# SIMULATION OF
# LARGE SCALE DOMESTIC
# WATER HEATING SYSTEMS

by

ANNETTE GERTH

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# ABSTRACT

Domestic hot water (DHW) systems account for almost 6% of the energy consumption in the United States. Today, most commonly, electric and gas water heaters are used. Costwise, solar thermal and heat pump water heating systems can usually barely compete with these conventional systems. An alternative, which has rarely been explored, is a combined use of solar thermal and heat pump system. Energywise, two free sources can be used. Synergetic effects between solar and heat pump system can be utilized: The performance of collectors is best at low temperatures, and the performance of heat pumps is best at high evaporator temperatures. Costwise, the initial investment is twofold. High initial costs are most likely acceptable with large-scale commercial or industrial applications.

This study investigates the thermal and economic performance of large scale domestic water heating systems using computer modeling techniques. The hot water load of a hospital models the large scale water usage. The study compares a conventional electric domestic water heating system to a solar domestic hot water system (SDHWS), a heat pump water heating system (HPWHS), and a combined parallel solar heat pump water heating system (PSHPS).

The thermal system performance is modeled with TRNSYS, a transient system simulation program, for the location Madison,WI. System parameters are varied. Heat pump water heater performance data are generated with EES (Engineering Equation Solver) and integrated in the TRNSYS simulation model. Results of the thermal simulation are used

to investigate the economic performance.

The TRNSYS simulations give the following thermal performances: A SDHWS at 3,000 $m^2$ collector area yields a 72.1% free fraction. A HPWHS with the capability to meet the entire load reaches a 65.9% free fraction. A PSHPS combining these system parameters, yields a free fraction as high as 84.4%. Smaller collector areas or heat pump capacities reach lower free fractions.

The economic performance depends to a large extend on the economic scenario and the related equipment costs. Striving for maximum LCS, the PSHPS is only competitive with a low related equipment cost, and assuming an economic scenario which promotes renewable energy application. The SDHWS is recommendable at the best and medium investigated economic scenarios and with low area-dependent costs, and high heat pump costs. The HPWHS yields maximum LCS mostly when the area-dependent costs are high, or the economic scenario does not promote renewable energy application.

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# NOMENCLATURE

## Acronyms

| | |
|---|---|
| COP | Coefficient of Performance |
| DHW | Domestic Hot Water |
| EES | Engineering Equation Solver |
| HPWHS | Heat Pump Water Heating System |
| LCC | Life Cycle Cost |
| LCS | Life Cycle Saving |
| NTU | Number of Transfer Units |
| PSHPS | Parallel Solar Heat Pump Water Heating System |
| SRCC | Solar Rating And Certification Corporation |
| SDHWS | Solar Domestic Hot Water System |
| TMY | Typical Meteorological Year |
| TRNSYS | Transient Systems Simulation Program |

## Roman symbols

| | |
|---|---|
| A | area |
| C | capacitance rate |
| c | cost |
| Cap | capacity |
| $c_p$ | specific heat |
| f | free fraction |
| $f_{cap}$ | capacity fraction |
| $f_{full}$ | free fraction of full capacity system |
| $F_R$ | collector heat removal factor |
| $f_{rel}$ | related free fraction |
| $G_T$ | irridiance on tilted collector surface |
| h | enthalpy |

| | |
|---|---|
| $\dot{m}$ | mass flow rate |
| m | mass |
| MBH | heat pump capacity [Btu/hr] |
| N | number of nodes |
| P | power input |
| p | pressure |
| $P_1$ | ratio of life cycle electric cost savings to the first-year electric cost savings |
| $P_2$ | ratio of life cycle expenditures incurred because of the additional capital investment to the initial investment |
| $\dot{Q}$ | heat flow rate |
| R | thermal resistance |
| T | temperature |
| t | time |
| T' | intermediate temperature |
| $\bar{T}$ | average temperature |
| U | internal energy |
| $U_L$ | collector overall heat loss coefficient |
| V | volume |
| v | ratio tank volume to collector area |
| $\dot{V}$ | volumetric flow rate |
| $\dot{W}$ | work input rate |
| x | sum |

**Greek symbols**

| | |
|---|---|
| $\Delta$ | difference |
| $\delta$ | deviation |
| $\varepsilon$ | effectiveness |
| $\eta$ | efficiency |
| $\rho$ | density |
| $(\tau\alpha)$ | transmittance absorptance product |

### Subscripts

| | |
|---|---|
| a | ambient |
| aux | auxiliary |
| backup | backup tank |
| c | collector |
| comp | compressor |
| cond | condensation |
| conv | conventional |
| db | dry bulb |
| del | delivered |
| desup | desuperheat |
| el | electric |
| env | environment |
| evap | evaporation |
| hot | hot water tank |
| i | inlet |
| i,s | constant entropy at point i |
| iso | isentropic |
| max | maximum |
| mech | mechanical |
| min | minimum |
| o | outlet |
| para | parasitic |
| r | refrigerant |
| s | source |
| sim | simulation data |
| solar | solar tank |
| T | tank |
| table | table data |
| u | useful |
| w | water |
| wb | wet bulb |

# CHAPTER ONE

# INTRODUCTION

This chapter, in a first part, gives the reasons why this study was done. Then, the goals are summarized. The following part introduces the way this study was performed. The last part gives some basic background on heat pumps.

## 1.1 Motivation

Domestic hot water (DHW) systems account for almost 6% of the energy consumption in the United States. Facing a decrease in fossil fuel resources and an increase in pollution, the ecological benefits of renewable and emission-free energy usage is well known. Mostly driven by federal and state tax incentives during the 1970s and early 1980s, solar domestic and heat pump water heating systems were installed. After the support had been canceled in the mid 1980s, the rate of new installed systems dropped immediately. Due to high initial equipment costs and still low conventional energy prices, these alternative systems can barely compete with conventional domestic hot water systems. The best economic performance is in areas where natural gas is unavailable, especially when electric water heaters are replaced.

An alternative, which has rarely been explored, is a combined use of solar energy and heat pump system. Energywise, two free sources can be used, from solar insolation and from the environment or indoor waste heat. Coupling of solar and heat pump system has synergetic effects: The performance of collectors is best at low temperatures, and the performance of heat pumps is best at high evaporator temperatures. The use of low temperature solar heated water as heat pump source increases the heat pump COP, and thus, the overall system COP. Costwise, the initial investment is twofold, for collector and heat pump equipment, with a high potential of energy costs savings.

Earlier studies of the late 1970s and early 1980s on solar assisted heat pump systems were focused on residential size forced air space heating systems (see section 1.4.3). The systems could not compete with conventional heating systems. Doing an investigation on domestic water heating nowadays is different in manifold ways: In contrast to building heating loads, the load varies non-seasonally only during the course of the day, with a high volume draw during the day and a low draw during the night. Heat pump water heaters perform differently than air conditioning heat pumps, because of the different properties of the heat transfer media. Last, but not least, the technologic development of the recent years, and changing economic conditions (energy and equipment costs) have to be taken into consideration.

It makes sense to specifically investigate large-scale commercial or industrial applications, where high initial costs are most likely acceptable. This study analyzes the hot water system of a hospital. A hospital with 220 beds uses as many as 24,000 gallons of hot water per day (see chapter 2.1). The consumption leads to annual energy costs of $ 40,000 or $ 140,000 using natural gas or electric costs, respectively. These numbers demonstrate that there is definitely a high energy and cost savings potential. Large scale equipment might perform differently than a residential sized one. Also, through economy of scale, relative equipment costs might be lower than in residential applications. A hospital uses large scale mechanical equipment for reason of air purification, conditioning, heating, sewage treatment etc. Waste heat from these processes might be easily recovered by integrating them into a heat pump system.

## 1.2 Objective

The objective of this study is to investigate the thermal and economic performance of large scale domestic water heating systems using computer modeling techniques. The hot water load of a hospital will model the large scale hot water usage. The study will compare a conventional electric domestic water heating system to a solar domestic hot water system (SDHWS), a heat pump water heating system (HPWHS), and a combined parallel solar heat pump water heating system (PSHPS).

The thermal system performance will be modeled with TRNSYS (Klein et al. (1996)), a transient system simulation program, for the location Madison,WI. System parameters in terms of solar collector area, storage size, and heat pump capacity will be varied. A crucial point of this study will be to generate usable heat pump water heater performance data for integrating in the TRNSYS simulation model.

Results of the thermal simulation will be used to investigate the economic performance in terms of allowable equipment costs and life cycle savings (LCS). Finally, recommendations will be given for system selection.

## 1.3 Methodology

The study investigates the three system types SDHWS, HPWHS, and PSHPS. In a first step, the thermal performance of the systems was predicted using the transient system simulation program TRNSYS. System parameters were varied. In a second step, the economic performance of the systems was estimated using the Engineering Equation Solver EES (Klein et al., 1995), and the results of the simulations were compared.

First, the SDHWS, then the HPWHS, and last the PSHPS were investigated. The SDHWS, in a first step, was simulated with the parameter settings of the VA hospital (see subchapter 1.4.1). The results were compared to the actual performance data of the hospital system. Furthermore, the results were compared to f-chart (Klein et al. (1994)) estimations. Before setting up the actual TRNSYS model of the HPWHS, the heat pump was modeled using EES. The heat pump model provides a wide range of performance data which can be used in connection with other investigations.

TRNSYS (Klein et al. (1996)) is a widely used, modular thermal process simulation program. The program allows to simulate time-dependent, dynamic processes. The system components, called types, are modeled in FORTRAN code and linked together to form the system. A TRNSYS deck describes how the different types shall interact with each other. The new TRNSYS version 15.0 for Windows was used.

F-chart (Klein et al. (1994)) is a solar energy system simulation program. The program uses the f-chart method, where numerical experiments are correlated in terms of dimensionless variables. The program allows the choice between different standard solar energy system types and locations.

The Engineering Equation Solver EES (Klein et al. (1995)) is a general equation-solving program capable of solving non-linear algebraic and differential equations without requiring the knowledge of an actual programming language. The program allows optimization and regression. A thermal and transport property library for a variety of substances is integrated.

## 1.4 Background

The first part of this section briefly introduces the SDHWS of the William S. Middleton Memorial Veterans Hospital in Madison, WI. The next section discusses basic heat pump technology and the thermodynamics applied to the EES heat pump model (chapter 2.4). The last section gives a brief overview of solar assisted heat pump system configurations.

### 1.4.1 William S. Middleton Memorial Veterans Hospital

The William S. Middleton Memorial Veterans Hospital is located in Madison, WI. Having a size of 220 beds, the hospital is about 20% larger than the average U.S. hospital (Longo et al. (1990)). The hospital used a solar domestic hot water system to meet part of its hot water load. The system was designed by Affiliated Engineers, Inc. Madison, WI in 1977/78, and was installed in 1979. In 1994, another story was added to the wing where the solar collectors were located. And, the solar panels were removed.

The solar domestic hot water system was added to the existing domestic hot water system, which was natural gas driven. According to information of the Hospital Engineering Department (Frazier, 1995), the investment for collectors, storage, pumps, piping and installation was $ 330,000, in 1979. In 1994, the hospital's gas bill was $ 39,360, and an estimated $ 7,300 could be saved. The actual hot water draw has not been measured. According to the cost information, the solar fraction is about 18.5 %. The collector relocation costs were estimated with $ 282,000. In face of a resulting payback period of 38 years, the Hospital Engineering Department decided not to relocate the collectors.

The purpose of investigating the system, is to have a guideline for component sizing to simulate the system, and a real life example for simulation result evaluation.

### 1.4.2 Heat Pump Fundamentals

The principle of a heat pump is to transfer heat from a lower temperature source to a higher temperature sink. Hence, a heat pump uses the energy contents of a medium at low temperature level for temperature rise. Therefore, a motor driven compressor is required, which circulates and modifies a working medium. The source medium can either be gaseous or liquid, and most commonly are air and water.

The vapor compression heat pump is the most common heat pump design. Vapor compression heat pumps used for domestic water heating are called heat pump water heater. They are available in large sizes (see chapter 2.4.1). The design consists of four basic components: The compressor, which is driven by an electric motor, the condenser, the evaporator, and the expansion valve.

Figure 1.1 shows the heat pump water heater cycle on pressure-enthalpy coordinates. The working medium is a refrigerant, here R134a.

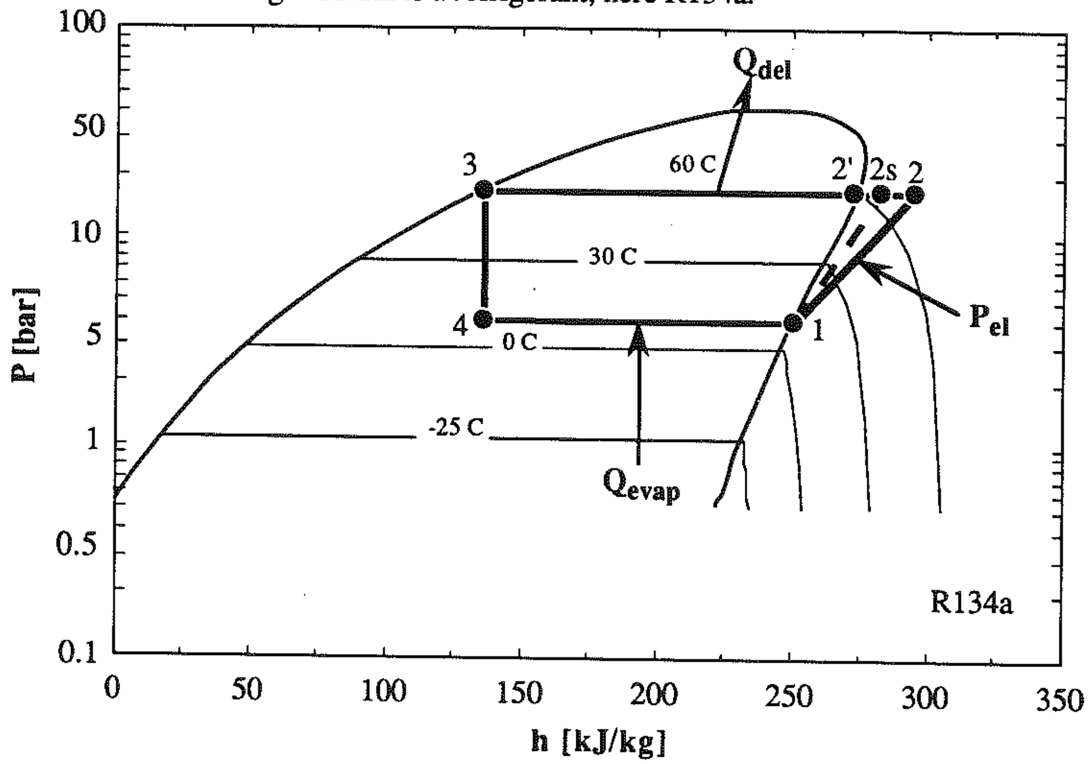



Figure 1.1 Vapor compression cycle (p-h diagram)

Refrigerant enters the compressor at point 1, where it is compressed to superheated vapor at higher temperature and pressure, point 2. An ideal cycle would follow the isentropic line to point 2s. Practically, because of heat transfer phenomena and irreversibilities, the compressor increases the enthalpy by a greater amount. The amount of increase is expressed by the isentropic efficiency (Reay and MacMichael (1988))

$$\eta_{iso} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1.1}$$

where $h_i$ is the enthalpy at point i. Mechanical losses do also decrease the compressor efficiency. The mechanical efficiency is defined as output over input

$$\eta_{mech} = \frac{\dot{m}_r(h_2 - h_1)}{P_{el}} \tag{1.2}$$

where $\dot{m}_r$ is the refrigerant mass flow rate, $h_i$ is the enthalpy at point i, and $P_{el}$ is the electric power input to the compressor motor.

In the condenser, energy is removed from the refrigerant with a cool external fluid, i.e. the water which should be heated. This heat transfer causes the superheated refrigerant to cool to saturation state, point 2', and then condense to point 3. Using the nomenclature of Equ. 1.2, the delivered heat to the water is

$$\dot{Q}_{del} = \dot{m}_r(h_2 - h_3) \tag{1.3}$$

The heat transfer is composed of the cooling and the condensing process as follows:

$$\dot{Q}_{del} = \dot{Q}_{desup} + \dot{Q}_{cond} \tag{1.4}$$

The relation between the delivered heat and the required compressor input is expressed in terms of the heat pump COP (coefficient of performance) (Mitchell and Braun (1996))

$$COP = \frac{\dot{Q}_{del}}{P_{el}} \tag{1.5}$$

Substituting $\dot{Q}_{del}$ from Equ.1.3, and substituting $P_{el}$ from Equ.1.2 into Equ.1.5 leads to

$$COP=\eta_{mech}\frac{h_2-h_3}{h_2-h_1} \tag{1.6}$$

Substituting the enthalpy difference ($h_2$ - $h_1$) from Equ.1.1 into Equ.1.6 simplifies the cycle COP calculation to

$$COP=\eta_{mech}\cdot\eta_{iso}\frac{h_2-h_3}{h_{2s}-h_1} \tag{1.7}$$

At point 3, the refrigerant enters an expansion valve which decreases the pressure and temperature. At point 4, lower temperature refrigerant enters the evaporator, evaporating as it removes energy from a warmer external fluid, the source. The refrigerant then reenters point 1 and repeats the cycle. The heat removal from the source to the evaporator is corresponding to Equ.1.3

$$\dot{Q}_{evap}=\dot{m}_r(h_1-h_4) \tag{1.8}$$

Figure 1.2 shows the equipment arrangement of a heat pump water heater. The heat pump cycle can be followed through the components as described above.

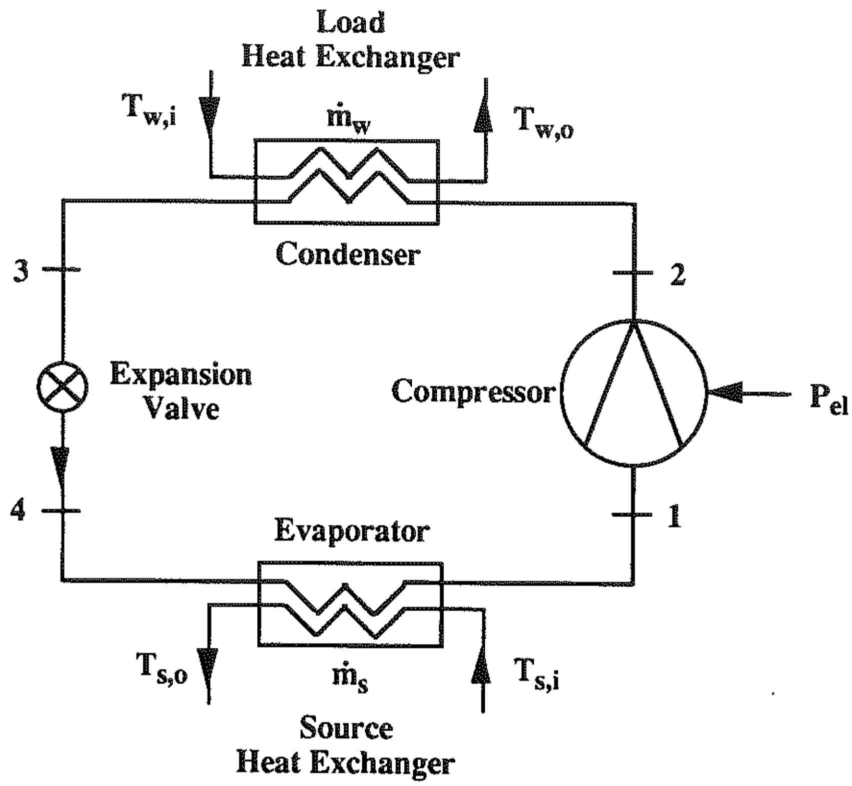



Figure 1.2 Heat pump water heater equipment arrangement

The heat transfer from the condenser to the water at flow rate $\dot{m}_w$ is through a counterflow heat exchanger. The water temperature is raised from the inlet temperature $T_{w,i}$ to the outlet temperature $T_{w,o}$. Usually, $T_{w,i}$ is the mains water temperature, and $T_{w,o}$ is the desired water set temperature. Expressing the delivered heat introduced in Equ.1.3 on the load side leads to

$$\dot{Q}_{del} = \dot{m}_w \cdot c_{p,w}(T_{w,o} - T_{w,i}) \tag{1.9}$$

where $c_{p,w}$ is the specific heat of water. Applying the NTU-effectiveness method for heat exchanger analysis (Incropera and DeWitt (1990)), the relation between $T_{w,i}$, $T_{w,o}$ and $T_2$, $T_3$ is found as a function of the effectiveness. The effectiveness $\varepsilon$ is the ratio of the actual heat transfer rate for a heat exchanger $\dot{Q}$ to the maximum possible heat transfer rate $\dot{Q}_{max}$

as follows

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{1.10}$$

According to Equ.1.4, the delivered heat at the condenser side is composed of cooling and condensing part. So, the effectiveness is composed of $\varepsilon_{desup}$ and $\varepsilon_{cond}$. Applying Equ.1.10, $\varepsilon_{desup}$ is (Incropera and DeWitt (1990))

$$\varepsilon_{desup} = \frac{C_w(T_{w,o} - T'_w)}{C_{min}(T_2 - T'_w)} \tag{1.11}$$

$C_w$ and $C_{min}$ are capacitance rates, the product of specific heat and mass flow rate. $c_w$ is the capacitance rate of the water load, and $C_{min}$ is the lower one of water load and refrigerant capacitance rates. The temperatures are defined as above. $T'_w$ is the water temperature between condensing and cooling process. Figure 1.3 qualitatively illustrates the course of temperatures, both at the load, and at the source heat exchanger (see below). Condensation and evaporation are represented by horizontal lines. Equ.1.10 gives $\varepsilon_{cond}$ as follows

$$\varepsilon_{cond} = \frac{(T'_w - T_{w,i})}{(T_3 - T_{w,i})} \tag{1.12}$$

$\varepsilon_{cond}$ is only a function of temperatures. During condensation, the refrigerant capacitance rate approaches infinity. Consequently, $c_w$ is equal to $c_{min}$. The effectiveness of a counterflow heat exchanger is determined by the following equation (Incropera and DeWitt (1990)

$$\varepsilon = \frac{1 - e^{-NTU(1-C_m)}}{1 - C_m \cdot e^{-NTU(1-C_m)}} \tag{1.13}$$

NTU is the number of transfer units, a dimensionless heat exchanger parameter. Commonly, NTU is between 2 and 5. $c_m$ is the heat capacitance ratio $c_{min}$ over $c_{max}$. Corresponding to $c_{min}$, which was defined earlier, $c_{max}$ is the higher one of water load and refrigerant capacitance rates. In case of condensation, $c_m$ is zero, and Equ.1.13 simplifies to

$$\varepsilon_{cond} = 1 - e^{-NTU_{cond}} \tag{1.14}$$

The heat transfer from the source to the evaporator is also through a counterflow heat exchanger. The source temperature is decreased from the inlet temperature $T_{s,i}$ to the outlet temperature $T_{s,o}$. The heat delivered from the environment is equals to $\dot{Q}_{evap}$ (Equ.1.8) and is

$$\dot{Q}_{env} = \dot{m}_s \cdot c_{p,s}(T_{s,i} - T_{s,o}) \tag{1.15}$$

$\dot{m}_s$ is the mass flow rate, and $c_{p,s}$ is the specific heat of the source. Neglecting effects from subcooling or superheating, the heat transfer is entirely through evaporation (see horizontal line in Figure 1.3). Corresponding to Equ.1.12, the effectiveness is

$$\varepsilon_{evap} = \frac{(T_{s,o} - T_{s,i})}{(T_1 - T_{s,i})} \tag{1.16}$$

Or, corresponding to Equ.1.14 for counterflow heat exchange

$$\varepsilon_{evap} = 1 - e^{-NTU_{evap}} \tag{1.17}$$

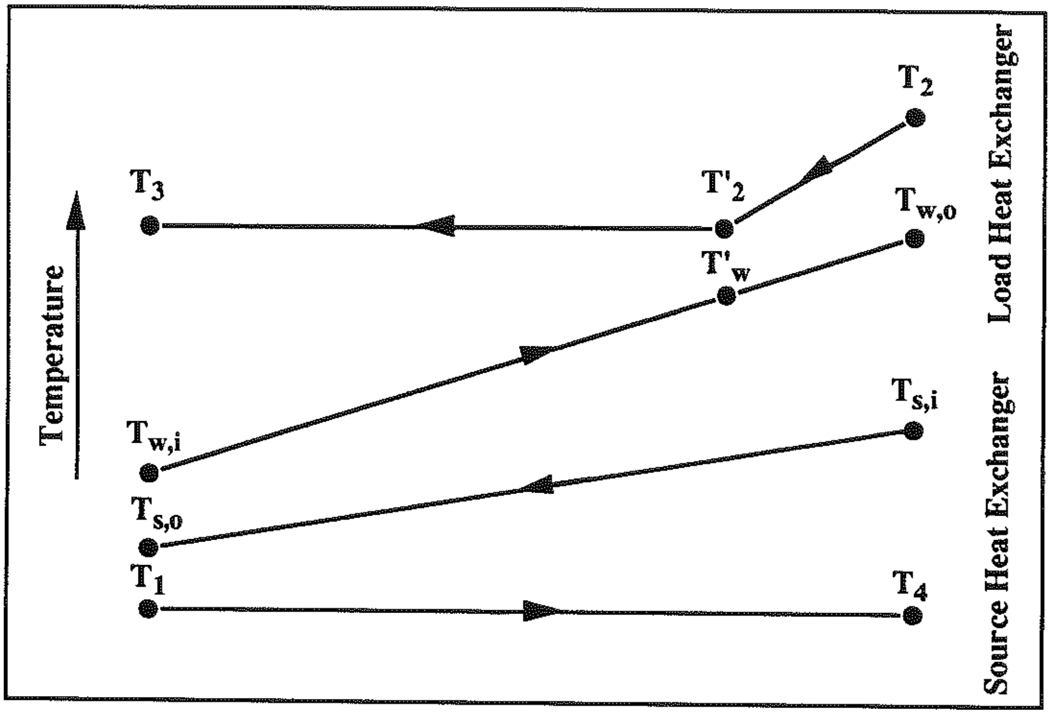



Figure 1.3 Temperatures at load and source heat exchangers

Heat pump water heaters use different types of sources, originating from the environment or indoor. The source medium is either air or water. Table 1.1 shows a selection of typical sources. The question of which type of source is the best cannot be easily answered. Ambient air is plentiful available and easily accessible. Having a specific heat about four times higher than air, water is the better heat transfer medium. With northern continental climate conditions, lake and especially ground water temperatures seasonally vary less than the ambient air temperature. Usually, indoor sources provide a comparatively high and constant temperature. The volumetric flow rate of waste heat from air conditioning fluctuates, whereas the waste water flow rate is year around evenly available. In this study, the systems are modeled with a ground water source heat pump for the reasons mentioned above. An additional advantage is that the ground water source is easy to model and simulate. Indoor source flows are hard to determine. The developed EES heat pump model (see chapter 2.4.1) has the capability to be adapted to other sources than ground water.

Table 1.1 Heat pump sources

| Origin | Medium | Source |
|---|---|---|
| environment | air | ambient |
| environment | water | lake |
| environment | water | ground |
| environment | air/water | radiation |
| indoor | air | waste heat |
| indoor | water | waste water |

The lack of a suitable heat pump size, or flexibility of controls, requires a battery of heat pumps, especially with large scale applications and when no additional back-up is used. There are different options to hook a set of heat pumps together. Figure 1.4 shows two typical heat pump configurations, series and parallel configuration (Reay and MacMichael (1988)). Certainly, more than two heat pumps can be hooked together. In a series system, the entire load flow rate $\dot{m}_w$ passes from one condenser heat exchanger through to the other. The water is gradually heated up. Consequently, the heat pumps must be able to handle the entire load flow rate, but they don't have to do the entire required temperature rise. In a parallel system, the water flow is divided into several flows, and each stream must be heated up to the required temperature by one heat pump. The heat pumps do only need the capability to handle a part of the load, but they have to be able to provide the entire required temperature rise. In this study, a parallel heat pump configuration is modeled (see chapter 2.4.1).

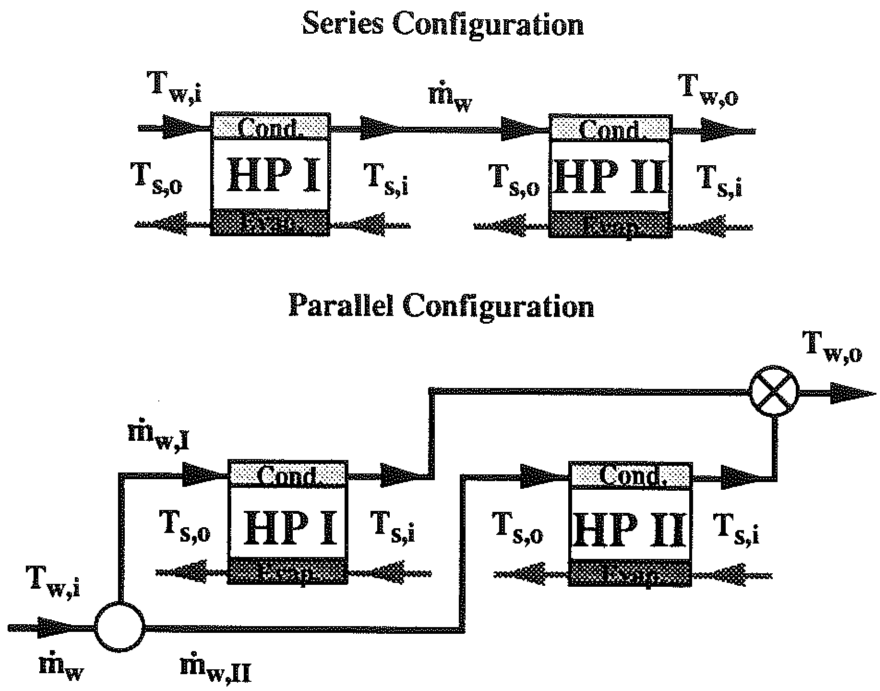



Figure 1.4 Heat pump configurations

The heat pumps don't necessarily have to meet the total water heating load. An additional back-up can be used. The heat pumps fulfill the function of pre-heaters. They are smaller sized, which decreases the investment costs. At the same time, more auxiliary heat is necessary. This option doesn't effect the heat pump COP as given by Equ.1.5, but lowers the system COP, which is defined as follows (Mitchell and Braun (1996))

$$COP_{system} = \frac{\dot{Q}_{load}}{\dot{W}_{comp} + \dot{W}_{aux} + \dot{W}_{para}} \qquad (1.18)$$

$\dot{Q}_{load}$ is the hot water load (see Equ.2.1). $\dot{W}_{comp}$ is the compressor power input, i.e. $P_{el}$ of Equ.1.2. $\dot{W}_{aux}$ is the auxiliary heat input, which is from either electric resistance, natural gas, or heating oil. This study is based on auxiliary heat from electric resistance with an efficiency of 1.0, which is of specific significance only when considering the economics (see chapter 4). $\dot{W}_{para}$ is the parasitic power input for circulator pumps and controllers.

The system COP is directly related to the free fraction (Equ.3.1) over

$$COP_{system} = \frac{1}{1-f} \tag{1.19}$$

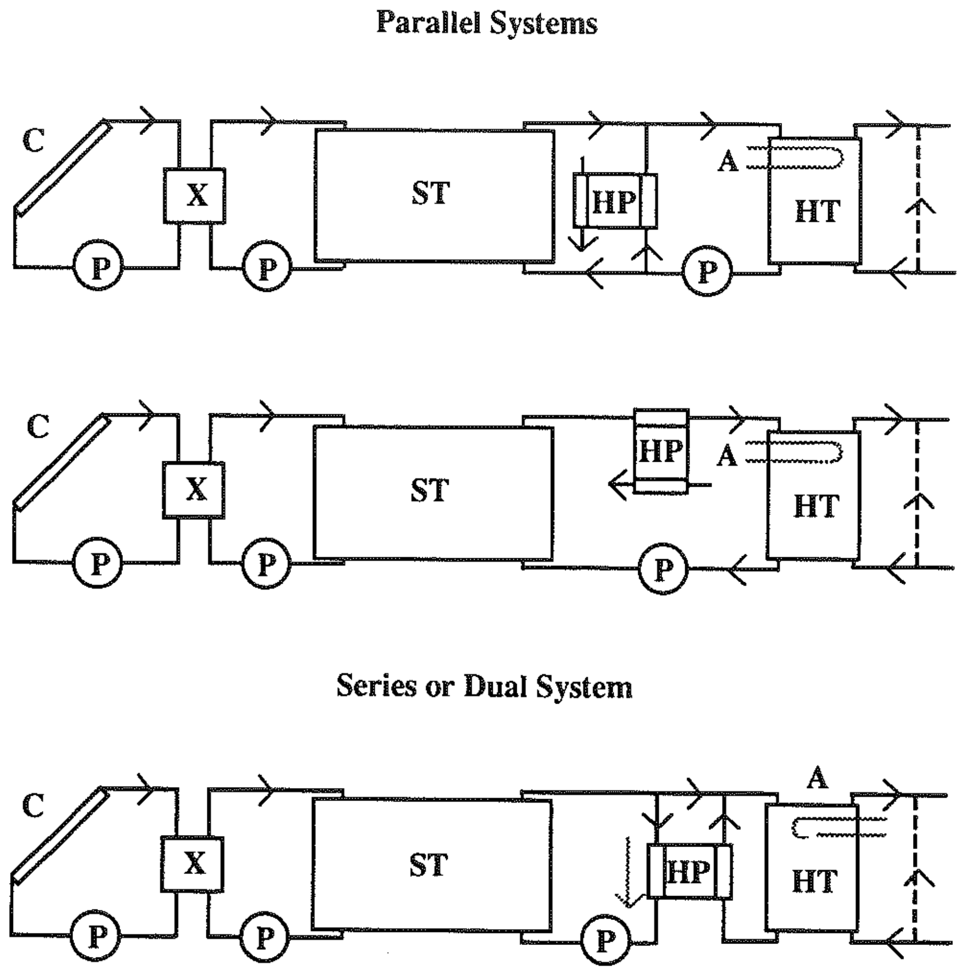
Neglecting the parasitic energy, having no auxiliary backup, and with $\dot{Q}_{load}$ and $\dot{W}_{comp}$ being equals to $\dot{Q}_{del}$ (Equ.1.3 and 1.9) and $P_{el}$, respectively, the system COP is equals to the heat pump COP.

### 1.4.3 Solar Assisted Heat Pump Systems

Generally, three different solar assisted heat pump system configurations can be found in literature: parallel, series, and dual source (Löf (1988), Duffie and Beckman (1991)).

In a parallel configuration, the heat pump serves as an independent energy source for the solar energy system. In a series configuration, the heat pump evaporator is supplied with energy from the solar energy system. Energy from the collector can also bypass the heat pump if its temperature is sufficient. The dual-source configuration combines parallel and series configuration with the heat pump evaporator supplied with energy from either solar energy or another source.

Figure 1.5 shows those three basic system configurations in a schematic way. The legend explains the system components. An electric auxiliary heating element is optional (faded).




**A Auxiliary Heating Element (optional)**
**C Collector**
**HP Heat Pump**
**HT Hot Water Tank**
**P Pump**
**X Heat Exchanger**

Figure 1.5 Basic solar assisted heat pump system configurations

The top and middle systems are two parallel system options. The upper parallel system is a system where hot water from the solar storage tank and from the heat pump flows into a hot water tank. There are two control strategy options: Hot water drawn from solar and from the heat pump is mixed and flows into the hot water tank. Or, only either the hot water from the solar storage tank, or from the heat pump serves the hot water tank. The lower parallel system shows a system where low temperature solar heated water is made up to the set temperature by the heat pump water heater.

The bottom system shows a series, or dual-source system. In a series system, the solar heated water, drawn from the storage tank, supplies the heat pump evaporator, cools down, and recirculates to the storage tank. The dual-source system uses two sources: the solar storage tank draw, and a collector-independent source like the parallel system uses. The controls selects the source with the higher available temperature. In the diagram, the solar-independent source is shown faded.

All the systems have in common that the heat pump is bypassed as soon as the storage tank draw temperature exceeds the domestic hot water set temperature.

It is difficult to qualitatively estimate how the systems compare in their thermal performance. A lot of factors effect each other, like collector and heat pump performance, choice of control strategy, and type of source. The performance of collectors is best at low temperatures. And, the performance of heat pumps is best at high evaporator temperature and low required water outlet temperature at the condenser side. Due to the thermophysical properties of refrigerants, the maximum possible outlet temperature is limited.

The following two paragraphs discuss advantages and disadvantages of the different systems. If not mentioned otherwise, a system without additional auxiliary heating element is assumed. The advantages and disadvantages of using an auxiliary heating element have been briefly summarized in the preceding section and will be discussed in a more detailed way in chapters 2.4 and 3.5.

