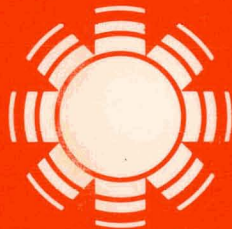SERI/TR-34-090   June 1979

Simulations and Economic Analyses of Desiccant Cooling Systems

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SIMULATIONS AND ECONOMIC
ANALYSES OF DESICCANT
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JUNE 1979

PREPARED UNDER TASK No. 3134.90

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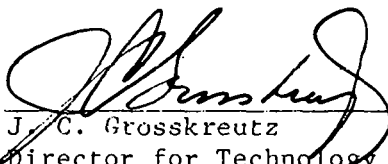
This report was prepared in compliance with Contract No. EG-77-C-01-4042 for the U.S. Department of Energy. It documents a portion of the research output of Task 31.34.90 in the Thermal Conversion Branch of the Research Division. The authors would like to recognize the contribution by John Mitchell, a consultant from the University of Wisconsin at Madison, who assisted in the preparations for the computer simulations of solar heating and desiccant cooling systems.

Frank Kreith
Branch Chief
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Approved for:

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SUMMARY

This report documents the progress to date in the development and analysis of computer simulations of solar-powered desiccant cooling using an axial-flow disc-type dehumidifier wheel, solar-powered space heating, and electrically driven, standard vapor-compression air-conditioning systems for residential use. Computer simulations for both solar and conventional heating and cooling systems were performed for 12-month heating and cooling seasons. Annual thermal performance and the resulting life cycle costs for both types of systems were analyzed and compared.

The heating/cooling season simulations were run for five U.S. cities representing a wide range of climatic conditions and insolation. With the information resulting from these simulations, the optimum air-conditioning system was chosen to maximize the conservation of fossil fuels and minimize operating costs. Because of the increasing use of residential air conditioning employing electrically driven vapor-compression coolers, the five locations were studied to determine if it would be beneficial (in terms of both economics and fossil fuel displacement) to displace fossil-fuel-powered vapor-compression coolers and natural gas space heaters with solar-powered heating and desiccant cooling systems.

The overall conclusions include the following:

- solar heating and desiccant cooling systems operate best in climates with nearly balanced heating and cooling loads (Section 3.7.3);

- solar desiccant coolers operate preferentially as dehumidifiers (Sections 3.6.2 and 3.7.3);

- the optimum desiccant cooler size for Washington, D.C., with auxiliary cooling and Dodge City, Kans., without auxiliary cooling is 4.5 kW with 35-m^2 collector area (Sections 3.7.3 and 5.3);

- under the cost assumptions of Section 5.2, the life-cycle cost for a 20-year solar heating/desiccant cooling system is nearly cost competitive with conventional residential heating and cooling (Section 5.3); and

- adding a desiccant cooler (with or without auxiliary cooling) to a solar heating system can increase fossil fuel displacement (Section 5.3) by a factor greater than 2, depending on the location.

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NOMENCLATURE

A	Outside wall area of house
A_c	Ceiling area of house
A_w	Window area of house
ASHRAE	American Society of Heating, Refrigerating, and Air Conditioning Engineers
AUX	Auxiliary cooling
b_n	Transfer function coefficient for temperature terms, house load model
CAP	Thermal house capacitance
CFM	Fan air flowrate
COP	Coefficient of performance
C_{pa}	Specific heat of dry air
C_{pw}	Specific heat of water
d_n	Transfer function coefficient for heat flux terms, house load model
DT	Temperature dead band for collector
ε	Evaporative cooler effectiveness
ε_i	Inlet evaporative cooler effectiveness
ε_o	Outlet evaporative cooler effectiveness
E	Fan efficiency
EER	Energy efficiency ratio
FRACT	Mass fraction of airflow that is dry
F_s	Shading fraction
H_{fg}	Heat of vaporization of water
H_T	Solar radiation
H_{TW}	Solar radiation on windows
i	Discount rate
INF	House air infiltration rate
LAT	Latent house load
$\dot{m}_C$	Collector system air flowrate
$\dot{m}_L$	Desiccant cooler system air flowrate
$\dot{m}_S$	Desiccant cooler system design air flowrate
m_T	Total mass of dry air and water vapor in house
MODEH	Heating/cooling mode indicator

NOMENCLATURE (Continued)

OMRI	Operating, maintenance, replacement, and insurance costs (annual)
p	Atmospheric pressure
p_w	Partial pressure of water vapor
P	System pressure drop
PP	Parasitic power
PW	Present worth of life-cycle costs
Q_B	Heat flow through house basement
Q_C	Total conduction through house walls and roof
Q_{IN}	Energy supplied to regenerate desiccant
Q_{INF}	House infiltration heat gain
Q_L	Constant house heat gains (people, lights, appliances, etc.)
Q_{LAT}	Latent cooling capacity of air conditioner
Q_{LOAD}	Total house cooling load
Q_n	Heat flux conducted through wall at time n
Q_o	Total heat conducted through ceiling
Q_{SEN}	Sensible cooling capacity of air conditioner
Q_{SHG}	House solar heat gain through windows
Q_{wall}	Heat conducted through house walls
$Q_{windows}$	Heat conducted through house windows
S/L	Ratio of sensible to latent cooling capacity components of air conditioner
SENS	Sensible house load
SHR	Sensible heat ratio (sensible capacity/total capacity of air conditioner)
T*	Saturation temperature
T_{AMB}	Ambient temperature
T_{CI}	Inlet cold airstream regenerator temperature
T_{CO}	Outlet cold airstream regenerator temperature
T_{COLL}	Average air temperature from collector during one time interval
T_{ENV}	Temperature of pebble bed environment
T_{EX}	Air temperature to house from inlet evaporative cooler
T_{HI}	Inlet hot airstream regenerator temperature
T_{HO}	Outlet hot airstream regenerator temperature

NOMENCLATURE (Concluded)

T_I	Dehumidifier inlet air temperature
T_{IN}	Evaporative cooler inlet air temperature
T_{MAX}	Upper temperature limit on desiccant cooler operation
T_{MAXC}	Maximum collector temperature allowed
T_{MIN}	Lower temperature limit on desiccant cooler operation
$\overline{T}_{OUT}$	Average collector outlet air temperature
T_{OUT}	Air temperature from storage or the collector to the dehumidifier (depending on operation mode)
T_{OUTEC}	Evaporative cooler outlet air temperature
T_{PEBED}	Average pebble bed temperature
T_R	Average house air temperature
T_{rc}	Constant room temperature, house load model
T'_{REG}	Regenerator temperature supplied to collector/storage system
T_{REG}	Regeneration temperature to dehumidifier
T_{RO}	Average house air temperature from previous time step
$T_{sa,eff,n}$	Effective sol-air temperature of roof surface and infiltration air
$T_{sa,n}$	Sol-air temperature of wall at time n
T_{SET}	House set temperature
T_{SS}	Steady-state house air temperature
T_{STOR}	Average storage temperature
TRNSYS	Transient Simulation Program developed by the University of Wisconsin
U_o	Overall heat transfer coefficient for house walls and windows
U_{WD}	Heat transfer coefficient for windows
U_{WL}	Heat transfer coefficient for walls
V_{STOR}	Pebble bed storage volume
W_{AMB}	Ambient air humidity ratio
W_{EX}	Air humidity ratio to house from inlet evaporative cooler
W_G	House interior moisture generation rate
W_I	Inlet humidity ratio to dehumidifier
W_{IN}	Inlet humidity ratio to evaporative cooler
W_R	House air humidity ratio
W_{RO}	House air humidity ratio from previous time step

SECTION 1.0

INTRODUCTION

This study evaluated residential solar desiccant cooling potential in order to identify optimum system configurations, climatic conditions, and economic parameters. In addition, since the U.S. Department of Energy (DOE) has given SERI the responsibility for managing the national solar cooling programs using desiccant processes, the system simulation techniques and analysis methods will be useful in evaluating other DOE contractor desiccant cooling systems.

Several air-conditioning configurations involving different physical, chemical, and electrical processes can be utilized to produce cooling effects that are adequate for residential buildings. These processes are summarized in Table 1-1.

Table 1-1. POTENTIAL AIR-CONDITIONING PROCESSES

Process	Open Cycle	Closed Cycle
Mechanical compression	Air cycle	Rankine cycle Brayton Stirling
Absorption	Desiccant	Absorption
Adsorption	Desiccant	Adsorption
Electronic transport	Thermionic emission	Peltier effect

The processes can be implemented in open or closed cycles. The closed cycle involves two separate process loops coupled by heat exchangers, with one loop for the refrigeration process and the other for the transfer of heat from the load. The open-cycle implementation eliminates the need for an interfacing heat exchanger by combining the refrigeration and heat transfer loops.

The open-cycle adsorption desiccant process represents an attractive alternative to conventional air-conditioning systems when coupled with a solar energy supply system. Most existing refrigeration and air-conditioning equipment operates on a closed mechanical compression cycle, specifically the Rankine vapor-compression cycle. There are several major advantages of a solar desiccant cooler based on a recyclable air desiccation process:

- It uses inexpensive materials and has potential to be manufactured at low cost.

- It uses air and inert inorganic materials and thus will not present corrosion or environmental problems.

- It can tolerate air leakage and be easily serviced, thus making its maintainability and reliability attractive.

- It can tolerate a wide range in solar input and still generate a usable output.

1

The potential for residential use of a solar desiccant cooler (using a silica gel desiccant) instead of an electrically driven, Rankine vapor-compression cycle unit was investigated in this study.

1.1 USE OF RESIDENTIAL AIR CONDITIONING IN THE UNITED STATES

According to the Federal Energy Administration (FEA) "Project Independence Report", residential buildings used approximately 70% of the 18.1 quads of energy consumed in the Household and Commercial Sector in 1972 (as defined by FEA, this sector accounts for 32% of all energy used in the United States) [1]. Of the roughly 12.7 quads used in residential buildings, about 72% (9.0 quads) was used for heating, cooling, and ventilation. Table 1-2 shows the growth in residential consumption of natural gas and electricity by end use for the period 1960-1968 [2].

Figure 1-1 illustrates FEA projections for growth in energy demand in the residential sector through 1990 by the major end uses. Although space heating dominates the energy used in residential buildings, air conditioning is projected to increase in energy use at an annual compound growth rate larger than for any other use. It is projected that by 1990 residential space heating and air conditioning will consume over 11 quads of energy, most of which will be obtained from fossil fuels. FEA stated that energy conservation measures to improve the thermal efficiency of both new and existing housing will be required to reduce residential energy demand [1]. The use of solar energy in residential space heating and air conditioning could have a significant impact on the displacement of fossil fuel in residential use.

1.2 OBJECTIVES

This study was initiated to evaluate the potential for solar desiccant air conditioning in terms of system performance and potential fossil fuel displacement. Although some work in the development of desiccant cooling has been performed, the interaction of the system components and their seasonal operation still require study, particularly when additional system operating complexities are introduced with solar power.

The portion of the task that is covered by this report involved the examination of one particular solar desiccant system configuration (see Section 2.2) in several different geographical locations. The overall objectives were:

- to determine the optimum use of solar desiccant coolers for particular climates (i.e., optimum cooler size, collector area, etc.);

- to determine the optimum size combination of a solar desiccant cooler and auxiliary vapor-compression cooler;

- to determine the total cost of a solar desiccant cooler versus a vapor-compression cooler;

Table 1-2. RESIDENTIAL CONSUMPTION OF NATURAL GAS AND ELECTRICITY BY END USE (Trillions of Btu) 1960-1968 [2]

	1960	1961	1962	1963	1964	1965	1966	1967	1968
Natural gas									
Space heating	2,188	2,316	2,666	2,589	2,755	2,816	3,000	3,157	3,236
Water heating	650	677	719	753	791	847	902	923	979
Cooking	316	314	318	318	319	321	323	323	325
Clothes drying	26	29	32	36	39	44	49	53	58
Refrigeration	32	26	21	16	12	9	7	6	5
Air conditioning	--	--	--	1	1	1	2	2	3
Total	3,212	3,362	3,756	3,713	3,917	4,038	4,283	4,464	4,606
Electricity[a]									
Refrigeration	122	137	152	167	183	200	215	232	250
Water heating	155	163	176	177	189	197	205	213	223
Space heating	29	38	45	55	81	106	116	145	164
Air conditioning	48	55	62	72	85	99	113	133	154
Television	59	63	69	75	83	93	105	116	128
Cooking	73	75	78	80	83	86	89	91	96
Food freezer	30	35	40	45	52	58	65	72	80
Clothes drying	23	25	28	30	34	38	42	46	51
Other[b]	203	194	197	209	198	185	205	226	244
Total	742	785	847	910	988	1,062	1,155	1,274	1,390

[a]At 3,413 Btu/kWh
[b]Dishwashers, washing machines, miscellaneous small appliances, and lighting
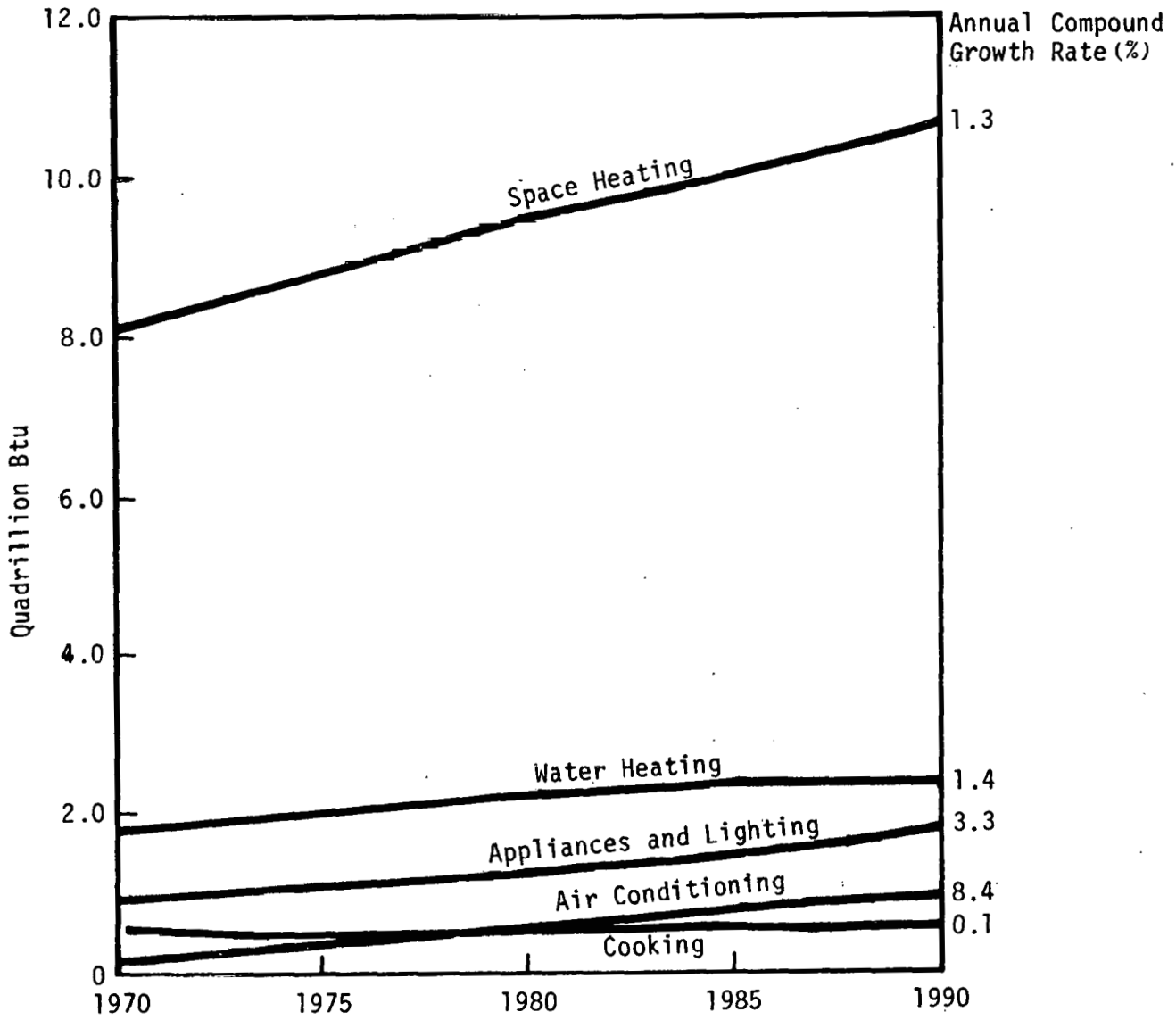
Figure 1-1. FEA-PROJECTED GROWTH IN RESIDENTIAL ENERGY DEMAND (1970-1990) BY END USE [1] (Note: Electricity measured at point-of-entry)

- to determine the potential for displacement of fossil fuel energy with a solar heating and desiccant cooling system; and

- to compare the cost and fossil fuel displacement of solar heating systems versus combined solar-heating/dessiccant-cooling systems.

In addition, DOE has given SERI the program management responsibilities for national solar cooling programs using desiccant processes. The techniques developed in this task will be useful in evaluating and comparing solar desiccant cooling systems developed by other DOE contractors.

1.3 METHODOLOGY

In order to achieve the stated objectives, the primary need was for computer models to simulate the annual operating performance of solar desiccant and vapor-compression coolers. A computer program to perform the economic analysis based on the results of the seasonal simulations was also required.

For the solar desiccant cooler simulation, it was decided to utilize TRNSYS, a transient simulation computer program developed by the University of Wisconsin at Madison [3]. The standard TRNSYS routines in conjunction with several special routines developed for desiccant cooler components [4] enabled seasonal simulations to be performed. For this study, certain TRNSYS routines were modified as described in Section 3.2.

To perform the vapor-compression cooler seasonal simulation and the economic analyses for both types of coolers, some additional computer programs (separate from TRNSYS) were developed specifically for this study.

Finally, appropriate sites had to be chosen for the simulations. SOLMET cities were chosen for study because of the availability of hourly meteorological data for use in the computer models. From the sites shown in Figure 1-2, the following five cities were chosen: Charleston, S. C.; Dodge City, Kans.; Ft. Worth, Tex.; Phoenix, Ariz.; and Washington, D.C. These five cities represent a large variation in insolation and climatic conditions. The annual insolation for the sites is shown in Figure 1-3, with climatic design conditions shown in Table 1-3.

Table 1-3. CLIMATIC DESIGN CONDITIONS

Location	Normal Annual Degree Days [5] 18.3°C Base		Design [6] Dry-bulb Temp. (°C)	Design [6] Wet-bulb Temp. (°C)
	Heating	Cooling		
Charleston, S.C.	2146	2078	34.4	27.2
Dodge City, Kans.	5046	1411	37.8	23.3
Ft. Worth, Tex.	2382	2587	38.3	25.6
Phoenix, Ariz.	1552	3508	42.8	24.4
Washington, D.C.	4211	1415	33.9	25.6

Figure 1-2. SOLMET SITES

Figure 1-3.   AVERAGE ANNUAL DIRECT INSOLATION (In Thousands of Btu/Ft^2 Yr) [7]

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SECTION 2.0

DESICCANT COOLER SYSTEM CHARACTERISTICS

Several desiccant cooler system configurations have been proposed by various organizations. These system concepts utilize different airflow configurations, dehumidifier designs, desiccant materials, and regenerative heat exchangers. Only desiccant cooler systems that could be simulated with existing TRNSYS routines were considered in this initial study. This section describes two open-cycle desiccant cooler configurations that could be modeled using existing TRNSYS routines and then discusses the choice for this study.

2.1 DESICCANT COOLER SYSTEMS

Most of the research and development for application of the adsorption process to solar cooling has been with the open desiccant cycle. The early work was done in Australia [Commonwealth Scientific and Industrial Research Organization (CSIRO)] and the United States [Institute of Gas Technology (IGT)]. The systems used two open process airstreams which were thermally coupled through rotary regenerators. The building airstream was dried in a desiccant bed, cooled in a regenerative heat exchanger, and refrigerated by evaporative cooling. The air drier was reactivated by an outside airstream that was solar heated to supply the desorption energy. The IGT approach, called Solar-MEC, required a gas-fired boost to achieve the required levels of desorption. In both systems, the desiccant bed operated as an adiabatic process.

The two system configurations used for this study operate in the ventilation and recirculation modes. The ventilation mode introduces ambient air into the conditioned space, while the recirculation mode recirculates air from the conditioned space. Both systems use the same components; the only differences are in the ducting and airflow paths.

The ventilation system configuration is shown in Figure 2-1. In the supply stream, ambient air is dehumidified, sensibly cooled, evaporatively cooled, and then introduced into the conditioned space. Air removed from the conditioned space is used in the regenerating stream, where it is first evaporatively cooled, then heated as it cools the supply stream, heated again by solar energy, cooled and humidified by the dehumidifier, and finally exhausted to the ambient air. This process is illustrated on the psychrometric diagram of Figure 2-2. (The numbers correspond to the bulk airstream conditions at the numbered system locations shown in Figure 2-1.)

For the recirculation system configuration, illustrated in Figure 2-3, the supply and regenerating airflow streams are separate. The supply stream air is removed from the conditioned space, dehumidified, sensibly cooled, evaporatively cooled, and then reintroduced into the conditioned space. In the regenerating stream, ambient air is evaporatively cooled, then heated as it cools the supply stream, heated again by solar energy, humidified and cooled as it regenerates the dehumidifier, and exhausted to the ambient air. This process is shown on a psychrometric plot in Figure 2-4; again, the
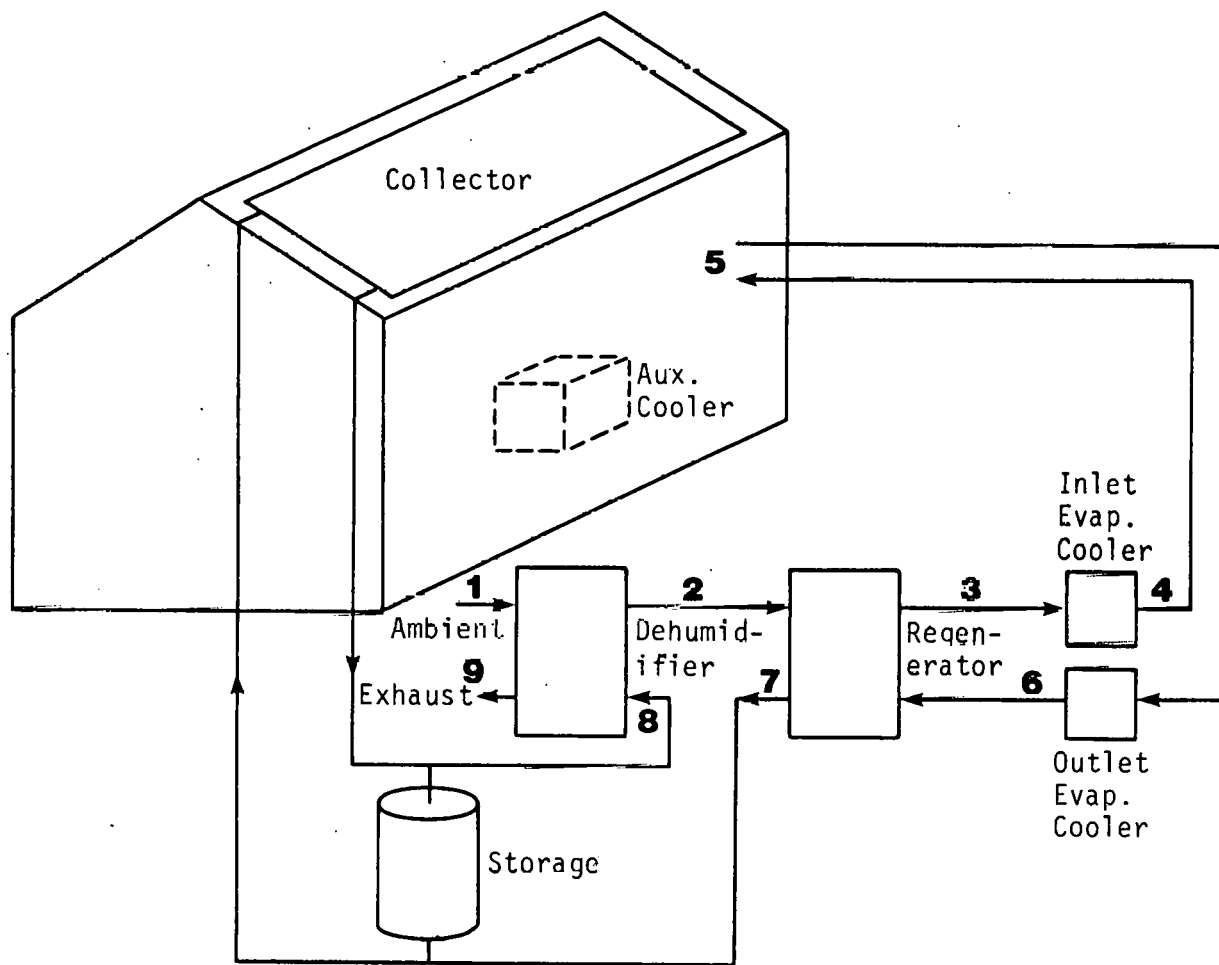
Figure 2-1. VENTILATION SYSTEM CONFIGURATION
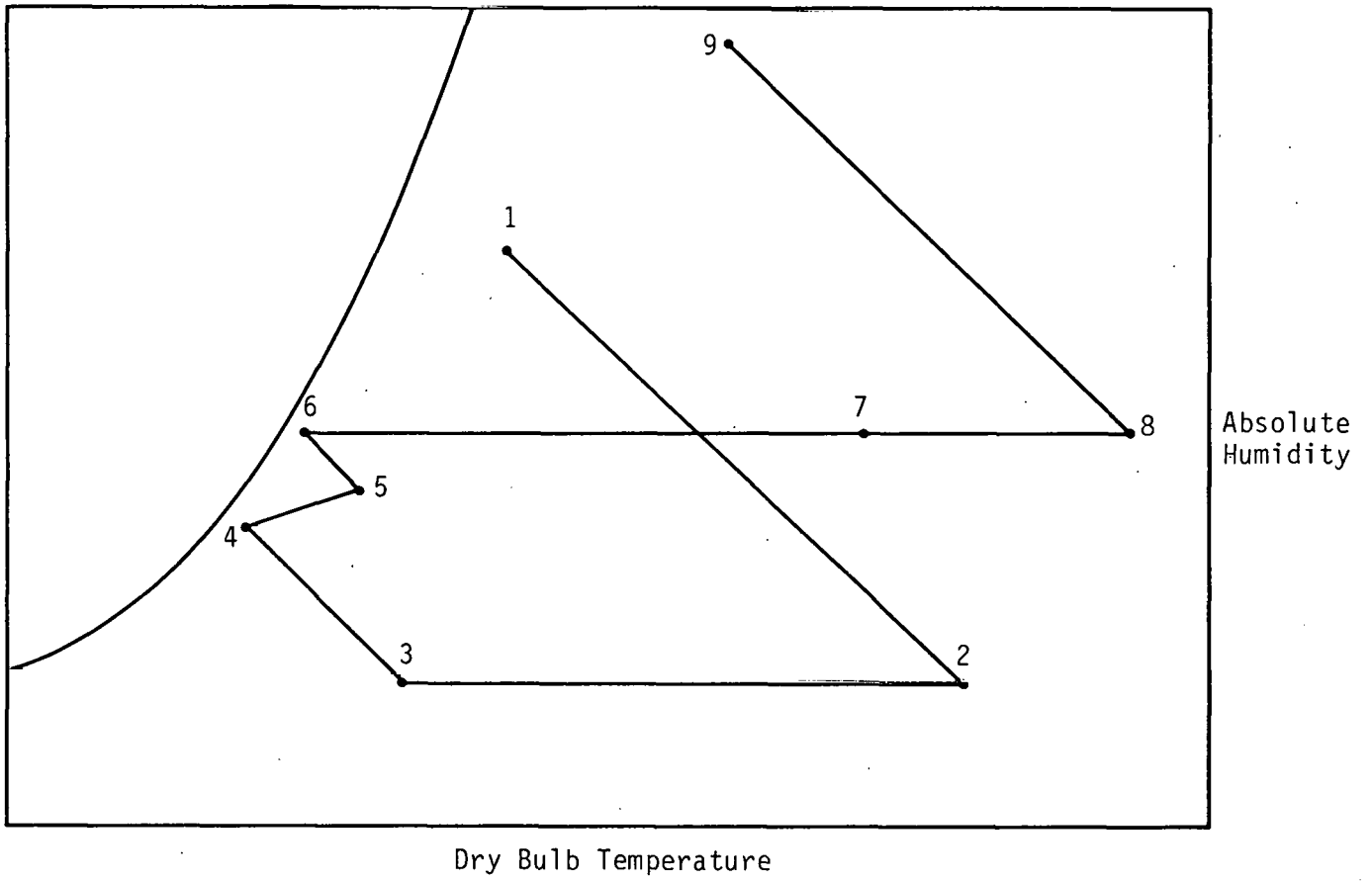
Figure 2-2. VENTILATION MODE PSYCHROMETRIC DIAGRAM

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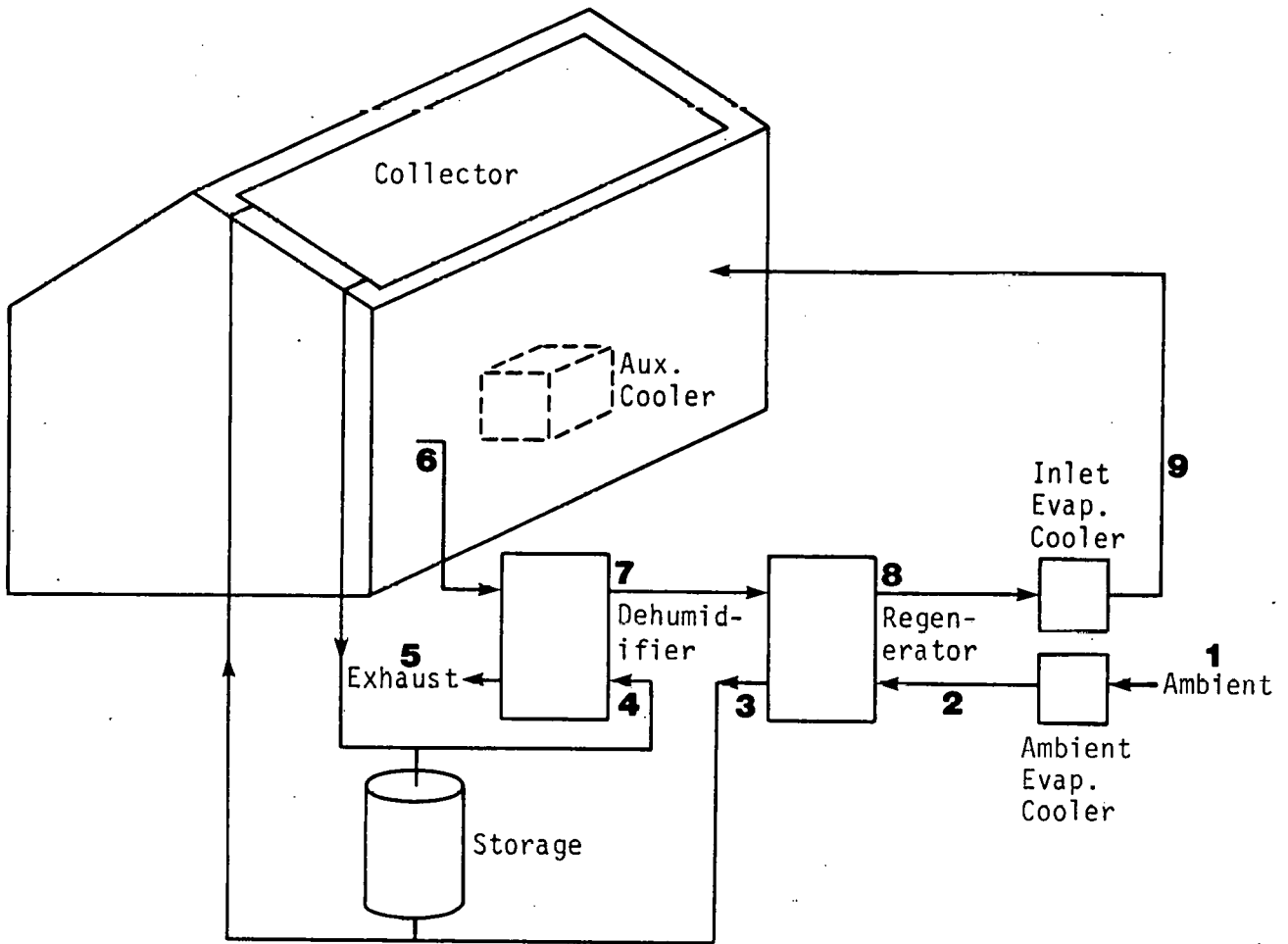
Figure 2-3. RECIRCULATION SYSTEM CONFIGURATION

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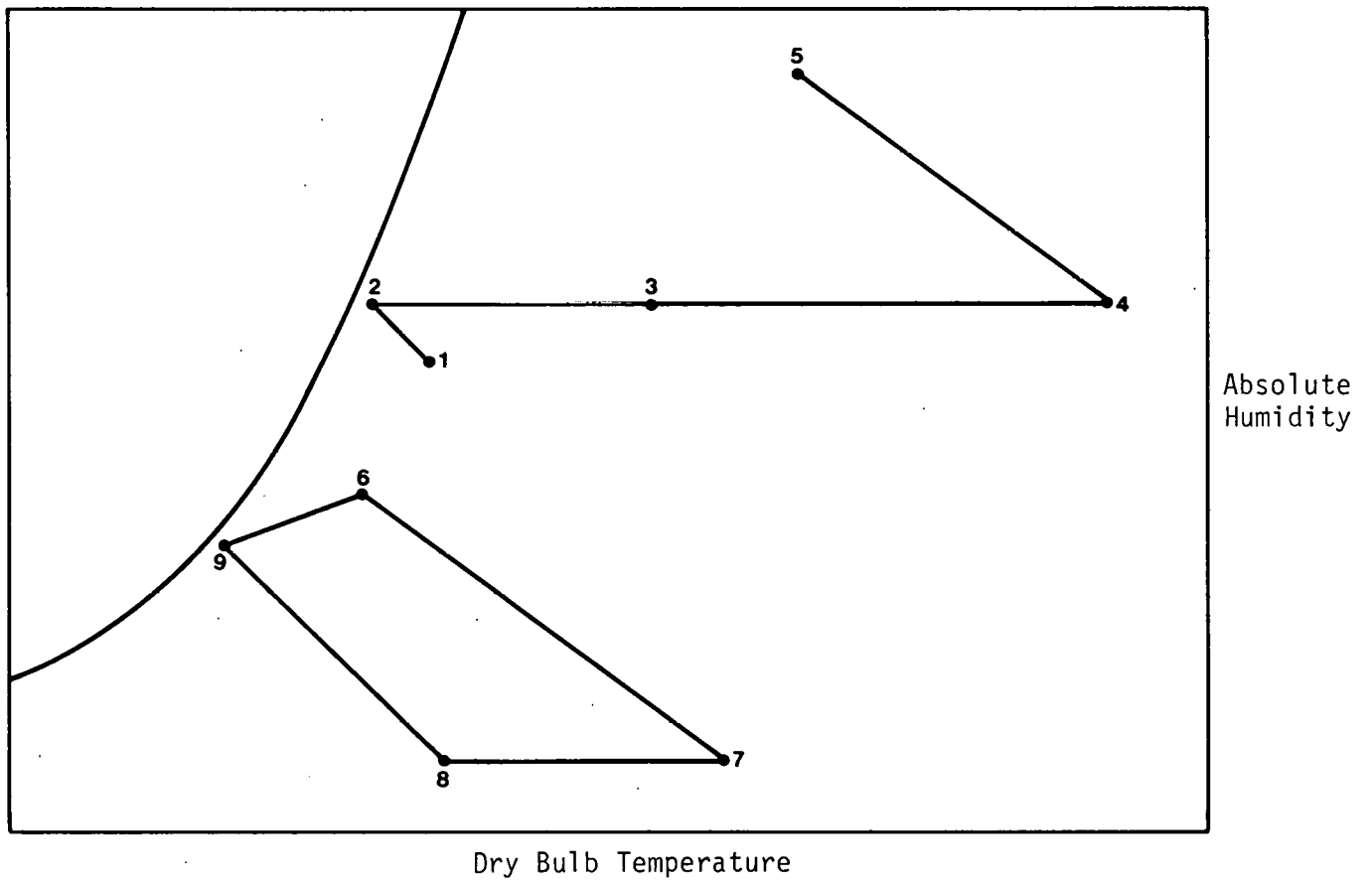
Figure 2-4. RECIRULATION MODE PSYCHROMETRIC DIAGRAM

numbers correspond to the bulk air properties at the system locations shown in Figure 2-3.

2.2 CHOICE OF SYSTEM FOR STUDY

Both the ventilation and recirculation modes were examined in previous studies [4,9]. It was found that the ventilation mode generally required less thermal energy input per unit of cooling output than the recirculation mode. However, it was also found that the recirculation mode was able to use thermal energy at a lower temperature than the ventilation mode. The lower operating temperature resulted in less electrical energy input for auxiliary cooling because solar desiccant cooling could be used more often.

Since a major objective of this study was to determine the potential for fossil fuel conservation, the recirculation mode was chosen for further examination. It is possible that, depending on local climate and energy costs, the ventilation mode could be superior to the recirculation mode in a particular location. For this initial study, however, no comparisons were made between these two modes. Thus, Figures 2-3 and 2-4 describe the basic desiccant cooling system used in this study.

SECTION 3.0

PERFORMANCE MODELING

3.1 CHOICE OF METHOD

Evaluation of desiccant system performance necessitates a study of the transient response of the house air state driven by external weather conditions, solar insolation, and machine operation. A steady-state analysis is inadequate because insufficient information is available that relates dynamic interaction of the desiccant cooling system with varying solar collector/storage energy output and ambient conditions. The effect of varying solar output, which would not be detected in a limited number of steady-state analyses, is discussed in Section 3.7.3.

Although several desiccant units presently exist or are under development [10-12], it was deemed beneficial to provide an indication of system feasibility using computer simulations. Precise modeling required detailed mathematical descriptions of all elements of the system and the capability of accurately predicting the dynamic behavior of component interaction. Yearly simulations were chosen to determine the annual performance of a solar-heating/desiccant-cooling system and compare it with the annual performance of a conventional heating and vapor-compression cooling system.

Computer programs that model desiccant coolers were developed previously. Airesearch, for example, uses a finite difference regression analysis of observed data to predict the system response to various sets of ambient and controlled space conditions. Average system performance can then be determined for the cooling season. This model does not meet the requirement of predicting dynamic behavior. Another model, written by the Institute of Gas Technology, is extremely detailed but runs in real time and is thus not appropriate for seasonal simulations.

The only simulation program available at the time this study began that modeled desiccant systems in the desired manner was software developed by the University of Wisconsin which was compatible with TRNSYS [3]. TRNSYS provided a framework for long term simulations run at short time step increments. An interface with meteorological data was provided, and the investigation of transient response was facilitated because TRNSYS was designed for the transient case. Furthermore, published results of previous desiccant system simulations using the TRNSYS software are available for comparison [4, 9]. Incorporation of other dehumidifier models with TRNSYS are possible and thereby allow direct comparisons of various desiccant dehumidifier units.

3.2 TRNSYS

The remainder of Section 3.0 (other than 3.6.2 and 3.7.3) provides an in-depth discussion of the various components of the computer model used in this study. A detailed understanding of this information is not required to interpret the results of the simulations. Complete treatments of these results are presented in Sections 3.6.2 and 3.7.3.

3.2.1 General Description

The TRNSYS program consists of several subroutines, each describing components of a solar energy system (e.g., collectors, pumps, storage). The program is driven by hourly meteorological data. An executive routine calls the component subroutines as necessary to simulate the transient conditions of the solar energy system. This modular approach reduces the complexity of modeling but increases the cost of computation, especially for long term simulations. Due to these high costs, TRNSYS is not useful as a design tool.

TRNSYS is quite useful, however, as a research tool when studying the transients involved in a particular design, as in the present case for a desiccant cooling system. Because the standard TRNSYS library does not include many of the components in a desiccant cooling system, these subroutines had to be developed. The basic desiccant component subroutines were developed by John S. Nelson [4]. Modifications were made to the routines as required for this study. The control strategy was overhauled and is discussed in detail in Section 3.7.2.

3.2.2 Component Models

The following subroutines were developed or modified to simulate the desiccant system:

- air collector/storage system,
- humidity converter,
- evaporative cooler,
- regenerator,
- dehumidifier,
- house,
- controller, and
- energy balancer.

The components were interconnected as shown in Figure 3-1.

3.2.2.1 Air Collector/Storage System

This model was developed by SERI from the original TRNSYS Type-22 air collector/storage routine. The modeled system consists of:

- flat-plate solar air collector,
- domestic hot water crossflow heat exchanger (not used),
- pebble bed storage unit,
- blower with differential temperature controller, and
- dampers to control flow stream.

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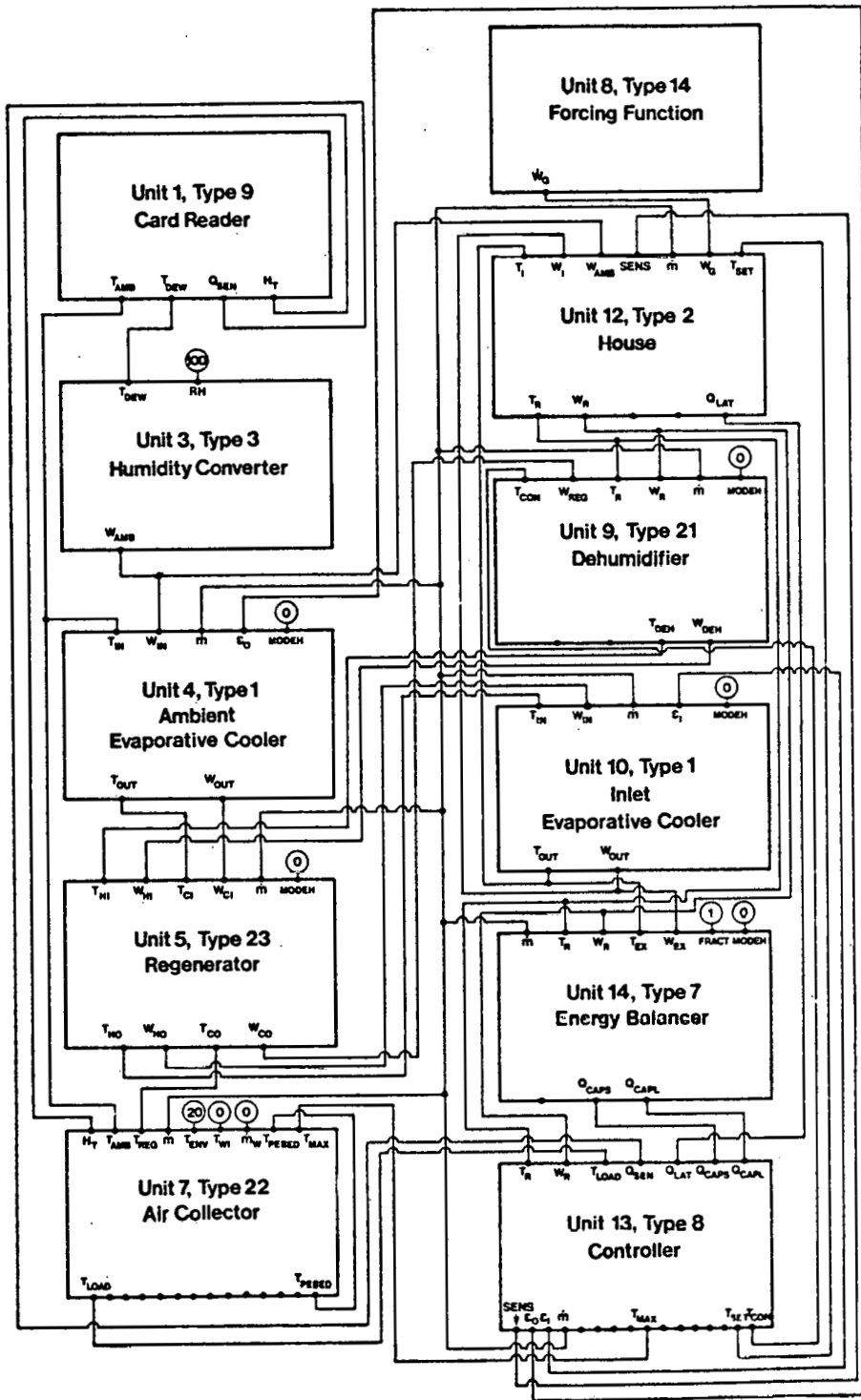
Figure 3-1. DESICCANT COOLER COMPONENT CONFIGURATION

The main difference between the standard TRNSYS routine and the one used in the desiccant simulation was the use of a variable flowrate through the collector to maintain a maximum desiccant regeneration temperature of 85°C whenever possible.

The collector fan was controlled as follows:

- The collector fan was turned off if:
  - -- collector outlet temperature was within 5°C of the collector inlet temperature, or
  - -- collector outlet temperature was less than the average storage temperature and the desiccant cooler was off.

- The collector fan provided a flowrate equal to that of the desiccant system if the collector outlet temperature was 5°C or more above the collector inlet temperature and less than 85°C (upper operation control temperature).

- The collector fan was on at a flowrate greater than the desiccant system (with excess flow to storage) so that the outlet temperature of the collector was 85°C if sufficient solar energy was available.

The energy flowed as follows:

- If there was no load (i.e. the desiccant cooler was not operating), collected energy was delivered to storage at a temperature less than 85°C but greater than the average storage temperature.

- If there was a load:
  - -- the load was met from storage if no solar energy was available;
  - -- the load was met from collected solar energy if possible, with excess solar energy being deposited into storage; or
  - -- the difference was made up from storage if collected energy was insufficient to meet the load.

Table 3-1 lists the values used for the parameters and inputs of the air collector/storage system.

3.2.2.2 Humidity Converter

The humidity converter routine was designed to take dew point temperatures and convert them to humidity ratios. The dew point temperature was inserted into the Keenan-Keyes formula to calculate the partial pressure of water vapor, p_w [6]. The humidity ratio of the ambient air W_{AMB} was calculated from the partial pressure of water vapor and the ratio of molecular weights:

$$W_{AMB} = \frac{\text{molecular weight water}}{\text{molecular weight air}} \times \frac{p_w}{p - p_w} \, ,$$

Table 3-1. PARAMETERS & INPUTS FOR AIR COLLECTOR/STORAGE SYSTEM

PARAMETERS

Collector area (m^2)	10 to 50
Flowrate (kg/s)	Variable
Specific heat of air (J/kg°C)	1005
Efficiency factor	0.78
Transmittance-absorptance product	0.77
Collector loss coeffecient (J/°C m^2s)	4.8
Controller ΔT for blower (°C)	5
Volume of pebble bed[a] (m^3)	3.9 to 19.5[a]
Effective pebble bed density (kg/m^2)	1533
Specific heat of pebbles (J/°C kg)	880
Pebble bed loss coefficient (J/°C m^2 s)	0.4772
Surface area of pebble bed (m^2)	13.7 to 40.1
Air viscosity (kg/m s)	0.002
Air density (kg/m^3)	0.987
Average pebble diameter (m)	0.0191
Void fraction in storage	0.33
Pressure drop through air ducts (N/m^2)	50.0

INPUTS

Solar radiation, H_T
Ambient temperature, T_{AMB}
Inlet Temperature, T_{HO}
Inlet flowrate, $\dot{m}$
Temperature of pebble bed environment = 20°C, T_{ENV}
Temperature of liquid in hot water heat exchanger (not used), T_{WI}
Flowrate of liquid in hot water heat exchanger (not used), $\dot{m}_W$
Average pebble bed temperature, T_{PEBED}
Maximum collector temperature (constant 85°C), T_{MAX}

[a]Varied as a function of collector area

where

p = atmospheric pressure.

The humidity ratio specified the mass of water in the air to be removed by the desiccant system and was defined as the mass of water vapor per mass of dry air.

3.2.2.3 Evaporative Cooler

The simulation of the evaporative cooler was based on equations given by Hollands [13]. The following assumptions were made for the evaporative cooler [4]:

- a Lewis number of unity,

- a rate at which water was supplied to the pad equal to the pad convective mass transfer rate,

- a temperature of the pad supply water equal to the thermodynamic wet bulb temperature of the incoming airstream, and

- a closely adiabatic saturation process.

The model calculated the saturation temperature and enthalpy of the incoming air. The outlet temperature T_{OUT} was calculated as a function of the inlet temperature T_{IN}, the saturation temperature T^*, and the given saturation effectiveness ϵ (i.e., the degree to which the outlet temperature approaches the saturation temperature):

$$T_{OUT} = (T^* - T_{IN}) \times \epsilon + T_{IN}$$

The outlet absolute humidity was calculated by equating the inlet and outlet enthalpies. Table 3-2 describes the inputs to the evaporative cooler.

Table 3-2. INPUTS TO EVAPORATIVE COOLER

Inlet temperature, T_{IN}
Inlet humidity ratio, W_{IN}
Flowrate of system, $\dot{m}$
Effectiveness of cooler, ϵ_I

3.2.2.4 Regenerator

The regenerator model was based on work done by Kays and London [21]. Assumptions were made for the regenerator as follows [4]:

- The effect of energy carry-over from one stream to another could be neglected.

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- The effects of leakage and water vapor carryover could be neglected.

- The radial temperature distribution in the airstream impinging on the face of the regenerator could be neglected.

- The regenerator was balanced and symmetrical (i.e., equal areas exposed to flow in both flow streams).

- The heat exchanger operated at 90% effectiveness.

The outlet temperatures were calculated as a function of effectiveness and inlet temperatures of the two flow streams:

$$T_{HO} = 0.9 \ (T_{HI} - T_{CI}) + T_{HI},$$
$$T_{CO} = 0.9 \ (T_{HI} - T_{CI}) + T_{CI},$$

where

T_{HO} = outlet temperature of hot airstream,

T_{HI} = inlet temperature of hot airstream,

T_{CO} = outlet temperature of cold airstream, and

T_{CI} = inlet temperature of cold airstream.

The specific humidity did not change from inlet to outlet. The inputs to the regenerator are described in Table 3-3.

Table 3-3. INPUTS TO REGENERATOR

Temperature of hot airflow stream, T_{HI}

Humidity ratio of hot airflow stream, W_{HI}

Temperature of cold airflow stream, T_{CI}

Humidity ratio of cold airflow stream, W_{CI}

Flowrate of airstream, $\dot{m}$

Heating/cooling mode indicator, MODEH

3.2.2.5 Dehumidifier

The equations used in the axial-flow disc-type dehumidifer model were developed from the work of I. L. Maclaine-Cross and P. J. Banks [14]. The assumptions used were as follows [4]:

- desiccant (silica gel) packed as a porous medium, the interstices of which were channels through which an air/water vapor mixture flowed;

- constant dry air density;

- constant dry silica gel density;

- constant interstitial fluid velocity;

- airstream flow and silica gel bed properties varying only in the direction of flow;

- moist silica gel and air/vapor mixture in thermal and sorption equilibrium at all locations along the flow path; and

- reversible sorption process (i.e., no hysteresis).

The model essentially used a graphical method of analysis developed by Banks-Close-Maclaine-Cross [15]. Silica gel data for the dehumidifier was developed from the work of P. J. Banks and D. J. Close [16]. The parameters and inputs for the dehumidifier are described in Table 3-4.

3.2.2.6 House

The house model, modified by SERI to include an option for auxiliary cooling, was connected to the desiccant system in order to ascertain the room conditions (the room temperature and humidity) on a dynamic basis.

The humidity in the room was calculated from a steady-state humidity W_{SS} defined as follows:

$$W_{SS} = \frac{\dot{m} \times W_{EX} + INF \times W_{AMB} + W_G}{\dot{m} + INF} ,$$

where

$$\dot{m} = \text{system flowrate,}$$
$$W_{EX} = \text{humidity from inlet evaporative cooler,}$$
$$INF = \text{infiltration rate,}$$
$$W_{AMB} = \text{ambient humidity, and}$$
$$W_G = \text{interior moisture generation.}$$

The room humidity ratio W_R was then determined:

$$W_R = W_{SS} - (W_{SS} - W_{RO}) - [1 - \exp(EXW)]/EXW ,$$

where

$$W_{RO} = \text{room humidity last time step, and}$$
$$EXW = (\dot{m} \times INF)/m_T, \text{ where } m_T = \text{total mass of dry air and water vapor.}$$

The temperature in the room was similarly computed from a steady-state temperature T_{SS} defined as follows:

$$T_{SS} = T_{EX} + SENS/(\dot{m} \times C_{p_a}),$$

Table 3-4. PARAMETERS AND INPUTS FOR DEHUMIDIFIER

PARAMETERS

Internal surface area per volume of packing (m^2/m^3)	1573
Length in flow direction (m)	0.0275
Frontal area exposed to flow (m^2)	0.74
Void fraction of packing	0.4
Rotational speed of matrix (r/s)	0.00153
Density of airstream (kg/m^3)	1.2
Effective conductance between packing and airstream (J/m^2 °C s)	58
Packing characteristic dimension (m)	0.000267
Friction factor	1.2
Fluid viscosity (kg/m s)	1500

INPUTS

Regenerating stream inlet temperature, T_{REG}

Regenerating stream inlet humidity ratio, W_{REG}

Supply stream inlet temperature, T_I

Supply stream inlet humidity ratio, W_I

System flowrate, $\dot{m}$

Heating/cooling mode indicator, MODEH

where

T_{EX} = temperature from inlet evaporative cooler,

SENS = sensible load, and

C_{p_a} = specific heat of dry air.

The room temperature was then determined from the steady-state temperature:

$$T_R = T_{SS} - (T_{SS} - T_{RO}) \times \exp(EXT),$$

where

T_{RO} = room temperature last time step, and

EXT = $\dot{m} \, C_{p_a}$/CAP, where CAP = thermal capacitance of house.

The room conditions, as calculated above, are subject to change if an auxiliary backup vapor-compression system is used. The operation of the auxiliary cooling system is discussed in Section 3.7.

The parameters and inputs for the house model are described in Table 3-5.

3.2.2.7 Controller

The controller was designed by SERI so that, for given room conditions, the desiccant system would operate in a manner which would not only return the room to conditions of specified comfort but also minimize both energy consumption and system operation time. The controller determined the following parameters:

- effectiveness of inlet evaporative cooler,
- effectiveness of outlet evaporative cooler,
- temperature of regeneration stream, and
- flowrate (on/off) of system.

The details of the control strategy are contained in Section 3.7.2. The parameters and inputs for the controller are described in Table 3-6.

3.2.2.8 Energy Balancer

The purpose of the energy balancer was to compute the sensible and latent capacities (Q_{SEN} and Q_{LAT}) of the desiccant cooler. These values were calculated as follows:

$$Q_{SEN} = \dot{m} \times [C_{p_a}(T_R - T_{EX}) + C_{p_w}(W_R \times T_R - W_{EX} \times T_{EX})],$$

$$Q_{LAT} = \dot{m} \, (W_{AMB} - W_{EX}) + H_{fg},$$

Table 3-5. PARAMETERS AND INPUTS FOR HOUSE

PARAMETERS

Mass of dry air in room (kg)	565.2
Infiltration rate (change/h)	0.5
Room capacitance (kJ/°C)	10,000
Mass of moisture used in house capacitance (kg)	5,652
Room temperature, initial (°C)	23.9
Room humidity ratio, initial	0.0085
Room set temperature (°C)	25.5
Room set relative humidity (%)	60

INPUTS

Temperature from inlet evaporative cooler, T_{EX}

Humidity ratio from inlet evaporative cooler, W_{EX}

Ambient humidity ratio, W_{AMB}

Sensible load, SENS

System flowrate, $\dot{m}$

Interior moisture generation, W_G

Room set temperature, T_{SET}

where

T_R = room temperature,

T_{EX} = temperature from inlet evaporative cooler,

C_{P_W} = specific heat of water,

W_{EX} = humidity ratio from inlet evaporative cooler, and

H_{fg} = heat of vaporization of water.

Table 3-7 lists the inputs to the energy balancer.

Table 3-6. PARAMETERS AND INPUTS FOR CONTROLLER

PARAMETERS

Minimum room temperature (°C)	20
Maximum room temperature (°C)	25.5
Maximum relative humidity (%)	60
Flowrate (kg/s at 9.0 kW cooling capacity)	0.45
Maximum regeneration temperature (°C)	85
Minimum regeneration temperature (°C)	45
Temperature deadband (°C)	0.5
Relative humidity deadband (%)	5
Auxiliary mode	1 = no auxiliary
	2 = auxiliary heat to desiccant
	3 = auxiliary cooling capacity
Specific heat of air (J/kg°C)	1005
House Capacitance (kJ/°C)	10,000

INPUTS

Room temperature, T_R

Room humidity ratio, W_R

Collector outlet temperature, T_{COLL}

Sensible load, SENS

Latent load, LAT

Sensible capacity, Q_{SEN}

Latent capacity, Q_{LAT}

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Table 3-7. INPUTS TO ENERGY BALANCER

System flowrate, $\dot{m}$

Room temperature, T_R

Room humidity ratio, W_R

Temperature from inlet evaporative cooler, T_{EX}

Humidity ratio from inlet evaporative cooler, W_{EX}

Fraction of dry air, FRACT

Heating/cooling mode indicator, MODEH

3.2.3 House Load Model

3.2.3.1 Transfer Function Approach

The standard TRNSYS transfer-function load model developed by M. J. Pawelski was used in this study [17]. The transfer function approach uses past heat flows and temperatures to predict future heat flow at any time. The driving force of the model is the sol-air temperature, which includes solar, convection, and radiation effects at the exterior boundary. Transfer function coefficients from the 1977 ASHRAE Handbook of Fundamentals [6] were used in the model to depict the type of wall used. Each wall may be modeled separately or, assuming similar construction, together using a weighted sol-air temperature. The house used in the present simulations was a two-story house with 167.3 m^2 of floor space and 178.4 m^2 of exterior wall space. The ASHRAE 90-75 standard was used in specifying the thermal design. The house was oriented north-south with a roof pitch equal to the latitude of the location. The building is characterized further in the following sections.

3.2.3.2. Walls

The heat transfer rates for the walls were calculated using the standard TRNSYS Type-17 wall model. The rates were calculated as follows:

- Heat conducted through walls, $\dot{Q}_{wall}$:

$$Q_{wall} = A\left(\sum_{n=0} b_n (T_{sa,n} - T_{rc}) - \sum_{n=1} d_n Q_n \right),$$

where

A = area of wall,

b_n = transfer function coefficients of temperature terms,

$T_{sa,n}$ = sol-air temperature of wall surface at time n,

T_{rc} = constant room temperature,

d_n = transfer function coefficients of heat flux terms, and

Q_n = heat fluxes at time n.

- Heat conducted through windows, $Q_{windows}$:

$$Q_{windows} = U_{WD} \times A_w (T_{AMB} - T_R),$$

where

U_{WD} = window heat transfer coefficient,

A_w = window area, and

T_{AMB} = ambient temperature.

- Solar heat gain from windows, Q_{SHG}:

$$Q_{SHG} = A_w (1 - F_s) \tau H_T,$$

where

F_s = shading fraction,

τ = transmissivity, and

H_T = total solar radiation incident on window.

The total rate of heat gain through the walls was determined by summing Q_{wall}, $Q_{windows}$, and Q_{SHG}.

Wall construction was chosen to be frame with 1-in. insulation (ASHRAE wall #38) with surface absorptance of 50% and infrared surface emittance of 80%.

An overall heat loss coefficient of $U_o = 1.60$ J/s m^2°C was used for the walls and windows. The allowable window area expressed as a fraction of wall area is:

$$F = \frac{U_o - U_{WL}}{U_{WD} - U_{WL}},$$

where

U_{WL} = heat transfer coefficient for wall, and

U_{WD} = heat transfer coefficient for window.

Given $U_{WL} = 1.012$ J/s m^2°C and $U_{WD} = 3.52$ J/s m^2°C and assuming double glazed windows, 12% of the total wall area should be windows. The window area was then distributed as shown in Table 3-8.

Table 3-8. WINDOW DISTRIBUTION IN HOUSE MODEL

Wall	Window Area (m_2)	% of Wall Area
North	2.23	5
East	4.01	9
West	4.01	9
South	11.15	25

Window shading factors were also used to express the amount of solar gain lost to shading effects. Shading factors of 20% in the winter months and 60% in the summer months were used.

3.2.3.3 Roof

The heat flux calculations for the roof were similiar to those described for the wall in the previous section. Generally, the total rate of heat conducted through the ceiling, Q_o, is

$$Q_o = A_c \left(\sum_{n=1} b_n (T_{sa,eff,n} - T_{rc}) - \sum_{n=1} d_n Q_n \right),$$

where

$$A_c = \text{area of ceiling, and}$$

$$T_{sa,eff,n} = \text{effective solar air temperature of roof surfaces and infiltration air.}$$

The U value of the roof was chosen as 0.28 J/s m^{2}°C, which represents 6 in. of insulation. The surface emittance and absorptance were the same as for the walls, 80% and 50% respectively.

3.2.3.4 Room and Basement

Heat transfer into the room was determined from the input of the walls and roof as described in the previous sections. In addition, other heat gains (or losses) were taken into account, such as heat generated by people, lights, and other appliances. The model also considered heat flows through the basement and infiltration losses.

