COMPUTER MODELING OF HEAT PUMP WATER HEATERS IN RESTAURANT APPLICATIONS

by

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Abstract

Restaurants using conventional water heaters have annual hot water heating costs of between two and ten thousand dollars. Recently, some restaurants have been using heat pump water heaters in an attempt to reduce this expense. Heat pump water heaters are ideally suited for this application, since restaurant kitchens have both a high demand for hot water and a nearly year round space cooling load. Experiments have measured the energy used by conventional and heat pump water heaters and have shown that the water heating cost could be reduced by up to 70 percent by using the heat pump water heaters. An additional benefit produced by the heat pump is air cooling for the kitchen. This cooling is a potential space conditioning savings which should be included in a cost comparison to conventional heaters.

In order to account for savings by both water heating and space cooling, a general model of a heat pump water heater was developed which accounts for changes in performance caused by changes in the condenser and evaporator environments. Individual models were made of the water storage tank and of the compressor, condenser, evaporator and capillary tubes of the vapor compression. All models were developed from fundamental equations. This model was used in conjunction with a restaurant building model to determine the annual performance.

As expected, the heat pump space cooling was found to be an air conditioning

savings during summer months and a heating requirement during winter months. Both the annual space conditioning savings and water heating savings of the HPWH were found to be very site specific. The coefficient of performance of the heat pump was found to be between 2 and 3 and largely dependent upon the water-draw schedule. The combined annual water heating and space conditioning savings of the HPWH was found to always be a net savings when compared to an electric resistance water heater, but that it could be either a net savings or loss when compared to a gas water heater.

Acknowledgements

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Chapter 1
Introduction

1.1 Background

Restaurants using electric resistance water heaters have hot water heating expenses of between \$2,000 and \$20,000 annually, using between 400 and 4,000 gallons of hot water daily. Additionally, restaurant kitchens have space cooling requirements over a large portion of the year. Heat pump water heaters (HPWH) in this application can supply the required water heating, and in the process also meet part of the cooling load. For a heat pump water heater with a coefficient of performance of 3, the annual water heating savings would be between \$1,300 and \$13,000, when compared to electric resistance heaters.

A heat pump is similar in operation to an air conditioner in that both transfer heat from a cold to a hot reservoir using the same thermodynamic cycle, a vapor compression cycle. Figure 1.1 shows a schematic of a typical vapor compression cycle, which consists of four components: compressor, condenser, expansion device, and evaporator. This cycle will be discussed in detail in chapter 2 but is mentioned here to point out that this cycle absorbs energy (Q_{evap}) from a low temperature reservoir, transfers energy (Q_{cond}) to a high temperature reservoir, and requires work (W). For a cycle having no losses to the environment from the compressor, expansion device, or interconnecting pipes, a first law balance on the cycle gives

$$Q_{cond} = Q_{evap} + W. \tag{1.1.1}$$

In pratice the expansion device and interconnecting pipes are well insulated and have negligible losses, but the compressor can have losses to the environment as high as 50% of the electric motor work (W) of the compressor. The difference between a heat pump and an air conditioner is simply the choice of which energy flow is utilized: a heat pump uses the heat rejection of the condenser (Q_{cond}) for heating, while an air conditioner uses the heat absorption of the evaporator (Q_{evap}) for cooling. In general, the vapor compression cycle is referred to as a refrigeration

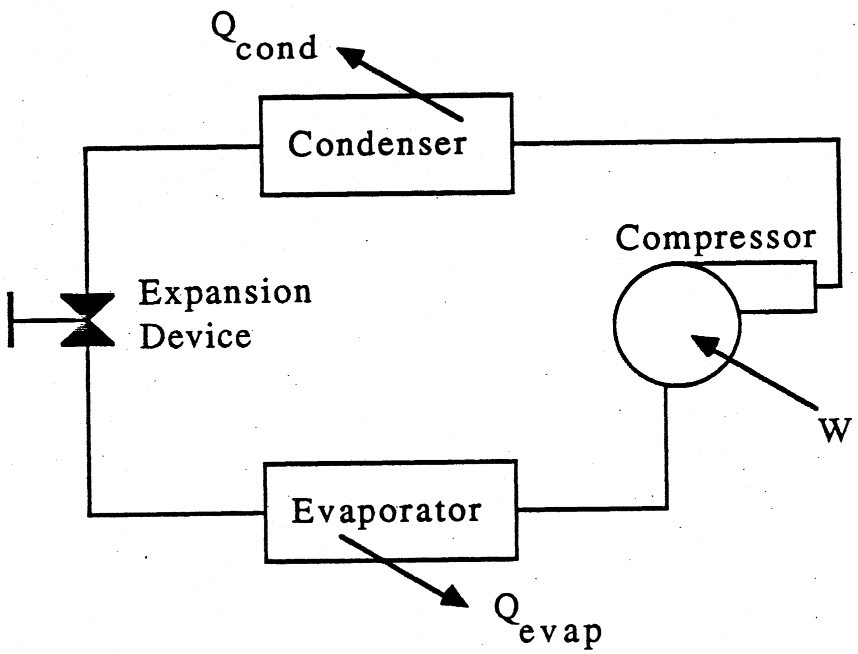

Figure 1.1 Schematic of A Typical Vapor Compression Cycle.

cycle when used for cooling and a heat pump cycle when used for heating, or for both heating and cooling. The performance of the vapor compression cycle is defined in terms of a coefficient of performance (COP), which is the ratio of useful energy transfer to compressor work. The coefficient of performance for the air conditioner is

$$COP_{a/c} = Q_{evap} / W \tag{1.1.2}$$

and for the heat pump it is

$$COP_{hp} = Q_{cond} / W. \tag{1.1.3}$$

For the cycle having no losses to the environment, the two COP's are related by

$$COP_{hp} = COP_{a/c} + 1. \tag{1.1.4}$$

In general, Q_{cond} and Q_{evap} are an unused energy transfer for the air conditioner and heat pump respectively. An ideal process for a vapor compression cycle would be one that uses Q_{cond} for heating and Q_{evap} for cooling. The application of a heat pump water heater in a restaurant kitchen is an attempt to utilize this ideal process: Q_{cond} can be used to heat water while Q_{evap} provides cooling to the kitchen.

1.2 Previous Work

Previous studies by the Dairy Equipment Company (DEC), a major heat pump water heater manufacturer, have compared the savings/losses of the heat pump water heater in restaurants to conventional electric or gas water heaters based solely on the water heating; no account has been made of the space conditioning savings/losses due to the air cooling effect of the heat pump. These studies experimentally measured the energy consumed by each water heater in several restaurant sites. For the restaurants considered, the heat pump water heaters had a coefficient of performance ranging between 2 and 3, giving a payback period (initial cost/annual savings) of 6 months to 3 years compared to electric water heaters and 2 to 3 years compared to gas water heaters. It was also noted that, since some states require gas water heaters to be within a fireproof room, this initial cost could be eliminated by using a heat pump water heater.

Computer models of the steady state performance of heat pumps have been developed by Domanski (1982), Krakow (1987) and others. Both Domanski's and Krakow's heat pump models consist of individual models of the heat pump components. Domanski develops very detailed and accurate models of each component which require detailed inputs of each component and considerable computer time, while Krakow's is less detailed, but is more suitable for mini and microcomputers. These models were developed specifically for predicting the performance of heat pumps being used for space heating and cooling, and are not directly applicable to water heating systems.

1.3 Purpose of Project

The goal of the project is to predict the savings/losses associated with using heat pump water heaters instead of conventional gas or electric water heaters in restaurant applications, with emphasis on the savings/losses associated with the heat pump cooling. Fundamental models of the heat pump water heater and restaurant are to be developed in order to predict the savings for restaurants having various space conditioning and hot water loads. The model includes the savings/losses of both the water heating and space cooling produced by the heat pump. In a second phase of this study, but not reported here, experimental measurements of the air conditioner and water heating savings/losses will be made at several restaurant sites. These measurements will be used to verify and improve the fundamental models, allowing the models to be confidently used to predict the savings for other restaurants.

1.4 Overview of Remaining Chapters

In the chapters 2 and 3, first principles models are developed of a vapor compression cycle, a water storage tank, and of the combined heat pump water heater. The combined model predicts the compressor work required, along with the water heating and space cooling produced. The performance of the combined HPWH is studied in the later part of chapter 3. A restaurant building model is developed in Chapter 4, which is used to estimate space cooling and heating loads. These loads are used to determine whether the cooling produced by the heat pump is a net air conditioning savings or a net heating expense. The heat pump water heater and building models are combined in Chapter 5 in a simulation program, which calculates the annual performance and net saving of the heat pump by calculating the hourly heat pump performance, space conditioning load, water heating savings, and air conditioning savings or heating expense, for each hour of the year. In Chapter 6, results from this simulation program are given for three typical restaurants. Also included are results from parametric studies of the heat pump water heater's performance and results from a life cycle savings analyses. Finally, in Chapter 7 conclusions and recommendations are given.

Chapter 2
System and Component Models of a Vapor Compression Cycle

There are many types and sizes of heat pumps; all, however, are similar to the schematic previously shown in Figure 1.1. To accurately predict the annual performance of a heat pump water heater, a model is required which will account for changes in performance caused by varying condenser or evaporator environments (i.e., reservoir temperatures). A complete model of a heat pump water heater will include a model of both the vapor compression cycle and of the water storage tank. The vapor compression cycle model will be further subdivided into models of each of its components: compressor, condenser, expansion device, and evaporator. The following paragraphs give a general description of the DEC heat pump water heater; the remainder of the chapter describes the individual component models and finally the heat pump water heater as a whole.

2.1 Description of the DEC Vapor Compression Cycle

A schematic of the DEC heat pump water heater is given in Figure 2.1. This schematic has two additional components compared to the schematic of the standard heat pump previously shown: a second condensing unit and an accumulator.

In the DEC heat pump cycle the compressor is a reciprocating compressor, the

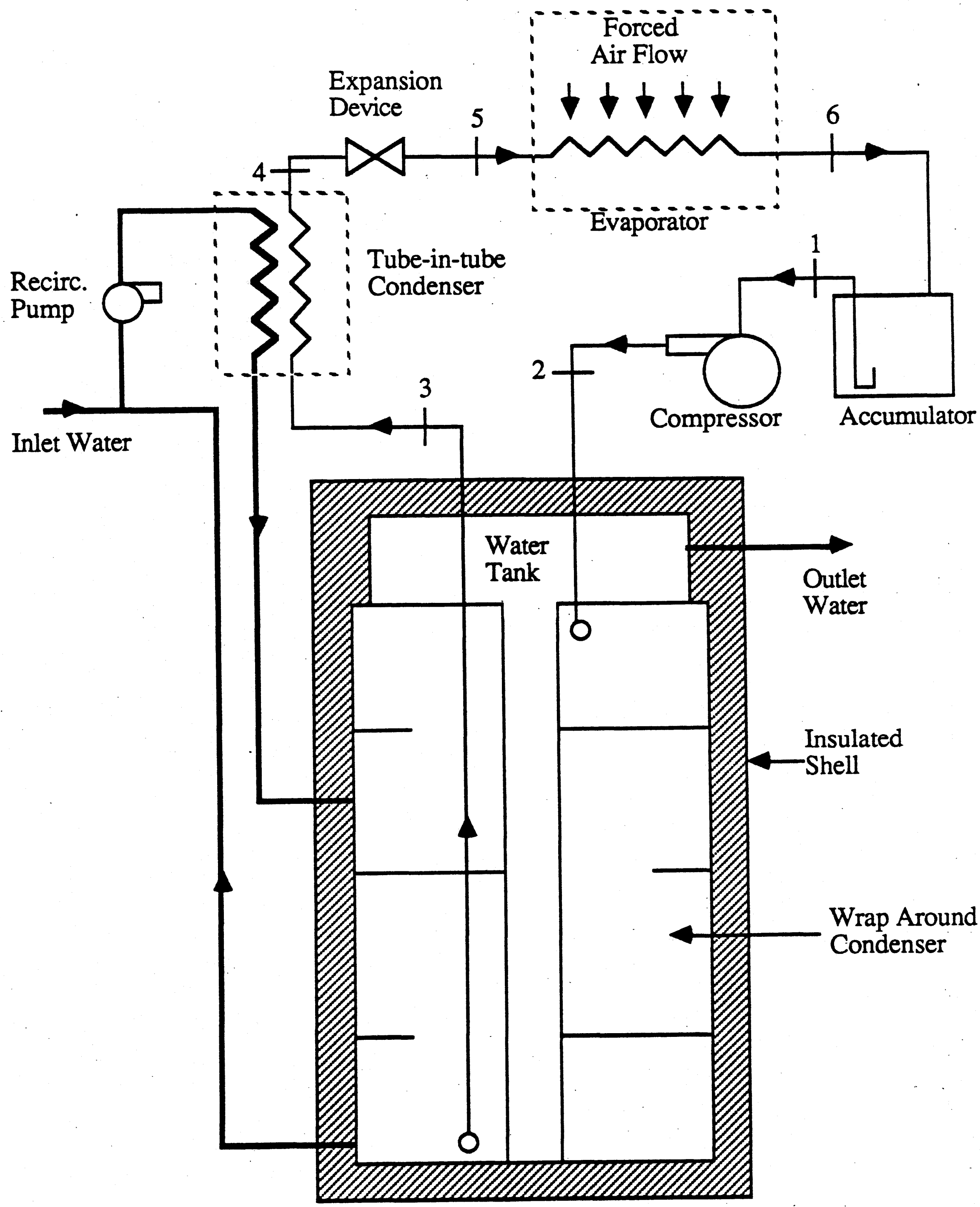

Figure 2.1 Schematic of the vapor compression cycle fot the DEC HP-120-27 water heater. Bold lines are water flow and other lines are refrigerant flow.

first condenser is a wrap-around heat exchanger on the storage tank, the second condenser is a water-to-refrigerant counterflow tube-in-tube heat exchanger, the expansion device is a capillary tube, and the evaporator is an air-to-refrigerant crossflow heat exchanger. The inlet water for the counterflow condenser is either the supply water or water recirculated from the bottom of the storage tank, depending upon whether hot water is being drawn. The storage tank volume of all DEC water heaters is 120 gallons.

A pressure-enthalpy plot of the DEC vapor compression cycle is given in Figure 2.2. The plot shows a saturated vapor at low pressure entering the compressor at state 1. The compressor performs work on the vapor during the compression, thereby superheating the vapor and increasing the pressure to the conditions at state 2. The vapor is desuperheated and partially condensed in the

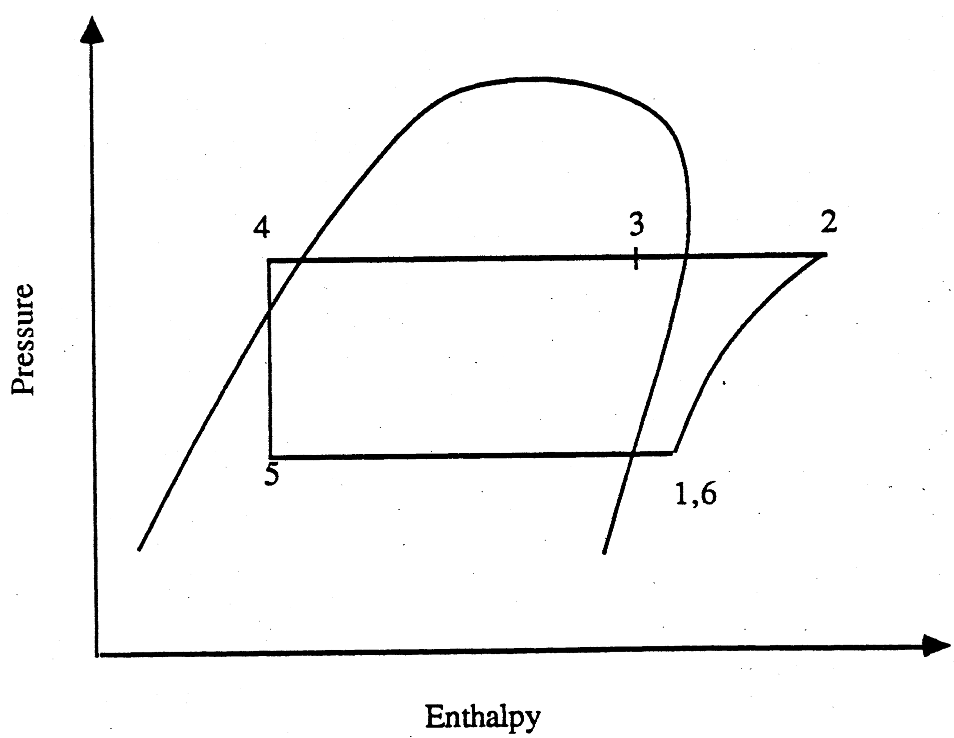

Figure 2.2 Pressure-Enthalpy Plot of the DEC vapor compression cycle

wrap around heat exchanger to state 3, and then further condensed and partially subcooled in the tube-in-tube heat exchanger to state 4. The fluid expands in the capillary tubes, which lowers the temperature and pressure to state 5. The two phase refrigerant at state 5 enters the evaporator where it is completely evaporated. The refrigerant next enters the accumulator at state 6 which, assuming heat transfer and pressure drops in the accumulator are negligible, is equal to state 1.

The vapor compression cycle of the DEC heat pump water heater will be modeled by individually modeling each of its five components (compressor, condensers, expansion device, and evaporaror). The accumulator is omitted from the cycle since it is assumed to have negligible heat transfer and pressure drop. The complete vapor compression cycle model, which is used to find the steady state performance of the heat pump, connects these five component models together in order of their physical occurrence.

2.2 Compressor Model

A hermetically-sealed reciprocating compressor is used in the DEC heat pump cycle, and also in most other small vapor compression cycles. As shown in Figure 2.3, the reciprocating compressor model includes an electric motor and a piston-cylinder assembly having suction and discharge valves. In building this compressor model it is assumed that the suction and discharge pressures and the inlet enthalpy of the refrigerant are known and that the refrigerant flow rate, compressor work, compressor input power, and outlet enthalpy are to be determined.

The inlet refrigerant is first used to cool the electric motor and then enters the cylinder where it is compressed and discharged. The pumping action of the compressor occurs by the piston moving to the right, reducing the pressure in the

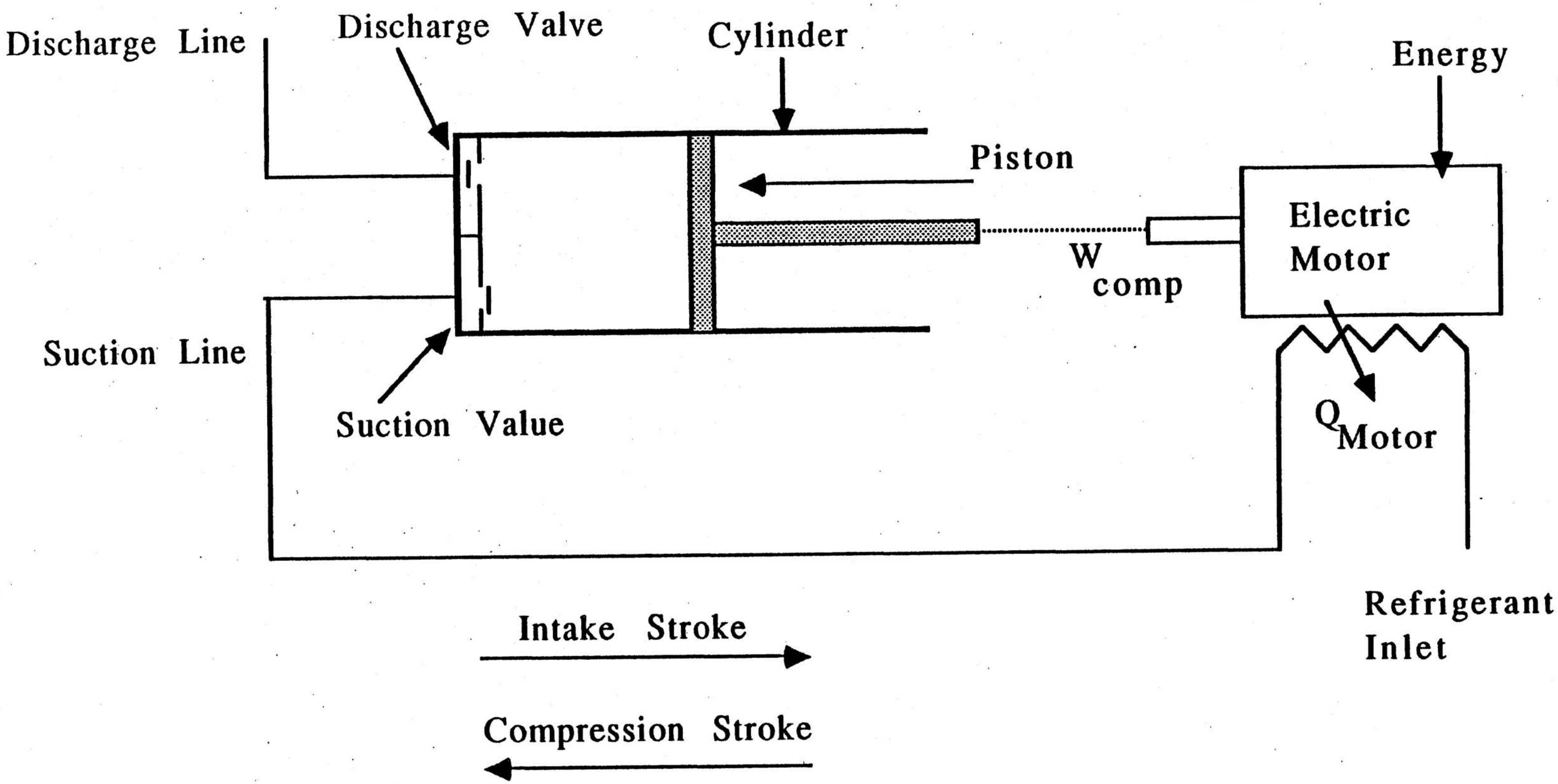

Figure 2.3 Diagram of the reciprocating compressor model.

cylinder until the suction valve opens and draws in low pressure refrigerant; after completing its inlet stroke, the suction valve closes and the piston is pushed back to the left, increasing the refrigerant pressure until the discharged valve opens and the refrigerant is forced out.

The energy loss of the electric motor goes partially to heating the inlet refrigerant and partially to heating the environment. The total energy given up by the motor (Q_{motor}) is equal to the difference between the electric energy supplied to the motor and the mechanical energy supplied to the piston. This energy difference is also labeled the work loss of the motor (W_{loss}) and is discussed in more detail later in this section. The actual percentage of this energy that goes to heating the refrigerant depends on the refrigerant and ambient temperatures, construction of the motor, and the refrigerant flow rate and flow paths around the motor. Since the detailed construction of the motor will not be known, it is assumed that the heat gained by the refrigerant is a fixed percentage of the motor work loss. The energy gains of the refrigerant and environment are defined as

$$Q_{r,motor} = Percent_{refrig}\, Q_{motor} \tag{2.2.1}$$

$$Q_{env,motor} = Percent_{env}\, Q_{motor} \tag{2.2.2}$$

The DEC heat pump loses approximately 10% of its work to heating the environment (DEC, 1987) and has a motor efficiency of about 60% (i.e., 60% of electric energy is converted to mechanical work). Using these two values, the heat transferred to the refrigerant and the environment was estimated to be 75% and 25%

of the motor losses respectively. The enthalpy of the refrigerant after being heated by the motor is calculated by

$$h_{r,in*} = h_{r,in} + \frac{Q_{r,motor}}{\dot{m}_r} \tag{2.2.3}$$

The flow rate of refrigerant through the compressor is dependent upon the inlet and outlet states and is described in terms of a volumetric efficiency. Volumetric efficiency is defined as

$$\eta_v = \frac{\text{volume flow rate entering compressor}}{\text{volume displacement rate of compressor}} \tag{2.2.4}$$

Stoecker and Jones (1982) show that the ideal volumetric efficiency can be calculated by

$$\eta_{v,ideal} = 100 - m\left(\frac{v_{suction}}{v_{discharge}} - 1\right) \tag{2.2.5}$$

$$m = 100\,\frac{V_C}{V_{cyl} - V_C} \tag{2.2.6}$$

where

m = percent clearance volume

V_C = clearance volume

V_{cyl} = cylinder volume

$v_{suction}$ = specific volume of the refrigerant at the inlet to the cylinder

$v_{discharge}$ = specific volume of the refrigerant at the exit of the cylinder

The ideal volumetric efficiency assumes that the specific volume of the refrigerant in the cylinder during the intake and exhaust is equal to the suction and discharge specific volumes respectively. This assumption neglects effects on the specific volumes caused by pressure differences across the suction and discharge valves, heat transfer in the cylinder during intake and exhaust, and leakage occuring in the valves and cylinder. Since the specific compressor details required to model these occurrences are not generally available, the ideal volumetric efficiency will be used as an approximation to the actual value.

The properties (temperature, pressure, enthalpy, entropy, quality, specific volume, and internal energy) of the refrigerant will be calculated by a Fortran program called FREON (Downing, 1971). Given any two independent properties, this program calculates the remaining properties.

The mass flow rate is calculated by

$$\dot{m}_r = \frac{PDR\left(\eta_{v,ideal} / 100 \right)}{v_{suction}} \tag{2.2.7}$$

Where the piston displacement rate (PDR) is the rate of volume swept by the piston. According equation 2.2.5, the volumetric efficiency can range in value from negative infinity to 1, but realistically it is confined between 0 and 1. The mass flow rate, therefore, is bounded between 0 and PDR/$v_{suction}$.

The compressor input power is generally defined in terms of two efficiencies: a compression efficiency and a motor efficiency. The work of compression (the work added to the refrigerant) is often defined in terms of an isentropic efficiency. Both Domanski (1982, p. 34) and Krakow (1987, p. 4), however, point out that the isentropic efficiency is not constant, but varies as a function of the pressure difference across the compressor. Domanski suggests using a polytropic efficiency along with an isentropic efficiency because the "polytropic efficiency is more consistent from one application to another and provides more consistent representation of average compressor performance" (1982 ,p. 37). The polytropic efficiency for compression is the isentropic efficiency of an infinitely small compression stage. It is independent of the pressure ratio, unlike the isentropic efficiency, and is defined as (Eastop, 1978)

$$\eta_{poly} = \frac{dh_I}{dh_a} \tag{2.2.8}$$

where

h_I = isentropic enthalpy

h_a = actual enthalpy.

For an isentropic compression the change in enthalpy is given as

$$dh_I = v\,dP \tag{2.2.9}$$

And for an ideal gas the equation of state and the change in enthalpy are defined as

$$P\,v = R\,T \tag{2.2.10}$$

$$dh_a = c_p\,dT \tag{2.2.11}$$

Substituting equations 2.2.10-12 into equation 2.2.9 gives

$$\eta_{poly} = \frac{R\,T\,dP}{c_p\,P\,dT}$$

or

$$\eta_{poly}\int_?^? \frac{dT}{T} = \frac{R}{c_p}\int_?^? \frac{dP}{P}$$

Integrating this equation gives

$$\eta_{poly}\ln\left(\frac{T_1}{T_2}\right) = \frac{k-1}{k}\ln\left(\frac{P_1}{P_2}\right)$$

or

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{\eta_{poly}\,k}} \tag{2.2.12}$$

The isentropic index (k) is defined as the ratio of specific heat at constant pressure to the specific heat at constant volume

$$k = \frac{c_p}{c_v} \tag{2.2.13}$$

Equation 2.2.13 can be written in terms of a polytropic index (n) as

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} \tag{2.2.14}$$

The polytropic index, therefore, is related to the isentropic index and polytropic efficiency by

$$\eta_{poly} = \frac{\frac{k-1}{k}}{\frac{n-1}{n}} \tag{2.2.15}$$

In the compressor model it is assumed that the polytropic efficiency is known and is constant, that the isentropic index is calculated using equation 2.2.13, and that the polytropic index is calculated from equation 2.2.15. Since both the specific heats vary during the compression cycle, an average value of the specific heats at the inlet and isentropic outlet are used to estimate an average k.

Using the value of the polytropic index calculated by equation 2.2.15, the isentropic efficiency of a steady flow process is given as

$$\eta_I = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{\frac{k}{k-1}}{\frac{n}{n-1}} \left[\frac{\left(P_2/P_1\right)^{\frac{k-1}{k}} - 1}{\left(P_2/P_1\right)^{\frac{n-1}{n}} - 1} \right] \tag{2.2.16}$$

The enthalpy difference across the compressor is calculated using this equation and the work of compression is found from

$$\dot{W}_{comp} = \dot{m}_r \left(h_2 - h_1 \right) \tag{2.2.17}$$

As previously mentioned, the compressor input power is often defined in terms of a motor efficiency, which is the ratio of compressor work to the input power and is given by

$$\eta_{motor} = \frac{\dot{W}_{comp}}{\dot{E}_{motor}} \tag{2.2.18}$$

Figure 2.4 is a plot of motor efficiency versus compressor load for a typical heat-pump motor, which shows that the motor efficiency varies as a function of the compressor load. Since the compressor load will vary with operating conditions at the evaporator and condenser, the motor efficiency cannot be assumed to be constant. The information in Figure 2.4 is used to find both the compressor input power (E_{motor}) and energy losses due to the conversion of electrical to mechanical energy (W_{losses}) as a function of the load fraction as follows:

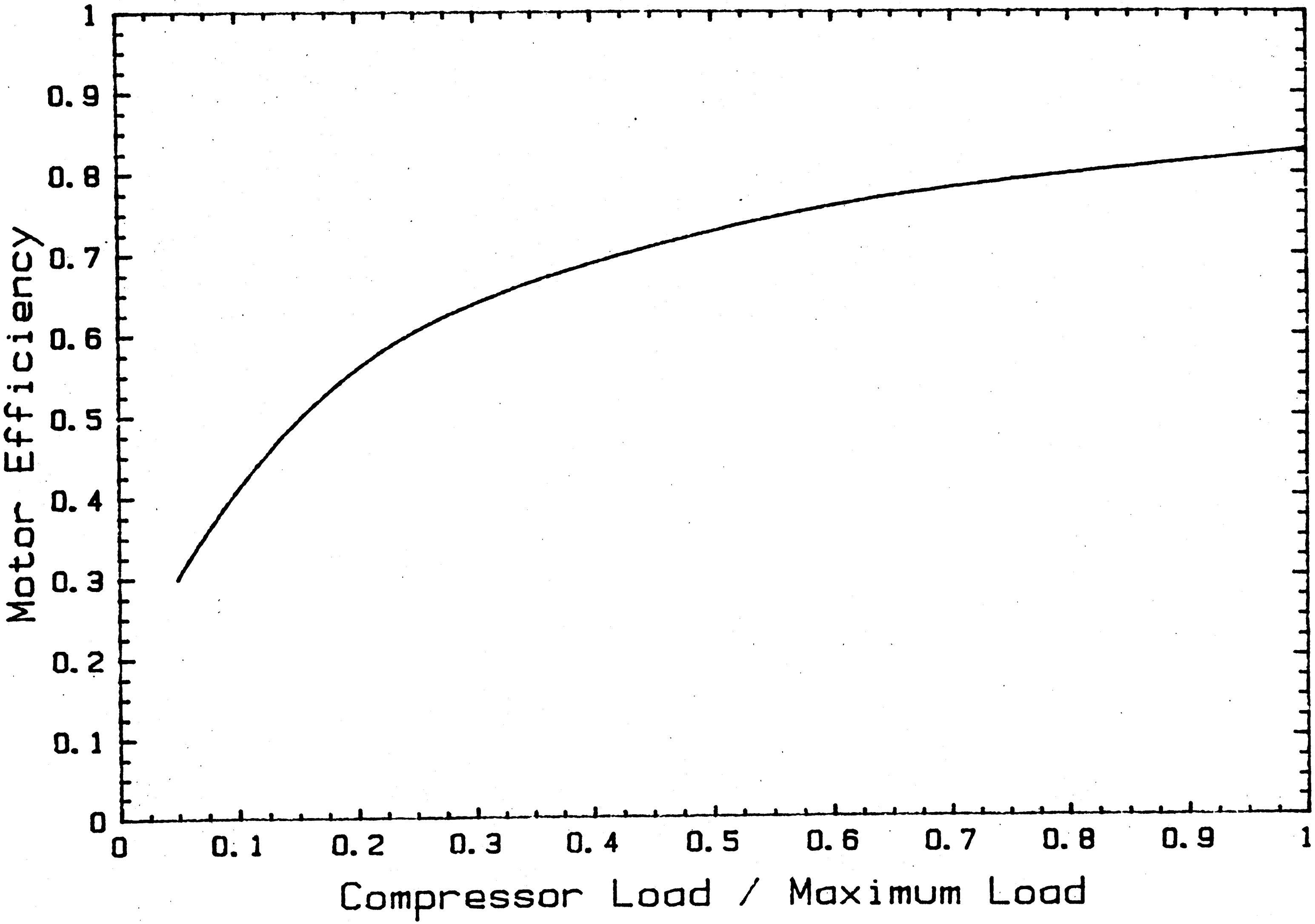

Figure 2.4 Motor efficiency versus compressor load for a typical heat pump motor (Domanski, 1982)

$$\dot{E}_{motor} = \frac{\dot{W}_{comp}}{\eta_{motor}} \tag{2.2.19}$$

$$\dot{W}_{losses} = \dot{E}_{motor} - \dot{W}_{comp} \tag{2.2.20}$$

Using these equations and the data in Figure 2.4, a plot was made of the compressor input power and the loss work versus the compressor load and is given in Figure 2.5. This graph shows that the loss work term is essentially constant for all load fractions. It is assumed, therefore, that the loss work term is constant and thus the compressor input power is calculated by

$$\dot{E}_{motor} = \dot{W}_{comp} + \dot{W}_{losses} \tag{2.2.21}$$

where the work of compression is calculated by equation 2.2.13 and the loss work term should be estimated from available manufacturer's data.

The preceeding equations are programmed into the Fortran subroutine COMP which is included in the Appendix. Inputs to this routine are the piston displacement rate (PDR), percent clearance (m), loss work ($\dot{W}_{loss}$), polytropic efficiency (η_{poly}), inlet refrigerant enthalpy, and the inlet and outlet refrigerant pressures; the outputs are the compressor input power ($\dot{E}_{motor}$) and refrigerant mass flow rate ($\dot{m}_r$).

Figures 2.6 and 2.7 are plots of the refrigerant flow rate and compressor input power versus the pressure ratio across the compressor calculated by the program for typical inputs. As expected, the volumetric efficiency and refrigerant flow rate decrease with increasing pressure ratio, while the input power increases to a

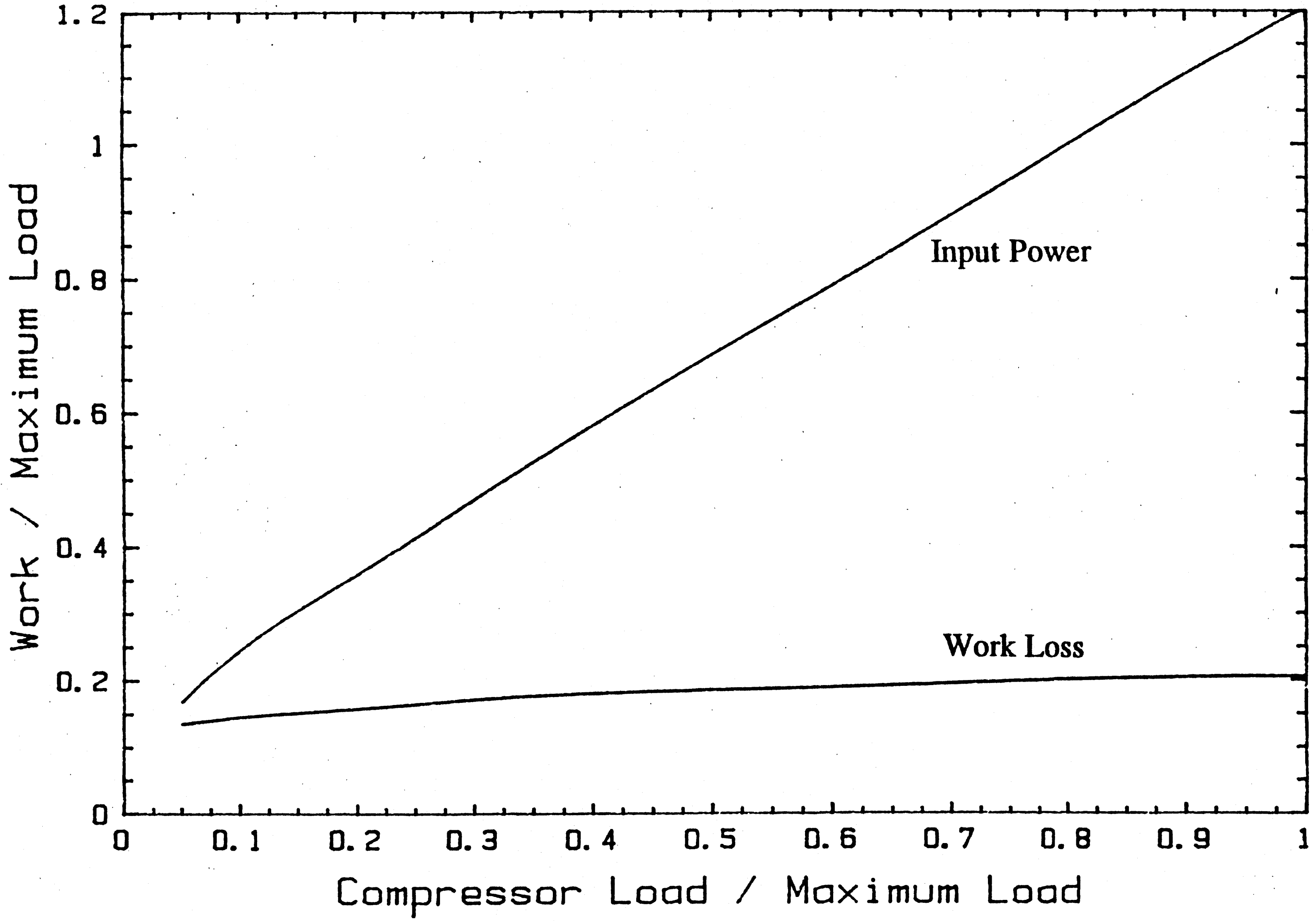

Figure 2.5 Compressor input power and loss work versus compressor load for a typical heat pump motor

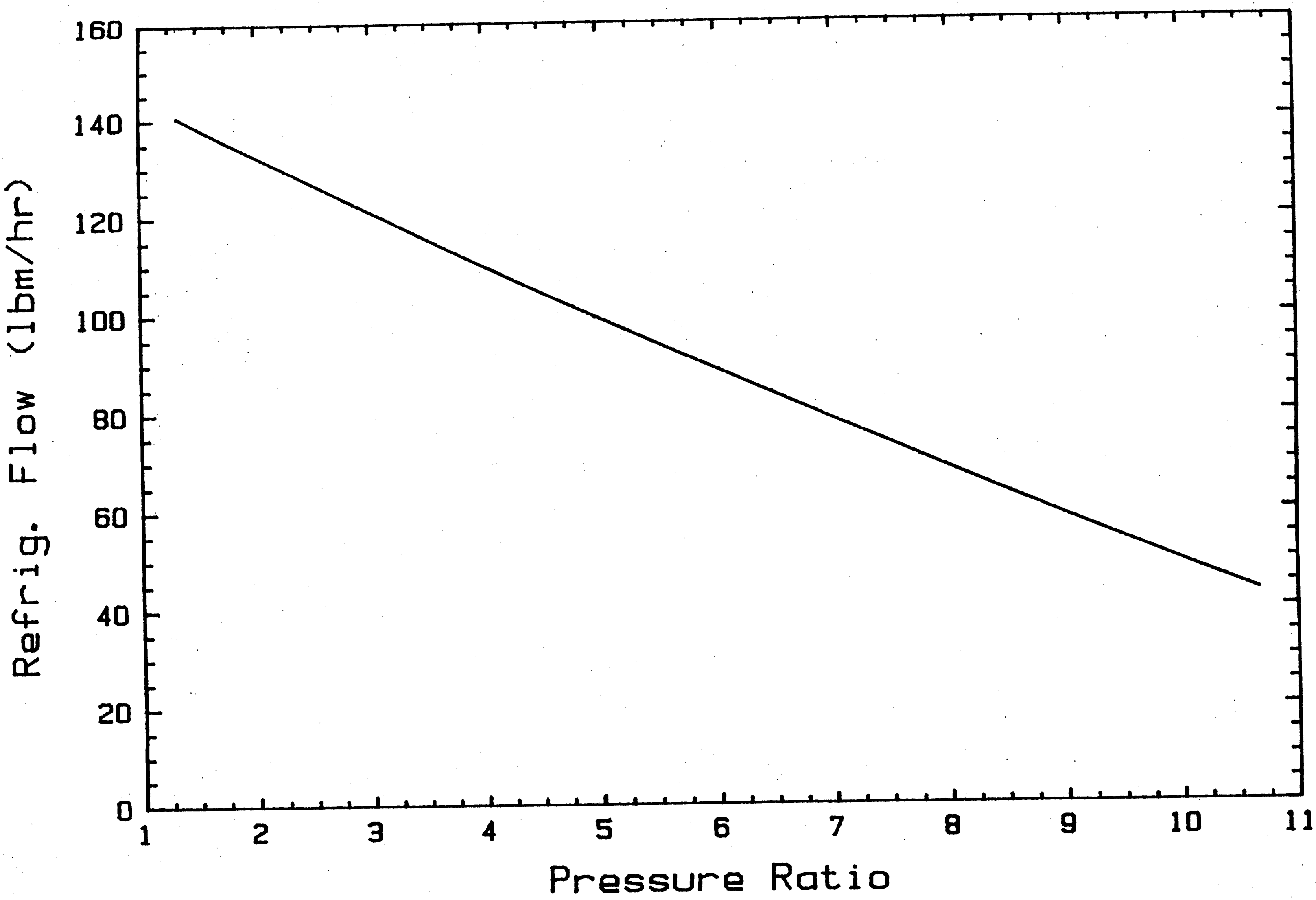

Figure 2.6 Refrigerant flow rate versus compressor pressure ratio

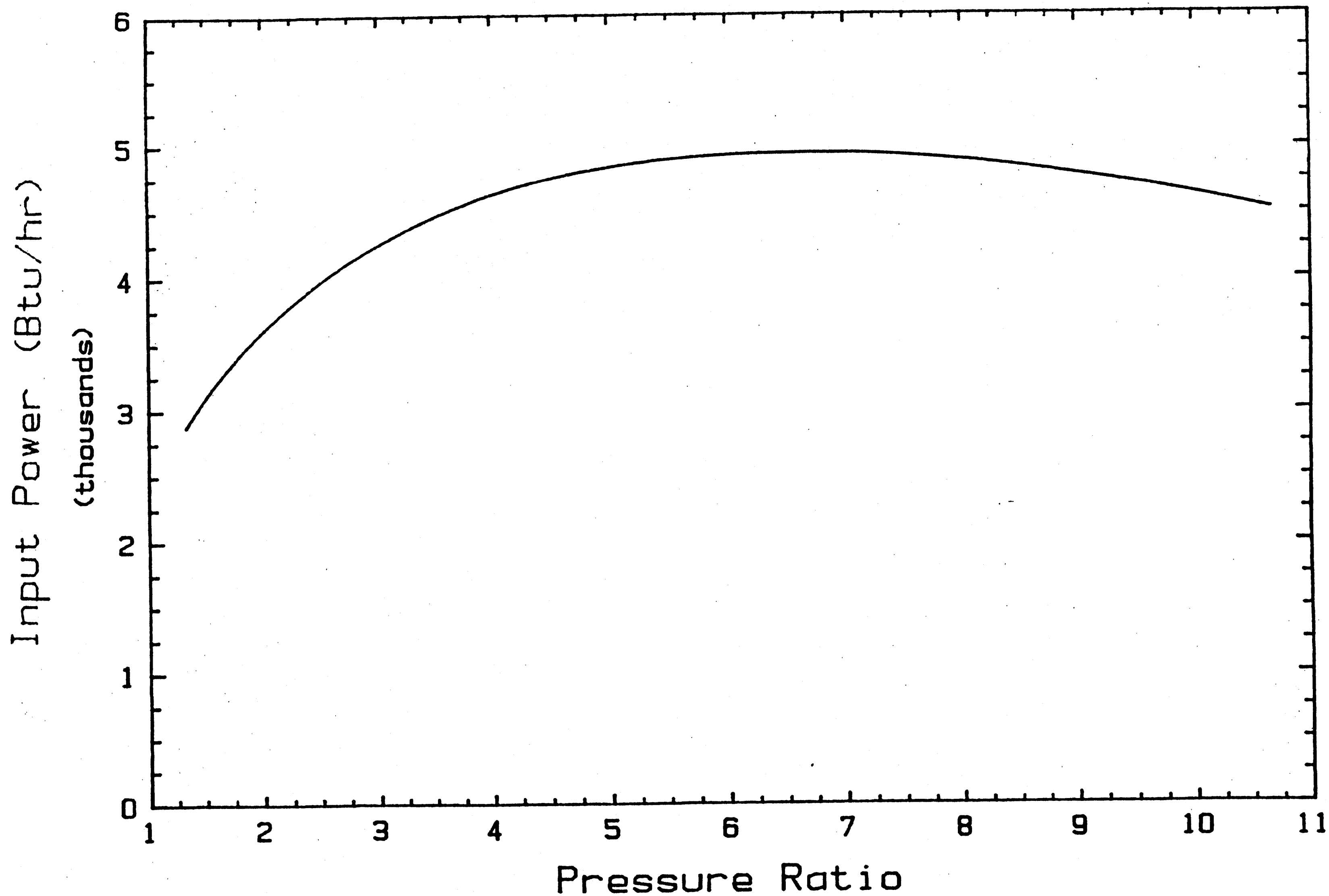

Figure 2.7 Compressor input power versus compressor pressure ratio

maximum and then decreases with increasing pressure ratio.

2.3 Condenser Models

The DEC heat pump condenses the refrigerant in two separate heat exchangers: a wrap-around heat exchanger and a tube-in-tube heat exchanger. The superheated refrigerant from the compressor enters a wrap-around heat exchanger on the storage tank where it is desuperheated and partially condensed. This exit refrigerant enters a counterflow tube-in-tube heat exchanger where it is condensed and subcooled. The cooling fluid running opposite the refrigerant in the tube-in-tube heat exchanger is the inlet water to the storage tank if hot water is being drawn, otherwise, it is water recirculated from the bottom of the tank, as shown in Figure 2.1.

2.3.1 Wrap-Around Heat Exchanger Model

The energy transfer from the refrigerant in the wrap-around heat exchanger can be viewed as taking place in three separate heat exchangers: a standard heat exchanger which cools the superheated vapor to a saturated gas; a condenser which condenses the refrigerant to a saturated liquid; and a standard heat exchanger which subcools the refrigerant. The limited amount of experimental data available shows only a small temperature difference between the top and bottom of the tank. Stratification, however, is reported to occur for certain operating conditions different from those during the collection of the data. As will be shown in section 2.6, calculating the heat transfer from a stratified tank to the refrigerant for a annual simulation would require a great deal of computer time. To simplify the model, the heat transfer is calculated assuming the refrigerant exchanges heat with the mean water temperature. Figure 2.8 shows the three regions of cooling and a temperature profile of both the refrigerant and water versus the tank height. The values of L1 and L2 will change with inlet conditions and therefore must be determined by the model.

The refrigerant entering the heat exchanger is assumed to be superheated since it is the exhaust from the compressor. The model is used to calculate the outlet state of the refrigerant, which could be superheated ($L_1 = L_2 = 0$), two phase ($L_1 = 0$), or subcooled. The following paragraphs describe the calculations for each of the three regions.