The parallel system uses two free energy sources: solar collector heated water, and a separate source at the heat pump evaporator. The various advantages related to a separate heat pump source were mentioned in the preceding section. The mixing control strategy applied to the upper system shown in Figure 1.5 has the advantage that water from the solar storage tank is drawn, which temperature is lower than the domestic hot water set temperature. When using no other auxiliary heater, on the other hand, the heat pump condenser side must have the capability to heat up water to a temperature which exceeds the required set temperature. Consequently, this control strategy improves the collector efficiency, but lowers the heat pump COP. The control strategy with either solar storage tank draw or heat pump operation causes the opposite effect. Additionally, stand-by losses are increased due to higher solar storage tank temperatures. Also, the solar storage tank draw is easier to control. The preheat system as shown in the middle diagram of Figure 1.5 seems to combine the advantages of the parallel configurations mentioned above: Solar storage tank draw for temperatures lower than set temperature, the heat pump performance is limited to the set temperature, and a lower temperature lift is required. On the other hand, it's difficult to find a suitable control strategy, which combines benefits from lower temperature solar storage tank water draw and heat storage.

The series system has several advantages: Pre-heated water from the solar storage tank is used on the evaporator side, which causes a higher average COP. Low tempered water from the solar tank is regularly drawn, which increases the collector efficiency. Also, the heat pump set-up is simpler, because no ground coupled heat exchanger is required. On the other hand, the benefits of the solar collector gains can only be used as long as the heat pump operates and processes the solar heated water (unless the solar storage tank temperature exceeds the set temperature and is bypassed). Another important point is that in northern continental climates, the solar gain during some winter periods can be so marginal that the water storage cannot provide sufficient temperature at the heat pump evaporator to exclude freezing. That means, this system is only feasible with an additional auxiliary heat. The dual system avoids the freezing problem by selecting the higher source temperature of the two available.

Earlier research on solar assisted heat pump systems gives simulation and test results. Most of the research was done in the late 1970s, early 1980s, as a result of the energy crisis. Most studies and tests were focused on residential size forced air space heating systems (Duffie and Beckman (1991), Freeman, Mitchell, and Audit (1979), Löf (1988), Anderson (1979)). Usually, the heat pumps were air-to-air or water-to-air, and another additional auxiliary source was used. Simulation studies (Freeman, Mitchell, and Audit (1979), Duffie and Beckman (1991)) compare the three basic systems mentioned in the beginning of the section. These studies came to the conclusion that the parallel system with mixing control strategy is the best configuration. With same heat pump size, collector type and area, the free fraction was highest. The reason for this is that in the series and dual-source systems, the heat pump must operate whenever the stored solar energy is below the required supply temperature. The additional compressor energy required more than was compensated by the combined advantages of higher collector efficiency and higher heat pump COP, as mentioned above.

In this study, it was not possible to actually compare different system configurations. It was decided to only investigate a parallel system with either solar storage tank draw or heat pump operation as described earlier in this section. The heat pump is modeled with a ground source. Chapter 2.4 describes the system and its control in a more detailed way.

Several reasons led to this decision: Earlier simulation studies recommend a parallel system. Originally, the idea of the study was to investigate a system which does not use any auxiliary source other than the heat pump. This premise requires the heat pump has the capability to deliver water at least at set temperature. In order to keep the heat pump outlet temperature as low as possible, it was decided that water from the solar storage tank should only be drawn when the storage temperature exceeds the set temperature. This consideration led to this particular control strategy choice. The control strategy was kept, when later in the course of the study, the TRNSYS system model was extended to a model with auxiliary heating element. Also, a parallel system is the easiest to model. The heat pump was modeled with a ground source for reasons pointed out in the preceding section.

# CHAPTER TWO

## SYSTEM DESCRIPTIONS

This chapter gives a description of the different domestic water heating system models, which are investigated in this study. It shows the assumptions made, describes the system parameters and the implementation in TRNSYS. Appendix B lists the TRNSYS decks written in context with this study.

### 2.1 Load

Generally, the hot water load of a hospital is characterized by a continuous, high volume water draw, which varies in the course of the day. Unlike a typical residential domestic hot water load, short-time peak demand periods are not common. In contrast to building heating loads, no significant seasonal variation occurs. And, being a hospital, 100% reliability must be provided.

The instantaneous hot water load is calculated as

$$\dot{Q}_{load}=\rho_w \cdot \dot{V}_w \cdot cp_w \cdot (T_{set}-T_{mains}) \tag{2.1}$$

where $\rho_w$ is the density, $\dot{V}_w$ the required volumetric flow rate, $c_{p,w}$ the specific heat, $T_{set}$ the required outlet water temperature, and $T_{mains}$ the mains inlet water temperature. The density and specific heat of water are assumed constant.

Table 2.1 Hot water use in hospitals and nursing homes (U.S. DOH (1983/84))

| | Clinical | Dietary | Laundry |
|---|---|---|---|
| Water flow rate per bed, l/s (gph) | .0033 (3) | .0021 (2) | .0021 (2) |
| Temperature, °C (°F) | 43 (110) | 49 (120) | 71 (160) |

For a hospital, the required volumetric flow rate is a function of number of beds and time. The U.S. Department of Health and Human Services Guideline for Construction and Equipment (1983/84) establishes the design hot water load for hospitals and medical facilities. The system must have sufficient capacity to supply water at the temperatures and amounts indicated in Table 2.1.

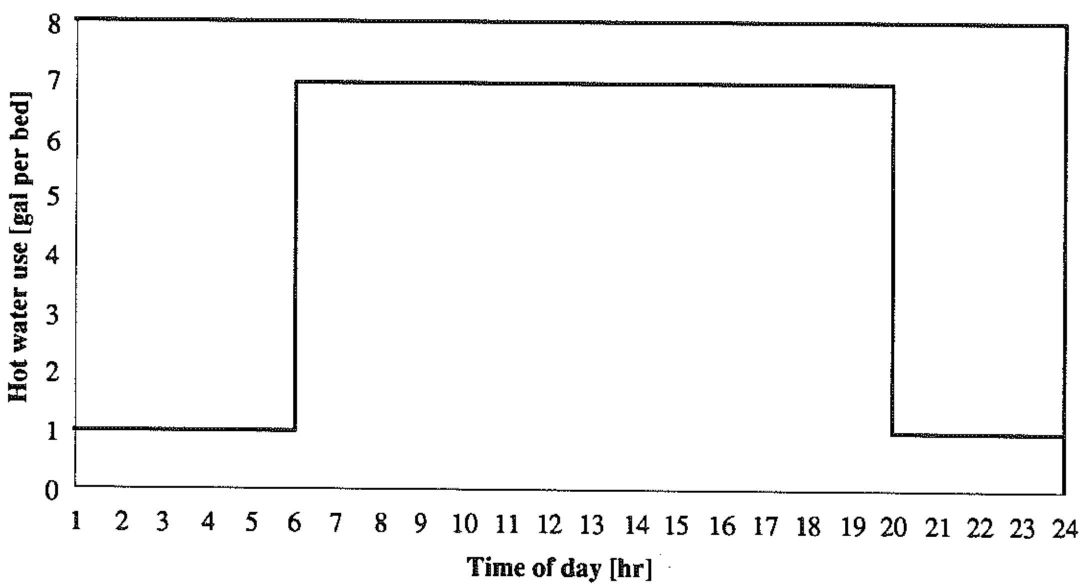



Figure2.1 Forcing function water draw for a typical hospital

The resulting load profile assumes that the recommended capacity is the constant day demand from 6 a.m. through 9 p.m. Night use has a constant lower level of 1 gph (.00105 l/s). Figure 2.1 shows the resulting forcing function used in Type 14, which models the water draw during the course of one day. Given a hospital with 220 beds, the volumetric flow rate is 1540 gph (5829 l/hr) during the day and 220 gph (833 l/hr) during the night. The hot water consumption over a 24 hour period is 23,760 gpd (89932 l/day). It is obvious that this idealized load profile does not reflect reality. As it was not possible to retrieve a more accurate model and as it accounts for design capacity, the idealized load is used in all system simulations.

The set temperature of 60 C is assumed to be constant. According to the A.O. Smith Corporation Catalog (1995), the current design practice is to provide the high temperature water demand for the laundry by a steam jet or a separate booster heater.

The catalog also suggests a "rule of thumb" for the required hot water load of 125 gallons (473 liters) of 140 °F (60 °C) water per bed per day. Given a hospital with 220 beds, the hot water consumption over a 24 hour period is 27,500 gpd (104,087 l/day). Although somewhat higher than the Health and Human Services Guideline, the recommendation is close to the calculations and assumptions, made above.

The mains water temperature slightly varies seasonally. The data reader Type 9 uses the F-CHART weather data file, which provides monthly average mains water temperatures for different U.S. locations.

Solving Equ. 2.1 for the maximum volumetric flow rate assumed above and a temperature lift of 95 °F (53 °C) yields a design instantaneous hot water load of about $1.2 \cdot 10^6$ Btu/hr (350 kW).

## 2.2 Electric Domestic Hot Water System

The electric domestic hot water system model is used to establish a reference model for the non-conventional systems. Figure 2.2 shows the system configuration. Appendix B.1 shows the resulting TRNSYS deck. The simulation load profile does not account for peak demands. Consequently, hot water storage is not necessary. Nevertheless, a tank is introduced into the system, because this configuration rather reflects reality, and provides an easy option to implement a more detailed load model, later.

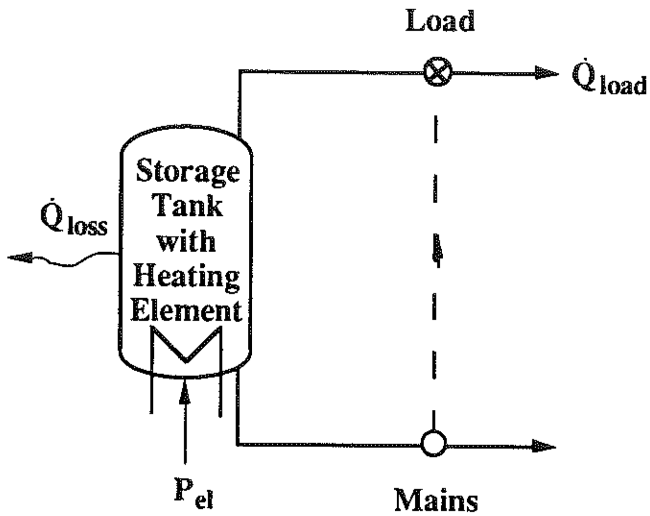



Figure 2.2 Electric domestic hot water system

The system energy balance on the tank yields

$$P_{el} - \dot{Q}_{load} - \dot{Q}_{loss} = \frac{dU}{dt} \tag{2.2}$$

$P_{el}$ is the electric power input, $\dot{Q}_{load}$ the energy rate delivered to the load as specified by Equ. 2.1. The loss energy rate is

$$\dot{Q}_{loss} = \frac{1}{R} \cdot A_T \cdot (T_T - T_{env}) \tag{2.3}$$

where R is the insulation R-value, $A_T$ the tank surface area, and $(T_T - T_{env})$ is the temperature difference between the tank water and the tank environment. The change in internal energy of the storage tank water is

$$\frac{dU}{dt} = (m \cdot c_{p,w}) \cdot \frac{d\overline{T}}{dt} \tag{2.4}$$

where $c_{p,w}$ is the specific heat, m the mass of the water in the tank, and $\frac{d\overline{T}}{dt}$ the time-dependent change in the average tank temperature.

### 2.2.1 Tank

The core of the system is the electric water heater. The thermostat and heater element are positioned at the bottom of the vertical water storage tank in order to provide a uniform high storage temperature. The tank and heating capacities are sized according to A.O. Smith Corporation Catalog (1995) , which assume a one hour storage capacity in order to cope with short time peak demands. Hence, the 220 bed model hospital requires a 1,500 gallon tank. A storage tank Type 4 is used. The tank is modeled to be fully mixed, because of the following three reasons: The stored water is heated at the very bottom of the tank, the water volume change rate is high, and the hot mass flow to the load is the only return mass flow from the tank. The tank is coated with a 4" (0.1 m) thick fiber glass insulation (ASHRAE (1993)). Table 2.2 shows the tank parameters.

Table 2.2 Tank parameters

| Parameter | | |
|---|---|---|
| Tank Volume | 1,500 [gal] | 5678 [l] |
| Tank Height | 13 [ft] | 3.96 [m] |
| Insulation R-Value | 16.0 [hr-ft$^2$-°F/Btu] | 0.79 [hr-m$^2$-°C/kJ] |
| Maximum Electric Power Input | 1500 [kW] | $5.4 \cdot 10^6$ [kJ] |

### 2.2.2 Controls

The heater is temperature controlled. It turns on whenever the tank temperature in the 4th node drops 5 °C below the set point temperature and stays on at its maximum power output until it reaches the set point. The controls is built into the storage tank Type 4.

A tempering valve is introduced to the system in case the tank outlet temperature is above the set point. Under this condition the tempering valve mixes mains water with a reduced water mass flow from the tank. The reduced flow rate can be found with mass and energy balances on the tempering valve:

$$\dot{m}_T + \dot{m}_{mains} = \dot{m}_{load} \tag{2.5}$$

$$\dot{m}_T \cdot c_{p,w} \cdot T_T + \dot{m}_{mains} \cdot c_{p,w} \cdot T_{mains} = \dot{m}_{load} \cdot c_{p,w} \cdot T_{set} \tag{2.6}$$

This concept is applied to all systems, investigated in this study, and is implemented in TRNSYS with a set of equations.

## 2.3 Solar Domestic Hot Water System

The system configuration, shown in Figure 2.3, corresponds to the solar domestic hot water system of the William S. Middleton Memorial Veterans Hospital in Madison, WI. Appendix B.2 shows the TRNSYS decks. The sun heats the circulating collector fluid. The heat is transferred by a heat exchanger to the solar storage water tank. An auxiliary heater makes up the water to the load to the desired set point temperature. For purpose of this simulation, an electric heater is used instead of a gas furnace. The system parameters correspond as far as possible, to information of the Hospital Engineering Department (Frazier (1995)) and the design company (Nelson and Kausch (1995)).

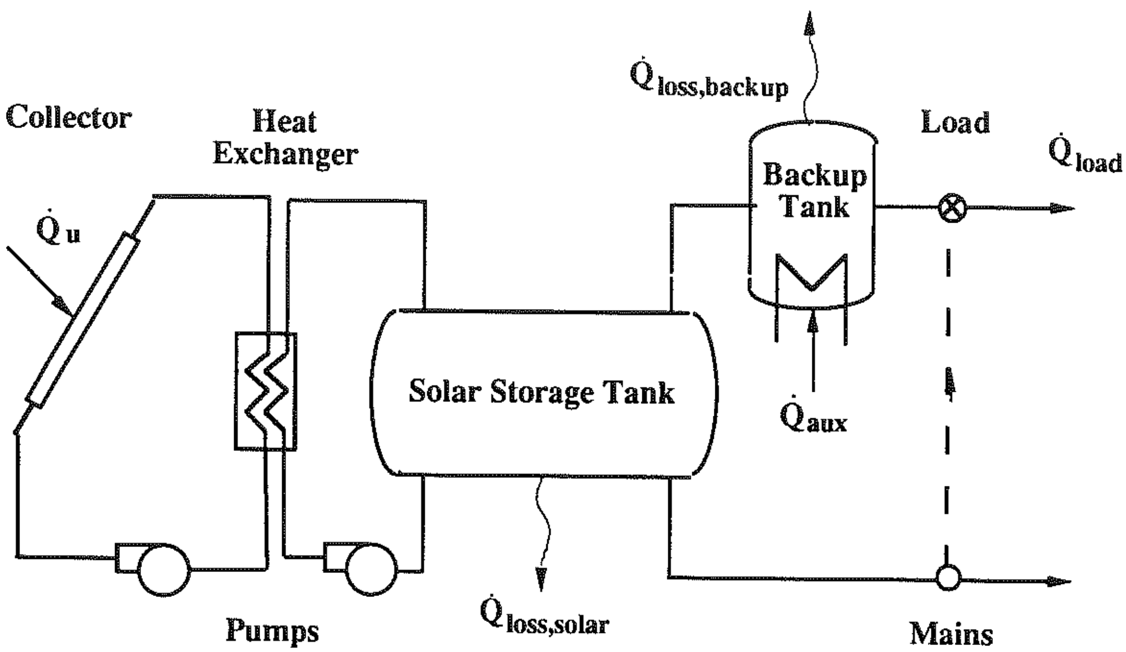



Figure 2.3 Solar domestic hot water system

Neglecting effects from pumping, the system energy balance yields

$$\dot{Q}_u + \dot{Q}_{aux} - \dot{Q}_{loss,solar} - \dot{Q}_{loss,backup} - \dot{Q}_{load} = \frac{dU}{dt}_{solar} + \frac{dU}{dt}_{backup} \quad (2.7)$$

$\dot{Q}_u$ is the net solar input, $\dot{Q}_{aux}$ the added auxiliary heat, in this case electric power input, $\dot{Q}_{loss,solar}$ and $\dot{Q}_{loss,backup}$ are the loss energy rates from the solar and the backup storage water tank, respectively as defined in Equ. 2.3, $\dot{Q}_{load}$ is the energy rate delivered to the load as specified in Equ. 2.1, and $\frac{dU}{dt}_{solar}$ and $\frac{dU}{dt}_{back\text{-}up}$ are the changes in internal energy of the tanks as characterized in Equ. 2.4. The useful energy gain from the solar collector field is (Duffie and Beckman (1990))

$$\dot{Q}_u = A_c \cdot F_R \ [G_T \ (\tau\alpha) - U_L \cdot (T_{c,i} - T_a)] \quad (2.8)$$

where $A_c$ is the collector area, $F_R$ the collector heat removal factor, $G_T$ the irridiance on the tilted collector surface, $(\tau\alpha)$ the transmittance absorptance product, $U_L$ the collector overall heat loss coefficient, $T_{c,i}$ the collector inlet temperature, and $T_a$ the ambient temperature. $F_R$, $(\tau\alpha)$, and $U_L$ are collector performance properties. $F_R(\tau\alpha)$ indicates the energy absorption, $F_RU_L$ counts for the energy loss.

### 2.3.1 Weather Data and Radiation Processor

The simulation uses weather data for the location for which the system performance is to be predicted. The Veteran Hospital's location is Madison. Madison represents a continental climate with hot summers and cold winters. The TRNSYS TMY database contains hourly weather data for different U.S. locations and is accessible by data reader Type 9. The data are based on the widely accepted SOLMET Typical Meteorological Year weather information. The global horizontal and the direct normal radiation are used in the radiation processor Type 16. The solar and the ambient temperature data are inputs of the solar collector Type 1.

### 2.3.2 Solar Collector

The solar collector field consists of flat-plate collectors with double glazing and non-selective surface, arranged in parallel. It was impossible to retrieve the original PPG collector performance data of 1976, which would ease evaluation. The Solar Rating & Certification Corporation rates and certifies solar collectors and water heating systems using standardized methods. Collector parameters typical to those listed in its directory were chosen (SRCC (1994)). The collector efficiency curve is modeled by a straight line with the Y intercept $F_R(\tau\alpha)$ and the slope $-F_RU_L$. It is assumed that the details of the collector performance do not significantly impact the general tendency of system performance results. An antifreeze fluid is used for freeze protection. A heat exchanger transfers the energy from the antifreeze to the water circulating through the solar storage tank. Table 2.3 shows the most important collector parameters used for TRNSYS.

Table 2.3 Collector parameters

| Parameter | | |
|---|---|---|
| Number of Collectors | 291 [-] | |
| Area per Collector | 17.5 [$ft^2$] | 1.63 [$m^2$] |
| Intercept Efficiency $F_R(\tau\alpha)$ | 0.7 [-] | |
| Negative of Slope $F_RU_L$ | 0.749 [Btu/hr-$ft^2$-°F] | 15.0 [kJ/hr-$m^2$-°C] |
| Incidence Angle Modifier $b_0$ | 1.0 [-] | |
| Collector Slope | 53 [°] | |
| Collector Flux | 13 [lb/hr-$ft^2$] | 63.47 [kg/hr-$m^2$] |
| Flux at Test Conditions | 10.24 [lb/hr-$ft^2$] | 50 [kg/hr-$m^2$] |
| Specific Heat Collector Fluid | 0.85 [Btu/lb-R] | 3.56 [kJ/kg-K] |
| Heat Exchanger Effectiveness | 0.5 [-] | |
| Tank Side Flux | 13 [lb/hr-$ft^2$] | 63.47 [kg/hr-$m^2$] |

### 2.3.3 Solar Water Storage Tank

Due to its high volume, the solar water storage tank has a horizontal shape. Generally, a horizontal tank does not stratify as much as a vertical tank. The tank is modeled by the algebraic tank (plug-flow) Type 38. This component model uses variable size segments of fluid, which allows a smaller simulation time step than a Type 4 tank. The important tank parameters are shown in Table 2.4.

Table 2.4 Tank parameters

| Parameter | | |
|---|---|---|
| Tank Volume | 10,000 [gal] | 37,850 [l] |
| Tank Height | 8.83 [ft] | 2.69 [m] |
| Insulation R-Value | 16.0 [hr-$ft^2$-°F/Btu] | 0.79 [hr-$m^2$-°C/kJ] |

### 2.3.4 Backup Tank with Auxiliary Heater

The auxiliary heater is located in a backup tank. The heating element must have the capability of heating up the total required hot water load for case when the water cannot be preheated by solar due to weather conditions. Therefore, the tank and heater are identical to the heater used for the conventional system (see subsection 2.2.1).

### 2.3.5 Pumps

Two pumps operate in the system: the collector loop pump and the tank loop pump. The collector loop pump is integrated in the collector model. The tank loop pump has the capability to produce the required mass flow rate of 30,000 kg/hr and has a power consumption of 3.73 kW. It is modeled by Type 3. The temperature lift of the water due to pressurizing is neglected. This simplification to the pump model is made in all systems, which are modeled in context of this study.

### 2.3.6 Controls

The pumps, and consequently the collectors, are controlled by an on-off controller. The controller compares the collector inlet temperature, measured at the bottom of the tank, with the collector exit fluid temperature or the mean plate temperature, when the collector fluid does not flow. Whenever the plate temperature at no-flow conditions exceeds the collector inlet temperature by a specific temperature difference $\Delta T_{on}$, the pumps are turned on. When the pump is on and the measured temperature difference falls below $\Delta T_{off}$, the controller turns the pump off. According to Duffie and Beckman (1991), the turn-off criterion must meet the following relation or the system will go unstable:

$$\Delta T_{off} \leq \frac{A_c \cdot F_R U_L}{\varepsilon \cdot (\dot{m} \cdot c_p)_{min}} \cdot \Delta T_{on} \tag{2.9}$$

The numerator of Equ. 2.9 counts for the useful energy gain when the pump turns on. $A_c$ is the collector area, $F_R U_L$ is modified for the heat exchanger (Duffie and Beckman (1991)). The denumerator indicates the heat removal. $\varepsilon$ is the heat exchanger effectiveness, $(\dot{m} \cdot c_p)_{min}$ the capacitance rate of the collector fluid. Solving Equ. 2.9 for the system parameters mentioned in the subsections above leads to

$$\frac{\Delta T_{on}}{\Delta T_{off}} \geq 8 \tag{2.10}$$

$F_R U_L$ yields 13.9 kJ/hr-m$^2$-°C (0.69 Btu/hr-ft$^2$-°F). The relation of Equ. 2.10 is independent of the collector area, because the mass flow rate through the collector is proportional to its area. An on/off differential controller Type 2 is used. $\Delta T_{off}$, which coresponds to the lower deadband temperature, is chosen to be 1 °C. $\Delta T_{on}$, which corresponds to the upper deadband temperature, is chosen to be 10 °C. The collector pumping system is shut down as soon as the water temperature exceeds the boiling point.

The control of the auxiliary heater is built into the backup tank as pointed out in subsection 2.2.2.

## 2.4 Heat Pump Domestic Hot Water System

The purpose of simulating a heat pump hot water system is to investigate the performance of a system configuration in which the hot water load is exclusively met by a heat pump water heater. Figure 2.4 shows the system configuration with heat pump and water storage tank. Appendix B.3 lists the TRNSYS decks used for the simulations. The heat pump is actually a battery of seven heat pumps, arranged in parallel, as pointed out in chapter 1.4.2. The number of heat pumps is chosen according to the load profile, as described in section 2.1. Each heat pump compressor is driven by an electric motor, the evaporators use environmental energy. External heat exchangers transfer the heat, released at the heat pump condensers, to the water, which flows to the tank. External heat exchangers are used, in order to provide a high flow velocity, and thus, good heat transfer.

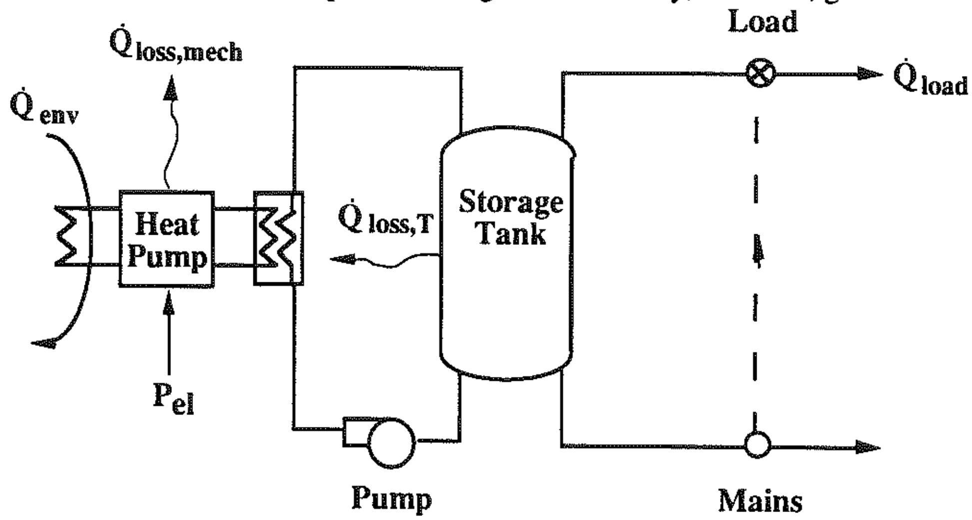



Figure 2.4 Heat pump domestic hot water system

Neglecting effects from pumping, the system energy balance yields

$$\dot{Q}_{env} + P_{el} - \dot{Q}_{loss,mech} - \dot{Q}_{loss,T} - \dot{Q}_{load} = \frac{dU}{dt} \qquad (2.11)$$

$\dot{Q}_{env}$ is the rate of energy contribution from the environment to the heat pump evaporator, $P_{el}$ is the electric power input to the compressor, $\dot{Q}_{loss,mech}$ is the loss energy rate resulting from mechanical inefficiency, $\dot{Q}_{loss,T}$ is the loss energy rate from the hot water storage tank as defined in Equ. 2.3, $\dot{Q}_{load}$ is the energy rate delivered to the load as specified in Equ. 2.1, and $\frac{dU}{dt}$ is the change in internal energy of the tank as characterized in Equ. 2.4. Heat pump details are given in chapter 1.4.2.

### 2.4.1 Heat Pump

The core of the system is the battery of heat pumps. For the purpose of modeling, simulating, and analyzing the performance of a heat pump domestic hot water system, the following two quantities of the heat pump performance must be known: the heat pump capacity under specific conditions and the resulting required electrical compressor power input in terms of the COP. The instantaneous capacity at the condenser depends on the following input variables: type of source, i. e. water or air, the given source inlet temperature, source mass flow rate, water inlet temperature and required water outlet temperature. Source temperature, source mass flow rate, and refrigerant mass flow rate have physical limits. For the water source, the outlet temperature has to be above freezing point.

<u>TRNSYS Types</u>

Currently, in TRNSYS, a general heat pump type does not exist. The available types require user-supplied performance data, usually retrieved from catalog tables. The data, dependent on the values of input variables, are provided in files. The model reads, interpolates and processes the data if required. The following three paragraphs give a brief description of the types 42, 20 and 71, which have the capability to model the performance of a heat pump.

Type 42 conditioning equipment (Klein et al. (1996)) models any piece of equipment whose performance can be characterized in terms of between one and three independent variables and between one and five dependent performance variables. The performance data are supplied in one single data file.

Type 20 dual-source heat pump (Klein et al. (1996)) models the performance of a heat pump having two evaporators: a liquid source to utilize heat from a solar system or other processes, and an ambient air source to be used when the outdoor temperature exceeds the liquid source temperature, or if the liquid source temperature approaches its freezing point. The model also allows a direct heating mode in which the hot liquid source bypasses the heat pump whenever its temperature exceeds a user specific minimum. Performance data are supplied in two data files as a function of the source temperature only.

Type 71 (Thornton (1995)) models a single-stage heat pump system. In the heating mode, energy is absorbed from a liquid stream at the evaporator and released to heat an air stream. In the cooling mode, energy is absorbed from the air and rejected to water. Performance data are supplied in six files as a function of mode, flow rates, and temperatures.

Catalog Data

Manufacturers of heat pump water heaters were contacted according to listings in Abrams and Sheedel (1989), and Thomas Register (1995) in order to obtain suitable catalog performance data. The product specifications of six manufacturers were investigated. It was not possible to retrieve any information from the companies beyond that given in the received catalogs.

Table 2.5 Catalog performance data of heat pump water heaters

| Company | Product series | Max. capacity [Btu/hr] | Source type | Capacity given for | | | COP |
|---|---|---|---|---|---|---|---|
| | | | | $T_{s,i}$ [F] | $\dot{m}_s$ [GPM/CFM] | $T_{w,o}$ [F] | |
| Fedders | SOCFO | 80,000 | water | 55-120 | 8.8 | 120-160 | yes |
| Drake | PWWH | 527,780 | water | 75 *) | 107 | 135 | no |
| | PAWH | 487,250 | air | 75 *) | - | 135 | no |
| Therma-Stor | HP **) | 20,700 | air | 67.5 | - | ΔT=60/80 | yes |
| Crispaire | WH-NT | 376,000 | air | Twb=45-75 | 9,200 | 95-130 | yes |
| | WH-HT | 276,000 | air | Tdb=75 Twb=63 | 8,600 | 150 | yes |
| ECU | WWH | 50,000 | water | 45-55 | 7.7-11.0 | - | yes |
| | AWH | 60,000 | air | Twb=59-71 | 1,900 | 95-130 | yes |
| | DS | 1,200,000 | air | - | - | - | - |
| Wallace | WRCB | 45,000 | air | - | - | 125 | - |

max. capacity: maximum available nominal capacity
COP: COP or electr. compressor power input given for each capacity
ΔT: temperature lift
Tdb: dry bulb temperature
Twb: wet bulb temperature
*): relation given for capacity as a function of source temperature
**): integrated backup heating element
-: no details given in catalog

Table 2.5 systematizes the performance data supplied by the companies. The first two columns specify the heat pump model by company and series name. The third column gives the maximum available nominal capacity of the series. The values show that heat pump water heaters are on the market which have the required capacity. Drake Industries, Inc. (1995), Crispaire Corporation (1995), and Energy Conservation Unlimited (1988) offer heat pumps with capacities larger than 200,000 Btu/hr. The performance tables for heat pumps with different nominal capacities show different results for same test conditions

(Crispaire, ECU). Hence, performance data of small scale heat pumps like Fedders Solar Products Company cannot be linearly extrapolated into large scale application. The fourth column indicates the source type. Air-source heat pump water heaters are more common than water-source. Air-source heat pumps usually provide air-conditioning or use waste heat from refrigeration systems. Crispaire gives the case study of a hospital in Huntsville, AL. The hospital kitchen is air-conditioned by a heat pump water heater which helps provide to meet the hot water load of the hospital. The following three columns show as a function of which variables ($T_{s,i}$, $\dot{m}_s$, or $T_{w,o}$) and in which range, the capacities are provided. The last column indicates if the performance table shows the COP or electrical compressor power input corresponding to each given capacity. Frequently, the catalogs give incomplete details (e.g. Drake, ECU, Wallace Energy Systems (1995)) and/ or only one test condition (e.g. Crispaire, Therma-Stor Products (1995)).

EES Model

The lack of suitable performance data which can be used to run the TRNSYS types mentioned above leads to the necessity to create a generic heat pump water heater model. It is beyond the scope of this study to model a new general TRNSYS heat pump type in terms of a FORTRAN code. Therefore EES was used to create a model which is capable to generate the performance data required for the existing TRNSYS types. Appendix A.1 shows the EES heat pump model which was developed in context with this study. The model follows the ideas outlined in the first paragraph of this subsection. The model processes the input data in terms of a given design capacity $\dot{Q}_{del,design}$, heat pump water outlet temperature $T_{w,o}$, which is usually the desired set temperature, source inlet temperature $T_{s,i}$, source mass flow rate $\dot{m}_s$, and heat pump water inlet temperature $T_{w,i}$, which is, assuming a stratified tank, usually close to the mains temperature. The simulation is based on thermal modeling and on approximations of catalog data. The simulation returns the performance data for the specified inputs, of which the energy released on the condenser side $\dot{Q}_{del}$ and the COP are of most interest. Figure 2.5 symbolizes the process.

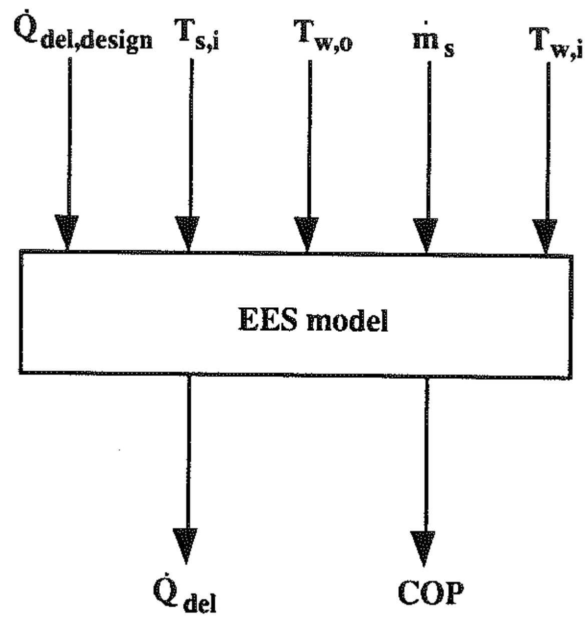



Figure 2.5 EES model

Heat pump cycle and heat transfer through the heat exchanger are modeled according to Equs. 1.1 through 1.17. Relations need to be found for the effectivenesses, efficiencies, and the heat pump capacity.

The refrigerant R134a is selected, because it is the state-of-art working fluid recommended for high condensing temperature application (Crispaire Corporation (1995)). The mechanical efficiency $\eta_{mech}$ is chosen to have the constant value 0.9 as recommended by Reay and MacMichael (1988).

The mechanical efficiency accounts for the whole working cycle. Effects of superheating or subcooling are neglected.

Isentropic efficiency $\eta_{iso}$ and NTUs are functions of the condensing temperature, or the set temperature. In order to find these relations, performance data of the Fedders Solar Products Company SOCFO80 heat pump are investigated. The performance table gives the values of COPs for six source inlet temperatures and five water outlet temperatures (see table 2.5). The COPs returned by the simulation are approximated to the table values. Therefore for each set temperature a set of values for $NTU_{evap}$, $NTU_{cond}$, and $\eta_{iso}$ is found, which minimizes the sum of deviations between the table and simulated values at different source temperatures. The deviation $\delta$ is found with the method of least square error

$$\delta = (COP_{table}-COP_{sim})^2 \tag{2.11}$$

where $COP_{table}$ refers to the table data, and $COP_{sim}$ to the simulation results. The sum x is given by

$$x = \sum_{i=1}^{6} \delta i \tag{2.12}$$

where $\delta_i$ is the deviation according to Equ. 2.11, and the subscript i indicates the different source inlet temperatures. The triples found for the five water outlet temperatures are curvefitted using linear regression. Figure 2.6 shows the resulting relations. The NTU values range between 2.5 and 5.0. These numbers are reasonable according to Incropera and DeWitt (1990). The equations for the NTUs are built in the EES model. The effectivenesses can be determined. It is assumed that the heat exchanger effectivenesses for heat transfer through condensing and through desuperheating are equal. The isentropic efficiency $\eta_{iso}$ is between 0.4 and 0.7.