The single-node room temperature was allowed to vary between upper and lower set temperatures. When the house was in comfort, that is, within this specified temperature range, the sensible load was zero. If the room temperature was out of this range, the heating load and minimum (for heating) or maximum (for cooling) temperature was output. The total load was calculated using a time-dependent transfer function of each component as a function of both current and previous loads. The components were as follows:

$$Q_{load} = Q_{SHG} + Q_C + Q_L + Q_{INF} + Q_B,$$

where

Q_C = conduction gains from walls and roof,

Q_L = constant heat gains (people, lights, appliances, etc.),

Q_{INF} = infiltration gains, and

Q_B = heat flow through basement.

Using a room capacitance of 10,000 kJ/°C and a building conductance (UA) of 1350 kJ/h°C, the heat inputs from the walls, windows, and roof were applied to the conditions of the room. Thermostat settings were 20.0° to 25.5°C with air infiltration of a half-house volume of air per hour.

Interior sensible heat generation was assumed to be 1260 kJ/h from lights and appliances and 1188 kJ/h from people. Moisture generation (i.e., latent load) was 0.26 kg/h from appliances and 0.24 kg/h from people.

The basement was assumed to be a crawl space 1.53 m deep. Ground-water temperature, used for calculating heat loss to the ground from the basement, was assumed to be a constant 10°C.

3.2.3.5 Determination of Cooling Season

The cooling season was determined from a full-year heating and cooling load calculation with the following assumptions:

Minimum room temperature: 18.8°C.

Maximum room temperature: 25.5°C.

Window shading factor: 0.4.

The cooling season was chosen such/that it would include less than 5% of the heating load. The season was rounded to the nearest half-month.

The preliminary determination of the cooling season was necessary because different assumptions were used in calculating the heating and cooling loads, as shown in Table 3-9.

Table 3-9. SEASONAL LOAD ASSUMPTIONS

	Heating Season	Cooling Season
Minimum room temperature (°C)	20	25.49
Maximum room temperature (°C)	25.5	25.51
Window shading factor	0.2	0.6

3.3 DESICCANT COOLER SYSTEM SIZING

In order to simulate the annual operation of a desiccant cooling system, the proper size relationships among the system components had to be determined. This included both the desiccant cooler components and the collector/storage system. The component sizes were dictated by:

- thermal performance requirements;

- economic requirements, including capital and operating costs; and

- physical size restrictions appropriate for residential heating/cooling systems.

The component configurations and required parameters for the computer simulation are discussed in Sections 2.2 and 3.2.2. This section discusses the component sizes used in the computer simulations.

Several different collector and storage size combinations were used with the desiccant systems to determine optimum desiccant thermal performance and economic trends. As previously discussed, the collector model included a pebble-bed storage tank. A general criterion for sizing collector/pebble-bed systems is to include 0.153 to 0.229 m^3 of pebbles per square meter of collector area when considering space heating only [18]. In this study, the proportion of pebble-bed volume to collector area was modified because of the addition of the solar desiccant air conditioner. An increased pebble-bed volume of 0.4 m^3 of pebbles per square meter of collector area was chosen, based on the results of initial desiccant cooler simulations. The resulting collector/storage sizes used in this study are shown in Table 3-10.

Table 3-10. COLLECTOR/STORAGE SIZE COMBINATIONS

Collector Area (m^2)	Pebble Bed Storage Size (m^3)
10.0	3.9
20.0	7.8
35.0	13.1
42.0	16.5
50.0	19.6

As discussed in Section 3.2.2, the collector/storage system included a variable air mass flowrate capability in order to control the maximum collector output temperature. The pressure drops through these components were calculated during each time increment to determine the parasitic power for the collector system fan motor. Based on flat-plate collector test data [23], correlations were developed for collector pressure drop as a function of air flowrate, including an allowance for entrance and exit effects plus duct losses, and these correlations were included in the computer program. To determine the pebble-bed storage system pressure drop, the Ergun Equation, which describes the pressure drop across packed columns, was used in the computer simulation [24].

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Since the desiccant cooler was to be designed for residential use, the nominal cooling capacities chosen for study were in the range of 2.3 to 9.0 kW, with a few simulations run for a 13.5-kW cooler. The chosen reference design was 9.0-kW cooling capacity with an air mass flowrate of 0.45 kg/s (corresponding to an average flow of 0.38 m^3/s). The cooler size and air flowrate combinations used in this study are shown in Table 3-11.

Table 3-11.   COOLER-SIZE/AIR-FLOWRATE COMBINATIONS

Desiccant Cooler Size (kW)	Air Flowrate (kg/s)
2.3	0.113
4.5	0.225
9.0 (reference)	0.450
13.5	0.675

The desiccant system components were sized in accordance with thermal, economic, and physical size requirements. Once the cooler size and corresponding air flowrates of Table 3-11 were established, the flow areas and flow lengths of the dehumidifier, regenerator, and evaporative coolers could be determined by specifying allowable pressure drops. The pressure drops should be as low as possible to minimize the electrical power required to drive the air fans.

An allowable pressure drop of 125.0 N/m^2 across the wheel for each flow (supply and regeneration) was selected for the reference design of the dehumidifier. A dehumidifier wheel diameter of 1.37 m was also specified. For the packed-bed silica-gel dehumidifier model used in this study, the resulting flow length (packed-bed depth) was 0.0275 m with a wheel rotation speed of 0.00153 r/s [19].

The regenerator and evaporative coolers were sized to be physically compatible with the dehumidifier. In this manner, the airflow path through the cooler has as few contractions and expansions as possible in order to minimize the system pressure drop. Reference 20 describes a solar desiccant air conditioner with the same basic airflow path, although the parasitic power experienced in that study was higher than predicted here due to design and operational problems.

The rotary regenerator was assumed to be a parallel passage type with a wheel diameter equal to that of the dehumidifier (1.37 m). Studies at the University of Wisconsin showed that the desiccant cooler performance was optimized as regenerator effectiveness increased [4]. For this study, a constant regenerator effectiveness of 0.90 was assumed. For a regenerator airflow path length of 0.05 m, a pressure drop of 25.0 N/m^2 was calculated for each flow direction [21].

The evaporative cooler flow area was assumed equal to 0.74 m^2, or half the frontal flow area of the dehumidifier and regenerator wheels. Identical

evaporative coolers were present in both the supply and regeneration streams. The effectiveness for each cooler was assumed to be capable of varying between 0.0 and 0.9. The effectiveness of the inlet evaporative cooler and that of the ambient evaporative cooler were important parameters in controlling the desiccant cooler performance and will be discussed in Section 3.7.2. For each evaporative cooler, the pressure drop was determined to be 50.0 N/m^2 for the reference design [21].

In addition to pressure drops through the individual system components, pressure drops were associated with other parts of the desiccant cooler system, including ducts and air filters. For each airflow stream, an additional pressure drop of 50 N/m^2 was included to account for duct losses (25 N/m^2) and an air filter (25 N/m^2). The pressure drops for the reference design system are summarized in Figure 3-2.

To determine electrical power requirements (parasitic power), a fan power equation was used in conjunction with the system pressure drops. Fan power was determined from the following equation [22]:

$$\text{Fan power in } J/s = \frac{\text{CFM}(\Delta P)}{72.2E} \text{ ,}$$

where

    CFM = air flowrate (m^3/s),

     P = system pressure drop (N/m^2), and

     E = fan efficiency.

Fan efficiency typically varies from 0.5 to 0.65; an average value of 0.58 was used. The computer simulation determined parasitic energy used by the cooler/collector system at each time increment.

Three fans were included in the system, one for each airflow stream (collector/storage, supply, and regeneration). For the reference design, the fan motors were sized as follows:

    Supply stream: 246.1 J/s.

    Regeneration stream: 372.9 J/s.

    Collector stream: variable speed (due to variable airflow as described in Section 3.2.2) with a minimum rating of 186.4 J/s.

An additional motor was included to drive the dehumidifier and regenerator wheels. This motor was sized at 186.4 J/s.

Since the desiccant cooler simulation was run at several cooler capacities (as shown in Table 3-11), the parasitic power requirement varied for each size. For each cooler capacity, the desiccant system components were sized so that the pressure drop through each remained the same as that for the reference design. The desiccant wheel flow length was maintained at 0.0275 m for all sizes, with the wheel diameter varying as shown in Table 3-12.

Collector

2●

Aux.
Cooler

4●

Inlet
Evap.
Cooler

1●

●3   ●8   ●5

Dehumid-
ifier

Regen-
erator

6
●7 ●Ambient

Exhaust   ●3   ●8

Ambient
Evap.
Cooler

●9 Storage

1. 50 N/m^2(0.2 in. water)
    Duct and Filter

2. 100 N/m^2(0.4 in. water)

3. 125 N/m^2(0.5 in. water)

4. 50 N/m^2(0.2 in. water)
    Duct and Filter

5. 50 N/m^2(0.2 in. water)

6. 50 N/m^2(0.2 in. water)
    Duct and Filter

7. 50 N/m^2(0.2 in. water)

8. 25 N/m^2(0.1 in. water)

9. 50 N/m^2(0.2 in. water)

Figure 3-2.   PRESSURE DROPS FOR REFERENCE SYSTEM

Table 3-12. DESICCANT-COOLER-SIZE/WHEEL-DIAMETER COMBINATIONS

Desiccant Cooler Size (kW)	Desiccant Wheel Diameter (m)
2.3	0.68
4.5	0.97
9.0	1.37
13.5	1.68

As a result, the parasitic power for each desiccant system capacity varied with the ratio of the system flowrate to the reference design flowrate. The computer program scaled the parasitic power for each simulation accordingly.

To be certain that the fan motor sizes were never allowed to drop below realistic values, mimimum motor sizes were included in the computer program. The minimum motor sizes were as follows:

Supply stream: 124.5 J/s.

Regeneration stream: 186.4 J/s.

Collector stream: 186.4 J/s.

If the calculated parasitic power requirement exceeded these values, the calculated value was used. However, if the calculated parasitic power was below any of the minimum motor ratings listed above, the minimum motor rating was used to determine parasitic power used by the desiccant cooling system.

3.4 DEFINITION OF HOUSE COMFORT CONDITIONS

To properly design heating, ventilating, and air-conditioning systems, an understanding of physiological principles and criteria for comfort is essential. The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) has sponsored research for many years on the prediction of thermal comfort. Primary parameters determining thermal comfort include dry-bulb temperature, relative humidity, air movement, and the degree of activity and type of clothing worn by individuals. ASHRAE has developed standards for comfort that serve as guidelines for the design of heating and air-conditioning systems.

Figure 3-3 shows the comfort zone recommended by ASHRAE for a wide range of applications, including homes, schools, and offices. [6]. The comfort zone is applicable to indoor environments ranging in altitude from sea level to 2134 m at air velocities less than 0.2286 m/s, and for lightly clothed occupants with sedentary activity. The upper boundaries of the zone are 25.0°C dry-bulb temperature and 60% relative humidity, with the lower boundaries specified as 22.8°C dry-bulb temperature and 20% relative humidity.
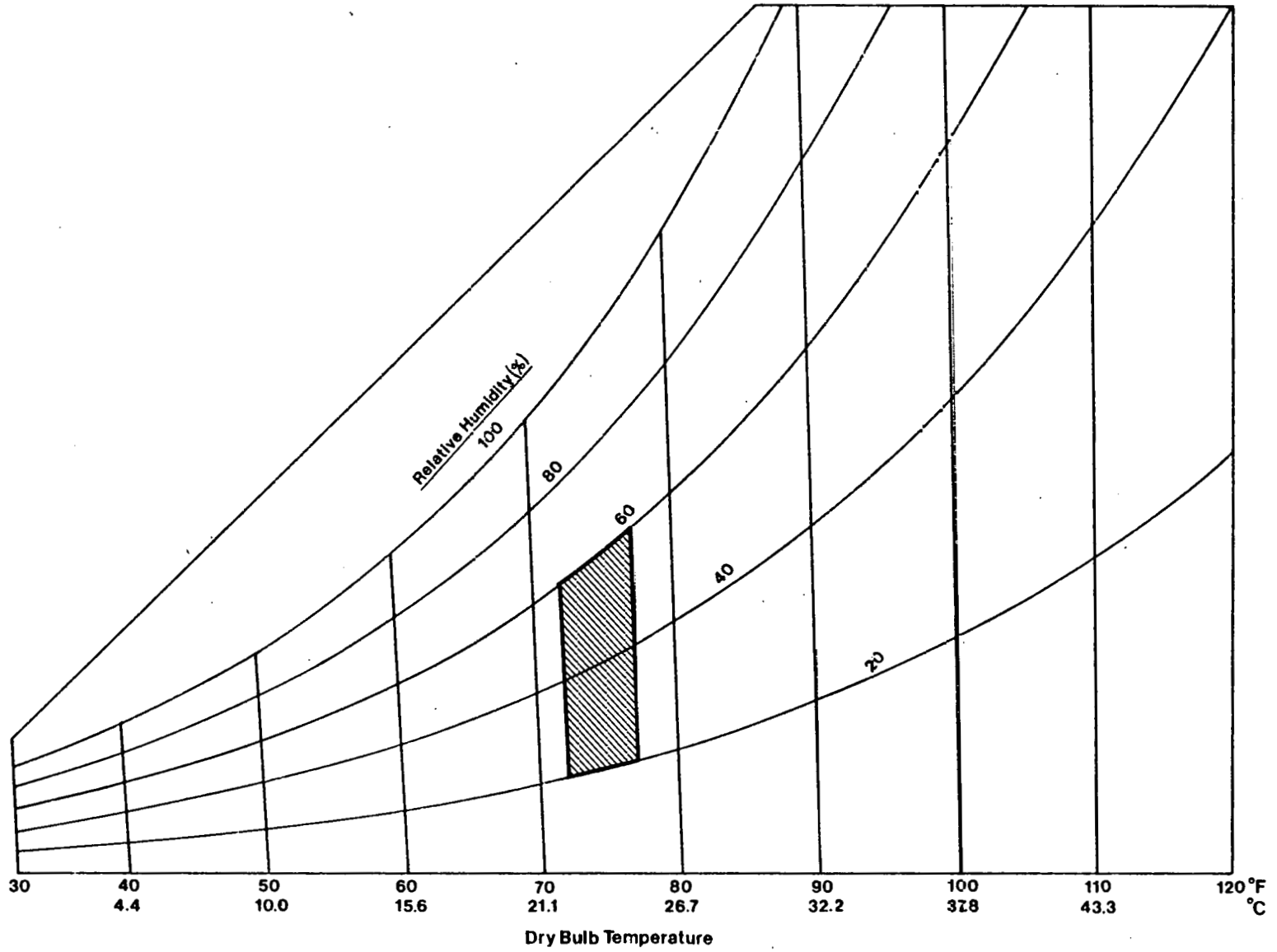
Figure 3-3. ASHRAE COMFORT ZONE (Shaded Area) [6]

The ASHRAE-recommended comfort zone was modified for the purpose of this study. With the currently increasing need for energy conservation and suggestions by the Federal Government for homeowners to increase thermostat settings in summer and decrease settings in winter, it was decided to increase the ranges of dry-bulb temperature and relative humidity that define the comfort zone. Figure 3-4 depicts the comfort zone used in the study. The upper comfort zone boundary was modified slightly to 25.5°C dry-bulb temperature and maintained at 60% relative humidity. A larger adjustment was made in the lower boundary, with the dry-bulb temperature specified as 20.0°C and no lower limit placed on relative humidity. These revisions to the control zone reduce the amount of energy consumed for maintaining comfort conditions in buildings, particularly during the heating season. Considering the large escalation in fuel costs during recent years and the potential for future fuel shortages, a revised comfort zone intended to conserve energy is appropriate. This comfort zone was used in simulations of both solar desiccant and vapor-compression air conditioners, as described in Sections 3.7.2 and 4.1.

3.5 COOLING SYSTEM OPERATIONAL CHARACTERISTICS

In describing the annual heating/cooling season simulations of solar desiccant and vapor-compression air conditioners, the remainder of this report employs several parameters commonly used to describe the operating performance of air-conditioning equipment. This section of the report briefly summarizes and defines these performance parameters.

The total cooling output of an air conditioner is composed of sensible and latent components which are defined as follows:

- Sensible cooling: cooling capacity associated with heat removal from an airstream, resulting in a drop in air temperature.

- Latent cooling: cooling capacity associated with moisture removal from an airstream due to a change in the water phase from gas to liquid, resulting in a drop in humidity.

Five other parameters used in this study were the following:

- Parasitic power: electrical input power to the air-conditioning equipment. For a vapor-compression cooler, electrical power drives the compressor and air fan, while a solar desiccant cooler uses electrical power for air fans only (in addition to auxiliary cooling, if any).

- Coefficient of performance (COP):   defined as the ratio of net heat removal (cooling capacity) to the rate of total energy input. For the solar desiccant cooler, only thermal input power (solar energy) was included in the COP calculation.
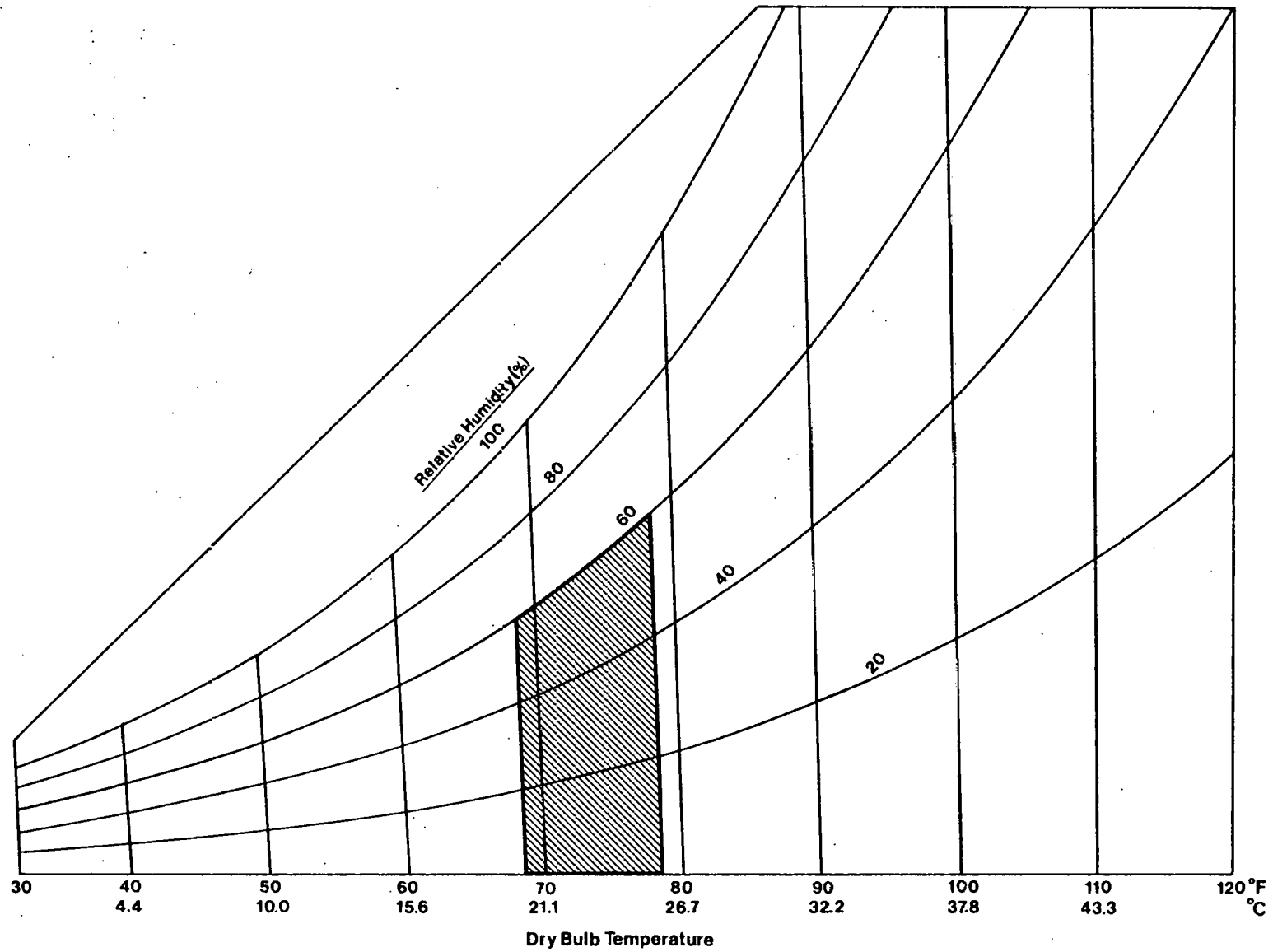
Figure 3-4.   COMFORT ZONE USED IN COMPUTER SIMULATIONS (Shaded Area)

- Energy efficiency ratio (EER): defined as the ratio of net cooling capacity in Btu/hr to the total rate of electrical input in watts. For the solar desiccant cooler, electrical power to the auxiliary vapor-compression cooler (if any) was included in the EER calculation.

- Duty cycle: fraction of the total time period that the cooler system was in operation.

- Percent in comfort: percentage of the total time period that the average house air conditions fell within the comfort zone (as defined on the psychrometric chart of Figure 3-4).

3.6 STEADY-STATE PERFORMANCE

3.6.1 Description of Analysis

The purpose of the steady-state analysis was to obtain a general indication of desiccant system performance. Nominal machine characteristics can be ascertained by examining the sensitivity response of the system to various ambient and indoor environments. Alteration of critical system parameters under steady-state conditions enables the formulation of an operational control strategy for the transient case, as described in Section 3.7.2.

The interaction of system components as shown in Figure 3-5 was studied. Air was circulated from the room into the dehumidifier where drying occurred by absorption of moisture by the desiccant. Heat exchange in the regenerator produced pre-cooling, and final cooling was provided by an inlet evaporative cooler regulated with an effectiveness ε_i. Ambient air cooled by another evaporative cooler (of effectiveness ε_o) was delivered to the regenerator as a heat sink exchange supply. The regeneration temperature T'_{REG}, was varied in 10°C increments to provide recharging of the moist desiccant gel. The energy needed to boost T'_{REG}, the exit temperature from the regenerator, to T'_{REG} (Q_{IN} in Figure 3-5) was assumed available; this is the energy supplied by solar storage or an auxiliary source during transient simulations.

3.6.2 Results

For a particular assignment of values to component control parameters (T_{REG}, ε_i, ε_o), the response of the system, quantified by the inlet temperature and humidity (T_I and W_I), can be investigated as a function of the driving conditions, namely the ambient and room temperatures and humidities (T_{AMB}, W_{AMB}, T_R, and W_R). Sensible and latent cooling capacities, Q_{SEN} and Q_{LAT} in Figure 3-5, then depend upon the temperature and humidity differentials that can be generated by the system between the room state and the inlet conditions. Steady-state performance was surveyed using TRNSYS by choosing meaningful sets of T_R, W_R, T_{AMB}, and W_{AMB} and repetitively inputting values for the control parameters until the latent and sensible capacities converged. The primary driving conditions of interest correspond to room
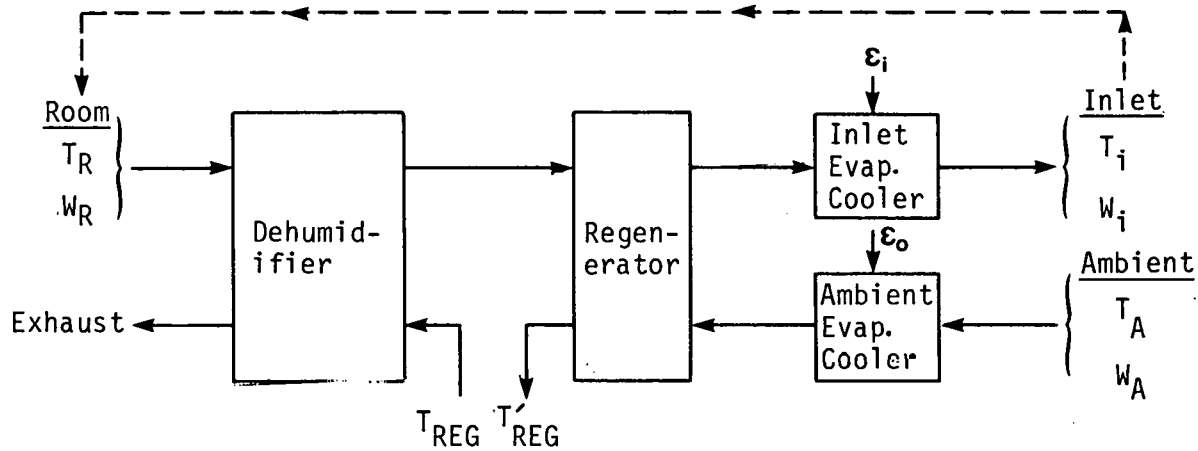
$$\Delta T = T_{REG} - T'_{REG}$$

$$Q_{IN} = \dot{m}\, C_{p,air}\, \Delta T$$

$$Q_{SEN} = \dot{m}\, [C_{p,air}\, (T_R - T_i) + C_{p,moist}(T_R W_R - T_i W_i)] = \text{sensible capacity}$$

$$Q_{LAT} = \dot{m}\, H_{L,H_2O}(W_R - W_i) = \text{latent capacity}$$

$$CAP = Q_{SEN} + Q_{LAT}$$

$$COP = CAP/Q_{IN}$$

where

$C_{p,air}$ = specific heat of dry air = 1005.0 $J/kg_{dry\ air}$

$C_{p,moist}$ = specific heat of moist air = 1884.3 J/kg_{H_2O}

H_{L,H_2O} = latent heat of vaporization of water = 2.466 X 10^6 J/kg_{H_2O}

T = temperature, °C

W = humidity ratio, $kg_{H_2O}/kg_{dry\ air}$

Figure 3-5. CONFIGURATION OF DESICCANT SYSTEM COMPONENTS USED TO GENERATE STEADY-STATE PERFORMANCE MAPS

S≡≥I☀

states located along the comfort zone borders (Figure 3-21), with the ambient temperature and humidity exceeding those of the room. Results for these cases (Figures 3-6 to 3-9) specify the system capabilities in attempting to restore comfort to each of the particular zones of discomfort.

Figures 3-10 to 3-13 illustrate the system reaction to other climatic circumstances. Machine response to a room state at the maximum comfortable temperature with elevated humidity is given by Figure 3-10. System performance is little affected by raising the outdoor temperature from 29.2°C (Figure 3-7) to 35°C (Figure 3-11). However, an increase in ambient humidity drastically degrades operation characteristics as seen by comparison of Figures 3-11 and 3-12. Conversely, Figures 3-6 and 3-13 exhibit the enhancement in performance resulting from milder ambient conditions.

To investigate further the sensitivity of performance, a design point (Table 3-13) was defined and the driving conditions and regeneration temperatures were varied. Total and latent system capacity and COP were then plotted (Figures 3-14 through 3-18). Figures 3-14 and 3-16 show relatively flat profiles of performance as functions of outdoor and indoor temperatures, respectively. On the other hand, substantial improvement in latent cooling capacity is attained at higher room humidities (Figure 3-17), and an inverse relationship is evidenced as the ambient humidity rises (Figure 3-15). In each case, system sensible cooling remains essentially constant. Thus, the desiccant machine operates optimally as a latent cooler in climates which are not excessively humid. System performance is seen to peak (Figure 3-18) near 85°C, in accordance with the choice of T_{REG} = 85°C for design-point operation. At lower regeneration temperatures, system capacity is small relative to the energy supplied (Q_{IN}). At higher temperatures, the large amount of energy supplied causes Q_{IN} to dominate the behavior of the COP.

A summary of the values used to generate all steady-state performance maps is presented in Table 3-13.

3.7 DYNAMIC PERFORMANCE

3.7.1 General Description

Long-term simulations of the transient behavior of the desiccant cooling system were performed by modifying the steady-state component configuration. The primary changes were the incorporations of a solar energy collector/storage unit, an interactive house model, and a controller capable of dynamically varying critical system parameters. An interface with hourly meteorological data and sensible load information was also added. The revised system arrangement is presented in Figure 3-19.

Regeneration demanded by the desiccant dehumidifier was provided by a solar collector air system in combination with an external thermal storage tank. A five-layer, thermally stratified pebble bed storage unit was modeled. All simulations assumed isothermal equilibrium at 40°C as the initial condition. A cylindrical tank was chosen with a height equal to twice its radius and with a volume V_{STOR} of 0.40 m^3 per m^2 of collector area.

Table 3-13. PERFORMANCE MAP PARAMETER SUMMARY

Figure	T_{AMB} (°C)	W_{AMB}	T_R (°C)	W_R	T_{REG} (°C)	ε_i	ε_o
3-6	29.2	0.0175	22.75	0.0104	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-7	29.2	0.0175	25.50	0.0124	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-8	29.2	0.0175	25.50	0.0100	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-9	29.2	0.0175	20.00	0.0080	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-10	29.2	0.0175	25.50	0.0140	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-11	35.0	0.0175	25.50	0.0124	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-12	35.0	0.0300	22.75	0.0124	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-13	22.0	0.0110	22.00	0.0100	45.0-85.0	0.0, 0.05, 1.0	0.0, 0.5, 1.0
3-14	28.0-40.0	0.0155	25.50	0.0124	85.0	1.0	1.0
3-15	35.0	0.005-0.019	25.50	0.0124	85.0	1.0	1.0
3-16	35.0	0.0155	20.0-27.0	0.0124	85.0	1.0	1.0
3-17	35.0	0.0155	25.50	0.005-0.017	85.0	1.0	1.0
3-18	35.0	0.0155	25.50	0.0124	45.0-105.0	1.0	1.0
Design point	35.0	0.0155	25.50	0.0124	85.0	1.0	1.0

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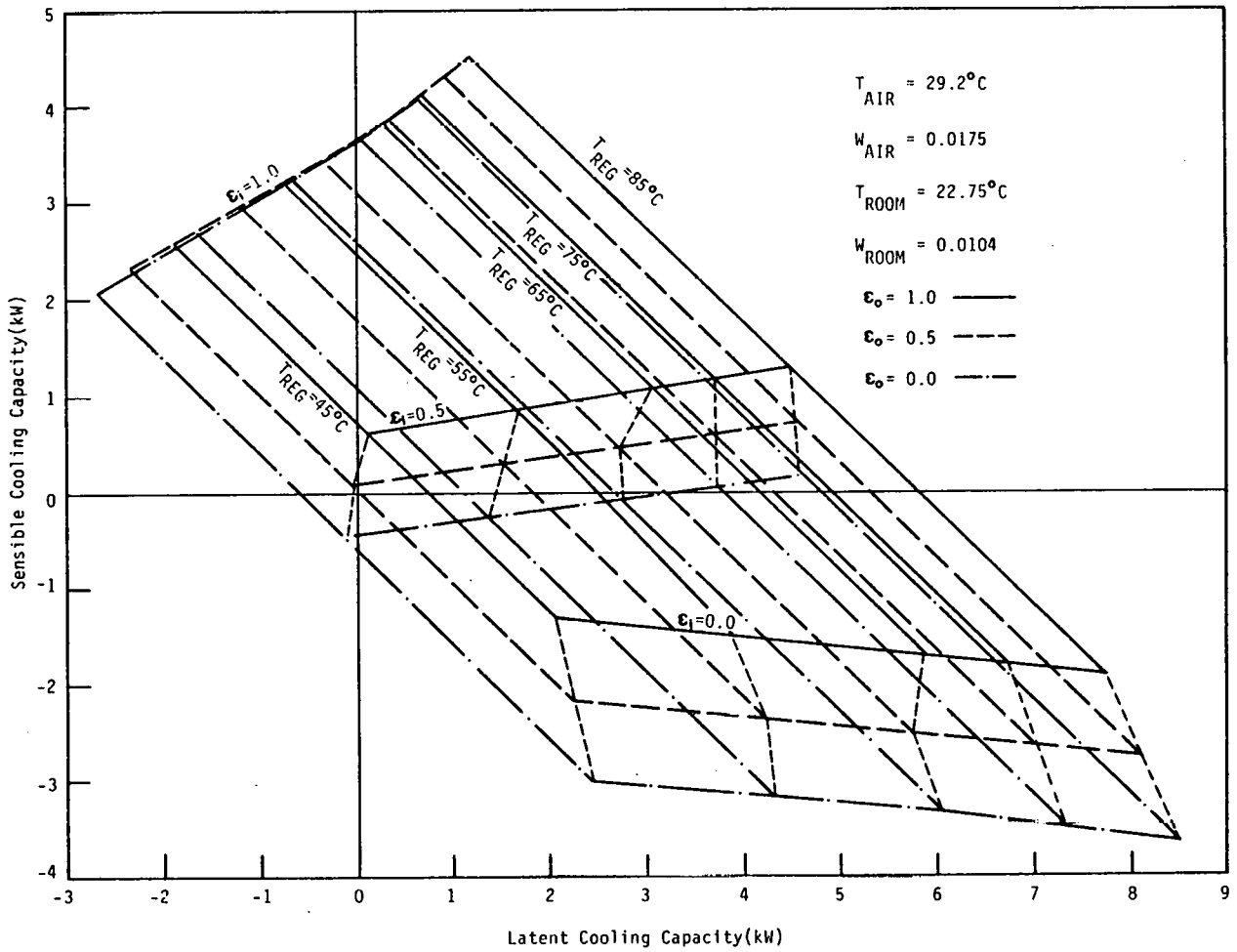
Figure 3-6. STEADY-STATE PERFORMANCE MAP: ZONES 3 - 6
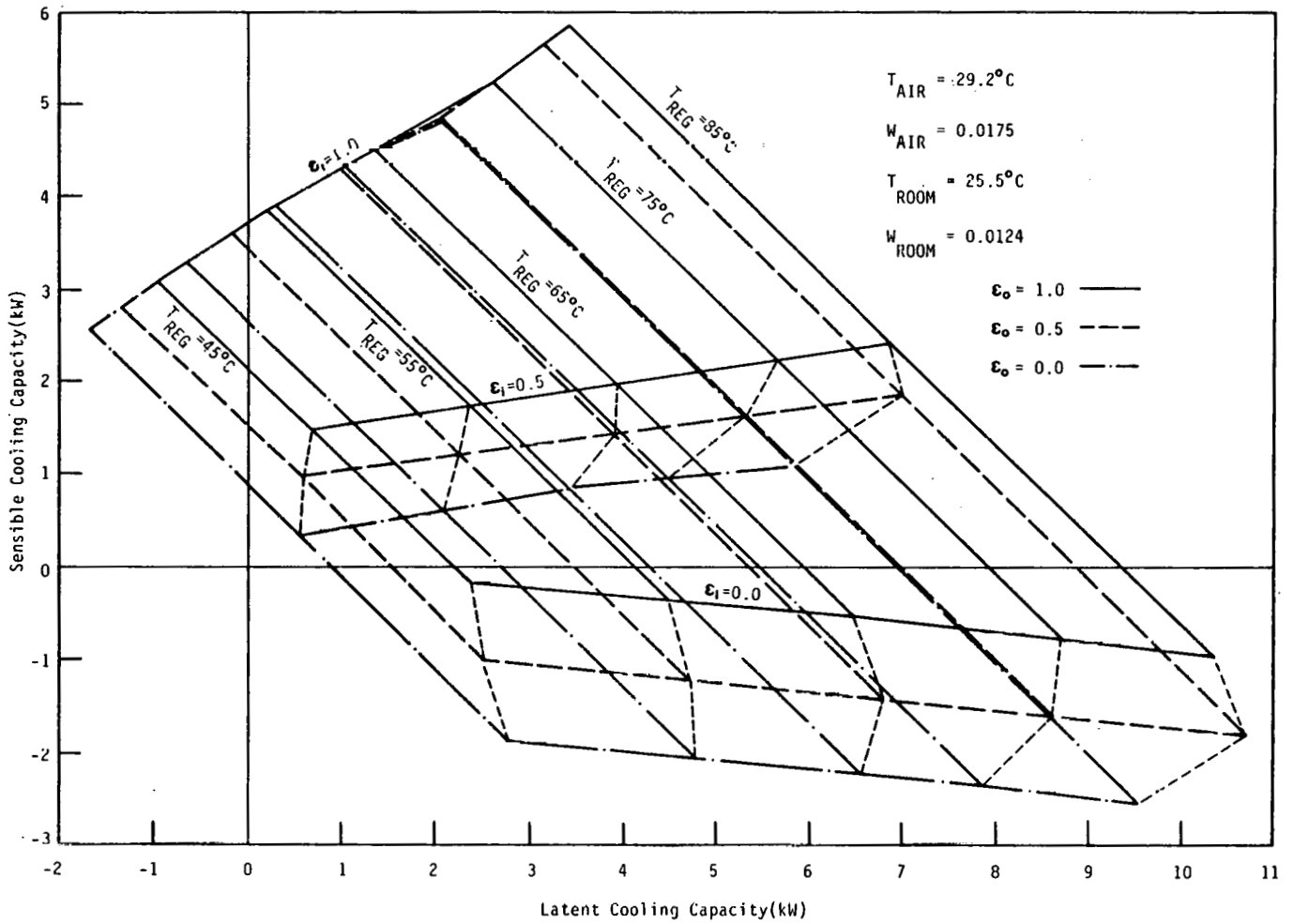
Figure 3-7. STEADY-STATE PERFORMANCE MAP: ZONES 4 - 6
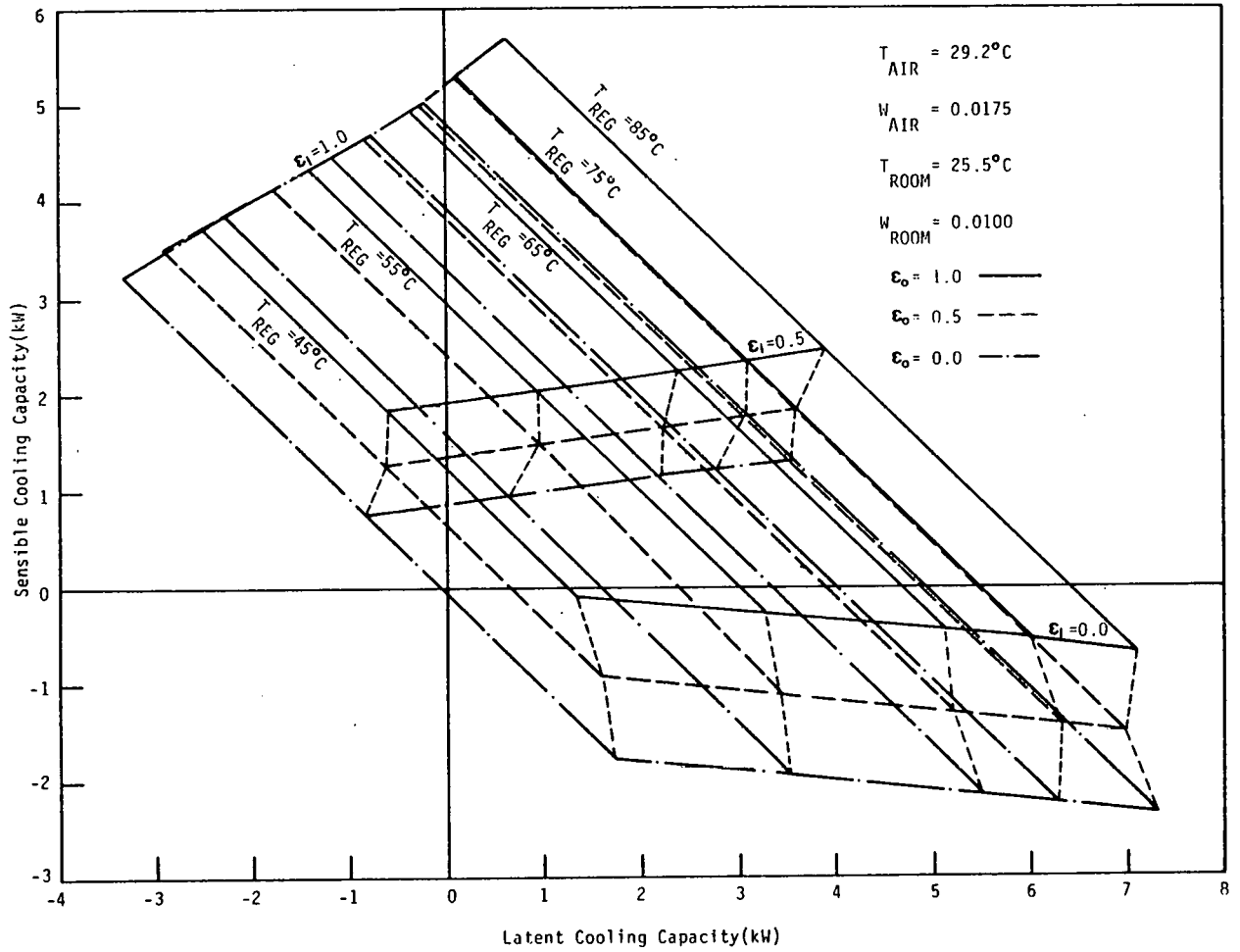
Figure 3-8. STEADY-STATE PERFORMANCE MAP: ZONES 5 - 6
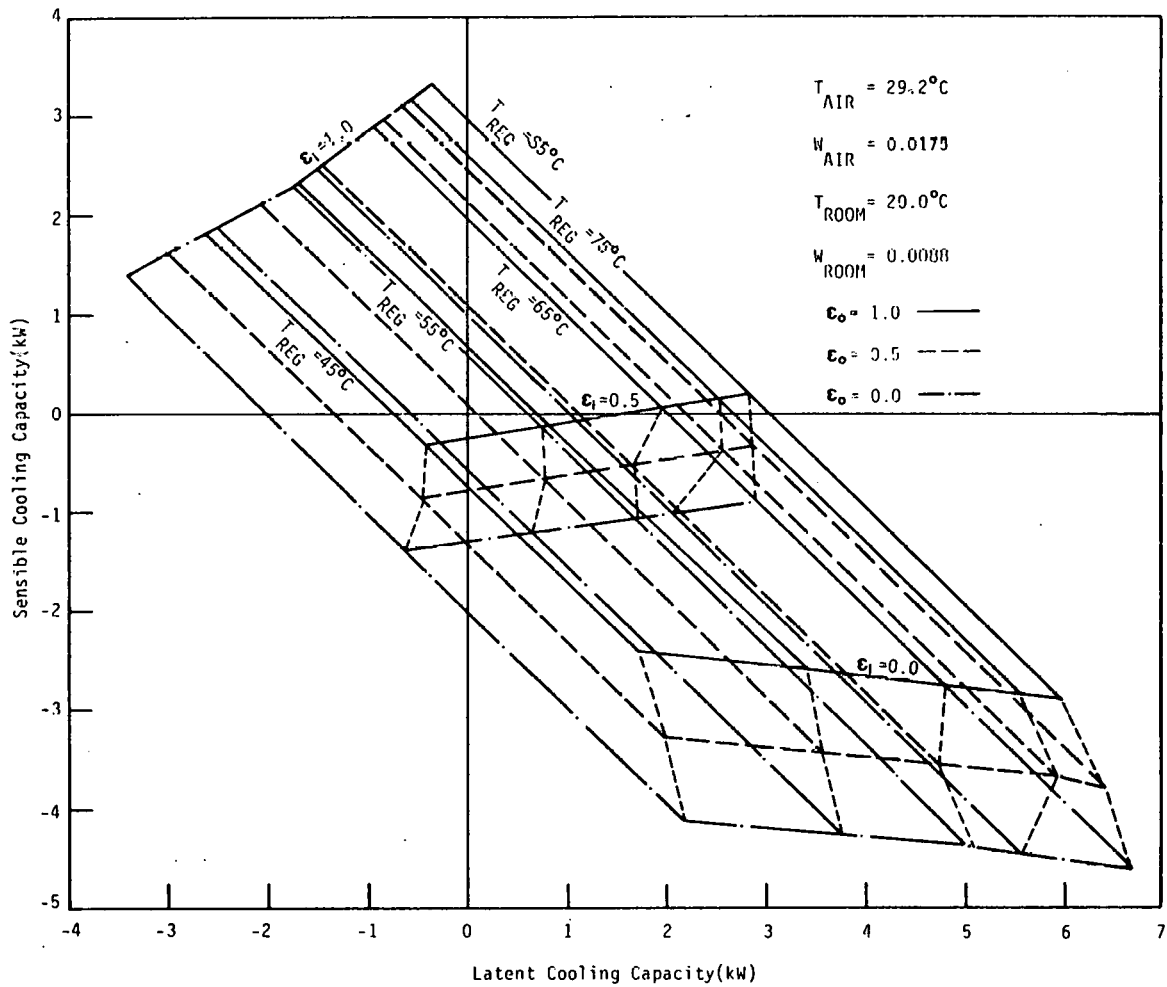
Figure 3-9. STEADY-STATE PERFORMANCE MAP: ZONES 2 - 6

SERI
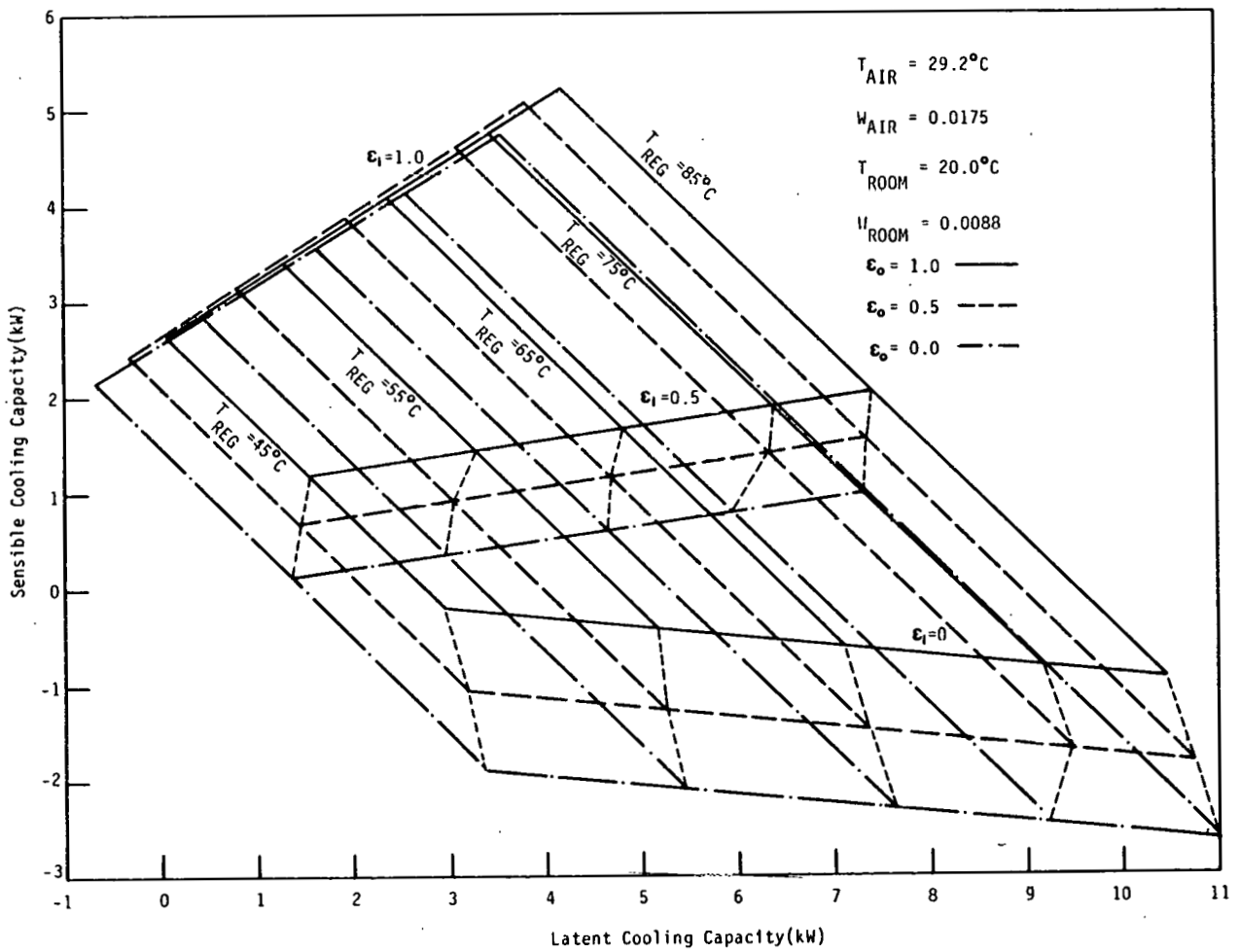
Figure 3-10. STEADY-STATE PERFORMANCE MAP: ZONES 3 - 4
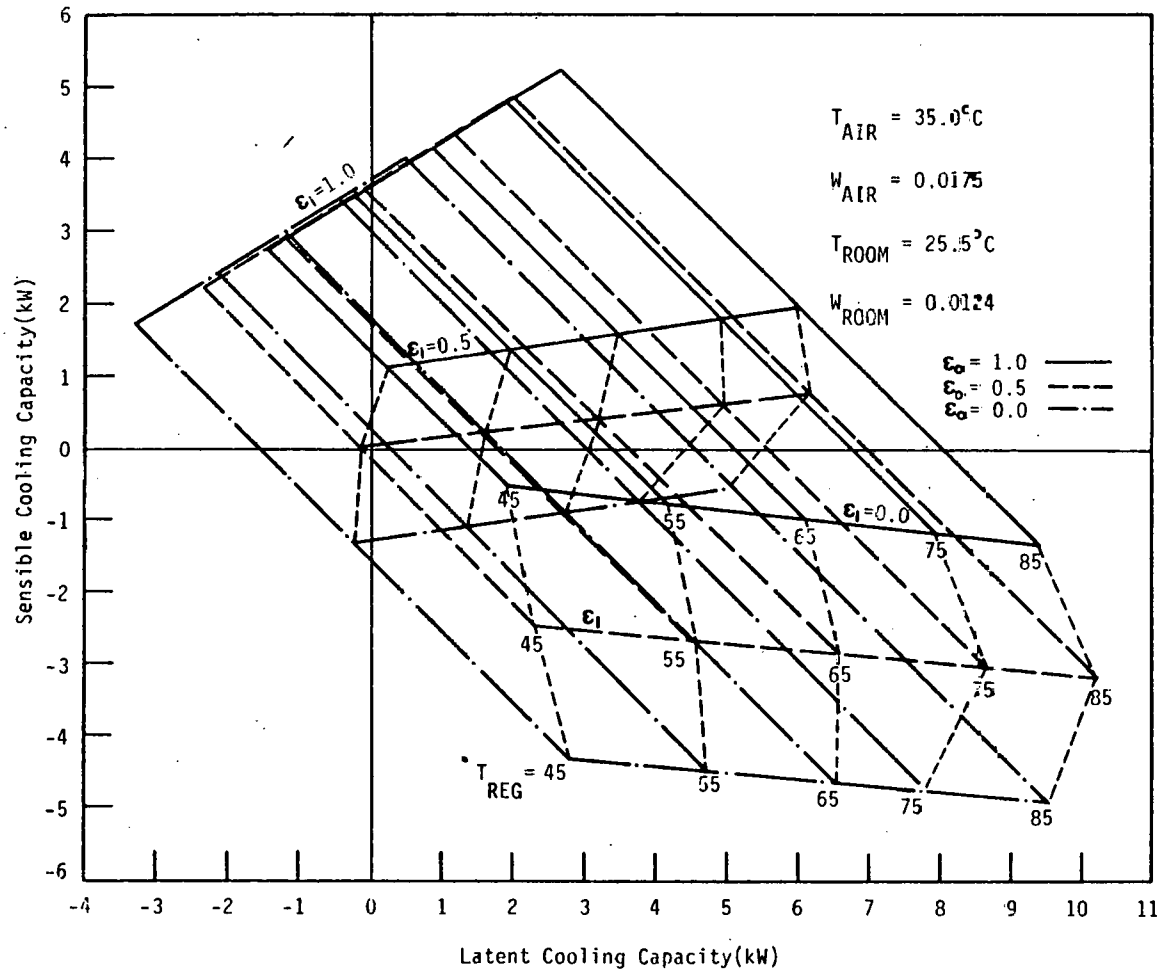
Figure 3-11.   STEADY-STATE PERFORMANCE MAP: SEVERE AMBIENT TEMPERATURE
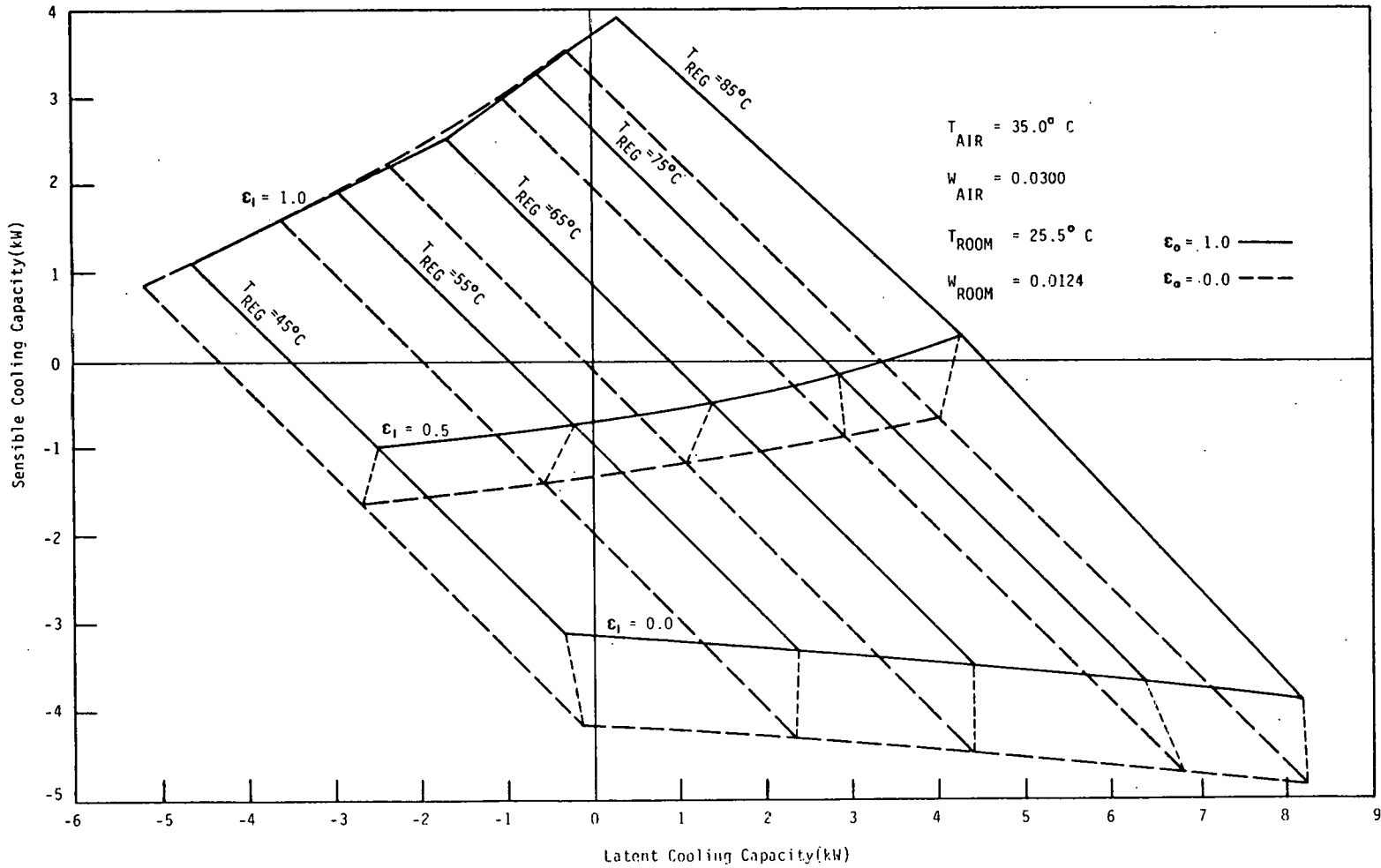
Figure 3-12. STEADY-STATE PERFORMANCE MAP: SEVERE AMBIENT TEMPERATURE AND HUMIDITY
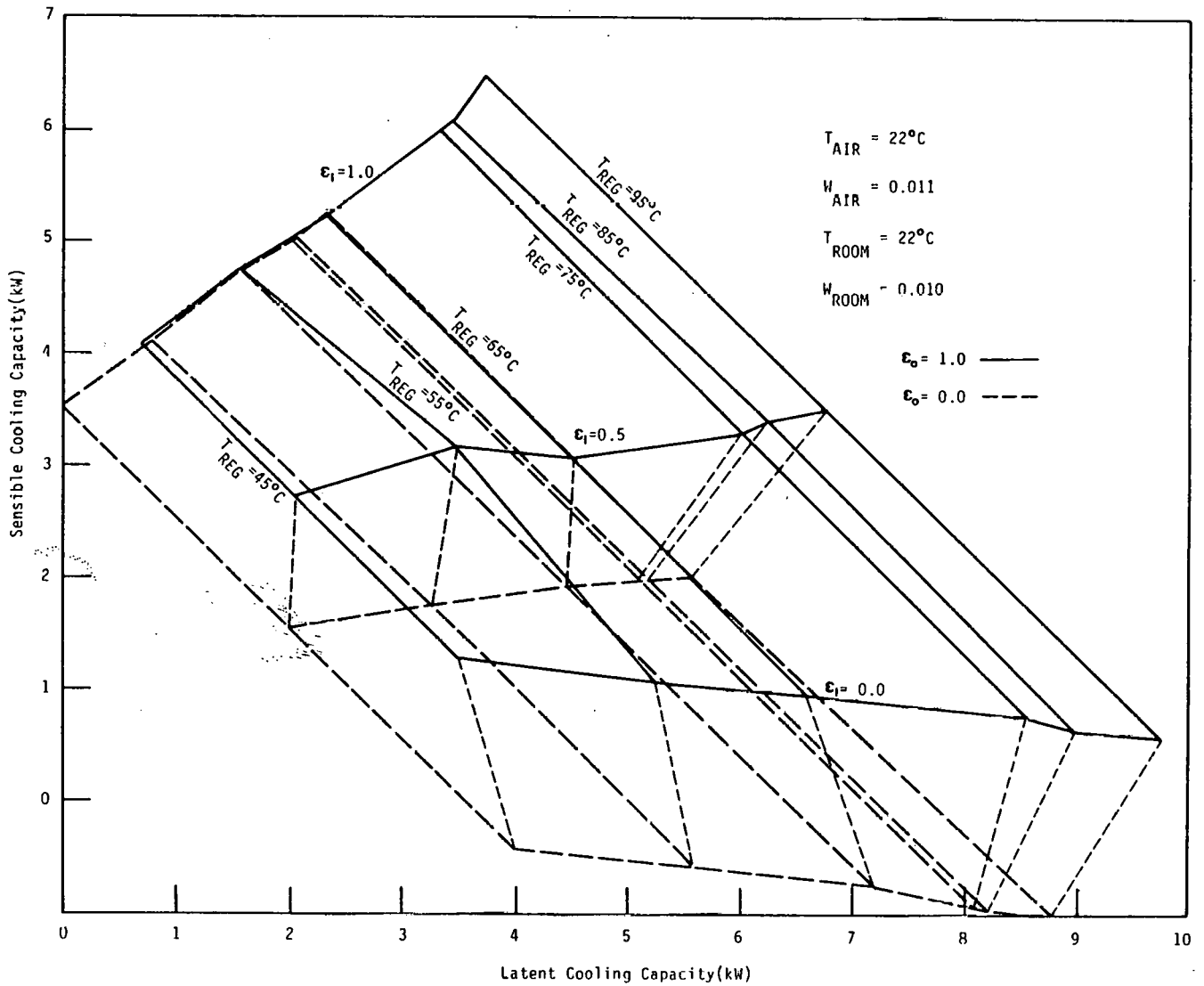
Figure 3-13. STEADY-STATE PERFORMANCE MAP: MILD AMBIENT CONDITIONS
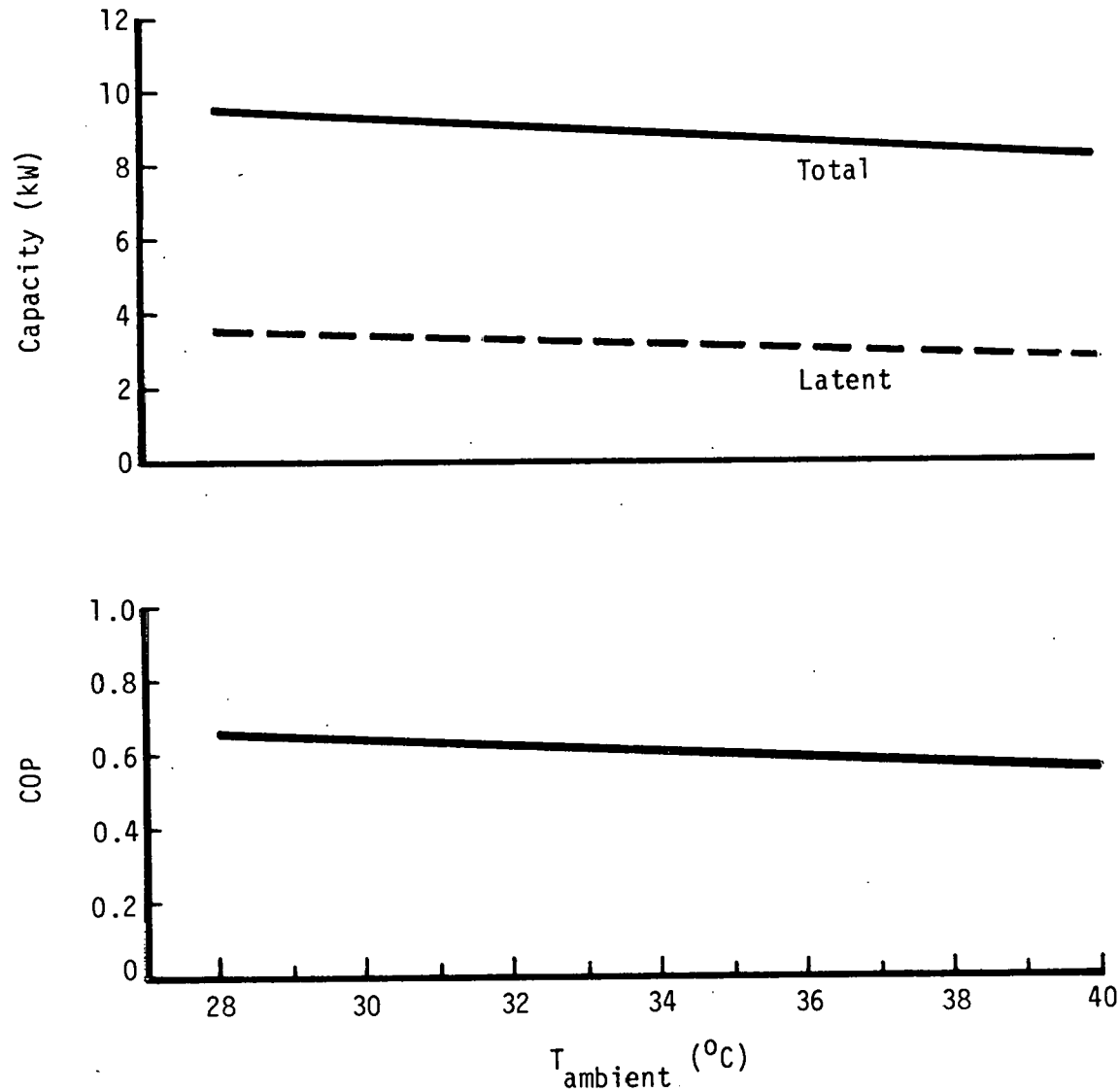
$T_{room} = 25.5^{o}C$

