric resistance heaters and oil-fired boilers. Energy savings have also been noted by Braud (1983) whose research found a seasonal reduction in energy consumption of 21% for a Louisiana residence.

### 2.3 Thermal Environmental Considerations for Swine

The thermal environment in which swine are raised directly impacts production and growth rates. The thermal environment for livestock considers such factors as air temperature, humidity, air velocity, ground surface temperature, and radiant energy from surfaces within the physical environment in which the animal is contained. Poultry and livestock are homeothermic, which means they maintain a relatively constant internal body
temperature despite temperatures changes within their environment. For swine the average internal body temperature is 102.5°F with a range of 101.6 to 103.6°F (Midwest Plan Service, 1987). In order to maintain temperature within this range, pigs must adjust their rate of heat production to balance heat loss and heat storage. Operating near the extremes of this temperature range result in lower growth and production rates while operating outside the temperature range can be fatal (Midwest Plan Service, 1987).

The temperatures for maximum swine performance are listed as 85 to 70°F for 12 to 75 pound pigs and between 60 and 70°F for 75 to 220 pound pigs. The minimum (lb feed)/(lb gain) ratio occurs within these ranges as well (Midwest Plan Service, 1987). According to Esmay (1978), adverse effects are apparent sooner from high than from low temperatures and livestock are considered to be depressed by temperatures over 75°F. Various methods of cooling livestock include providing cooled drinking water, increasing convective cooling, providing a cooled slab, evaporative cooling, inspired-air cooling and air conditioning.

Brumm and Shelton (1987) studied the influence of reduced nocturnal temperatures on the performance of weaned pigs. While increased gain was associated with the treatment, an additional benefit of reduced utility input costs by over 30 percent was also realized.

Nienaber et al. (1987) evaluated the effects of cyclic temperatures on growing (44-88 lb) and finishing (140-220 lb) swine. Negative performance response occurred for the finishing swine which endured cyclic temperature patterns of ±22°F at 68°F and at 41°F while swine maintained at constant temperatures of 68°F or 41°F were not affected. Growing swine were not affected by the either treatment.

Special considerations need to be made for the various swine types. Newborn pigs, for example, need an environment with temperatures which range between 90 and 95°F for the first three days of life while the sow is most comfortable at 60-65°F. For 3-week-old pigs, a temperature of 85°F is recommended and for growing-finishing pigs, a temperature between
60 and 70°F is recommended (Midwest Plan Service, 1987). Another consideration may be the use of artificial cooling for breeding and gestating swine.
3 EXPERIMENTAL SET-UP

3.1 Facility

A 1.5 ton (18,000 Btuh) water-to-water heat pump system was installed at Iowa State University’s Swine Nutrition and Management Research Center (SNMRC), located approximately two miles northwest of Ames, Iowa. The system utilized the Center’s cistern— an in-ground, 20,000 gallon well-water storage tank—as the ground-couple. The heat pump itself was located in the Center’s machine shop. The shop, a 590 square foot room with a 15 foot ceiling, occupies a quarter of a larger building on its east side (Figure 3.1).

The heat pump’s use was strictly experimental since the primary source of heating for the machine shop, a gas-fired furnace, also provides heat for the adjacent feed mill. Also, cooling was not a necessity for the machine shop as it is used irregularly.

3.2 Equipment

A 5-ton (60,000 Btuh) Command Aire Heat Pump (Model WHP611) was the initial heat pump installed for this research. The unit failed before any experimental data could be collected and was replaced with a Fedders Solar/Compression Furnace (Model SOCF-020700). The Fedders unit, a water-to-water heat pump, was rated at 20,000 Btuh heating. Its configuration provided for heating only. In order to collect cooling data the piping itself was reversed so that the condenser discharged waste heat to the cistern while the evaporator supplied cooling to the machine shop.

The 20,000 gallon well-water storage tank is an Owens-Corning (Model D-6 20,000) fiberglass tank (Figure 3.2). Its inner diameter is 10.0 feet. The ribbed, cylindrical midsection is 27.0 feet long and the endcaps are spherical with a radius of 5.0 feet. It has a
Figure 3.1 Machine shop location/layout
22.0 inch flanged manway with a two foot long extension, as shown. Well-water is supplied through a 6.0 inch NPT fitting located at the bottom of the tank on the left-hand side. It is discharged within the tank through a four inch diameter, 7.0 foot extension to the fitting. Water is withdrawn from the bottom of the tank through the right-hand side 6.0 inch NPT fitting. The tank's thermal characteristics are discussed in section 6.3 and Appendix E.

The primary link between the heat pump and the cistern was a custom designed heat exchanger (Figure 3.3). The main portion of the heat exchanger consisted of 500 feet of one-inch polyethylene piping (Schedule 40) submerged in the well-water storage tank. The piping was purchased in 100 foot-long sections bundled in four-foot coils. The four-foot coil shape was maintained within the tank by one-inch polyvinyl chloride (PVC) piping. The PVC was connected to the polyethylene heat exchanger, providing supply and return lines from the heat pump. A horizontal section of PVC expanded the coil over a length of twenty feet. Two vertical, eight-foot sections maintained the coil's vertical position in the tank. To prevent the coil from floating in the tank, two twelve-foot lengths of one-inch PVC were threaded through the four-foot opening of the coil and through four, eight-inch concrete blocks. Supply and return lines to the tank were established through the two-foot long manway extension. Two PVC-to-polyethylene fittings were designed and constructed for this connection with allowance for two 0.25 inch diameter temperature probes. To complete the link with the heat pump, two 70-foot sections of polyethylene tubing connected at the manway extension were buried in the ground at a depth of 6 feet. Access to the machine shop was gained through a six-inch plastic drain tile which penetrated the cement floor next to the east wall.

Water pumping was accomplished by two, high head Grundfos circulating pumps (Model UP-26-96 BF). The performance curve, nearly linear, indicated 30 feet of head at
Figure 3.3 Heat exchanger coil configuration
0 gpm and 0 feet of head at 25 gpm. Standard power requirements for each pump were 205 watts of input for a single phase of 115 volts.

Two First Company fan coil units (Model 6HBC-3) were used to distribute the heating and cooling to the shop. Maximum heating capacity of each unit was 24,100 Btuh at 5.5 gpm and an entering water temperature of 120 F. The total cooling capacity of each unit, at an entering water temperature of 45 F, was 22,200 Btuh at 5.7 gpm, a dry bulb temperature of 84 F and a wet bulb temperature of 67 F or at 17,800 Btuh with a flow rate of 5.7 gpm, a dry bulb temperature of 75 F and a wet bulb temperature of 63 F.

3.3 Data Acquisition

Temperatures, volumetric flow rates and power consumption were monitored throughout the system. Thermocouples were located strategically within the storage tank surrounding the heat exchanger as well as at supply and return entrances to the heat exchanger, and at supply and return entrances to the evaporator and condenser sides of the heat pump. Volumetric flow rates were recorded for water on both evaporator and condenser sides of the heat pump and for water use from the well-water storage tank. Power consumption of the heat pump and its two associated water pumps was recorded utilizing a watt transducer. All data, except the well-water flow rate, were recorded automatically using the Campbell Scientific (Model CR10) data acquisition system. Sensor locations are shown in Figure 3.4. The well-water flow rate was recorded manually.

Temperatures throughout the system were recorded using T-type (copper-constantan) thermocouples. The output from this sensor is millivolts. A positive linear relationship exists for voltages from -6.258 to 20.869 mV and corresponding temperatures from -454 to 752 F with a cold junction reference temperature of 32 F. This temperature range far exceeded the
Figure 3.4 Data acquisition sensor locations
temperature range encountered in the current research. It was, therefore, more than adequate for measuring system temperatures.

The water flow rates on indoor and outdoor sides of the heat pump were monitored by Rho Sigma (Model RS 6805) flow meters. The output from the meters is in the form of a pulse and corresponds to gallons. The resolution is 0.25 gallons per pulse. Pulse amplitude (high) is 2.0 to 5.5 volts while pulse amplitude (low) is from 0 to 0.8 V. Pulse width and spacing are each a minimum of 2.5 microseconds. The calibration procedure is described in Appendix A.

An Ohio Semitronics (Model PC5-061C) watt transducer was used to measure instantaneous power. Output from 0 to 10 volts direct current corresponds to a full scale of 20 kilowatts. Response time is 250 milliseconds and accuracy is ± 0.5% of full scale.

Because the CR10 computer accepts a voltage range of -2.5 to 2.5 Vdc, a voltage divider was constructed to step down the output for compatibility with the CR10. The calculations for the voltage divider and subsequent calibration of the unit are outlined in Appendix A.

The Campbell Scientific CR10 and an AM416 relay multiplexer collected data from the sensors. Sixteen T-Type thermocouples were connected to the multiplexer. The CR10 contained a 10TCRT thermocouple reference. The two flow meters were connected to the pulse input channels on the CR10 while the watt transducer was connected to one of the six paired analog inputs.

The program written to collect and process data (Appendix B) sampled sensors every minute except pulse inputs which were automatically sampled at a frequency of 8 Hz. Polynomial curve fits were input for the specific heat and density of water as a function of temperature. At each sampling, the heat rate was calculated for both condenser and evaporator sides of the heat pump as well as for the heat exchanger located in the well-water storage tank. The following formula was used for the calculation:
\[ \dot{q} = \rho \dot{V} c \Delta T \]

where

\[ \dot{q} = \text{the heat rate, Btu/h} \]
\[ \rho = \text{the density of water, lbm/ft}^3 \]
\[ \dot{V} = \text{the volumetric flow of water, ft}^3/\text{h} \]
\[ c = \text{the specific heat of water, Btu/lbm-F} \]
\[ \Delta T = \text{the temperature difference between inlet and outlet, F} \]

Calculations were completed at each sampling then averaged every ten minutes in order to reduce the amount of data to be stored while at the same time maintaining the integrity of the time-dependent and variable interdependent calculation.
4 HEAT EXCHANGER DESIGN

4.1 Basic Premise

An energy balance approach was considered in determining the heat transfer rate from the polyethylene piping to arrive at an appropriate length of piping to serve as a heat exchanger for the experimental set up. Because the flow in a tube was completely enclosed, this approach was applied to determine how the total convection heat transfer, $q_{conv}$, was related to the temperature differences at tube inlet and outlet (Incropera and DeWitt, 1985).

In considering the control volume of Figure 4.1, the mass flow rate, $m$, is constant and the convection heat transfer occurs at the inner surface. Fluid kinetic and potential energy changes, as well as energy transfer by conduction in the axial direction may be considered negligible. Now, only thermal energy and flow work are of importance. Flow work is the result of fluid moving through a control surface (Moran and Shapiro, 1986). On a per unit mass of fluid basis, it may be expressed as the product of the fluid pressure $p$ and specific volume $v$: The process of applying the conservation of energy to the control volume is outlined below.

![Figure 4.1 Control volume for internal flow in a tube.](image)
\( T_m \) is the mean fluid temperature. From the diagram, the differential approach provides

\[
dq_{\text{conv}} + m(cT_m + pv) - \left[ m(cT_m + pv) + m \frac{d(cT_m + pv)dx}{dx} \right] = 0
\]

or, more simply and on a per mass basis, the difference in specific enthalpy between inlet and outlet provides the energy transfer from the piping as

\[
q_{\text{conv}} = h_o - h_i
\]

The right hand side of Equation 4.2 is the sum of the internal specific energy and flow work and may be written as

\[
h_o - h_i = u_o - u_i + v(p_o - p_i)
\]

where

\[
u_o - u_i = \int_{T_{m,i}}^{T_{m,o}} c(T) \, dT
\]

so

\[
h_o - h_i = \int_{T_{m,i}}^{T_{m,o}} c(T) \, dT + v(p_o - p_i)
\]

then

\[
q_{\text{conv}} = \int_{T_{m,i}}^{T_{m,o}} c(T) \, dT + v(p_o - p_i)
\]

Based on a maximum \( \Delta p \) for the system of 32 feet of pump head, the second term on the right may be neglected (32 ft / 778 ft lb/Btu = 0.04 Btu/lb vs. \( cdT \) of approximately 5.0 Btu/lb for a \( dT = 5.0 \) F). The resulting equation for the heat transfer rate (including mass flow) is

\[
q_{\text{conv}} = \dot{m}c_p(T_{m,o} - T_{m,i})
\]

In order to proceed with the ground-coupled heat exchanger, certain assumptions were made. They are:

1. Steady-state conditions.
2. Constant properties.
3. Incompressible liquid.
4. Negligible kinetic and potential energy changes.
5. Uniform convection coefficient at outer surface.

4.2 Procedure

The overall resistance of the polyethylene piping, $R_{tot}$, was determined by calculating the inside and outside convective heat transfer coefficients and combining them with the conductivity of the piping. $R_{tot}$ was then achieved by analogy to a resistance network. The energy balance procedure outlined in 4.1 was reworked to yield an equation which provides the length of piping required to achieve the necessary heat transfer. The procedure is outlined below.

The resistance network was set up as follows:

$$
T_i \rightarrow \frac{1}{h_c} \rightarrow R_{pipe} \rightarrow \frac{1}{h_o} \rightarrow T_\infty
$$

where

- $h_c = \text{Inside convective heat transfer coefficient, Btu/h ft}^2 \text{ F}$
- $h_o = \text{Outside convective heat transfer coefficient, Btu/h ft}^2 \text{ F}$
- $T_i = \text{Inside fluid temperature, F}$
- $T_\infty = \text{Outside fluid temperature, F}$

Total resistance of the network is then

$$R_{tot} = \frac{1}{h_c} + R_{pipe} + \frac{1}{h_o}$$
To determine the internal convective heat transfer coefficient, the Nusselt number identified by Incropera and DeWitt (1985) was incorporated.

\[ Nu_D = 0.023 \, Re_D^{0.8} \, Pr^n \]

where

\[ Re_D = \text{Reynolds number based on pipe inner diameter, unitless} \]
\[ Pr = \text{Prandtl number, unitless} \]
\[ n = 0.4 \text{ for heating or 0.3 for cooling, unitless} \]

The Reynolds number was further defined as:

\[ Re_D = 4 \, \dot{m}/(\pi D_i \mu) \]

where

\[ \dot{m} = \text{Mass flow rate, lb}_{m}/h \]
\[ D_i = \text{Internal pipe diameter, ft} \]
\[ \mu = \text{Fluid viscosity, lb}/ft\,h \]

The final calculation for the internal convective heat transfer coefficient was then

\[ h_{ci} = \frac{Nu_D k}{D_i} \]

where \( k \) is the thermal conductivity of the fluid in Btu/h-ft-F.