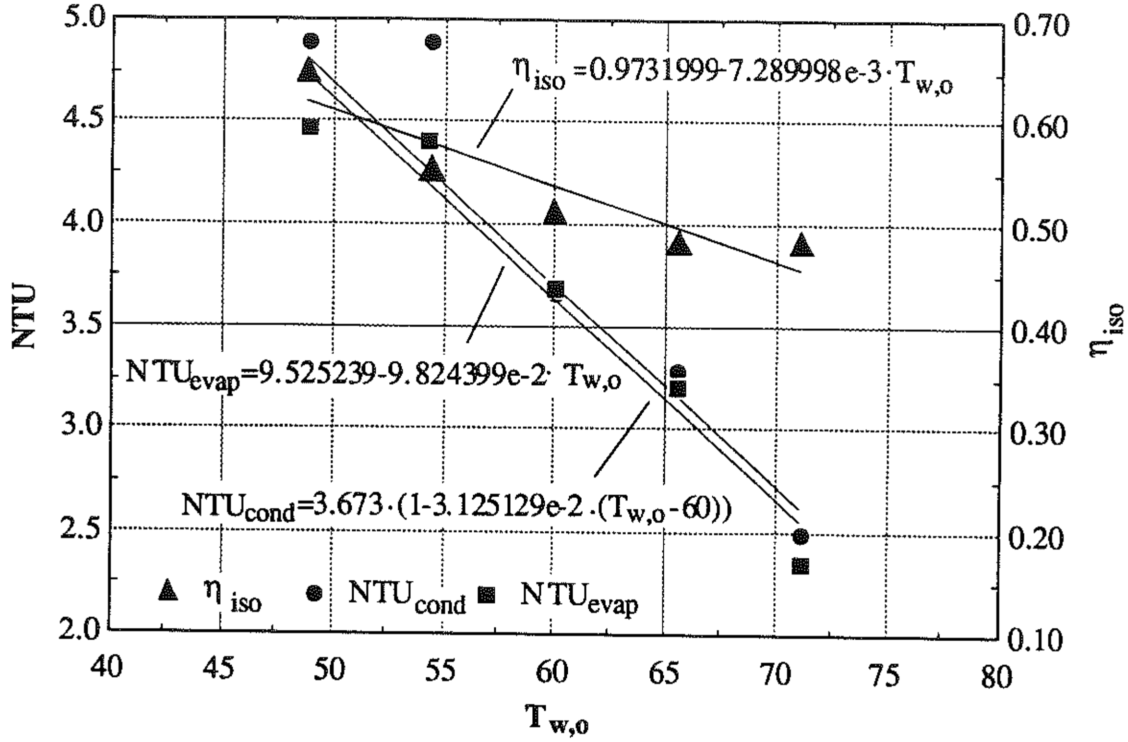



Figure 2.6 Relations for NTUs and $\eta_{iso}$

It is not possible to determine the heat pump capacity for specific input conditions without knowing compressor and electric motor details. Therefore the relation of the capacity for changing input conditions is approximated using catalog data. Drake Industries (1995) gives a relation for the capacity as a simple linear function of the source temperature. For every 10 °F rise or fall in source temperature the capacity will increase or decrease by approximately 6% compared to the capacity at test condition, listed in the performance table. Another simple linear relation is found for the capacity as a function of the set temperature by investigating and linearly curvefitting Fedders Solar Products Company SOCFO80 performance data. Combing these two relations, and introducing a design condition the capacity MBH yields

$$MBH = MBH_{design} \cdot \left(1 + \Delta T_s \cdot \frac{6}{1{,}000} + \Delta T_{set} \cdot \frac{5}{1{,}000}\right) \qquad (2.13)$$

where $\Delta T_s$ and $\Delta T_{set}$ are the difference between the design and actual source temperature and set temperature (i.e. $T_{w,o}$), respectively. The design capacity $MBH_{design}$ is the capacity the heat pump is able to deliver at design conditions, i.e. the worst case with lowest possible source temperature and highest necessary set temperature. Note that Equ. 2.13 is given for pound-inch units.

The relation given in Equ. 2.13 is independent of the source mass flow rate and the heat pump inlet temperature. Figure 2.7 shows the influence of the source mass flow rate on the heat pump COP. The three curves are results of simulations for same design capacity and heat pump inlet temperature, but for different settings for source and heat pump outlet temperatures. With increasing mass flow rate, the COP increases in an asymptotic way. Mass flow rates higher than a certain rate yield approximately identical heat pump performance. Hence, choosing a suitable rate, the influence of the source mass flow on the heat pump performance can be neglected.

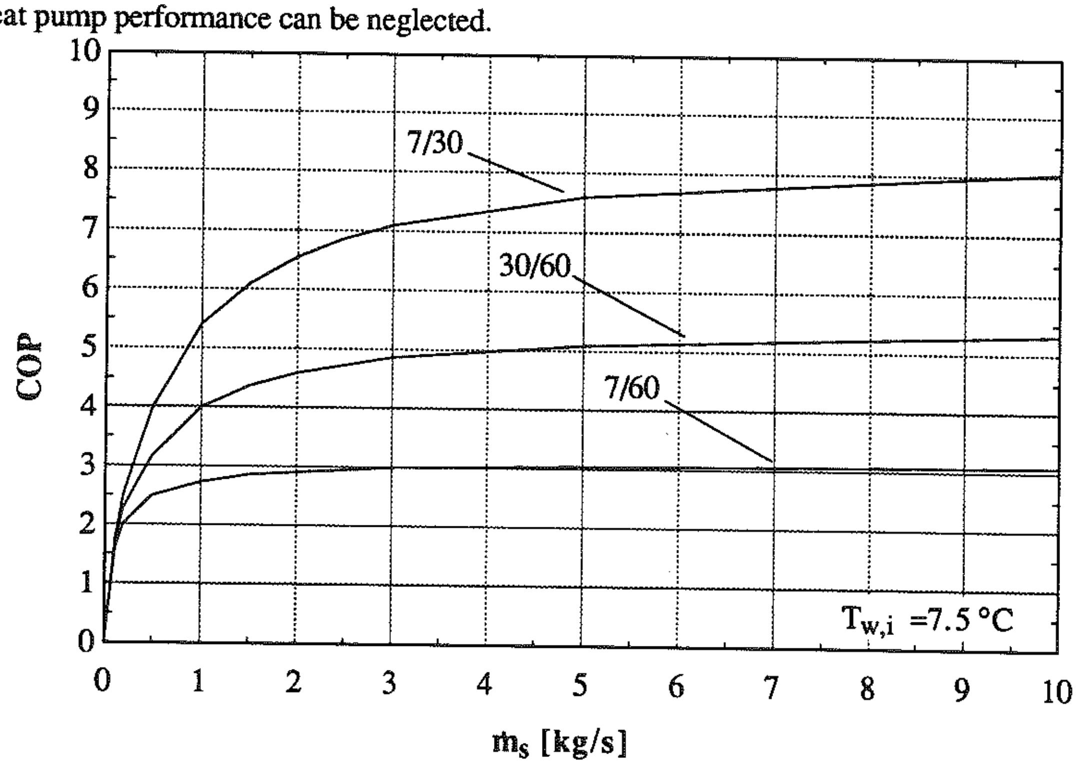



Figure 2.7 COP vs. $\dot{m}_s$ for different settings $T_{s,i}/T_{w,o}$

Figure 2.8 shows the influence of the heat pump inlet temperature on the COP. The three curves are results of simulations corresponding to those shown in Figure 2.7. The source mass flow rate is 2.5 kg/s. The COP decreases slightly with increasing heat pump inlet temperature. Hence, the influence of the heat pump inlet temperature on the heat pump performance can be neglected.

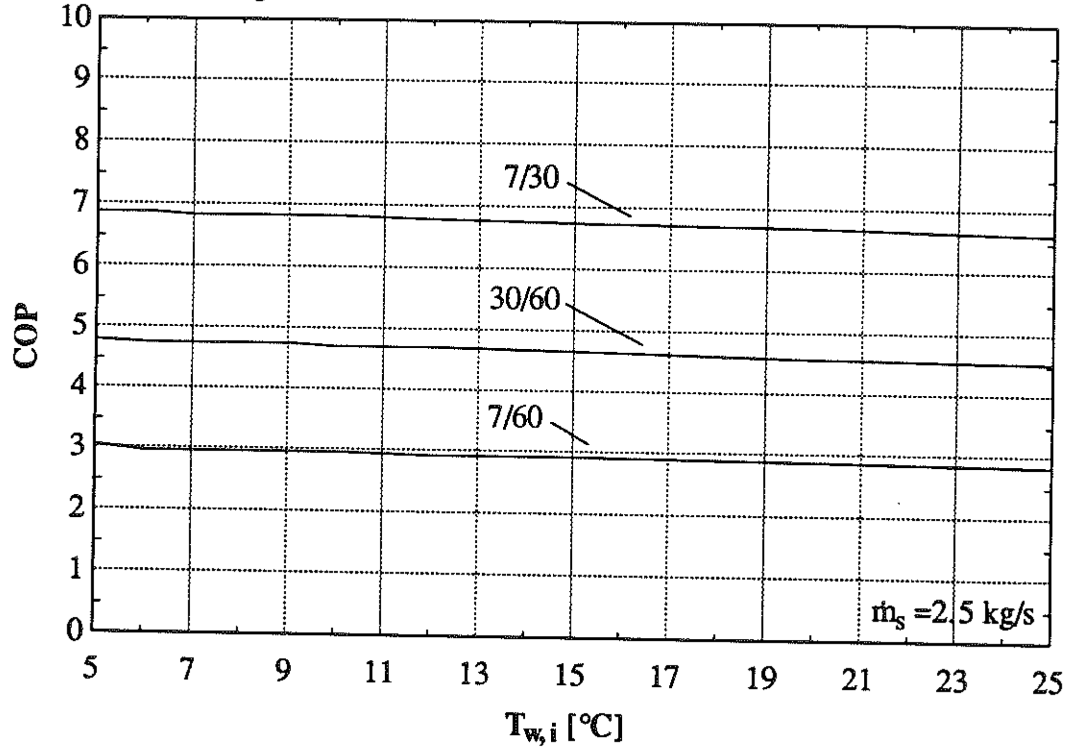



Figure 2.8 COP vs. $T_{w,i}$ for different settings $T_{s,i}/T_{w,o}$

<u>Validation</u>

The validation of the model is difficult and ambiguous. As mentioned before, a variety of heat pumps are on the market, and most of the available performance data are incomplete.

Figures 2.9 and 2.10 compare the heat pump performance according to the EES simulation and to the Fedders Solar Products Company SOCFO80 catalog data. The simulations were carried out using identical values for design capacity, heat pump inlet temperature, and source mass flow rate. Figure 2.9 shows the delivered heat vs. the source temperature for different heat pump water outlet temperatures $T_{w,o}$. The calculated delivered heat is less sensitive to the source temperature than the delivered heat according to the Fedders catalog performance data, because this relation was found using Drake Industries, Inc. (1995) catalog information.

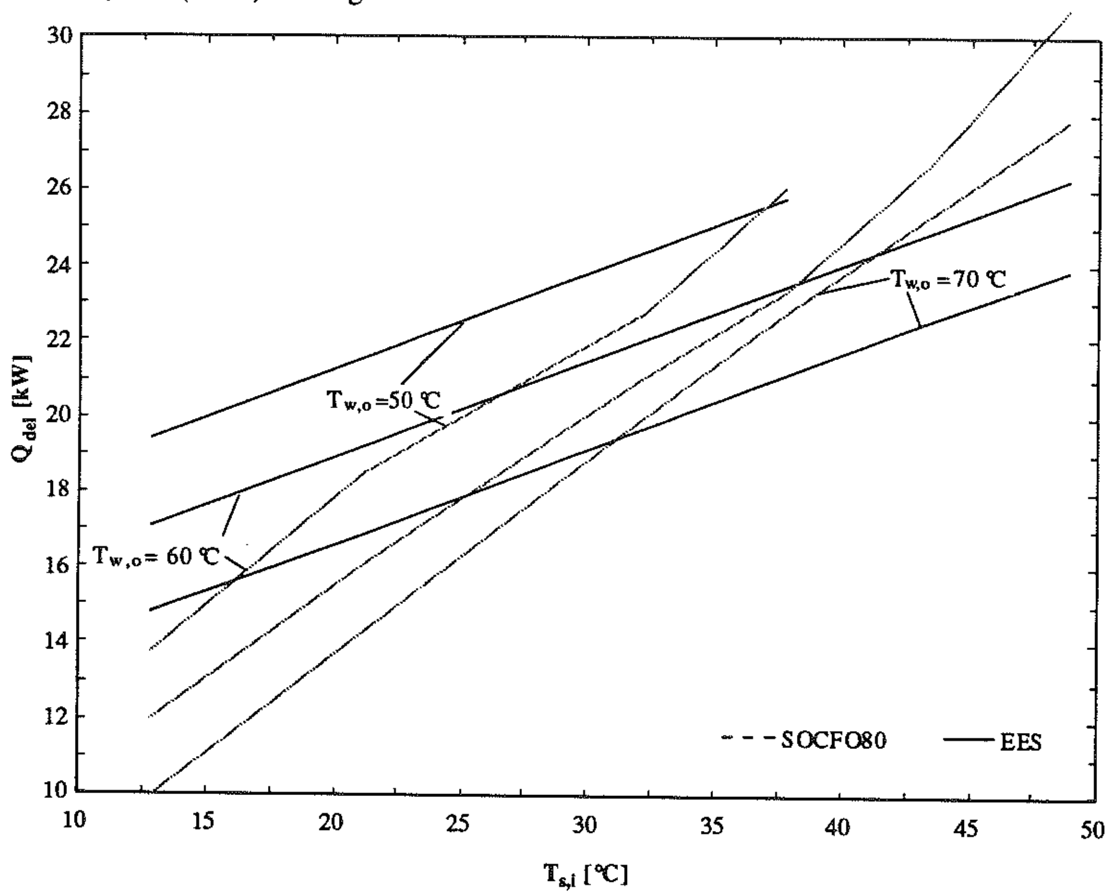



Figure 2.9 Simulation and catalog data $Q_{del}$ vs. $T_{s,i}$ for different settings $T_{w,o}$

Figure 2.10 shows the COP vs. the source temperature for different heat pump outlet temperatures. For an outlet temperature of 50 °C, the calculated COPs are up to 20% higher than the catalog data COPs. For higher heat pump outlet temperatures, the calculated COPs show sufficient coincidence with the catalog data. With a set temperature of 60 °C, the design water outlet operating point of the heat pump, the COPs give reasonable values.

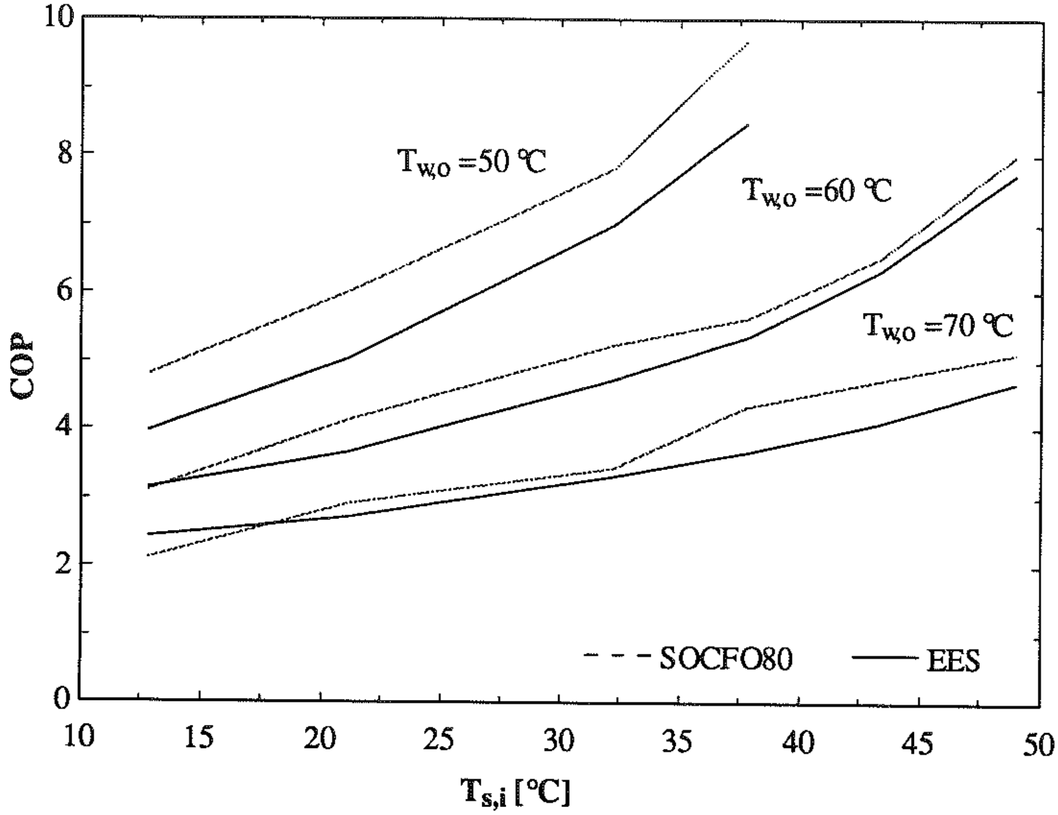



Figure 2.10 Simulation and catalog data COP vs. $T_{s,i}$ for different settings $T_{w,o}$

Performance

The heat pump performs for heat pump outlet temperatures which are at least as high as the source temperature, but do not exceed 85 °C. With source temperature and heat pump outlet temperature being the same, the COP theoretically reaches infinity. Above 85 °C heat pump outlet temperature, a two phase condition does not exist anymore. The COP approaches unity.

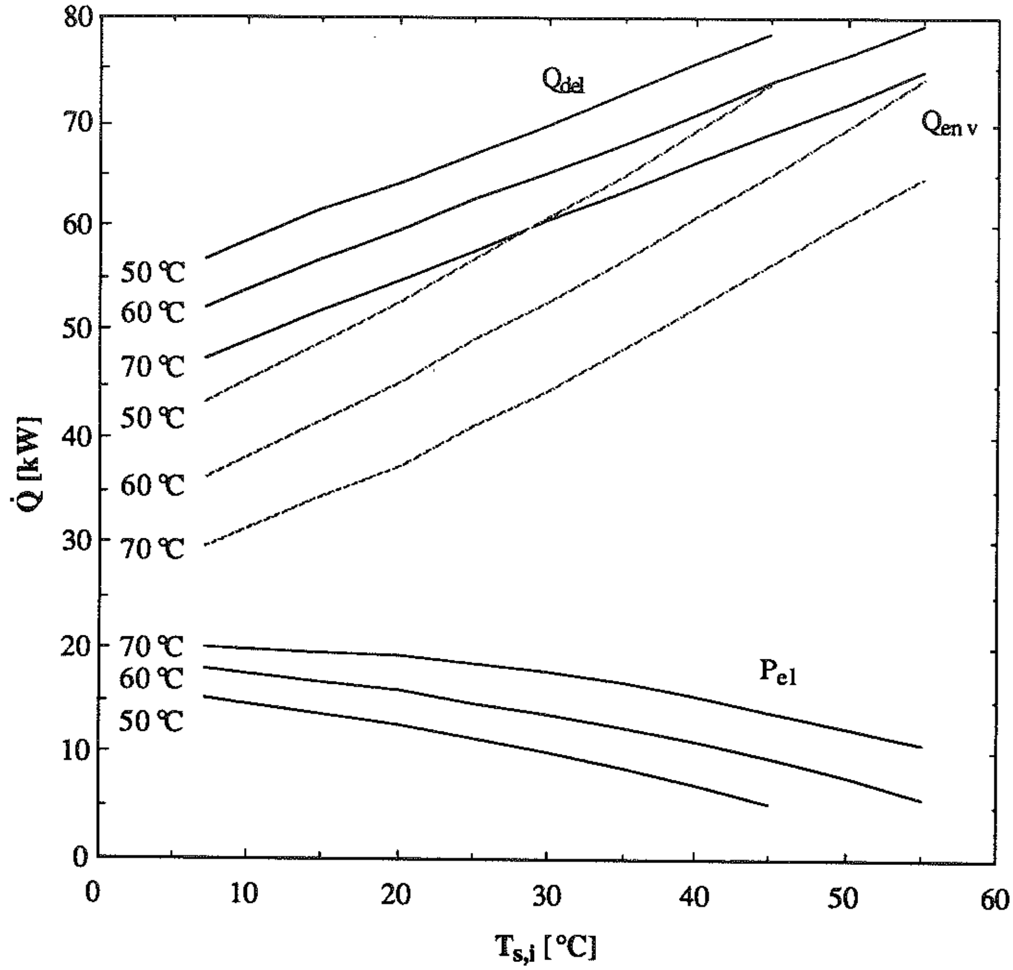



Figure 2.11 Heat pump performance vs. $T_{s,i}$ for different settings $T_{w,o}$

Figures 2.11 and 2.12 show the heat pump performance for a design capacity of 53 kW at $T_S$=7 °C and $T_{w,o}$=60 °C. The source mass flow rate is 2.5 kg/s, and the heat pump inlet temperature is 7.5 °C. Figure 2.11 shows the relation of delivered heat $\dot{Q}_{del}$, supplied heat from the environment $\dot{Q}_{env}$, and compressor power $P_{el}$ to the source temperature and heat pump outlet temperature. Delivered and supplied heat increase with increasing source temperature and decreasing outlet temperature. The compressor power decreases. The course of the curves illustrates the relation between $\dot{Q}_{env}$, $P_{el}$, and $\dot{Q}_{del}$ pointed out in subsection 1.4.2. Figure 2.12 shows the resulting COP versus the source temperature for different heat pump outlet temperatures. The COP increases with increasing source temperature and decreasing outlet temperature. Appendix A.2 shows a table with the heat pump performance data $\dot{Q}_{del}$, COP, and $\dot{Q}_{evap}$ for different heat pump water outlet and source inlet temperatures.

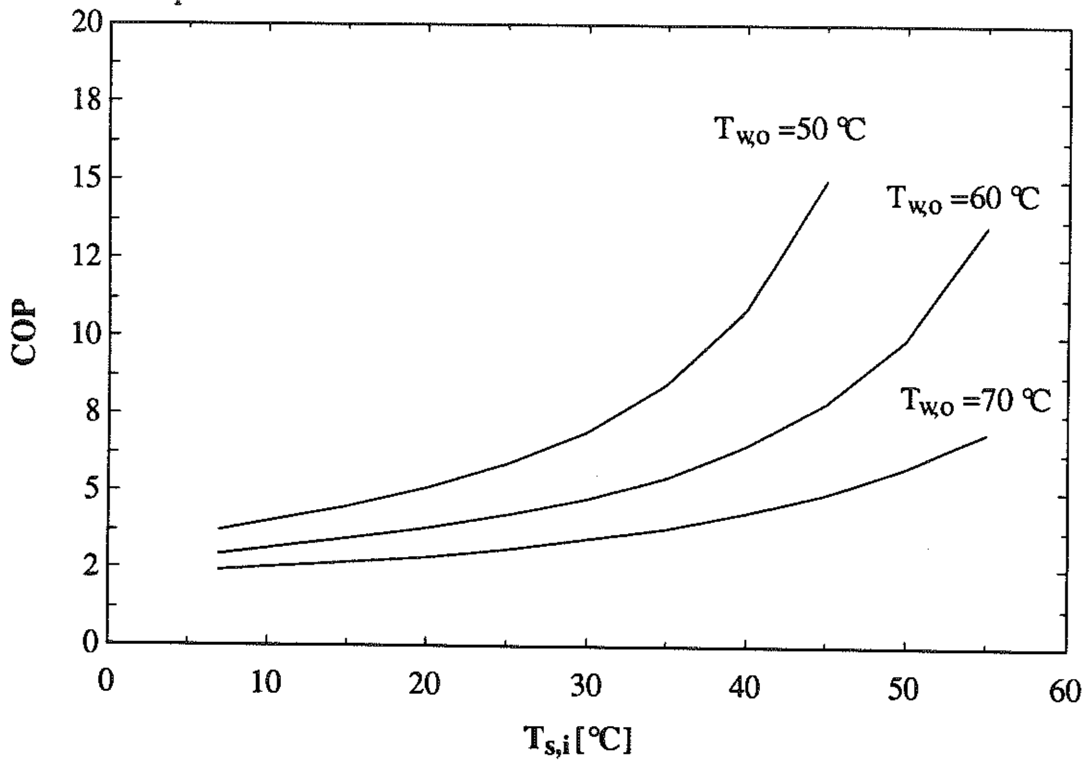



Figure 2.12 COP vs. $T_{s,i}$ for different settings $T_{w,o}$

Sizing

The heat pump capacity must be capable to meet the entire load. Using Equ. 2.1, and assuming a heat pump inlet temperature of 7.5 °C, the required capacity of each heat pump is 53 kW. This design capacity has to be delivered at the lowest possible source temperature, which is assumed to be 7 °C. The ground source temperature of a specific location can be approximated using the average annual ambient temperature of this location. The Madison average annual ambient temperature is 7.2 °C.

The source mass flow rate has to meet three requirements as mentioned earlier: The mass flow rate must be high enough that it does not significantly influence the COP (Figure 2.7), and that the source leaving temperature does not drop below the freezing point at lowest possible source entering temperature. On the other hand, the source mass flow rate must not exceed an amount which can practically be obtained. Comparing information given by Figures 2.7, 2.13, and Drake Industries, Inc. (1995) catalog data, a mass flow rate of 2.5 kg/s at each heat pump evaporator is reasonable.

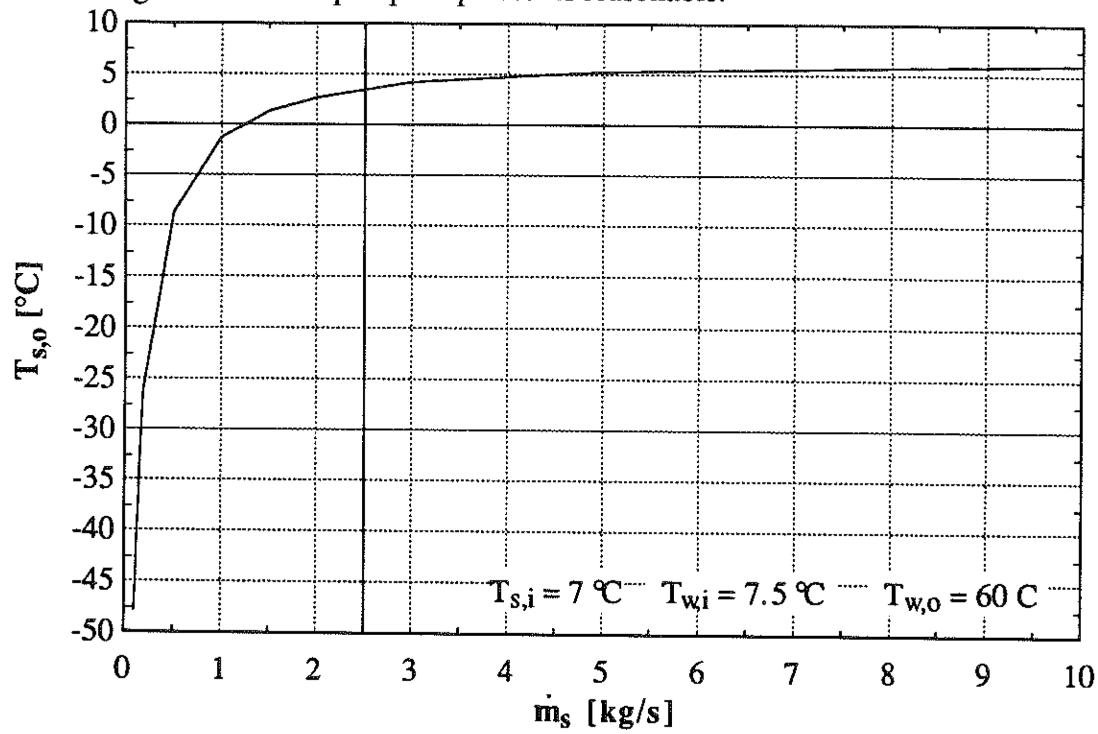



Figure 2.13 $T_{s,o}$ vs. $\dot{m}_s$

Implementation in TRNSYS

TRNSYS Type 42 conditioning equipment is used to process the performance data generated by the EES model. Type 42 is the most flexible and has the capability to handle the two independent input variables, which are $T_{s,i}$ and $T_{w,o}$. The dependent performance variables are the corresponding delivered heat $\dot{Q}_{del}$ and the COP. A relation for the delivered heat dependent on the mass flow rate through the condenser heat exchanger and the temperature lift is given by Equ. 1.9. The heat pump outlet temperature is obtained by rearranging this equation. The mass flow rate through each condenser heat exchanger is constant at 842 kg/h. Providing sufficient capacity, the heat pump outlet temperature balances into a value sufficient to meet the required set temperature at design water draw.

### 2.4.2 Hot Water Tank

The hot water tank is sized according to the recommendations given in subsection 2.2.1. With the cold water from the mains entering the bottom of the tank, and the hot water from the heat pump entering the top of the tank, stratification is assumed. A 5-node-tank Type 4 is used.

### 2.4.3 Pumps

Each heat pump requires a circulation pump to move the water through the heat exchanger. Each pump has the capability to produce the required mass flow rate of 842 kg/h, and has a power consumption of 106 W. The pumps are modeled by one single Type 3, varying the mass flow rate and power consumption with help of a control function.

### 2.4.4 Controls

The operation of the pumps, which is coupled to the operation of the heat pump compressors, is both, time and temperature controlled. During the night, only one heat pump operates. By 6 a.m., a timer turns on the other six available pumps. By 8 p.m., the pumps are turned off, again. The timer is identical to the forcing function type 14 used to model the load profile (see section 2.1). The pumps shut down whenever the temperature measured in the third node of the tank exceeds the set temperature of 60 C, and turns on whenever it drops below. An on/off differential controller Type 2 is used. With the mains water temperature being the only time-dependent variable besides of the load, the system operates in almost steady state. In this case the temperature controller is rather a safety device.

## 2.5 Parallel Solar Heat Pump Domestic Hot Water System

In context of this study, a parallel solar heat pump system was investigated as introduced in chapter 1.4.3. Figure 2.14 shows the configuration according to Löf (1988). Appendix B.4 gives the TRNSYS decks. Either the solar or the heat pump domestic hot water system operates, as described in sections 2.3 and 2.4, respectively.

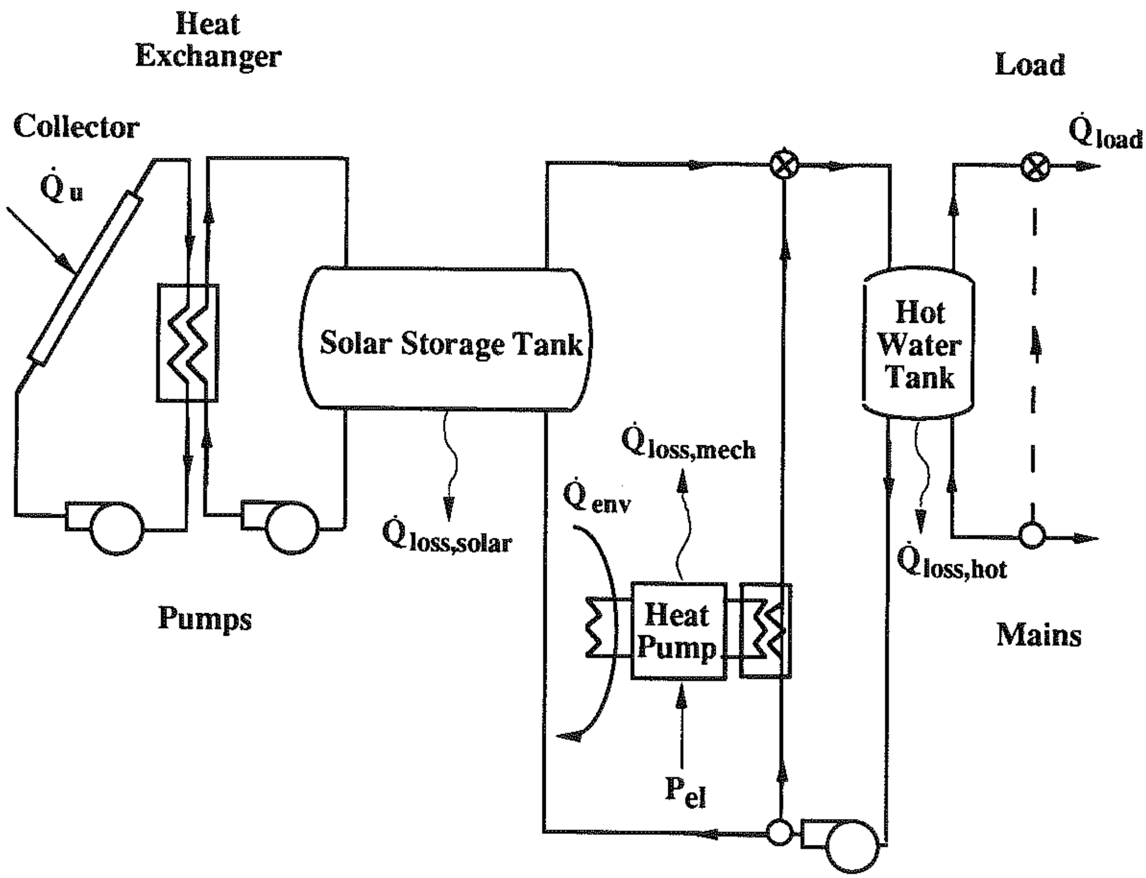



Figure 2.14 Parallel solar heat pump domestic hot water system

Combining Equ. 2.7 and 2.11, and using identical nomenclature, the system energy balance yields

$$\dot{Q}_u + \dot{Q}_{env} + P_{el} - \dot{Q}_{loss,solar} - \dot{Q}_{loss,hot} - \dot{Q}_{loss,mech} - \dot{Q}_{load} = \frac{dU}{dt}_{solar} + \frac{dU}{dt}_{hot} \quad (2.14)$$

The system components are sized according to sections 2.3 and 2.4. The heat pump has the capability to meet the total load, because the solar gain is not reliable. The following subsection will characterize the control strategy, which is specific to the parallel system.

### 2.5.1 Controls

The solar collector and the heat pump subsystems are controlled in accordance with subsections 2.3.6 and 2.4.4, respectively. There are two ways to supply hot water to the hot water tank: Hot water is either drawn from the solar storage tank, or is supplied by the heat pump. The hot water supply to the hot water tank is temperature controlled. An on/off differential controller Type 2 is used. Whenever the water temperature at no-draw conditions measured at the top of the solar storage tank exceeds the set temperature by 10 °C, hot water is drawn from the solar storage tank and replaced by cold water from the bottom of the hot water tank. The heat pump compressors shut down. When water is drawn from the solar storage tank and the measured temperature falls below 1 °C above the set temperature, the draw is terminated and the heat pump compressors turn on. The chosen deadbands allow stable controls (see subsection 2.3.6).

# *Chapter Three*

## System Simulations

The first two sections of this chapter show the methodologies used simulating the systems with TRNSYS, and analyzing the system performances. The later sections describe the simulations which were run for each system, and show, evaluate, and validate the obtained results. Appendix B lists the used TRNSYS decks.

### 3.1 Methodology

Before running time-consuming one-year simulations, one-day simulations of an average day (April 9th) were analyzed. In order to exclude effects of initialization, one week simulations were run, using the same weather and radiation data for each day. It was assumed that a steady periodic state was reached by the 7th day. Hourly values of temperatures and mass flow rates of the 7th day were investigated in terms of reasonability. The analysis should ensure that the system components and controls are designed properly.

Annual simulations were run with 6 days pre-simulation, as well. The weather and radiation data of the last days of December were used. The results are summarized in monthly simulation summaries. The TRNSYS simulation summary, Type 28, has the capability to integrate and average simulation outputs for each month of the year. Additionally, the system energy balance can be checked. The monitored simulation outputs are: the monthly system energy inputs and outputs, the monthly average collector and tank temperatures, and the monthly average heat pump COPs. The average annual deviation of the system energy balance check compared to an ideally closed energy balance never exceeded 0.5%.

The total annually required amount of auxiliary energy is used to calculate the free fraction. The free fraction is given by

$$f = \frac{Q_{conv} - Q_{el}}{Q_{conv}} \tag{3.1}$$

$Q_{conv}$ is the annual energy requirement of a conventional domestic hot water system as described in section 2.2. $Q_{el}$ is the sum of back-up, heat pump, and parasitic energy requirement of the investigated system. Parasitic energy includes the energy required by circulator pumps and controllers. For purpose of this study, the parasitic energy requirement is neglected. Later on, the validity of this assumption will be discussed.

Annual simulations were run for different system parameter settings. Collector area, solar storage tank capacity per collector area unit, and heat pump capacity were varied.

The results of the thermal analysis are a basis for economic considerations. The annual fuel savings are calculated using the free fraction. The parameter settings influence the additional initial investment.

## 3.2 Simulation Time Step

TRNSYS requires the user to specify a simulation time step which is fixed throughout the simulation. For a thermal process, the largest possible value of time step for which the integration algorithm is estimated to be stable is (Klein et al. (1996))

$$\Delta t = \frac{(m \cdot c_p)}{(\dot{m} \cdot c_p)_1 + (\dot{m} \cdot c_p)_2} \tag{3.2}$$

where $(m \cdot c_p)$ is the thermal capacitance and $(\dot{m} \cdot c_p)_1$, $(\dot{m} \cdot c_p)_2$ are the thermal capacitance rates. Regarding the capacitance of a single tank node and assuming constant density and specific heat leads to

$$\Delta t = \frac{V_T}{N \cdot (\dot{V}_1 + \dot{V}_2)} \tag{3.3}$$

where $V_T$ is the total tank volume, N the number of nodes, and $\dot{V}_1$, $\dot{V}_2$ are the volumetric flow rates through the tank, e.g. the collector and the mains flow rate. The variables have to be chosen for an extreme case with smallest total tank volume, highest number of nodes, and largest occurring volume flow rates. The obtained critical time step is a first guess value, which must be evaluated for each case during the system simulations.