$W_{room} = 0.0124$

$W_{ambient} = 0.0155$

$T_{regeneration} = 85^{o}C$

Figure 3-14.   STEADY-STATE SENSITIVITY TO AMBIENT TEMPERATURE

Figure 3-15. STEADY-STATE SENSITIVITY TO AMBIENT HUMIDITY
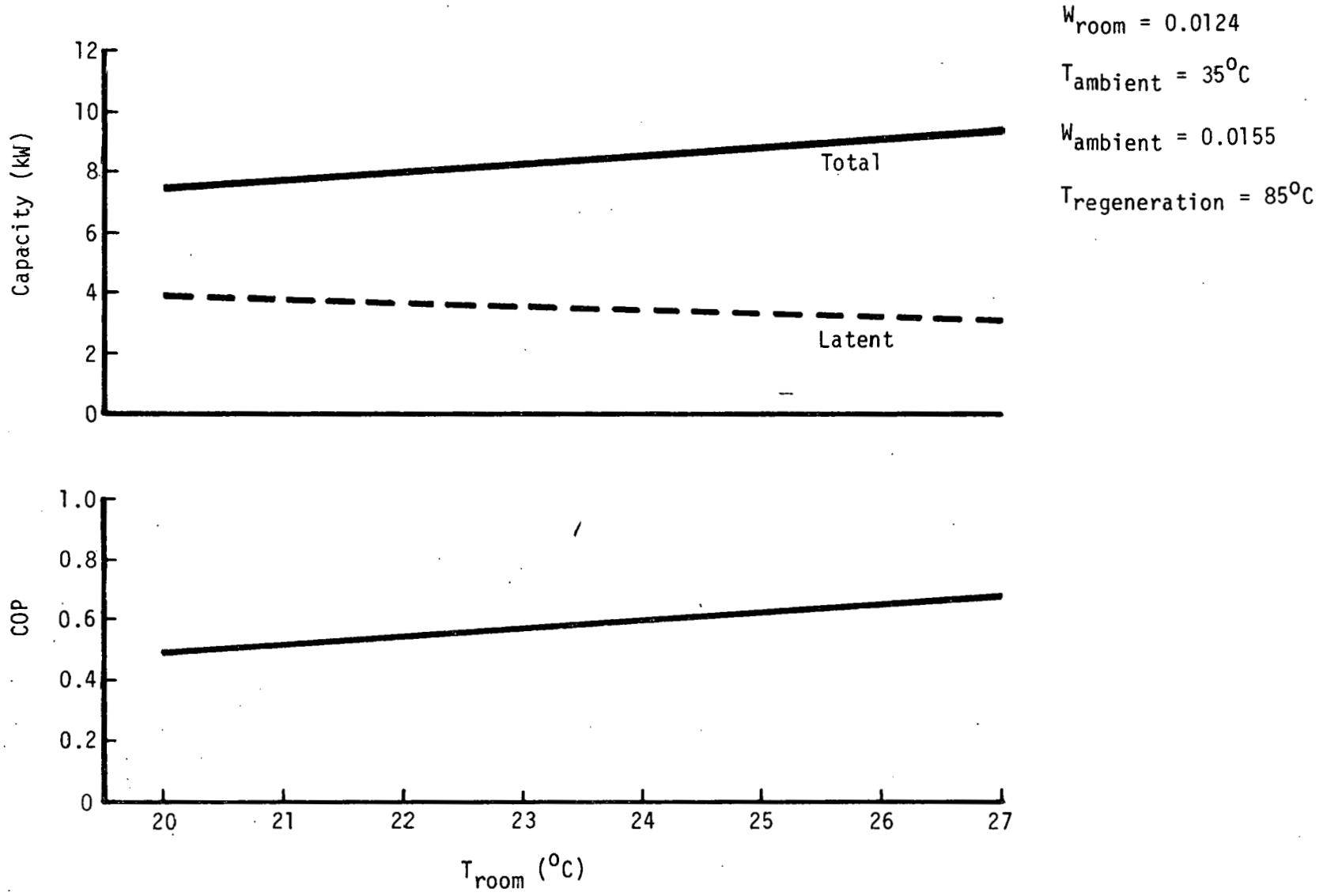
Figure 3-16. STEADY-STATE SENSITIVITY TO ROOM TEMPERATURE
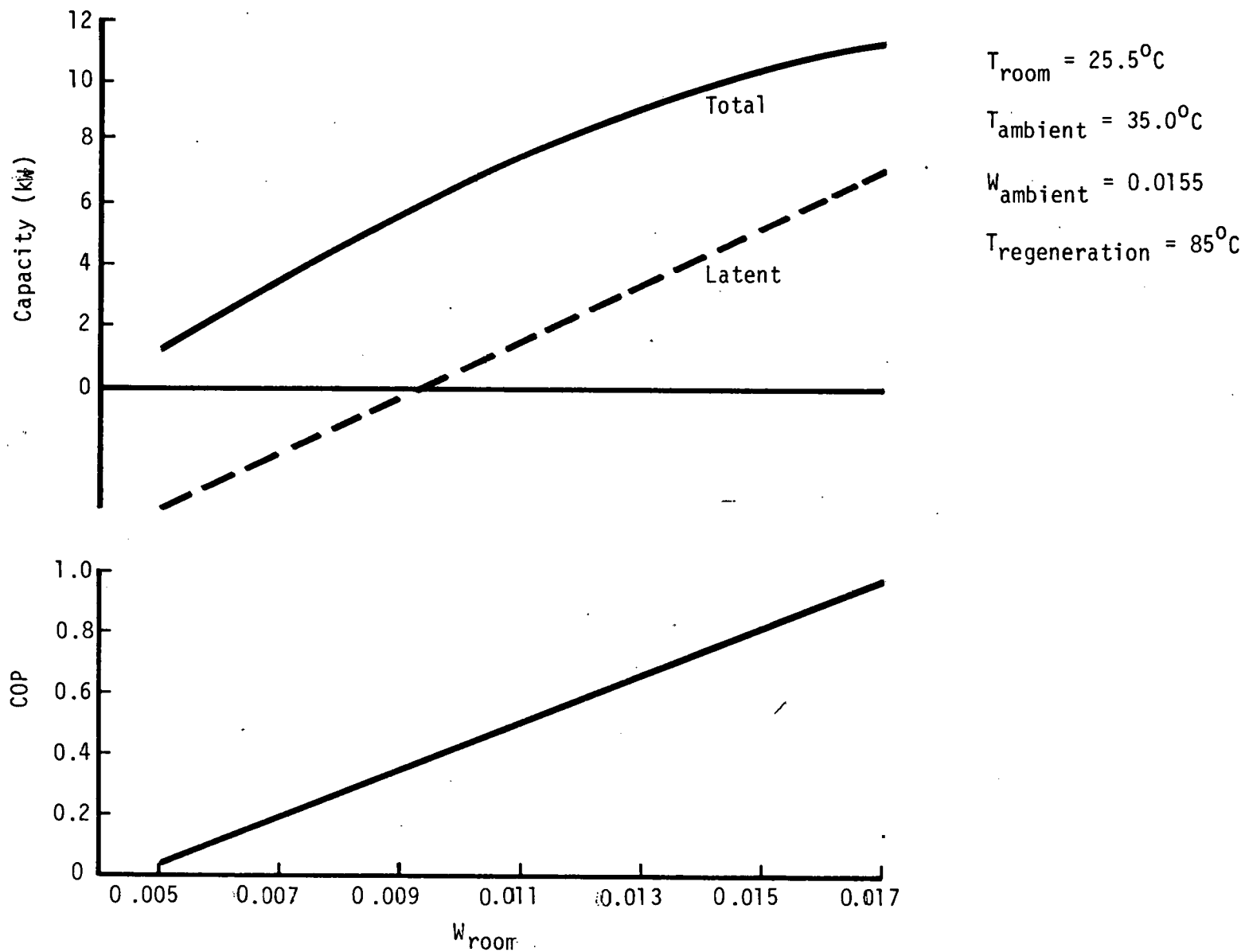
Figure 3-17. STEADY-STATE SENSITIVITY TO ROOM HUMIDITY

$T_{room} = 25.5^oC$

$W_{room} = 0.0124$

$T_{ambient} = 35^oC$

$W_{ambient} = 0.0155$
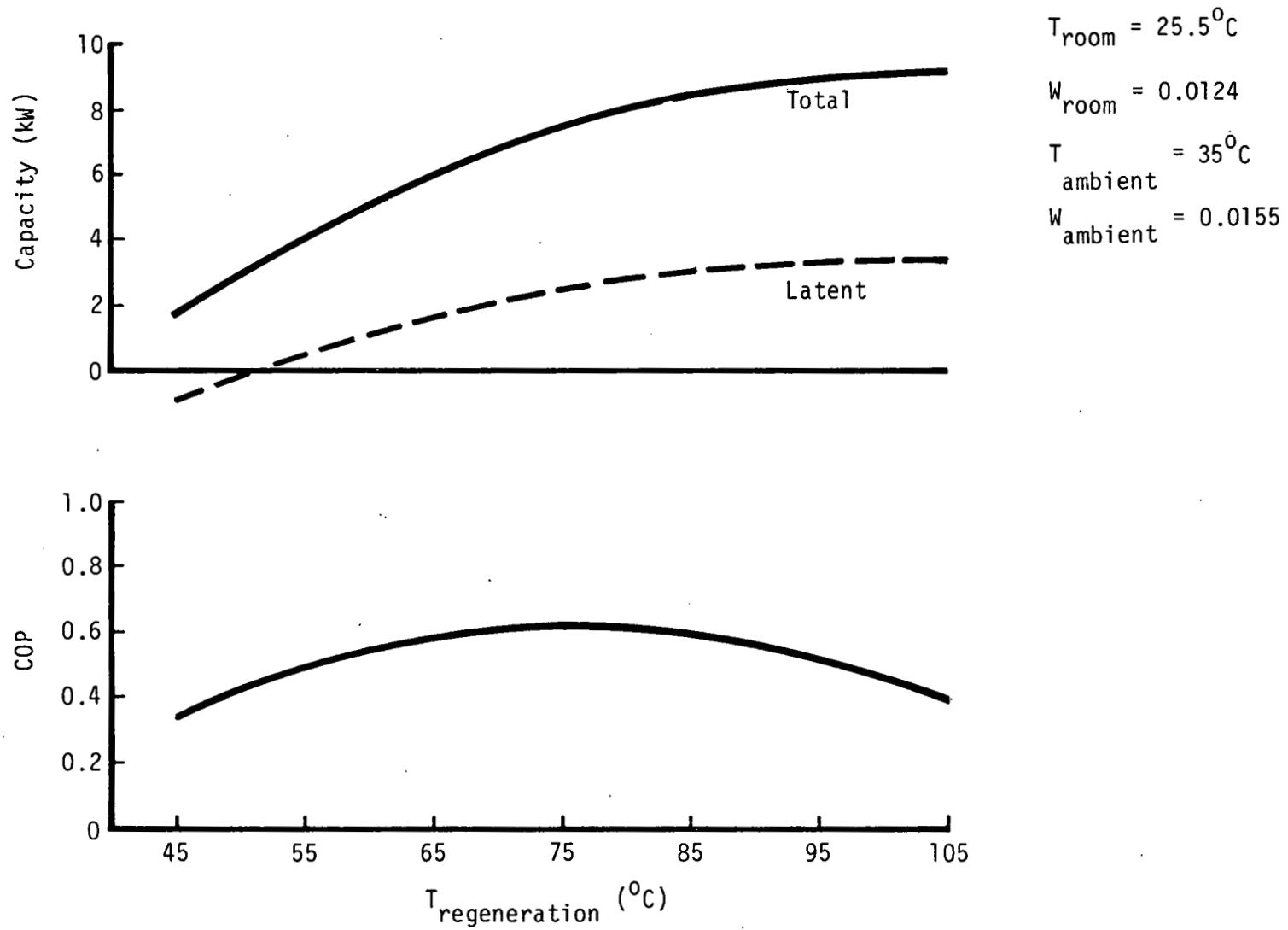
Figure 3-18.   STEADY-STATE SENSITIVITY TO REGENERATION TEMPERATURE
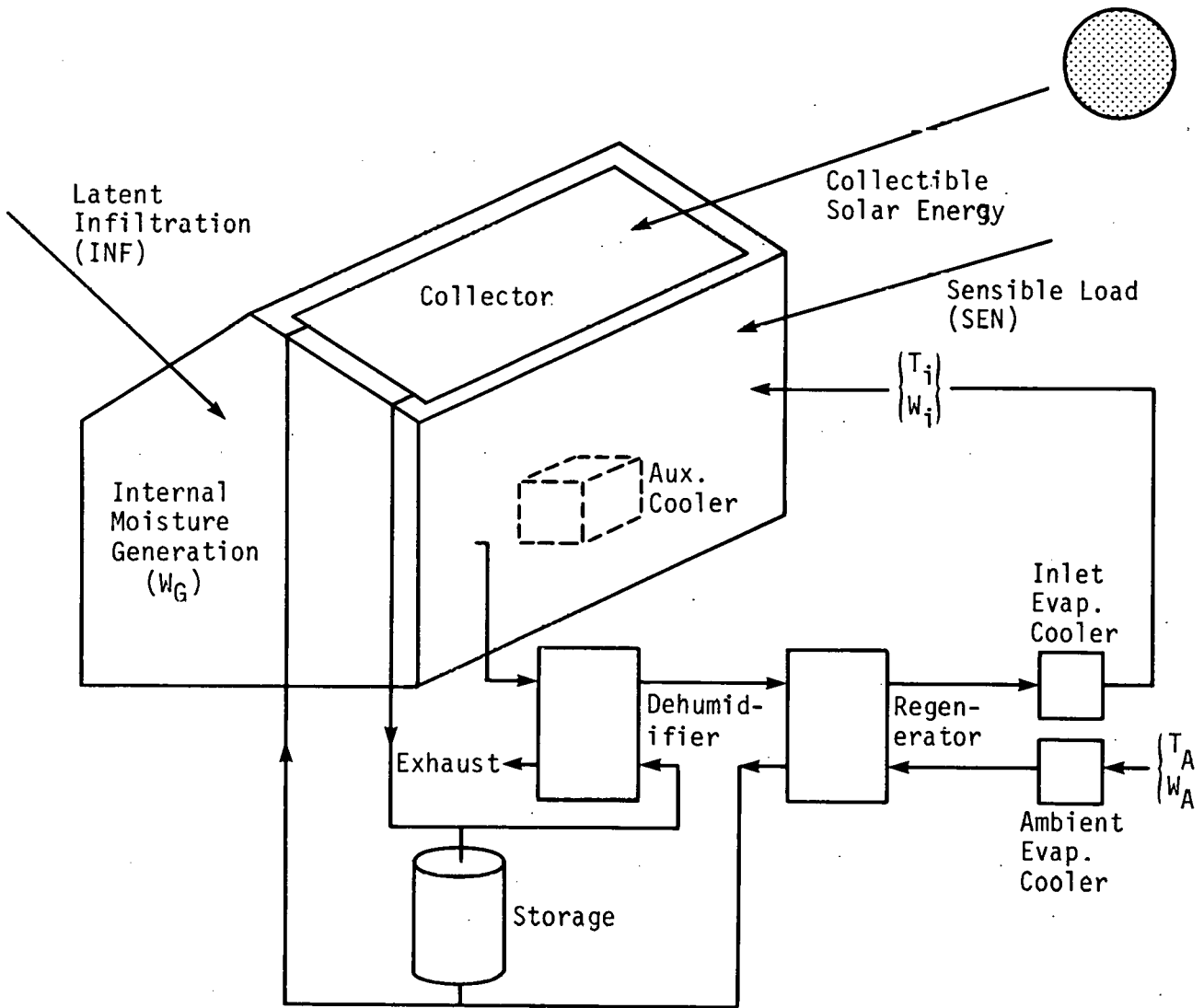
Figure 3-19. DESICCANT SYSTEM COMPONENT CONFIGURATION FOR DYNAMIC SIMULATIONS

As discussed in Section 3.7.2, upper and lower operation temperature limits (T_{MAX} and T_{MIN}) for the desiccant system were appropriate. Whenever cooling capacity was required and the temperature available from the collector/storage system (T_{REG}) was less than T_{MIN}, air flow to the dehumidifier was turned off ($\dot{m}_L = 0$). To ensure that energy was never used at an outlet temperature exceeding the maximum ($T_{REG} > T_{MAX}$), the flowrate through the collector was increased ($\dot{m}_C > \dot{m}_L$) when necessary to regulate the heat delivered either to storage or directly to the drier. The standard TRNSYS collector subsystem was modified to provide collector/storage usage as summarized in Table 3-14.

The transient house model was sensitive to external latent and sensible loads, internal moisture generation, and the cooling capacities of the desiccant machine and an optional auxiliary vapor-compression unit. Calculation of the latent and sensible cooling capacities of the desiccant system was identical to that for the steady-state case (Figure 3-5) except that the room conditions were allowed to fluctuate in response to the driving forces. Initial values for the indoor temperature and humidity were selected to be $T_R = 23.9°C$ and $W_R = 0.0085$ $kg_{H_2O}/kg_{dry\ air}$, respectively. Thereafter, the room temperature variation was determined by the sensible machine capacity, if any, and the sensible load was computed as described in Section 3.2.3. Similiarly, the room humidity was a function of the desiccant dehumidification capacity and the latent load experienced by the house.

The latent load consisted of two contributions: the transposition of indoor and outdoor moisture due to infiltration, and the formation of water vapor due to internal, water fixtures, household appliances, and residents. A representative daily profile of internal water vapor generation, illustrated in Figure 3-20, was assumed. The instantaneous latent load LAT was then computed:

$$LAT = [(W_{AMB} - \overline{W}_R)\ INF + W_G]\ H_{fg},$$

where

$$\overline{W}_R = \text{average room humidity } (kg_{H_2O}/kg \text{ dry air}),$$
$$INF = \text{rate of infiltration} = 0.282\ kg_{dry\ air}/h,$$
$$W_G = \text{internal rate of moisture generation, and}$$
$$H_{fg} = \text{latent heat of vaporization of water} = 2.466 \times 10^6\ J/kg_{H_2O}.$$

3.7.2 Control Strategy

The primary objective of the control strategy was to maintain the climatic conditions inside the house within the specified comfort limits. During this process, it was desirable to minimize the consumption of energy and the operation time of the machine. This was achieved in the controller by the assignment of values to control parameters. The strategy for choosing these values was based on the results of the steady-state analysis. Implementation of several auxiliary power mode options and accurate bookkeeping of the house loads and system capacities were additional functions performed by the controller.

Table 3-14. SOLAR COLLECTOR/STORAGE USAGE[a]

Desiccant System Flow Logic			Collector System Flow Logic	
Zone	T_{REG}	$\dot{m}_L$	T_{COLL}	$\dot{m}_C$
1, 2, 6		0	$\leq (T'_{REG} + DT)$	0
3, 4, 5	$< T_{MIN}$	0	$< T_{STOR}$	0
3, 4, 5	$\geq T_{MIN}$	$\dot{m}_S$	$(T_{REG} + DT) < T_{COLL} < T_{MAX}$	$\dot{m}_S$
			$= T_{MAX}$	$> \dot{m}_S$

		Collector-Desiccant Interaction

$\dot{m}_L$	$\dot{m}_C$	Action
0	0	None
0	$= \dot{m}_S$	Deliver collected solar energy to storage at $T < T_{MAX}$
$\dot{m}_S$	$> \dot{m}_S$	Deliver collected solar energy to storage at $T = T_{MAX}$
$\dot{m}_S$	0	Meet load with energy in storage
$\dot{m}_S$	$= \dot{m}_S$	Meet load with collected solar energy in parallel with storage
$\dot{m}_S$	$> \dot{m}_S$	Meet load with collected solar energy and deliver excess to storage

[a]Zone = control zone based on defined house comfort conditions (see Section 3.7.2 and Figure 3-21)

T_{REG} = temperature available from storage/collector system

T_{MIN} = lower operation control temperature = 45°C

$\dot{m}_L$ = load flowrate demanded by desiccant system

$\dot{m}_S$ = desiccant system design flowrate

T_{COLL} = temperature available from collector

T'_{REG} = temperature input to collector from regenerator

DT = temperature dead band = 5°C

T_{STOR} = average storage temperature

T_{MAX} = upper operation control temperature = 85°C

$\dot{m}_C$ = collector flowrate
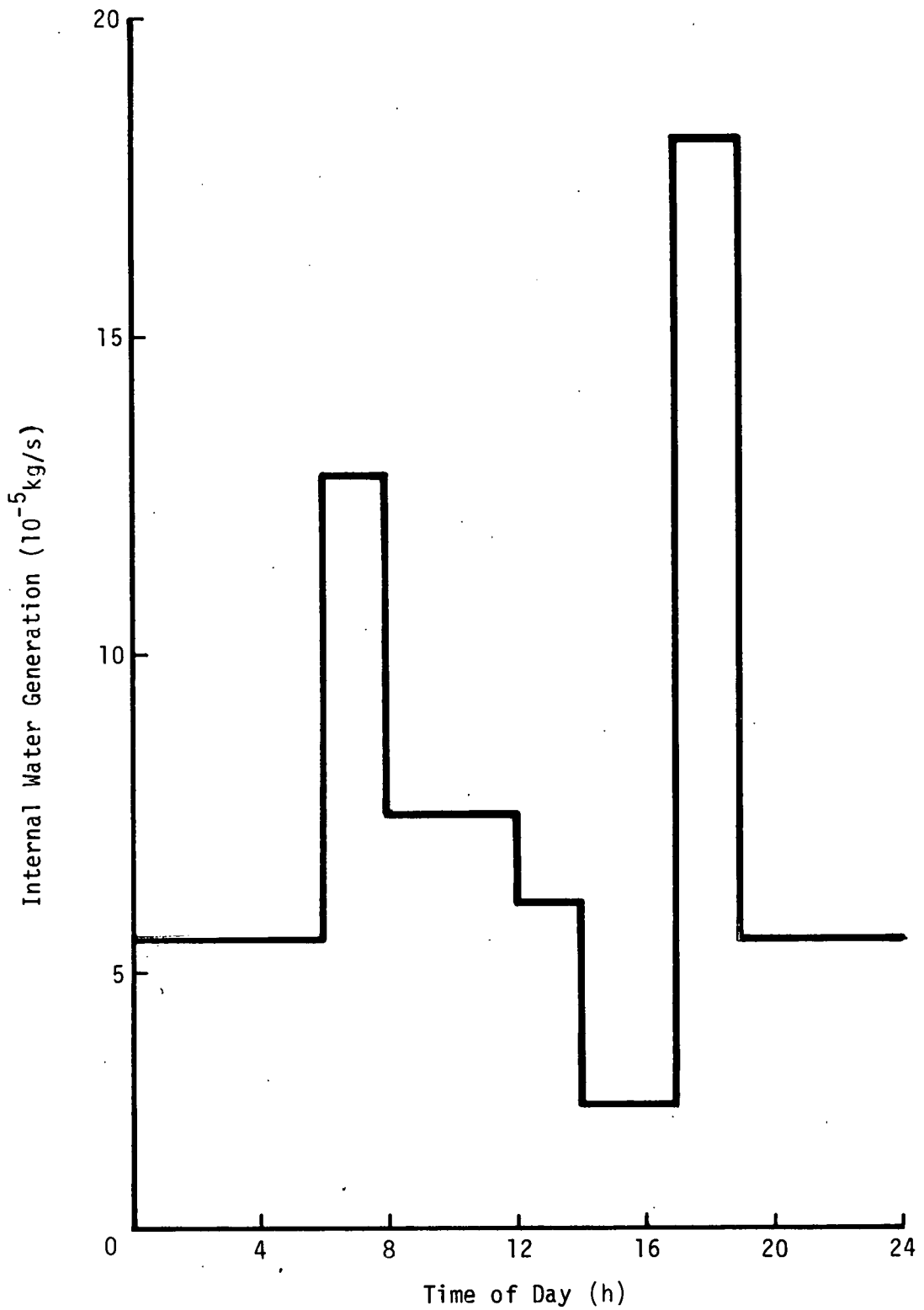
Figure 3-20. DIURNAL VARIATION OF HOUSE INTERNAL WATER VAPOR GENERATION

Nelson has shown that a set of parameters suitable for characterizing the performance of the desiccant system includes the temperature of the regeneration stream T_{REG} and the effectiveness of the inlet and outlet evaporative coolers (ϵ_i and ϵ_o) [4]. It can be seen from the steady-state performance maps that T_{REG} and, to a lesser extent, ϵ_o govern the magnitude of the attainable cooling capacity, whereas ϵ_i regulates the ratio of the sensible to latent components (S/L). The strategy was then to vary these control parameters to maximize latent cooling when indoor conditions became too humid, maximize sensible cooling when indoor conditions were too hot, and adjust S/L appropriately when both humidity and temperature exceeded specified thresholds.

Studies by the University of Wisconsin and Solaron Corporation used solar energy supplemented by auxiliary heat to provide the desired regeneration temperature at all times [25, 26]. It was implicitly assumed that all drifting of room conditions were comfort/noncomfort interactions; that is, no passage from one noncomfort zone to another ever occurred. An incremental control loop was used to approximate a continuous feedback control system. Based upon the room conditions at the start of each time step, the control parameters were adjusted to meet the house loads. Because optimum system performance (COP) is inversely related to the regeneration stream temperature, T_{REG} was initialized to a minimum value and then increased with each time step in an effort to enhance system operating efficiency. An upper limit cutoff of T_{REG} (max) = 85°C was found to be practical. Wisconsin concluded that further refinements were needed in the control strategy to significantly improve the operation of the desiccant system.

Rather than optimizing the efficient use of solar energy by the desiccant cooler (COP), the emphasis of the control strategy used in this study was to maximize the time spent in comfort (zone 6) and minimize the duty cycle of the desiccant cooler. This should result in the minimum amount of electrical input and thus lower operating costs. The controller was modified to reflect this operating philosophy. Whenever an excursion out of comfort occurred, regeneration of the desiccant material was demanded. A lower cutoff regeneration temperature, required by the dynamic behavior of the gel and evident from the performance maps, was chosen as T_{REG} (min) = 45°C. The maximum temperature between T_{REG} (min) and T_{REG} (max) available from storage was then used, resulting in the generation of the greatest attainable cooling capacity. Obvious trade-offs exist between this method and the attempt to ration the collected solar heat by applying an incremental scheme. By gradually elevating the system capacity until the loads are just met, system COP is maximized by minimizing solar energy usage. However, deviations from comfort occur with greater frequency because just matching the cooler capacity with the house loads controls along the comfort boundaries. This tends to increase operation time and thereby increase the expenditure of electrical energy. On the other hand, operating at maximum available capacity ensures the highest probability of returning to comfort in the shortest time period, and less electrical energy (fan usage) should be, required. In addition, this strategy allows a deeper penetration into the comfort zone.

Based on the definition of house comfort conditions, six unique control zones can be defined (Figure 3-21). Unlike prior simulations that controlled to a maximum absolute humidity, it was decided to use relative humidity as the
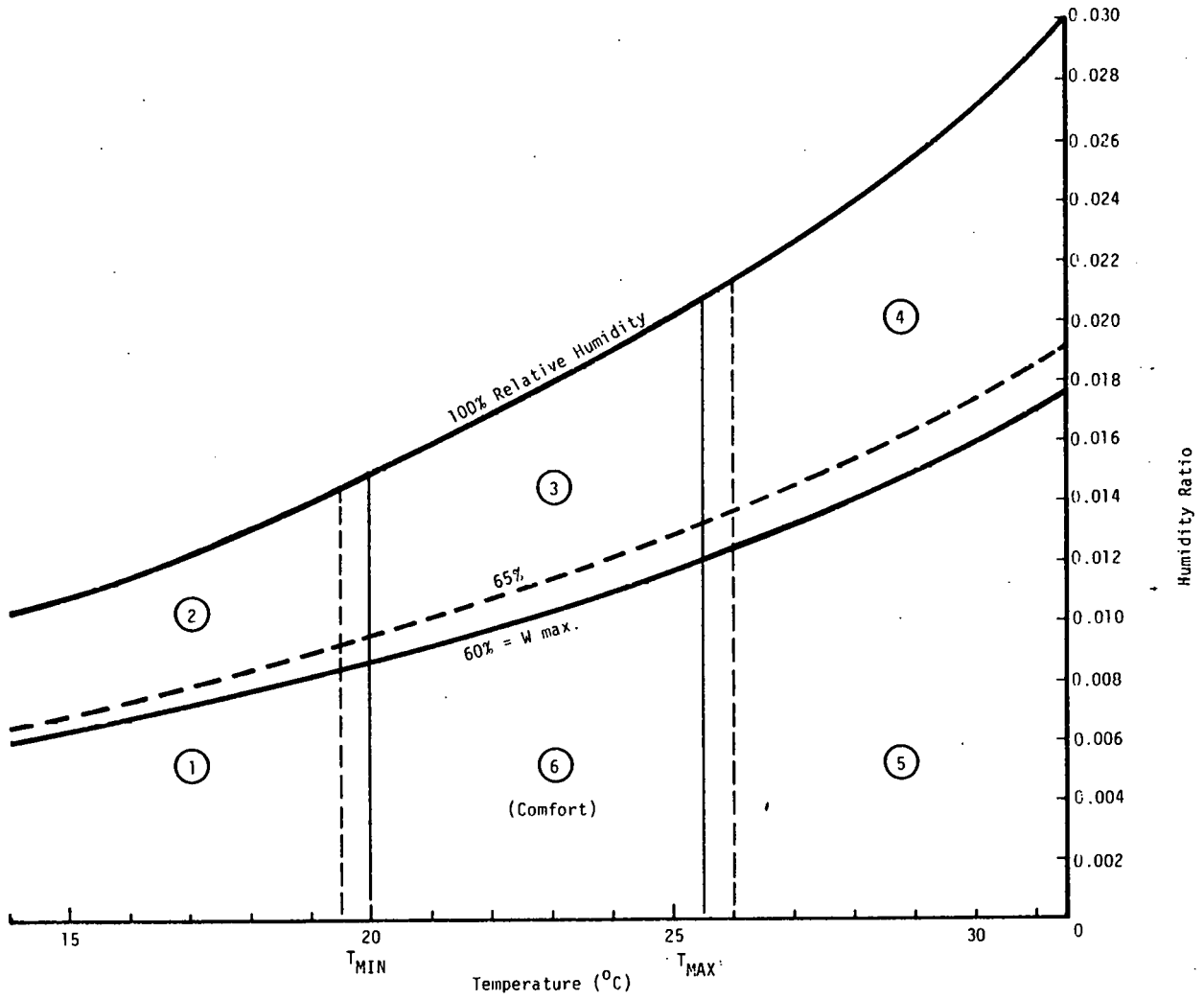
Figure 3-21. CONTROL ZONES BASED ON HOUSE COMFORT CONDITIONS (Dead bands are located between solid and dashed lines)

control criterion. The reason for adopting this control parameter was the wide variance (34%) in humidity ratio at the upper and lower temperature limits (T_{MAX} and T_{MIN}) at 60% relative humidity. "Dead bands" were employed to suppress the effect of small drifts out of the comfort zone due to fluctuations in room conditions caused by the occasional inability of the machine to meet the total load. Whenever the house resided in a dead band, it was considered to be in comfort but the desiccant cooling system functioned with the appropriate zonal strategy as described in the following paragraphs.

Sensible heating is required to return the zone-1 room state to comfort. The desiccant unit does not operate in this zone. However, the amount of heating needed to return to comfort (zone 6) was computed in the model and compared with the external house load due to insolation. If ambient conditions were sufficient to restore comfort within one time increment, then no additional action was taken. Otherwise, the house heating load was met by auxiliary heating. Collected/stored solar energy was used for heating during the heating season and strictly for cooling during the rest of the year. Heating loads occurring during the cooling season were always met by auxiliary heating.

Zone-2 strategy dictates that the house be heated and dehumidified. Figure 3-9 presents the performance of the desiccant system at the crossover point between zone 2 and comfort. Operating with $\varepsilon_i = \varepsilon_o = 0$ produces negative sensible (heating) capacity concurrent with positive latent (dehumidification) capacity. However, preliminary simulations showed that zone 2 was reached only occasionally during the transition periods (between heating and cooling seasons). Attempts to operate the desiccant system as outlined above resulted in an approximately isothermal transition from zone 2 to zone 1. This was caused by the inability of the desiccant system heating capacity to compensate for the predominantly negative (heating) loads, the net effect of which was dehumidification only. Consequently, it was decided that the optimum approach would be to duplicate the zone-1 control strategy in zone 2, resulting in the transition of room conditions from zone 2 to zone 3. The unused solar capacity was then used for dehumidification by the zone 3 strategy.

Ideally, only dehumidification is necessary in zone 3. Nominal system performance for this condition is displayed in Figure 3-6. In this case, it is desirable to operate with zero sensible cooling capacity, a good approximation to this being $\varepsilon_o = 0.9$ and $\varepsilon_i = 0.4$. In practice, the decrease in ambient temperatures each evening results in increased relative humidity and negative sensible house loads. If the humidity drives the indoor state into zone 3 (point A to point B in Figure 3-22), a latent capacity sufficient to bring about a change in humidity ratio $\Delta W = W(B) - W(C)$ is required to restore comfort, with the room temperature assumed constant. The accompanying negative evening load, however, lowers the room temperature by an amount $\Delta T = T(C) - T(D)$. Thus, pure dehumidification of magnitude ΔW, in conjunction with a decrease in room temperature, results in room state D rather than C and comfort is not achieved. For this case, a better strategy is to operate the system with both evaporative coolers turned off ($\varepsilon_o = \varepsilon_i = 0$ in Figure 3-6), thereby generating heating capacity to offset the sensible load, and increasing the latent capacity (Table 3-15).

Conditions represented by zone 4 require both latent and sensible cooling to return to comfort. System performance at the intersection of zones 4 and 6 is displayed in Figure 3-7. The ratio of the sensible to latent loads
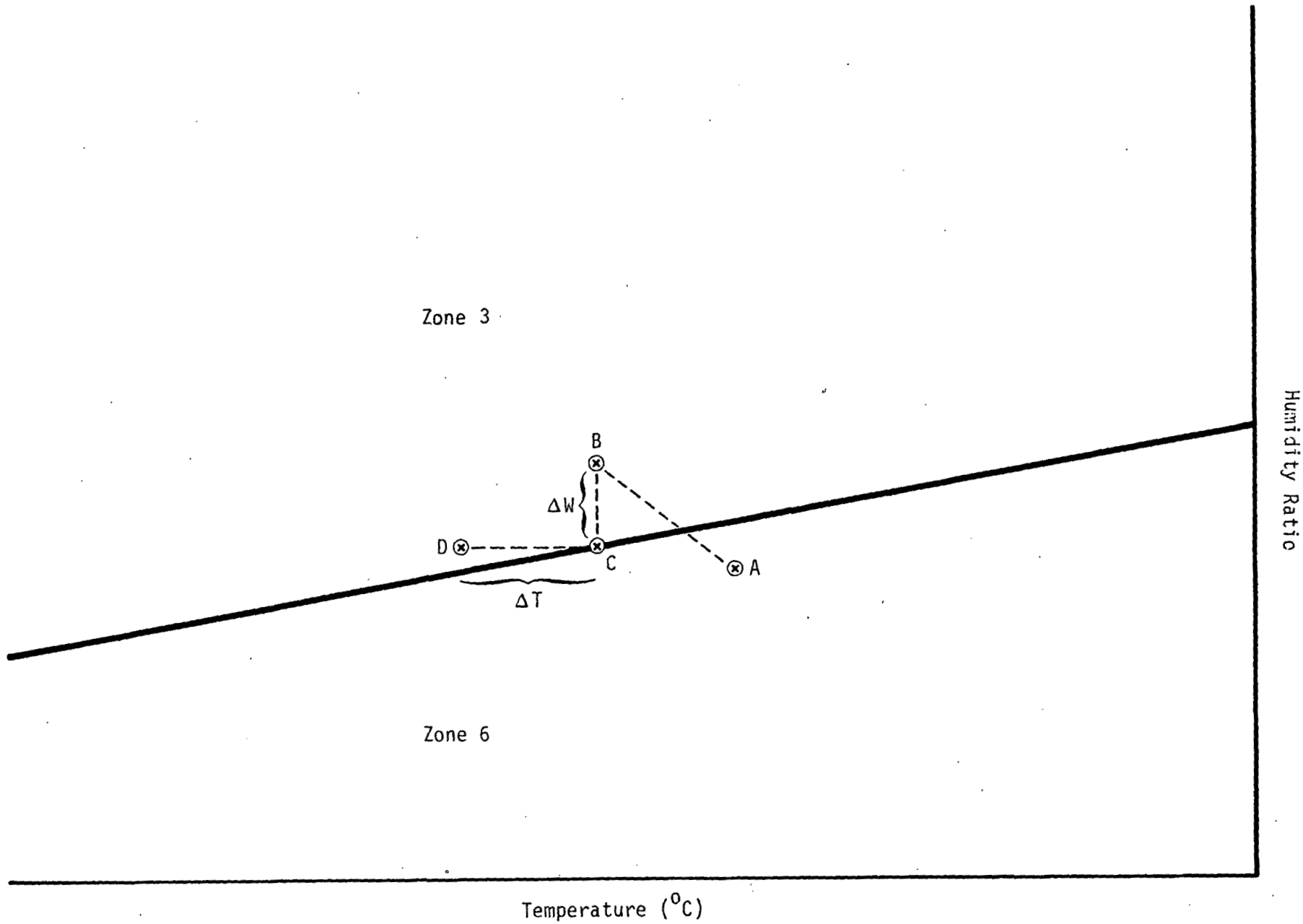
Figure 3-22. ILLUSTRATION OF FAILURE TO RESTORE COMFORT BY DEHUMIDIFICATION ALONE

Table 3-15. DESICCANT SYSTEM CONTROL STRATEGY[a]

Zone	$\dfrac{SEN}{\lvert LAT \rvert}$	T_{REG}	$\dot{m}$	ε_i	ε_o
1	--	--	off	--	--
2	--	--	off	--	--
3	--	< 45°	off	--	--
	< 0	> 45°	on	0.0	0.0
	> 0	> 45°	on	0.4	0.9
4	--	< 45°	off	--	--
	> 1.0	> 45°	on	0.9	0.9
	< 1.0	45° to 50°	on	0.40 + 0.10 X	0.9
	< 1.0	50° to 55°	on	0.40 + 0.15 X	0.9
	< 1.0	55° to 60°	on	0.40 + 0.20 X	0.9
	< 1.0	60° to 65°	on	0.40 + 0.25 X	0.9
	< 1.0	65° to 70°	on	0.40 + 0.30 X	0.9
	< 1.0	70° to 75°	on	0.40 + 0.35 X	0.9
	< 1.0	75° to 80°	on	0.40 - 0.40 X	0.9
	< 1.0	80° to 85°	on	0.40 + 0.45 X	0.9
5	--	< 45°	off	--	--
	--	45° to 50°	on	0.7	0.9
	--	50° to 55°	on	0.8	0.9
	--	55° to 60°	on	0.9	0.9
	--	60° to 65°	on	0.9	0.9
	--	65° to 70°	on	0.9	0.9
	--	70° to 75°	on	0.9	0.9
	--	75° to 80°	on	0.9	0.9
	--	80° to 85°	on	0.9	0.9
6	--	--	off	--	--

[a] SEN = sensible house load
LAT = latent house load
T_{REG} = regeneration temperature (°C)
$\dot{m}$ = flowrate (kg/sec)
ε_i = inlet evaporative cooler effectiveness
ε_o = ambient evaporative cooler effectiveness
X = SEN/LAT

experienced by the house should be reflected in the S/L capacity supplied. When the sensible load exceeded the latent element, the sensible heating ratio SHR (sensible capacity/total capacity) was maximized by operating with $\epsilon_i = \epsilon_o$ = 0.9. Conversely, when the S/L load $\leq$ 1, the inlet cooler effectiveness was appropriately proportioned to furnish cooling to counteract both components of the load. Details of this strategy are given in Table 3-15. Implementation of the simulated controller in a practical sense would require monitoring of both indoor and outdoor temperatures and humidities.

Room state 5 demands sensible cooling only for restoration of comfort. This corresponds to operation along the line QLAT = 0 in Figure 3-8. Achievement of this control was facilitated by optimum outlet cooler effectiveness (ϵ_o = 0.9) combined with a scheme parameterizing ϵ_i as a function of T_{REG} (max) as specified in Table 3-15.

Zone 6 represents comfort conditions. No operation of the desiccant system is needed. The action of the control strategy in this case was merely to shut off the machine.

The controller was allowed an auxiliary cooling option to supplement desiccant system performance. In this mode, the portion of the latent and/or sensible load that could not be met by the solar desiccant cooler was provided by an assumed auxiliary vapor-compression unit. This guaranteed an immediate return to comfort whenever the room state entered zones 3, 4, or 5. Several runs for various cities were made in which only the sensible component was supplemented. A second auxiliary alternative, whereby the regeneration stream temperatures could be augmented to T_{REG} = 85°C at all times, was installed in the controller but not applied in the present study. Frequency of auxiliary usage and failure of the desiccant system to operate when demanded [due to T_{REG} (max) < 45°C] was recorded.

A convention was established to account for the house loads, desiccant cooler system capacities, auxiliary heating, and auxiliary cooling to assure an accurate balance of energy. The possibility of both positive and negative house loads and desiccant cooler system capacities necessitated a special energy accounting method. All heating loads occuring in zones 1 and 2 were assumed to be met by an auxiliary heating supply. No cooling loads or capacities were allowed in these zones. In zone 3, the control strategy permitted a desiccant cooler sensible heating (negative) capacity. Similarily, because zone 5 attempted to operate at zero capacity, ambient and room conditions could precipitate a desiccant cooler latent heating (humidification) capacity. To handle these cases in zones 3 through 6, whenever a desiccant system heating capacity (sensible or latent) was experienced, it was added to the house cooling load. All sensible and latent loads and capacities in these four zones were then algebraically summed. Table 3-16 gives a summary of this accounting system.

3.7.3 Results of Case Studies

The simulations were used to investigate such questions as which climate was most appropriate for desiccant system operation, what system capacity and collector size were optimum, and which auxiliary cooling strategy most enhanced desiccant system performance. Five cities were chosen on the basis

Table 3-16. ENERGY ACCOUNTING CONVENTION[a]

Zone	QLOADS	QCAPS	QCAPL	HLOADS	CLOADS	CLOADL	HCAPS	HCAPL	CCAPS	CCAPL
1, 2	> XLOADS	--	--	0	0	0	0	0	0	0
	0 < QLOADS < XLOADS	--	--	XLOADS-QLOADS	0	0	0	0	0	0
	QLOADS < 0 < XLOADS	--	--	XLOADS	0	0	0	0	0	0
3, 4, 5		< 0	< 0	0	QLOADS+HCAPS	QLOADL+HCAPL	|QCAPS|	|QCAPL|	0	0
		< 0	> 0	0	QLOADS+HCAPS	QLOADL+HCAPL	|QCAPS|	0	0	QCAPL
		> 0	< 0	0	QLOADS+HCAPS	QLOADL+HCAPL	0	|QCAPL|	QCAPS	0
		> 0	> 0	0	QLOADS+HCAPS	QLOADL+HCAPL	0	0	QCAPS	QCAPL
6		0	0	0	QLOADS	QLOADL	0	0	0	0

[a] QLOADS = sensible house load
QCAPS = sensible desiccant system capacity
QCAPL = latent desiccant system capacity
HLOADS = sensible auxiliary heat required
CLOADS = sensible cooling house load
CLOADL = latent cooling house load
HCAPS = sensible heating capacity of desiccant system
HCAPL = latent heating capacity of desiccant system
CCAPS = sensible cooling capacity of desiccant system
CCAPL = latent cooling capacity of desiccant system
XLOADS = sensible heating required to boost room temperature back to comfort
$= CAP_H (T_{MIN} - T_{ROOM})/\Delta t$
CAP_H = capacity of the house = 10^7 J/°C
Δt = time step = 1800 s
T_{MIN} = minimum comfortable room temperature = 20°C

of their diverse climatic characteristics: Phoenix, Arizona (very hot, dry); Dodge City, Kansas (warm, dry); Fort Worth, Texas (hot, humid); Washington, D.C. (warm, humid); and Charleston, South Carolina (warm, very humid). For each city, a series of yearly simulations were made. Results of solar heating during the winter months were incorporated with results of the cooling season to allow annual economic analyses. A range of collector sizes and desiccant system capacities were simulated at each of the geographic locations of interest. The auxiliary strategies were as follows: no auxiliary cooling, auxiliary cooling driven by the sensible load, and auxiliary cooling designed to match both the sensible and latent cooling requirements that could not be met by the desiccant system. A summary is given in Table 3-17 of the simulations.

To determine the relative degree of desirability of solar heating and cooling compared to solar heating only and to permit amortized cost estimates, both the heating and the cooling seasons were simulated. The cooling season was treated as described in Section 3.7.1. A solar space heating unit was modeled for the heating season as shown in Figure 3-23. Solar water heating was not considered.

The objective of the dynamic study was to ascertain the most favorable climate for solar heating and desiccant cooling and the combination of system size and collector area yielding the best thermal performance. The principal findings can be summarized as follows:

- Climates which experience both high heating loads in the winter and high cooling loads in the summer are the most suitable candidates for solar heating and desiccant cooling.

- Washington, D.C., is the most promising of the five cities examined for the application of solar heating and desiccant cooling (due to its climatic conditions).

- The desiccant system which was modeled is preferentially a latent rather than sensible cooler.

- Thermal performance is best understood in terms of the dynamic interaction of the desiccant machine with the solar collector/storage system outlet temperature.

- The optimum combinations of system sizes for Washington, D.C., are 4.5-kW capacity at 35-m^2 collector area and 9.0-kW capacity at 50-m^2 collector area.

For the various cities included in this study, Figure 3-24 displays the percentage of the heating load provided by solar energy as a function of collector size. Geographical optimization of collector areas for solar heating is controlled by economic considerations. Projections based upon a series of F-Chart analyses [27] appear in Table 3-18 with the corresponding heating loads met by solar energy. It can be seen that, with the exception of Phoenix, cost is minimized when 30% to 50% of the seasonal heating load is provided by solar energy. The discrepancy of Phoenix is due to the relatively low heating load and high insolation. Since so little heating is needed, a higher percentage must be met by solar energy to allow an appreciable total fuel savings to compensate for collector costs. A more detailed discussion of heating season economics is presented in Section 5.3.

Table 3-17. PARAMETERS USED IN DESICCANT COOLER SIMULATIONS

Run No.	Cooler Size (kW)	Collector Area (m^2)	Storage Volume (m^2)	Auxiliary Cooling
WASH[a] 1	9.0	35	13.1	None
WASH 2	9.0	35	13.1	Sensible
WASH 3	4.5	35	13.1	None
WASH 4	4.5	35	13.1	Sensible
WASH 5	2.3	35	13.1	None
WASH 6	2.3	35	15.1	Sensible
WASH 7	9.0	50	19.6	None
WASH 8	9.0	50	19.6	Sensible
WASH 9	4.5	50	19.6	None
WASH 10	4.5	50	19.6	Sensible
WASH 12	2.3	50	19.6	Sensible
WASH 13	9.0	20	7.8	Sensible
WASH 14	4.5	20	7.8	Sensible
WASH 15	2.3	20	7.8	Sensible
WASH 16	9.0	42	16.5	Sensible
WASH 17	4.5	42	16.5	Sensible
WASH 18	2.3	42	16.5	Sensible
WASH 19	9.0	42	16.5	Sensible + Latent
WASH 20	9.0	20	7.8	Sensible + Latent
WASH 21	9.0	35	13.1	Sensible + Latent
WASH 22	9.0	50	19.6	Sensible + Latent
WASH 23	9.0	10	3.9	Sensible + Latent
WASH 24	4.5	10	3.9	Sensible + Latent
WASH 25	4.5	35	13.1	Sensible + Latent
WASH 26	4.5	50	19.6	Sensible + Latent
WASH 27	2.3	10	3.9	Sensible + Latent
WASH 28	2.3	35	13.1	Sensible + Latent
WASH 29	2.3	50	19.6	Sensible + Latent
DODG[b] 1	9.0	35	13.1	None
DODG 2	4.5	35	13.1	None
DODG 3	4.5	35	13.1	Sensible
DODG 4	2.3	35	13.1	None
DODG 5	2.3	35	13.1	Sensible
DODG 6	9.0	20	7.8	None
DODG 7	4.5	20	7.8	None
DODG 8	2.3	20	7.8	None
DODG 9	9.0	42	16.5	None
DODG 10	4.5	42	16.5	None
DODG 11	2.3	42	16.5	None
DODG 12	4.5	35	13.1	Sensible + Latent
CHAR[c] 8	9.0	50	19.6	Sensible
CHAR 10	9.0	50	19.6	None
CHAR 11	4.5	35	13.1	Sensible
CHAR 12	4.5	35	13.1	None
CHAR 13	13.5	50	19.6	Sensible
CHAR 14	4.5	35	13.1	Sensible + Latent
FTWR[d] 1	9.0	35	13.1	None
FTWR 2	4.5	35	13.1	None
FTWR 3	4.5	35	13.1	Sensible
FTWR 5	2.3	35	13.1	Sensible
FTWR 6	4.5	35	13.1	Sensible + Latent
PHNX[e] 1	9.0	35	13.1	None
PHNX 2	9.0	35	13.1	Sensible
PHNX 4	4.5	35	13.1	Sensible
PHNX 7	4.5	45	13.1	Sensible + Latent

[a]WASH = Washington, D.C.   [c]CHAR = Charleston, S.C.   [e]PHNX = Phoenix, Ariz.
[b]DODG = Dodge City, Kans.   [d]FTWR = Fort Worth, Tex.

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Collectible
Solar Energy
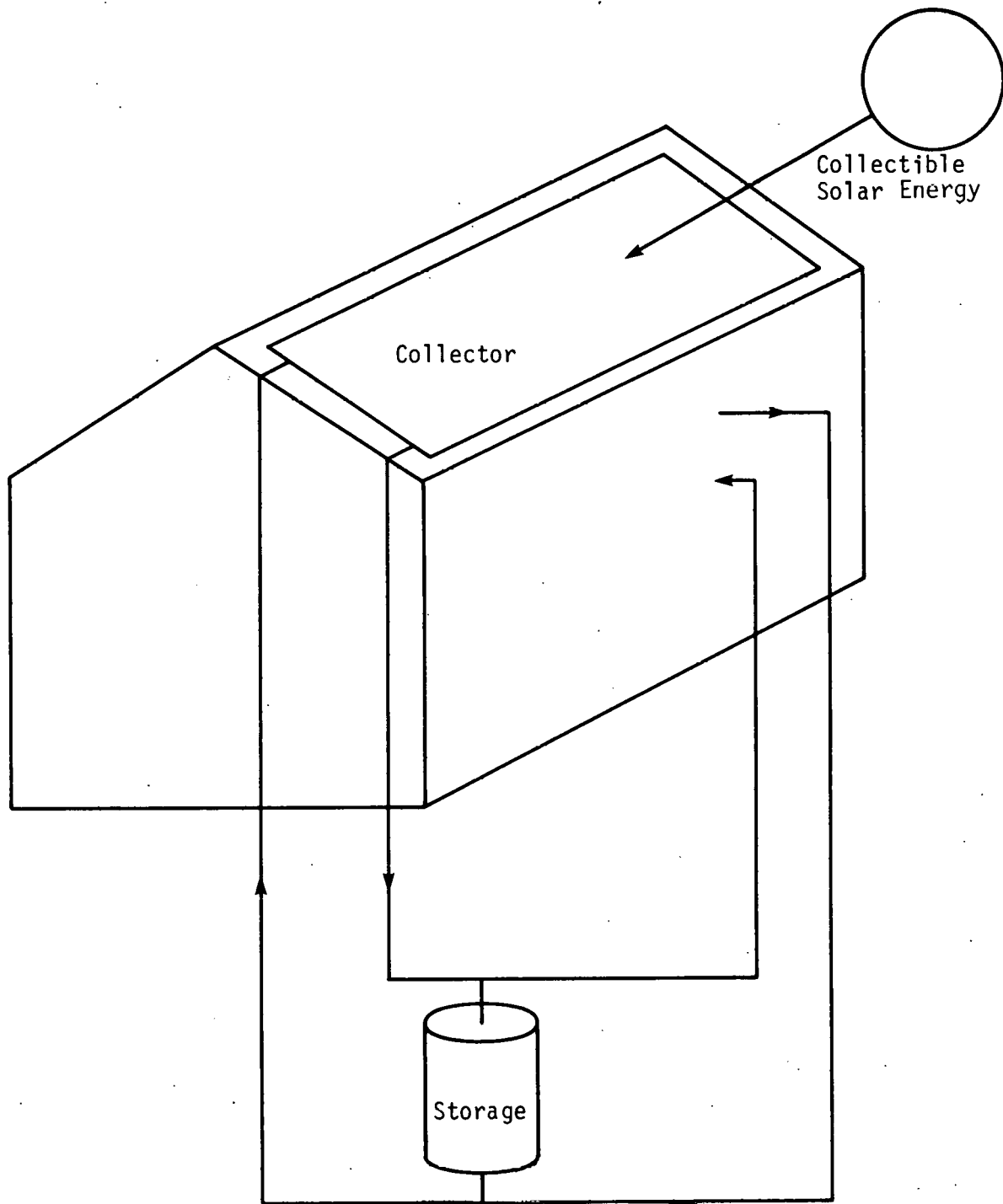
Collector

Storage

Figure 3-23. SOLAR COLLECTOR/STORAGE SYSTEM CONFIGURATION DURING HEATING SEASON
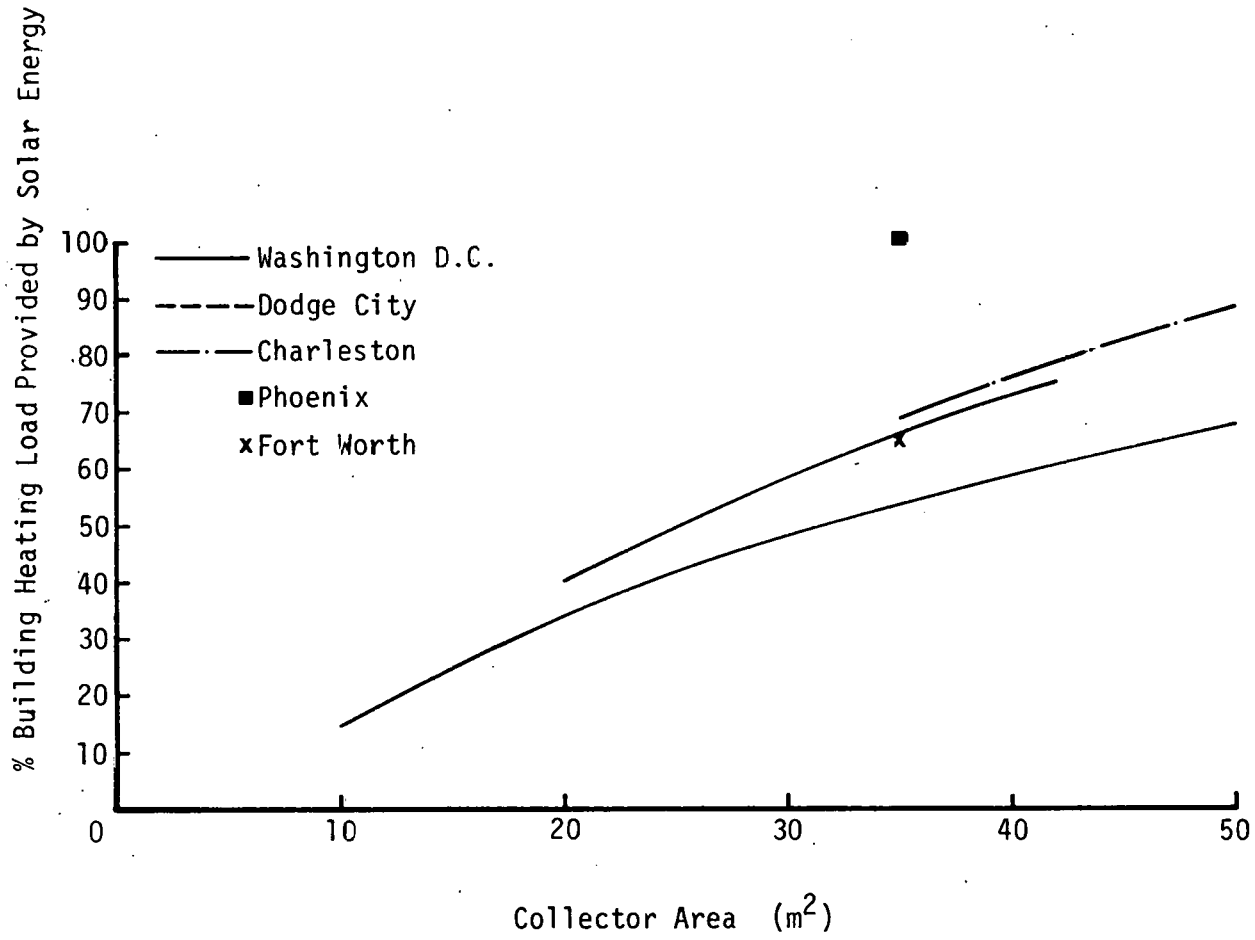
Figure 3-24.   PERFORMANCE OF SOLAR SYSTEM DURING HEATING SEASON

Areas having mild winter months (lower heating loads) generally experience more severe cooling seasons (higher cooling loads) and vice versa. This makes it unlikely that the optimal collector sizes for the two seasons will be identical. If the total loads are not significantly high, then the net fuel saved will not offset collector costs. The implication is that the climate most appropriate for both solar heating and cooling is one experiencing high heating and cooling loads.

For each of the runs listed in Table 3-17, monthly and annual summaries of all heating and cooling loads, capacities, and auxiliary requirements were computed, as well as such parameters as COP, EER, and the percentage of time the conditioned space was within the defined comfort zone. Sample reports are included in Appendix A and the annual performance recaps appear in Table 3-19.

Table 3-18. PERCENTAGE OF HEATING LOAD SUPPLIED BY SOLAR ENERGY

City	Heating Load $(10^{10}J)$	% of Heating Load Supplied by Solar Energy at Collector Area of $35m^2$[a]	Optimum Collector Area for Heating (m^2)[b]	% of Heating Load Supplied by Solar Energy at Optimum Collector Area
Phoenix, Ariz.	0.76	100	14	95[c]
Charleston, S.C	2.60	69	12	33[c]
Fort Worth, Tex.	3.24	65	16	45[c]
Washington, D.C.	4.81	54	18	30[a]
Dodge City, Kans.	5.65	66	26	51[a]

[a]See Figure 3-24
[b]Ref. 27; for an assumed average conventional fuel cost = $10/GJ, collector cost = $110/m², and annual payback of capital = 0.1
[c]F-Chart results

Figures 3-25a, b, and c summarize the effects of desiccant cooler size and collector area on delivered cooling capacity for Washington, D.C. These plots show that the amount of parasitic energy used is relatively unaffected by collector area or desiccant machine size. This is because smaller machines demand less power but operate for longer time periods than larger units.