The outside convective heat transfer coefficient was determined using another equation for the Nusselt number, based on free convection, from Incropera and DeWitt (1985). The equation is written as follows:

\[ Nu_D = \{0.6 + 0.387 \, Ra_D^{1/6} \} + (0.559/Pr)^{9/16} )^{8/27} \]

where

\[ Ra_D = \text{Rayleigh number based on outside pipe diameter, unitless} \]
\[ Pr = \text{Prandtl number, unitless} \]

The Rayleigh number was further defined as

\[ Ra_D = \alpha (T_s - T_\infty) D_o^{3/4} \]
where

\[ g = \text{Gravitational constant, ft/s}^2 \]

\[ \beta = \text{Volumetric thermal expansion coefficient, } {1/{F}} \]

\[(T_s - T_\infty) = \text{Temperature difference between the external surface and fluid, F} \]

\[ D_o = \text{Outside pipe diameter, ft} \]

\[ \nu = \text{Kinematic viscosity, ft}^2/{h} \]

\[ \alpha = \text{Thermal diffusivity, } {ft}^2/{h} \]

The external convective heat transfer coefficient was then calculated as

\[ h_{co} = \frac{N_{\text{Pr}} \alpha}{D_o}. \]

The energy balance approach demonstrated in equations 4.1 to 4.7 was modified for the purpose of determining the piping length needed. Figure 4.2 provides the parameters used in development of the equation which yielded length.

![Figure 4.2 Differential element of pipe](image)
The energy balance:

\[ \text{Energy in} = \text{Energy out} \]

\[ \dot{m}c \ T_{in} = \dot{m}c \ T_{out} + dq \]

where

\[ dq = \frac{(T - T_{\infty}) \ dA}{R_{tot}} \]

continuing

\[ \dot{m}c \ (T_{in} - T_{out}) = \frac{(T - T_{\infty}) \ dA}{R_{tot}} \]

\[ - \ \dot{m}c \ dT = \frac{(T - T_{\infty}) \ dA}{R_{tot}} \]

\[ \int_{A_i}^{A_f} \frac{l}{\dot{m}c \ R_{tot}} \ dA = \int_{T_{in}}^{T_{out}} \frac{l}{(T - T_{\infty})} \ dT \]

\[ \frac{A_f}{\dot{m}c \ R_{tot}} = \ln \left[ \frac{T_{out} - T_{\infty}}{T_{in} - T_{\infty}} \right] \]

The equation for length is then written as

\[ L = - \ln \left[ \frac{T_{out} - T_{\infty}}{T_{in} - T_{\infty}} \right] \frac{\dot{m}c \ R_{tot}}{\pi D_i} \]

The internal film resistance was determined to be 0.003 h ft-F/Btu for an internal flow rate of three gallons per minute. Considering laminar flow conditions on the exterior of the pipe, the external film resistance was determined to be 0.019 h ft-F/Btu. The pipe resistance was 0.028 h ft-F/Btu. A total thermal resistance of 0.050 h ft-F/Btu was calculated for the combined effect of the internal and external heat transfer coefficient and pipe resistance.

Alternate calculations of total resistance using an internal flow rate of ten gallons per minute yielded a resistance which was only 4.0 percent lower than the resistance at three gallons per minute. The length of piping required for the 3.0 gpm flow rate was determined
to be 300 feet while for the 10.0 gpm flow rate, the length was calculated to be 425 feet. The 500 feet of piping used within the tank was considered a conservative length.
The experimental portion of this research consisted of continuous run-time periods for the heat pump during both heating and cooling seasons. Continuous operation was conducted for the purpose of demonstrating the resilience of the well-water storage tank in maintaining consistent water-source temperatures despite the thermal impact of the heat pump. The 1.5 ton unit installed, however, met with performance difficulties from start-up. The expansion valve had to be replaced. Because the unit is no longer manufactured and limited specification information was available, the expansion valve installed as a replacement created a condition which required the water flow rate through the evaporator to be finely balanced to prevent refrigerant floodback. A flow rate of approximately 0.5 gpm was necessary. Subsequent performance of the heat pump was highly variable and resulted in an erratic heat rate from the evaporator side of the heat pump. The end result was that the 1.5 ton heat pump unit operated at approximately one third its rated capacity.

### 5.1 Heat Pump Performance

Heating mode data was collected from March 17, 1994 to April 1, 1994 while cooling mode data was taken from May 18, 1994 to May 25, 1994. During each period of time, the heat pump ran continuously. The impact of the heat pump on the well-water storage tank was minimal during the heating mode. A comparison of well-water storage tank temperature to the rate of heat extracted from the tank during this mode of operation is shown in Figure 5.1. The tank temperature is observed to fluctuate as erratically as the rate of heat extracted by the heat exchanger. Upon closer inspection, however, the overall variability occurred within a range of less than one degree Fahrenheit over a 14 day period of time. From this perspective, the tank temperature was considered relatively constant throughout the period of continuous operation. During the cooling mode, a comparison of tank temperature to the rate of heat
exhausted to the tank from the heat pump condenser (Figure 5.2), indicated a gradual 2 F increase in tank temperature over the eight-day continuous run period. The heat rate through the heat exchanger from the condenser is noticeably more stable than that displayed in Figure 5.1. Also, the water flow rate through the condenser during this time was approximately 4.6 gpm compared to an average 0.5 gpm during the heating mode. The sharp drop in $q_{hx}$ during Julian day 143 may have been the result of the heat pump being shut off. A review of the data for that day indicated that only enough power to run the circulating pumps was being recorded for a 5 hour period of time during midday. Upon close inspection of Figure 5.2, it is interesting to note that the temperature of the tank began to drop very shortly after the heat pump was off, then began to rise when the heat pump came back on and is indicative of the tank’s resilience to thermal impact.

Heat pump efficiency is given a rating called coefficient of performance (COP). This rating is the heating or cooling capacity of the unit divided by the power input. For heating and cooling the COPs are calculated respectively, as follows:

$$COP_h = \frac{\dot{q}_h}{P \times 3.412}$$

$$COP_c = \frac{\dot{q}_c}{P \times 3.412}$$

where

- $\dot{q}_h$ = Heating capacity of unit, Btu/h
- $\dot{q}_c$ = Cooling capacity of unit, Btu/h
- $P$ = Power input to unit, W
- $3.412$ = Conversion from watts to Btu/h
Heating Mode Data

Figure 5.1 Tank water temperature compared to the rate of heat extracted by the HX

Cooling Mode Data

Figure 5.2 Tank water temperature compared to the rate of heat exhausted by the HX
Another used rating for refrigeration is the energy efficiency ratio (EER) which is the cooling capacity (Btuh) divided by the power input (watts) or in equation form

\[ EER = \frac{\dot{q}}{P} \]

The average COP\(_h\) achieved by the heat pump was 1.39 while the cooling COP was 1.06. Or, in terms of the energy efficiency ratio, \( EER = 3.62 \text{ Btu/h/W} \). Performance of this kind is low in comparison to heat pumps operating at full capacity but is the kind of performance expected for off-design conditions.

The energy input to and output from the heat pump may also be analyzed. For heating, the heat capacity (condenser-side) is the sum of the power input and the energy gained through the heat exchanger (evaporator-side) of the heat pump. In equation form,

\[ \dot{q}_{\text{cond}} = \dot{q}_{\text{evap}} + P \]

This is represented graphically in Figure 5.3. Inspection of the graph indicates once again how variable the evaporator heat rate was. The evaporator heat rate curve shows the greatest amount of fluctuation and its influence on the condenser curve is noticeable. Of further mention is the relationship among curves as suggested by the equation for \( \dot{q}_{\text{cond}} \) above. When the amount of heat extracted by the evaporator decreases, the capacity of the heat pump, \( \dot{q}_{\text{cond}} \), also decreases, but less so, and a subtle increase in power, as expected, is observed.

For cooling, the cooling capacity (evaporator-side) is the difference between the heat exhausted to the heat exchanger (condenser-side) and the power input. This may be written as

\[ \dot{q}_{\text{evap}} = \dot{q}_{\text{cond}} - P \]

Figure 5.4 depicts the heat pump cooling mode energy exchange. Note the closeness in proximity of the curves for power and evaporator heat rate. Recall that these two parameters establish the cooling coefficient of performance—the ratio of capacity (\( \dot{q}_{\text{evap}} \)) to power—and was given as 1.06 above. This relationship is easily visualized in Figure 5.4.
Figure 5.3 Heat pump energy input/output; experimental

Figure 5.4 Heat pump energy input/output; experimental
5.2 Heat Exchanger Effectiveness

Heat exchanger effectiveness may be described in terms of inlet and outlet temperatures to/from the heat exchanger coil and the average temperature of the well-water \( T_{\text{tank}} \) in the storage tank. An equation defined by Feiereisen et al. (1982) for tanks with negligible stratification is written as

\[
e = \frac{(T_{\text{coil.in}} - T_{\text{coil.out}})}{(T_{\text{coil.in}} - T_{\text{tank}})}
\]

The average effectiveness of the heat exchanger was 0.71 for the heating mode and 0.99 for the cooling mode. The average cooling mode effectiveness of nearly 100% indicated the positive effect of a low flow rate (approximately 0.5 gpm) on heat transfer from the coil. Low flow rates resulted in the outlet temperature from the heat exchanger to become nearly identical to the temperature of the storage tank well water.

5.3 Analysis of Uncertainty

The probable uncertainty in determining the heat pump condenser and evaporator heat rates was accomplished by an analysis of the uncertainties in the individual measuring devices for the range of values over which they applied. The formulas used for heat pump heat rate follow.

\[
\dot{q}_i = \rho c \dot{V}_i (T_f - T_h)
\]

\[
\dot{q}_o = \rho c \dot{V}_i (T_s - T_d)
\]

The uncertainty in the calculation was due to the inaccuracies in measurement of temperatures and volumetric flowrates. The density and specific heat of water were considered to constant. The measurement, device, and uncertainty of each device are listed in Table 5.1. Table 5.2 provides the average temperature, flow rate and power used for the analysis.
Table 5.1 Percent of uncertainty in each measurement device

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Device</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>Thermocouple</td>
<td>± 2 F</td>
</tr>
<tr>
<td>Volumetric Flow Rate</td>
<td>Rotometer</td>
<td>± 5 %</td>
</tr>
<tr>
<td>Power Consumption</td>
<td>Watt Transducer</td>
<td>± 0.5 %</td>
</tr>
</tbody>
</table>

Table 5.2 Values used in the analysis of uncertainty

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Heating Value</th>
<th>Heating Uncertainty</th>
<th>Cooling Value</th>
<th>Cooling Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>T₄ (F)</td>
<td>48.2</td>
<td>± 0.2</td>
<td>98.1</td>
<td>± 0.2</td>
</tr>
<tr>
<td>T₅ (F)</td>
<td>48.6</td>
<td>± 0.2</td>
<td>59.9</td>
<td>± 0.2</td>
</tr>
<tr>
<td>T₆ (F)</td>
<td>77.4</td>
<td>± 0.2</td>
<td>55.6</td>
<td>± 0.2</td>
</tr>
<tr>
<td>T₇ (F)</td>
<td>104</td>
<td>± 0.2</td>
<td>57.8</td>
<td>± 0.2</td>
</tr>
<tr>
<td>V₁ (gpm)</td>
<td>0.487</td>
<td>± 0.024</td>
<td>4.52</td>
<td>± 0.226</td>
</tr>
<tr>
<td>V₀ (gpm)</td>
<td>4.63</td>
<td>± 0.23</td>
<td>0.475</td>
<td>± 0.237</td>
</tr>
<tr>
<td>P (Btu/h)</td>
<td>6070</td>
<td>± 30.4</td>
<td>6700</td>
<td>± 33.5</td>
</tr>
</tbody>
</table>

The analysis of uncertainty was completed following the approach demonstrated by Henry et al. (1991) and outlined below. For the indoor side of the heat pump, the uncertainty is written as
\[ dq_i = \left[ \left( \frac{\partial q_i}{\partial V_i} \cdot \omega_v \right)^2 + \left( \frac{\partial q_i}{\partial T_7} \cdot \omega T_7 \right)^2 + \left( \frac{\partial q_i}{\partial T_6} \cdot \omega T_6 \right)^2 \right]^{1/2} \]

and for the outdoor side the uncertainty is determined by

\[ dq_i = \left[ \left( \frac{\partial q_o}{\partial V_o} \cdot \omega_v \right)^2 + \left( \frac{\partial q_o}{\partial T_5} \cdot \omega T_5 \right)^2 + \left( \frac{\partial q_o}{\partial T_4} \cdot \omega T_4 \right)^2 \right]^{1/2} \]

Results of the analysis are tabulated in Table 5.3. An error of uncertainty of 74.6% is indicated for the energy rate, \( \dot{q}_o \), on the outdoor side of the heat pump (evaporator). The high level of uncertainty was traced to the combined effect of low heat rate (small temperature for the flow rate) and the water flow rate during the heating mode. While disconcerting, this result helps explain the erratic heat pump behavior described in Section 5.1. It further identifies the heat rate to the evaporator as the unstable heat pump parameter. This conclusion is also supported, though to a lesser degree, by the percent of error for the heat rate to the indoor side (evaporator) of the heat pump, \( \dot{q}_i \), during cooling.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Heating</th>
<th>Cooling</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \dot{q}_i ) (Btu/h)</td>
<td>6570 ± 5.1%</td>
<td>4940 ± 13.9%</td>
</tr>
<tr>
<td>( \dot{q}_o ) (Btu/h)</td>
<td>882 ± 74.6%</td>
<td>9088 ± 5.0%</td>
</tr>
<tr>
<td>P (Btu/h)</td>
<td>6070 ± 0.5%</td>
<td>5700 ± 0.5%</td>
</tr>
</tbody>
</table>

Table 5.3 Results of the analysis of uncertainty
6 SYSTEM MODEL

6.1 Rationale for Model

A mathematical model was developed for the purpose of making predictions regarding the availability of thermal energy from a cistern-based, ground-coupled heat pump system with the prospect of reducing energy consumption at a commercial swine facility. Its development allowed for the manipulation of variables which were beyond those limited by the experimental set-up and which influenced energy availability.