## 3.3 Conventional System

The total annual energy requirement was estimated by running a one year TRNSYS simulation of the conventional system according to subsection 2.2. Appendix B.1 shows the TRNSYS deck. The total annual energy requirement is used to calculate the free fraction as defined in Equ. 3.1. The TRNSYS simulation yields $7.264 \cdot 10^9$ kJ/year. F-chart (Klein et al. (1994)) gives $7.273 \cdot 10^9$ kJ/year for the same load requirement, but different load profile. The F-chart result deviates from the TRNSYS result by 0.1%. The total annual energy requirement can also be estimated on basis of the VA hospital data as described in subsection 1.4.1. Assuming a gas price of 4.55 $/$10^6$ Btu for a commercial application as listed by the Wisconsin Energy Statistics 1995 (Wisconsin State Department of

Administration (1995)), the total energy requirement yields $9.126 \cdot 10^9$ kJ. Comparison with the TRNSYS result leads to a gas boiler efficiency of 0.8. This is a reasonable value. Hence, both, the F-chart and the VA hospital energy requirements, support the TRNSYS simulation very well.

### 3.4 Solar Domestic Hot Waster System

For purpose of validation, an annual TRNSYS simulation of the system as described in section 2.3 was run. Appendix B.2 shows the TRNSYS deck. The first twelve columns of the chart in Figure 3.1 show the monthly average daily auxiliary energy requirement from January through December according to the TRNSYS simulation. The last column shows the annual average daily energy requirement. The auxiliary energy requirement is higher during the winter months than during the summer months. The daily total energy requirement does not significantly vary over the course of the year. The annual average free fraction is 19.8% compared to 18.5% observed at the VA hospital. One reason, why the TRNSYS simulation yields a higher free fraction than observed with the VA hospital, is, that the TRNSYS model does not count for any pipe and duct losses. In a hospital, where the domestic hot water is served by a single central unit, long pipes and ducts with high heat losses are inevitable.

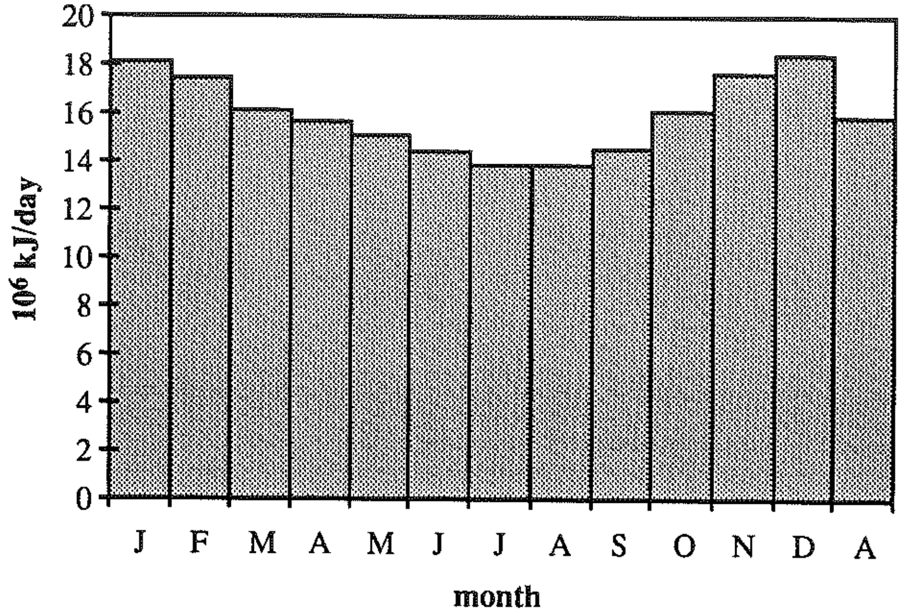



Figure 3.1 SDHWS monthly average daily auxiliary energy requirement

The result of another annual TRNSYS simulation was compared to an F-chart simulation. The F-chart model {R22} assumes a vertical one node solar storage tank. An instantaneous heater as back-up heats the water to the desired temperature. In the TRNSYS simulation, the solar storage tank is modeled by a Type 4 tank with one node. The auxiliary heater is modeled by Type 6 heater. The TRNSYS simulation yields a free fraction of 18.1%, whereas the F-chart simulation yields only 16.0%. The deviation is comparatively large. According to Duffie/Beckman 1990 {R6}, the F-chart model is generally rather conservative. Furthermore, the F-chart program was developed for residential size systems with residential water draw than for rather commercial applications.

After the validation, annual simulations with different parameter settings for collector area and tank volume per collector area unit were run. Appendix B.2 shows the TRNSYS deck. Collector area and solar storage tank size are the most significant system parameters which influence the free fraction. The collector area was varied in a range from 500 to 3,000 $m^2$. For areas larger than 3,000 $m^2$, the average monthly solar storage tank temperature exceeds 70 °C from March through September. This fact indicates, that a lot of energy is dumped. Duffie and Beckman (1991) recommend a ratio of tank volume to collector area unit in a range from 40 $l/m^2$ to 100 $l/m^2$. The VA hospital system was operated at a tank volume to collector area ratio of 77 $l/m^2$ which is within the recommended range. Simulations were run for the lower and upper limit of recommended ratios. Table 3.1 lists the simulation runs. The tank height was adjusted to the volume according to A.O. Smith Corporation catalog (1995). The last column shows the estimated free fractions. Figure 3.10 summarizes the free fractions in a graphical way in comparison with the free fractions obtained for the other systems. which were investigated. As expected, the free fraction increases with increasing collector area. With 3,0000 $m^2$, a state of saturation is reached. The collector area has more impact on the free fraction than the tank size. For small collector areas, the tank size hardly effects the free fraction, at all, because the solar heated water is instantaneously used. For large collector areas, the larger tank increases the free fraction by about 10%. With large collector areas, the water is heated more than required during the day, and this energy is used during the night.

Table 3.1 Simulation runs solar domestic water heating system

| # coll. [-] | tank vol. [gal] | tank ht. [ft] | area [$m^2$] | vol./area [$l/m^2$] | free fraction [-] |
|---|---|---|---|---|---|
| 300 | 5,000 | 6.8 | 488 | 40 | 0.182 |
| 600 | 10,000 | 8.8 | 975 | 40 | 0.338 |
| 900 | 15,000 | 9.7 | 1463 | 40 | 0.469 |
| 1,200 | 20,000 | 11.4 | 1951 | 40 | 0.585 |
| 1,500 | 25,000 | 12.1 | 2439 | 40 | 0.667 |
| 1,800 | 30,000 | 12.9 | 2926 | 40 | 0.721 |
| 300 | 12,500 | 9.2 | 488 | 100 | 0.195 |
| 600 | 25,000 | 12.1 | 975 | 100 | 0.359 |
| 900 | 40,000 | 14.7 | 1463 | 100 | 0.498 |
| 1,200 | 50,000 | 16.1 | 1951 | 100 | 0.623 |
| 1,500 | 65,000 | 17.8 | 2439 | 100 | 0.717 |
| 1,800 | 80,000 | 19.6 | 2926 | 100 | 0.788 |

### 3.4 Heat Pump Water Heating System

Originally, a full design capacity heat pump water heating system was assumed which has the capability to meet the entire hot water demand (see chapter 2.4). Using a full capacity heat pump, energy is saved in comparison to an electric resistance heater, as long as the heat pump COP is larger than unity. On the other hand, a full capacity heat pump means high initial costs.

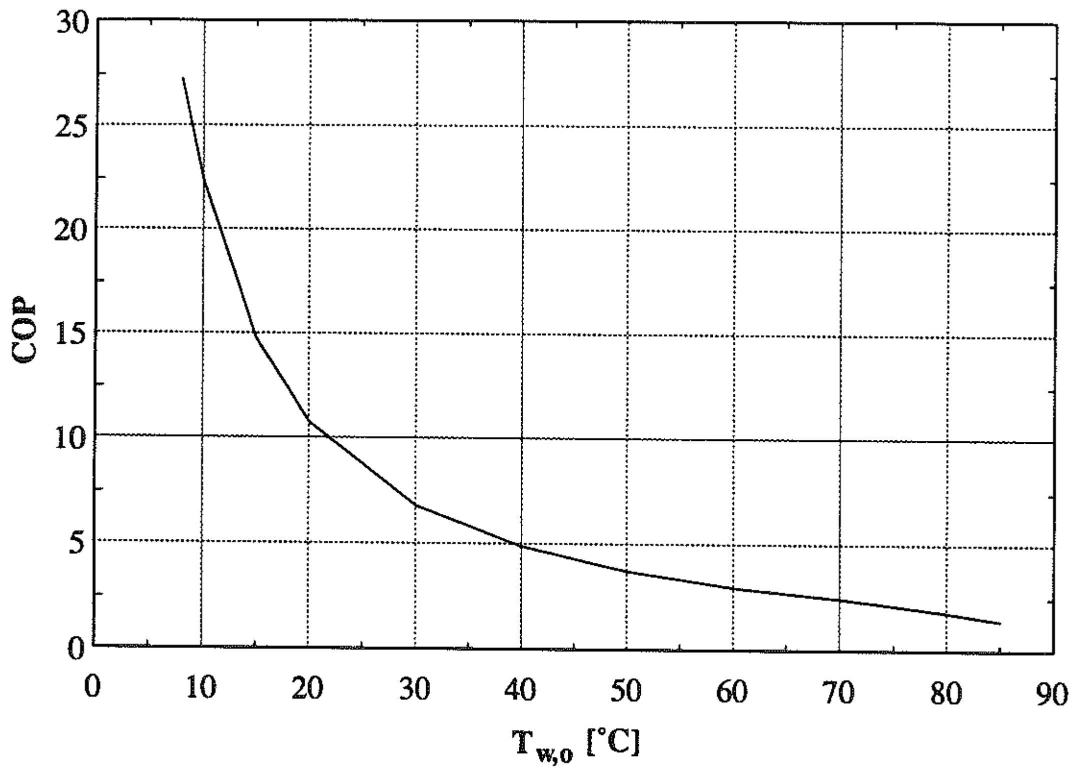



Figure 3.2 COP vs. $T_{w,o}$ for $T_{s,i}$=7 °C

Another effect, which has already been shown in Figure 2.12, is that the COP increases with decreasing heat pump outlet temperature. Figure 3.2 illustrates this phenomenon more clearly. The figure shows the COP for different heat pump water outlet temperatures, but with the same source inlet temperature and heat pump water inlet temperature. With the heat pump water outlet temperature approaching 7 °C, which is the same as the source temperature, the COP reaches theoretically infinity. With the heat pup outlet temperature reaches high values, the COP approaches unity. This phenomenon has the following effect on the energy savings: Reducing the heat pump design capacity, and maintaining the condenser mass flow rate, lowers the heat pump outlet temperature, and thus, increases the COP. Hence, reducing the heat pump capacity by a certain proportion means that the energy savings are reduced by a smaller proportion.

These considerations led to the conclusion to investigate systems with different heat pump design capacities. The systems with heat pumps smaller than full design capacity use the heat pump as a pre-heater. The pre-heated water is backed up by an electric auxiliary heater as used with the solar domestic water heating system.

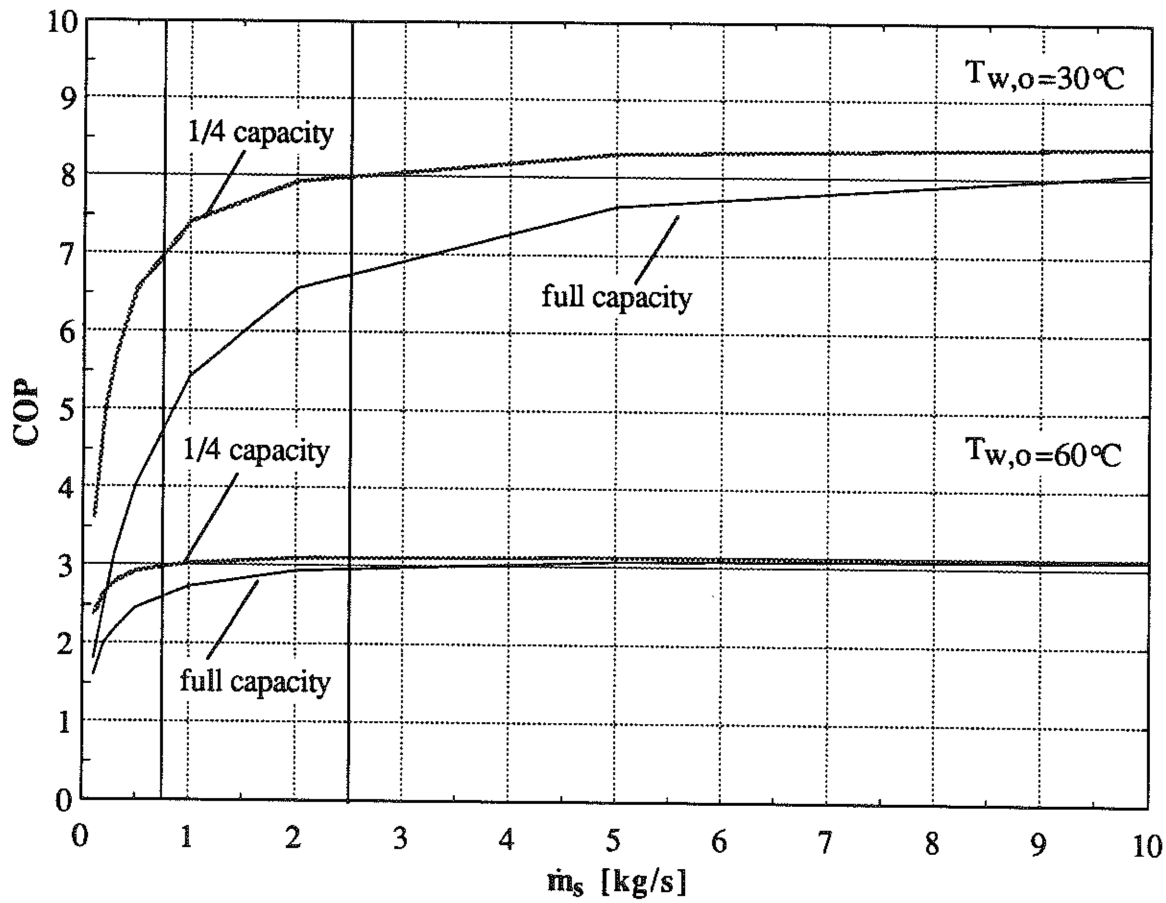



Figure 3.3 COP vs. $\dot{m}_s$ for different design capacities, usual conditions

An easy way to implement systems with different design capacities in TRNSYS is using the full capacity performance data file for Type 42 as described in subsection 2.4.1 (and also shown in Appendix A.2), multiplying the delivered heat by a heat pump design capacity fraction, and hence, keeping the COP performance identical to the full design capacity. Figure 3.3 shows the COP vs. the source mass flow rate for full and quarter capacity. The plain vertical lines illustrate that the COP of the quarter capacity heat pump for a mass flow rate 0.625 kg/s does not significantly deviate from the COP of a full capacity heat pump at 2.5 kg/s mass flow rate, which is four times as much. This fact justifies the simplification done above.

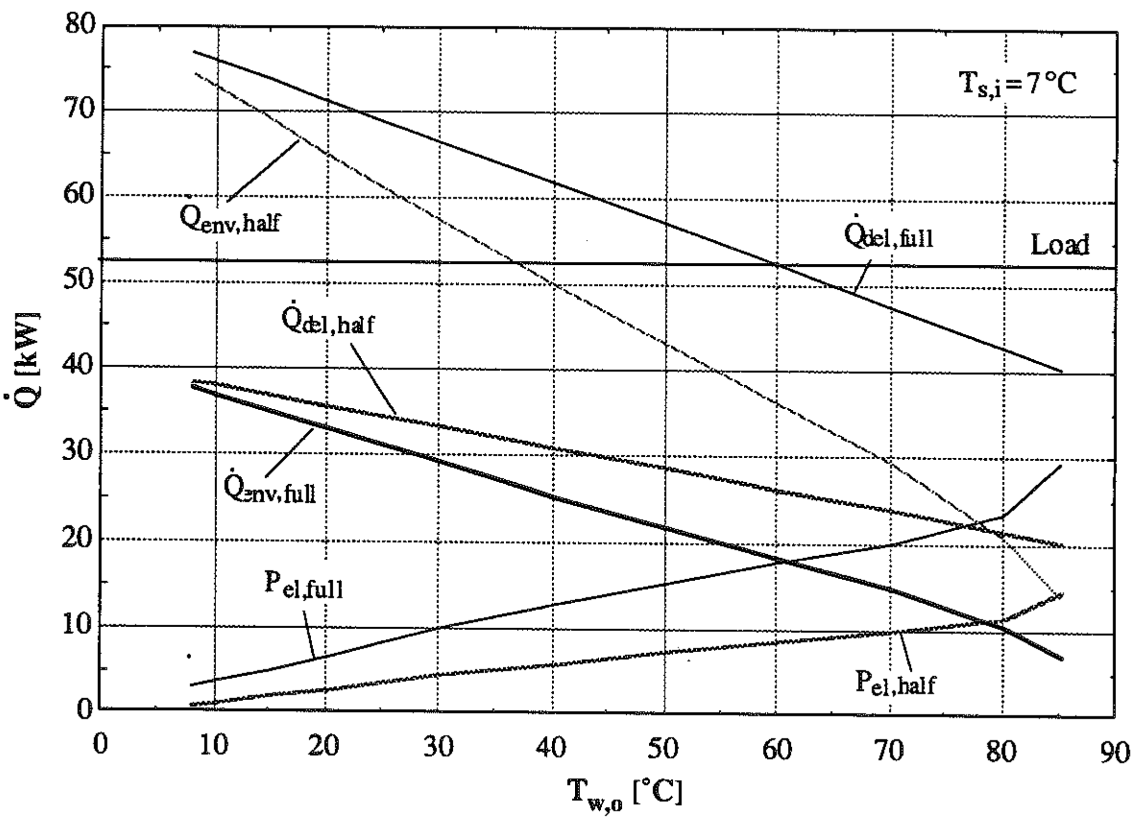



Figure 3.4 Energy flows on heat pump vs. $T_{w,o}$ for full and half capacity

Figure 3.4 shows the energy flows for heat pumps of full and half design capacity, and constant source temperature and heat pump inlet temperature. The full capacity heat pump meets the load of 53 kW for $T_{w,o}$=60 °C. The half capacity heat pump meets only half of the lad at each condition. The energy delivered by the source $Q_{env}$, and the compressor energy $P_{el}$ are half, as well. The delivered heat increases with decreasing $T_{w,o}$.

Rearranging Equ. 1.9 for the heat pump outlet temperature as pointed out n section 2.4.2, and assuming for this time constant heat pump inlet temperature, and condenser mass flow rate, which leads to steady state conditions, one sees that for each capacity, a balance temperature is reached. Hence, the lower the capacity, the lower the balance temperature. And, the lower the balance temperature, and thus, the heat pump outlet temperature, the higher the COP.

The TRNSYS simulations are not steady state simulations for the following two reasons: The heat pump inlet temperature varies slightly, which is related to the mains temperature. And, the condenser mass flow rate switches between day and night, which causes delays due to the controls system. These two factors cause only slight differences in daily system performance over the course of a year. Therefore, investigating the different heat pump water heating systems, this paper presents only annual daily-average energy consumption.

Annual TRNSYS simulations according to section 2.4 were run for heat pump design capacity fractions $f_{cap}$ from 1 to 0.03125 (=1/32). Appendix B.3 shows the TRNSYS deck. Figure 3.5 compares the annual average daily energy consumption of the heat pump compressor and the auxiliary heater for heat pump design capacity fractions ranging from 1 to 0.03125. For full capacity ($f_{cap}$=1), the auxiliary heater requirement is zero. With decreasing capacity fraction, the auxiliary heater energy consumption increases, whereas the compressor energy consumption decreases. The horizontal line at $20 \cdot 10^6$ kJ/day, which frames the chart is the average annual daily total energy requirement, i.e. at a heat pump fraction equals zero, the energy requirement is entirely met by a heater.

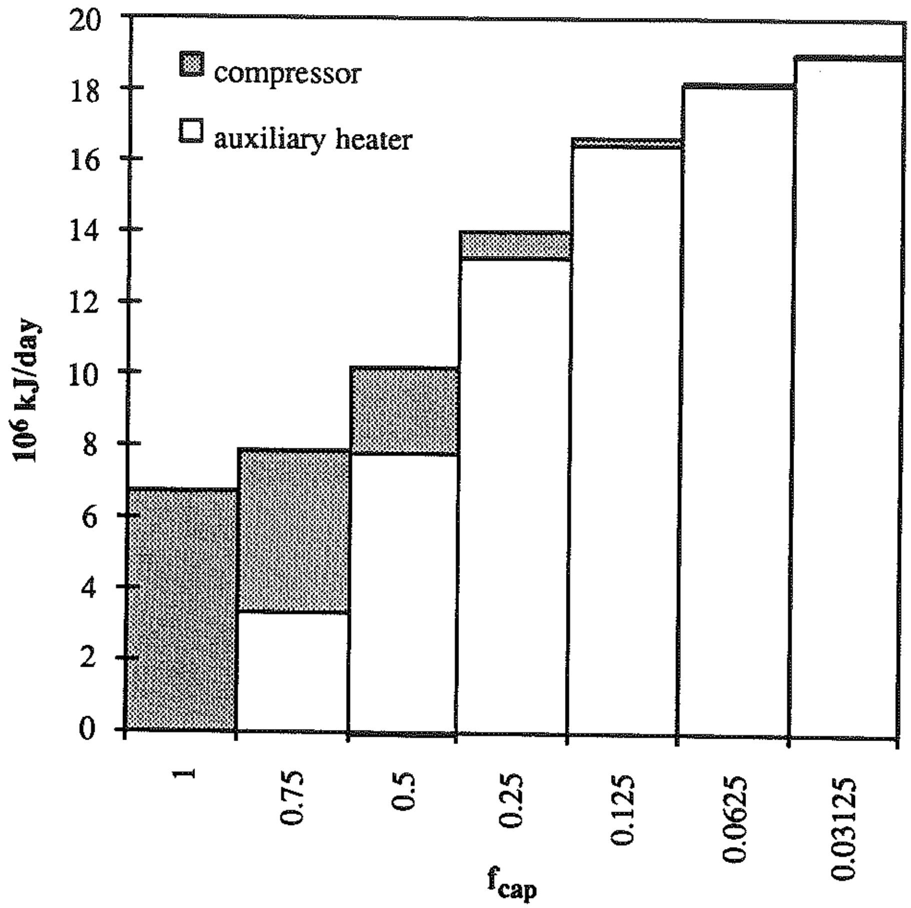



Figure 3.5 Annual average daily energy usage for different heat pump capacities

Figure 3.6 shows the resulting free fraction vs. the heat pump capacity fraction. The black dots indicate the capacity fractions for which simulations were run. With full capacity, the free fraction is 65.9%. The smaller the heat pump capacity fraction, the smaller the free fraction.

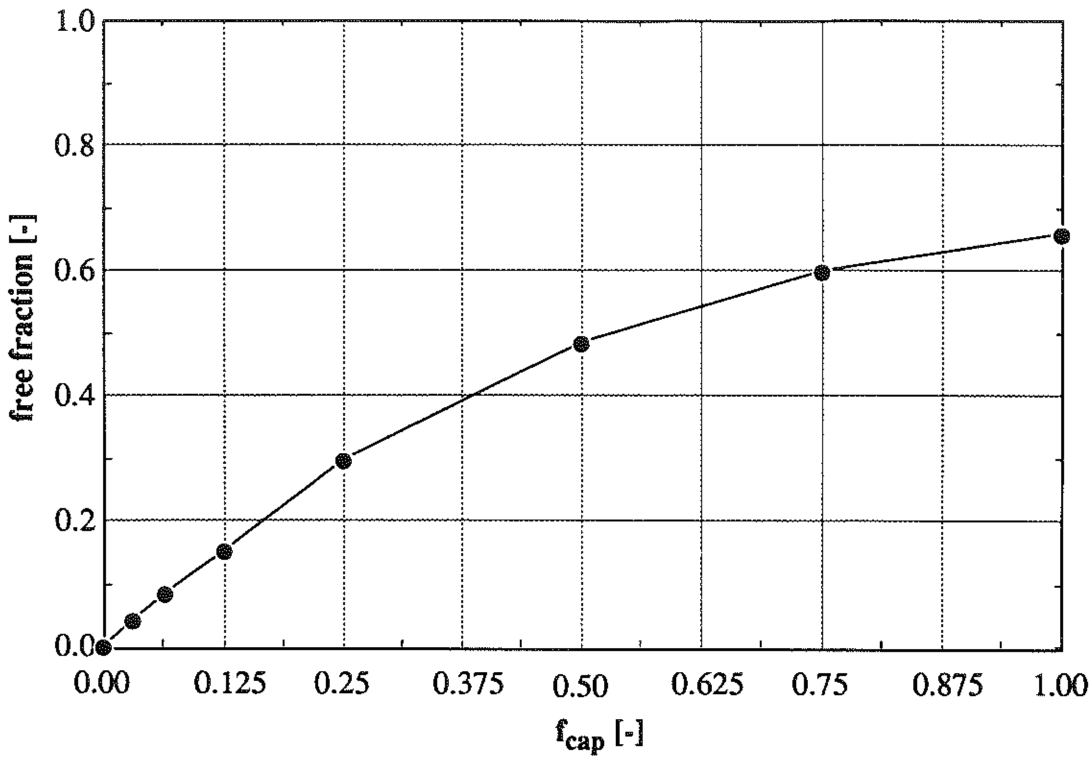



Figure 3.6 Free fraction vs. heat pump capacity fraction

Figure 3.7 compares the obtained heat pump and system COPS for different heat pump capacity fractions The heat pump and system COPS were calculated according to Equ. 1.5 and 1.19, respectively, neglecting parasitic energy requirement. With increasing heat pump fraction, the average annual heat pump COP decreases exponentially, whereas the average annual system COP increases almost linearly at a slope smaller than 1. For a capacity fraction equal 0.03125, the system COP is almost unity (bold horizontal line), as it was for the conventional system. For full capacity, the pump and system COP are both equal to 2.93. The curves intersect when the total added electric energy equals the compressor energy input.

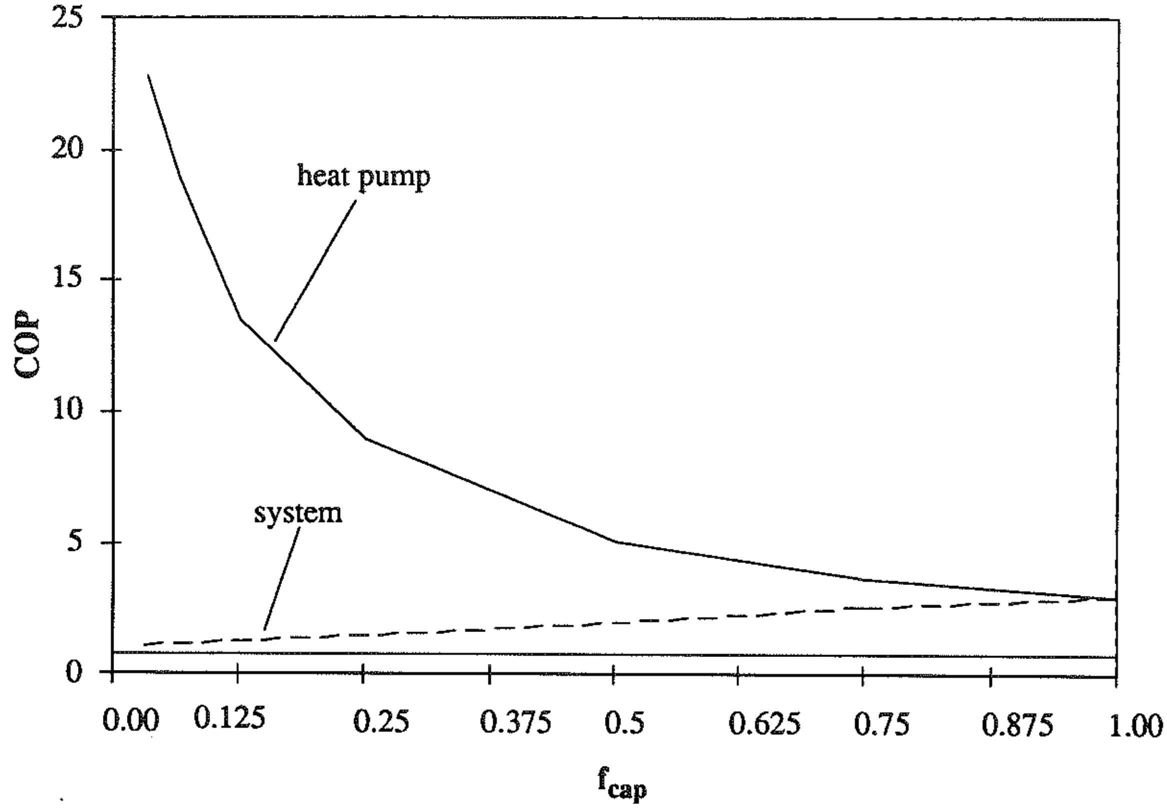



Figure 3.7 Heat pump and system COPS vs. heat pump capacity fraction

The effect of decreasing heat pump COP with increasing heat pump capacity fraction can also be illustrated by introducing a related free fraction. The related free fraction relates the free fraction at a certain capacity to the free fraction obtained by a full capacity system. The related free fraction is given by the ratio

$$f_{rel} = \frac{f}{f_{full}} \tag{3.4}$$

where f is the free fraction as defined in Equ.3.1, and $f_{full}$ is the free fraction obtained by the full capacity system, in this case 0.659. Figure 3.8 shows the related fraction vs. the heat pump capacity fraction. If there was not the effect of increasing heat pump capacity with decreasing heat pump outlet temperature, the heat pump fraction would be equals the related free fraction, i.e. the curve would be a straight line with slope=1. One sees that the related fraction is always equal to (at full capacity) or larger than the corresponding heat pump capacity fraction. This fact will be of specific significance for the economic analysis of chapter 4.

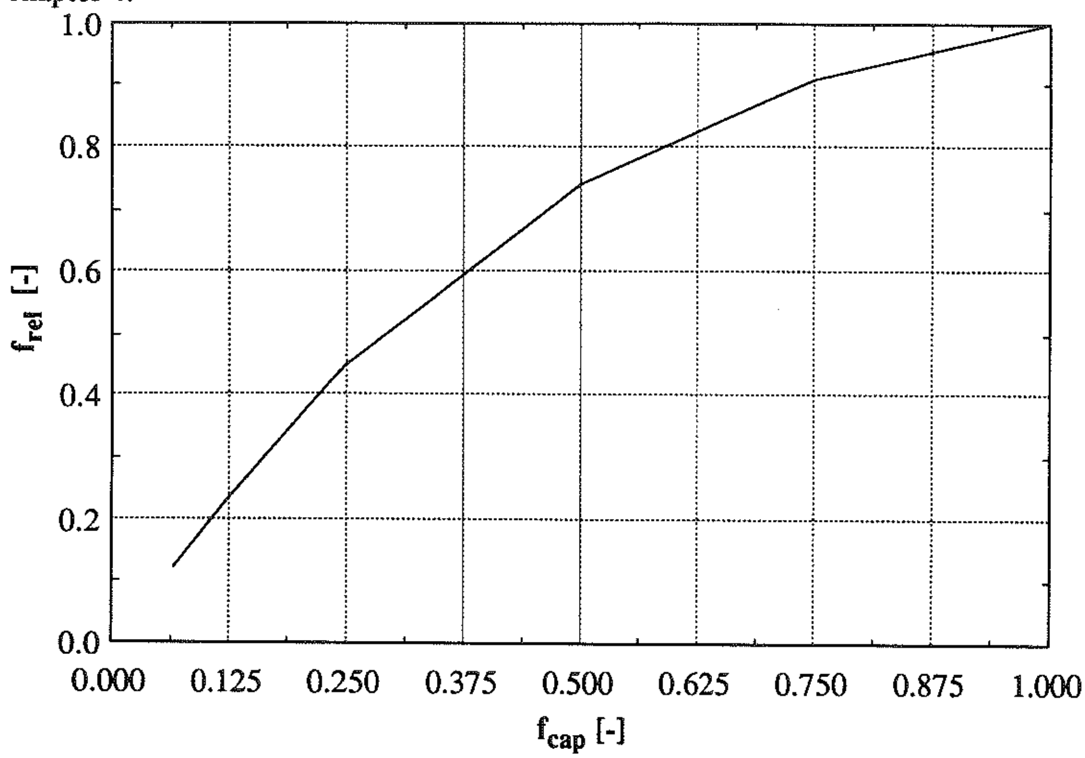



Figure 3.8 Related free fraction vs. heat pump capacity fraction

### 3.6 Parallel Solar Heat Pump Domestic Hot Water System

The parallel system is a combination of the solar domestic water heating system and the heat pump water heating system as described in section 2.5. Appendix B.3 shows the TRNSYS deck. Consequently, annual simulations were run varying the following three parameters: collector area, tank volume per collector area unit, and heat pump capacity.

Table 3.2 Simulation runs parallel solar heat pump water heating system

| # coll. [-] | tank vol. [gal] | $f_{cap}$ [-] | area [$m^2$] | vol./area [$l/m^2$] | free fraction [-] |
|---|---|---|---|---|---|
| 300 | 5,000 | 1.0 | 488 | 40 | 0.687 |
| 600 | 10,000 | 1.0 | 975 | 40 | 0.72 |
| 900 | 15,000 | 1.0 | 1463 | 40 | 0.748 |
| 1,200 | 20,000 | 1.0 | 1951 | 40 | 0.784 |
| 1,500 | 25,000 | 1.0 | 2439 | 40 | 0.821 |
| 1,800 | 30,000 | 1.0 | 2926 | 40 | 0.844 |
| 300 | 12,500 | 1.0 | 488 | 100 | 0.696 |
| 600 | 25,000 | 1.0 | 975 | 100 | 0.732 |
| 900 | 40,000 | 1.0 | 1463 | 100 | 0.773 |
| 1,200 | 50,000 | 1.0 | 1951 | 100 | 0.810 |
| 1,500 | 65,000 | 1.0 | 2439 | 100 | 0.849 |
| 1,800 | 80,000 | 1.0 | 2926 | 100 | 0.883 |
| 300 | 5,000 | 0.5 | 488 | 40 | 0.535 |
| 600 | 10,000 | 0.5 | 975 | 40 | 0.587 |
| 900 | 15,000 | 0.5 | 1463 | 40 | 0.629 |
| 1,200 | 20,000 | 0.5 | 1951 | 40 | 0.682 |
| 1,500 | 25,000 | 0.5 | 2439 | 40 | 0.734 |
| 1,800 | 30,000 | 0.5 | 2926 | 40 | 0.768 |
| 300 | 12,500 | 0.5 | 488 | 100 | 0.549 |
| 600 | 25,000 | 0.5 | 975 | 100 | 0.605 |
| 900 | 40,000 | 0.5 | 1463 | 100 | 0.657 |
| 1,200 | 50,000 | 0.5 | 1951 | 100 | 0.717 |
| 1,500 | 65,000 | 0.5 | 2439 | 100 | 0.769 |
| 1,800 | 80,000 | 0.5 | 2926 | 100 | 0.821 |
| 300 | 5,000 | 0.25 | 488 | 40 | 0.360 |
| 600 | 10,000 | 0.25 | 975 | 40 | 0.433 |
| 900 | 15,000 | 0.25 | 1463 | 40 | 0.492 |
| 1,200 | 20,000 | 0.25 | 1951 | 40 | 0.568 |
| 1,500 | 25,000 | 0.25 | 2439 | 40 | 0.638 |

| | | | | | |
|---|---|---|---|---|---|
| 1,800 | 30,000 | 0.25 | 2926 | 40 | 0.696 |
| 300 | 12,500 | 0.25 | 488 | 100 | 0.378 |
| 600 | 25,000 | 0.25 | 975 | 100 | 0.455 |
| 900 | 40,000 | 0.25 | 1463 | 100 | 0.541 |
| 1,200 | 50,000 | 0.25 | 1951 | 100 | 0.620 |
| 1,500 | 65,000 | 0.25 | 2439 | 100 | 0.698 |
| 1,800 | 80,000 | 0.25 | 2926 | 100 | 0.757 |

Table 3.2 lists the simulation runs. Collector area and tank volume were varied in the same range as the solar domestic water heating system simulation runs as listed in Table 3.1. Each of these parameter settings was run for three heat pump capacity fractions: full capacity, half capacity, and quarter capacity. Again, the last column of the table lists the obtained free fraction, and Figure 3.10 shows the free fractions in a graphical way in comparison with the free fractions obtained for the other systems which were investigated. As expected, the free fraction increases with increasing heat pump fraction, collector area, and tank volume per unit area. The tank volume effects the system performance at large collector areas more than observed with the solar domestic water heating system. The curves start at zero collector area with free fractions corresponding to those of the heat pump water heating systems. Again, with 3,000 m$^2$, a state of saturation is reached, and the monthly average solar storage tank temperature exceeds 70 °C from March through September.