COP and EER were chosen as measures of performance. A benchmark desiccant system was defined to be of a 4.5-kW nominal capacity and 35-m² collector area. This configuration was simulated in the cities of interest both with and without auxiliary cooling. The single variable that best characterized climatic conditions was the relative humidity corresponding to the design-point wet- and dry-bulb temperatures for each city [6]. Plots of COP and EER as functions of this design-point relative humidity (Figures 3-26a and b) indicate that desiccant system performance peaks near 50% and that, therefore, Washington, D.C., is the optimum climate of the five cities studied.

TABLE 3-19. ANNUAL DESICCANT SYSTEM PERFORMANCE
(all energy units in joules)

Run No.	Building Load			Cooling Capacity Supplied		Auxiliary Supplied		% Bldg. Load by Solar Energy		Parasitic Energy	COP	EER	Duty Cycle	% in Comfort
	Sensible Cooling	Latent Cooling	Heat	Sensible	Latent	Heat	Cooling	Heat	Cooling					
WASH 1	1.28 E10	6.74 E9	4.81 E10	1.32 E10	6.54 E9	2.19 E10	--	54	--	3.05 E9	0.71	22.07	0.19	59
2	1.29 E10	7.56 E9	4.81 E10	7.91 E9	7.33 E9	2.19 E10	5.39 E9	54	74	2.99 E9	0.57	13.68	0.19	81
3	1.29 E10	6.84 E9	4.81 E10	1.34 E10	6.81 E9	2.19 E10	--	54	--	3.44 E9	0.77	20.07	0.34	57
4	1.29 E10	7.94 E9	4.81 E10	6.17 E9	7.90 E9	2.19 E10	7.23 E9	54	66	2.84 E9	0.61	12.69	0.27	88
5	1.29 E10	6.63 E9	4.81 E10	1.34 E10	6.64 E9	2.19 E10	--	54	--	5.27 E9	0.89	12.99	0.56	49
6	1.29 E10	7.85 E9	4.81 E10	4.04 E9	7.84 E9	2.19 E10	9.39 E9	54	56	4.05 E9	0.62	9.30	0.42	91
7	1.29 E10	8.34 E9	4.81 E10	1.33 E10	8.15 E9	1.56 E10	--	68	--	3.48 E9	0.59	21.08	0.22	72
8	1.29 E10	8.80 E9	4.81 E10	9.77 E9	8.61 E9	1.56 E10	3.57 E9	68	84	3.30 E9	0.54	15.87	0.21	91
9	1.29 E10	8.46 E9	4.81 E10	1.35 E10	8.50 E9	1.56 E10	--	68	--	3.86 E9	0.66	19.45	0.39	70
10	1.29 E10	8.81 E9	4.81 E10	6.90 E9	8.82 E9	1.56 E10	6.52 E9	68	71	3.01 E9	0.58	13.53	0.30	96
12	1.29 E10	8.43 E9	4.81 E10	4.26 E9	8.45 E9	1.56 E10	9.17 E9	68	58	4.02 E9	0.60	9.71	0.42	98
13	1.29 E10	5.32 E9	4.81 E10	4.48 E9	5.14 E9	3.17 E10	8.80 E9	34	52	2.27 E9	0.62	10.87	0.14	65
14	1.29 E10	6.12 E9	4.81 E10	4.17 E9	6.05 E9	3.17 E10	9.15 E9	34	53	2.37 E9	0.64	10.98	0.21	72
15	1.29 E10	6.33 E9	4.80 E10	3.17 E9	6.29 E9	3.17 E10	1.02 E10	34	48	3.53 E9	0.66	8.82	0.35	76
16	1.30 E10	8.25 E9	4.81 E10	8.98 E9	8.01 E9	1.88 E10	4.34 E9	61	80	3.18 E9	0.56	14.82	0.20	86
17	1.29 E10	8.41 E9	4.81 E10	6.57 E9	8.41 E9	1.88 E10	6.83 E9	61	69	2.96 E9	0.59	13.07	0.29	93
18	1.29 E10	8.25 E9	4.81 E10	4.17 E9	8.26 E9	1.88 E10	9.25 E9	61	57	4.11 E9	0.51	9.48	0.43	96
19	1.29 E10	9.25 E9	4.81 E10	9.62 E9	6.58 E9	1.88 E10	6.15 E9	61	72	3.08 E9	0.56	13.76	0.19	100
20	1.28 E10	8.49 E9	4.81 E10	5.48 E9	2.80 E9	3.17 E10	1.32 E10	34	39	2.12 E9	0.58	9.92	0.12	100
21	1.29 E10	9.02 E9	4.81 E10	8.80 E9	5.59 E9	2.19 E10	7.69 E9	55	65	2.90 E9	0.57	12.62	0.18	100
22	1.29 E10	9.48 E9	4.82 E10	1.02 E10	7.57 E9	1.56 E10	4.80 E9	68	79	3.19 E9	0.54	15.11	0.20	100

TABLE 3-19. ANNUAL DESICCANT SYSTEM PERFORMANCE (Continued)
(all energy units in joules)

Run No.	Building Load Sensible Cooling	Latent Cooling	Heat	Cooling Capacity Supplied Sensible	Latent	Auxiliary Supplied Heat	Cooling	% Bldg. Load by Solar Energy Heat	Cooling	Parasitic Energy	COP	EER	Duty Cycle	% in Comfort
WASH 23	1.27 E10	7.96 E9	4.81 E10	1.20 E9	6.34 E8	4.00 E10	1.91 E10	14	9	6.44 E8	0.63	8.63	0.03	100
24	1.27 E10	8.13 E9	4.81 E10	2.85 E9	1.58 E9	4.01 E10	1.67 E10	14	21	1.52 E9	0.61	8.79	0.13	100
25	1.28 E10	8.80 E9	4.81 E10	6.53 E9	6.58 E9	2.19 E10	8.89 E9	55	60	2.64 E9	0.60	12.11	0.25	100
26	1.28 E10	8.97 E9	4.81 E10	6.87 E9	7.48 E9	1.56 E10	7.52 E9	68	66	2.67 E9	0.58	13.37	0.26	100
27	1.28 E10	8.14 E9	4.81 E10	2.66 E9	2.05 E9	4.02 E10	1.64 E10	14	22	2.35 E9	0.63	8.09	0.21	100
28	1.28 E10	8.43 E9	4.81 E10	3.82 E9	5.24 E9	2.19 E10	1.24 E10	55	42	2.97 E9	0.59	9.24	0.30	100
29	1.28 E10	8.47 E9	4.81 E10	3.89 E9	5.64 E9	1.56 E10	1.20 E10	68	44	2.93 E9	0.57	9.51	0.30	100
DODG 1	1.16 E10	3.33 E9	5.65 E10	1.20 E10	3.59 E9	1.91 E10	—	66	—	3.07 E9	0.56	17.38	0.18	82
2	1.14 E10	3.46 E9	5.66 E10	1.21 E10	3.78 E9	1.91 E10	—	66	—	2.91 E9	0.59	18.69	0.32	76
3	1.14 E10	3.18 E9	5.66 E10	6.01 E9	3.48 E9	1.91 E10	6.01 E9	66	61	1.86 E9	0.53	12.42	0.18	99
4	1.13 E10	3.25 E9	5.66 E10	1.19 E10	3.49 E9	1.91 E10	—	66	—	4.06 E9	0.69	12.95	0.47	61
5	1.14 E10	2.77 E9	5.66 E10	3.40 E9	2.99 E9	1.91 E10	8.60 E9	66	43	2.20 E9	0.52	9.08	0.23	99
6	1.16 E10	2.27 E9	5.64 E10	1.18 E10	2.38 E9	3.36 E10	—	40	—	2.26 E9	0.79	21.45	0.16	60
7	1.14 E10	2.38 E9	5.65 E10	1.20 E10	2.62 E9	3.36 E10	—	40	—	2.32 E9	0.81	21.53	0.25	55
8	1.13 E10	2.53 E9	5.65 E10	1.19 E10	2.72 E9	3.36 E10	—	40	—	3.89 E9	0.88	12.83	0.48	56
9	1.16 E10	3.55 E9	5.65 E10	1.21 E10	3.84 E9	1.42 E10	—	75	—	2.51 E9	0.52	21.66	0.18	94
10	1.15 E10	3.60 E9	5.66 E10	1.21 E10	3.92 E9	1.42 E10	—	75	—	2.63 E9	0.56	20.86	0.31	80
11	1.13 E10	3.32 E9	5.65 E10	1.19 E10	3.57 E9	1.42 E10	—	75	—	3.83 E9	0.67	13.80	0.46	61
12	1.14 E10	3.18 E9	5.66 E10	6.00 E9	3.39 E9	1.91 E10	6.08 E9	66	61	1.65 E9	0.53	12.93	0.18	99

TABLE 3-19. ANNUAL DESICCANT SYSTEM PERFORMANCE (Concluded)
(all energy units in joules)

Run No.	Building Load			Cooling Capacity Supplied		Auxiliary Supplied		% Bldg. Load by Solar Energy		Parasitic Energy	COP	EER	Duty Cycle	% in Comfort
	Sensible Cooling	Latent Cooling	Heat	Sensible	Latent	Heat	Cooling	Heat	Cooling					
PHNX 1	5.13 E10	1.39 E9	7.57 E9	5.15 E10	1.47 E9	0	--	100	--	6.06 E9	0.72	29.87	0.29	52
2	5.13 E10	1.94 E9	7.55 E9	3.00 E10	1.98 E9	0	2.12 E10	100	60	5.32 E9	0.49	13.17	0.25	99
4	5.11 E10	1.85 E9	7.52 E9	1.89 E10	1.90 E9	0	3.26 E10	100	39	3.89 E9	0.43	10.77	0.28	99
7	5.11 E10	1.89 E9	7.52 E9	1.89 E10	1.87 E9	0	3.27 E10	100	39	3.89 E9	0.43	10.76	0.28	99
CHAR 8	1.21 E10	1.21 E10	2.60 E10	9.45 E9	1.24 E10	2.86 E9	3.42 E9	89	86	5.07 E9	0.51	13.41	0.24	86
10	1.20 E10	1.15 E10	2.60 E10	1.28 E10	1.19 E10	2.86 E9	--	89	--	5.20 E9	0.54	16.20	0.25	72
11	1.19 E10	1.11 E10	2.60 E10	5.91 E9	1.15 E10	8.10 E9	6.87 E9	69	72	4.10 E9	0.57	12.09	0.33	81
12	1.19 E10	9.66 E9	2.59 E10	1.27 E10	1.00 E10	8.10 E9	--	69	--	4.54 E9	0.73	17.08	0.37	51
13	1.22 E10	1.15 E10	2.60 E10	1.01 E10	1.16 E10	2.86 E9	2.84 E9	89	88	7.25 E9	0.47	9.99	0.19	80
14	1.17 E10	1.27 E10	2.61 E10	6.38 E9	9.19 E9	8.10 E9	9.92 E9	69	61	3.24 E9	0.55	12.07	0.29	100
FTWR 1	2.39 E10	9.39 E9	3.24 E10	2.41 E10	9.27 E9	1.13 E10	--	65	--	5.28 E9	0.71	21.57	0.25	48
2	2.38 E10	9.72 E9	3.24 E10	2.45 E10	9.83 E9	1.13 E10	--	65	--	5.88 E9	0.77	19.90	0.49	46
3	2.38 E10	1.09 E10	3.24 E10	1.07 E10	1.10 E10	1.13 E10	1.36 E10	65	61	4.40 E9	0.55	12.25	0.35	95
5	2.39 E10	1.05 E10	3.24 E10	6.59 E9	1.05 E10	1.13 E10	1.78 E10	65	49	5.49 E9	0.61	9.44	0.48	98
6	2.37 E10	1.11 E10	3.25 E10	1.08 E10	9.63 E9	1.13 E10	1.49 E10	65	58	3.73 E9	0.59	12.45	0.31	100
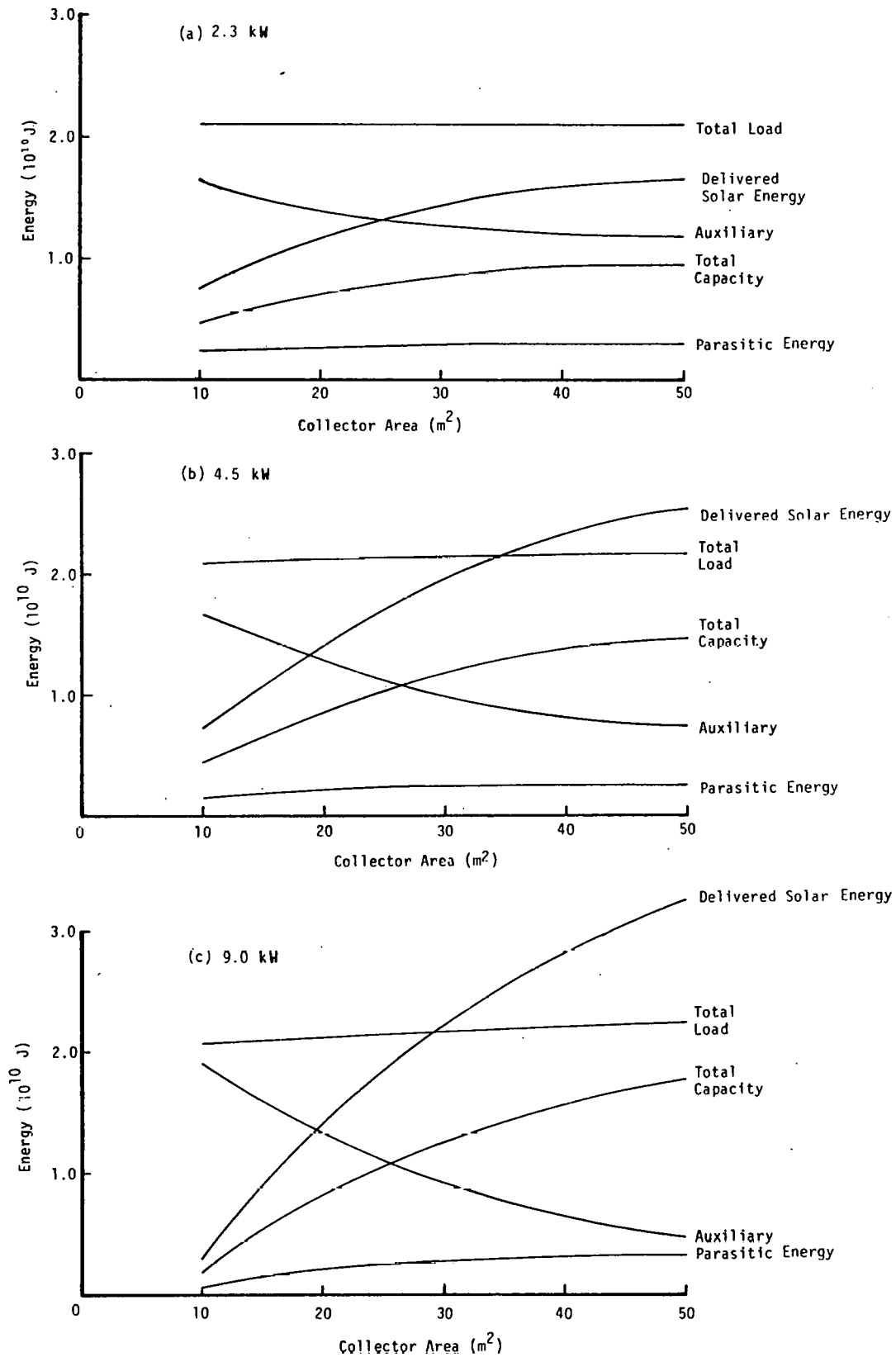
Figure 3-25. SIMULATED ANNUAL PERFORMANCES OF SOLAR DESICCANT COOLERS WITH SENSIBLE AND LATENT AUXILIARY COOLERS IN WASHINGTON, D.C.

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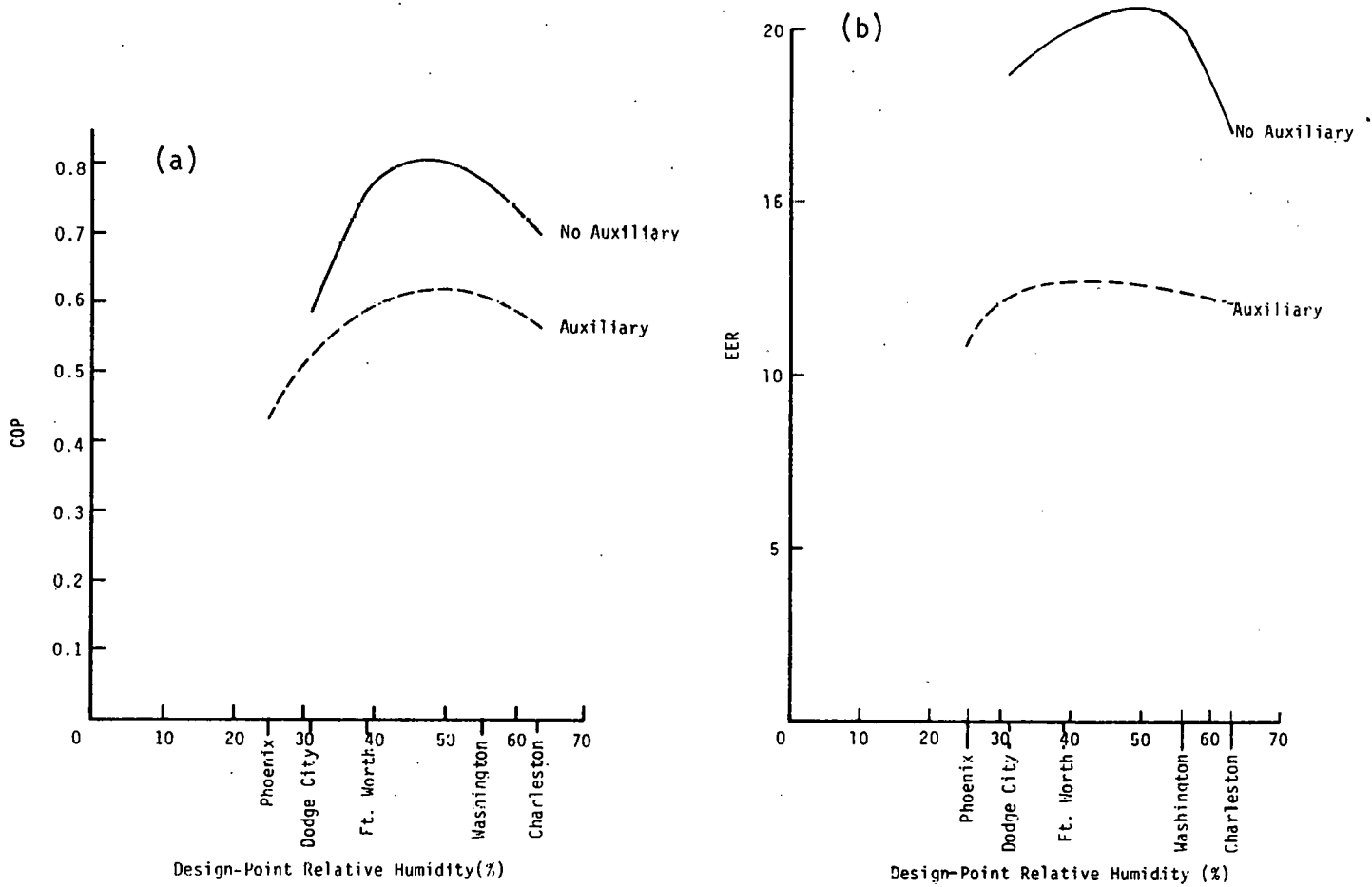
Figure 3-26. EFFECT OF DESIGN-POINT RELATIVE HUMIDITY ON (a) COP AND (b) EER - 4.5-KW COOLER WITH 35-M^2 COLLECTOR, WITH AND WITHOUT AUXILIARY COOLING

The steady-state analysis (Section 3.6.2) showed that the modeled desiccant system was preferentially a latent rather than sensible cooler. For the dynamic analysis, runs in which auxiliary sensible cooling was provided were chosen to see how well the latent component of the load could be met by the desiccant machine. The percentage of time which the house spent in zone 3 (out of comfort, too humid) in each city was again plotted against design-point relative humidity (Figure 3-27). General agreement was found with the steady-state sensitivity curve of capacity versus ambient humidity; namely, as outside humidity increased, system performance decreased and a greater amount of time was spent out of comfort.

To further illustrate the effect of climate upon system operation, the variation of the percentage of the latent cooling load that was provided by solar energy, as a function of the percentage of the total latent cooling load, was studied. The percentage of the latent cooling supplied by solar energy was calculated as follows:

$$\frac{\text{Latent cooling capacity supplied by solar energy}}{\text{Latent cooling load}} \times 100\%.$$

The results are plotted in Figure 3-28 and they agree with the steady-state performance. As the latent load became more appreciable, the cooling capacity dropped and the percentage of the latent load supplied by solar energy decreased. Thus, the response of the modeled desiccant system was more sensitive to latent than sensible conditions.

As explained in Section 3.7.2, whenever the collector/storage system could not deliver 45°C, the desiccant system did not operate. The percentage of time that cooling capacity was required but could not be provided due to insufficient outlet temperature was recorded for each simulation run (see Table 3-20). Inability to operate was plotted in Figure 3-29 as a function of collector area and system capacity for Washington, D.C., for two auxiliary cooling strategies. Proper matching of collector area with desiccant system size minimized the percentage of time in which the system was unable to operate. Two conclusions can then be made if it is desired that the desiccant system operate 90% of the time that it is needed. First, a considerable increase in collector area (roughly 15 m^2) is required to maintain a 90% availability (10% downtime) if only sensible auxiliary cooling is provided rather than both sensible and latent auxiliary. This implies, on the basis of minimizing collector area, that the strategy of providing sensible and latent auxiliary cooling is more favorable than supplying sensible auxiliary only. Second, the appropriate choice of collector size versus system capacity is 20 m^2 at 2.3 kW, 35 m^2 at 4.5 kW, and 50 m^2 at 9.0 kW.

Additional support for the above combinations of collector size and system capacity was obtained from frequency histograms of collector flowrate. As mentioned in Section 3.7.1, if the collectible solar energy resulted in an outlet temperature greater than 85°C, the collector flowrate $\dot{m}_C$ was increased until exactly 85°C was available. The percentage of time that $\dot{m}_C$ exceeded the desiccant system flowrate $\dot{m}_S$ was therefore a measure of how oversized the collector area was relative to desiccant capacity. These percentages are presented in Table 3-20 and are displayed as a function of collector area in Figure 3-30. When it is assumed that 30% represents the maximum percentage of time that collector flowrate should exceed desiccant flowrate, agreement is

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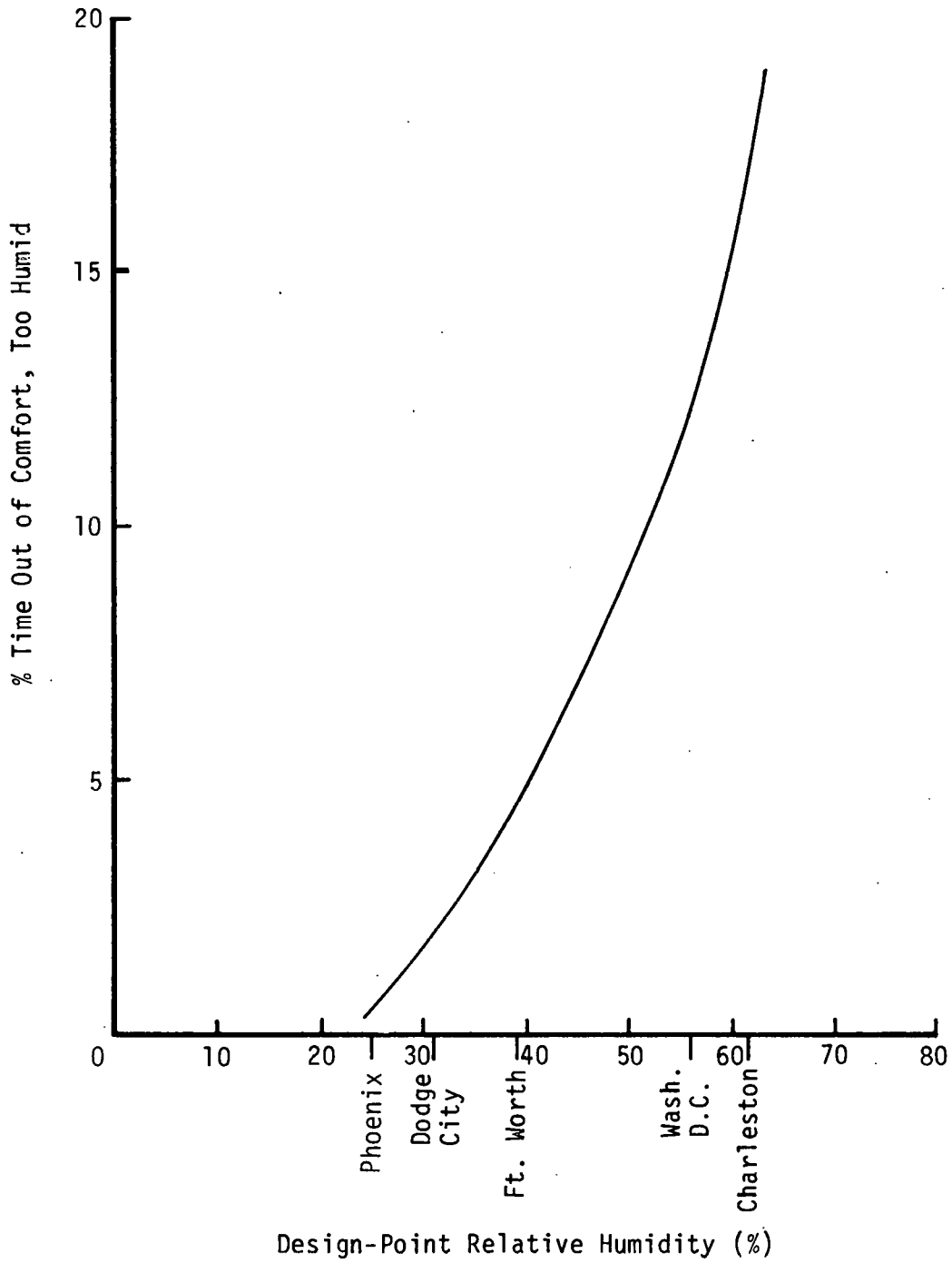
Figure 3-27. EFFECT OF DESIGN-POINT RELATIVE HUMIDITY ON COMFORT - 4.5-KW COOLER WITH 35-M^2 COLLECTOR

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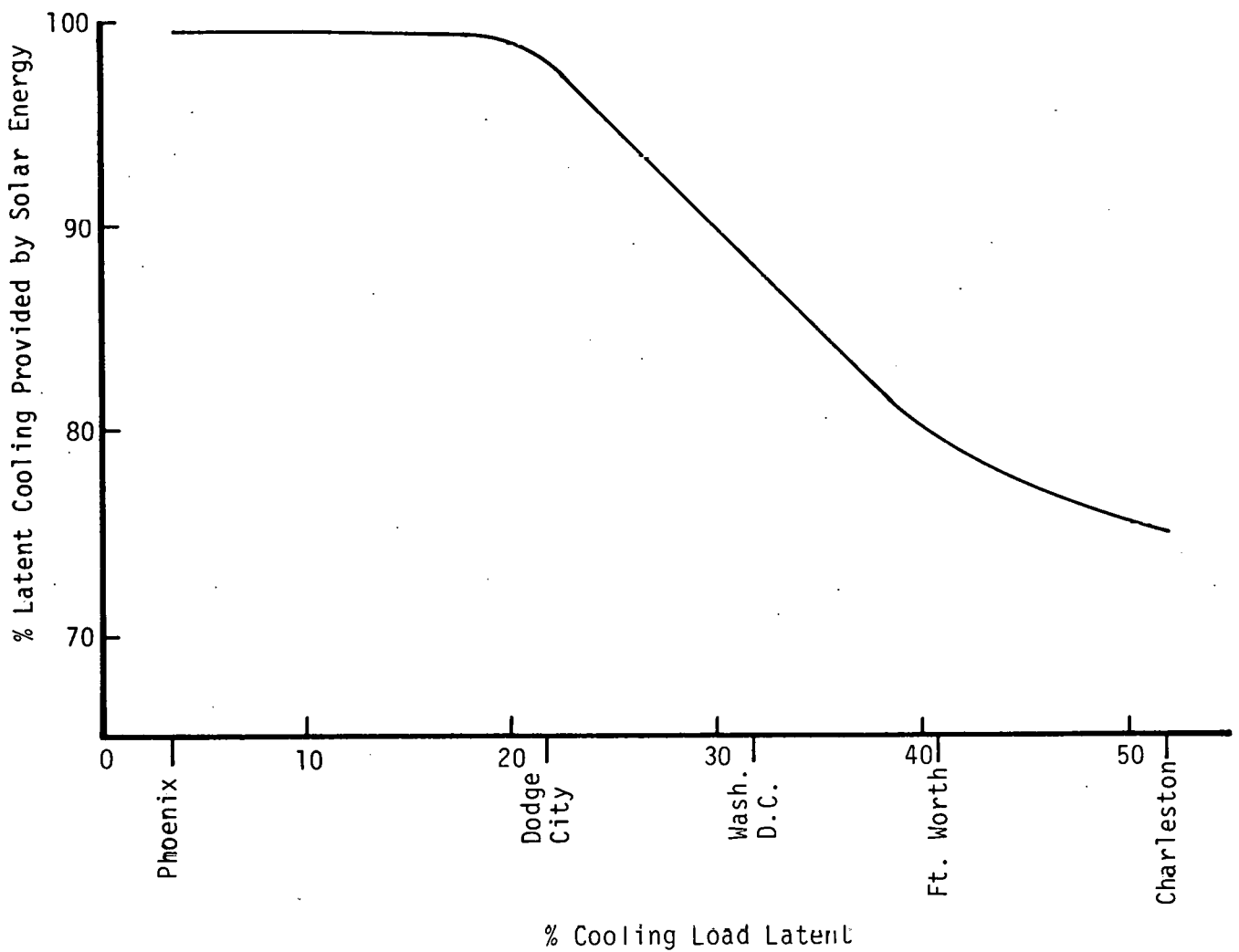
Figure 3-28.   PERCENTAGE OF LATENT COOLING LOAD PROVIDED BY SOLAR ENERGY VS. PERCENTAGE OF TOTAL COOLING LOAD THAT IS LATENT - 4.5-KW COOLER WITH 35-M^2 COLLECTOR

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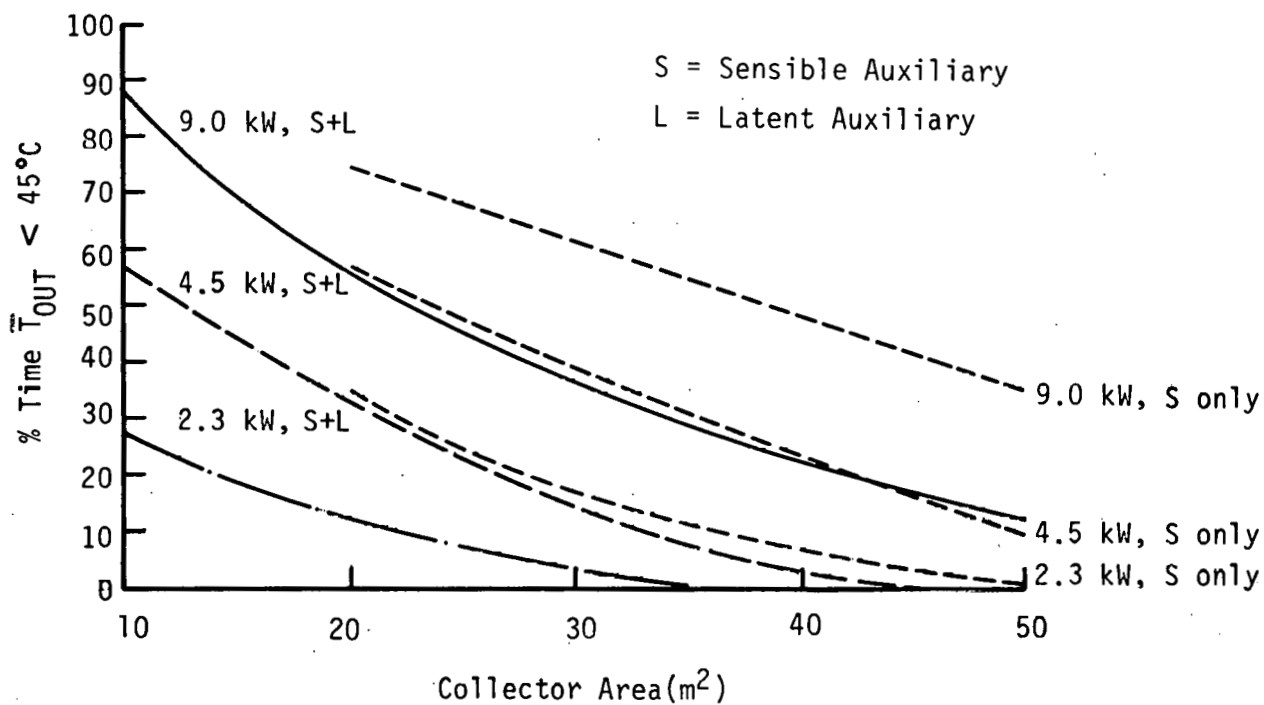
Figure 3-29. PERCENTAGE OF TIME SYSTEM CANNOT PROVIDE COOLING BECAUSE OF INSUFFICIENT REGENERATION TEMPERATURE $_{OUT}$ (WASHINGTON, D.C.)

Table 3-20. SYSTEM PERFORMANCE PARAMETERS

Run No.	Average Temperature Available from Solar Collector/ Storage (°C)	% of Time Cooling Capacity Required and $\bar{T}_{OUT}$< 45°C (machine off)	% of Time Auxiliary Required	% of Time Collector Flow- rate > System Flowrate
WASH 1	57.0	63.0	0.0	0.0
WASH 2	58.5	54.4	12.4	0.0
WASH 3	60.7	38.5	0.0	37.9
WASH 4	65.1	31.2	14.0	38.3
WASH 5	67.0	9.4	0.0	57.9
WASH 6	69.8	10.0	15.6	59.0
WASH 7	56.7	47.3	0.0	6.0
WASH 8	64.5	35.3	9.6	5.8
WASH 9	66.3	16.3	0.0	55.5
WASH 10	70.6	9.7	12.7	58.8
WASH 12	74.4	1.0	15.3	69.7
WASH 13	49.3	75.2	15.8	0.0
WASH 14	56.7	58.7	16.1	1.6
WASH 15	61.6	35.7	16.7	29.4
WASH 16	61.6	45.1	10.7	0.5
WASH 17	67.9	19.4	13.4	50.4
WASH 18	72.2	3.2	15.4	64.2
WASH 19	62.7	20.6	24.5	0.7
WASH 20	50.5	55 5	28.4	0.0
WASH 21	59.7	29.0	25.5	0.0
WASH 22	65.7	12.2	23.0	6.1
WASH 23	42.8	89.3	30.7	0.0
WASH 24	49.3	56.9	30.0	0.0
WASH 25	67.6	6.6	27.3	39.0
WASH 26	72.4	1.0	26.4	60.6
WASH 27	57.6	29.3	30.7	0.0
WASH 28	74.0	0.1	30.0	64.3
WASH 29	76.7	0.0	29.9	73.6
DODG 1	65.9	36.4	0.0	0.4
DODG 2	70.1	5.9	0.0	55.3
DODG 3	75.1	0.0	11.1	65.2
DODG 4	76.3	0.0	0.0	73.1
DODG 5	78.4	0.0	13.5	74.6
DODG 6	54.3	65.5	0.0	0.0
DODC 7	61.3	49.9	0.0	15.3
DODG 8	66.9	7.2	0.0	50.6
DODG 9	69.6	7.5	0.0	4.4
DODG 10	73.5	0.3	0.0	66.0
DODG 11	78.0	0.0	0.0	76.5
DODG 12	75.2	0.0	18.1	66.0
CHAR 8	62.5	38.2	8.7	0.0
CHAR 10	60.8	44.8	0.0	6.6
CHAR 11	61.9	33.9	12.8	38.0
CHAR 12	58.6	39.5	0.0	38.0
CHAR 13	57.7	54.3	7.5	0.0
CHAR 14	66.5	6.9	31.6	40.1
FTWR 1	56.1	58.6	0.0	0.0
FTWR 2	60.4	23.1	0.0	49.2
FTWR 3	68.4	10.9	19.1	52.9
FTWR 5	74.3	2.2	22.2	69.6
FTWR 6	70.2	1.6	32.1	56.2
PHNX 1	60.1	46.8	0.0	0.2
PHNX 2	65.6	7.3	20.5	0.1
PHNX 4	74.0	0.7	24.2	67.5
PHNX 7	74.0	0.3	28.9	67.5

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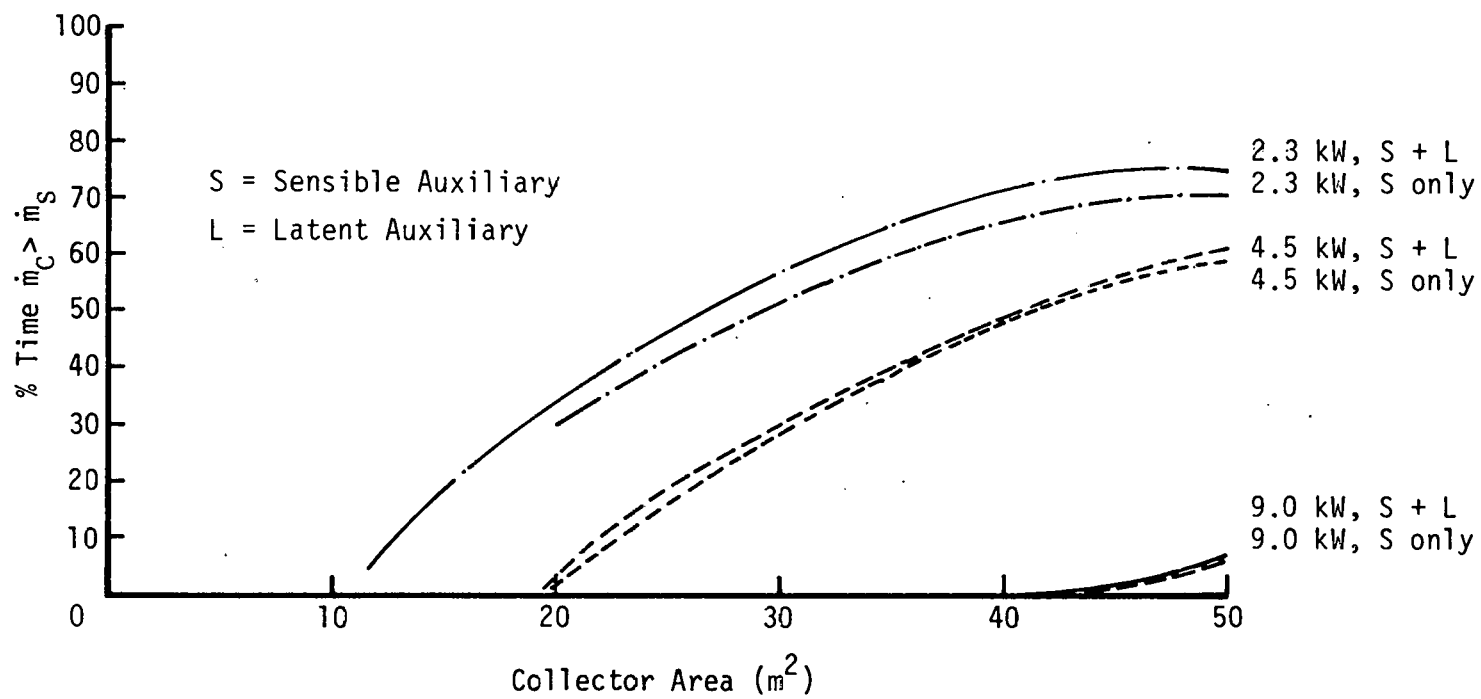
Figure 3-30. PERCENTAGE OF TIME COLLECTOR FLOWRATE $\dot{m}_C$ EXCEEDS DESICCANT SYSTEM FLOWRATE $\dot{m}_S$ (WASHINGTON, D.C.)

found with the matching of 20 m^2 with 2.3 kW and 35 m^2 with 4.5 kW. Note that the collector area required by a machine of 9.0-kW capacity is undersized even at 50 m^2.

To determine which of the combinations of collector area and system capacity was most desirable for Washington, D.C., the percentage of the total house cooling load supplied by solar energy was plotted against collector area (Figure 3-31). The capacity of the 9.0-kW system was far too great for small collectors. Usable solar energy was rapidly depleted and the system remained off most of the time. Another way of viewing this is that the average temperature from the collector available for regeneration was low and, thus, the corresponding percentage of time that the collector outlet temperature was less than 45°C (and, hence, the system inoperative) was high (see Figure 3-29). On the other hand, when sufficient collector area was provided, the 9.0-kW machine capacity could meet a greater percentage of the building load than the smaller systems. The percentages for the three system configurations under consideration were as follows:

Collector Area (m^2)	Capacity (kW)	% of Building Load Met
20	2.3	32
35	4.5	60
50	9.0	79

The sensible and latent components of the cooling load exhibited the same general behavior as the total cooling load (see Figures 3-32a and b).

Thus, the larger collector/capacity systems clearly exhibit greatly improved performance over the 20-m^2/2.3-kW combination. This conclusion is further supported by a similar anlaysis of EER versus collector area (see Figure 3-33). For the sensible and latent auxiliary cooling strategy, the EER at 50-m^2 collector area and 9.0-kW capacity was 15.11 and at 35 m^2 and 4.5 kW it was 12.11, whereas at 20 m^2 and 2.3 kW the EER was only 9.51.

When system sizing is viewed from the standpoint of COP, the importance of the dynamic interaction between the desiccant system and the collector/storage unit becomes evident. Figure 3-34 is a plot of system COP as a function of collector area for the system sizes under consideration. At very low collector areas, the 9.0-kW machine is quite oversized, as discussed previously, and hence the system rarely was capable of operating. When it did operate, it was primarily driven directly by solar energy; that is, storage remained depleted. Thus, while the system operated with a low duty cycle (see Table 3-19), high temperatures were available during those times it did operate, resulting in efficient use of supplied energy and a high COP. With large collector areas, more solar energy was collected, some of which could be stored at temperatures above 45°C (the lower level temperature control). This storage was depleted a smaller fraction of time than in the case of the smaller collector areas, and the desiccant system could operate at the correspondingly lower temperature range, resulting in a smaller COP. This behavior could not have been predicted from the steady-state analysis; it can only be understood in terms of the dynamic interaction of the desiccant cooler with the solar collector/storage system outlet temperature.

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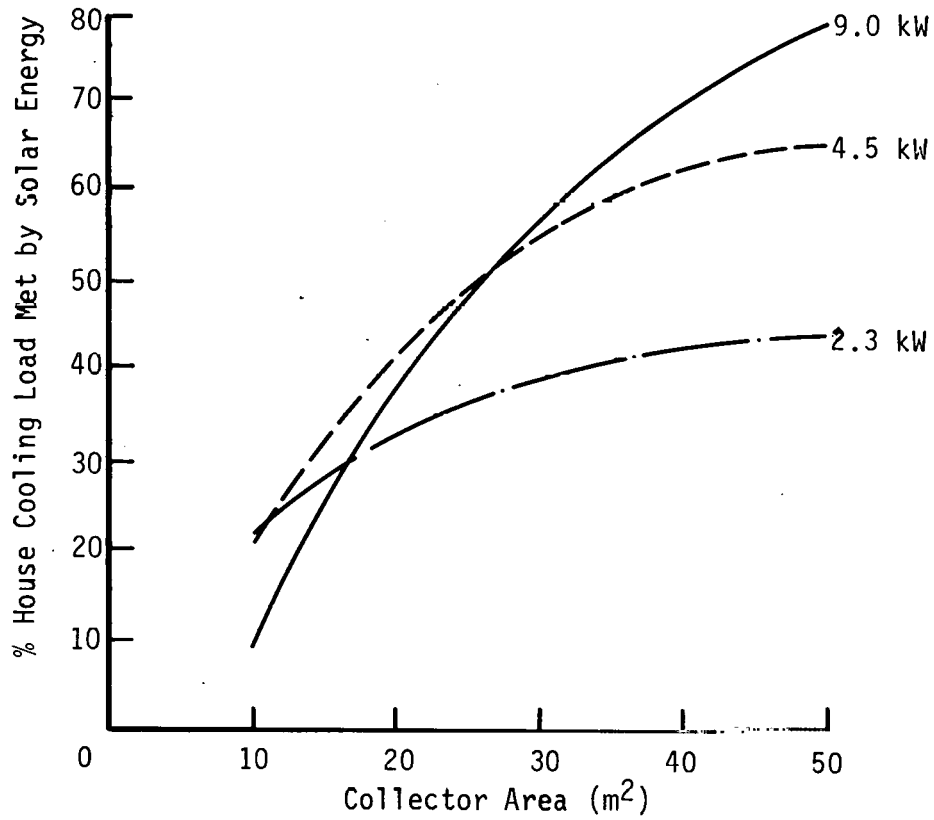
Figure 3-31.   PERCENTAGE OF ANNUAL COOLING LOAD MET BY SOLAR ENERGY (WASHINGTON, D.C.;
AUXILIARY COOLING SUPPLIED)

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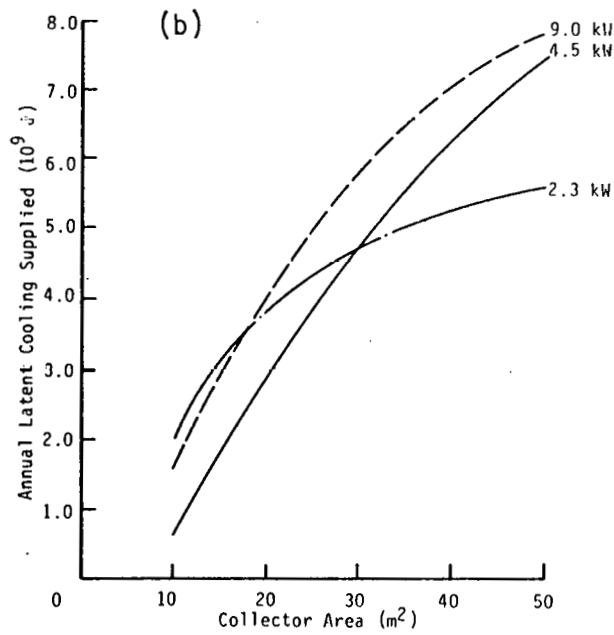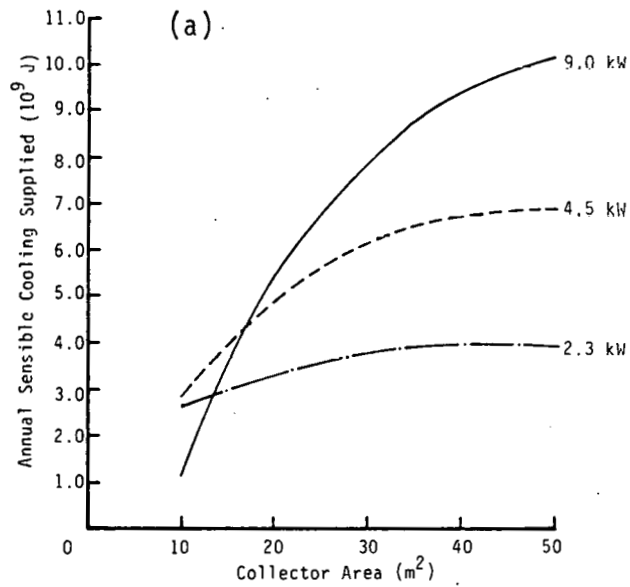
Figure 3-32. ANNUAL COOLING SUPPLIED BY SOLAR DESICCANT COOLER (WASHINGTON, D.C.; AUXILIARY COOLING SUPPLIED): (a) SENSIBLE COOLING AND (b) LATENT COOLING

S = Sensible Auxiliary

L = Latent Auxiliary

9.0 kW, S + L

4.5 kW, S + L

2.3 kW, S + L

EER
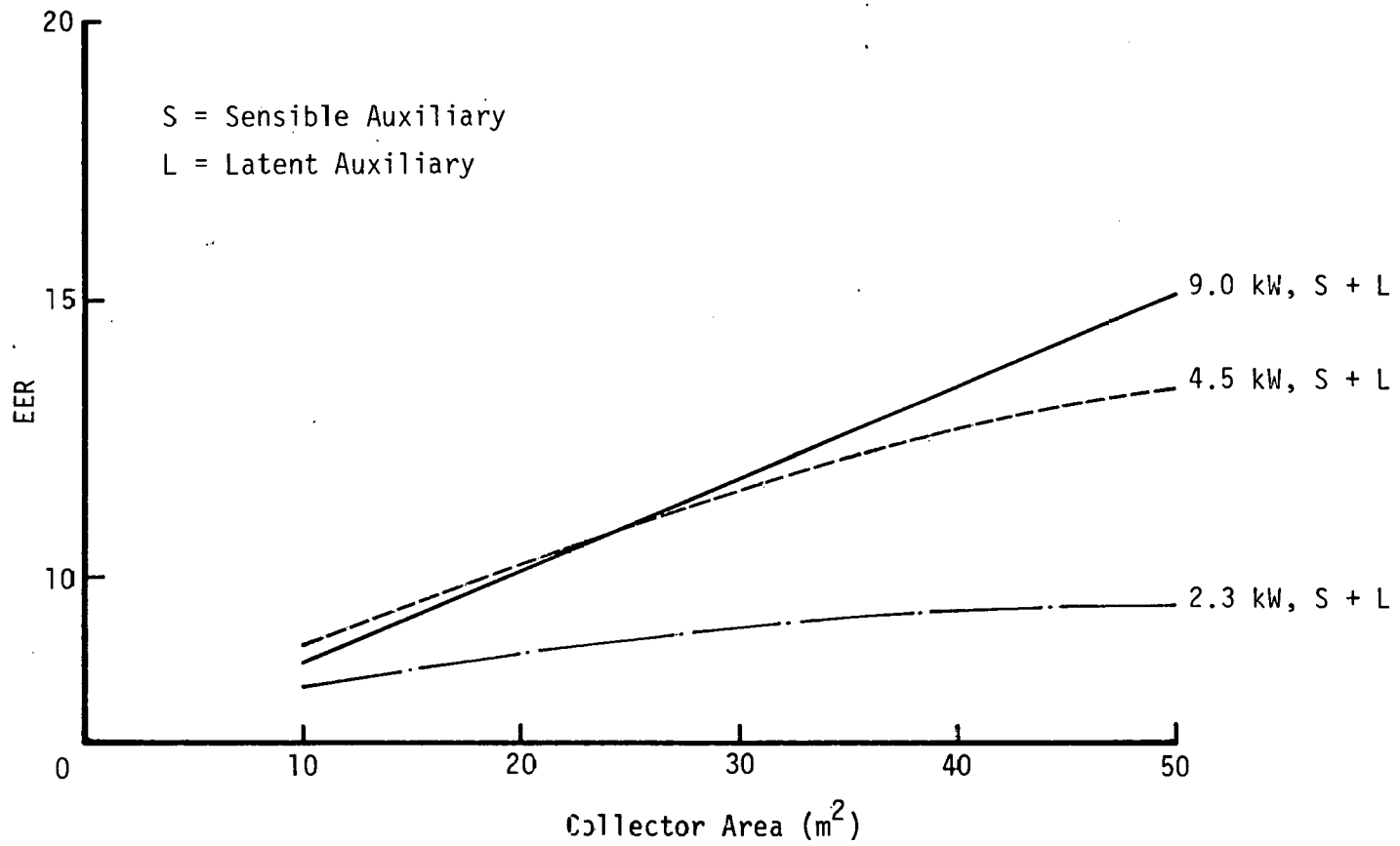
Collector Area (m^2)

Figure 3-33. EER VS. COLLECTOR AREA (WASHINGTON, D.C.)
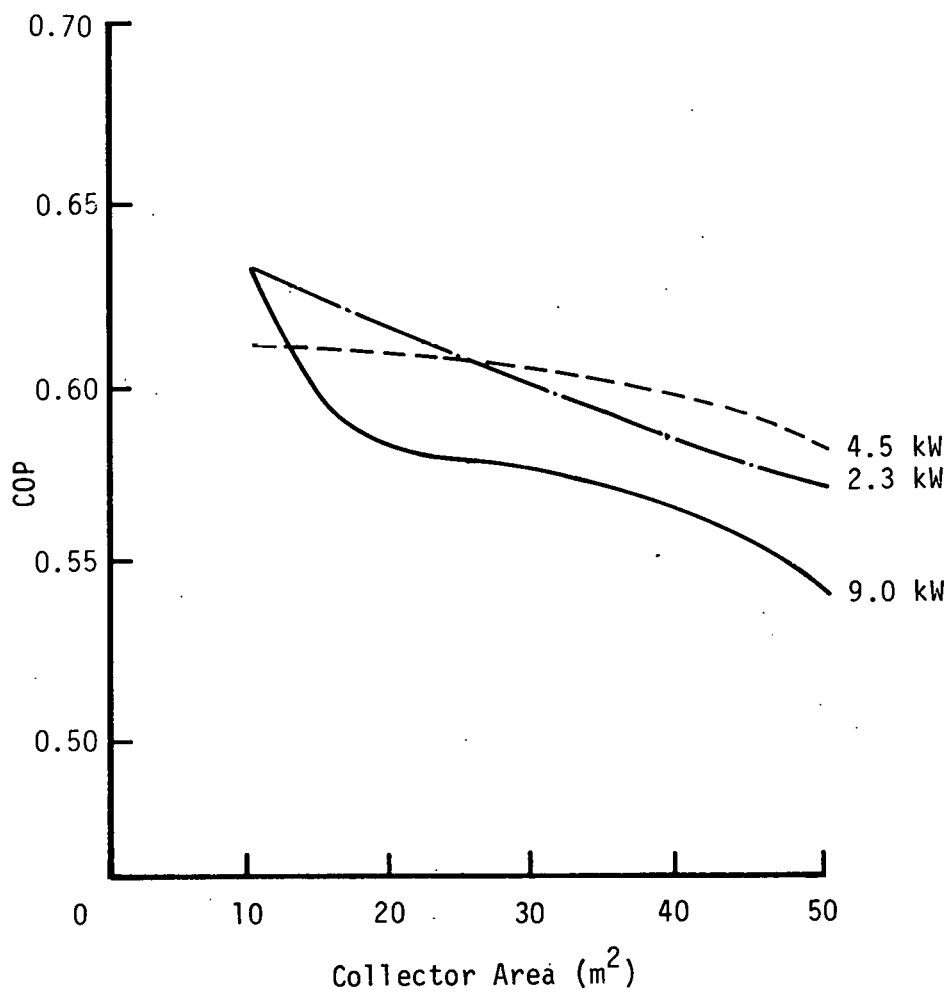
Figure 3-34. COP VS. COLLECTOR AREA (WASHINGTON, D.C.)

This suggests that a more meaningful characterizing variable for the desiccant cooler (rather than COP or collector area) is the average temperature available from the collector/storage unit (i.e., regeneration temperature). Such information is available from frequency histograms of the outlet temperature from the collector/storage system and is compiled in Table 3-20. The translation from collector area to average outlet temperature $\overline{T}_{OUT}$ is depicted in Figure 3-35.

Figure 3-36 shows average outlet temperature plotted versus COP. The reason for the decreasing COP as $\overline{T}_{OUT}$ increases is similar to the argument given in the steady-state analysis. At high regeneration (outlet) temperatures, the supplied energy increases more rapidly than the cooling capacity. This appears to be an inherent characteristic of the desiccant systems.

To further illustrate the effect on system parameters of increasing collector outlet temperatures, a plot of $\overline{T}_{OUT}$ versus EER for Washington, D.C., is presented in Figure 3-37. As discussed earlier, parasitic power requirements are essentially independent of collector area and machine size. Therefore, since a larger machine provides a larger fraction of the total house cooling load, as shown in Figure 3-38, the cooling capacity supplied per unit of parasitic power (i.e., EER) is greater for the systems of larger size.

The smaller machine with 20-m^2 collector area and 2.3-kW desiccant cooler had the most favorable COP (see Figure 3-36), but the objective of the present control strategy was to maximize capacity and machine off-time and not to optimize system COP. For this reason, the results presented in Figures 3-31 and 3-33 support the selection of a 9.0-kW/50-m^2 or a 4.5-kW/35-m^2 system, with both sensible and latent auxiliary cooling supplied, as the most suitable system for Washington, D.C. For the 4.5-kW cooler, a 35-m^2 collector area is most favorable since the incremental increase in cooling capacity and EER does not justify the increase from 35- to 50-m^2 collector area. As previously discussed, a 50-m^2 collector area is oversized for a 4.5-kW desiccant cooler. The 9-kW cooler performs best with a 50-m^2 collector area.
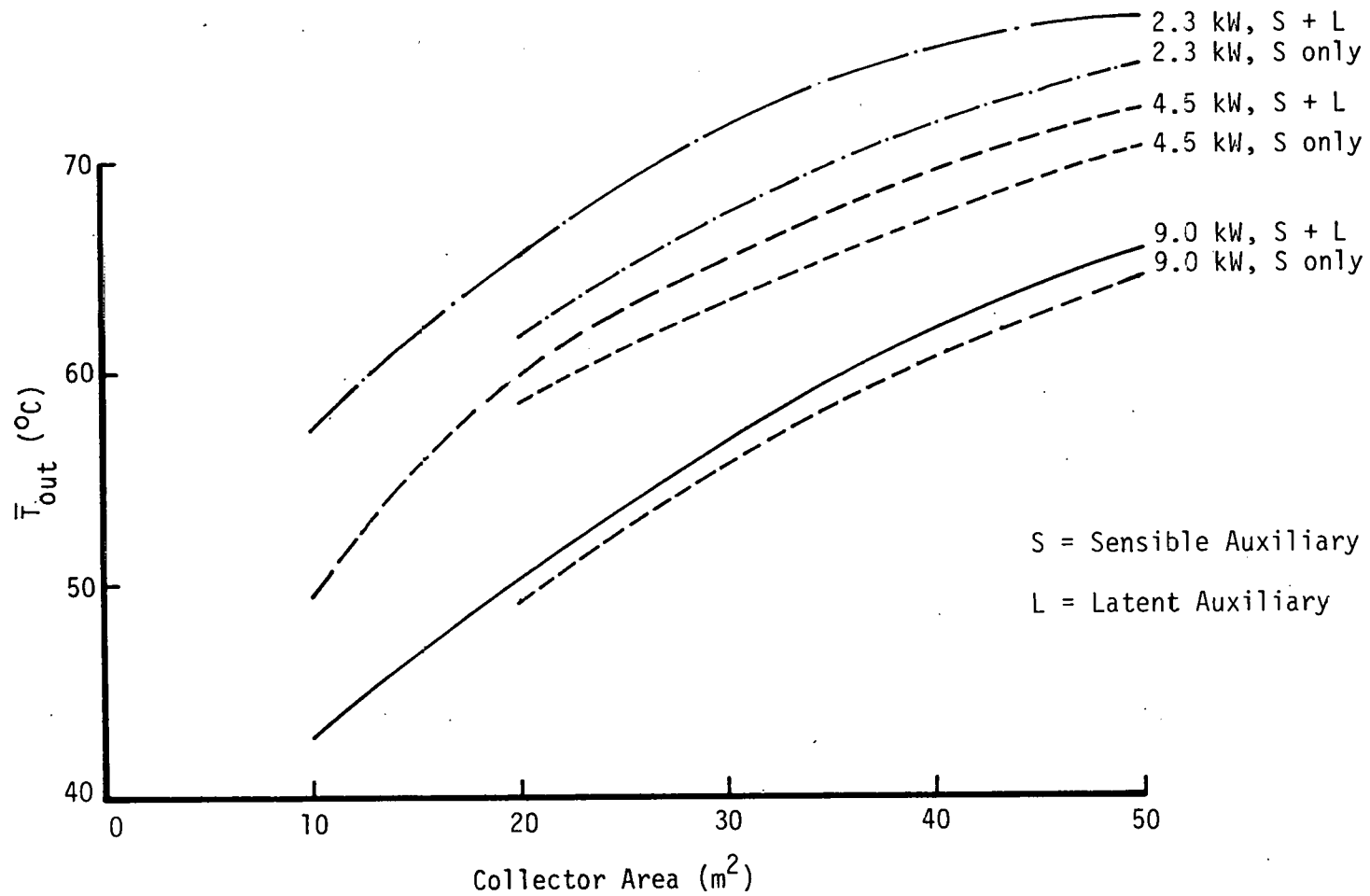
Figure 3-35. REGENERATION TEMPERATURE $\overline{T}_{OUT}$ VS. COLLECTOR AREA (WASHINGTON, D.C.)