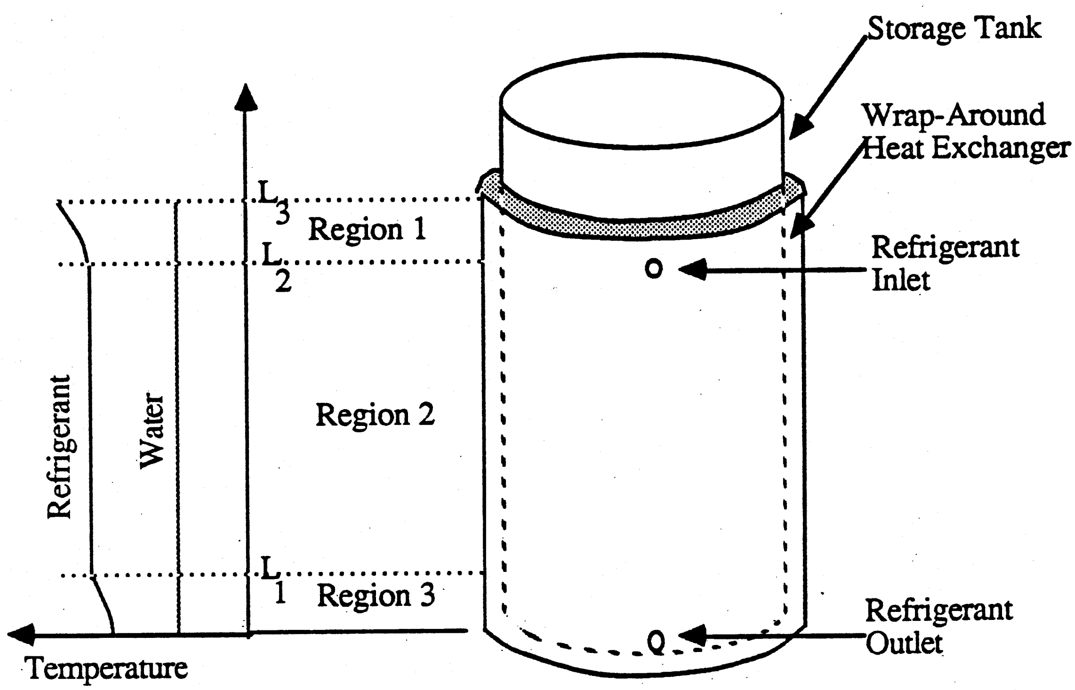

Figure 2.8 Schematic of the wrap-around condenser

Region I

The heat transfer in each region is between the refrigerant and the mean water temperature, and is calculated using a log mean temperature difference (LMTD) approach. The LMTD method defines the heat transfer in terms of a conductance, a surface area, and a log mean temperature difference (LMTD) as

$$q = A\,U\,(LMTD) \tag{2.3.1}$$

where the LMTD between flows a and b is defined by

$$LMTD = \frac{\left(T_{a,out} - T_{b,in}\right) - \left(T_{a,in} - T_{b,out}\right)}{\ln\left[\left(T_{a,out} - T_{b,in}\right) \Big/ \left(T_{a,in} - T_{b,out}\right)\right]} \tag{2.3.2}$$

Since the refrigerant in region 1 is assumed to exchange heat with the mean water temperature, the LMTD is

$$LMTD_1 = \frac{T_{r,out} - T_{r,in}}{\ln\left[\left(T_{r,out} - T_{H_2O,in}\right) \Big/ \left(T_{r,in} - T_{H_2O,out}\right)\right]} \tag{2.3.3}$$

The model assumes the conductance-area product (UA) is known and that conductance is constant over the entire height of the heat exchanger (i.e., tank). This assumption should have only a small affect on the final result, because the major portion of the energy transfer occurs in the two phase region where the UA of the refrigerant is approximately constant. Using this assumption, the heat transfer in

Region 1 is given as

$$q_1 = (LMTD)_1\,(UA)_{WA,\,tot}\left(\frac{L_3 - L_2}{L_3}\right) \tag{2.3.4}$$

An energy balance on the refrigerant shows the change in energy of the refrigerant to be equal to the heat transfer in this region

$$q_1 = \dot{m}_r\left(h_{r,in} - h_{r,out,1}\right) \tag{2.3.5}$$

Region II

The heat transfer in region 2 is also calculated using the LMTD method and is given as

$$q_2 = (LMTD)_2\,(UA)_2 \tag{2.3.6}$$

Since the refrigerant in region 2 is condensing, the refrigerant temperature is uniform and equal to the condensing temperature. Since the tank temperature is also uniform, the log mean temperature difference reduces to

$$LMTD_2 = T_{r,sat} - T_{H_2O} \tag{2.3.7}$$

Equation 2.3.6 can be expressed in terms of a conductance-area product of the entire heat exchanger by

$$q_2 = (LMTD)_2 (UA)_{WA,\,tot}\left(\frac{L_2 - L_1}{L_3}\right) \tag{2.3.8}$$

and, similar to region 1, an energy balance on the refrigerant gives

$$q_2 = \dot{m}_r\left(h_{r,out,1} - h_{r,out,2}\right) \tag{2.3.9}$$

Region III

The equations describing the heat transfer in Region III, which are similar to those for Region I

$$q_3 = (LMTD)_3 (UA)_{WA,\,tot}\frac{L_2}{L_3} \tag{2.3.10}$$

$$LMTD_3 = \frac{T_{r,out,3} - T_{sat}}{\ln\left[\left(T_{r,out,3} - T_{H_2O}\right)\Big/\left(T_{sat} - T_{H_2O}\right)\right]} \tag{2.3.11}$$

$$q_3 = \dot{m}_r\left(h_{r,sat,v} - h_{r,out,3}\right) \tag{2.3.12}$$

The nine equations 2.3.3-5, 7-12 can be solved for the nine unknowns which are the lengths, LMTD's, and heat transfer in each region. Since the intermediate refrigerant states are implicit in the equations, a secant iteration method is used. The total energy delivered to the storage tank is equal to the sum of the heat transfers in the three regions

$$q_{WA} = q_1 + q_2 + q_3 \tag{2.3.13}$$

An additional assumption made in this model is that the overall conductance is constant for all refrigerant flow rates. The heat-transfer coefficient on the refrigerant side of the heat exchanger, and therefore the overall conductance, will vary with flow rate, but the range of refrigerant flow rates is small enough that the effect on the conductance should be negligible.

The above model was programed into a Fortran subroutine (COND). A plot of the total heat transfer versus the refrigerant flow rate was made for typical values of $(UA)_{WA}$, T_{H_2O}, $P_{r,in}$ and $h_{r,in}$, and is shown in Figure 2.9. This figure shows that the rate of increase in heat transfer with increasing refrigerant flow depends upon the outlet state of the refrigerant; it increases rapidly if the entering fluid is subcooled but slower if two phase or superheated.

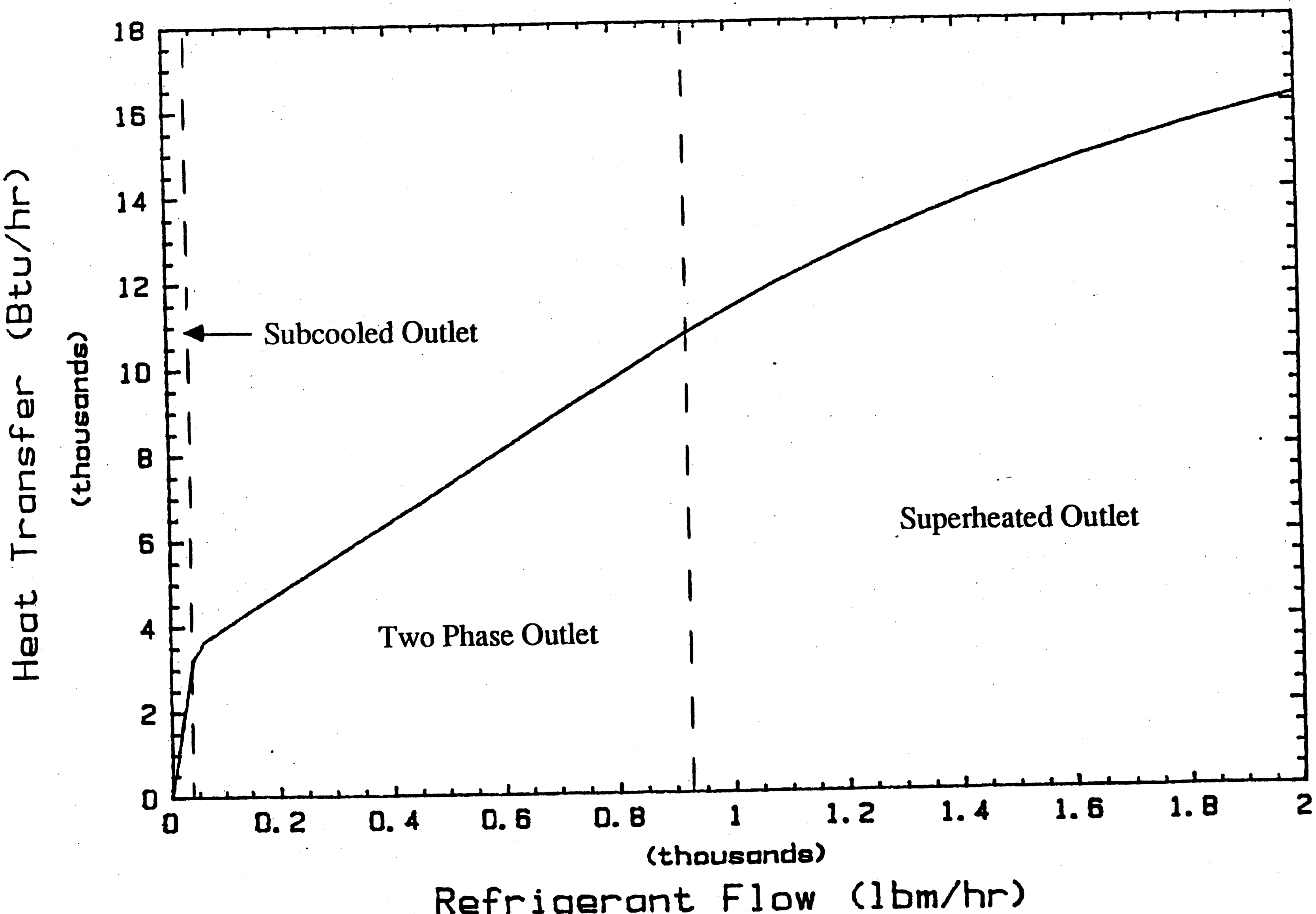

Figure 2.9 Heat transfer of the wrap-around condenser versus the refrigerant flow rate

2.3.2 Tube-in-Tube Heat Exchanger Model

The refrigerant exiting from the wrap-around heat exchanger enters the tube-in-tube heat exchanger where it is cooled by a counterflow of water. It is assumed that the state of the entering refrigerant is known and is either a two phase mixture or a subcooled liquid and that the exiting refrigerant may be either two phase or subcooled. Further, it is assumed that the mass flow rate and inlet temperature of the water are known. Figure 2.10 shows a plot of the refrigerant and water temperatures versus the length of the heat exchanger. This figure shows that, unlike the wrap-around heat exchanger, the water temperature varies over the length of the heat exchanger. The heat transfer can be viewed as occurring in two heat exchangers: a condensing heat exchanger (Region I) and a subcooling heat exchanger (Region II).

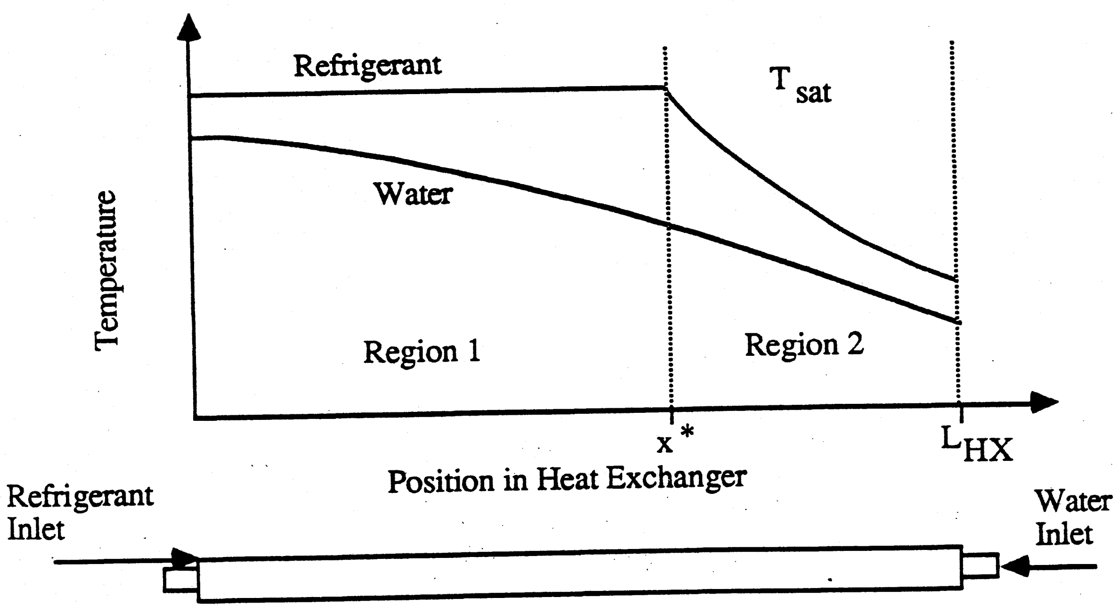

Figure 2.10 Refigerant and water temperatures versus the tube-in-tube heat exchanger length

An energy balance on the refrigerant in Region I shows the heat transfer to be equal to the change in energy of the refrigerant

$$q_1 = \dot{m}_r\left(h_{r,in} - h_{r,x*}\right) \tag{2.3.14}$$

Similarly, an energy balance on the water gives

$$q_1 = \dot{m}_{H_2O}\, c_{pH_2O}\left(T_{H_2O,out} - T_{H_2O,x*}\right) \tag{2.3.15}$$

Since the outlet and intermediate water temperatures are unknown, the effectiveness method for calculating heat transfer is used, instead of the LMTD method used for the wrap-around heat exchanger. The effectiveness method defines the heat transfer between streams a and b as

$$q = \varepsilon\left(\dot{m}\, c_p\right)_{min}\left(T_{a,in} - T_{b,in}\right) \tag{2.3.16}$$

where the effectiveness for a single-pass counter-flow heat exchanger is defined as

$$\varepsilon = \frac{1 - \exp\left[-NTU\,(1 - C)\right]}{1 - C\exp\left[-NTU\,(1 - C)\right]} \tag{2.3.17}$$

$$NTU = \frac{UA}{\left(\dot{m}\, c_p\right)_{min}} \tag{2.3.18}$$

$$C = \frac{(\dot{m}\,c_p)_{min}}{(\dot{m}\,c_p)_{max}} \tag{2.3.19}$$

The specific heat for a condensing gas is effectively infinite. In region 1, therefore, the water has the minimum capacitance rate, C is zero, and the effectiveness equations reduce to

$$q_1 = \varepsilon_1 (\dot{m}\,c_p)_{H_2O} (T_{r,sat} - T_{H_2O,x*}) \tag{2.3.20}$$

$$\varepsilon_1 = 1 - \exp[-NTU_1] \tag{2.3.21}$$

$$NTU_1 = \frac{(UA)_{TT} \frac{x^*}{L_{TT}}}{(\dot{m}\,c_p)_{H_2O}} \tag{2.3.22}$$

The overall heat-transfer coefficient is assumed to be a known, constant over the entire length of the heat exchanger, and independent of the refrigerant flow rate. The equations describing the heat transfer in Region II are derived in a manner similar to those for Region I, and are

$$q_2 = \dot{m}_r (h_{r,x*} - h_{r,out}) \tag{2.3.23}$$

$$q_1 = \dot{m}_{H_2O}\, c_{pH_2O} (T_{H_2O,x*} - T_{H_2O,in}) \tag{2.3.24}$$

$$q_2 = \varepsilon_2 \left(\dot{m}\, c_p\right)_{min} \left(T_{r,sat} - T_{H_2O,in}\right) \tag{2.3.25}$$

$$\varepsilon_2 = \frac{1 - \exp\left[-NTU_2 \left(1 - C_2\right)\right]}{1 - C_2 \exp\left[-NTU_2 \left(1 - C_2\right)\right]} \tag{2.3.26}$$

$$NTU_2 = \frac{(UA)_{TT} \dfrac{L_{TT} - x^*}{L_{TT}}}{\left(\dot{m}\, c_p\right)_{min}} \tag{2.3.27}$$

$$C_2 = \frac{\left(\dot{m}\, c_p\right)_{min}}{\left(\dot{m}\, c_p\right)_{max}} \tag{2.3.28}$$

Finally, the solution to these equations is iterative since the intermediate water temperature and refrigerant state are unknown and implicit in the equations. The equations were solved in a Fortran subroutine called SUBCOOLHX using a secant iteration method. A plot is shown in Figure 2.11 of the heat transfer versus the refrigerant flow rate for typical input values. This figure shows that the heat transfer increases with flow rate until the outlet becomes two phase, and then it is constant. The heat transfer is constant because the refrigerant is two phase the entire length of the tube and so increasing the flow does not change the temperature seen by the cooling water. This is only true because it was assumed that the UA is constant and independent of flow rate. The enthalpy of the exiting refrigerant, however, continues to decrease with increasing flow and approaches the inlet enthalpy.

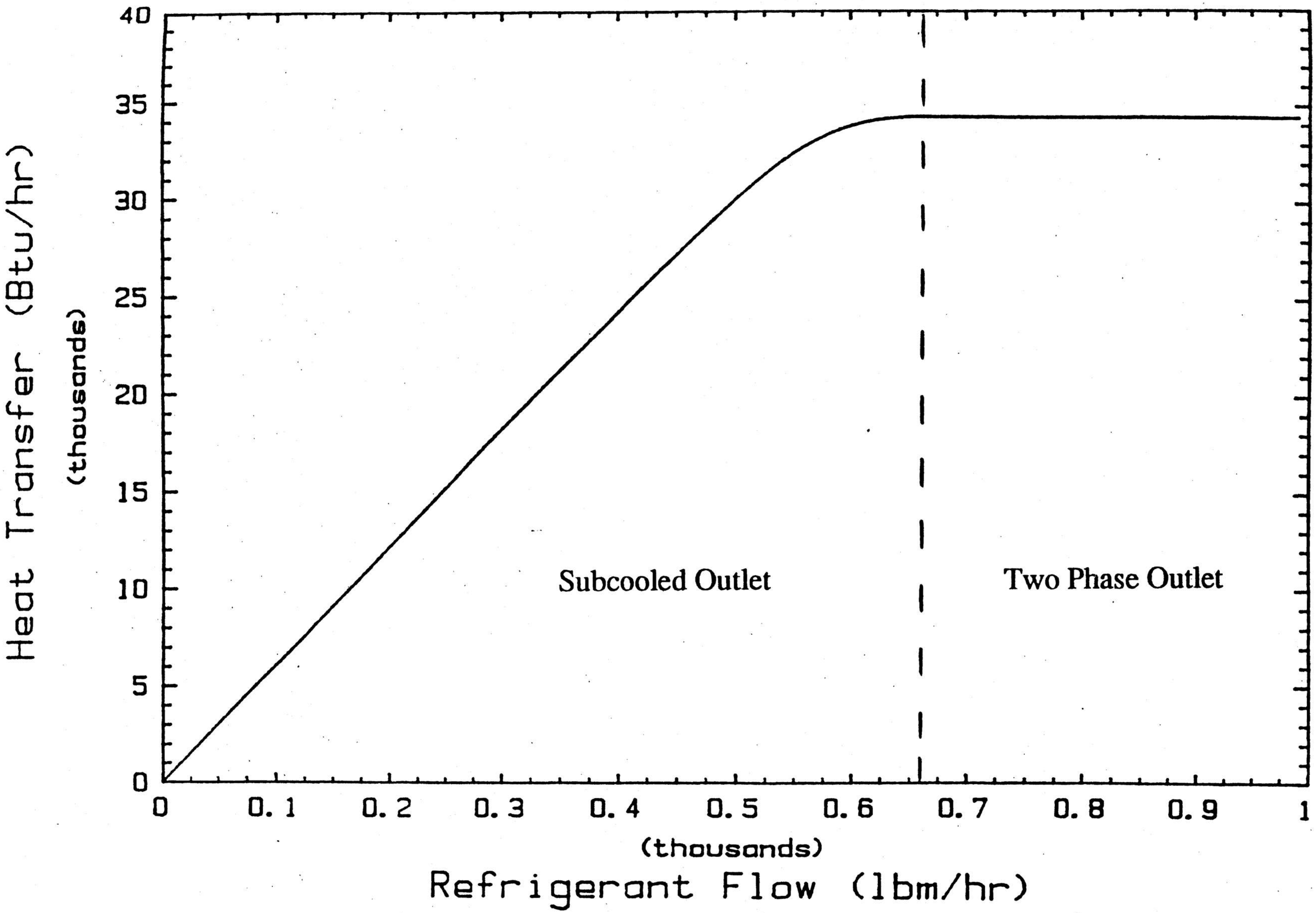

Figure 2.11 Heat transfer of the tube-in-tube condenser versus refrigerant flow rate

2.4 Capillary Tube Model

A capillary tube is a long, small diameter tube used to create a flow restriction between the condenser and evaporator. Unlike other expansion devices used in vapor compression cycles, the capillary tube offers no level or superheat control, but only provides a pressure drop between the condenser and evaporator, determined by the inlet and outlet states of the tube.

The refrigerant leaving the condenser in the DEC HPWH is divided into two flow paths, which remain separate until combined at the exit of the evaporator. To calculate the pressure drop in the capillary tubes, it is assumed that the pressure drop in each path is the same. The pressure drop calculations, therefore, are made for one tube with half the total flow.

The fluid exiting the condenser is either a subcooled liquid or a two phase mixture. Figure 2.12 illustrates this point by showing a pressure-enthalpy plot of the vapor compression cycle having three possible pressure drop paths between the condenser and evaporator. If the fluid entering the capillary tube is a subcooled

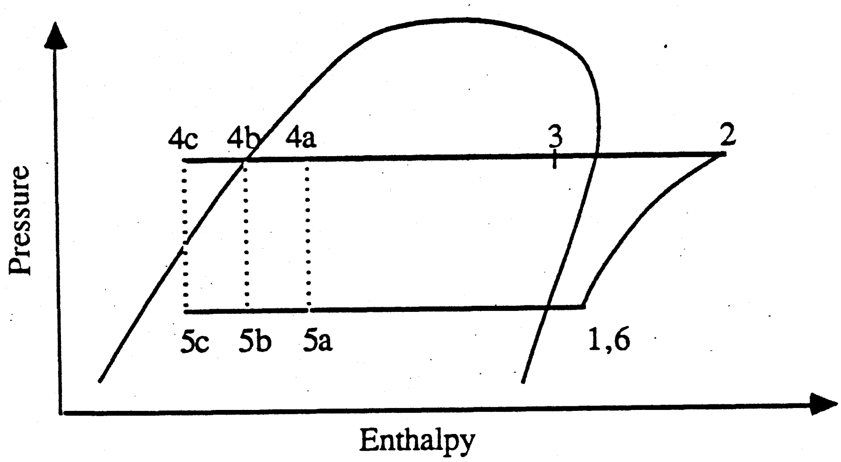

Figure 2.12 Pressure-enthalpy plot showing three possible pressure drop paths between the condenser and the evaporator

liquid, the pressure drop in the tube between the inlet and the length at which the fluid becomes a saturated liquid is determined by a momentum balance on the fluid

$$\left[(P_{in} - P^*) - f\,\frac{x^*}{D\,v}\,\frac{V^2}{2}\right] A = \dot{m}_r (V^* - V_i) \qquad (2.4.1)$$

where P*, V* and x* refer to the pressure, velocity, and length at which the refrigerant becomes a saturated liquid. Since the tube has a constant cross-sectional area and since the density of a subcooled refrigerant is relatively constant, the two velocities on the right side of the equation are approximately equal and therefore this term reduces to zero. Assuming an isoenthalpic expansion, the quality and enthalpy at x* are equal to zero and the inlet enthalpy respectively. These two properties completely define the thermodynamic state and therefore the pressure is determined by

$$P^* = P\left(h^*{=}h_{in},\ \text{quality}{=}0\right) \qquad (2.4.2)$$

Equation 2.4.1 can be rearranged to

$$x^* = \frac{2\,(P_{in} - P^*)\,D\,v}{f\,V_i^2} \qquad (2.4.3)$$

where the velocity is approximated as the inlet velocity.

In performing these calculations, it is first assumed that P* is less than the exit pressure, and therefore, the refrigerant is subcooled the entire length of the tube. Next the length x* is calculated using equation 2.4.3, and the assumption is checked by comparing this length to the capillary tube length. If this assumption is found to be correct, the exit pressure is calculated using a rearranged form of equation 2.4.3 where x* and P* are exchanged with the tube length and the exit pressure. If the assumption is incorrect, then two phase flow exists in at least part of the tube length, in which case the pressure drop in the portion of the tube with a subcooled fluid is calculated using equations 2.4.2 and 2.4.3.

The steady state pressure drop in the two phase portion of the tube is calculated using the method outlined by Stoecker and Jones (1982). Figure 2.13 shows an incremental length of the tube with inlet position a and outlet position b, showing the important parameters in this analysis. A mass balance on the incremental length is

$$\dot{m}_r = \rho_a V_a A = \rho_b V_b A$$

or

$$\frac{\dot{m}_r}{A} = \frac{V_a}{v_a} = \frac{V_b}{v_b} \tag{2.4.4}$$

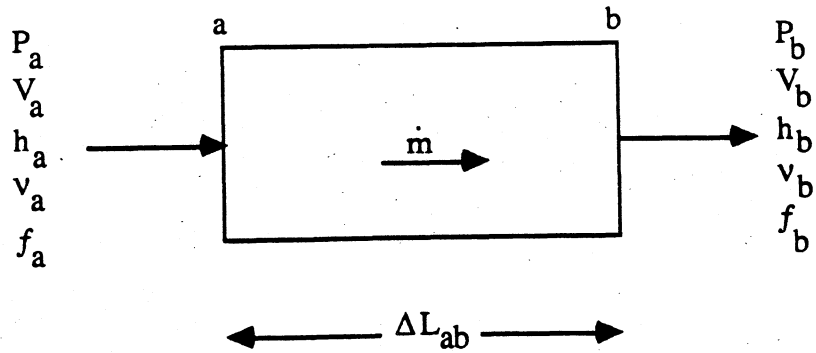

Figure 2.13 Incremental section of capillary tube

An energy balance on this section, neglecting heat losses to the surroundings, is

$$h_a + \frac{V_a^2}{2} = h_b + \frac{V_b^2}{2} \tag{2.4.5}$$

Finally, a momentum balance yields

$$\left[(P_a - P_b) - f\frac{\Delta L}{D v}\frac{V^2}{2}\right]A = \dot{m}_r(V_b - V_a) \tag{2.4.6}$$

At any point in the saturation dome, the enthalpy and specific volume are defined in terms of the quality by

$$h = h_f(1 - x) + h_g x \tag{2.4.7}$$

$$v = v_f(1 - x) + v_g x \tag{2.4.8}$$

Equation 2.4.4 is inserted into the momentum equation to yield

$$\left[(P_a - P_b) - f\frac{\Delta L}{D}\frac{V}{2}\frac{\dot{m}_r}{A}\right]A = \dot{m}_r(V_b - V_a) \tag{2.4.9}$$

The velocity in this equation will be assumed to be the arithmetic average of the inlet and outlet velocities, and the friction factor for the flow is said (Stoecker, 1982) to be approximated by

$$f = \frac{0.33}{Re^{0.25}} = \frac{0.33}{\left(\frac{V D \rho}{\mu}\right)} \tag{2.4.10}$$

The viscosity of the refrigerant in the vapor dome is related to the quality by

$$\mu = \mu_f(1 - x) + \mu_g x \tag{2.4.11}$$

The friction factor in equation 2.4.9 is taken to be the arithmetic average of the inlet and outlet friction factors.

Stoecker and Jones show that equations 2.4.4, 5, 7 and 8 can be reduced to

$$x_b = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \tag{2.4.12}$$

where

$$a = \frac{1}{2}\left(v_{g,b} - v_{f,b}\right)\left(\frac{\dot{m}_r}{A}\right)^2$$

$$b = \left(h_{g,b} - h_{f,b}\right) + v_{f,b}\left(v_{g,b} - v_{f,b}\right)\left(\frac{\dot{m}_r}{A}\right)^2$$

$$c = \left(h_{f,b} - h_a\right) + \frac{1}{2} v_{f,b}^2 \left(\frac{\dot{m}_r}{A}\right)^2 - \frac{V_a^2}{2}$$

The calculation procedure for each interval, which assumes the inlet conditions are known and the outlet conditions are to be determined, is as follows:

1) Select an outlet temperature T_b
2) Use T_b to calculate P_b, h_{fb}, h_{gb}, v_{fb}, and v_{gb}
3) Solve equation 2.4.12 for x_b
4) Use x_b to calculate values of h_b, v_b, u_b, and V_b
5) Calculate the outlet friction factor and the average friction factor using equations 2.4.10 and 2.4.11.
6) Calculate ΔL using equation 2.4.9.

This procedure is repeated until the sum of the ΔL's is greater than or equal to the capillary tube length. If the input mass flow is found to be greater than the choked mass flow rate, the procedure is stopped because this is physically unrealistic. This condition is checked by making sure the change in entropy in each section is positive, since a negative change in entropy would indicate that choked flow actually existed in the tube.

The preceding calculation method was programed into a Fortran subroutine called EXPAN. Using a typical capillary tube geometry and inlet condition, this program was run for a range of refrigerant flow rates, and a plot was made of the pressure ratio across the tube versus the flow rate (Figure 2.14). This plot shows that the pressure drop increases smoothly with mass flow up to a maximum mass flow of 146 lbm/hr. Increasing the input mass flow further is an attempt to make the flow rate in the tube greater than the choked mass flow. When the mass flow is greater than this maximum value, the outputs of the routine are a flag indicating choked flow and the position in the tube where the choking condition was found.

2.5 Evaporator Model

The DEC evaporator is a forced-air crossflow heat exchanger. As mentioned in section 2.4, the refrigerant flow is divided into two paths, separating at the inlet of the capillary tubes and joined again at the outlet of the evaporator. The two flow paths through the evaporator were designed to have equal heat transfer, and therefore, it is equivalent to model the heat exchanger using a single flow path having the total refrigerant flow and using an effective overall heat-transfer coefficient.

Figure 2.15 shows a schematic of the evaporator and a plot of both refrigerant and inlet air temperatures versus the evaporator position. It is assumed that the entering refrigerant is a two phase mixture, but that it may leave as either two phase or superheated. The evaporator can be viewed as two heat exchangers, an evaporator and a standard heat exchanger, as indicated in Figure 2.15 by regions 1 and 2 respectively.

All or part of the exterior surface of the evaporator may be below the dew point temperature of the cooling air, in which case the air stream would condense, increasing the evaporator's effectiveness. In addition to finding the

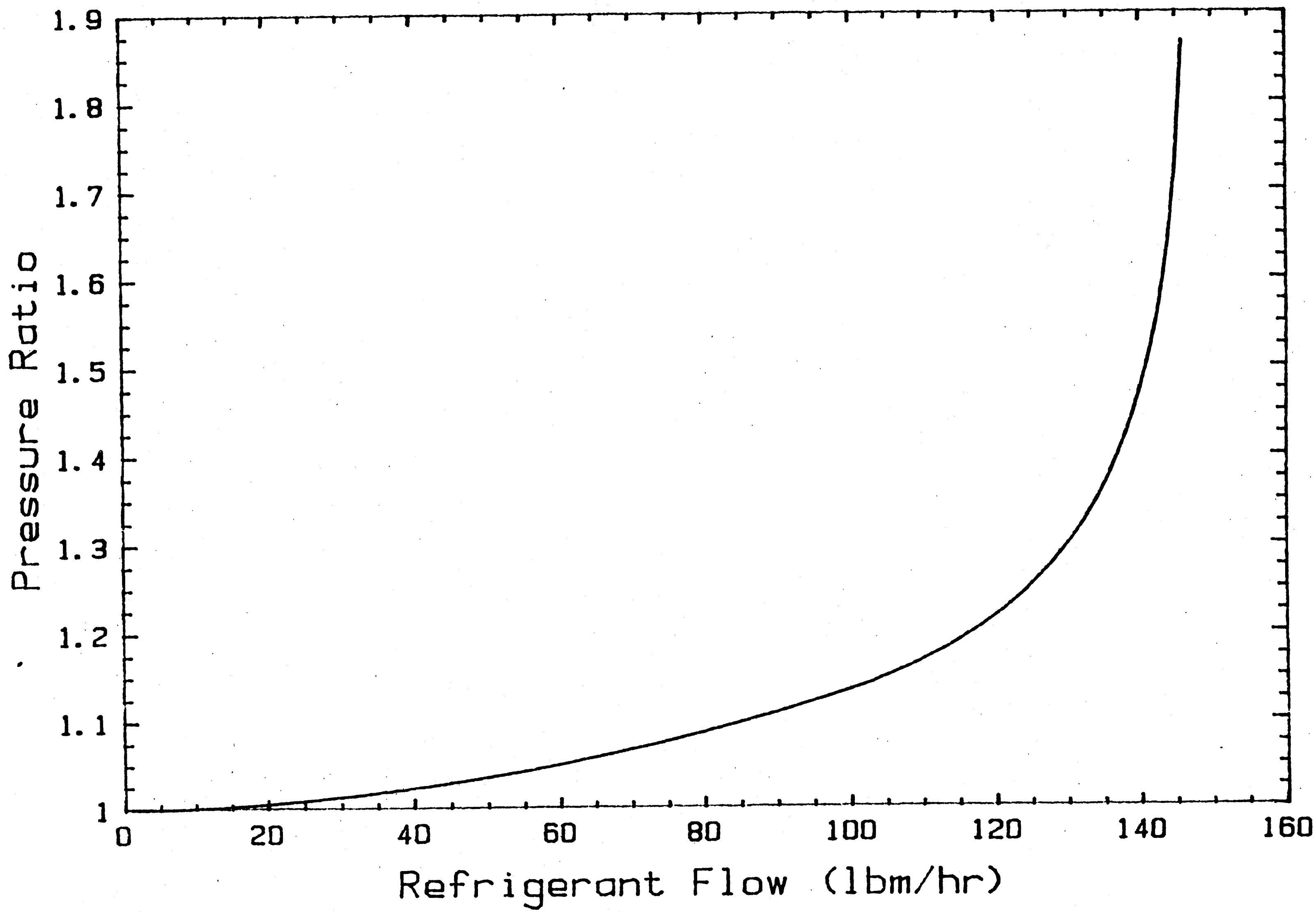

Figure 2.14 Pressure ratio across the capillary tube versus refrigerant flow rate

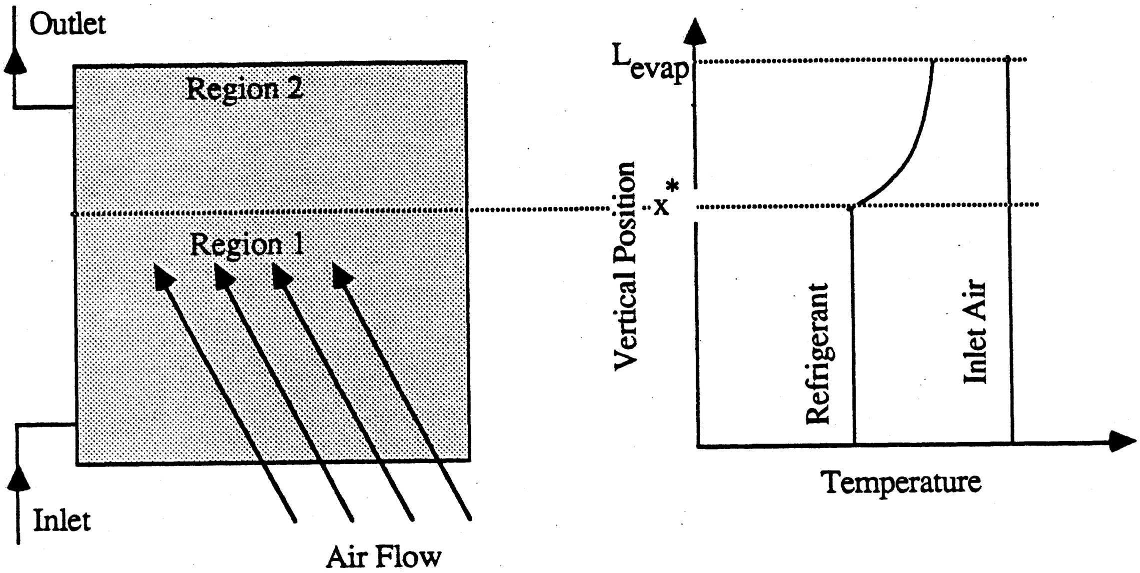

Figure 2.15 Diagram of the evaporator model showing air and temperatures versus the height of the evaporator

regions of the two heat exchangers, the evaporator model must also determine if the evaporator is all dry, all wet, or somewhere in between, and then calculate the heat transfer corresponding to that situation. The following paragraphs consider these three cases separately.

Completely Dry Exterior Surface

The exterior temperature of the evaporator must be known in order to determine if an air stream is condensing. Detailed information concerning the evaporator construction and the individual heat-transfer coefficients of the air and refrigerant stream would be needed to calculate the actual exterior surface temperature. To simplify the calculations, it is assumed that the exterior surface temperature at any point on the evaporator is approximately equal to the average of the air and refrigerant temperatures at that location.

$$T_{e,s} = \frac{T_r + T_{air}}{2} \tag{2.5.1}$$

If $T_{e,s}$ at the evaporator entrance is higher than the dew point of the air stream, then the evaporator will be completely dry. The completely dry heat-exchanger's performance can be calculated by the effectiveness method, and similar to the tube-in-tube heat exchanger model for the condensing region, the governing equations for the portion of the heat exchanger with an evaporating refrigerant (Region I) are:

$$q_1 = \dot{m}_r\left(h_{r,x*} - h_{r,in}\right) \tag{2.5.2}$$

$$q_1 = \varepsilon_1 \left(\dot{m}\, c_p\right)_{air}\left(T_{air} - T_{r,sat}\right) \tag{2.5.3}$$

$$\varepsilon_1 = 1 - \exp\left(-NTU_1\right) \tag{2.5.4}$$

$$NTU_1 = \frac{(UA)_{evap}\dfrac{x^*}{L_{evap}}}{\left(\dot{m}\, c_p\right)_{air}} \tag{2.5.5}$$

Likewise, the equations for the region with a superheated refrigerant (Region II) are:

$$q_2 = \dot{m}_r\left(h_{r,out} - h_{r,x*}\right) \tag{2.5.6}$$

$$q_2 = \varepsilon_2 \left(\dot{m}\, c_p\right)_{air} \left(T_{air} - T_{r,sat}\right) \tag{2.5.7}$$

$$\varepsilon_2 = \frac{NTU_2}{\left[\dfrac{NTU_2}{1 - \exp\left(-NTU_2\right)} + \dfrac{C_2\, NTU_2}{1 - \exp\left(-C_2\, NTU_2\right)} - 1\right]} \tag{2.5.8}$$

$$NTU_2 = \frac{\left(UA\right)_{evap} \dfrac{L_{evap} - x^*}{L_{evap}}}{\left(\dot{m}\, c_p\right)_{min}} \tag{2.5.9}$$

$$C_2 = \frac{\left(c_p\, \dot{m}\right)_{min}}{\left(c_p\, \dot{m}\right)_{max}} \tag{2.5.10}$$

Completely Wet Exterior Surface

If the exterior evaporator temperature ($T_{e,s}$) at the evaporator exit is below the air stream dew point, then the exterior surface is completely wet. Unfortunately, $T_{e,s}$ at the exit is an unknown. The calculation of a completely wet evaporator is outlined temporarily, assuming that the evaporator is known to be completely wet.

Braun (1986) has developed an enthalpy effectiveness model to calculate the heat transfer between a wet surface and an air stream. He defines this heat transfer as

$$q_{wet} = \varepsilon_{wet}\, \dot{m}_{air} \left(h_{air,in} - h^*_{air,r,in}\right) \tag{2.5.11}$$

and analogous to dry crossflow heat exchangers, the effectiveness is

$$\varepsilon_{wet} = \frac{NTU_{wet}}{\left[\frac{NTU_{wet}}{1-\exp(-NTU_{wet})} + \frac{m^* \, NTU_{wet}}{1-\exp(-m^* \, NTU_{wet})} - 1\right]} \tag{2.5.12}$$

where $h^*_{air,r,in}$ is the enthalpy of saturated air at the inlet refrigerant temperature and m^* is defined as

$$m^* = \frac{\dot{m}_{air} C_s}{(\dot{m}\, c_p)_r} \tag{2.5.13}$$

Braun defines C_S to be a saturation specific heat given as

$$C_s = \left.\frac{dh^*_{sat}}{dT}\right|_{T=T_r} = \frac{h^*_{air,r,in} - h^*_{air,r,out}}{T_{r,in} - T_{r,out}} \tag{2.5.14}$$

The number of transfer units (NTU) is defined differently than for a dry heat exchanger by (note that UA has units of lb_m/hr)

$$NTU_{wet} = \frac{(UA)_{wet}}{\dot{m}_{air}} \tag{2.5.15}$$

$$(UA)_{wet} = \frac{1}{\frac{C_s}{h_{c,r} A_i} + \frac{c_{pm}}{h_{c,air} A_o}} \tag{2.5.16}$$

Using this enthalpy effectiveness model, the resulting equations for Region I are

$$q_{wet,1} = \dot{m}_r\left(h_{r,in} - h_{r,x*}\right) \tag{2.5.17}$$

$$q_{wet,1} = \dot{m}_{air}\,\varepsilon_{wet,1}\left(h^*_{air,r,in} - h_{air,in}\right) \tag{2.5.18}$$

$$\varepsilon_{wet,1} = 1 - \exp\left(-NTU_{wet,1}\right) \tag{2.5.19}$$

$$NTU_{wet,1} = \frac{\left((UA)_{evap,wet}\right)}{\dot{m}_{air}}\,\frac{x^*}{L_{evap}} \tag{2.5.20}$$

The resulting equations for Region II are

$$q_{wet,2} = \dot{m}_r\left(h_{r,x*} - h_{r,out}\right) \tag{2.5.21}$$

$$q_{wet,2} = \dot{m}_{air}\,\varepsilon_{wet,2}\left(h^*_{air,r,x*} - h_{air,in}\right) \tag{2.5.22}$$

$$\varepsilon_{wet,2} = \frac{NTU_{wet,2}}{\left[\dfrac{NTU_{wet,2}}{1-\exp\left(-NTU_{wet,2}\right)} + \dfrac{m^*\,NTU_{wet,2}}{1-\exp\left(-\,m^*\,NTU_{wet,2}\right)} - 1\right]} \tag{2.5.23}$$

$$NTU_{wet,2} = \frac{\left((UA)_{evap,wet}\right)}{\dot{m}_{air}}\,\frac{L_{evap} - x^*}{L_{evap}} \tag{2.5.24}$$

Similar to the dry heat exchanger, the solution is iterative, since the length x* is initially unknown, and also since the outlet refrigerant temperature is needed to estimate C_S.

The calculations are made assuming a completely wet surface. When this solution is completed, this assumption is checked by comparing the calculated value of the evaporator surface temperature ($T_{e,s}$) to the air stream dew point. If $T_{e,s}$ is lower than the dew point, then the evaporator is completely wet.

Partially Wet Exterior Surface

According to Braun, the heat transfer from the partially wet surface can be approximated as the maximum of the heat flows calculated assuming completely wet and dry surfaces. He shows that this will always be an underprediction, and usually accurate to within 5 percent.

Combined Evaporator Solution

In performing the heat transfer calculation for the evaporator, it is assumed that the wet and dry overall heat-transfer coefficients are known and that they are constant with respect to position and time, and that the inlet temperatures and flow rates of the air and refrigerant are known and the heat transfer and outlet conditions are to be found. The steps in determining the performance of the evaporator are summarized as follows:

1) Make calculations assuming the exterior surface is completely dry.
2) Determine if surface is completely dry. If it is, the solution is finished.
3) If the exterior surface is at least partially wet, calculate the solution for a

completely wet surface.

4) Determine if the surface is completely wet. If so, the solution is finished.

5) If the surface is found to be only partially wet, then approximate the heat transfer as the maximum of the two solutions.

Again this calculation method was programed into a Fortran subroutine (EVAP). Using this suboutine, a plot was made of the evaporator heat-transfer versus the refrigerant flow rate for both completely dry (T_{air} = 75°F, T_{wb} = 55°F) and wet (Tair = 75°F, Twb = 75°F) evaporators and is given in Figure 2.16. Like the tube-in-tube heat exchanger, when the evaporator becomes two phase over the entire length, increasing flow has no effect on the heat transfer. This plot also shows that at the low flow rates the dry and wet evaporators give the same heat transfer, because in both cases the refrigerant is heated to its maximum value of 75°F. For higher refrigerant flow rates, the air is the limiting flow (i.e., has smallest capacitance rate), and therefore the wet evaporator has a higher heat transfer because it heats the refrigerant by both sensible and latent cooling of the air.

2.6 Combined Vapor Compression Model

The completed component models are used to determine the steady-state performance of a vapor compression cycle for any given component sizes and any condenser or evaporator operating conditions. A Fortran subroutine called VCC_LOOP was created which calls the component models in the order in which they physically occur. The compressor was chosen to be the first component called, followed by the wrap-around condenser, the tube-in-tube condenser, the capillary tube, and the evaporator. The refrigerant states needed for the vapor compression cycle "loop" calculations are the condenser and evaporator pressures and the

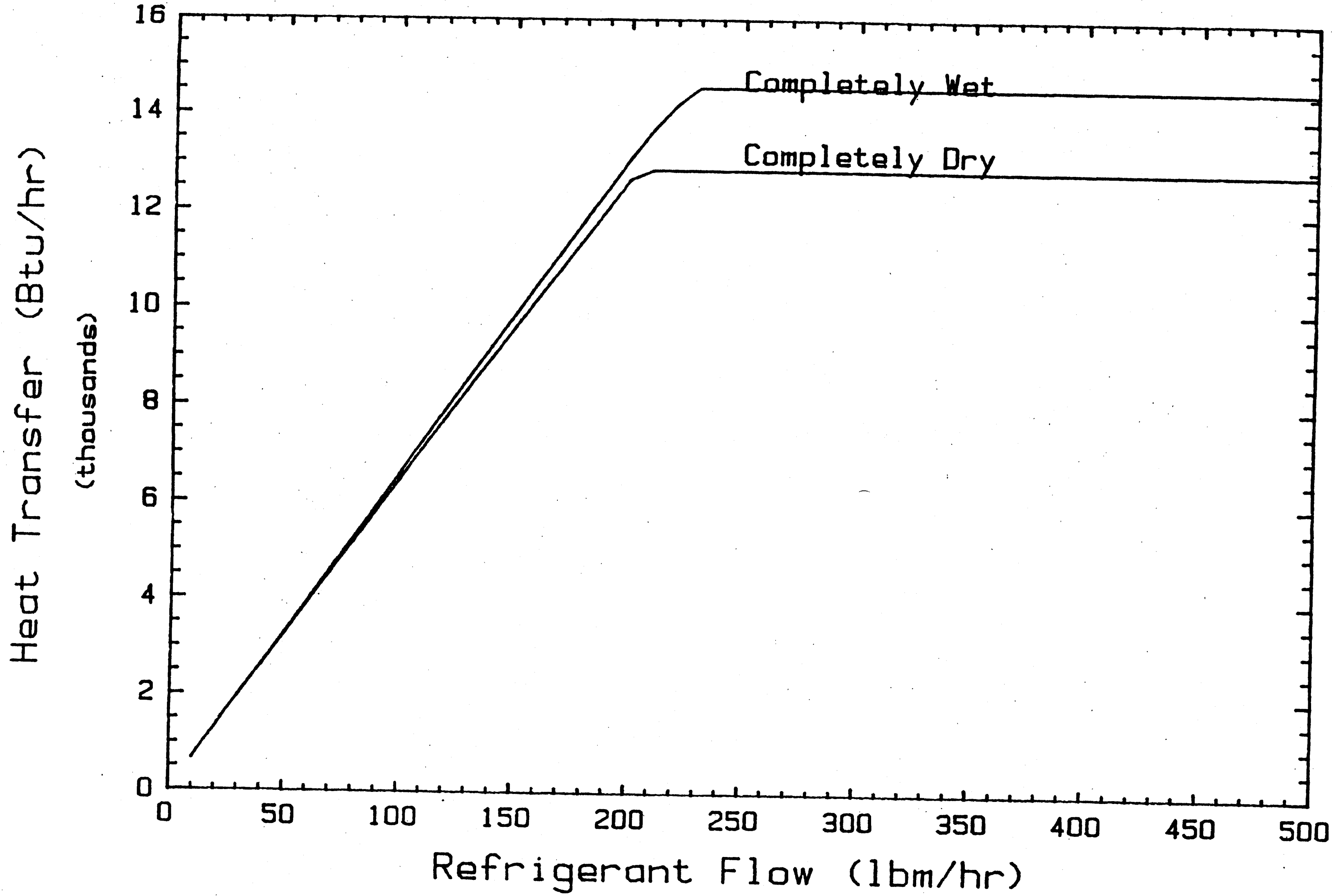

Figure 2.16 Evaporator heat transfer versus refrigerant flow rate

enthalpy at the inlet of the compressor. Given these values, and all component parameters (lengths, UA's, efficiencies, etc.), the model performs a loop calculation. In this calculation, each of the components following the compressor uses the outlet enthalpy from the previous component as its inlet enthalpy. The steady-state performance of the vapor compression cycle is found when the input values of the compressor inlet enthalpy and high and low pressures are consistent with the calculated enthalpy at the compressor and the pressure drop across the capillary tube. The output from this subroutine is the calculated performance of the cycle and a comparison of the input states to the calculated values. The comparisons made are

$$\text{Residual}_1 = \frac{h_{1,\text{input}} - h_{1,\text{calculated}}}{h_{1,\text{calculated}}} \tag{2.6.1}$$

$$\text{Residual}_2 = \frac{P_{\text{low,calculated}} - P_{\text{low,input}}}{P_{\text{low,input}}} \tag{2.6.2}$$

The capillary tube model is unable to predict a low pressure when the refrigerant flow rate calculated by the compressor model is higher than choked flow through the capillary tube. In this case the following comparison is made

$$\text{Residual}_2 = \frac{L_{\text{cap. tube}} - L^*}{L_{\text{cap. tube}}} \tag{2.6.3}$$

where L* is the capillary tube length where a positive change in entropy was found.