Cistern size, heat exchanger length and material characteristics, heat pump capacity, flow rates and soil characteristics are variables capable of being manipulated by the model. It is viewed as a valuable tool in assessing the potential for use in such sensitive thermal environments as breeding, gestation, farrowing and nursery.

6.2 Assumptions Used in Development

Assumptions used for the heat pump revolve around the thermodynamic principle of conservation of energy. Basically, energy input equals energy output with the assumption being that no energy is stored. For heating, the rated capacity of a unit is equal to the sum of electrical energy input to the unit, and heat extracted from the ground-source. For cooling, the cooling capacity is the heat exhausted to the ground-source minus the electrical energy input. The equation form of each is shown below.

\[ \dot{q}_{hpi} = P + \dot{q}_{hpo} \quad \text{Heating} \]

\[ \dot{q}_{hpi} = \dot{q}_{hpo} \cdot P \quad \text{Cooling} \]

where

\[ \dot{q}_{hpi} = \text{Heating/cooling capacity of unit. Btu/h} \]

\[ \dot{q}_{hpo} = \text{Heat rate from/to ground source. Btu/h} \]

\[ P = \text{Electrical energy supplied. Btu/h} \]
In addition, the heat pump roomside temperature was assumed constant. This resulted in four one-dimensional curve fits based on source water temperature. Two were for the capacity of the unit—one for heating and one for cooling. The remaining two curve fits were for the unit's power consumption—heating and cooling. The data were obtained from the manufacturer and based on the Air-Conditioning and Refrigeration Institute's Standard 320 (American Air Filter, 1986).

The heat exchanger submerged within the storage tank was assumed to consist totally of polyethylene piping, Schedule 40, even though approximately thirty feet of its length was PVC piping. Convection coefficients for the piping were determined for the purpose of calculating total thermal resistance in a radial direction. The internal flow rate was assumed to be 3.0 gpm while external free convection on a horizontal circular cylinder was assumed to be the fluid condition outside the piping. As previously mentioned, a change in the internal flow rate to 10 gpm resulted in a total resistance which was only four percent lower than that calculated at 3 gpm.

The polyethylene tubing within the ground was assumed to be buried at a depth of six feet for its entire length even though on approach to the storage tank it reached a depth of one foot. This length amounted to no more than five percent of the buried length. Also, the length of piping within the machine shop was purposely neglected. It amounted to less than five percent of the total heat exchanger length.

The depth below ground surface of the top of the well-water storage tank was assumed to be two feet. Ground temperatures at four depths, based upon tank diameter, were calculated as if the tank was buried below ground even though it is buried in a mound. Because ground water temperature at a depth beyond 30 to 60 feet is very nearly equal to the annual average air temperature, as reported by Kusuda and Achenbach (1965), the inlet temperature of well-water to the tank was assumed to be 52°F based on data from Kusuda and
Achenbach (1965). Water within the tank was considered well-mixed. This was verified by ten tank temperatures monitored at various horizontal and vertical positions. These temperatures differed by less than 0.5 F. The tank well water temperature, therefore, was considered uniform. In turn, the outlet water temperature from the tank to the Center, was assumed to be the temperature of the water in the tank. The water in the tank was considered to be at a height which would contact 75% of the cylindrical surface area (approximately 90% of its volume). The wall of the tank was considered a plane one-dimensional wall for purposes of heat transfer.

Ground temperatures were estimated using the following equation from Bose et al. (1985):

\[
T_g = T_m - \frac{A_s \exp(-x/(\pi \times 365 \alpha)^2)}{\cos(2\pi/365 \times [D - D_o - x/(2 \times 365/\pi \alpha)^2])}
\]

where

- \(T_m\) = Mean earth temperature, F
- \(A_s\) = Annual surface temperature amplitude, F
- \(\alpha\) = Soil thermal diffusivity, ft^2/day
- \(x\) = Soil depth, ft
- \(D\) = Day of year (Jan 1 = 1), days
- \(D_o\) = Phase constant, day of minimum surface temperature, days
6.3 Description of Development

Initial development began by defining the physical parameters of the well-water storage tank. Formulas for surface area and volume were developed based upon tank diameter and length of the cylindrical segment of the tank. As mentioned previously, each endcap consists of a hemisphere the radius of which is half the tank diameter. The tank was divided into four sections of equal area along the horizontal direction of the tank's cylindrical midsection as shown in Figure 6.1. This provided four levels or depths at which ground temperature, and thereby heat rates, were calculated. The rate equations of Fourier and Newton provided the basis for calculating the rate of heat transfer across the wall of the tank. For conduction, a form of Fourier's law is expressed as

$$q_x = -kA \Delta T$$

where

- \( k \) = Thermal conductivity, Btu/h ft °F
- \( A \) = The wall area normal to the direction of heat transfer, ft²
- \( \Delta T \) = Temperature difference, °F

And from Newton's law of cooling,

$$q = hA \Delta T$$

where

- \( h \) = Convection heat transfer coefficient, Btu/h ft² °F
- \( A \) = The wall area normal to the direction of heat transfer, ft²
- \( \Delta T \) = Temperature difference, °F
Figure 6.1 Tank sectional surface areas
These two equations may be combined by utilizing a parameter known as the overall heat transfer coefficient. The equation is as follows:

\[ q = UA\Delta T \]

where

- \( U \) = Overall heat transfer coefficient, Btu/h ft\(^2\) F
- \( A \) = The wall area normal to the direction of heat transfer, ft\(^2\)
- \( \Delta T \) = Temperature difference, F

The overall heat transfer coefficient is related to the total thermal resistance, \( R_{\text{tot}} \), by

\[ UA = 1/R_{\text{tot}} \]

and

\[ R_{\text{tot}} = 1/hA + L/kA + R_{\text{soil}} \]

where \( L \) is the tank thickness.

The total resistance across the wall of the well-water storage tank was calculated for three wall configurations and two fluid conditions—air and water—for a combination of six different total resistances. When paired with two types of tank areas, ribbed and non-ribbed, twelve equations for calculating the overall heat transfer coefficient-area product, \( UA \), were the outcome.

The analysis in section 4.2 for design of the heat exchanger was incorporated into the system model. Instead of solving for length as in 4.2, however, the equations for the model were developed to yield outlet temperature. For a known length of piping, the outlet temperature is a function of inlet temperature, internal flow rate, total pipe resistance, ground temperature or tank water temperature, and volume of water within the tank as well as its flow rate. The formulas used for this development are located in Appendix D.
Heat pump modelling was accomplished by curve fitting manufacturer's heating/cooling capacities and power input, to the water-source inlet temperature. This resulted in four equations, capacity versus inlet temperature and power versus inlet temperature, two each for heating and two for cooling for each unit considered. One additional set of curves, one for power and one for capacity, was established to evaluate a 10-ton heat pump but in the heating mode only. The manufacturer's data listed water-source temperatures as low as 60 F. Extrapolation below this temperature was considered reasonable after discussion with the HVAC technician (Kapaun, 1994) who supplied the manufacturer's data and considering that the best fit equations for the units used were linear.

Conservation of energy principles were incorporated throughout the model. Figure 6.2 shows the various heat rate losses/gains across control volume and the parameters needed to determine them.

The program created to test the model was written in FORTRAN. The FORTRAN code is located in Appendix C. An input file contained most of the constants and variables required of the program. The user input whether heating or cooling was requested, the numeric day of the year the test was to begin, the number of days for which it was to run, and the flow rate of water through the outdoor side of the heat pump. The file input variables are described in Appendix C. Output provided is on an hourly basis.

6.4 Verification of Model

The heat pump model was developed from experimental data of capacity and power curve fit to the experimental water-source inlet temperature. A two degree polynomial curve fit was used for heat pump capacity and a one degree polynomial was used to describe the power input during the heating mode. For the cooling mode, one degree polynomials were used for curve fits of both power and heat pump capacity. Appendix D contains the curve fit
Figure 6.2  Energy gains/losses and other parameters used in development of the mathematical model
equations. No attempt was made to model the heat pump more closely because of its erratic performance (described in Chapter 5). However, because temperatures were considered more indicative of energy gains and losses within the system, various system parameters were manipulated in an effort to improve the numerical model. Improvements to the numerical model are considered in section 6.7.

The system model was verified by comparison to experimental data. Temperatures and heat rates were compared. The results are shown graphically in Figures 6.3 through 6.10. Figures 6.3 (heating) and 6.4 (cooling) compare experimental heat exchanger inlet and tank temperatures, $T_{\text{hx, in (exp)}}$ and $T_{\text{tank (exp)}}$, respectively, to those calculated by the model, $T_{\text{hx, in (model)}}$ and $T_{\text{tank (model)}}$. For the heating mode, the difference between tank temperatures, experimental versus model, ranged from -1.97 to 0.27 F and heat exchanger inlet temperatures differed from -2.41 to -0.47 F. Model heat exchanger inlet temperatures differed most from the experimental temperatures during the cooling mode, from a low of -1.12 F to a high of 4.37 F while tank temperature differences ranged between -1.18 and 0.32 F.

The sharp decline in experimental heat exchanger temperature (Figure 6.4) which occurred during Julian day 143, was excluded from this analysis because it did not represent the continuous operation being evaluated. Tank water temperature differences exhibited the smallest variation, ranging from -1.18 to 0.32 F. Figures 6.5 and 6.6 compare experimental and model heat exchanger inlet and outlet temperatures. Inlet temperature differences ranged from -0.47 to -2.41 F during the heating mode and from -1.12 to 4.37 F, as mentioned previously, during the cooling mode. Outlet temperature differences ranged from 0.0 to -1.83 F during the heating mode and from -1.25 to 0.31 F during the cooling mode.

Temperatures on the outdoor side of the heat pump are represented in Figures 6.7 and 6.8. The difference between experimental and model inlet temperatures ranged from -0.04 to
Figure 6.3 Tank and heat exchanger inlet temperature data: experimental vs. model

Figure 6.4 Tank and heat exchanger inlet temperature data: experimental vs. model
Figure 6.5  Heat exchanger inlet outlet temperatures: experimental vs. model

Figure 6.6  Heat exchanger inlet outlet temperatures: experimental vs. model
Figure 6.7 Heat pump evaporator inlet/outlet temperatures: experimental vs. model

Figure 6.8 Heat pump condenser inlet/outlet temperatures: experimental vs. model
-2.20 F for heating and from -6.96 to -14.00 F for cooling. For outlet temperature differences, the range was from -2.42 to -0.13 F for heating and from -13.86 to -0.95 F for cooling.

A comparison of experimental and model heat rates and power associated with the heat pump are shown in Figure 6.9 for heating data and in Figure 6.10 for cooling data. There is wide variability between experimental and model data. Much of this variability is due to the erratic performance of the heat pump. Because of this, the average experimental heat pump heat rates and power were used for comparison. For heating, the difference in heat capacity \( q_{\text{indoor}} \) ranged as high as 10.91 % while the power (Power) rate varied by as little as 4.65 %. The difference between experimental and model heat rates on the outdoor side of the heat pump \( q_{\text{outdoor}} \) on the other hand, ranged from -31.87 to 23.95 % (heating mode). For the cooling mode, the difference in cooling capacity \( q_{\text{indoor}} \) ranged from 30.10 to 34.06 %, the difference in the outdoor heat rate \( q_{\text{outdoor}} \) ranged from 16.23 to 17.52 % and the Power difference was from -39.82 to -38.50 %.

Once again, this wide variability was traced back to the fact that the heat pump was operating at nearly one third its rated capacity and that flow rates had to be finely balanced in order for the heat pump to operate within a range which prevented refrigerant floodback. Regardless of the heat pump performance, however, the temperatures provided a more accurate representation of the system model as they are the result of energy exchanges beyond the heat pump. Favorable temperature comparisons of model to experimental data suggested that the system model as developed, is a reasonable model of the cistern-based heat pump system.
Figure 6.9 Heat pump heat rates and power: experimental vs. model

Figure 6.10 Heat pump heat rates and power: experimental vs. model
6.5 Prediction Capabilities

The system model is configured such that various capacities for the heat pump may be considered for either the current heat exchanger-cistern arrangement or one of several other variations. Curve fit equations (Appendix D) for two EnerCon water-source heat pumps were programmed into the model. One is a 1.5 ton unit, the other is a 5 ton unit. Each were programmed for three distinct water-source flow rates, \( V_w \), for both heating and cooling. The 1.5 ton unit performed well with the system as currently configured except with the lowest water-source flow rate, 2.4 gpm, in the heating mode. After 117 hours of operation, the evaporator outlet temperature became 32.0°F. For the heating mode, the 5 ton unit had to be reconfigured to prevent evaporator outlet from freezing. Output for each unit is shown in Figures 6.11 through 6.18. Figures 6.11 and 6.13 demonstrate the influence of heat extraction on tank temperature for 1.5 ton and 5 ton units, respectively. Likewise, Figures 6.12 and 6.14 show the influence of exhaust heat on tank temperature for the 1.5 ton and 5 ton units. Unless indicated otherwise, the system configuration is that of the set up at the SNMRC which was as follows:

\[
\begin{align*}
D &= 10\ ft, \ \text{the diameter of the tank} \\
L &= 27\ ft, \ \text{the length of the cylindrical section of the tank} \\
H_x &= 530\ ft, \ \text{the length of the heat exchanger in the tank} \\
V_t &= 3.9\ gpm, \ \text{the flow rate of well water through the tank}
\end{align*}
\]

These variables were manipulated to allow for the 5 ton unit to operate in the heating mode. Any variations are shown on the appropriate graph. Heat pump capacity data are also tabulated in Table 6.1. Capacities listed are those which existed after ten days of continuous run time. For the heating mode, this period was from January 21 to 31 and for cooling, from July 21 to 31. Where capacities are not listed, the low flow rate of water through the heat exchanger created freezing conditions at the heat pump.
Figure 6.11 The influence of heat extraction on tank temperature - 1.5 ton unit

Figure 6.12 The influence of exhaust heat on tank temperature - 1.5 ton unit
Heating Mode

\[ D = 12 \text{ ft} \]
\[ L = 35 \text{ ft} \]
\[ H_x = 150 \text{ ft} \]
\[ V_t = 6 \text{ gpm} \]
\[ V_0 = 20 \text{ gpm} \]

Figure 6.13 The influence of heat extraction on tank temperature - 5 ton unit

Cooling Mode

\[ D = 10 \text{ ft} \]
\[ V_t = 3.9 \text{ gpm} \]
\[ t = 77 \text{ ft} \]
\[ V_0 = 18.9 \text{ gpm} \]
\[ H_x = 30 \text{ ft} \]

Figure 6.14 The influence of exhaust heat on tank temperature - 5 ton unit
Figure 6.15 Capacity compared to tank and water source temperatures - 1.5 ton unit

Figure 6.16 Capacity compared to tank and water source temperatures - 1.5 ton unit
Figure 6.17 Capacity compared to tank and water source temperatures - 5 ton unit

Figure 6.18 Capacity compared to tank and water source temperatures - 5 ton unit
evaporator outlet. Also tabulated are the coefficients of performance. Recall that these indicate the amount of heating/cooling provided for each unit of electrical energy input.

