

SIMULATION OF SOLAR HEAT PUMP SYSTEMS
AND
THE "PARALLEL" SYSTEM DESIGN PROCEDURE

BY

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NOMENCLATURE

A_c	collector area
C_{min}	minimum capacitance rate of heat exchanger flow streams
C_p	specific heat
COP	instantaneous heat pump coefficient of performance
COP_m	monthly heat pump coefficient of performance
COP_s	seasonal heat pump coefficient of performance
\overline{COP}	instantaneous heat pump system coefficient of performance
\overline{COP}_m	monthly heat pump system coefficient of performance
\overline{COP}_s	seasonal heat pump system coefficient of performance
F	fraction of the load (space and/or hot water) supplied by solar and ambient air sources
f	fraction of the load (space and/or hot water) supplied by solar energy
F_R	collector heat removal factor
HAA	hours of heat pump operation in the air-to-air heating mode
HWA	hours of heat pump operation in the water-to-air heating mode
\dot{m}	mass flow rate
N	number of hours in temperature range ("bin") or number of collector glazings
\dot{Q}, q	energy transfer rate
QAIR	energy extracted by the heat pump from ambient air in the air-to-air heating mode
QAUX	auxiliary energy requirements (space and/or hot water)
QDH	energy supplied to house in direct heating mode by solar energy

Nomenclature (cont.)

QLOAD	heating (space and/or hot water) or cooling (space) load
QRH	energy rejected to the house by the heat pump in the heating mode
QRAA	energy rejected by the heat pump in the air-to-air heating mode
QRWA	energy rejected by the heat pump in the water-to-air heating mode
QU	useful energy collected by solar operation
T	temperature
T_{\min}	minimum storage tank temperature
UA	house heating requirement per room-ambient temperature difference
U_L	collector loss coefficient
V	storage tank volume
WAH	heat pump electrical input in heating mode
ϵ	heat exchanger effectiveness
η	heat pump compressor motor efficiency
ρ	density
$\tau\alpha$	collector transmittance-absorptance product

Subscripts

amb	ambient
b	"bin"
c	collector or cooling
dh	direct heating

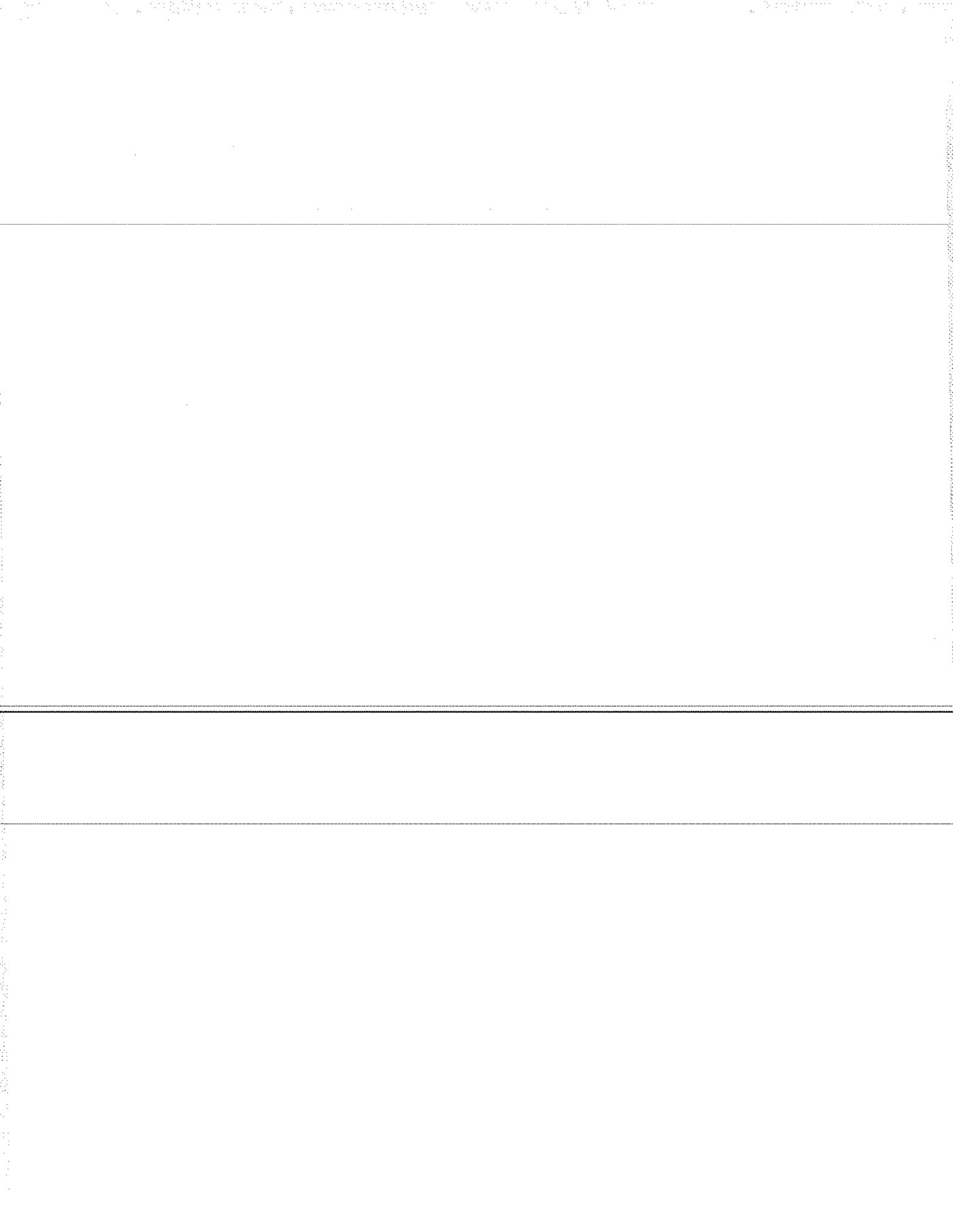
Nomenclature (cont.)

h	heating or hot
hw	hot water
m	monthly
s	seasonal
sp	house, space

ABSTRACT

The work presented here on solar heat pump systems for residential applications is divided into two parts. The first part looks at system simulations done with the transient simulation program, TRNSYS. Five basic solar heat pump systems are simulated and compared from a thermal performance point of view under similar and varying operating conditions. The five systems are "conventional solar," "conventional heat pump," "parallel," "in-line," and "dual source." Operating conditions of importance are collector area and construction (e.g. number of collector glazings), main storage tank volume, house capacity, heating loads for space and/or hot water, cooling loads, minimum storage tank temperature, minimum source temperature for heat pump operation and direct heating with solar energy, heat pump source (ambient air or storage medium), heat pump operating characteristics, system control options (time of day for operation of solar and/or heat pump system), and geographical location. From these simulations, it is determined that "parallel" and "dual source" systems yield the best thermal performance.

The second part of this work presents a general design procedure for the "parallel" system. Although, the "parallel" and "dual source" systems had almost identical thermal performance, the "parallel" system being simpler in construction and therefore, more cost effective merits its choice over the "dual source" system. This general design procedure which incorporates the commonly used procedure (the "Bin"



1.0 INTRODUCTION

1.1 BACKGROUND

Various heating and cooling systems are under study for reductions of the consumption of non-renewable energy resources and, hopefully, the cost of delivered energy. Solar energy and heat pump systems are two of these that look promising. Alone, each system has its shortcomings. Combining the two systems may compensate for some of the individual shortcomings and further reduce the cost of delivered energy. Just how much energy can be saved, and whether the savings will pay for the additional complexity and expense of the combined system has not yet been determined.

In residential applications a heat pump using ambient air as a source in the heating mode is commonly found. Less common is one using a moderate temperature water source since the water source is either not available or too expensive to obtain and use. In the heating mode the coefficient of performance, COP (the ratio of heat output to input electrical energy), and capacity of an air source heat pump increase as ambient temperature increases, and at low ambient temperatures the decrease in capacity is such that the COP approaches unity.

With an air source heat pump in the heating mode, moisture in the outdoor air condenses and freezes on the evaporator coil at or near the freezing point of water requiring that the heat pump periodically switch to the cooling mode to defrost the outdoor coil. This results in a waste of energy and a reduction in system perform-

ance. In Fig. 2.5 from [1] it can be seen that as the ambient temperature decreases, capacity decreases, and the house load increases. Also as seen in Fig. 2.5, at low ambient temperatures below the balance point (a condition where the heating capacity of the heat pump meets the heating demand exactly) an auxiliary heater is required to meet the remainder of the load. The auxiliary heater is usually a strip resistance heater operating at a COP equal to unity. This too reduces overall system performance.

A solar energy system is designed to supply a fraction of the house heating load by solar energy. Although it is possible to design a solar energy system to provide all of the heating requirements, it is not economically feasible to do so. The system would be oversized almost all of the time. A properly designed solar energy system would ~~meet the entire space and service hot water heating requirements during~~ spring, summer and fall, but require supplemental heating in winter. This need for supplemental heating in winter exists because there are times when little or no solar energy is collected, and that which is collected is at storage temperatures not high enough, relative to room temperature, to be useful for direct heating. This situation occurs during the same season that the heat pump system performance is poor, and provides the motivation for combining a solar system with a heat pump system. Additional motivation is provided since heat pump systems can both heat and cool while most solar energy systems lack the capability of space cooling.

A combination of solar energy and heat pump systems appears to alleviate many individual shortcomings. The storage tank or rock bed heated by useful energy from the collector can serve as the heat pump heat source. The problem of frost on the liquid source evaporator could be eliminated because of an ethylene glycol solution in the collector loop lowering the freezing point of the collector loop fluid. The heat pump would see a higher source temperature and, therefore, require less auxiliary. Also, a smaller heat pump might be allowed at these higher source temperatures because capacity and COP would be greater. Since the heat pump would keep the collector fluid cooler, collector radiation and convection losses would decrease and collector performance would improve. Finally, because of the lower collector fluid temperature a single glazing of the collector may be economically justifiable.

There are many possible ways to combine solar energy and heat pump systems. In simulating system performance, many design conditions must be varied and performance parameters compared. Design conditions include collector area and construction, storage size, control options, heat pump source selections (ambient air or storage), heat pump performance, and geographical location. Performance parameters are heat pump and system COP, fraction of the house heating load met by solar energy, and percent auxiliary requirements.

One goal of this study is to compare performance parameters of various systems, and determine the best system configuration from an energy savings and cost point of view.

1.2 LITERATURE SURVEY

In an earlier literature survey done by Freeman [2] it was found that most solar heat pump systems have been evaluated with many simplifications in system performance. The results obtained by this approach may be valid only over a relatively short time period when average conditions prevail, and that more realistic results can only be obtained by considering the time and weather dependency of solar energy systems for an entire heating season.

Freeman also found from work done in the nineteen fifties that solar heat pump systems have a relatively high economic feasibility in the south, and that evaporator-side storage systems require much smaller heat pumps than do condenser-side storage. The General Electric Phase 0 report [3] continued the study of solar heat pump systems by looking at six types of buildings in nine locations with various heating and cooling modes. Systems with dual evaporators selecting the highest temperature heat source by switching between ambient and storage showed a significant thermal advantage. These are some of the basic principles of operation that are used in most of the systems studied to date.

Proceedings of the NSF-sponsored workshops held at Pennsylvania State University [4], June 12-14, 1975, summarized development of solar heat pump systems for heating and cooling of buildings. Studies reported by Rittleman, Gilman, Jardine, Bridgers and Dubin are based on a specific detailed design, while Freeman and Drucker presented

more general studies. In all studies, various assumptions were made to reduce the complexity and quantity of calculations. The use of "average" or "design" weather conditions of radiation and air temperature, constant collector plate and inlet temperature, constant heat pump COP with temperature, and constant storage temperature are a few of the commonly used assumptions. Most of these studies demonstrated a substantial energy savings using solar heat pump systems. None of these studies compared different solar heat pump system configurations to each other and to "conventional solar" and "conventional heat pump" systems. Also, very few specific system design recommendations were made.

Another summarization of solar heat pump systems development was at the proceeding of ISES [5], Winnipeg, Canada, August 15-20, 1976. A study done by Marvin and Mumma for a residence in the Great Lakes area using a commercially available heat pump showed that a system using solar energy for only space heating backed up with an air-to-air heat pump using only ambient air and electric resistance heating performed better than the more popular solar assisted heat pump system that have heat pumps using either ambient or storage depending on which one has the higher temperature.

Simulations of the detailed dynamic behavior of solar assisted heat pump systems have been made by Freeman [2], Karman et al. [6], and Bosio and Suryanarayana [7]. Freeman described a general purpose analytical heat pump model that could be used with the transient simulation program, TRNSYS [8]. Using this heat pump model and

TRNSYS, Freeman looked at an "in-line" system where the heat pump using solar as its source is located between the solar collection loop and the heat load loop, and the "parallel" system where the heat pump using ambient air as its source operates on the heat load independently of the solar system operation.

Bosio and Suryanarayana did a seasonal simulation for Madison, Wisconsin, on what has previously been called an "in-line" system. Unlike the "in-line" system looked at by Freeman, Bosio and Suryanarayana's system did not have the capability of bypassing the heat pump and doing direct heating with solar energy. Nor, did they simulate a "conventional solar" or "conventional heat pump" system and make a comparison on system performance. They mentioned that the main components of a solar assisted heat pump system were the collector, thermal storage and heat pump, and that it was important to vary collector area and operating temperature, and storage capacity, but nothing was mentioned about the effect on system performance by varying heat pump characteristics. They conclude that there is a savings in fossil fuel with a solar assisted heat pump system, and that under certain combinations of interest, oil and electric rates such a system can be competitive with a domestic oil furnace system.

Karman et al. continued Freeman's research on the more usual solar heat pump systems. The performance of the heat pump is determined empirically by interpolating performance data from manufacturers' specifications. Karman et al. also looked at a "dual source" system which has the option of going to ambient air for a source if the ambient temperature exceeds storage temperature. From seasonal

simulations of combined system performance in three locations (Madison, Wisconsin; Albuquerque, New Mexico; and Charleston, South Carolina), they concluded that the "dual source" system is the best from an energy savings standpoint but that economics indicate no clear-cut incentive for the general utilization of solar heat pump systems. However, they did not consider "parallel" performance and compare it to "dual source" performance.

A more unusual solar assisted heat pump system study was done by Abbaspour and Glicksman [9] which looked at low and high side temperature storage (storage at evaporator and condenser of the heat pump). Their simulation study in New York City used monthly averaged temperature data reduced to hourly data, assuming the solar energy is uniform from sunrise to sunset. They found that the solar heat pump system is not economically competitive with conventional heating systems on a total annual cost basis. A two-storage system achieves a modest improvement in performance over a single storage system, but this increase in performance may be outweighed by the additional complexity involved.

To date, very few people have agreed on the best way to combine a heat pump and a solar energy system to produce the greatest savings in consumed energy. Also, no one has arrived at a general design procedure for solar heat pump systems that would allow a person to readily evaluate system performance or design a system to meet a given load.

1.3 PURPOSE

Work at the University of Wisconsin Solar Energy Laboratory has been aimed toward evaluating combinations of solar and heat pump systems, formulating general guidelines for designing solar heat pump systems, and arriving at general design procedures for predicting solar heat pump system performance.

The work presented here is divided into two parts. The first part looks at system simulations done with the transient simulation program, TRNSYS [8]. Five basic systems are simulated and compared under similar and varying conditions. The five systems are "conventional solar," "conventional heat pump," "parallel," "in-line," and "dual source." These systems will be described in detail later.

The second part of this work presents a design procedure for the best performing system. This incorporates the commonly used procedure (the "Bin" method) of determining the performance of an air-to-air heat pump and the f -Chart [10] procedure of determining a solar energy system performance.

2.0 SYSTEM SIMULATIONS

2.1 MODELING OF SYSTEM COMPONENTS

System simulations for this study were performed using the University of Wisconsin solar energy system simulation program, TRNSYS [8]. TRNSYS is a modular solar energy system simulation program written in FORTRAN that solves for the transient performance of system components (collectors, storage, heating and cooling loads, heat pumps, etc.) joined together by information flow which usually represents pipes, ducts, and wires in response to time varying forcing functions like meteorological data. TRNSYS component models are self-contained sub-routines having constant "parameters," user supplied, describing the modeled hardware, time varying "inputs" representing time dependent information flowing into the model, and time varying "outputs" representing time dependent information flow out of the model. TRNSYS includes all the mechanics necessary to govern input/output operations and solve the components' simultaneous algebraic and differential equations repeatedly. A brief review of the major TRNSYS models used is presented below. For a more complete discussion see reference [8].

2.1.1 WEATHER DATA

This work is primarily concerned with comparing performance of different systems in response to identical forcing functions (meteorological data). Therefore, whether it is meteorological data representative of the expected life of the system (10 to 20 years), or an

"average," a "design," or "typical" year is not extremely important.

This being the case, "design" years consisting of actual hourly readings from local weather bureaus of dry bulb temperature and total radiation on a horizontal surface are used as the weather model for the simulations.

2.1.2 THE COLLECTOR

The collector model used in this work is the flat-plate collector model of Hottel and Whillier [11]. The heat removal factor, transmittance-absorptance product, and loss coefficient are held constant throughout the simulations, and have typical values arrived at by the methods outlined by Duffie and Beckman [12].

Hourly values of solar energy incident on the tilted collector surface are determined from hourly values of total radiation on a horizontal surface by the Liu and Jordan method [13]. This method involves taking the total radiation on a horizontal surface and separating it into beam and diffuse components. The beam component is corrected for incidence angle on the tilted collector surface and the diffuse component is assumed to be evenly distributed throughout the collector-to-sky view factor.

2.1.3 STORAGE

All solar heat pump systems studied here used a completely insulated liquid storage tank (no losses). The liquid storage tank is modeled as a fully-mixed tank.

2.1.4 SPACE HEATING AND COOLING LOADS

The building used in every simulation is a well insulated single-family residence of 120 square meters floor area. The thermal capacitance of the walls and roofs are modeled using a finite difference representation throughout most of the simulations. Internal heat generations and solar heat gains are included. Many simulations have a service hot water load in addition to a space heating load. Actual values of thermal capacitance, internal heat gains, and service hot water load are presented in the discussion of system simulations.

2.1.5 HEAT PUMP

A quasi steady-state heat pump model is used in the simulations. The "transient" behavior of the heat pump is empirically determined by interpolating performance data (absorbed energy rate, rejected energy rate, and electrical input) supplied to the model from data obtained from heat pump manufacturers' specifications. Figure 2.4 shows the heating capacity, COP, and electrical input of a "standard" three-ton heat pump used in simulations.

2.2 SYSTEMS DESCRIPTION

The components, operating characteristics, and performance parameters of the more usual heat pump, solar energy, and solar energy heat pump systems will be discussed to provide a basis for understanding

what types of systems have been studied and how system performance is analyzed and compared.

2.2.1 THE "CONVENTIONAL SOLAR" SYSTEM

A schematic of a conventional liquid solar energy system is shown in Fig. 2.1. Figures 2.1 through 2.8 are obtained from reference [1]. The liquid collector loop consists of a flat plate solar collector, pressure relief valve, storage tank heat exchanger, storage tank, and pump. Solar energy is absorbed in the collector and transferred to the liquid storage tank. This collected energy is then available to provide service hot water heating and space heating. An auxiliary energy supply needed for periods when insufficient solar energy is available is obtained from natural gas, oil, electricity or wood.

The performance of a solar energy system can be represented by the fraction of the house heating load (space and/or service hot water) that is supplied by solar energy. A general curve showing the annual space and hot water load supplied by solar as a function of collector area for a house is shown in Fig. 2.2. As previously mentioned, although it is possible to design a solar energy system to meet all of the heating requirements, it is not economically feasible to do so, and the system would be oversized almost all of the time.

2.2.2 THE "CONVENTIONAL HEAT PUMP" SYSTEM

A heat pump is a refrigeration system that removes heat from one area and supplies it to another. In a mechanical air conditioner,

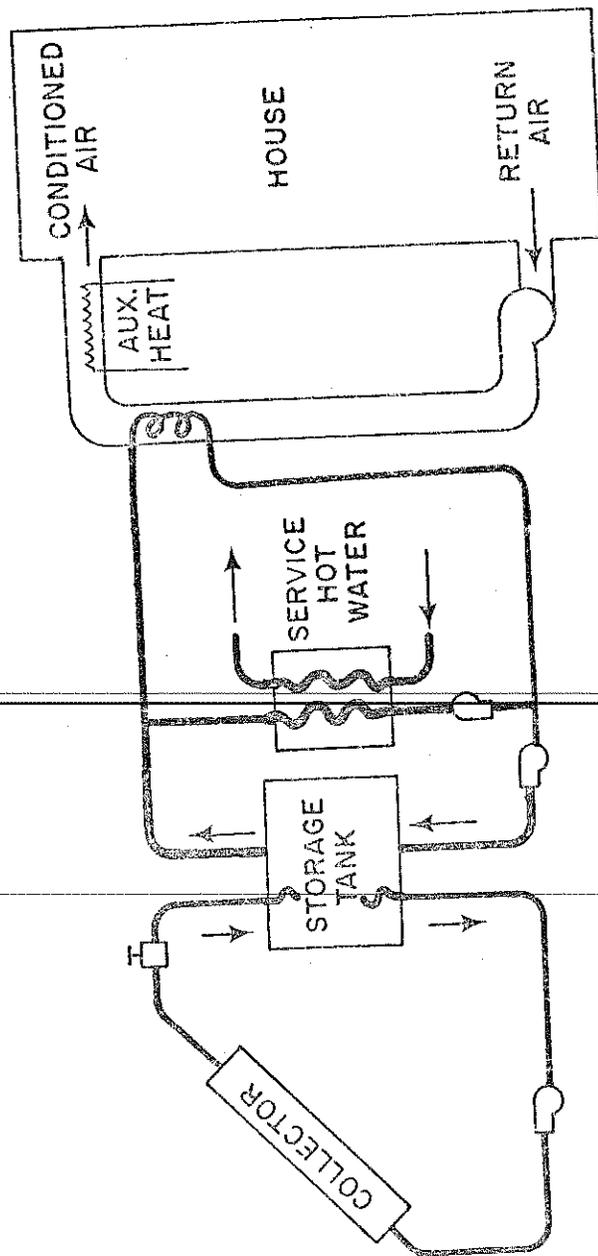


Figure 2.1 "Conventional Solar" System

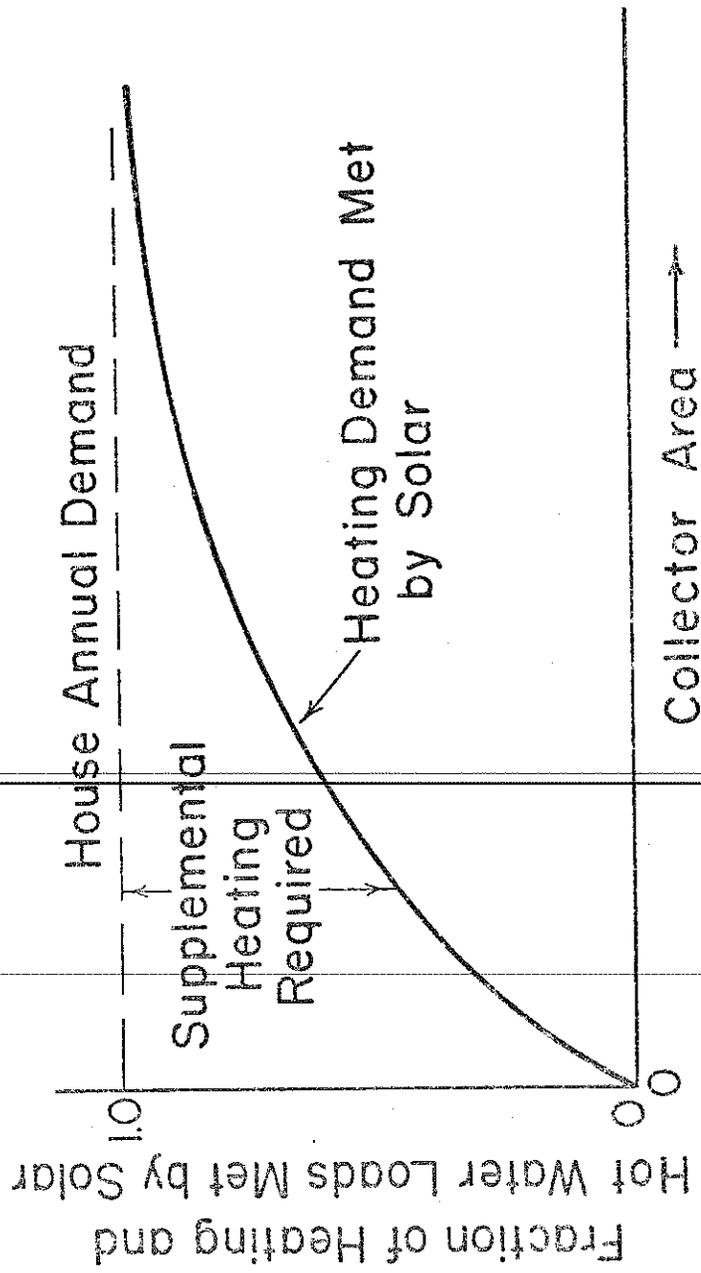


Figure 2.2 Fraction by Solar versus Collector Area

cooling is the only desired effect, while in a heat pump, the desired effect may be either heating or cooling. In residential heat pump applications, an air-to-air heat pump in the heating mode takes heat from the cool outside air and "pumps" it into the room, or in the cooling mode, it removes heat from the room and "pumps" it to the warmer outside air.