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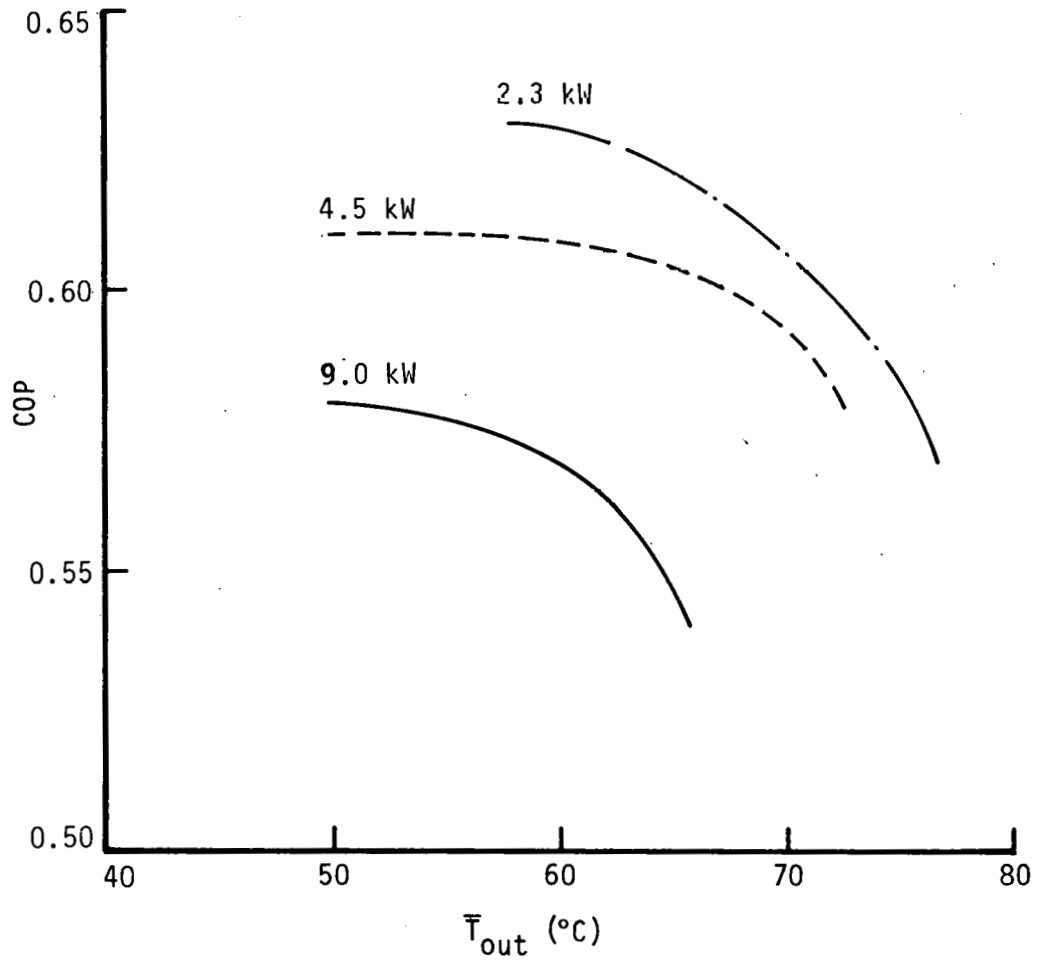
Figure 3-36. COP VS. REGENERATION TEMPERATURE $\overline{T}_{OUT}$ (WASHINGTON, D.C.)
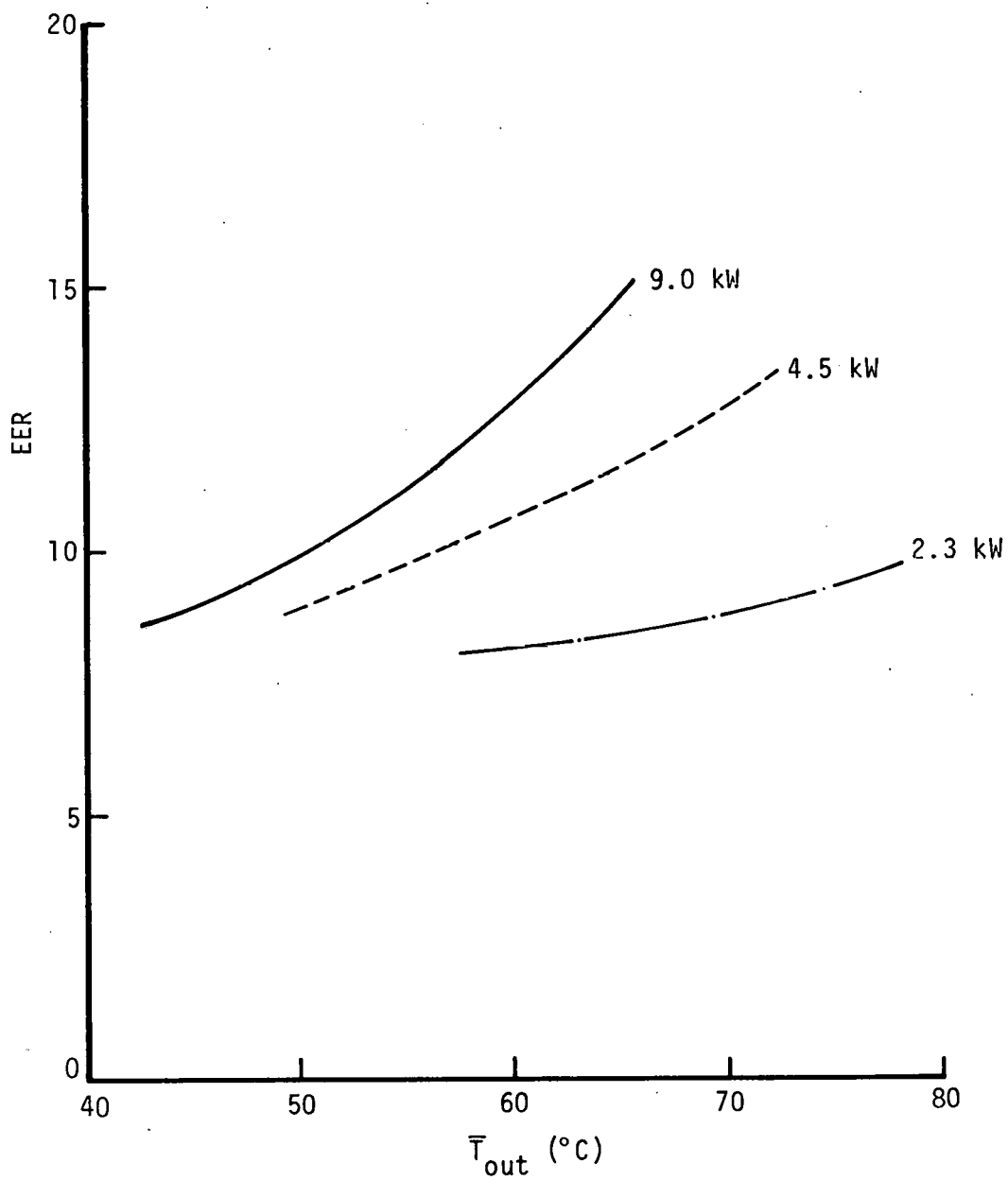
Figure 3-37. EER VS. REGENERATION TEMPERATURE $\overline{T}_{OUT}$ (WASHINGTON, D.C.)
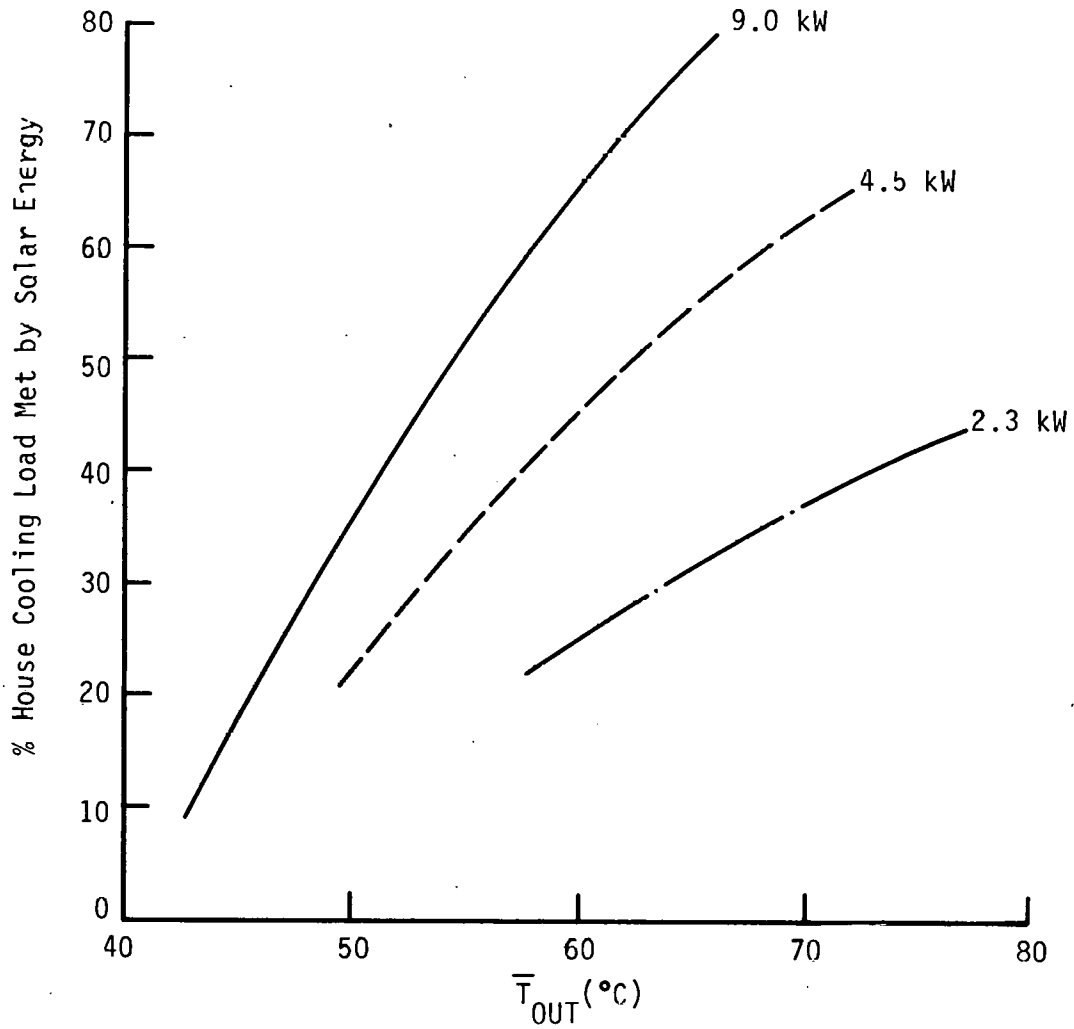
Figure 3-38. PERCENTAGE OF ANNUAL COOLING LOAD MET BY SOLAR ENERGY VS. REGENERATION
TEMPERATURE T_{OUT} (WASHINGTON, D.C.)

SECTION 4.0

STANDARD VAPOR-COMPRESSION AIR-CONDITIONING SIMULATION

In order to evaluate the performance characteristics and cost of a solar desiccant cooler, an accepted basis of comparison is required. For the purposes of this study, an electrically driven, standard vapor-compression air conditioner, used commonly throughout the United States, was chosen for comparison. As with the desiccant cooler, the vapor-compression cooler simulation was run for a 12-month cooling/heating season. Important parameters required to make a comparison between the two cooler types were calculated, including cooling capacity supplied, heat supplied, electrical energy required, COP, EER, and percentage of time the house was in the comfort zone.

4.1 DESCRIPTION OF SIMULATION

The vapor-compression cooler simulations were performed by a computer program entirely separate from TRNSYS. However, the desiccant and vapor-compression simulations shared many features so that direct comparisons of the results could be made. The same basic house model, load calculation model, house comfort zone, SOLMET ambient weather data for each location, and time increment (one-half hour) were used.

Figure 4-1 illustrates the ideal single-stage vapor-compression refrigeration cycle. The energy transfers are given by

$$_2Q_3 = h_3 - h_2,$$

$$_3W_4 = h_4 - h_3,$$

$$_4Q_1 = h_3 - h_1, \text{ and}$$

$$h_1 = h_2,$$

where

Q = heat transfer (J/kg),

W = work transfer (J/kg), and

h = enthalpy (J/kg).

The theoretical coefficient of performance is defined as

$$COP = \frac{h_3 - h_2}{h_4 - h_3}.$$

The vapor-compression simulation was written with two basic assumptions:

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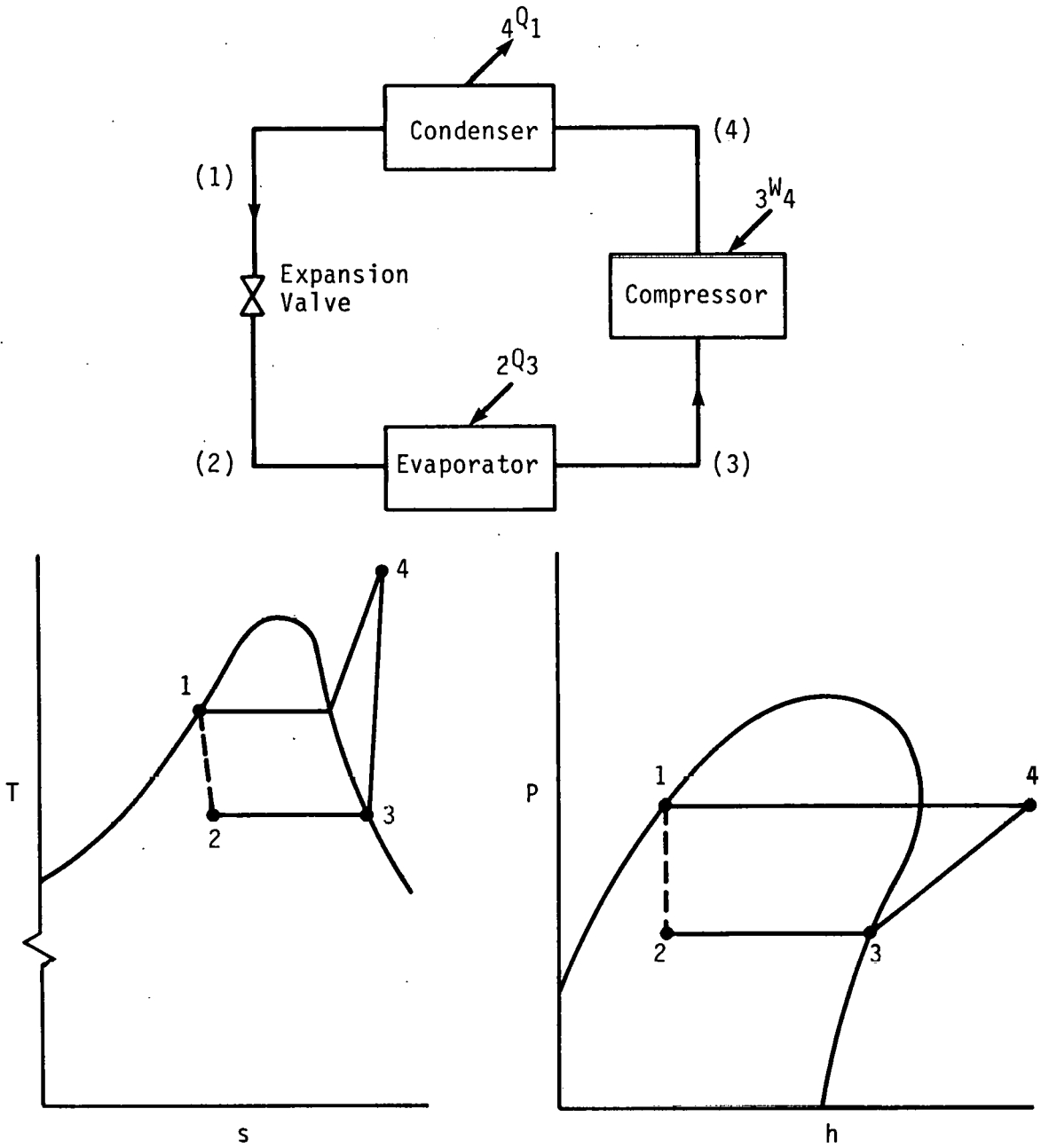
Figure 4-1. THEORETICAL SINGLE-STAGE VAPOR-COMPRESSION REFRIGERATION CYCLE

- The vapor-compression machine had a constant cooling capacity of 9.0 kW (sensible + latent).

- The average COP for the cooler was 2.5.

These and the other basic input parameters of the simulation are summarized in Table 4-1.

Table 4-1. SUMMARY OF INPUT PARAMETERS FOR VAPOR-COMPRESSION AIR-CONDITIONING SIMULATION

Operating capacity (kW) [tons]	9.0 [2.5]
Average vapor-compression COP (without reheat)	2.5
Flowrate (m^3/s)	0.48
Assumed air pressure drops (N/m^2)	125
House comfort zone	Defined in Figure 3-4
House model	Described in Section 3.2.2.6
House cooling/heating loads	Described in Section 3.2.3
SOLMET hourly ambient weather data	
Charleston, S.C.	1963
Dodge City, Kans.	1955
Ft. Worth, Tex.	1960
Phoenix, Ariz.	1962
Washington, D.C.	1955
Simulation time increment (h)	0.5
Simulation time period	12-month heating/cooling season

The simulated operation of the vapor-compression air conditioner is illustrated in Figure 4-2. Two different room conditions, 25.5°C with 60% relative humidity and 25.5°C with 20% relative humidity, are shown on the psychrometric chart with the corresponding paths for room air passing through the vapor-compression cooler evaporator.

Point 1 on Figure 4-2 represents room air at 25.5°C dry-bulb temperature and 60% relative humidity. As the air passes through the evaporator, it is sensibly cooled along a line of constant humidity ratio W until it reaches saturation (dew-point temperature) at point 2. From point 2 to point 3 the air undergoes both sensible and latent cooling as its dry-bulb temperature and humidity ratio decrease. The sum of the sensible cooling from points 1 to 2, sensible cooling from points 2 to 3, and latent cooling from points 2 to 3 equals 9.0 kW.

In the same manner, point 4 represents room air at 25.5°C dry-bulb temperature and 20% relative humidity. As the air passes through the evaporator, it is sensibly cooled to point 5. At this point the cooling capacity of 9.0 kW has
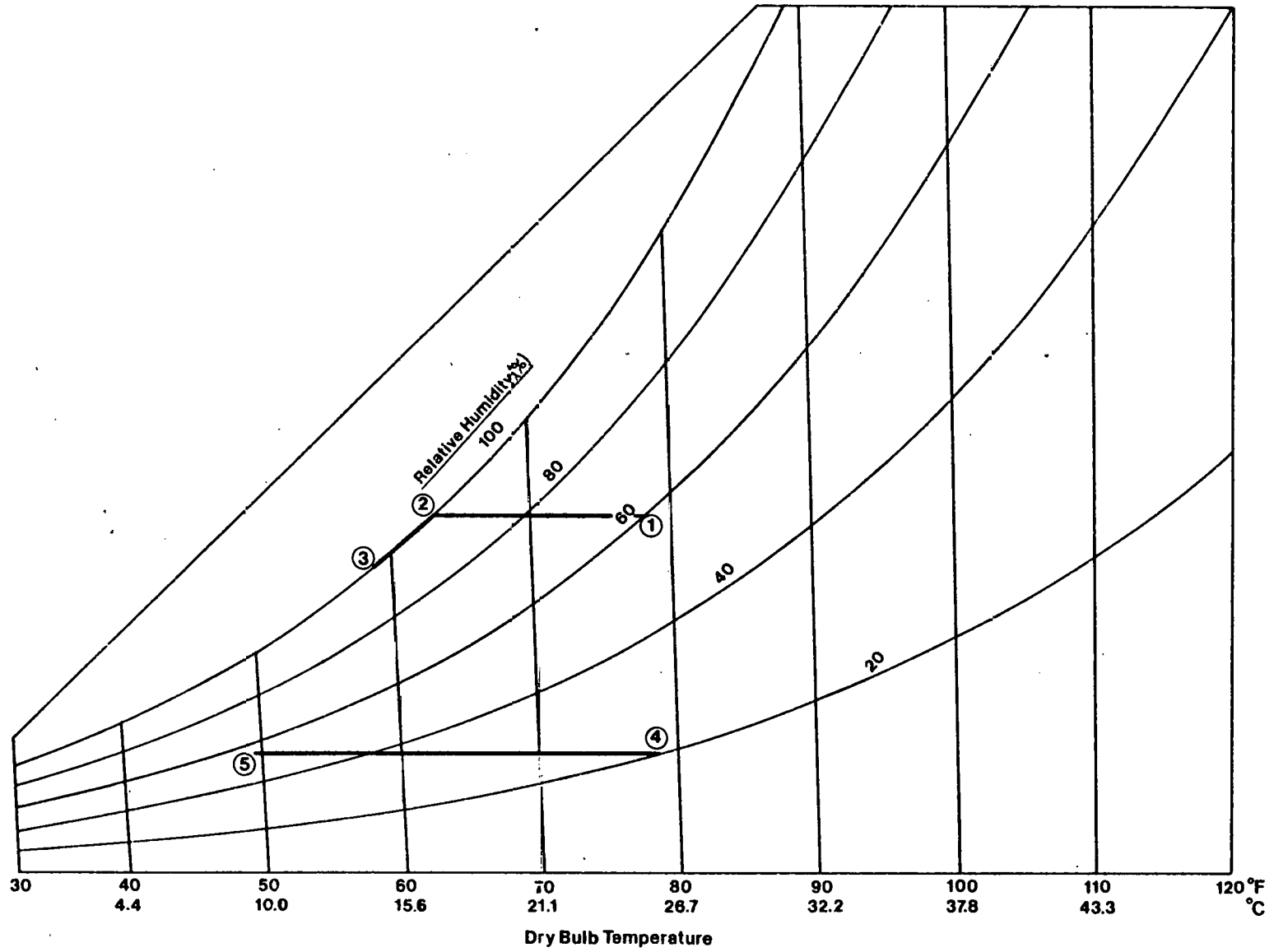
Figure 4-2.   TYPICAL AIR STATE TRANSITIONS FOR VAPOR-COMPRESSION COOLER SIMULATION

been reached and, therefore, no latent cooling is supplied. This condition could occur in very dry climates such as Phoenix, Ariz. Correlations for enthalpy as a function of humidity ratio and specific heat of moist air allow the states indicated in Figure 4-2 to be determined [28].

Two sets of simulations were performed for each of the five cities under study:

- a standard operating mode for a residential vapor-compression cooler, and

- a revised operating mode using reheat for the room air immediately after the air leaves the cooler and before it is reintroduced into the house.

Although commonly used in commercial applications, reheat is not usually employed in residential systems. Reheat was included in one set of simulations in order to match latent house load to cooler latent capacity for the desiccant and vapor-compression systems. The desiccant cooler system can meet latent loads as well as sensible loads and can even supply latent capacity alone if required. However, the vapor-compression cooler must supply enough sensible cooling to bring the room air to saturation before any latent capacity can be supplied. In humid climates such as Charleston, S. C., or Washington, D.C., the vapor-compression cooler might supply too much sensible cooling in meeting the large latent loads. Therefore, when more latent than sensible cooling was required to maintain the house in comfort, reheat was used whereby heat was added to the cooled air leaving the vapor-compression machine in an amount equal to the difference between the sensible cooling supplied and the sensible house load.

The controller required to operate the vapor-compression cooler was simpler than that for the desiccant cooler (see Section 3.7.2). When room conditions were above comfort in dry-bulb temperature or relative humidity, the vapor-compression system was simply turned on. The proportions of sensible and latent cooling supplied were determined by the state of the moist air introduced into the evaporator. It would be possible to change the proportions of sensible to latent cooling by varying the rate of airflow through the cooler while maintaining the 9.0-kW total cooling capacity. A variable air mass flow capability was included in the controller for the vapor-compression simulation but the feature was not used in this study.

The computer program written for the vapor-compression machine simulated the conditions for an annual cooling/heating season in essentially the same manner as did the program for the desiccant cooler. Moist air at room conditions entered the cooler, 9.0 kW of cooling capacity was supplied, reheat (if any) was added, and the airstream was reintroduced into the house. The house model determined the resulting house air conditions, and the controller continued cooler operation if the room air was out of comfort in dry-bulb temperature and/or relative humidity; or, if the room was in comfort, the controller turned off the cooler. If the room conditions were such that heating was required, the heating load was assumed to be met with an auxiliary heater so that the room dry-bulb temperature never dropped below 20°C. However, the room conditions could go higher than 25.5°C dry-bulb temperature and 60%

relative humidity if the vapor-compression cooler could not meet the sensible and latent house loads. Energy accounting for cooler capacity, house loads, etc., was identical to that described for the desiccant cooler. Figure 4-3 is a flow diagram showing basic steps in the computer program.

4.2 RESULTS

The vapor-compression simulation was run for the five cities used in the desiccant simulation. The results are shown in Appendix B (Tables B-1 through B-10). The size of the vapor-compression cooler was not varied from 9.0 kW.

The information shown in the tables of Appendix B was defined in the same manner as for the desiccant cooler with the exception of reheat (see Section 4.1) and COP. Although the average COP for the vapor-compression cooler was assumed to be 2.5, the overall system COP was calculated as

$$\text{COP overall} = \frac{\text{Cooling capacity}}{\text{Cooling capacity}/2.5 + \text{reheat}}$$

The reheat was considered part of the vapor-compression system and, as a heat input, was included in the calculation of cooling COP shown in Tables B-6 to B-10. Tables B-1 to B-5 are the results of the vapor-compression simulation without reheat.

Investigation of the two sets of results (with and without reheat) indicates that the net energy requirements for each are nearly the same. For all five cities, the results for the standard operating mode without reheat (Tables B-1 to B-5) show that the heating load increased by an amount approximately equal to the reheat requirements of Tables B-6 to B-10. Thus, in the standard operating mode, the vapor-compression cooler drove the room conditions to the lower comfort boundary in temperature (20°C) in an effort to meet the latent load and maintain comfort conditions of relative humidity (60% upper limit). The excess sensible cooling capacity of the cooler was offset by the house natural-gas-fired space heater. In the reheat mode, the excess sensible cooling capacity was met by a natural-gas-fired air heater between the cooler and the house. In either case, the annual electrical and natural gas requirements were essentially the same. If the cooler had been controlled by temperature only, the house relative humidity would have drifted in and out of the comfort zone and no extra energy for reheat or space heating would have been required.

Several items of information from the tables in Appendix B were required for the economic analysis (see section 5.0), including total supplied heat, cooling capacity, reheat, and parasitic energy.
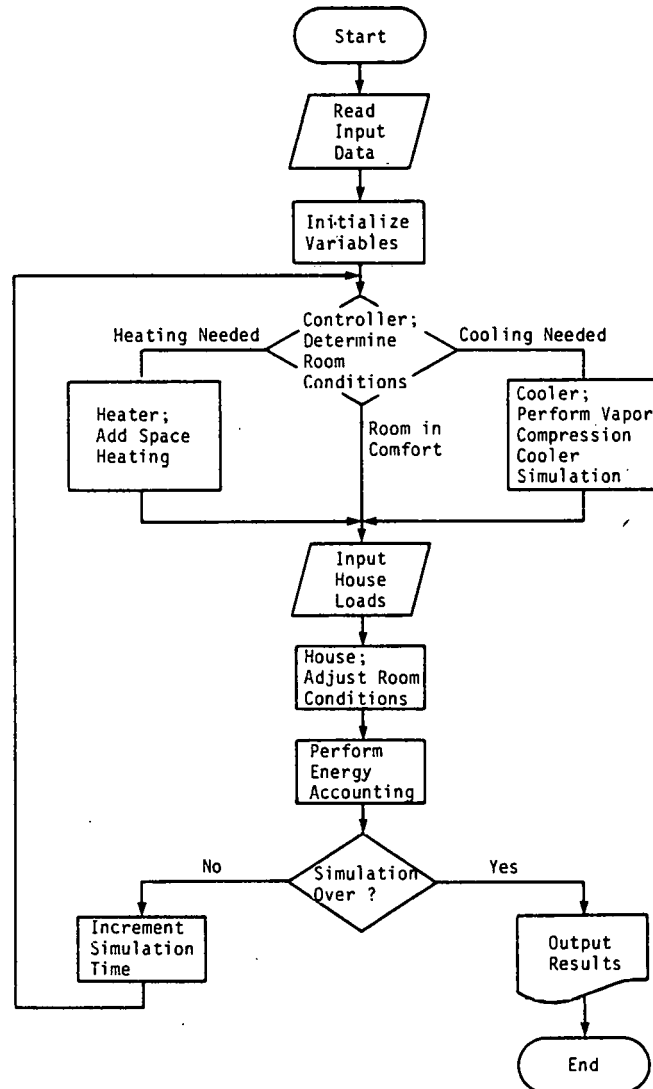
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Figure 4-3. VAPOR-COMPRESSION COOLER SIMULATION FLOW DIAGRAM

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SECTION 5.0

ECONOMIC ANALYSIS

The potential for using solar space heating and/or desiccant cooling in residential buildings ultimately depends on the economics of solar heating and cooling systems as compared to conventional systems. Local energy costs (electricity and natural gas) and the amount of conventional energy displaced by solar energy determine if the capital cost of the solar system can be recovered within an acceptable time period through energy cost savings. The previous sections of this report describe simulations of solar and conventional systems for residential space heating and cooling. The results of these simulations, namely, power input required to provide annual heating and cooling, were used in an economic analysis computer program to make direct comparisons between solar and conventional systems. Also considered was the potential for fossil fuel conservation.

5.1 METHODOLOGY

The economic analysis program was used to discount life-cycle costs of the solar and conventional systems, resulting in present worths of the costs for each system. The present worth was determined for all annual heating/cooling season simulations of the solar and conventional systems in each of the five cities studied. Cost/benefit ratios for the solar systems were then calculated from the present worth figures.

In order to determine the present worth of life-cycle costs, several parameters were required for the economic analysis. These included interest rates, inflation rates, and energy-cost escalation rates. Since a 20-year economic lifetime was assumed for both the solar and conventional systems, forecasts were made for the economic parameters (assumed as constants) over the lives of the systems. The values used in the analysis (base year of 1978) are shown in Table 5-1.

Table 5-1. PARAMETERS USED IN ECONOMIC ANALYSIS

System lifetime	20 years
General inflation rate	6%
Interest rate	9%
Discount rate	3%
Natural gas escalation rate	5%
Electricity escalation rate	5%

By using these parameters, the annual cash flow (outlay) for each system was discounted to the present by using the following equation [29]:

$$PW = A \times \frac{(1+i)^n - 1}{i\,(1+i)^n} \, ,$$

where

PW = present worth ($),
A = annual cash outlay ($),
i = discount rate, and
n = number of years from present to expenditure of outlay A.

All cash outlays and the present worth were assumed to be in constant dollars. Due to the natural gas and electricity escalation rates (that is, the rate at which the costs of these fuels increase in excess of the general inflation rate), the annual cash outlay increased over the life of the system.

In addition to the economic parameters of Table 5-1, annual cash outlays for solar and conventional systems are influenced by capital equipment costs; installation costs; operation, maintenance, replacement, and insurance (OMRI) costs; and potential government tax credits or subsidies for residential solar installations. The values used in this study are described in Section 5.2. Since the economic parameters and costs used in the life-cycle analysis can vary widely over a 20-year period, the results are useful only for general comparisons between system types. The analyses may not reflect true costs in the latter years of the system lifetimes.

Once the present worth of the life-cycle costs for solar and conventional systems was calculated, the solar system cost/beneift ratio was determined. For the purposes of this study, "benefit" was defined as the present worth of life-cycle costs to maintain house air conditions within the comfort zone over 20 consecutive 12-month heating/cooling seasons by use of conventional fuels and air-conditioning methods (that is, natural gas heating and electrically driven, vapor-compression air conditioning). "Cost" was defined as the present worth of the life-cycle costs to maintain comfort conditions for the same time period using solar energy for heating and desiccant air conditioning, with auxiliary heating and cooling as required. Thus, a cost/benefit ratio equal to 1.0 represents a case where the net solar system costs equal the conventional system costs over 20-year lifetimes. For a cost/benefit ratio less than 1.0, the solar system results in a net savings over the cost of a conventional system, and for a ratio greater than 1.0, the conventional system results in a net savings over the cost of the solar system.

Solar systems that supply only space heating also were compared. Cost/benefit ratios were determined for solar space heating systems with natural gas heaters for auxiliary heat and vapor-compression air conditioners for the total cooling loads. With this information, incremental changes in cost/benefit ratios were examined for cases in which combined solar desiccant coolers and solar space heating systems replaced conventional coolers with solar space heating systems.

An important aspect of the solar heating and cooling systems is the potential for conservation of fossil fuels. As fossil fuels become more scarce, the impetus for their displacement by solar energy for heating and cooling may equal economic incentives. The percentage of fossil fuel displaced by solar energy was determined for each of the annual simulations performed in this study. The fossil fuel displacement was also determined for solar heating systems with vapor-compression air conditioning.

To determine the fossil fuel displaced by solar heating and cooling systems, the power inputs to both the solar and conventional systems were obtained from the computer simulations for annual heating and cooling seasons. To convert electrical power to its equivalent in fossil fuel, an overall conversion and transmission efficiency of 33% was assumed. The following equations were used to determine fossil fuel displacement:

$$DISP = \frac{HEAT\ DISP \times HEAT\ LOAD + (BASE\ THERMAL - SOLAR\ THERMAL)}{HEAT\ LOAD + BASE\ THERMAL},$$

where

DISP = fraction of fossil fuel energy displaced by solar energy,

HEAT LOAD = total annual house heating load,

HEAT DISP = fraction of HEAT LOAD met by solar energy

BASE THERMAL = fossil fuel energy required for cooling season with conventional air conditioning, and

SOLAR THERMAL = fossil fuel energy required for cooling season with solar desiccant cooler.

BASE THERMAL and SOLAR THERMAL were determined as follows:

$$BASE\ THERMAL = 3.0 \times \frac{COOLING}{COP} + PP + REHEAT,$$

where

COOLING = annual vapor-compression cooling supplied to house,

COP = vapor-compression cooler coefficient of performance (assumed to be 2.5),

PP = electrical power to vapor-compression cooler air fan, and

REHEAT = natural gas reheat (if any);

and

$$SOLAR\ THERMAL = 3.0 \times PPS + \frac{AUX\ COOL}{COP},$$

where

PPS = electrical power to desiccant cooler air fans and drive motor,

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    AUX COOL = total annual cooling supplied to house by auxiliary vapor-
                  compression cooler, and

    COP   = auxiliary vapor-compression cooler coefficient of perform-
                  ance (assumed to be 2.5).

The factor of 3.0 in the BASE THERMAL and SOLAR THERMAL accounts for the 33%
conversion efficiency from fossil fuels to delivered electrical power. All
auxiliary natural gas space heating and reheat values include an assumed
heater efficiency of 65%.

For the case in which solar energy was utilized solely for space heating in
conjunction with a vapor-compression cooler, the fraction of displaced fossil
fuel energy, DISP H, was calculated in a similar manner. The equation was as
follows:

$$\text{DISP H} = \frac{\text{HEAT DISP} \times \text{HEAT LOAD}}{\text{HEAT LOAD} + \text{BASE THERMAL}} \cdot$$

5.2 ASSUMED CAPITAL AND OPERATING COSTS

In order to determine life-cycle costs for the solar and conventional systems,
assumptions were made regarding capital equipment costs, installation costs,
OMRI costs, and potential government solar tax incentives or subsidies. Due
to uncertainty of the values of these costs and incentives, the program was
written so that the present worth of life-cycle costs was determined for
several increments in collector cost, percentage of initial capital cost for
OMRI, and percentage of inital solar equipment cost subsidized by the govern-
ment. In addition, the complete analysis was performed for five base years
(year of system purchase and startup): 1978, 1985, 1990, 1995, and 2000.

Because of the very limited amount of experience with desiccant cooler
systems, detailed system cost figures were not available. The cost of the
installed desiccant cooler was assumed to vary with cooling capacity as
follows:

    Desiccant Capital Cost = $500.00 + $800.00/ton cooling capacity.

The desiccant system cost is plotted as a function of capacity in Figure 5-1a.

The average cost for an installed vapor-compression cooler was estimated from
manufacturers' quotations. A cost of $400.00 per ton of cooling capacity was
used in the economic analysis. This cost is plotted as a function of capacity
in Figure 5-1b.

The collector/storage system cost was a major portion of the total solar
heating/cooling system capital cost. Due to differences in collector cost
projections, the installed collector/storage system costs were varied from
$108/m^2 to $323/m^2 in increments of $54/m^2.

OMRI costs vary widely for different systems. For this study, OMRI costs were
taken as constant annual percentages of the initial capital investment. The
OMRI percentage was varied from 0.5% to 5.0% in increments of 0.5%.
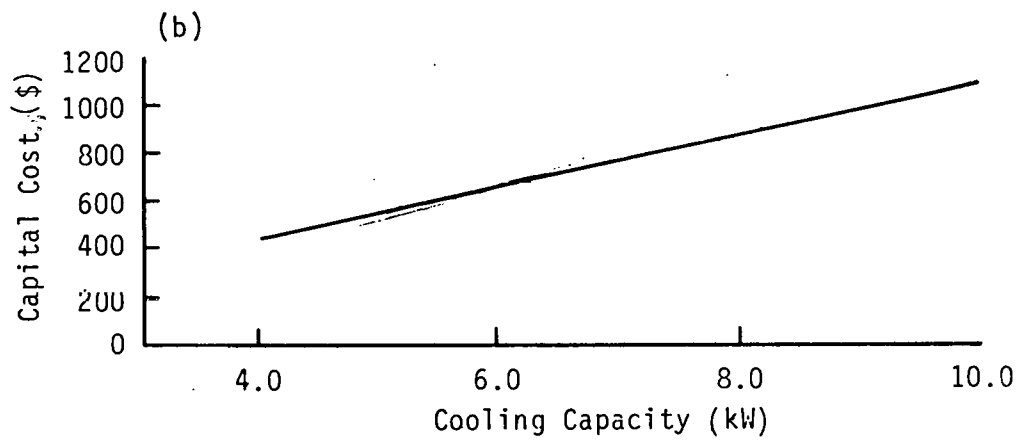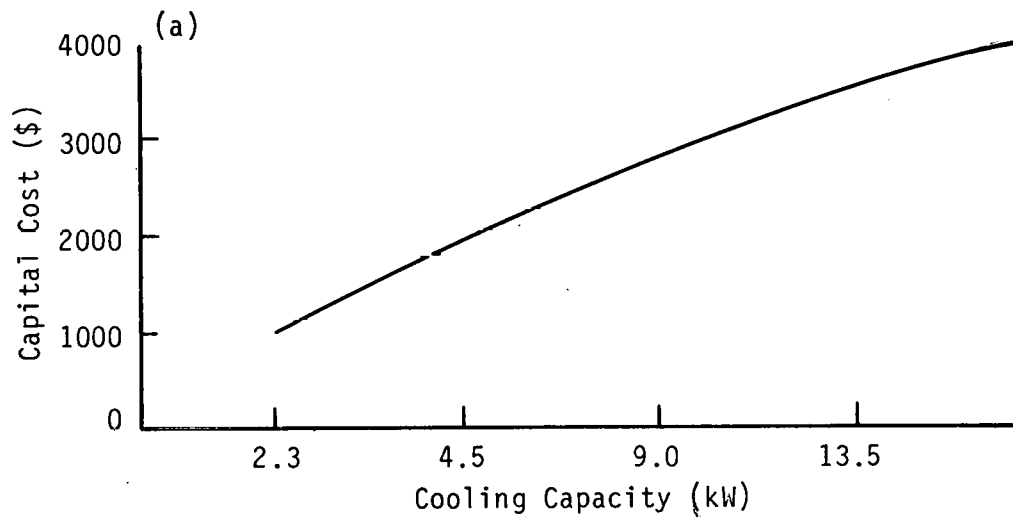
Figure 5-1. CAPITAL COST VS. CAPACITY FOR (a) DESICCANT COOLER AND
(b) VAPOR-COMPRESSION COOLER

Allowances were made in the economics program for potential government subsidies for solar systems. Two cases were studied: subsidies of 0% and 25% of the initial capital cost of solar equipment.

Table 5-2 summarizes the ranges of cost figures used in the economic analysis program and the values upon which most of the study conclusions were based.

Table 5-2. EQUIPMENT-RELATED COSTS USED IN ECONOMIC ANALYSIS FOR DETERMINATION OF LIFE-CYCLE PRESENT WORTH AND COST/BENEFIT RATIO

Cost Parameter	Parameter Range	Parameter Increment	Value(s) Used in Study
Desiccant cooler capital cost ($)	----	----	500 + 800/ton
Vapor-compression cooler capital cost ($)	----	----	400/ton
Collector/storage system cost ($/m^2)	107.65 to 322.95	53.83	107.65 and 215.30
OMRI (%)	0.5 to 5.0	0.5	1.0
Government subsidy of initial capital cost (%)	0 to 25.0	----	25.0

The last of the major inputs required for the economic anlysis were electricity and natural gas rates for the five cities. These rates were obtained directly from the utility serving each city, assuming a monthly consumption of 500 kWh of electrical energy and 340 m^3 of natural gas. The electrical and natural gas rates for each city are shown in Table 5-3.

Table 5-3. ELECTRICITY AND NATURAL GAS RATES FOR RESIDENTIAL BUILDINGS (AUGUST 1978)[a]

City	Electricity Cost ($/kWh)	Natural Gas Cost ($/kWh)
Charleston, S.C.	0.0526	0.0120
Dodge City, Kans.	0.0499	0.0049
Fort Worth, Tex.	0.0396	0.0094
Phoenix, Ariz.	0.0603	0.0090
Washington, D.C.	0.0410	0.0150

[a]Rates are based on monthly consumption of 500 kWh of electrical energy and 340 m^3 of natural gas.

5.3 RESULTS

Utilizing the economics program and cost data described in the previous section, the present worth of 20-year life-cycle costs was determined for each of the annual desiccant simulations described in Section 3.7.3. The resulting cost/benefit ratios for base years 1985 and 1990 are presented in Appendix C. A sample economics computer program output is included in Appendix D.

The general conclusions resulting from the economic analysis with a base year of 1985 include the following:

- Humid climates with high conventional fuel costs such as exist in Washington, D.C., and Charleston, S.C., are economically the preferred conditions for desiccant coolers.

- Based on life-cycle costs and fossil fuel displacement, a 4.5-kW desiccant cooler system with a 35-m^2 collector area and auxiliary cooling is the preferred configuration for Washington, D.C.; without auxiliary cooling, it is also the preferred configuration for Dodge City.

- The solar heating/desiccant-cooling system approaches cost competitiveness in Washington, D.C., (including auxiliary cooling) but not in Dodge City (without auxiliary cooling) due to low costs for conventional fossil fuels in Dodge City.

In order to illustrate the effects of climate on the operation of solar heating and desiccant cooling systems, cost/benefit was plotted versus ambient design relative humidity in Figures 5-2 and 5-3 for the five cities. Design relative humidity is a good indication of climate since it is a function of both dry- and wet-bulb temperatures. In Figure 5-2, the cost/benefit ratios were determined using actual 1978 electricity and natural gas prices from Table 5-3 (escalated to base years 1985 and 1990) for each city. Figure 5-3 shows cost/benefit ratios resulting from economic analyses using Washington, D.C., 1978 electricity and natural gas prices for all five cities. These figures clearly show that the effect of increasing relative humidity on solar cost-effectiveness is a decreasing cost/benefit ratio. Since a desiccant cooler generally has a larger latent capacity, this is the expected result. Thus, humid climates such as exist in Washington, D.C., and Charleston, S.C., are economically the preferred climate types for desiccant cooling systems.

The curves in Figure 5-2 do not show a continuous decrease in cost/benefit with increasing design relative humidity because of the varying energy costs in the five cities. Dodge City shows the largest variation because of its high annual heat load and very low natural gas price, a combination that favors conventional systems over solar systems. However, the curves in Figure 5-3 show increasing cost-effectiveness with increasing ambient design relative humidity when uniform energy costs (1978 Washington, D.C., costs) are assumed.

Figures 5-2 and 5-3 are not generic and cannot be used to obtain cost/benefit ratios for cities other than the five studied. Locations with the same design relative humidity can have different heating loads, cooling loads, and annual solar insolation, all of which are factors that affect the cost/benefit ratio.
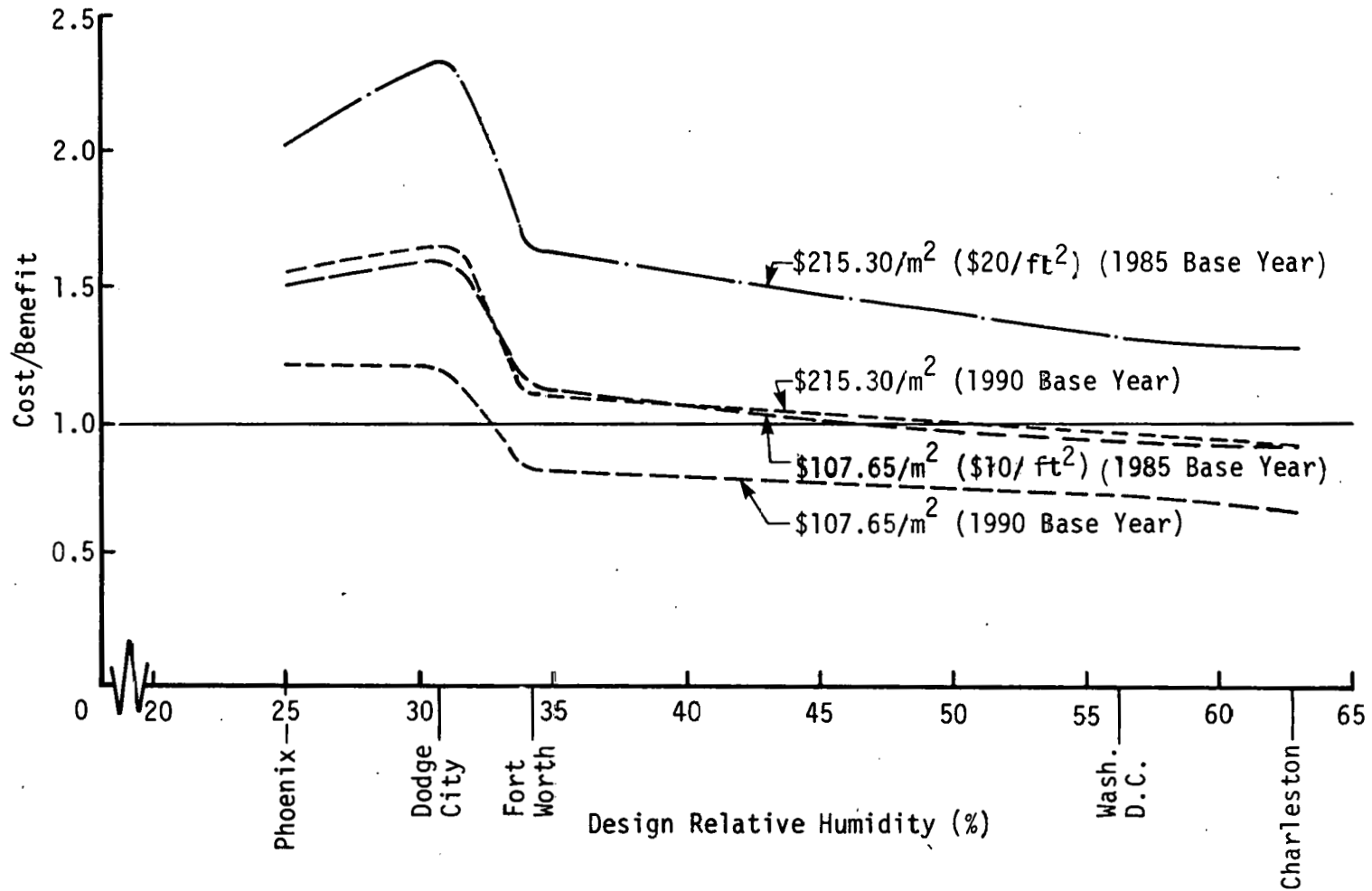
Figure 5-2. COST/BENEFIT VS. DESIGN RELATIVE HUMIDITY (1985 BASE YEAR)
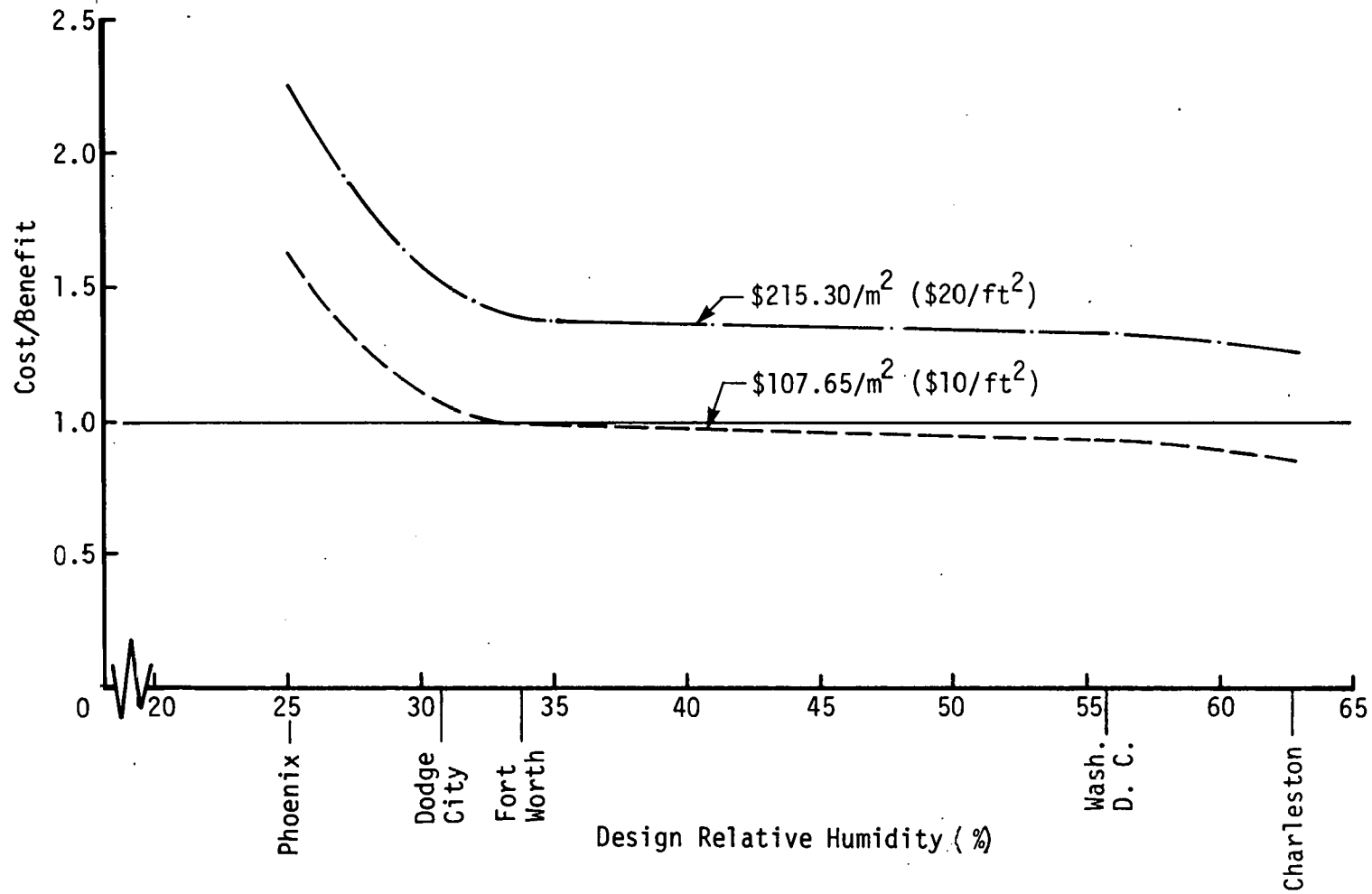
Figure 5-3.   COST/BENEFIT VS. DESIGN RELATIVE HUMIDITY USING WASHINGTON, D.C., ENERGY COSTS
FOR ALL CITIES (1985 BASE YEAR)

Figures 5-4 and 5-5 show the effects of the choice of base year (year of system purchase and startup) on the cost/benefit ratio. Under the economic parameter assumptions of Section 5.2, Figure 5-4 shows cost/benefit ratio versus base year for $108/m^2 collectors. Figure 5-5 shows the same plots for $215/m^2 collectors. Although it is impossible to predict accurately economic trends for the time periods encompassed, the two figures nevertheless illustrate that the cost/benefit ratio should decrease and eventually drop below 1.0 as conventional energy costs increase. As the base year is extended further into the future, collector costs become less important and the cost/benefit ratios for systems with collector costs of either $108/m^2 or $215/m^2 drop below 1.0.

The remainder of this section of the report deals with Washington, D.C., and Dodge City, Kansas. Washington, D.C., was chosen for further study because of the superior thermal performance of the desiccant cooler system in that climate and the potential for cost-effectiveness. Dodge City represents a location where the solar heating and desiccant cooling system described in this report could potentially provide comfort in a residential building without an auxiliary vapor-compression cooler (see Table 3-19).

An important characteristic to identify is the fraction of fossil fuel energy displaced by the solar energy systems. This is economically important since it determines whether a solar system is cost-effective. The percentage of fossil fuel energy displaced as a function of collector area is shown for Washington, D.C., and Dodge City in Figures 5-6 and 5-7, respectively.

Both figures show that the percentage of fossil fuel energy displaced was not significantly influenced by desiccant cooler capacity; instead, collector area appears to be the dominant influence. Two factors were responsible for this lack of influence by cooler capacity. First, superposition of the solar heating and cooling system performances diluted the impact of small differences in cooling system performance. This occurred in spite of the fact that cooling was responsible for 35% of the primary energy consumed for environmental control in a Washington, D.C., home and 25% in a Dodge City home. Second, cooling supplied by solar energy did not translate directly into displaced fossil fuel energy. Substantial parasitic energy was required in the solar desiccant cooling system. Thus, only a portion of the fossil fuel energy associated with providing a given amount of cooling was displaced through the use of the solar desiccant unit.

The analysis indicated that a small desiccant system tended to be more cost-effective than a larger system and that larger desiccant systems displaced a greater percentage of fossil fuel, although not by a significant amount. In Section 3.7.3 it is concluded that either a 4.5- or 9.0-kW cooler with a 35-m^2 collector is generally the optimum combination for Washington, D.C. Thus, the more cost-effective 4.5-kW system, which displaces nearly the same percentage of fossil fuel as a 9.0-kW system, is the preferred system size for Washington, D.C.