The serrated heat exchanger, heat rate curves in Figure 6.11 through 6.14 resulted when model heat exchanger inlet and outlet temperature differences became the same on a variable but recurring basis. The fact that tank temperatures began to level out in both heating and cooling modes indicated that even under continuous operation, the system approached a steady state condition where tank temperatures were reasonable for both continued heat pump operation and water consumption.

Figures 6.15 through 6.18 compare heating and cooling capacities to the inlet water source temperature. The effect upon the tank water temperature is also observed. Variations in system components are noted in Figure 6.17.

Table 6.1 Output from 1.5 and 5 ton units (after 10-day run)

<table>
<thead>
<tr>
<th>Unit</th>
<th>Flow rate (gpm)</th>
<th>HEATING</th>
<th>COOLING</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Capacity (MBtuh)</td>
<td>T\textsubscript{tank} (F)</td>
</tr>
<tr>
<td>1.5</td>
<td>2.4</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td></td>
<td>4.9</td>
<td>20.8</td>
<td>44.5</td>
</tr>
<tr>
<td></td>
<td>6.0</td>
<td>20.7</td>
<td>45.0</td>
</tr>
<tr>
<td>5.0</td>
<td>8.0</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td></td>
<td>15.9</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td></td>
<td>20.0</td>
<td>63.2*</td>
<td>40.3</td>
</tr>
</tbody>
</table>

* D = 12 ft. L = 35. Hx = 1530 ft. Vt = 6 gpm
6.6 Sensitivity Analysis of the Various System Heat Rates

The various heat rates depicted in Figure 6.2 were analyzed after adjusting the system parameters of the soil thermal diffusivity around the tank, $\alpha_s$, internal flow rate of the heat exchanger, $v_i$, and the total resistance of the tubing in the tank, $R_{tu}$. Individual heat rates were compared from treatment to treatment as well as to one another within treatments. Table 6.2 provides the conditions evaluated for the heating mode and Table 6.3 provides the conditions for the cooling mode. The 1.5 ton heat pump model was used for the purpose of generating the heat rates. Though not a heat rate, power (P) is included in the table. The units given are appropriate.

Table 6.2 is divided into four sections labeled for ease of discussion. Section No 1 contains the original system configuration with $R_{tu}$ listed as 0.05 ft$^2$ h F/Btu and $\alpha_s$ as 0.35 ft$^2$/day. A change in the tubing resistance to 0.04 ft$^2$ hr F/Btu, in No 2, to simulate non-laminar conditions on the outside of the heat exchanger coil, resulted in minor changes in heat rates. An increase of 50 % in the thermal diffusivity of the soil around the tank in section No 3, however, created dramatic increases in the rate of heat transfer from the tank as noted by the bolded heat rates corresponding to $\dot{q}_t$. The heat transfer from the tank for all three flow rates increased by 80 % from conditions listed in section No 1. Increasing the thermal diffusivity by 100 % in section No 4, resulted in an increase of 164 % in the tank heat transfer rate when compared to the conditions listed in No 1.

A comparison of heat rates within each treatment provided valuable information as to the relative importance each system component had regarding heat transfer. Considering the original configuration of section No 1 only, the greatest amount of heat transfer resulted from the tank heat rate, $\dot{q}_t$. The second highest was from the capacity of the heat pump, $\dot{q}_p$, followed by the heat rate to the heat pump evaporator, $\dot{q}_{pe}$, or the heat rate from the heat exchanger, $\dot{q}_{he}$, depending on which flow rate is considered. The heat transfer rate
Table 6.2 The effect of changes to system parameters on heat rates: heating mode

<table>
<thead>
<tr>
<th>Heat Rate (MBtu/h)</th>
<th>Volumetric Flow Rate (gpm)</th>
<th>Parameters Evaluated</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2.4</td>
<td>4.9</td>
</tr>
<tr>
<td>P</td>
<td>4.44</td>
<td>4.40</td>
</tr>
<tr>
<td>qo</td>
<td>16.16</td>
<td>16.27</td>
</tr>
<tr>
<td>qi</td>
<td>20.60</td>
<td>20.67</td>
</tr>
<tr>
<td>qt</td>
<td>-28.83</td>
<td>-57.36</td>
</tr>
<tr>
<td>qhx</td>
<td>16.00</td>
<td>16.37</td>
</tr>
<tr>
<td>qp1</td>
<td>-0.60</td>
<td>-0.46</td>
</tr>
<tr>
<td>qp2</td>
<td>0.68</td>
<td>0.18</td>
</tr>
<tr>
<td>qww</td>
<td>-4.60</td>
<td>-4.69</td>
</tr>
</tbody>
</table>

|                   | 4.46 | 4.43 | 5.60 | No. 2 |
| P                 | 16.11 | 16.20 | 15.46 |
| qo                | 20.57 | 20.62 | 21.05 |
| qi                | -28.77 | -57.20 | -86.12 |
| qhx               | 16.06 | 16.47 | 15.81 |
| qp1               | -0.66 | -0.55 | -0.53 |
| qp2               | 0.62 | 0.09 | 0.03 |
| qww               | -4.60 | -1.72 | -4.55 |

|                   | 4.44 | 4.40 | 5.58 | No. 3 |
| P                 | 16.17 | 16.28 | 15.34 |
| qo                | 20.60 | 20.68 | 20.92 |
| qt                | -51.76 | -103.22 | -155.36 |
| qhx               | 15.98 | 16.34 | 15.51 |
| qp1               | -0.59 | -0.44 | -0.44 |
| qp2               | 0.69 | 0.19 | 0.06 |
| qww               | -4.84 | -4.93 | -4.72 |
Table 6.2 The effect of changes to system parameters on heat rates; heating mode (continued)

<table>
<thead>
<tr>
<th>Heat Rate (MBtu/h)</th>
<th>Volumetric Flow Rate (gpm)</th>
<th>Parameters Evaluated</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2.4</td>
<td>4.9</td>
</tr>
<tr>
<td>P</td>
<td>4.44</td>
<td>4.40</td>
</tr>
<tr>
<td>q_o</td>
<td>16.18</td>
<td>16.29</td>
</tr>
<tr>
<td>q_i</td>
<td>20.61</td>
<td>20.68</td>
</tr>
<tr>
<td>q_t</td>
<td>-76.01</td>
<td>-151.72</td>
</tr>
<tr>
<td>Q_{hx}</td>
<td>15.96</td>
<td>16.32</td>
</tr>
<tr>
<td>q_{p1}</td>
<td>-0.58</td>
<td>-0.43</td>
</tr>
<tr>
<td>q_{p2}</td>
<td>0.70</td>
<td>0.20</td>
</tr>
<tr>
<td>q_{ww}</td>
<td>-5.10</td>
<td>-5.19</td>
</tr>
</tbody>
</table>

From/to the piping in the ground, \( \dot{q}_{p1} \) and \( \dot{q}_{p2} \) respectively, was relatively low by comparison to other system components.

When reviewing the original configuration of section No 1 for the cooling mode (Figure 6.3), the rate of heat transfer from the tank, \( \dot{q}_r \), is exceeded by the heat rate from the condenser, \( \dot{q}_{p} \), only once and at a flow rate of 2.4 gpm. Increasing the soil thermal diffusivity to 0.35 ft\(^2\)/day and then 0.70 ft\(^2\)/day, results in heat rate increases of 118 % and 243 %, respectively. As can be seen from the data, other heat rates remain essentially the same.
Table 6.3 The effect of changes to system parameters on heat rates: cooling mode

<table>
<thead>
<tr>
<th>Heat Rate (MBtuh)</th>
<th>Volumetric Flow Rate (gpm)</th>
<th>Parameters Evaluated</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2.4</td>
<td>4.9</td>
</tr>
<tr>
<td>P</td>
<td>4.72</td>
<td>4.78</td>
</tr>
<tr>
<td>qo</td>
<td>24.80</td>
<td>24.75</td>
</tr>
<tr>
<td>qi</td>
<td>20.08</td>
<td>19.97</td>
</tr>
<tr>
<td>qt</td>
<td>19.26</td>
<td>38.24</td>
</tr>
<tr>
<td>qhx</td>
<td>-23.61</td>
<td>-23.97</td>
</tr>
<tr>
<td>qp1</td>
<td>0.43</td>
<td>0.21</td>
</tr>
<tr>
<td>qp2</td>
<td>-1.51</td>
<td>-0.74</td>
</tr>
<tr>
<td>qww</td>
<td>6.53</td>
<td>6.63</td>
</tr>
</tbody>
</table>

| P                | 4.72 | 4.79 | 4.80 | No 2 |
| qo               | 24.80 | 24.75 | 24.74 |
| qi               | 20.08 | 19.96 | 19.93 | $R_{nw} = 0.05$ |
| qt               | 41.85 | 83.41 | 125.02 |
| qhx              | -23.58 | -23.94 | -23.93 | $\alpha = 0.50$ |
| qp1              | 0.42 | 0.20 | 0.14 |
| qp2              | -4.52 | 0.75 | -0.63 |
| qww              | 6.77 | 6.86 | 6.86 |

| P                | 4.73 | 4.79 | 4.81 | No 3 |
| qo               | 24.80 | 24.75 | 24.73 |
| qi               | 20.07 | 19.96 | 19.93 | $R_{nw} = 0.05$ |
| qt               | 65.85 | 131.43 | 197.05 |
| qhx              | -23.56 | -23.90 | -23.90 | $\alpha = 0.70$ |
| qp1              | 0.40 | 0.18 | 0.13 |
| qp2              | -1.52 | -0.76 | -0.64 |
| qww              | 7.02 | 7.11 | 7.11 |
6.7 Improvements to the Original Model

The importance of soil thermal diffusivity on the rate of heat transfer from the well water storage tank was established from the sensitivity analysis of section 6.6. Additional parameters considered to impact upon heat transfer were, the water level in the storage tank, and the flow rate of well water through the tank. In the original model, these two parameters were held constant. In an effort to determine possible improvements to the original model, two cyclic patterns of water use were evaluated along with three values of soil thermal diffusivity. One additional parameter, the thermal resistance of the tubing in the ground, $R_t$, was discovered to provide final adjustments to the model and bring it even closer to the experimental output. Table 6.4 provides a matrix of the various combinations evaluated. The two cycles of water use are referred as Model 1 and Model 2. Both are modifications of the original model. The initial condition of the tank was full as defined in Section 6.2, for each model. For Model 1, during the first 8 hours of operation, the tank was drained at 11.7 gpm—three times the average of 3.9 gpm—with no replacement well water. During the following 16 hours, the tank was restored to full capacity at a well water flow rate of 5.85 gpm—1.5 times the average 3.9 gpm. For Model 2 operation, the tank remained full until 6:00 a.m. For the next 14 hours, until 8:00 p.m., the tank discharged at a rate of 6.69 gpm with no recharge. Then, from 8:00 p.m. to 12:00 a.m., the tank was fully recharged. Model 1 and Model 2 water use cycles are depicted graphically in Appendix E, Figures E.3 and E.4, respectively. The Model 1 pattern was developed as a simple discharge/charge cycle to determine what kind of effect, if any, it had on heat pump system. Once it was established that cyclic water use had a considerable effect on the system, Model 2 was developed. Its pattern more accurately depicts the water use at the SNMRC where the tank is discharged
throughout most of the daytime and charging occurs relatively quickly (within a few hours) in the evening.

A matrix of the parameters adjusted with each of the two water use cycles is displayed in Table 6.4.

### Table 6.4 Matrix of parameters evaluated for system model improvement

<table>
<thead>
<tr>
<th>$R_g$</th>
<th>$\alpha_{te}$</th>
<th>Model 1</th>
<th>$\alpha_{te}$</th>
<th>Model 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.20</td>
<td>0.35</td>
<td>0.50</td>
<td>0.20</td>
</tr>
<tr>
<td>0.2</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>0.5</td>
<td>X</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
</tbody>
</table>

The graphical output from these parameters was compared to the tank and heat exchanger inlet temperature data - experimental versus model of Figure 6.3, based on the original model. Figures 6.19 and 6.20 demonstrate the influence Model 1 and $R_g$ when the thermal diffusivity is held constant at 0.20 ft$^2$/day (first column data in Table 6.4). The pairing of graphs continues in this manner resulting in six paired graphs, one pair for each column in the matrix of Table 6.4.

Upon close inspection of the graphs, it is clear that in each paired case, the higher thermal resistance of the piping in the ground brings the model data closer to the experimental. Within Model 1 data, a thermal diffusivity of 0.2 ft$^2$/day and a piping thermal resistance of 0.5 hr-ft$^2$-F/Btu provide the best fit to the experimental data as seen in Figure 6.20. Among Model 2 data, the same thermal diffusivity and thermal resistance as in Figure 6.20, provided the best fit to experimental (see Figure 6.26). A comparison between Figures 6.20 and 6.26 shows that Model 2 is a better model than Model 1 in predicting experimental
data temperature profiles.

Figures 6.31 through 6.33 add further support that cyclic water use is responsible for the variation in temperatures and energy rates within the system. A closer match to the experimental data is also evident when comparing Figures 6.31, 6.32 and 6.33 to Figures 6.5, 6.7 and 6.9, respectively.