The components necessary for providing either heating or cooling with an air-to-air heat pump are shown schematically in Fig. 2.3. Following the refrigerant flow through the cycle shows that the refrigerant in the vapor state enters a reciprocating compressor and has work done on it bringing it to a relatively high pressure and temperature. The refrigerant condenses to a liquid as it passes through the condenser heat exchanger coil and, in the heating mode as room air circulates over the condenser coil, the heat of condensation ($q_{\text{condenser}}$) is transferred to the room. The warm condensed liquid refrigerant flows through tubing to the expansion valve where it experiences a large pressure drop across the valve causing a drop in fluid temperature. This cold liquid refrigerant then enters the evaporator heat exchanger coil located outdoors where heat ($q_{\text{evaporator}}$) is transferred from the cold ambient air to the even colder refrigerant. The refrigerant vaporizes as it passes through the evaporator and flows to the suction side of the compressor completing the cycle.

In the cooling mode, the room air is cooled by reversing the functions of the evaporator and condenser coils. Rotation of the reversing valve would send the hot refrigerant from the discharge side

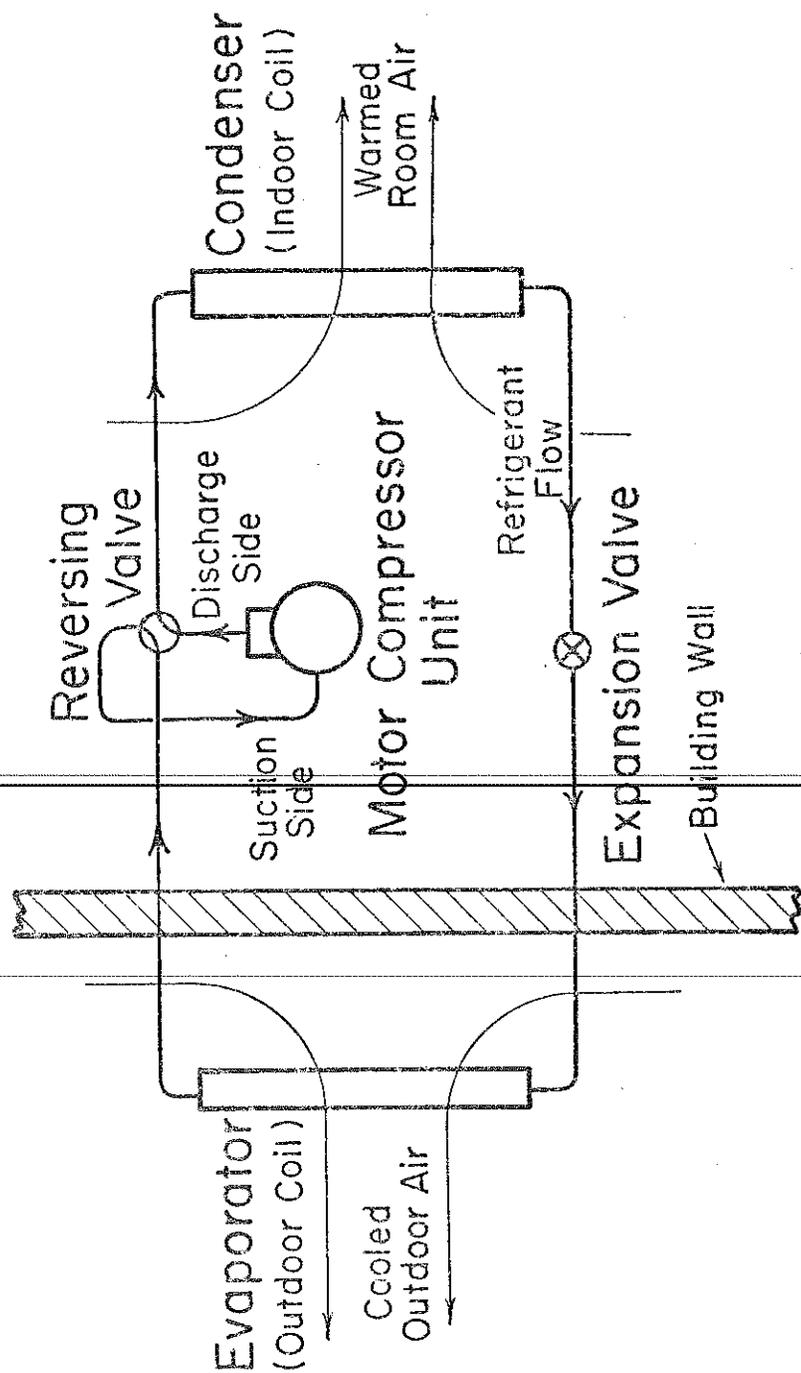


Figure 2.3 Air-to-Air Heat Pump

of the compressor to the outdoor coil (now the condenser) to reject heat outdoors. After passing through the expansion valve, the cold refrigerant passes through the indoor coil (now the evaporator) where heat is absorbed by the refrigerant cooling the room air.

In residential applications an air-to-air heat pump is commonly found. Other types of heat pumps are water-to-air and water-to-water units which use water from a lake or well as the heat source or sink. In water-to-water units the water rather than the refrigerant is switched between the condenser and evaporator coils to change the unit from heating to cooling.

The heat supplied to the room by an air-to-air heat pump comes from both the outside air and the work done on the refrigerant by the compressor ($w_{\text{compressor}}$). Usually, the heat extracted from outside air is greater than the compressor work. An energy balance on the heat pump yields:

$$q_{\text{evaporator}} + w_{\text{compressor}} = q_{\text{condenser}} .$$

The instantaneous coefficient of performance, COP, is the ratio of the rate of heat output (heating mode) or the rate of heat absorbed (cooling mode) to the total electrical power input (w_{electric}). The total electrical power input is the summation of electrical power to operate the heat exchanger fans (w_{fans}) and actual compressor work ($w_{\text{compressor}}/\eta$) where the motor efficiency (η) is less than unity.

$$w_{\text{electric}} = w_{\text{fans}} + w_{\text{compressor}}/\eta .$$

For heating, the coefficient of performance is

$$\text{COP}_h = \frac{q_{\text{condenser}}}{w_{\text{electric}}} .$$

The COP for heating is usually in the range of two to four. This means that two to four units of heat are delivered to the room for every unit of electricity supplied. The COP is strongly dependent on indoor and outdoor temperatures. As the difference between indoor and outdoor temperature increases, more electrical power input is required, thus decreasing the COP.

For cooling, the coefficient of performance is defined as

$$\text{COP}_c = \frac{q_{\text{evaporator}}}{w_{\text{electric}}} .$$

For the same source and sink temperatures, the cooling COP is less than that for heating by about one, putting it in the range of one to three.

During the heating mode with an air-to-air heat pump and at outside temperatures at or near the freezing point of water, frost may form on the outdoor coil and reduce the heat pump COP. The heat pump operates in the cooling mode briefly to remove the frost. While operating in the defrost mode, the heating COP is negative which lowers the seasonal COP (the total seasonal heating capacity divided by seasonal electrical input).

The capacity of a heat pump is the amount of heating (or cooling) done by the unit and equals the condenser (or evaporator) heat transfer

rate. Like the COP, the capacity also depends on source and sink temperatures and decreases as the difference between the two temperatures becomes greater. Figure 2.4 shows typical three-ton heat pump operating characteristics used in simulations for a room temperature of 20°C (68°F). (One ton of heating represents 12,000 Btu/hr.)

One disadvantage of the heat pump is that its performance is not well matched to house heating requirements. Figure 2.5 shows house heating requirements for a typical home in Madison, Wisconsin, as a function of outside air temperature. The slope of the heating requirement line is a constant house UA (overall heat transfer coefficient of the house times the house surface area). Also shown is the capacity curve for the three-ton heat pump previously mentioned. The intersection of the two curves is called the balance point. At this point, the heat pump capacity exactly meets the house heating demand. At higher ambient temperatures, the heat pump capacity exceeds the house heating requirement and needs to operate only a fraction of the time. At temperatures below the balance point, the heat pump operates 100% of the time but is unable to meet the load alone. Supplementary heat, usually in the form of electrical resistance heating, is required. A larger heat pump in the same home will shift the balance point to a lower ambient air temperature, but there will still be a need for supplementary heat. However, in going to a larger heat pump, the initial cost of the unit increases. To install a heat pump that can meet the largest heating load becomes uneconomical because of the higher initial cost.

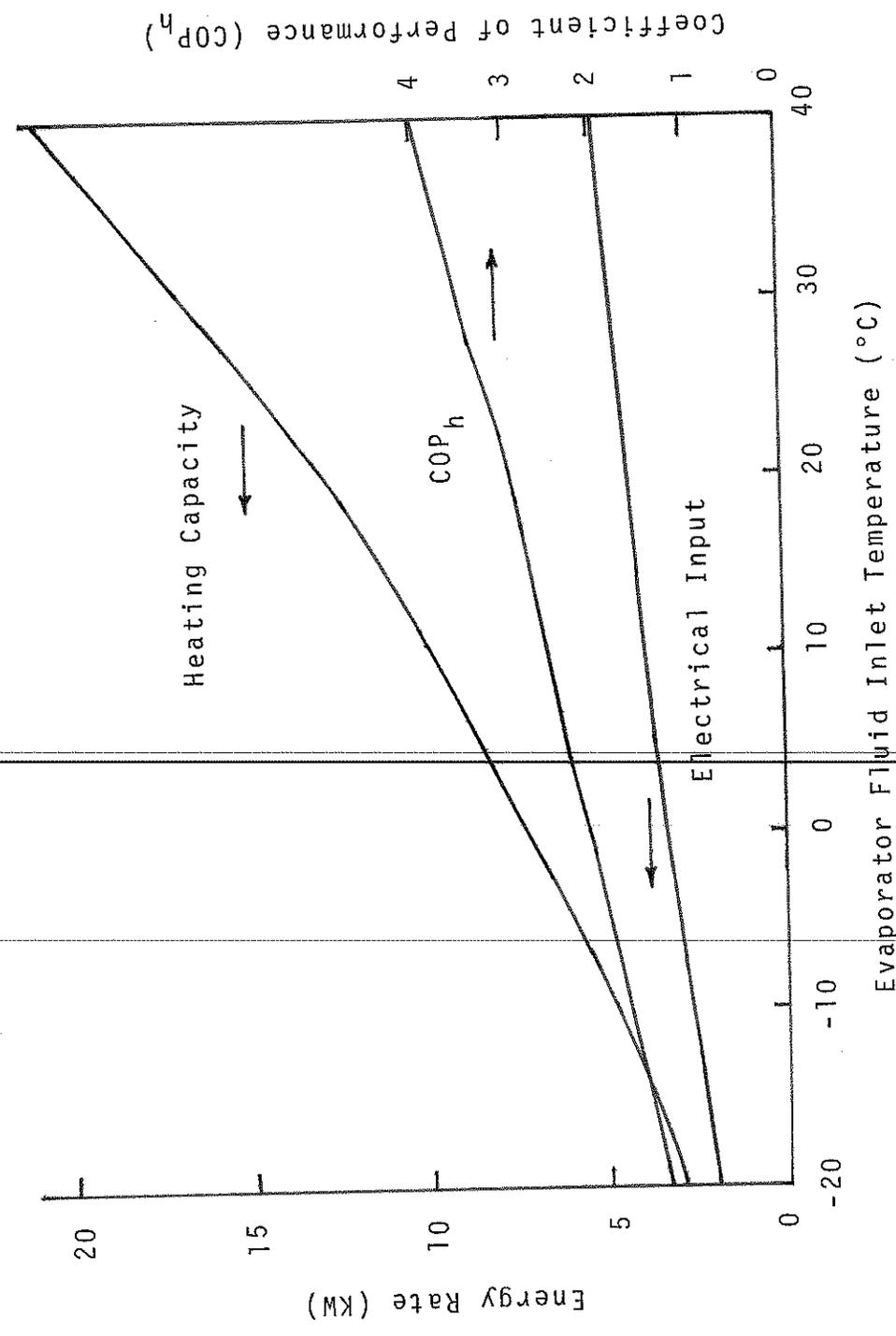


Figure 2.4 Three-Ton Heat Pump Operating Characteristics for Heating

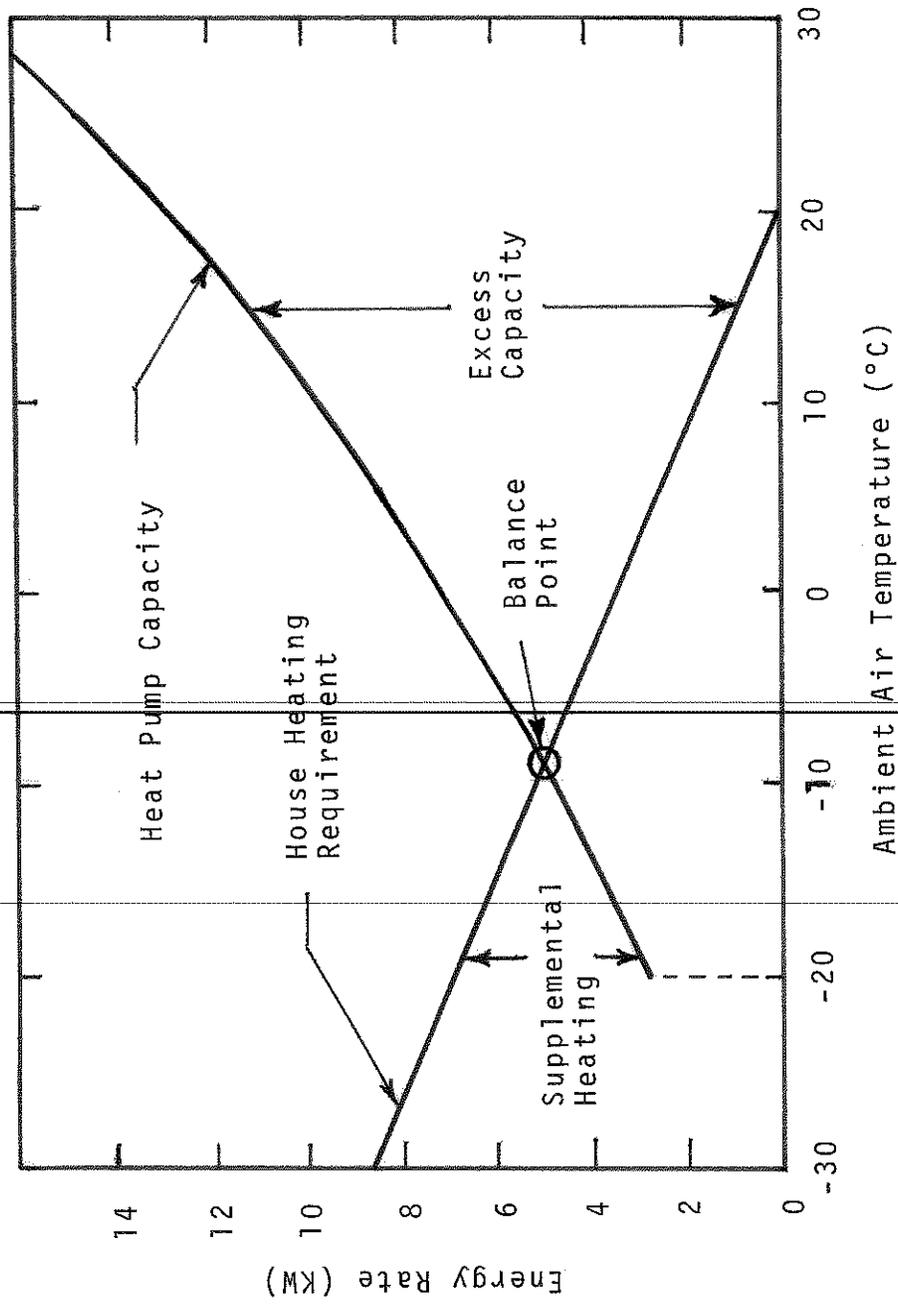


Figure 2.5 House Heating Requirement and Heat Pump Capacity vs. Ambient Air Temperature

The seasonal system COP, denoted as \overline{COP}_s , is the total seasonal space heating supplied (heat pump and supplemental heating) divided by the sum of the seasonal auxiliary energy and the electricity input (auxiliary input and electricity used in running the compressor and fans). \overline{COP}_s is usually about two.

$$\overline{COP}_s = \frac{QRH + QAUX_{sp}}{WAH + QAUX_{sp}} .$$

2.2.3 THE "PARALLEL" SYSTEM

The simplest solar heat pump system arrangement is a "conventional solar" system operating independently of the heat pump which acts as an auxiliary energy source. This arrangement is termed a "parallel" system and is shown schematically in Fig. 2.6. This figure shows available solar energy being collected to increase storage tank temperature while the air-to-air heat pump meets the house heating requirement. The solar system collects available energy whenever possible to do direct heating and supply service hot water.

In the heating mode, direct solar heating would be used whenever possible. This direct solar heating for the space heating load can happen only when the storage tank temperature is above a preset level, T_{dh} . The heat pump, using ambient air as the source, would be turned on whenever there was insufficient solar energy to meet the load with direct heating. The heat pump's built-in auxiliary heater (electrical resistance heater) would be used when neither direct heating nor rejected heat from the heat pump could meet the load.

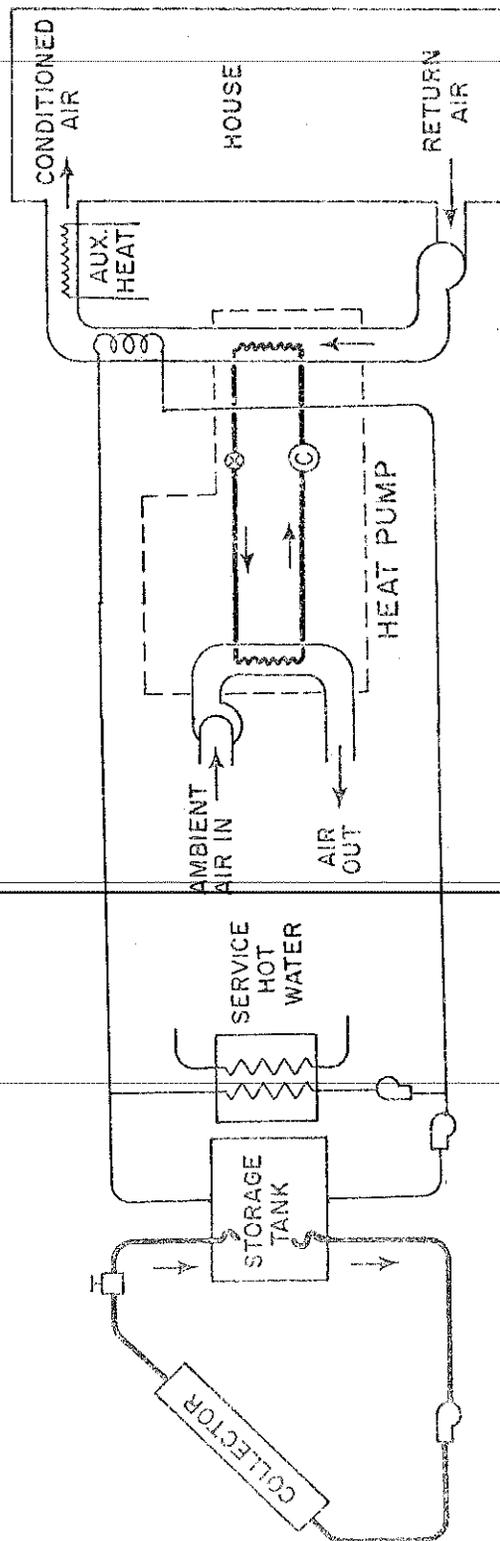


Figure 2.6 The "Parallel" System

In the cooling mode, the solar system needs to collect energy only to meet the service hot water load. The air-to-air heat pump cycles on and off to meet the house cooling load. There is no cooling auxiliary so the heat pump must be sized properly to meet the cooling load at room design temperature. If the heat pump does not meet the instantaneous cooling load, the room temperature would rise above its design condition. The rejected heat from the heat pump goes to the ambient air.

2.2.4 THE "IN-LINE" SYSTEM

By placing the heat pump between the solar system and the house load, as shown in Fig. 2.7, a combined system termed an "in-line" system is arrived at. The heat pump evaporator has access to the storage tank and the condenser is located in the house heating duct.

The advantage of this system is that the heat pump uses the relatively warm storage tank as a source, which increases its COP, and allows the solar system to operate at a lower temperature, which increases its efficiency. Direct and electrical resistance heating are also possible with this configuration, but the heat pump can never use ambient air as a source.