Figure 3.9 gives more detailed TRNSYS simulation results for a system with the following parameters: full heat pump capacity, collector area of 2,439 m$^2$, and storage tank volume per collector area unit of 100 l/m$^2$. The first twelve columns show the monthly average daily auxiliary energy requirement from January through December. The last column shows the annual average daily energy requirement. The seasonal differences in auxiliary energy requirement are much smaller than for a solar domestic water heating system, because the heat pump performs steadily without regard to climate changes. The free fraction varies between 75% in January and 95% in July, and it yields an annual average of 85%. The horizontal line at $20 \cdot 10^6$ kJ/day, which frames the chart, is the average annual daily total energy requirement.

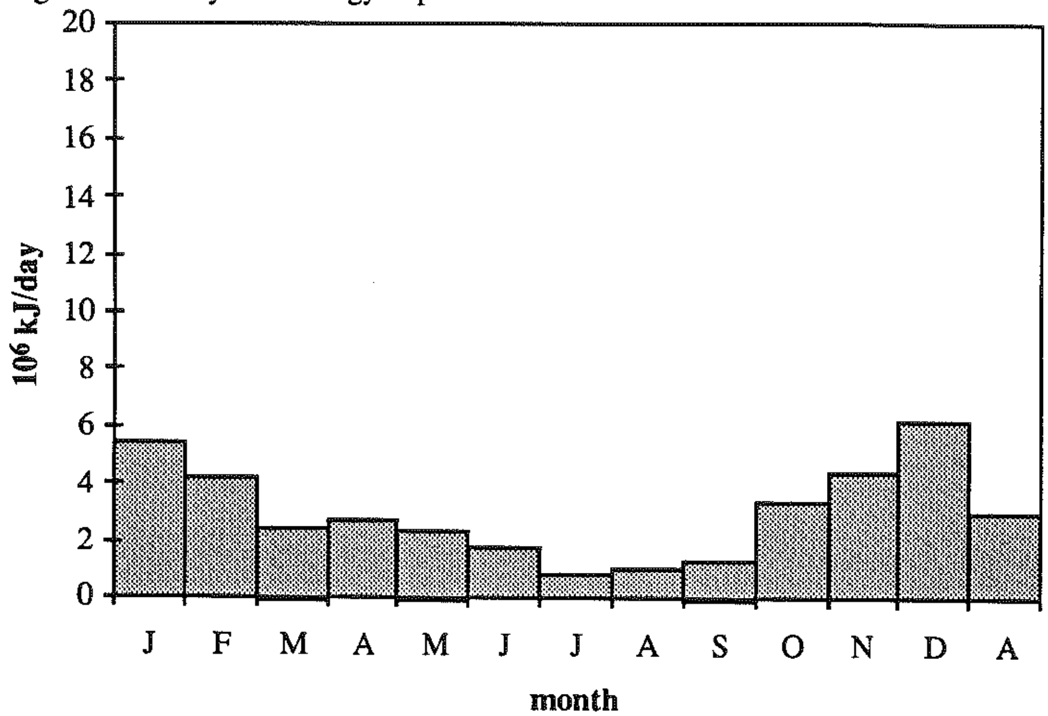



Figure 3.9 PSHPS monthly average daily auxiliary energy requirement

### 3.7 Comparison of Systems

Figure 3.10 gives an overview of the simulation results. The chart shows the free fraction vs. the collector area. Parameters are the heat pump capacity fraction, and the tank volume per collector area unit, as described in the preceding sections.

The chart shows the course of the free fractions of three heat pump water heating systems: *full capacity, half capacity, and quarter capacity. The course of the free fractions* is represented by the dashed, horizontal, straight lines. The free fraction is independent of the collector area, and higher for the larger the heat pump capacity.

Furthermore, the chart shows there the free fractions obtained by the simulation of the solar domestic water heating system depend on the collector area. The course of the free fraction of the small tank solar domestic water heating system (labeled solar 40) is represented by the solid, bold line starting at zero free fraction for zero collector area. The course of the solar fraction of the large tank solar domestic water heating system (labeled solar 100) is represented by the solid, thin line starting at zero free fraction for zero collector area.

Last, the chart shows the dependency of the free fractions for the parallel solar heat pup water heating system on the collector area. Three heat pump capacity fractions are considered corresponding to the heat pump water heating systems, which were investigated. Two tank sizes were considered corresponding to the solar domestic water heating systems, which were investigated. The course of the free fractions is represented by the dotted-dashed lines. The lines start at zero collector area at the free fraction o the corresponding heat pump water heating system. The bold lines represent the small tank system (labeled parallel x-40), and the thin lines represent the large tank system (labeled parallel x-100). x indicates the heat pump capacity fraction.

In terms of the free fraction, the solar domestic water heating system can only compete with a heat pump system for areas larger than 2,100 $m^2$, and 2,400 $m^2$ for high and low tank volume, respectively. The heat pump water heating systems without solar-assistance yield very high fractions With increasing collector area, the free fraction of the full capacity parallel system increases with a smaller slope than the free fraction of the solar domestic water heating system. The larger the collector area is, the smaller is the difference between parallel and solar domestic water heating system. Prejudging the economic analysis of the following chapter, for parallel systems, large collector areas seem to make no sense.

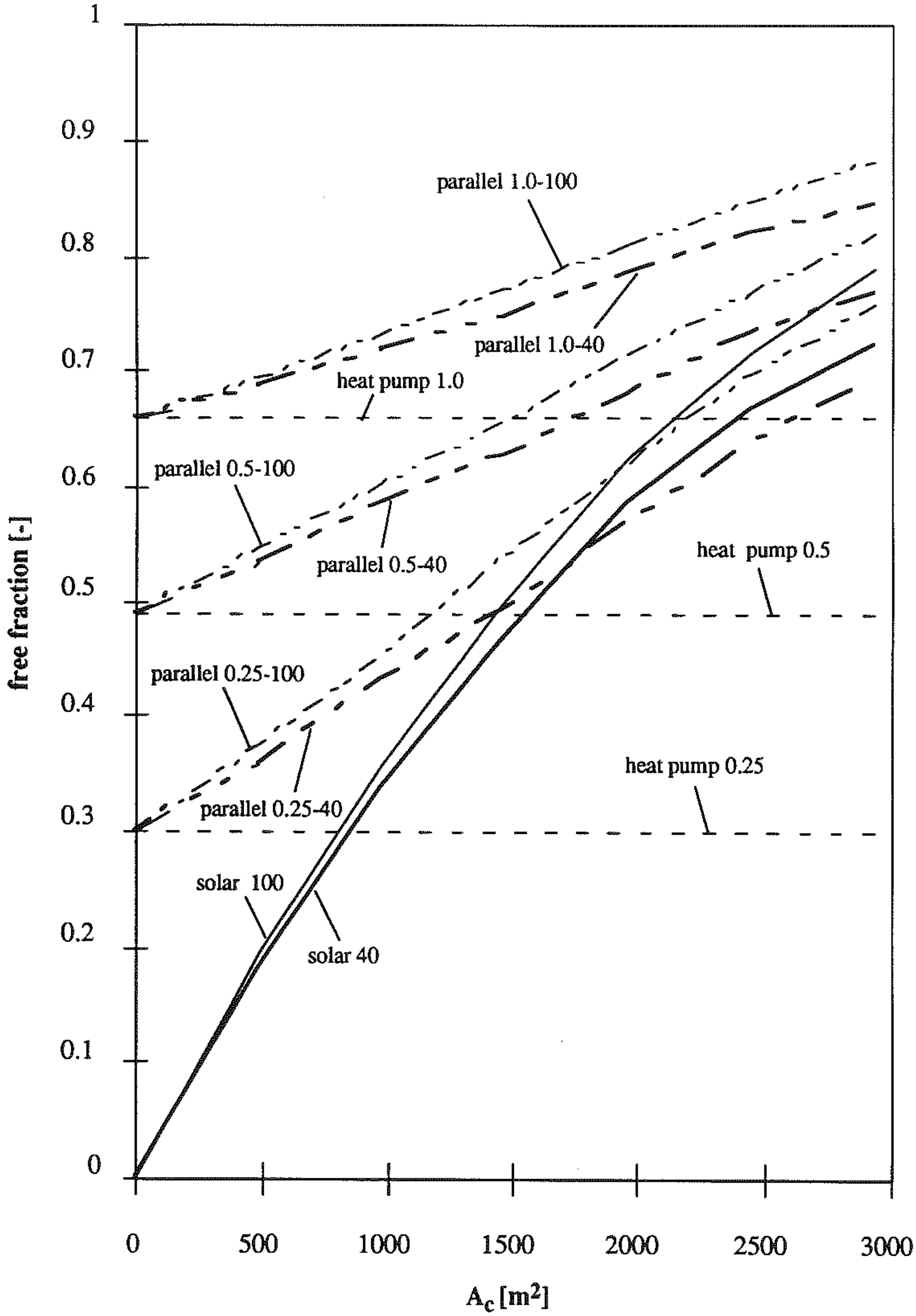



Figure 3.10 Free fraction vs. collector area

# CHAPTER FOUR

## ECONOMICS

In this chapter, the systems, which were modeled and simulated as described in the preceding two chapters, will be analyzed with regard to their economic performance.

Investigating the economic feasibility of alternative systems is important, because these systems have to compete on the market with reliable conventional systems. The decision in favor or opposed to a certain system, is usually based on cost savings rather than on resource and pollution savings.

The economic analysis, executed in context with this study, is a rough estimation. The purpose of the analysis is to give a guideline to which system configuration and with which parameter settings shows the best economic performance assuming different economic conditions. The settings of collector area, tank size, and heat pump capacity are the driving factors of both, the initial costs and the free fraction, which influences the annual costs in terms of energy costs significantly. The economic conditions include assumptions for equipment costs, for electricity cost, and for general market conditions.

The first section of this chapter will briefly introduce the method of economic analyses, the $P_1$,$P_2$ method. The following sections will analyze the systems, investigated in this study. The last section will compare the systems and discuss the results. Appendix A.3 shows the EES program which was written to execute the economic analysis.

## 4.1 $P_1$,$P_2$ Method

The $P_1$,$P_2$ method was introduced by Duffie and Beckman (1991). The $P_1$,$P_2$ method is a quick and convenient way of doing life cycle savings analysis of solar process systems. Life cycle savings (LCS) is defined as the difference between the life cycle costs of a conventional system and the life cycle cost of the solar plus auxiliary energy system. Life cycle cost (LCC) is the sum of all the costs associated with an energy delivery system over its life time or over a selected period of analyses. LCC are expressed in today's dollars, and take into account the time value of money.

Using the nomenclature of this paper, the LCS can be written as

$$LCS = P_1 \cdot c_{el} \cdot Q_{conv} \cdot f - P_2 \cdot c_s \tag{4.1}$$

The first part of the right side expresses the energy cost savings, and the second part stands for the additional initial costs. $P_1$ is the ratio of the life cycle electric cost savings to the first-year electric cost savings, and $P_2$ is the ratio of the life cycle expenditures incurred because of the additional capital investment to the initial investment. $c_{el}$ is the first year's electrical cost in \$/kWh. $Q_{conv}$ is the total annual load in kWh. f is the free fraction as defined in Equ. 3.1. $c_s$ is the total cost in \$ of the installed solar energy and heat pump equipment above the cost of the conventional energy system.

$P_1$ depends on assumptions for the following economic parameters: period of economic analysis, income tax rate, inflation, and discount rate. $P_2$ depends on these same parameters but, in addition, depends on the down payment, mortgage interest rate, resale value, and other investment related parameters. For purpose of simplification, $P_1$ and $P_2$ were not explicitly calculated, but two settings were assumed: $P_1$ = 10/$P_2$ = 1, and $P_1$ =

$5/P_2 = 1$. The first setting is typical of residential situations, the second setting is typical for commercial applications as recommended by Mitchell and Braun (1996). The higher $P_1$, the greater the energy cost savings, and hence, the better the economic performance. As the costs of commercial energy use are tax deductible, $P_1$ for commercial applications is lower than for residential ones.

The economic analysis was executed assuming two different first year's electric costs for operating the auxiliary resistance heater and the heat pump compressor: $c_{el} = 0.06$ \$/kWh, and $c_{el} = 0.10$ \$/kWh. The first value represents the current average Wisconsin electric rate for commercial application according to Wisconsin Energy Statistics 1995 (Wisconsin State Department of Administration (1995)). The second value assumes a potential future higher rate, which is desirable to promote solar energy technology application.

Combining the settings of $P_1$, $P_2$, and $c_{el}$, Table 4.1 shows the three economic scenarios which were investigated. The first row number set is the best case in favor of solar energy use with high $P_1$ and $c_{el}$. The last row number set is the worst case regarding solar energy use with low $P_1$ and $c_{el}$. The middle row number set is in-between the two extreme scenarios.

Table 4.1 Economic scenarios

| Scenario | $P_1$ | $P_2$ | $c_{el}$ [\$/kWh] |
|---|---|---|---|
| best | 10 | 1 | 0.10 |
| medium | 5 | 1 | 0.10 |
| worst | 5 | 1 | 0.06 |

The value of the total annual load $Q_{tot}$ is taken from the conventional system (see chapter 3.3), and is $2.018 \cdot 10^6$ kWh. Hence, the conventional system, which is the base system for the LCS, is a system whose free fraction and additional initial costs are zero.

The value of the free fraction f of a specific system is the free fraction obtained by the corresponding simulation. The values are shown in sections 3.4 to 3.7. The free fraction can be dependent on the collector area, the tank size, and the heat pump capacity.

The total cost $c_s$ is composed of collector-area-dependent costs and area-independent costs as follows:

$$c_s = c_A \cdot A_c + c_{hp} \cdot Cap \cdot f_{cap} \tag{4.2}$$

The first summand expresses the collector-area-dependent costs of the solar domestic water heating system. $A_c$ is the collector area in $m^2$. $c_A$ includes the costs in $\$/m^2$ of both collectors and tank, and is composed of

$$c_A = (c_C + c_T \cdot v_T) \tag{4.3}$$

$c_C$ is the collector cost per collector area unit in $\$/m^2$, $c_T$ is the tank cost per volume unit in \$/liter, and $v_T$ is the ratio of tank volume to collector area in $l/m^2$ as introduced in chapter 3.4. In this study collector and tank costs are considered together. The second summand of Equ. 4.2 represents the area-independent costs of the heat pump system: $c_{hp}$ is the heat pump cost per unit of capacity at standard conditions in \$/kW. Standard conditions are $T_s$ = 100 °F (or 38 °C), and $T_{hp,o}$ = 140 °F (or 60 °C), and they are taken from Fedders Solar Product Company performance catalog. Cap is the design capacity at standard conditions, which totals 490 kW for the seven heat pumps according to Figure 2.11. $f_{cap}$ is the heat pump capacity fraction as introduced in chapter 3.5. During the economic analysis, these parameters, determining $c_s$, were varied as will be described in the following sections of this chapter. For the solar domestic water heating system, only the first summand of Equ. 4.2 is relevant. For the heat pump water heating system, only the second summand is relevant. For the parallel system, both the first and the second summand are relevant. Other additional costs than for collectors, tank, and heat pump are not considered. It is assumed that other additional costs as for controls, piping etc. are small compared to the cost drivers mentioned above.

The $P_1$,$P_2$ method was applied in two ways: In a first step, the economic break-even was determined. Setting Equ. 4.1 equals zero and rearranging for $c_s$ leads to

$$c_s = \frac{P_1}{P_2} \cdot c_{el} \cdot Q_{conv} \cdot f \tag{4.4}$$

Knowing the break-even of equipment cost, gives a rough idea about the economic feasibility of a system.

In a second step, equipment costs $c_s$ were assumed, and the resulting LCS were calculated according to Equ. 4.4. This approach allows determining optimum system parameters by maximizing the LCS at given costs.

Due to the large system scale, it is difficult to determine a realistic cost range of $c_A$ and $c_{HP}$. References like the Means Mechanical Cost Data (1995) or the Solar Resource Guide (Poplawski (1993)) focus on residential size systems. Estimates by distributors and suppliers are difficult to obtain for a system of this scale. On the one hand, manufacturing and installation costs should be lower than with a residential size system. But on the other hand, large scale systems are rather customized solutions. Taking these considerations into account, it was decided to do the LCS analysis for a range of $c_A$=100 to 400 \$/m$^2$, and $c_{HP}$=250 to 1,000 \$/kW.

## 4.2 Solar Domestic Hot Water System

Applying Equ. 4.4 for the solar domestic hot water system, and using the nomenclature introduced in the preceding section, leads to allowable collector-area-dependent costs as following

$$c_A = \frac{P_1}{A_C \cdot P_2} \cdot c_{el} \cdot Q_{conv} \cdot f \qquad (4.5)$$

For any given economic scenario, the allowable collector-area-dependent costs are only a function of the collector area $A_c$, because the free fraction is also only a function of the collector area. The free fractions at corresponding collector areas were obtained from the results given in chapter 3.4. Also, according to chapter 3.4, two ratios of tank volume to collector area were analyzed.

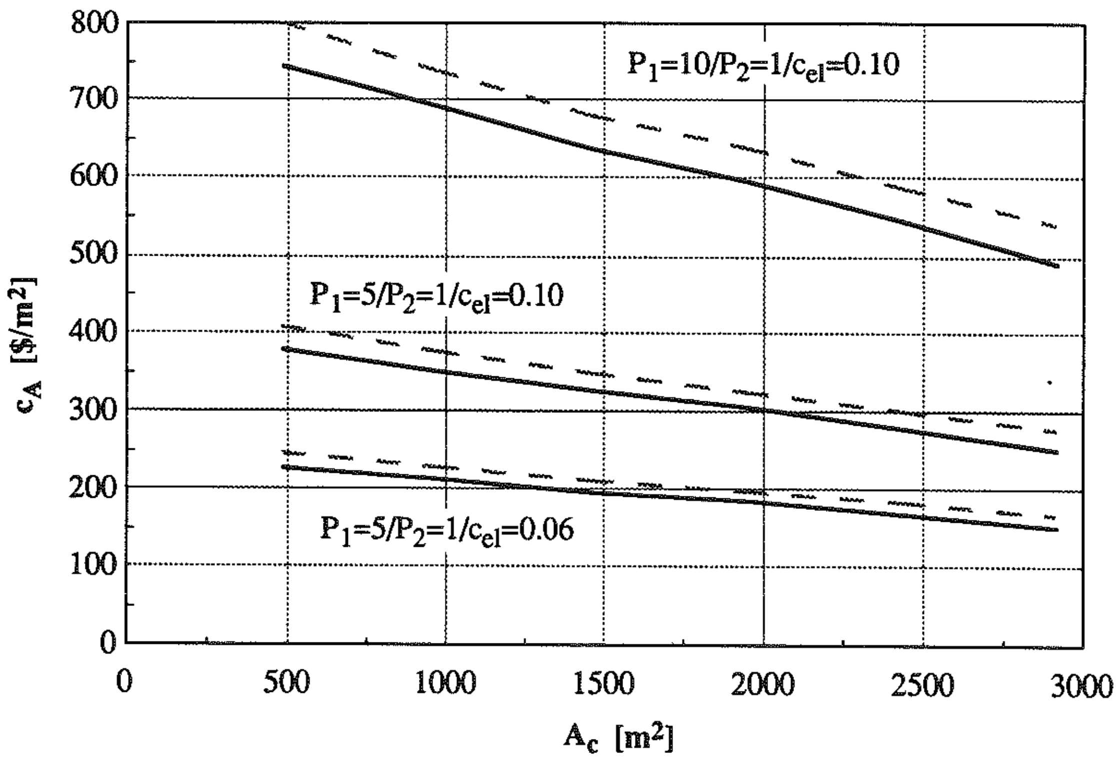



Figure 4.1 Allowable area-dependent costs vs. collector area for SDHWS

Figure 4.1 shows the allowable area-dependent costs vs. the collector area. Three economic scenarios were investigated according to Table 4.1. In each case, the allowable collector cost decreases with increasing collector area. Considering the worst economic scenario, the allowable collector cost has a range from about 250 \$/m$^2$ for a 500 m$^2$ area, to 150 \$/m$^2$ for 3,000 m$^2$. The plain lines characterize the systems with 40 l/m$^2$ tank volume, and the dashed lines characterize the systems with 100 l/m$^2$ tank volume. The 60 l/m$^2$ larger tank volume allows an additional cost of 25 to 50 \$/m$^2$ dependent on the economic scenario. This results in an allowable tank cost of 0.42 to 0.84 \$/l. Means Mechanical Cost Data (1995) gives 0.50 to 2.00 \$/l depending on volume and quality.

LCS

The LCS were estimated applying Equ. 4.1 on the SDHWS. Only the small tank system was investigated as a function of collector areas. As mentioned above, the large tank system allows an additional investment, but the general economic performance does not significantly differ from the small tank system. The total cost $c_s$ given by Equ. 4.2 is only composed of the first summand, the collector-area-dependent costs $c_A$. The LCS were estimated for three values of $c_A$: \$ 100, \$ 200, and \$ 400 (see preceding subchapter).

Figure 4.2 compares the LCS in 1000 \$ vs. the collector area for these three values of $c_A$. The solid bold curves represent the best economic scenario, the dashed ones the medium, and the solid plain ones the worst scenario. The curves start at the origin of the axes system. Zero collector area corresponds to the conventional system, which is the base system, and consequently has zero LCS.

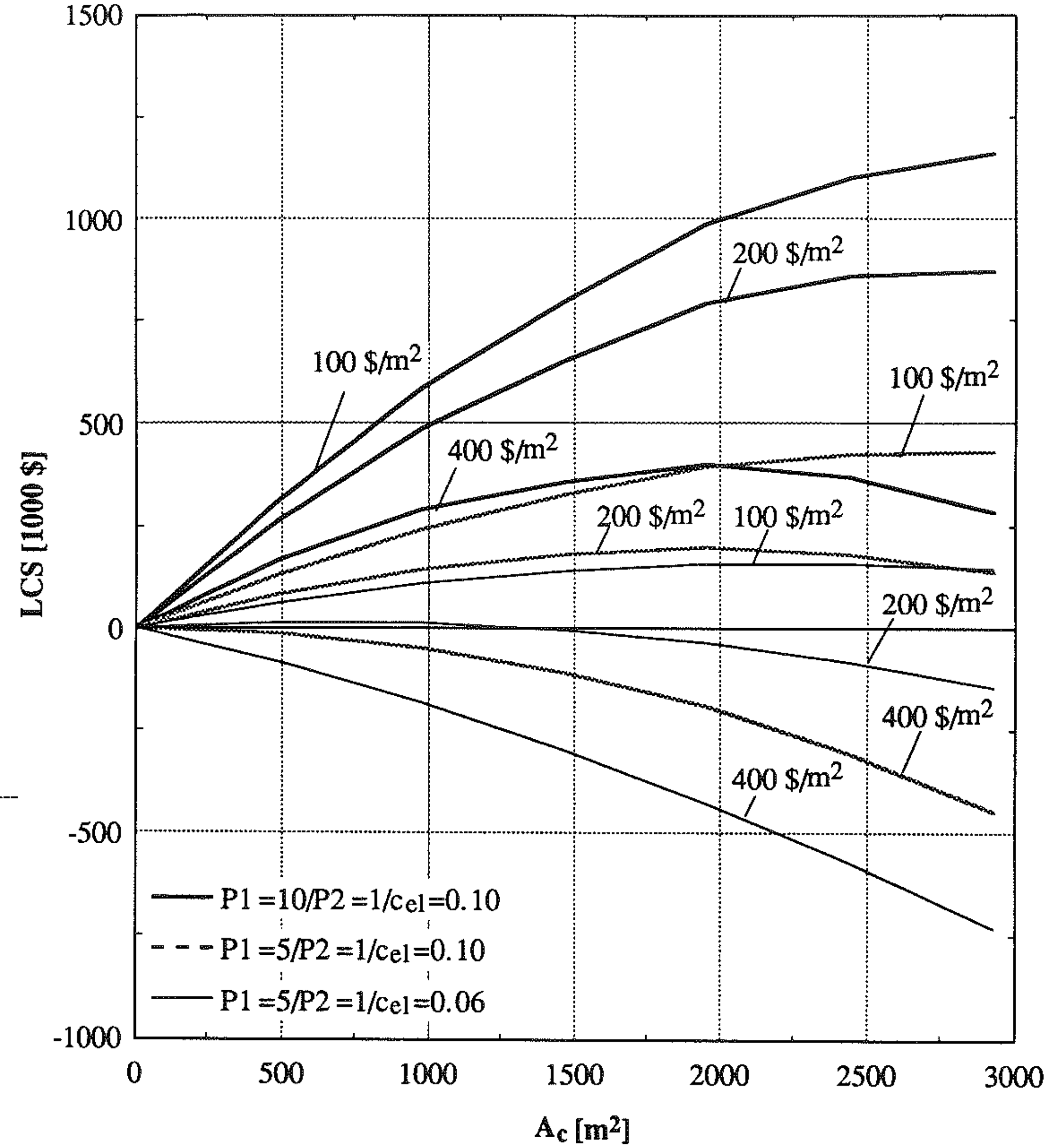



Figure 4.2 LCS vs. collector area for SDHWS

The location of the maximum LCS as a function of the collector area can be analytically determined by differentiating Equ. 4.1 with respect to $A_c$, and setting it equals zero:

$$\frac{\partial LCS}{\partial A_c} = 0 \tag{4.6}$$

Executing the derivation in Equ. 4.6, using Equ. 4.1, and rearranging for $\frac{\partial f}{\partial A_c}$ leads to:

$$\frac{\partial f}{\partial A_c} = \frac{P_2 \cdot c_A}{P_1 \cdot c_{el} \cdot Q_{conv}} \tag{4.7}$$

$\frac{\partial f}{\partial A_c}$ is the derivative of the free fraction with respect to the collector area, i.e. the gradient of the SDHWS curve f vs. $A_c$ shown in Fig. 3.10. The larger the collector area, the smaller the gradient $\frac{\partial f}{\partial A_c}$ gets. There is no maximum LCS at all, if the value of the right side of Equ. 4.7 is higher than the maximum gradient at $A_c$ equal to zero. If there is any maximum, the following can be concluded: The lower the collector cost is, and the better the economic scenario is, the larger the collector area at maximum LCS is.

The amount of maximum LCS shows the same dependencies as the collector area: The lower the collector cost is, and the better the economic scenario is, the higher the maximum LCS is.

Considering the best economic scenario, the SDHWS pays for each investigated collector-area dependent cost. The optimum collector area is between 2,000 $m^2$ and 3,000 $m^2$, and the LCS are between \$ 300,000 and \$ 1,200,000, respectively. The medium economic scenario yields smaller LCS. $c_A$=\$ 400 is at the borderline of economic feasibility, since the LCS barely get positive, at all. And, for areas larger than about 300 $m^2$, the LCS is negative, which coincides with Figure 4.1. As far as the worst economic scenario is concerned, the borderline of economic feasibility is already reached at $c_A$=\$ 200, where the LCS also barely reaches a positive value.

### 4.3 Heat Pump Water Heating System

Applying Equ. 4.4 for the heat pump water heating system, and using the nomenclature introduced in the preceding section, leads to allowable heat pump costs as following

$$c_{HP} = \frac{P_1}{Cap \cdot f_{cap} \cdot P_2} \cdot c_{el} \cdot Q_{conv} \cdot f \tag{4.8}$$

For any given economic scenario, the allowable heat pump costs are only a function of the heat pump capacity fraction $f_{cap}$, because, again, the free fraction is also only a function of the capacity fraction. The free fractions at corresponding capacity fractions were obtained from the results given in chapter 3.5. Figure 4.3 shows the allowable heat pump costs vs. the heat pump capacity fraction. Three economic scenarios were investigated according to Table 4.1. In each case, the allowable collector cost decreases with increasing heat pump capacity fraction. At first glance, this result seems surprising. Considering the worst economic scenario, the allowable heat pump cost has a range from about 1,700 $/kW for a 0.01325 fraction to 800 $/kW for full capacity. But, the LCS at given heat pump cost are not proportional to these values. If the heat pump capacity fraction is low, the heat pump capacity is low. The LCS, resulting from the equal difference between actual and maximum allowable heat pump cost per kW, are higher the larger the total heat pump capacity is.

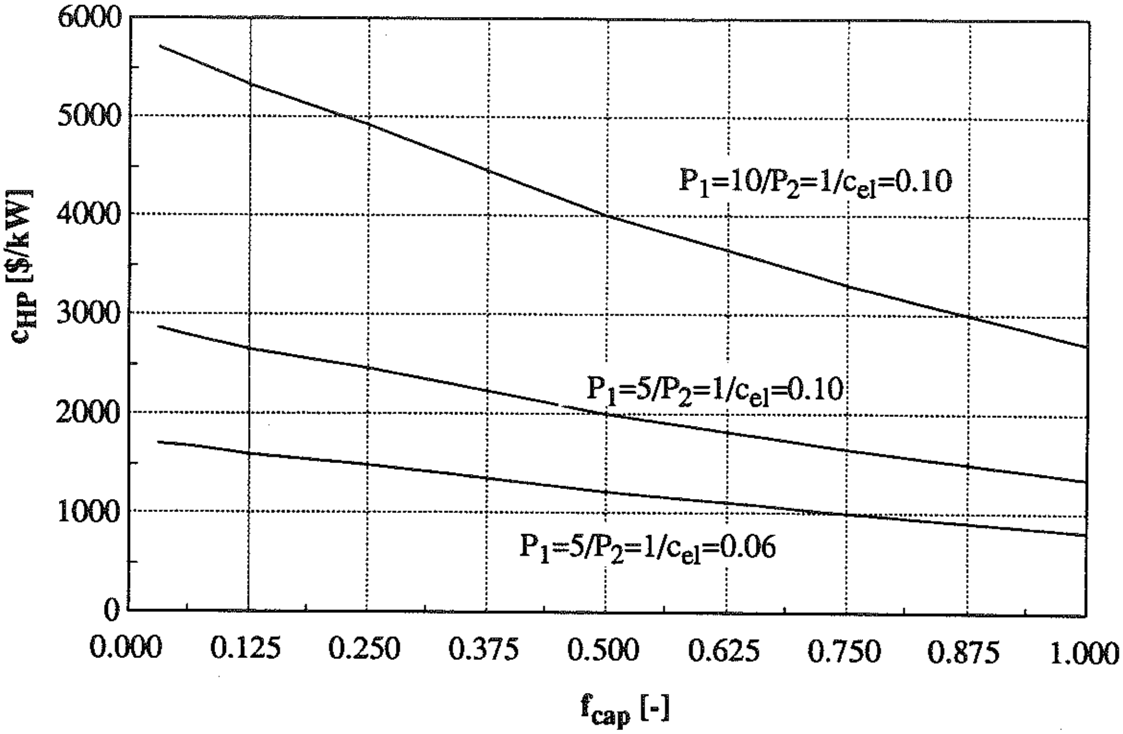



Figure 4.3 Allowable heat pump costs vs. capacity fraction for HPWHS

LCS

The LCS were estimated applying Equ. 4.1 on the HPWHS. The total cost $c_S$ given by Equ. 4.2 is only composed of the second summand, the heat pump costs $c_{HP}$. The LCS were estimated for three values of $c_{HP}$: 250 $/kW, 500 $/kW, and 1000 $/kW (see subchapter 4.1). Figure 4.4 compares the LCS in 1000 $ vs. the heat pump capacity factor for these three values of $c_{HP}$. The analyzed economic scenarios, the general course of the curves, and the way of presentation is identical to the SDHWS of the preceding subchapter, and especially Figure 4.2. The location of the maximum LCS on the x-axis, i.e. the heat pump capacity factor at maximum LCS can be analytically determined substituting in Equ. 4.7, $A_c$ for $f_{cap}$, and $c_A$ for $c_{HP}$. Consequently, the maximum LCS of the HPWHS show the same performance as the maximum LCS of the SDHWS as described in the preceding subchapter.

The best economic scenario reaches the highest LCS. The peak is at full capacity for $c_{HP}$=250 \$/kW and 500 \$/kW, and is 0.75 for $c_{HP}$=1000 \$/kW. The maximum LCS are between \$1,200,000 and \$800,000. The medium economic scenario yields significantly lower LCS. Again, the peak is at full capacity for $c_{HP}$=250 \$/kW and 500 \$/kW, and is 0.5 for $c_{HP}$=1000 \$/kW. The LCS are between \$ 500,000 and \$ 200,000. Assuming the worst economic scenario, the LCS still reach a positive maximum in each case. The maximums are found at full capacity for $c_{HP}$=250 \$/kW, at 0.75 for 500 \$/kW, and at 0.25 for $c_{HP}$=1000 \$/kW. Hence, with increasing heat pump costs per kW, the optimum heat pump capacity fraction decreases. This observation corresponds to the course of the allowable $c_{HP}$ as shown in Figure 4.3. The maximum LCS are between \$280,000 and \$60,000.

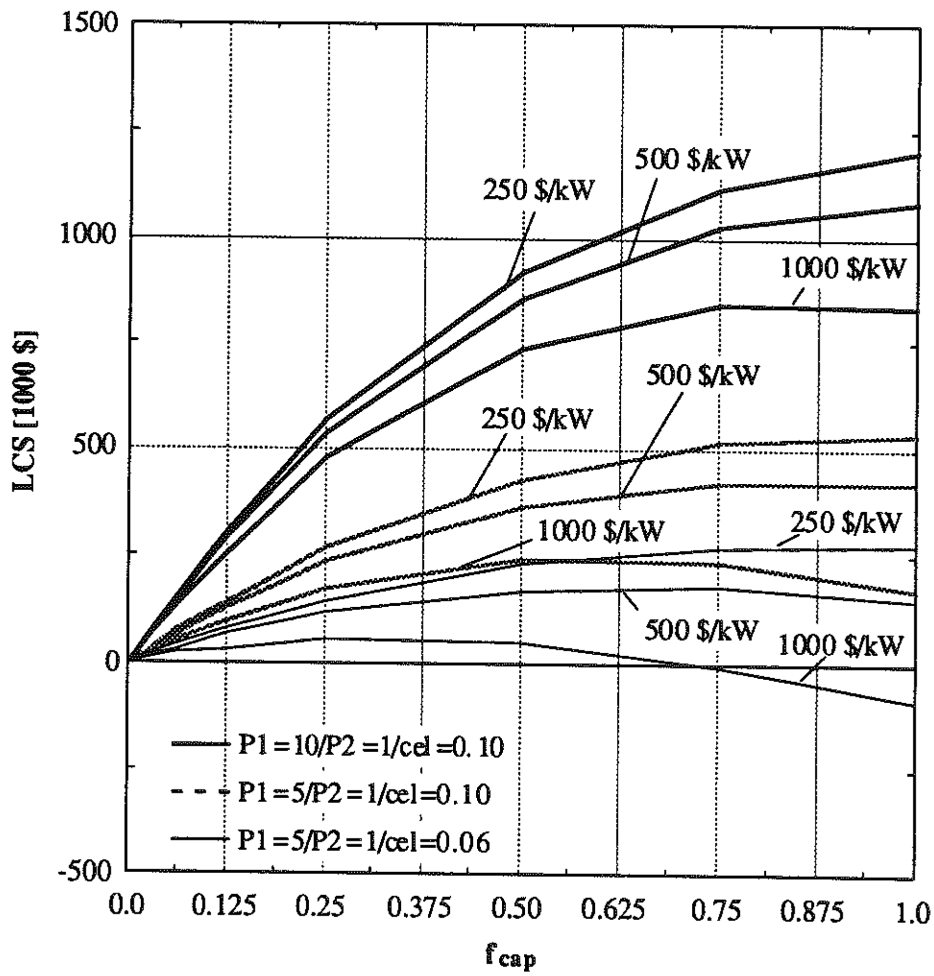



Figure 4.4 LCS vs. heat pump capacity factor for HPWHS

### 4.4 Parallel System

Investigating the parallel system on allowable costs is more complex than for the SDHWS and the HPWHS, because collector-area-dependent and independent costs are involved. As this study focuses on solar assistance and for purpose of simplification, only the allowable area-dependent costs $c_A$ were investigated at three given heat pump costs $c_{HP}$ per kW. Applying Equ. 4.2 and 4.4 for the parallel system, and using the nomenclature introduced in subchapter 4.1, leads to allowable collector-area dependent costs as follows:

$$c_A = \frac{1}{A_C}\left(\frac{P_1}{P_2} c_{el} \cdot Q_{conv} \cdot f - c_{HP} \cdot Cap \cdot f_{cap}\right) \tag{4.9}$$

For any given economic scenario, the allowable collector-area-dependent costs are a function of the collector area $A_c$, the heat pump capacity fraction fcap, and the heat pump costs $c_{HP}$. The performance of $c_A$ is hard to predict. On the one hand, the free fraction in the minuend of Equ. 4.9 is larger the larger $A_c$ and $f_{cap}$ are. On the other hand, $f_{cap}$ is part of the subtrahend, and $A_c$ is the denominator of Equ. 4.9. The free fractions at corresponding collector and heat pump capacities were obtained from the results given in chapter 3.6. Also, according to chapter 3.6, two ratios of tank volume to collector area were analyzed. Figure 4.5 shows the allowable area-dependent costs vs. the collector area. Three heat pump capacity fractions were investigated: The left plot is for full, the middle one for half, and the right one for a quarter capacity. Three economic scenarios and heat pump costs per kW were investigated according to Table 4.1 and the preceding subchapter, respectively. For purpose of clarity, the figure shows only the results for the small storage systems. The large storage systems allow an additional cost very similar to the SDHWS (see subchapter 4.2). Generally, the allowable area-dependent-costs decrease with increasing collector area in a hyperbolic way. The higher the related heat pump costs, the lower the allowable $c_A$. The larger the heat pump capacity the wider the range of $c_A$. The subtrahend is larger and has more weight. In the case of the full capacity system, the allowable cost drops to negative for the worst economic scenario and the highest considered heat pump cost. The curve is flipped vertically in relation to the other curves. With the worst economic scenario, and $c_{HP}$ being 500 \$/kW, $c_A$ has a range from 350 \$/m$^2$ to 90 \$/m$^2$ with increasing collector area, and from 320 \$/m$^2$ to 120 \$/m$^2$, for full and quarter capacity, respectively. Half capacity is in-between these ranges.