This analysis reflects the importance of creating a model which closely characterizes the daily water use patterns and utilizes appropriate thermal properties to define the model. Soil thermal diffusivity and cyclic water patterns are most influential in developing an accurate model.
Figure 6.19 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use

Figure 6.20 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use
Figure 6.21 Tank and heat exchanger inlet temperature data; experimental vs. model for cyclic well water use

Figure 6.22 Tank and heat exchanger inlet temperature data; experimental vs. model for cyclic well water use
Heating Mode

Model 1
\( c_1 = 0.50 \)
\( R_g = 0.20 \)

Figure 6.23 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use

Heating Mode

Model 1
\( c_1 = 0.50 \)
\( R_g = 0.50 \)

Figure 6.24 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use
Figure 6.25 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use

Figure 6.26 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use
Figure 6.27 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use.

Figure 6.28 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use.
Figure 6.29 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use

Figure 6.30 Tank and heat exchanger inlet temperature data: experimental vs. model for cyclic well water use
Figure 6.31 Heat exchanger inlet/outlet temperature data: experimental vs. model for cyclic well water use

Figure 6.32 Heat pump inlet/outlet temperature data: experimental vs. model for cyclic well water use
Figure 6.33 Heat pump heat rates and power: experimental vs. model for cyclic well water use
7 IMPLICATIONS FOR A SWINE FACILITY

7.1 Thermal Environmental Requirements

Swine require differing levels of thermal comfort for various stages of growth and production. These stages are often broken down into such categories as 1) farrowing, 2) nursery, 3) growing and finishing, and 4) breeding and gestation. Two primary variables used to control the animal thermal environment are temperature and ventilation rate. Table 7.1, developed from material found in the Midwest Plan Service Structures and Environment Handbook (1987), shows a break down of these categories and provides design data for ventilation rate, winter room temperature, and supplemental heating requirements based on maturity. Weight and age categories are also provided.

<table>
<thead>
<tr>
<th>Animal Category</th>
<th>Age</th>
<th>Weight (lb)</th>
<th>Ventilation, cfm/hd</th>
<th>Winter Room Temp (F)</th>
<th>Supplemental Heat Btu/hr/hd</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sow and Litter</td>
<td>to 3 wks</td>
<td>to 12</td>
<td>20 80 500</td>
<td>80</td>
<td>4000</td>
</tr>
<tr>
<td>Prenursery pig</td>
<td>3-4 wks</td>
<td>12-30</td>
<td>2 10 25</td>
<td>85</td>
<td>350</td>
</tr>
<tr>
<td>Nursery pig</td>
<td>5-8 wks</td>
<td>30-75</td>
<td>3 15 35</td>
<td>75</td>
<td>350</td>
</tr>
<tr>
<td>Growing pig</td>
<td>75-150</td>
<td>7 24 75</td>
<td>60</td>
<td>600</td>
<td></td>
</tr>
<tr>
<td>Finishing pig</td>
<td>150-220</td>
<td>10 35 120</td>
<td>60</td>
<td>600</td>
<td></td>
</tr>
<tr>
<td>Gestating sow</td>
<td>325</td>
<td>12 40 150</td>
<td>60</td>
<td>1000</td>
<td></td>
</tr>
<tr>
<td>Boar</td>
<td>400</td>
<td>14 50 300</td>
<td>60</td>
<td>1000</td>
<td></td>
</tr>
</tbody>
</table>

Table 7.1 Design data for swine
For the sow and litter, two different temperature schemes are typically necessary. The sow is most comfortable at 60-65°F, while a newborn pig requires a dry, draft-free environment at a temperature from 90-95°F. And nursery conditions range from 85°F for a 3-week-old pig down to approximately 70°F for an 8-week-old pig (Midwest Plan Service, 1987). One approach to accommodate these separate conditions is to maintain the room air between 65 and 75°F and provide supplemental heat to the creep area. As shown in Table 7.1, supplemental heat is a design requirement for every category of swine where slotted floors are involved. It is also required for a sow and litter, and for the nursery pig where bedded or scraped floors are the floor condition. Many variables need to be controlled in managing the animal environment, this research focused on the potential for a cistern-based heat pump system to provide supplemental and space heating requirements for the Swine Nutrition and Management Research Center. In particular, the feasibility of using the heat pump system for farrowing and nursery environments were considered.

7.2 Thermal Environmental Evaluation

An analysis of the heating requirements for the farrowing and nursery units at the SNMRC was conducted. The space heating requirements were evaluated first followed by supplemental heating needs. In terms of determining the additional heating needed for space conditioning, a modified version of a formula from Albright (1990) was utilized:

\[
\dot{q}_s + \dot{q}_h + \dot{q}_{hl} + \dot{q}_{vi} = \dot{q}_b + \dot{q}_{vo}
\]

where

\[
\begin{align*}
\dot{q}_s &= \text{Sensible heat gain from animals, Btu/h} \\
\dot{q}_h &= \text{Additional sensible heat required to maintain indoor design temperature, Btu/h} \\
\dot{q}_{hl} &= \text{Sensible heat gain from heat lamps, Btu/h} \\
\dot{q}_{vi} &= \text{Sensible heat from inlet ventilation air, Btu/h} \\
\dot{q}_b &= \text{Space heating requirements, Btu/h} \\
\dot{q}_{vo} &= \text{Supplemental heat requirements, Btu/h}
\end{align*}
\]
\( q_{bh} \) = Sensible heat loss through building envelope, Btu/h

\( q_{v,a} \) = Sensible heat contained in the ventilation air exiting the building, Btu/h

This formula assumes negligible sensible heat gain from mechanical sources, from the sun, and from evaporation. Also, the loss of sensible heat through the floor, essentially a perimeter effect, is considered relatively insignificant. Of interest in this equation is the additional space heating, \( q_{bh} \), needed to maintain the thermal environment.

In addition to the sensible energy balance, a mass balance was used to account for moisture. It is written as

\[
\dot{m}_p + \dot{m}_{vi} = \dot{m}_{vo}
\]

where

\( \dot{m}_p \) = Rate of moisture produced within the building, lb_m/h

\( \dot{m}_{vi} \) = Rate at which moisture enters the building by ventilation air, lb_m/h

\( \dot{m}_{vo} \) = Rate at which moisture exits the building by ventilation air, lb_m/h

The farrowing and nursery units are configured as shown in Figure 7.1 (Huss et al. 1986). The farrowing and nursery building is 58 ft by 107 ft. There are four farrowing rooms each with nine crates mounted on raised woven wire floors. The nursery consists of four rooms each with 24.4 ft by 4 ft pens with woven wire flooring. The analysis assumed a maximum capacity of eight pigs per pen for a total of 768 pigs (35 lb average) in the nursery while the farrowing unit was assumed to have 36 sows (400 lb average) with litters. Heat lamps rated at 250 W were used for each sow and litter.

The indoor design conditions were a temperature of 70 F and a relative humidity of 70%, while for outdoor design these two conditions were -10 F and 80%, respectively. The thermal resistance for the walls was evaluated at 22 hr-ft²-F/Btu. The ceiling was considered to have a total thermal resistance of 40 hr-ft²-F/Btu.
Figure 7.1 Farrowing and nursery building
A minimum cold weather ventilation rate of 3024 cfm was determined using 20 cfm per head for sow and litter and 3 cfm per head for nursery pigs as recommended by Midwest Plan Service (1987). The mass balance yielded a ventilation rate of 3170 cfm to control moisture. Since this rate exceeded the cold weather minimum rate of 3024, it was used in the heat balance calculations. The thirty six heat lamps provided an additional 30,700 Btuh of heating to the building.

The final space heating requirement based upon the conditions listed was determined to be 106,700 Btuh. This rate is nearly 70% greater than the capacity available from the 5 ton unit configured as shown in Table 6.1. In an attempt to meet the needs of this space heating requirement, a 10 ton unit was incorporated into the system model. The well water storage tank size was increased to approximately 50,000 gallons--diameter of 15 feet and cylindrical length of 30 feet. Heat exchanger length and well water flow rate were then systematically increased and the heat pump capacity recorded after a period of continuous run of 10 days. The results are displayed in Table 7.2.

Table 7.2 10 ton heat pump*, heating capacity (MBtuh) as a function of heat exchanger length and well water flow rate

<table>
<thead>
<tr>
<th>HX Length (ft)</th>
<th>Well Water Flow Rate (gpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>4</td>
</tr>
<tr>
<td>1000</td>
<td>---</td>
</tr>
<tr>
<td>2000</td>
<td>---</td>
</tr>
<tr>
<td>3000</td>
<td>81.9</td>
</tr>
<tr>
<td>4000</td>
<td>82.7</td>
</tr>
<tr>
<td>5000</td>
<td>83.2</td>
</tr>
</tbody>
</table>

*Other parameters: D = 15 ft. L = 30 ft. Vo = 44.0 gpm
Ill-configured systems which resulted in freezing conditions at evaporator outlet were left blank. The greatest heat capacity provided by the 10 ton unit occurred with 5000 feet of heat exchanger coil and a 12 gpm well water flow rate. With this configuration, the 10 ton unit provided 91,200 Btuh of heating capacity. This was less than 15% below the space heating requirements estimated for the farrowing and nursery units combined. However, a well water flow rate of 12 gpm in the 50,000 gallon storage tank resulted in a turnover rate for the volume of water in the tank of less than 2.9 days. The well water usage rate at the SNMRC was recorded at approximately 4 gpm for the 20,000 gallon cistern resulting in a turnover rate of nearly 3.5 days. In light of this comparison, a 10 ton heat pump system operating at 12 gpm is considered to use an unreasonable amount of water and at least three times more than required at the SNMRC. Larger commercial swine facilities might require such water use, but they would most certainly have increased space heating needs as well and hence, require a heat pump size greater than 10 tons.

The supplemental heating requirements were considered separately from the space heating requirements. Essentially, supplemental heating is provided by heat pads located in the pens. Supplemental heat requirements for farrowing and nursery units at the Swine Nutrition and Management Research Center are outlined in Table 7.3.

<table>
<thead>
<tr>
<th>Animal Category</th>
<th>Number in Category</th>
<th>Supplemental Heat/Category</th>
<th>Total Energy Required</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sow and Litter</td>
<td>36</td>
<td>3000</td>
<td>108,000</td>
</tr>
<tr>
<td>Nursery Pig</td>
<td>768</td>
<td>350</td>
<td>268,800</td>
</tr>
</tbody>
</table>

A minimum heating capacity of 108,000 Btuh is difficult to achieve with a reasonably sized cistern-based heat pump system for reasons discussed previously. After considering several
system configurations, it appears evident that a cistern-based heat pump system can supply only a portion of the heating needs of a commercial swine facility.

An alternative evaluation considered the use of the heat pump system to provide energy for conditions other than design conditions. The 5 ton unit operating at a well water flow rate of 6 gpm. was used for the purpose of this evaluation. Its configuration, shown in Table 6.1., indicates a minimum output of 63,200 Btu/h heating which occurred after ten days of continuous operation. With cyclic operation, its capacity can range as high as 70,000 Btu/h. The building heat rates were evaluated starting at the design temperature of -10 F. The calculations continued up to 55 F by 5 degree increments. The results are shown in Figure 7.2 and indicate that the 5 ton heat pump can provide for the space heating needs of the farrowing and nursery building from 5 F to 30 F at which temperature heating is no longer required. Between the outdoor design temperature of -10 F and 5 F, additional heating would be necessary.

An additional consideration to be made is that the heat pump offers the advantage of being used for cooling as well as heating. Its use for cooling may prove helpful in maintaining consistent levels of growth and production which can suffer during period of heat stress. And perhaps more importantly, it may decrease or prevent the pig mortality rate which occurs during hot weather.

The review of literature found that air-conditioning was a marginal advantage, if at all, especially in light of the additional cost. Since the heat pump system as evaluated, has demonstrated the potential to provide for some of the heating needs of a swine facility, the additional costs for cooling are only those for electricity. And electrical costs are tempered by the heat pump's coefficient of performance, which range from 2.7 to 4.0 for the systems evaluated.
Building Heat Rates

\[ \dot{q}_h = \dot{q}_b + \dot{q}_v - \dot{q}_s - \dot{q}_l \]

Figure 7.2 SNMRC farrowing and nursery unit heat rates
8 CONCLUSIONS

8.1 Summary

The results of this research indicated that a cistern-based heat pump system as configured is energy efficient and feasible within certain limitations. Experimental data for both cooling and heating were collected. The COPs for heating and cooling were determined to be 1.39 and 1.06, respectively. The experimental heat exchanger effectiveness was calculated to be 0.71 for heating and 0.99 for cooling. Similarly, the numerical model predicted 0.72 and 1.00 for heating and cooling, respectively. Even though the capacity of the existing heat pump was less than desired, the data collected through experimentation provided substantial enough information to verify the system's mathematical model. The model, in turn, was used to evaluate three additional heat pump capacities, 1.5-ton, 5-ton and 10-ton. The existing cistern-heat exchanger configuration as well as system configurations involving the manipulation of the heat exchanger size, tank size and well water flow rate were evaluated.

The outcome of analysis indicated that while a 1.5-ton heat pump is well suited for the current 20,000 gallon cistern, a 5-ton heat pump required a larger tank and heat exchanger, and a higher well water flow rate in order for it to be viable for heating. The 10-ton heat pump, evaluated for the purpose of providing space heating or supplemental heating requirements of the SNMRC farrowing and nursery unit, was shown to provide as much as 75% of the heating needs but required a 50,000 gallon cistern and a well water flow rate approximately five times higher than the current flow rate to perform at that level.

At best, the cistern-based heat pump system appears to be capable of providing some of the supplemental heating needs of the SNMRC. Not taken into consideration, however, is the heat pump system's additional capability to provide cooling. It is expected that because
the heat pump system for heating alone is feasible, that utilizing the system for cooling purposes during periods of potential animal heat stress outweighs the added cost of electricity to operate the unit.