In the heating mode, direct heating would be used whenever the storage tank temperature is above T_{dh} . When the storage tank temperature is below T_{dh} and is above its minimum, T_{min1} (usually a temperature slightly higher than the freezing point of the storage fluid), stored energy is used as a source for the water-to-air heat pump. This

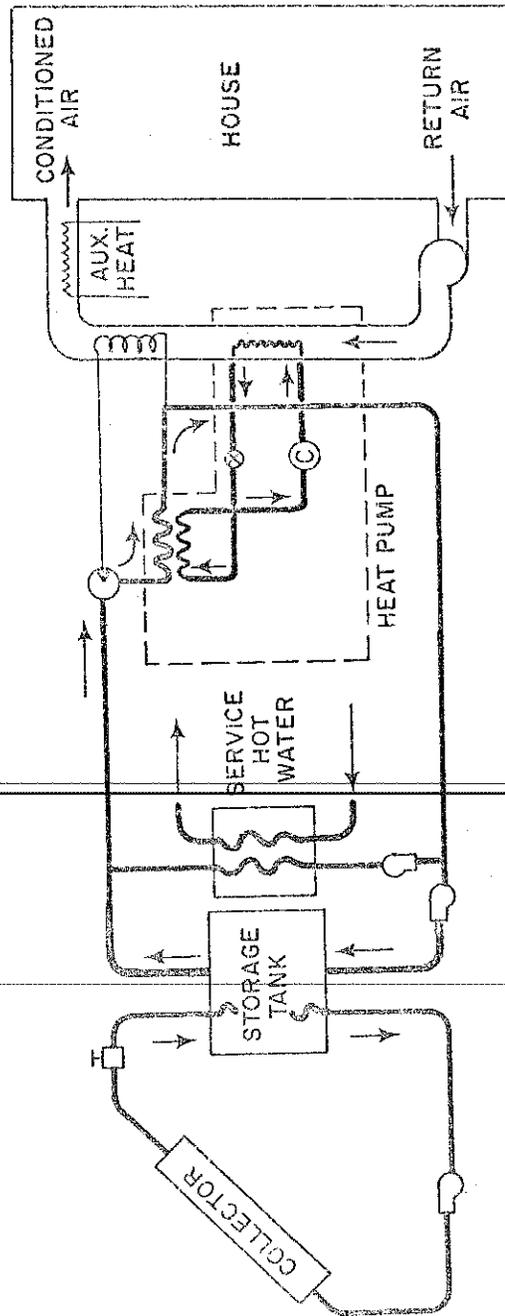


Figure 2.7 The "In-Line" System

mode of operation is shown in Fig. 2.7. If the storage tank temperature is less than T_{min1} , the heat pump's built-in auxiliary heater would be used. In all three modes of operation, available solar energy could be collected and stored if possible, and stored energy could be used as a preheat for service hot water.

The cooling mode operation for the "in-line" system is similar to the "parallel" system except that the rejected heat from the heat pump goes to either the storage tank or to a cooling tower.

2.2.5 THE "DUAL SOURCE" SYSTEM

The "dual source" system, shown in Fig. 2.8, combines the "parallel" and the "in-line" systems to use a heat pump with two evaporators, one in conjunction with the storage tank and the other outdoors. This would allow the heat pump to use either the collected solar energy or ambient air as its source, depending on which temperature is higher. This system would appear to have the best performance, but as will be shown later, this is not necessarily the case.

In the heating mode, the "dual source" system using a liquid solar loop would operate in the direct heating mode, as shown in Fig. 2.8, if the storage tank temperature is greater than T_{dh} . If the storage tank temperature is greater than the ambient temperature, T_{amb} , and between T_{dh} and T_{min1} , the fluid from the storage tank is used as the heat source for the heat pump. However, if the storage tank temperature is less than T_{amb} and is between T_{dh} and T_{min1} , the ambient air

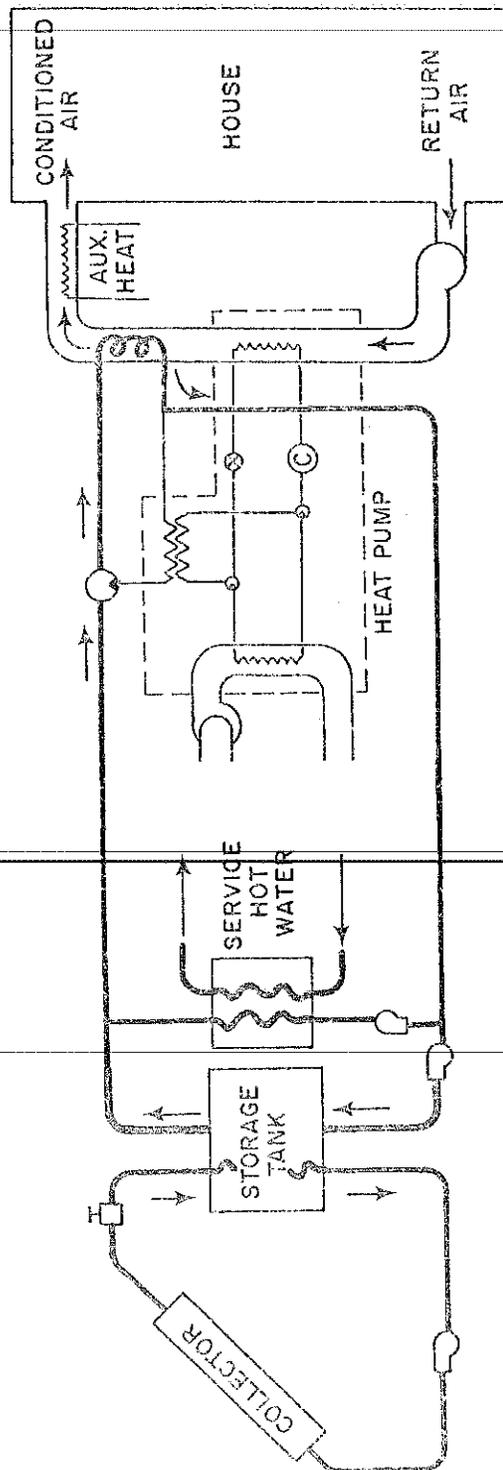


Figure 2.8 The "Dual Source" System

is used as the source for the heat pump. The heat pump's built-in auxiliary heater would be used if the house heating requirement is still not met by direct heating with solar and rejected heat from the heat pump. The solar system would continue to operate and collect available energy to raise the storage tank temperature and supply service hot water if possible.

The cooling mode operation for the "dual source" system is also a combination of the "parallel" and "in-line" systems. The service hot water load is met by the solar system with rejected heat from the heat pump going to either the storage tank, cooling tower, or ambient air.

2.3 DISCUSSION OF SIMULATIONS

2.3.1 GENERAL DESIGN PARAMETERS

In simulating system performance for evaluating the best performing solar heat pump system, many design parameters must be varied in an orderly fashion and performance measures compared. Important design parameters include collector area and construction (e.g. number of collector glazings), main storage tank volume, house capacitance, heating loads for space and/or hot water, cooling loads, minimum storage tank temperature, minimum source temperature for heat pump operation and direct heating, heat pump source (ambient air or storage), heat pump performance, system control options (time of day for operation of solar and/or heat pump system), and geographical location. Performance measures of major importance are monthly and seasonal heat

pump and system COP and the fraction of the house heating and service hot water load met by solar and air sources. Section 2.3.3 presents a more detailed discussion of performance measures.

In order to find the best performing solar heat pump system, system performance of one system must be compared with performance of the others under similar design and operating conditions of storage medium and space heating and service hot water loads. The purpose of this work is to compare dual source system performance to parallel, in-line, conventional solar, and conventional heat pump system performance using a common storage medium (liquid) for solar and solar heat pump systems. Simulation results from Karman et al. [6] of solar heat pump systems (dual source and in-line) showed a general increase in conventional fuel savings with increasing collector area using either pebble bed or liquid storage media. Liquid storage systems are easier to simulate than pebble bed storage systems. Therefore, since a common storage medium is needed for solar and solar heat pump systems, and since liquid storage systems are easier to simulate, liquid storage systems were used throughout all solar and solar heat pump system simulations.

2.3.2 SPECIFIC DESIGN PARAMETERS

Control options and actual numerical values for component sizes were selected as typical for residential applications. Values of collector area, A_c , were selected by starting with zero and increasing in size until the savings in consumed energy was approximately 80

percent. This resulted in using collector areas of 10, 30, and 60 square meters for Madison, Wisconsin. The values of the collector heat removal factor, F_R , loss coefficient, U_L , and transmittance-absorptance product, $\tau\alpha$, given in Table 2.1 for zero, one, and two collector glazings are typical and chosen by the methods outlined by Duffie and Beckman [12]. Other collector loop parameters used throughout all of the simulations are also shown in Table 2.1.

The base value of main storage tank volume used in the simulations was a water storage tank volume to collector area ratio, V/A_C , of $0.075 \text{ m}^3/\text{m}^2$ as recommended by Beckman, Klein, and Duffie [10]. The ratio was varied over a wide range to evaluate the sensitivity to this parameter. Initially, performance of all systems was compared using a volume to area ratio of $0.075 \text{ m}^3/\text{m}^2$. From these simulations, the better performing solar heat pump systems (parallel and dual source) were selected and further observed for different V/A_C .

To see if house capacitance had an effect on the decision of the best performing solar heat pump system, simulations were done in Madison, Wisconsin for a house with a heating UA of $620 \text{ KJ/hr-}^\circ\text{C}$ and with either a constant room temperature of 20°C or a capacitance of $50,000 \text{ KJ/}^\circ\text{C}$ and floating room temperature. The load heat exchanger located in the house heating duct has an effectiveness of 0.75, and a value of $1300 \text{ KJ/hr-}^\circ\text{C}$ was used for the return air capacitance rate.

The service hot water load used in the simulations is represented by a repeating daily flow during the period 6:00 A.M. to 1:00 A.M.

TABLE 2.1
Collector Loop Parameters

Collector Type	F_R	U_L (KJ/hr-m ² -°C)	$\tau\alpha$
Zero-Glazed, Water	0.73	108.0	0.90
Single-Glazed, Water	0.90	30.0	0.85
Double-Glazed, Water	0.90	21.0	0.76

Parameter	Value
Transfer Medium Specific Heat, C_{p_h}	3.35 KJ/kg-°C
Hot Side Flow Rate Per Collector Area, \dot{m}_h/A_c	50 kg/hr-m ²
Collector-Storage Heat Exchanger Effectiveness, ϵ	0.8
Cold Side Flow Rate Per Collector Area, \dot{m}_c/A_c	40 kg/hr-m ²
Cold Side Transfer Medium Specific Heat, C_{p_c}	4.19 KJ/kg-°C
Relief Valve Maximum Temperature	100°C
Collector Tilt From Horizontal, s	
Madison, Wisconsin	59°C
Albuquerque, New Mexico	51°C

(See reference [8] for more information on this profile.) The total daily demand of hot water is 279.5 kg at a required temperature of 60°C or higher. The main water temperature for Madison is 10°C. This results in an annual service hot water energy requirement of 21.37 GJ. A value of 15°C was used for the main water temperature in Albuquerque, New Mexico. The domestic hot water preheat storage tank, like the main storage tank, is completely insulated (no losses). A value of 0.5 was used for the service hot water heat exchanger effectiveness.

The minimum main storage tank temperature for direct heating, T_{dh} , was chosen to be 30°C for all of the simulations. Since the room temperature was either constant at 20°C or fluctuated closely around 20°C, the storage tank temperature had to be approximately 10°C higher than the room temperature before any direct heating with solar energy would occur. A temperature in the main storage tank that is higher than the room temperature imposes a slight penalty on solar system performance because of a higher return temperature to the collectors and less useful energy collected. However, a smaller load heat exchanger can be used and excessive cycling of the storage water circulating pump for direct heating at low storage tank temperatures can be avoided which yields longer pump life and lower maintenance cost. For simulations using water as the storage medium, a minimum allowable storage temperature of 5°C was used. Simulations were also done with an antifreeze storage medium which allowed a minimum tank temperature of -20°C.

To observe the effect of geographical location on system performance, two locations with different weather characteristics were used in the simulations. The two locations chosen were Madison, Wisconsin and Albuquerque, New Mexico. The majority of the simulations were done in Madison, Wisconsin.

The conventional control strategy used allows heat pump operation in the heating mode when direct heating by solar energy is unable to meet the house load. This strategy does not, for example, allow solar energy to be stored during the day and used at night while the heat pump operates mainly during the day when ambient source temperatures are high. To see what might happen to system performance with a "smart" controller, the control strategy shown in Table 2.2 and Fig. 2.9 was used on a parallel system in Madison and compared to parallel system performance using the conventional control strategy.

T_{low} is the lowest possible ambient temperature encountered during the heating season, and for Madison, T_{low} is -30°C . T_{star} is a parameter that limits use of the heat pump when storage is charged above a critical value indicative of the line separating zones A and B in Fig. 2.9. If T_{star} equals T_{dh} , the strategy for zone A is used at all ambient temperatures, which is the conventional control strategy. However, if T_{star} equals infinity, the strategy for zone B is used at all ambient temperatures. For T_{star} between T_{dh} and infinity, the strategy for zones A and B share control for various combinations of ambient and storage tank temperatures.

TABLE 2.2

Conventional Control Strategy

Day	Night
1) Solar	1) Solar
2) Heat Pump	2) Heat Pump
3) Electric Backup	3) Electric Backup

"Smart" Controller Strategy

	Zone A	Zone B
	1) Solar	1) Heat Pump
Day	2) Heat Pump	2) Electric Backup
	3) Electric Backup	
Night	1) Solar	1) Solar
	2) Heat Pump	2) Heat Pump
	3) Electric Backup	3) Electric Backup

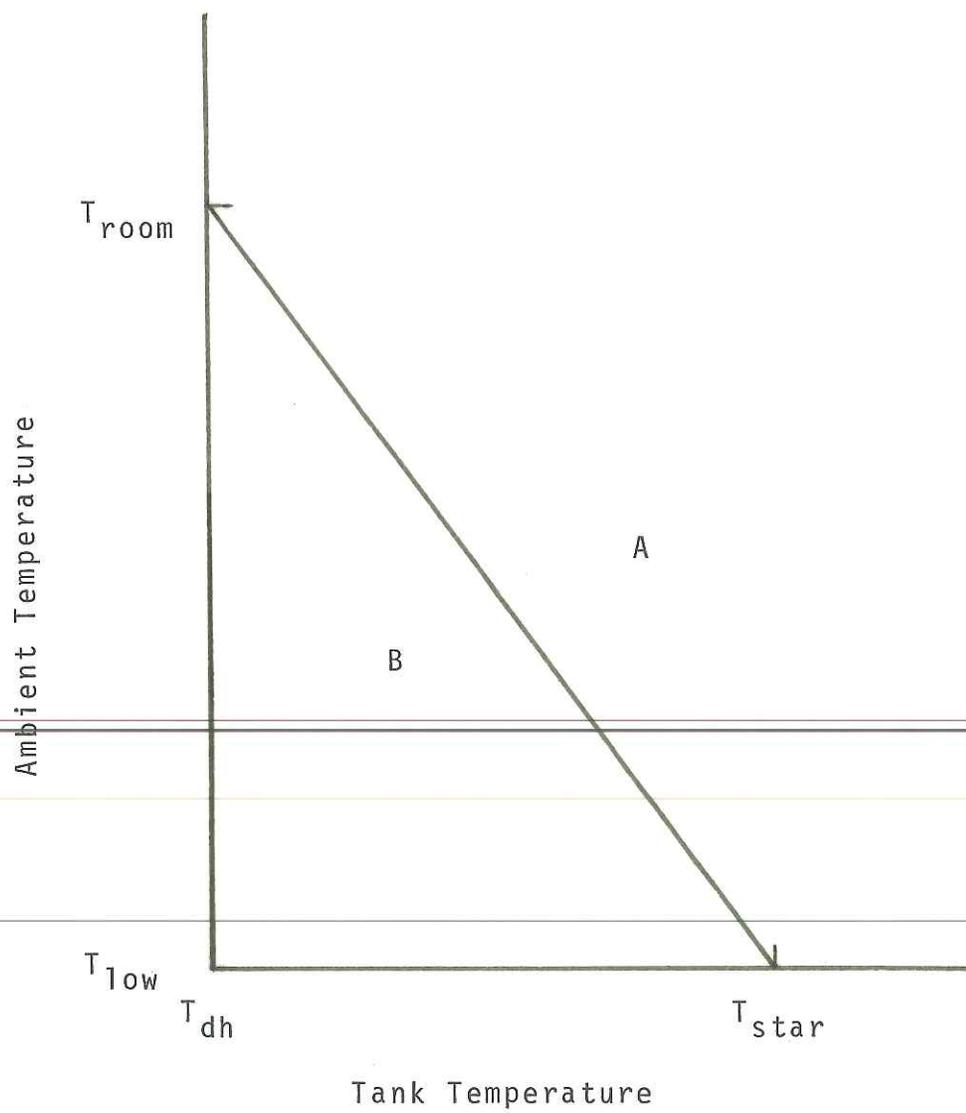


Figure 2.9 "Smart" Controller Zoning

The heat pump employed in these simulations is a "standard" three-ton unit. Figure 2.4 shows the heat pump operating characteristics for heating used in the simulations. Figure 2.10 shows the operating characteristics for cooling. Freeman [2] showed that the size of the heat pump was not critical as long as it was large enough to meet the design cooling load. Conventional air-to-air heat pump installations are generally sized for the cooling load to insure proper dehumidification. However, when heating is the prime concern, the unit should be oversized so that electrical resistance auxiliary heat is not often required. Excessive compressor cycling sometimes caused by oversizing can be avoided by widening the thermostat dead-band range and using the conventional control strategy which utilizes direct heating by solar energy when storage temperatures are sufficiently high.

The data presented in Figs. 2.4 and 2.10 represents the heat pump heating and cooling operating characteristics used in the simulations for both the air and liquid source heat pumps as well as a special dual source heat pump having both an air and liquid evaporator. Since actual performance data is lacking for dual source heat pumps, it is assumed that their performance at a given source inlet temperature is identical in either the air or liquid source modes. This is equivalent to assuming that the evaporator heat exchangers have been designed to have equal heat transfer effectivenesses.

To determine if a better performing heat pump has an effect on the determination of the best performing solar heat pump system, heat

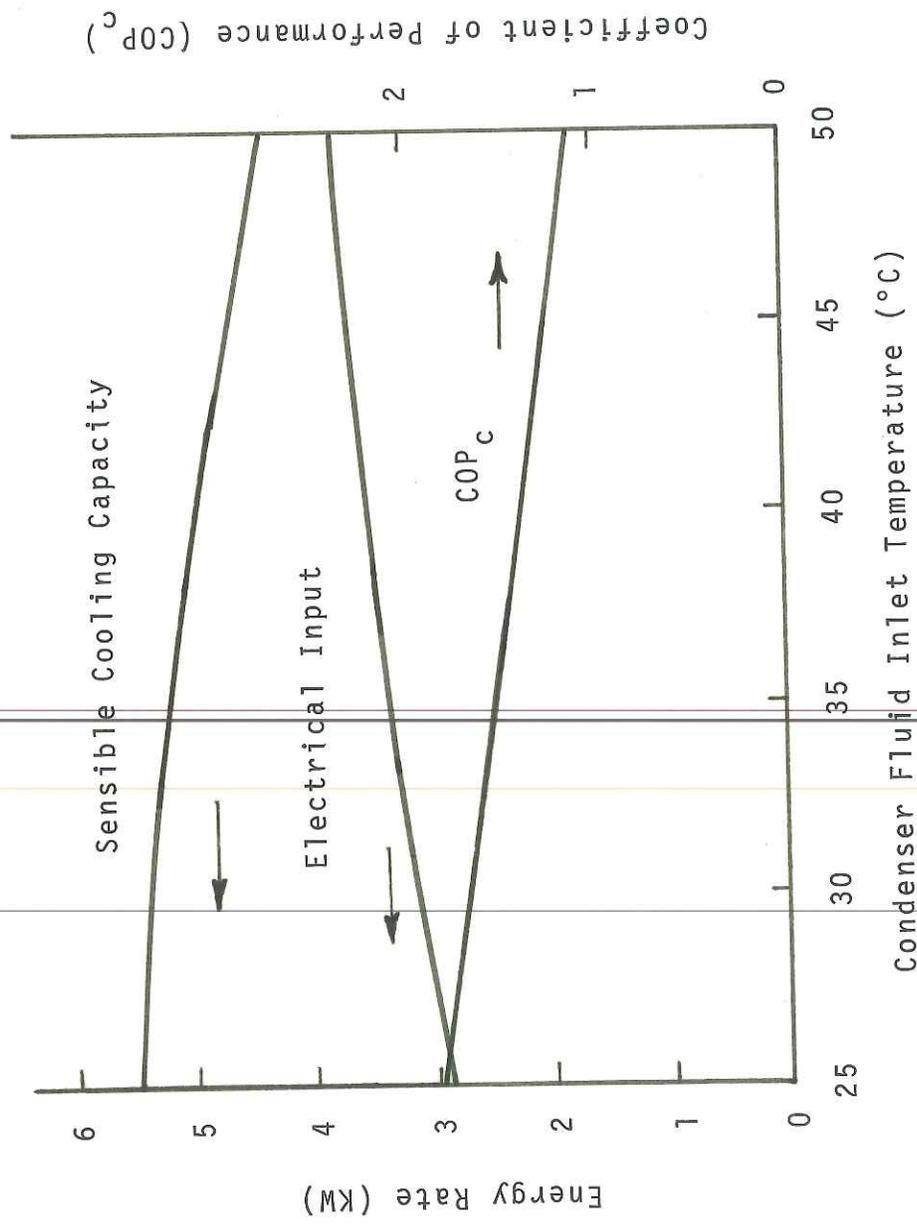


Figure 2.10 Three-Ton Heat Pump Operating Characteristics for Cooling

pumps using heating data reflecting a 25, 50, and 100 percent improvement in COP for the "standard" three-ton heat pump were used in Madison. Simulations were done on the conventional heat pump, parallel, dual source, and in-line systems for a house with a floating room temperature and capacitance of 50,000 KJ/°C. Thirty square meters of single-glazed collectors were used with a storage volume to collector area ratio of $0.075 \text{ m}^3/\text{m}^2$. The improved heat pump operating characteristics are shown in Fig. 2.11 for a room temperature of 20°C. (Note that the same heating capacity is used in all three cases.)

In the cooling mode, the systems having an air-to-air heat pump use the outside air condenser as the sink. In the case of the in-line system having a water condenser, it is assumed that an extra storage tank has been supplied and energy is rejected by either "night-sky radiation" through the collectors at night or else a cooling tower.

Accurate evaluations of cooling loads are not of major concern in this work. The solar collectors and storage continue to operate in the summer to supply the service hot water load.