The allowable $c_A$ of the parallel system is more sensitive to the area than the one of the SDHWS, because of the higher free fraction. But, as discussed in the preceding section, the allowable cost is not an indicator for the best economic performance for the case where the actual area-dependent cost is lower than the maximum allowable cost. The succeeding section will compare the different systems in a more detailed way.

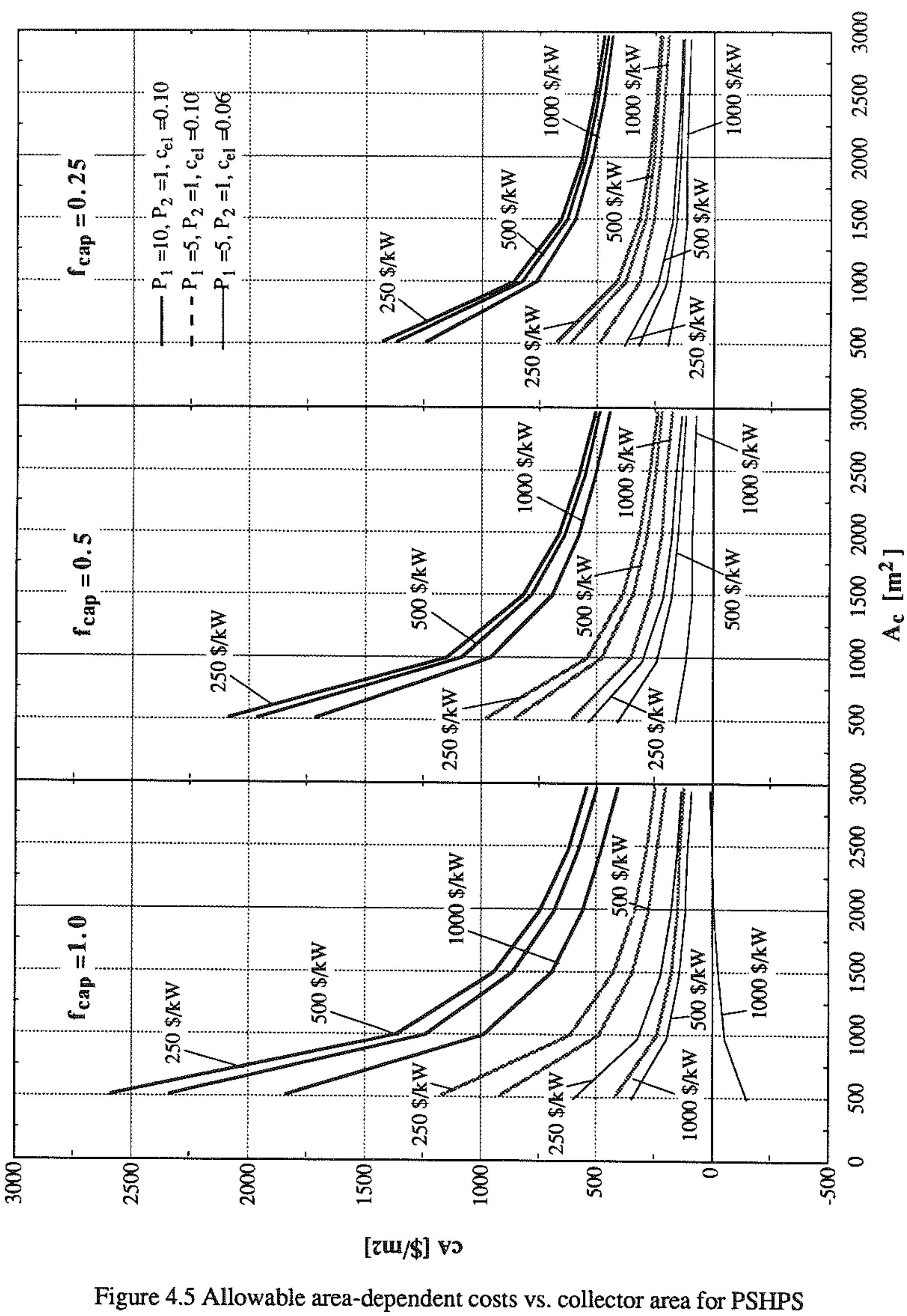



Figure 4.5 Allowable area-dependent costs vs. collector area for PSHPS

LCS

The LCS were estimated applying Equ. 4.1 to the PSHPS. The total cost $c_s$ is given by Equ. 4.2. The LCS were estimated for a combination of the three values of area-dependent and the three values of area-independent costs as used with the SDHWS (chapter 4.2) and the HPWHS (chapter 4.3), respectively. Figures 4.6 to 4.8 compare the LCS in 1000 \$ vs. the collector area $A_c$. Again, the collector area was chosen to be the independent variable because the focus of this study is on solar. The curve for $A_c$=0 represents the HPWHS. Each figure shows the LCS for one area-dependent cost $c_A$: Figure 4.6 for $c_A$=100 \$/m$^2$, Figure 4.7 for $c_A$=200 \$/m$^2$, and Figure 4.8 for $c_A$=400 \$/m$^2$. Like Figure 4.5, each figure is divided into three sections: the left one shows the performance of a parallel system with full heat pump capacity, the middle one a system with half capacity, and the right one with quarter capacity. Each plot shows curves for three related heat pump costs and three economic scenarios identical to those pointed out in the preceding paragraph.

The location of the maximum LCS on the x-axis, i.e. the collector area at maximum LCS can be analytically determined using Equ. 4.7. The location of the maximum on the x-axis is not a function of $c_{HP}$, though the value of the maximum LCS is. On the plots, parallel curves illustrate this phenomenon. With the maximum being located at $A_c$=0, the HPWHS shows better economic performance than any corresponding parallel system. This is the case for $c_A$=400 \$/m$^2$ (Figure 4.8). The increased free fraction cannot economically justify the high additional costs for solar collectors. The LCS drops far below zero. With

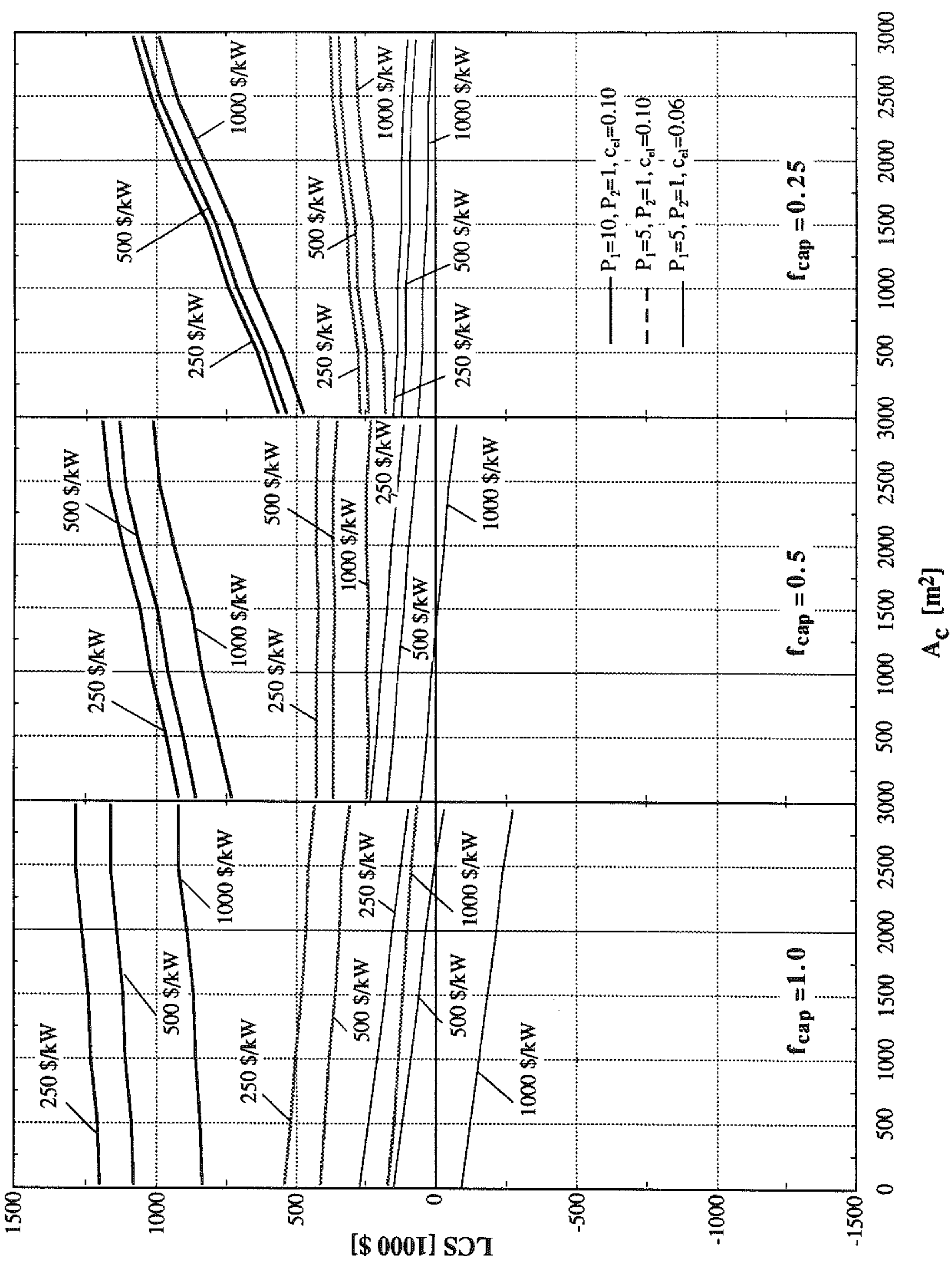



Figure 4.6 LCS vs. $A_c$ at $c_A$=100 \$/m$^2$ for PSHPS

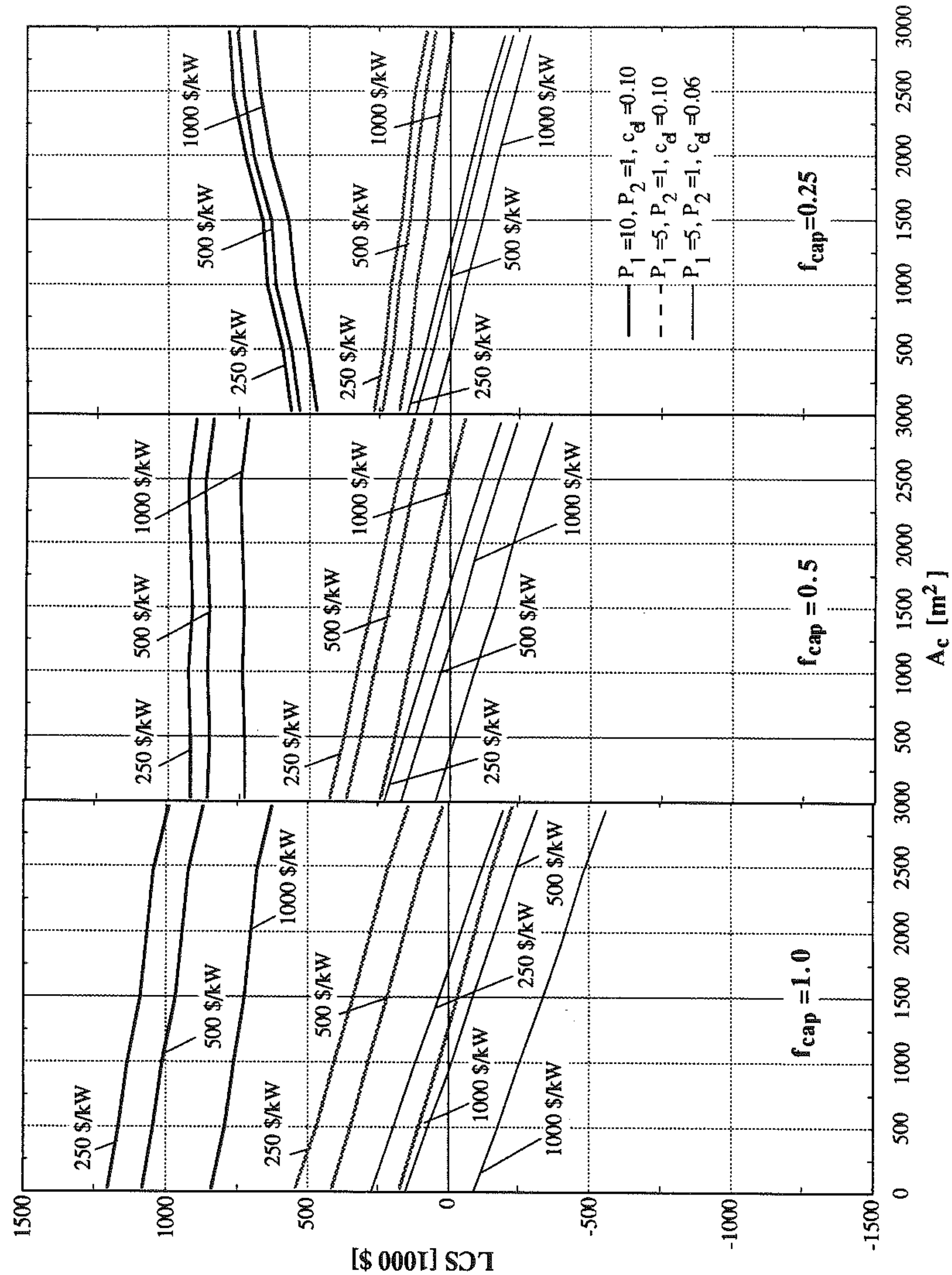



Figure 4.7 LCS vs. $A_c$ at $c_A$=200 \$/m$^2$ for PSHPS

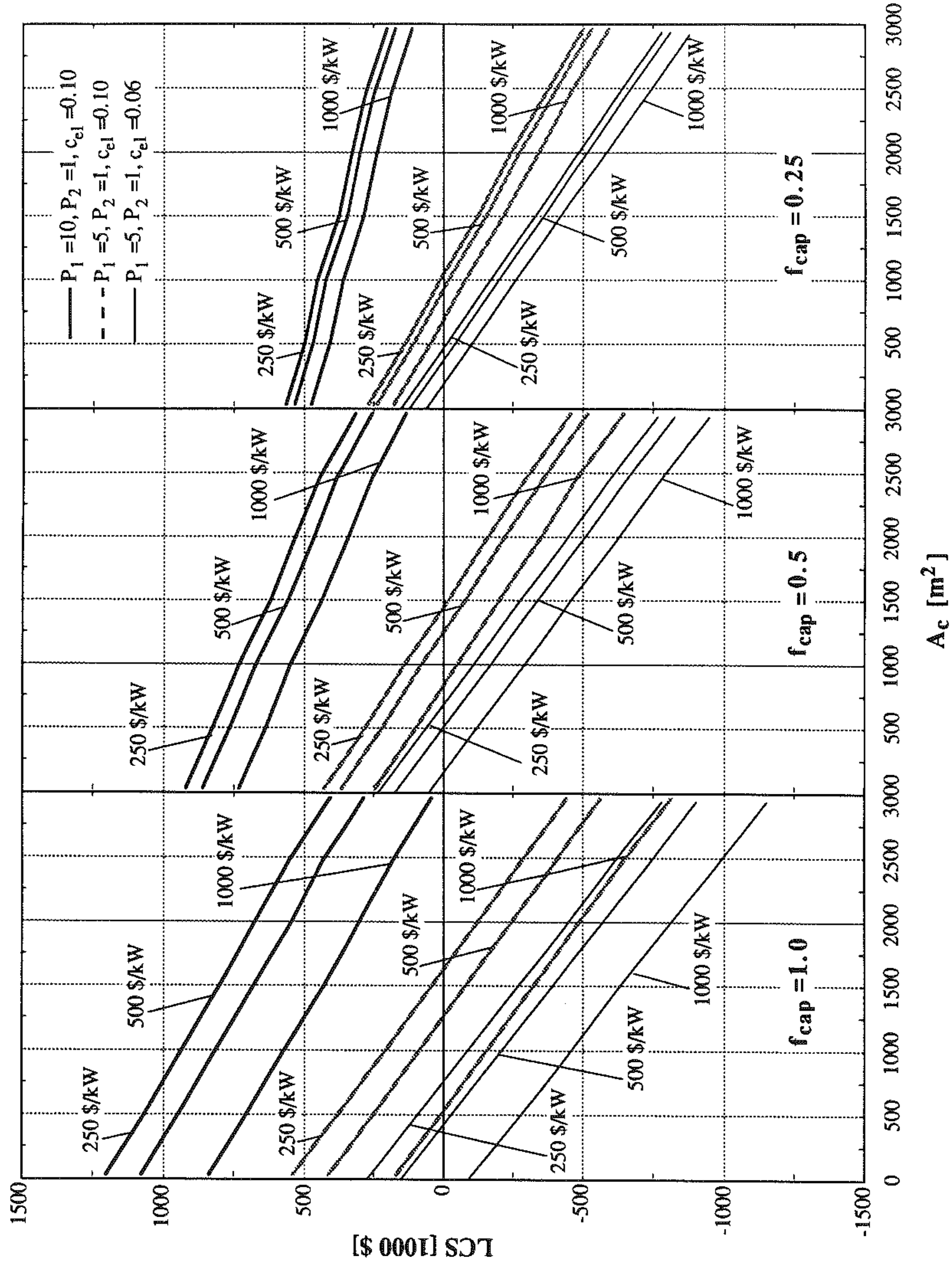



Figure 4.8 LCS vs. $A_c$ at $c_A$=400 $/m$^2$ for PSHPS

$c_A$=200 \$/m$^2$ (Figure 4.7), the parallel system can only compete with the HPWHS at quarter capacity and best economic assumptions. The economic benefits due to the higher free fraction overcompensate the additional equipment costs. The maximum LCS are reached at $A_c$=3000 m$^2$ and yield around \$ 750,000. The value of $c_{HP}$ shows only minor effect on the LCS, because the heat pump costs are small compared to the collector area-dependent costs. At half capacity and best economic scenario, the LCS are nearly independent of collector area over the entire area range. Higher free fraction and higher additional investment balance each other. With $c_A$=400 \$/m$^2$ (Figure 4.6), the parallel system can compete with the HPWHS under best economic assumptions. The maximum LCS is between \$ 1,300,000 and \$ 900,000, and between \$ 1,100,000 and \$ 1,000,000 for the full capacity system and the quarter capacity system, respectively. The half capacity system yields maximum LCS in-between. Regarding the medium economic scenario, only the quarter capacity system can compete with the HPWHS, yielding \$ 300,000 to \$ 400,000 maximum LCS.

## 4.5 Comparative Discussion

After having investigated the economic performance of each system type separately in the preceding sections, the question remains which system is recommendable under given specific economic conditions. Optimum system selection from an economic point of view can be approached in different ways. The profit-oriented approach selects the system which is reaches the maximum LCS. The environment-oriented approach selects the system which yields maximum free fraction at zero LCS. A hospital administration might go for the later option, because a hospital is not as profit-oriented run as a commercial corporation.

This section investigates both the profit-oriented and the environment-oriented approach. The LCS were obtained from subchapters 4.2 to 4.4. No actual optimization was executed. Each economic scenario and each combination of $c_A$ and $c_{HP}$ were separately analyzed. First, the system with maximum LCS was determined. Therefore, the maximum obtained LCS of each system were compared among each other. If the maximum LCS of different systems were identical or close, the system with the higher free fraction was preferred. Then, the system with maximum free fraction was determined. Therefore, the maximum obtained free fractions at LCS $\geq 0$ of each system were compared among each other. If the maximum fractions of different systems were identical or close, the system with the higher LCS was preferred.

Figure 4.9 shows the results of the analysis. The legend in the upper right section illustrates how to read the chart. The upper section shows the maximum LCS, the lower section the maximum free fraction. The left section takes into account the best economic scenario, the middle section the medium, and the right section the worst scenario. Each column represents one specific system. Each system type, i.e. SDHWS, HPWHS, or PSHPS, is represented by a specific column pattern. The horizontal bar contains the nine combinations of area-dependent and area-independent costs repeated for each economic scenario. The upper number is $c_A$ in $/m^2$, and the lower number is $c_{HP}$ in $/kW.

Regarding the upper section, the height of the column represents the LCS reached by the system with the best economic performance. A column reaching the border of the plot, would mean $ 1,500,000 LCS. The number set at the top of the column shows the system performance. The upper number is the actual LCS in 1,000 $. The lower number is the corresponding free fraction in % according to chapter 3. The bottom number set shows the system parameters in terms of collector area and heat pump capacity fraction. Consequently, a system with $A_c$=0 is a HPWHS, and a system with fcap=0 is a SDHWS. The collector areas are rounded to full 500 $m^2$.

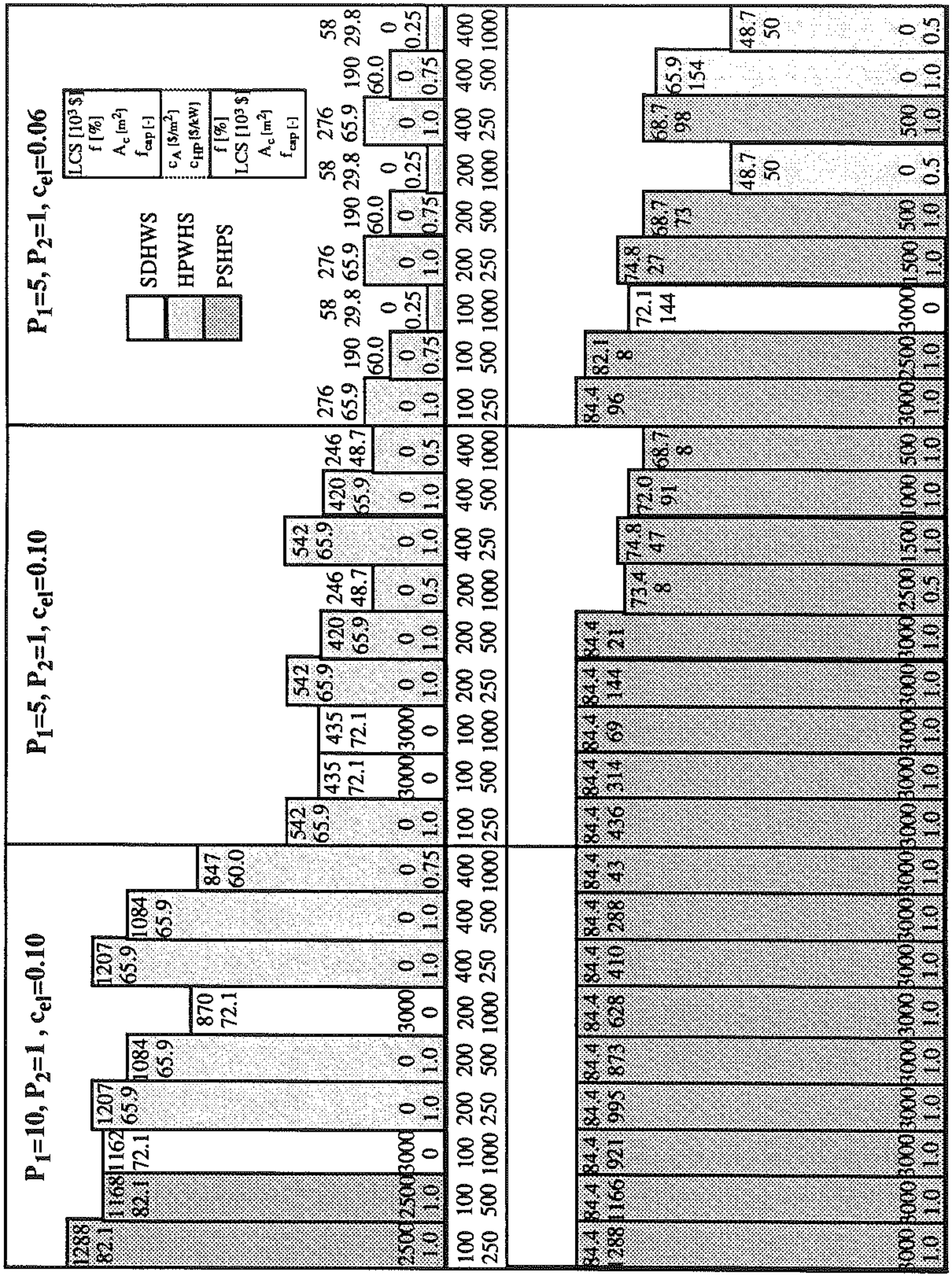



Figure 4.9 Summary system choice maximum LCS and maximum free fraction

The height of the column in the lower section indicates the magnitude of the free fraction reached by the system with best thermal performance and LCS≥0. A column reaching the border of the section would mean 100% free fraction. The numbers in the columns represent the same parameters as pointed out regarding the upper section. The top number set has switched: The upper number is the optimized free fraction. The lower number is the corresponding LCS.

In all cases, a system exists which yields positive maximum LCS regardless of the economic scenario and the related equipment costs. Hence, the installation of an alternative domestic hot water system always pays. The system choice and the obtainable LCS depend to a high extend on the assumed economic scenario, and on $c_A$ and $c_{HP}$, as well. The PSHPS performs the better the better the chosen economic scenario is, because the PSHPS yields the highest free fraction. Low initial area-dependent and independent costs especially support the PSHPS with a high volume of equipment. It makes sense that a low $c_A$ and a high $c_{HP}$ are in favor for the SDHWS. Vise versa, with regard to the HPWHS.

The full heat pump capacity PSHPS is the best system choice at best economic scenario and low $c_A/c_{HP}$. The LCS are higher than \$ 1,000,000. The free fraction is very high (82.1%). The SDHWS is the best system choice at best and medium economic scenario and low $c_A$/high $c_{HP}$. The free fraction is 72.1% which is the fraction at the largest investigated collector area. A collector area larger than 3,000 $m^2$ has the potential to obtain higher LCS. In all other cases, especially at worst economic scenario, the HPWHS shows the best economic performance. With high $c_{HP}$, only a small, quarter capacity, heat pump makes sense. The LCS are significantly lower (\$ 58,000), but the free fraction is only about 30%.

Focusing on maximum free fraction, the full capacity large collector area PSHPS is the best choice in most of the cases. At best and medium economic scenario, the free fraction reaches to 84.4%. The LCS are so high that there is a large potential to increase the free fraction by enlarging the collector area beyond 3000 $m^2$ till the LCS drops below zero. At the medium economic scenario and high $c_A$, the free fraction is still around 70%. The LCS are significantly lower. The 73.4% of the $c_A$=200 \$/$m^2$/$c_{HP}$=1,000 \$/kW might be outperformed by a large area SDHWS, because the 72.1% at 3,000 $m^2$ area still yield \$ 142,000 LCS. Otherwise, the large area SDHWS is only the best choice at worst economic scenario and low $c_A$ and high $c_{HP}$. With \$ 144,000 LCS, the system has still a potential of increasing its free fraction by increasing the collector area. The HPWHS yields maximum free fraction at worst economic scenario and high $c_A$/high $c_{HP}$. At first glance, this seems to be surprising, but one has to take into account that at these conditions, the SDHWS barely reaches positive LCS, at all. The best choice is a half capacity system, which means a comparatively low additional investment, but still almost 50% free fraction. There is still a potential to yield a higher fraction by varying the heat pump capacity.

# *Chapter Five*

## Conclusions and Recommendations

This chapter will first summarize the results of the thermal and economic system analyses. Then, recommendations for future work on the subject of this study will be given.

### 5.1 Conclusions

The TRNSYS simulations give the following thermal performances: A SDHWS at 3,000 $m^2$ collector area yields a 72.1% free fraction. A HPWHS with the capability to meet the entire load reaches a 65.9% free fraction. A PSHPS combining these system parameters, yields a free fraction as high as 84.4%. From an ecological point of view, this PSHPS system is highly recommendable. Smaller collector areas or heat pump capacities reach lower free fractions.

The economic performance depends to a large extend on the economic scenario and the related equipment costs. Striving for maximum LCS, the PSHPS is only competitive with a low related equipment cost, and assuming an economic scenario which promotes renewable energy application. The free fraction is over 80%, and the LCS are over $ 1,000,000, which makes renewable energy application economically interesting. Otherwise, the higher free fraction does not compensate for the higher additional costs. The

SDHWS is recommendable at the best and medium economic scenarios and with low area-dependent costs, and high heat pump costs. The HPWHS yields maximum LCS mostly when the area-dependent costs are high. Assuming an economic scenario which does not promote renewable energy application, the HPWHS yields maximum LCS regardless of the related equipment costs. The free fractions are between 30 and 66%, and the LCS are between \$ 58,000 and \$ 276,000.

Being interested in a highest possible free fraction at no economic loss, the PSHPS is mostly the best choice. Only when assuming the worst economic scenario does the LCS drop below zero at such low free fraction that the SDHWS and the HPWHS become competitive. The free fraction is between 48.7 and 84.4%.

The results are only a rough estimation and meant to be a guideline for system selection and design. Before designing an actual system, climate conditions, available collector, tank and heat pump designs, and current market conditions have to be taken into consideration and further investigations have to be executed.

## 5.2 Recommendations for Future Work

In this study, only a parallel solar assisted heat pump system with a specific control strategy is investigated. The following suggestions are also valid for the SDHWS and the HPWHS.

The TRNSYS system model is simple and could be refined in different ways, in order to obtain a more accurate and reliable performance prediction. For example, the hot water draw could be modeled in a more accurate way, taking into account short period peaks. A suitable tool is the water draw modeling program WATSIM (EPRI (1992)).

The system was only simulated for collector areas up to 3,000 $m^2$. It might be interesting to investigate larger areas, not only to provide higher free fraction but also to optimize the LCS.

The results of the thermal performance are entirely for Madison, WI. Not only solar radiation and ambient air temperature, but also the ground source temperature are functions of location. Under other climate conditions, collector and heat pump performance are different and will lead to a different free fraction, which effects the LCS.

The system control strategy could have a major impact on the performance. In this study, only a parallel system with either solar storage tank draw or heat pump operation was investigated. Solar storage tank draw is only given when the storage tank temperature exceeds the domestic hot water set temperature. Other control strategies as discussed in chapter 1.4.3 work with a more frequent storage tank draw at a lower temperature. These control strategies could significantly increase the collector efficiency, avoid cycling, and decrease storage tank standby losses. These control strategies seem to be most promising in combination with an auxiliary heating element. The heat pump water outlet temperature can also be kept low so that the COP is not effected. The remaining required temperature lift to set point is made up by auxiliary heat. Appendix B.5 shows a system where low temperature solar heated water is made up by the heat pump water heater. The system was not investigated in a detailed way. This could be the subject of a future study.

Other system configuration options exist: the series system and the dual system (see chapter 1.4.3). With the solar heated water being the heat pump source, the heat pump COP is significantly increased. It would be of interest to compare the performance of the different system configurations. Appendices B.6 and B.7 propose TRNSYS decks for series and dual systems with and without auxiliary heating element. The systems were not thoroughly investigated. A future study could execute the required simulations and analyses.

The EES heat pump model was developed to model the heat pump performance for different source temperatures. This provides the opportunity to run simulations for different or varying ground source temperatures as required for other climate conditions than Madison, and the series and dual systems. The option to use waste heat in terms of drain water should also be investigated.

It would be valuable, if a generic heat pump model, in the form of a TRNSYS type, was developed. This TRNSYS type would be helpful for further heat pump systems research done at the Solar Energy Laboratory.

The economic analysis is based on electricity costs. Today, large scale domestic hot water plants are usually fired by natural gas or heating oil, because electricity is more expensive. Basing the economic analysis on today's natural gas or heating oil costs would significantly lower the obtained LCS. More general, accurate and reliable predictions could be done by extending and refining the economic analysis.