8.2 Future Considerations to Assist Numerical Model Improvement

Time and rate of well water use, soil thermal diffusivity, and thermal resistance of the piping in the ground have been identified as important parameters to consider in the development of a numerical model for a cistern-based heat pump system. It is recommended that future research in this area include the monitoring of the time and rate of well water use. Data regarding the level of water in the tank and the temperatures of the well water at inlet to and outlet from the tank should also be collected. And, because the thermal diffusivity of the soil was demonstrated to have a strong effect on heat transfer from the tank, an effort should be made to measure this property. Temperature probes in the ground at various depth around the tank would provide support for heat transfer calculations. Finally, cyclic operation of the heat pump should be considered for the purpose of determining the thermal resilience of the well water storage tank to operation other than continuous.

8.3 Future Considerations for a Commercial Swine Facility

The heat pump system discussed in the current research has been shown to be a viable source of supplemental heating for a commercial swine facility such as the SNMRC. It has, however, essentially evaluated only the heating capabilities of the system. The potential for use during periods of hot weather may be as important as (perhaps more important than) its use in cold weather. Because of the energy efficiency of the ground-coupled heat pump, it appears most reasonable that additional research focus on the utilization of cooling available from the cistern-based heat pump system. Consideration should be made for its use in
breeding and gestation units as well as farrowing and nursing. The influence on mortality rate would seem to be an important issue followed by the effect on growth and production rates.
REFERENCES


ACKNOWLEDGMENTS

I wish to thank all those who have shown their support for my pursuit of this degree. My wife, Marcia, and son, Rick, have been behind me 100%. Now it's their turn. Thanks also to Dr. Dwaine S. Bundy for being my major professor and ultimately having faith that I would accomplish my degree sooner than later. Finally, thanks to Dr. Steven J. Hoff who pushed me to evaluate my research to accomplish a polished dissertation.
APPENDIX A

CALIBRATION PROCEDURES
Rho Sigma Flow Meters

The Rho Sigma pulsing flow meters were calibrated by measuring the volume of water which flowed through them. The volume of water was measured 15 gallons at time in a tub calibrated to 15 gallons. Measurement proceeded until 105 gallons of water had passed through the flow meters. The manufacturer recommended a minimum of 100 gallons. During the process, pulse information recorded by the Campbell Scientific and was checked against analog output from the flow meter accuracy. It was the same. Also, the analog output from the flow meters was evaluated against the measured volume of water. For the room side flow meter, the measurements were 1.2% high while for the tank side flow meter, the measurements were 1.9% high. Because both were high, a multiplier based on the average percent greater than actual was used. A value of 0.985 was assigned the multiplier. No offset was used.

Thermocouples

The T-type thermocouples were calibrated by recording the temperature of the fluid in which they were immersed. Two fluid environments were used. The first fluid was air with the ambient temperature measured with a calibrated mercury thermometer at approximately 26.0 C. An ice bath was used as the second fluid. It was measured at temperature of approximately 0.1 C. The resulting temperature measurements were less than one percent off and fluctuated on each side of the calibration temperatures. Due to this small amount of variability and because the sixteen thermocouples were read through one channel of the CR10 from a multiplexer, a multiplier of 1.0 and no offset were programmed for the thermocouples.
**Watt Transducer**

The Ohio Semitronics watt transducer, with an output of 0 to 10 Vdc for a full scale input of 20 kW, was factory calibrated and checked 100% for voltage and current linearity, power factor, and initial set point. Temperature effects (-10 C to 60 C) on full scale output was listed as ± 0.1%.

**Voltage Divider**

A voltage divider utilizing four 1.5 K resistors was constructed according to the diagram shown below (Figure A.1). As stated in the text, the purpose of the voltage divider was to reduce the full scale voltage output of 10 Vdc from the watt transducer to correspond to a maximum input voltage of 2.5 Vdc acceptable for data acquisition by the Campbell Scientific CR10. The measured resistance for each resistor is included on the diagram.

![Voltage divider schematic](image)

**Figure A.1 Voltage divider schematic**
Equations:

\[ V = i R \quad \text{and} \quad V_x = i R_x \quad \Rightarrow \quad i = \frac{V_x}{R_x} \]
\[ V_y = i R_y \quad \Rightarrow \quad i = \frac{V_y}{R_y} \]

therefore

\[ \frac{V_x}{R_x} = \frac{V_y}{R_y} \]

or

\[ V_y = \frac{R_x V_x}{R_y} \]

substituting for \( V_x \)

\[ V_x = 10 - V_y \]

\( V_y \) can be written as

\[ V_y = \frac{R_y (10 - V_y)}{R_x} \]

Values for \( R_x \) and \( R_y \) are substituted in and yield the maximum voltage seen by the CR10 from a 10 Vdc source.

Nomenclature:

\[ V = \text{Voltage, volts (associated with x and y branches)} \]
\[ i = \text{Current, amps} \]
\[ R = \text{Resistance, ohms (associated with x and y branches)} \]

The circuit was calibrated using a dc voltage source. Voltage output across the \( V_y \) branch was recorded for voltage input ranging from 0.00 to 10.00 Vdc in one volt increments. The correlation coefficient, \( r \), was 1.00.

Due to the factory calibration of the watt transducer and the high correlation for the voltage divider circuit, no multiplier or offset were required for this channel in the CR10 program.
APPENDIX B

DATA ACQUISITION PROGRAM
Program: RICK7

Flag Usage:
Input Channel Usage:
Excitation Channel Usage:
Control Port Usage:
Pulse Input Channel Usage:
Output Array Definitions:

* 1 Table 1 Programs
  01: 5 Sec. Execution Interval

  01: P2 Volt (DIFF)
      01: 1 Rep
      02: 25 2500 mV 60 Hz rejection Range
      03: 6 IN Chan
      04: 40 Loc :
      05: 1.0 Mult
      06: 0 Offset

  02: P92 If time is
      01: 0 minutes into a
      02: 10 minute interval
      03: 10 Set high Flag 0 (output)

  03: P77 Real Time
      01: 110 Day, Hour-Minute

  04: P71 Average
      01: 1 Rep
      02: 40 Loc

  05: P86 Do
      01: 20 Set low Flag 0 (output)

  06: P End Table 1

* 2 Table 2 Programs
  01: 60 Sec. Execution Interval

  01: P3 Pulse
      01: 2 Reps
      02: 1 Pulse Input Chan
      03: 2 Switch closure
      04: 18 Loc :
      05: 0.985 Mult
      06: 0.0000 Offset

  02: P11 Temp 107 Probe
      01: 1 Rep
      02: 1 IN Chan
      03: 3 Excite all reps w/EXchan 3
      04: 1 Loc :
      05: 1 Mult
      06: 0.0000 Offset
Table 2

03: P14 Thermocouple Temp (DIFF)
   01: 1 Rep
   02: 21 2.5 mV 60 Hz rejection Range
   03: 4 IN Chan
   04: 1 Type T (Copper-Constantan)
   05: 1 Ref Temp Loc
   06: 42 Loc :
   07: 1 Mult
   08: 0.0000 Offset

04: P86 Do
   01: 44 Set high Port 4

05: P87 Beginning of Loop
   01: 00 Delay
   02: 16 Loop Count

06: P86 Do
   01: 73 Pulse Port 3

07: P14 Thermocouple Temp (DIFF)
   01: 1 Rep
   02: 21 2.5 mV 60 Hz rejection Range
   03: 3 IN Chan
   04: 1 Type T (Copper-Constantan)
   05: 1 Ref Temp Loc
   06: 2-- Loc :
   07: 1 Mult
   08: 0.0000 Offset

08: P95 End

09: P86 Do
   01: 54 Set low Port 4

10: P35 Z=X-Y
    01: 13 X Loc
    02: 12 Y Loc
    03: 20 Z Loc :

11: P35 Z=X-Y
    01: 15 X Loc
    02: 14 Y Loc
    03: 21 Z Loc :

12: P35 Z=X-Y
    01: 17 X Loc
    02: 16 Y Loc
    03: 22 Z Loc :

13: P33 Z=X+Y
    01: 12 X Loc
    02: 13 Y Loc
    03: 26 Z Loc :
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</table>

If time is
0 minutes into a
10 minute interval
Set high Flag 0 (output)

Real Time
Day, Hour-Minute
Page 5  Table 2

35:  P71  Average
   01: 21  Reps
   02:  2  Loc

36:  P71  Average
   01:  1  Rep
   02: 42  Loc

37:  P86  Do
   01: 20  Set low Flag 0 (output)

38:  P   End Table 2

*  3  Table 3 Subroutines

01:  P   End Table 3

*  A  Mode 10 Memory Allocation
   01: 42  Input Locations
   02: 64  Intermediate Locations
   03: 0.0000 Final Storage Area 2

*  C  Mode 12 Security
   01:  0  LOCK 1
   02:  0  LOCK 2
   03: 0000  LOCK 3

!
## Input Location Assignments (with comments):

**Key:**
- T = Table Number
- E = Entry Number
- L = Location Number

<table>
<thead>
<tr>
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<th>E:</th>
<th>L:</th>
</tr>
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| 2: | 7: | 2: Loc:
| 2: | 1: | 18: Loc:
| 2: | 10: | 20: Z Loc:
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| 2: | 27: | 20: Z Loc:
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| 2: | 25: | 21: Z Loc:
| 2: | 28: | 21: Z Loc:
| 2: | 31: | 21: Z Loc:
| 2: | 12: | 22: Z Loc:
| 2: | 26: | 22: Z Loc:
| 2: | 29: | 22: Z Loc:
| 2: | 32: | 22: Z Loc:
| 2: | 13: | 26: Z Loc:
| 2: | 14: | 27: Z Loc:
| 2: | 15: | 28: Z Loc: |
APPENDIX C

SYSTEM MODEL PROGRAM
HEAT PUMP SYSTEM MODEL
APRIL, 1994
JIM LEARY

THIS PROGRAM IS DESIGNED TO SIMULATE A GROUND-COUPLED HEAT
PUMP SYSTEM WHICH UTILIZES AN IN-GROUND WELL-WATER STORAGE
(CYLINDRICALLY SHAPED WITH HEMISPHERICAL END CAPS) AS A PRIMARY
SINK/SOURCE OF ENERGY.

THE MODEL ACCOUNTS FOR HEAT PUMP CAPACITY AND STORAGE TANK
SIZE AS WELL AS TUBING LENGTH FOR TUBING IN THE GROUND--
CONNECTING THE HEAT PUMP TO THE TANK--AND TUBING USED AS THE
HEAT EXCHANGER IN THE TANK.

AN OVERALL ENERGY BALANCE IS USED TO PREDICT ENERGY SUPPLIED
BY THE HEAT PUMP FROM THE SYSTEM, AND POWER USED.

INPUT VARIABLES

D = TANK DIAMETER (FT)
L = LENGTH OF CYLINDRICAL SECTION OF TANK (FT)
DTo = OUTSIDE DIAMETER OF TUBING (in)
LTG = LENGTH OF TUBING IN GROUND (FT)
LTW = LENGTH OF TUBING IN WATER (FT)
RT = TOTAL RADIAL RESISTANCE OF TUBING IN WATER (HR-FT^2-F/BTU)
RTG = TOTAL RESISTANCE FOR PIPE IN GROUND (HR-FT^2/BTU)
V0 = OUTDOOR SIDE OF HEAT PUMP VOLUMETRIC FLOW RATE (GPM)
Vi = INDOOR SIDE OF HEAT PUMP VOLUMETRIC FLOW RATE (GPM)
Vt = VOLUMETRIC FLOW RATE OF WELL WATER THROUGH TANK (GPM)
Th = MEAN ANNUAL GROUND TEMPERATURE (F)
STA = ANNUAL SURFACE TEMPERATURE AMPLITUDE (F)
DAM = DAY OF MAXIMUM ANNUAL TEMPERATURE
DAY = DAY OF THE YEAR TO EVALUATE GROUND TEMP. (JAN 1 = 1)
U1w = OVERALL HEAT TRANS COEF FOR CYLINDER IN AIR (BTU/H-FT^2-F)
U1r = OVERALL HEAT TRANS COEF FOR RIB OF CYLINDER IN AIR (BTU/H-FT^2-F)
U1EC = OVERALL HEAT TRANS COEF FOR ENDCAP IN AIR (BTU/H-FT^2-F)
U1w = OVERALL HEAT TRANS COEF FOR CYLINDER IN WATER (BTU/H-FT^2-F)
U1r = OVERALL HEAT TRANS COEF FOR RIB OF CYL IN WATER (BTU/H-FT^2-F)
U1EC = OVERALL HEAT TRANS COEF FOR ENDCAP IN WATER (BTU/H-FT^2-F)
AL1 = THERMAL DIFFUSIVITY OF SOIL AROUND TUBING (FT^2/DAY)
AL2 = THERMAL DIFFUSIVITY OF SOIL AROUND TANK (FT^2/DAY)
T12 = HEAT PUMP ENTERING WATER TEMPERATURE FROM ROOM (F)
T14 = INITIAL ESTIMATE OF H2O TEMP ENTERING HP FROM OUTSIDE (F)
DLTA = MAXIMUM TEMPERATURE DIF ALLOWED FOR T14=T14 (F)
TIME = THE NUMBER OF DAYS OVER WHICH TO RUN PROGRAM (DAY)

PROGRAM HPSYS
IMPLICIT DOUBLE PRECISION (A-H,O-Z)
CHARACTER*20 SFILE
DIMENSION TG(5),UAT(4)
REAL*8 L,LTG,LTW,Mo,Mi,Mt
INTEGER TIME
COMMON//T13,T14,T15,T16,TWI,TTW,ATW,ATG,RT,RTG,Mo,Mt,VLB

************* THE STATEMENT FUNCTION FOR GROUND TEMPERATURE *************

TE(H,AL,DAY) = Tm - STA*EXP(-H*SQR(PI/(365.0*AL)))*
& COS((2.0*PI)/365.0*(DAY-DAYO-(H/2.0)*SQR(365.0/(FI*AL))))

***********************************************************************

PRINT*, 'ENTER A "1" FOR HEATING OR A "2" FOR COOLING.'
READ*,LOAD
PRINT*,''
PRINT*, 'ENTER THE DAY TO START AND THE NUMBER OF DAYS TO RUN.'
READ*,DAY,TIME
PRINT*, 'ENTER THE NAME OF YOUR INPUT FILE.'
READ(5,10) SFILE
10 FORMAT(A20)

OPEN(UNIT=10, FILE=SFILE, STATUS='OLD')