The next step is to find an iteration technique which will find the condenser and evaporator pressures such that the two residual comparisons are zero. The iteration technique found to consistently converge on the solution is a combination of a secant method and a half interval methods. As shown in the flow chart in Figure 2.17, the secant method uses an inner loop to converge on the condenser pressure, which gives $Residual_1$ equal to zero, while a half interval technique was used in the major loop to converge on the evaporator pressure, which gives $Residual_2$ equal to zero. This iteration scheme is programed in a Fortran subroutine called VCC. This routine was found to take approximately a minute of computer time on a MicroVax to converge on each solution. Several more sophisticated iteration techniques were tried, but did not work because of the discontinuous definition of $Residual_2$.

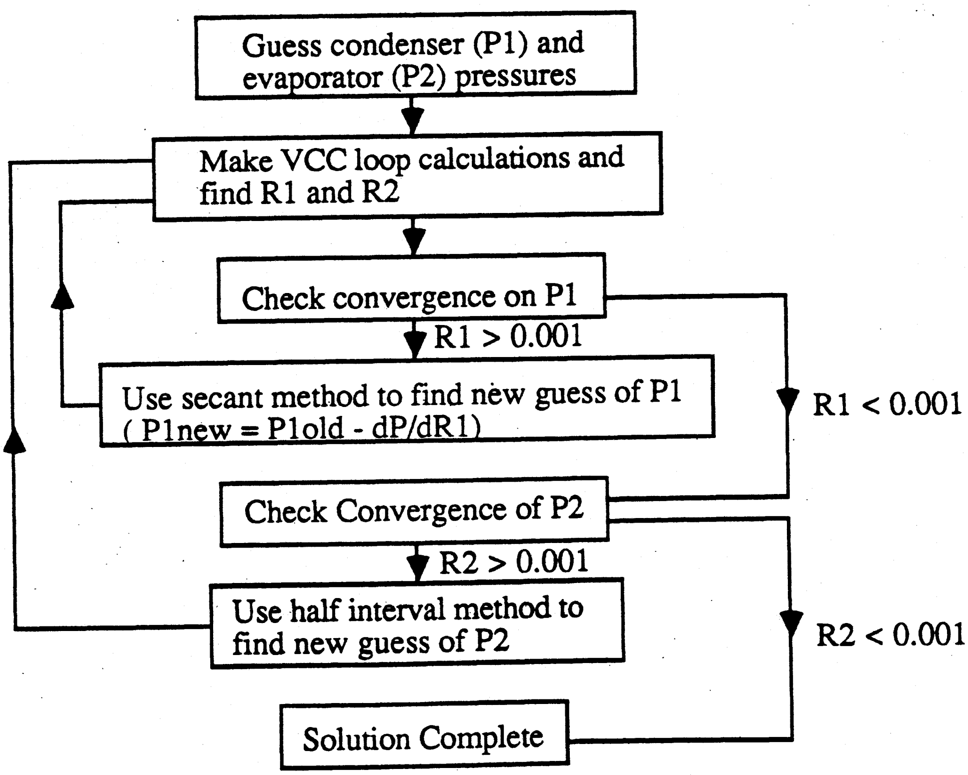

Figure 2.17 Flowchart of the vapor compression cycle program

2.7 Analysis of DEC HPWH Data

The available data of the performance of the DEC HP-120-27 heat pump water heater is given in Table 2.1. The data is of a tank being heated up from 55°F to 135°F using only the heat pump heater (i.e., no backup heating) and having no hot water draw. The volume of water in the tank during this test was 110 gallons (920 lb_m). This data is used to compare the performance predicted by the vapor compression to that of the actual machine. The performance predicted by the vapor compression cycle model, however, is the instantaneous heat transfer to the water, refrigerant pressure, and the compressor input power, whereas the performance data given is the mean tank temperature, integrated input power over a time step, and the instantaneous refrigerant pressure.

Time (hours)	Average Water Temperature (°F)	Total Power * (KW)	Instantaneous Pressure Suction/Discharge (psia)
0.0	57.4	0.0	75 / 152
1.0	76.7	1225	83 / 191
2.0	95.2	2540	85 / 235
3.0	112.3	4000	88 / 285
4.0	128.7	5610	95 / 340
4.5	136.3	6440	98 / 370

* Integrated power during heating ($\int_{t=0}^{t=t} \dot{E}_{HP}\, dt$)

Table 2.1 Performance data of the DEC HP-120-27

The predicted pressures can be directly compared to the data, but the input power and heating capacity must be compared indirectly. Using the data, linear regression was used to find equations which express the integrated input power and the temperature of the water as a function of time. The equations found are

$$\int \dot{E}_{HP}\, dt = 1110 + 3810\, t + 233\, t^2 \quad \text{(Btu)} \tag{2.7.1}$$

$$\overline{T}_{H_2O} = 57.4 + 19.9\, t - 0.522\, t^2 \quad (^\circ F) \tag{2.7.2}$$

The accuracy of a regression equation to predict the actual response (eg. temperature or integrated power) of a system (eg. heat pump) is often given in terms of an R^2 value ("R-squared"). The R^2 value is the percentage of the variations in the data that are accounted for by the regression equation and is defined as

$$R^2 = 1 - \frac{\overline{\sigma}_{equ.}}{\overline{\sigma}_{data}} \tag{2.7.3}$$

where

$\overline{\sigma}_{equ}$ = mean standard deviation of the equation's response to actual response

$\overline{\sigma}_{data}$ = mean standard deviation of the actual response to the average response.

Therefore, a R^2 value of 1.0 indicates that all variations in the data are exactly accounted for by the regression equation (i.e., equations predict the data exactly), and a value of 0.0 indicates that none of the variations in the data are accounted for. Both equations 2.7.3 and 2.7.4 have R^2 values of 0.999; therefore, both equations accurately predict the actual data.

The instantaneous power to the compressor of the heat pump is the derivative of equation 2.7.1, which is

$$\dot{E}_{HP} = 3810 + 466\,t \quad \text{(Btu/hr)} \tag{2.7.4}$$

The instantaneous heating capacity is related to the temperature by

$$Q_{H_2O} = m_{H_2O}\, c_p \frac{dT_{H_2O}}{dt} \tag{2.7.5}$$

Substituting in the mass of the water, specfic heat, and the water temperature given by equation 2.7.2, into equation 2.7.5 gives

$$Q_{H_2O} = 18240 - 958\,t \quad \text{(Btu/hr)} \tag{2.7.6}$$

Now equations 2.7.4 and 2.7.6, along with the pressure data given in Table 2.1, are comparable to the performance predicted by the model. This comparison is made in the following section.

2.8 Comparison of the Vapor Compression Cycle Model to Data

Input values for the vapor compression cycle model were determined from catalog information when available and approximated using engineering judgment otherwise. These input values are given in Table 2.2, where the italicized values are those which were approximated. Using these input values, the vapor compression cycle model was run for several condenser water temperatures corresponding to those of the test data. Figure 2.18 compares the performance predicted by the model to the test data (equations 2.7.6 and 2.7.7). This plot shows that the power is

Variable	Units	Initial Input	Best Fit Input
Refrigerant		22	22
m		0.055	0.08
PDR	ft^3/sec	0.047	0.0458
η_{poly}		*0.8*	0.8
W_{loss}	Btu/hr	*2000*	2350
$(UA)_{WA}$	Btu/hr-°F	450	450
L_{TT}	ft	14.5	14.5
$(UA)_{TT}/L$	Btu/hr-ft-°F	140	140
m_{H2O}	lbm/hr	2000	2000
A_{CT}	ft^2	0.0000191	0.0000191
L_{CT}	ft	2.5	2.5
$(UA)_{evap}$	Btu/hr-°F	*1600*	380
$\dot{m}_{air}$	lbm/hr	5625	5625

Table 2.2 Input values to VCC program. Italicized values in column 3 are the values approximated without information.

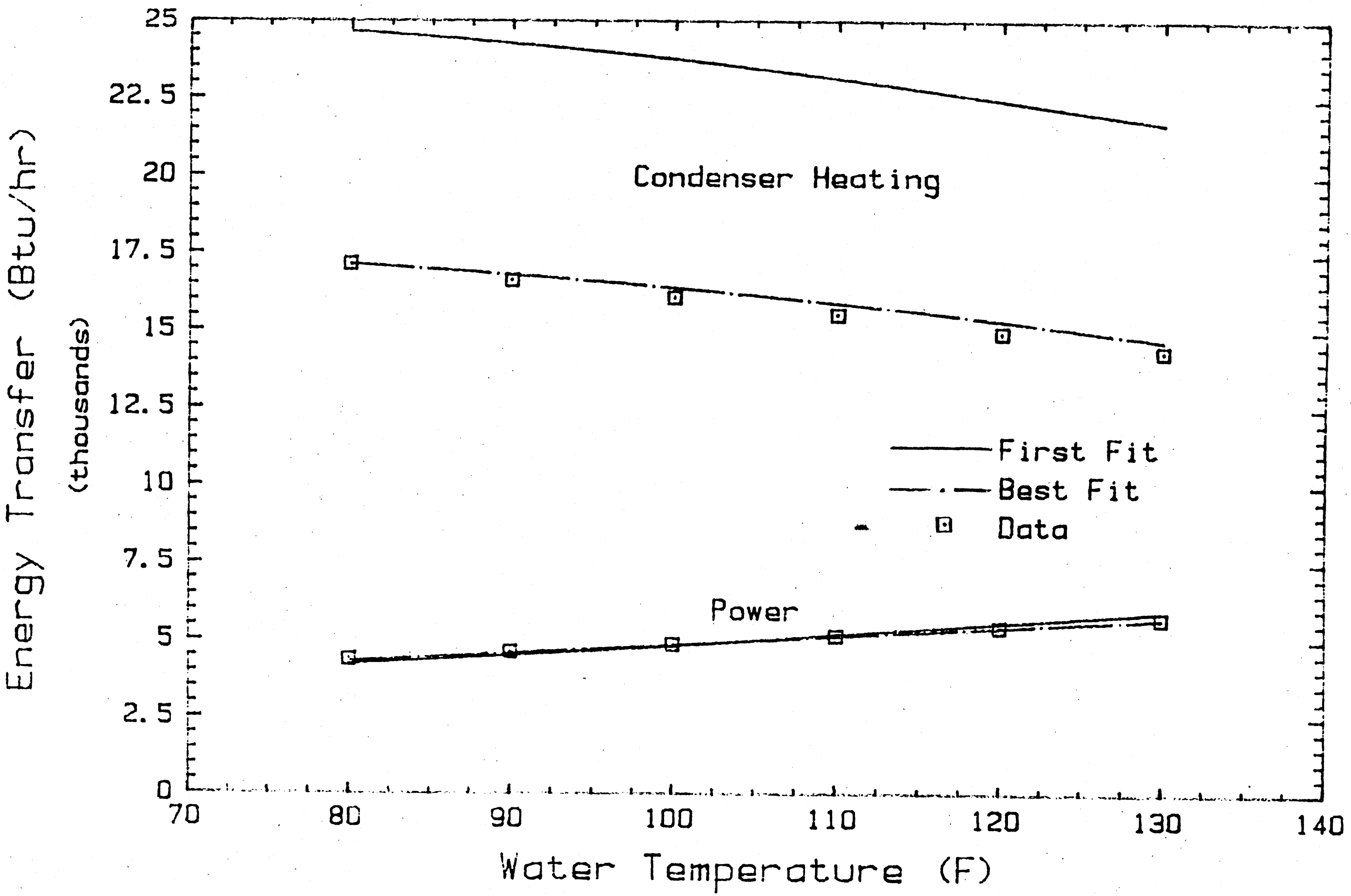

Figure 2.18 Comparison of vapor compression cycle model to performance data for both the initial inputs and and inputs giving the best fit

accurately predicted by the original inputs, but that the condenser water heating is over-predicted by about 50%.

The second column of input values in Table 2.2 are the values which were found to give the best comparison to the test data. This comparison is also given in Figure 2.18, showing that the model predicts the measured performance to within 3%. The largest change in the input values from the original values is the conductance-area product of the evaporator. The original value was approximated to be four times larger than that needed to give accurate results. Table 2.3 shows the comparison of the predicted condenser and evaporator pressures to data, using the input values giving the best fit to the power and capacity data. This comparison assumes the suction and discharge pressures given in Table 2.1 are equal to the pressures in the evaporator and condenser respectively. The model closely predicts the pressures for the higher water temperatures, but is off by up to 15% for the

Average Tank Temperature	Condenser Pressure Actual / Predicted	Evaporator Pressure Actual / Predicted
136.3	370 / 361	98 / 97
128.7	340 / 331	95 / 93
112.3	285 / 272	88 / 87
95.2	235 / 219	85 / 81
76.7	191 / 170	83 / 77
57.4	152 / 129	75 / 73

Table 2.3 Comparison of condenser and evaporator pressure to data

lower temperatures. The vapor compression cycle model is used next to predict heat pump performance for condenser and evaporator conditions different from the test values.

2.9 Curve Fitting Computer-Generated Performance Data

As mentioned in section 2.6, the vapor compression cycle model takes approximately a minute of computer time on a MicroVax to calculate each steady-state performance. The final HPWH model will be incorporated into an annual simulation program, which will require the heat pump performance to be calculated at least once an hour for each hour of the year. Using the vapor compression cycle model directly in this simulation would require at least 7 days of computer time for each simulation. Therefore, instead of using the vapor compression cycle model directly in the simulation program, it will be used to generate performance data over a range of operating conditions, which can be curve fit using a regression analysis.

In order to generate data for curve fitting, it is necessary to know what affects the performance of the heat pump under normal operating conditions. In general, the performance of a given heat pump is only affected by the environmental conditions of the evaporator and condenser, which for the heat pump water heater would be the water temperatures in each condenser, water flow rate through the tube-in-tube condenser, the kitchen air dry and wet bulb temperatures, and the air flow rate over the evaporator. The DEC heat pump has constant water and air flows in and over the condenser and evaporator respectively. The kitchen dry bulb temperatures will vary throughout the day and year, but these changes should be relatively small and therefore negligible. The wet bulb temperature will also vary throughout the day and year; however, the wet bulb temperature is only important when the water in the cooling air condenses on the evaporator. Since this seldom occurs (DEC, 1987), it is assumed that the air side of the evaporator is always dry and only performs sensible cooling. The performance of the heat pump, therefore, is only affected by changes in the water temperatures in the two condensers.

The heat pump performance data of interest are the heat transfers in the heat exchangers and the compressor input power. Water temperatures in the two condensers vary between 50° and 150°F with the tube-in-tube heat exchanger water temperature always being less than or equal to the water temperature in the wrap-around heat exchanger. Therefore, the vapor compression cycle model was used to generate performance data over this range of wrap-around and tube-in-tube heat exchanger water temperatures. This data was curve fit using linear regression analysis (Box, 1978) and the following equations were found to fit the data

$$E_{motor} = 1980 + 29.3\, T_{TT} - 0.0214\, T_{TT}^2 + 0.01607\, T_{TT} T_{WA} \quad \text{(Btu/hr)} \qquad (2.9.1)$$

$$Q_{WA} = 2880 + 55.0\, T_{TT} + \frac{264}{(T_{WA} - T_{TT})^{1/2}} - 0.1005\, T_{TT}^2$$
$$- 37.2\, T_{WA} + 0.0484\, T_{WA} T_{TT} \quad \text{(Btu/hr)} \qquad (2.9.2)$$

$$Q_{TT} = 13940 - 22.5\, T_{TT} - \frac{259}{(T_{WA} - T_{TT})^{1/2}} - 0.258\, T_{TT}^2$$
$$+ 38.7\, T_{WA} - 0.0830\, T_{WA} T_{TT} \quad \text{(Btu/hr)} \qquad (2.9.3)$$

Equations 2.9.1 - 3 all have R^2 values ranging between 0.995 and 0.998 and therefore are good fits to the generated data.

Figure 2.19 and 2.20 are plots of the heat transfer to the water by each condenser and of the motor power versus the water temperature of the wrap-around condenser. Curves are given for inlet water temperatures to the tube-in-tube condenser of 50° and 100°F, each showing good agreement between the generated data and the regression equations. This plot also shows that, for a constant inlet water temperature to the tube-in-tube condenser, the heat transfer in the tube-in-tube condenser increases and the heat transfer in the wrap-around heat exchanger decreases as the water temperature of the wrap-around condenser increases. The combined heat transfer to the water is basically constant with respect to changes in the water temperature of the wrap-around heat exchanger (i.e., the increase in the tube-in-tube heat transfer is equal to the decrease in the wrap-around condenser). This is because the condensing units are oversized relative to the other components in the vapor compression cycle, and therefore increasing the water temperature seen by the first condenser just shifts more of the cooling to the second condenser.

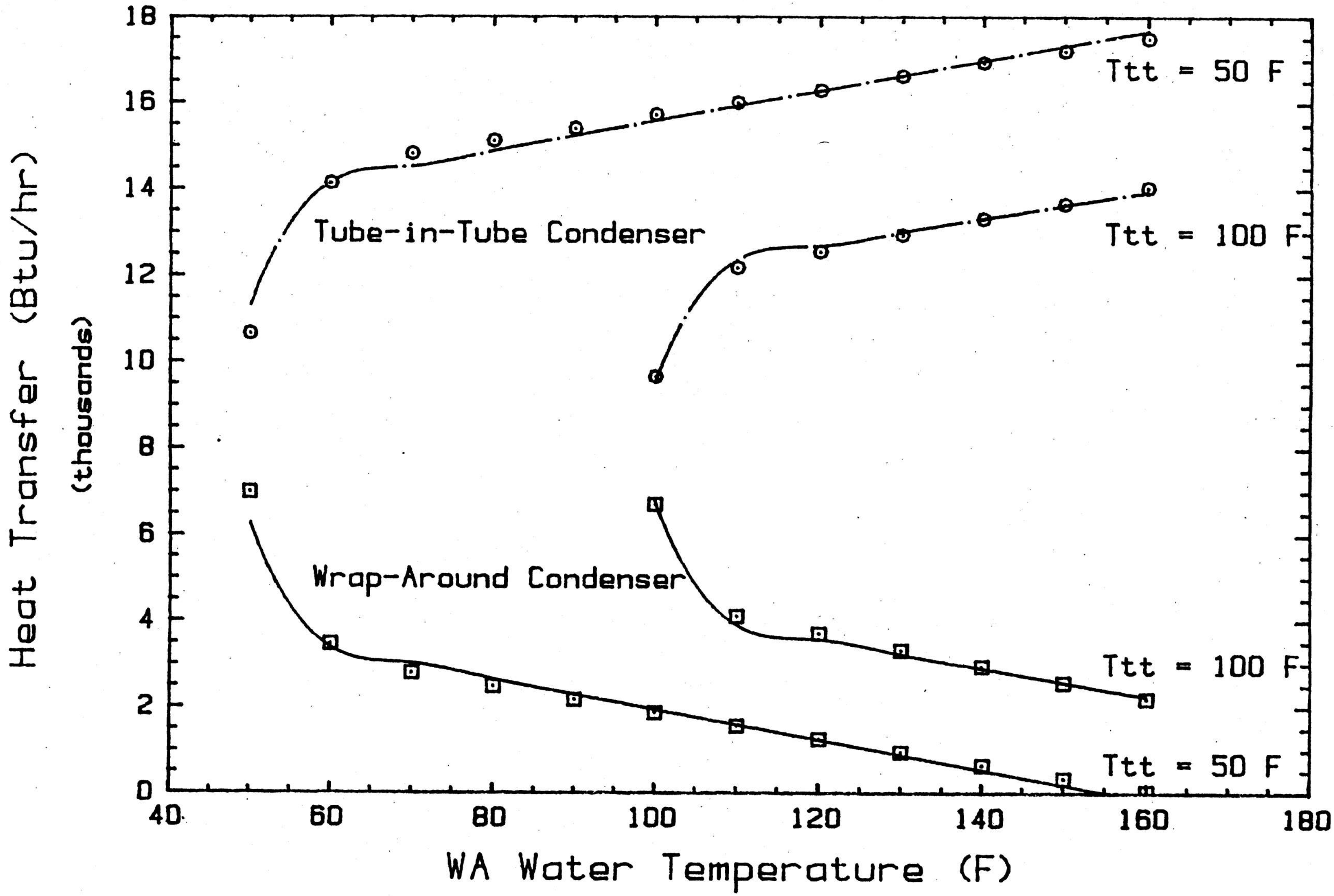

Figure 2.19 Curve fit of heat-transfer data generated by the vapor compression cycle model

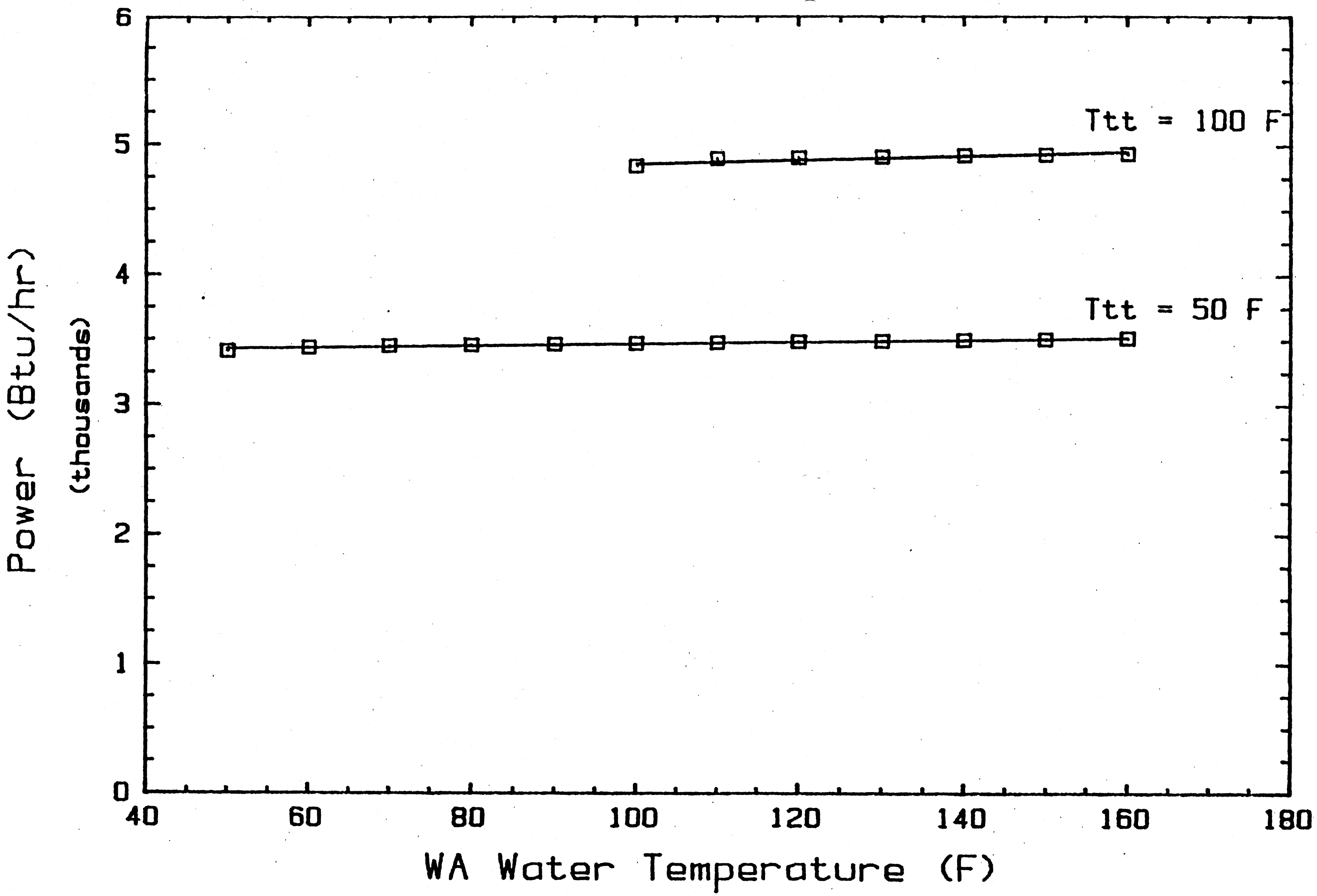

Figure 2.20 Curve fit of power data generated by the vapor compression cycle model

Chapter 3
Heat Pump Water Heater Model

3.1 Description of the DEC Heat Pump Water Heater

As shown in Figure 3.1, the DEC HP-120-27 water heater consists of a storage tank, a heat pump, two electric resistance backup heaters, and controllers for each heater. The hot water outlet is at the top of the tank; the cold water inlet is at the bottom. The wrap-around heat exchanger of the heat pump covers approximately the bottom 85% of the tank. When hot water is being drawn from the tank, an equal amount of cold water is supplied to the tank, keeping the total mass of water in the tank constant. While the heat pump is operating, the recirculation pump draws water through the tube-in-tube heat exchanger, which cools the refrigerant and heats the water. As previously shown, the heat pump's performance increases with cooler water being drawn through the tube-in-tube condenser. The heat pump water heater was designed, therefore, to supply the coolest water available to this heat exchanger. When the heat pump is operating and hot water is being drawn, the tube-in-tube heat exchanger uses the cold supply water; otherwise the tube-in-tube uses water from the bottom of the tank. For low hot water draw rates, a combination of the supply water and water drawn from the bottom of the tank is used to meet the required flow for the tube-in-tube heat exchanger. In each case, the recirculation pump attempts to draw all of the water from the supply line, but uses water from the bottom of the

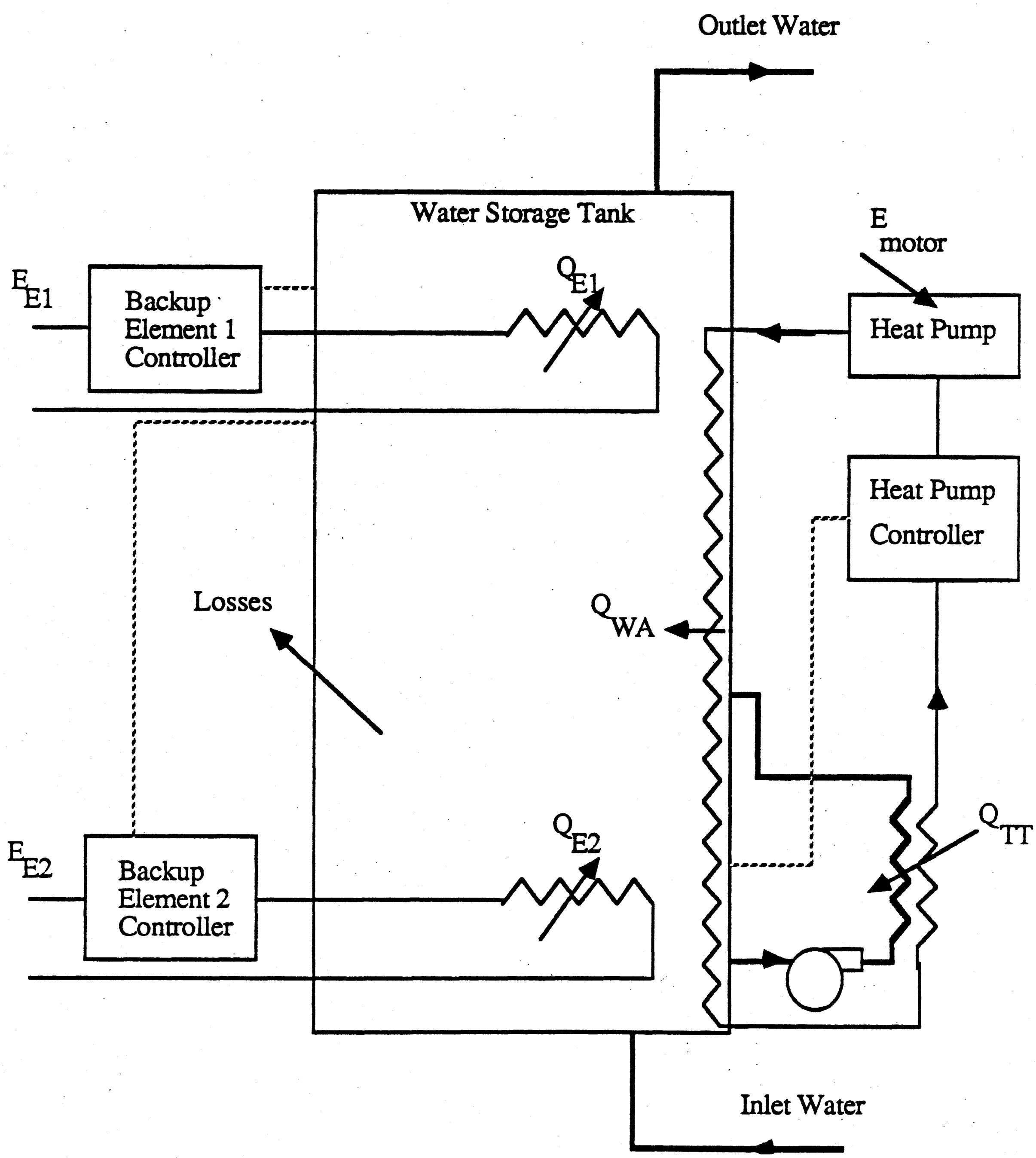

Figure 3.1 Schematic of the HP-120-27 heat pump water heater. Water lines are shown in bold lines, refrigerant in other lines.

tank to meet part or all of the required flow if there is limited or no hot water draw. The inlet for this water drawn from the tank is approximately a foot from the bottom of the tank, while the outlet is near the center of the tank.

The water heater has two 6 KW electric resistance heating elements (backup heaters) to help in water heating during periods of high hot water draws, and these heaters are located at the top and bottom of the tank. Each heater in the tank has an individual temperature controller, which is located near the top of the tank for the two backup heaters and at the bottom of the tank for the heat pump heater. The heat pump temperature controller is near the bottom of the tank because it is always the coolest, and therefore, the heat pump would be the first heater to turn on. The backup heaters' controllers, on the other hand, are located near the top of the tank so that they will turn on only if the water supplied to the load (i.e., water near the top of tank) is too cold. The heat pump turns on when the temperature at its sensor falls below 132°F and it turns off when heated above 140°F (DEC, 1987). The backup elements operate similarly, and the high and low control temperatures are 135°F, 127°F and 120°F, 112°F for the top and bottom elements respectively.

3.2 Water Storage Model

An important consideration when modeling a hot water storage tank is whether the tank is fully mixed or stratified. A fully mixed tank is one that has a uniform temperature throughout, whereas a stratified tank has warmer water near the top of the tank and colder near the bottom. A stratified tank is desirable for the DEC heat pump water heater since the water on the bottom of the tank, which is used by the heat pump tube-in-tube condenser, would be colder, thus increasing the heat pump's performance. Two operating conditions of this storage tank which cause stratification are: (1) drawing hot water from the top of the tank and replacing it with cold water at the bottom; (2) backup heating with the top element. A single heater in a tank, in general, tends to fully mix the water above it, and therefore supply equal heating to this water. Water heating by the heat pump heater only, therefore, tends to mix the entire tank. When one or both of the backup heaters is operating the mixing of the tank depends upon the relative size of the heaters, the operating condition of each heater, and the rate of hot water draw. Since both the performance of the heat pump and the control of the heaters depend upon the local tank temperature, a stratified tank model was used.

A stratified tank model is not in contradiction with the model of the wrap-around condenser, where it was assumed that the heat transfer in this condenser could be calculated using the mean water temperature in the portion of the tank covered by the condenser. The heat-transfer in this condenser, however, could be calculated more accurately if the water temperature at each level of the tank is known, but this would require the heat transfer, and therefore the performance of the heat pump, to be calculated at the same time as the tank temperatures, which, as noted in section 2.6, would require considerable computer time.

A simple method of modeling stratified tanks is to divide the tank into several horizontal sections, or nodes (Klein, 1983) as shown in Figure 3.2. Each node is modeled as fully mixed and accounts for heat addition by heaters, for losses to the environment, and for energy exchanges between adjacent nodes caused by water drawn through the node. An energy balance written about the i^{th} tank segment is

$$m_i c_p \frac{dT}{dt} = \dot{m}_{Draw} c_p \left(T_{i-1} - T_i \right) + \dot{Q}_i + (UA)_i \left(T_{env} - T_i \right) \tag{3.2.1}$$

The temperatures in each node are determined by integrating the energy balance equation for each node. For this model, the equations were integrated using Euler's integration, which assumes that changes occur slowly enough that the derivatives can be approximated by a forward difference method. For the i^{th} node, the temperature equation becomes

$$T_i^t = T_i^{t-\Delta t} + \frac{\Delta t}{m_i c_p}\left[\dot{m}_{Draw} c_p \left(T_{i-1}^{t-\Delta t} - T_i^{t-\Delta t}\right) + \dot{Q}_i^{t-\Delta t} + (UA)_i \left(T_{env} - T_i^{t-\Delta t}\right)\right] \tag{3.2.2}$$

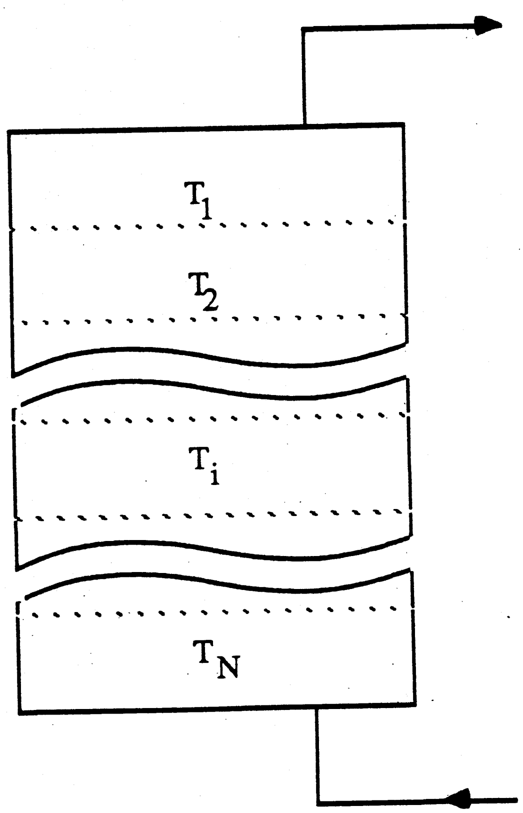

Figure 3.2 Diagram of general stratified tank model

The water temperature in any node cannot be greater than the water temperature in the node above it, since the hot water would rise from the lower node to the higher. Therefore, the model assumes that energy supplied to the tank by heaters is added into the node specified as having that heater, until the temperature of that node is equal to that of the node above it. These nodes are now considered to be mixed and energy is then added to them equally.

The computer model, therefore, checks the node temperatures calculated using equation 3.2.2, at the end of each time step in the Euler solution. If the temperature in any node is greater than the node above it, then the nodes are mixed and the new mixed temperature of the nodes is calculated by

$$T = \frac{m_i T_i + m_{i+1} T_{i+1}}{m_i + m_{i+1}} \tag{3.2.3}$$

3.3 Combined HPWH Model

A three-node tank model was chosen and the nodes are shown in Figure 3.3. The first node is the bottom of the tank and includes the inlet and outlet water from the tube-in-tube condenser, the lower backup element, the lower portion of the wrap-around condenser, the heat pump controller, and the cold water inlet. The second node is the middle of the tank and includes the top portion of the wrap-around condenser and the controller for the bottom backup element. The third node is the top of the tank and includes the top backup element, the controller for the top element, and the hot water outlet. The bottom, middle, and top nodes are 42%, 40%, and 18% of the tank respectively (DEC, 1987).

The performance of the heat pump is defined in terms of the mean water temperature seen by the wrap-around condenser and the water temperature supplied to the tube-in-tube heat exchanger. The mean water temperature of the wrap-around condenser is equal to the average temperature of nodes 1 and 2 and is calculated by

$$T_{WA} = \frac{Ptnk_1\, T_1 + Ptnk_2\, T_2}{Ptnk_1 + Ptnk_2} \qquad (3.3.1)$$

where

$Ptnk_1$ = percent of volume in node 1 (42%)

$Ptnk_2$ = percent of volume in node 2 (40%)

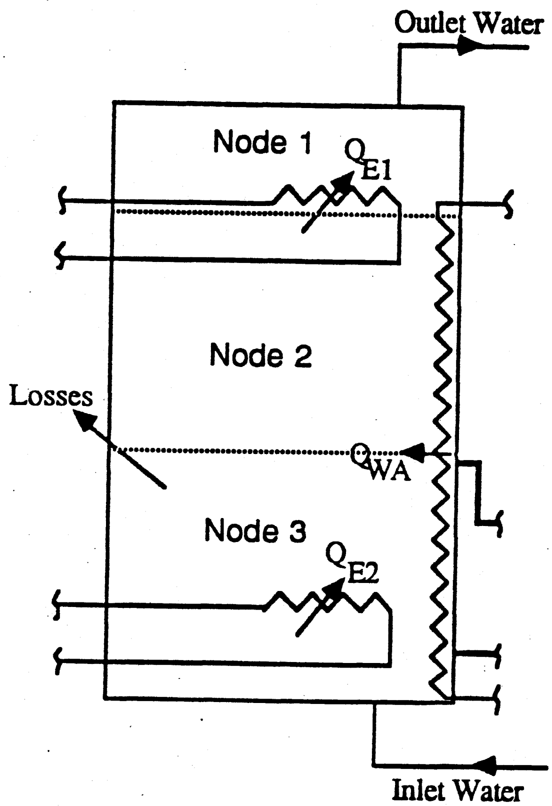

Figure 3.3 Node description for the HP-120-27 stratified tank model

T_1 = temperature in node 1

T_2 = temperature in node 2

The temperature of the water supplied to the tube-in-tube condenser depends upon the operating conditions and is calculated by the following:

$$\text{if } \dot{m}_{Draw} > \dot{m}_{TT}, \quad T_{TT} = T_{mains}$$

$$\text{if } \dot{m}_{Draw} = 0, \quad T_{TT} = T_1$$

$$\text{if } 0 < \dot{m}_{Draw} < \dot{m}_{TT}, \quad T_{TT} = \frac{T_{mains}\,\dot{m}_{Draw} + T_1\left(\dot{m}_{TT} - \dot{m}_{Draw}\right)}{\dot{m}_{TT}} \tag{3.3.2}$$

The water heating by the wrap-around condenser going to nodes 1 and 2 is weighted according to the area of the wrap-around condenser in each node, and is given by

$$Q_{wa,1} = Pwa_1\, Q_{wa}$$
$$Q_{wa,2} = Pwa_2\, Q_{wa} \tag{3.3.3}$$

where Pwa_1 and Pwa_2 are the percentage of the wrap-around condenser in each node and are equal to 51% and 49% respectively (DEC, 1987), and Q_{wa} is given by equation 2.9.2. The water heating and power consumption of an electric resistance heater are equal, and for this unit are 6 KW.

The overall coefficient of performance of the HPWH depends upon the amount of heating performed by each heater and upon the coefficient of performance of the heat pump when it is operating (recall that $Q_{E1} = E_{E1}$ and $Q_{E2} = E_{E2}$)

$$COP_{HPWH} = \frac{Q_{HP} + Q_{E1} + Q_{E2}}{E_{motor} + E_{E1} + E_{E2}} \tag{3.3.4}$$

3.4 Model Comparison to Data

The DEC HP-27-120 data given in Table 2.1 is of a cold tank being heated from 57.4 °F to 136.3 °F, using only the heat pump heater (i.e., no backups) (DEC, 1987). By turning off the backup heaters in the HPWH model, it can be operated in a manner similar to the test, and the results can be compared to the data. Since the vapor compression cycle model was forced to fit the heat pump performance data, this will not be a judge of the accuracy of the model in general; rather, it will only verify that the HPWH model has been programed as outlined. This comparison was made and the results are graphically shown in Figure 3.4. As expected, this figure shows agreement between the predicted and actual tank temperatures. The time to heat the tank was 4.5 hours with an overall coefficient of performance of 3.3.

3.5 Parametric Study of the HPWH's Performance

The performance of the HPWH is usually judged by its coefficient of performance and by the maximum number of gallons of hot water that it can deliver during an hour. As noted in section 3.3, the overall coefficient of performance of the heat pump water heater depends upon both the amount of water heating supplied by each heater and on the coefficient of performance of the heat pump. In actual operation, the heat supplied by each heater will depend upon the water draw

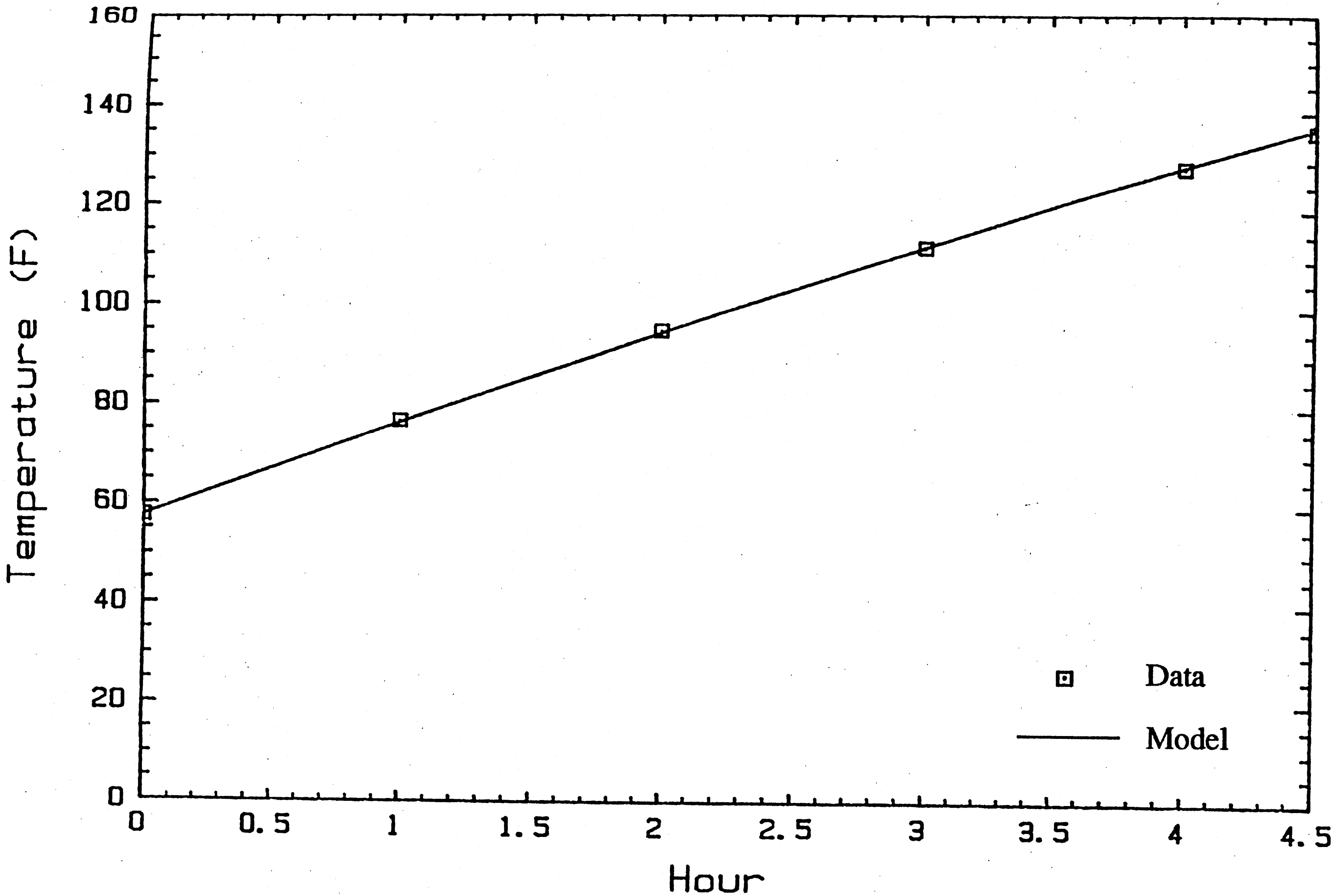

Figure 3.4 Comparison of the heat pump water heater model to data

schedule, so the overall coefficient of performance will vary with the rate and duration of draw. The maximum COP of the water heater is obtained when no heating is done by the backup elements. The maximum number of gallons of hot water that can be delivered, however, increases with greater use of the backup elements. Thus, there is a trade-off between the COP of the water heater and the maximum rate at which hot water is drawn.

There are an infinite number of possible daily hot water draw schedules for the water heater in restaurant applications. The three types of draw schedules which are considered here are shown in Figure 3.5. The first schedule is uniform draw over a single period; the second and third schedules are uniform and equal draw over two and three periods respectively. Since the major use of hot water in restaurants is for dish washing and cleanup, draw schedule 1 is similar to a restaurant that washes dishes and cleans evenly throughout the day; schedule 2 is similar to a restaurant serving lunch and dinner and cleaning up immediately after each meal; and schedule 3 is similar to a restaurant serving three meals a day and cleaning up after each one. The time between the periods of draw is assumed to be long enough that the tank can be completely reheated before the beginning of the next draw period. The heat pump water heater model predicts a recovery time of 2.5 hours to heat up a tank from 55°F to 140 °F. In practice, the mixed tank temperature is never as cold as 55 °F, so the maximum recovery time is considered to be two hours.

Assuming the tank recovers completely between periods of draw allows each schedule to be viewed as uniform draw over a single time span. Schedules 2 and 3 are still considered to have 2 and 3 periods of draw per day respectively, but since the tank fully recovers between these periods, the performance during each period is equal to and independent of the other draw periods. The single time span for draw

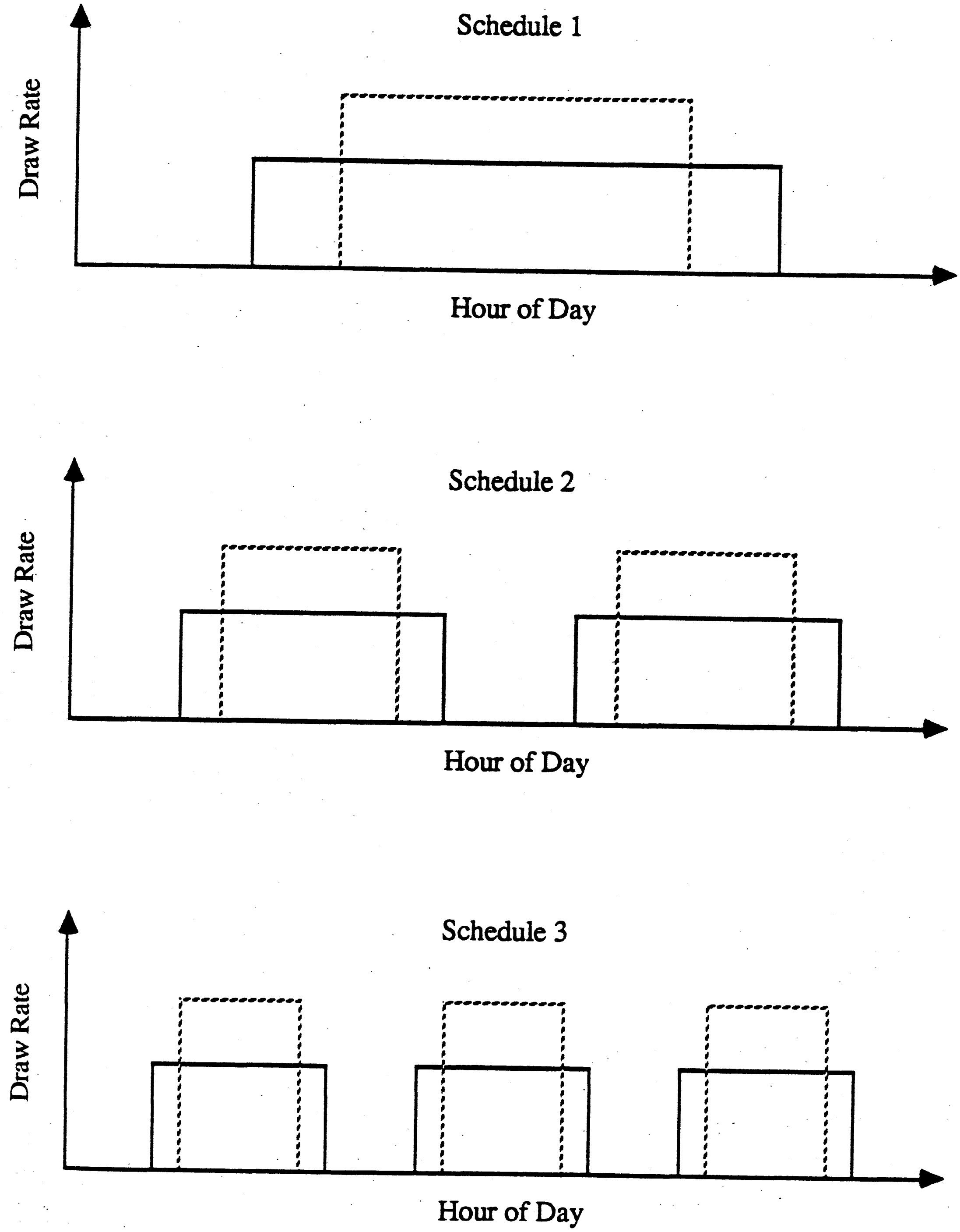

Figure 3.5 Plots of three types of hot water draw schedules, with two possible distributions given for each schedule. Each plot has the same total daily draw.

schedule 1 can have a maximum draw time of 22 hours (24 hours in a day minus the 2 hour recovery); draw schedules 2 and 3 have maximum times of 10 and 6 hours respectively. In the first schedule, the total daily draw is distributed over the time span; in the second, half the total draw is distributed over the single time span; in the third, one-third of the total is distributed over the single time span.