# APPENDIX A

# EES PROGRAMS

## A.1 Heat Pump Water Heater

```
"
HEAT PUMP WATER HEATER MODEL

This program models the performance of a heat pump water heater, which has the capability to meet the
entire water heating load. The model is based on thermal modeling and on approximations of catalog data.

The SI system is the default unit system.

INPUT VARIABLES

Source inlet temperature T_s_i [°C], i.e. the temperature of the source entering the heat exchanger at the
evaporator side.
Set temperature T_w_o [°C], i.e. the temperature of the water leaving the heat exchanger at the condenser
side, in this case the hot water set temperature.
The input variables can be varied in a parametric table.
"

"CONSTANTS"

"mass flow rate source"mdot_s=2.5"[kg/s]"
"water inlet temperature, i.e. mains water temperature"T_w_i=7.5"[°C]"
"specific heat capacity water"cp_w=4.19"[kJ/kg-K]"

"DESIGN CAPACITY"
"
The design capacity has to meet the required hot water mass flow rate (220 GPH)
@ maximum possible temperature lift (=54C from 6 C to 60 C).
"

MBH_des=220*3.785/3600*4.19*3413/1000*54"[1000 Btu/hr]"

"UNIT CONVERSIONS"

"SI -> BU"
T_w_o=(T_w_o_F-32)*5/9
mdot_w=GPH/3600*3.785
T_s_i=(EWT-32)*5/9
```

```
mdot_s=GPM/60*3.785
T_w_i=(T_w_i_F-32)*5/9

"RELATIONS FOR EFFECTIVENESS & EFFICIENCY"

"isentropic efficiency curvefit (Fedders performance table)"
eta_iso=.9731999-7.289998e-3*T_w_o
"NTU evaporator side curvefit (Fedders performance table)"
NTU_evap=9.525239-9.824399e-2*T_w_o
"NTU condenser side curvefit (Fedders performance table)"
NTU_cond=3.673*(1-3.125129e-2*(T_w_o-60))
"mechanic efficiency counts for all losses caused by friction, pressure drops etc."
eta_mech=.9
"relation for effectiveness evaporator side"
epsilon_evap=1-exp(-NTU_evap)
"relation for effectiveness condenser side condensing"
epsilon_cond=1-exp(-NTU_cond)
"relation for effectiveness condenser side desuperheating"
epsilon_desup=epsilon_cond

"specific heat capacity refrigerant"
cp_r=SpecHeat(R134a, T=((T_2+Tprime_2)/2),p=p_3)"[kJ/kg-K]"
"capacitance rate water" C_w=cp_w*mdot_w"[kJ/s-K]"
"capacitance rate refrigerant" C_r=cp_r*mdot_r"[kJ/s-K]"
"determination of minimum capacitance rate" C_min=min(C_r,C_w)"[kJ/s-K]"


"REFRIGERATION CYCLE"

T_2=Temperature(R134a,h=h_2,p=p_2)

T_1=T_4
Tprime_2=T_3

p_3=Pressure(R134a,T=T_3,X=0)
p_2=p_3
p_1=Pressure(R134a,T=T_1,X=1)
p_4=p_1

h_1=Enthalpy(R134a,T=T_1,X=1)
h_3=Enthalpy(R134a,T=T_3,X=0)
h_4=h_3

hprime_2=Enthalpy(R134a,T=T_3,X=1)

"100% isentropic efficiency"
s_1=Entropy(R134a,T=T_1,X=1)
h_2_s=Enthalpy(R134a,P=p_2,S=s_1)

eta_iso*(h_2-h_1)=(h_2_s-h_1)
COP*(h_2-h_1)=(h_2-h_3)*eta_mech
P_el=Q_del/COP"kW"
```

"
CONDENSER

The course of the capacity is curvitted where the influence of T_s_i is considered according to performance table Drake HP, and the influence of T_w_o is considered by linear curvefit according to performance table Fedders SOCFO80.
"

```
MBH=MBH_des+deltaT_EWT/10*MBH_des*6/100+deltaT_set_F*MBH_des*5/1000
deltaT_EWT=EWT-EWT_ref
deltaT_set_F=Tset_F_ref-Tset_F
"reference temperatures"
"lowest possible entering source temperature"EWT_ref=45
"highest possible exiting water temperature"Tset_F_ref=140
"unit conversion"Qdot_del=MBH*1000/3413
"calculation of water mass flow"Qdot_del=mdot_w*cp_w*(T_w_o-T_mains)
"calculation of refrigerant mass flow"Qdot_del=mdot_r*(h_2-h_3)
"condensing"epsilon_c=(Tprime_w-T_w_i)/(T_3-T_w_i)
"desuperheating"C_min*(T_2-Tprime_w)*epsilon_desup=C_w*(T_w_o-Tprime_w)

"EVAPORATOR"

Qdot_evap=mdot_s*cp_w*(T_s_i-T_s_o)
Qdot_evap=mdot_r*(h_1-h_4)
epsilon_evap=(T_s_o-T_s_i/(T_1-T_s_i)
```

## A.2 Heat Pump Performance Data Table

According to EES program (Appendix A.1) @ $\dot{Q}_{del,design}$=53 kW, $T_{w,i}$=7.5 °C, $\dot{m}_s$ =2.5 kg/s
First row: $T_{w,o}$ [°C], first column: $T_{s,i}$ [°C], each cell: $\dot{Q}_{del}$ [kW] - COP - $\dot{Q}_{evap}$ [kW]

| | 40 | 50 | 60 | 70 | 85 |
|---|---|---|---|---|---|
| 7 | 45.9-5.1-37.9 | 41.2-3.8-31.5 | 36.5-3.0-25.6 | 31.8-2.4-20.0 | 24.7-1.3-8.2 |
| 15 | 50.4-6.4-43.2 | 45.7-4.6-36.6 | 41.0-3.5-30.2 | 36.3-2.7-24.2 | 29.3-1.4-11.0 |
| 20 | 53.3-7.7-47.0 | 48.6-5.2-40.1 | 43.9-3.9-33.6 | 39.1-2.9-27.0 | 32.1-1.5-13.0 |
| 25 | 56.1-9.5-50.7 | 51.4-6.1-43.7 | 46.7-4.3-36.9 | 42.0-3.2-30.1 | 34.9-1.6-15.3 |
| 30 | 58.9-12.6-54.6 | 54.2-7.3-47.4 | 49.5-4.9-40.4 | 44.8-3.5-33.2 | 37.7-1.7-17.8 |
| 35 | 61.8-18.2-58.6 | 57.0-9.0-51.3 | 52.3-5.7-44.0 | 47.6-3.9-36.6 | 40.6-1.8-20.5 |
| 40 | - | 59.9-11.8-55.2 | 55.2-6.8-47.7 | 50.5-4.4-40.1 | 43.4-2.0-23.5 |
| 45 | - | 62.7-16.9-59.3 | 58.0-8.3-51.6 | 53.3-5.1-43.8 | 46.2-2.1-26.7 |
| 50 | - | - | 60.8-10.8-55.6 | 56.1-6.0-47.6 | 49.0-2.3-30.2 |
| 55 | - | - | 63.6-15.3-59.8 | 58.9-7.3-51.6 | 51.8-2.6-33.7 |

## A.3 Economics

```
"
ECONOMICS

This program calculates the economic performance (break-even point and LCS) of alternative DHW system types.
"

"Constants:"
"annual energy usage conv. system"Q_conv=2017939"[kWh]"
"full heat pump capacity"Cap=490"[kW]"

"
Parameters in parametric table:
free fraction f [-]
costs collectors + tank together c_A [$/sm]
heat pump costs c_HP [$/kW]
collector area A_c [sm]
heat pump capacity fraction f_cap [-]
present worth factors PW_1, PW_2[-]
electricity costs c_el= [$/kWh]
LCS [$] -> LCS=0 at break-even point
"

"heat pump capacity" HP= f_cap*Cap"[kW]"
"LCS" LCS=PW_1*C - PW_2*I"[$]"
"Annual electric costs"
C= (1- f) * C_conv"[$]"
C_conv=Q_conv*c_F"[$]"
"Initial costs"
I=  A_c * c_A+ HP * c_HP"[$]"
```