2.3.3 PERFORMANCE MEASURES

The simulation results of major interest in this work for solar heat pump systems are long-term integrated energy quantities. These include the total heat gain of the solar collectors (QU), the total space and service hot water heating load (QLOAD), the total auxiliary energy added by the furnace and service hot water heater (QAUX), the total energy removed by the heat pump from the ambient air in the

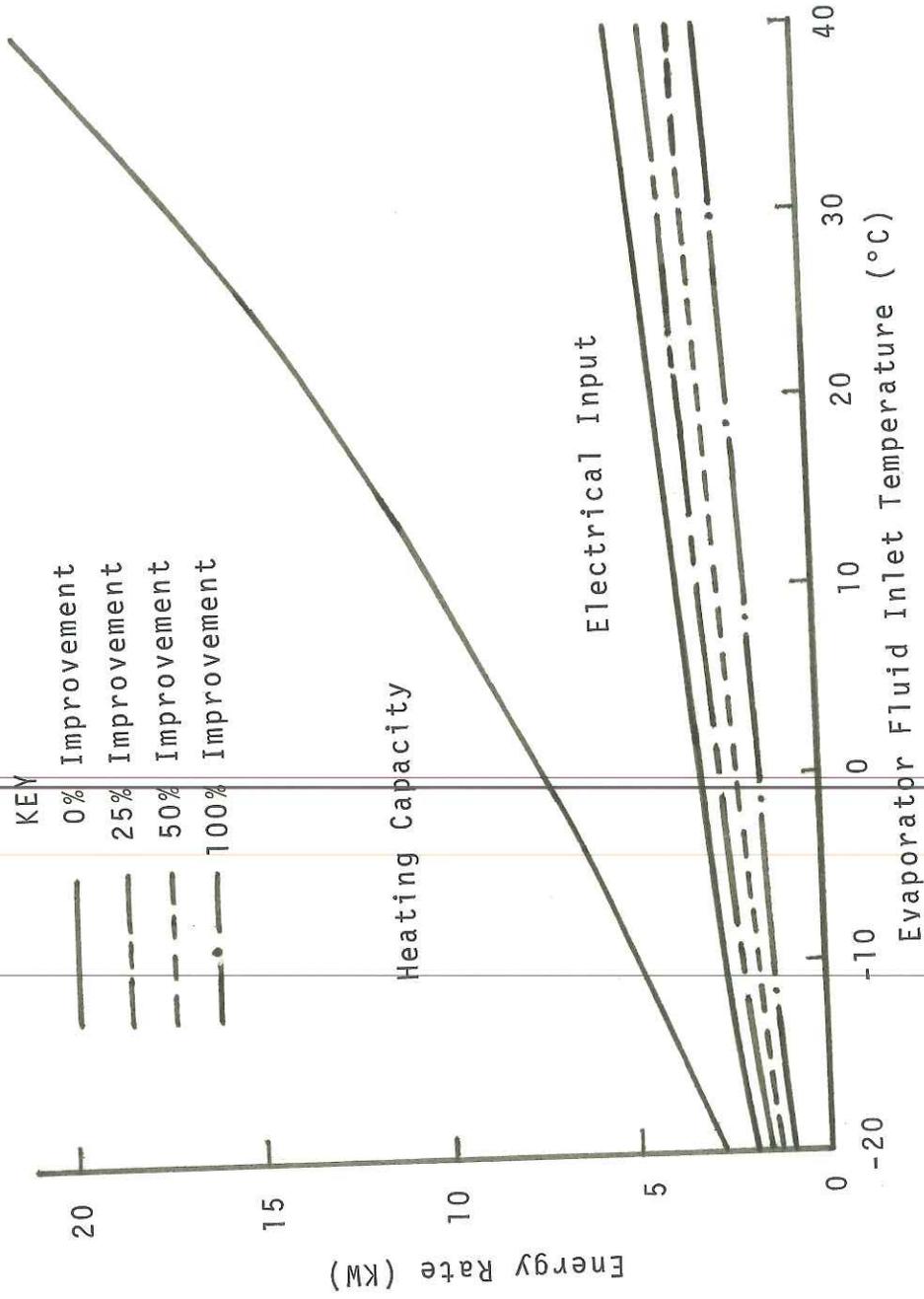


Figure 2.11 Improved Heat Pump Operating Characteristics for Heating

heating mode (QAIR), and the total heat pump electrical input for compressor and pumps or fans (WAH). For a heating season, the system energy balance assuming negligible change in stored energy in the tank is

$$QU + QAIR + QAUX + WAH = QLOAD.$$

The single most informative indicator of solar heat pump system performance is the fraction of the total load met by solar and ambient air sources, defined as

$$F = (QLOAD - WAH - QAUX)/QLOAD.$$

When considering only solar energy system performance, the fraction of the load met by solar (f) is of prime concern. Considering only the space heating load, f_{sp} is defined as

$$f_{sp} = QDH/QLOAD_{sp}$$

where QDH is the total direct space heating by solar energy and QLOAD_{sp} is the total space heating load. For the hot water load, f_{hw} is defined as

$$f_{hw} = 1 - (QAUX_{hw}/QLOAD_{hw})$$

where QAUX_{hw} is the auxiliary energy added to the hot water heater and QLOAD_{hw} is the total hot water heating load. The fraction of the total space and hot water heating load met by solar energy is defined as

$$f = (QDH + QLOAD_{hw} - QAUX_{hw})/QLOAD.$$

In addition to WAH, a long-term integrated energy quantity of interest in heat pump performance is the energy rejected to the house in the heating mode (QRH). QRH is the summation of the energy rejected in the water-to-air heating mode (QRWA) and the air-to-air heating mode (QRAA). Also of interest is the total number of hours of operation spent in the water-to-air heating mode (HWA) and the air-to-air heating mode (HAA). Heat pump and system COP will also be evaluated on monthly and seasonal bases when relevant.

2.3.4 MADISON SIMULATION RESULTS

2.3.4.1 BASE CASES

The initial simulations were done for Madison, Wisconsin. These simulations were chosen to study the effects of collector area and construction, ambient energy source, and solar contribution. The annual space heating and service hot water load is 85.3 GJ. Results of these simulations using a house heating UA of 620 KJ/hr-°C and constant room temperature of 20°C are shown in Figs. 2.12 and 2.13. These simulations used an energy rate control. That is, solar, heat pump, and auxiliary operate when needed to exactly meet the instantaneous house heating load for a constant room temperature. The 2.43 GJ cooling load for this house was met entirely by the heat pump.

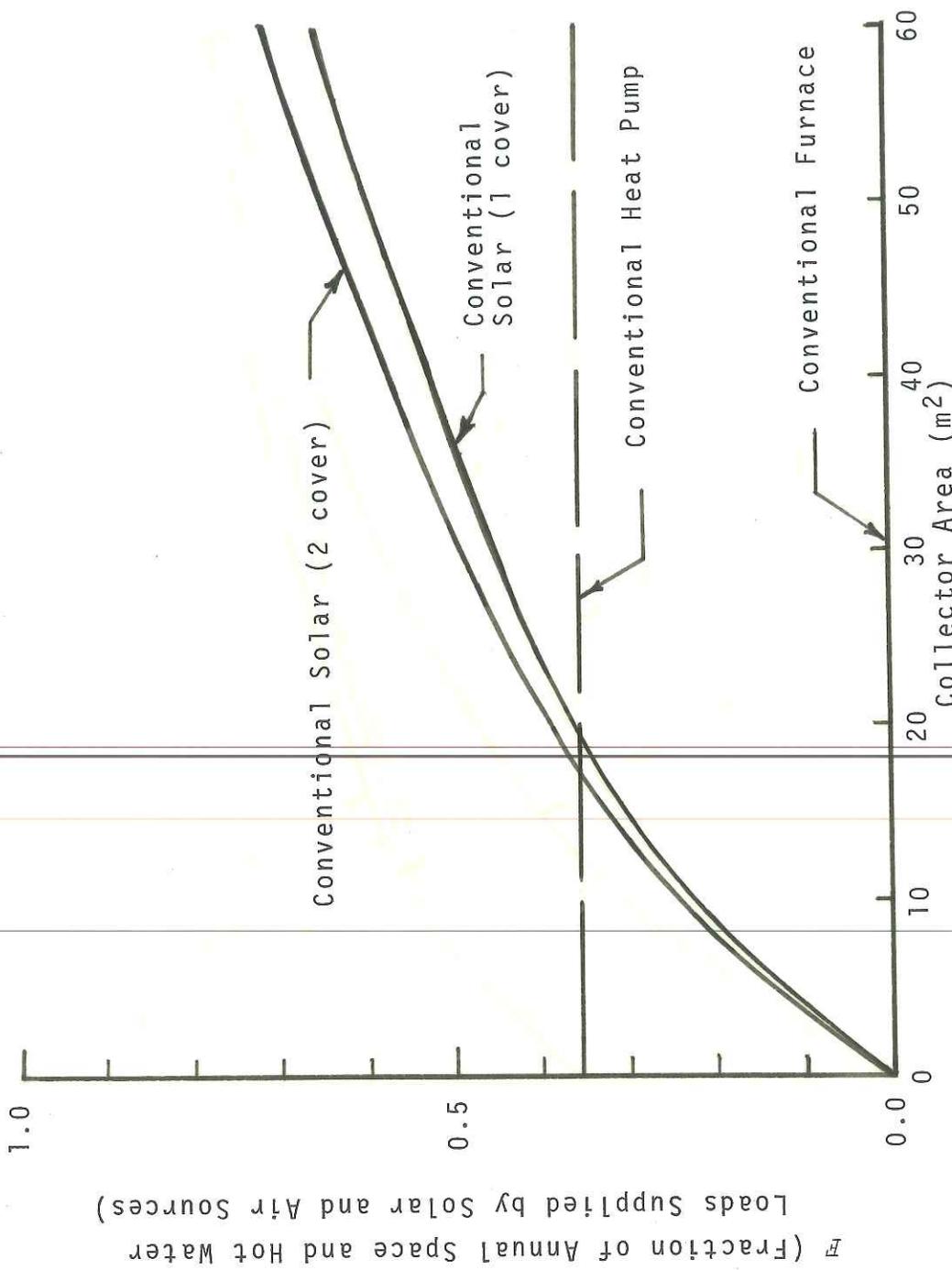
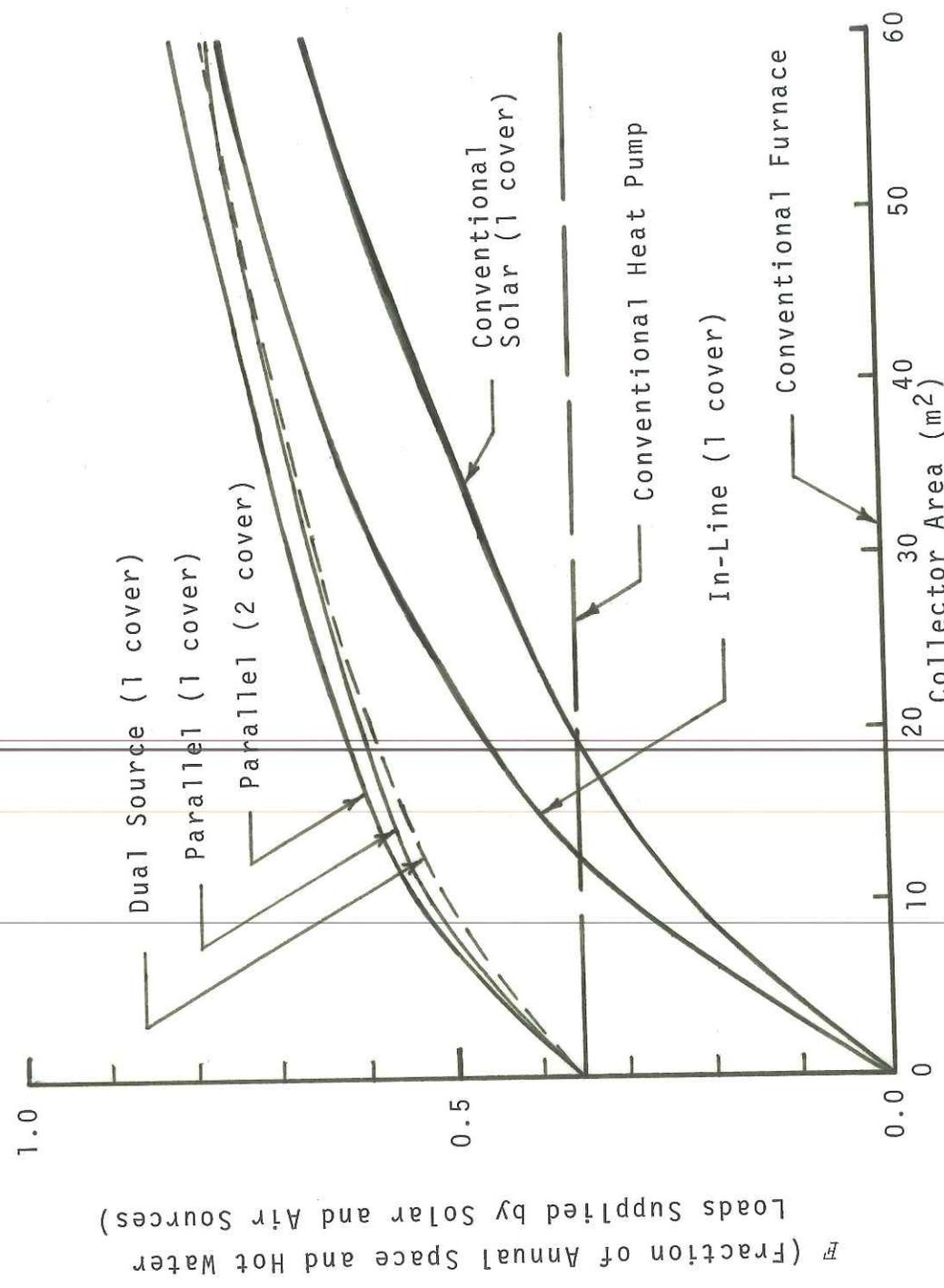


Figure 2.12 F vs. A_c



F (Fraction of Annual Space and Hot Water Loads Supplied by Solar and Air Sources)

Figure 2.13 F vs. A_c

While doing the conventional solar and parallel system simulations, it was observed that two cover collectors yielded a relatively small savings in "conventional" fuel over one cover collectors. Therefore, most of the solar heat pump simulations were performed with less expensive, single-glazed collectors.

Figures 2.12 and 2.13 and Table 2.3 allow comparisons to be made between conventional furnace (electrical resistance heating), conventional solar, conventional heat pump (air-to-air), in-line, parallel, and dual source systems. For conventional furnace and conventional heat pump, F does not depend on collector area. The collector size necessary for the conventional solar system to consume less auxiliary energy than a conventional heat pump system is about 19 m^2 .

For parallel systems, the solar part of the system supplies F of the load shown in Figs. 2.12 and 2.13. The air-to-air heat pump supplies a portion of the remaining load from the ambient air. However, the heat pump system COP is less than that in a conventional heat pump system since the solar system supplies heat during sunny and warm weather while the heat pump operates mainly in cold weather, particularly at night, when the COP is low. Thus, the amount of auxiliary energy required by the combined system is reduced but not by as large an amount as might be expected. At zero collector area, the parallel system performs like a conventional heat pump system. At very large collector areas, the solar system meets a large fraction of the load forcing the heat pump to operate under very cold weather at a COP close to one. As a result, the parallel system performs much like a conventional solar system.

TABLE 2.3
 Base Cases Simulation Results for Madison
 $Q_{LOAD} = 85.3 \text{ GJ}$ $V/A_c = 0.075 \text{ m}^3/\text{m}^2$

System Description	A_c (m^2)	QDH (GJ)	QRWA (GJ)	QRAA (GJ)	QAUX _{sp} (GJ)	QAUX _{hw} (GJ)	WAH (GJ)	HWA (hr)	HAA (hr)	F (—)
Conventional Heat Pump	—	—	—	58.70	5.23	21.37	28.35	—	2575	0.356
Conventional Solar 1 cover	10	8.27	—	—	55.69	10.08	—	—	—	0.229
	30	23.81	—	—	40.15	6.89	—	—	—	0.449
	60	39.89	—	—	24.06	5.29	—	—	—	0.656
2 cover	10	9.36	—	—	54.60	9.48	—	—	—	0.249
	30	26.98	—	—	36.98	6.30	—	—	—	0.493
	60	44.40	—	—	19.56	4.52	—	—	—	0.718
In-Line 1 cover	10	3.24	21.39	—	39.33	12.27	7.81	562	—	0.304
	30	13.77	35.27	—	14.92	8.59	12.40	868	—	0.579
	60	30.78	27.32	—	5.86	6.22	9.04	606	—	0.752
Parallel 1 cover	10	8.27	—	50.23	5.46	10.07	24.63	—	2264	0.529
	30	23.81	—	34.99	5.16	6.88	17.42	—	1642	0.655
	60	39.89	—	18.67	5.40	5.28	9.30	—	862	0.766
2 cover	10	9.36	—	49.29	5.31	9.58	24.19	—	2225	0.542
	30	26.98	—	31.66	5.32	6.38	15.74	—	1463	0.678
	60	44.40	—	16.38	3.18	4.63	8.30	—	788	0.811
Dual Source 1 cover	10	3.31	21.21	34.72	4.72	12.22	25.14	556	1627	0.507
	30	13.61	35.16	13.08	2.11	8.62	19.11	867	640	0.650
	60	30.78	27.29	3.99	1.90	6.23	11.22	605	228	0.773

Figure 2.13 shows that an in-line system performs identical to a conventional solar system at zero collector area because the heat pump cannot use ambient air as a source. For a given collector area, the in-line system performance is only slightly better than a one cover conventional solar system, and is definitely not as good as the parallel or dual source system. Since the in-line system uses the storage as its source, the storage medium and circulating fluid temperatures are lower than in the conventional solar system which increases collector efficiency and collection of useful energy, but does less direct heating by solar energy as seen in Table 2.3. It appears that the advantage of lower storage and collector temperatures is very nearly cancelled by the inability of the system to utilize the direct heating mode as often as the conventional solar system.

As shown in Fig. 2.13, the dual source system performs like the conventional heat pump system at zero collector area. For a given collector area, the dual source system performs better than the in-line system. Performance of the dual source system will always be better than the in-line system, all things being the same, because the heat pump in the dual source system will always see the higher source temperature (storage or ambient), whereas the in-line heat pump can only use storage as its source.

Surprisingly, the parallel system performs as well as, if not better than, the dual source system. In comparison to the parallel system, Table 2.3 shows that more heat is supplied by the heat pump in the dual source system, but more electrical work is required

because the heat pump operates more since there is less direct heating by solar energy. The advantage of lower storage and collector temperatures is cancelled by the inability of the system to utilize the direct heating mode as often as the parallel system. At large collector areas, the dual source system, like the in-line system, behaves like a conventional solar system since the solar part of the system meets most of the load.

2.3.4.2 VARIABLE VOLUME RESULTS

Since the parallel and dual source system performed best in these initial simulations, these two systems were selected to study the effects of increasing and decreasing storage volume to collector area ratio using 30 m^2 of single-glazed collectors. The range of V/A_c values used in the simulations is 0.01875 to $5.00 \text{ m}^3/\text{m}^2$. The seasonal results of these two solar heat pump systems are illustrated in Fig. 2.14 for varying conditions of V/A_c . It is seen that extremely small values of V/A_c tend to slightly hurt performance of both systems when compared to V/A_c in the range of 0.075 to $1.200 \text{ m}^3/\text{m}^2$. The dual source system performance remained quite constant beyond $1.200 \text{ m}^3/\text{m}^2$ while the parallel system performance decreased. An explanation for this is that the stored energy in the parallel system is not really useful for space heating until the storage temperature exceeds 30°C , while stored energy can be used by the dual source system when the storage temperature exceeds 5°C . Storage temperatures with large storage volumes seldom exceeded 30°C .

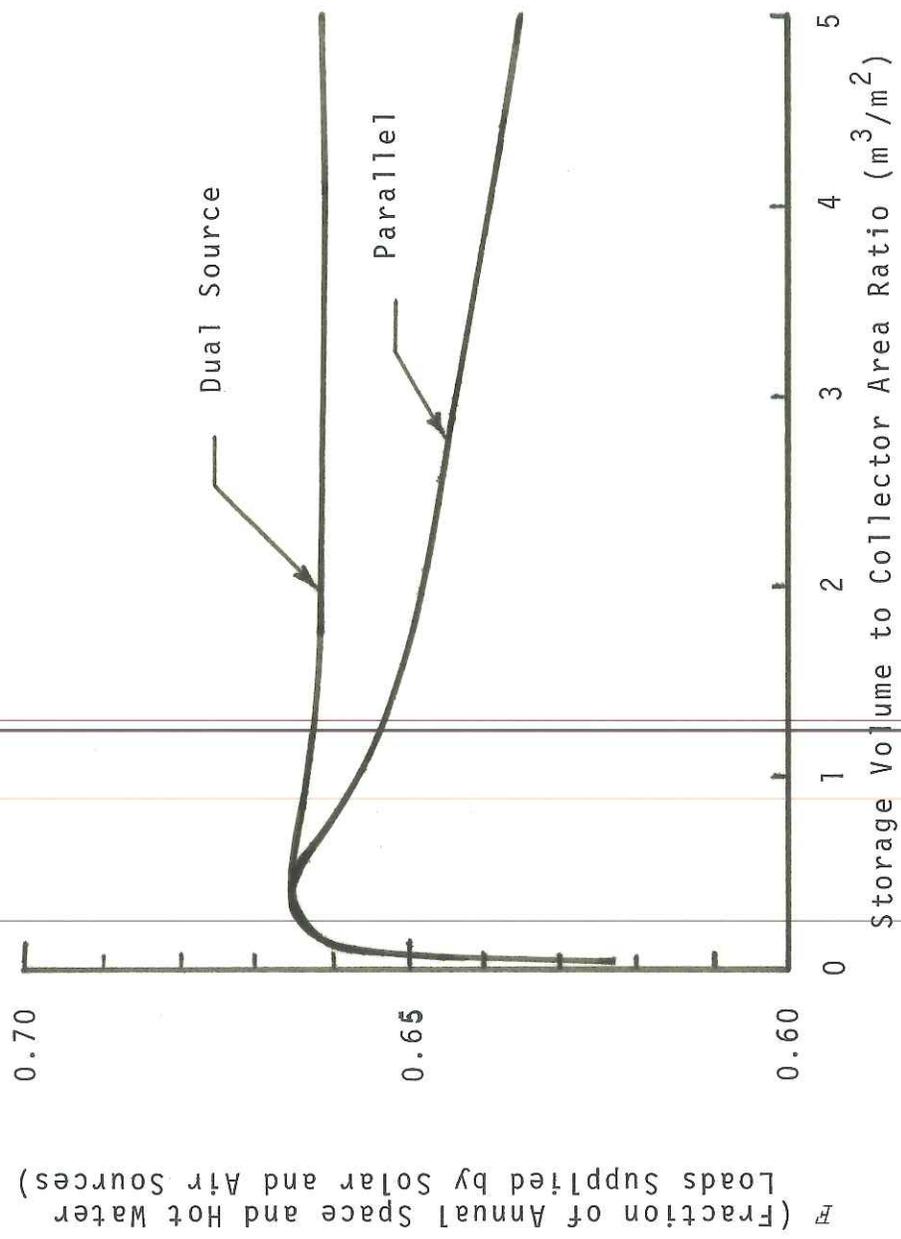


Figure 2.14 F vs. V/A_c

F (Fraction of Annual Space and Hot Water Loads Supplied by Solar and Air Sources)

to allow direct heating with solar energy which required the heat pump to operate more often at a lower source temperature (ambient) for the parallel system than the dual source system. An optimum value of V/A_c for both systems in Madison is approximately $0.3 \text{ m}^3/\text{m}^2$. Although not quite at the optimum value, a V/A_c of $0.075 \text{ m}^3/\text{m}^2$ yields a performance that is only 0.2 percent lower than that for $0.3 \text{ m}^3/\text{m}^2$ and is used extensively throughout the remainder of the simulations.