Using the HPWH model, graphs were made of the overall coefficient of performance of the heat pump water heater versus the rate of hot water draw from the tank. These are shown in Figures 3.6-8 for draw schedules 1-3 respectively. In each graph three curves are given corresponding to total daily draws of 300, 500, and 700 gallons. The inputs to the program for these graphs are given in Table 3.1, and the evaporator inlet air temperature is 75 °F.

Each of these plots shows that the performance has a maximum performance between draws of 25 and 30 gallons per hour and that the overall coefficient of performance varies between 1.3 and 3.0. The initial increase in performance with increased draw rate occurs while the heat pump heater is meeting the load without use of the backups. The performance increases because a larger portion of the water flow through the tube-in-tube condenser is the cold supply water, which increases the effectiveness of this condenser, and therefore increases the performance of the heat pump. The performance decreases when the backup heaters begin to contribute to the water heating, and it continues to decrease as the draw rate is increased, because the backup heaters contribute a larger and larger portion of the total heating.

These plots also show that the performance decreases with increased total daily draw for draw rates which are greater than the draw at the maximum performance, but that the shapes of the curves are similar. The backup heaters are off until the temperatures in the top two nodes of the tank fall below the set point temperatures of

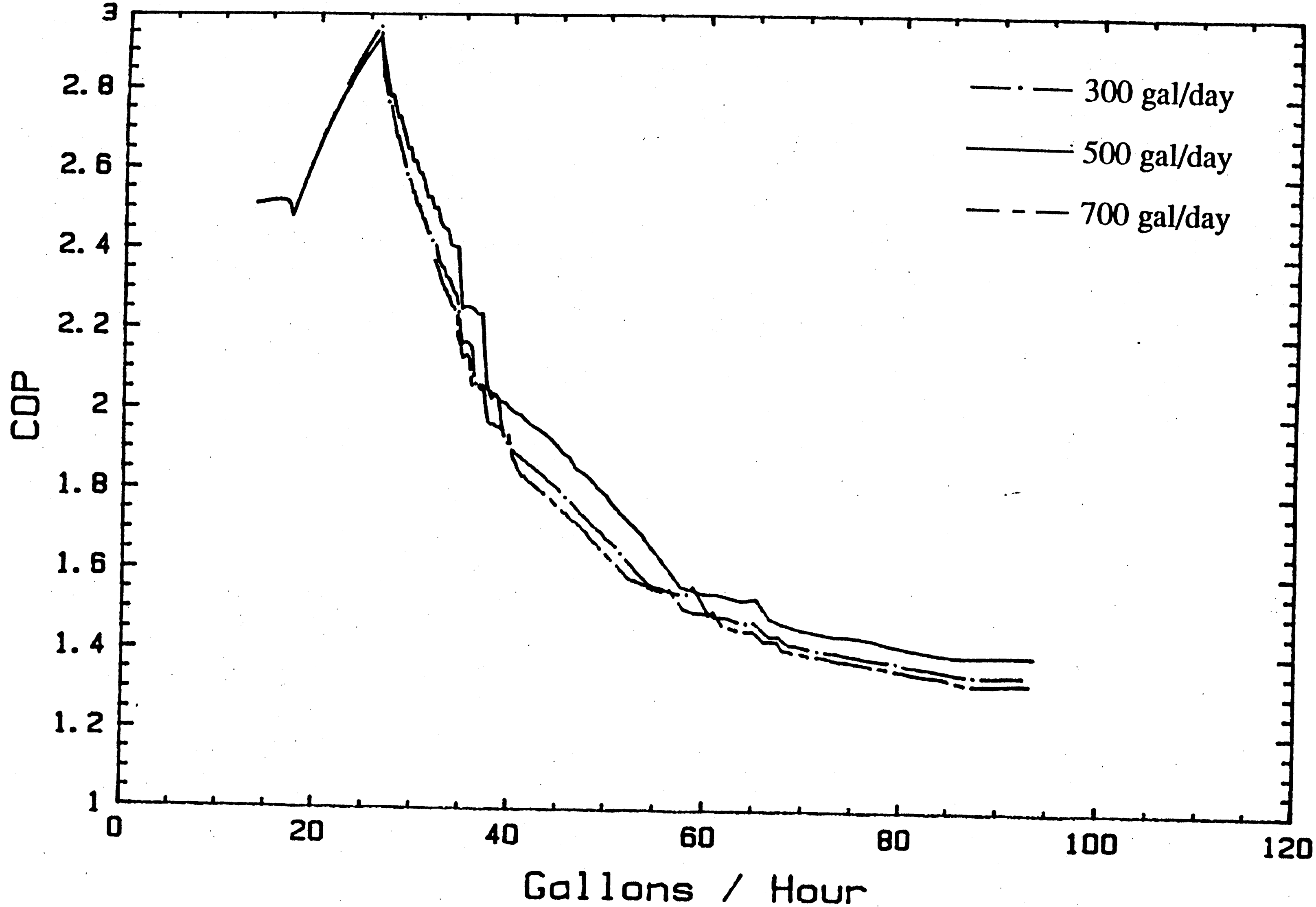

Figure 3.6 Performance of the HPWH for draw schedule 1

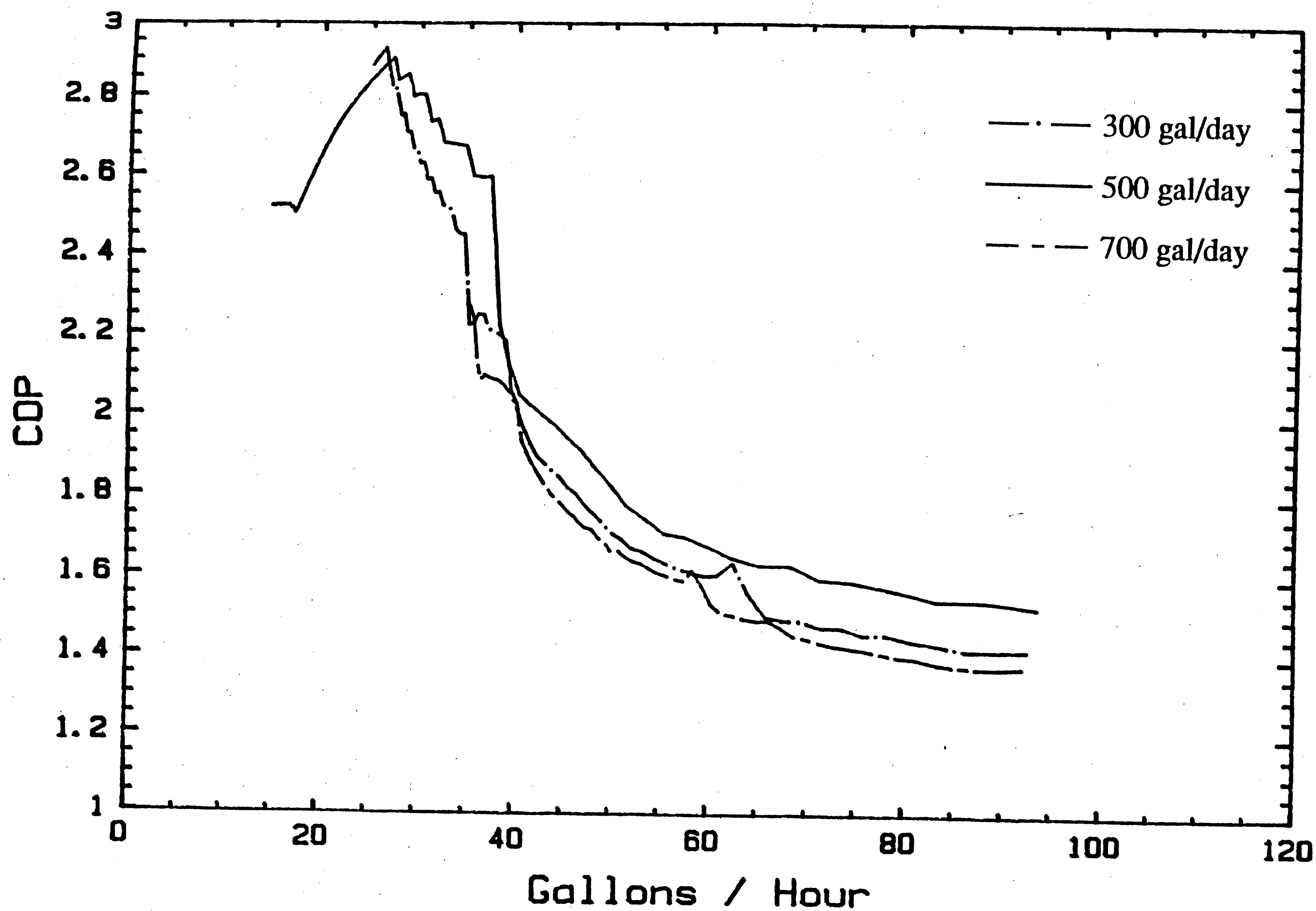

Figure 3.7 Performance of the HPWH for draw schedule 2

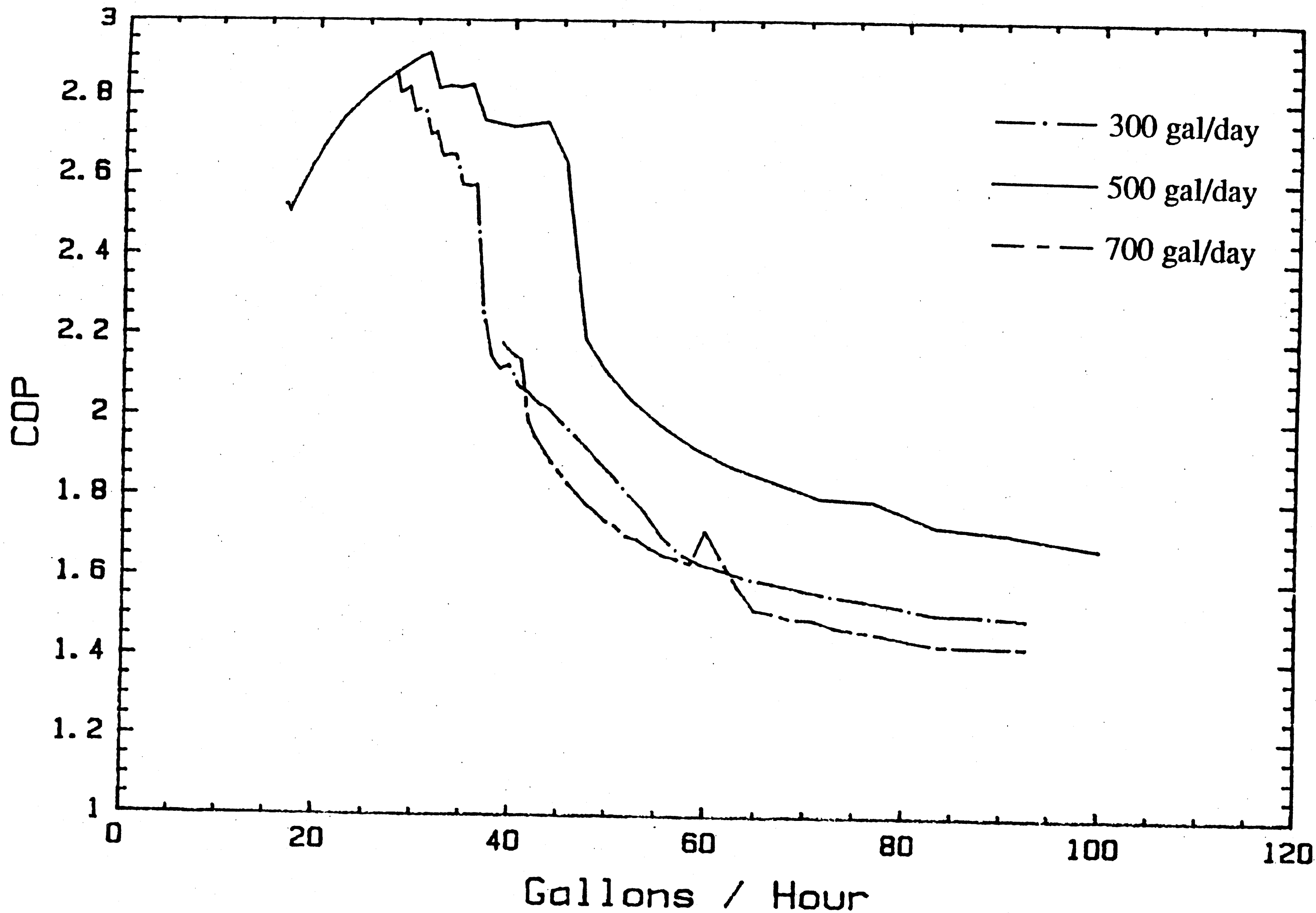

Figure 3.8 Performance of the HPWH for draw schedule 3

Variable	Units	Input Value
$(UA)_{tank}$	Btu/hr-°F	0.0
T_{env}	°F	75
V_{tank}	ft^3	14.7
$T_{hp,h}$	°F	140
$T_{hp,l}$	°F	132
$\dot{m}_{H2O,TT}$	lbm/hr	2000
T_{mains}	°F	55
$T_{e1,h}$	°F	135
$T_{e1,l}$	°F	127
$T_{e2,h}$	°F	120
$T_{e2,l}$	°F	112
Pwa_1		0.551
Pwa_2		0.489
Pwa_3		0.0
$Ptnk_1$		0.421
$Ptnk_2$		0.404
$Ptnk_3$		0.175

Table 3.1 Inputs to the HPWH program.

these heaters. The longer the period of draw for a given flow rate (greater than the draw associated with maximum performance), therefore, the larger the portion of total heating by the backup elements. For example, if hot water is drawn at a given rate just up to the point at which the backup element was ready to turn on, and then stopped, the water heater would have the same coefficient of performance as the heat pump. Increasing the period of draw further would degrade this performance because a portion of the heating would be supplied by the backup heaters; the portion of backup heating would increase for even longer periods of draw. The performance decreases, therefore, with increased daily draw, because it takes a longer period of time to supply the total draw, allowing the backup elements to contribute a larger portion of the total heating.

The jagged shapes of the curves are caused by the backup elements cycling on and off. The 300 gallons/day curve on Figure 3.7, for example, decreases in performance between 26 and 37 gallon/hr in a step-like manner due to the cycling of the top backup heater. When the draw rate is increased slightly above 26 gallons per hour, the temperature in the top node of the tank at the end of the draw period falls below the setpoint temperature of the top backup heater, and thus the heater turns on and decreases the performance of the water heater. Increasing the draw rate further causes the backup element to turn on earlier in the draw period, allowing it to do a larger portion of the total water heating, which further reduces the overall performance. When the backup heater is on long enough to completely reheat the top element, this heater is turned off, and increasing the draw rate again results in increased performance. Each following step in performance in this region is an additional cycling of the top backup heater. The large drop in performance at 37 gallons/hour is caused by the bottom element turning on and completely heating the bottom two nodes of the tank. The rise in performance at 62 gallons/hour occurs because both the top and bottom backup heaters have been turned off, and the heat pump performance increases with increased draw rate.

Comparing all nine curves in Figures 3.6-8 shows that the overall coefficient of performance of the water heater is largely affected by the rate of draw, and neglecting the curve of 300 gallons per day on schedule 3, it is seen that the performance is basically independent of the draw schedule. The water heater performs differently for schedule 3 with 300 gallons/day because the hot water load is met largely by the hot water in the tank at the beginning of the draw. Since the total daily draw is distributed evenly to the three periods of draw per day, the draw over the single period studied is 100 gallons, which is less than the 110 gallons in the storage tank.

In generating the data for Figures 3.6-8, the draw rate was increased until the temperature in the top node continued to decrease, even with the top backup element on. This indicates that the water heaters are incapable of meeting the heating load created by this draw schedule. The maximum draw rate of hot water is seen from the graphs to be about 93 gallons/hour for each schedule, except for the curve of 300 gallons/day on schedule 3. As mentioned in the previous paragraph, the draw for this schedule is less than the volume in the storage tank, and therefore, it is expected to supply the entire draw in a single hour (100 gal/hr).

Figure 3.9 is a plot of the node temperatures in the tank versus time for one draw period of schedule 2 having a total draw of 500 gallons/day drawn at a rate of 50 gallons/hr. The water temperature in the top node fluctuates between 127 °F and 135 °F, which are the setpoint temperatures of the top backup heater. The temperature falls in this node until the temperature is below 127 °F, at which point

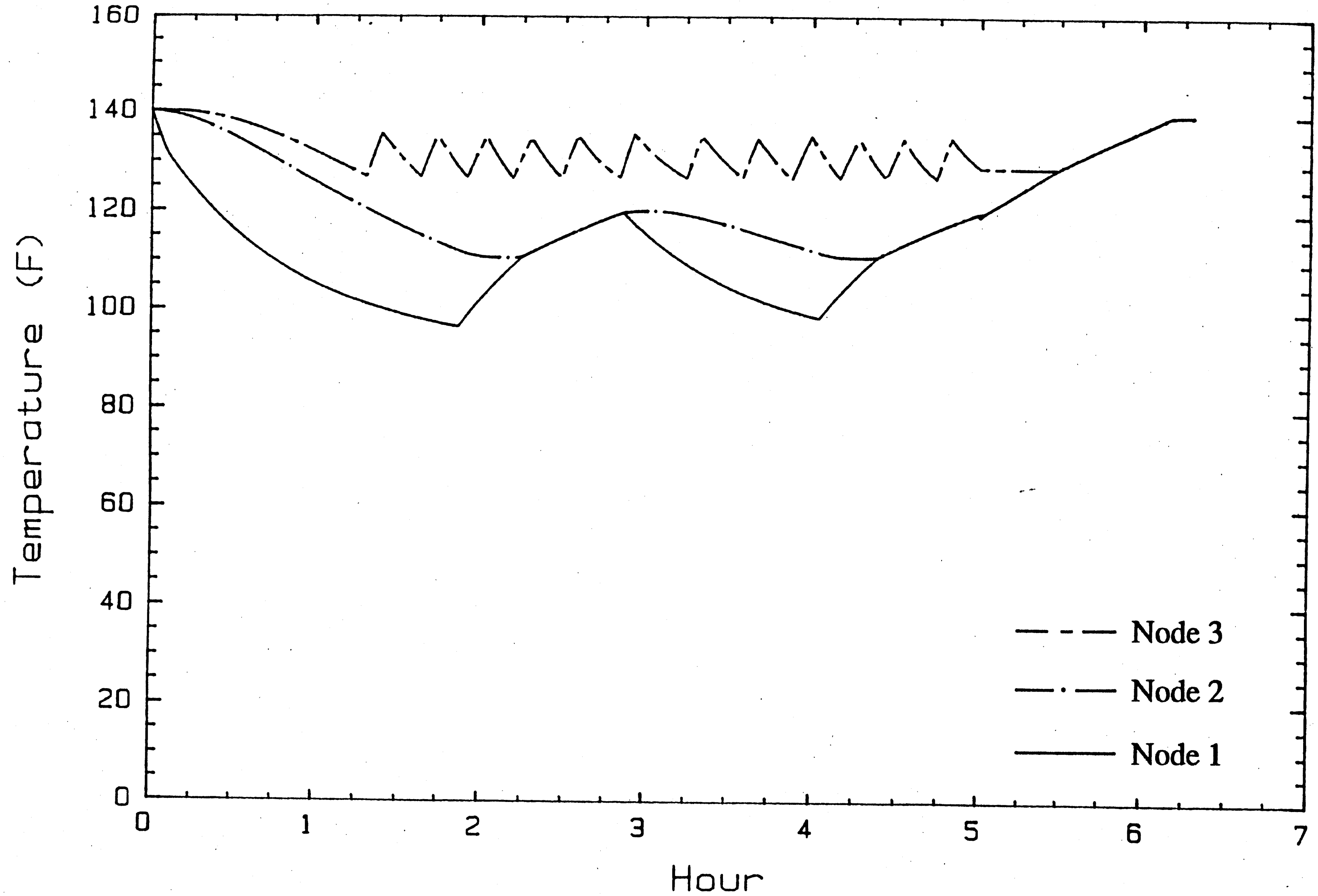

Figure 3.9 Node temperature versus time for draw schedule 2, 500 gal/day at 50 gal/hr

the top backup heater is turned on, and the node is heated to 135 °F. The water temperatures in the bottom two nodes fall until the temperature in the middle node is below the setpoint temperature for the bottom backup heater, at which point the bottom backup heater is turned on. This backup heater heats the bottom node to a temperature equal to the middle node's, and then heats both nodes equally to the high setpoint temperature of the bottom backup heater.

In order to study the effects of changing several parameters of the HPWH model, the base performance is taken to be described by schedule 2 having a daily draw of 500 gallons. As mentioned in the previous paragraphs, both the overall coefficient of performance and the maximum rate of draw of the HPWH are basically independent of both the draw schedule and the total daily draw, and therefore the changes in performance of this schedule should be typical of the changes for the other schedules.

Several parameters of the heat pump water heater are varied in order to determine their effects on the overall performance. The parameters varied are:

1) evaporator inlet air temperature,

2) storage volume, and

3) setpoint temperatures of the top backup heater.

Figure 3.10 (page 87) is a plot of the performance of the HPWH versus draw rate for evaporator air temperatures of 75 °F (base case) and 85 °F. The figure shows that the performance increases by about 20% at low draw rates and by about 6% at high draw rates for this increase in temperature. Figure 3.11 (page 88) is a plot of the performance for three storage tank volumes which shows that performance increases about 5 - 10% by increasing the storage volume by 50%, and decreases 5 - 10% by decreasing the storage volume 50%. A plot of the performance for three different setpoint temperatures of the top backup heater is given in Figure 3.12 (page 89). In each case the difference between the high and low setpoint temperatures is 8°F. Increasing the setpoint temperatures from 127°F and 135°F to 132°F and 140°F decreases the performance by about 5%, while decreasing these temperatures to 122°F and 130°F increases the performance by about 5%.

The graph of the node temperatures versus time given in Figure 3.9 shows that the bottom backup heater is turned on in the middle of the draw period even though the water supplied to the load (node 1) is maintained within the setpoint temperatures of the top backup element. The bottom heater completely heats the bottom two nodes to its high setpoint temperature. Since water heating by the backup heaters is less efficient than water heating by the heat pump heater, the bottom nodes of the tank could be heated more efficiently using only the heat pump. For the draw schedule of Figure 3.9, turning off the bottom backup heater increases the overall coefficient of performance from 1.71 to 2.06. Eliminating this resistance heater completely from the water heater, however, would also decrease the maximum number of gallons of hot water that it could deliver in an hour, since the overall water heating would be reduced.

An alternate water heater design could have a single backup element at the top of the tank having twice the heating capacity (ie. 12 KW). Figure 3.13 (page 90) graphically compares the overall performance of the existing water heater to water heaters having a single backup heater at the top of the tank of either 6 or 12 KW. This plot shows the overall coefficient of performance versus draw rate for each water heater subjected to the base draw schedule. The performances of the water heaters having the single backup element are equal to the existing water heater's performance for low draw rates, but are about 20% higher at high draw rates. The water heater with the single 6 KW backup element can supply a maximum of 61 gallons of hot water per hour, compared to 93 for the existing water heater, and 100 for the water heater with a 12 KW backup heater. This graph, therefore, indicates that both the performance and the maximum hot water supplied per hour can be increased by having a single backup heater of 12 KW at the top of the tank.

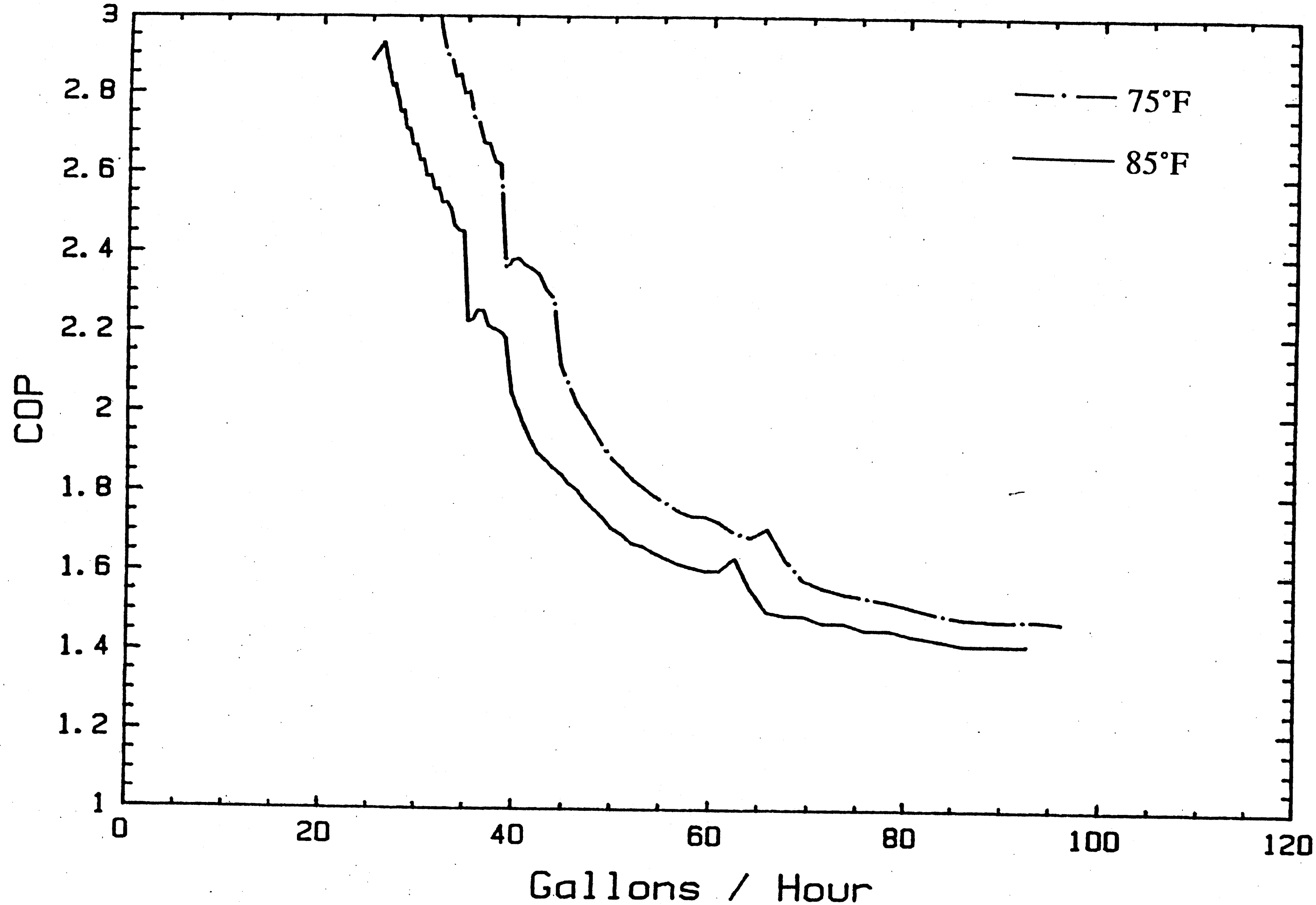

Figure 3.10 Overall coefficient of performance versus draw rate for evaporator temperatures of 75°F and 85°F

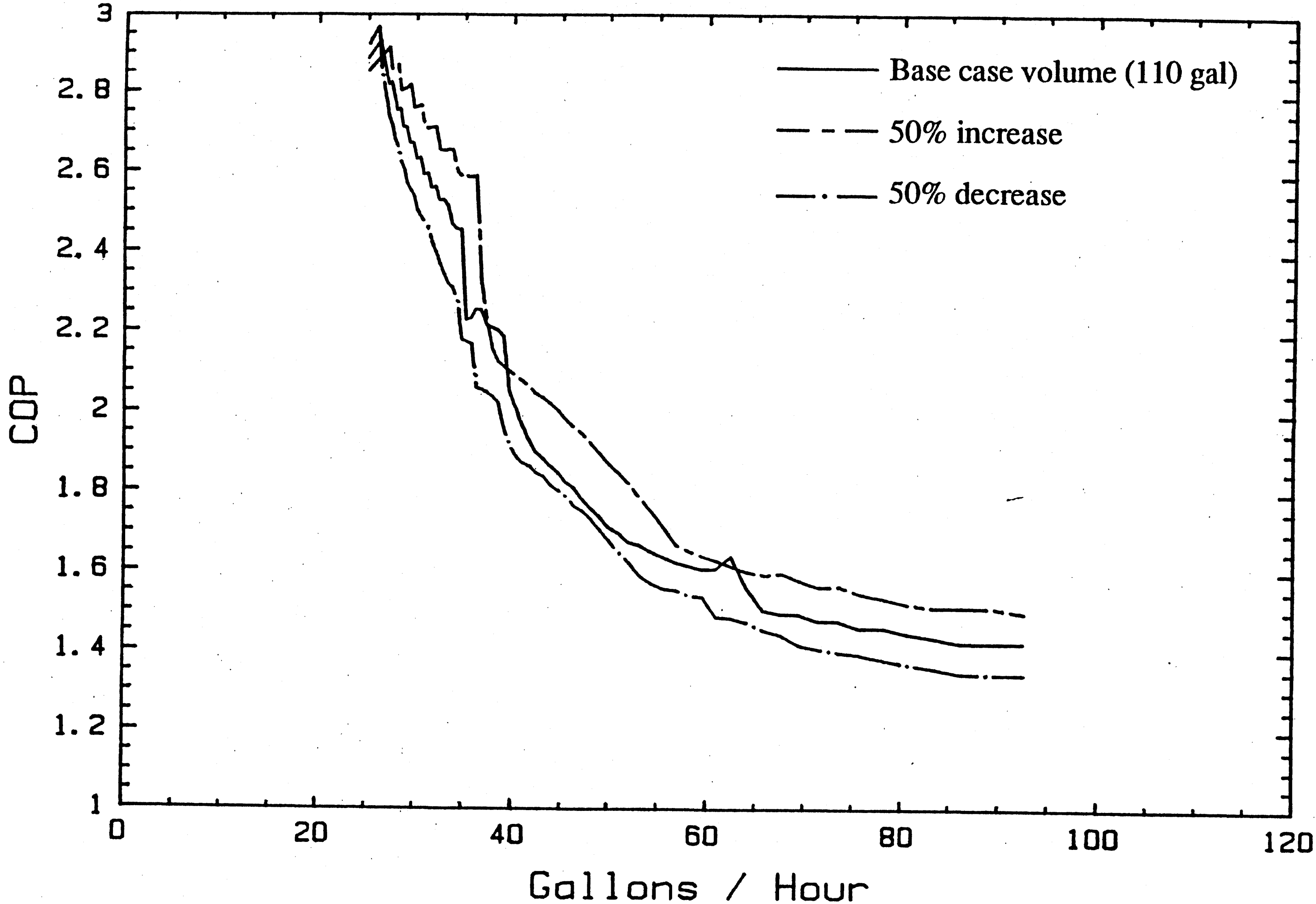

Figure 3.11 Overall coefficient of performance versus draw rate for storage volumes of 50%, 100%, and 150% of the base case volume.

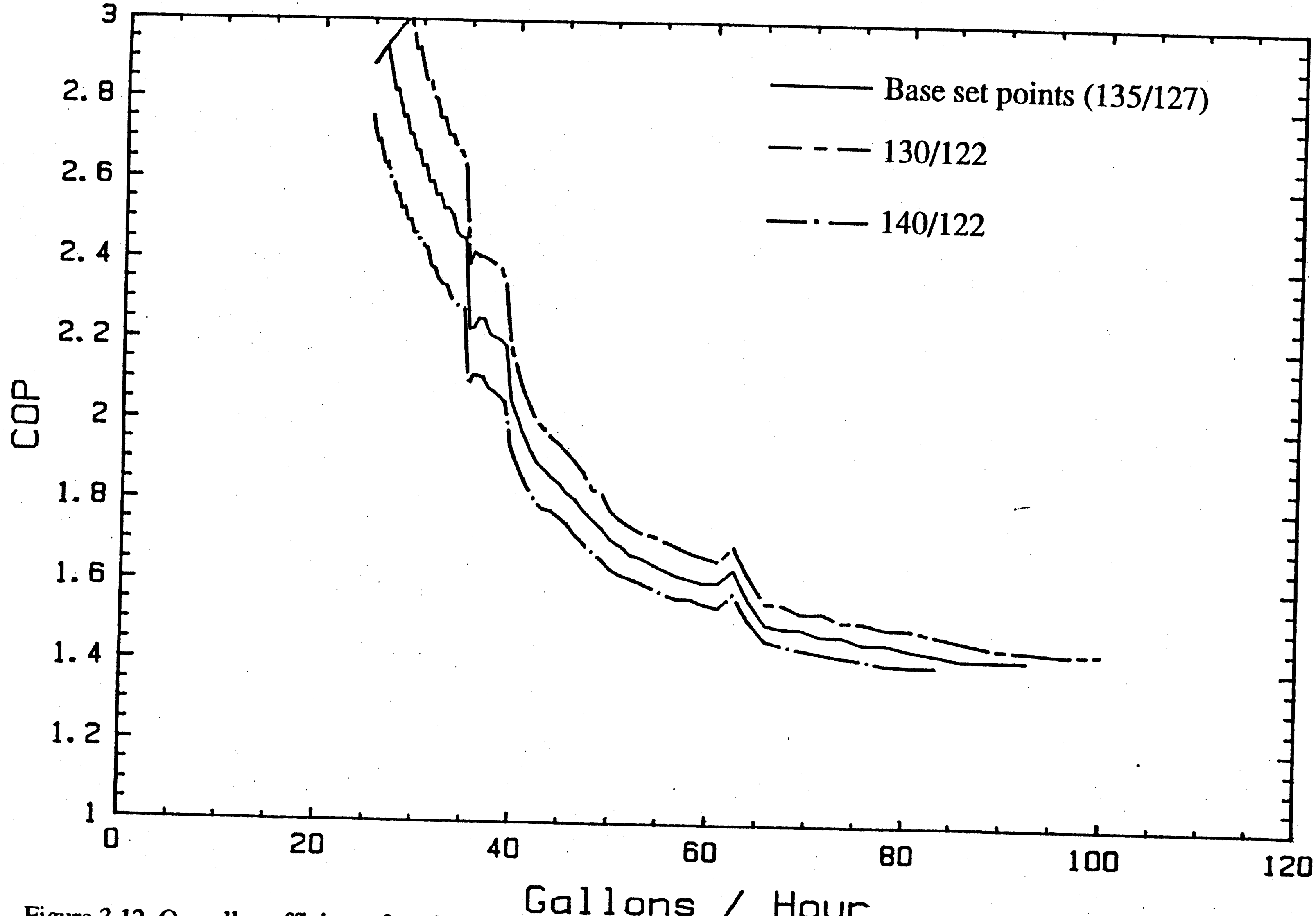

Figure 3.12 Overall coefficient of performance versus draw rate for three pairs of set point temperatures for the top backup heater

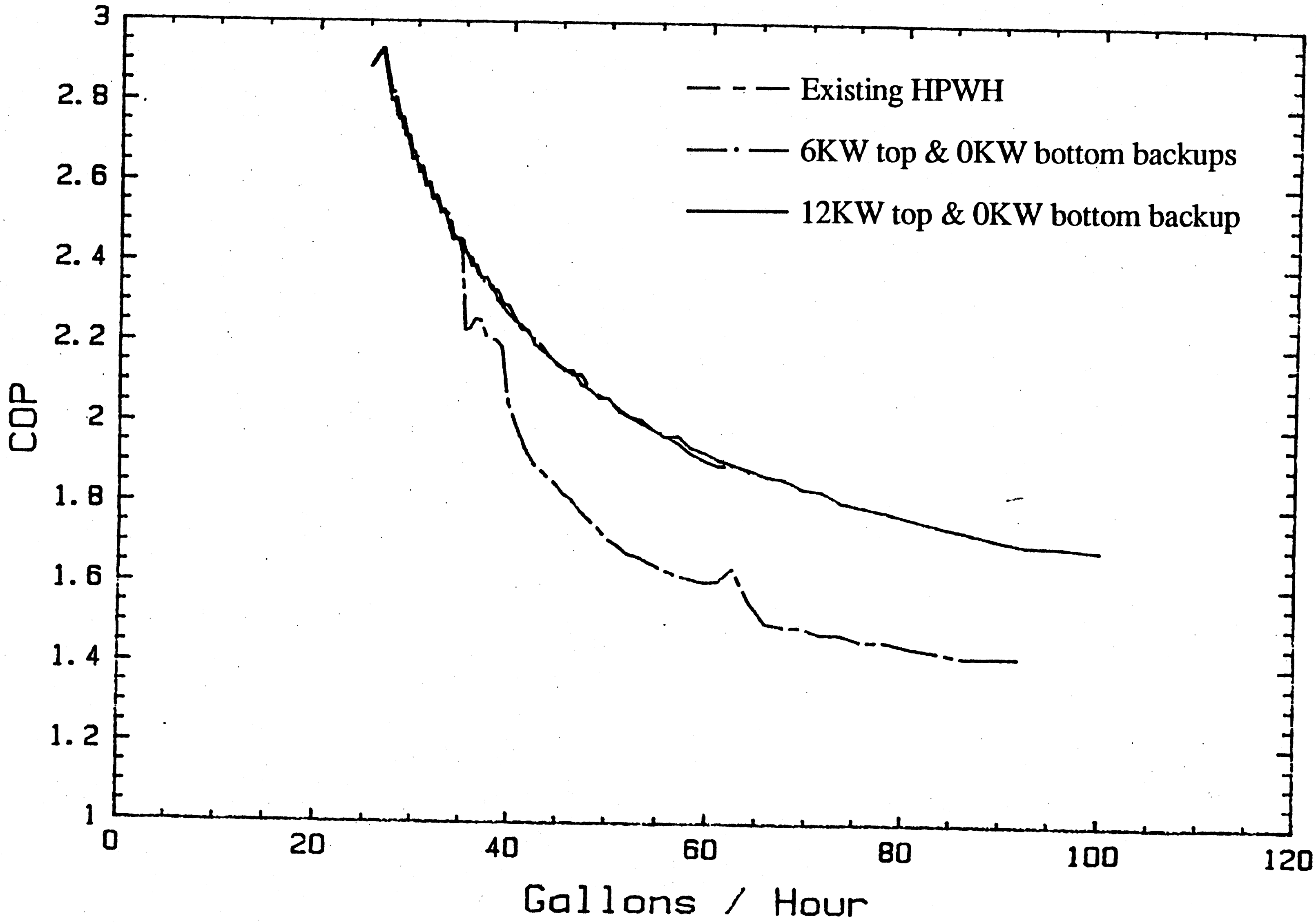

Figure 3.13 Performance of three HPWHs having different backup heaters

Chapter 4
Restaurant Building Model

4.1 General Description

The air cooling produced by the heat pump over a year may be either a net air conditioning savings or a net heating expense. In order to determine whether this cooling is useful, a building model is required to predict cooling and heating loads. The building model developed calculates hourly space-conditioning requirements considering effects such as inside and outside temperatures, skin losses/gains, ventilation, infiltration, and internal gains caused by cooking, people, and electric lighting and appliances. Throughout this chapter, heat added to the building is referred to as a heat gain, while heat removed is labeled a heat loss.

Figure 4.1 is a schematic of the building model showing the major heat flows and gains being considered. Data from several restaurants of monthly electric, gas, and water usage is used to estimate the magnitude of the heating, cooling, and hot water loads. The purpose of this model, however, is to generate space conditioning loads that could exist in a restaurant, rather than to predict exactly the cooling and heating loads of a specific building.

For this study, the restaurant is assumed to be open for 18 hours per day, opening at 6 o'clock in the morning and closing at 12 o'clock in the evening. The building is assumed to have one heating zone that is perfectly controlled to a specified temperature. It is further assumed that the heat pump water heater's performance neither affects nor is affected by the building latent load; therefore, the latent load will be neglected from the building model. The remaining chapter separately describes the calculation of each heat flow and gain and then of the combined building space conditioning load.

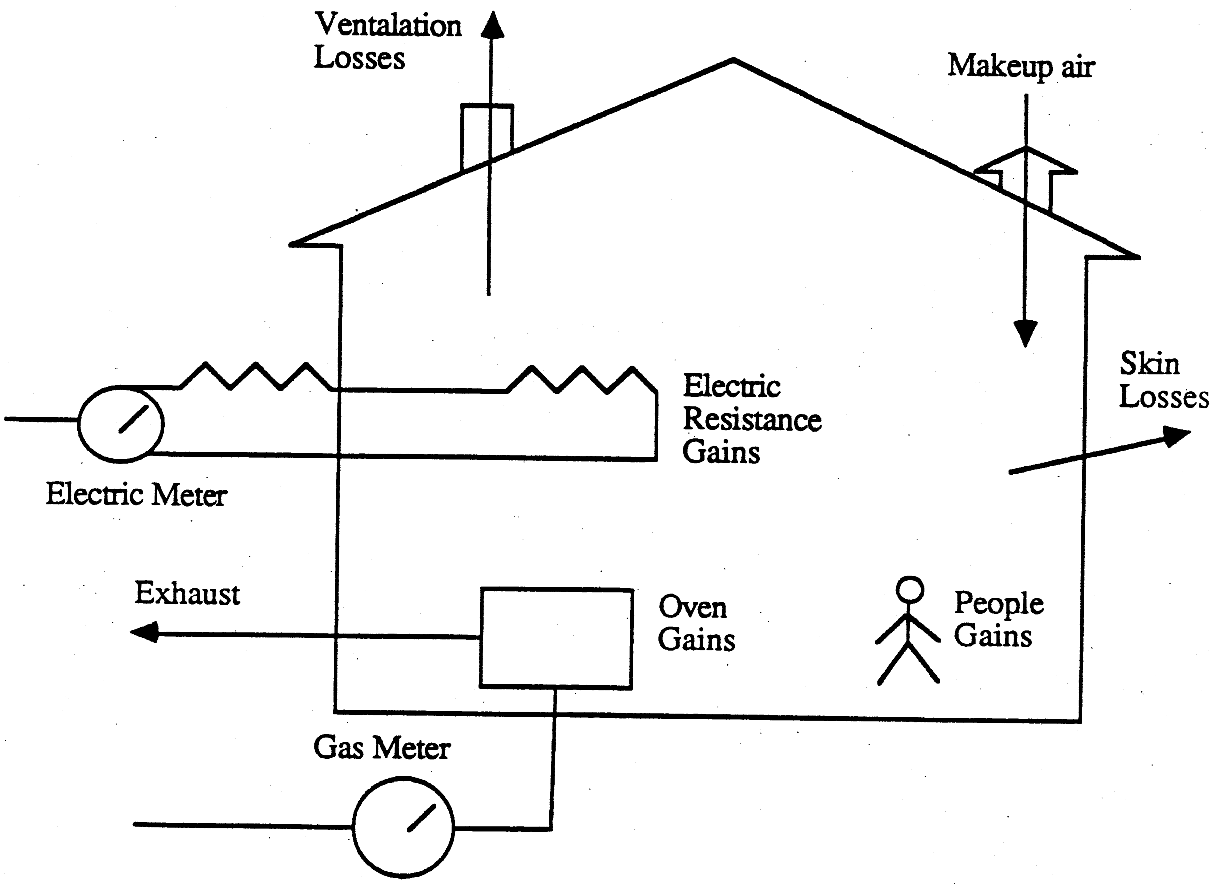

Figure 4.1 Diagram of the building model showing heat flows and gains being considered

4.2 Skin Load

The skin load is the energy transfer through the walls, ceiling, and floor of a building caused by a temperature difference between the inside and outside air; the load may be either a gain or a loss. This energy flow consists of convection and radiation on the inside and outside surfaces of the walls and conduction through the walls. A simplified method for calculating the skin load is to assume the thermal resistances of the walls, ceiling, and floor can be combined to give an overall resistance such that

$$Q_{skin} = \frac{A_{bld,surf}\left(T_{inside} - T_{amb}\right)}{R_{overall}} \tag{4.2.1}$$

This is usually expressed in terms of an overall conductance-area product, which is the inverse of the resistance

$$Q_{skin} = (UA)_{overall}\left(T_{inside} - T_{amb}\right) \tag{4.2.2}$$

It is assumed that the conductance-area product of the building is constant throughout the year. The inside temperature is considered constant throughout the year while the outside temperature is estimated using TMY (Typical Meterological Year) weather data for Madison, Wisconsin. A typical conductance-area product is estimated using standard construction for commercial buildings in Wisconsin (0.14 Btu/hr-ft2-F (Mitchell, 1987)), and the surface area of a fast food restaurant with dimensions 65'x50'x10' (5600 square feet). Using these values, the conductance-area product is about 800 Btu/hr-F, which is used as the base value.

4.3 Ventilation and Infiltration

Ventilation is a major contributor to the heating/cooling loads in restaurants. State health standards require restaurants to be well-ventilated in order to supply fresh air for both workers and customers. The ventilation required is largely determined by the number of occupants and by the size and type of ovens in the restaurant. The ventilation load is calculated by

$$Q_{vent} = \dot{m}_{vent}\, c_{p,air}\left(T_{inside} - T_{amb}\right) \tag{4.3.1}$$

The ventilation mass flow for a typical fast food restaurant is estimated using both the Department of Industry, Labor and Human Relations Ventilation Codes (DILHR, 1985) and the HVAC specification for a Madison restaurant to be 4500 lbm/hr (1000 CFM).

An infiltration load could be calculated in a manner similar to the ventilation load, but it is neglected since it is usually only a small percentage of the ventilation load.

4.4 Internal Gains

Internal gains are defined here as sources of heat addition to the building other than skin, ventilation, infiltration, or furnace gains. The major internal gains in a restaurant are from cooking, people, and electric appliances. A portion of the hot water heating ends up in the restaurant as an internal gain, but it is negligible relative to the cooking and electric gains.

4.4.1 People Gains

People transfer both sensible and latent heat to their surroundings; the latent gains, however, are neglected here, since only sensible cooling benefits of the heat pump water heater are being considered. An estimate of internal gains by people is 225 Btu/person-hr (Mitchell, 1983). The number of people in the restaurant during any hour is an input to the model. A Madison area restaurant supplied two months of data of the number of people in the restaurant each hour of the day. This restaurant is similar in size to the restaurants in this study and was used as an order of magnitude estimate of the occupancy of restaurants. The base occupancy schedule of the building is given in Table 4.1.

4.4.2 Cooking Gains

Ovens are a large source of internal gains in restaurants. Part of the heat generated by the ovens ends up as an internal gain, and part ends up as exhaust gas. The total heat generated by the ovens can be calculated by

$$Q_{oven} = \eta_{oven} \dot{m}_{gas} HHV \qquad (4.4.1)$$

where

η_{ovens} = efficiency of ovens in converting gas to heat, and

$\dot{m}_{gas}HHV$ = energy flow rate of gas based on the higher heating value.(Btu/hr)

The gas energy flow for a typical restaurant is estimated from monthly data of gas usage for several restaurants in the Madison area (WP&L, 1987). The major uses of gas in restaurants are cooking and space heating. Since there is no space heating

Hour of Day (AM)	Occupancy	Hour of Day (PM)	Occupancy
1	0	1	35
2	0	2	20
3	0	3	20
4	0	4	20
5	0	5	35
6	0	6	35
7	20	7	35
8	20	8	20
9	10	9	20
10	10	10	20
11	20	11	20
12	50	12	20

Table 4.1 Base Occupancy Schedule

load during summer months, the base gas usage of the ovens is estimated using the gas consumed over these months. Assuming that the ovens' gas usage is the same for each day of the year, is uniform over the open hours, and is zero during closed hours, allows the gas energy flow to be estimated by

$$\dot{m}_{gas}\,HHV = \frac{\text{avg. monthly summer gas supply (Btu)}}{(30\ \text{days/month})(\#\ \text{open hours/day})} \qquad (4.4.2)$$

For the data available this energy flow is estimated to be 134,000 Btu/hr. A fixed percentage of this heat generated by the ovens is assumed to end up in the kitchen as an internal gain. For this study the combination of this percentage and the oven efficiency is taken to be 20%, which gives an hourly building gain of 26,800 Btu/hr during open hours.