# APPENDIX B

## TRNSYS DECKS

**B.1 Electric Domestic Hot Water System**

**B.2 Solar Domestic Hot Water Systems**

**B.3 Heat Pump Water Heating Systems**

**B.4 Parallel Solar Heat Pump Water Heating Systems**

**B.5 Parallel Solar Heat Pump Water Heating Systems (Pre-heating)**

**B.6 Series Solar Heat Pump Water Heating Systems**

**B.7 Dual Solar Heat Pump Water Heating Systems**

```
ASSIGN C:\annette\CONV1.LST          6
ASSIGN C:\annette\CONV1.OUT          38

*******************************************************
*  This deck predicts the thermal performance of a hospital
*  CONVENETIONAL ELECTRIC DOMESTIC HOT WATER
*  SYSTEM
*******************************************************

*** FILE ASSIGNMENT ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16

***  SYSTEM PARAMETERS  ***

EQUATIONS  3
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 3 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS

EQUATIONS 3
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([5,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

EQUATIONS 5 ELECTRIC WATER HEATER
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* tank nodes
NODES = 1
* maximum output
QMAX=1500*3600

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START =  1
* simulation stop time
STOP =  8760
* simulation time step
STEP = .5

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

**** DATA READERS ***

UNIT 29 TYPE 14   FORCING FUNCTION FOR WATER DRAW
HOSPITAL
```

```
* water draw: gal/hr-bed
PARAMETERS 12
* TO VO T1 V0 T1 V1 T2 V1 T2 V0 T3 V0
0.0 1.0 6.0 1.0 6.0 7.0 20.0 7.0 20.0 1.0 24.0 1.0
* OUTPUTS: 1,VABAR 2,V

UNIT 39     TYPE 9   DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE  N  dT(HOURS) TMAINS  LU  FRMT
   -2   1    24    -1 1 0   16   0
* OUTPUTS: 1, TMAINS

*** SYSTEM COMPONENTS ***

UNIT 5 TYPE 4  ELECTRIC WATER HEATER
PARAMETERS 20
* MODE  VOL  CPF  RHO   UT   HI   AUXMOD   NODE1
NODETI
  1  ELTNK  4.19  1000  ULOSS  ELHGT    1     1     1
* TSET  DTDB  QAUX1    NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   0   QMAX     1     1    TSET    0    0
* UAFLUE  TFLUE  TBOIL
  0.0    TENV   100
INPUTS 7
* TH  MH  TL   ML   TENV
  0,0  0,0  39,1  MLOAD  0,0  0,0  0,0
  0.0  0.0  TI   100.0   TENV  1.0  0.0
DERIVATIVES NODES
TSET TSET TSET  TSET TSET TSET TSET TSET TSET TSET
*UNIT 5 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load   5,Qenv,loss
*6,Qs 7,dEtank 8,Qtotal

***  OUTPUT ***

UNIT   38  TYPE  28  SIMULATION  SUMMARY  &  ENERGY
BALANCE CHECK
PARAMETERS 22
*DTP  TON  TOFF  LU  OMODE
 -1  START  STOP  38  2  1  0  0  -4  -4   0  -4  0    -4 0 -2 2 -4 0 -2
2 -4
INPUTS  6
*DE  QAUX  QLOSS QLOAD TLOAD TAV
 5,7  5,8  5,5  5,6 5,3 5,12
LABELS  6
 QAUX[kJ]  DE[kJ]  QLOSS[kJ]  QLOAD[kJ] TLOAD[C] TAV[C]
CHECK 0.10  1, -2,  -3, -4,

END
```

```
ASSIGN C:\annette\S2.LST        6
ASSIGN C:\annette\S2.OUT        38

*******************************************************
*  This deck predicts the thermal performance of the
*  VA HOSPITAL SOLAR DOMESTIC HOT WATER SYSTEM
*******************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*    required, the water draw from the tank is reduced and additional water
* from the mains    is added to keep the required water flow and reach the
* required temperature.
```

```
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

EQUATIONS 10 STORAGE TANK
* tank volume & unit conversion
TNKSIZE =  1.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1
* tank losses & unit conversion
RVAL=16.0
RVAL1=RVAL*.0489194
ULOSS=1/RVAL
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2

EQUATIONS 1 BACKUP HEATER
* maximum output
Qmax=1500*3600

EQUATIONS 12 COLLECTORS
* number of collectors in array
COL = 291
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001
```

```
* maximum collector flow rate
MMAXCOL=30000
* maximum pump power input
PMAXCOL=14000
* controls deadbands
DEADH = 15
DEADL = 1
CONST=1

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 1 OUTPUT
* total energy requirement (for comparison)
QT=7264000000

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START =  1
* simulation stop time
STOP =  8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120

TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19   TYPE 9   DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1     14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2    1     24     -1 1 0   16    0
* OUTPUTS: 1, TMAINS

UNIT 16 TYPE 16 RADIATION PROCESSOR MADISON
PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY      LAT  SC
SHIFT  SMOOTH IE
   7      1       1     STRTDAY    LAT  SC  SHIFT    2     -1
INPUTS 7
* I(kJ/m2-hr)  td1    td2    RHOG    BETA1    GAMMAI
 19,4    19,3    19,19  19,20  RHOG     SLOPE     GAMMAI
 0.0     0.0       0.0    0.0     RHOG     SLOPE     GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
```

```
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp   EFFMD  G   ao   a1   a2  EFF CpHX
OPTMD  bo  b1
  1   1  AREA  CPHOT   1    50  FRta  FRUL  0.  EFF 4.2   1    0.1
0.0
INPUTS 10
* Ti  mCOLL(kg/hr) mHX  Tamb  It   I   Id   RHOG  THETA
BETA(SLOPE)
 3,1  3,2          3,2  19,5  16,6 16,4  16,5   0,0   16,9   16,10
 TI   0.0          0.0  20.0   0.0  0.0   0.0  RHOG    0.0    40.0
* OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL   100
INPUTS 4
* Th   Tl    TIN  GAMMAI
 1,4  4,1   1,1   2,1
 15.  TI    100    0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAXCOL  4.19  PMAXCOL    0.
INPUTS 3
*  Ti    mi   GAMMA
 4,1    4,2   2,1
  TI    0.0    0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 6 TYPE 6 EXTRA AUX. HEATER
PARAMETERS 5
*Qmax Tset Cpf UA ETA
QMAX TSET 4.19 1.114 1.0
INPUTS 4
*Ti Mi GAMMA Tenv
 4,3 4,4 CONST 0,0
 60.0 0.0 1.0 TENV
*UNIT 6 OUTPUTS:
*1,To 2,Mo 3,Qaux 4,Qloss 5,Qfluid

UNIT 4 TYPE 38  SOLAR STORAGE TANK (HORIZONTAL)
PARAMETERS 11
*MODE   VOL      HT    HR  CPF  RHO  k CONFIG UA    RI  TI
 2    TSIZE    HEIGHT  HR  4.19 1000 0   2   ULOSS 1   30
INPUTS 5
* TH   MH   TL    ML     TENV
 1,1   1,2  39,1  MLOAD  0,0
 0.0   0.0  TI    0.0    TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

***  OUTPUT ***

UNIT 38 TYPE 28 ENERGY BALANCE CHECK
PARAMETERS  35
*DTP  TON  TOFF  LU  OMODE
 -1   8761  STOP  38  2  1  0  0  0 -4  -4  -4  0  -4  0  -4
 0 0 -1 AREA 1 2 -4 -1 QT  -12  4 -1 QT  2 -4 0 -2 2 -4
INPUTS  8
*DE  QAUX QU QLOSS QLOAD   QU I0 T4-1
 DE  6,8  1,3 QLOSS  QLOAD  1,3 16,1 4,3
LABELS  8
QU[kJ]  QAUX[kJ]  DE[kJ]  QLOSS[kJ]  QLOAD[kJ]   EFFIC   SF T -
1-AV
CHECK 0.10  1, 2, -3, -4, -5

END
```

```
ASSIGN C:\annette\S1.LST        6
ASSIGN C:\annette\S1.OUT        38

*******************************************************
* This deck predicts the thermal performance of a hospital
* SOLAR DOMESTIC HOT WATER SYSTEM
*******************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT       16
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI         14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 7 STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1
* tank losses & unit conversion
RVAL=16.0
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 3 STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2

EQUATIONS 5 BACKUP HEATER
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
NODES = 1
* maximum output
Qmax=1500*3600

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
```

```
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 5 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 1 OUTPUT
* total energy requirement (for comparison)
QT=7264000000

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START =  1
* simulation stop time
STOP =  8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19   TYPE 9   DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1     14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2   1     24     -1 1 0   16   0
* OUTPUTS: 1, TMAINS
```

```
UNIT 16 TYPE 16 RADIATION PROCESSOR MADISON
PARAMETERS 9
* RADMODE TRACKMODE TILTMODE DAY    LAT SC
SHIFT SMOOTH IE
  7    1    1   STRTDAY  LAT SC SHIFT  2  -1
INPUTS 7
* I(kJ/m2-hr) td1  td2  RHOG  BETA1  GAMMAI
 19,4  19,3  19,19 19,20 RHOG  SLOPE  GAMMAI
 0.0  0.0    0.0  0.0  RHOG  SLOPE  GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp  EFFMD G  ao  a1  a2 EFF CpHX
OPTMD  bo  b1
  1  1  AREA  CPHOT  1   50 FRta  FRUL  0.  EFF 4.2  1   0.1
0.0
INPUTS 10
* Ti mCOLL(kg/hr) mHX  Tamb  It  I  Id  RHOG THETA
BETA(SLOPE)
 3,1  3,2       3,2  19,5  16,6 16,4  16,5  0,0   16,9   16,10
 TI  0.0        0.0  20.0  0.0 0.0  0.0  RHOG   0.0   40.0
* OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL  100
INPUTS 4
* Th  Tl   TIN  GAMMAI
 1,4  4,1   1,1   2,1
 15.  TI    100   0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX   Cp   Pmax(KJ/HR)  fpar
  MMAXCOL  4.19  PMAXCOL   0.
INPUTS 3
*  Ti   mi  GAMMA
 4,1   4,2  2,1
  TI   0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 6 TYPE 4 BACKUP HEATER
PARAMETERS 20
*MODE  VOL  CPF  RHO  UT  HI   AUXMOD  NODE1
NODETI
  1  ELTNK  4.19  1000  ULOSS ELHGT   1     1   1
*TSET  DTDB  QAUX1   NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   0   QMAX    1    1    TSET   0   0.0
*UAFLUE  TFLUE  TBOIL
 0.0   TENV  100
INPUTS 5
* TH  MH  TL   ML    TENV
 0,0 0,0  4,3   4,4   0,0
 0.0 0.0  0.0  0.0   TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 6 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*       7,dEtank 8,Qaux1

UNIT 4 TYPE 38  SOLAR STORAGE TANK (HORIZONTAL)
PARAMETERS 11
*MODE  VOL    HT   HR  CPF  RHO  k CONFIG UA   RI  TI
 2   TSIZE   HEIGHT  HR  4.19 1000 0   2  ULOSS 1   30
INPUTS 5
* TH  MH  TL   ML    TENV
 1,1  1,2  39,1  MLOAD  0,0
 0.0  0.0  TI   0.0   TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
```

```
*       7,dEtank 8,Qaux1

***  OUTPUT ***

UNIT 38 TYPE 28 ENERGY BALANCE CHECK
PARAMETERS  35
*DTP  TON  TOFF  LU  OMODE
 -1   8761  STOP  38  2  1  0  0  0 -4  -4  -4  0  -4  0  -4
 0 0 -1 AREA 1 2 -4 -1 QT  -12  4 -1 QT  2 -4 0 -2 2 -4
INPUTS  8
*DE  QAUX QU QLOSS QLOAD   QU I0 T4-1
 DE  6,8  1,3 QLOSS  QLOAD  1,3 16,1 4,3
LABELS  8
QU[kJ]  QAUX[kJ]  DE[kJ]  QLOSS[kJ]  QLOAD[kJ]   EFFIC   SF T  -
1-AV
CHECK 0.10  1, 2, -3, -4, -5

END
```

```
ASSIGN C:\annette\HP1.LST      6
ASSIGN C:\annette\HP1.OUT      48

*******************************************************
*  This deck predicts the thermal performance of a hospital
*  FULL CAPACITY HEAT PUMP WATER HEATING SYSTEM
*******************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE8.DAT  42


***  SYSTEM PARAMETERS  ***

EQUATIONS 3
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

EQUATIONS 1 HEAT PUMP
TSOURCE=7.2

EQUATIONS 9 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[13,1]
QCOND=HPCON*[49,1]*[42,1]*3600
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW

EQUATIONS 2
*HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 7 STORAGE TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
```

```
NODES = 5
* tank losses & unit conversion
RVAL=16.0
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START =  1
* simulation stop time
STOP =  8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2    1    24    -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

*** SYSTEM COMPONENTS ***

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2  3  10 5
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
 0      10 60
*OUTPUTS 2
*1,CAP 2,COP 3,QEVAP

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER T3
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22   85
INPUTS 4
* Th   Tl    TIN  GAMMAI
 TSET 14,14   THPO   22,1
 60.0  TI     100    0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
  MMAX   4.19   2500          0.0
INPUTS 3
*  Ti    mi   GAMMA
  14,1  14,2   PCON
  TI    0.0    0.0
```

```
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL
PARAMETERS 20
*MODE  VOL  CPF  RHO  UT  HI   AUXMOD  NODE1
NODETI
  1  ELTNK  4.19  1000  ULOSS ELHGT    1    1    1
*TSET  DTDB  QAUX1   NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   0   0     1    1    TSET   0   0.0
*UAFLUE  TFLUE  TBOIL
 0.0   TENV  100
INPUTS 5
* TH  MH   TL   ML   TENV
 THPO 13,2 39,1  ML   0,0
 TI   0.0  TI   0.0  TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtn 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*       7,dEtank 8,Qaux1

***  OUTPUT ***

UNIT 48 TYPE 28 MONTHLY SIM. SUM. & ENERGY BALANCE
CHECK HEAT PUMP
PARAMETERS  16
*DTP  TON  TOFF  LU  OMODE
 -1  START  STOP  48  2  1 0  0 -4  -4  0  -4  0  -4 0 -4
INPUTS  5
*DE   QCOND  QLOSS QLOAD PEL
 14,7 QCOND  14,5  14,6  PEL
LABELS  5
QCOND[kJ]  DE[kJ]  QLOSS[kJ]  QLOAD[kJ] PEL[KJ]
CHECK 0.9  1, -2, -3, -4

END
```

```
ASSIGN C:\annette\HP2.LST      6
ASSIGN C:\annette\HP2.OUT      48

*********************************************************
*  This deck predicts the thermal performance of a hospital
*  HEAT PUMP WATER HEATING SYSTEM W/ AUX. HEATER
*********************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FIL13.DAT  42


***  SYSTEM PARAMETERS  ***

EQUATIONS 3
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*    required, the water draw from the tank is reduced and additional water
* from the mains    is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

EQUATIONS 2 HEAT PUMP
TSOURCE=7.2
* capacity fraction
FCAP=0.125

EQUATIONS 9 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[13,1]
QCOND=HPCON*[49,1]*[42,1]*3600*FCAP
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW

EQUATIONS 2
*HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 7 STORAGE TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
```

```
ELHGT=-HEIEL
* nodes
NODES = 5
* maximum power aux. heater
Qmax= 1500*3600
* tank losses & unit conversion
RVAL=16.0
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START = 1
* simulation stop time
STOP = 8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2   1    24    -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

*** SYSTEM COMPONENTS ***

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2  3  10 5
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
0      10 60
*OUTPUTS 2
*1,CAP 2,COP 3,QEVAP

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER T3
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22  85
INPUTS 4
* Th   Tl    TIN  GAMMAI
  TSET THPI   THPO   22,1
  60.0  TI     100    0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
  MMAX   4.19   2500         0.0
```

```
INPUTS 3
* Ti   mi   GAMMA
  14,1  14,2   PCON
  TI   0.0  0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL W/
AUX. HEATER
PARAMETERS 20
*MODE  VOL  CPF   RHO   UT   HI    AUXMOD   NODE1
NODETI
  1   ELTNK  4.19  1000  ULOSS ELHGT    1      1      1
*TSET  DTDB  QAUX1    NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   0    QMAX      1      1      TSET     0     0.0
*UAFLUE  TFLUE  TBOIL
 0.0    TENV   100
INPUTS 5
* TH   MH    TL    ML    TENV
 THPO 13,2 39,1  ML    0,0
 TI    0.0  TI    0.0  TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtn 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

***  OUTPUT ***

UNIT 48 TYPE 28 MONTHLY SIM. SUM. & ENERGY BALANCE
CHECK HEAT PUMP
PARAMETERS  20
*DTP  TON  TOFF  LU  OMODE
 -1  8761  STOP  48  2  1 0  0 0 -4 -4  -4  0  -4  0  -4 0 -4 0 -4
INPUTS  7
*DE   QCOND  QAUX QLOSS QLOAD PEL PELPUMP
 14,7 QCOND  14,8  14,5  14,6 PEL 13,3
LABELS  7
QAUX[KJ] QCOND[kJ]  DE[kJ]  QLOSS[kJ]  QLOAD[kJ] PEL[KJ]
PELPUMP[KJ]
CHECK 0.1  1, 2, -3, -4, -5

END
```

```
ASSIGN C:\annette\PSHP5.LST          6
ASSIGN C:\annette\PSHP5.OUT          58

*****************************************************
*  This deck predicts the thermal performance of a hospital
*  PARALLEL SOLAR ASSISTED FULL CAPACITY HEAT PUMP
*  WATER HEATING SYSTEM WITH EITHER STORAGE TANK
*  DRAW OR HEAT PUMP OPERATION
*****************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE9.DAT       42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 3 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
```

```
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]


EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 1 HEAT PUMP
TSOURCE=7.2

EQUATIONS 12 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[13,1]
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]


EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 6 HEAT PUMP TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
NODES = 5
* heat pump tank inputs
TINLET=THPO*(1-[12,1])+[4,3]*[12,1]
MINLET=[4,4]*[12,1]+[13,2]


EQUATIONS 7 ENERGY BALANCE -> UNIT 58
```

```
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP
* circulators energy
PELPUMP=[3,3]+[13,3]+[23,3]


*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START =  1
* simulation stop time
STOP =  8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19   TYPE 9   DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1     14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2   1     24     -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR  PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY      LAT  SC
SHIFT  SMOOTH IE
   7       1       1    STRTDAY    LAT  SC  SHIFT    2     -1
INPUTS 7
* I(kJ/m2-hr)  td1    td2    RHOG    BETA1    GAMMAI
 19,4    19,3    19,19  19,20  RHOG     SLOPE     GAMMAI
 0.0     0.0        0.0     0.0    RHOG     SLOPE     GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp   EFFMD  G   ao    a1    a2  EFF CpHX
OPTMD  bo  b1
```

```
 1  1 AREA CPHOT  1   50 FRta FRUL 0. EFF 4.2  1   0.1
0.0
INPUTS 10
* Ti mCOLL(kg/hr) mHX Tamb  It  I  Id  RHOG THETA
BETA(SLOPE)
 3,1 3,2       3,2 19,5 16,6 16,4 16,5  0,0   16,9  16,10
 TI  0.0       0.0 20.0  0.0 0.0  0.0  RHOG  0.0   40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL   100
INPUTS 4
* Th  Tl   TIN  GAMMAI
 1,4  4,1   1,1   2,1
 15.  30    100   0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX   Cp   Pmax(KJ/HR)  fpar
 MMAXCO  4.19  PMAXCO        0.
INPUTS 3
*  Ti   mi   GAMMA
 4,1    4,2   2,1
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38  SOLAR STORAGE TANK HORIZONTAL
PARAMETERS 11
*MODE  VOL     HT   HR  CPF  RHO  k CONFIG UA   RI  TI
 2   TSIZE   HEIGHT  HR  4.19 1000 0  2  ULOSS  1   TI
INPUTS 5
* TH   MH   TL    ML     TENV
 1,1   1,2  14,1  MLOAD  0,0
 30.0 30.0  30.0   0.0   TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*       7,dEtank 8,Qaux1

UNIT 12 TYPE 2 MASS FLOW CONTROL FROM SOLAR
STORAGE TANK (0/1)
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  10      1     100
INPUTS 4
* Th  Tl   TIN  SMCONI
 4,3  TSET  0,0  12,1
 60  60    100   0.
*OUTPUTS: 1,SMCONo (CONTROL FUNCTION)

UNIT 23 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX   Cp   Pmax(KJ/HR)  fpar
 MLOAD  4.19  2800          0.
INPUTS 3
*  Ti   mi   GAMMA
 4,3   4,4   12,1
  TI   0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP)
PARAMETERS 5
*LU NX NY NX1 NX2
42 2  3  10 5
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
0     10 60
*OUTPUTS 2
*1,CAP 2,COP 3,QEVAP

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER T3
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22  85
INPUTS 4
```

```
* Th  Tl   TIN  GAMMAI
  TSET THPI   THPO   22,1
  60.0  TI     100    0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
  MMAX   4.19   2800        0.0
INPUTS 3
*  Ti    mi   GAMMA
   14,1  14,2   PCON
   TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HEAT PUMP STORAGE TANK VERTICAL
PARAMETERS 20
*MODE   VOL   CPF   RHO   UT   HI    AUXMOD   NODE1
NODETI
   1    ELTNK  4.19  1000  ULOSS ELHGT     1       1       1
*TSET   DTDB   QAUX1    NODE2   NODET2   TSET2   DTDB2
QAUX2
 TSET    0     0        1       1        TSET     0     0.0
*UAFLUE   TFLUE   TBOIL
  0.0     TENV    100
INPUTS 5
* TH    MH     TL     ML     TENV
  TINLET MINLET 39,1  ML    0,0
  TI    0.0  TI    0.0   TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

***  OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK
PARAMETERS 26
*DTP  TON  TOFF  LU  OMODE
 -1   8761  STOP  30  2  1  0  0 0 0-4  -4  -4 -4 0  -4  0  -4  0
-4 0 -4 0 -4 0 -4
INPUTS  10
*DE  QAUX QCOND QU QLOSSS QLOSSH QLOAD  PELHP
QLOSO PELPUMP
DE   14,8 QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
PELPUMP
LABELS  10
QAUX[KJ] QU[kJ]  QCOND[kJ]  DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ]
QLOAD[kJ]  PEL[kJ]
QLOSO[KJ] PELPUMP[KJ]
CHECK 0.10  1, 2, 3, -4, -5, -6, -7

END
```

```
ASSIGN C:\annette\PSHP11.LST          6
ASSIGN C:\annette\PSHP11.OUT          58

*******************************************************
*  This deck predicts the thermal performance of a hospital
*  PARALLEL SOLAR ASSISTED HEAT PUMP WATER
*  HEATING SYSTEM WITH EITHER STORAGE TANK DRAW
*  OR HEAT PUMP OPERATION AND AUX. HEATING
*******************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FIL13.DAT       42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 3 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
```

```
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 2 HEAT PUMP
TSOURCE=7.2
* capacity fraction
FCAP=0.125

EQUATIONS 12 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[13,1]
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220*FCAP
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]


EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 7 HEAT PUMP TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
NODES = 5
* maximum power aux. heater
QMAX= 1500*3600
* heat pump tank inputs
TINLET=THPO*(1-[12,1])+[4,3]*[12,1]
MINLET=[4,4]*[12,1]+[13,2]
```

```
EQUATIONS 7 ENERGY BALANCE -> UNIT 58
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP
* circulators energy
PELPUMP=[3,3]+[13,3]+[23,3]

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START =  1
* simulation stop time
STOP =  8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19   TYPE 9   DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1     14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2    1     24     -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR  PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY      LAT  SC
SHIFT  SMOOTH IE
  7       1       1     STRTDAY    LAT  SC  SHIFT   2    -1
INPUTS 7
* I(kJ/m2-hr)  td1    td2    RHOG    BETA1    GAMMAI
19,4    19,3    19,19  19,20  RHOG    SLOPE    GAMMAI
0.0     0.0       0.0    0.0    RHOG    SLOPE    GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
```

```
* MODE N  AREA  Cp   EFFMD  G   ao    a1    a2  EFF CpHX
OPTMD  bo  b1
  1   1  AREA  CPHOT   1    50  FRta  FRUL  0.  EFF 4.2   1    0.1
0.0
INPUTS 10
* Ti  mCOLL(kg/hr) mHX  Tamb   It    I    Id    RHOG  THETA
BETA(SLOPE)
 3,1  3,2          3,2  19,5  16,6 16,4  16,5   0,0    16,9   16,10
 TI   0.0          0.0  20.0   0.0  0.0   0.0   RHOG    0.0    40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL   100
INPUTS 4
* Th   Tl    TIN  GAMMAI
 1,4  4,1    1,1   2,1
 15.  30     100   0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAXCO  4.19  PMAXCO          0.
INPUTS 3
*  Ti    mi   GAMMA
 4,1    4,2   2,1
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38  SOLAR STORAGE TANK HORIZONTAL
PARAMETERS 11
*MODE   VOL      HT    HR  CPF  RHO  k CONFIG UA    RI  TI
 2    TSIZE    HEIGHT  HR  4.19 1000 0   2   ULOSS  1    TI
INPUTS 5
* TH   MH   TL    ML     TENV
 1,1   1,2  14,1  MLOAD  0,0
 30.0 30.0  30.0   0.0   TENV

*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

UNIT 12 TYPE 2 MASS FLOW CONTROL FROM SOLAR
STORAGE TANK (0/1)
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  10      1     100
INPUTS 4
* Th   Tl    TIN  SMCONI
 4,3  TSET   0,0   12,1
 60  60     100    0.
*OUTPUTS: 1,SMCONo (CONTROL FUNCTION)

UNIT 23 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MLOAD  4.19   2800           0.
INPUTS 3
*  Ti    mi   GAMMA
 4,3    4,4   12,1
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2  3  10 5
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
0      10 60
*OUTPUTS 2
*1,CAP 2,COP 3,QEVAP

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER THPI
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
```

```
 11 1    3  85
INPUTS 4
* Th  TI   TIN  HPCONI
 TSET THPI  THPO  22,1
 60.0 TI   60  0.
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX   Cp   Pmax(KJ/HR)  fpar
 MMAX  4.19  2800      0.0
INPUTS 3
*  Ti   mi  GAMMA
  14,1  14,2  PCON
  TI   0.0  0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL
PARAMETERS 20
*MODE  VOL  CPF  RHO  UT  HI   AUXMOD  NODE1
NODETI
  1  ELTNK  4.19  1000  ULOSS ELHGT   1    1    1
*TSET  DTDB  QAUX1   NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET  0   QMAX    1    1    TSET   0   0.0
*UAFLUE  TFLUE  TBOIL
 0.0   TENV  100
INPUTS 5
* TH  MH   TL   ML   TENV
 TINLET MINLET 39,1  ML   0,0
 TI  0.0  TI   0.0  TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*      7,dEtank 8,Qaux1

*** OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK
PARAMETERS 26
*DTP  TON  TOFF  LU  OMODE
 -1  8761  STOP  30  2  1  0  0 0 0-4  -4  -4 -4 0  -4  0  -4  0
-4 0 -4 0 -4 0 -4
INPUTS  10
*DE  QAUX QCOND QU QLOSSS QLOSSH QLOAD  PELHP
QLOSO PELPUMP
DE   14,8 QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
PELPUMP
LABELS  10
QAUX[KJ] QU[kJ]  QCOND[kJ]  DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ]
QLOAD[kJ]  PEL[kJ]
QLOSO[KJ] PELPUMP[KJ]
CHECK 0.10  1, 2, 3, -4, -5, -6, -7

END
```

```
ASSIGN C:\annette\PSHP6.LST          6
ASSIGN C:\annette\PSHP6.OUT          58

*****************************************************
* This deck predicts the thermal performance of a hospital
* PARALLEL SOLAR ASSISTED FULL CAPACITY HEAT PUMP
* WATER HEATING SYSTEM WITH SIMULTANOUS STORAGE
* TANK DRAW AND HEAT PUMP OPERATION
*****************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE5.DAT       42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 3 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
```

```
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]


EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 1 HEAT PUMP
TSOURCE=7.2

EQUATIONS 13 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[4,3]
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
THPO=THPOC
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]


EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]*[12,1]
* pump control (on day/off night)
PCON=[49,1]/7

EQUATIONS 4 HEAT PUMP TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
NODES = 5

EQUATIONS 6 ENERGY BALANCE -> UNIT 58
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
```

```
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START = 1
* simulation stop time
STOP = 8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19   TYPE 9   DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1    14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
  -2   1    24    -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR  PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY     LAT  SC
SHIFT  SMOOTH IE
  7      1      1     STRTDAY    LAT  SC  SHIFT    2     -1
INPUTS 7
* I(kJ/m2-hr)  td1    td2    RHOG    BETA1    GAMMAI
 19,4   19,3    19,19  19,20  RHOG    SLOPE    GAMMAI
 0.0    0.0       0.0    0.0     RHOG    SLOPE    GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp   EFFMD  G   ao    a1    a2  EFF CpHX
OPTMD  bo  b1
  1   1  AREA  CPHOT   1    50  FRta  FRUL  0.  EFF 4.2   1    0.1
0.0
INPUTS 10
* Ti  mCOLL(kg/hr) mHX  Tamb  It   I   Id   RHOG  THETA
BETA(SLOPE)
 3,1  3,2         3,2  19,5  16,6 16,4  16,5   0,0    16,9   16,10
```

```
 TI  0.0         0.0  20.0  0.0  0.0   0.0  RHOG    0.0    40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11 DEADH DEADL   100
INPUTS 4
* Th   TI    TIN  GAMMAI
 1,4  4,1   1,1   2,1
 15.  30     100    0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX    Cp   Pmax(KJ/HR)   fpar
 MMAXCO  4.19  PMAXCO          0.
INPUTS 3
*  Ti    mi   GAMMA
 4,1    4,2   2,1
  TI    0.0    0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38  SOLAR STORAGE TANK HORIZONTAL NO
AUX. HEAT
PARAMETERS 11
*MODE   VOL      HT    HR  CPF  RHO  k CONFIG UA    RI  TI
 2    TSIZE    HEIGHT  HR  4.19 1000 0   2   ULOSS  1   TI
INPUTS 5
* TH   MH   TL    MMAX  TENV
 1,1   1,2  14,1  13,2  0,0
 30.0  30.0  30.0    0.0    TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*          7,dEtank 8,Qaux1

UNIT 12 TYPE 2 HEAT PUMP CONTROLLER THPI
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22   100
INPUTS 4
* Th   TI    TIN  HPCONI
  TSET  4,3  0,0   12,1
  60.0  TI    100    0.
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2  3  9   4
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
 0      10 60
*OUTPUTS 2
*1,CAP 2,COP

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER TANK14 T3
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22   100
INPUTS 4
* Th   TI    TIN  HPCONI
  TSET 14,15  0,0   22,1
  60.0  TI     100    0.
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX    Cp   Pmax(KJ/HR)   fpar
  MMAX   4.19   2800         0.0
INPUTS 3
*  Ti    mi   GAMMA
  14,1  14,2   PCON
   TI   0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump
```

```
UNIT 14 TYPE 4  HEAT PUMP STORAGE TANK VERTICAL
*MODE   VOL   CPF   RHO   UT   HI    AUXMOD   NODE1
NODETI
  1   ELTNK  4.19  1000  ULOSS ELHGT     1       1      1
*TSET  DTDB  QAUX1    NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   5    0       1      1      TSET    0    0.0
*UAFLUE  TFLUE  TBOIL
 0.0    TENV   100
INPUTS 5
* TH   MH    TL    ML    TENV
 THPO 4,4   39,1   ML    0,0
 TI   0.0  TI   0.0  TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

***  OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK
PARAMETERS 22
*DTP  TON  TOFF  LU  OMODE
 -1   8761  STOP  58  2  1  0  0 0 -4  -4  -4 0  -4  0  -4  0  -4 0 -4 0 -4
INPUTS  8
*DE  QCOND QU QLOSSS QLOSSH QLOAD  PEL  QLOSO
DE   QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
LABELS  8
QU[kJ]  QCOND[kJ]  DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ] QLOAD[kJ]
PEL[kJ] QLOSO[KJ]
CHECK 0.90  1, 2, -3, -4, -5, -6

END
```

```
ASSIGN C:\annette\PSHP10.LST          6
ASSIGN C:\annette\PSHP10.OUT          58

*********************************************************
*  This deck predicts the thermal performance of a hospital
*  PARALLEL SOLAR ASSISTED HEAT PUMP WATER
*  HEATING SYSTEM WITH  SIMULTANOUS STORAGE
*  TANK DRAW AND HEAT PUMP OPERATION AND AUX.
*  HEAT
*********************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT         16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE6.DAT        42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI           14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 3 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
FRta =  7.0000E-0001
```

```
* slope of efficiency curve & unit conversion
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 2 HEAT PUMP
TSOURCE=7.2
* heat pump capacity fraction (parameter)
FCAP=0.5

EQUATIONS 13 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[4,3]
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220*FCAP
* pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPOC=THPI+QCOND/(4.19*MHPCHH)
THPO=THPOC
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]

EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]
* pump control (on day/off night)
PCON=[49,1]/7

EQUATIONS 5 HEAT PUMP TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
NODES = 5
* maximum electr. power input
Qmax=1500*3600

EQUATIONS 6 ENERGY BALANCE -> UNIT 58
```

```
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START =  1
* simulation stop time
STOP =  8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19   TYPE 9   DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1     14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2   1     24     -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR  PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY      LAT  SC
SHIFT  SMOOTH IE
  7       1       1     STRTDAY    LAT  SC  SHIFT    2     -1
INPUTS 7
* I(kJ/m2-hr)  td1    td2    RHOG     BETA1     GAMMAI
 19,4    19,3    19,19   19,20  RHOG     SLOPE     GAMMAI
 0.0     0.0        0.0     0.0     RHOG     SLOPE     GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp   EFFMD  G   ao    a1    a2  EFF CpHX
OPTMD  bo  b1
  1   1  AREA  CPHOT   1    50  FRta  FRUL  0.  EFF 4.2   1    0.1
0.0
INPUTS 10
```

```
* Ti  mCOLL(kg/hr) mHX  Tamb  It   I   Id   RHOG  THETA
BETA(SLOPE)
 3,1  3,2          3,2  19,5  16,6 16,4  16,5   0,0    16,9    16,10
 TI   0.0          0.0  20.0   0.0  0.0   0.0   RHOG    0.0    40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL   100
INPUTS 4
* Th   TI    TIN  GAMMAI
 1,4  4,1   1,1   2,1
 15.  30    100   0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAXCO  4.19  PMAXCO          0.
INPUTS 3
*  Ti    mi   GAMMA
 4,1    4,2   2,1
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38  SOLAR STORAGE TANK HORIZONTAL NO
AUX. HEAT
PARAMETERS 11
*MODE   VOL      HT    HR  CPF  RHO  k CONFIG UA    RI  TI
 2    TSIZE    HEIGHT  HR  4.19 1000 0   2   ULOSS  1   TI
INPUTS 5
* TH   MH   TL    MMAX  TENV
 1,1   1,2  14,1  13,2  0,0
 30.0  30.0  30.0   0.0   TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2  3  9   4
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
 0      10 60
*OUTPUTS 2
*1,CAP 2,COP

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER THPI
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  2     0     70
INPUTS 4
* Th   TI    TIN   HPCONI
 TSET 4,3   THPO   22,1
 60.0  TI    70    0.
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAX    4.19   2800         0.0
INPUTS 3
*  Ti    mi   GAMMA
 14,1  14,2   PCON
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL
PARAMETERS 20
*MODE   VOL   CPF   RHO   UT   HI    AUXMOD    NODE1
NODETI
  1   ELTNK  4.19  1000  ULOSS ELHGT    1       1      1
*TSET  DTDB  QAUX1    NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   0    QMAX   1     1     TSET    0    0.0
```

```
*UAFLUE   TFLUE   TBOIL
  0.0     TENV   100
INPUTS 5
* TH   MH    TL    ML    TENV
  THPO 4,4   39,1   ML    0,0
  TI    0.0  TI    0.0   TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*         7,dEtank 8,Qaux1

***  OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK
PARAMETERS 24
*DTP  TON  TOFF  LU  OMODE
 -1   START  STOP  58  2  1  0  0 0 0 -4 -4  -4  -4 0  -4
 0  -4  0  -4 0 -4 0 -4
INPUTS  9
*DE  QAUX QCOND QU QLOSSS QLOSSH QLOAD  PEL  QLOSO
DE   14,8 QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
LABELS  9
QU[kJ]  QCOND[kJ]  QAUX[KJ] DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ]
QLOAD[kJ]  PEL[kJ] QLOSO[KJ]
CHECK 0.90  1, 2, 3, -4, -5, -6, -7

END
```

```
ASSIGN C:\annette\SSHP1.LST          6
ASSIGN C:\annette\SSHP1.OUT          58

********************************************************
*  This deck predicts the thermal performance of a hospital
*  SERIES SOLAR ASSISTED FULL CAPACITY HEAT PUMP
*  WATER HEATING SYSTEM
*  (no freeze control)
********************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE5.DAT       42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 5 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2
*determination of solar storage tank inputs (unit 4)
TL4=[42,3]*(1-[12,1])+[39,1]*[12,1]
ML4=[23,2]*HPCON+ML*[12,1]

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
```

```
* intercept efficiency
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 2 HEAT TRANSFER AT EVAPORATOR
* evaporator inlet temperature from solar storage tank
TSOURCE=[4,3]
* maximum mass flow rate through evaporator heat exchanger according
* to heat pump data file
MMAXEV=63000

EQUATIONS 13 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[4,3]
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
THPO=THPOC
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]

EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]*(1- [12,1])
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 8 HEAT PUMP TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
NODES = 5
```

```
* determination of heat pump tank inputs (unit 14)
* determination of heat pump tank inputs (unit 14)
TI14=THPO*(1-[12,1])*[22,1]*[32,1]
MI14=[13,2]*(1-[12,1])*[22,1]*[32,1]
TL14=[39,1]*(1-[12,1])+[4,3]*[12,1]
ML14=ML

EQUATIONS 6 ENERGY BALANCE -> UNIT 58
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START = 1
* simulation stop time
STOP = 8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19   TYPE 9   DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1     14
*OUTPUTS:3,Idn 4,I 5,Tdb
```

```
UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39   TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE   N  dT(HOURS) TMAINS   LU  FRMT
   -2    1     24     -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR  PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY      LAT  SC
SHIFT  SMOOTH IE
   7       1        1     STRTDAY    LAT  SC  SHIFT    2    -1
INPUTS 7
* I(kJ/m2-hr)  td1    td2    RHOG    BETA1    GAMMAI
 19,4    19,3    19,19   19,20   RHOG     SLOPE    GAMMAI
 0.0     0.0       0.0     0.0     RHOG     SLOPE    GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***
```

```
UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N AREA Cp EFFMD G ao a1 a2 EFF CpHX
OPTMD bo b1
  1 1 AREA CPHOT 1 50 FRta FRUL 0. EFF 4.2 1 0.1
0.0
INPUTS 10
* Ti mCOLL(kg/hr) mHX Tamb It I Id RHOG THETA
BETA(SLOPE)
 3,1 3,2        3,2 19,5 16,6 16,4 16,5 0,0 16,9 16,10
 TI 0.0         0.0 20.0 0.0 0.0 0.0 RHOG 0.0 40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow Tmax
  11 DEADH DEADL 100
INPUTS 4
* Th Tl TIN GAMMAI
  1,4 4,1 1,1 2,1
  15. 30 100 0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX Cp Pmax(KJ/HR) fpar
 MMAXCO 4.19 PMAXCO 0.
INPUTS 3
* Ti mi GAMMA
 4,1 4,2 2,1
  TI 0.0 0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38 SOLAR STORAGE TANK HORIZONTAL NO
AUX. HEAT
PARAMETERS 11
*MODE VOL HT HR CPF RHO k CONFIG UA RI TI
 2 TSIZE HEIGHT HR 4.19 1000 0 2 ULOSS 1 TI
INPUTS 5
* TH MH TL ML TENV
 1,1 1,2 TL4 ML4 0,0
 30.0 30.0 30.0 0.0 TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*       7,dEtank 8,Qaux1

UNIT 12 TYPE 2 HP BYPASS CONTROL
PARAMETERS 4
* NSTK dThigh dTlow Tmax
  11 2 0 100
INPUTS 4
* Th Tl TIN BYPI
 4,3 TSET 0,0 12,1
 60 60 100 0.
*OUTPUTS: 1,BYPo (CONTROL FUNCTION)

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP
HP-FILE5.DAT)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2 3 9 4
INPUTS 3
*GAMMA X1 X2
 HPCON TSOURCE THPO
0 10 60
*OUTPUTS 3
*1,CAP 2,COP 3,TLEAVE

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER T3
PARAMETERS 4
* NSTK dThigh dTlow Tmax
  11 DHC22 DLC22 85
INPUTS 4
* Th Tl TIN HPCONI
 TSET 14,14 THPO 22,1
 60.0 TI 85 0.
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)
```

```
UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX   Cp   Pmax(KJ/HR)  fpar
  MMAX  4.19  2800      0.0
INPUTS 3
*  Ti   mi  GAMMA
   14,1  14,2  PCON
   TI   0.0  0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL NO
AUX. HEATER
PARAMETERS 20
*MODE  VOL  CPF   RHO   UT   HI    AUXMOD   NODE1
NODETI
  2  ELTNK  4.19  1000  ULOSS ELHGT    1     1     1
*TSET  DTDB  QAUX1   NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   0    0     1    1     TSET    0    0.0
*UAFLUE  TFLUE  TBOIL
 0.0    TENV   100
INPUTS 5
* TH  MH   TL   ML   TENV
 TI14 MI14  TL14  ML14   0,0
 TI   0.0  TI   0.0  TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*       7,dEtank 8,Qaux1

UNIT 23 TYPE 3 PUMP HEAT PUMP EVAPORATOR
PARAMETERS 4
* mMAX   Cp   Pmax(KJ/HR)  fpar
 MMAXEV  4.19  2500      0.0
INPUTS 3
*  Ti   mi  GAMMA
  42,3  4,4   PCON
  TI   0.0  0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

*** OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK SERIES SYSTEM
PARAMETERS 22
*DTP  TON  TOFF  LU  OMODE
 -1  8761  STOP  58  2  1  0  0 0 -4  -4  -4 0  -4  0  -4  0  -4 0 -4 0 -4
INPUTS  8
*DE  QCOND  QU  QLOSSS QLOSSH  QLOAD  PEL QLOSO
DE   QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
LABELS  8
QU[kJ]  QCOND[kJ]  DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ] QLOAD[kJ]
PEL[kJ] QLOSO[KJ]
CHECK 0.90  1, 2, -3, -4, -5, -6

END
```

```
ASSIGN C:\annette\SSHP2.LST           6
ASSIGN C:\annette\SSHP2.OUT           58

*****************************************************
* This deck predicts the thermal performance of a hospital
* SERIES SOLAR ASSISTED FULL CAPACITY HEAT PUMP
* WATER HEATING SYSTEM
*****************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE7.DAT       42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 6 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2
*determination of solar storage tank inputs (unit 4)
TL4=TLEAVE*(1-[12,1])+[39,1]*[12,1]
ML4=[23,2]*HPCON+ML*[12,1]
* bypass temperature
TBYP=60


CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
```

```
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 4 HEAT TRANSFER AT EVAPORATOR
* evaporator inlet temperature from solar storage tank
TSOURCE=[4,3]
* maximum mass flow rate through evaporator heat exchanger according
* to heat pump data file
MMAXEV=63000
MEV=MAX(.00001,[23,2])
* evaporator outlet temperature in solar storage tank
TLEAVE=[4,3]-[42,3]*[49,1]*HPCON*3600/(MEV*4.19)

EQUATIONS 14 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[4,3]
*heat pump capacity fraction (parameter)
FCAP=0.5
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220*FCAP
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
THPO=THPOC
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]

EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[32,1]*[22,1]*(1- [12,1])
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 9 HEAT PUMP TANK
```

```
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
NODES = 5
* electr. power input aux. heater
QMAX=1500*3600
* determination of heat pump tank inputs (unit 14)
TI14=THPO*(1-[12,1])*[22,1]*[32,1]
MI14=[13,2]*(1-[12,1])*[22,1]*[32,1]
TL14=[39,1]*(1-[12,1])+[4,3]*[12,1]
ML14=ML

EQUATIONS 6 ENERGY BALANCE -> UNIT 58
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START = 1
* simulation stop time
STOP = 8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19 TYPE 9 DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE LU
 -1 14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0 6.0 1.0 6.0 7.0 20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39 TYPE 9 DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE N dT(HOURS) TMAINS LU FRMT
  -2 1 24 -1 1 0 16 0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0 6.0 1.0 6.0 7.0 20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR PARAMETERS 9
* RADMODE TRACKMODE TILTMODE DAY LAT SC
SHIFT SMOOTH IE
  7 1 1 STRTDAY LAT SC SHIFT 2 -1
INPUTS 7
* I(kJ/m2-hr) td1 td2 RHOG BETA1 GAMMAI
```

```
19,4  19,3   19,19  19,20  RHOG    SLOPE    GAMMAI
0.0   0.0     0.0   0.0   RHOG   SLOPE   GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp   EFFMD  G   ao    a1    a2  EFF CpHX
OPTMD  bo  b1
  1   1  AREA  CPHOT   1    50  FRta  FRUL  0.  EFF 4.2   1    0.1
0.0
INPUTS 10
* Ti  mCOLL(kg/hr) mHX  Tamb   It    I    Id    RHOG  THETA
BETA(SLOPE)
 3,1  3,2          3,2  19,5  16,6 16,4  16,5   0,0    16,9    16,10
 TI   0.0          0.0  20.0   0.0  0.0   0.0   RHOG    0.0    40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL   100
INPUTS 4
* Th   Tl    TIN  GAMMAI
  1,4  4,1   1,1   2,1
  15.  30    100   0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
  MMAXCO  4.19  PMAXCO         0.
INPUTS 3
*  Ti    mi   GAMMA
  4,1    4,2   2,1
   TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38  SOLAR STORAGE TANK HORIZONTAL NO
AUX. HEAT
PARAMETERS 11
*MODE   VOL      HT    HR  CPF  RHO  k CONFIG UA    RI  TI
  2   TSIZE   HEIGHT  HR  4.19 1000 0   2   ULOSS  1   TI
INPUTS 5
* TH   MH   TL    ML     TENV
 1,1   1,2  TL4   ML4    0,0
 30.0  30.0  30.0   0.0    TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*         7,dEtank 8,Qaux1

UNIT 12 TYPE 2 HP BYPASS CONTROL
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  0     0     100
INPUTS 4
* Th   Tl    TIN    BYPI
  4,3  TBYP   0,0   12,1
  60  60     100   0.
*OUTPUTS: 1,BYPo (CONTROL FUNCTION)

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP
HP-FILE6.DAT)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2  3  10   5
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
0     10     60
*OUTPUTS 3
*1,CAP 2,COP 3,QEVAP

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER THPI
PARAMETERS 4
```

```
* NSTK dThigh dTlow  Tmax
  11  DHC22   DLC22  85
INPUTS 4
* Th  TI   TIN  HPCONI
  TSET THPI  THPO   22,1
  60.0  TI   60.0   1.0
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)

UNIT 32 TYPE 2 HEAT PUMP CONTROLLER TSOURCE
(FREEZE CONTROL)
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  0   0  85
INPUTS 4
* Th       TI  TIN   FRCONI
  TLEAVE  0,0 0,0   32,1
  TI       1.0 60.0  1.0
*OUTPUTS: 1,FRCONo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP CONDENSER
PARAMETERS 4
* mMAX    Cp   Pmax(KJ/HR)   fpar
 MMAXCON   4.19   2500        0.0
INPUTS 3
*  Ti   mi   GAMMA
   14,1  14,2   PCON
   TI   0.0   1.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL W/
AUXILIARY HEATER
PARAMETERS 20
*MODE   VOL   CPF   RHO   UT   HI    AUXMOD   NODE1
NODETI
  2   ELTNK  4.19  1000  ULOSS ELHGT     1       1       1
*TSET  DTDB  QAUX1    NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET   0    QMAX      1      1       TSET    0    0.0
*UAFLUE   TFLUE   TBOIL
  0.0    TENV   100
INPUTS 5
* TH  MH    TL   ML    TENV
  TI14 MI14  TL14  ML14    0,0
  TI    0.0  TI    0.0  TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

UNIT 23 TYPE 3 PUMP HEAT PUMP EVAPORATOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
  MMAXEV   4.19   2500        0.0
INPUTS 3
*  Ti    mi   GAMMA
   TLEAVE  4,4   PCON
   TI    0.0   1.0
*OUTPUTS: 1,To 2,mo 3,Ppump

***  OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK SERIES SYSTEM
PARAMETERS 22
*DTP  TON  TOFF  LU  OMODE
 -1   8761  STOP  58  2  1  0  0 0 -4  -4  -4 0  -4  0  -4  0  -4 0 -4 0 -4
INPUTS  8
*DE  QCOND  QU  QLOSSS QLOSSH  QLOAD  PEL QLOSO
DE   QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
LABELS  8
QU[kJ]  QCOND[kJ]  DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ] QLOAD[kJ]
PEL[kJ] QLOSO[KJ]
CHECK 0.90  1, 2, -3, -4, -5, -6

END
```

```
ASSIGN C:\annette\DSHP1.LST          6
ASSIGN C:\annette\DSHP1.OUT          58

*******************************************************
* This deck predicts the thermal performance of a hospital
* DUAL SOURCE SOLAR ASSISTED FULL CAPACITY HEAT
* PUMP WATER HEATING SYSTEM
*******************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE5.DAT       42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

***  SYSTEM PARAMETERS  ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank.  The following equations
* account for the tempering valve.  They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 5 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2
*determination of solar storage tank inputs (unit 4)
TL4=[42,3]*(1-[12,1])+[39,1]*[12,1]
ML4=[23,2]*HPCON+ML*[12,1]

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
```

```
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 3 HEAT TRANSFER AT EVAPORATOR
* ground temperature
TGR=7.2
* evaporator inlet temperature
TSOURCE=[4,3]*(1-[32,1])+TGR*[32,1]
* maximum flow rate through evaporator hx according to heat pump
* data file
MMAXEV=63000

EQUATIONS 13 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[4,3]
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
THPO=THPOC
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]

EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]*(1- [12,1])
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 8 HEAT PUMP TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
```

```
NODES = 5
* determination of heat pump tank inputs (unit 14)
* temperature from heat source is either heat pump outlet temperature or
* solar storage tnak outlet temperature, when heat pump is bypassed
TI14=THPO*(1-[12,1])
* mass flow from heat source is either condenser pump flowrate or
* MLOAD
MI14=[13,2]*(1-[12,1])
* temperature of replacement fluid is either TMAINS or irrelevant, when
* heat pump is bypassed
TL14=[39,1]*(1-[12,1])+[4,3]*[12,1]
* mass flow to load = mass flow of replacement fluid is either MLOAD
* or zero, when heat pump is bypassed
ML14=ML

EQUATIONS 6 ENERGY BALANCE -> UNIT 58
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START = 1
* simulation stop time
STOP = 8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120
TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19  TYPE 9  DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE LU
 -1   14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39  TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE  N  dT(HOURS) TMAINS  LU  FRMT
  -2   1    24    -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR
PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY    LAT  SC
SHIFT  SMOOTH IE
  7     1      1    STRTDAY   LAT  SC  SHIFT   2   -1
INPUTS 7
```

```
* I(kJ/m2-hr) td1  td2  RHOG  BETA1  GAMMAI
 19,4  19,3  19,19  19,20  RHOG  SLOPE  GAMMAI
 0.0  0.0   0.0  0.0  RHOG  SLOPE  GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp  EFFMD  G  ao  a1  a2  EFF CpHX
OPTMD  bo  b1
  1  1  AREA  CPHOT  1   50  FRta  FRUL  0.  EFF 4.2  1   0.1
0.0
INPUTS 10
* Ti  mCOLL(kg/hr) mHX  Tamb  It  I  Id  RHOG  THETA
BETA(SLOPE)
 3,1  3,2      3,2  19,5  16,6 16,4  16,5  0,0   16,9  16,10
 TI  0.0       0.0  20.0  0.0  0.0  0.0  RHOG  0.0  40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL  100
INPUTS 4
* Th  Tl   TIN  GAMMAI
 1,4  4,1  1,1  2,1
 15.  30   100  0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX   Cp   Pmax(KJ/HR)  fpar
 MMAXCO  4.19  PMAXCO     0.
INPUTS 3
*  Ti   mi  GAMMA
 4,1   4,2  2,1
  TI   0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38  SOLAR STORAGE TANK HORIZONTAL NO
AUX. HEAT
PARAMETERS 11
*MODE  VOL    HT   HR  CPF  RHO  k CONFIG UA   RI  TI
 2   TSIZE   HEIGHT  HR  4.19 1000 0  2  ULOSS  1   TI
INPUTS 5
* TH   MH   TL   ML    TENV
 1,1  1,2  TL4  ML4   0,0
 30.0  30.0  30.0   0.0   TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*       7,dEtank 8,Qaux1

UNIT 12 TYPE 2 HP BYPASS CONTROL
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11 DEADH DEADL 100
INPUTS 4
* Th   Tl   TIN   BYPI
 4,3  TSET   0,0   12,1
 60  60    100   0.
*OUTPUTS: 1,BYPo (CONTROL FUNCTION)

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP
HP-FILE5.DAT)
PARAMETERS 5
*LU NX NY NX1 NX2
 42 2  3  9 4
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
0     10 60
*OUTPUTS 3
*1,CAP 2,COP 3,TLEAVE

UNIT 22 TYPE 2 HEAT PUMP CONTROLLER T3
```

```
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22   85
INPUTS 4
* Th   TI      TIN   HPCONI
 TSET 14,14   THPO   22,1
 60.0  TI     85    0.
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)

UNIT 32 TYPE 2 HEAT PUMP CONTROLLER TGR
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22   100
INPUTS 4
* Th   TI    TIN  GCONI
 TGR  4,3   0,0   32,1
 60.0  TI     100    0.
*OUTPUTS: 1,GCONo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAX   4.19   2500        0.0
INPUTS 3
*  Ti    mi   GAMMA
  14,1  14,2   PCON
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL
PARAMETERS 20
*MODE   VOL   CPF    RHO    UT    HI     AUXMOD    NODE1
NODETI
  1    ELTNK  4.19  1000  ULOSS ELHGT     1       1       1
*TSET   DTDB   QAUX1    NODE2   NODET2   TSET2   DTDB2
QAUX2
 TSET    0     0       1       1       TSET     0     0.0
*UAFLUE   TFLUE   TBOIL
 0.0     TENV    100

INPUTS 5
* TH   MH    TL    ML    TENV
 TI14 MI14  TL14  ML14    0,0
 TI    0.0  TI    0.0   TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

UNIT 23 TYPE 3 PUMP HEAT PUMP EVAPORATOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAXEV   4.19   2500        0.0
INPUTS 3
*  Ti    mi   GAMMA
  42,3  4,4   PCON
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

***  OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK SERIES SYSTEM
PARAMETERS 22
*DTP  TON  TOFF  LU  OMODE
 -1   8761  STOP  58  2  1  0  0 0 -4  -4  -4 0  -4  0  -4  0  -4 0 -4 0 -4
INPUTS  8
*DE  QCOND QU QLOSSS QLOSSH QLOAD  PEL  QLOSO
DE   QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
LABELS  8
QU[kJ]  QCOND[kJ]  DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ] QLOAD[kJ]
PEL[kJ] QLOSO[KJ]
CHECK 0.90  1, 2, -3, -4, -5, -6

END
```

```
ASSIGN C:\annette\DSHP2.LST          6
ASSIGN C:\annette\DSHP2.OUT          58

********************************************************
* This deck predicts the thermal performance of a hospital
* DUAL SOURCE SOLAR ASSISTED HEAT PUMP WATER
* HEATING SYSTEM WITH AUXILIARY HEATER
********************************************************

*** FILE ASSIGNMENTS ***

* mains water temperature according to f-chart weather data
ASSIGN C:\annette\MDSMAIN.DAT        16
* heat pump data file according to EES model
ASSIGN C:\annette\HP-FILE6.DAT       42
* TMY weather data file Madison,WI
ASSIGN c:\annette\MADISN.WI          14

*** SYSTEM PARAMETERS ***

EQUATIONS 4
* latitude Madison, WI
LAT=43.1
* hot water set temperature
TSET = 60
* tank environment temperature
TENV = 18
* initial water mains temperature Madison
TI=6.4

EQUATIONS 6 LOAD
* unit conversion
FACTOR = 3.7853
BEDS=220
SCALE = FACTOR*BEDS
* determine the required draw from the tank. The following equations
* account for the tempering valve. They result from simplified mass
* and energy balances where Cp is assumed constant. For case, that
* the water from the mains is heated to a higher temperature than
*   required, the water draw from the tank is reduced and additional water
* from the mains   is added to keep the required water flow and reach the
* required temperature.
TDIFF = MAX(0.000001,([4,3]-[39,1]))
TNKDRW = MIN(1,((TSET-[39,1])/TDIFF))
MLOAD = [29,2]*TNKDRW*SCALE

CONSTANTS 4 SOLAR STORAGE TANK
* tank volume (parameter) & unit conversion
TNKSIZE =  2.0000E+0004
TSIZE = .0037854*TNKSIZE
* tank height (parameter) & unit conversion
HEIGHT1 = 11.5
HEIGHT = .3048*HEIGHT1

EQUATIONS 3 TANK LOSSES
RVAL=16.0
* unit conversion
RVAL1=RVAL*.0489194
ULOSS=1/RVAL

EQUATIONS 5 SOLAR STORAGE TANK
*height of collector return to tank above bottom of tank
HR=HEIGHT
*height of thermostat above bottom of tank
HTH=HEIGHT-0.2
*height of auxiliary above bottom of tank
HA=HEIGHT-0.2
*determination of solar storage tank inputs (unit 4)
TL4=[42,3]*(1-[12,1])+[39,1]*[12,1]
ML4=[23,2]*HPCON+ML*[12,1]

CONSTANTS 7 COLLECTORS
* number of collectors in array (parameter)
COL = 450
* area of a single collector
AREA1 =  1.7500E+0001
AREA = COL*AREA1*.0929
* intercept efficiency
```

```
FRta =  7.0000E-0001
* slope of efficiency curve & unit conversion
FRUL1 =  7.4000E-0001
FRUL=20.4418*FRUL1
* collector slope
SLOPE =  5.3000E+0001

EQUATIONS 6 COLLECTORS
* maximum collector flow rate
MMAXCOL=30000*COL/291
* maximum pump power input
PMAXCOL=14000*COL/291
* controls deadbands
DEADH = 25
DEADL = 2
CONST=1
* control solar storage tank draw
SDCON=[12,1]

EQUATIONS 3 HEAT EXCHANGER
* effectiveness
EFF =  5.0000E-0001
* specific heat of collector side fluid & unit conversion
CPH1 =  8.5000E-0001
CPHOT = 4.1868*CPH1

EQUATIONS 5 RADIATION PROCESSOR
RHOG=2.0000E-01
STRTDAY = INT(1+START1/24)
GAMMAI=0.0000E+0
SC=4871
SHIFT=0.0

EQUATIONS 3 HEAT TRANSFER AT EVAPORATOR
* ground temperature
TGR=7.2
* evaporator inlet temperature
TSOURCE=[4,3]*(1-[32,1])+TGR*[32,1]
* maximum flow rate through evaporator hx according to heat pump
* data file
MMAXEV=63000

EQUATIONS 13 HEAT TRANSFER FROM HEAT PUMP
CONDENSER TO WATER STREAM
MHPCH=[13,2]
THPI=[4,3]
QCOND=HPCON*[49,1]*[42,1]*3600*BEDS/220
*pump mass flow limit
MMAX=5900
*heat pump water outlet temperature
MHPCHH=MAX(0.1,MHPCH)
THPO=THPI+QCOND/(4.19*MHPCHH)
THPO=THPOC
* mass flow through heat pump heat exchanger
DELTAT = MAX(0.000001,([14,3]-[39,1]))
DRAW = MIN(1,((TSET-[39,1])/DELTAT))
ML=[29,2]*SCALE*DRAW
THPOO=THPO*HPCON
TLO=[14,3]
MLO=[14,4]

EQUATIONS 2 HEAT PUMP CONTROLLER T3
DHC22=2
DLC22=2

EQUATIONS 2 HEAT PUMP CONTROL
*heat pump control (on day/off night)
HPCON=[22,1]*(1- [12,1])
* pump control (on day/off night)
PCON=HPCON*[49,1]/7

EQUATIONS 9 HEAT PUMP TANK
* tank volume
ELTNK=1500*.0037854
*tank height
HEIEL = .3048*13.00
ELHGT=-HEIEL
* nodes
```

```
NODES = 5
* maximum electr. power input
Qmax=1500*3600
* determination of heat pump tank inputs (unit 14)
* temperature from heat source is either heat pump outlet temperature or
* solar storage tnak outlet temperature, when heat pump is bypassed
TI14=THPO*(1-[12,1])
* mass flow from heat source is either condenser pump flowrate or
* MLOAD
MI14=[13,2]*(1-[12,1])
* temperature of replacement fluid is either TMAINS or irrelevant, when
* heat pump is bypassed
TL14=[39,1]*(1-[12,1])+[4,3]*[12,1]
* mass flow to load = mass flow of replacement fluid is either MLOAD
* or zero, when heat pump is bypassed
ML14=ML

EQUATIONS 6 ENERGY BALANCE -> UNIT 58
QLOSSS=[4,5]
QLOSSH=[14,5]
QLOAD=[14,6]
QLOSO=[4,6]
DE=[4,7] + [14,7]
* electr. power input heat pump
COP=MAX([42,2],0.0001)
PEL=HPCON*QCOND/COP

*** SIMULATION PARAMETERS ***

EQUATIONS 3
* simulation start time
START = 1
* simulation stop time
STOP = 8760
* simulation time step
STEP = .1

SIMULATION START STOP STEP
LIMITS 120 120 120

TOLERANCES 0.001 0.001
WIDTH 72

* ** DATA READERS ***

UNIT 19  TYPE 9  DATA READER FOR WEATHER MADISON
PARAMETERS 2
* MODE  LU
 -1    14
*OUTPUTS:3,Idn 4,I 5,Tdb

UNIT 29 TYPE 14 FORCING FUNCTION FOR WATER DRAW
HOSPITAL
* water draw: gal/hr-bed
PARAMETERS 12
* t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
* OUTPUTS: 1,vbar 2,v

UNIT 39  TYPE 9  DATA READER FOR MAINS WATER
TEMPERATURE
* changes monthly but read in daily from f-chart file
PARAMTERS 8
* MODE  N  dT(HOURS) TMAINS   LU  FRMT
  -2   1    24    -1 1 0   16   0
* OUTPUTS: 1, TMAINS

UNIT 49 TYPE 14 TIMER CONTROL LEVELS DAY / NIGHT
PARAMETERS 12
*t0 v0 t1 v0 t1 v1 t2 v1 t2 v0 t3 v0
0.0 1.0  6.0 1.0  6.0 7.0  20.0 7.0 20.0 1.0
24.0 1.0
*OUTPUTS:1,TCBAR 2,TC

UNIT 16 TYPE 16 RADIATION PROCESSOR
PARAMETERS 9
* RADMODE TRACKMODE TILTMODE  DAY     LAT  SC
SHIFT  SMOOTH IE
```

```
   7      1      1    STRTDAY   LAT  SC  SHIFT   2    -1
INPUTS 7
* I(kJ/m2-hr)  td1   td2   RHOG   BETA1   GAMMAI
 19,4   19,3   19,19  19,20  RHOG    SLOPE    GAMMAI
 0.0   0.0      0.0   0.0    RHOG    SLOPE    GAMMAI
*OUTPUTS: 1,Io 2,THETAz 3,GAMMAs 4,I 5,Id 6,IT1 7,IbT1 8,IdT1
9,THETA1 10,BETA1 11,IT1
* Inputs 7,8: INext(IF SMOOTH=1) 19,24 19,23  0.0 0.0

*** SYSTEM COMPONENTS ***

UNIT 1 TYPE 1 COLLECTOR
PARAMETERS 14
* MODE N  AREA  Cp   EFFMD  G   ao    a1    a2  EFF CpHX
OPTMD  bo  b1
  1   1  AREA  CPHOT   1    50  FRta  FRUL  0.  EFF 4.2   1    0.1
0.0
INPUTS 10
* Ti  mCOLL(kg/hr) mHX  Tamb   It    I    Id    RHOG  THETA
BETA(SLOPE)
 3,1  3,2          3,2  19,5  16,6 16,4  16,5   0,0    16,9    16,10
 TI   0.0          0.0  20.0   0.0  0.0   0.0  RHOG    0.0     40.0
*OUPUTS: 1, To 2,mo 3,Qgain(KJ/HR) 4,Tco

UNIT 2 TYPE 2 COLLECTOR PUMP CONTROLLER
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DEADH  DEADL   100
INPUTS 4
* Th   TI    TIN  GAMMAI
 1,4  4,1    1,1   2,1
 15.  30     100    0.
*OUTPUTS: 1,GAMMAo (CONTROL FUNCTION)

UNIT 3 TYPE 3 PUMP COLLECTOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAXCO  4.19  PMAXCO          0.
INPUTS 3
* Ti    mi   GAMMA
 4,1    4,2   2,1
  TI   0.0    0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 4 TYPE 38  SOLAR STORAGE TANK HORIZONTAL NO
AUX. HEAT
PARAMETERS 11
*MODE   VOL      HT    HR  CPF  RHO  k CONFIG UA    RI  TI
 2    TSIZE    HEIGHT  HR  4.19 1000 0   2   ULOSS  1   TI
INPUTS 5
* TH   MH   TL    ML     TENV
 1,1   1,2  TL4   ML4    0,0
 30.0 30.0  30.0   0.0    TENV
*UNIT 4 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

UNIT 12 TYPE 2 HP BYPASS CONTROL
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11 DEADH DEADL 100
INPUTS 4
* Th   TI    TIN    BYPI
 4,3  TSET   0,0   12,1
 60  60     100    0.
*OUTPUTS: 1,BYPo (CONTROL FUNCTION)

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT
PARAMETERS 5
*LU NX NY NX1 NX2
42 2  3  9 5
INPUTS 3
*GAMMA X1 X2
 HPCON  TSOURCE THPO
0      10 60
*OUTPUTS 3
*1,CAP 2,COP 3,TLEAVE
```

```
UNIT 22 TYPE 2 HEAT PUMP CONTROLLER T3
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22   85
INPUTS 4
* Th  Tl     TIN  HPCONI
 TSET 14,14   THPO   22,1
 60.0  TI     100    0.
*OUTPUTS: 1,HPCONo (CONTROL FUNCTION)

UNIT 32 TYPE 2 HEAT PUMP CONTROLLER TGR
PARAMETERS 4
* NSTK dThigh dTlow  Tmax
  11  DHC22  DLC22   100
INPUTS 4
* Th   Tl    TIN  GCONI
 TGR  4,3   0,0   32,1
 60.0  TI    100    0.
*OUTPUTS: 1,GCONo (CONTROL FUNCTION)

UNIT 13 TYPE 3 PUMP HEAT PUMP
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAX   4.19   2500       0.0
INPUTS 3
*  Ti    mi   GAMMA
  14,1  14,2   PCON
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

UNIT 14 TYPE 4  HOT WATER STORAGE TANK VERTICAL
PARAMETERS 20
*MODE  VOL  CPF   RHO   UT   HI    AUXMOD    NODE1
NODETI
  1   ELTNK  4.19  1000  ULOSS ELHGT     1       1      1
*TSET  DTDB  QAUX1    NODE2  NODET2  TSET2  DTDB2
QAUX2
 TSET    0    0      1      1      TSET     0     0.0
*UAFLUE   TFLUE   TBOIL
  0.0    TENV   100
INPUTS 5
* TH   MH    TL    ML    TENV
 TI14 MI14  TL14  ML14   0,0
 TI    0.0  TI    0.0  TENV
DERIVATIVES NODES
TSET TSET TSET TSET TSET TSET TSET TSET TSET
*UNIT 14 OUTPUTS:
*OUTPUTS: 1,Trtn 2,m_rtnCOLL 3,Tload 4,m_load 5,Qenv,loss 6,Qs
*        7,dEtank 8,Qaux1

UNIT 23 TYPE 3 PUMP HEAT PUMP EVAPORATOR
PARAMETERS 4
* mMAX    Cp    Pmax(KJ/HR)   fpar
 MMAXEV   4.19   2500       0.0
INPUTS 3
*  Ti    mi   GAMMA
  42,3  4,4   PCON
  TI    0.0   0.0
*OUTPUTS: 1,To 2,mo 3,Ppump

***  OUTPUT ***

UNIT 58 TYPE 28 ENERGY BALANCE CHECK SERIES SYSTEM
PARAMETERS 22
*DTP  TON  TOFF  LU  OMODE
 -1  8761  STOP  58  2  1  0  0 0 -4  -4  -4 0  -4  0  -4  0  -4 0 -4 0 -4
INPUTS  8
*DE  QCOND QU QLOSSS QLOSSH QLOAD  PEL  QLOSO
DE   QCOND  1,3 QLOSSS QLOSSH  QLOAD  PEL QLOSO
LABELS  8
QU[kJ]  QCOND[kJ]  DE[kJ]  QLOSSS[kJ]  QLOSSH[kJ] QLOAD[kJ]
PEL[kJ] QLOSO[KJ]
CHECK 0.90  1, 2, -3, -4, -5, -6

END
```

# APPENDIX C

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