2.3.4.3 EFFECT OF HOUSE CAPACITANCE

All of the solar heat pump system simulation results presented so far have used an energy rate control. To determine if using a house with a floating room temperature and large house capacitance ($50,000 \text{ KJ}/^\circ\text{C}$) would alter the conclusion that parallel and dual source are the two best performing solar heat pump systems, simulations were performed on the parallel, dual source, in-line, and conventional heat pump systems for 30 m^2 of single-glazed collectors and a V/A_c of $0.075 \text{ m}^3/\text{m}^2$. The house had a UA of $620 \text{ KJ}/\text{hr}\text{-}^\circ\text{C}$. The solar, heat pump, and auxiliary operate when needed by a temperature control. When any of the three means of heating are required, they operate at full capacity. This will at times increase room temperature into the idle range. In first stage heating, the solar system does direct heating, if possible, when the room temperature falls below 20°C . If the room temperature falls to 19.5°C (implying that solar energy cannot meet the load), second stage heating (heat pump and auxiliary) meets the load. In the case of the conventional heat pump, first

stage heating is done by the heat pump and second stage by auxiliary.

The room temperature idle range is from 20 to 25°C. For room temperatures above 25°C, the heat pump operates in the cooling mode. The service hot water load was not considered in these simulations.

Seasonal values of the fraction of space heating load met by solar and air sources for the four systems are presented in Table 2.4 where $\gamma_1 = 20^\circ\text{C}$ is first stage and $\gamma_2 = 19.5^\circ\text{C}$ is second stage heating. The annual cooling load was 2.27 GJ and the space heating load approximately 68.7 GJ. Again, the parallel system performed as well as the dual source system, and both definitely performed better than the in-line system. Table 2.4 shows that house capacitance does not alter the decision of the best performing solar heat pump system. Also, because the service hot water load was not considered, these results cannot be directly compared to earlier.

2.3.4.4 EFFECT OF MINIMUM STORAGE TEMPERATURE

Additional simulations were performed for solar heat pump systems using 30 m² of single-glazed collectors a V/A_c of 0.075 m³/m² to investigate the effect of using antifreeze in the storage tank to allow the use of storage down to -20°C. This minimum storage temperature is denoted as T_{\min} . Seasonal values of the fraction of the total space and hot water load met by solar and air sources for in-line, parallel, and dual source systems are presented in Table 2.5. Values of F for $T_{\min} = 5^\circ\text{C}$ are shown in Table 2.5 for comparison. These simulations were done with the energy rate control used in the

TABLE 2.4

Temperature Control Results

House Capacitance = 50,000 KJ/°C QLOAD_{sp} = 68.7 GJ
 $\gamma_1 = 20^\circ\text{C}$ $\gamma_2 = 19.5^\circ\text{C}$

System	Fraction of Annual Space Heating Load Met by Solar and Air Sources (F_{sp})
Conventional Heat Pump	0.514
In-Line	0.663
Parallel	0.715
Dual Source	0.720

TABLE 2.5

Energy Rate Control

$A_c = 30 \text{ m}^2$ $V/A_c = 0.075 \text{ m}^3/\text{m}^2$ QLOAD = 85.3 GJ

Fraction of Annual Space and Hot Water Load Met by Solar and Air Sources (F)

System	$T_{\text{min1}} = 5^\circ\text{C}$		$T_{\text{min1}} = -20^\circ\text{C}$	
	One Cover	Two Cover	One Cover	Zero Cover
	In-Line	0.579	—	0.632
Parallel	0.655	0.678	0.653	—
Dual Source	0.650	—	0.647	0.496

initial simulations. The parallel and dual source systems showed no change in system performance, while the in-line system showed a noticeable improvement. However, the in-line system performance is still worse than the parallel or dual source performance. Therefore, it is felt that there is no advantage in using antifreeze in the storage tank. Zero cover collector simulations were also performed for the dual source and in-line systems. The results in Table 2.5 indicate that dual source and in-line systems with coverless collectors result in performance better than a conventional heat pump with no solar collectors at all ($F = 0.356$), and slightly better than conventional solar with single-glazed collectors ($F = 0.449$).

2.3.4.5 EFFECT OF CONTROL STRATEGY

In all of the results presented so far, the parallel system performance has been as good as, if not better than, the dual source system performance. This, plus its simplicity in construction, makes it the best solar heat pump system from a thermal point of view. Therefore, the parallel system was chosen to be simulated with the aforementioned "smart" controller. The reason for using the smart controller is to allow the heat pump to operate at more favorable times (during the day when ambient temperature is usually warmer than at night) and to store collected solar energy for possible night use. It was hoped that the expected improvement in heat pump performance would outweigh the expected decline in solar performance. The results using the smart controller shown in Table 2.6 consist of

TABLE 2.6

"Smart" Controller Results

T_{star} (°C)	QLOAD = 85.3 GJ		$T_{dh} = 30^{\circ}C$		$V/A_c = 0.075 \text{ m}^3/\text{m}^2$	$T_{room} = 20^{\circ}C$		$T_{low} = -30^{\circ}C$	
	QU (GJ)	QDH (GJ)	QRH (GJ)	QDH (GJ)		WAH (GJ)	COP _s (-)	QAUX _{sp} (GJ)	QAUX _{hw} (GJ)
30	38.3	23.8	34.9	34.9	17.5	1.99	5.14	6.88	0.654
40	38.0	23.4	35.5	35.5	17.7	2.01	5.01	6.80	0.654
50	37.8	23.2	35.8	35.8	17.9	2.00	4.88	6.70	0.654
∞	37.3	22.4	36.8	36.8	18.2	2.02	4.71	6.43	0.656

solar and heat pump performance measures for varying values of T_{star} . Table 2.6 shows that the increase in heat pump performance, QRH, was counterbalanced by a decrease in solar performance, QDH and QU. QDH and QU slightly decreased as T_{star} increased because while storing collected solar energy for possible night use, collector losses increased with a higher return temperature to the collector inlet from storage. The seasonal heat pump COP (COP_s) remained essentially constant at two. The results obtained here for varying values of T_{star} indicate no advantage to using the smart controller. Therefore, all subsequent simulations were done using the conventional control strategy used in previous simulations.

2.3.4.6 EFFECT OF IMPROVED HEAT PUMP PERFORMANCE

So far, all of the results shown for solar heat pump systems have been for systems using the "standard" three-ton heat pump. To see if a better performing heat pump will alter the decision of the best performing system, simulations were done on the conventional heat pump, parallel, dual source, and in-line systems using the improved heat pump operating characteristics shown in Fig. 2.11. Figure 2.15 and Table 2.7 show the results of these simulations performed by temperature control. As in the case of the other temperature control simulations, the service hot water load was not considered. The parallel, dual source, and conventional heat pump system performance improved with improvement in heat pump performance. The parallel system performed just as well as the dual source system. The in-line

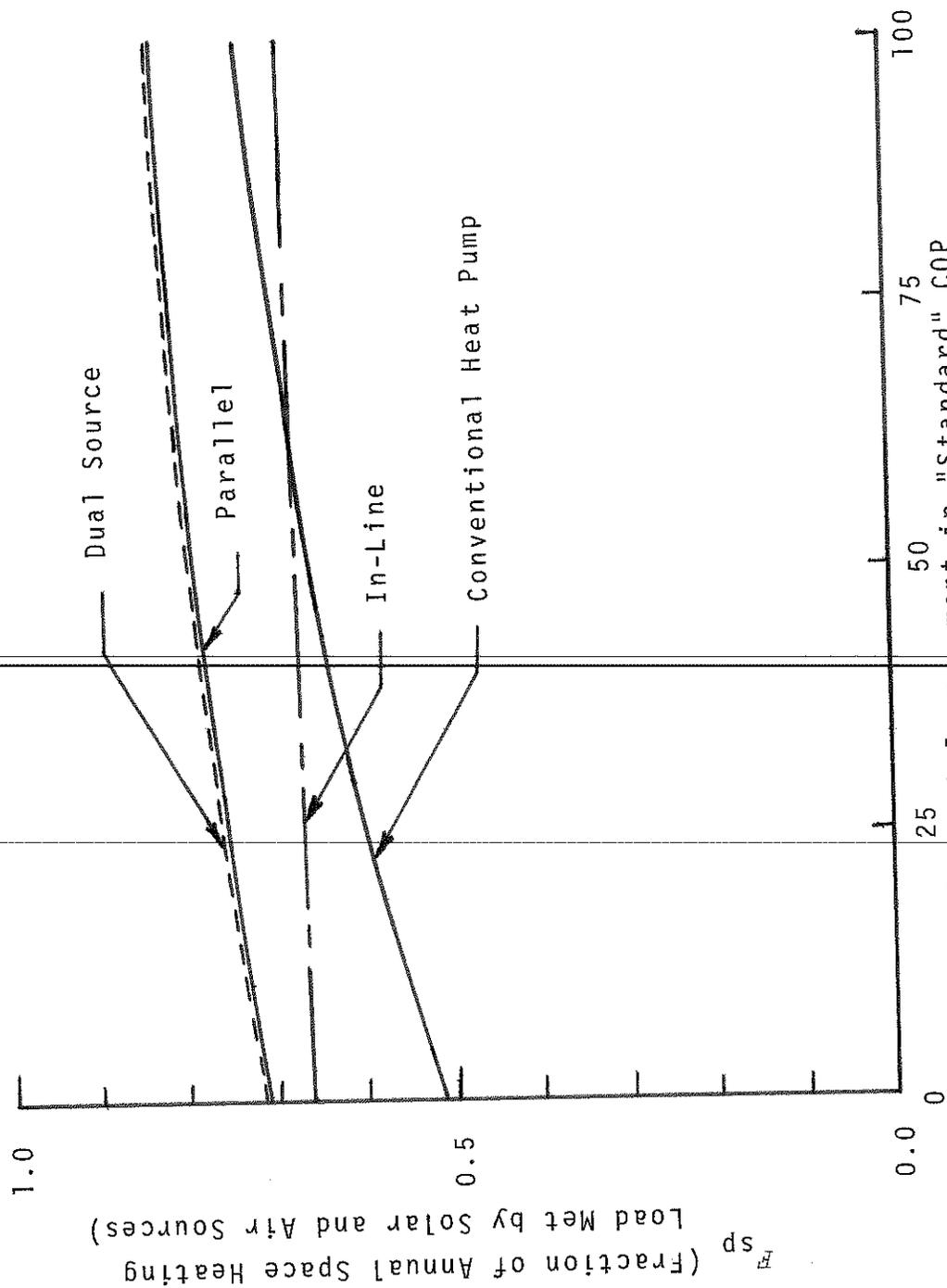


Figure 2.15 E_{sp} vs. Percent Improvement in "Standard" COP

TABLE 2.7
Improved Heat Pump Results

System	Percent Improvement in "Standard" COP	Fraction of Annual Space Heating Load Met by Solar and Air Sources (F_{sp})
Conventional Heat Pump	0	0.514
	25	0.600
	50	0.658
	100	0.730
Parallel	0	0.715
	25	0.760
	50	0.790
	100	0.828
Dual Source	0	0.720
	25	0.763
	50	0.792
	100	0.832
In-Line	0	0.663
	25	0.665
	50	0.672
	100	0.680

system performance barely improved with better heat pumps. The reason for this is that while improving the heat pump performance by decreasing the electrical power input to the heat pump to yield the same heating capacity as the "standard" unit, the rate of energy absorbed by the unit must increase. In doing this, the heat pump using storage as its source keeps the storage temperature lower than the "standard" unit did. Thus, the improvement in heat pump COP was counterbalanced by the lower source temperature.

2.3.4.7 ANALYSIS OF SYSTEM OPERATION

The parallel system has always performed as well as, if not better than, the dual source system in all of the simulations done in Madison. The in-line system never performed as well as either the parallel or the dual source system. These are observations, and the reasons why these systems perform like this must now be answered.

In order to answer this question, the contribution of performance measures in attempting to meet the annual space heating and service hot water energy requirements must be looked at closely. The performance measures of major importance are QU, QAIR, QAUX, WAH, HWA, HAA, and storage tank temperature frequency (number of hours that storage temperature was in a specified temperature range, or "bin"). In looking at previous simulation results, a well sized solar heat pump system for this house, having a UA of 620 KJ/hr-°C, is one consisting of the "standard" three-ton heat pump, 30 m² of single-glazed

collectors, and a storage volume to collector area ratio of $0.075 \text{ m}^3/\text{m}^2$.

All five systems have been simulated for an entire heating season by using the energy rate control method for a constant room temperature of 20°C .

Table 2.8 and Figs. 2.16 through 2.18 help explain what happens in solar heat pump system operation. The conventional heat pump and conventional solar systems are shown for comparison. Percentages of the annual space heating and service hot water load (85.3 GJ) met by QU, QAIR, QAUX, and WAH for the parallel, dual source, in-line, conventional solar, and conventional heat pump systems are shown in Table 2.8 and Fig. 2.16. Figure 2.16 shows system performance in terms of "free" (QU and QAIR) and "purchased" (QAUX and WAH) energy. Figure 2.17 shows HWA and HAA as a function of source temperature.

Figure 2.18 shows the main storage tank temperature frequency distribution for the parallel, dual source, in-line, and conventional solar systems. Although the solar part of all four systems operates during the heating and cooling seasons to meet the service hot water load, the frequency distribution for 5°C temperature "bins" shown in Fig. 2.18 is only for the heating season since the heating season is of major importance in determining the best performing solar heat pump system.

First of all, in looking at collected solar energy (QU) in Table 2.8 and Fig. 2.16, it can be seen that solar collector performance is better for dual source and in-line than parallel and conventional solar. This happens because storage for the dual source

TABLE 2.8

Percent of Annual Space and Service
Hot Water Load Met by Performance Measures

<u>System</u>	<u>QU</u>	<u>QAIR</u>	<u>QAUX</u>	<u>WAH</u>
Parallel	44.9	20.6	14.1	20.4
Dual Source	57.8	7.2	12.6	22.4
In-Line	57.9	—	27.6	14.5
Conventional Solar	44.9	—	55.1	—
Conventional Heat Pump	—	35.6	31.2	33.2

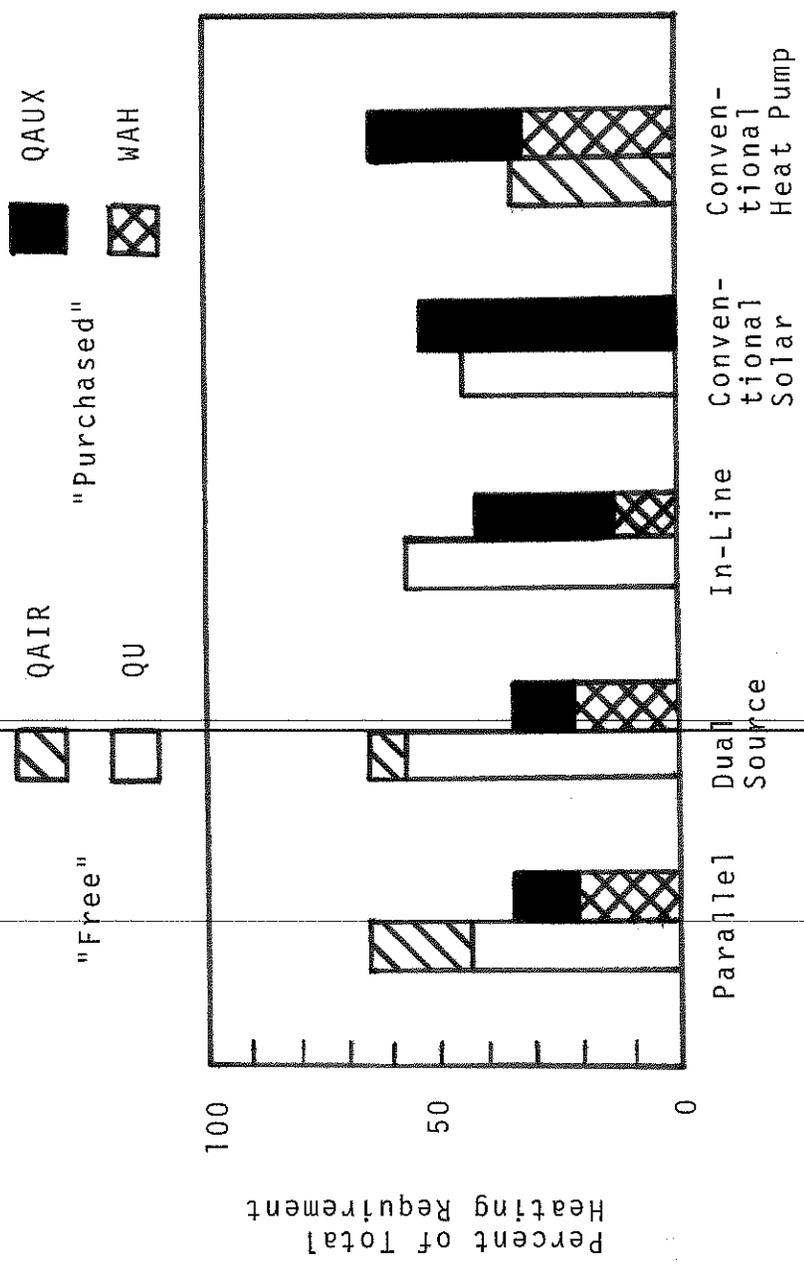


Figure 2.16 "Free" and "Purchased" Energy

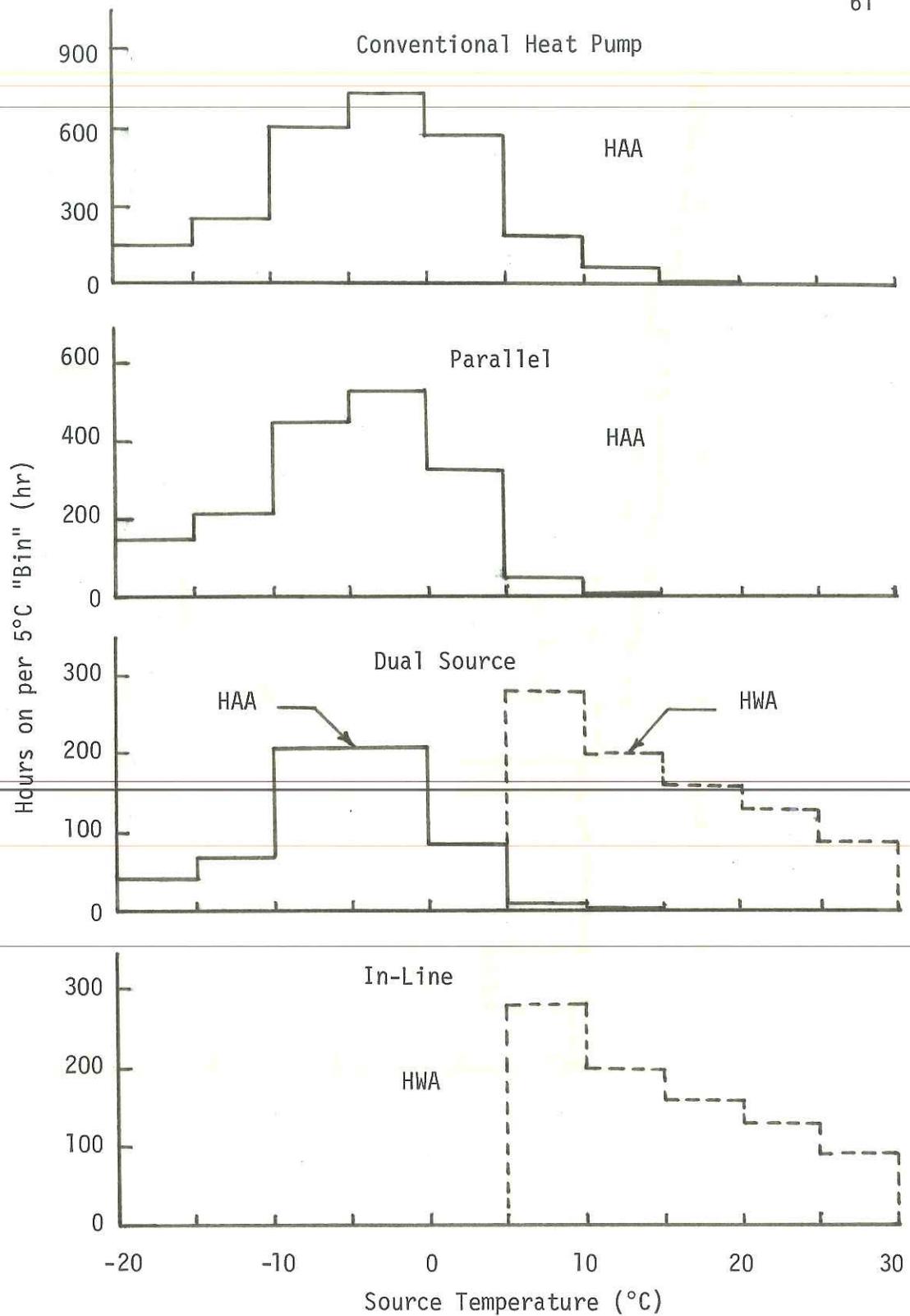


Figure 2.17 HWA and HAA vs. Source Temperature

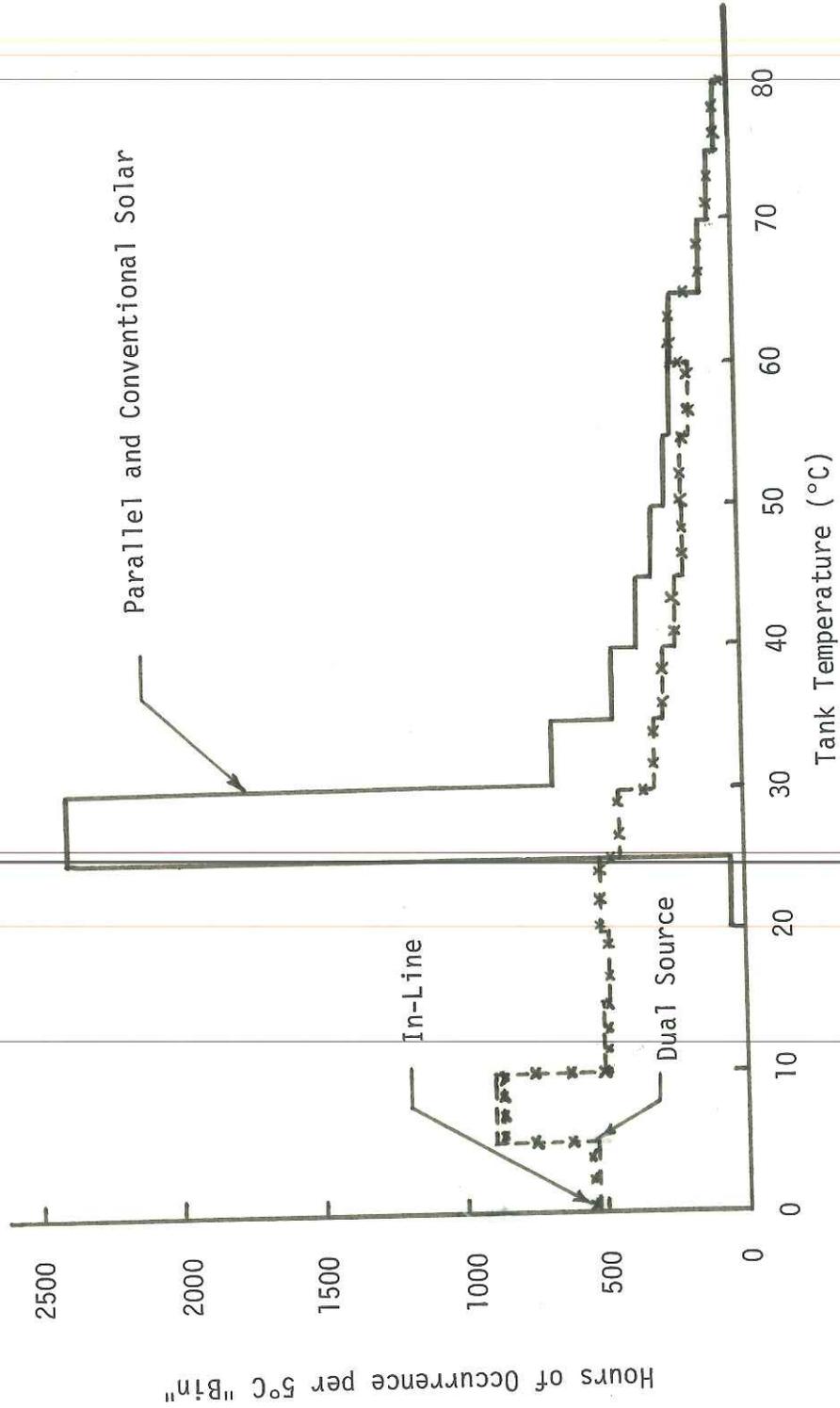


Figure 2.18 Frequency Distribution of Main Storage Tank Temperature

and in-line systems is more frequently at a lower temperature, as seen in Fig. 2.18. Since the storage temperature was generally lower, the return temperature to the collectors was lower resulting in lower collector radiation and convection losses and higher QU. QU is practically identical for the dual source and in-line systems and is reflected in the practically identical frequency distributions of main storage tank temperature seen in Fig. 2.18. This storage temperature being generally lower than parallel and conventional solar storage also indicates that most of the space heating was done by heat pump operation for the dual source and in-line systems, and very little by direct heating with solar energy.