4.4.3 Electric Resistance Gains

Electric resistance gains, another major source of internal gains in restaurants, are defined here as the electricity entering the restaurant that ends up as heat added to the building. For this study, it is assumed that all electricity used inside the restaurant is eventually an internal gain. This is usually true with the minor exception of radiation losses of the lighting through windows. The total electric resistance gains are estimated using monthly data in a manner similar to that of the ovens' gains. The summer and winter electric usage is different, basically due to the summer air-conditioning load. Therefore, the data for the winter months is used to estimate the base electric load.

$$\text{Base Load} = \frac{\text{avg. monthly winter electric use (KWH)}}{(30\text{ days/month})(\#\text{ open hours/day})} \qquad (4.4.3)$$

The base load is estimated to be 52.3 KWH. This base electric load ends up partially as internal resistance gains and partially as losses to the environment. These losses are the result of outdoor lighting, outside condensers of the freezers and refrigerators, etc.. A fixed percentage of 30% of the base electric load is assumed to be internal gains, and these gains are assumed to be uniformly distributed throughout the open hours of the restaurant. The percent internal gains was initially taken to be larger than 30%, however, the resulting building had a year-round cooling requirement. This percentage was decreased, however, because restaurants in Madison typically have cooling loads between 7 and 10 months of the year, and the base case building was to be typical of restaurants in Madison.

4.5 Furnace and Air-Conditioner Models

Furnace and air-conditioner models are needed in order to associate a dollar value with the decreased cooling or increased heating load caused by the heat pump water heater space cooling. The restaurants being considered have gas furnaces and electric air conditioners. The performance of the furnace is described in terms of a thermal efficiency by

$$Q_{furn} = \eta_{furn}\, \dot{m}_{gas}\, LHV \tag{4.5.1}$$

where LVH is the lower heating value of the fuel. Traditional combustion furnaces have a steady-state efficiency of approximately 80%. In actual operation, however, the efficiency is lower due to the furnace cycling on and off and due to standby losses. In general, larger heating loads cycle the furnace less often, giving a higher furnace efficiency. The furnace, therefore, has its highest efficiency in the winter and somewhat lower values in the spring and fall, with an annual variation of between zero and twenty percent. This seasonal variation is neglected, however, since this variation is specific to a given system and since the relation to weather is unclear. An average efficiency of 60% is used as a base value.

Because the air-conditioner's condenser is located outdoors, its performance is strongly dependent upon the weather. The coefficient of performance of an ideal Carnot cycle varies with the high and low reservoir temperatures according to

$$COP_{Carnot} = \frac{T_{low}}{T_{high} - T_{low}} \tag{4.5.2}$$

In practice, the coefficient of performance of an air conditioner is always less than the Carnot Cycle's, because of the inefficiencies of the motor and because of temperature differences between the working fluid and the reservoirs that exist in actual systems. A simple method of accounting for variations in the coefficient of performance of an air conditioner is to assume that the temperature differences at the condenser and evaporator are equal and constant, and also that the coefficient of performance is a fixed percentage of the the value predicted by the equation 4.5.2. These assumptions result in the following equation

$$COP_{A/C} = P_{Carnot} \frac{T_{amb} + \Delta T}{\left(T_{inside} - \Delta T\right) - \left(T_{amb} + \Delta T\right)} \tag{4.5.3}$$

The fixed temperature difference was taken to be 35°F. Using performance data of an air conditioner at ambient temperatures of 75°F and 85°F, the fixed percentage (P_{Carnot}) was determined to be 40%. Figure 4.2 is a plot of the air conditioner performance versus ambient temperature for an inside temperature of 75°F. The coefficient of performance is a minimum during the summer months and increases as the ambient temperature decreases. The cost of cooling, therefore, is much greater in the summer months.

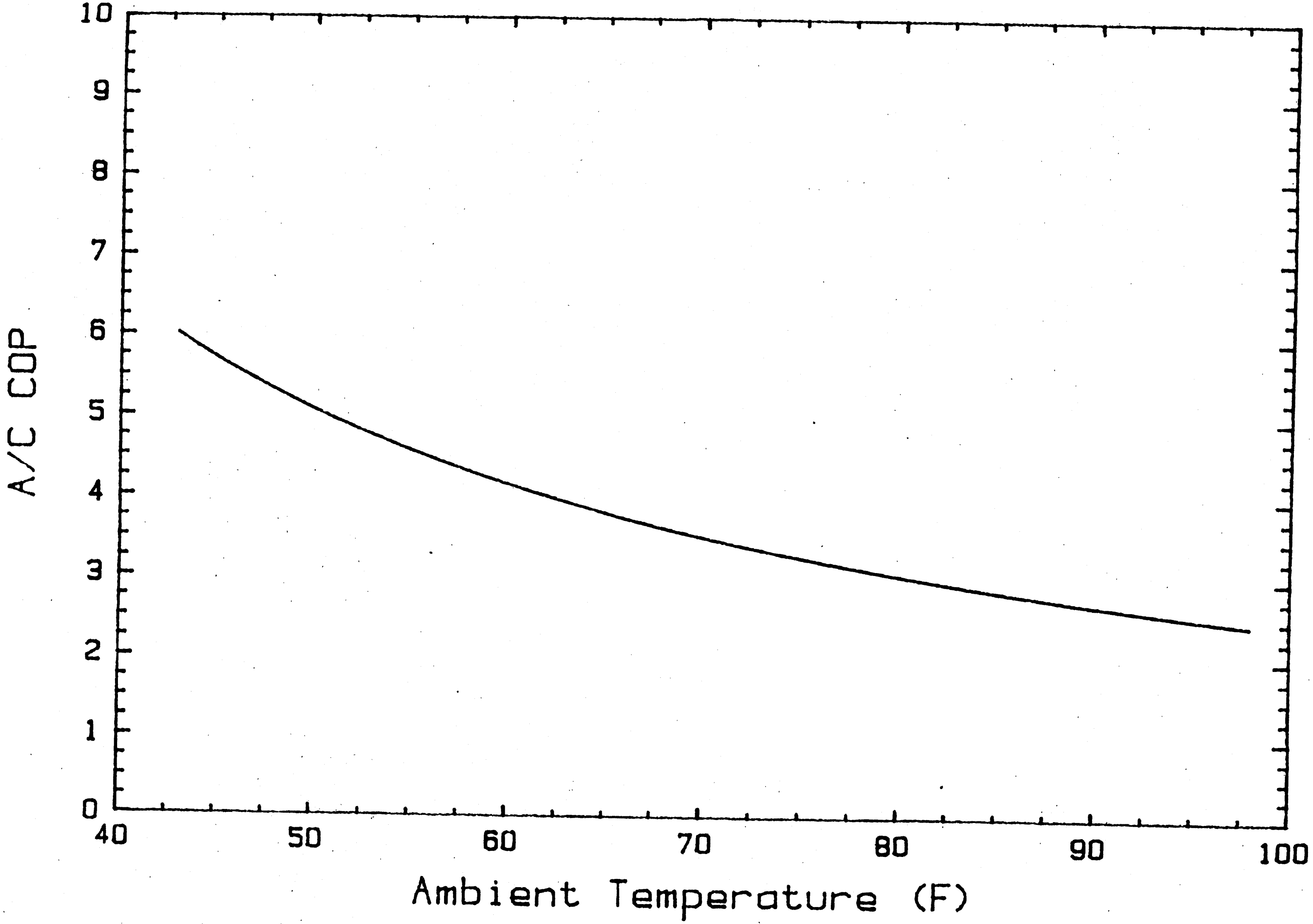

Figure 4.2 Variation of the air conditioner's coefficient of performance with ambient temperature

4.6 Overall Building Model

The air conditioner and heating loads are determined by summing all the energy flows and gains previously described.

$$E_{net} = Q_{skin} + Q_{vent} + Q_{people} + Q_{ovens} + Q_{electric} \tag{4.6.1}$$

A positive value of the net building energy indicates a cooling load, while a negative value indicates a heating load. Solar gains, losses of the hot water storage, and gains from the hot water being used have been neglected by assuming that they are small relative to the other gains. This assumption is not valid for every building design and hot water usage, but the model predict building loads typical of many restaurants, which is the intent. A Fortran program was made of the building model presented in this chapter. Figure 4.3 is a graph of the building load during open hours of the restaurant predicted by the model, for the base building parameters specified throughout this chapter and for Madison weather data. This plot is of the average hourly building load during open hours for each day of the year, and shows that the building has approximately a 9 month cooling load and 3 month heating load. This is typical of restaurants in the Madison area, which report cooling loads of 7 to 10 months of the year.

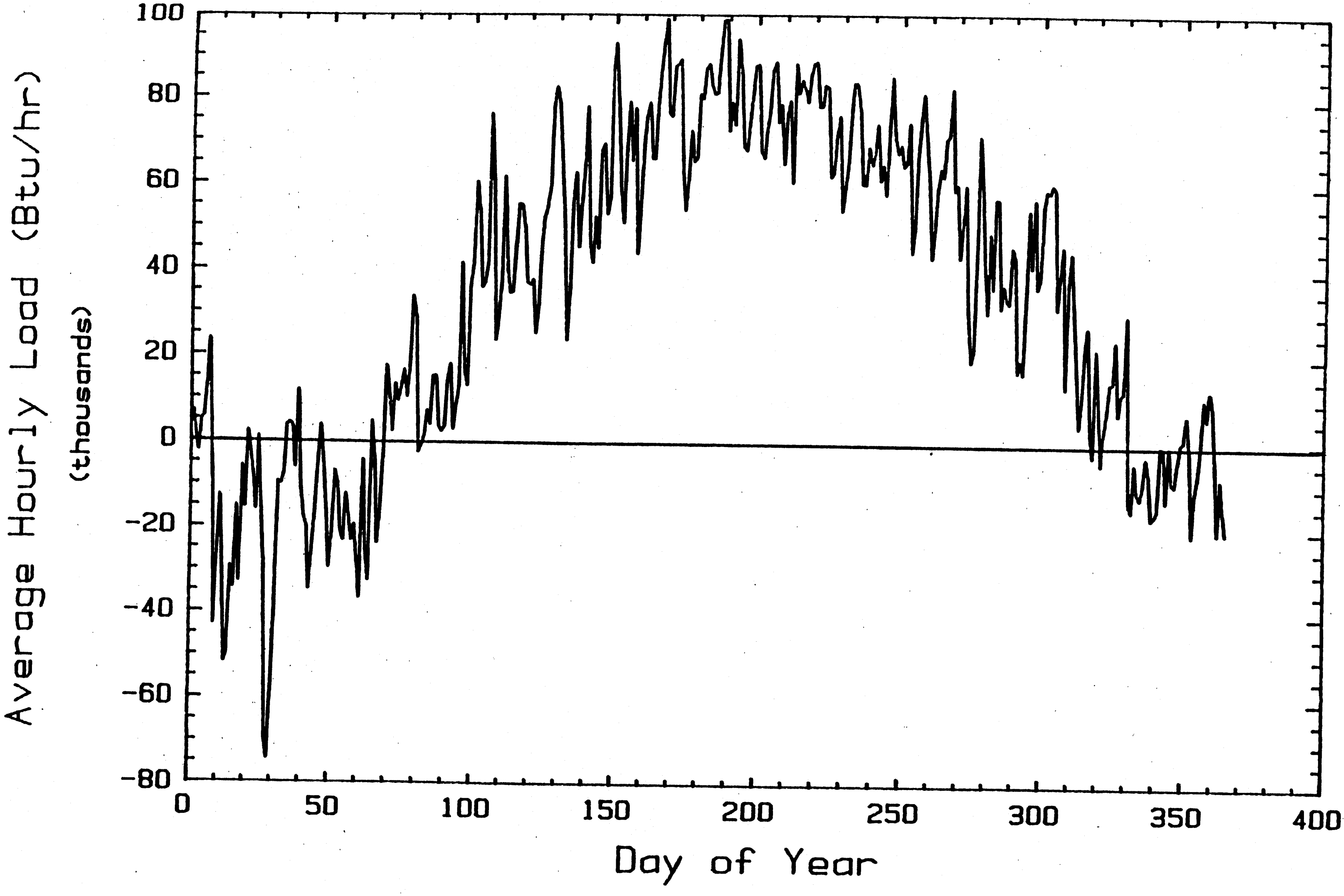

Figure 4.3 Average hourly building load during open hours versus day of the year

Chapter 5
Simulation Program

5.1 Description of Simulation Program

The heat pump water heater and building models are combined in an annual simulation program which performs hourly calculations of the building's space conditioning load and the HPWH's performance for each hour of the year. In each hour the program is used to determine whether the space cooling produced by the heat pump is an air conditioning savings or a heating requirement, and also to calculate the associated savings or expense. The HPWH cooling is an air-conditioner savings if the net building hourly energy is greater than the cooling. The amount of savings depends upon the cost of electricity (C_{KWH}) and the coefficient of performance of the air conditioner for that hour, and is calculated by

$$Sav_{A/C} = \frac{HPWH_{A/C}}{COP_{A/C}} C_{KWH} \tag{5.1.1}$$

The $HPWH_{A/C}$ cooling is a heating expense if the net building energy is negative. The expense associated with this cooling depends upon the efficiency of the furnace and the price of the heating fuel, and is calculated by

$$\mathrm{Exp}_{SH} = \frac{\mathrm{HPWH}_{A/C}}{\eta_{furn}} C_{Therm} \tag{5.1.2}$$

If the net building hourly energy is positive, but less than the HPWH cooling, then the cooling is a savings while eliminating the positive building load, but is an expense for cooling greater than the net building heating load. These savings and expenses are calculated by

$$\mathrm{Sav}_{A/C} = \frac{E_{Net}}{\mathrm{COP}_{A/C}} C_{KWH} \tag{5.1.3}$$

$$\mathrm{Exp}_{SH} = \frac{\mathrm{HPWH}_{A/C} - E_{Net}}{\eta_{furn}} C_{Therm} \tag{5.1.4}$$

The annual space conditioning savings associated with the HPWH cooling is determined by summing all the hourly savings and expenses throughout the year

$$\mathrm{Annual\ Sav}_{SC} = \sum \mathrm{Sav}_{A/C} - \sum \mathrm{Exp}_{SH} \tag{5.1.5}$$

In addition to determining the air conditioning savings, the simulation program is used to calculate the water heating savings of the HPWH compared to electric or gas water heaters. The program assumes that each water heater performs the same amount of water heating. The annual water heating savings of the HPWH compared to an electric water heater is calculated by

$$Sav_{WH} = \left[Q_{H_2O} - \int \dot{E}_{motor}\, dt \right] C_{KWH} \tag{5.1.6}$$

For a gas water heater the savings depend upon the thermal efficiency of the heater, and the savings are calculated by

$$Sav_{WH} = \frac{Q_{H_2O}}{\eta_{GWH}} C_{Therm} - C_{KWH} \int \dot{E}_{motor}\, dt \tag{5.1.7}$$

The total savings of the HPWH relative to the electric or gas water heaters is the sum of the space conditioning and water heating savings.

WP&L, the project sponsor, has electricity and gas charge rates for small commericial users (i.e., restaurants) of $0.061/KWH and $0.43/Therm respectively.

5.2 Input Hot Water Draw and Internal Gains

Whether the HPWH cooling is a savings or an expense is dependent on when the building load occurs in the day relative to the hot water load. In a restaurant the building load is largely controlled by the internal gains, and therefore, the scheduling of hot water draw and internal gains are important considerations. If hot water is only needed during hours when there are no gains, then the HPWH cooling is generally a heating expense; otherwise it is generally an air conditioning savings. There is an infinite number of combinations of gains and hot water draw schedules. In general, however, fast food restaurants' gains are fairly uniform during open hours, and they are assumed to be exactly uniform throughout the open hours as outlined in chapter 4. The restaurants for this study are assumed to be open between

7 A.M. until 1 A.M..

In section 3.5 it was shown that the performance of the HPWH was largely dependent upon the rate of water draw, but only slightly affected by either the type of draw schedule or amount of daily draw. Therefore, as in section 3.5, the base draw schedule is defined as having two periods of water draw (ie. draw schedule 2) with a total daily draw of 500 gallons.

To be consistent with Chapter 3, the draw schedules considered in the simulation program have at least two hours of no draw between the two draw periods to allow the storage tank to be completely reheated. Various draw rates are investigated, two of which are shown in Figure 5.1, along with the building gains schedule. In each case the first draw periods end at 3 P.M. and the second draw period ends at 2 A.M., and the total daily hot water draw is 500 gallons. The building's electric gains and oven gains are evenly distributed over the open hours of the restaurant (7 A.M. until 1 A.M.).

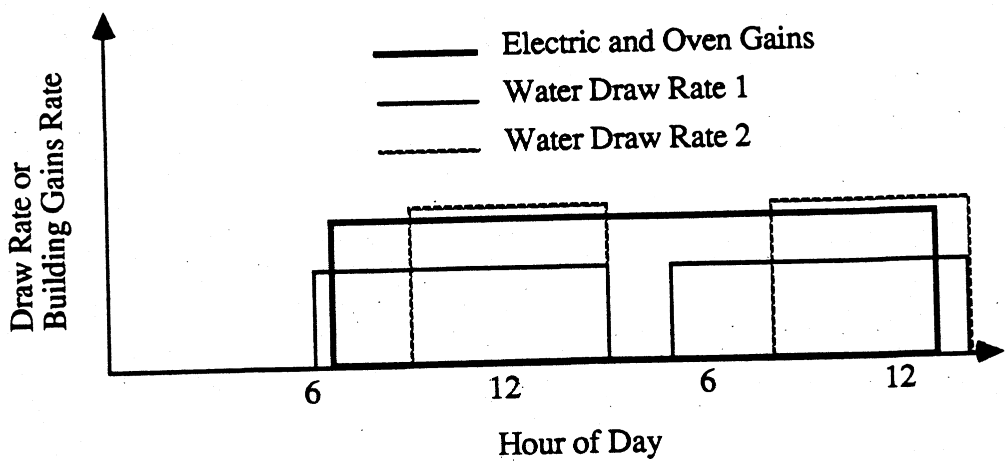

Figure 5.1 Water draw schedule and building electric and oven gains schedule, which shows the relative positioning of the schedules.

Chapter 6
Simulation Studies and Results

In this chapter, the simulation program of Chapter 5 is used to study the savings of a typical fast food restaurant using a HPWH instead of gas or electric resistance water heaters, and to determine the effects that various HPWH parameters, draw rates, and building loads have on the savings. The input values for the base case building model were given throughout chapters 4 and 5 and are summarized in Table 6.1. The input values for the HPWH model are as previously given in Table 3.1.

The simulation program calculates hourly savings for each hour of the year as outlined in Chapter 5. The savings presented in this chapter are either the average hourly savings over a month or the annual savings. In each case these savings are determined from the hourly values.

6.1 Base Case Restaurant

The simulation results for the base case restaurant with a draw rate of 36 gallons per hour (i.e., 500gal/14hr) is given graphically in Figures 6.1 and 6.2. Figure 6.1 is a plot of the daily average HPWH savings over the open hours of the restaurant for each month of the year when compared to electric resistance water heaters. This graph consists of four curves. The first curve is the savings in space conditioning which can be positive or negative depending upon whether the $HPWH_{A/C}$ cooling is

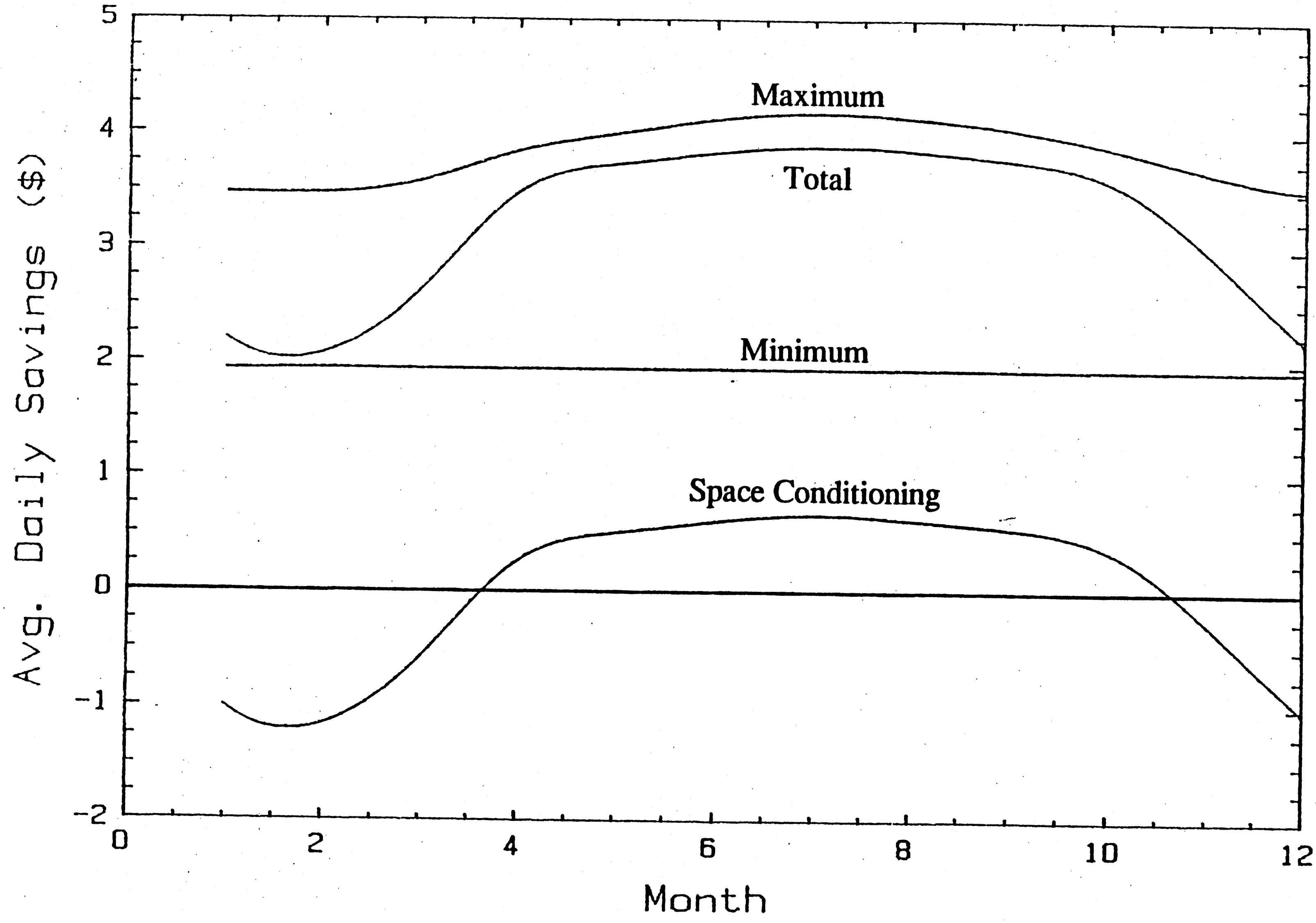

Figure 6.1 Monthly HPWH savings compared to an electric water heater

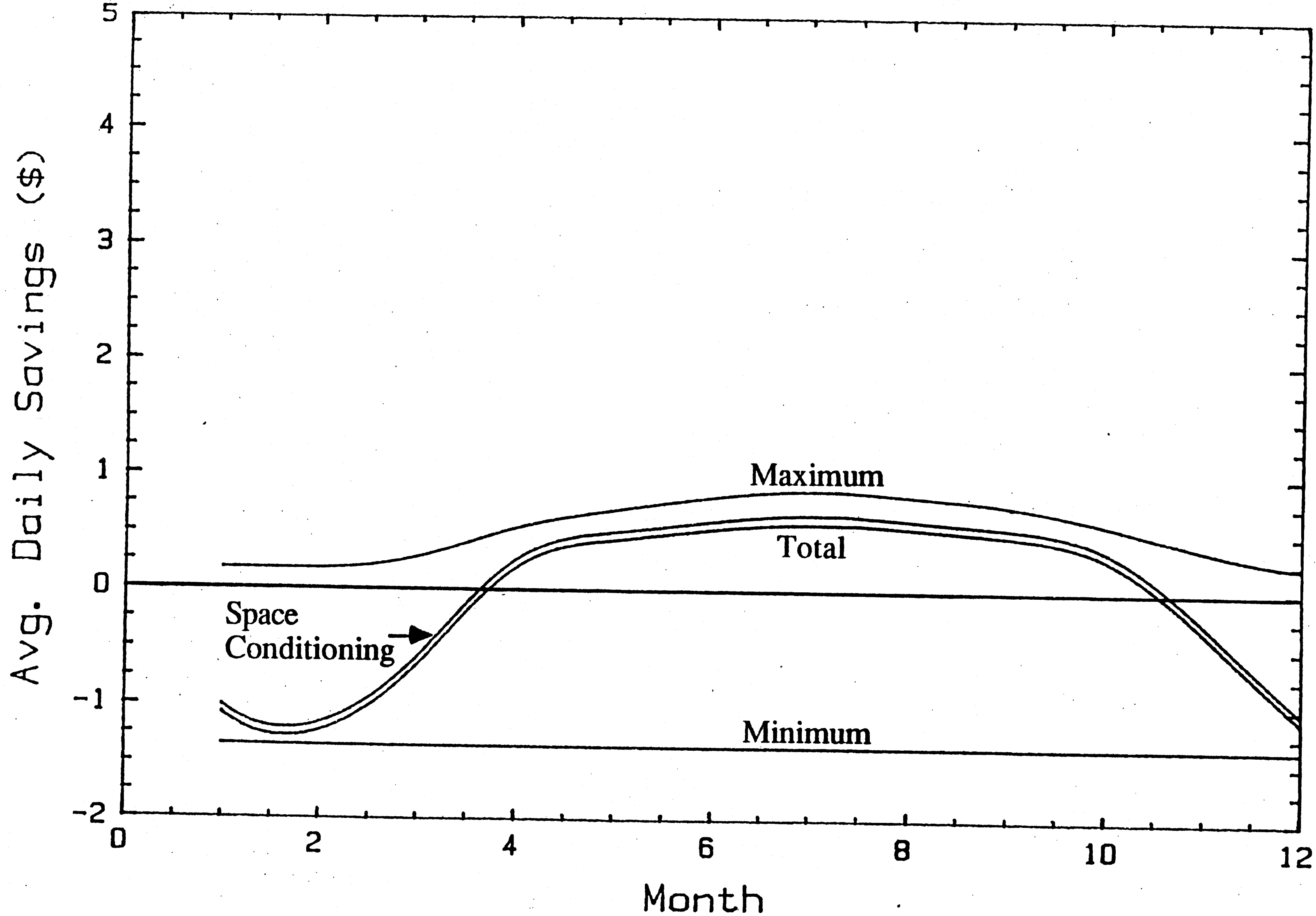

Figure 6.2 Monthly HPWH savings compared to a gas water heater

Variable Description	Units	Input Value
% oven gains	(0 to 1)	0.2
% electric gains	(0 to 1)	0.3
$(UA)_{Bld,overall}$	Btu / hr °F	800
Ventilation	CFM	1000
Inside Temperature	°F	75
η_{furn}	(0 to 1)	0.6
η_{GWH}	(0 to 1)	0.55
C_{KWH}	$/KWH	0.0613
C_{Therm}	$/Therm	0.43

Table 6.1 Inputs for base case building model

a net air conditioning savings or a net heating expense. The second curve is the combined space conditioning savings and water heating savings of the HPWH (labeled total). The third curve is the total savings (i.e. combined space conditioning and water heating savings) if all cooling produced by the heat pump were an air conditioning benefit (labeled maximum). The last curve is the total savings if all the cooling were a heating requirement (labeled minimum). The maximum and minimum curves show the range of effect that the space conditioning savings could have on the total savings for buildings having different cooling and heating requirements. The maximum and minimum curves do not apply to the HPWH in general, but are the maximum and minimum savings for the the draw schedule, air conditioner, and energy prices of this specific case. For example, the maximum savings could be higher than shown if a draw schedule which gave a higher HPWH coefficient of performance was used. The calculations of the savings were outlined in section 5.1.

The graph in Figure 6.1 shows that the daily total savings of the HPWH range between $2 and $4 over the year. The variation in the total savings is due to changes in the amount of the HPWH cooling that is useful and in the performance of the air conditioner, as can be seen in equation 5.1.1.

Figure 6.1also shows that the air cooling produced by the HPWH is a net air conditioning savings for about 7 months of the year, but is a net heating requirement the rest of the year. In Chapter 4 it was noted that the base case building had approximately a 9 month cooling load. The air conditioner savings, however, only occurs in about 7 of these months because the cost of heating is greater than the cost of cooling (about twice as great in this case), and therefore the transition months between cooling and heating end up as a net heating expense.

The maximum savings curve varies from a high during the summer to a low during the winter. Since each day of the year has the same water draw schedule and the same evaporator air temperature, the performance of the heat pump and the amount of cooling produced is the same each day. Further, since the amount of HPWH cooling is the same each day, and since the maximum savings assumes that all this cooling is useful, the the variation in the maximum savings is due only to changes in the value of this cooling throughout the year caused by changes in the performance of the air conditioner. The performance of the air conditioner changes with ambient temperatures as previously shown in Figure 4.2. The air conditioner's performance decreases with increasing ambient temperatures, and thus the value of the HPWH cooling increases with ambient temperature. The maximum savings, therefore, are largest during the summer when the HPWH cooling has its greatest value.

The minimum savings curve, unlike the maximum savings curve, is constant throughout the year because it is assumed that the furnace efficiency is unaffected by the weather and is a constant 60%. Since both the amount of HPWH cooling and the furnace efficiency are the same each day, the daily cost of reheating the HPWH cooling is the same throughout the year.

Figure 6.2 is a similar plot to 6.1 and shows the HPWH savings compared to a gas water heater for the base case conditions and a draw rate of 36 gal/hr. Unlike the comparison to the electric water heater, the HPWH is found to be less expensive than the gas water heater only during the summer months. The daily savings of the HPWH is approximately \$0.60 during the summer, but about a \$1.20 loss during the winter. The total savings and space conditioning savings curves are close in this graph because the HPWH water heating savings compared to the gas heater are relatively small (actually a \$0.07 daily loss).

For the base case, the total savings over the year of the HPWH when compared to an electric water heater is about \$1200, but is about a \$30 loss when compared to a gas water heater. The effect of the HPWH cooling on the space conditioning costs over the year was a net loss of \$12. The HPWH coefficient of performance over the year for this case is 2.23.

In Chapter 3 the performance of the HPWH was shown to be strongly dependent upon the rate of hot water draw. The simulation program was exercised, therefore, for the base case building for draw rates other than 36 gal/hr. For each new draw rate, the period over which water is drawn is also changed so that the total daily draw in each case is constant (500 gallons/day for base case). Figure 6.3 is a plot of the predicted HPWH annual savings versus hot water draw rate. This graph gives curves of the total savings compared to gas and electric water heaters, and of the space conditioning savings. This plot shows that for the base case building the HPWH is more economical than the electric water heater for all draw rates, and that the gas water heating is less costly than the HPWH at low draw rates, but more costly at high draw rates. The curves of the total savings have a shape similar to the curves given in Chapter 3 of the overall HPWH coefficient of performance versus draw rate. This is expected, since the savings are directly proportional to the HPWH's COP. The curves in Figure 6.3 are much smoother, however, than the jagged curves of Chapter 3, because 7 data points were used in Figure 6.3, whereas nearly a hundred points were used in Chapter 3. Only 7 data points of the annual savings were generated because of the computer time required to make each simulation run (4 minutes per run). These points, however, show the major effects of changing the draw rate.

For the base case building, the air conditioning savings caused by the HPWH are approximately equal to the additional heating expense, and therefore, the air cooling of the HPWH is has a negligible contribution to the total savings. For restaurants having space conditioning loads different than the base case, the HPWH space conditioning savings will have an effect on the total savings. Increased building loads could occur in restaurants having more internal gains or in restaurants located further south. These cases are studied in the next section.

6.2 Parametric Studies of the Annual Savings

This section examines the effects that changing water heater and building model parameters have on the annual savings. In each case, the HPWH savings are

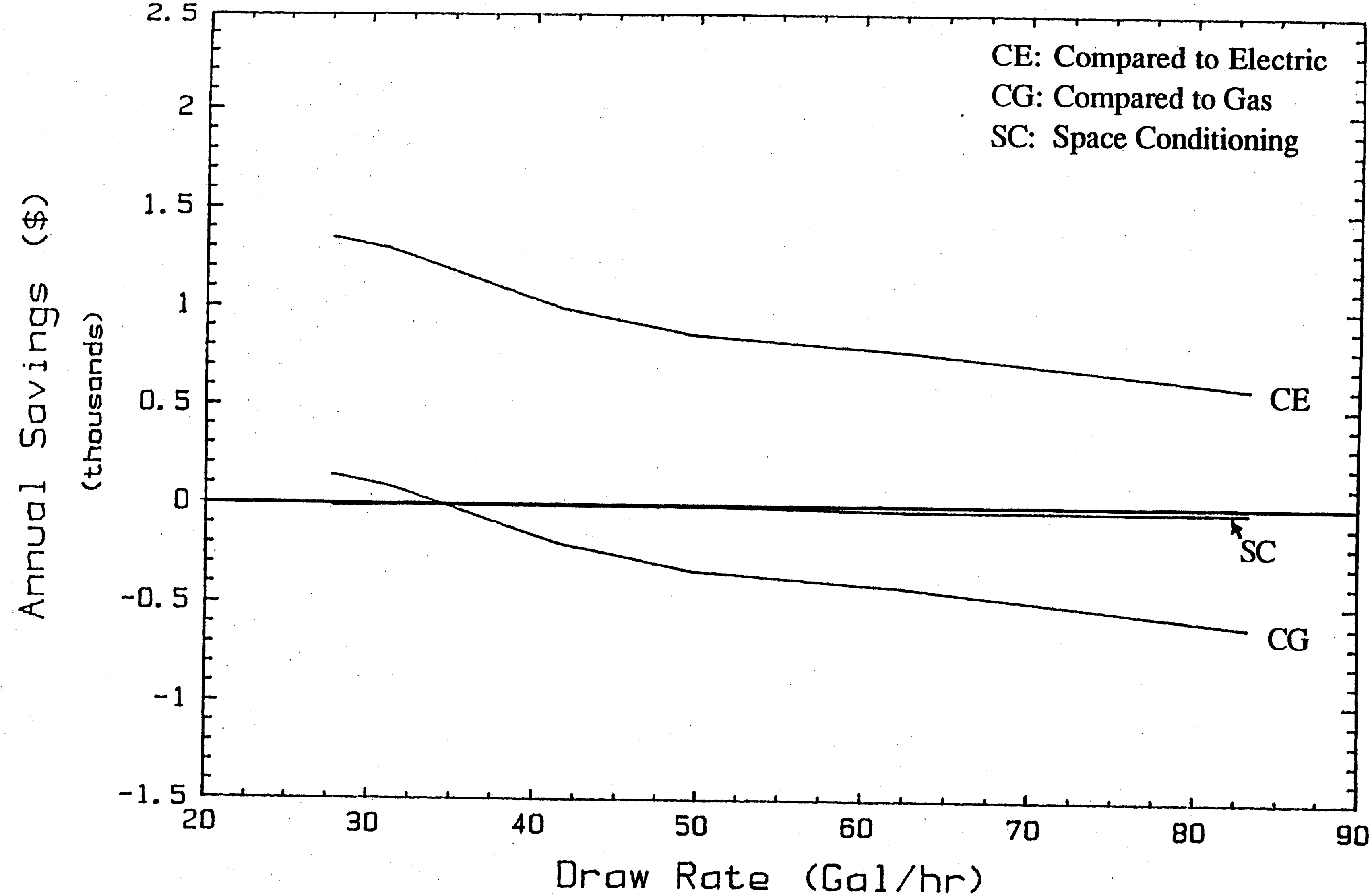

Figure 6.3 Annual savings versus hot water draw rate for the base case building

compared to the base case savings in a graph similar to that in Figure 6.3. The parameters which are studied are:

1) building gains
2) gas heater efficiency
3) temperature of air flow over the heat pump evaporator
4) total daily draw
5) tank volume
6) weather
7) electricity and gas costs
8) changing HPWH to have a single backup heater of 12 KW, located at the top of the tank.

6.2.1 Effects of Building Gains

Increasing the internal gains of the base case building would increase the air-conditioner load throughout the year, and would thus increase air conditioner savings caused by the HPWH air cooling. Figure 6.4 is a plot which compares the HPWH savings of the base case building to a building having half the internal gains and to one having twice the gains. This plot shows that doubling the gains increases the space conditioning savings, and therefore the total savings, between \$60 and \$170. Likewise, halving the gains decreases the savings by a similar amount. In plots of the average hourly building load for each day for the year, similar to Figure 4.3, it is noted that doubling the gains gives a year-round air-conditioning load, whereas halving the gains gives a 6 month cooling load.

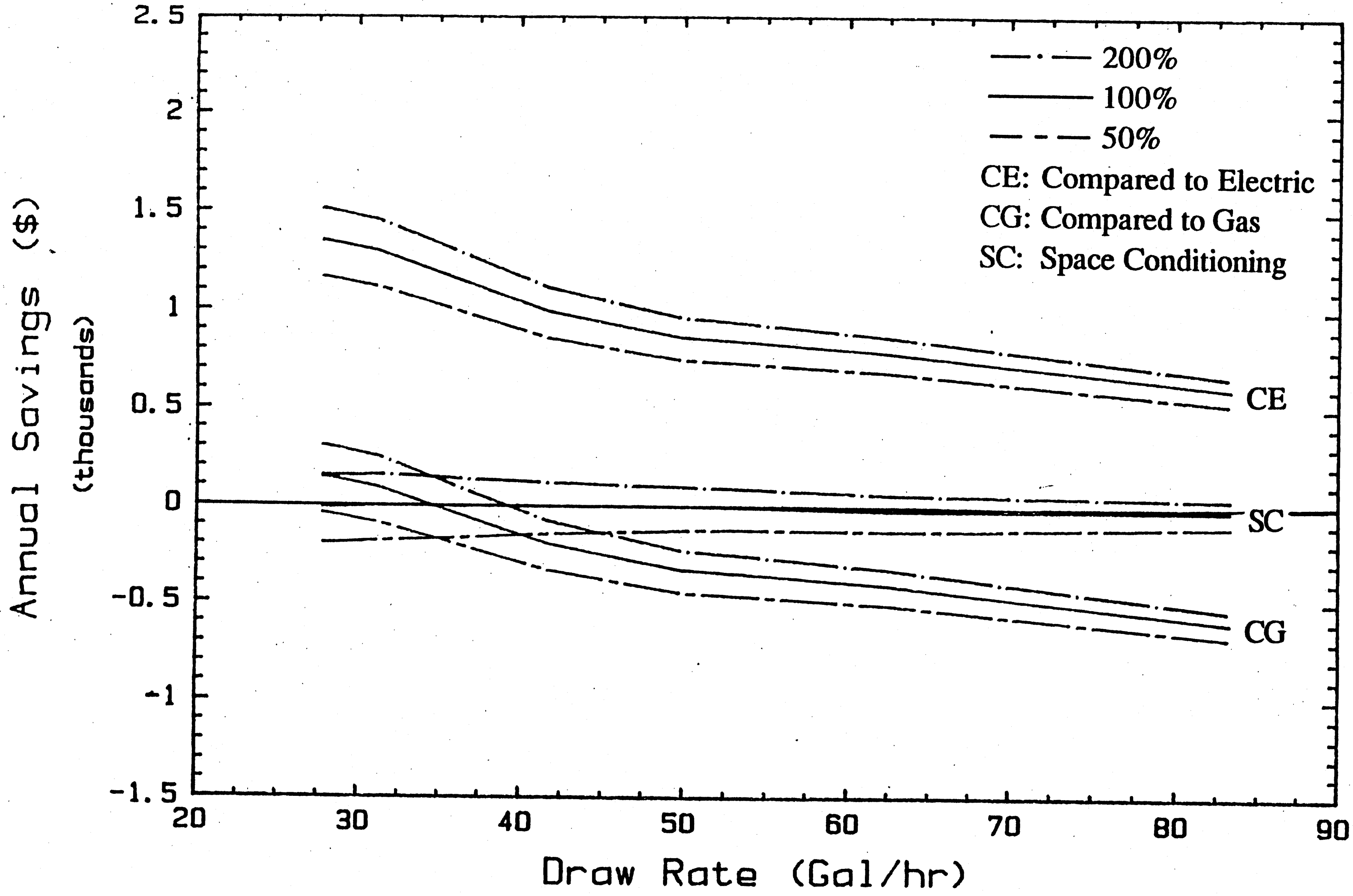

Figure 6.4 Annual savings versus hot water draw rate for buildings having 50%, 100%, and 200% of the base case gains

6.2.2 Effects of Gas Water Heater Efficiency

The efficiency of the gas water heater is 55% for the base case. Figure 6.5 is a plot of the savings of the HPWH for gas water heater efficiencies of 45%, 55%, and 65%. Changing this efficiency does not have an effect on the HPWH savings or the space conditioning savings, but only affects the HPWH savings compared to the gas water heater. The HPWH savings compared to gas heaters increases about $200 by decreasing the efficiency from 55% to 45%, and decreases about $130 by increasing it from 55% to 65%. These savings are seen to be independent of the draw rate. As expected, the savings of the HPWH compared to gas heaters increase when compared to less efficient heaters.

6.2.3 Effects of Evaporator Air Temperature

It was shown in Chapter 3 that the performance of the HPWH increases between 6% and 20% by increasing the air temperature flowing over the heat pump evaporator from 75°F to 85°F. As shown in Figure 6.6 this increased performance reduces the annual HPWH heating cost between $70 and $200. Therefore, it is best to position the evaporator in the warmest location in the kitchen. This adds little or no initial cost, yet significantly decreases the HPWH heating costs.

6.2.4 Effects of Total Daily Draw

Figure 6.7 gives a comparison of the HPWH savings for daily draws of 300, 500, and 700 gallons. The water heating savings, as expected, increase with increasing total draw and are approximately proportional to the amount of water being heated. The savings are not exactly proportional to the total draw since the overall coefficient of performance of the HPWH is slightly affected by the change in

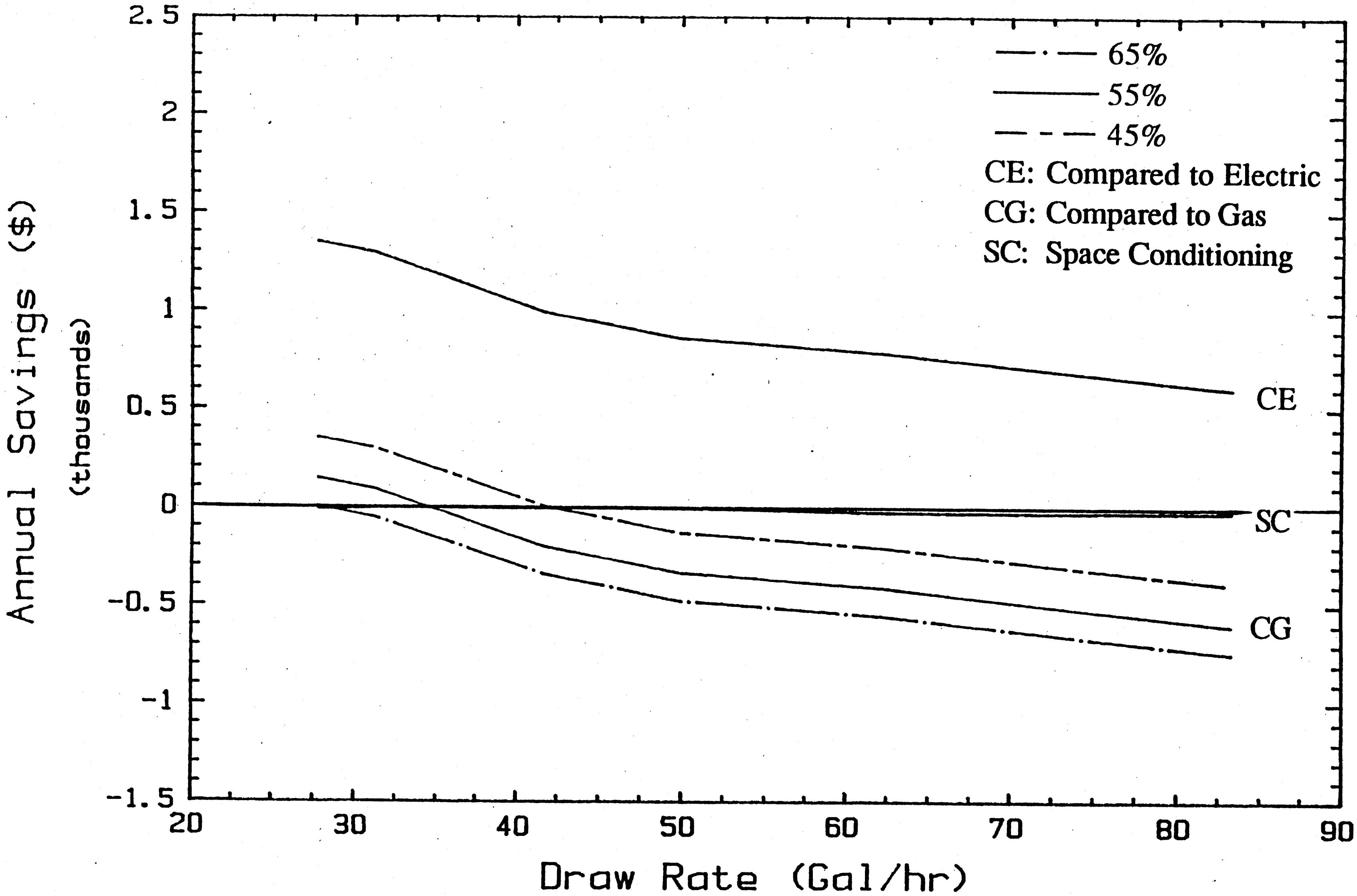

Figure 6.5 Annual savings versus hot water draw rate for gas water heater efficiencies of 45%, 55%, and 65%

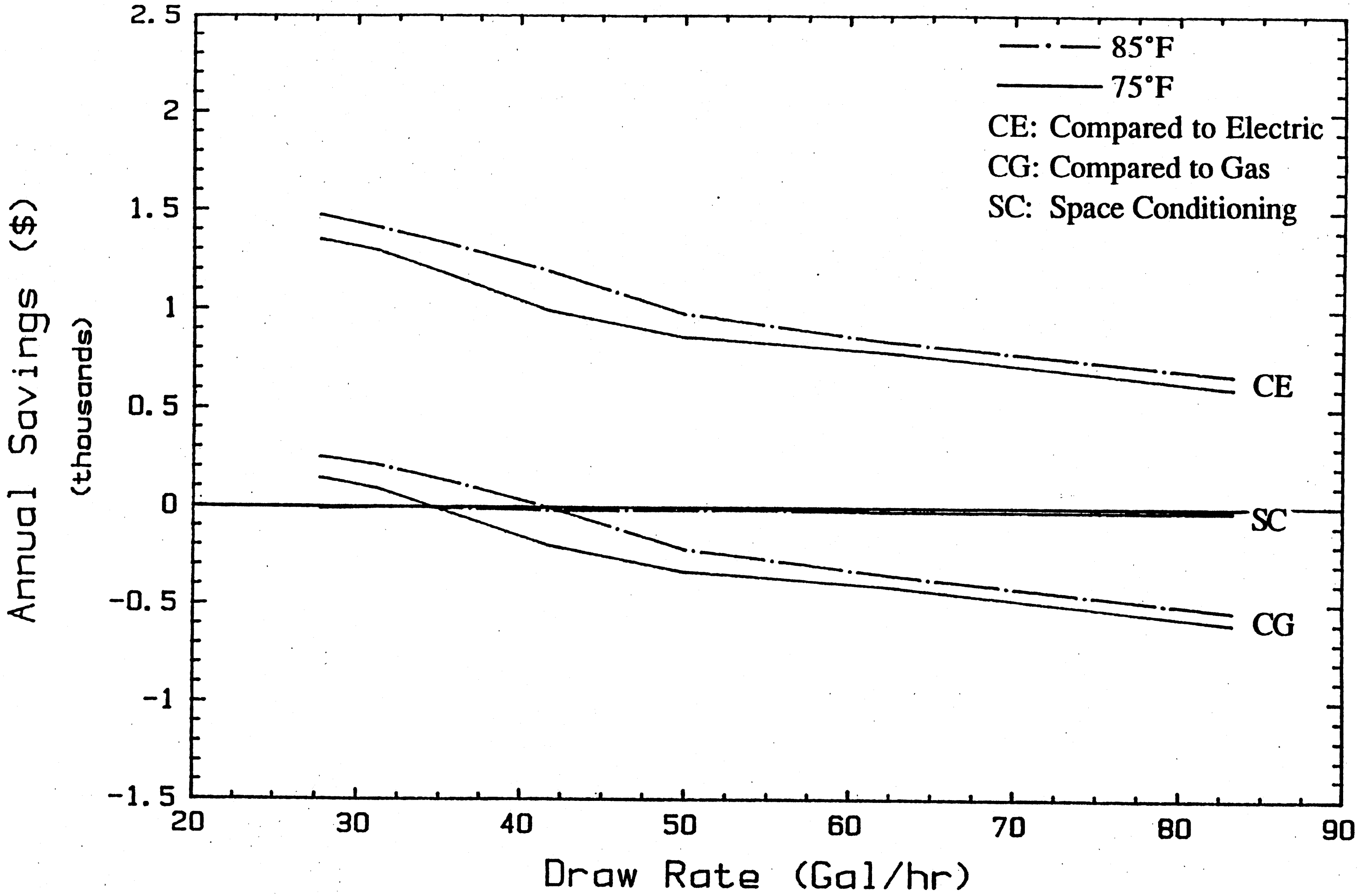

Figure 6.6 Annual savings versus hot water draw rate for evaporator air temperatures of 75°F and 85°F

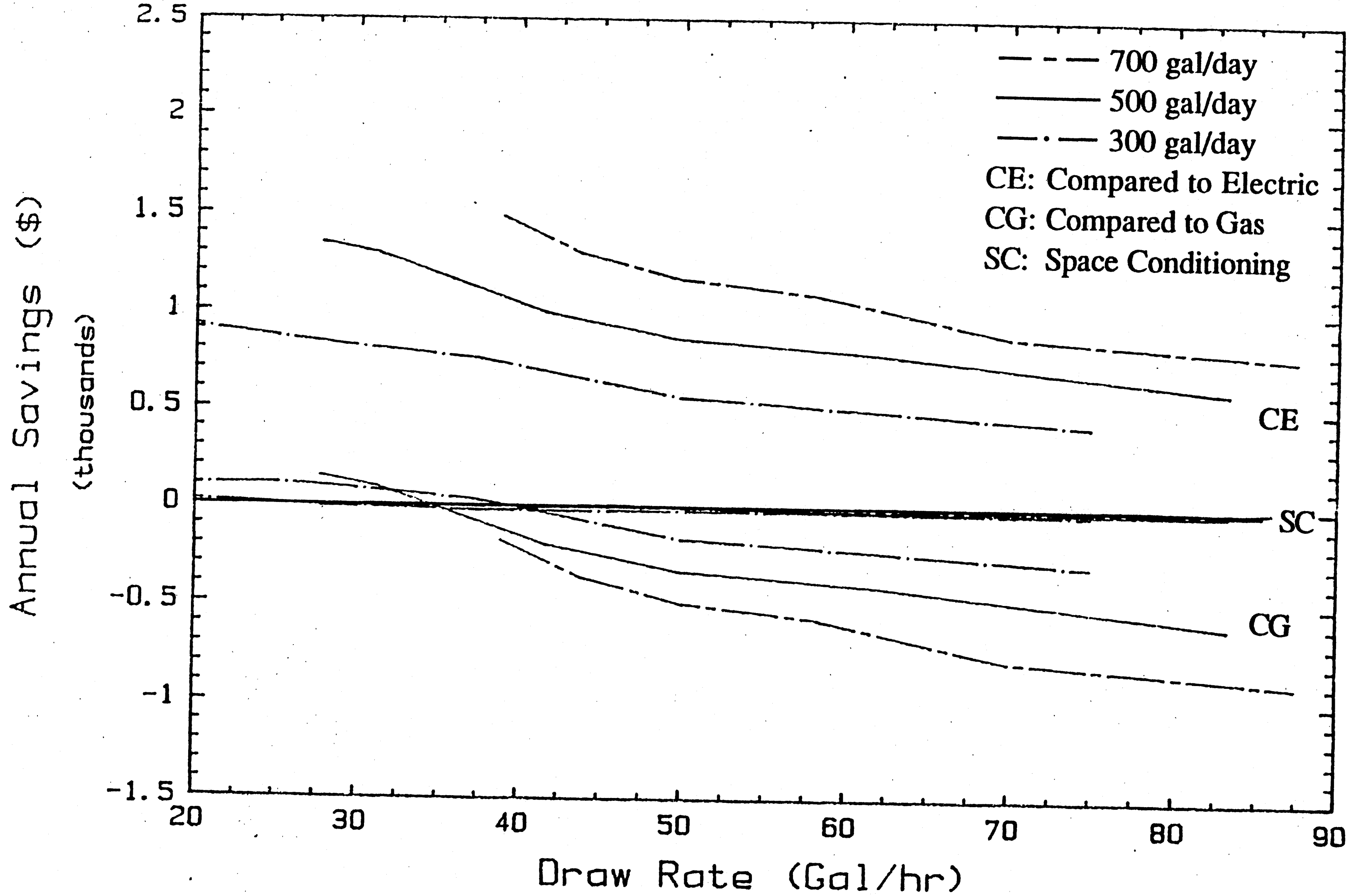

Figure 6.7 Annual savings versus hot water draw rate for daily hot water draws of 300, 500, and 700 gallons

total draw, as previously shown in Chapter 3 in Figures 3.6-8.