Figures 5-8 through 5-13 show the effects of collector area and the percentage of fossil fuel energy displaced on the cost/benefit ratio for solar heating alone and solar heating plus a 9.0-, 4.5-, or 2.3-kW desiccant cooler for Washington, D.C. The base year is 1985, and both sensible and latent

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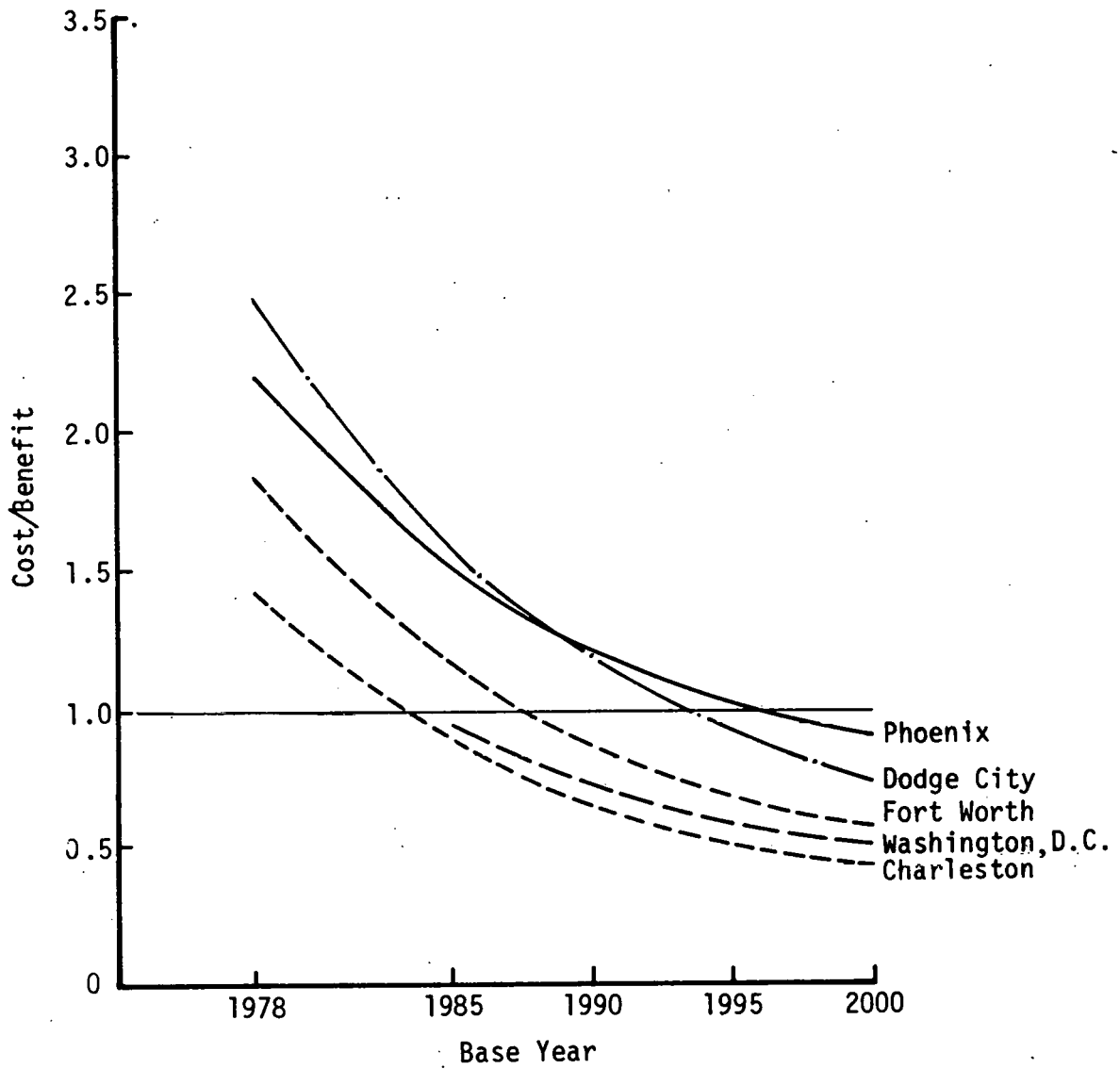
Figure 5-4. COST/BENEFIT VS. BASE YEAR - 35-M^2 COLLECTOR AT \$107.65/M^2 (\$10/FT^2)
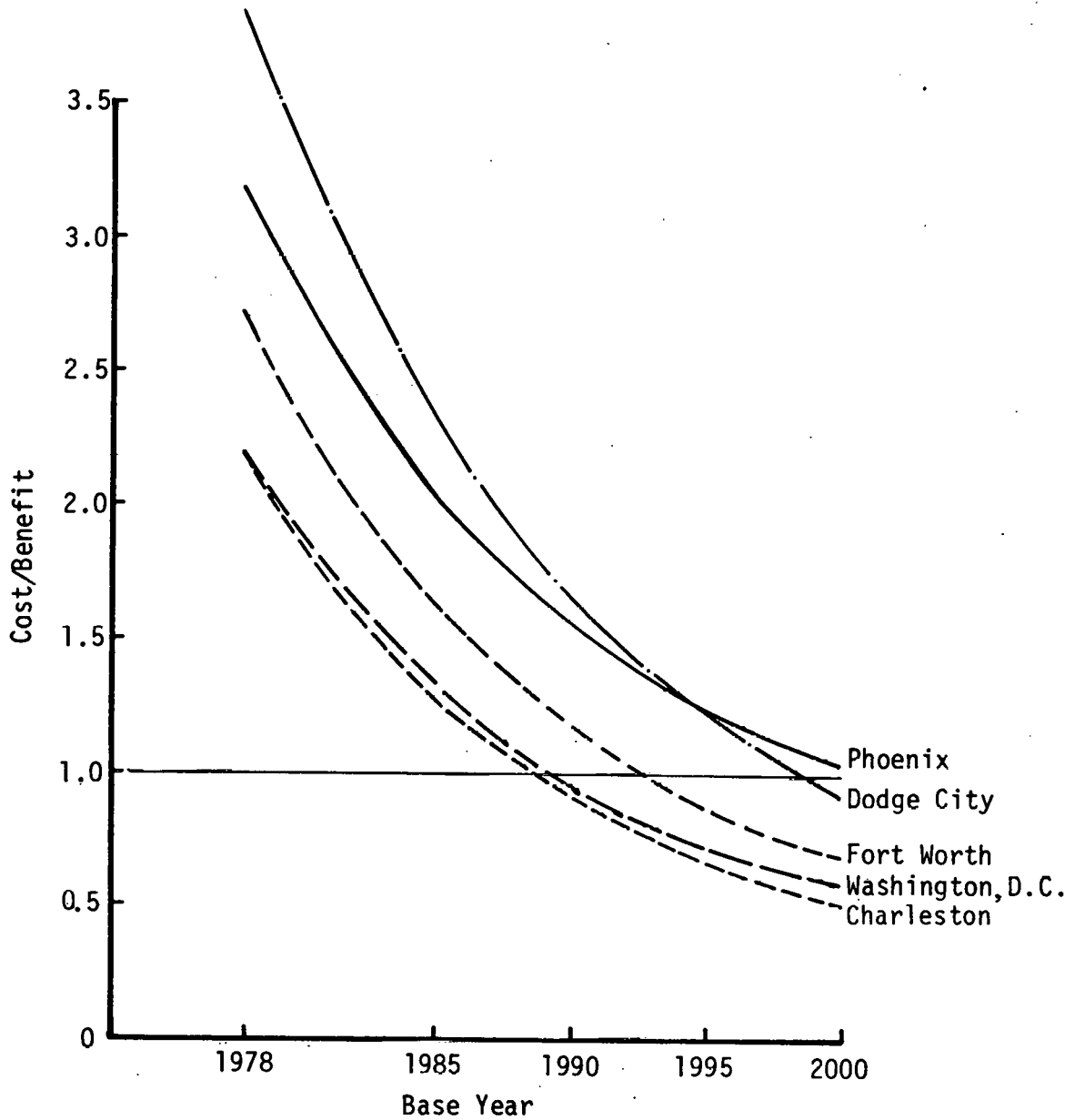
Figure 5-5. COST/BENEFIT VS. BASE YEAR - 35-M^2 COLLECTOR AT $215.30/M^2 ($20/FT^2)

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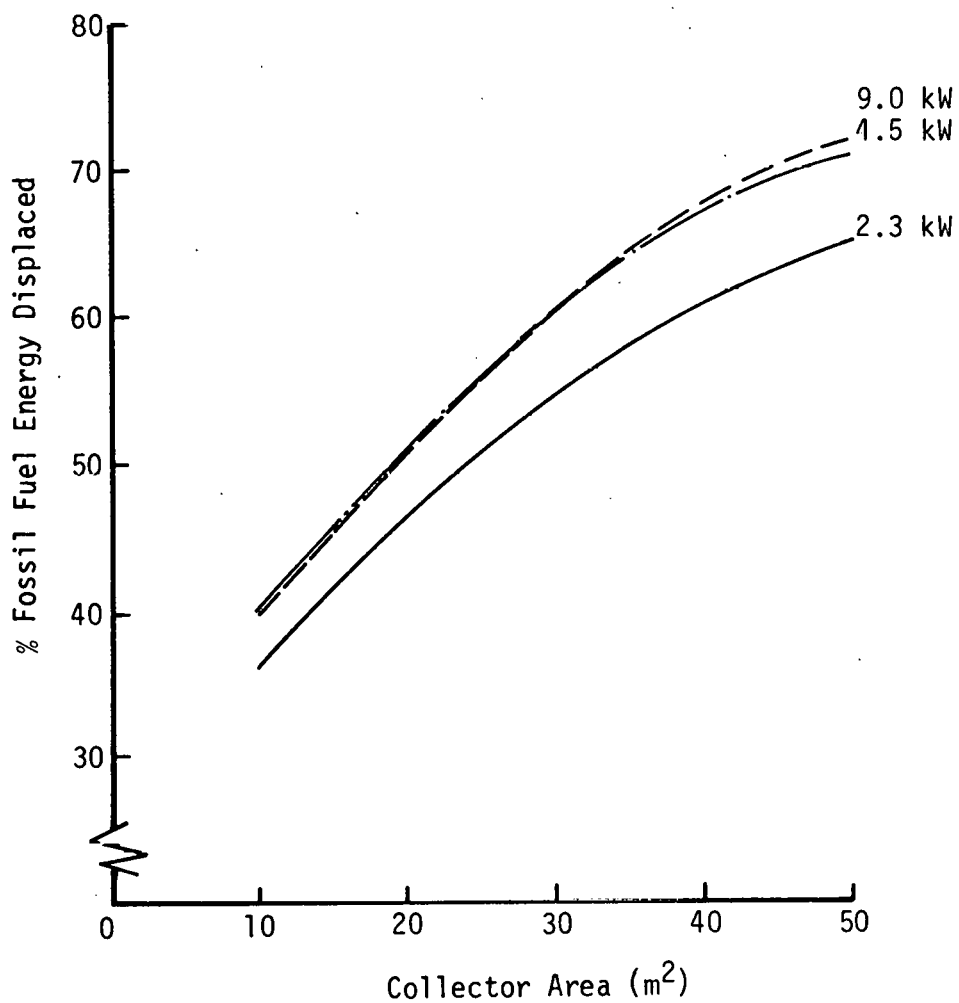
Figure 5-6.   FOSSIL FUEL ENERGY DISPLACED VS. COLLECTOR AREA - WASHINGTON, D.C., WITH AUXILIARY COOLING

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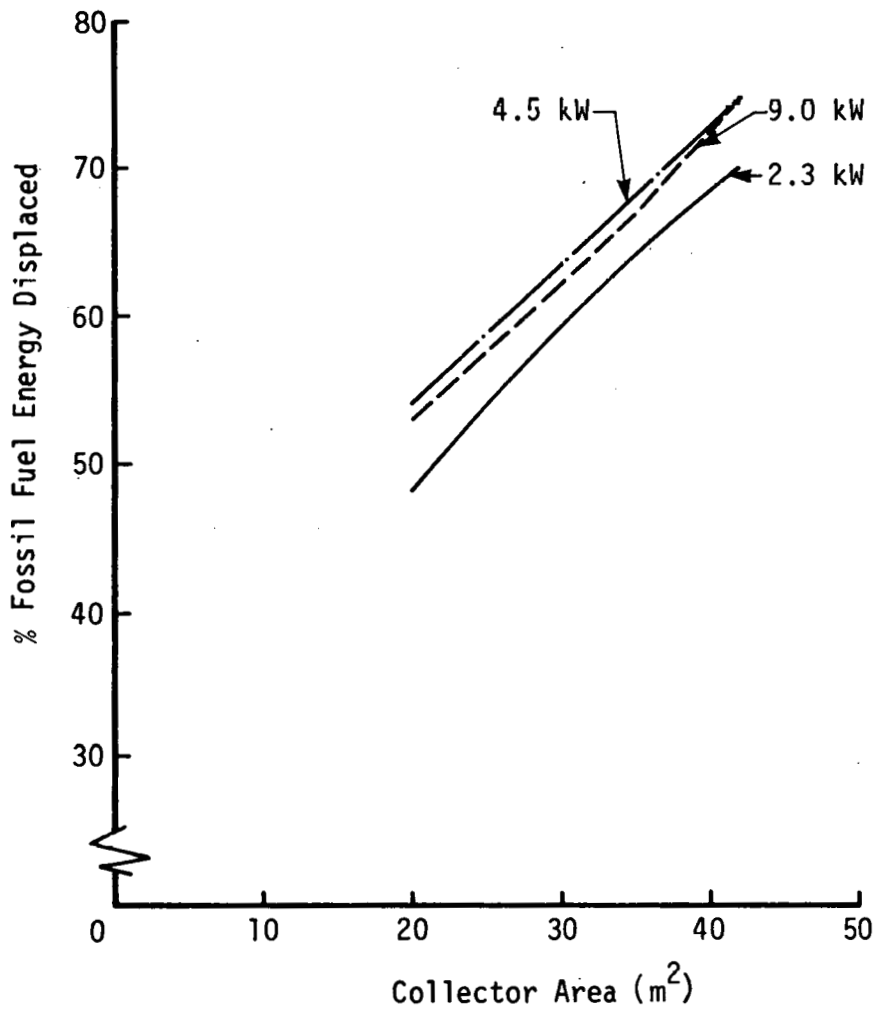
Figure 5-7. FOSSIL FUEL ENERGY DISPLACED VS. COLLECTOR AREA - DODGE CITY, KANS., WITHOUT AUXILIARY COOLING

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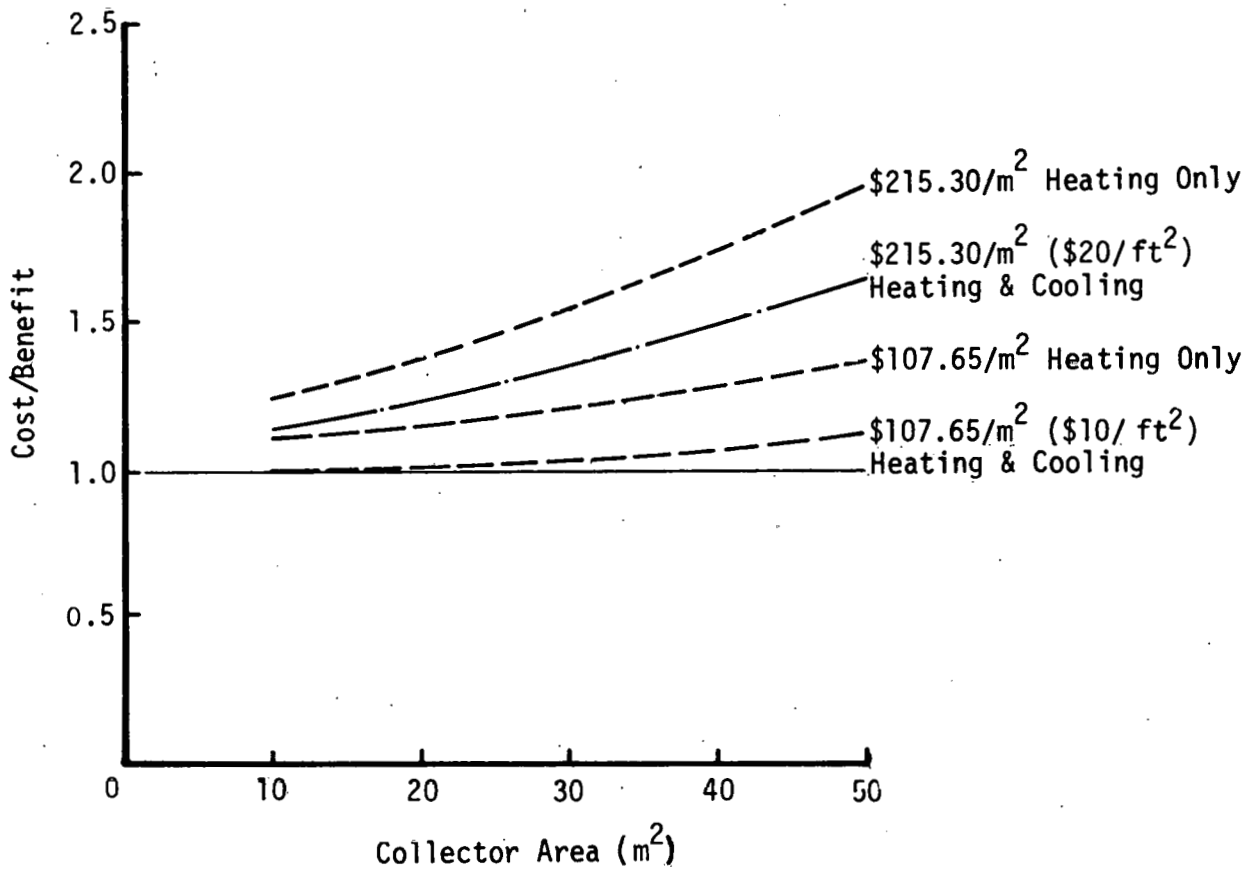
Figure 5-8. COST/BENEFIT VS. COLLECTOR AREA OF A 9.0-KW COOLER WITH AUXILIARY IN WASHINGTON, D.C. (1985 BASE YEAR)
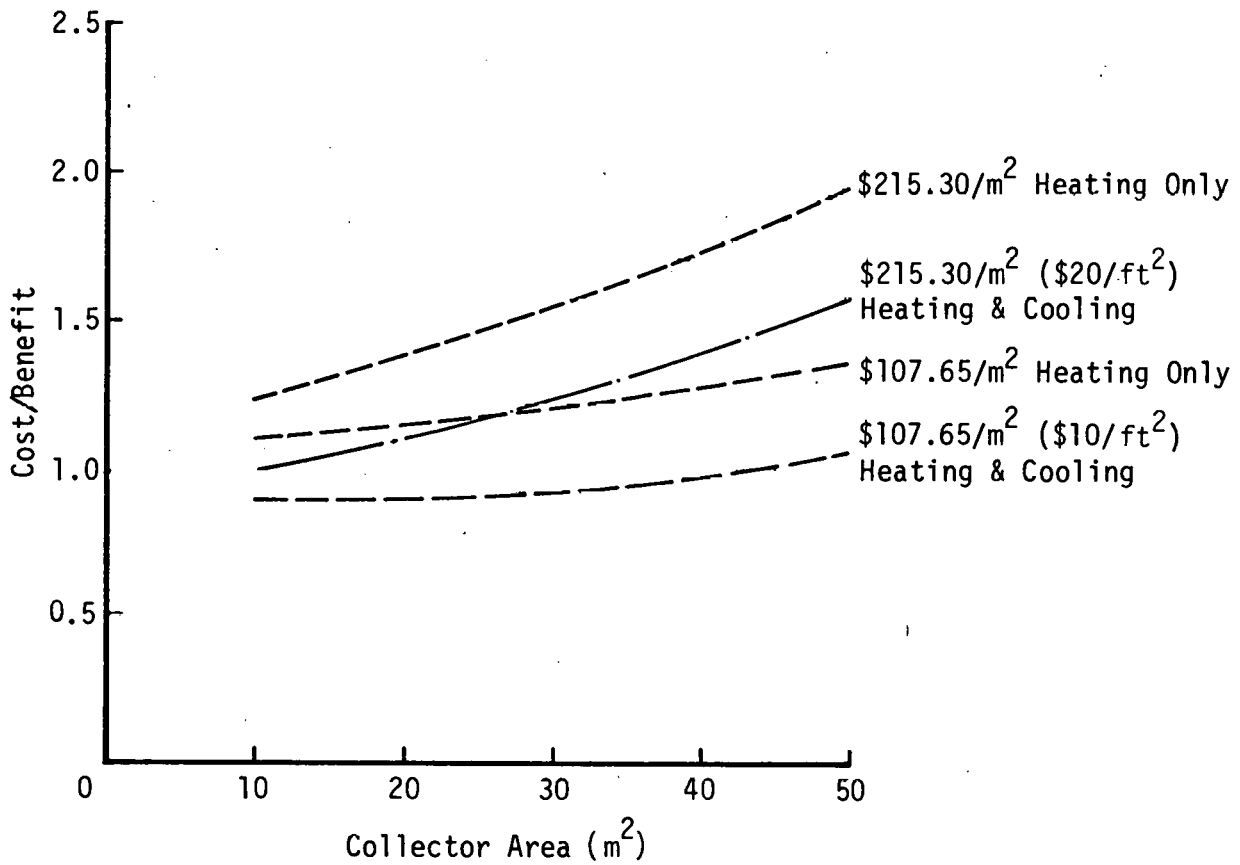
Figure 5-9. COST/BENEFIT VS. COLLECTOR AREA OF A 4.5-KW COOLER WITH AUXILIARY IN WASHINGTON, D.C. (1985 BASE YEAR)

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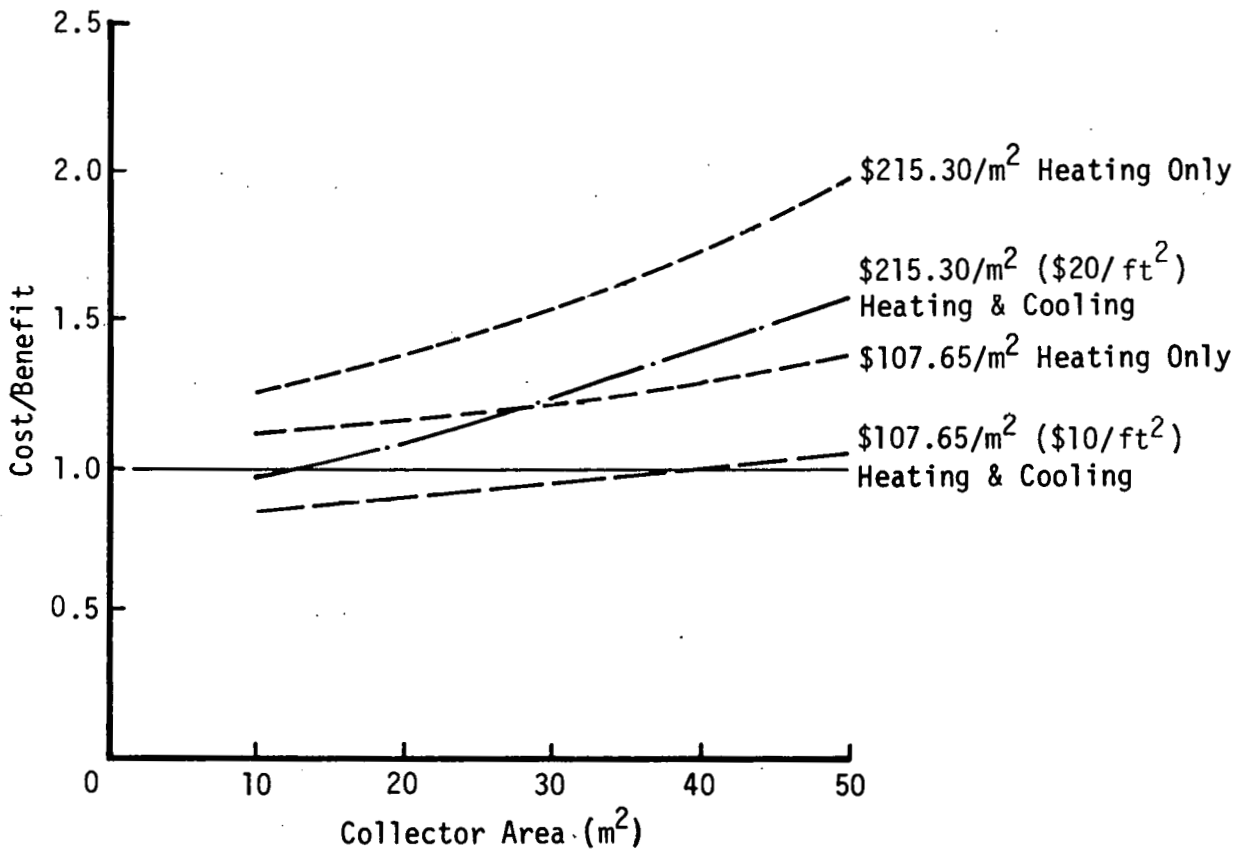
Figure 5-10. COST/BENEFIT VS. COLLECTOR AREA OF A 2.3-KW COOLER WITH AUXILIARY IN WASHINGTON, D.C. (1985 BASE YEAR)
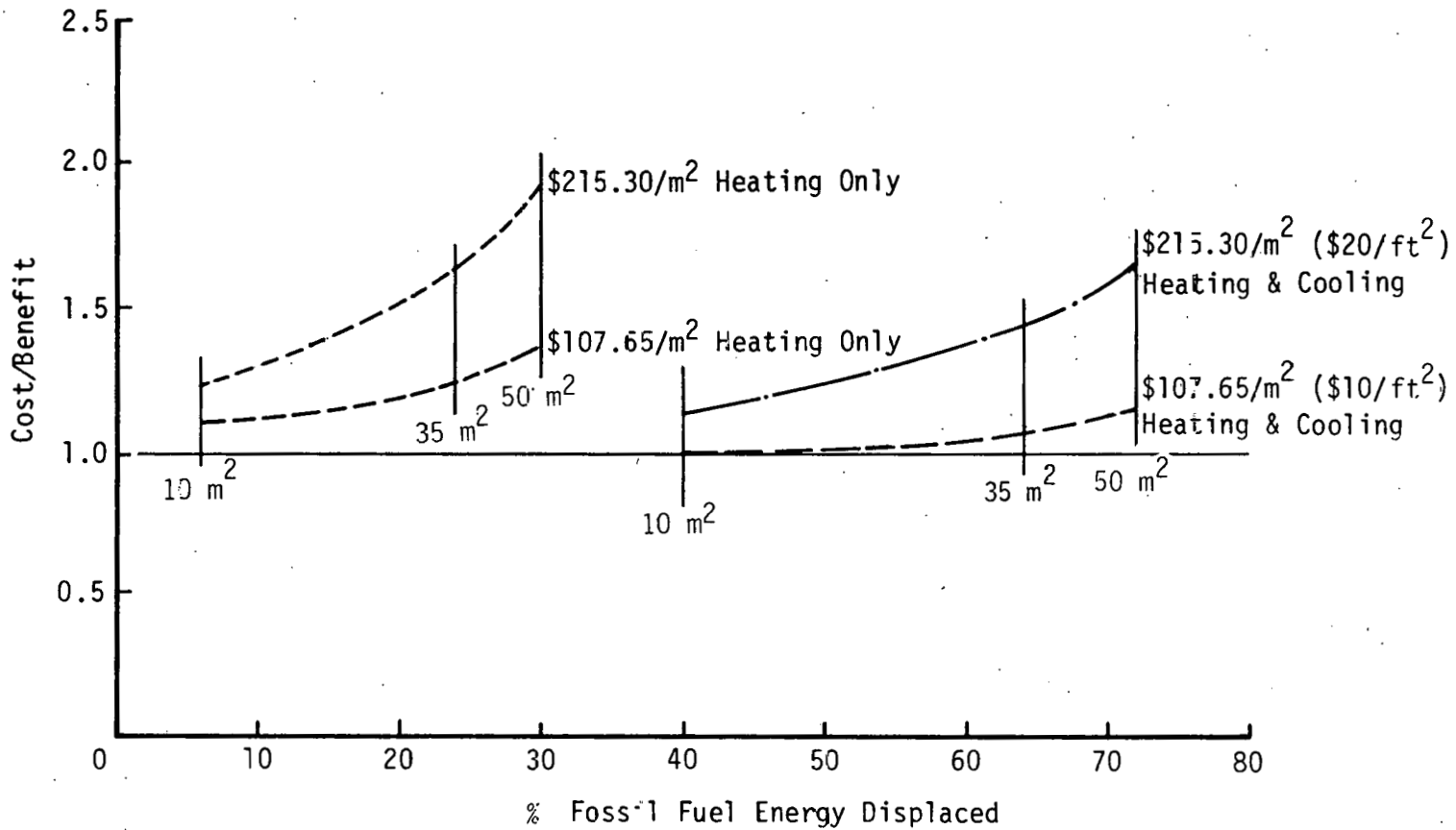
Figure 5-11. COST/BENEFIT VS. FOSSIL FUEL ENERGY DISPLACED BY A 9.0-KW COOLER WITH AUXILIARY AT VARIOUS COLLECTOR AREAS AND COSTS IN WASHINGTON, D.C. (1985 BASE YEAR)
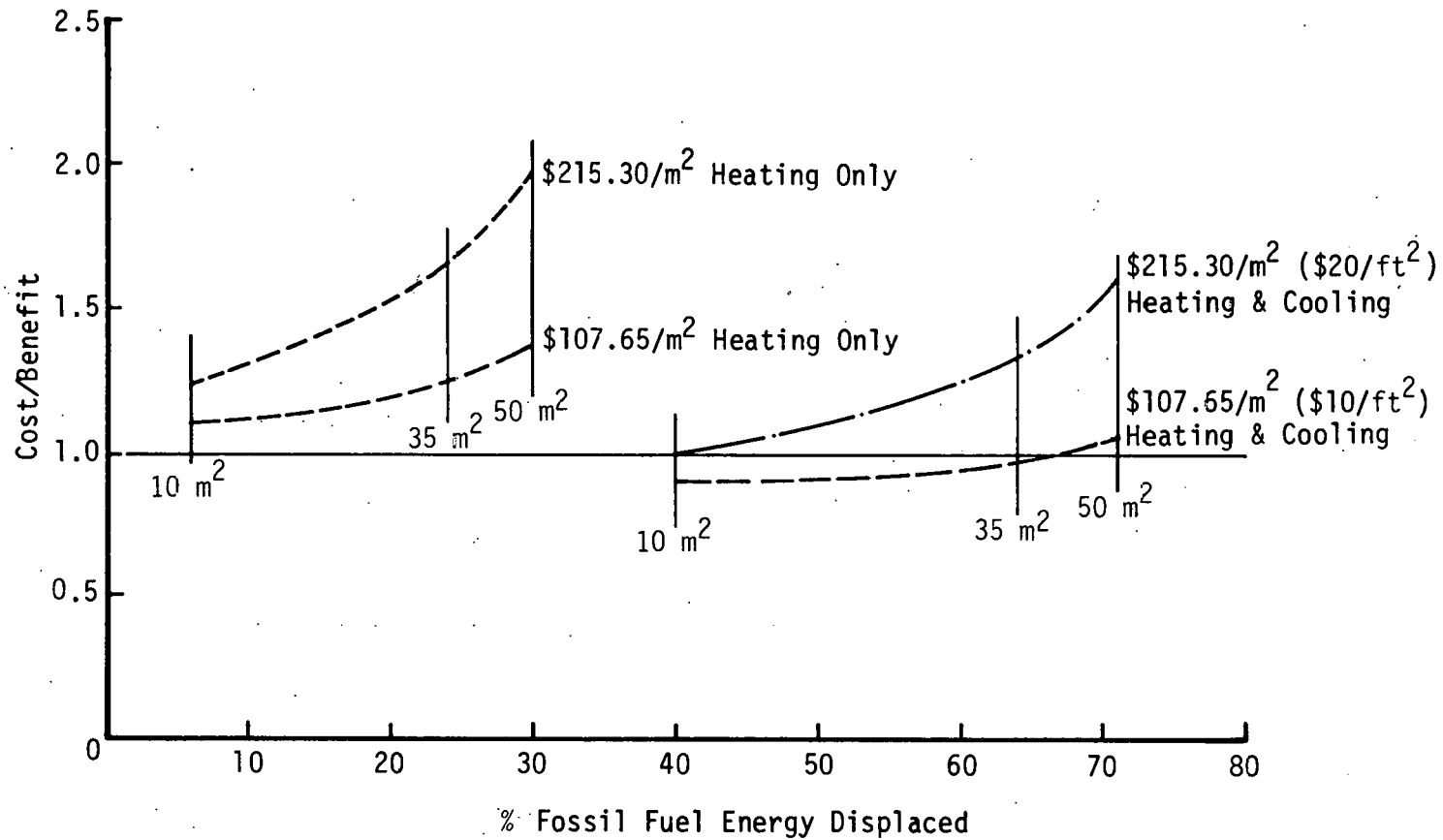
Figure 5-12. COST/BENEFIT VS. FOSSIL FUEL ENERGY DISPLACED BY A 4.5-KW COOLER WITH AUXILIARY AT VARIOUS COLLECTOR AREAS AND COSTS IN WASHINGTON, D.C. (1985 BASE YEAR)
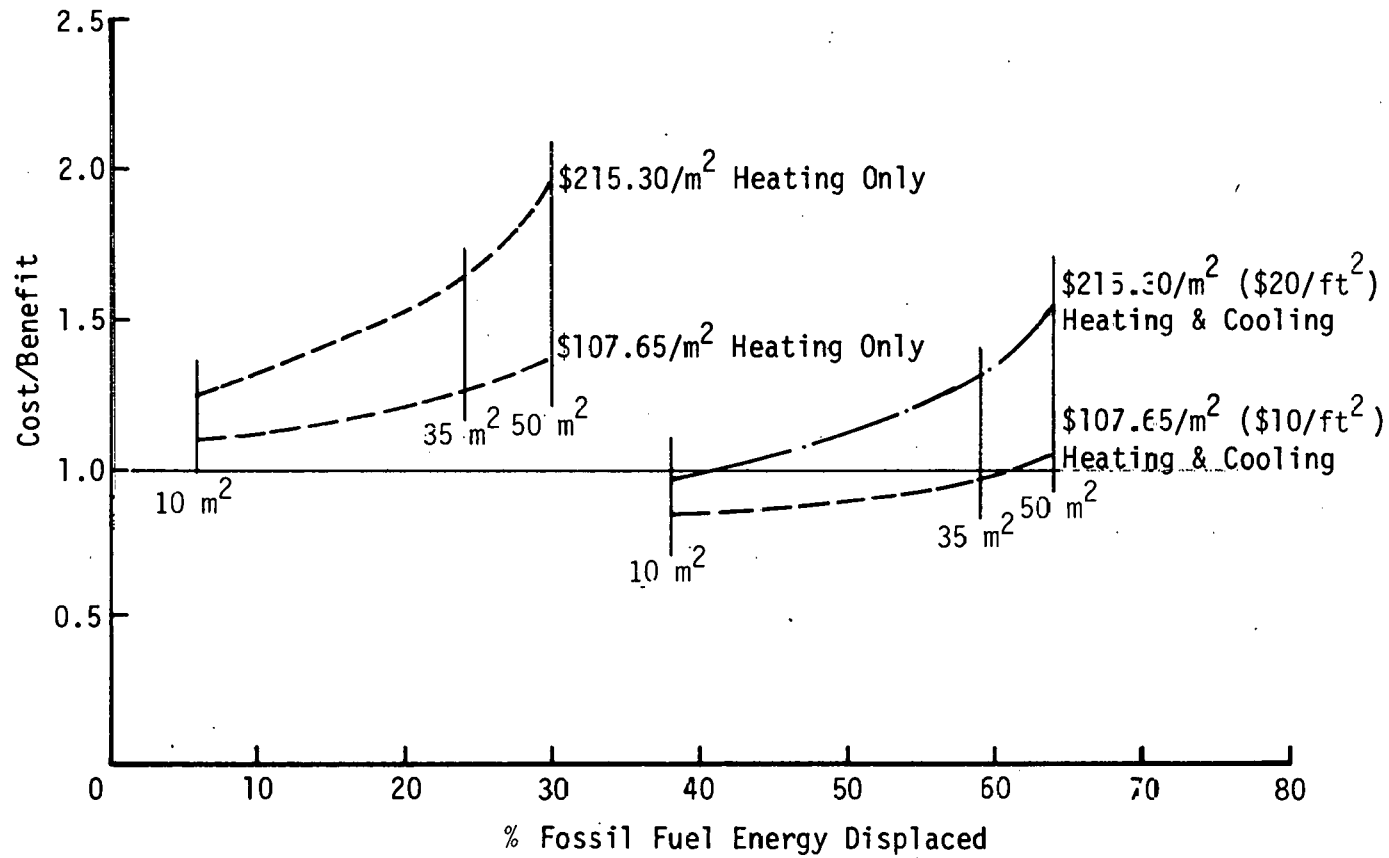
Figure 5-13. COST/BENEFIT VS. FOSSIL FUEL ENERGY DISPLACED BY A 2.3-KW COOLER WITH
AUXILIARY AT VARIOUS COLLECTOR AREAS AND COSTS IN WASHINGTON, D.C. (1985
BASE YEAR)

auxiliary cooling are provided. Three basic conclusions can be drawn from these six figures:

- Under the economic assumptions of this study (Section 5.2), solar heating in Washington, D.C., is not cost competitive (i.e., the cost/benefit ratio is never less than 1.0). Domestic hot water was not included in this analysis. Its inclusion would increase benefits without appreciable increase in cost and, therefore, a cost/benefit ratio higher than 1.0 might warrant installation of such a system if the cost of the collectors were in the $108 to $140/m^2$ range.

- If a desiccant cooler is added to the solar heating system, the cost/benefit ratio falls below 1.0 for certain cases, indicating cost-effectiveness. For the 4.5-kW cooler with a 35-m^2 collector array (Figure 5-9), the cost/benefit ratio is 0.94 with $108/m^2$ collectors. With solar hot water included, this system should remain cost-effective with collector costs in the $140 to $161/m^2$ range. For $108/m^2$ collector cost, the 9.0-kW system (Figure 5-8) is cost-effective only with the lowest collector areas, while the 2.3-kW system (Figure 5-10) is cost-effective with all but the largest collector sizes.

- When a desiccant cooling system is added to a solar heating system, a significant increase in the displaced fossil fuel energy can be achieved. This is the case for the 9.0-, 4.5-, and 2.3-kW cooler sizes (Figures 5-11, 5-12, and 5-13, respectively). As previously discussed, the addition of a desiccant cooler to a solar heating system may result in cost-effectiveness. However, even if the cost/benefit ratio is greater than 1.0, the addition of a desiccant cooling system results in larger allowable collector area and greater fossil fuel displacement for the same cost/benefit as a heating-only system. For example, in Figure 5-12 a heating system with collector area of 10 m^2 has a cost/benefit ratio of 1.23 and a fossil fuel displacement of 6% at a collector cost of $215/m^2$. For the same cost/benefit ratio and collector cost, a solar heating and 4.5-kW desiccant cooling system would have a collector area of approximately 30 m^2 and a 59% fossil fuel displacement. Thus, if solar heating can be justified (either economically or for fossil fuel conservation) for residential use in Washington, D.C., the addition of desiccant cooling will significantly increase the potential fossil fuel displacement at the same cost/benefit ratio.

Figures 5-14 through 5-20 show the same plots for Dodge City. These results are for a solar system with no auxiliary cooling, so the house air conditions leave the comfort zone during the times when the desiccant cooler cannot meet the house cooling load.

As in the case of Washington, D.C., solar heating is not cost competitive in Dodge City under the cost assumptions of this study. In addition, solar heating plus desiccant cooling is never cost competitive for 9.0-kW (Figure 5-14) or 4.5-kW (Figure 5-15) desiccant systems. The only cost competitive system is a 2.3-kW desiccant cooler (Figure 5-16) with a 20-m^2 collector area and a collector cost of $108/m^2$. The cost/benefit ratios tend to be higher in Dodge City than in Washington, D.C., and rise at much faster rates with increasing collector area.

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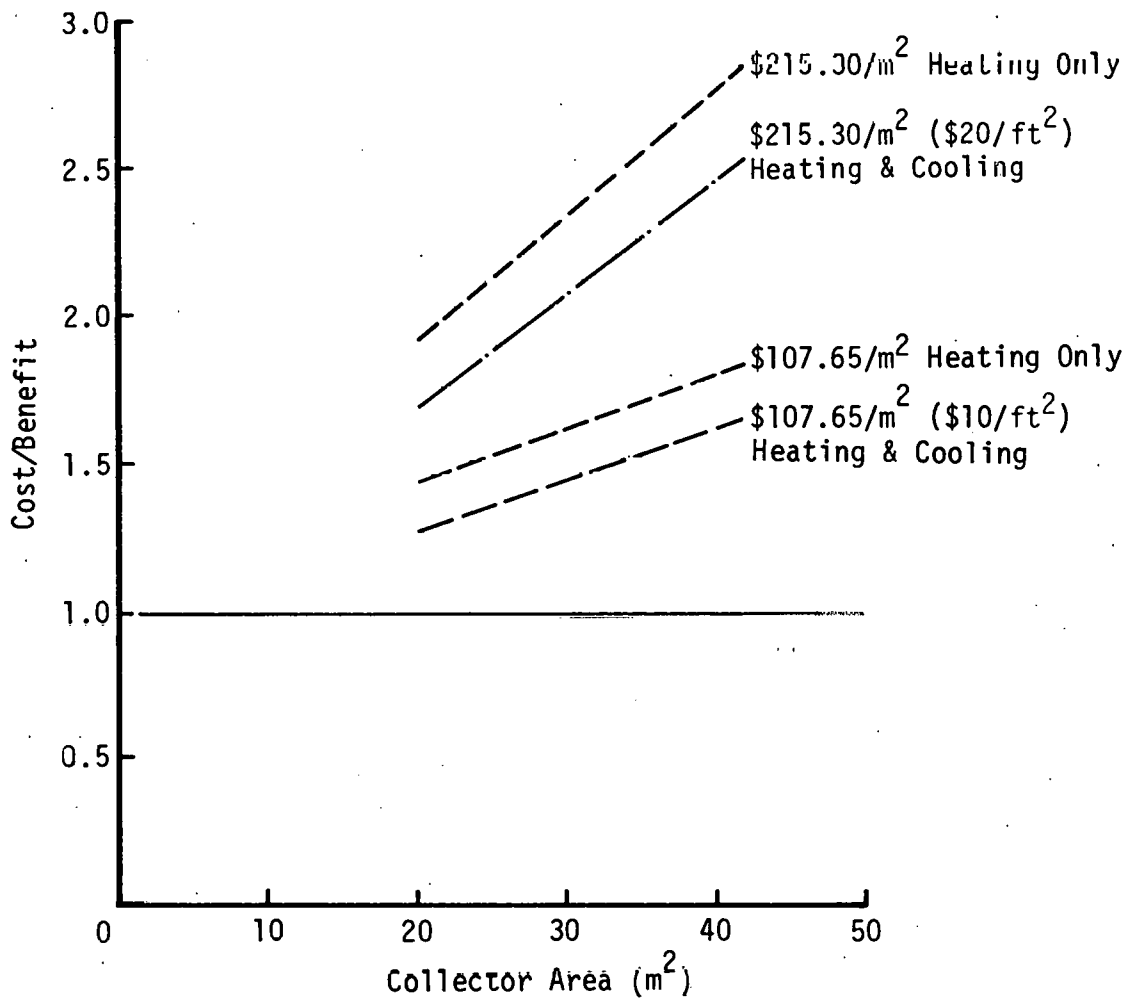
Figure 5-14. COST/BENEFIT VS. COLLECTOR AREA OF A 9.0-KW COOLER WITHOUT AUXILIARY IN DODGE CITY (1985 BASE YEAR)
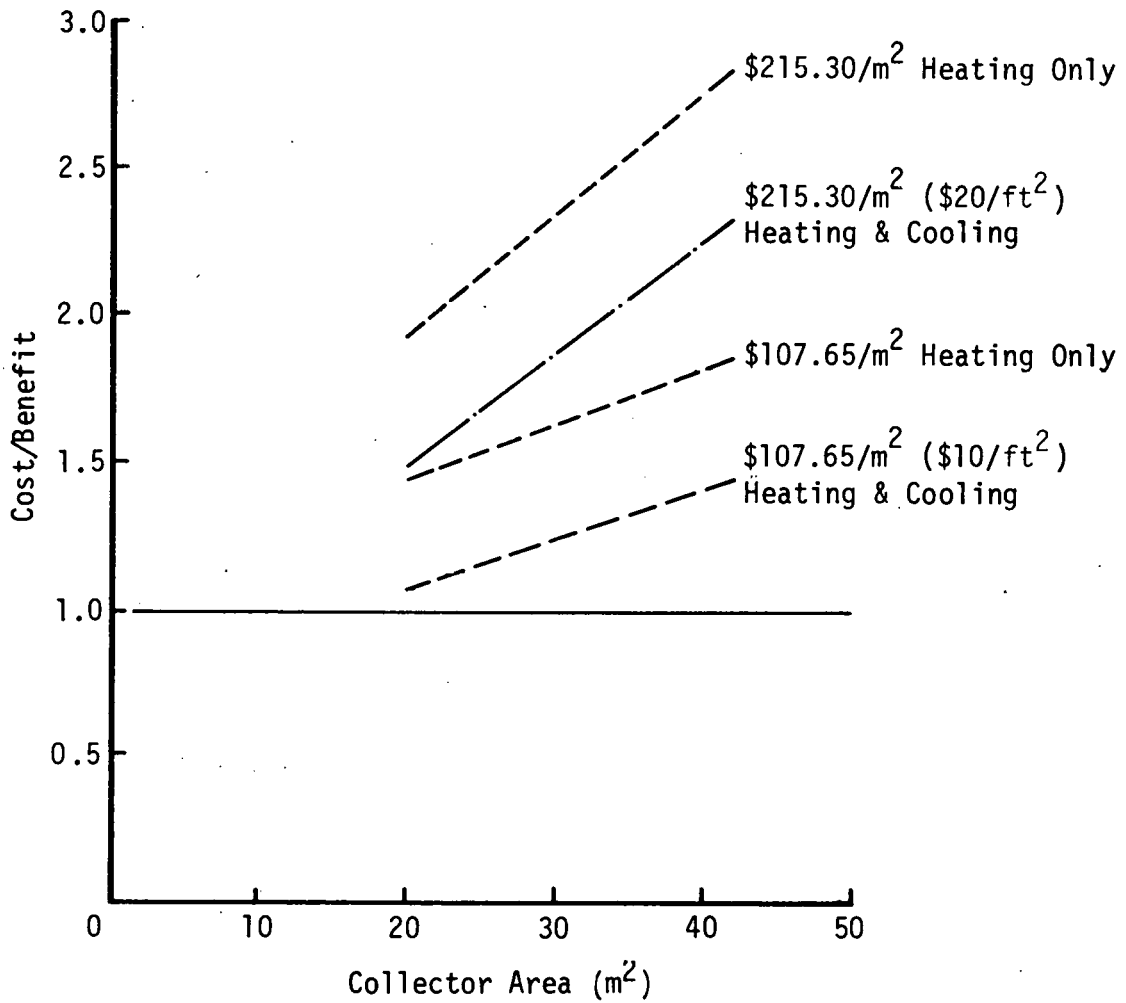
Figure 5-15. COST/BENEFIT VS. COLLECTOR AREA OF A 4.5-KW COOLER WITHOUT AUXILIARY IN DODGE CITY (1985 BASE YEAR)
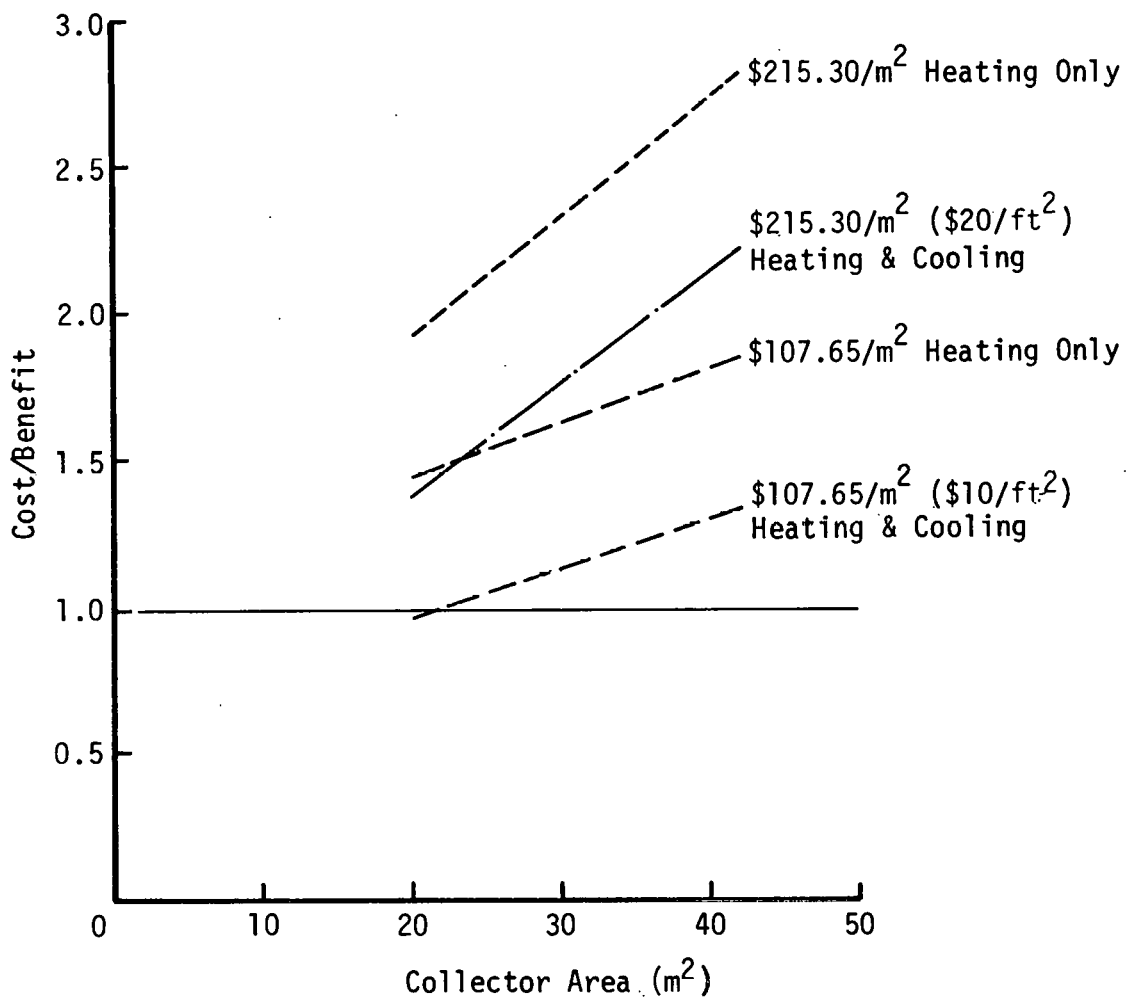
Figure 5-16. COST/BENEFIT VS. COLLECTOR AREA OF A 2.3-KW COOLER WITHOUT AUXILIARY IN DODGE CITY (1985 BASE YEAR)
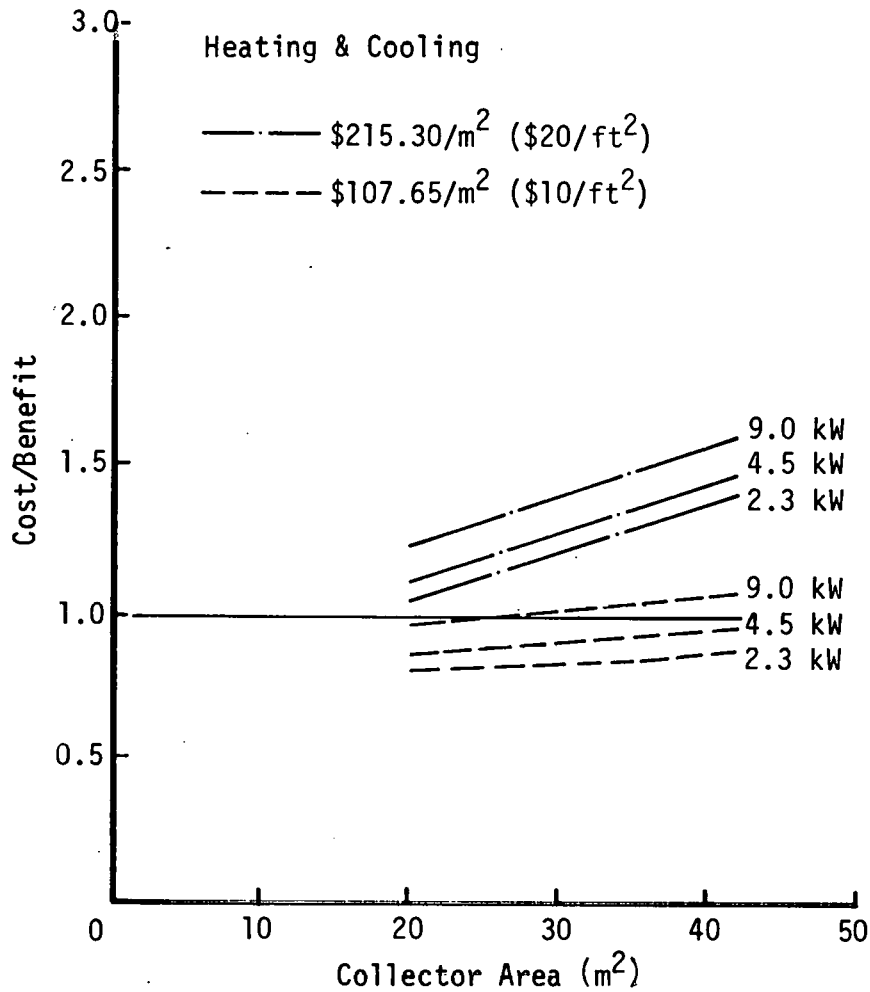
Figure 5-17. COST/BENEFIT VS. COLLECTOR AREA OF HEATING AND COOLING SYSTEMS OF VARIOUS
SIZES AND COSTS IN DODGE CITY (1985 BASE YEAR; WASHINGTON, D.C.,ENERGY COSTS)

3.0

2.5

$215.30/m² ($20/ft²)
Heating & Cooling

2.0

Cost/Benefit

$215.30/m²
Heating Only

$107.65/m² ($10/ft²)
Heating & Cooling

42 m²

1.5

$107.65/m²
Heating Only
20 m²

35 m²

35 m²

42 m²

20 m²

1.0

0.5

0

0   10   20   30   40   50   60   70   80

% Fossil Fuel Energy Displaced
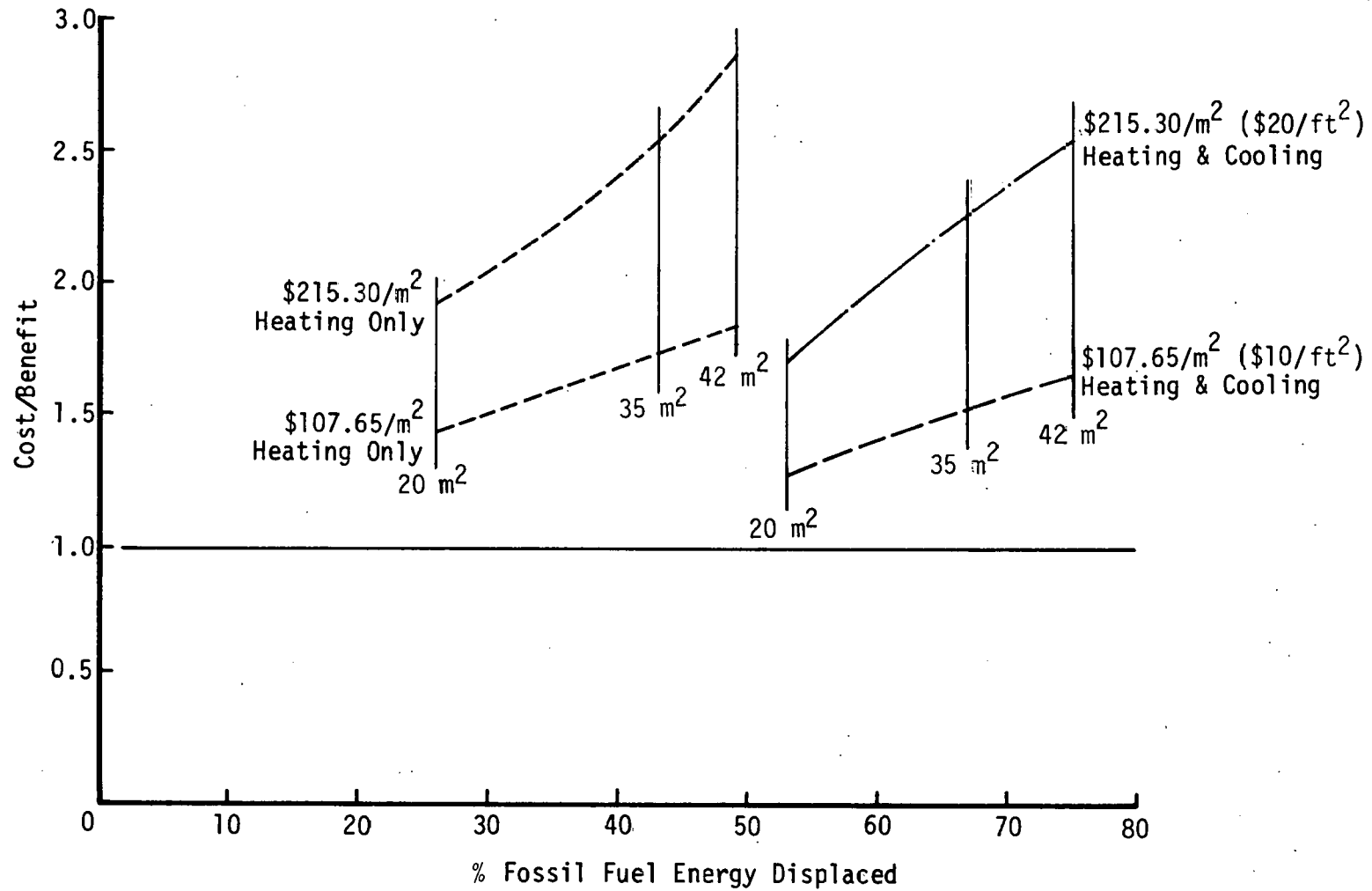
Figure 5-18.   COST/BENEFIT VS. FOSSIL FUEL ENERGY DISPLACED BY A 3.0-KW COOLER WITHOUT
AUXILIARY AT VARIOUS COLLECTOR AREAS AND COSTS IN DODGE CITY (1985 BASE YEAR)
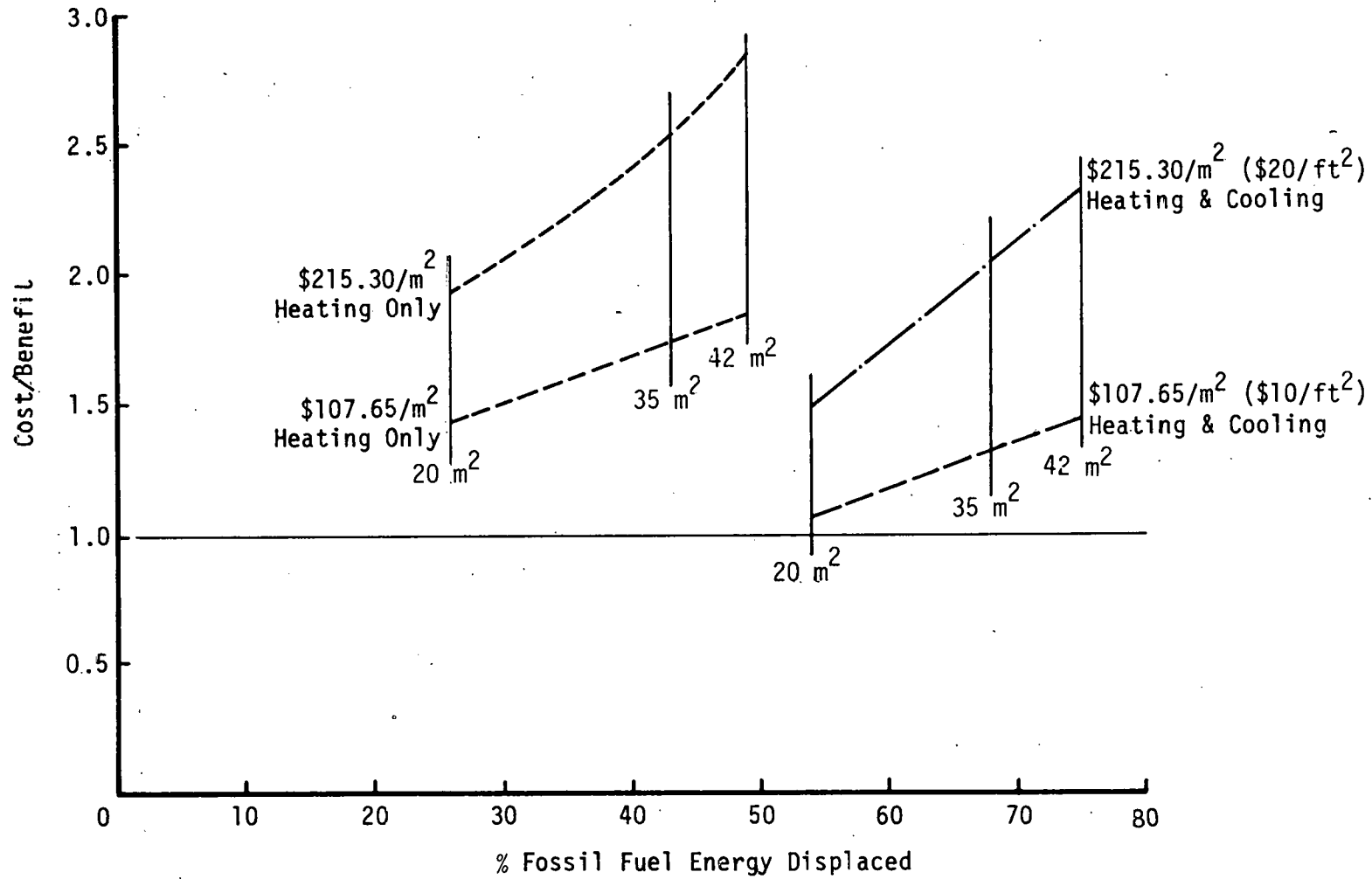
Figure 5-19. COST/BENEFIT VS. FOSSIL FUEL ENERGY DISPLACED BY A 4.5-KW COOLER WITHOUT
AUXILIARY AT VARIOUS COLLECTOR AREAS AND COSTS IN DODGE CITY (1985 BASE YEAR)
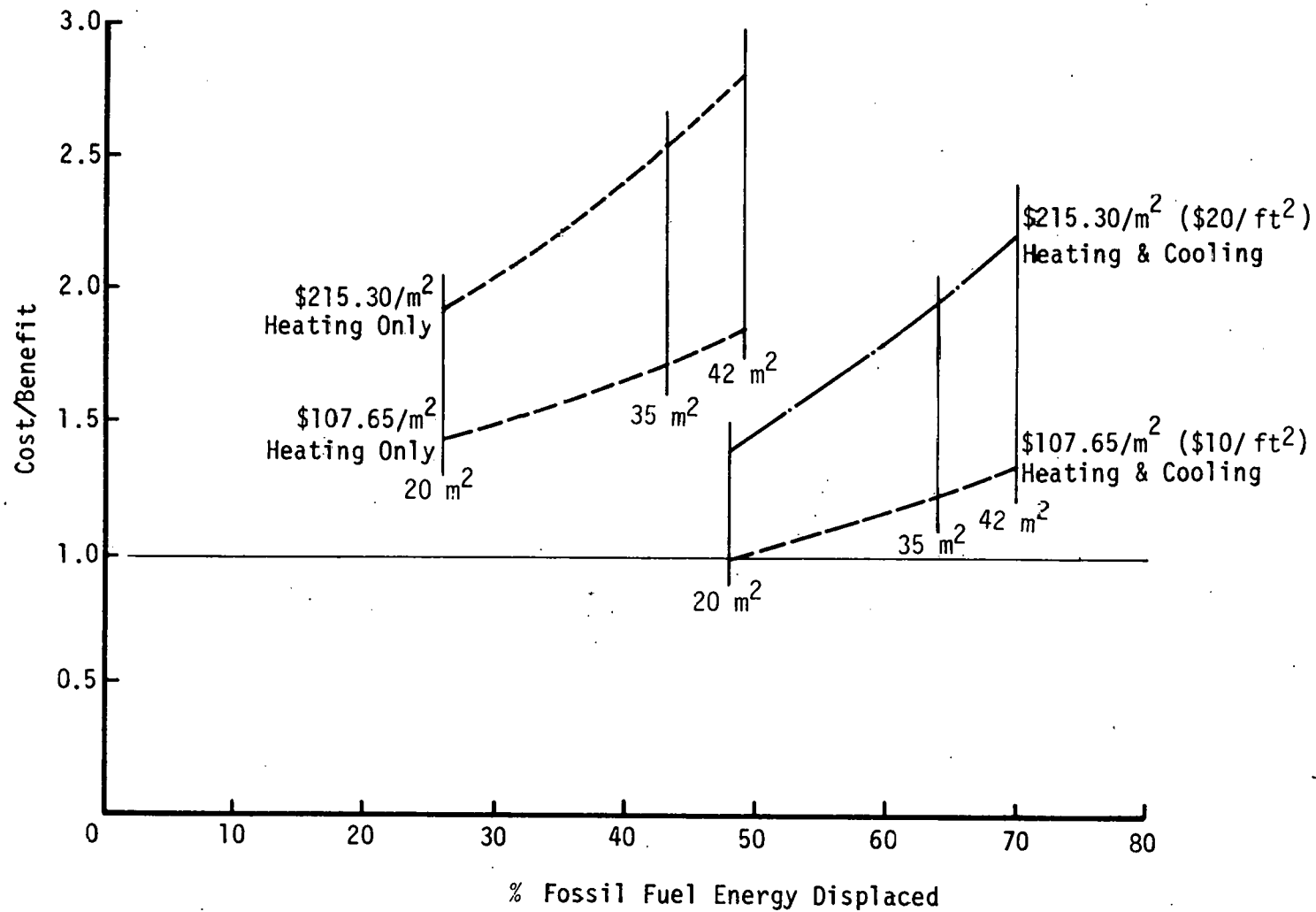
Figure 5-20. COST/BENEFIT VS. FOSSIL FUEL ENERGY DISPLACED BY A 2.3-KW COOLER WITHOUT
AUXILIARY AT VARIOUS COLLECTOR AREAS AND COSTS IN DODGE CITY (1985 BASE YEAR)

As previously discussed, the high cost/benefit ratios for Dodge City result from the combination of low natural gas rates and a relatively high heating load. To illustrate this point, Figure 5-17 is a plot of cost/benefit ratio versus collector area for Dodge City using Washington, D.C., electricity and natural gas rates. A large decrease in the cost/benefit ratio occurs; the desiccant systems are cost-effective for part or all of the collector sizes from 20 to 42 m^2 at $108/$m^2$. Thus, with sufficiently high-cost conventional energy sources (on the order of Washington, D.C., 1978 prices), solar heating and desiccant coolers in the 2.3- to 4.5-kW range, without auxiliary cooling, can be cost-effective in Dodge City at collector costs up to approximately $161/$m^2$. The addition of solar hot water systems would result in further increases in cost-effectiveness.

Figures 5-18, 5-19, and 5-20 show cost/benefit ratio plotted versus displaced fossil fuel for 9.0-, 4.5-, and 2.3-kW solar heating-only and heating/desiccant-cooling systems in Dodge City at collector costs of $107.65/$m^2$ and $215.30/$m^2$. The percentage of fossil fuel displaced increases with the addition of desiccant cooling in all cases. As before, desiccant cooling plus heating results in a larger allowable collector area and greater fossil fuel displacement for the same cost/benefit ratio as a heating-only system. A solar heating system with a 20-m^2 collector area displaces 26% of the fossil fuel requirement. With $215/$m^2$ collectors the cost/benefit ratio is 1.92. The addition of a 4.5-kW desiccant cooler results in approximately a 35-m^2 collector area and a fossil fuel displacement of 65% with the same cost/benefit ratio and collector cost (see Figure 5-19). This represents an increase in fossil fuel displacement by a factor greater than two.