For the parallel and conventional solar systems, main storage tank temperatures are generally higher than those for the dual source and in-line systems and result in more direct heating by solar energy and lower QU because of a higher return temperature to the collectors. Since the heat pump in the parallel system is not coupled with the storage tank, the solar performance for the parallel system is identical to that for the conventional solar system.

Secondly, since the parallel and dual source systems can use the ambient air as a source, they are able to extract energy from the ambient air (QAIR). The in-line system does not have an air source evaporator and is unable to use this available energy. The dual source system did not extract as much energy from the ambient air as parallel system because the storage temperature for the dual source system was often higher than the ambient air temperature when heating

was required. This is also reflected by the area under the HWA and HAA curves in Fig. 2.17 for the dual source system.

For the parallel system, more direct heating with solar energy and greater use of ambient air as a source compensates for a lower QU than the dual source system. On the other hand, the dual source system using storage more frequently as a source for heat pump operation yields higher QRH and QU than the parallel system compensating for less amounts of QAIR and direct heating. Both the parallel and dual source systems have different operating characteristics which tend to balance out to similar thermal performances. The heat pump operates approximately the same total number of hours for the heating season in either case.

The parallel system obtains a sizeable amount of energy from the ambient air, as shown in Table 2.8, while the in-line system obtains none. The heat pump in the parallel system will generally see a smaller load than in the in-line system because of more direct heating done by solar energy with the parallel system. Even though it sees a smaller load, the heat pump in the parallel system will operate more often than the in-line heat pump because of a lower source temperature. However, since the parallel system heat pump sees a smaller load, more of the load can be met by heat pump operation (COP greater than unity) requiring less auxiliary. The parallel system performance would improve with a larger capacity heat pump performing better at low ambient temperatures. However, the in-line performance would probably remain about the same as in the "standard" heat pump case

because while increasing QRWA, more energy must be extracted out of storage lowering the storage temperature even further and allowing less direct heating.

There are two important points to keep in mind when considering a heat pump's source. First, QAIR is readily available to the heat pump as a source whereas, QU must be made available to the heat pump. Thus, energy is required only once to transfer QAIR to the house (heat pump operation only), while QU requires energy both in the solar collector loop operation and the heat pump operation. Second, these simulations have not included the amount of energy required to run the pumps for circulating the collecting medium. However, this power is negligible in comparison to the other energy quantities. Therefore, the parallel system performance looks even better when these two points are considered.

In summary, simulation results for Madison and explanations of observations show that the parallel and dual source systems are comparable over the range of parameters studied. Also, because of the ability to use direct heating with solar energy and ambient air as a source, the parallel and dual source systems are definitely better than the in-line system. Simulation results for Albuquerque will now be presented to show that independent of geographical location, the parallel and dual source systems have the best thermal performance.

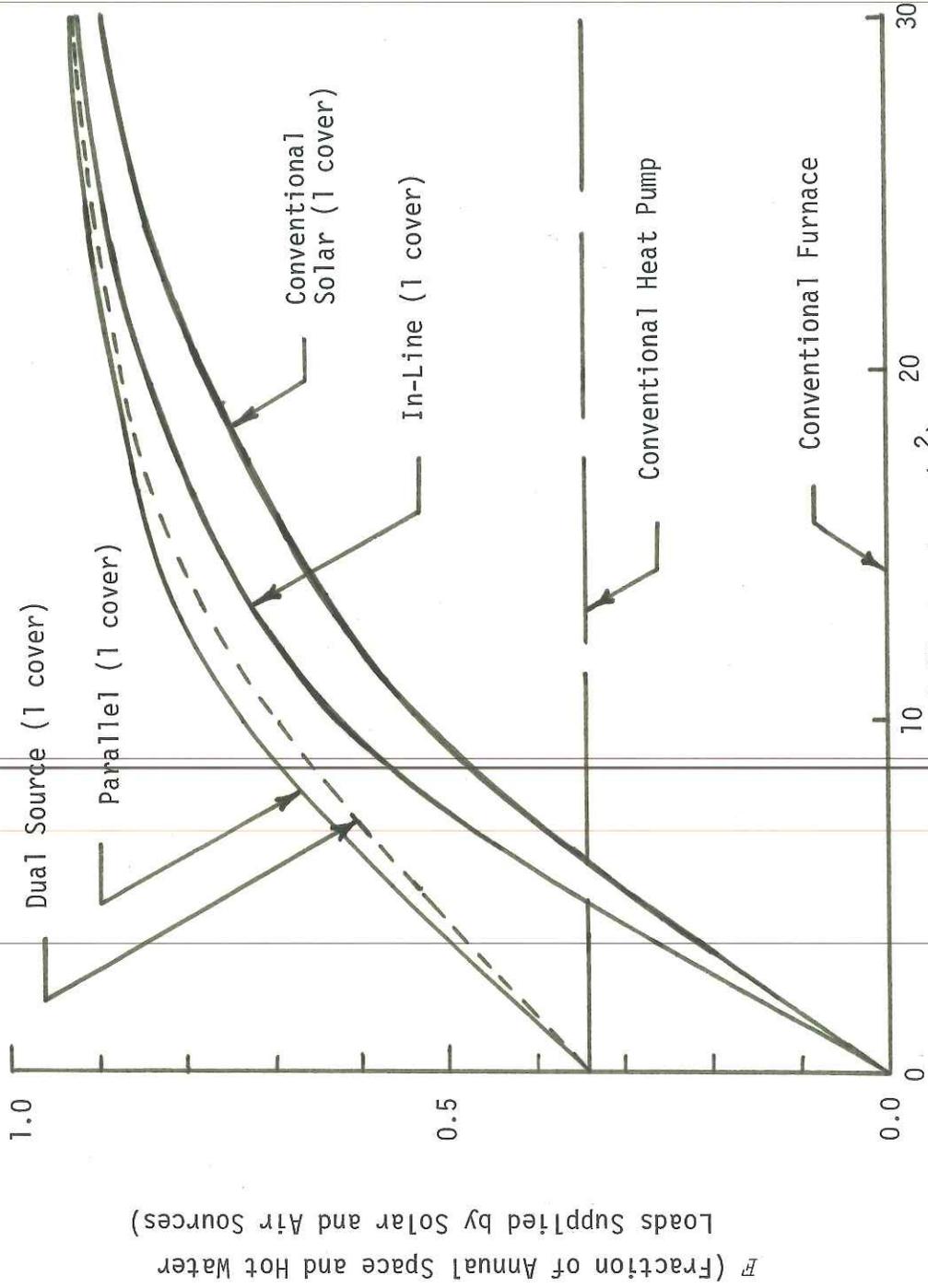
2.3.5 ALBUQUERQUE SIMULATION RESULTS

Simulations were done for Albuquerque, New Mexico, using a house heating UA of 620 KJ/hr-°C, constant room temperature of 20°C, the "standard" three-ton heat pump, V/A_C of $0.075 \text{ m}^3/\text{m}^2$, and energy rate control. Albuquerque has a milder climate than Madison and only 30 m^2 of single-glazed collectors are needed to achieve a savings in consumed energy of approximately 90 percent for the solar heat pump systems. The annual space heating and service hot water load was 50.92 GJ. Figure 2.19 and Table 2.9 show the results of simulations of the conventional furnace, solar, solar heat pump, and heat pump systems.

Comparisons between conventional furnace (electrical resistance heating), conventional solar, conventional heat pump (air-to-air), in-line, parallel, and dual source systems can be made using Fig. 2.19 and Table 2.9. The collector size necessary for the conventional solar system to consume less auxiliary energy than a conventional heat pump system is about 6 m^2 .

As seen for Albuquerque, the parallel and dual source systems perform like a conventional heat pump system, and the in-line system performs like a conventional solar system at zero collector area. Also, at large collector areas the parallel, dual source, and in-line systems behave like a conventional solar system since the solar part of the system meets most of the load.

Also, as was the case in Madison, the parallel system performed slightly better than the dual source system and substantially better than the in-line system. Table 2.9 shows that more heat is supplied



F (Fraction of Annual Space and Hot Water Loads Supplied by Solar and Air Sources)

Figure 2.19 F vs. A_c

TABLE 2.9

Albuquerque Simulation Results

$$Q_{LOAD} = 50.92 \text{ GJ} \quad V/A_c = 0.075 \text{ m}^3/\text{m}^2$$

System Description	A_c (m^2)	QDH (GJ)	QRWA (GJ)	QRAA (GJ)	QAUX _{sp} (GJ)	QAUX _{hw} (GJ)	WAH (GJ)	HWA (hr)	HAA (hr)	F
Conventional Heat Pump	—	—	—	31.51	0.18	19.23	13.98	—	1174	0.344
Conventional Solar 1 cover	10	12.63	—	—	19.06	4.85	—	—	—	0.530
	30	28.06	—	—	3.63	1.93	—	—	—	0.891
In-Line 1 cover	10	7.21	17.82	—	6.66	6.14	6.16	427	—	0.628
	30	26.66	4.41	—	0.62	2.10	1.43	95	—	0.918
Parallel 1 cover	10	12.63	—	18.88	0.18	4.85	8.70	—	757	0.730
	30	28.06	—	3.44	0.19	1.93	1.68	—	155	0.925
Dual Source 1 cover	10	7.23	17.76	6.56	0.14	6.14	9.27	425	281	0.695
	30	26.66	4.41	0.55	0.07	2.10	1.74	95	32	0.923

by the heat pump in the dual source system than in the parallel system, but more electrical work is required because the heat pump operates more since there is less direct heating by solar energy.

Figure 2.19 and Table 2.9 in comparison to Figs. 2.12 and 2.13 and Table 2.3 show similar trends in system performance for the parallel, dual source, and in-line systems with increasing collector area. Also, the comparisons show that for both locations, the parallel and dual source systems are the best performing solar heat pump systems from a thermal point of view.

2.4 CONCLUSIONS AND RECOMMENDATION

Simulation results for Madison and Albuquerque show that the parallel and dual source systems are comparable over the range of parameters studied. The in-line system never performed better than either the dual source or parallel system. For collector areas greater than zero, the parallel, dual source, and in-line systems will always perform better than the conventional solar system. Above a certain collector size for a given storage volume, the in-line system will perform better than the conventional heat pump system. The parallel and dual source systems will always perform better than the conventional heat pump system.

For a set value of A_c and V/A_c , the collected solar energy (QU) and collector efficiency will generally be better for dual source and in-line than for parallel and conventional solar systems because the storage for dual source and in-line systems will generally be more

frequently at a lower temperature. The lower storage temperature for dual source and in-line systems also indicates that a greater fraction of the space heating load was met by heat pump operation, and very little by direct heating with solar energy.

Since the parallel and dual source systems can use the ambient air as a source, they are able to extract energy from the ambient air (QAIR). The in-line system does not have an air source evaporator and is unable to use this available energy. The dual source system does not extract as much energy from the ambient air as the parallel system because the storage temperature for the dual source system was often higher than the ambient air temperature when heating was required.

Parallel and conventional solar systems do more direct heating by solar energy and have lower QU than the dual source and in-line systems because the main storage tank temperatures are generally higher. Also, the solar performance for the parallel system is identical to that for the conventional solar systems since the heat pump in the parallel system is not coupled with the storage tank.

For the parallel system, more direct heating and greater use of ambient air as a source compensates for a lower QU than the dual source system. However, the dual source system using storage more frequently as a source for heat pump operation yields higher QRH and QU than the parallel system compensating for less amounts of QAIR and direct heating. Both the parallel and dual source systems have different operating characteristics which tend to balance out to

similar thermal performances. The heat pump operates approximately the same total number of hours for the heating season in either case.

The heat pump in the parallel system will generally see a smaller load than in the in-line system because of more direct heating done by solar energy with the parallel system. Even though it sees a smaller load, the heat pump in the parallel system will operate more often than the in-line heat pump because of a lower temperature. However, since the parallel system heat pump sees a smaller load, more of the load can be met by heat pump operation (COP greater than unity) requiring less auxiliary.

In summary, the dual source system will always perform better than the in-line system because it consistently sees higher source temperatures for heat pump operation in the water-to-air heating mode, and it is able to use ambient air as a source at a COP greater than unity when the storage temperature is at its minimum. The parallel system will always perform better than the conventional solar system because backup heating is frequently obtained by a heat pump at a COP greater than unity. The parallel system will always do more direct heating with solar energy than either the dual source or in-line system because of the higher storage temperature. The parallel system will always extract more energy from ambient air than the dual source system. Since the power required to run the pumps for circulating the fluid in the solar collection and direct heating loops is negligible in comparison to the energy required to transfer heat through the heat pump, the parallel system always uses less energy than both the

dual source and in-line systems. The dual source and in-line systems require energy twice to heat the house when operating in the water-to-air heating mode (first to collect and store and second to operate the heat pump). Also, since fewer components are needed to construct and operate the parallel system than the dual source and in-line systems, the parallel system is less expensive.

The parallel system is recommended as the best way to combine a solar system with a heat pump system to obtain the greatest savings in conventional fuels with the least complexity. Following up on this recommendation, the parallel system is chosen as the system requiring a general design procedure for designing a system and/or readily evaluating system performance for a given load.

3.0 DESIGN PROCEDURE

3.1 CONVENTIONAL HEAT PUMP SYSTEM

Conventional air-to-air heat pump installations are generally sized for the design cooling load to insure proper dehumidification. However, when heating is the prime concern, the unit is usually oversized so that electrical resistance auxiliary heat is not often required. The actual size chosen for installation depends on the economics as well as the estimated seasonal performance. Although, the economics are important, this section concentrates on the procedure for estimating the seasonal performance of a conventional heat pump system so that a basis for understanding the design procedure for the parallel system can be obtained.

Estimating the seasonal performance of a conventional heating system (gas, oil, or electrical furnace, etc.) using the degree day method is relatively simple because the efficiency of the system is assumed to be constant regardless of ambient temperature. However, because the efficiency of the heat pump changes with variations in ambient temperature, the seasonal performance of the heat pump system must be dealt with differently.

The most common measure of heat pump system performance is the seasonal system COP, denoted as \overline{COP}_S . \overline{COP}_S is the total annual space heating load that the heat pump system must meet by heat pump rejected heat (QRH) and electrical resistance heat ($QAUX_{sp}$) divided by the total annual electrical input to the heat pump system (WAH and $QAUX_{sp}$).

The instantaneous COP is sometimes used but has less meaning because it is only a measure of the instantaneous efficiency of a heat pump and does not take into account the auxiliary electrical heat that is required below the system balance point.

In calculating $\overline{\text{COP}}_s$ by the "Bin" procedure, U.S. Government weather information based on 10 year averages is used to determine the number of hours per year that fall within each 5°F (2.78°C) temperature bin in a given area. Two sources of obtaining this information are the National Climatic Center [14] and the Air Force Manual 88-8, Chapter 6. This bin information is given on both a monthly and seasonal basis for a certain location's heating season. Note, this temperature bin information for the United States is given in degree Fahrenheit rather than degree Celsius.

The seasonal system COP is more frequently calculated than the monthly system COP, denoted as $\overline{\text{COP}}_m$, because $\overline{\text{COP}}_s$ is a faster method of determining the seasonal performance of a heat pump system. $\overline{\text{COP}}_s$ can be obtained from monthly calculations by summing up system performances (QRH, QAUX_{sp} , and WAH) for each month of the heating season.

Since the parallel system design procedure presented later requires that calculations be done on a monthly basis, the Bin procedure will now be discussed on a monthly basis rather than the more common seasonal basis so that similarities and differences can be seen.

Figure 2.5 is very helpful in understanding the calculations of system performance by the Bin procedure. It should be noted that the

house heating requirement is less than the structure heat loss because of internal heat sources. A reduction of approximately 7°F (3.89°C) from the design room temperature of 75°F (23.9°C) is assumed to make up this difference for residential applications.

The following is a step by step description of the Bin procedure for determining the system performance on a monthly basis:

- (1) Calculate the house heating UA.
- (2) Draw the constant house heating requirement on a figure similar to Fig. 2.5. Note that the internal heat sources are accounted for by drawing this straight line with the slope equal to the UA from (1) and intersecting the ambient temperature axis at approximately 68°F (20°C).
- (3) List the temperature differences for each bin (constant room temperature of 68°F minus the bin midpoint temperature). The bin midpoint temperature is, for example, 22°F for the 5°F temperature bin covering ambient temperatures from 20 to 24°F inclusive.
- (4) Calculate the hourly heating requirement for each bin (house heating UA times (3)).
- (5) List the hourly heating capacity of the heat pump for each temperature bin (information is obtained from manufacturer's data). As shown in Fig. 2.5, sketch these values on the same figure with the house heating requirement. From this and (2), the system balance point is determined.

- (6) Calculate the theoretical fraction of running time of the heat pump system for each bin. This is the hourly house heating requirement divided by the hourly heat pump heating capacity. The fraction is set equal to one below the system balance point indicating that the heat pump operates 100 percent of the time but still requires supplemental heat to meet the remaining house heating requirements.
- (7) List the hourly electrical input to the heat pump (compressor work plus indoor and outdoor fan motors) for each temperature bin. This is obtained from manufacturer's data.
- (8) List the number of hours (N_{amb}) that fall within each of the 5°F temperature bins as given by the sources aforementioned.
- (9) Calculate the hourly electrical resistance heating required below the system balance point for each temperature bin ((4) minus (5)).
~~At or above the system balance point (4) minus (5) is set equal to zero because the heating capacity of the heat pump easily meets the heating requirement.~~
- (10) Calculate the monthly heat rejected (QRH) by the heat pump for each temperature bin ((5) times (6) times (8)).
- (11) Calculate the monthly electrical input to the heat pump (WAH) for each temperature bin ((6) times (7) times (8)).
- (12) Calculate the monthly electrical resistance heat ($QAUX_{sp}$) for each temperature bin ((8) times (9)).
- (13) Obtain the total monthly QRH by summing up all of the values calculated in (10).

- (14) Obtain the total monthly WAH by summing up all of the values calculated in (11).
- (15) Obtain the total monthly $QAUX_{sp}$ by summing up all of the values calculated in (12).
- (16) The monthly system COP is

$$\overline{COP}_m = \frac{QRH_m + QAUX_{sp_m}}{WAH_m + QAUX_{sp_m}} \cdot$$

- (17) Repeat steps (10) through (16) for each month in the heating season.
- (18) Sum up the monthly values for (13), (14), and (15) for the entire heating season and calculate the seasonal COP;

$$\overline{COP}_s = \frac{QRH + QAUX_{sp}}{WAH + QAUX_{sp}}$$

where QRH = the summation of QRH_m ,
 WAH = the summation of WAH_m , and
 $QAUX_{sp}$ = the summation of $QAUX_{sp_m}$.

- (19) F_{sp} can be calculated for the entire heating season;

$$F_{sp} = (QLOAD_{sp} - WAH - QAUX_{sp}) / QLOAD_{sp}.$$

The parallel system design procedure is similar to this Bin procedure. There are differences, however. The reasons for these

differences and how they were accounted for in obtaining the parallel system design procedure will now be discussed.

3.2 ARRIVING AT PARALLEL SYSTEM DESIGN PROCEDURE

The parallel system is a combination of the conventional solar and conventional heat pump (air-to-air) systems. As discussed previously, the solar part of the parallel system operates whenever possible to meet the house heating requirements. If direct heating with solar energy cannot meet the entire house heating requirements, the heat pump system operates to meet the remainder of the load. Since the heat pump does not use the storage medium as a source in the parallel system, the solar system performance is identical to that of the conventional solar system. However, since the heat pump acts as a backup to the solar system, its performance differs from that of the conventional heat pump system. In fact, TRNSYS simulations show that the monthly system performance of the heat pump as a backup to a solar system declines for all months from that of the conventional heat pump as f_{spm} (the fraction of the monthly space heating load met by direct heating with solar energy) increases because the heat pump is forced to operate more frequently at less favorable times (generally at night when it is colder than during the day).

The fact that the solar part of the parallel system performs identical to the conventional solar system simplifies the design procedure for the parallel system. The solar performance of a parallel system can be readily evaluated by the f -Chart [10] method.

To this point, the following observations for continued research have been made. First, the solar performance of the parallel system can be readily obtained by using f -Chart. Second, the monthly heat pump system COP in the parallel system is worse than the monthly conventional heat pump system COP. Third, the design procedure for a parallel system must be geographical location and heat pump operating characteristics independent.

The air-to-air heat pump acting as a backup to the solar system operates in the same temperature bins as the conventional heat pump. However, it operates fewer hours than the conventional heat pump because there are times when solar can meet 100 percent of the space heating load and the heat pump does not have to operate. Since the heat pump in the parallel system uses ambient air as a source and operates in the same temperature bins as the conventional heat pump, it was hoped that a modification of the Bin procedure in connection with f -Chart would yield satisfactory results of predicting parallel system performance.

The first attempt at finding a design procedure for the parallel system originated from the observation that monthly heat pump system performance decreased as f_{sp_m} increased. For any given month, the greatest the heat pump system COP will ever be is when f_{sp_m} equals zero because of being forced to operate at less favorable times as f_{sp_m} increases.