READ(10,*)D,L,DTo,LTG,LTW,RT,RTG,SZ,Vi,Vt,Tm,STA,DAYO,U1W, &
U1R,U1EC,UW,UR,UEC,AL1,AL2,TWI,T12,T14,DLTA

PRINT*, 'UNIT SIZE IS: ',SZ
IF(SZ.EQ.1)THEN
  PRINT*, 'SELECT SOURCE WATER FLOWRATE (2.4, 4.9 or 6.0) GPM'
  READ*,Vo
ELSEIF(SZ.EQ.5)THEN
  PRINT*, 'SELECT SOURCE WATER FLOWRATE (8, 15.9, or 20) GPM'
  READ*,Vo
ELSEIF(SZ.EQ.10)THEN
  Vo = 44.0
ELSE
  PRINT*, 'ENTER THE SOURCE WATER FLOWRATE.'
  READ*,Vo
ENDIF

C FORMAT OUTPUT STATEMENT TITLES -- ECHO CHECK

OPEN(UNIT=6, FILE='TOUT.DAT', STATUS='NEW')

WRITE(6,20) D,L,DTo,LTG,LTW,RT,RTG,SZ,Vi,Vt,Tm,STA,DAYO,DAY,U1W, &
U1R,U1EC,UW,UR,UEC,AL1,AL2,TWI,T12,T14,DLTA,TIME
20 FORMAT(1X,'TANK DIAMETER: ',F4.1,' FT',/,&
1X,'LENGTH OF CYLINDRICAL SECTION: ',F4.1,' FT',/,&
1X,'OUTSIDE DIAMETER OF TUBING: ',F5.3,' in',/,&
1X,'LENGTH OF TUBING IN GROUND: ',F5.1,' FT',/,&
1X,'LENGTH OF TUBING IN TANK: ',F6.1,' FT',/,&
1X,'RESISTANCE OF TUBING IN WATER: ',F5.3,' H-FT^2-P/ BTU',/,&
1X,'RESISTANCE OF TUBING IN GROUND: ',F5.3,' H-FT^2-P/ BTU',/,&
1X,'HEAT PUMP SIZE IS: ',12,' TON',/,&
1X,'OUTDOOR SIDE HP FLOWRATE: ',F6.3,' GPM',/,&
1X,'INDOOR SIDE HP FLOWRATE: ',F4.1,' GPM',/,&
1X,'FLOWRATE OF WELL WATER: ',F4.1,' GPM',/,&
1X,'MEAN ANNUAL GROUND TEMPERATURE: ',F4.1,' F',/,
99

& IX, 'ANNUAL SURFACE TEMP AMPLITUDE: ', F4.1, ' F', /,
& IX, 'DAY OF MINIMUM SURFACE TEMP: ', F6.2, /,
& IX, 'DAY OF YEAR TO BEGIN EVALUATION: ', F6.2, /,
& IX, 'OVERALL HTC FOR CYL IN AIR: ', F6.4, ' BTU/H-FT^2-F', /,
& IX, 'OVERALL HTC FOR RIB IN AIR: ', F6.4, ' BTU/H-FT^2-F', /,
& IX, 'OVERALL HTC FOR ENDCAP IN AIR: ', F6.4, ' BTU/H-FT^2-F', /,
& IX, 'OVERALL HTC FOR CYL IN WATER: ', F6.4, ' BTU/H-FT^2-F', /,
& IX, 'OVERALL HTC FOR RIB IN WATER: ', F6.4, ' BTU/H-FT^2-F', /,
& IX, 'OVERALL HTC FOR ENDCAP IN WATER: ', F6.4, ' BTU/H-FT^2-F', /,
& IX, 'THERMAL DIFFUSIVITY AROUND TUBING: ', F4.2, ' FT^2/DAY', /,
& IX, 'THERMAL DIFFUSIVITY AROUND TANK: ', F4.2, ' FT^2/DAY', /,
& IX, 'TEMPERATURE OF WELL WATER: ', F4.1, ' F', /
& IX, 'INDOOR WATER TEMP ENTERING HEAT PUMP: ', F4.1, ' F', /,
& IX, 'OUTDOOR WATER TEMP ENTERING HEAT PUMP: ', F4.1, ' F', /,
& IX, 'MAXIMUM TEMP DIF FOR ABS(T14N-T14): ', F5.3, ' F', /
& IX, 'RUN PROGRAM FOR THIS LENGTH OF TIME: ', I2, ' DAYS', ///

*********** WRITE TITLES FOR OUTPUT ********************************************

IF (LOAD.EQ.1) THEN
WRITE(6,22)
ELSE
WRITE(6,25)
ENDIF
WRITE(6,27)SZ,Vo,LTW,LTG,ALl,AL2
WRITE(6,30)

******************** WRITE TITLES FOR OUTPUT ********************************************

* CALCULATE STORAGE TANK SURFACE AREAS *

PI = 3.14159
R = D/2.0
AC = PI*D*L ! CYLINDER SURFACE AREA
AS = 4.0*PI*(R**2.0) ! SPHERE SURFACE AREA

C*********** CALCULATE SECTIONAL AREAS OF CYLINDER & SPHERE **************

AC1 = AC/4.0 ! THESE REPRESENT LENGTHWISE
AC2 = AC1
AC3 = AC1
AC4 = AC1

AS2 = 2.0*PI*(R**2.0)*( COS(Pi/4.0) - COS(Pi/2.0) )
AS3 = AS2
AS1 = AS/2.0 - AS2
AS4 = AS1

C*********** CALCULATE OUTER AREA OF TUBING (IN SQ FT) ***************
ATG = PI*(DTo/12.0)*LTG ! SQ FT OF TUBING IN GROUND
ATW = PI*(DTo/12.0)*LTW ! SQ FT OF TUBING IN TANK

***************************************************************************
* CALCULATE 12 COMBINATIONS OF UAs FOR THE TANK
* UNITS ARE (BTU/HR-F)
***************************************************************************

UAC1R = U1R*AC1*0.318 ! 31.8% OF AREA IS RIBBED; AIR
UAC1W = U1W*AC1*0.682 ! 68.2% OF AREA IS WALL; AIR
UAS1  = U1EC*AS1       ! TOP SECTION OF SPHERE; AIR

UAC2R = UR*AC2*0.318 ! SECOND QUARTER RIBBED; WATER
UAC2W = UW*AC2*0.682 ! SECOND QUARTER WALL; WATER
UAS2  = UEC*AS2       ! SECOND QUARTER END CAP; WATER

UAC3R = UAC2R ! THIRD QUARTER SYMMETRIC TO
UAC3W = UAC2W ! SECOND QUARTER IN AREA AND
UAS3  = UAS2         ! OVERALL HT COEF PRODUCT; WATER

UAC4R = UR*AC4*0.318 ! FOURTH QUARTER RIBBED; WATER
UAC4W = UW*AC4*0.682 ! FOURTH QUARTER WALL; WATER
UAS4  = UAS4         ! FOURTH QUARTER END CAP; WATER

***************************************************************************
* CALCULATE UA TOTALS FOR THE FOUR QUADRANTS OF THE TANK
***************************************************************************

UAT(1) = UAC1R + UAC1W + UAS1
UAT(2) = UAC2R + UAC2W + UAS2
UAT(3) = UAC3R + UAC3W + UAS3
UAT(4) = UAC4R + UAC4W + UAS4

***************************************************************************
* CALCULATE MASS FLOWRATES
***************************************************************************

Mi = Vi*500.34 ! INDOOR (lbm/HR)
Mo = Vo*500.34 ! OUTDOOR (lbm/HR)
Mt = Vt*500.34 ! WELL WATER (lbm/HR)

***************************************************************************
* CALCULATE SOIL DEPTH AT THE MIDDLE OF EACH QUARTER SECTION OF THE TANK
***************************************************************************

Y = R*SIN((45.0/2.0)*PI/180.0)
D1 = 2.0 ! TOP OF TANK AT DEPTH OF 2 FT
D2 = D1+R-Y
D3 = D1+R+Y
D4 = D1+D
D6 = 6.0 ! TUBING IN GROUND IS 6 FT DEEP

***************************************************************************
* CALCULATE VOLUME OF WATER IN TANK ASSUMING THAT THE TOP QUARTER OF
* CYLINDER SURFACE AREA AND CORRESPONDING SPHERE AREA ARE IN AIR.
***************************************************************************

VC1 = (PI*(R**2.0)*L)/2.0 ! 1/2 OF CYLINDER VOLUME
AV1 = PI*(R**2.0)*45.0/360.0 ! 1/8 CYLINDER X-SECTIONAL AREA
X = ((R**2.0)/2.0)**0.5 ! HEIGHT OF H2O ABOVE MID-TANK
AV2 = X**2.0
AV3 = AV1
VC2 = (AV1+AV2+AV3)*L ! CYL VOL OF H2O ABOVE MID-TANK
VC = VC1+VC2 ! VOLUME OF H2O IN CYLINDER
VS1 = 2.0/3.0*PI*(R**3.0) ! 1/2 VOLUME OF SPHERE
VSU = PI*((R-X)**2.0)* (R-(R-X)/3.0) ! SPHERE VOL OF AIR
VS = VS1-VSU ! VOLUME OF WATER IN SPHERE
VT = VC+VS ! VOLUME OF WATER IN TANK (FT^3)
VSL = VSl-VSU ! 1/2 VOLUME OF SPHERE
VS = VS1-VSL ! VOLUME OF WATER IN SPHERE
VT = VC+VS ! VOLUME OF WATER IN TANK (FT^3)
VGAL = VT*7.4805 ! GALLONS: 7.4805 GAL/FT^3
VLBM = VT*62.4 ! MASS: 62.4 lbm/FT^3
PRINT*, 'VGAL= ',VGAL

**********************************************************************
* CALCULATE GROUND TEMPERATURES, TG(I), AS A FUNCTION OF SOIL
* DEPTH, SOIL THERMAL DIFFUSIVITY, AND DAY OF THE YEAR (H,AL,DAY).
* THESE ARE DETERMINED BY THE STATEMENT FUNCTION "TE(H,AL,DAY)"
* WHICH IS LOCATED BEFORE ANY EXECUTABLE STATEMENTS, JUST AFTER
* THE DECLARATION STATEMENTS.
**********************************************************************

DAYi = DAY
HR = 1.0

40 DO 50 I=1,5
AL = AL2
IF(I.EQ.1)THEN
  H=D1
  TG(I) = TE(H,AL,DAY)
ELSE IF(I.EQ.2)THEN
  H=D2
  TG(I) = TE(H,AL,DAY)
ELSE IF(I.EQ.3)THEN
  H=D3
  TG(I) = TE(H,AL,DAY)
ELSE IF(I.EQ.4)THEN
  H=D4
  TG(I) = TE(H,AL,DAY)
ELSE
  AL=AL1
  H=D6
  TG(I) = TE(H,AL,DAY)
END IF

50 CONTINUE

**********************************************************************
**** BALANCE TEMPERATURES IN THE LOOP BEFORE RUNNING HEAT PUMP ****
**********************************************************************

IF(DAY.EQ.DAYi)THEN
QHPO=0.0
60 CALL EBAL(HR,LOAD,QHPO,UAT,TG,T14N)
  DIF=ABS(T14N-T13)
  PRINT*,'T14N= ',T14N,'T13= ',T13
  IF (DIF.GT.DLTA) THEN
    T14=(T14N+T14)/2.0
    GOTO 60
  ENDIF
ENDDIF
ENDIF
90 IF(SZ.EQ.1)THEN
   GOTO 100
ELSEIF(SZ.EQ.5)THEN
   GOTO 110
ELSEIF(SZ.EQ.10)THEN
   GOTO 111
ELSE
   GOTO 112
ENDIF

100 CALL HPUMP(LOAD,Vo,T14,QHPI,P)
   GOTO 115
110 CALL HP5TON(LOAD,Vo,T14,QHPI,P)
   GOTO 115
111 CALL HP10TON(T14,QHPI,P)
   GOTO 115
112 CALL EXPTON(LOAD,Vo,T14,QHPI,P)

115 IF(LOAD.EQ.1)THEN
    QHPO = QHPI - P ! HEATING
  ELSE
    QHPO = QHPI + P ! COOLING
  ENDIF

  CALL EBAL(HR,LOAD,QHPO,UAT,TG,T14N)
  T14 = T14N

*************** CALCULATE HOURLY DATA ***********************

  QO = QHPO
  QI = QHPI
  PD = P

*************** OUTPUT RESULTS ********************************

  WRITE(6,120) DAY,HR,QI,QO,PD,T13,T14,T15,T16,TTW
120 FORMAT(IX,I3,3X,I3,3F8.0,5(2X,F5.1))

  IF(((DAY-DAYi).LT.TIME)THEN
     DAY = DAY + 1.0/24.0
     HR = HR + 1.0
  GO TO 40
  ENDIF

  CLOSE(UNIT=6)
  STOP
END

************************************************************************

* SUBROUTINE HPUMP FOR 1.5 TON HEAT PUMP
************************************************************************

SUBROUTINE HPUMP(LOAD,Vo,T14,QHPI,P)
REAL*6 Vo,T14,QHPI,P
INTEGER LOAD
IF((LOAD.EQ.1).AND.(Vo.EQ.2.4))THEN
   QHPI = 130.2*T14 + 14230.0 ! HEATING WITH
   P = (5.25*T14 + 1365.0)*3.412 ! Vo AT 2.4 GPM
ELSEIF((LOAD.EQ.1).AND.(Vo.EQ.4.9))THEN
    QHPI = 138.25*T14 + 14363.0  ! HEATING WITH
    P = (5.65*T14 + 1366.0)*3.412  ! Vo AT 4.9 GPM
ELSEIF((LOAD.EQ.1).AND.(Vo.EQ.6.0))THEN
    QHPI = 140.55*T14 + 14400.0  ! HEATING WITH
    P = (5.75*T14 + 1368.0)*3.412  ! Vo AT 6.0 GPM
ELSEIF((LOAD.EQ.2).AND.(Vo.EQ.2.4))THEN
    QHPI = -54.15*T14 - 22830.0  ! COOLING WITH
    P = (9.8*T14 + 868.0)*3.412  ! Vo AT 2.4 GPM
ELSEIF((LOAD.EQ.2).AND.(Vo.EQ.4.9))THEN
    QHPI = -53.65*T14 + 23062.0  ! COOLING WITH
    P = (8.9*T14 + 885.0)*3.412  ! Vo AT 4.9 GPM
ELSE
    QHPI = -53.6*T14 + 23167.0  ! COOLING WITH
    P = (8.55*T14 + 892.0)*3.412  ! Vo AT 6.0 GPM
ENDIF
RETURN
END