SECTION 6.0

CONCLUSIONS AND RECOMMENDATIONS

This report describes the results of the development and analysis of computer simulations of solar space heating, solar desiccant cooling (using an axial-flow, disc-type dehumidifier with silica gel), and electrically driven vapor-compression air-conditioning systems for residential heating and cooling. The objective of the study was to evaluate residential solar desiccant cooling potential and identify optimum system configurations, climatic conditions, and economic parameters. The system chosen for study was a disc-type desiccant bed configuration in the recirculation mode (Figures 2-3 and 2-4).

The TRNSYS computer program, in conjunction with special desiccant cooler routines, proved to be a valuable tool in evaluating the seasonal performance of solar heating/desiccant-cooler systems at the five locations studied. The ability to perform a transient analysis is important because the effects of varying ambient conditions and solar energy input to a desiccant cooler cannot be detected in a limited number of steady-state analyses. The results from the TRNSYS simulations, the economic analyses, and the vapor-compression cooler simulation programs allow evaluation and comparison of the various systems.

As described in the previous sections, several solar heating/desiccant-cooling annual simulations were performed for five U.S. cities with diverse climatic conditions. Five conclusions regarding system performance were drawn from these simulations:

- Those climates that experience both high heating loads in the winter and high cooling loads in the summer are the most suitable candidates for solar heating and desiccant cooling. By having large heating and cooling loads, the solar system can be matched to the heating and cooling energy requirements without additional collector area being required for the cooling season. The solar system can then be utilized year-round to the maximum extent.

- Washington, D.C., is the most promising of the five cities examined for the application of a solar heating and desiccant cooling system with a vapor-compression cooler backup. This results from the favorable climatic conditions, namely, the high heating and cooling loads and high relative humidity. Dodge City, Kans., shows promise for solar heating and desiccant cooling without vapor-compression cooler backup. Although the room conditions in a Dodge City home would occasionally leave the comfort zone during peak cooling loads, a reasonable degree of comfort could be maintained during the summer months without auxiliary cooling. Other locations with similar climatic conditions have the same promise as these two cities.

- The desiccant cooler system which was modeled is preferentially a latent rather than sensible cooler. The steady-state simulations clearly showed that the potential latent cooling capacity is larger than the potential sensible capacity at a given desiccant regeneration temperature. Thus, the modeled desiccant cooler system would

provide optimum cooling performance in humid climates such as Washington, D.C., and would perform less efficiently in dry climates with large sensible cooling loads, such as Phoenix, Ariz.

- The thermal performance of the desiccant cooler is best understood in terms of the interaction of the desiccant system with the solar collector/storage outlet (regeneration) temperature. Due to the cyclic nature of the solar input, the average regeneration temperature from the collectors is a major factor in the optimum coupling of solar collectors with desiccant coolers.

- For combined solar space heating/desiccant-cooling systems, the optimum matches of system sizes and collector areas for Washington, D.C., (with auxiliary cooling) are a 4.5-kW desiccant cooler with 35-m^2 collector area and a 9.0-kW desiccant cooler with 50-m^2 collector area. This selection is based on the objective of maximizing both cooling capacity and desiccant cooler off-time (that is, maximizing EER). These system sizes are smaller than those typically found in Washington, D.C., homes, so the vapor-compression backup would meet a portion of the cooling load. This strategy is preferable to large solar desiccant coolers that are sized to meet the full cooling load with no backup for two reasons: 1) solar desiccant coolers sized to meet peak cooling loads would be underutilized during a major portion of the cooling season; and 2) the collector area required for such a desiccant cooler would probably far exceed that required for heating, even if the heating and cooling loads were nearly equal, resulting in underutilization of the collectors during the heating season. Sizing desiccant coolers for peak cooling loads would tend to reduce the cost-effectiveness of the solar systems.

With the economics analysis program and cost assumptions described in Section 5.0, the present worth of 20-year life-cycle costs was determined for each of the annual solar space heating/desiccant-cooler simulations (Section 3.7.3) and natural-gas-space-heating/vapor-compression-cooler simulations (Section 4.0). The percentage of fossil fuel displaced by the solar systems was also determined. Conclusions resulting from this analysis for combined heating and cooling systems (assuming a 1985 system installation) are as follows:

- Economic considerations indicate that humid climates with relatively high conventional fuel costs are the preferred conditions for the solar desiccant cooler modeled in this study.

- Analysis of life-cycle cost and fossil fuel displacement indicates that a 4.5-kW desiccant cooler with a 35-m^2 collector area and auxiliary vapor-compression cooling backup is the preferred configuration for Washington, D.C.; with no backup cooling, it is the preferred configuration for Dodge City.

- The solar heating/desiccant-cooling system with backup approaches cost competitiveness in Washington, D.C., at installed collector costs under $215/$m^2$, but not in Dodge City (without backup), due to low costs for conventional fossil fuels in the latter location.

- When a desiccant cooling system is added to a solar space heating system, a significant increase in displaced fossil fuel energy can be achieved. For the same cost/benefit ratio (cost-effectiveness) and collector cost per square meter, the addition of a desiccant cooler results in a larger economically allowable collector area and a significant increase in fossil fuel displacement. Thus, if residential solar space heating can be justified (either economically or by fossil fuel conservation), then the addition of a desiccant cooler in certain climates can significantly increase fossil fuel displacement without affecting system cost-effectiveness.

Potential system configurations for climates not examined in detail in this study can be proposed. For the hot, dry climate of Phoenix, Ariz., a system consisting of a relatively small solar-regenerated desiccant wheel and a large evaporative cooler could potentially provide the required degree of comfort. In such a dry climate, the large evaporative cooler should be able to provide the required sensible cooling. The addition of a desiccant wheel would maintain the relative humidity of the conditioned space within acceptable limits. For a warm, very humid climate such as Charleston, S.C., a large solar-regenerated desiccant wheel combined with a relatively small evaporative cooler should be the optimum system. A large desiccant wheel is required to meet the large latent cooling load.

Thus, it should be possible to alter the cooling capacities of the individual components of a solar desiccant cooling system, namely, the desiccant wheel (latent) and evaporative cooler (sensible), in order to match the system to local climatic conditions. This is a distinct advantage over other cooler types for which the ratio of sensible to latent cooling output is fixed. System costs should be reduced by matching the sizes of the components with local requirements to minimize excessive latent or sensible capacities. However, for each case the total solar heating/desiccant-cooler system must be compared on a life-cycle basis to the conventional system employing energy from fossil fuels to determine if the solar desiccant cooler will be cost-effective.

In spite of the apparent potential for solar space heating/desiccant-cooler systems in certain climatic conditions, additional work remains to be done to prepare solar desiccant cooler systems for residential application. The major areas of work include:

- selection of the optimum desiccant dehumidifier bed configuration (disc-type, as modeled in this study, or alternates, such as cross-cooled isothermal beds or concentric drums);

- optimization of other desiccant cooler system components, such as the regenerative heat exchangers and evaporative coolers;

- construction and testing of prototype desiccant cooler systems;

- definition of expected desiccant cooler system capital costs and methods, if any, of cost reduction;

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- minimization of parasitic power required for desiccant coolers; and

- comparison of solar desiccant coolers with competing cooler types in terms of cost-effectiveness and fossil fuel displacement.

SERI and several other organizations under contract to the U.S. Department of Energy are continuing work in these areas in order to define the future role of solar desiccant cooling. The analysis techniques developed in this study should prove to be useful in comparing the performance of various cooling systems and in determining which system(s) should ultimately be carried through the commercialization stage for the residential market.

SECTION 7.0

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SERI

APPENDIX A

SAMPLE RESULTS FROM SIMULATION OF ANNUAL SOLAR HEATING AND DESICCANT
COOLING PERFORMANCE

SERI◉

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**********************************
*                                *
*  DESICCANT COOLING SIMULATION  *
*                                *
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INPUT PARAMETERS

LOCATION : WASHINGTON, D.C.   LATITUDE : 38.83 DEGREES N

YEAR : 1959

HOUSE DESCRIPTION :

        TWO STORY, 167.3 M2 FLOOR SPACE
        178.4 M2 OUTSIDE WALL SPACE
        PITCH OF ROOF : 39.0 DEGREES
        NORTH-SOUTH ORIENTATION
        U = 1.33 J/SEC-M2-C (WALLS + WINDOWS)
        U =   .28 J/SEC-M2-C (ROOF)

COOLER UNIT DESCRIPTION :

        NOMINAL OPERATING CAPACITY AT 29.4 C AND 50 RH : 4.5 KW
        DEHUMIDIFIER :
            PACKED BED
            GEL TYPE : SILICA
            PARTICLE SURFACE AREA PER UNIT DEHUMIDIFIER VOLUME :   1573 M2/M3
            WHEEL DIAMETER :   .97 M
            FLOW LENGTH : .0279 M
            WHEEL RPM :   .09180

SENSIBLE AND LATENT AUXILIARY COOLING TO HOUSE

COLLECTOR :

        FLAT PLATE (AIR)
        AREA :   35.0 M2

STORAGE :

        PEBBLE BED
        VOLUME :   13.1 M3

SOLAR DESICCANT   WASH25
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
INCIDENT SOLAR (J)	1.14E+10	1.32E+10	1.91E+10	1.80E+10	2.05E+10	2.05E+10	1.79E+10	1.98E+10	2.01E+10	1.56E+10	1.06E+10	9.36E+09	1.96E+11
USEFUL SOLAR COLLECTED (J)	4.12E+09	5.03E+09	7.46E+09	5.79E+09	6.21E+09	6.93E+09	6.35E+09	7.44E+09	6.86E+09	5.50E+09	3.98E+09	3.36E+09	6.90E+10
COLLECTOR EFFICIENCY, PCT	36.	38.	39.	32.	30.	34.	36.	38.	34.	35.	37.	36.	35.

BUILDING LOAD

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	5.76E+07	1.87E+09	4.06E+09	5.12E+09	6.14E+09	3.29E+09	1.10E+09	0.	0.	2.16E+10
SENSIBLE (J)	0.	0.	0.	-5.45E+07	9.33E+08	2.89E+09	2.88E+09	3.92E+09	1.87E+09	3.89E+08	0.	0.	1.28E+10
LATENT (J)	0.	0.	0.	1.12E+08	9.34E+08	1.18E+09	2.24E+09	2.22E+09	1.41E+09	7.11E+08	0.	0.	8.80E+09
HEATING (J)	1.27E+10	9.17E+09	6.65E+09	1.56E+09	1.06E+08	6.25E+07	1.31E+07	2.45E+07	5.81E+07	1.54E+09	6.52E+09	9.70E+09	4.81E+10
TOTAL (J)	1.27E+10	9.17E+09	6.65E+09	1.62E+09	1.97E+09	4.13E+09	5.13E+09	6.16E+09	3.34E+09	2.64E+09	6.52E+09	9.70E+09	6.98E+10

COOLING CAPA-
CITY SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
(J)	0.	0.	0.	1.46E+08	1.55E+09	2.33E+09	2.85E+09	3.28E+09	2.10E+09	8.48E+08	0.	0.	1.31E+10
SENSIBLE (J)	0.	0.	0.	4.29E+07	6.38E+08	1.35E+09	1.48E+09	1.76E+09	9.88E+08	2.78E+08	0.	0.	6.53E+09
LATENT (J)	0.	0.	0.	1.03E+08	9.12E+08	9.83E+08	1.37E+09	1.52E+09	1.11E+09	5.70E+08	0.	0.	6.58E+09

SOLAR ENERGY
SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	7.90E+08	2.34E+09	3.89E+09	4.47E+09	5.30E+09	3.49E+09	1.65E+09	0.	0.	2.19E+10
HEATING (J)	4.26E+09	4.87E+09	6.65E+09	1.56E+09	1.06E+08	6.25E+07	1.31E+07	2.45E+07	5.81E+07	1.54E+09	3.84E+09	3.28E+09	2.59E+10

AUXILIARY
SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	0.	4.19E+08	1.81E+09	2.20E+09	2.86E+09	1.22E+09	3.79E+08	0.	0.	8.89E+09
HEATING (J)	8.39E+09	4.29E+09	0.	0.	0.	0.	0.	0.	0.	0.	2.78E+09	6.42E+09	2.19E+10

PERCENT BLDG
LOAD BY SOLAR

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	100.	79.	56.	56.	53.	63.	69.	0.	0.	60.
HEATING (J)	34.	53.	100.	100.	100.	100.	100.	100.	100.	100.	58.	34.	55.

PERCENT COLLEC-
TED SOLAR USED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	26.	38.	56.	70.	71.	51.	61.	0.	0.	55.
HEATING (J)	100.	97.	89.	27.	2.	1.	0.	0.	1.	28.	96.	98.	88.

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP.	OCT	NOV	DEC	TOTAL
TOTAL PARASITIC ENERGY (J)	0.	0.	0.	4.90E+07	2.85E+08	4.32E+08	6.23E+08	6.35E+08	4.21E+08	1.98E+08	0.	0.	2.64E+09
COOLING COP	3.00	0.00	0.00	.18	.66	.60	.64	.62	.60	.51	0.00	0.00	.60
COOLING EER	0.00	0.00	0.00	10.18	14.83	12.22	11.47	11.79	12.47	11.98	0.00	0.00	12.11
DESICCANT DUTY CYCLE	0.00	0.00	0.00	.03	.15	.25	.37	.38	.25	.23	0.00	0.00	.25
PCT IN COMFORT	100.	100.	100.	99.	100.	100.	100.	100.	100.	100.	100.	100.	100.

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*                                *
*  DESICCANT COOLING SIMULATION  *
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INPUT PARAMETERS

LOCATION : PHEONIX, ARIZONA   LATITUDE : 33.43 DEGREES N

YEAR : 1962

HOUSE DESCRIPTION :

      TWO STORY, 167.3 M2 FLOOR SPACE
      173.4 M2 OUTSIDE WALL SPACE
      PITCH OF ROOF : 33.4 DEGREES
      NORTH-SOUTH ORIENTATION
      U = 1.33 J/SEC-M2-C (WALLS + WINDOWS)
      U =   .28 J/SEC-M2-C (ROOF)

COOLER UNIT DESCRIPTION :

      NOMINAL OPERATING CAPACITY AT 29.4 C AND 50 RH : 4.5 KW
      DEHUMIDIFIER :
         PACKED BED
         GEL TYPE : SILICA
         PARTICLE SURFACE AREA PER UNIT DEHUMIDIFIER VOLUME :   1573 M2/M3
         WHEEL DIAMETER :   .97 M
         FLOW LENGTH : .0275 M
         WHEEL RPM :   .09180

SENSIBLE AND LATENT AUXILIARY COOLING TO HOUSE

COLLECTOR :

      FLAT PLATE (AIR)
      AREA :   35.0 M2

STORAGE :

      PEBBLE BED
      VOLUME :   13.1 M3

SOLAR DESICCANT
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
INCIDENT SOLAR (J)	1.86E+10	1.88E+10	2.58E+10	2.95E+10	3.07E+10	2.91E+10	2.83E+10	2.90E+10	2.49E+10	2.65E+10	2.03E+10	1.80E+10	2.99E+11
USEFUL SOLAR COLLECTED (J)	6.89E+09	6.66E+09	9.45E+09	1.14E+10	1.19E+10	1.23E+10	1.23E+10	1.28E+10	1.04E+10	9.94E+09	6.91E+09	6.90E+09	1.18E+11
COLLECTOR EFFICIENCY, PCT	37.	35.	37.	39.	39.	42.	44.	44.	42.	38.	34.	38.	39.

BUILDING LOAD

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	2.63E+08	3.51E+09	4.20E+09	8.43E+09	1.17E+10	1.24E+10	8.60E+09	3.43E+09	4.94E+08	0.	5.30E+10
SENSIBLE (J)	0.	0.	2.98E+08	3.52E+09	4.21E+09	8.30E+09	1.12E+10	1.19E+10	7.80E+09	3.41E+09	5.12E+08	0.	5.11E+10
LATENT (J)	0.	0.	-3.46E+07	-4.89E+06	-4.01E+06	1.26E+08	5.38E+08	4.66E+08	7.95E+08	2.34E+07	-1.33E+07	0.	1.89E+09
HEATING (J)	2.96E+09	9.32E+08	1.03E+09	1.50E+08	9.82E+07	3.71E+05	0.	0.	0.	1.89E+08	2.65E+08	1.90E+09	7.52E+09
TOTAL (J)	2.96E+09	9.32E+08	1.29E+09	3.66E+09	4.30E+09	8.43E+09	1.17E+10	1.24E+10	8.60E+09	3.62E+09	7.58E+09	1.90E+09	6.05E+10

COOLING CAPACITY SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
	0.	0.	2.16E+08	1.74E+09	1.91E+09	3.07E+09	4.17E+09	4.14E+09	3.43E+09	1.63E+09	4.72E+08	0.	2.08E+10
SENSIBLE (J)	0.	0.	2.15E+08	1.74E+09	1.91E+09	2.96E+09	3.67E+09	3.64E+09	2.75E+09	1.55E+09	4.72E+08	0.	1.89E+10
LATENT (J)	0.	0.	1.20E+06	1.26E+06	2.90E+06	1.07E+08	5.04E+08	5.00E+08	6.76E+08	7.53E+07	6.83E+04	0.	1.87E+09

SOLAR ENERGY SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	9.08E+08	4.28E+09	4.87E+09	7.71E+09	9.31E+09	9.38E+09	6.90E+09	3.82E+09	1.27E+09	0.	4.84E+10
HEATING (J)	2.96E+09	9.32E+08	1.03E+09	1.50E+08	9.82E+07	3.71E+05	0.	0.	0.	1.89E+08	2.65E+09	1.90E+09	6.79E+09

AUXILIARY SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	1.64E+08	1.92E+09	2.36E+09	5.25E+09	7.38E+09	8.15E+09	5.05E+09	2.07E+09	3.39E+08	0.	3.27E+10
HEATING (J)	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.

PERCENT BLDG LOAD BY SOLAR

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	57.	47.	45.	37.	36.	34.	40.	44.	58.	0.	39.
HEATING (J)	100.	100.	100.	100.	100.	100.	100.	100.	100.	100.	100.	100.	100.

PERCENT COLLECTED SOLAR USED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	39.	37.	41.	63.	76.	73.	67.	38.	23.	0.	54.
HEATING (J)	43.	14.	11.	1.	1.	0.	0.	0.	0.	2.	4.	28.	23.

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
TOTAL PARASITIC ENERGY (J)	0.	0.	4.82E+07	3.17E+08	3.49E+08	5.70E+08	7.92E+08	7.81E+08	6.23E+08	3.03E+08	1.02E+08	0.	3.89E+09
COOLING COP	0.00	0.00	.24	.41	.39	.40	.45	.44	.50	.43	.37	0.00	.43
COOLING EER	0.00	0.00	11.40	11.51	11.28	10.64	10.54	10.38	10.96	11.17	11.64	0.00	10.76
DESICCANT DUTY CYCLE	0.00	0.00	.09	.18	.19	.34	.47	.47	.38	.16	.06	0.00	.28
PCT IN COMFORT	100.	100.	97.	99.	100.	100.	100.	100.	100.	99.	98.	100.	99.

A-6

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*  OESICCANT COOLING SIMULATION *        FTWR 6
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INPUT PARAMETERS

LCCATION : FORT WORTH, TEXAS LATITUDE : 32.83 DEGREES N

YEAR : 1960

HOUSE DESCRIPTION :

    TWO STORY, 167.3 M2 FLCOR SPACE
    178.6 M2 OUTSIDE WALL SPACE
    PITCH CF ROOF : 33.G DEGREES
    NORTH-SOUTH ORIENTATION
    U = 1.33 J/SEC-M2-C (WALLS + WINDOWS)
    U =   .28 J/SEC-M2-C (ROOF)

CCOLER UNIT DESCRIPTION :

    NOMINAL OPERATING CAPACITY AT 29.4 C AND 50 RH : 4.5 KW
    DEHUMIDIFIER :
        PACKED BEC
        GEL TYPE : SILICA
        PARTICLE SURFACE AREA PER UNIT DEHUMIDIFIER VOLUME :   1573 M2/M3
        WHEEL DIAMETER :   .97 M
        FLOW LENGTH : .0275 M
        WHEEL RPM :   .09180

SENSIBLE AND LATENT AUXILIARY COOLING TO HOUSE

COLLECTOR :

    FLAT PLATE (AIR)
    AREA :   35.C M2

STORAGE :

    PEBBLE BED
    VOLUME :   13.1 M3

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
INCIDENT SOLAR (J)	1.33E+10	1.46E+10	1.76E+10	2.03E+10	2.32E+10	2.43E+10	2.40E+10	2.40E+10	2.28E+10	1.95E+10	1.59E+10	1.06E+10	2.30E+11
USEFUL SOLAR COLLECTED (J)	4.92E+09	5.44E+09	6.12E+09	6.70E+09	7.81E+09	9.26E+09	9.43E+09	9.40E+09	8.69E+09	6.93E+09	5.81E+09	3.77E+09	8.43E+10
COLLECTOR EFFICIENCY,PCT	37.	37.	35.	33.	34.	38.	39.	39.	38.	36.	37.	36.	37.
BUILDING LOAD													
COOLING (J)	0.	0.	0.	8.96E+08	2.67E+09	7.03E+09	8.27E+09	8.22E+09	5.55E+09	2.23E+09	0.	0.	3.49E+10
SENSIBLE (J)	0.	0.	0.	1.49E+08	1.50E+09	5.27E+09	5.93E+09	5.77E+09	4.29E+09	8.35E+08	0.	0.	2.37E+10
LATENT (J)	0.	0.	0.	7.47E+08	1.17E+09	1.75E+09	2.34E+09	2.46E+09	1.26E+09	1.40E+09	0.	0.	1.11E+10
HEATING (J)	8.06E+09	7.26E+09	6.07E+09	1.83E+08	6.76E+07	0.	0.	0.	6.27E+07	1.25E+08	1.83E+09	8.83E+09	3.25E+10
TOTAL (J)	8.06E+09	7.26E+09	6.07E+09	1.08E+09	2.74E+09	7.03E+09	8.27E+09	8.22E+09	5.62E+09	2.36E+09	1.83E+09	8.83E+09	6.73E+10
COOLING CAPACITY SUPPLIED	0.	0.	0.	1.69E+09	1.93E+09	3.74E+09	4.34E+09	4.26E+09	3.13E+09	1.97E+09	0.	0.	2.04E+10
SENSIBLE (J)	0.	0.	0.	3.08E+08	9.00E+08	2.27E+09	2.42E+09	2.37E+09	1.89E+09	6.31E+08	0.	0.	1.08E+10
LATENT (J)	0.	0.	0.	7.81E+08	1.03E+09	1.47E+09	1.92E+09	1.89E+09	1.24E+09	1.33E+09	0.	0.	9.63E+09
SOLAR ENERGY SUPPLIED													
COOLING (J)	0.	0.	0.	2.36E+09	3.10E+09	6.24E+09	7.09E+09	7.01E+09	5.24E+09	3.69E+09	0.	0.	3.46E+10
HEATING (J)	4.88E+09	5.24E+09	5.14E+09	1.83E+08	6.76E+07	0.	0.	0.	6.27E+07	1.25E+08	1.83E+09	3.70E+09	2.08E+10
AUXILIARY SUPPLIED													
COOLING (J)	0.	0.	0.	1.30E+08	7.05E+08	3.19E+09	3.83E+09	3.87E+09	2.56E+09	5.98E+08	0.	0.	1.49E+10
HEATING (J)	3.18E+09	2.02E+09	9.23E+08	0.	0.	0.	0.	0.	0.	0.	0.	5.13E+09	1.13E+10
PERCENT BLDG LOAD BY SOLAR													
COOLING (J)	0.	0.	0.	89.	73.	54.	53.	52.	55.	77.	0.	0.	58.
HEATING (J)	61.	72.	85.	100.	100.	100.	100.	100.	100.	100.	100.	42.	65.
PERCENT COLLECTED SOLAR USED													
COOLING (J)	0.	0.	0.	35.	40.	67.	75.	75.	60.	52.	0.	0.	59.
HEATING (J)	99.	96.	84.	3.	1.	0.	0.	0.	1.	2.	31.	98.	80.
TOTAL PARASITIC ENERGY (J)	0.	0.	0.	2.34E+08	3.49E+08	6.56E+08	7.92E+08	7.78E+08	5.30E+08	3.88E+08	0.	0.	3.73E+09
COOLING COP	0.00	0.00	0.00	.46	.61	.60	.61	.61	.60	.55	0.00	0.00	.59
COOLING EER	0.00	0.00	0.00	14.54	14.12	12.25	12.00	11.93	12.49	13.95	0.00	0.00	12.45
DESICCANT DUTY CYCLE	0.00	0.00	0.00	.13	.19	.40	.47	.46	.32	.22	0.00	0.00	.31
PCT IN COMFORT	100.	100.	100.	99.	100.	100.	100.	100.	100.	100.	100.	100.	100.

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*  DESICCANT COOLING SIMULATION *              DODG12
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INPUT PARAMETERS

LOCATION : DODGE CITY, KS   LATITUDE : 37.77 DEGREES N

YEAR : 1955

HOUSE DESCRIPTION :

        TWO STORY, 167.3 M2 FLOOR SPACE
        179.4 M2 OUTSIDE WALL SPACE
        PITCH OF ROOF : 40.0 DEGREES
        NORTH-SOUTH ORIENTATION
        U = 1.33 J/SEC-M2-C (WALLS + WINDOWS)
        U =   .28 J/SEC-M2-C (ROOF)

COOLER UNIT DESCRIPTION :

        NOMINAL OPERATING CAPACITY AT 29.4 C AND 50 RH : 4.5 KW
        DEHUMIDIFIER :
           PACKED BED
           GEL TYPE : SILICA
           PARTICLE SURFACE AREA PER UNIT DEHUMIDIFIER VOLUME :   1573 M2/M3
           WHEEL DIAMETER :   .97 M
           FLOW LENGTH : .0275 M
           WHEEL RPM :   .09190

SENSIBLE AND LATENT AUXILIARY COOLING TO HOUSE

COLLECTOR :

        FLAT PLATE (AIR)
        AREA :   35.0 M2

STORAGE :

        PEBBLE BED
        VOLUME :   13.1 M3

SOLAR DESICCANT
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
INCIDENT SOLAR (J)	1.76E+10	1.81E+10	2.41E+10	2.52E-10	2.34E+10	2.15E+10	2.64E+10	2.64E+10	2.33E+10	2.40E+10	1.82E+10	1.68E+10	2.65E+11
USEFUL SOLAR COLLECTED (J)	7.02E+09	6.92E+09	9.22E+09	8.73E-09	7.12E+09	6.91E+09	1.01E+10	1.00E+10	7.49E+09	8.39E+09	7.11E+09	6.67E+09	9.56E+10
COLLECTOR EFFICIENCY,PCT	40.	38.	38.	35.	30.	32.	38.	38.	32.	35.	39.	40.	36.

BUILDING LOAD

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	6.67E+07	2.45E+08	1.54E+09	6.11E+09	5.24E+09	1.41E+09	0.	0.	0.	1.46E+10
SENSIBLE (J)	0.	0.	0.	1.01E+08	-4.23E+05	7.92E+08	4.97E+09	4.26E+09	1.31E+09	0.	0.	0.	1.14E+10
LATENT (J)	0.	0.	0.	-3.41E+07	2.46E+08	7.51E+08	1.13E+09	9.83E+08	9.98E+07	0.	0.	0.	3.18E+09
HEATING (J)	1.22E+10	1.21E+10	7.70E+09	1.72E+09	2.60E+08	1.56E+08	1.71E+05	1.04E+07	5.22E+07	1.97E+08	9.14E+09	1.30E+10	5.66E+10
TOTAL (J)	1.22E+10	1.21E+10	7.70E+03	1.78E+09	5.05E+08	1.70E+09	6.11E+09	5.25E+09	1.46E+09	1.97E+08	9.14E+09	1.30E+10	7.12E+10

COOLING CAPACITY SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
	0.	0.	0.	1.73E+09	5.87E+08	1.46E+09	3.44E+09	2.96E+09	7.66E+08	0.	0.	0.	9.39E+09
SENSIBLE (J)	0.	0.	0.	1.67E+08	2.31E+08	5.91E+08	2.32E+09	1.99E+09	6.93E+08	0.	0.	0.	6.00E+09
LATENT (J)	0.	0.	0.	6.06E+06	3.57E+08	8.69E+08	1.12E+09	9.68E+08	7.36E+07	0.	0.	0.	3.39E+09

SOLAR ENERGY SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	9.58E+08	1.18E+09	2.62E+09	6.01E+09	5.28E+09	1.73E+09	0.	0.	0.	1.77E+10
HEATING (J)	7.07E+09	6.68E+09	7.70E+09	1.72E+09	2.60E+08	1.56E+09	1.71E+05	1.04E+07	5.22E+07	1.97E+08	7.10E+09	6.49E+09	3.68E+10

AUXILIARY SUPPLIED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	3.68E+07	6.43E+07	4.02E+08	2.63E+09	2.28E+09	6.61E+08	0.	0.	0.	6.08E+09
HEATING (J)	5.16E+09	5.42E+09	0.	0.	0.	0.	0.	0.	0.	0.	2.04E+09	6.52E+09	1.91E+10

PERCENT BLDG LOAD BY SOLAR

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	82.	90.	78.	57.	57.	54.	0.	0.	0.	61.
HEATING (J)	58.	55.	100.	100.	100.	100.	100.	100.	100.	100.	78.	50.	66.

PERCENT COLLECTED SOLAR USED

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
COOLING (J)	0.	0.	0.	19.	17.	38.	60.	53.	34.	0.	0.	0.	40.
HEATING (J)	100.	97.	84.	20.	4.	2.	0.	0.	1.	2.	100.	97.	71.

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
TOTAL PARASITIC ENERGY (J)	0.	0.	0.	5.42E+07	1.29E+08	2.78E+08	5.61E+08	4.89E+08	1.41E+08	0.	0.	0.	1.65E+09
COOLING COP	0.00	0.00	0.00	.20	.50	.56	.57	.56	.44	0.00	0.00	0.00	.53
COOLING EER	0.00	0.00	0.00	10.37	14.38	14.48	12.84	12.77	12.03	0.00	0.00	0.00	12.93
DESICCANT DUTY CYCLE	0.00	0.00	0.00	.03	.06	.15	.32	.28	.16	0.00	0.00	0.00	.18
PCT IN COMFORT	100.	100.	100.	99.	98.	99.	100.	100.	100.	100.	100.	100.	99.

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                    * DESICCANT COOLING SIMULATION *              CHAR14
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INPUT PARAMETERS

LOCATION : CHARLESTON, SC   LATITUDE : 32.90 DEGREES N

YEAR : 1963

HOUSE DESCRIPTION :

      TWO STORY, 167.3 M2 FLOOR SPACE
      178.4 M2 OUTSIDE WALL SPACE
      PITCH OF ROOF : 33.0 DEGREES
      NORTH-SOUTH ORIENTATION
      U = 1.33 J/SEC-M2-C (WALLS + WINDOWS)
      U =   .29 J/SEC-M2-C (ROOF)

COOLER UNIT DESCRIPTION :

      NOMINAL OPERATING CAPACITY AT 29.4 C AND 50 RH : 4.5 KW
      DEHUMIDIFIER :
         PACKED BED
         GEL TYPE : SILICA
         PARTICLE SURFACE AREA PER UNIT DEHUMIDIFIER VOLUME : 1573 M2/M3
         WHEEL DIAMETER : .97 M
         FLOW LENGTH : .0275 M
         WHEEL RPM : .09180

SENSIBLE AND LATENT AUXILIARY COOLING TO HOUSE

COLLECTOR :

      FLAT PLATE (AIR)
      AREA :   35.0 M2

STORAGE :

      PEBBLE BED
      VOLUME : 13.1 M3

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	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
INCIDENT SOLAR (J)	1.29E+10	1.33E+10	2.10E+10	2.09E+10	1.98E+10	1.79E+10	2.07E+10	2.01E+10	1.98E+10	2.16E+10	1.60E+10	1.38E+10	2.17E+11
USEFUL SOLAR COLLECTED (J)	4.65E+09	4.96E+09	7.16E+09	6.39E+09	6.87E+09	6.57E+09	7.66E+09	7.68E+09	7.37E+09	7.30E+09	5.90E+09	5.38E+09	7.79E+10
COLLECTOR EFFICIENCY,PCT	37.	37.	34.	31.	35.	37.	37.	38.	37.	34.	37.	39.	36.
BUILDING LOAD													
COOLING (J)	0.	0.	0.	5.30E+08	2.50E+09	4.95E+09	5.76E+09	6.75E+09	3.26E+09	7.05E+08	0.	0.	2.45E+10
SENSIBLE (J)	0.	0.	0.	1.29E+08	1.07E+09	2.31E+09	3.21E+09	3.60E+09	1.29E+09	1.18E+08	0.	0.	1.17E+10
LATENT (J)	0.	0.	0.	4.01E+08	1.43E+09	2.64E+09	2.55E+09	3.15E+09	1.98E+09	5.87E+08	0.	0.	1.27E+10
HEATING (J)	7.78E+09	6.75E+09	6.20E+08	1.27E+08	1.63E+08	3.35E+07	8.68E+06	0.	7.63E+07	2.46E+08	2.11E+09	8.14E+09	2.61E+10
TOTAL (J)	7.78E+09	6.75E+09	6.20E+08	6.56E+08	2.66E+09	4.99E+09	5.77E+09	6.75E+09	3.34E+09	9.50E+08	2.11E+09	8.14E+09	5.05E+10
COOLING CAPACITY SUPPLIED	0.	0.	0.	6.83E+08	1.92E+09	2.65E+09	3.46E+09	3.40E+09	2.42E+09	1.03E+09	0.	0.	1.56E+10
SENSIBLE (J)	0.	0.	0.	1.98E+08	6.65E+08	1.17E+09	1.62E+09	1.60E+09	7.65E+08	3.69E+08	0.	0.	6.38E+09
LATENT (J)	0.	0.	0.	4.86E+08	1.26E+09	1.48E+09	1.85E+09	1.80E+09	1.66E+09	6.57E+08	0.	0.	9.19E+09
SOLAR ENERGY SUPPLIED													
COOLING (J)	0.	0.	0.	1.32E+09	3.58E+09	4.56E+09	5.67E+09	5.99E+09	4.33E+09	2.20E+09	0.	0.	2.76E+10
HEATING (J)	4.58E+09	4.74E+09	6.20E+08	1.27E+09	1.63E+08	3.35E+07	8.68E+06	0.	7.63E+07	2.46E+08	2.11E+09	5.24E+09	1.73E+10
AUXILIARY SUPPLIED													
COOLING (J)	0.	0.	0.	1.13E+08	8.34E+08	2.33E+09	2.21E+09	3.18E+09	1.03E+09	2.21E+08	0.	0.	9.92E+09
HEATING (J)	3.20E+09	2.01E+09	0.	0.	0.	0.	0.	0.	0.	0.	0.	2.90E+09	8.10E+09
PERCENT BLDG LOAD BY SOLAR													
COOLING (J)	0.	0.	0.	86.	70.	53.	61.	52.	70.	82.	0.	0.	61.
HEATING (J)	59.	70.	100.	100.	100.	100.	100.	100.	100.	100.	100.	64.	69.
PERCENT COLLECTED SOLAR USED													
COOLING (J)	0.	0.	0.	37.	52.	69.	74.	78.	59.	30.	0.	0.	59.
HEATING (J)	98.	96.	9.	2.	2.	1.	0.	0.	1.	3.	36.	98.	56.
TOTAL PARASITIC ENERGY (J)	0.	0.	0.	1.37E+08	3.99E+08	5.63E+08	7.00E+08	7.58E+08	4.75E+08	2.10E+08	0.	0.	3.24E+09
COOLING COP	0.00	0.00	0.00	.52	.54	.58	.61	.57	.56	.47	0.00	0.00	.56
COOLING EER	0.00	0.00	0.00	14.96	12.84	11.37	12.23	11.06	13.33	14.27	0.00	0.00	12.07
DESICCANT DUTY CYCLE	0.00	0.00	0.00	.14	.22	.34	.41	.45	.29	.10	0.00	0.00	.29
PCT IN COMFORT	100.	100.	100.	99.	100.	100.	100.	100.	100.	99.	100.	100.	100.

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APPENDIX B

RESULTS FROM VAPOR-COMPRESSION COOLER SIMULATION

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Table B-1

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*  VAPOR COMPRESSION COOLING SIMULATION  *
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INPUT PARAMETERS

LOCATION : WASHINGTON, DC   YEAR : 1955
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.00 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD													
COOLING (J)	0.	0.	0.	2.53E+08	2.59E+09	4.57E+09	5.54E+09	6.26E+09	3.67E+09	1.85E+09	0.	0.	2.47E+10
SENSIBLE (J)	0.	0.	0.	2.04E+08	1.79E+09	3.33E+09	3.81E+09	4.32E+09	2.49E+09	1.29E+09	0.	0.	1.72E+10
LATENT (J)	0.	0.	0.	5.87E+07	7.90E+08	1.24E+09	1.73E+09	1.94E+09	1.13E+09	5.65E+08	0.	0.	7.51E+09
HEATING (J)	1.27E+10	9.17E+09	6.65E+09	3.15E+09	1.00E+09	1.76E+03	3.85E+09	3.03E+09	2.83E+09	3.89E+09	6.62E+09	9.70E+09	6.63E+10
TOTAL (J)	1.27E+10	9.17E+09	6.65E+09	3.41E+09	5.58E+09	6.33E+09	9.39E+09	9.29E+09	6.50E+09	5.74E+09	6.62E+09	9.70E+09	9.10E+10
HEAT SUPP (J)	1.27E+10	9.17E+09	6.65E+09	3.15E+09	1.00E+09	1.76E+09	3.85E+09	3.03E+09	2.83E+09	3.89E+09	6.62E+09	9.70E+09	6.63E+10
COOLING CAPA-													
CITY SUPPLIED (J)	0.	0.	0.	5.51E+08	4.62E+09	5.99E+09	1.04E+10	1.05E+10	6.47E+09	3.58E+09	0.	0.	4.21E+10
SENSIBLE (J)	0.	0.	0.	4.08E+08	3.29E+09	4.38E+09	7.15E+09	7.24E+09	4.45E+09	2.49E+09	0.	0.	2.94E+10
LATENT (J)	0.	0.	0.	1.42E+08	1.33E+09	1.62E+03	3.26E+09	3.26E+09	2.02E+09	1.09E+09	0.	0.	1.27E+10
REHEAT (J)	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TOTAL PARASITIC													
ENERGY (J)	0.	0.	0.	6.34E+03	5.21E+04	8.02E+04	1.19E+05	1.31E+05	7.68E+04	4.01E+04	0.	0.	5.04E+05
COOLING COP	0.00	0.00	0.00	2.50	2.50	2.50	2.50	2.50	2.50	2.50	0.00	0.00	2.50
COOLING EER	0.00	0.00	0.00	8.53	8.53	8.53	8.53	8.53	8.53	8.53	0.00	0.00	8.53
VAPOR COMP													
DUTY CYCLE	0.00	0.00	0.00	.04	.19	.26	.43	.44	.29	.31	0.00	0.00	.23
PCT IN COMFORT	100.	100.	100.	99.	84.	83.	42.	50.	68.	85.	100.	100.	59.

Table B-2

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**************************************************
*                                                *
*    VAPOR COMPRESSION COOLING SIMULATION   *
*                                                *
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INPUT PARAMETERS

LOCATION : WASHINGTON, DC   YEAR : 1955
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.00 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD													
COOLING (J)	0.	0.	0.	6.90E+07	1.99E+09	4.17E+09	5.22E+09	6.32E+09	3.40E+09	1.14E+09	0.	0.	2.23E+10
SENSIBLE (J)	0.	0.	0.	1.49E+07	1.05E+09	2.95E+09	2.87E+09	3.92E+09	1.92E+09	4.15E+08	0.	0.	1.31E+10
LATENT (J)	0.	0.	0.	5.40E+07	9.42E+08	1.22E+09	2.34E+09	2.39E+09	1.48E+09	7.25E+08	0.	0.	9.17E+09
HEATING (J)	1.27E+10	9.17E+09	6.65E+09	2.83E+09	1.04E+09	5.01E+18	2.78E+07	4.36E+07	5.83E+08	2.06E+09	6.62E+03	9.70E+09	5.19E+10
TOTAL (J)	1.27E+10	9.17E+09	6.65E+09	2.90E+09	3.03E+09	4.67E+09	5.24E+09	6.36E+09	3.98E+09	3.20E+09	6.62E+03	9.70E+09	7.42E+10
HEAT SUPP (J)	1.27E+10	9.17E+09	6.65E+09	2.83E+03	1.04E+09	5.01E+08	2.79E+07	4.36E+07	5.93E+08	2.06E+09	6.62E+03	9.70E+09	5.19E+10
COOLING CAPA-													
CITY SUPPLIED (J)	0.	0.	0.	5.67E+08	3.94E+09	5.90E+09	9.31E+09	1.00E+10	6.08E+09	5.19E+09	0.	0.	3.90E+10
SENSIBLE (J)	0.	0.	0.	4.24E+08	2.94E+09	4.52E+09	6.84E+09	7.46E+09	4.52E+09	2.33E+09	0.	0.	2.90E+10
LATENT (J)	0.	0.	0.	1.43E+18	1.01E+09	1.38E+09	2.47E+09	2.59E+09	1.56E+09	4.66E+08	0.	0.	1.00E+10
REHEAT (J)	0.	0.	0.	3.75E+08	1.86E+09	1.51E+09	3.89E+09	3.43E+09	2.55E+09	1.93E+09	0.	0.	1.54E+10
TOTAL PARASITIC													
ENERGY (J)	0.	0.	0.	6.66E+03	5.08E+04	7.82E+04	1.27E+05	1.41E+05	8.45E+04	4.44E+04	0.	0.	5.33E+05
COOLING COP	0.00	0.00	0.00	.94	1.15	1.52	1.22	1.35	1.22	1.03	0.00	0.00	1.26
COOLING EER	0.00	0.00	0.00	3.21	3.92	5.20	4.18	4.60	4.17	3.50	0.00	0.00	4.29
VAPOR COMP													
DUTY CYCLE	0.00	0.00	0.00	.05	.16	.25	.39	.42	.24	.27	0.00	0.00	.27
PCT IN COMFORT	100.	100.	100.	100.	99.	97.	99.	95.	99.	99.	100.	100.	99.

Table B-3

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* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
*                                                                            *
*   VAPOR COMPRESSION COOLING SIMULATION   *
*                                                                            *
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
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INPUT PARAMETERS

LOCATION : DODGE CITY, KS   YEAR : 1955
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.03 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD													
COOLING (J)	0.	0.	0.	2.34E+08	1.01E+09	2.54E+09	6.32E+09	5.39E+09	1.45E+09	0.	0.	0.	1.70E+10
SENSIBLE (J)	0.	0.	0.	2.34E+08	7.42E+08	1.82E+09	5.22E+09	4.56E+09	1.40E+09	0.	0.	0.	1.40E+10
LATENT (J)	0.	0.	0.	-4.34E+07	2.69E+08	7.13E+08	1.11E+09	8.33E+08	5.93E+07	0.	0.	0.	2.93E+09
HEATING (J)	1.22E+10	1.21E+10	7.70E+09	2.45E+09	2.61E+09	3.00E+09	8.53E+08	8.18E+08	1.79E+08	1.97E+08	9.14E+09	1.30E+10	6.43E+10
TOTAL (J)	1.22E+10	1.21E+10	7.70E+09	2.68E+09	3.63E+09	5.53E+09	7.18E+09	6.21E+09	1.64E+09	1.97E+08	9.14E+09	1.30E+10	9.12E+10
HEAT SUPP (J)	1.22E+10	1.21E+10	7.70E+09	2.45E+09	2.61E+09	3.00E+09	8.53E+08	8.18E+08	1.79E+08	1.97E+08	9.14E+09	1.30E+10	6.43E+10
COOLING CAPACITY SUPPLIED	0.	0.	0.	3.56E+08	1.56E+09	4.20E+09	7.44E+09	6.37E+09	1.45E+09	0.	0.	0.	2.14E+10
SENSIBLE (J)	0.	0.	0.	3.14E+08	1.20E+09	3.99E+09	6.05E+09	5.26E+09	1.43E+09	0.	0.	0.	1.74E+10
LATENT (J)	0.	0.	0.	4.25E+07	3.51E+08	1.10E+09	1.34E+09	1.10E+09	2.38E+07	0.	0.	0.	4.01E+09
REHEAT (J)	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TOTAL PARASITIC ENERGY (J)	0.	0.	0.	4.46E+03	1.65E+04	4.75E+04	9.21E+04	8.02E+04	1.78E+04	0.	0.	0.	2.59E+05
COOLING COP	0.00	0.00	0.00	2.50	2.50	2.50	2.50	2.50	2.50	0.00	0.00	0.00	2.50
COOLING EER	0.00	0.00	0.00	8.53	8.53	8.53	8.53	8.53	8.53	0.00	0.00	0.00	8.53
VAPOR COMP DUTY CYCLE	0.00	0.00	0.00	.03	.06	.13	.31	.26	.13	0.00	0.00	0.00	.13
PCT IN COMFORT	100.	100.	100.	93.	97.	92.	87.	86.	98.	100.	100.	100.	92.

Table B-4

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********************************************
*                                          *
*  VAPOR COMPRESSION COOLING SIMULATION    *
*                                          *
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INPUT PARAMETERS

LOCATION : DODGE CITY, KS   YEAR : 1955
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.00 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

		JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD														
COOLING	(J)	0.	0.	0.	1.63E+08	5.67E+08	1.70E+09	6.06E+09	5.15E+09	1.46E+09	0.	0.	0.	1.51E+10
SENSIBLE	(J)	0.	0.	0.	2.19E+08	2.93E+08	9.39E+08	4.97E+09	4.25E+09	1.40E+09	0.	0.	0.	1.21E+10
LATENT	(J)	0.	0.	-0.	-5.66E+07	2.74E+08	7.58E+08	1.09E+09	9.04E+08	5.85E+07	0.	0.	0.	3.03E+09
HEATING	(J)	1.22E+10	1.21E+10	7.70E+09	2.38E+09	1.79E+09	1.08E+09	5.65E+08	3.34E+07	1.79E+08	1.97E+08	9.14E+09	1.30E+10	5.93E+10
TOTAL	(J)	1.22E+10	1.21E+10	7.70E+09	2.55E+09	2.35E+09	2.78E+09	6.06E+09	5.19E+09	1.64E+09	1.97E+08	9.14E+09	1.30E+10	7.49E+10
HEAT PUMP	(J)	1.22E+10	1.21E+10	7.70E+09	2.38E+09	1.79E+09	1.08E+09	5.65E+08	3.34E+07	1.79E+08	1.97E+08	9.14E+09	1.30E+10	5.98E+10
COOLING CAPA-														
CITY SUPPLIED	(J)	0.	0.	0.	3.73E+08	1.56E+09	4.08E+09	7.16E+09	6.40E+09	1.46E+09	0.	0.	0.	2.10E+10
SENSIBLE	(J)	0.	0.	0.	3.32E+08	1.20E+09	3.11E+09	5.98E+09	5.40E+09	1.43E+09	0.	0.	0.	1.75E+10
LATENT	(J)	0.	0.	0.	4.11E+07	3.50E+08	9.69E+08	1.18E+09	1.00E+09	2.88E+07	0.	0.	0.	3.57E+09
REHEAT	(J)	0.	0.	0.	1.05E+09	9.97E+08	2.13E+09	8.78E+08	1.03E+09	0.	0.	0.	0.	5.04E+09
TOTAL PARASITIC														
ENERGY	(J)	0.	0.	0.	5.14E+03	1.78E+04	5.16E+04	9.31E+04	8.45E+04	1.78E+04	0.	0.	0.	2.70E+05
COOLING COP		0.00	0.00	0.00	1.47	1.02	1.08	1.91	1.78	2.50	0.00	0.00	0.00	1.56
COOLING EER		0.00	0.00	0.00	5.03	3.49	3.70	6.53	6.09	8.53	0.00	0.00	0.00	5.34
VAPOR COMP														
DUTY CYCLE		0.00	0.00	0.00	.03	.06	.18	.30	.27	.13	0.00	0.00	0.00	.18
PCT IN COMFORT		100.	100.	100.	100.	100.	99.	94.	94.	98.	100.	100.	100.	97.

Table B-5

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**********************************************
*                                            *
*   VAPOR COMPRESSION COOLING SIMULATION   *
*                                            *
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INPUT PARAMETERS

LOCATION : CHARLESTON, SC   YEAR : 1963
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.00 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

		JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD														
COOLING	(J)	0.	0.	0.	1.16E+09	3.92E+09	5.67E+09	5.99E+09	6.52E+09	4.61E+09	1.97E+09	0.	0.	2.99E+10
SENSIBLE	(J)	0.	0.	0.	8.33E+08	2.76E+09	3.77E+09	4.05E+09	4.25E+09	3.12E+09	1.45E+09	0.	0.	2.02E+10
LATENT	(J)	0.	0.	0.	3.29E+08	1.16E+09	1.90E+09	1.94E+09	2.27E+09	1.49E+09	5.16E+08	0.	0.	9.61E+09
HEATING	(J)	7.78E+09	6.75E+09	6.26E+09	1.92E+09	4.40E+09	4.35E+09	4.09E+09	4.01E+09	4.28E+09	5.23E+09	2.11E+09	8.14E+09	5.17E+10
TOTAL	(J)	7.76E+09	6.75E+09	6.26E+09	3.08E+09	8.33E+09	1.00E+10	1.01E+10	1.05E+10	8.90E+09	5.25E+09	2.11E+09	8.14E+09	8.16E+10
HEAT SUPP	(J)	7.78E+09	6.75E+09	6.26E+09	1.92E+09	4.40E+09	4.35E+09	4.09E+09	4.01E+09	4.28E+09	3.28E+09	2.11E+09	8.14E+09	5.17E+10
COOLING CAPA-														
CITY SUPPLIED	(J)	0.	0.	0.	2.29E+09	7.01E+09	1.06E+10	1.14E+10	1.26E+10	8.22E+09	3.17E+09	0.	0.	5.53E+10
SENSIBLE	(J)	0.	0.	0.	1.65E+09	5.00E+09	7.09E+09	7.72E+09	8.03E+09	5.58E+09	2.51E+09	0.	0.	3.77E+10
LATENT	(J)	0.	0.	0.	6.45E+08	2.01E+09	3.50E+09	3.71E+09	4.33E+09	2.55E+09	9.61E+08	0.	0.	1.76E+10
REHEAT	(J)	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TOTAL PARASITIC														
ENERGY	(J)	0.	0.	0.	2.59E+04	7.94E+04	1.20E+05	1.32E+05	1.46E+05	9.45E+04	3.75E+04	0.	0.	6.35E+05
COOLING COP		0.00	0.00	0.00	2.50	2.50	2.50	2.50	2.50	2.50	2.50	0.00	0.00	2.50
COOLING EER		0.00	0.00	0.00	8.53	8.53	8.53	8.53	8.53	8.53	8.53	0.00	0.00	8.53
VAPOR COMP														
DUTY CYCLE		0.00	0.00	0.00	.18	.29	.45	.47	.51	.35	.14	0.00	0.00	.36
PCT IN COMFORT		100.	100.	100.	95.	79.	29.	32.	14.	57.	93.	100.	100.	54.

Table B-6

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**************************************
*                                    *
*  VAPOR COMPRESSION COOLING SIMULATION  *
*                                    *
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INPUT PARAMETERS

LOCATION : CHARLESTON, SC   YEAR : 1963
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.00 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD													
COOLING (J)	0.	0.	0.	6.32E+08	2.65E+09	5.11E+09	5.93E+09	6.89E+09	3.46E+09	3.47E+09	0.	0.	2.55E+10
SENSIBLE (J)	0.	0.	0.	2.38E+08	1.20E+09	2.38E+09	3.21E+09	3.59E+09	1.37E+09	3.79E+08	0.	0.	1.24E+10
LATENT (J)	0.	0.	0.	3.94E+08	1.45E+09	2.74E+09	2.69E+09	3.30E+09	2.03E+03	5.69E+08	0.	0.	1.32E+10
HEATING (J)	7.76E+09	6.75E+09	6.20E+08	7.28E+08	1.04E+09	1.10E+08	2.62E+07	0.	4.83E+03	1.42E+09	2.11E+09	8.14E+09	2.92E+10
TOTAL (J)	7.78E+09	6.75E+09	6.26E+08	1.36E+09	3.69E+09	5.23E+09	5.92E+09	6.89E+09	3.88E+09	2.36E+09	2.11E+09	9.14E+09	5.47E+10
HEAT SUPP (J)	7.78E+09	6.75E+09	6.20E+08	7.28E+08	1.04E+09	1.10E+08	2.62E+07	0.	4.83E+03	1.42E+09	2.11E+09	8.14E+09	2.92E+10
COOLING CAPA-CITY SUPPLIED (J)	0.	0.	0.	2.25E+09	5.77E+09	1.10E+10	1.04E+10	1.22E+10	8.64E+09	3.47E+09	0.	0.	5.48E+10
SENSIBLE (J)	0.	0.	0.	1.67E+09	5.03E+09	8.01E+09	7.51E+09	8.77E+09	6.36E+09	2.60E+09	0.	0.	4.01E+10
LATENT (J)	0.	0.	0.	5.79E+08	1.75E+09	3.00E+09	2.83E+09	3.42E+09	2.28E+09	8.63E+08	0.	0.	1.47E+10
REHEAT (J)	0.	0.	0.	1.36E+09	3.80E+09	5.58E+09	4.31E+09	5.10E+09	4.91E+09	2.16E+09	0.	0.	2.72E+10
TOTAL PARASITIC ENERGY (J)	0.	0.	0.	3.12E+04	9.65E+04	1.64E+05	1.53E+05	1.83E+05	1.29E+05	4.49E+04	0.	0.	7.99E+05
COOLING COP	0.00	0.00	0.00	1.00	1.04	1.10	1.23	1.22	1.03	.98	0.00	0.00	1.11
COOLING EER	0.00	0.00	0.00	3.40	3.56	3.77	4.20	4.17	3.52	3.33	0.00	0.00	3.81
VAPOR COMP DUTY CYCLE	0.00	0.00	0.00	.18	.28	.47	.43	.51	.37	.14	0.00	0.00	.35
PCT IN COMFORT	1.0.	100.	100.	100.	99.	98.	98.	97.	99.	100.	100.	100.	99.

Table B-7

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**************************************************
*                                                *
*      VAPCR CCMPRESSION COOLING SIMULATION       *
*                                                *
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INPUT PARAMETERS

LOCATION : FORT WORTH, TX   YEAR : 1960
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.00 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD													
COOLING (J)	0.	0.	-3.06E+06	2.15E+09	3.41E+09	7.15E+09	8.16E+09	7.70E+09	6.13E+09	3.71E+09	0.	0.	3.34E+10
SENSIBLE (J)	0.	0.	-3.19E+06	1.54E+09	2.39E+09	5.41E+09	5.80E+09	5.41E+09	4.77E+09	2.61E+09	0.	0.	2.79E+10
LATENT (J)	0.	0.	1.29E+05	6.11E+08	1.02E+09	1.74E+09	2.36E+09	2.29E+09	1.36E+09	1.10E+09	0.	0.	1.05E+10
HEATING (J)	8.06E+09	7.26E+09	6.07E+09	3.46E+09	3.08E+09	1.34E+09	2.15E+09	2.08E+09	1.40E+09	4.40E+09	1.83E+09	8.83E+09	4.99E+10
TOTAL (J)	8.06E+09	7.26E+09	6.06E+09	5.61E+09	6.48E+09	8.49E+09	1.03E+10	9.78E+09	7.53E+09	8.10E+09	1.83E+09	8.83E+09	8.83E+10
HEAT SUPP (J)	8.06E+09	7.26E+09	6.07E+09	3.46E+09	3.08E+09	1.34E+09	2.15E+09	2.08E+09	1.40E+09	4.40E+09	1.83E+09	8.83E+09	4.99E+10
COOLING CAPA-													
CITY SUPPLIED (J)	0.	0.	8.10E+06	3.88E+09	5.89E+09	9.20E+09	1.19E+10	1.16E+10	7.45E+09	6.76E+09	0.	0.	5.67E+10
SENSIBLE (J)	0.	0.	5.62E+06	2.83E+09	4.21E+09	6.86E+09	8.34E+09	8.04E+09	5.75E+09	4.87E+09	0.	0.	4.09E+10
LATENT (J)	0.	0.	2.48E+06	1.05E+09	1.68E+09	2.34E+09	3.55E+09	3.59E+09	1.70E+09	1.89E+09	0.	0.	1.59E+10
REHEAT (J)	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TOTAL PARASITIC													
ENERGY (J)	0.	0.	1.71E+02	4.37E+04	6.77E+04	1.22E+05	1.63E+05	1.56E+05	1.01E+05	7.65E+04	0.	0.	7.30E+05
COOLING COP	0.00	0.00	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	0.00	0.00	2.50
COOLING EER	0.00	0.00	8.53	8.53	8.53	8.53	8.53	8.53	8.53	8.53	0.00	0.00	8.53
VAPOR COMP													
DUTY CYCLE	0.00	0.00	.51	.17	.24	.39	.49	.48	.32	.28	0.00	0.00	.34
PCT IN COMFORT	100.	100.	100.	91.	84.	74.	54.	47.	85.	83.	100.	100.	74.