6.2.5 Effects of Storage Volume

In Chapter 3 it was shown that increasing the water storage volume increased the performance of the HPWH. Figure 6.8 is a plot of the annual savings for tank volumes having 50%, 100%, and 150% of the base case volume of 110 gallons. This figure shows that a 50% increase in the storage capacity reduces the annual HPWH heating costs an average of about $70, and similarly, a 50% smaller tank increases the cost about $70. These curves assume that the performance of the wrap-around condenser is unaffected by the size of the storage tank. In practice, increasing the tank size would also increase the size of this condenser, which in turn, would increase the condenser's performance. The actual savings effect caused by changing the tank size would therefore be further magnified by this change in performance.

6.2.6 Effects of Weather

Weather plays an important role in determining the HPWH space cooling savings because both the building load and the air-conditioner performance are largely dependent upon the ambient temperature. Figure 6.9 shows the annual savings for the base case building for the weather at 3 locations: Madison, Wisconsin; Nashville, Tennessee; and Miami, Florida. This figure shows that the space conditioning savings are positive for all draw rates if the building is located in either Nashville or Miami, but is negative if in Madison. The net annual space conditioning savings are between $100 and $300 in Miami, between $50 and $100 in Nashville, and about a $25 loss in Madison. Since the total HPWH savings are a

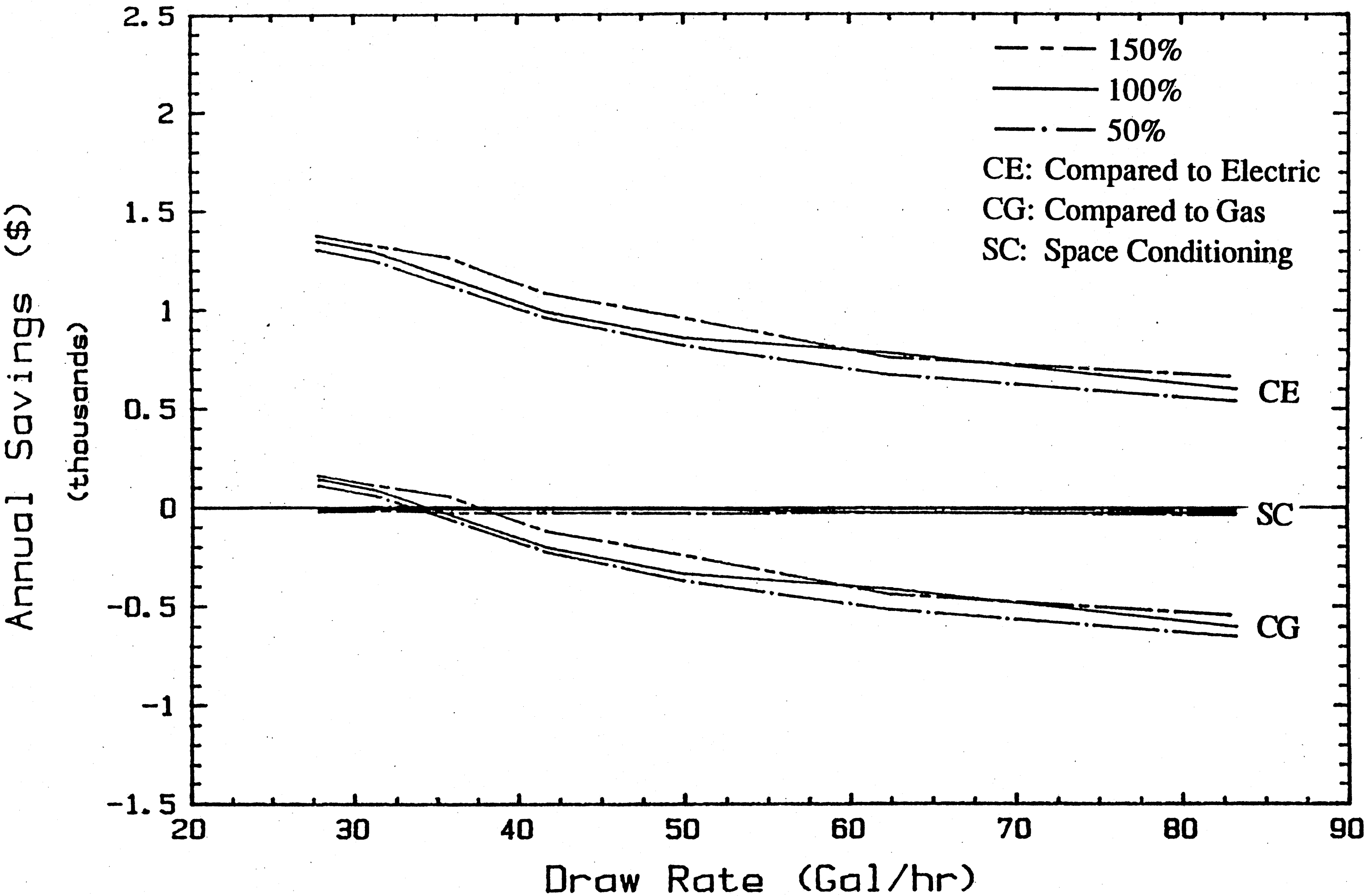

Figure 6.8 Annual savings versus hot water draw rate for storage volumes of 50%, 100%, and 150% of the base case volume

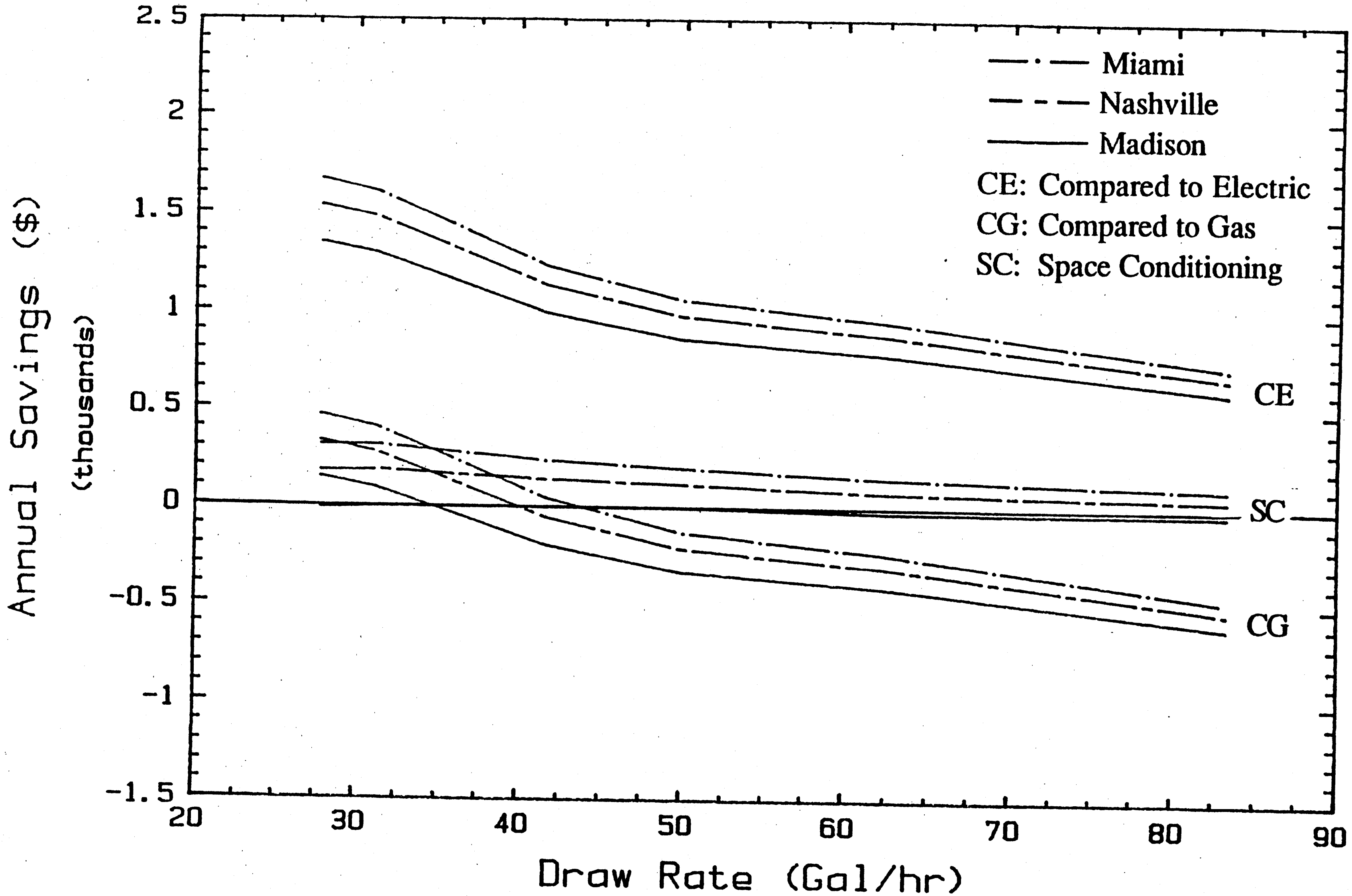

Figure 6.9 Annual savings versus hot water draw rate for 3 geographic locations: Madison, WI; Nashville, TN; Miami, FL

combination of the space cooling and the water heating savings, the total savings are increased by an amount equal to the space conditioning savings. In practice, the savings could be even greater, since the evaporator air temperature could be warmer than the 75°F assumed for Madison.

6.2.7 Effects of Electricity and Gas Prices

An important consideration in the comparison of electric water heaters (i.e., heat pump or electric resistance) to gas heaters is the relative price of gas to electricity. For the base case, the ratio of electricity to gas prices is 4.2. Figure 6.10 is a plot of the savings for cost ratios of 2, 3, and 4.2. This graph is different from preceding graphs in this section in that the ordinate of the graph is savings divided by the cost of electricity. This makes the plot more general, since it can be used to estimate the savings for any gas or electricity costs with these price ratios, whereas the previous ordinate would have limited the usefulness of this plot to the specific values of the energy prices used. For example, the total savings of the HPWH compared to a gas heater at a draw rate of 50 gal/hr and having gas and electricity prices of $0.05/KWH and $0.73/Therm respectively (i.e., price ratio of 3), is $575 (i.e., 11500*0.05). This plot shows that, as the relative price of gas increases, the HPWH savings compared to gas water heaters increases, but space conditioning savings decrease. This is expected because, as the price of gas increases, so does the gas water heating cost and the space heating cost.

6.2.8 Effect of Having a Single 12KW Backup Heater

In Chapter 3 it was noted that the performance of the HPWH could be increased at the high draw rates by having a single 12KW backup heater at the top of the tank.

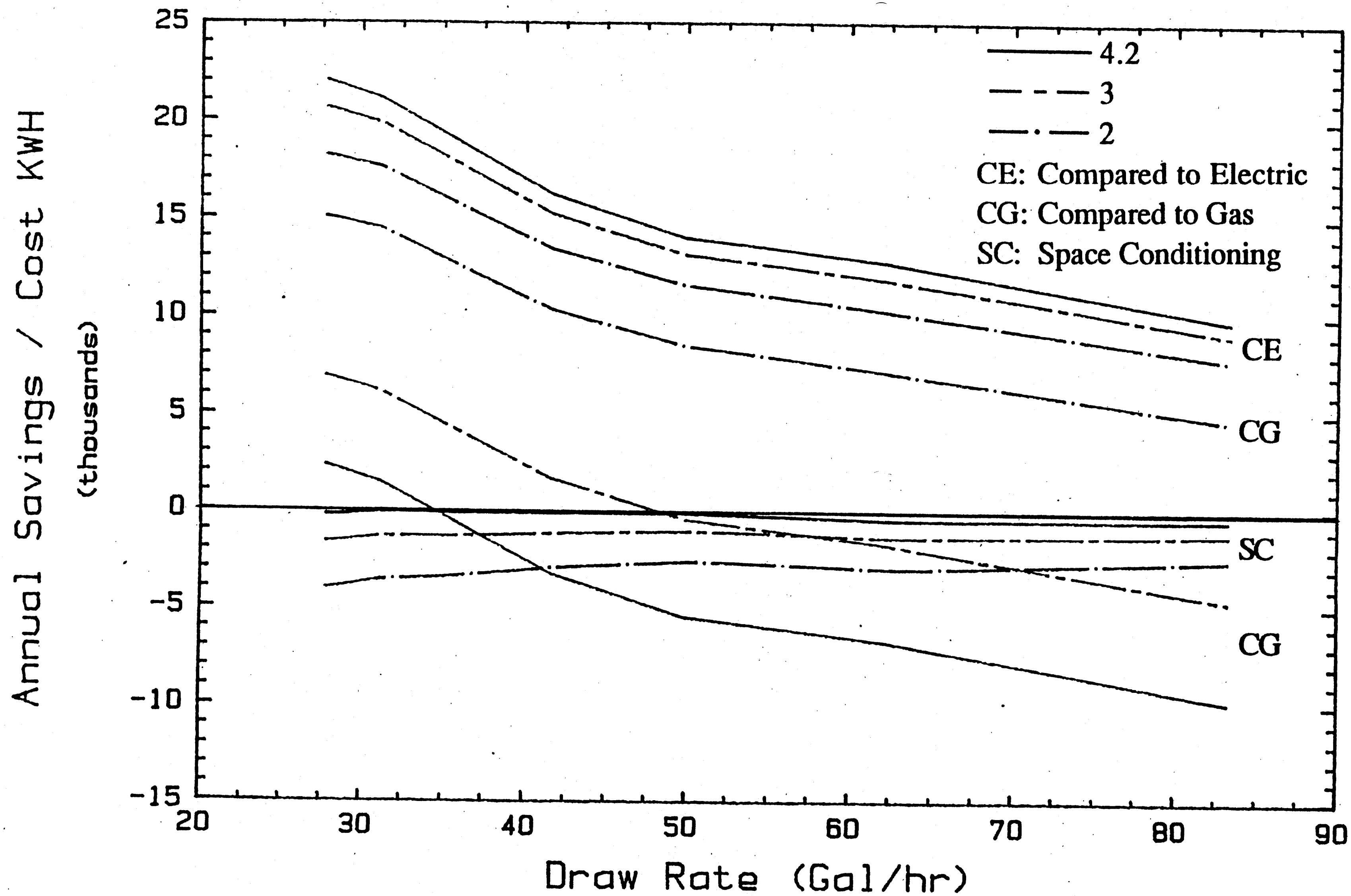

Figure 6.10 Annual savings divided by electricity cost versus hot water draw rate for price ratios of electricity to gas of 2, 3, and 4.2

Figure 6.11 compares the annual savings of the standard water heater to the water heater with the 12KW backup heater. This plot shows that the savings are unchanged at low draw rates where no backup heating is required, but increase by up to $230 at high draw rates. The cost to incorporate this alternative design should be relatively small since both the existing and alternative water heaters have an equal amount of backup heating capacity (12KW), and since the cost of the controller for the second backup heater could be saved. This is considered further in the economics section (section 6.4).

6.3 Other Potential Savings and Expenses

In comparing HPWHs to conventional water heaters, two additional items which may affect the savings comparison are electric demand charges and construction requirements for gas water heaters. Although these items are mentioned for the sake of completeness, they are not thoroughly investigated since they did not apply to the restaurants being considered in this study. WP&L's electricity pricing structure does not have demand charges for small commercial users (i.e., restaurants). In locations where restaurants are charged for electrical demand, this expense will affect the savings comparison of the water heaters. In these locations, the savings of the HPWH compared to electric water heaters would increase because of the decreased electrical demand, while the savings compared to gas water heaters would decrease. The electrical demand of electric water heaters of equivalent size to the HP-120-27 depends upon the hot water draw schedule, but ranges between 0 and 18KW; likewise, the demand of the HPWH ranges between 0 and 14KW. Therefore, the electric demand would decrease between 0 and 4KW

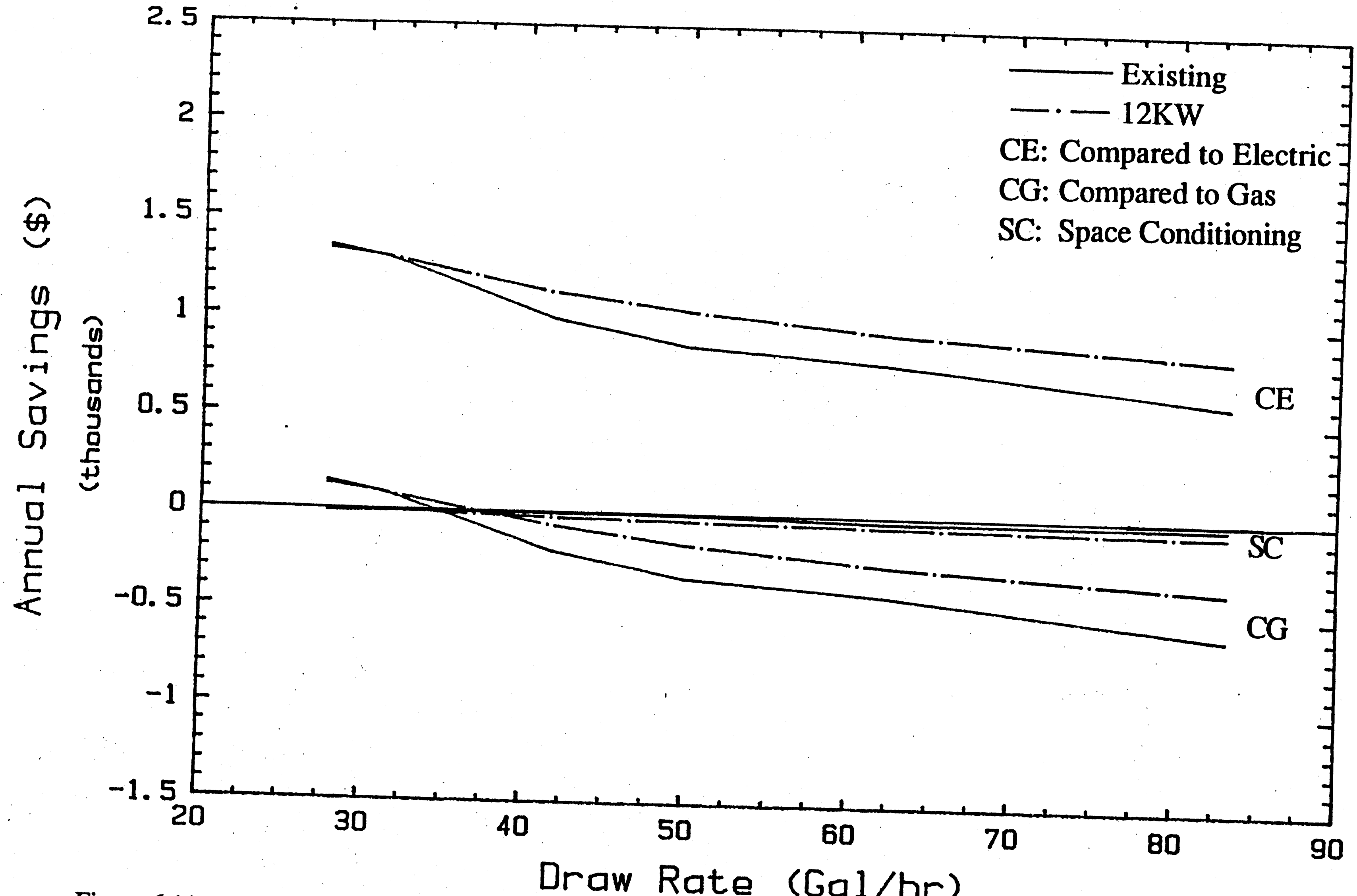

Figure 6.11 Annual savings versus hot water draw rate of existing HPWH to one having a single 12KW backup heater at the top of the tank

(i.e., 18-14) when compared to an electric water heater, but would increase between 0 and 14KW when compared to a gas water heater. Figure 6.12 is a plot of annual demand costs versus electric demand for three charge rates. This plot shows that the increase in savings compared to electric water heaters could be as high as $160, while the decrease in savings compared to gas could be as large as $560. An additional demand savings could occur due to reduced air conditioner load.

Some states require that gas water heaters be in fireproof rooms. The additional construction expense of this room could be avoided by using a HPWH instead of the gas water heater. This savings would only apply when gas and electric water heaters are being considered for new construction; it would not affect the comparison when considering replacement water heaters for restaurants already having a fireproof room.

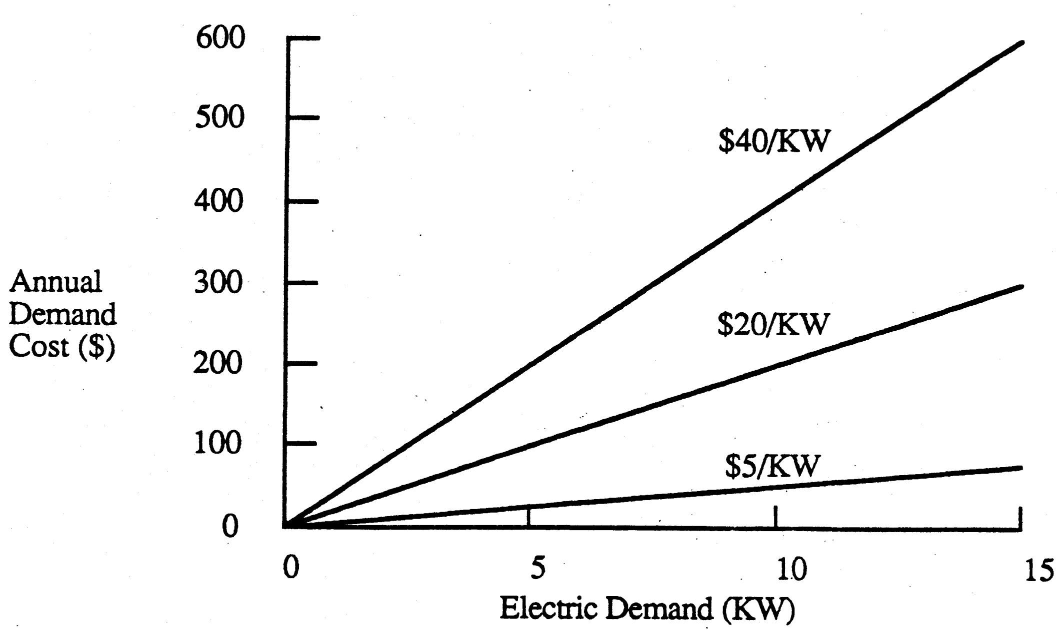

Figure 6.12 Annual demand cost versus electric demand for three demand rates

6.4 Economics of Heat Pump Water Heaters

This section presents an economic comparison of the HPWH to conventional water heaters and gives typical payback periods. Also included is an economic comparison of the existing water heater to a similar water heater having increased storage and to one having having a single 12KW backup heater at the top of the tank.

6.4.1 Economic Theory

The life cycle savings associated with the HPWH is the difference between the net savings of the water heater compared to a conventional water heater and the HPWH's cost over a given period of analysis. The life cycle savings (S_{LC}) can be expressed as (Duffie and Beckman, 1980):

$$S_{LC} = P_1 S - P_2 C_E \tag{6.4.1}$$

where

S = annual savings in current dollars

C_E = cost of equipment

P_1 is the ratio of life cycle savings to the first year's savings and accounts for the fact that after taxes only a percent of the savings is a profit. P_2 is the ratio of the life cycle expenditures due to additional capital investments to the initial investment and accounts for the fact that a percent of the cost of the water heater can be depreciated for several years, resulting in a tax savings. Assuming that no money is borrowed to purchase the HPWH, that the HPWH has no resale value at the end of the analysis, and that there is no increase in property taxes due to the purchase of the HPWH, P_1

and P_2 are defined as

$$P_1 = (1 - t_i)\ PWF(N, i, d) \tag{6.4.2}$$

$$P_2 = 1 + t_p\,(1 - t_i)\ PWF(N, i, d) - t_i\ PWF(N_{min}, 0, d) / N_d - \tag{6.4.3}$$

where

N = number of years of analysis

i = general inflation rate

d = market discount rate

t_i = income tax rate

t_p = property tax rate

N_d = depreciation lifetime in years

N_{min} = minimum of N and N_d

PWF is a present worth factor which accounts for changes in the value of the dollar due to inflation, and it accounts for the fact that the invested money (i.e., selling price of the HPWH) could have been invested in an alternate investment having a return rate of d. The present worth factor is given as

$$PWF(N, i, d) = \frac{1}{(d - i)}\left[1 - \left(\frac{1 + i}{1 + d}\right)^N\right] \tag{6.4.4}$$

The payback period is defined here as the time needed for the cumulative fuel savings caused by using a HPWH instead of a conventional water heater to equal the total initial investment (i.e., selling price). This is equivalent to a zero S_{LC} in equation 6.4.1. In the following sections it is assumed that the economic parameters are as given in Table 6.2

6.4.2 Economic Comparison of HPWH to Conventional Water Heaters

The annual savings required to give a three year payback can be calculated by setting S_{LC} equal to zero and the analysis period equal to three years (N=3) in equation 6.4.1 and solving for the annual savings required.

$$S = C_E P_2 / P_1 \tag{6.4.5}$$

Variable	Value
i	4%
d	8%
t_i	Federal 34%
	State 8% (WI)
	Total 42%
t_p	2.7%
N_d	5 years
P_1 (for N=3)	1.55
P_2 (for N=3)	0.775

Table 6.2 Economic parameters used in this section and the resulting P_1 and P_2 for a three year analysis period (ie. N=3)

The cost of the equipment in the analysis for a HPWH being placed in a new building is the incremental cost of the HPWH over the cost of an alternative water heater. Gas and electric water heaters having the same heating capacity as the HP-120-27 currently cost about $1300, whereas the HP-120-27 costs about $3000. For an incremental equipment cost of $1700, annual savings required to give a three year payback is calculated using equation 6.4.5 to be $850. In Chapter 6 it was noted that the annual savings of the HPWH compared to electric water heaters ranged between $600 and $1400, and between $150 savings and a $600 loss compared to a gas water heater. The electric water heater, therefore, could have a 3 year payback, but a gas water heater could not. If the HPWH is to replace an existing water heater for efficiency purposes only, the equipment cost is equal to the price of the HPWH (assuming no resale value of the old water heater). For an equipment cost of $3000, the required annual savings to give a three year payback is calculated to be $1500. For this case it is unlikely that either the gas or electric water heaters could have a three year payback. These calculations neglect the electric demand and construction savings mentioned in section 6.3. In restaurants where these savings occur, the payback period would decrease. Figure 6.13 is a plot of the payback time versus the annual savings for the economic parameters used in this section (new values of P_1 and P_2 are calculated for each point).

6.4.3 Economic Comparison of Alternative HPWH Designs

A means of comparing the existing HPWH to an alternative design is to determine the maximum increase in the selling price of the water heater that could be paid back in a three year period by the increased savings. This increase in initial cost is calculated with equation 6.4.1, letting S_{LC} be zero and using a three-year analysis period.

$$\Delta C_E = P_1 \, \Delta S \, / \, P_2 \tag{6.4.6}$$

In section 6.2.5 it is shown that an average of about $70 can be saved by increasing the tank size by 50%. The increase in selling price that can be paid back in a three-year period is calculated using equation 6.4.6 to be $140.

In section 6.2.8 it is shown that by having a single 12KW backup heater at the top of the tank the annual savings can be increased by as much as $230. From Figure 6.11 it appears that an average savings is closer to $200. Using equation 6.4.6 with an increased savings of $200, the increase in selling price that could be paid back in three years by increased savings is calculated to be $410.

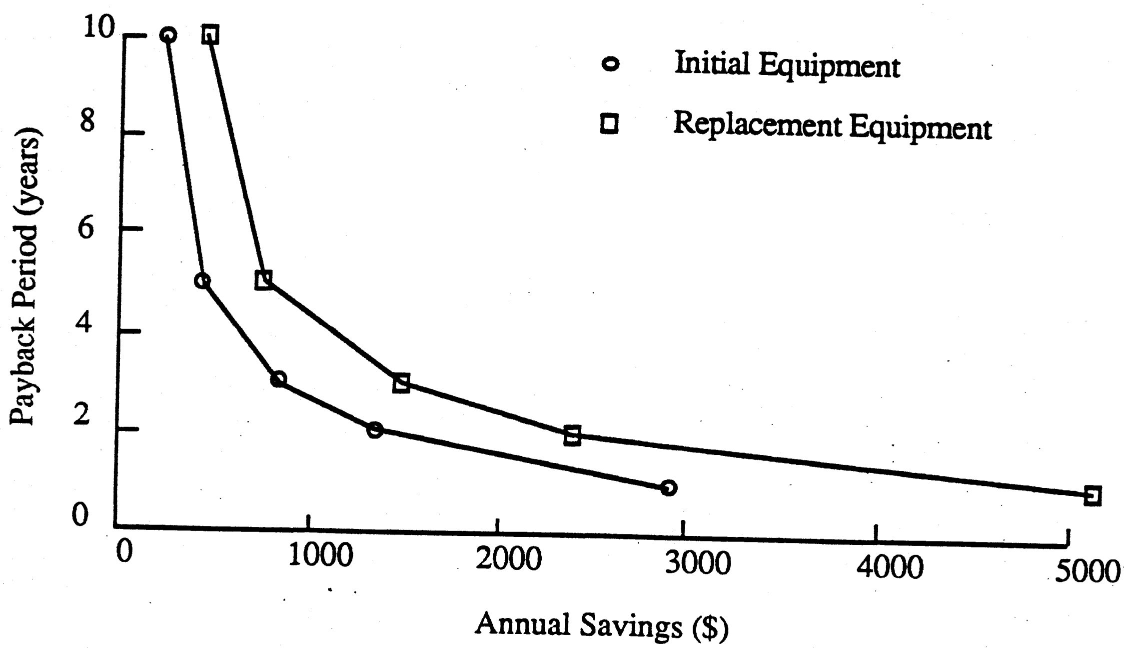

Figure 6.13 Plot of payback period versus first year's savings

Chapter 7
Conclusions and Recommendations

7.1 Conclusions

The goal of this project was to study the performance of HPWHs in restaurant applications and to compare their annual water heating costs to conventional water heaters. Savings due to both water heating and reduced space conditioning costs were considered. In Chapter 3 the performance of the HPWH was found to have a wide range of coefficient-of-performances, varying between 1.3 and 3.3. The major source of variability was found to be caused by the rate at which hot water is drawn from the tank. Two other factors shown to affect the coefficient of performance are the evaporator air temperature and the total daily draw. Raising the setpoint temperatures of the top backup heater was seen to have only a small effect on the performance.

Two design alternatives to the existing HPWH were considered. The first alternative was to have increased storage, and it was determined that increasing the tank size by 50% increased the overall performance of the HPWH by about 7%. The second design alternative was to have a single 12KW backup heater near the top of the tank. This change had no effect at low draw rates, but increased the performance by 20% at the higher draw rates.

In Chapter 6 the annual cost of water heating using a HPWH was compared to conventional water heaters and the savings in space conditioning caused by the HPWH air cooling were investigated. The total savings of the HPWH compared to gas and electric water heaters also have significant variations. Both the cost of the water heating and the space conditioning savings varied with changing building and HPWH parameters. The space conditioning savings were found to be largely dependent upon the magnitude of the internal gains of the building, upon the geographic location, and upon the prices of gas and electricity. For the base case restaurant in Madison having internal gains such that the building had a 9 month cooling load, the space conditioning savings were found to be insignificant.

The savings of the HPWH were found to be dependent upon the gas water heater efficiency, the prices of gas and electricity, the total daily draw, and the parameters mentioned above as affecting the performance of the HPWH. For the base case building, the annual savings compared to electric water heaters varied between \$600 and \$1400, and between \$150 savings and a \$600 loss compared to gas. Payback periods of the HPWH were calculated to range between 2 and 10 years compared to electric, while it may or may not pay for itself through savings when compared to gas. The total savings were noted approximately proportional to the total daily draw. The HPWH savings increased by up to \$200 by raising the evaporator air temperature 10°F. The evaporator, therefore, should be located in the warmest part of the kitchen. In Madison the the evaporator should be located in the warmest part of the kitchen even at the expense of losing the evaporator-air cooling up an exhaust, since this cooling had little affect on the total savings.

Since the total savings of the HPWH are dependent upon many effects unique to each restaurant, no general conclusions can be made. By use of the graphs given in Chapter 6, however, the savings can be estimated for restaurants where building loads and HPWH conditions are known . More accurate estimates can be obtained by using the computer model developed.

Finally, the HPWH's annual savings were found to increase by up to $230 by having a single 12KW backup heater at the top of the storage tank instead of 6KW heaters at both the top and bottom of the tank. It was shown in section 6.4.3 that a $410 increase in the selling price of the HPWH due to this design change could be paid back in 3 years by the increased savings. Increasing the water tank size was determined to increase savings by about $70. A $140 increase in selling price due to the cost of a larger tank could be paid back in 3 years.

7.2 Recommendations

WP&L intends to collect data of the performance of the heat pump water heaters at three restaurant sites. The collected data will allow both water heating savings and space conditioning savings to be determined. This savings can be used to verify and refine the computer model developed. The current HPWH program is specific to the HP-120-27 water heater but could be generalized to include HPWHs of other sizes (i.e., capacities).

Experimental measurements should be performed of the HP-120-27 water heater and of a similar water heater having a single 12KW heater at the top of the tank to determine if the 20% increase in performance could be achieved in practice.

Appendix A
VCC


```
C*****************************************************************************

C    DESCRIPTION: ROUTINE TO SOLVE FOR THE SOLUTION OF THE VCC_ROUTINES
C    AUTHOR      : BYRON PRITCHARD
C    MACHINE     : SOLAR ENERGY LAB'S MICRO VAX
C    DATE        : FALL 1987
C*****************************************************************************

C Driver for the program VCC allowing the performance to be calculated
C    between specified temperature ranges of TTANK and TINCFHXL
c Input file is PARAM.DAT
c Output file is VCCNEW8.OUT

c Variable Definitions
c   Note:  variables not defined are defined in VCC
c    TTANKH   : HIGH TTANK TEMPERATURE FOR PERFORMANCE CALCULATIONS
C    TTANKL   : LOW TTANK TEMPERATURE FOR PERFORMANCE CALCULATIONS
C    TTANKS   : TTANK TEMPERATURE STEP. THE NUMBER OF PERFORMANCE
C               CALCULATIONS BETWEEN TTANKL AND TTANKH IS DETERMINED
C               BY THIS VARIABLE
C    TINCFHXH : HIGH TINCFHX TEMPERATURE FOR PERFORMANCE
C                 CALCULATIONS
C    TINCFHXL : LOW TINCFHX TEMPERATURE FOR PERFORMANCE CALCULATIONS
C    TINCFHXS : TINCFHX TEMPERATURE STEP, SIMILAR TO TTANKS

      real comp(4),condua,subhx(4),expan(3),evap(3)

C OPEN DATA FILES
      open(unit=11,file='vccnew8.out',status='new')
      open(unit=13,file='PARAM.DAT',status='OLD')
      READ(13,*)
      READ(13,*)
      READ(13,*)
      READ(13,*)
      READ(13,*) COMP(1),COMP(2),COMP(3),COMP(4),CONDUA,SUBHX(1),
```

```
     .        SUBHX(2),SUBHX(3),SUBHX(4),EXPAN(1),EXPAN(2),EXPAN(3),
     .        EVAP(1),EVAP(2),EVAP(3),IFREON,TINAIR,TWBAIR,TTANKH,
     .        TTANKL,TTANKS,TINCFHXH,TINCFHXL,TINCFHXS

C CALL VCC FOR EACH SET OF TEMPERATURES
      DO TTANK=TTANKL,TTANKH,TTANKS
        DO TINCFHX=TINCFHXL,TINCFHXH,TINCFHXS
          CALL VCC(COMP,CONDUA,SUBHX,EXPAN,EVAP,IFREON,TINAIR,
     .         TWBAIR,TTANK,TINCFHX,QEVAP,QCOND,QHX,WDOT)
         write(11,'(6F11.1)') ttank,tincfhx,qcond,qhx,WDOT
        END DO
      END DO

C CLOSE DATA FILES
      CLOSE(11)
      CLOSE(13)
      END


      SUBROUTINE VCC(COMP,CONDUA,SUBHX,EXPAN,EVAP,IFREON,TINAIR,
     .         TWBAIR,TTANK,TINCFHX,QEVAP,QCOND,QHX,WDOT)
C**VAIABLE DEFINATION
C    CONDUA  : UA OF THE CONDENSER (BTU/HR-F)
C    COMP(1) : PERCENT CLEARENCE ()
C    COMP(2) : PISTON DISPLACEMENT RATE (FT**3/SEC)
C    COMP(3) : ISENTROPIC EFFICIENCY ()
C    COMP(4) : WORK LOSS IN ELECTRICAL TO MECHANICAL CONVERSION (BTU/HR)
C    ERESID  : RESIDUAL IN INTERATIVE SOLUTION FRO P1
C    EERR    : PERCENT ERROR LIMIT FOR CONVERGENCE ON P1
C    EVAP(1) : UA OF HX IF AIR SIDE IS DRY (BTU/HR-F)
C    EVAP(2) : UA OF HX IF AIR SIDE IS WET (BTU/HR-F)
C    EVAP(3) : MASS FLOW OF AIR (LBM/HR)
C    EXPAN(1): TEMPERATURE STEP IN CAPILLARY TUBE ROUTINE (F)
C    EXPAN(2): CROSS-SECTIONAL AREA OF THE CAP. TUBE (FT**2)
C    EXPAN(3): LENGTH OF THE CAP. TUBE (FT)
```

Appendix A
VCC


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C   H        : ENTHAPLY (BTU/LBM)
C   ICHOKED : FLAG SPECIFING WHETHER CHOKING OCCURED IN CAPPILLARY
C              TUBE
C   ICOUNT  : COUNTER ON MAJOR LOOP
C   ICOUNTE : COUNTER ON INTERNAL LOOP
C   IFREON  : REFRIGERANT NUMBER (IE. FREON 22 IS IFREON=22)
C   ILIMIT  : FLAG SPECIFING IF MRESIDS HAVE BEEN FOUND ON BOTH SIDES OF
C              ZERO
C   LENSTR  : LENGTH AT WHICH CHOKED FLOW OCCURED IN THE CAPILLARY
C              TUBE (FT)
C   MDOT    : MASS FLOW OF THE FREON (LBM/HR)
C   MERR    : PERCENT ERROR LIMIT FOR CONVERGENCE ON P2
C   MRESID  : RESIDUAL IN INTERATIVE SOLUTION FOR P2
C   MRESID_DIFF: PERCENT DIFFERENCE IN MRESID BETWEEN TIME STEPS
C   P1      : PRESSURE IN THE EVAPORATOR (PSIA)
C   P1AIR   : PRESSURE OF FREON IF AT SAME TEMPERATURE AS TINAIR (PSIA)
C   P2      : PRESSURE IN THE CONDENSER (PSIA)
C   P2TANK  : PRESSURE OF FREON IF AT SAME TEMPERATURE AS TTANK (PSIA)
C   QCOND   : HEAT TRANSFER AT BTHE CONDENSER (BTU/HR)
C   QEVAP   : HEAT TRANSFER AT THE EVAPORATOR (BTU/HR)
C   QHX     : HEAT TRANSFER IN THE HEAT EXCHANGER (BTU/HR)
C   S       : ENTROPY
C   SUBHX(1): HEAT EXCHANGER LENGTH (FT)
C   SUBHX(2): UA PER FOOT OF LENGTH (BTU/HR-FT-F)
C   SUBHX(3): SPECIFIC HEAT OF THE COOLING FLUID (BTU/LBM-F)
C   SUBHX(4): MASS FLOW OF THE COOLING FLUID (LBM/HR)
C   T       : TEMPERATURE (F)
C   TINAIR  : INLET AIR TEMPERATURE AT THE EVAPORATOR (F)
C   TINCFHX : INLET TEMP. OF COOLING FLUID IN THE HEAT EXCHANGER (F)
C   TOUTCFHX: OUTLET TEMPERATURE OF THE COOLING FLUID IN THE HX (F)
C   TTANK   : TEMPERATURE OF WATER TANK TEMPERATURE (F)
C   TWBAIR  : WET BULB TEMPERATURE OF EVAPORATOR INLET AIR (F)
C   U       : INTERNAL ENERGY
C   V       : SPECIFIC VOL
C   WDOT    : WORK OF THE MOTOR (BTU/HR)
```

Appendix A
VCC


```
C --DECLARING VARIABLES
      IMPLICIT NONE
      REAL COMP(4),SUBHX(4),EXPAN(3),EVAP(3),P1,P2,TINAIR,
     .   TWBAIR,TTANK,TINCFHX,QEVAP,QCOND,T,S,V,U,WDOT,MDOT,H2,H3,
     .   H4,H5,H6,P3,P4,P5,P6,TOUTCFHX,MRESID,MRESIDN,MRESIDO,
     .   ERESID,ERESIDN,ERESIDO,P2N,P2O,P1N,P1O,P1AIR,H,DELP,
     .   EERR,MERR,CONDUA,LENSTR,QHX,POUT,H1,P2TANK,MRESID_DIFF
      INTEGER IFREON,ICHOKED,ICOUNT,ICOUNTE,ILIMIT

C --SET PERCENTAGE ERROR LIMITS
      EERR = .00001
      MERR = .01

C --CALCULATE UPPER LIMIT ON THE LOW PRESSURE
      CALL FREON(TINAIR,P1AIR,H,S,1.0,V,U,IFREON,15)

C --CALCULATE LOWER LIMIT ON THE HIGH PRESSURE
      CALL FREON((ttank+tincfhx)/2.,P2TANK,H,S,1.0,V,U,IFREON,15)

C --SET INITIAL GUESSES FOR THE CONDENSER AND EVAPORATOR PRESSURES
      P1O = P1AIR*0.50
      P2 = P2TANK*1.10

C --CHECK TO MAKE SURE GUESS MAKES SENSE
  133 IF(P2 .LT. P1O) THEN
        P2 = P2*1.05
        P1O = P1*0.95
        GOTO 133
      END IF

C --PRINT HEADER OF ITERATION RESULTS TO SCREEN
      WRITE(6,5)

C --SET INITIAL VALUES OF FLAGS AND COUNTERS
      ILIMIT = 0
```

```
      ICOUNT = 0


C*****TOP OF MASS ITERATION LOOP (MAJOR LOOP)*****
 10   CONTINUE
      ICOUNT = ICOUNT + 1
      ICOUNTE = 0

C  --STOP PROGRAM IF SOLUTION IS NOT FOUND AFTER 40 ITERATIONS
      IF(ICOUNT .GT. 40) THEN
        WRITE(6,*) ' SOLUTION NOT FOUND AFTER 40 ITERATIONS.'
        STOP
      END IF
C  --GET ONE THE TWO LOOP RESULTS FOR THE SECANT METHOD USED TO
C      CONVERGE
C  --ON P1
      CALL VCC_LOOP_CALC(P1O,P2,ERESIDO,MRESID,COMP,CONDUA,SUBHX,EXPAN,
     .              EVAP,IFREON,TINAIR,TWBAIR,TTANK,TINCFHX,QEVAP,
     .              QCOND,QHX,WDOT)


C*****TOP OF ENTHALPY ITERATION LOOP*****
C  --GET SECOND LOOP RESULT FOR SECANT METHOD WITH SMALL CHANGE IN P1
      P1N = P1O*1.01
      CALL VCC_LOOP_CALC(P1N,P2,ERESIDN,MRESID,COMP,CONDUA,SUBHX,EXPAN,
     .              EVAP,IFREON,TINAIR,TWBAIR,TTANK,TINCFHX,QEVAP,
     .              QCOND,QHX,WDOT)
 20   CONTINUE
      ICOUNTE = ICOUNTE + 1

C  --UPDATE GUESS ON P1
      IF(ABS(ERESIDN-ERESIDO) .LT. 0.000001) THEN
        DELP = 2.0*DELP
        P1 = P1N - DELP
      ELSE IF(ERESIDN*ERESIDO .GT. 0.0) THEN
        DELP = ERESIDN*(P1N - P1O)/(ERESIDN - ERESIDO)
```

```
      P1 = P1N - DELP
    ELSE
      DELP = ERESIDN*(P1N - P1O)/(ERESIDN - ERESIDO)
      P1 = P1N - DELP
    END IF

C  --CHECK TO ASSURE P1 IS A VALID GUESS
    IF(P1 .GT. P1AIR) THEN
      P1 = (P1AIR + P1N)/2.0
    END IF
    IF(P1 .LT. 0) P1 = (P1O+P1N)/2.0
    IF(P1 .LT. .5*P1O) P1=0.75*P1O

C  --GET NEW LOOP RESULTS FOR SECANT METHOD WITH NEW VALUE OF P1
    CALL VCC_LOOP_CALC(P1,P2,ERESID,MRESID,COMP,CONDUA,SUBHX,EXPAN,
   .            EVAP,IFREON,TINAIR,TWBAIR,TTANK,TINCFHX,QEVAP,
   .            QCOND,QHX,WDOT)

C  --CHECK IF SOLUTION HAS CONVERGED
    IF(ABS(ERESID) .GT. EERR) THEN
      ERESIDO = ERESIDN
      ERESIDN = ERESID
      P1O = P1N
      P1N = P1
      GOTO 20
    ELSE IF(ABS(MRESID) .LT. MERR) THEN
C     --CONVERGENCE IS REACHED. SO GOTO END
      GOTO 30
    END IF

C  --START CALCULATIONS FOR MASS LOOP.
C  --NEED TWO LOOP RESULTS OF THE P1 SOLUTION TO PROCEED. ONE MRESID
C  --MUST BE GREATER THAN ZERO THE OTHER LESS THAN ZERO.
C  --SAVE THE MRESID'S AS MRESIDN .GT. 0, MRESIDO .LT. 0.
    IF(ICOUNT .EQ. 1) THEN
```

```
        MRESIDO = MRESID
        P2O = P2
        P2 = P2*1.10
        GOTO 10
      ELSE IF(ICOUNT .EQ. 2) THEN
        IF(MRESID .GT. MRESIDO) THEN
          MRESIDN = MRESID
          P2N = P2
        ELSE
          MRESIDN = MRESIDO
          MRESIDO = MRESID
          P2N = P2O
          P2O = P2
        END IF
      ELSE
        IF(MRESID .GT. 0) THEN
          MRESID_DIFF = MRESID - MRESIDN
          MRESIDN = MRESID
          P2N = P2
        ELSE
          MRESID_DIFF = MRESID - MRESIDO
          MRESIDO = MRESID
          P2O = P2
        END IF
      END IF

C   --PRINT ITERATION RESULTS TO THE SCREEN TO AMMUSE THE USER
      WRITE(6,40) ICOUNT,P1,P2,MRESID,ERESID,WDOT,QCOND+QHX

C   --CHECK TO MAKE SURE AN MRESID IS FOUND ON BOTH SIDES OF ZERO. IF NOT
C   --(ILIMIT=0) THEN GUESS A VALUE OF P2 LIKELY TO GIVE THE NEEDED MRESID
      IF(MRESIDN.GT.0.0 .AND. MRESIDO.GT.0.0 .AND. ILIMIT.EQ.0) THEN
        P2 = P2*.9
 111    IF(P2 .LT. P2TANK .AND. TINCFHX .GT. .99*TTANK) THEN
          P2 = P2*1.01
          GOTO 111
```

```
      END IF
      IF(P2 .LT. P1) P1 = P2*.7
      GOTO 10
    ELSE IF(MRESIDN.LT.0..AND.MRESIDO.LT.0..AND. ILIMIT.EQ.0) THEN
      P2 = P2*1.1
      GOTO 10
    ELSE
      ILIMIT = 1
    END IF

C   --SET NEW GUESS OF P2
    P2 = (P2N + P2O)/2.0
    IF(ABS(MRESID_DIFF/MRESID) .LT. .01) THEN
      IF(MRESID .GT. 0) THEN
        P2 = P2*.999
      ELSE
        P2 = P2*1.001
      END IF
    END IF

C   --DO LOOP CALCULATIONS. IF SOLUTION NOT CONVERGED, USE LOOP RESULTS
C   --IN THE P1-SECANT METHOD ITERATION
    CALL VCC_LOOP_CALC(P1,P2,ERESID,MRESID,COMP,CONDUA,SUBHX,EXPAN,
   .              EVAP,IFREON,TINAIR,TWBAIR,TTANK,TINCFHX,QEVAP,
   .              QCOND,QHX,WDOT)
    IF(ABS(MRESID) .LT. MERR .AND. ABS(ERESID) .LT. EERR) THEN
      GOTO 30
    ELSE
      ERESIDO = ERESID
      P1O = P1
      GOTO 10
    END IF

 30 CONTINUE
C   --SOLTION CONVERGED. PRINT OUT FINAL ITERATION RESULTS AND RETURN
```

```
      WRITE(6,40) ICOUNT,P1,P2,MRESID,ERESID,WDOT,QCOND+QHX

C  --FORMAT STATEMENTS
    5 FORMAT('  I    P1    P2   MRESID    ERESID     WDOT
     .QCOND')
   40  FORMAT(X,I3,2(X,F6.2),X,F10.7,X,F10.7,X,F10.0,X,F10.0)

      RETURN
      END
```

```
C*******************************************************************************

      SUBROUTINE VCC_LOOP_CALC(P1,P2,ERESID,MRESID,COMP,CONDUA,
     .                  SUBHX,EXPAN,EVAP,IFREON,TINAIR,
     .                  TWBAIR,TTANK,TINCFHX,QEVAP,QCOND,
     .                  QHX,WDOT)
C***ROUTINE DESCRIPTION
C  --THIS SUBROUTINE CONNECTS THE INDIVIDUAL COMPONENTS INTO A VAPOR
C  --COMPRESSION CYCLE, PERFORMS THE CALCULATIONS AROUND THE LOOP
C  --AND RETURNS THE RESIDUALS.