After observing this decline in system COP, it was thought that plotting the ratio of system coefficients of performance as a function of f_{sp_m} (the ratio of the parallel system monthly system COP to the

monthly conventional system COP) would yield a means of predicting the heat pump system performance in the parallel system. Values of this ratio as a function of f_{sp_m} were first calculated from output of TRNSYS simulations using the "standard" three-ton heat pump and 10, 30, and 60 m² of collector area in Madison and 10 and 30 m² in Albuquerque, and then plotted. A fairly linear relationship was found. The ratio of monthly system coefficients of performance is equal to unity at f_{sp_m} equal to zero and decreases as f_{sp_m} increases.

This attempt in finding a general parallel system design procedure worked quite well when using the "standard" three-ton heat pump. However, when the TRNSYS output of simulations using the improved heat pump operating characteristics shown in Fig. 2.11 was plotted on the same graph as that of the "standard" three-ton heat pump, a substantial deviation from linearity was observed at small values of f_{sp_m} . This only occurred at small values of f_{sp_m} because the deviation from linearity was due to $QAUX_{sp}$ being almost equal to WAH. At large values of f_{sp_m} , $QAUX_{sp}$ is zero. These results proved that the initial attempt in finding a general parallel system design procedure was not independent of heat pump operating characteristics.*

It was then decided that finding a way to modify the Bin procedure for predicting the heat pump performance when acting as a solar system backup was the proper direction to head. The Bin procedure is independent of geographical location and heat pump operating characteristics. In order to obtain the modified Bin procedure, it was necessary to estimate first the amount of space heating load the air-

* Graph showing these relationships is in Appendix A3.

to-air heat pump system sees per temperature bin after the solar system attempts to meet the load, and second the maximum number of hours available per temperature bin for heat pump operation. It is very important that the heat pump system performance in the parallel system be analyzed bin by bin for a month rather than a season.

A histogram component was developed to be compatible with TRNSYS and yield information on $QLOAD_{sp}$, QDH , N_{amb} (the total number of hours that the ambient temperature was in a bin), and HAA for each ambient temperature bin of the month. By using this histogram, information on f_{sp_b} (the fraction of the space heating load met by direct heating with solar energy per temperature bin) and HAA below the system balance point was observed for each temperature bin in Madison TRNSYS simulations of 10, 30, and 60 m^2 .

Histogram outputs showed that f_{sp_b} was approximately f_{sp_m} . A plot of the ratio of f_{sp_b} to f_{sp_m} versus T_{amb} showed that at very cold ambient temperatures for the month, f_{sp_b} was approximately zero, and that at the other end of the temperature scale for the month f_{sp_b} was approximately unity. This however, was only at the very extremes and that for better than 90 percent of the ambient temperature range for the month, f_{sp_b} was approximately equal to f_{sp_m} . Since the majority of the total hours for the month are in temperature bins where f_{sp_m} approximately equals f_{sp_b} , the assumption that f_{sp_m} equals f_{sp_b} was used in the parallel system design procedure. Thus, the amount of space heating load that the heat pump system sees per bin is $(1-f_{sp_b})$ times $QLOAD_{sp_b}$.

Histogram outputs also showed that the maximum allowable hours for heat pump operation between the minimum allowable ambient temperature for heat pump operation and the system balance point was less than N_{amb} . The reason being, as mentioned earlier, is that even though f_{sp_b} was generally less than unity and sometimes very close to zero, there were times in a temperature bin when f_{sp_b} was unity and therefore, the heat pump was off. (See Appendix A4.)

Three attempts were made at estimating the available number of hours for heat pump operation below the system balance point temperature. The first attempt was to use N_{amb} times $(1-f_{sp_b})$. This resulted in the parallel system heat pump having a \overline{COP}_m identical to that for the conventional heat pump system. The uniform reduction in heat pump operation throughout all temperature bins resulted in a uniform reduction in performance measures for the month, whereas the actual number of hours that the heat pump operated below the system balance point was generally somewhat less than $(1-f_{sp_b})$ times N_{amb} . This actual number of hours required that more electrical resistance heating be used and resulted in a reduction in heat pump system performance when compared to the conventional heat pump system.

The second attempt was based on a correlation of the average tank temperature per bin with f_{sp_b} . This was attempted because first, histogram outputs from TRNSYS showed that for good values of house heat exchanger effectiveness (e.g. 0.75), the heat pump system seldom operated at the same time the solar system did. Second, if the average tank temperature per bin could be obtained, the average rate at

which direct heating was done could be found per bin. With this and the total space heating load met by solar energy per bin, an estimate of the number of hours of direct heating with solar energy per bin could then be obtained. A reduction of N_{amb} by this estimated hours of direct heating per bin yields the maximum hours of heat pump operation per bin below the system balance point. However, no correlation between the average tank temperature per bin and f_{sp_b} could be found to yield results similar to those obtained using TRNSYS.

The third attempt was to correlate the TRNSYS output of the hours of heat pump operation in the air-to-air heating mode (HAA) below the system balance point with T_{amb_b} and f_{sp_m} . It was observed that a linear approximation of HAA with T_{amb} below the system balance point resulted in values of heat pump operation close to those actually seen in TRNSYS simulations. As f_{sp_m} increased, the slope of HAA increased. This slope increased negatively from the system balance point temperature to the minimum ambient temperature for heat pump operation. The slope of HAA between the balance point and the ambient temperature cutoff for heat pump operation was plotted as a function of f_{sp_m} for TRNSYS simulation results of 10, 30, and 60 m^2 in Madison. A parabola fit this plot fairly well resulting in a slope equal to the square root of 4 times f_{sp_m} . Attempts at finding a point on the line representing the available number of hours for heat pump operation below the system balance point as a function of both f_{sp_m} and T_{amb_b} resulted in the equation

$$N_{hp_b} = N_{bin} - \left| (4.0 \times f_{sp_m})^{\frac{1}{2}} \times T_{amb_b} \right|$$

where N_{hp_b} is the available number of hours for heat pump operation per bin, N_{bin} is N_{amb} just below the system balance point, and the negative absolute value of the slope times T_{amb_b} is used to ensure that N_{hp_b} always decreases as T_{amb_b} decreases.

Two points to mention is that first N_{hp_b} is always between zero and N_{amb} . Therefore, if N_{hp_b} is calculated to be less than zero, N_{hp_b} equals zero. If N_{hp_b} is calculated to be greater than N_{amb} , then N_{hp_b} equals N_{amb} . Second, this equation for N_{hp_b} only applies below the system balance point and when f_{sp_m} is less than unity.

Above the system balance point, the heat pump operates only a fraction of the available number of hours determined by $(1-f_{sp_b})$ times $Q_{LOAD_{sp_b}}$ divided by the heating capacity of the heat pump per bin. Therefore, an estimation of the maximum number of hours for heat pump operation is not needed above the system balance point.

Now that a means of estimating the heat pump heating load and hours of operation has been determined, a design procedure has been found for the parallel system. A step by step description of the parallel system design procedure will now be presented followed by an example and a comparison of design and TRNSYS results.

3.3 PARALLEL SYSTEM

3.3.1 CALCULATION PROCEDURE AND EXAMPLE

The following is a step by step description of the parallel system design procedure for determining the system performance on a monthly basis (the first seven steps are the same as for the Bin method):

- (1) Calculate the house heating UA.
- (2) Draw the constant house heating requirement on a figure similar to Fig. 2.5.
- (3) List the temperature differences for each bin.
- (4) Calculate the hourly space heating requirement for each bin (house heating UA times (3)).
- (5) ~~List the hourly heating capacity of the heat pump for each temperature bin. Sketch these values on the same figure with the house heating requirement. From this and (2), the system balance point is determined.~~
- (6) List the hourly electrical input to the heat pump (compressor work plus indoor and outdoor fan motors) for each temperature bin.
- (7) List the number of hours (N_{amb}) that fall within each of the 5°F temperature bins as published by the sources aforementioned.
- (8) Calculate f_{sp_m} from f -Chart. Use this value as f_{sp_b} .
- (9) Calculate the heat pump system load per bin ($(1-f_{sp_b})$ times (4)).

(10) Calculate the theoretical fraction of running time of the heat pump system for each bin.

a) Above or at the balance point, it is equal to (9) divided by (5).

b) Below the balance point the fraction is one.

(11) Assign a value to N_{bin} . The value assigned should be that of N_{amb} in the first bin below the balance point.

(12) Calculate the hours of heat pump operation per bin.

a) Above or at the balance point,

$$N_{hp_b} = N_{amb} \times (10a)$$

b) Below the balance point,

$$N_{hp_b} = N_{bin} - \left| (4.0 \times f_{sp_m})^{\frac{1}{2}} \times T_{amb_b} \right|$$

(13) Calculate the monthly QRH by the heat pump for each temperature bin ((12) times (5)).

(14) Calculate the monthly WAH to the heat pump for each temperature bin ((12) times (6)).

(15) Calculate the monthly $QAUX_{sp}$ for each temperature bin (((7) times (9)) minus (13)). Note that $QAUX_{sp}$ equals zero if above the balance point.

(16) Obtain the total monthly QRH by summing up all of the values calculated in (13).

(17) Obtain the total monthly WAH by summing up all of the values calculated in (14).

(18) Obtain the total monthly $QAUX_{sp}$ by summing up all of the values calculated in (15).

(19) The monthly system COP is

$$\overline{COP}_m = \frac{QRH_m + QAUX_{sp_m}}{WAH_m + QAUX_{sp_m}} .$$

(20) Repeat steps (8) through (19) for each month in the heating season.

(21) Sum up the values in (16), (17), and (18) for an entire heating season and calculate the seasonal system COP;

$$\overline{COP}_s = \frac{QRH + QAUX_{sp}}{WAH + QAUX_{sp}} .$$

(22) F_{sp} can be calculated for the entire heating season;

$$F_{sp} = (QLOAD_{sp} - WAH - QAUX_{sp})/QLOAD_{sp} .$$

An example of using this procedure will now be presented for two temperature bins. One bin is below the balance point and the other is above it. The house is located in Madison, Wis. and has a UA of 620 KJ/hr-°C. The solar system part of the parallel system consists of 30 m² of single-glazed collectors with a V/A_c 0.075 m³/m². The "standard" three-ton heat pump system with electrical resistance auxiliary stands as the backup to the solar system.

For the region below the balance point:

(1) UA = 620 KJ/hr-°C.

(2) See Fig. 2.5.

(3) The bin chosen is from -14 to -16°C . Therefore, -15°C is the bin midpoint temperature. $\Delta T = (20 - (-15)) = 35^{\circ}\text{C}$.

$$(4) \text{QLOAD}_{sp} = UA\Delta T = 620 \times 35 = 21700 \text{ KJ/hr.}$$

$$(5) \text{QRH}_b = 13500 \text{ KJ/hr.}$$

The balance point temperature is shown in Fig. 2.5 to be approximately -9°C .

$$(6) \text{WAH}_b = 8600 \text{ KJ/hr.}$$

$$(7) N_{amb} = 36 \text{ hr.}$$

$$(8) \text{Using the } f\text{-Chart procedure, } f_{sp_m} = 0.26 = f_{sp_b}.$$

$$(9) \text{HPL}_{sp_b} = (1 - f_{sp_b}) \times \text{QLOAD}_{sp_b} = 16058 \text{ KJ/hr.}$$

$$(10) \text{Fraction} = 1.$$

(11) The first bin below the balance point temperature stated in (5)

has the bin midpoint temperature of -11°C . Therefore, $N_{bin} = N_{amb}$ at -11°C which is 33 hours.

$$(12) N_{hp_b} = 33 - |(4.0 \times 0.26)^{\frac{1}{2}} \times (-15)| \\ = 17.7 \text{ hr.}$$

$$(13) \text{QRH}_{m_b} = N_{hp_b} \times \text{QRH}_b \\ = 17.7 \times 13500 \\ = 238950 \text{ KJ.}$$

$$(14) \text{WAH}_{m_b} = N_{hp_b} \times \text{WAH}_b \\ = 17.7 \times 8600 \\ = 152220 \text{ KJ.}$$

$$\begin{aligned}
 (15) \text{QAUX}_{sp_b} &= N_{amb} \times \text{HPL}_{sp_b} - \text{QRH}_{m_b} \\
 &= 36 \times 16058 - 238950 \\
 &= 339138 \text{ KJ} .
 \end{aligned}$$

For the region above the balance point:

(1), (2), and (8) are the same as in the previous example.

$$(3) \Delta T = (20 - (-1)) = 21^\circ\text{C}$$

The bin midpoint temperature selected is -1°C .

$$(4) \text{QLOAD}_{sp_b} = UA \Delta T = 620 \times 21 = 13020 \text{ KJ/hr.}$$

$$(5) \text{QRH}_b = 24830 \text{ KJ/hr.}$$

$$(6) \text{WAH}_b = 11510 \text{ KJ/hr.}$$

$$(7) N_{amb} = 90 \text{ hr.}$$

$$\begin{aligned}
 (9) \text{HPL}_{sp_b} &= (1 - f_{sp_b}) \times \text{QLOAD}_{sp_b} \\
 &= (1 - 0.26) \times 13020 \\
 &= 9635 \text{ KJ/hr} .
 \end{aligned}$$

$$\begin{aligned}
 (10) \text{Fraction} &= \text{HPL}_{sp_b} / \text{QRH}_b \\
 &= 9635 / 24830 \\
 &= 0.388 .
 \end{aligned}$$

(11) Not needed.

$$\begin{aligned}
 (12) N_{hp_b} &= N_{amb} \times \text{Fraction} \\
 &= 90 \times 0.388 \\
 &= 34.92 \text{ hr} .
 \end{aligned}$$

$$\begin{aligned}
 (13) \text{QRH}_{m_b} &= N_{hp_b} \times \text{QRH}_b \\
 &= 34.92 \times 24830 \\
 &= 867064 \text{ KJ} .
 \end{aligned}$$

$$\begin{aligned}
 (14) \text{ WAH}_{m_b} &= N_{hp_b} \times \text{WAH}_b \\
 &= 34.92 \times 11510 \\
 &= 401929 \text{ KJ} .
 \end{aligned}$$

$$(15) \text{ QAUX}_{sp_{m_b}} = 0 \text{ (above the system balance point).}$$

These calculations would be done for each temperature bin. The remaining steps of (21) and (22) would then follow to obtain the desired seasonal performance. For comparison, N_{hp_b} from TRNSYS simulations is 18 hours at -15°C and 37 hours at -1°C . (The example showed 17.7 and 34.9 hours.)

Since this is a tedious exercise, a FORTRAN program for doing these calculations is presented in Appendix A1. Appendix A2 describes the terminology used in the program. This program was used to generate the data necessary to make the following comparisons to TRNSYS outputs. A word of caution pertaining to the program in the Appendix is that the units stated in Appendix A2 must be used with the program as it stands.

3.3.2 COMPARISON OF TEST RESULTS

The parallel system design procedure was used to generate results so that a comparison to TRNSYS simulation results could be obtained. It was important to test out this design procedure to make certain that it was independent of geographical location, solar system, and heat pump operating characteristics.

The parallel system used for this test was one consisting of a house heating UA of $620 \text{ KJ/hr-}^\circ\text{C}$, constant room temperature of 20°C ,

V/A_c of $0.075 \text{ m}^3/\text{m}^2$, and single-glazed collectors. Yearly simulations were done for Madison using 10, 30, and 60 m^2 of collector area, the "standard" three-ton heat pump, and the improved heat pump (50% improvement in COP). Yearly simulations were also done for Albuquerque using 10 and 30 m^2 of collector area and the "standard" three-ton heat pump.

The performance measure of major importance is the fraction of the monthly space heating load met by solar and air sources (F_{sp_m}). Using the exact same weather for both the design procedure and TRNSYS, values of F_{sp_m} were generated for the months of January through June and September through December for Madison and Albuquerque.

The results of these calculations are shown in Fig. 3.1. The parallel design procedure results agree with the TRNSYS results extremely well. The dash reference line in Fig. 3.1 signifies that the design procedure results match TRNSYS results exactly. The values of F_{sp_m} calculated by the design procedure were consistently within 3% of those from TRNSYS. The design procedure almost always predicted the performance of the parallel system to be slightly better than TRNSYS. This is mostly due to the assumption of $f_{sp_m} = f_{sp_b}$. In cold months, f_{sp_b} is almost unity at the high temperature bins. Also shown in Fig. 3.1 is the calculated standard deviation of 0.0216 signifying that design results matched TRNSYS results exceptionally well.

3.4 CONCLUSIONS AND RECOMMENDATIONS

The test results presented show that the parallel system design procedure arrived at yields excellent results for the conditions

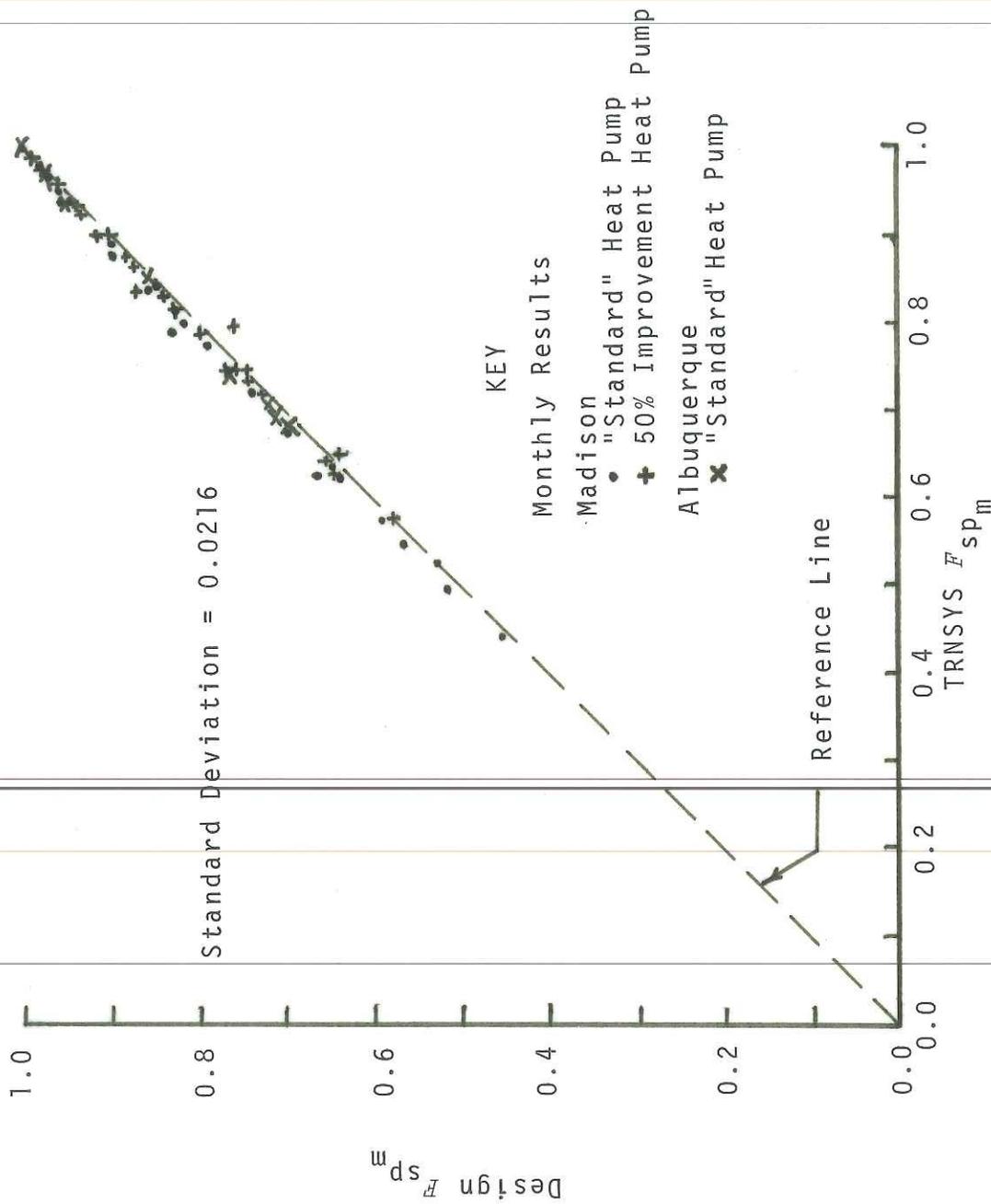


Figure 3.1 Design vs. TRNSYS Test Results

considered. The results obtained for the months when all of the ambient temperature bins were above the system balance point show a monthly system COP identical to that of the conventional heat pump system. This is due to the assumption that $f_{sp_m} = f_{sp_b}$. However, these are moderate heating months and in comparison to the annual space heating load, this slight inaccuracy is quite negligible.

Under all cases tested to date, the assumption of $f_{sp_m} = f_{sp_b}$ and the equation used to predict the available number of hours for heat pump operation below the system balance point are adequate. However, it is strongly recommended that both of these be further tested. The fraction of the space heating load met by direct heating with solar energy per temperature bin should be studied more carefully. Also, the effect of varying the load heat exchanger effectiveness on the value of f_{sp_m} should be evaluated.

In summary, the parallel system design procedure has worked for the situations presented. The control strategy required for this parallel system is the conventional control strategy described in Chapter 2. The system balance point determined by the house heating requirement line and heating capacity of the heat pump should be used for all values of f_{sp_m} . The system balance point should not be shifted to a lower ambient temperature as f_{sp_m} increases because there will be times in a temperature bin when solar does not do any direct heating and the heat pump acts identical to the conventional heat pump. Finally, further testing of the stated assumption of $f_{sp_m} = f_{sp_b}$ and hours of heat pump operation is recommended.