******************************************************************************
* SUBROUTINE HPSTON FOR A 5 TON HEAT PUMP
******************************************************************************

SUBROUTINE HPSTON(LOAD,Vo,T14,QHPI,P)
REAL*8 Vo,T14,QHPI,P
INTEGER LOAD
IF((LOAD.EQ.1).AND.(Vo.EQ.8.0))THEN
    QHPI = 394.8*T14 + 45425.0  ! HEATING WITH
    P = (25.25*T14 + 4218.0)*3.412  ! Vo AT 8.0 GPM
ELSEIF((LOAD.EQ.1).AND.(Vo.EQ.15.9))THEN
    QHPI = 422.4*T14 + 45592.0  ! HEATING WITH
    P = (27.65*T14 + 4187.0)*3.412  ! Vo AT 15.9 GPM
ELSEIF((LOAD.EQ.1).AND.(Vo.EQ.20.0))THEN
    QHPI = 457.35*T14 + 45761.0  ! HEATING WITH
    P = (32.05*T14 + 4119.0)*3.412  ! Vo AT 20.0 GPM
ELSEIF((LOAD.EQ.2).AND.(Vo.EQ.8.0))THEN
    QHPI = -174.05*T14 + 72541.0  ! COOLING WITH
    P = (37.4*T14 + 3085.0)*3.412  ! Vo AT 8.0 GPM
ELSEIF((LOAD.EQ.2).AND.(Vo.EQ.15.9))THEN
    QHPI = -174.4*T14 + 74828.0  ! COOLING WITH
    P = (34.4*T14 + 2830.0)*3.412  ! Vo AT 15.9 GPM
ELSE
    QHPI = -175.4*T14 + 75397.0  ! COOLING WITH
    P = (33.5*T14 + 2774.0)*3.412  ! Vo AT 20.0 GPM
ENDIF
RETURN
END

******************************************************************************
* SUBROUTINE HP10TON
******************************************************************************

SUBROUTINE HP10TON(T14,QHPI,P)
REAL*8 T14,QHPI,P

    QHPI = 31231.7 + 1431.0*T14  ! HEATING AT Vo=44gpm
    P = (4321.67 + 102.5*T14)*3.412
RETURN
END

******************************************************************************
SUBROUTINE EXPTON
***********************************************************************
SUBROUTINE EXPTON(LOAD,V0,T14,QHPI,P)
REAL*8 V0,T14,QHPI,P
INTEGER LOAD

IF(LOAD.EQ.1) THEN ! HEATING
QHPI = -1.32906E4 + 420*T14
P = -1344.47 + 152.897*T14
ELSE
QHPI = -125.0*T14 + 12500.0 ! COOLING
P = 51.9868*T14 + 899.35 ! COOLING POWER (BTU/HR)
ENDIF
RETURN
END
***********************************************************************

SUBROUTINE EBAL
***********************************************************************
SUBROUTINE EBAL(HR,LOAD,QHPO,UAT,TG,T14N)
IMPLICIT DOUBLE PRECISION (A-H,O-Z)
INTEGER I
REAL*8 Mo,Mt,MCR
DIMENSION UAT(4),TG(5)
COMMON//T13 ,T14,T15,T16,TWI,TTW,ATW,ATG,RT,RTG,Mo,Mt,VLM

UA = 0
UATG = 0

DO 60 I= 1,4
UA = UA + UAT(I) ! SUM OF UA-PRODUCTS
UATG = UATG + UAT(I)*TG(I) ! SUM OF UA(I)*TG(I)
60 CONTINUE

CP = 1.001 ! BTU/lbm-F
MCR = Mo*CP*RT ! FT^2
MCRG = Mo*CP*RTG ! FT^2

********** T13 IS THE EXIT TEMP TO THE OUTSIDE FROM THE HP **********
IF(LOAD.EQ.1) THEN
T13 = T14 - QHPO/(Mo*CP)
ELSE
T13 = T14 + QHPO/(Mo*CP)
ENDIF

********** T16 IS THE ENTRANCE TEMP TO THE HX COIL IN THE TANK ********
T16 = (T13-TG(5))*EXP(-ATG/MCRG) + TG(5)

********** CALCULATE THE TEMPERATURE OF THE TANK WATER **********
C
TTW = (Mt*CP*TWI+UATG+Mo*CP*T16+(1.0-EXP(-ATW/MCR)))/
&  (Mt*CP-UA+Mo*CP*(1.0-EXP(-ATW/MCR)))
TTW = (TWI+(Mo*CP*T16*(1.0-EXP(-ATW/MCR))+UATG)*
& ((1-EXP((-Mt*HR)/VLBM))/(Mt*CP)) / 
& (1+ (UA-Mo*CP*(EXP(-ATW/MCR)-1.0))*(1.0-EXP((-Mt*HR)/VLBM))) / 
& (Mt*CP))

********** T15 IS THE EXIT TEMP FROM THE HX COIL IN THE TANK **********
T15 = (T16-TTW)*EXP(-ATW/MCR) + TTW

****************** T14N IS A NEW T14--ENTERS HP OUTDOOR SIDE ************
T14N = (T15-TG(5))*EXP(-ATG/MCRG) + TG(5)

RETURN
END
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APPENDIX D

EQUATION DEVELOPMENT AND CURVE FITS FOR SYSTEM MODEL
The mathematical model was based on the energy gains and losses throughout the system. Those gains and losses, depicted in Figure D.1 as heat rates, \( q \), are described below.

\[ q_t = \rho V_{gw} c (T_1 - T_{gw}) \] energy gained/lost by well water flow rate

\[ q_{hpo} = \rho V_o c (T_4 - T_5) \] energy to/from the outdoor side of the heat pump

\[ q_{hpi} = \rho V_i c (T_7 - T_6) \] energy to/from the indoor side of the heat pump

\[ q_{hx} = \rho V_o c (T_3 - T_2) \] energy to/from the heat exchanger

\[ q_{g1} = U_p A_p (T_{g1} - T_p) \] heat exchange between the piping and the ground.

\[ q_{g2} = U_p A_1 (T_{g2} - T_1) \] heat exchange between the tank and the ground

\textit{Power} = the power consumed by the heat pump and circulating pumps
Figure D.1 Energy gains/losses and other parameters used in mathematical model development
Formulas used in the development of the heat pump system model follow.

\[ T_1 = T_s - \frac{\dot{q}_{ex}}{\dot{m}_o c_p} \]

\[ T_2 = (T_4 - T_{g1})\exp\left(\frac{A_g}{\dot{m}_o c_p R_g}\right) + T_{g1} \]

\[ T_{gw} + \dot{m}_o c_p T_2[1 - \exp\left(-\frac{A_w}{\dot{m}_o c_p R_w}\right)] + \Sigma U A T_{g2}\left\{1 - \exp\left(-\frac{\dot{m}_t}{m_t}\right)\right\} \]

\[ T_t = \frac{T_{gw} + \dot{m}_o c_p T_2[1 - \exp\left(-\frac{A_w}{\dot{m}_o c_p R_w}\right)] + \Sigma U A T_{g2}\left\{1 - \exp\left(-\frac{\dot{m}_t}{m_t}\right)\right\}}{\dot{m}_i c_p} \]

\[ T_3 = (T_2 - T_t)\exp\left(-\frac{A_w}{\dot{m}_o c_p R_w}\right) + T_t \]

\[ T_{5n} = (T_3 - T_{g1})\exp\left(-\frac{A_2}{\dot{m}_o c_p R_k}\right) + T_{g1} \]
Also used were the following formulas:

\[ q_{hpo} = q_{hpi} - P \quad \text{(Heating Mode)} \]
\[ q_{hpo} = q_{hpi} + P \quad \text{(Cooling Mode)} \]

**Curve Fit Equations**

**Experimental Heat Pump Unit**

**Heating**

\[ q_{hpi} = -1.32906E4 + 420*T_5 \quad \text{(Btu/h)} \]
\[ P = -1344.47 + 152.897*T_5 \quad \text{(Btu/h)} \]

**Cooling**

\[ q_{hpi} = -125*T_5 + 12500 \quad \text{(Btu/h)} \]
\[ P = 51.9868*T_{14} + 899.35 \quad \text{(Btu/h)} \]

**1.5 Ton Model Unit**

**Heating** (2.4 gpm water flow rate)

\[ q_{hpi} = 130.2*T_5 + 14230 \quad \text{(Btu/h)} \]
\[ P = (5.25*T_5 + 1365)*3.412 \quad \text{(Btu/h)} \]

**Heating** (4.9 gpm water flow rate)

\[ q_{hpi} = 138.25*T_5 + 14363 \quad \text{(Btu/h)} \]
\[ P = (5.65*T_5 + 1366)*3.412 \quad \text{(Btu/h)} \]

**Heating** (6.0 gpm water flow rate)

\[ q_{hpi} = 140.55*T_5 + 14400 \quad \text{(Btu/h)} \]
\[ P = (5.75*T_5 + 1368)*3.412 \quad \text{(Btu/h)} \]
Cooling (2.4 gpm water flow rate)

\[ q_{hpi} = -54.14 \times T_5 + 22830 \]  (Btu/h)

\[ P = (9.8 \times T_5 + 868) \times 3.412 \]  (Btu/h)

Cooling (4.9 gpm water flow rate)

\[ q_{hpi} = -53.65 \times T_5 + 23062 \]  (Btu/h)

\[ P = (8.9 \times T_5 + 885) \times 3.412 \]  (Btu/h)

Cooling (6.0 gpm water flow rate)

\[ q_{hpi} = -53.6 \times T_5 + 23167 \]  (Btu/h)

\[ P = (8.55 \times T_5 + 892) \times 3.412 \]  (Btu/h)

5 Ton Model Unit

Heating (8.0 gpm water flow rate)

\[ q_{hpi} = 394.8 \times T_5 + 45425 \]  (Btu/h)

\[ P = (25.25 \times T_5 + 4218) \times 3.412 \]  (Btu/h)

Heating (15.9 gpm water flow rate)

\[ q_{hpi} = 422.4 \times T_5 + 45592 \]  (Btu/h)

\[ P = (27.65 \times T_5 + 4187) \times 3.412 \]  (Btu/h)

Heating (20.0 gpm water flow rate)

\[ q_{hpi} = 470.85 \times T_5 + 45761 \]  (Btu/h)

\[ P = (32.05 \times T_5 + 4119) \times 3.412 \]  (Btu/h)

Cooling (8.0 gpm water flow rate)

\[ q_{hpi} = -174.05 \times T_5 + 72541 \]  (Btu/h)

\[ P = (37.4 \times T_5 + 3085) \times 3.412 \]  (Btu/h)
Cooling (15.9 gpm water flow rate)

\[ q_{hp} = -174.4T + 74828 \quad \text{(Btu/h)} \]
\[ P = (34.4T + 2830) \times 3.412 \quad \text{(Btu/h)} \]

Cooling (20.0 gpm water flow rate)

\[ q_{hp} = -175.4T + 75397 \quad \text{(Btu/h)} \]
\[ P = (33.8T + 2774) \times 3.412 \quad \text{(Btu/h)} \]

10 Ton Model Unit

Heating (44 gpm water flow rate)

\[ q_{hp} = 1431T + 31231.7 \quad \text{(Btu/h)} \]
\[ P = (102.5T + 4521.67) \times 3.412 \quad \text{(Btu/h)} \]
APPENDIX E

STORAGE TANK CHARACTERISTICS
USED FOR SYSTEM MODEL DEVELOPMENT
The composite rib construction of the Owens-Corning well water storage tank is shown in Figure E.1.

![Composite Rib Construction Diagram](image)

**Figure E.1** Storage tank composite rib construction

In developing the system model, the rib section was evaluated as a rectangular cross section as shown in Figure E.2 where the length dimension was taken as one half of the sum of the base and top lengths of the trapezoid.

![Rectangular Cross-Section Diagram](image)

**Figure E.2** Rectangular cross-section used to development tank model
The overall heat transfer coefficients were then determined using the formulas below for the various sections of the tank.

\[
U_\text{r} = \left( \frac{1}{h_{\text{wi}}} + \frac{2t_f}{k_f} + \frac{t_a}{k_a} + \frac{2}{h_{\text{ri}}} \right)^{-1} \quad \text{Rib section of tank}
\]
\[
U_\text{w} = \left( \frac{1}{h_{\text{wi}}} + \frac{t_f}{k_f} \right)^{-1} \quad \text{Plain wall section of tank}
\]
\[
U_\text{e} = \left( \frac{1}{h_{\text{wi}}} + \frac{t_f}{k_f} \right)^{-1} \quad \text{End cap section of tank}
\]

where

- \( h_{\text{ri}} \) = Heat transfer coefficient on each surface inside the rib. Btu/h ft\(^2\) F
- \( h_{\text{wi}} \) = Heat transfer coefficient on the inside wall of the tank (air/water). Btu/h ft\(^2\) F
- \( k_a \) = Conductance of the air. Btu/h ft F
- \( k_f \) = Conductance of the fiberglass. Btu/h ft F
- \( t_a \) = Depth of air inside the rib. ft
- \( t_f \) = Thickness of the fiberglass. ft

There are six overall heat transfer coefficients for the tank. One for each type of tank cross-section—ribbed, nonribbed and end cap—and two fluid conditions—water and air—for each of the three tank sections.

The overall heat transfer values calculated and used in the system model are displayed in Table E.1.
Table E.1 Overall heat transfer coefficients for various tank sections (Btu/h-ft²-F)

<table>
<thead>
<tr>
<th></th>
<th>Ribbed</th>
<th>Non Ribbed</th>
<th>End Cap</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>0.037</td>
<td>0.234</td>
<td>0.197</td>
</tr>
<tr>
<td>Water</td>
<td>0.041</td>
<td>0.747</td>
<td>0.467</td>
</tr>
</tbody>
</table>
Water Use Cycles—Models 1 and 2

Figure E.3 Model 1 water use cycle

Figure E.4 Model 2 water use cycle