C***SEE ROUTINE VCC FOR VARIABLE DEFINATION***

C  --DECLARE VARIABLES
      REAL COMP(4),SUBHX(4),EXPAN(3),EVAP(3),P1,P2,TINAIR,H2,H3,
     .   TWBAIR,TTANK,TINCFHX,QEVAP,QCOND,T,S,V,U,WDOT,MDOT,
     .    H4,H5,H6,P3,P4,P5,P6,TOUTCFHX,MRESID,ERESID,CONDUA,
     .   LENSTR,QHX,POUT,H1
      INTEGER IFREON,ICHOKED

C  --CALCULATE VALUE OF H1 (SAT. BECAUSE LEAVING AN ACCUMULATOR)
       CALL FREON(T,P1,H1,S,1.0,V,U,IFREON,25)

C  --PERFORM CALCULATIONS AROUND THE VCC LOOP
       CALL COMPRESSOR_P(COMP(1),COMP(2),COMP(3),COMP(4),IFREON,P1,
     &             H1,P2,WDOT,MDOT,H2)
      CALL CONDENSER(CONDUA,P2,H2,MDOT,IFREON,TTANK,QCOND,H3,P3)
      CALL SUBCOOLHX(SUBHX(1),SUBHX(2),IFREON,SUBHX(3),TINCFHX,SUBHX(4),
     .             H3,P3,MDOT,TOUTCFHX,H4,P4,QHX)

C  --P5 IS EQUAL TO P1 SINCE PRESSURE DROPS HAVE BE IGNORED IN ALL UNITS
C  --EXCEPT THE COMPRESSOR AND THE EXPANSION DEVICE
      P5 = P1

C  --TWO FLOW PATHS FOR THE DEC UNIT EXIST, THEREFORE USE HALF THE FLOW
```

```
      MDOT = MDOT/2.0
      CALL EXPANSION(IFREON,EXPAN(1),EXPAN(2),EXPAN(3),MDOT,P4,
     .          H4,POUT,H5,ICHOKED,LENSTR)

C  --CALCULATE MRESID USING LENGTHS IF CHOKED AND PRESSURES OTHERWISE
      IF(ICHOKED .EQ. 1) THEN
        MRESID = LENSTR/EXPAN(3) - 1.0
      ELSE
        MRESID = POUT/P5 - 1.0
      END IF

C  --RETURN TO FULL FLOW
      MDOT = MDOT*2.0
      CALL EVAPORATOR(EVAP(1),EVAP(2),IFREON,TINAIR,TWBAIR,EVAP(3),
     .        H5,P5,MDOT,QEVAP,H6,P6)

C  --CALCULATE THE E RESIDUAL
      ERESID = (H1 - H6)/H6

      RETURN
      END
```

Appendix A
COMP


```
      SUBROUTINE COMPRESSOR_P(M,PDR,NP,W_LOSSES,IFREON,PIN,HINCAN,POUT,
     &                     WDOT,MDOT,HOUT)
C--DEFINE VARIABLES
C    M              : PERCENT CLEARANCE (0.0 TO 1.0)
C    PDR            : PISTON DISPLACEMENT RATE (FT**3/SEC)
C    EFFC           : ISENTROPIC EFFICIENCY
C    P,T,H,S,V,U,X (IN) : PRESSURE (PSIA), TEMPERATURE (F), ENTHALPY
C                   (BTU/LBM), ENTROPY (BTU/LBM-F), SPECIFIC VOLUME
C                   (FT**3/LBM), INTERNAL ENERGY (BTU/LBM) AND
C                   QUALITY AT THE INLET TO THE CYLINDER
C    P,T,H,S,V,U,X (OUT): SAME AS ABOVE FOR THE OUTLET POSITION
C    WDOT           : RATE OF WORK BY THE COMPRESSOR (BTU/HR)
C    MDOT           : MASS FLOW RATE OF FREON (LBM/HR)
C    WPERLB         : WORK INTO FREON PER POUND OF FREON (BTU/LBM)
C    HINCAN         : ENTHALPY AT INLET OF THE COMPRESSOR
C    HOUTSTR        : ENTHALPY AT THE OUTLET IF COMPRESSOR WAS
C                     ISENTROPIC
C    EFFCV          : VOLUMETRIC EFFICIENCY
C    W_LOSSES       : WORK LOSSES DUE TO CONVERTION OF ELCTRICAL TO
C                   MECHNICAL WORK
C    WISEN          : ISENTROPIC WORK (BTU/LBM)

C--DECLARE VARIABLES
      IMPLICIT NONE
      REAL *4 M,PDR,EFFC,PIN,HIN,POUT,WDOT,MDOT,HOUT,TOUT,
     .    VOUT,XOUT,VIN,XIN,SIN,WPERLB,HOUTSTR,TSTR,X,VSTR,U,
     .    W_LOSSES,WISEN,TIN,UIN,SOUT,UOUT,II1,II2,II,PI,NP,EFFCV,
     .    CP1,CV1,CP2,CV2,TINP,PINP,SINP,XINP,VINP,UINP,TINM,
     .    PINM,SINM,XINM,VINM,UINM,SP,XP,VP,UP,POUTP,POUTM,
     .    SM,XM,UM,VM,PR,C1,C2,HOUTP,HOUTM,TP,TM,HINP,HINM,USTR,
     .    HINCAN
      INTEGER IFREON,L

      DO L=1,5
C--ADD WORK LOSS TERM TO FREON (INLET FREON IS USED TO COOL THE MOTOR
```

```
C--WINDINGS)
     IF(MDOT .GT. .00001) THEN
       HIN = HINCAN + W_LOSSES/MDOT
     ELSE
       HIN = HINCAN
     END IF
C--ESTABLISH INLET STATE
     CALL FREON(TIN,PIN,HIN,SIN,XIN,VIN,UIN,IFREON,23)
C--CALCULATE ISENTROPIC WORK
     CALL FREON(TSTR,POUT,HOUTSTR,SIN,X,VSTR,USTR,IFREON,24)
     WISEN = (HOUTSTR - HIN)
C--CALCULATE AVERAGE ISENTROPIC INDEX
     HINP = HIN*1.001
     CALL FREON(TINP,PIN,HINP,SINP,XINP,VINP,UINP,IFREON,23)
     CP1 = (HINP - HIN)/(TINP - TIN)
     CALL FREON(TIN*1.001,PINP,HINP,SINP,XINP,VIN,UINP,IFREON,16)
     CV1 = (UINP - UIN)/(TIN*1.001 - TIN)
     II1 = CP1/CV1
     CALL FREON(TP,POUT,HOUTSTR*1.001,SP,XP,VP,UP,IFREON,23)
     CP2 = (HOUTSTR*1.001 - HOUTSTR)/(TP - TSTR)
     CALL FREON(TSTR*1.001,POUTP,HOUTP,SP,XP,VSTR,UP,IFREON,16)
     CV2 = (UP - USTR)/(TSTR*1.001 - TSTR)
     II2 = CP2/CV2
     II = (II1 + II2)/2.0
C--CALCULATE THE POLYTROPIC INDEX
     PI = II*NP/(II*NP - II + 1.0)
C    PI = II/(II - NP*II + NP)
C--CALCULATE THE PRESSURE RATIO
     PR = POUT/PIN
C--CALCULATE THE WORK PER POUND OF REFRIGERANT
     C1 = (PI - 1.0)/PI
     C2 = (II - 1.0)/II
C    HOUT = HIN + WISEN*(1.0-II)*(PR**C1-1.0)/((PR**C2-1.0)*(1.0-PI))
     HOUT = HIN + WISEN*NP*(PR**C1-1.0)/(PR**C2-1.0)
     WPERLB = HOUT - HIN
```

```
C--ESTABLISH OUTPUT STATE
    CALL FREON(TOUT,POUT,HOUT,SOUT,XOUT,VOUT,UOUT,IFREON,23)
C--CALCULATE VOLUMETRIC EFFICIENCY, MASS FLOW AND WORK
    EFFCV = 1 - M*(VIN/VOUT -1)
    MDOT = (PDR*3600)*EFFCV/VIN
    WDOT = MDOT*WPERLB + W_LOSSES
    END DO
    RETURN
    END
```

Appendix A
COND


```
      SUBROUTINE CONDENSER(UA,PIN,HIN,MDOT,IFREON,TTANK,QCOND,
     &                              HOUT,POUT)
C*************************************************************************
C  GENERAL DESCRIPTION:  PROGRAM TO CALCULATE THE ENERGY TRANSFER
C                    IN A CONDENSER. THE INLET REFRIGERANT IS
C                    ASSUMED TO BE SUPPERHEATED; THE EXIT CAN BE
C                    SUPERHEATED, 2 PHASE OR, SUBCOOLED. THE PROGRAM
C                    CALCULATES THE ENERGY TRANSFER BY DIVIDING THE
C                    CONDENSER UP INTO 3 REGIONS (SUPERHEAT, CONDENSING,
C                    AND SUBCOOLING).  IN EACH REGION THE TEMPERATURE IS
C                    CALCULATED USING AN LMTD APPROACH.
C  AUTHOR:              BYRON PRITCHARD
C  MACHINE:             MICROVAX
C  LANGUAGE:            FORTRAN
C  DATE:                SUMMER 1987
C*************************************************************************

C--DEFINE VARIABLES

C  --DECLARE VARIABLES
      IMPLICIT NONE
      REAL *4 TIN,PIN,HIN,S,X,V,U,LMTD,HSATG,TSAT,MDOT,UA,PERC1,HSATF,
     .      TTANK,Q1,Q2,QCOND,TOUTO,TOUT,TOUTN,RESIDN,RESIDO,HOUT,
     .      POUT,PERC2,P,LOGTERM
      INTEGER IFREON,IFLAG

C  --CALCULATE CONSTANTS
      CALL FREON(TIN,PIN,HIN,S,X,V,U,IFREON,23)
      CALL FREON(TSAT,PIN,HSATG,S,1.0,V,U,IFREON,25)
      CALL FREON(TSAT,PIN,HSATF,S,0.0,V,U,IFREON,25)

C  --IF TSAT IS LESS THAN TTANK THEN ASSUME THE SUPERHEATED STEAM IS
C  --COOLED UNTIL TSAT = TTANK.  THIS IS REASONABLE SINCE A CONDENSER
C  --SHOULD BE GREATLY OVERSIZED AS A DESUPERHEAT EXCHANGER
      IF(TSAT .LT. TTANK) THEN
```

```
      TOUT = TTANK
      CALL FREON(TOUT,PIN,HOUT,S,X,V,U,IFREON,12)
      QCOND = MDOT*(HIN - HOUT)
      POUT = PIN
      RETURN
     END IF

C  --REGION 1: SUPERHEATED FREON
     LOGTERM = ALOG((TSAT-TTANK)/(TIN-TTANK))
     IF(ABS(LOGTERM) .LT. .00001) THEN
       LMTD = TSAT - TTANK
     ELSE
       LMTD = (TSAT-TIN)/LOGTERM
     END IF
     Q1 = MDOT*(HIN-HSATG)
     PERC1 = Q1/(LMTD*UA)
     IF(PERC1 .GT. 1.0) THEN
       TOUTO = TSAT
       RESIDO = Q1 - UA*LMTD
C      --GUESS NEW OUTLET FREON TEMPERATURE
       TOUTN = TIN - (TIN - TSAT)*.95
C      --MAKE SURE TOUTN IS GREATER THAN TTANK
 10    IF(TOUTN .LT. TTANK) THEN
         TOUTN = (TOUTO + TTANK)/2.0
       END IF
112    CALL FREON(TOUTN,PIN,HOUT,S,X,V,U,IFREON,12)
      LMTD = (TOUTN-TIN)/ALOG((TOUTN-TTANK)/(TIN-TTANK))
      RESIDN = MDOT*(HIN-HOUT) - UA*LMTD
      TOUT = TOUTN - RESIDN*(TOUTN-TOUTO)/(RESIDN-RESIDO)
      IF(ABS(TOUT-TOUTN).GT..001 .AND. ABS(TOUT-TTANK).GT..001) THEN
        TOUTO = TOUTN
        TOUTN = TOUT
        RESIDO = RESIDN
        GOTO 10
      END IF
```

```
113   QCOND = MDOT*(HIN-HOUT)
        POUT = PIN
        RETURN
      END IF
C  --REGION 2: CONDENSING FREON
      LMTD = TSAT - TTANK
      Q2 = MDOT*(HSATG-HSATF)
      PERC2 = PERC1 + Q2/(LMTD*UA)
      IF(PERC2 .GT. 1.0) THEN
        Q2 = UA*(1.0-PERC1)*LMTD
        HOUT = HSATG - Q2/MDOT
        QCOND = Q1 + Q2
        POUT = PIN
        RETURN
      END IF
C  --REGION 3: SUBCOOLED FREON
      TOUTO = TSAT
      LMTD = TSAT - TTANK
      RESIDO = -UA*(1.0-PERC2)*LMTD
C  --GUESS ANOTHER TOUT
      TOUTN = .95*TSAT
C  --MAKE SURE TOUTN IS GREATER THAN TTANK
 111  IF(TOUTN .LE. TTANK) THEN
        TOUTN = TOUTN*1.01
        GOTO 111
      END IF
      IFLAG = 0.0
 20   LMTD = (TOUTN-TSAT)/ALOG((TOUTN-TTANK)/(TSAT-TTANK))
      CALL FREON(TOUTN,P,HOUT,S,0.0,V,U,IFREON,15)
      RESIDN = MDOT*(HSATF - HOUT) - UA*(1.0-PERC2)*LMTD
      TOUT = TOUTN - RESIDN*(TOUTN-TOUTO)/(RESIDN-RESIDO)
      IF(TOUT .LT. TTANK) THEN
        IF(IFLAG .EQ. 1) THEN
C         --THE OUTLET MUST BE PRETTY CLOSE TO TTANK TO MAKE IT HERE
C         --TWICE, THEREFORE ASSUME IT IS EQUAL TO TTANK
          TOUT = TTANK
```

```
      CALL FREON(TOUT,P,HOUT,S,0.0,V,U,IFREON,15)
      QCOND = MDOT*(HSATF-HOUT) + Q1 + Q2
      POUT = PIN
      RETURN
    END IF
    IFLAG = 1.0
    TOUT = TTANK*1.01
  END IF
  IF(ABS(TOUT-TOUTN) .GT. .01) THEN
    TOUTO = TOUTN
    TOUTN = TOUT
    RESIDO = RESIDN
    GOTO 20
  END IF
  QCOND = MDOT*(HSATF-HOUT) + Q1 + Q2
  POUT = PIN
  RETURN
  END
```

Appendix A
SUBCOOLED_HX


```
      SUBROUTINE SUBCOOLHX(HXLEN,UAPERFT,IFREON,CPCF,TINCF,MDOTCF,
     .                HIN,PIN,MDOT,TOCF,HOUT,POUT,QHX)
C***********************************************************************
C  GENERAL DESCRIPTION:  PROGRAM TO CALCULATE THE ENERGY TRANSFER
C                   IN A COUNTER FLOW HEAT EXCHANGER BETWEEN
C                   A REFRIGERANT (EITHER SUBCOOLED OR IN THE
C                   VAPOR DOME) AND A CONSTANT Cp FLUID.  THE
C                   PROGRAM ASSUMES THE SUBCOOLING TO BE SMALL
C  AUTHOR:             BYRON PRITCHARD
C  MACHINE:            MICROVAX
C  LANGUAGE:           FORTRAN
C  DATE:            SUMMER 1987
C***********************************************************************

C--DEFINE VARIABLES
C    CPCF         : SPECIFIC HEAT OF THE COOLING FLUID (BTU/LBM-F)
C    CPF          : SPECIFIC HEAT OF THE FREON (BTU/LBM-F)
C    EFF1         : EFFECTIVENESS IN TUBE WHERE THE FREON IS IN THE
C                 VAPOR DOME
C    EFF2         : EFFECTIVENESS IN TUBE WHERE FREON IS SUBCOOLED
C    HIN,PIN      : INLET CONDITIONS OF THE FREON: ENTHALPY (BTU/LBM)
C                 PRESSURE (PSIA)
C    HOUT,POUT    : OUTLET CONDITIONS OF THE FREON
C    HXLEN        : LENGTH OF THE HEAT EXCHANGER
C    IFREON       : FREON NUMBER (ie. FREON #22>> IFREON=22)
C    MCPCF        : MASS FLOW TIMES THE SPECIFIC HEAT OF THE COOLING FLUID
C    MCPF         : MASS FLOW TIMES THE SPECIFIC HEAT OF THE FREON (BTU/HR-F)
C    MCPMAX       : THE MAXIMUM OF 'MCPF' AND 'MCPCF'
C    MCPMIN       : THE MINIMUM OF 'MCPF' AND 'MCPCF'
C    MDOT         : MASS FLOW RATE OF THE FREON (LBM/HR)
C    MDOTCF       : MASS FLOW RATE OF THE COOLING FLUID (LBM/HR)
C    Q1           : HEAT FLOW IN TUBE REGION CORRESPONDING TO EFF1 (BTU/HR)
C    Q2           : HEAT FLOW IN TUBE REGION CORRESPONDING TO EFF2 (BTU/HR)
C    RESIDN,RESIDO: DIFFERENCE BETWEEN THE ACTUAL Q1 AND THE Q1 GIVE BY
C                 THE CURRENT GUESS OF XSTR
```

```
C    TINCF       : THE COOLING FLUID INLET TEMPERATURE (F)
C    TMCF        : THE COOLING FLUID TEMPERATURE AT XSTR (F)
C    TOCF        : THE COOLING FLUID OUTLET TEMPERATURE (F)
C    UAPERFT     : THE UA PER FOOT LENGTH OF HEAT EXCHANGER (BTU/HR
C              SUBCOOLING FIRST APPEARS (FT)
C    XSTRO,XSTRN : OLD AND NEW GUESSES OF XSTR IN ITERATIVE SOLUTION (FT)
C    T,P,H,S,V,U,X: TEMPERATURE (F), PRESSURE (PSIA), ENTHALPY (BTU/LBM),
C     (SAT OR IN)  ENTROPY (BTU/LBM-F), SPECIFIC VOLUME (FT**3/LBM),
C              INTERNAL ENERGY (BTU/LBM), QUALITY. 'IN' CORRESPONDS TO
C              THE INLET STATE:

C--DECLARE VARIABLES
      IMPLICIT NONE
      REAL *4 HXLEN,UAPERFT,CPCF,TINCF,MDOTCF,HIN,PIN,MDOT,TOCF,HOUT,
     .    POUT,TSAT,HSAT,SSAT,VSAT,USAT,SIN,VIN,UIN,XIN,MCPCF,MCPF,
     .    Q1,Q2,EFF1,EFF2,TIN,CPF,MCPMIN,MCPMAX,XSTR,XSTRO,XSTRN,
     .    TMCF,PSAT,RESIDN,RESIDO,QHX
      INTEGER IFREON
      COMMON Q1,Q2,XSTRN,TMCF,EFF1,EFF2

      MCPCF = MDOTCF*CPCF

C--CHECK IF INLET FREON IS SUBCOOLED
      CALL FREON(TSAT,PIN,HSAT,SSAT,0.0,VSAT,USAT,IFREON,25)
      IF(HIN .LE. HSAT) THEN
C-----MAKE CALCULATIONS FOR SUBCOOLED FLUID THE ENTIRE LENGTH OF TUBE
        CALL FREON(TIN,PSAT,HIN,SIN,0.0,VIN,UIN,IFREON,35)
        MCPF = MDOT*CPF(PIN,IFREON)
        IF(MCPF .LT. MCPCF) THEN
          MCPMIN = MCPF
          MCPMAX = MCPCF
        ELSE
          MCPMIN = MCPCF
          MCPMAX = MCPF
        END IF
```

```
      CALL EFF2CALC(HXLEN-XSTR,UAPERFT,MCPMIN,MCPMAX,EFF2)
      Q2 = EFF2*MCPMIN*(TIN - TINCF)
      TOCF = TINCF + Q2/MCPCF
      HOUT = HIN - Q2/MDOT
    ELSE
C-----MAKE CALCULATIONS FOR 2 PHASE FLOW THE ENTIRE LENGTH OF THE TUBE
      EFF1 = 1 - EXP(-UAPERFT*HXLEN/MCPCF)
      Q1 = EFF1*MCPCF*(TSAT - TINCF)
      HOUT = HIN - Q1/MDOT
      TOCF = TINCF + Q1/MCPCF
C-----CHECK TO SEE IF THE OUTLET IS ACTUALLY TWO PHASE
      IF(HOUT .LT. HSAT) THEN
C--------IF NOT TWO PHASE OUTLET THE ITERATIVE SOLVE FOR THE SOLUTION
C       SET UP VALUES TO START THE ITERATION
        MCPF = MDOT*CPF(PIN,IFREON)
        IF(MCPF .LT. MCPCF) THEN
          MCPMIN = MCPF
          MCPMAX = MCPCF
        ELSE
          MCPMIN = MCPCF
          MCPMAX = MCPF
        END IF
        Q1 = MDOT*(HIN-HSAT)
        XSTRO = HXLEN
        RESIDO = Q1 - EFF1*MCPCF*(TSAT - TINCF)
        XSTRN = .5*HXLEN
C--------START ITERATION
10      EFF1 = 1.0 - EXP(-UAPERFT*XSTRN/MCPCF)
        CALL EFF2CALC(HXLEN-XSTRN,UAPERFT,MCPMIN,MCPMAX,EFF2)
        Q2 = EFF2*MCPMIN*(TSAT - TINCF)
        TMCF = TINCF + Q2/MCPCF
        RESIDN = Q1 - EFF1*MCPCF*(TSAT - TMCF)
C--------CHECK CONVERGENCE
        IF(ABS(RESIDN/(Q1+Q2)) .GT. .001) THEN
C-----------FIND NEW GUESS OF XSTR
          IF(ABS((RESIDN-RESIDO)/(XSTRN-XSTRO)) .LT. .00001) THEN
```

```
          IF(RESIDN .GT. 0) XSTR = XSTRN*1.5
          IF(RESIDN .LT. 0) XSTR = XSTRN*.667
        ELSE
          XSTR = XSTRN - RESIDN*(XSTRN-XSTRO)/(RESIDN-RESIDO)
        END IF
        IF(XSTR .GT. HXLEN) XSTR = (HXLEN + XSTRN)/2.0
        IF(XSTR .LT. 0.0) XSTR = XSTRN/2.0
        RESIDO = RESIDN
        XSTRO = XSTRN
        XSTRN = XSTR
        GOTO 10
      ELSE
C-----------CALCULATE OUTPUTS USING THE CONVERGED SOLUTION
        TOCF = TMCF + Q1/MCPCF
        HOUT = HSAT - Q2/MDOT
      END IF
    END IF
  END IF
C--NO PRESSURE DROP IS BUILT INTO THE SOLUTION
  QHX = MDOT*(HIN-HOUT)
  POUT = PIN
  RETURN
  END

  SUBROUTINE EFF2CALC(LEN,UAPERFT,MCPMIN,MCPMAX,EFF2)
C--CALCULATES EFFECTIVENESS FOR A ONE-PASS COUNTERFLOW HEAT
C  EXCHANGER
  IMPLICIT NONE
  REAL *4 EFF2,NTU,UAPERFT,LEN,MCPMIN,MCPMAX,C
  NTU = UAPERFT*LEN/MCPMIN
  C = MCPMIN/MCPMAX
  EFF2 = (1.0 - EXP(-NTU*(1-C)))/(1.0 - C*EXP(-NTU*(1-C)))
  RETURN
  END
```

```
      FUNCTION CPF(P,IFREON)
C--FUNCTION TO CALCULATE THE SPECIFIC HEAT OF A SUBCOOLED LIQUID
      CALL FREON(T,P,H,S,0.0,V,U,IFREON,25)
      TRANK = T + 460.0
      tplus = TRANK*1.01 - 460
      CALL FREON(TPLUS,PSAT,HPLUS,S,0.0,V,U,IFREON,15)
      tMINUS = TRANK*.99 - 460
      CALL FREON(TMINUS,PSAT,HMINUS,S,0.0,V,U,IFREON,15)
      CPF = (HMINUS - HPLUS)/(TMINUS - TPLUS)
      RETURN
      END
```

```
      SUBROUTINE EXPANSION(IFREON,DTEMP,CAPAREA,CAPLEN,MDOT,PIN,
     .                HIN,POUT,HOUT,ICHOKED,LEN)
C*****************************************************************************
C
C GENERAL DESCRIPTION:  CALCULATES PRESSURE DROP IN A CAPILLARY TUBE
C                  IF NOT CHOKED FLOW.  IF FLOW IS CHOKED IT INFORMS
C                  THE MAIN PROGRAM BUT DOES NOT CALCULATE THE EXIT
C                  CONDITIONS
C AUTHOR          : BYRON PRITCHARD
C MACHINE         : MICRO VAX
C DATE            : SUMMER '87
C
C*****************************************************************************

C--INPUT-OUTPUT DEFINITION--------------------------------------------

C   CAPAREA:  CROSSECTIONAL AREA OF THE CAPILLARY TUBE (FEET)
C   CAPLEN :  LENGTH OF THE CAPILLARY TUBE (FT**2)
C   DTEMP  :  TEMPERATURE CHANGE TO MARCH THROUGH TWO-PHASE
C                  SOLUTION WITH.
C            THE PROGRAM MAY REDUCE THIS NEAR THE END OF THE SOLUTION SO
C            THE FINAL 'LEN' = 'CAPLEN' WITHIN .1% (DEGREES F)
C   ICHOKED:  FLAG THAT TELLS MAIN PROGRAM WHETHER THE FLOW CHOKED
C   IFREON :  GIVES FREON NUMBER (ie. FREON22 >> 'IFREON' = 22)
C   MDOT   :  MASS FLOW RATE THROUGH CAPILLARY TUBE (LBM/HR)
C   PIN, HIN :  PRESSURE (PSIA) AND ENTHALPY (BTU/LBM) AT CONDENSER EXIT
C   PH, HOUT :  PRESSURE (PSIA) AND ENTHALPY (BTU/LBM) AT TUBE EXIT

C--VARIABLE DEFINITION------------------------------------------------

C   DELPSC      : PRESSURE DIFFERENCE BETWEEN 'PIN' AND THE PRESSURE AT
C               FLUID AT HIN = HSAT
C   DELTAL      : CHANGE IN LENGTH CORRESPONDING TO THE CHANGE IN TEMP
C               'WDTEMP'
C   DELTAS      : CHANGE IN ENTROPY CORRESPONDING TO THE CHANGE IN TEMP
```

```
C              'WDTEMP'
C   DIA         : CAPILLARY TUBE INSIDE DIAMETER
C   FRICT       : MEAN FRICTION FACTOR
C   FRICTA (OR B): FRICTION FACTOR AT LOCATION A (OR B)
C   LEN         : CUMMULATIVE LENGTH IN SOLUTION
C   MU          : VISCOSITY AT A (MUA) OR B (MUB)
C   RE          : REYNOLDS NUMBER AT A (REA) OR B (REB)
C   VEL:        : VELOCITY AT A (VELA), B (VELB) AND AVERAGE (VELBAR)
C   WDTEMP      : WORKING VALUE OF DTEMP.  IT MAY BE DIFFERENT THAN
C                 DTEMP TO ASSURE THE FINAL 'LEN' = 'CAPLEN' WITHIN .1%
C   A,B,C       : CONSTANTS DEFINED BY THE SOLUTION BY STOECKER
C   HF,SF,VF,UF : ENTHALPY (BTU/LBM), ENTROPY (BTU/LBM-F), SPECIFIC
C          VOLUME (FT**3/LBM) AND INTERNAL ENERGY (BTU/LBM) FOR SAT. FLUID
C   HG,SG,VG,UG : SAME AS ABOVE FOR SATURATED GAS
C   P,H,T,S,V,U,X: PRESSURE (PSIA), ENTHALPY (BTU/LBM), TEMPERATURE(DEG F),
C   (LOCATIONS A,  ENTROPY (BTU/LBM-F), SPECIFIC VOLUME (FT**3/LBM),
C   B, AND SAT )  INTERNAL ENERGY (BTU/LBM) AND QUALITY AT LOCATIONS A,
C                 B, AND SAT

C--DECLARE VARABLES
      IMPLICIT NONE
      REAL*4 DTEMP,CAPAREA,MDOT,PIN,HIN,POUT,HOUT,DIA,TA,PA,SA,VA,UA,XA,
     .     VELA,MUA,REA,FRICTA,TB,PB,HB,SB,XB,VB,UB,HFB,SFB,VFB,
     .     UFB,HGB,SGB,VGB,UGB,A,B,C,VELB,REB,FRICTB,FRICT,VELBAR,
     .     DELTAL,HA,MUB,LEN,CAPLEN,DELTAS,WDTEMP,TSAT,
     .     HSAT,SSAT,VSAT,USAT,DELPSC,PSAT
      INTEGER IFREON,ICHOKED,ISUBCOL

C--SET INITIAL VALUES
      ICHOKED = 0
      LEN = 0.0
      DIA = SQRT(CAPAREA*4.0/3.14159)
      WDTEMP = DTEMP

C--SET STATE 'A' EQUAL TO STATE '3' DEFINED BY 'HIN' AND 'PIN'
```

```
C  AND CALCULATE NECESSARAY VALUES FOR STATE 'A'
     PA = PIN
     HA = HIN
     CALL FREON(TSAT,PA,HSAT,SSAT,0.0,VSAT,USAT,IFREON,25)
     IF(HSAT .GT. HA) THEN
       CALL FREON(TA,PSAT,HA,SA,0.0,VA,UA,IFREON,35)
       ISUBCOL = 1
     ELSE
       CALL FREON(TA,PA,HA,SA,XA,VA,UA,IFREON,23)
       ISUBCOL = 0
     END IF
     VELA = MDOT*VA/CAPAREA
     CALL VISC(IFREON,TA,XA,MUA)
     REA = VELA*DIA/(MUA*VA)
     FRICTA = 0.33/REA**0.25

C--CHECK IF FLUID IS INITIALLY SUBCOOLED
C  IF SUBCOOLED CALCULATE LENGTH TO GIVE PRESSURE DROP BETWEEN 'PA'
C  AND THE PRESSURE WHERE 'HA' = HSAT LIQUID
     IF(ISUBCOL .EQ. 1) THEN
       XB = 0.0
       CALL FREON(TB,PB,HA,SB,XB,VB,UB,IFREON,35)
       DELPSC = PA - PB
       VELB = MDOT*VB/CAPAREA
       CALL VISC(IFREON,TB,0.0,MUB)
       REB = VELB*DIA/(MUB*VB)
       FRICTB = 0.33/REB**0.25
       FRICT = (FRICTA + FRICTB)/2.0
       VELBAR = (VELA + VELB)/2.0
       LEN = DELPSC*DIA*2.0*CAPAREA/(FRICT*VELBAR*MDOT)*6.0093E+10
C-----IF 'LEN' IS GREATER THAN 'CAPLEN' THEN CALCULATE THE PRESSURE DROP
C     JUST FOR 'CAPLEN' AND RETURN TO MAIN PROGRAM
C     OTHERWISE SET STATE 'A' TO 'B' AND PROCEDE
       IF(LEN .GT. CAPLEN) THEN
         VELB = VELA + CAPLEN*(VELB - VELA)/LEN
         REB = VELB*DIA/(MUB*VB)
```

```
        FRICTB = 0.33/REB**0.25
        FRICT = (FRICTA + FRICTB)/2.0
        VELBAR = (VELA + VELB)/2.0
        POUT = PIN - CAPLEN*FRICT*VELBAR*MDOT/
     .       (6.0093E+10*DIA*2.0*CAPAREA)
        HOUT = HA
        RETURN
      ELSE
        PA = PB
        SA = SB
        XA = XB
        VA = VB
        TA = TB
        VELA = VELB
        REA = REB
        FRICTA = FRICTB
      END IF
    END IF

C--SOLVE FOR PRESSURE DROP IN PORTION OF THE TUBE WHERE 2 PHASE FLOW
C EXIST USING THE CALCULATION PROCEDURE OUTLINED IN STOECKER
    DO WHILE(LEN .LE. CAPLEN)
      TB = TA - WDTEMP
      CALL FREON(TB,PB,HFB,SFB,0.0,VFB,UFB,IFREON,15)
      CALL FREON(TB,PB,HGB,SGB,1.0,VGB,UGB,IFREON,15)
      A = (VGB-VFB)**2*(MDOT/CAPAREA)**2/2.0
      B = 3.2467E+11*(HGB-HFB) + VFB*(VGB-VFB)*(MDOT/CAPAREA)**2
      C = 3.2467E+11*(HFB-HA) + (MDOT/CAPAREA)*VFB**2/2.0-VELA**2/2.0
      XB = (-B + SQRT(B**2 - 4.0*A*C))/(2.0*A)
      CALL FREON(TB,PB,HB,SB,XB,VB,UB,IFREON,15)
      IF(XB .LT. 0.0 .OR. XB .GT. 1.0) THEN
        WRITE(6,*) ' QUALITY=',XB,' IS OUT OF RANGE (0 TO 1)'
c         STOP
      END IF
      VELB = MDOT*VB/CAPAREA
```

```
      CALL VISC(IFREON,TB,XB,MUB)
      REB = VELB*DIA/(MUB*VB)
      FRICTB = 0.33/REB**0.25
      FRICT = (FRICTA + FRICTB)/2.0
      VELBAR = (VELA + VELB)/2.0
      DELTAL = -(2.0*CAPAREA*DIA)/(MDOT*VELBAR*FRICT)*
     .         (MDOT*(VELB-VELA)/CAPAREA + (PB-PA)*6.0093E+10)
      LEN = LEN + DELTAL
      DELTAS = SB - SA
C-----IF THE CUMMULATIVE LENGTH IS GREATER THAN .1% OF THE 'CAPLEN' THEN
C   HALF THE TEMPERATURE STEP (WDTEMP) AND RESET 'TB' AND 'LEN' AND
C   REDO THIS STEP
      IF(LEN .GT. CAPLEN+.001*CAPLEN) THEN
        TB = TA + WDTEMP
        WDTEMP = WDTEMP/2.0
        LEN = LEN - DELTAL
      ELSE
C--------CHECK TO SEE IF A CHOCKED CONDITION HAS OCCURED
C     IF IT HAS RETURN TO MAIN WITH 'ICHOKED' = 1
C     OTHERWISE SET STATE 'A' EQUAL TO 'B' AND CONTINUE
        IF(DELTAL .LT. 0.0 .OR. DELTAS .LT. 0.0) THEN
          ICHOKED = 1
          HOUT = HB
          POUT = PB
          RETURN
        END IF
        PA = PB
        HA = HB
        SA = SB
        XA = XB
        VA = VB
        TA = TB
        VELA = VELB
        REA = REB
        FRICTA = FRICTB
      END IF
```

```
      END DO

C--SET UP OUTPUT AND RETURN
      POUT = PB
      HOUT = HB
      RETURN
      END

!-----------------------------------------------------------------
C--SUBROUTINE TO FIND VISCOUSITY OF A REFRIGERANT IN SATURATION DOME
      SUBROUTINE VISC(IFREON,T,X,MU)
      IMPLICIT NONE
      REAL *4 T,X,MU,TDEGC,MUG,MUF
      INTEGER IFREON
      IF(X .GT. 1.0) X=1.0
      IF(X .LT. 0.0) X=0.0
      IF(IFREON .EQ. 22) THEN
        TDEGC = (T-32)/1.8
        MUF = 0.0002367 - 1.715E-6*TDEGC + 8.869E-9*TDEGC**2
        MUF = MUF/4.1338E-4
        MUG = 11.945E-6 + 50.06E-9*TDEGC + 0.256E-9*TDEGC**2
        MUG = MUG/4.1338E-4
      ELSE
        WRITE(6,*) ' NO VISCOSITY INFO ON REFRIGERANT ',IFREON
        STOP
      END IF
      MU = MUF*(1.0 - X) + MUG*X
      RETURN
      END
```

Appendix A
EVAP


```
      SUBROUTINE EVAPORATOR(UADRY,UAWET,IFREON,TINAIR,TWBAIR,MDOTAIR,
     .                HIN,PIN,MDOT,QEVAP,HOUT,POUT)
C***************************************************************************
C GENERAL DESCRIPTION:  PROGRAM TO CALCULATE THE ENERGY TRANSFER
C                    IN A CROSS FLOW AIR HEAT EXCHANGER
C AUTHOR:            BYRON PRITCHARD
C MACHINE:            MICROVAX
C LANGUAGE:            FORTRAN
C DATE:             SUMMER 1987
C***************************************************************************

C--DEFINE VARIABLES
C   CPCAIR     : SPECIFIC HEAT OF THE AIR (BTU/LBM-F)
C   CPF2      : SPECIFIC HEAT OF THE FREON (BTU/LBM-F)
C   CPAIRS     : EFFECTIVE SATURATION SPECIFIC HEAT OF AIR
C   EFF1      : EFFECTIVENESS IN TUBE WHERE THE FREON IS IN THE
C             VAPOR DOME
C   EFF2      : EFFECTIVENESS IN TUBE WHERE FREON IS SUBCOOLED
C   HIN,PIN    : INLET CONDITIONS OF THE FREON: ENTHALPY (BTU/LBM)
C             PRESSURE (PSIA)
C   HINAIR     : ENTHALPY OF THE AIR AT THE INLET
C   HOUTAIR     : ENTHALPY OF THE AIR AT THE OUTLET
C   HOUT,POUT   : OUTLET CONDITIONS OF THE FREON
C   HWFS       : SATURATION ENTHALPY OF THE AIR IF AT THE FREON SAT. TEMP
C   HWFSA,B     : HWFS AT TWO TEMPS USED TO FIND CPAIRS
C   IFREON     : FREON NUMBER (ie. FREON #22>> IFREON=22)
C   MCPAIR     : MASS FLOW TIMES THE SPECIFIC HEAT AIR
C   MCPF      : MASS FLOW TIMES THE SPECIFIC HEAT OF THE FREON (BTU/HR-F)
C   MCPMAX      : THE MAXIMUM OF 'MCPF' AND 'MCPCF'
C   MCPMIN     : THE MINIMUM OF 'MCPF' AND 'MCPCF'
C   MDOT      : MASS FLOW RATE OF THE FREON (LBM/HR)
C   MDOTAIR     : MASS FLOW RATE OF THE AIR (LBM/HR)
C   PER2P     : PERCENT OF THE HEAT EXCHANGER IN THE VAPOR DOME REGION
C   PER2PO,PER2PN: OLD AND NEW GUESSES OF PER2P IN ITERATIVE SOLUTION
C   Q1        : HEAT FLOW IN TUBE REGION CORRESPONDING TO EFF1 (BTU/HR)
```

```
C    Q2          : HEAT FLOW IN TUBE REGION CORRESPONDING TO EFF2 (BTU/HR)
C    QEVAP       : HEAT TRANSFER FROM THE EVAPORATOR
C    QEVAPDRY    : QEVAP IF THE SURFACE IS TOTALLY DRY
C    QEVAPWET    : QEVAP IF THE SURFACE IS TOTALLY WET
C    RESIDN,RESIDO: DIFFERENCE BETWEEN THE ACTUAL Q1 AND THE Q1 GIVE BY
C                THE CURRENT GUESS OF XSTR
C    TINAIR      : THE AIR INLET TEMPERATURE (F)
C    TAIRDP      : DEW PONIT OF THE AIR
C    TWBAIR      : WET BULB TEMPERATURE OF THE INLET AIR
C    T,P,H,S,V,U,X: TEMPERATURE (F), PRESSURE (PSIA), ENTHALPY (BTU/LBM),
C     (SAT OR IN)  ENTROPY (BTU/LBM-F), SPECIFIC VOLUME (FT**3/LBM),
C                INTERNAL ENERGY (BTU/LBM), QUALITY.  'IN' CORRESPONDS TO
C                THE INLET STATE:
C    UADRY       : UA WHEN AIR SIDE SURFACE IS DRY
C    UAWET       : UA WHEN AIR SIDE SURFACE IS WET

C--DECLARE VARIABLES
      IMPLICIT NONE
      REAL *4 PSYDAT(9),INFO(10)
      REAL *4 HIN,PIN,MDOT,HOUT,TAIRDP,TINAIR,TWBAIR,QEVAPWET,
     .      POUT,TSAT,HSAT,SSAT,VSAT,USAT,MCPF,QEVAP,QEVAPDRY,UAWET,
     .       Q1,Q2,EFF1,EFF2,MCPMIN,MCPMAX,PER2P,PER2PN,PER2PO,
     .      PSAT,RESIDN,RESIDO,CPAIR,MCPAIR,MDOTAIR,UADRY,
     .      HINAIR,HOUTAIR,HWFS,CPAIRS,HWFSA,HWFSB,CPF2,TWALL,TOUT,
     .      TOUTN,V,U,X,S,CPAIRSN
      INTEGER IFREON,IBEWET

      IBEWET = 0
      CPAIR = 0.24
      MCPAIR = MDOTAIR*CPAIR

C  --CALCULATE THE STATE FOR PIN AND QUALITY=1
      CALL FREON(TSAT,PIN,HSAT,SSAT,1.0,VSAT,USAT,IFREON,25)
C  --CALCULATE THE DEW POINT
      PSYDAT(1) = 1.0
```

```
      PSYDAT(2) = TINAIR
      PSYDAT(3) = TWBAIR
      CALL PSYCH(INFO,2,1,0,PSYDAT)
      TAIRDP = PSYDAT(5)
      HINAIR = PSYDAT(7)
C  --MAKE CALCULATIONS ASSUMING A COMPLETELY DRY SURFACE ON THE AIR
C     SIDE
C  --MAKE CALCULATIONS FOR TWO PHASE FLOW THE ENTIRE LENGTH OF THE
C     TUBE
      EFF1 = 1 - EXP(-UADRY/MCPAIR)
      QEVAPDRY = EFF1*MCPAIR*(TINAIR - TSAT)
      HOUT = HIN + QEVAPDRY/MDOT
C  --CHECK TO SEE IF THE OUTLET IS ACTUALLY TWO PHASE
      IF(HOUT .GT. HSAT) THEN
C    --IF NOT TWO PHASE OUTLET THEN ITERATIVE SOLVE FOR PER2P
C    --SET UP VALUES TO START THE ITERATION
        PER2PO = 1.0
        Q1 = MDOT*(HSAT-HIN)
        RESIDO = Q1 - EFF1*MCPAIR*(TINAIR-TSAT)
        PER2PN = 0.5
C    --START ITERATION
 10     EFF1 = 1 - EXP(-UADRY*PER2PN/MCPAIR)
        RESIDN = Q1 - EFF1*MCPAIR*PER2PN*(TINAIR-TSAT)
        IF(ABS((RESIDN)/Q1) .GT. .001) THEN
          PER2P = PER2PN - RESIDN*(PER2PN-PER2PO)/(RESIDN-RESIDO)
          PER2PO = PER2PN
          PER2PN = PER2P
          RESIDO = RESIDN
          GOTO 10
        END IF
        MCPF = MDOT*CPF2(PIN,IFREON)
        IF(MCPF .GT. MCPAIR) THEN
          MCPMAX = MCPF
          MCPMIN = MCPAIR
        ELSE
          MCPMAX = MCPAIR
```

```
        MCPMIN = MCPF
      END IF
      CALL EFF2EVAP(UADRY,PER2PN,MCPMIN,MCPMAX,EFF2)
      Q2 = EFF2*MCPMIN*(1.0-PER2PN)*(TINAIR-TSAT)
      QEVAPDRY = Q1 + Q2
    END IF
C --CHECK TO SEE IF AIR SIDE SURFACE IS DRY
C --ASSUMING TWALL IS APPROX THE AVERAGE OF FREON AND AIR TEMPS
    TWALL = 0.5*(TSAT + TINAIR)
    IF(TWALL .LT. TAIRDP) THEN
C     --SURFACE IS WET (TWALL IS LESS THAN TDP)
      IBEWET = 1
C     --FIRST MAKE CALCULATIONS AS IF THE FREON IS 2 PHASE THE ENTIRE
C       LENGTH.