A P P E N D I X

A1 PROGRAM LISTING

```

C HEAT PUMP F-CHART * PARALLEL SYSTEM *
C SI UNITS SEEN ON OUTPUT ARE NEEDED TO RUN THIS PROGRAM.
C FRACTION BY SOLAR IS ASSUMED UNIFORM THROUGHOUT THE MONTH.
  REAL N,NHP,NBIN
  DIMENSION TITLE(12),MONTH(12),QLOAD(18),UNIT(5),QLHP(18)
  DIMENSION WAH(18),WA(18),QRH(18),QR(18),QAUX(18),NHP(18)
  COMMON/TINC/TINC,TMIN,QRDATA(7),WADATA(7),T(18),NB,N(18),B,TBP
C READ IN TITLE, UNITS, MONTH.
C READ IN FRACTION OF SPACE LOAD MET BY SOLAR, HOUSE UA, NO. OF
C HEAT PUMP PERFORMANCE POINTS, AND NO. OF BINS.
C READ IN AMBIENT TEMPERATURE (T) AND
C AMBIENT TEMPERATURE FREQUENCY PER BIN (N).
  READ(5,501,END=999) (TITLE(I),I=1,12)
  WRITE(6,601) (TITLE(I),I=1,12)
  WRITE(6,602)
  WRITE(6,603)
  READ(5,502) (UNIT(I),I=1,5)
  WRITE(6,604) (UNIT(I),I=1,5)
  1 READ(5,503,END=999) (MONTH(I),I=1,12)
  WRITE(6,605) (MONTH(I),I=1,12)
  READ(-,-) F,UA,NQ,NB
  WRITE(6,606) F,UA,NQ,NB
C READ IN AMBIENT TEMPERATURE AT EQUAL INTERVALS
C STARTING WITH THE COLDEST MONTHLY TEMPERATURE.
  READ(-,-) (T(I),I=1,NB)
  WRITE(6,666) (T(I),I=1,NB)
C READ IN AMBIENT TEMPERATURE FREQUENCY (N(I))
C CORRESPONDING TO (T(I)),
  READ(-,-) (N(I),I=1,NB)
  WRITE(6,667) (N(I),I=1,NB)
C READ IN HEAT PUMP CAPACITY DATA IN EQUAL
C TEMPERATURE INCREMENTS STARTING WITH THE
C CAPACITY CORRESPONDING TO TMIN.
C THE SIZE OF THE INCREMENT IS USER SPECIFIED BY TINC.
  READ(-,-) (QRDATA(I),I=1,NQ)
  WRITE(6,607) (QRDATA(I),I=1,NQ)
C READ IN HEAT PUMP WORK DATA CORRESPONDING
C TO CAPACITY DATA.
  READ(-,-) (WADATA(I),I=1,NQ)
  WRITE(6,677) (WADATA(I),I=1,NQ)
C READ IN MINIMUM AMBIENT TEMPERATURE FOR HEAT PUMP OPERATION,
C CONSTANT TEMPERATURE INCREMENT CORRESPONDING TO HEAT PUMP CAPACITY
C DATA, MAXIMUM AMBIENT AND CONSTANT ROOM TEMPERATURE FOR HEATING, ONE
C HALF OF BIN RANGE DEGREE C, AND THE BALANCE POINT TEMPERATURE.
  READ(-,-) TMIN,TINC,TMAX,TROOM,B,TBP
  WRITE(6,608) TMIN,TINC,TMAX,TROOM,B,TBP
C IF SOLAR MEETS ENTIRE SPACE LOAD, THE HEAT PUMP IS OFF.
  IF(F .GE. 1.) GO TO 90
C CALCULATING THE SPACE HEATING LOAD (QLOAD(I))
C AND THE HEAT PUMP HEATING LOAD (QLHP(I)).
  DO 20 I=1,NB
  IF (T(I) .GE. TMAX) GO TO 15
  QLOAD(I)=UA*(TROOM-T(I))
  QLHP(I)=(1,-F)*QLOAD(I)
  GO TO 20
C NO SPACE HEATING LOAD.
  15 QLOAD(I)=0.
  QLHP(I)=0.
  20 CONTINUE

```

```

C DETERMINING HEAT PUMP PERFORMANCE AND BACK UP ENERGY NEEDED TO MEET
C THE SPACE HEATING LOAD.
  CALL DATA(QR,WA)
  CALL BPT(NBIN,TBIN)
  DO 40 I=1,NB
    IF (T(I) .LT. TMIN) GO TO 50
C DETERMINING THE HOURS OF HEAT PUMP OPERATION.
    FRAC=QLHP(I)/QR(I)
    BF=QLOAD(I)/QR(I)
    IF(BF .LT. 1. .OR. F .LE. 0.) GO TO 41
    IF(BP .LT. 1.01 .AND. BP .GT. 1.) GO TO 43
    NHP(I)=-ABS(SQRT(4.*F)*T(I))+NBIN
    IF(NHP(I) .LT. 0.) NHP(I)=0.
    IF(NHP(I) .GT. N(I)) NHP(I)=N(I)
    GO TO 42
  43 FRAC=1.-F
  41 IF(F .LE. 0. .AND. BP .GE. 1.) FRAC=1.
    NHP(I)=N(I)*FRAC
  42 QRH(I)=NHP(I)*QR(I)
    QAUX(I)=N(I)*QLHP(I)-QRH(I)
    WAH(I)=NHP(I)*WA(I)
    GO TO 40
C HEAT PUMP IS OFF.
  50 QRH(I)=0.
    WAH(I)=0.
    QAUX(I)=N(I)*QLHP(I)
  40 CONTINUE
C SUMMING UP THE ENERGY QUANTITIES PER BIN FOR THE MONTH.
  SQLOAD=0.
  SQLHP=0.
  SQR=0.
  SWA=0.
  SQAUX=0.
  DO 60 I=1,NB
    SQR=QRH(I)+SQR
    SWA=WAH(I)+SWA
    SQAUX=QAUX(I)+SQAUX
  SQLOAD=N(I)*QLOAD(I)+SQLOAD
  SQLHP=N(I)*QLHP(I)+SQLHP
  60 CONTINUE
C DETERMINING THE MONTHLY HEAT PUMP COP.
  COPHP=SQR/SWA
C DETERMINING THE MONTHLY HEAT PUMP SYSTEM COP.
  COPHPS=(SQR+SQAUX)/(SWA+SQAUX)
  WRITE(6,609) SQLOAD,SQLHP,COPHP,SQR,SQAUX,SWA,COPHPS
  501 FORMAT(20A4)
  502 FORMAT(5A5)
  503 FORMAT(20A3)

```

```

601 FORMAT('1',40X,20A4)
602 FORMAT('0',40X,'THE FOLLOWING UNITS MUST BE USED')
603 FORMAT('0',15X,'DIMENSIONAL - UNIT SYSTEM'/
1      ' ',15X,'DIMENSIONS: MASS,M      TEMP,T      LENGTH,L      ENERGY
2,E TIME,0')
604 FORMAT(' ',15X,'USER UNITS: ',5(A5,5X))
605 FORMAT('0',15X,'MONTH OF YEAR ... ',20A3)
606 FORMAT('0',15X,'FRACTION BY SOLAR,F = ',F6.4,' HOUSE UA = ',
1F6.1,' NUMBER OF CAPACITY POINTS,NQ = ',I2,'/
2      ' ',15X,'NUMBER OF TEMPERATURE BINS,NB = ',I2)
666 FORMAT('0',15X,'TBIN..... ',9F7.2,'/',15X,9F7.2)
667 FORMAT('0',15X,'NBIN..... ',9F7.2,'/',15X,9F7.2)
607 FORMAT('0',15X,'QRDATA ..... ',7F10.1)
677 FORMAT('0',15X,'WADATA ..... ',7F10.1)
608 FORMAT('0',15X,'TMIN = ',F6.2,' TINC = ',F6.2,' TMAX = ',F6.2,'
1 TROOM = ',F6.2,' B = ',F6.2,' TBP = ',F6.2)
609 FORMAT('0',15X,'SQLOAD = ',E11.4,' SOLHP = ',E11.4,' COPHP = ',
1F5.3,' SQR = ',E11.4,'/',15X,' SQAUX = ',E11.4,' SWA = ',E11.4,
2' COPHPS = ',F5.3)
GO TO 100
90 WRITE(6,610)
610 FORMAT('0',15X,'F = 1.      HEAT PUMP IS OFF.')
```

```

C
C HEAT PUMP DATA INTERPOLATER.
C
```

```

SUBROUTINE DATA(QR,WA)
DIMENSION QR(18),WA(18)
COMMON/TINC/TINC,TMIN,QRDATA(7),WADATA(7),T(18),NB,N(18),B,TBP
DO 1 I=1,NB
J=(T(I)-TMIN)/TINC
R=((T(I)-TMIN)-FLOAT(J)*TINC)/TINC
J=J+1
QR(I)=QRDATA(J)+R*(QRDATA(J+1)-QRDATA(J))
WA(I)=WADATA(J)+R*(WADATA(J+1)-WADATA(J))
1 CONTINUE
RETURN
```

```

C
C FINDING THE FREQUENCY OF AMBIENT TEMPERATURE(NBIN)
C JUST BELOW THE BALANCE POINT TEMPERATURE.
C
```

```

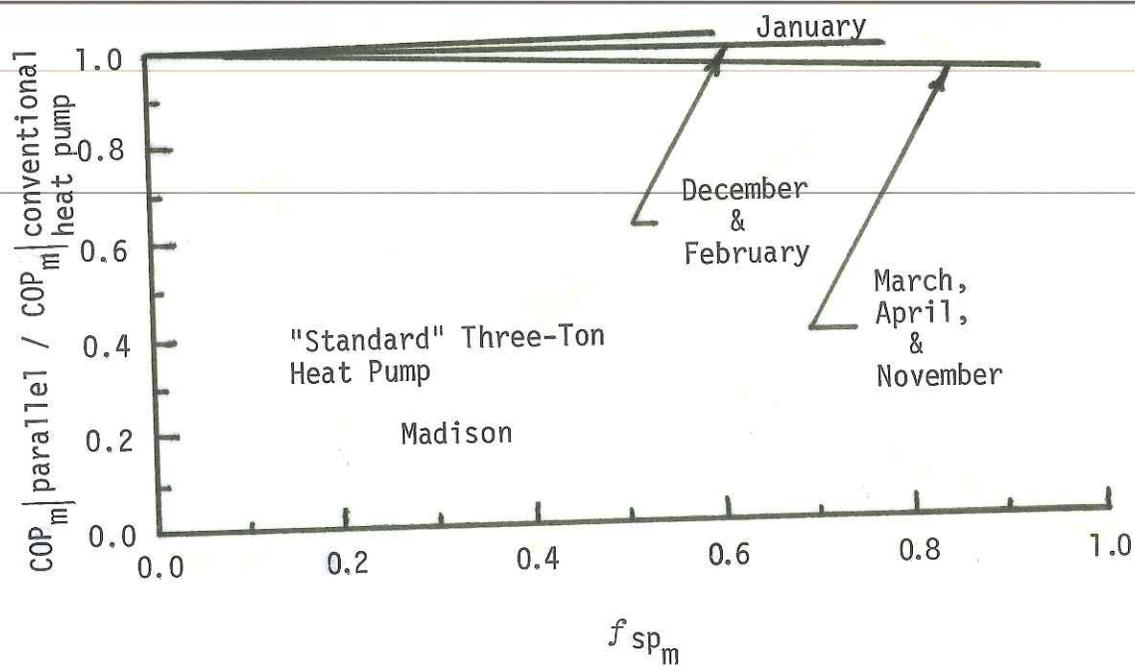
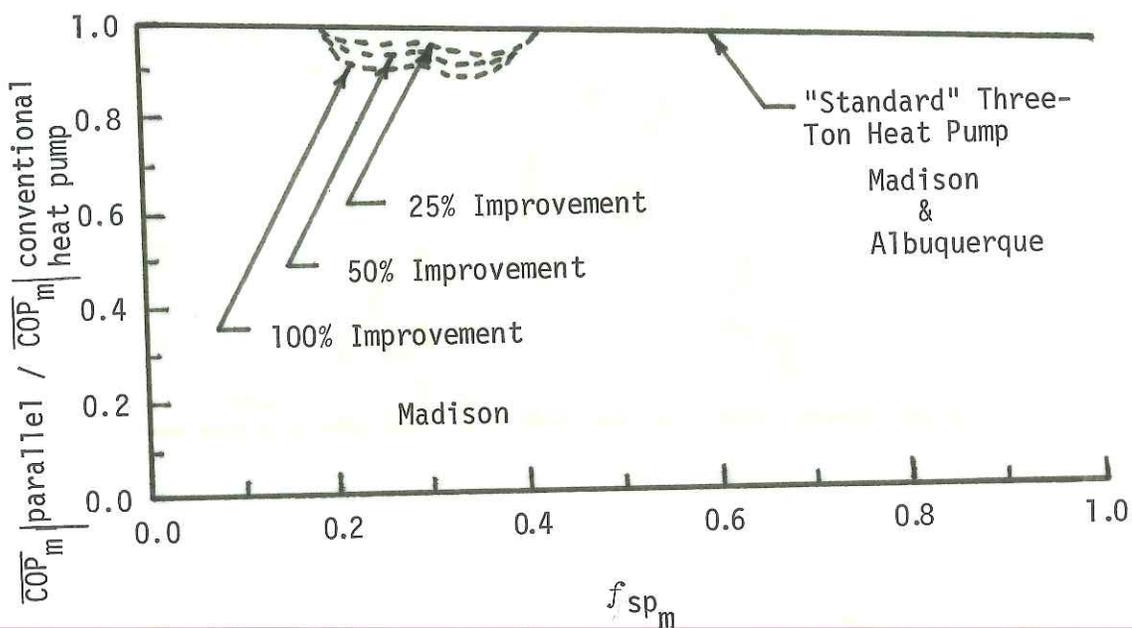
SUBROUTINE BPT(NBIN,TBIN)
COMMON/TINC/TINC,TMIN,QRDATA(7),WADATA(7),T(18),NB,N(18),B,TBP
TBIN=-30.
NBIN=0.
DO 1 I=1,NB
XBIG=T(I)+B
XLIT=T(I)-B
IF(TBP .LE. XBIG .AND. TBP .GE. XLIT) TBIN=T(I)
IF(TBIN .GT. -29.) NBIN=N(I)
IF (TBIN .GT. -29.) RETURN
1 CONTINUE
END
```

A2 DESCRIPTION OF TERMS

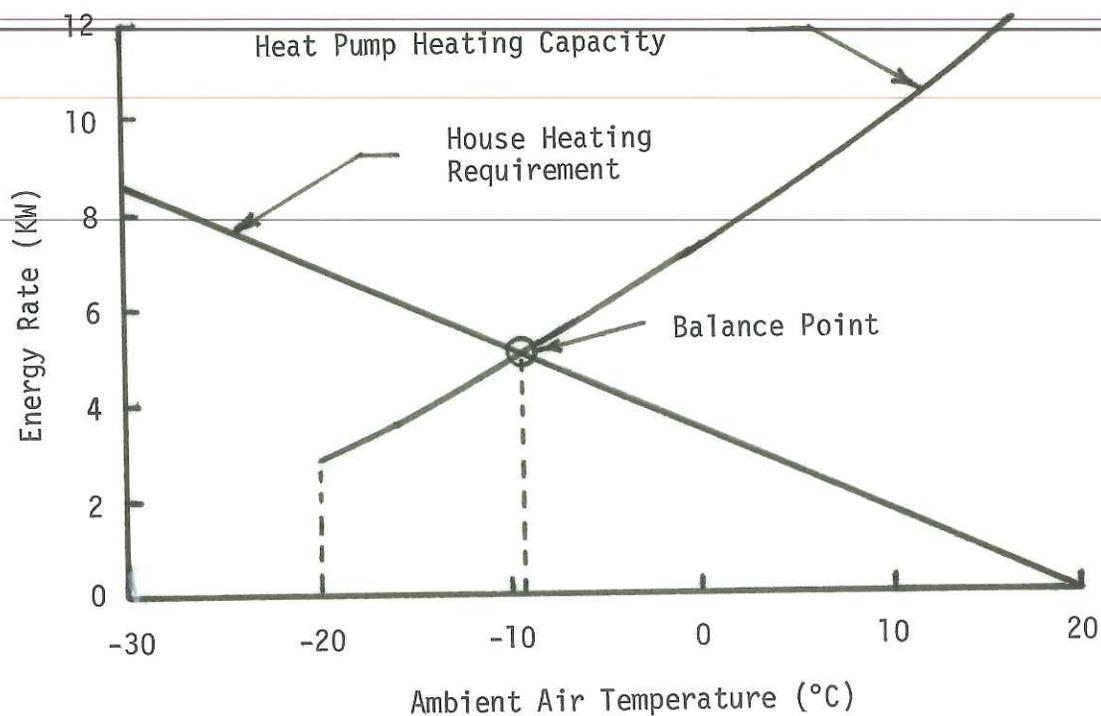
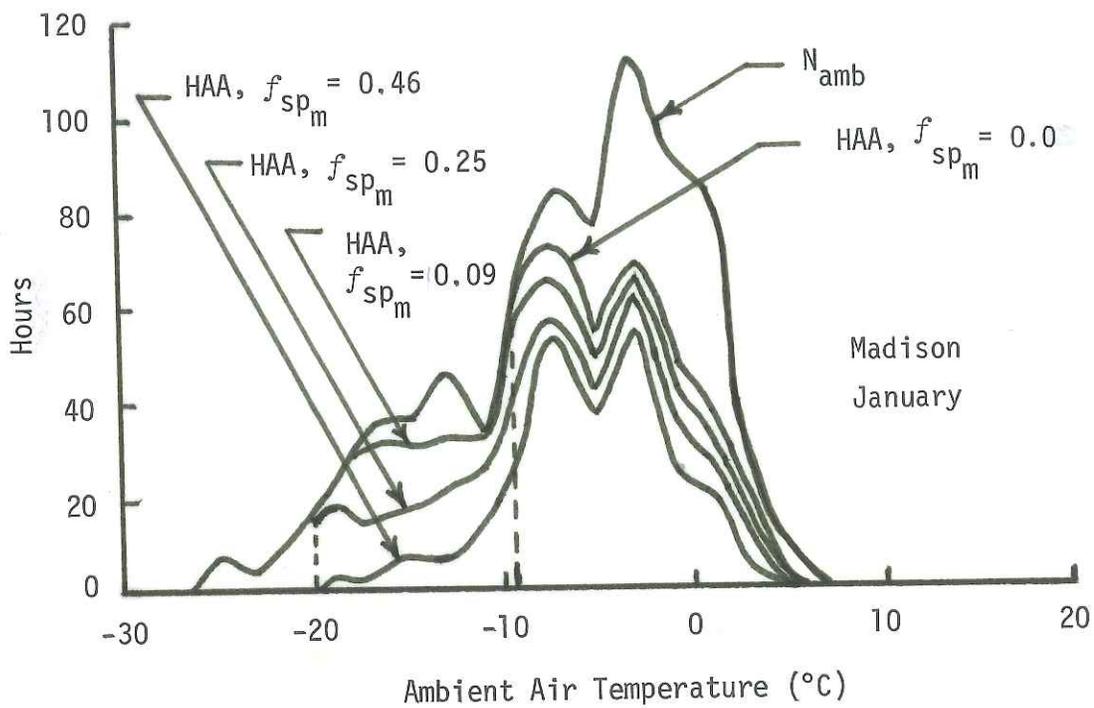
<u>TERM</u>	<u>DESCRIPTION</u>	<u>UNITS</u>
B	One Half of Bin Temperature Range	°C
BP	Ratio of Hourly Space Heating Load to Hourly Heat Pump Heating Capacity per Bin	—
COPHP	Monthly Heat Pump COP	—
COPHPS	Monthly Heat Pump System COP	—
F	Fraction of Space Heating Load Met by Direct Heating with Solar Energy for the Month	—
FRAC	Ratio of Hourly Space Heating Load for the Heat Pump System to Hourly Heat Pump Heating Capacity per Bin	—
MONTH	Month of Heating Season (Alpha Characters)	—
N(I)	Number of Hours That Ambient Temperature is in Bin I	hr
NB	Number of Temperature Bins	—
NBIN	N_{amb} in First Bin Below the System Balance Point	hr
NHP(I)	Number of Hours of Heat Pump Operation in Bin I	hr
NQ	Number of Heat Pump Heating Capacity Data Points	—
QAUX(I)	Monthly Auxiliary Energy Required in Bin I	KJ
QLHP(I)	Hourly Space Heating Load for Heat Pump System in Bin I	KJ/hr
QLOAD(I)	Hourly Space Heating Load for Parallel System in Bin I	KJ/hr
QR(I)	Heat Pump Heating Capacity Rate in Bin I	KJ/hr
QRDATA(I)	Heat Pump Heating Capacity Rate Data in Bin I	KJ/hr
QRH(I)	Monthly Heat Rejected by Heat Pump in Bin I	KJ
SQAUX	Monthly total of Auxiliary Required	KJ

<u>TERM</u>	<u>DESCRIPTION</u>	<u>UNITS</u>
SQLHP	Monthly Total Space Heating Load for Heat Pump System	KJ
SQLOAD	Monthly Total Space Heating Load for Parallel System	KJ
SQR	Monthly Total Heat Rejected by Heat Pump	KJ
SWA	Monthly Total Electrical Input to Heat Pump for Compressor Work and Fans	KJ
T(I)	Bin Midpoint Temperature in Bin I	°C
TBIN	Bin Midpoint Temperature in First Bin Just Below the System Balance Point	°C
TBP	System Balance Point Temperature	°C
TINC	Constant Ambient Temperature Increment for Which Heat Pump Data is Supplied	°C
TITLE(I)	User Supplied Title Describing the System to be Tested (Alpha Characters)	—
TMAX	Maximum Ambient Temperature for Heat Pump Operation (This Value Must be Less Than or Equal to TROOM)	°C
TMIN	Minimum Ambient Temperature for Heat Pump Operation	°C
TROOM	Constant Room Temperature for Heating	°C
UA	House Heating UA	KJ/hr-°C
UNIT(I)	User Supplied Units (Must be Those Units Seen in This Section; Alpha Characters)	—
WA(I)	Heat Pump Electrical Power Input in Bin I	KJ/hr
WADATA(I)	Heat Pump Electrical Power Input Data Corresponding to Capacity Data in Bin I	KJ/hr
WAH(I)	Monthly Work Input to Heat Pump in Bin I	KJ

A3 MONTHLY SYSTEM COP AND MONTHLY HEAT PUMP COP RATIOS VS. f_{sp_m}



A4 HISTOGRAM OUTPUTS OF N_{amb} AND HAA



REFERENCES

1. Mitchell, J. W., Freeman, T. L. and Beckman, W. A., "Combined Solar-Heat Pump Systems for Residential Heating," paper from the Solar Energy Laboratory, University of Wisconsin, Madison, Wisconsin, (March 1977).
2. Freeman, T. L., "Computer Modeling of Heat Pumps and the Simulation of Solar Energy-Heat Pump Systems," M.S. Thesis, Engineering, University of Wisconsin, Madison, Wisconsin, (1975).
3. General Electric, Solar Heating and Cooling of Buildings, Phase 0 Feasibility and Planning Study Final Report, vol. 2, book 1, (May 1974).
4. Pennsylvania State University, Workshop on Solar Energy Heat Pump Systems for Heating and Cooling Buildings, (June 1975).
5. Proceedings of the International Solar Energy Society, vol. 3, Winnipeg, Canada, (August 15-20, 1976).
6. Karman, V. D., Freeman, T. L. and Mitchell, J. W., "Simulation Study of Solar Heat Pump Systems," Proceedings of ISES, Winnipeg, Canada, (August 15-20, 1976).
7. Bosio, R. C. and Suryanarayana, N. V., "Solar Assisted Heat Pump System: A Parametric Study for Space Heating of a Characteristic House in Madison, Wisconsin," ASME paper 75-WA/SOL-8, presented at Winter Meeting, (1975).
8. Solar Energy Laboratory of the University of Wisconsin, "TRNSYS - A Transient Simulation Program," Engineering Experiment Station Report 38 (March 1975).
9. Abbaspour, M. and Glicksman, L. R., "The Proper Use of Thermal Storages for a Solar Assisted Heat Pump Heating System," ASME paper 76-WA/HT-76, presented at Winter Meeting, (1976).
10. Beckman, W. A., Klein, S. A. and Duffie, J. A., Solar Heating Design by the f-Chart Method, John Wiley & Sons, Inc., (1977).
11. Hottel, H. C. and Whillier, A., "Evaluation of Flat-Plate Solar-Collector Performance," The Use of Solar Energy, vol. II, Thermal Processes, Part I, Section A.
12. Duffie, J. A. and Beckman, W. A., Solar Energy Thermal Processes, John Wiley & Sons, Inc., (1974).

13. Liu, B. Y. H. and Jordan, R. C., "The Interrelationship and Characteristic Distribution of Direct, Diffuse, and Total Solar Radiation," Solar Energy, vol. IV, pp. 1-19, (July 1960).
14. U.S. Department of Commerce - Weather Bureau, "Decennial Census of United States Climate - Summary of Hourly Observations - Madison, Wisconsin," 1951-1960.