C     --APPROXIMATE THE UAWET ASSUMING EQUAL RESISTANCES TO HEAT
C     --TRANSFER ON THE AIR AND FREON SURFACES
C       CALL FCPAIRS(CPAIRS,TSAT-1.,TSAT+1.)
C       UAWET = 1/(0.5*CPAIRS/UADRY + 0.5*CPAIR/UADRY)


     EFF1 = 1.0 - EXP(-UAWET/MDOTAIR)
C    --AGAIN ASSUMING THE WALL TEMP IN CONTACT WITH THE AIR IS APPROX.
C    --THE AVERAGE OF TSAT AND TINAIR
     PSYDAT(1) = 1.0
     PSYDAT(2) = TWALL
     PSYDAT(4) = 0.99999
     CALL PSYCH(INFO,2,2,0,PSYDAT)
     HWFS = PSYDAT(7)
     QEVAPWET = EFF1*MDOTAIR*(HINAIR - HWFS)
     HOUTAIR = HINAIR - EFF1*(HINAIR - HWFS)
     HOUT = HIN + QEVAPWET/MDOT
```

```
C      --CHECK TO SEE IF OUTLET IS TWO PHASE
       IF(HOUT .GT. HSAT) THEN
C        --NOT TWO PHASE OUTLET SO ITERATIVELY SOLVE FOR PER2P
C        --SET UP VALUES TO START THE ITERATION
         Q1 = MDOT*(HSAT - HIN)
         PER2PO = 1.0
         RESIDO = Q1 - EFF1*MDOTAIR*(HINAIR-HWFS)
         PER2PN = 0.5
C        --START SECANT ITERATION.
 20      EFF1 = 1.0 - EXP(-UAWET*PER2PN/MDOTAIR)
         RESIDN = Q1 - EFF1*PER2PN*MDOTAIR*(HINAIR-HWFS)
         IF(ABS((RESIDN)/Q1) .GT. .001) THEN
           PER2P = PER2PN - RESIDN*(PER2PN-PER2PO)/(RESIDN-RESIDO)
           PER2PO = PER2PN
           PER2PN = PER2P
           RESIDO = RESIDN
           GOTO 20
         END IF
         MCPF = MDOT*CPF2(PIN,IFREON)
C      --CALCULATING THE AIR SATURATION ENTHALPY
C      --ASSUME TOUTN = TSAT + 5
         TOUTN = TSAT + 5.0
C      --START OF A SUCCESSIVE GUESS METHOD FOR ITERATION ON CPAIRS
C      --IF A WET EVAPORATOR IS USED MUCH, THIS ITERATION METHOD SHOULD
C      --BE IMPROVED TO AT LEAST A SECANT METHOD!!!
 30      CALL FCPAIRS(CPAIRS,TSAT,TOUTN)
         MCPAIR = MDOTAIR*CPAIRS
         CALL EFF2EVAP2(UAWET,PER2PN,MCPF,MDOTAIR,CPAIRS,EFF2)
       Q2 = EFF2*MDOTAIR*(HINAIR - HWFS)
       HOUT = HSAT + Q2/MDOT
       CALL FREON(TOUT,PIN,HOUT,S,X,V,U,IFREON,23)
       IF(ABS(TOUTN-TOUT) .GT. 1.0) THEN
         TOUTN = (TOUT+TOUTN)/2
         GOTO 30
       END IF
       QEVAPWET = Q1 + Q2
```

```
        END IF
      END IF
      IF(IBEWET .EQ. 0) THEN
        QEVAP = QEVAPDRY
      ELSE IF(QEVAPDRY .GT. 0.0) THEN
        QEVAP = AMAX1(QEVAPDRY,QEVAPWET)
      ELSE
C       --IF THE HEAT TRANSFER IS NEGITIVE THEN THE EVAPORATOR IS HEATING
C       --THE AIR AND THEREFORE ACTS LIKE A DRY HEAT EXCHANGER
        QEVAP = QEVAPDRY
      END IF
      HOUT = HIN + QEVAP/MDOT
C       print*,mdot,ibewet,QEVAPWET,QEVAPDRY,HOUT
C     --NO PRESSURE DROP IS BUILT INTO THE SOLUTION
      POUT = PIN
      RETURN
      END

      SUBROUTINE FCPAIRS(CPAIRSN,THIGH,TLOW)
      REAL PSYDAT(9),INFO(10)
      PSYDAT(1) = 1.0
      PSYDAT(2) = THIGH
      PSYDAT(4) = 0.99999
      CALL PSYCH(INFO,2,2,0,PSYDAT)
      HWFSA = PSYDAT(7)
      PSYDAT(1) = 1.0
      PSYDAT(2) = TLOW
      PSYDAT(4) = 0.999999
      CALL PSYCH(INFO,2,2,0,PSYDAT)
      HWFSB = PSYDAT(7)
      CPAIRSN = (HWFSA - HWFSB)/(THIGH - TLOW)
      RETURN
      END
```

```
    SUBROUTINE EFF2EVAP(UA,PER2P,MCPMIN,MCPMAX,EFF2)
C--CALCULATES THE EFFECTIVENESS FOR A MIXED CROSSFLOW HEAT
EXCHANGER
    IMPLICIT NONE
     REAL *4 EFF2,NTU,UA,PER2P,MCPMIN,MCPMAX,C
    NTU = UA*(1-PER2P)/MCPMIN
    C = MCPMIN/MCPMAX
    IF(ABS(NTU*C) .LT. .0001) THEN
       EFF2 = 0.0
    ELSE
     EFF2 = NTU/(NTU/(1.-EXP(-NTU))+NTU*C/(1.-EXP(-NTU*C))-1)
    END IF
    RETURN
    END

    SUBROUTINE EFF2EVAP2(UAWET,PER2P,MCPF,MDOTAIR,CPAIRS,EFF2)
C--CALCULATES THE EFFECTIVENESS FOR A MIXED CROSSFLOW HEAT
EXCHANGER
C--USING BRAUN'S ENTHALPY EFFECTIVENESS MODEL
    IMPLICIT NONE
     REAL *4 EFF2,NTU,UAWET,PER2P,MCPF,MDOTAIR,MSTR,CPAIRS
    NTU = UAWET*(1.-PER2P)/MDOTAIR
     MSTR = MDOTAIR*CPAIRS/MCPF
    EFF2 = NTU/(NTU/(1.-EXP(-NTU))+NTU*MSTR/(1.-EXP(-NTU*MSTR))-1)
    RETURN
    END

    FUNCTION CPF2(P,IFREON)
C--FUNCTION TO CALCULATE THE SPECIFIC HEAT OF A SUPERHEATED LIQUID
    CALL FREON(T,P,H,S,1.0,V,U,IFREON,25)
    TRANK = T + 460.0
    tplus = TRANK*1.01 - 460.
    CALL FREON(TPLUS,PSAT,HPLUS,S,1.0,V,U,IFREON,15)
    tMINUS = TRANK*.99 - 460.
    CALL FREON(TMINUS,PSAT,HMINUS,S,1.0,V,U,IFREON,15)
```

```
CPF2 = (HMINUS - HPLUS)/(TMINUS - TPLUS)
RETURN
END
```

APPENDIX A
HPWH


```
      SUBROUTINE HPWH(VOL,TANKUA,TRNDT,HPDT,THPH,THPL,H2ODRAW,
     .  HXMDOT,ISTATUS,TTANK,TAIR,TMAINS,POWER_SUM,HPAC_SUM,Q_H2O_SUM,
     .    POW_LOS_SUM,TE1H,TE1L,TE2H,TE2L,PTNK1,PTNK2,PTNK3,PWA1,PWA2,
     .    PWA3,ICHP,ICE1,ICE2,TMIXEDHIGH,TMIXEDLOW,TMIXEDSUM,
     .    TSPLYHIGH,TSPLYLOW,TSPLYAVG,COP)

C*****************************************************************************
C   DESCRIPTION: PROGRAM MODELS A DAIRY EQUIPMENT COMPANY HEAT PUMP
C                WATER HEATER (HP-120-27), WITH 75 DEGF EVAP. AIR TEMPERATURE
C   AUTHOR:     BYRON PRITCHARD
C   MACHINE:    SEL MICROVAX
C   DATE:       SEPTEMBER 1987
C*****************************************************************************

C --DEFINITION OF VARIABLES
C   CPH2O  : SPECIFIC HEAT OF THE WATER
C   HPAC_SUM:  AIR COOLING OVER THE TIME STEP (BTU)
C   HPAC   : RATE OF AIR COOLING PRODUCED BY THE HPWH (BTU/HR)
C   HPDT   : TIME STEP FOR HEAT PUMP WATER HEATER'S EULER SOLUTION (HRS)
C   H2ODRAW: GALLONS OF HOT WATER DRAWN OVER TIME STEP (GAL)
C   HXMDOT : MASS FLOW RATE THROUGH THE HX (LBM/HR)
C   I      : COUNTER
C   ISTATUS: ON/OFF FLAG OF THE HPWH ()
C   NSTEP  : NUMBER OF STEPS IN THE EULER SOLUTION ()
C   MDOT   : AVERAGE MASS FLOW OF DRAWN WATER OVER TIME STEP (LBM/HR)
C   POWER  : RATE OF ENERGY CONSUMPTION OF THE HPWH (BTU/HR)
C   POWER_SUM: CONSUMED ENERGY OVER TIME STEP (BTU)
C   QHP_WA : RATE OF WATER HEATING PRODUCED BY THE HPWH WRAP-AROUND
C          HEAT EXCHANGER(BTU/HR)
C   QHP_TIT: RATE OF WATER HEATING PRODUCED BY THE HPWH TUBE-IN-TUBE
C          HEAT EXCHANGER(BTU/HR)
C   RHOH2O : DENSITY OF THE WATER (LBM/FT**3)
C   TAIR   : TEMPERATURE OF THE AIR THE HPWH LOSSES HEAT TO (F)
C   TANKUA : UA OF THE TANK (BTU/HR-F)
C   THPH   : TEMPERATURE AT WHICH THE HPWH SHUTS OFF (F)
```

APPENDIX A
HPWH


```
C   THX    : TEMPERATURE OF WATER GOING TO THE HX (F)
C   THPL   : TEMPERATURE AT WHICH THE HPWH TURNS ON (F)
C   TMAINS : TEMPERATURE OF WATER SUPPLIED TO THE HPWH (F)
C   TRNDT  : TIME STEP FOR MAIN PROGRAM (IE. TRNSYS TIME STEP) (HRS)
C   VOL    : VOLUME OF WATER IN THE STORAGE TANK (FT**3)

C --DECLARE VARIABLES
      IMPLICIT NONE
      REAL VOL,TRNDT,HPDT,THPH,THPL,TTANK,POWER,TTANKI,CPH2O,
     .   RHOH2O,H2ODRAW,MDOT,POWER_SUM,HPAC_SUM,HPAC,QHP_WA,TMAINS,
     .   TANKUA,TAIR,HXMDOT,THX,POW_LOS,TE1H,TE1L,TE2H,TE2L,
     .   POWERHP,POWERE1,POWERE2,QE1,QE2,Q_H2O,POW_LOS_SUM,
     .   TMIXEDHIGH,TMIXEDLOW,TMIXEDSUM1,TMIXEDSUM,Q_H2O_SUM,QHP_TIT,
     .   PTNK1,PTNK2,PTNK3,PWA1,PWA2,PWA3,T(3),TMIXED,TMIX_WA,
     .   TSPLYHIGH,TSPLYLOW,TSPLYAVG,TSPLYSUM,MSPLY,COP,t3last,
     .   time
      INTEGER NSTEP,ISTATUS,I,IHOUR,IMONTH,IDAY,ICHP,ICE1,ICE2
      COMMON IHOUR,IMONTH,IDAY

C --DEFINE CONSTANTS AND SET INITIAL VALUE OF VAIABLES
      RHOH2O = 62.4
      CPH2O = 1.00
      POW_LOS_SUM = 0.0
      POWER_SUM = 0.0
      HPAC_SUM = 0.0
      TMIXEDSUM1 = 0.0
      Q_H2O_SUM = 0.0
      MDOT = H2ODRAW*RHOH2O/(7.48*TRNDT)
      NSTEP = IFIX(TRNDT/HPDT + .1)

C --SET INITIAL VALUES OF TANK TEMPERATURES
      IF(IMONTH .EQ. 1 .AND. IDAY .EQ. 1 .AND. IHOUR .EQ. 1) THEN
        T(1) = TTANK
        T(2) = TTANK
        T(3) = TTANK
```

APPENDIX A
HPWH


```
        MSPLY = 0.0
        TSPLYHIGH = TTANK
        TSPLYLOW = TTANK
        TMIXEDHIGH = TTANK
        TMIXEDLOW = TTANK
        TSPLYSUM = 0.0
        time = 0.0
      END IF

C  --START EULER SOLUTION
      DO I=1,NSTEP
C      --DETERMINE STATUS OF THE HPWH
        IF(T(ICHP) .LT. THPL) THEN
          ISTATUS = 1
        ELSE IF(T(ICHP) .GT. THPH) THEN
          ISTATUS = 0
        END IF
        IF(ISTATUS .EQ. 1) THEN
C         --APPROXIMATE THE HEAT EXCHANGER WATER TEMPERATURE
          IF(MDOT .LT. .001) THEN
            THX = T(1)
          ELSE IF(MDOT .LT. HXMDOT) THEN
            THX = (T(1)*(HXMDOT-MDOT) + TMAINS*MDOT)/HXMDOT
          ELSE
            THX = TMAINS
          END IF
          TMIX_WA = T(1)*PWA1 + T(2)*PWA2 + T(3)*PWA3
          CALL HPWH_VCC(TMIX_WA,THX,POWERHP,QHP_WA,QHP_TIT,POW_LOS)
C         --DETERMINE POWER AND HEATING OF THE BACKUP ELEMENTS
          IF(T(ICE1) .LT. TE1L) THEN
            QE1 = 6.0*3413.0
            POWERE1 = 6.0*3413.0
          ELSE IF(T(ICE1) .GT. TE1H) THEN
            QE1 = 0.0
            POWERE1 = 0.0
          END IF
```

```
  IF(T(ICE2) .LT. TE2L) THEN
    QE2 = 6.0*3413.0
    POWERE2 = 6.0*3413.0
  ELSE IF(T(ICE2) .GT. TE2H) THEN
    QE2 = 0.0
    POWERE2 = 0.0
  END IF
  Q_H2O = QE1 + QE2 + QHP_WA + QHP_TIT
  HPAC = QHP_WA + QHP_TIT - (POWERHP - POW_LOS)
  POWER = POWERHP + POWERE1 + POWERE2
 ELSE
  HPAC = 0.0
  POWER = 0.0
  POW_LOS = 0.0
  QE1 = 0.0
  QE2 = 0.0
  Q_H2O = 0.0
  POWERE1 = 0.0
  POWERE2 = 0.0
  POWERHP = 0.0
  QHP_WA = 0.0
  QHP_TIT = 0.0
 END IF
 CALL STRATIFIED(T(1),T(2),T(3),QE1,QE2,QHP_WA,QHP_TIT,TANKUA,
.       TAIR,HPDT,RHOH2O,VOL,CPH2O,TMAINS,MDOT,PTNK1,PTNK2,
.         PTNK3,PWA1,PWA2,PWA3)

c   --check to see if load is being met
 if(t3last .lt. tell .and. t(3) .lt. t3last) then
   if( t3last .gt. 0.001) print*,' **',trndt,t(3)
 end if
 t3last = t(3)

 POWER_SUM = POWER_SUM + POWER*HPDT
 HPAC_SUM  = HPAC_SUM + HPAC*HPDT
```

```
      POW_LOS_SUM = POW_LOS_SUM + POW_LOS*HPDT
      Q_H2O_SUM = Q_H2O_SUM + Q_H2O*HPDT
      TMIXED = PTNK1*T(1) + PTNK2*T(2) + PTNK3*T(3)
      IF(TMIXED .LT. TMIXEDLOW) TMIXEDLOW = TMIXED
      IF(TMIXED .GT. TMIXEDHIGH) TMIXEDHIGH = TMIXED
      TMIXEDSUM1 = TMIXEDSUM1 + TMIXED
      IF(MDOT .GT. 0.001) THEN
        MSPLY = MSPLY + MDOT
        IF(T(3) .LT. TSPLYLOW) TSPLYLOW = T(3)
        IF(T(3) .GT. TSPLYHIGH) TSPLYHIGH = T(3)
        TSPLYSUM = TSPLYSUM + T(3)*MDOT
        TSPLYAVG = TSPLYSUM/MSPLY
      END IF
      time = time + hpdt
c      WRITE(30,*) time,T(1),T(2),T(3)
c      PRINT*,time,T(1),T(2),T(3)
    END DO
    TMIXEDSUM = TMIXEDSUM + TMIXEDSUM1/FLOAT(NSTEP)
    TTANK = TMIXED
    IF(POWER_SUM .GT. 0.00001) THEN
      COP = Q_H2O_SUM/POWER_SUM
    ELSE
      COP = 0.0
    END IF
    RETURN
    END

    SUBROUTINE HPWH_VCC(TTANK,THX,POWER,QWA,QTIT,POW_LOS)
C  --MODELS THE HPWH VAPOR COMPRESSION CYCLE AS FUNCTION OF TTANK
C     AND THX
    IMPLICIT NONE
    REAL TTANK,THX,POWER,QWA,QTIT,POW_LOS
c curve fits to 75 F ambient data
    QWA = 2881.4 + 55.0*THX + 263.8*(TTANK - THX + 0.01)**(-.5)
   .     - 0.1005*THX**2 - 37.18*TTANK + 0.04844*THX*TTANK
    QTIT = 13935 - 22.48*THX - 259.3*(TTANK - THX + 0.01)**(-.5)
```

```
 .    - 0.2582*THX**2 + 38.75*TTANK - 0.08302*THX*TTANK
   POWER = 1979.2 + 29.311*THX - 0.021418*THX**2 +
 .       0.016076*THX*TTANK
c curve fits to 85 F ambient data
c     QWA = 4100 + 48.22*THX + 291.4*(TTANK - THX + 0.01)**(-.5)
c  .     - 0.0918*THX**2 - 50.0*TTANK + 0.1315*THX*TTANK
c     QTIT = 14605 - 22.16*THX - 289.*(TTANK - THX + 0.01)**(-.5)
c  .     - 0.20447*THX**2 + 52.175*TTANK - 0.1744*THX*TTANK
c     POWER = 1855.3 + 31.0*THX - 0.021545*THX**2 +
c  .       0.017620*THX*TTANK
   POW_LOS = 2400.*(0.25)
   RETURN
   END

   SUBROUTINE STRATIFIED(T1,T2,T3,QE1,QE2,QHP_WA,QHP_TIT,TANKUA,
 .              TAIR,HPDT,RHOH2O,VOL,CPH2O,TMAINS,MDOT,
 .              PTNK1,PTNK2,PTNK3,PWA1,PWA2,PWA3)
C
C THIS PROGRAM CALCULATE THE TEMPS IN A THREE NODE STRATIFIED TANK
C
C VARIABLE DEFINITION
C    T'S ARE TEMPERATURES
C    1, 2, 3 ARE THE THREE NODE, 1 BEING AT THE BOTTOM OF THE TANK
C    OLD REFERS TO VALUES FROM PREVIOUS TIME STEP
C
C BACKUP ELEMENT 1 IS IN NODE 3 AND BACKUP ELEMENT 2 IS IN NODE 1
C
   IMPLICIT NONE
   REAL T1,T2,T3,T1NEW,T2NEW,T3NEW,QE1,QE2,QHP_WA,QHP_TIT,TANKUA,
 .   TAIR,HPDT,RHOH2O,VOL,CPH2O,TMAINS,MDOT,PTNK1,PTNK2,PTNK3,
 .   PWA1,PWA2,PWA3

C --CALCULATE NEW TEMPERATURES FOR EACH NODE
   T1NEW = T1 + (qHP_TIT + qHP_WA*PWA1 + qE2 - TANKUA*PTNK1*
 .     (T1 - TAIR) - MDOT*CPH2O*(T1 - TMAINS))*HPDT/
```

```
     .     (RHOH2O*VOL*PTNK1*CPH2O)
      T2NEW = T2 + (QHP_WA*PWA2 - TANKUA*PTNK2*(T2 - TAIR)
     .   - MDOT*CPH2O*(T2 - T1))*HPDT/(RHOH2O*VOL*PTNK2*CPH2O)
      T3NEW = T3 + (QHP_WA*PWA3 + qE1 - TANKUA*PTNK3*(T3 - TAIR)
     .   - MDOT*CPH2O*(T3 - T2))*HPDT/(RHOH2O*VOL*PTNK3*CPH2O)
      T1 = T1NEW
      T2 = T2NEW
      T3 = T3NEW
C --CHECK FOR TEMPERATURE INVERSIONS
      IF(T1 .GT. T2) THEN
C       --NODES 1 AND 2 MIXED
        T1 = (T1*PTNK1 + T2*PTNK2)/(PTNK1 + PTNK2)
        T2 = T1
      END IF
      IF(T1 .EQ. T2 .AND. T2 .GT. T3) THEN
C       --NODES 1, 2, AND 3 MIXED
        T2 = (T2*(PTNK1 + PTNK2) + T3*PTNK3)/(PTNK1 + PTNK2 + PTNK3)
        T1 = T2
        T3 = T2
      END IF
      IF(T1 .NE. T2 .AND. T2 .GT. T3) THEN
C       --NODES 2 AND 3 MIXED
        T2 = (T2*PTNK2 + T3*PTNK3)/(PTNK2 + PTNK3)
        T3 = T2
        IF(T1 .GT. T2) THEN
C         --NODES 1, 2 and 3 MIXED
          T2 = (T2*(PTNK2 + PTNK3) + T1*PTNK1)/(PTNK1 + PTNK2 + PTNK3)
          T3 = T2
          T1 = T2
        END IF
      END IF
      RETURN
      END
```

Appendix A
EMOR


```
C  EMOR (ENERGY MODELING OF RESTAURANTS)
C     CALCULATES THE BUILDING LOAD AND DETERMINES THE HPWH
C     SAVINGS COMPARED TO GAS AND ELECTRIC WATER HEATERS

C  DEFINING VARIABLES
C     PEOPLE  : NUMBER OF PEOPLE IN THE BUILDING DURING THIS HOUR
C     PEO_GAIN: ENERGY GAIN TO RESTAURANT DUE TO PEOPLE (BTU/HR)
C     OVE_GAIN: ENERGY GAIN TO RESTAURANT DUE TO OVENS (BTU/HR)
C     OVENEFF : EFFICIENCY OF OVEN BURNERS' CONVERSION OF FUEL TO HEAT
C     OVENGAS : THERMS OF GAS USED IN THAT HOUR (10**5 BTU)
C     PER_INTL: PERCENT OF BASE ELECTRICITY CONTRIBUTING TO
C     INTERNAL GAINS
C     ELE_GAIN: ENERGY GAIN TO RESTAURANT DUE TO ELEC GAINS (BTU/HR)
C     ELEC    : ELECTRIC USE IN THAT HOUR (KWHR)
C     UA_GAIN : ENERGY GAIN TO RESTAURANT DUE TO SKIN EXCHANGE (BUT/HR)
C     UA      : OVERALL UA OF BUILDING (BTU/HR-F)
C     TAMB    : AMBIENT TEMPERATURE (F)
C     TROOM   : TEMPERATURE INSIDE THE RESTAURANT (F)
C     CFM_GAIN: ENERGRY GAIN TO RESTAURANT DUE TO AIR EXCHANGES (BTU/HR)
C     CFM     : AIR EXCHANGE RATE (FT**3/MIN)
C     CPAIR   : SPECIFIC HEAT OF AIR (BTU/LBM-F)
C     RHOAIR  : DENSITY OF AIR (LBM/FT**3)
C     NET_ENGY: NET ENERGY GAIN IN THE BUILDING FOR THAT HOUR (BTU)
C     TSTEP   : TIME STEP FOR SIMULATION (HR)
C     VOL     : VOLUME OF THE WATER TANK (FT**3)
C     TANKUA  : UA OF THE TANK (BTU/HR-F)
C     HPDT    : TIME STEP FOR HEAT PUMP WATER HEATER'S EULER SOLUTION (HRS)
C     THIGH   : TEMPERATURE AT WHICH THE HPWH SHUTS OFF (F)
C     TLOW    : TEMPERATURE AT WHICH THE HPWH TURNS ON (F)
C     H2ODRAW : GALLONS OF HOT WATER DRAWN OVER TIME STEP (GAL)
C     HXMDOT  : MASS FLOW RATE THROUGH THE HX (LBM/HR)
C     ISTATUS : ON/OFF FLAG OF THE HPWH ()
C     TTANKI  : TANK TEMPERATURE (F)
C     TMAINS  : TEMPERATURE OF WATER SUPPLIED TO THE HPWH (F)
C     POWER   : CONSUMED ENERGY OVER TIME STEP (BTU)
```

Appendix A
EMOR


```
C      HPAC   : AIR COOLING OVER THE TIME STEP (BTU)

C   DECLARE VARIABLES
    IMPLICIT NONE
    REAL PEOPLE(25),OVENGAS(25),ELEC(25),H2ODRAW(25),PEO_GAIN,
  .   OVE_GAIN,OVENEFF,
  .   PER_INTL,UA,TROOM,VOL,TANKUA,TSTEP,HPDT,THPH,THPL,
  .   HXMDOT,TTANK;TMAINS,FUR_EFF,RM_VOL,CPAIR,RHOAIR,TAMB,
  .   AC_SAV_Y,H2O_SAV_ELE_Y,H2O_SAV_GAS_Y,TOT_SAV_ELE_Y,
  .   TOT_SAV_GAS_Y,ELE_GAIN,UA_GAIN,CFM_GAIN,
  .   AC_SAV_M,H2O_SAV_ELE_M,H2O_SAV_GAS_M,TOT_SAV_ELE_M,
  .   TOT_SAV_GAS_M,CFM,NET_ENGY,POWER_SUM,HPQCOND_SUM,
  .   AC_SAV_D,H2O_SAV_ELE_D,H2O_SAV_GAS_D,TOT_SAV_ELE_D,
  .   TOT_SAV_GAS_D,POWER,HPAC,GOOD_AC,BAD_AC,
  .   AC_SAV_H,H2O_SAV_ELE_H,H2O_SAV_GAS_H,TOT_SAV_ELE_H,
  .   TOT_SAV_GAS_H,ACCOP,COST_THRM,COST_KWH,HPHW,POW_LOS,
  .   GAS_EFF,DUMMY,H2O_COST_HP_H,H2O_COST_HP_D,
  .   H2O_COST_HP_M,H2O_COST_HP_Y,HPCOP,SUM_ACCOP,
  .   SUM_NETHPAC, AVG_ACCOP,MAX_SAV_ELE_H,MAX_SAV_GAS_H,
  .   MIN_SAV_ELE_H,MIN_SAV_GAS_H,MAX_SAV_ELE_D,
  .   MAX_SAV_GAS_D, MIN_SAV_ELE_D,MIN_SAV_GAS_D,
  .   MAX_SAV_ELE_M,MAX_SAV_GAS_M,MIN_SAV_ELE_M,
  .   MIN_SAV_GAS_M,MAX_SAV_ELE_Y,MAX_SAV_GAS_Y,
  .   MIN_SAV_ELE_Y,MIN_SAV_GAS_Y,SUM_GOOD_AC_D,
  .   SUM_GOOD_AC_M, SUM_GOOD_AC_Y, SUM_BAD_AC_D,
  .   TE1H,TE1L,TE2H,TE2L,D,TMIXEDSUM,TMIXEDLOW,TMIXEDHIGH,
  .   NETHPAC,HPQCOND,PTNK1,PTNK2,PTNK3,PWA1,PWA2,PWA3,
  .   TSPLYAVG,TSPLYHIGH,TSPLYLOW,COP,SUM_BAD_AC_M,
  .   SUM_BAD_AC_Y
    INTEGER ISTATUS,IMONTH,IDAY,NDAYS(12),IHOUR,I,ICHP,ICE1,ICE2
    CHARACTER *20 FILE
    COMMON IHOUR,IMONTH,IDAY

    DATA NDAYS/31,28,31,30,31,30,31,31,30,31,30,31/
```

Appendix A
EMOR

```
C --OPEN INPUT AND OUTPUT FILES
    OPEN(UNIT=12,FILE='PARAM.DAT',STATUS='OLD')
    OPEN(UNIT=13,FILE='OUTPUT.DAT',STATUS='NEW')
    OPEN(UNIT=14,FILE='[trnsys.weather]madison.all'
   .   ,STATUS='OLD',read only)

C --WRITE HEADER IN OUTPUT FILE
    WRITE(13,11)
    WRITE(13,12)
    WRITE(13,13)

C --READ IN INPUT PARAMETERS
    READ(12,*)
    READ(12,*)
    READ(12,*)
    READ(12,*)
    READ(12,*) OVENEFF,PER_INTL,UA,TROOM,VOL,TANKUA,
   .        HPDT,THPH,THPL,HXMDOT,TTANK,TMAINS,FUR_EFF,
   .        GAS_EFF,COST_KWH,COST_THRM,CFM,
   .        TE1H,TE1L,TE2H,TE2L,PTNK1,PTNK2,PTNK3,PWA1,PWA2,PWA3,
   .        ICHP,ICE1,ICE2
    READ(12,'(A20)') FILE
    READ(*,'(A20)') FILE
    read(*,*) vol,gas_eff,cost_kwh,cost_thrm
    OPEN(UNIT=11,FILE=FILE,STATUS='OLD')

C --SET VALUES OF CONSTANTS
    CPAIR = 0.24
    RHOAIR = 0.075
    TSTEP = 1.0

C --SET THE INITIAL STATUS OF THE HEAT PUMP TO OFF
    ISTATUS = 0

C --SET INITIAL VALUE OF YEAR SUMS EQUAL TO ZERO
    TMIXEDSUM = 0.0
```

```
      TMIXEDLOW = 300
      TSPLYLOW = 300
      SUM_ACCOP    = 0.0
      SUM_NETHPAC  = 0.0
      AC_SAV_Y     = 0.0
      H2O_SAV_ELE_Y = 0.0
      H2O_SAV_GAS_Y = 0.0
      TOT_SAV_ELE_Y = 0.0
      TOT_SAV_GAS_Y = 0.0
      H2O_COST_HP_Y = 0.0
      MAX_SAV_ELE_Y = 0.0
      MIN_SAV_ELE_Y = 0.0
      MAX_SAV_GAS_Y = 0.0
      MIN_SAV_GAS_Y = 0.0
      SUM_GOOD_AC_Y = 0.0
      POWER_SUM    = 0.0
      HPQCOND_SUM  = 0.0
      SUM_BAD_AC_Y  = 0.0
c     idday = 0

C   --READ IN AND STORE HOURLY DATA OF PEOPLE, OVEN GAS, ELEC AND
C      H2ODRAW
      DO I=1,24
       READ(11,*) DUMMY,PEOPLE(I),OVENGAS(I),ELEC(I),H2ODRAW(I)
      END DO

      DO IMONTH=1,12
C    --PRINT THE MONTH TO AMMUSE THE USER
c       PRINT*,IMONTH

C    --SET INITIAL VALUE OF MONTH SUMS EQUAL TO ZERO
        AC_SAV_M     = 0.0
        H2O_SAV_ELE_M = 0.0
        H2O_SAV_GAS_M = 0.0
        TOT_SAV_ELE_M = 0.0
```

```
      TOT_SAV_GAS_M = 0.0
      H2O_COST_HP_M = 0.0
      MAX_SAV_ELE_M = 0.0
      MIN_SAV_ELE_M = 0.0
      MAX_SAV_GAS_M = 0.0
      MIN_SAV_GAS_M = 0.0
      SUM_GOOD_AC_M = 0.0
      SUM_BAD_AC_M  = 0.0

      DO IDAY=1,NDAYS(IMONTH)
C        --SET INITIAL VALUE OF DAY SUMS EQUAL TO ZERO
         AC_SAV_D      = 0.0
         H2O_SAV_ELE_D = 0.0
         H2O_SAV_GAS_D = 0.0
         TOT_SAV_ELE_D = 0.0
         TOT_SAV_GAS_D = 0.0
         H2O_COST_HP_D = 0.0
         MAX_SAV_ELE_D = 0.0
         MIN_SAV_ELE_D = 0.0
         MAX_SAV_GAS_D = 0.0
         MIN_SAV_GAS_D = 0.0
         SUM_GOOD_AC_D = 0.0
         SUM_BAD_AC_D  = 0.0
c     sumn = 0.0
c     idday = idday + 1
         DO IHOUR=1,24
           READ(14,*) DUMMY,dummy,dummy,dummy,TAMB
            tamb = tamb*1.8/10. + 32.0
           PEO_GAIN = PEOPLE(IHOUR)*225.0
           OVE_GAIN = (OVENGAS(IHOUR)*100000.)*OVENEFF
           ELE_GAIN = ELEC(IHOUR)*PER_INTL*3413.0
           UA_GAIN  = UA*(TAMB - TROOM)
           IF(OVENGAS(IHOUR) .GT. 0.0) THEN
             CFM_GAIN = CFM*60.0*RHOAIR*CPAIR*(TAMB - TROOM)
            ELSE
             CFM_GAIN = 0.0
```

```
        END IF

C       --ENERGY BALANCE ON RESTAURANT
        NET_ENGY = (PEO_GAIN+OVE_GAIN+ELE_GAIN+UA_GAIN+CFM_GAIN)*
     .            TSTEP

C PRINTS OUT DATA OF AVERAGE HOURLY BUILDING LOAD DURING OPEN HOURS FOR
C EACH DAY OF THE YEAR
c     if(ihour .gt. 7) then
c       sumn = sumn + net_engy
c     end if
c     if(ihour .eq. 24) then
c       avgsumn = sumn/18.0
c      write(100,*) idday,avgsumn
c     end if

        CALL HPWH(VOL,TANKUA,TSTEP,HPDT,THPH,THPL,
     .         H2ODRAW(IHOUR),HXMDOT,ISTATUS,TTANK,TROOM,
     .          TMAINS,POWER,HPAC,HPQCOND,POW_LOS,
     .          TE1H,TE1L,TE2H,TE2L,PTNK1,PTNK2,PTNK3,PWA1,
     .          PWA2,PWA3,ICHP,ICE1,ICE2,
     .         TMIXEDHIGH,TMIXEDLOW,TMIXEDSUM,TSPLYHIGH,
     .          TSPLYLOW,TSPLYAVG,COP)

C       --LET HPAC BE THE NET COOLING PRODUCED BY THE HPWH
        NETHPAC = HPAC - POW_LOS

C       --DETERMINE IF HPAC CONTRIBUTES TO THE HEATING OR A/C LOAD
        IF(NETHPAC .GT. 0.00001) THEN
          IF(NET_ENGY .GT. NETHPAC) THEN
            GOOD_AC = NETHPAC
            BAD_AC  = 0.0
          ELSE IF(NET_ENGY .GT. 0.0) THEN
            GOOD_AC = NET_ENGY
            BAD_AC  = NETHPAC - NET_ENGY
```

```
      ELSE
        GOOD_AC = 0.0
        BAD_AC  = NETHPAC
      END IF
    ELSE
      GOOD_AC = 0.0
      BAD_AC = 0.0
    END IF
    CALL AC_COP(TAMB,TROOM,ACCOP)
    IF(NETHPAC .GT. 0.0001 .AND. NET_ENGY .GT. 0.0001) THEN
      SUM_ACCOP = ACCOP*NETHPAC + SUM_ACCOP
      SUM_NETHPAC = SUM_NETHPAC + NETHPAC
    END IF

C        --CALCULATE THE HOURLY RESULTS
  IF(GOOD_AC .GT. 0.001) THEN
       AC_SAV_H = ((GOOD_AC/ACCOP)/3413.)*COST_KWH -
     .            ((BAD_AC/FUR_EFF)/100000.0)*COST_THRM
  ELSE
       AC_SAV_H = -((BAD_AC/FUR_EFF)/100000.0)*COST_THRM
  END IF
    H2O_SAV_ELE_H = (HPQCOND/3413.)*COST_KWH
     .            - (POWER/3413.)*COST_KWH
    H2O_SAV_GAS_H = (HPQCOND)/(GAS_EFF*100000.0)*COST_THRM
     .            - (POWER/3413.)*COST_KWH
    TOT_SAV_ELE_H = H2O_SAV_ELE_H + AC_SAV_H
    TOT_SAV_GAS_H = H2O_SAV_GAS_H + AC_SAV_H
    MAX_SAV_ELE_H = H2O_SAV_ELE_H +
     .            (NETHPAC/3413./ACCOP)*COST_KWH
    MIN_SAV_ELE_H = H2O_SAV_ELE_H-(NETHPAC/100000./FUR_EFF)*
     .            COST_THRM
    MAX_SAV_GAS_H = H2O_SAV_GAS_H +
     .            (NETHPAC/3413./ACCOP)*COST_KWH
    MIN_SAV_GAS_H = H2O_SAV_GAS_H-(NETHPAC/100000./FUR_EFF)*
     .            COST_THRM
    H2O_COST_HP_H = (POWER/3413.)*COST_KWH
```

```
  POWER_SUM     = POWER_SUM + POWER
  HPQCOND_SUM   = HPQCOND_SUM + HPQCOND

C    --SUM THE HOURLY SAVINGS TO DETERMINE THE DAYLY SAVINGS
  AC_SAV_D      = AC_SAV_H      + AC_SAV_D
  H2O_SAV_ELE_D = H2O_SAV_ELE_H + H2O_SAV_ELE_D
  H2O_SAV_GAS_D = H2O_SAV_GAS_H + H2O_SAV_GAS_D
  TOT_SAV_ELE_D = TOT_SAV_ELE_H + TOT_SAV_ELE_D
  TOT_SAV_GAS_D = TOT_SAV_GAS_H + TOT_SAV_GAS_D
  H2O_COST_HP_D = H2O_COST_HP_H + H2O_COST_HP_D
  MAX_SAV_ELE_D = MAX_SAV_ELE_H + MAX_SAV_ELE_D
  MIN_SAV_ELE_D = MIN_SAV_ELE_H + MIN_SAV_ELE_D
  MAX_SAV_GAS_D = MAX_SAV_GAS_H + MAX_SAV_GAS_D
  MIN_SAV_GAS_D = MIN_SAV_GAS_H + MIN_SAV_GAS_D
  SUM_GOOD_AC_D = SUM_GOOD_AC_d + GOOD_AC
  SUM_BAD_AC_D  = SUM_BAD_AC_d  + BAD_AC
 END DO

C    --SUM THE DAILY SAVINGS TO DETERMINE THE MONTHLY SAVINGS
  AC_SAV_M      = AC_SAV_M      + AC_SAV_D
  H2O_SAV_ELE_M = H2O_SAV_ELE_M + H2O_SAV_ELE_D
  H2O_SAV_GAS_M = H2O_SAV_GAS_M + H2O_SAV_GAS_D
  TOT_SAV_ELE_M = TOT_SAV_ELE_M + TOT_SAV_ELE_D
  TOT_SAV_GAS_M = TOT_SAV_GAS_M + TOT_SAV_GAS_D
  H2O_COST_HP_M = H2O_COST_HP_M + H2O_COST_HP_D
  MAX_SAV_ELE_M = MAX_SAV_ELE_M + MAX_SAV_ELE_D
  MIN_SAV_ELE_M = MIN_SAV_ELE_M + MIN_SAV_ELE_D
  MAX_SAV_GAS_M = MAX_SAV_GAS_M + MAX_SAV_GAS_D
  MIN_SAV_GAS_M = MIN_SAV_GAS_M + MIN_SAV_GAS_D
  SUM_GOOD_AC_M = SUM_GOOD_AC_M + SUM_GOOD_AC_D
  SUM_BAD_AC_M  = SUM_BAD_AC_M  + SUM_BAD_AC_D
 END DO

C    --SUM THE MONTHLY SAVINGS TO DETERMINE THE YEARLY SAVINGS
  AC_SAV_Y      = AC_SAV_Y      + AC_SAV_M
```

```
      H2O_SAV_ELE_Y = H2O_SAV_ELE_Y + H2O_SAV_ELE_M
      H2O_SAV_GAS_Y = H2O_SAV_GAS_Y + H2O_SAV_GAS_M
      TOT_SAV_ELE_Y = TOT_SAV_ELE_Y + TOT_SAV_ELE_M
      TOT_SAV_GAS_Y = TOT_SAV_GAS_Y + TOT_SAV_GAS_M
      H2O_COST_HP_Y = H2O_COST_HP_Y + H2O_COST_HP_M
      MAX_SAV_ELE_Y = MAX_SAV_ELE_M + MAX_SAV_ELE_Y
      MIN_SAV_ELE_Y = MIN_SAV_ELE_M + MIN_SAV_ELE_Y
      MAX_SAV_GAS_Y = MAX_SAV_GAS_M + MAX_SAV_GAS_Y
      MIN_SAV_GAS_Y = MIN_SAV_GAS_M + MIN_SAV_GAS_Y
      SUM_GOOD_AC_Y = SUM_GOOD_AC_M + SUM_GOOD_AC_Y
      SUM_BAD_AC_Y  = SUM_BAD_AC_Y  + SUM_BAD_AC_M

C     --PRINT THE MONTHLY RESULTS
      D = NDAYS(IMONTH)
      WRITE(13,10) AC_SAV_M/D,H2O_SAV_ELE_M/D,H2O_SAV_GAS_M/D,
     .         TOT_SAV_ELE_M/D,TOT_SAV_GAS_M/D,H2O_COST_HP_M/D,
     .         MAX_SAV_ELE_M/D,MIN_SAV_ELE_M/D,
     .         MAX_SAV_GAS_M/D,MIN_SAV_GAS_M/D,
     .         SUM_GOOD_AC_M/D,SUM_BAD_AC_M/D
     END DO
C   --CALCULATE THE AVERAGE COP OF THE HPWH AND A/C
     HPCOP = HPQCOND_SUM/POWER_SUM
     AVG_ACCOP = SUM_ACCOP/SUM_NETHPAC

C   --PRINT THE YEARLY RESULTS
     WRITE(13,13)
     WRITE(13,10) AC_SAV_Y,H2O_SAV_ELE_Y,H2O_SAV_GAS_Y,
     .        TOT_SAV_ELE_Y,TOT_SAV_GAS_Y,H2O_COST_HP_Y,
     .        MAX_SAV_ELE_Y,MIN_SAV_ELE_Y,
     .        MAX_SAV_GAS_Y,MIN_SAV_GAS_Y,SUM_GOOD_AC_Y,
     .        SUM_BAD_AC_Y
     open(unit=17,file='sim.out',status='old',access='append')
     WRITE(17,17) AC_SAV_Y,TOT_SAV_ELE_Y,TOT_SAV_GAS_Y,
     .        H2O_COST_HP_Y,hpcop,SUM_GOOD_AC_Y,SUM_BAD_AC_Y
  17 FORMAT(5(F8.2,2x),2(F11.0,2X))
```

```
  close(17)
  WRITE(13,14) HPCOP,AVG_ACCOP,TMIXEDSUM/8760.,TMIXEDHIGH,
 .         TMIXEDLOW,TSPLYAVG,TSPLYHIGH,TSPLYLOW

C  --FORMAT STATEMENTS
 10 FORMAT(10(F8.2,2x),2(F10.0,2X))
 11 FORMAT(6x,'AC   H2O_ELE   H2O_GAS   TOT_ELE   TOT_GAS   H2O ELE
 .  MAX ELE   MIN ELE   MAX GAS   MIN GAS      HP AC      HP Q')
 12 FORMAT(5X,'SAV   SAVINGS   SAVINGS   SAVINGS   SAVINGS   HP COST
 .  SAVINGS   SAVINGS   SAVINGS   SAVINGS      BTUS      BTUS')
 13 FORMAT(' ----------------------------------------------------
 .---------------------------------------------')
 14 FORMAT(//,' AVERAGE HEAT PUMP COP =',F5.2,
 .       /,' AVERAGE A/C COP       =',F5.2,
 .       /,' AVERAGE MIXED TANK TEMP    =',F5.1,
 .       /,' HIGH MIXED TANK TEMP       =',F5.1,
 .       /,' LOW MIXED TANK TEMP        =',F5.1,
 .       /,' AVERAGE SUPPLY TANK TEMP   =',F5.1,
 .       /,' HIGH SUPPLY TANK TEMP      =',F5.1,
 .       /,' LOW SUPPLY TANK TEMP       =',F5.1)

C  --CLOSE THE INPUT AND OUTPUT FILES
   CLOSE(11)
   CLOSE(12)
   CLOSE(13)
   CLOSE(14)
   END


   SUBROUTINE AC_COP(TAMB,TROOM,COP)
   DELT = 35.0
   IF(TAMB .GT. 15) THEN
     COP = .40*(TROOM + DELT + 460.)/(TAMB-TROOM+2.*DELT)
   ELSE
     COP = .40*(TROOM + DELT + 460.)/(15.0-TROOM+2.*DELT)
   END IF
```

```
IF(COP .LT. 0) THEN
  WRITE(6,*) ' AC MODEL GAVE A COP OF LESS THAN 0'
  STOP
END IF
RETURN
END
```

Appendix B

Data of usage of electricity, gas, and water for a typical fastfood restaurant. Data was supplied by WP&L.

Date (Month/day)	Electricity (KWH)	Gas (Therm)	Water (100 ft^3)
11/19	25320	1270	46
10/21	27360	831	44
9/22	31920	730	54
8/21	33360	636	68
7/22	36960	715	72
6/20	28080	639	62
5/21	28920	689	48
4/22	28080	1080	55
3/21	24480	1598	36
2/20	24840	2067	36
1/21	28320	2414	41
12/18	23040	2135	36
11/19	25560	1050	45
10/21	26760	954	54
9/20	29520	672	54
8/21	31680	677	63
7/23	36480	716	92
6/21	30480	667	70
5/22	28080	705	46
4/23	27360	1152	56
3/22	23760	1286	51
2/21	23880	2094	56
1/22	27120	2290	61
12/19	24840	1493	41

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Nomenclature

Note: Variables which are used only once are defined locally

Symbol	Definition
A	area
A_i	inside surface area
A_o	outside surface area
C	ratio of capacitance rates between two fluid streams
C_{KWH}	electricity cost per KWH
COP	coefficient of performance
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
C_s	saturation specific heat
C_{Therm}	gas cost per Therm
D	diameter of capillary tube
E	input energy to the HPWH or backup elements
Exp	additional space heating expense

Symbol	Definition
h	enthaply
h_c	heat transfer coefficient
h*	enthalpy at x*
h*	enthalpy of saturated air in the evaporator model
k	isentropic index
L	length or position in a component
LMTD	log mean temperature difference
$\dot{m}$	mass flow rate
m	percent clearance in the piston cylinder assembly of the compressor
m_i	mass in the i^{th} node of the stratified tank model
m*	as defined on page 46, equation 2.5.13
NTU	number of transfer units
P	pressure
P*	pressure at x*
PDR	piston displacement rate of the compressor
Q, q	heat transfer
R	universal gas constant

Symbol	Definition
Sav	savings of the HPWH
T	temperature
t	time
U	conductance
V*	velocity at x*
W	work
x	quality of refrigerant in the capillary tube model
x*	unknown length or height where refrigerant property values are to be found

Greek

ε	effectiveness
f	friction factor in capillary tube
η_{poly}	polytropic efficiency in the compression process
η_v	volumetric efficiency of the compressor
ρ	density of refrigerant
μ	viscosity of the refrigerant in the capillary tube model
ν	specific volume

Symbol	Definition

Subscripts

air	air
A/C	air conditioner
f	saturated fluid
g	saturated gas
HP	heat pump
HPWH	heat pump water heater
H_2O	water
in	inlet to a component
losses	heat losses form the compressor motor
motor	compressor motor
out	outlet to a component
r	refrigerant
s	outlet condition if the process took place isentropically to the same pressure
TT	tube-in-tube condenser
sat	saturated condition of the refrigerant
WA	wrap-around condenser
1,2,3	refering to regions 1, 2, or 3 of the condenser and evaporator models

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