Table B-8

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**************************************************
*                                                *
*  VAPOR COMPRESSION COOLING SIMULATION  *
*                                                *
**************************************************
```

INPUT PARAMETERS

LOCATION : FORT WORTH, TX   YEAR : 1960
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.00 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD													
COOLING (J)	0.	0.	-3.06E+06	1.03E+09	2.74E+09	7.15E+09	8.54E+09	8.46E+09	5.69E+09	1.26E+09	0.	0.	3.59E+10
SENSIBLE (J)	0.	0.	-3.13E+06	3.37E+08	1.56E+09	5.26E+09	5.93E+09	5.76E+09	4.36E+09	9.31E+08	0.	0.	2.41E+10
LATENT (J)	0.	0.	1.29E+05	7.23E+08	1.18E+09	1.89E+09	2.61E+09	2.76E+09	1.32E+09	1.83E+09	0.	0.	1.18E+10
HEATING (J)	8.06E+09	7.26E+09	6.07E+09	1.30E+09	7.62E+08	0.	0.	0.	2.26E+08	1.04E+09	1.03E+09	4.03E+09	8.53E+10
TOTAL (J)	8.06E+09	7.26E+09	6.06E+09	2.33E+09	3.50E+09	7.15E+09	8.54E+09	8.46E+09	5.91E+09	5.27E+09	1.03E+09	8.03E+09	7.12E+10
HEAT SUPP (J)	8.06E+09	7.26E+09	6.07E+09	1.30E+09	7.62E+08	0.	0.	0.	2.26E+08	1.04E+09	1.03E+09	8.03E+09	3.53E+10
COOLING CAPA-CITY SUPPLIED (J)	0.	0.	8.10E+06	3.75E+09	4.80E+09	8.94E+09	1.10E+10	1.12E+10	7.47E+09	6.51E+09	0.	0.	5.37E+10
SENSIBLE (J)	0.	0.	5.62E+06	2.79E+09	3.58E+09	6.97E+09	8.22E+09	8.46E+09	5.95E+09	4.85E+09	0.	0.	4.38E+10
LATENT (J)	0.	0.	2.48E+05	9.62E+08	1.21E+09	1.97E+09	2.74E+09	2.83E+09	1.52E+09	1.67E+09	0.	0.	1.29E+10
REHEAT (J)	0.	0.	5.62E+06	2.40E+09	2.03E+09	1.59E+09	2.16E+09	2.52E+09	1.47E+09	1.82E+09	0.	0.	1.60E+10
TOTAL PARASITIC ENERGY (J)	0.	0.	1.71E+02	5.14E+04	6.40E+04	1.22E+05	1.56E+05	1.62E+05	9.77E+04	1.22E+04	0.	0.	7.45E+05
COOLING COP	0.00	0.00	.91	.96	1.22	1.73	1.67	1.60	1.68	1.01	0.00	0.00	1.43
COOLING EER	0.00	0.00	3.12	3.28	4.15	5.90	5.72	6.47	5.72	3.46	0.00	0.00	4.89
VAPOR COMP DUTY CYCLE	0.00	1.00	.50	.16	.20	.38	.45	.47	.32	.27	0.00	0.00	.32
PCT IN COMFORT	100.	100.	100.	100.	99.	93.	92.	93.	94.	99.	100.	100.	96.

Table B-9

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******************************************
*                                        *
*  VAPOR COMPRESSION COOLING SIMULATION  *
*                                        *
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INPUT PARAMETERS

LOCATION : PHOENIX, AZ   YEAR : 1962
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.03 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

		JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD														
COOLING	(J)	0.	0.	3.76E+08	3.82E+09	4.37E+09	8.44E+09	1.17E+10	1.24E+10	8.79E+09	3.71E+09	8.17E+08	0.	5.45E+10
SENSIBLE	(J)	0.	0.	4.51E+08	3.82E+09	4.37E+09	8.31E+09	1.12E+10	1.19E+10	7.95E+09	3.75E+09	8.33E+08	0.	5.25E+10
LATENT	(J)	0.	0.	-7.45E+07	-4.58E+06	2.51E+06	1.35E+08	5.77E+08	4.99E+08	8.40E+08	-3.56E+07	-1.56E+07	0.	1.92E+09
HEATING	(J)	2.96E+09	9.32E+08	1.30E+09	3.90E+08	2.45E+08	1.77E+06	0.	0.	4.70E+08	5.45E+08	1.55E+09	1.90E+09	1.03E+10
TOTAL	(J)	2.96E+09	9.32E+08	1.67E+09	4.21E+09	4.62E+09	8.45E+09	1.17E+10	1.24E+10	9.26E+09	4.26E+09	2.36E+09	1.90E+09	6.47E+10
HEAT SUPP	(J)	2.96E+09	9.32E+08	1.30E+09	3.90E+08	2.45E+08	1.77E+06	0.	0.	4.70E+08	5.45E+08	1.55E+09	1.90E+09	1.03E+10
COOLING CAPA-CITY SUPPLIED	(J)	0.	0.	5.35E+08	3.89E+09	4.46E+09	8.55E+09	1.19E+10	1.27E+10	9.41E+09	3.84E+09	8.75E+08	0.	5.62E+10
SENSIBLE	(J)	0.	0.	4.95E+08	3.89E+09	4.46E+09	8.46E+09	1.14E+10	1.21E+10	8.47E+09	3.82E+09	4.75E+08	0.	5.39E+10
LATENT	(J)	0.	0.	3.85E+07	0.	0.	9.32E+07	5.80E+08	5.76E+08	9.44E+08	1.87E+07	0.	0.	2.25E+09
REHEAT	(J)	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
TOTAL PARASITIC ENERGY	(J)	0.	0.	6.86E+03	5.23E+04	6.24E+04	1.32E+05	1.85E+05	2.65E+05	1.42E+05	5.62E+04	1.06E+04	0.	8.52E+05
COOLING COP		0.00	0.00	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	0.00	2.50
COOLING EER		0.00	0.00	8.53	8.53	8.53	8.53	8.53	8.53	8.53	8.53	8.53	0.00	8.53
VAPOR COMP DUTY CYCLE		0.00	0.00	.10	.17	.18	.37	.50	.53	.40	.16	.05	0.00	.30
PCT IN COMFORT		100.	100.	98.	93.	93.	83.	79.	77.	80.	93.	99.	100.	87.

Table B-10

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*****************************************
*                                       *
*   VAPOR COMPRESSION COOLING SIMULATION *
*                                       *
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INPUT PARAMETERS

LOCATION : PHOENIX, AZ   YEAR : 1962
VAPOR COMPRESSION COOLER NOMINAL OPERATING CAPACITY : 9.30 KW

VAPOR COMPRESSION
SYSTEM PERFORMANCE

		JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
BUILDING LOAD														
COOLING	(J)	0.	0.	3.23E+09	3.82E+09	4.37E+09	8.44E+09	1.17E+10	1.24E+10	8.64E+09	3.79E+09	8.17E+08	0.	5.42E+10
SENSIB	(J)	0.	0.	3.99E+08	3.82E+09	4.37E+09	8.31E+09	1.12E+10	1.19E+10	7.80E+09	3.74E+09	8.33E+08	0.	5.23E+10
LATENT	(J)	0.	0.	-7.61E+07	-4.58E+05	2.51E+06	1.35E+08	5.77E+08	4.99E+08	8.43E+08	-3.47E+07	-1.58E+07	0.	1.93E+09
HEATING	(J)	2.96E+09	9.32E+08	1.24E+09	3.90E+08	2.45E+08	1.77E+06	0.	0.	0.	5.42E+08	1.55E+09	1.90E+09	9.76E+09
TOTAL	(J)	2.96E+09	9.32E+08	1.56E+09	4.21E+09	4.62E+09	8.45E+09	1.17E+10	1.24E+10	8.64E+09	4.25E+09	2.36E+09	1.90E+09	6.40E+10
HEAT SUPP	(J)	2.96E+09	9.32E+08	1.24E+09	3.90E+08	2.45E+08	1.77E+06	0.	0.	0.	5.42E+08	1.55E+09	1.90E+09	9.76E+09
COOLING CAPA-														
CITY SUPPLIED		0.	0.	5.67E+08	3.89E+09	4.46E+09	8.55E+09	1.19E+10	1.27E+10	9.23E+09	3.84E+09	8.75E+08	0.	5.63E+10
SENSIBLE	(J)	0.	0.	5.25E+08	3.89E+09	4.46E+09	8.46E+09	1.14E+10	1.21E+10	8.40E+09	3.82E+09	8.75E+08	0.	5.39E+10
LATENT	(J)	0.	0.	4.20E+07	0.	0.	9.32E+07	5.80E+08	5.76E+08	8.32E+08	2.01E+07	0.	0.	2.14E+09
REHEAT	(J)	0.	0.	1.04E+09	0.	0.	0.	0.	0.	4.69E+08	0.	0.	0.	5.72E+08
TOTAL PARASITIC														
ENERGY	(J)	0.	0.	7.89E+03	5.23E+04	6.24E+04	1.32E+05	1.85E+05	2.05E+05	1.41E+05	5.64E+04	1.06E+04	0.	8.52E+05
COOLING COP		0.00	0.00	1.72	2.50	2.50	2.50	2.50	2.50	2.22	2.50	2.50	0.00	2.44
COOLING EER		0.00	0.00	5.85	8.53	8.53	8.53	8.53	8.53	7.57	8.53	8.53	0.00	8.32
VAPOR COMP														
DUTY CYCLE		0.00	0.00	.10	.17	.18	.37	.50	.53	.40	.16	.05	0.00	.30
PCT IN COMFORT		100.	100.	99.	93.	93.	83.	79.	77.	84.	93.	99.	100.	87.

APPENDIX C

RESULTS OF COST/BENEFIT ANALYSIS

SERI☀

SERI

	Desiccant Cooler Size (kW)	Collector Area (m²)	Cost/Benefit Ratio			
			1985 Base Year		1990 Base Year	
Run No.[a]			$107.65/m²	$215.30/m²	$107.65/m²	$215.30/m²
WASH 2	9.0	35	1.03	1.41	0.78	1.01
WASH 4	4.5	35	0.95	1.33	0.73	0.96
WASH 6	2.3	35	0.93	1.31	0.72	0.96
WASH 8	9.0	50	1.13	1.67	0.81	1.14
WASH 10	4.5	50	1.05	1.59	0.77	1.10
WASH 12	2.3	50	1.03	1.57	0.76	1.09
WASH 13	9.0	20	0.98	1.20	0.79	0.92
WASH 14	4.5	20	0.88	1.10	0.73	0.86
WASH 15	2.3	20	0.85	1.07	0.71	0.85
WASH 16	9.0	42	1.08	1.53	0.79	1.07
WASH 17	4.5	42	1.00	1.46	0.75	1.03
WASH 18	2.3	42	0.98	1.43	0.74	1.02
WASH 19	9.0	42	1.08	1.54	0.80	1.08
WASH 20	9.0	20	1.01	1.22	0.82	0.95
WASH 21	9.0	35	1.06	1.43	0.80	1.03
WASH 22	9.0	50	1.13	1.67	0.81	1.14
WASH 23	9.0	10	1.01	1.12	0.86	0.93
WASH 24	4.5	10	0.89	1.00	0.78	0.85
WASH 25	4.5	35	0.94	1.32	0.73	0.96
WASH 26	4.5	50	1.05	1.59	0.77	1.10
WASH 27	2.3	10	0.85	0.96	0.75	0.82
WASH 28	2.3	35	0.93	1.31	0.73	0.96
WASH 29	2.3	50	1.04	1.58	0.77	1.10
DODG 1	9.0	35	1.54	2.28	1.10	1.57
DODG 2	4.5	35	1.32	2.06	0.95	1.42
DODG 3	4.5	35	1.62	2.37	1.22	1.69
DODG 4	2.3	35	1.21	1.95	0.87	1.34
DODG 5	2.3	35	1.58	2.33	1.21	1.68
DODG 6	9.0	20	1.27	1.70	0.95	1.22
DODG 7	4.5	20	1.07	1.49	0.81	1.08
DODG 8	2.3	20	0.97	1.39	0.75	1.02
DODG 9	9.0	42	1.64	2.53	1.14	1.70
DODG 10	4.5	42	1.43	2.32	1.00	1.57
DODG 11	2.3	42	1.33	2.22	0.93	1.50
DODG 12	4.5	35	1.59	2.33	1.18	1.65
CHAR 8	9.0	50	1.10	1.66	0.77	1.11
CHAR 11	4.5	35	0.90	1.29	0.66	0.91
CHAR 13	13.5	50	1.25	1.82	0.88	1.23
CHAR 14	4.5	35	0.89	1.28	0.66	0.90
FTWR 3	4.5	35	1.17	1.66	0.88	1.19
FTWR 5	2.3	35	1.13	1.62	0.87	1.17
FTWR 6	4.5	35	1.16	1.65	0.87	1.18
PHNX 2	9.0	35	1.56	2.08	1.21	1.54
PHNX 4	4.5	35	1.50	2.02	1.21	1.53
PHNX 7	4.5	35	1.51	2.02	1.21	1.54

[a]See Table 3-17 for type of auxiliary cooling (if any) used with each run.

SERI

APPENDIX D

SAMPLE OUTPUT FROM ECONOMICS ANALYSIS COMPUTER PROGRAM

SERI

```
****************************************          WASH25
*                                      *
* DESICCANT SYSTEM ECONOMIC ANALYSIS *
*          BASE YEAR = 1978           *
*                                      *
****************************************
```

6.00 PCT. ECONOMIC INFLATION   5.00 PCT. ELECTRICITY ESCALATION
0.00 PCT. SOLAR EQUIP. INFLATION   5.00 PCT. AUX. HEAT ESCALATION
3.00 PCT. DISCOUNT RATE

   35.00 M**2 COLLECTOR AREA   1.02E+11 J/YR TOTAL HEATING LOAD
8.99E+09 J/YR AUX. COOLING REQUIRED   3.37E+10 J/YR AUX. HEATING REQUIRED
   5.50 KW AUX. COOLING CAPACITY   9.00 KW VAPOR COMPRESSION COOL CAP
2.26E+09 J/YR DESICCANT PARASITIC POWER 1.69E+10 J/YR VAP. COMP. PARASITIC POW
   2.50 COP OF AUX. COOLING UNIT   2.50 COP OF VAPOR COMPRESSION UNIT

      ASSUMED 1978 COSTS   COSTS ESCALATED TO 1978
              .0411 ELECTRICITY RATES ($/KW-HR)   .0410
              .0150 AUXILIARY HEATING ($/KW-HR)   .0150
           1500.00   DESICCANT SYSTEM ($)   1500.00
              0.00   STORAGE UNIT ($)   0.00
            610.00   AUXILIARY COOLER UNIT ($)   610.00
            300.00   SYSTEM INSTALLATION ($)   300.00
           1000.00 VAPOR COMPRESSION UNIT ($)   1000.00

   PRESENT WORTH ($) BASED ON 20 YEAR LIFETIME:
         DESICCANT SYSTEM   VAPOR COMPRESSION SYSTEM
              1.50E+02 PARASITIC POWER 1.11E+03
              8.19E+02   HEATING   2.48E+03
              2.36E+02   COOLING
                          TOTAL   4.59E+03

PCT. GOV.	PCT. COST	COLLECTOR	COLLECTOR	INITIAL	OMRI	TOTAL	COST
SUBSIDY	FOR OMRI	COST	CAPITAL	CAPITAL	PRESENT	PRESENT	BENEFIT
		($/M**2)	COST ($)	COST ($)	WORTH ($)	WORTH ($)	RATIO
0.00	.50	107.65	3767.75	6177.75	459.55	7840.92	1.71
0.00	.50	161.48	5651.63	8061.63	599.68	9864.94	2.15
0.00	.50	215.30	7535.50	9945.50	739.82	11888.95	2.59
0.00	.50	269.13	9419.37	11829.38	879.96	13912.96	3.03
0.00	.50	322.95	11303.25	13713.25	1020.09	15936.97	3.47
0.00	1.00	107.65	3767.75	6177.75	919.09	8300.47	1.81
0.00	1.00	161.48	5651.63	8061.63	1199.37	10464.62	2.28
0.00	1.00	215.30	7535.50	9945.50	1479.64	12628.77	2.75
0.00	1.00	269.13	9419.37	11829.38	1759.91	14792.91	3.22
0.00	1.00	322.95	11303.25	13713.25	2040.19	16957.06	3.69
0.00	1.50	107.65	3767.75	6177.75	1378.64	8760.02	1.91
0.00	1.50	161.48	5651.63	8061.63	1799.05	11064.30	2.41
0.00	1.50	215.30	7535.50	9945.50	2219.46	13368.59	2.91
0.00	1.50	269.13	9419.37	11829.38	2639.87	15672.87	3.41
0.00	1.50	322.95	11303.25	13713.25	3060.28	17977.16	3.92
0.00	2.00	107.65	3767.75	6177.75	1838.19	9219.56	2.01
0.00	2.00	161.48	5651.63	8061.63	2398.73	11663.98	2.54
0.00	2.00	215.30	7535.50	9945.50	2959.28	14108.41	3.07
0.00	2.00	269.13	9419.37	11829.38	3519.82	16552.43	3.61
0.00	2.00	322.95	11303.25	13713.25	4080.37	18997.25	4.14
0.00	2.50	107.65	3767.75	6177.75	2297.73	9679.11	2.11
0.00	2.50	161.48	5651.63	8061.63	2998.42	12263.67	2.67
0.00	2.50	215.30	7535.50	9945.50	3699.10	14848.23	3.23
0.00	2.50	269.13	9419.37	11829.38	4399.78	17432.78	3.80
0.00	2.50	322.95	11303.25	13713.25	5100.46	20017.34	4.36
0.00	3.00	107.65	3767.75	6177.75	2757.28	10138.66	2.21
0.00	3.00	161.48	5651.63	8061.63	3598.10	12863.35	2.80

0.00	3.00	215.30	7535.50	9945.50	4438.92	15588.05	3.40
0.00	3.00	269.13	9419.37	11829.38	5279.74	18312.74	3.99
0.00	3.00	322.95	11303.25	13713.25	6120.56	21037.43	4.58
0.00	3.50	107.65	3767.75	6177.75	3216.43	10598.20	2.31
0.00	3.50	161.48	5651.63	8061.63	4197.78	13463.13	2.93
0.00	3.50	215.30	7535.50	9945.50	5178.74	16327.86	3.56
0.00	3.50	269.13	9419.37	11829.38	6159.69	19192.70	4.18
0.00	3.50	322.95	11303.25	13713.25	7140.65	22057.63	4.81
0.00	4.00	107.65	3767.75	6177.75	3676.37	11057.75	2.41
0.00	4.00	161.48	5651.63	8061.63	4797.46	14062.72	3.06
0.00	4.00	215.30	7535.50	9945.50	5918.56	17067.68	3.72
0.00	4.00	269.13	9419.37	11829.38	7039.65	20072.65	4.37
0.00	4.00	322.95	11303.25	13713.25	8160.74	23077.62	5.03
0.00	4.50	107.65	3767.75	6177.75	4135.92	11517.30	2.51
0.00	4.50	161.48	5651.63	8061.63	5397.15	14662.40	3.19
0.00	4.50	215.30	7535.50	9945.50	6658.38	17807.50	3.88
0.00	4.50	269.13	9419.37	11829.38	7919.61	20952.61	4.56
0.00	4.50	322.95	11303.25	13713.25	9180.83	24097.71	5.25
0.00	5.00	107.65	3767.75	6177.75	4595.47	11976.84	2.61
0.00	5.00	161.48	5651.63	8061.63	5996.83	15262.08	3.32
0.00	5.00	215.30	7535.50	9945.50	7398.20	18547.32	4.04
0.00	5.00	269.13	9419.37	11829.38	8799.56	21832.56	4.76
0.00	5.00	322.95	11303.25	13713.25	10200.93	25117.80	5.47
25.00	.50	107.65	3767.75	4633.31	459.55	6296.49	1.37
25.00	.50	161.48	5651.63	6046.22	599.68	7849.53	1.71
25.00	.50	215.30	7535.50	7459.12	739.82	9402.57	2.05
25.00	.50	269.13	9419.37	8872.03	879.96	10955.61	2.39
25.00	.50	322.95	11303.25	10284.94	1020.09	12508.66	2.73
25.00	1.00	107.65	3767.75	4633.31	919.09	6756.03	1.47
25.00	1.00	151.48	5651.63	6046.22	1199.37	8449.21	1.84
25.00	1.00	215.30	7535.50	7459.12	1479.64	10142.39	2.21
25.00	1.00	269.13	9419.37	8872.03	1759.91	11835.57	2.58
25.00	1.00	322.95	11303.25	10284.94	2040.19	13528.75	2.95
25.00	1.50	107.65	3767.75	4633.31	1378.64	7215.58	1.57
25.00	1.50	161.48	5651.63	6046.22	1799.05	9048.90	1.97
25.00	1.50	215.30	7535.50	7459.12	2219.46	10882.21	2.37
25.00	1.50	269.13	9419.37	8872.03	2639.87	12715.53	2.77
25.00	1.50	322.95	11303.25	10284.94	3060.28	14548.84	3.17
25.00	2.00	107.65	3767.75	4633.31	1838.19	7675.13	1.67
25.00	2.00	161.48	5651.63	6046.22	2398.73	9648.58	2.11
25.00	2.00	215.30	7535.50	7459.12	2959.28	11622.03	2.53
25.00	2.00	269.13	9419.37	8872.03	3519.82	13595.48	2.96
25.00	2.00	322.95	11303.25	10284.94	4080.37	15568.94	3.39
25.00	2.50	107.65	3767.75	4633.31	2297.73	8134.67	1.77
25.00	2.50	161.48	5651.63	6046.22	2998.42	10248.26	2.23
25.00	2.50	215.30	7535.50	7459.12	3699.10	12361.85	2.69
25.00	2.50	269.13	9419.37	8872.03	4399.78	14475.44	3.15
25.00	2.50	322.95	11303.25	10284.94	5100.46	16589.13	3.61
25.00	3.00	107.65	3767.75	4633.31	2757.28	8594.22	1.87
25.00	3.00	161.48	5651.63	6046.22	3598.10	10847.94	2.36
25.00	3.00	215.30	7535.50	7459.12	4438.92	13101.67	2.85
25.00	3.00	269.13	9419.37	8872.03	5279.74	15355.40	3.35
25.00	3.00	322.95	11303.25	10284.94	6120.56	17609.12	3.84
25.00	3.50	107.65	3767.75	4633.31	3216.43	9053.77	1.97
25.00	3.50	161.48	5651.63	6046.22	4197.78	11447.63	2.49
25.00	3.50	215.30	7535.50	7459.12	5178.74	13841.49	3.02
25.00	3.50	269.13	9419.37	8872.03	6159.69	16235.35	3.54
25.00	3.50	322.95	11303.25	10284.94	7140.65	18629.21	4.06
25.00	4.00	107.65	3767.75	4633.31	3676.37	9513.31	2.07
25.00	4.00	161.48	5651.63	6046.22	4797.46	12047.31	2.62
25.00	4.00	215.30	7535.50	7459.12	5918.56	14581.31	3.18
25.00	4.00	269.13	9419.37	8872.03	7039.65	17115.31	3.73
25.00	4.00	322.95	11303.25	10284.94	8160.74	19649.31	4.28
25.00	4.50	107.65	3767.75	4633.31	4135.92	9972.86	2.17

25.00	4.50	161.48	5651.63	6046.22	5397.15	12646.99	2.76
25.00	4.50	215.30	7535.50	7459.12	6658.38	15321.13	3.34
25.00	4.50	269.13	9419.37	8872.03	7919.61	17995.26	3.92
25.00	4.50	322.95	11303.25	10284.94	9180.83	20669.40	4.50
25.00	5.00	137.65	3767.75	4633.31	4595.47	10432.41	2.27
25.00	5.00	161.48	5651.63	6046.22	5996.83	13246.68	2.89
25.00	5.00	215.30	7535.50	7459.12	7398.20	16060.95	3.50
25.00	5.00	269.13	9419.37	8872.03	8799.56	18875.22	4.11
25.00	5.00	322.95	11303.25	10284.94	10200.93	21689.49	4.73

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                    ******************************************        WASH25
                    *                                        *
                    *  DESICCANT SYSTEM ECONOMIC ANALYSIS  *
                    *           BASE YEAR = 1985             *
                    *                                        *
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6.00 PCT. ECONOMIC INFLATION   5.00 PCT. ELECTRICITY ESCALATION
0.00 PCT. SOLAR EQUIP. INFLATION   5.00 PCT. AUX. HEAT ESCALATION
3.00 PCT. DISCOUNT RATE

   35.00 M**2 COLLECTOR AREA   1.02E+11 J/YR TOTAL HEATING LOAD
8.89E+09 J/YR AUX. COOLING REQUIRED   3.37E+10 J/YR AUX. HEATING REQUIRED
   5.50 KW AUX. COOLING CAPACITY   9.00 KW VAPOR COMPRESSION COOL CAP
2.26E+09 J/YR DESICCANT PARASITIC POWER 1.68E+10 J/YR VAP. COMP. PARASITIC POW
   2.50 COP OF AUX. COOLING UNIT   2.50 COP OF VAPOR COMPRESSION UNIT

        ASSUMED 1978 COSTS   COSTS ESCALATED TO 1985
               .0410 ELECTRICITY RATES ($/KW-HR)   .0851
               .0150 AUXILIARY HEATING ($/KW-HR)   .0311
           1500.00   DESICCANT SYSTEM ($)   1506.00
              0.00   STORAGE UNIT ($)   0.00
            610.00   AUXILIARY COOLER UNIT ($)   917.21
            300.00   SYSTEM INSTALLATION ($)   451.09
           1000.00   VAPOR COMPRESSION UNIT ($)   1506.63

   PRESENT WORTH ($) BASED ON 20 YEAR LIFETIME:
           DESICCANT SYSTEM   VAPOR COMPRESSION SYSTEM
               3.11E+02 PARASITIC POWER   2.31E+03
               1.70E+03   HEATING   5.14E+03
               4.90E+02   COOLING
                            TOTAL   8.96E+03

PCT. GOV. SUBSIDY	PCT. COST FOR OMFI	COLLECTOR COST ($/M**2)	COLLECTOR CAPITAL COST ($)	INITIAL CAPITAL COST ($)	OMRI PRESENT WORTH ($)	TOTAL PRESENT WORTH ($)	COST BENEFIT RATIO
0.00	.50	107.65	3767.75	6636.05	493.64	9628.62	1.37
0.00	.50	161.48	5651.63	8519.93	633.78	11652.63	1.30
0.00	.50	215.30	7535.50	10403.80	773.91	13676.64	1.53
0.00	.50	269.13	9419.37	12287.68	914.05	15700.65	1.75
0.00	.50	322.95	11303.25	14171.55	1054.18	17724.66	1.98
0.00	1.00	107.65	3767.75	6636.05	987.28	10122.25	1.13
0.00	1.00	161.48	5651.63	8519.93	1267.55	12286.40	1.37
0.00	1.00	215.30	7535.50	10403.80	1547.82	14450.55	1.61
0.00	1.00	269.13	9419.37	12287.68	1828.10	16614.70	1.85
0.00	1.00	322.95	11303.25	14171.55	2108.37	18778.85	2.10
0.00	1.50	107.65	3767.75	6636.05	1480.92	10615.89	1.19
0.00	1.50	161.48	5651.63	8519.93	1901.33	12920.18	1.44
0.00	1.50	215.30	7535.50	10403.80	2321.73	15224.46	1.70
0.00	1.50	269.13	9419.37	12287.68	2742.14	17528.75	1.96
0.00	1.50	322.95	11303.25	14171.55	3162.55	19833.03	2.21
0.00	2.00	107.65	3767.75	6636.05	1974.55	11109.53	1.24
0.00	2.00	161.48	5651.63	8519.93	2535.10	13553.95	1.51
0.00	2.00	215.30	7535.50	10403.80	3095.65	15998.37	1.79
0.00	2.00	269.13	9419.37	12287.68	3656.19	18442.79	2.06
0.00	2.00	322.95	11303.25	14171.55	4216.74	20887.22	2.33
0.00	2.50	107.65	3767.75	6636.05	2468.19	11603.17	1.30
0.00	2.50	161.48	5651.63	8519.93	3168.88	14187.73	1.58
0.00	2.50	215.30	7535.50	10403.80	3869.56	16772.29	1.87
0.00	2.50	269.13	9419.37	12287.68	4570.24	19356.84	2.16
0.00	2.50	322.95	11303.25	14171.55	5270.92	21941.40	2.45
0.00	3.00	107.65	3767.75	6636.05	2961.83	12096.81	1.35
0.00	3.00	161.48	5651.63	8519.93	3802.65	14821.50	1.65
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|---|---|---|---|---|---|---|---|
| 0.00 | 3.00 | 215.30 | 7535.50 | 10403.80 | 4643.47 | 17546.20 | 1.96 |
| 0.00 | 3.00 | 269.13 | 9419.37 | 12287.68 | 5484.29 | 20270.89 | 2.26 |
| 0.00 | 3.00 | 322.95 | 11303.25 | 14171.55 | 6325.11 | 22995.58 | 2.57 |
| 0.00 | 3.50 | 107.65 | 3767.75 | 6636.05 | 3455.47 | 12590.45 | 1.41 |
| 0.00 | 3.50 | 161.48 | 5651.63 | 8519.93 | 4436.43 | 15455.28 | 1.73 |
| 0.00 | 3.50 | 215.30 | 7535.50 | 10403.80 | 5417.38 | 18320.11 | 2.05 |
| 0.00 | 3.50 | 269.13 | 9419.37 | 12287.68 | 6398.34 | 21184.94 | 2.37 |
| 0.00 | 3.50 | 322.95 | 11303.25 | 14171.55 | 7379.29 | 24049.77 | 2.68 |
| 0.00 | 4.00 | 107.65 | 3767.75 | 6636.05 | 3949.11 | 13084.39 | 1.46 |
| 0.00 | 4.00 | 161.48 | 5651.63 | 8519.93 | 5070.20 | 16089.05 | 1.80 |
| 0.00 | 4.00 | 215.30 | 7535.50 | 10403.80 | 6191.29 | 19094.02 | 2.13 |
| 0.00 | 4.00 | 269.13 | 9419.37 | 12287.68 | 7312.39 | 22098.99 | 2.47 |
| 0.00 | 4.00 | 322.95 | 11303.25 | 14171.55 | 8433.48 | 25103.95 | 2.80 |
| 0.00 | 4.50 | 107.65 | 3767.75 | 6636.05 | 4442.75 | 13577.72 | 1.52 |
| 0.00 | 4.50 | 161.48 | 5651.63 | 8519.93 | 5703.98 | 16722.83 | 1.87 |
| 0.00 | 4.50 | 215.30 | 7535.50 | 10403.80 | 6965.20 | 19867.93 | 2.22 |
| 0.00 | 4.50 | 269.13 | 9419.37 | 12287.68 | 8226.43 | 23013.04 | 2.57 |
| 0.00 | 4.50 | 322.95 | 11303.25 | 14171.55 | 9487.66 | 26158.14 | 2.92 |
| 0.00 | 5.00 | 107.65 | 3767.75 | 6636.05 | 4936.39 | 14071.36 | 1.57 |
| 0.00 | 5.00 | 161.48 | 5651.63 | 8519.93 | 6337.75 | 17356.60 | 1.94 |
| 0.00 | 5.00 | 215.30 | 7535.50 | 10403.80 | 7739.12 | 20641.84 | 2.30 |
| 0.00 | 5.00 | 269.13 | 9419.37 | 12287.68 | 9140.48 | 23927.08 | 2.67 |
| 0.00 | 5.00 | 322.95 | 11303.25 | 14171.55 | 10541.85 | 27212.32 | 3.04 |
| 25.00 | .50 | 107.65 | 3767.75 | 4977.04 | 493.64 | 7969.63 | .89 |
| 25.00 | .50 | 161.48 | 5651.63 | 6389.95 | 633.78 | 9522.64 | 1.06 |
| 25.00 | .50 | 215.30 | 7535.50 | 7802.85 | 773.91 | 11075.69 | 1.24 |
| 25.00 | .50 | 269.13 | 9419.37 | 9215.76 | 914.05 | 12628.73 | 1.41 |
| 25.00 | .50 | 322.95 | 11303.25 | 10628.67 | 1054.18 | 14181.77 | 1.58 |
| 25.00 | 1.00 | 107.65 | 3767.75 | 4977.04 | 987.28 | 8463.24 | .94 |
| 25.00 | 1.00 | 161.48 | 5651.63 | 6389.95 | 1267.55 | 10156.42 | 1.13 |
| 25.00 | 1.00 | 215.30 | 7535.50 | 7802.85 | 1547.82 | 11849.60 | 1.32 |
| 25.00 | 1.00 | 269.13 | 9419.37 | 9215.76 | 1828.10 | 13542.78 | 1.51 |
| 25.00 | 1.00 | 322.95 | 11303.25 | 10628.67 | 2108.37 | 15235.96 | 1.70 |
| 25.00 | 1.50 | 107.65 | 3767.75 | 4977.04 | 1480.92 | 8956.88 | 1.00 |
| 25.00 | 1.50 | 161.48 | 5651.63 | 6389.95 | 1901.33 | 10790.20 | 1.20 |
| 25.00 | 1.50 | 215.30 | 7535.50 | 7802.85 | 2321.73 | 12623.51 | 1.41 |
| 25.00 | 1.50 | 269.13 | 9419.37 | 9215.76 | 2742.14 | 14456.83 | 1.61 |
| 25.00 | 1.50 | 322.95 | 11303.25 | 10628.67 | 3162.55 | 16290.14 | 1.82 |
| 25.00 | 2.00 | 107.65 | 3767.75 | 4977.04 | 1974.55 | 9450.52 | 1.06 |
| 25.00 | 2.00 | 161.48 | 5651.63 | 6389.95 | 2535.10 | 11423.97 | 1.28 |
| 25.00 | 2.00 | 215.30 | 7535.50 | 7802.85 | 3095.65 | 13397.42 | 1.50 |
| 25.00 | 2.00 | 269.13 | 9419.37 | 9215.76 | 3656.19 | 15370.87 | 1.72 |
| 25.00 | 2.00 | 322.95 | 11303.25 | 10628.67 | 4216.74 | 17344.33 | 1.94 |
| 25.00 | 2.50 | 107.65 | 3767.75 | 4977.04 | 2468.19 | 9944.16 | 1.11 |
| 25.00 | 2.50 | 161.48 | 5651.63 | 6389.95 | 3168.88 | 12057.75 | 1.35 |
| 25.00 | 2.50 | 215.30 | 7535.50 | 7802.85 | 3869.56 | 14171.33 | 1.58 |
| 25.00 | 2.50 | 269.13 | 9419.37 | 9215.76 | 4570.24 | 16284.92 | 1.82 |
| 25.00 | 2.50 | 322.95 | 11303.25 | 10628.67 | 5270.92 | 18398.51 | 2.05 |
| 25.00 | 3.00 | 107.65 | 3767.75 | 4977.04 | 2961.83 | 10437.80 | 1.17 |
| 25.00 | 3.00 | 161.48 | 5651.63 | 6389.95 | 3802.65 | 12691.52 | 1.42 |
| 25.00 | 3.00 | 215.30 | 7535.50 | 7802.85 | 4643.47 | 14945.25 | 1.67 |
| 25.00 | 3.00 | 269.13 | 9419.37 | 9215.76 | 5484.29 | 17198.97 | 1.92 |
| 25.00 | 3.00 | 322.95 | 11303.25 | 10628.67 | 6325.11 | 19452.70 | 2.17 |
| 25.00 | 3.50 | 107.65 | 3767.75 | 4977.04 | 3455.47 | 10931.43 | 1.22 |
| 25.00 | 3.50 | 161.48 | 5651.63 | 6389.95 | 4436.43 | 13325.30 | 1.49 |
| 25.00 | 3.50 | 215.30 | 7535.50 | 7802.85 | 5417.38 | 15719.16 | 1.75 |
| 25.00 | 3.50 | 269.13 | 9419.37 | 9215.76 | 6398.34 | 18113.02 | 2.02 |
| 25.00 | 3.50 | 322.95 | 11303.25 | 10628.67 | 7379.29 | 20506.88 | 2.29 |
| 25.00 | 4.00 | 107.65 | 3767.75 | 4977.04 | 3949.11 | 11425.37 | 1.28 |
| 25.00 | 4.00 | 161.48 | 5651.63 | 6389.95 | 5070.20 | 13953.07 | 1.56 |
| 25.00 | 4.00 | 215.30 | 7535.50 | 7802.85 | 6191.29 | 16493.07 | 1.84 |
| 25.00 | 4.00 | 269.13 | 9419.37 | 9215.76 | 7312.39 | 19027.07 | 2.12 |
| 25.00 | 4.00 | 322.95 | 11303.25 | 10628.67 | 8433.48 | 21561.07 | 2.41 |
| 25.00 | 4.50 | 107.65 | 3767.75 | 4977.04 | 4442.75 | 11918.71 | 1.33 |

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|---|---|---|---|---|---|---|---|
| 25.00 | 4.50 | 161.48 | 5651.63 | 6389.95 | 5703.98 | 14592.85 | 1.63 |
| 25.00 | 4.50 | 215.30 | 7535.50 | 7802.85 | 6965.20 | 17266.98 | 1.93 |
| 25.00 | 4.50 | 269.13 | 9419.37 | 9215.76 | 8226.43 | 19941.12 | 2.23 |
| 25.00 | 4.50 | 322.95 | 11303.25 | 10628.67 | 9487.66 | 22615.25 | 2.52 |
| 25.00 | 5.00 | 137.65 | 3767.75 | 4977.04 | 4936.39 | 12412.35 | 1.39 |
| 25.00 | 5.00 | 161.48 | 5651.63 | 6389.95 | 6337.75 | 15226.62 | 1.70 |
| 25.00 | 5.00 | 215.30 | 7535.50 | 7802.85 | 7739.12 | 18040.89 | 2.01 |
| 25.00 | 5.00 | 269.13 | 9419.37 | 9215.76 | 9140.48 | 20855.16 | 2.33 |
| 25.00 | 5.00 | 322.95 | 11303.25 | 10628.67 | 10541.85 | 23669.44 | 2.64 |

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**********************************************          WASH25
*                                            *
*   DESICCANT SYSTEM ECONOMIC ANALYSIS       *
*            BASE YEAR = 1990                *
*                                            *
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6.00 PCT. ECONOMIC INFLATION         5.00 PCT. ELECTRICITY ESCALATION
0.00 PCT. SOLAR EQUIP. INFLATION     5.00 PCT. AUX. HEAT ESCALATION
3.00 PCT. DISCOUNT RATE

  35.00 M**2 COLLECTOR AREA          1.02E+11 J/YR TOTAL HEATING LOAD
8.89E+09 J/YR AUX. COOLING REQUIRED  3.37E+10 J/YR AUX. HEATING REQUIRED
  5.30 KW AUX. COOLING CAPACITY          9.00 KW VAPOR COMPRESSION COOL CAP
2.25E+09 J/YR DESICCANT PARASITIC POWER 1.68E+10 J/YR VAP. COMP. PARASITIC POW
  2.50 COP OF AUX. COOLING UNIT          2.50 COP OF VAPOR COMPRESSION UNIT

     ASSUMED 1978 COSTS                   COSTS ESCALATED TO 1990
            .0410 ELECTRICITY RATES ($/KW-HR)    .1434
            .0150 AUXILIARY HEATING ($/KW-HR)    .0525
           1500.00    DESICCANT SYSTEM ($)      1500.00
              0.00      STORAGE UNIT ($)           0.00
            610.00   AUXILIARY COOLER UNIT ($)   1227.44
            300.00    SYSTEM INSTALLATION ($)     603.66
           1000.00   VAPOR COMPRESSION UNIT ($)  2012.20

   PRESENT WORTH ($) BASED ON  20 YEAR LIFETIME:
           DESICCANT SYSTEM              VAPOR COMPRESSION SYSTEM
              5.25E+02 PARASITIC POWER  3.90E+03
              2.80E+03    HEATING       8.66E+03
              8.25E+02    COOLING
                         TOTAL          1.46E+04
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PCT. GOV. SUBSIDY	PCT. COST FOR OMRI	COLLECTOR COST ($/M**2)	COLLECTOR CAPITAL COST ($)	INITIAL CAPITAL COST ($)	OMRI PRESENT WORTH ($)	TOTAL PRESENT WORTH ($)	COST BENEFIT RATIO
0.00	.50	107.65	3767.75	7098.85	528.06	11837.74	.81
0.00	.50	161.48	5651.63	8982.72	668.20	13861.76	.95
0.00	.50	215.30	7535.50	10866.60	808.34	15885.77	1.09
0.00	.50	269.13	9419.37	12750.47	948.47	17909.78	1.23
0.00	.50	322.95	11303.25	14634.35	1088.61	19933.79	1.37
0.00	1.00	107.65	3767.75	7098.85	1056.13	12365.81	.85
0.00	1.00	161.48	5651.63	8982.72	1336.40	14529.96	1.00
0.00	1.00	215.30	7535.50	10866.60	1616.68	16694.11	1.15
0.00	1.00	269.13	9419.37	12750.47	1896.95	18858.25	1.29
0.00	1.00	322.95	11303.25	14634.35	2177.22	21022.40	1.44
0.00	1.50	107.65	3767.75	7098.85	1584.19	12893.87	.88
0.00	1.50	161.48	5651.63	8982.72	2004.60	15198.16	1.04
0.00	1.50	215.30	7535.50	10866.60	2425.01	17502.44	1.20
0.00	1.50	269.13	9419.37	12750.47	2845.42	19806.73	1.36
0.00	1.50	322.95	11303.25	14634.35	3265.83	22111.01	1.52
0.00	2.00	107.65	3767.75	7098.85	2112.26	13421.94	.92
0.00	2.00	161.48	5651.63	8982.72	2672.80	15866.36	1.09
0.00	2.00	215.30	7535.50	10866.60	3233.35	18310.78	1.26
0.00	2.00	269.13	9419.37	12750.47	3793.90	20755.20	1.42
0.00	2.00	322.95	11303.25	14634.35	4354.44	23199.62	1.59
0.00	2.50	107.65	3767.75	7098.85	2640.32	13950.00	.96
0.00	2.50	161.48	5651.63	8982.72	3341.01	16534.56	1.13
0.00	2.50	215.30	7535.50	10866.60	4041.69	19119.12	1.31
0.00	2.50	269.13	9419.37	12750.47	4742.37	21703.68	1.49
0.00	2.50	322.95	11303.25	14634.35	5443.05	24288.23	1.67
0.00	3.00	107.65	3767.75	7098.85	3168.39	14478.07	.99
0.00	3.00	161.48	5651.63	8982.72	4009.21	17202.76	1.18

0.00	3.00	215.30	7535.50	10866.60	4850.03	19927.46	1.37
0.00	3.00	269.13	9419.37	12750.47	5690.85	22652.15	1.55
0.00	3.00	322.95	11303.25	14634.35	6531.66	25376.84	1.74
0.00	3.50	107.65	3767.75	7098.85	3696.45	15006.13	1.03
0.00	3.50	161.48	5651.63	8982.72	4677.41	17870.96	1.23
0.00	3.50	215.30	7535.50	10866.60	5658.36	20735.79	1.42
0.00	3.50	269.13	9419.37	12750.47	6639.32	23600.62	1.62
0.00	3.50	322.95	11303.25	14634.35	7620.28	26465.46	1.82
0.00	4.00	107.65	3767.75	7098.85	4224.52	15534.20	1.07
0.00	4.00	161.48	5651.63	8982.72	5345.61	18539.16	1.27
0.00	4.00	215.30	7535.50	10866.60	6466.70	21544.13	1.48
0.00	4.00	269.13	9419.37	12750.47	7587.79	24549.10	1.68
0.00	4.00	322.95	11303.25	14634.35	8708.89	27554.07	1.89
0.00	4.00	322.95	11303.25	14634.35	8708.89	27554.07	1.89
0.00	4.50	107.65	3767.75	7098.85	4752.58	16062.26	1.10
0.00	4.50	161.48	5651.63	8982.72	6013.81	19207.37	1.32
0.00	4.50	215.30	7535.50	10866.60	7275.04	22352.47	1.53
0.00	4.50	269.13	9419.37	12750.47	8536.27	25497.57	1.75
0.00	4.50	322.95	11303.25	14634.35	9797.50	28642.68	1.97
0.00	5.00	107.65	3767.75	7098.85	5280.65	16590.33	1.14
0.00	5.00	161.48	5651.63	8982.72	6682.01	19875.57	1.36
0.00	5.00	215.30	7535.50	10866.60	8083.38	23160.81	1.59
0.00	5.00	269.13	9419.37	12750.47	9484.74	26446.05	1.81
0.00	5.00	322.95	11303.25	14634.35	10886.11	29731.29	2.04
25.00	.50	107.65	3767.75	5324.14	528.06	10063.03	.69
25.00	.50	161.48	5651.63	6737.04	668.20	11616.18	.80
25.00	.50	215.30	7535.50	8149.95	808.34	13169.12	.90
25.00	.50	269.13	9419.37	9562.86	948.47	14722.16	1.01
25.00	.50	322.95	11303.25	10975.76	1088.61	16275.20	1.12
25.00	1.00	107.65	3767.75	5324.14	1056.13	10591.10	.73
25.00	1.00	161.48	5651.63	6737.04	1336.40	12284.28	.84
25.00	1.00	215.30	7535.50	6149.95	1616.68	13977.46	.96
25.00	1.00	269.13	9419.37	9562.86	1896.95	15670.64	1.08
25.00	1.00	322.95	11303.25	10975.76	2177.22	17363.81	1.19
25.00	1.50	107.65	3767.75	5324.14	1584.19	11119.16	.76
25.00	1.50	161.48	5651.63	6737.04	2004.60	12952.48	.89
25.00	1.50	215.30	7535.50	8149.95	2425.01	14785.79	1.01
25.00	1.50	269.13	9419.37	9562.86	2845.42	16619.11	1.14
25.00	1.50	322.95	11303.25	10975.76	3265.83	18452.43	1.27
25.00	2.00	107.65	3767.75	5324.14	2112.26	11647.23	.80
25.00	2.00	161.48	5651.63	6737.04	2672.80	13620.68	.93
25.00	2.00	215.30	7535.50	8149.95	3233.35	15594.13	1.07
25.00	2.00	269.13	9419.37	9562.86	3793.90	17567.58	1.21
25.00	2.00	322.95	11303.25	10975.76	4354.44	19541.04	1.34
25.00	2.50	107.65	3767.75	5324.14	2640.32	12175.29	.84
25.00	2.50	161.48	5651.63	6737.04	3341.01	14288.88	.98
25.00	2.50	215.30	7535.50	8149.95	4041.69	16402.47	1.13
25.00	2.50	269.13	9419.37	9562.86	4742.37	18516.06	1.27
25.00	2.50	322.95	11303.25	10975.76	5443.05	20629.65	1.42
25.00	3.00	107.65	3767.75	5324.14	3168.39	12703.36	.87
25.00	3.00	161.48	5651.63	6737.04	4009.21	14957.08	1.03
25.00	3.00	215.30	7535.50	8149.95	4850.03	17210.81	1.18
25.00	3.00	269.13	9419.37	9562.86	5690.85	19464.53	1.34
25.00	3.00	322.95	11303.25	10975.76	6531.66	21718.26	1.49
25.00	3.50	107.65	3767.75	5324.14	3696.45	13231.42	.91
25.00	3.50	161.48	5651.63	6737.04	4677.41	15625.28	1.07
25.00	3.50	215.30	7535.50	8149.95	5658.36	18019.14	1.24
25.00	3.50	269.13	9419.37	9562.86	6639.32	20413.01	1.40
25.00	3.50	322.95	11303.25	10975.76	7620.28	22806.87	1.57
25.00	4.00	107.65	3767.75	5324.14	4224.52	13769.49	.94
25.00	4.00	161.48	5651.63	6737.04	5345.61	16293.48	1.12
25.00	4.00	215.30	7535.50	8149.95	6466.70	18827.43	1.29
25.00	4.00	269.13	9419.37	9562.86	7587.79	21361.49	1.47
25.00	4.00	322.95	11303.25	10975.76	8708.89	23895.43	1.64

25.00	4.50	107.65	3767.75	5324.14	4752.58	14287.55	.98
25.00	4.50	161.48	5651.63	6737.04	6013.81	16961.69	1.16
25.00	4.50	215.30	7535.50	8149.95	7275.04	19635.82	1.35
25.00	4.50	269.13	9419.37	9562.86	8536.27	22309.96	1.53
25.00	4.50	322.95	11303.25	10975.76	9797.50	24984.09	1.71
25.00	5.00	107.65	3767.75	5324.14	5280.65	14815.62	1.02
25.00	5.00	161.48	5651.63	6737.04	6682.01	17629.89	1.21
25.00	5.00	215.30	7535.50	8149.95	8083.38	20444.16	1.40
25.00	5.00	269.13	9419.37	9562.86	9484.74	23258.43	1.60
25.00	5.00	322.95	11303.25	10975.76	10886.11	26072.70	1.79

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* DESICCANT SYSTEM ECONOMIC ANALYSIS *
*        BASE YEAR = 1995          *
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6.00 PCT. ECONOMIC INFLATION       5.00 PCT. ELECTRICITY ESCALATION
0.00 PCT. SOLAR EQUIP. INFLATION    5.00 FCT. AUX. HEAT ESCALATION
3.00 PCT. DISCOUNT RATE

 35.00 M**2 COLLECTOR AREA            1.02E+11 J/YR TOTAL HEATING LOAD
8.89E+09 J/YR AUX. COOLING REQUIRED   3.37E+10 J/YR AUX. HEATING REQUIRED
 5.55 KW AUX. COOLING CAPACITY          9.00 KW VAPOR COMPRESSION COOL CAP
2.26E+09 J/YR DESICCANT PARASITIC POWER 1.68E+10 J/YR VAP. COMP. PARASITIC POW
 2.50 COP OF AUX. COOLING UNIT           2.50 COP OF VAPOR COMPRESSION UNIT
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    ASSUMED 1978 COSTS                    COSTS ESCALATED TO 1995
            .0410 ELECTRICITY RATES ($/KW-HR)    .2417
            .0150 AUXILIARY HEATING ($/KW-HR)    .0884
         1500.00      DESICCANT SYSTEM ($)      1500.00
            0.00          STORAGE UNIT ($)         0.00
          610.00   AUXILIARY COOLER UNIT ($)    1642.59
          300.00    SYSTEM INSTALLATION ($)      807.83
         1000.00   VAPOR COMPRESSION UNIT ($)   2692.77
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   PRESENT WORTH ($) BASED ON  20 YEAR LIFETIME:
        DESICCANT SYSTEM              VAPOR COMPRESSION SYSTEM
            8.84E+02 PARASITIC POWER   5.57E+03
            4.82E+03      HEATING      1.46E+04
            1.39E+03      COOLING
                          TOTAL        2.39E+04
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PCT. GOV. SUBSIDY	PCT. COST FOR OMRI	COLLECTOR COST ($/M**2)	COLLECTOR CAPITAL COST ($)	INITIAL CAPITAL COST ($)	OMRI PRESENT WORTH ($)	TOTAL PRESENT WORTH ($)	COST BENEFIT RATIO
0.00	.50	107.65	3767.75	7718.17	574.13	15387.80	.64
0.00	.50	161.48	5651.63	9602.05	714.27	17411.81	.73
0.00	.50	215.30	7535.50	11485.92	854.41	19435.83	.81
0.00	.50	269.13	9419.37	13369.80	994.54	21459.84	.9?
0.00	.50	322.95	11303.25	15253.67	1134.68	23483.85	.98
0.00	1.00	107.65	3767.75	7718.17	1148.27	15961.94	.67
0.00	1.00	151.48	5651.63	9602.05	1428.54	18126.09	.76
0.00	1.00	215.30	7535.50	11485.92	1708.82	20290.23	.85
0.00	1.00	269.13	9419.37	13369.80	1989.09	22454.38	.94
0.00	1.00	322.95	11303.25	15253.67	2269.36	24618.53	1.03
0.00	1.50	107.65	3767.75	7718.17	1722.40	16536.07	.69
0.00	1.50	161.48	5651.63	9602.05	2142.81	18840.36	.79
0.00	1.50	215.30	7535.50	11485.92	2563.22	21144.64	.89
0.00	1.50	269.13	9419.37	13369.80	2983.63	23448.93	.98
0.00	1.50	322.95	11303.25	15253.67	3404.04	25753.21	1.08
0.00	2.00	107.65	3767.75	7718.17	2296.54	17110.21	.72
0.00	2.00	161.48	5651.63	9602.05	2857.08	19554.63	.82
0.00	2.00	215.30	7535.50	11485.92	3417.63	21999.05	.92
0.00	2.00	269.13	9419.37	13369.80	3978.18	24443.47	1.02
0.00	2.00	322.95	11303.25	15253.67	4538.72	26887.89	1.13
0.00	2.50	107.65	3767.75	7718.17	2870.67	17684.34	.74
0.00	2.50	161.48	5651.63	9602.05	3571.36	20268.90	.85
0.00	2.50	215.30	7535.50	11485.92	4272.04	22853.46	.96
0.50	2.50	269.13	9419.37	13369.80	4972.72	25438.01	1.07
0.00	2.50	322.95	11303.25	15253.67	5673.40	28022.57	1.17
0.00	3.00	107.65	3767.75	7718.17	3444.81	18258.48	.77
0.00	3.00	161.48	5651.63	9602.05	4285.63	20983.17	.88

0.00	3.00	215.30	7535.50	11485.92	5126.45	23707.86	.99
0.00	3.00	269.13	9419.37	13369.80	5967.27	26432.56	1.11
0.00	3.00	322.95	11303.25	15253.67	6808.08	29157.25	1.22
0.00	3.50	107.65	3767.75	7718.17	4318.94	18832.61	.79
0.00	3.50	161.48	5651.63	9602.05	4999.90	21697.44	.91
0.00	3.50	215.30	7535.50	11485.92	5980.85	24562.27	1.03
0.00	3.50	269.13	9419.37	13369.80	6961.81	27427.10	1.15
0.00	3.50	322.95	11303.25	15253.67	7942.76	30291.93	1.27
0.00	4.00	107.65	3767.75	7718.17	4593.08	19546.75	.81
0.00	4.00	161.48	5651.63	9602.05	5714.17	22411.71	.94
0.00	4.00	215.30	7535.50	11485.92	6835.26	25416.68	1.07
0.00	4.00	269.13	9419.37	13369.80	7956.35	28421.65	1.19
0.00	4.00	322.95	11303.25	15253.67	9077.45	31426.61	1.32
0.00	4.50	107.65	3767.75	7718.17	5167.21	19960.88	.84
0.00	4.50	161.48	5651.63	9602.05	6428.44	23125.98	.97
0.00	4.50	215.30	7535.50	11485.92	7689.67	26271.09	1.10
0.00	4.50	269.13	9419.37	13369.80	8950.90	29416.19	1.23
0.00	4.50	322.95	11303.25	15253.67	10212.13	32561.30	1.36
0.00	5.00	107.65	3767.75	7718.17	5741.35	20555.02	.86
0.00	5.00	161.48	5651.63	9602.05	7142.71	23840.26	1.00
0.00	5.00	215.30	7535.50	11485.92	8544.08	27125.50	1.14
0.00	5.00	269.13	9419.37	13369.80	9945.44	30410.74	1.27
0.00	5.00	322.95	11303.25	15253.67	11346.81	33695.98	1.41
25.00	.50	107.65	3767.75	5788.63	574.13	13458.26	.56
25.00	.50	161.48	5651.63	7201.54	714.27	15011.30	.63
25.00	.50	215.30	7535.50	8614.44	854.41	16564.35	.69
25.00	.50	269.13	9419.37	10027.35	994.54	18117.39	.76
25.00	.50	322.95	11303.25	11440.25	1134.68	19670.43	.82
25.00	1.00	107.65	3767.75	5788.63	1248.27	14032.39	.59
25.00	1.00	161.48	5651.63	7201.54	1428.54	15725.57	.66
25.00	1.00	215.30	7535.50	8614.44	1708.82	17418.75	.73
25.00	1.00	269.13	9419.37	10027.35	1989.09	19111.93	.80
25.00	1.00	322.95	11303.25	11440.25	2269.36	20805.11	.87
25.00	1.50	107.65	3767.75	5788.63	1722.40	14606.53	.61
25.00	1.50	161.48	5651.63	7201.54	2142.81	16439.85	.69
25.00	1.50	215.30	7535.50	8614.44	2563.22	18273.16	.77
25.00	1.50	269.13	9419.37	10027.35	2983.63	20106.48	.84
25.00	1.50	322.95	11303.25	11440.25	3404.04	21939.79	.92
25.00	2.00	107.65	3767.75	5788.63	2296.54	15180.66	.64
25.00	2.00	161.48	5651.63	7201.54	2857.08	17154.12	.72
25.00	2.00	215.30	7535.50	8614.44	3417.63	19127.57	.80
25.00	2.00	269.13	9419.37	10027.35	3978.18	21101.12	.88
25.00	2.00	322.95	11303.25	11440.25	4538.72	23074.47	.97
25.00	2.50	107.65	3767.75	5788.63	2870.67	15754.80	.66
25.00	2.50	161.48	5651.63	7201.54	3571.36	17868.39	.75
25.00	2.50	215.30	7535.50	8614.44	4272.04	19981.98	.84
25.00	2.50	269.13	9419.37	10027.35	4972.72	22095.57	.93
25.00	2.50	322.95	11303.25	11440.25	5673.40	24209.15	1.01
25.00	3.00	107.65	3767.75	5788.63	3444.81	16328.93	.68
25.00	3.00	161.48	5651.63	7201.54	4285.63	18582.66	.78
25.00	3.00	215.30	7535.50	8614.44	5126.45	20836.38	.87
25.00	3.00	269.13	9419.37	10027.35	5967.27	23090.11	.97
25.00	3.00	322.95	11303.25	11440.25	6808.08	25343.83	1.06
25.00	3.50	107.65	3767.75	5788.63	4018.94	16903.07	.71
25.00	3.50	161.48	5651.63	7201.54	4999.90	19296.93	.81
25.00	3.50	215.30	7535.50	8614.44	5980.85	21690.79	.91
25.00	3.50	269.13	9419.37	10027.35	6961.81	24084.65	1.01
25.00	3.50	322.95	11303.25	11440.25	7942.76	26478.52	1.11
25.00	4.00	107.65	3767.75	5788.63	4593.08	17477.20	.73
25.00	4.00	161.48	5651.63	7201.54	5714.17	20011.20	.84
25.00	4.00	215.30	7535.50	8614.44	6835.26	22545.20	.94
25.00	4.00	269.13	9419.37	10027.35	7956.35	25079.20	1.05
25.00	4.00	322.95	11303.25	11440.25	9077.45	27613.20	1.16
25.00	4.50	107.65	3767.75	5788.63	5167.21	18051.34	.76

25.00	4.50	161.48	5651.63	7201.54	6428.44	20725.47	.87
25.00	4.50	215.30	7535.50	8614.44	7689.67	23399.61	.98
25.00	4.50	269.13	9419.37	10027.35	8950.90	26073.74	1.09
25.00	4.50	322.95	11303.25	11440.25	10212.13	28747.88	1.20
25.00	5.00	107.65	3767.75	5788.63	5741.35	18625.47	.78
25.00	5.00	161.48	5651.63	7201.54	7142.71	21439.74	.90
25.00	5.00	215.30	7535.50	8614.44	8544.08	24254.01	1.02
25.00	5.00	269.13	9419.37	10027.35	9945.44	27068.29	1.13
25.00	5.00	322.95	11303.25	11440.25	11346.81	29882.56	1.25

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   ********************************************        WASH25
   *                                          *
   * DESICCANT SYSTEM ECONOMIC ANALYSIS *
   *          BASE YEAR = 2000               *
   *                                          *
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6.00 PCT. ECONOMIC INFLATION	5.00 PCT. ELECTRICITY ESCALATION
3.00 PCT. SOLAR EQUIP. INFLATION	5.00 PCT. AUX. HEAT ESCALATION
3.00 PCT. DISCOUNT RATE	

35.00 M**2 COLLECTOR AREA	1.02E+11 J/YR TOTAL HEATING LOAD
8.89E+09 J/YR AUX. COOLING REQUIRED	3.37E+10 J/YR AUX. HEATING REQUIRED
5.55 KW AUX. COOLING CAPACITY	9.00 KW VAPOR COMPRESSION COOL CAP
2.26E+09 J/YR DESICCANT PARASITIC POWER 1.68E+10 J/YR VAP. COMP. PARASITIC POW	
2.50 COP OF AUX. COOLING UNIT	2.50 COP OF VAPOR COMPRESSION UNIT

ASSUMED 1978 COSTS		COSTS ESCALATED TO 2000
.0410	ELECTRICITY RATES ($/KW-HR)	.4073
.0150	AUXILIARY HEATING ($/KW-HR)	.1490
1500.00	DESICCANT SYSTEM ($)	1500.00
0.00	STORAGE UNIT ($)	0.00
610.00	AUXILIARY COOLER UNIT ($)	2198.16
300.00	SYSTEM INSTALLATION ($)	1081.06
1000.00	VAPOR COMPRESSION UNIT ($)	3603.54

PRESENT WORTH ($) BASED ON 20 YEAR LIFETIME:

DESICCANT SYSTEM		VAPOR COMPRESSION SYSTEM
1.49E+03	PARASITIC POWER	1.11E+04
8.12E+03	HEATING	2.46E+04
2.34E+03	COOLING	
	TOTAL	3.93E+04

PCT. GOV. SUBSIDY	PCT. COST FOR OM&I	COLLECTOR COST ($/M**2)	COLLECTOR CAPITAL COST ($)	INITIAL CAPITAL COST ($)	OM&I PRESENT WORTH ($)	TOTAL PRESENT WORTH ($)	COST BENEFIT RATIO
0.00	.50	107.65	3767.75	8546.97	635.79	21139.08	.54
0.00	.50	161.48	5651.63	10430.84	775.92	23163.09	.59
0.00	.50	215.30	7535.50	12314.72	916.06	25187.10	.64
0.00	.50	269.13	9419.37	14198.59	1056.20	27211.11	.69
0.00	.50	322.95	11303.25	16082.47	1196.33	29235.12	.74
0.00	1.00	107.65	3767.75	8546.97	1271.57	21774.86	.55
0.00	1.00	161.48	5651.63	10430.84	1551.85	23939.01	.61
0.00	1.00	215.30	7535.50	12314.72	1832.12	26103.16	.66
0.00	1.00	269.13	9419.37	14198.59	2112.39	28267.31	.72
0.00	1.00	322.95	11303.25	16082.47	2392.67	30431.46	.77
0.00	1.50	107.65	3767.75	8546.97	1907.36	22410.65	.57
0.00	1.50	161.48	5651.63	10430.84	2327.77	24714.94	.63
0.00	1.50	215.30	7535.50	12314.72	2748.18	27019.22	.69
0.00	1.50	269.13	9419.37	14198.59	3168.59	29323.51	.75
0.00	1.50	322.95	11303.25	16082.47	3589.00	31627.79	.81
0.00	2.00	107.65	3767.75	8546.97	2543.15	23046.44	.59
0.00	2.00	161.48	5651.63	10430.84	3103.69	25490.86	.65
0.00	2.00	215.30	7535.50	12314.72	3664.24	27935.28	.71
0.00	2.00	269.13	9419.37	14198.59	4224.78	30379.70	.77
0.00	2.00	322.95	11303.25	16082.47	4785.33	32824.12	.84
0.00	2.50	107.65	3767.75	8546.97	3178.93	23682.22	.60
0.00	2.50	161.48	5651.63	10430.84	3879.62	26266.78	.67
0.00	2.50	215.30	7535.50	12314.72	4580.30	28851.34	.73
0.00	2.50	269.13	9419.37	14198.59	5280.98	31435.90	.80
0.00	2.50	322.95	11303.25	16082.47	5981.66	34020.45	.87
0.00	3.00	107.65	3767.75	8546.97	3814.72	24318.01	.62
0.00	3.00	161.48	5651.63	10430.84	4655.54	27042.71	.69

0.00	3.00	215.30	7535.50	12314.72	5496.36	29767.40	.76
0.00	3.00	269.13	9419.37	14198.59	6337.18	32492.09	.83
0.00	3.00	322.95	11303.25	16082.47	7178.00	35216.79	.90
0.00	3.50	107.65	3767.75	8546.97	4450.51	24953.80	.64
0.00	3.50	161.48	5651.63	10430.84	5431.46	27818.63	.71
0.00	3.50	215.30	7535.50	12314.72	6412.42	30683.46	.78
0.00	3.50	269.13	9419.37	14198.59	7393.37	33548.29	.85
0.00	3.50	322.95	11303.25	16082.47	8374.33	36413.12	.93
0.00	4.00	107.65	3767.75	8546.97	5386.29	25589.58	.65
0.00	4.00	161.48	5651.63	10430.84	6207.38	28594.55	.73
0.00	4.00	215.30	7535.50	12314.72	7328.48	31599.52	.80
0.00	4.00	269.13	9419.37	14198.59	8449.57	34604.49	.88
0.00	4.00	322.95	11303.25	16082.47	9570.66	37609.45	.96
0.00	4.50	107.65	3767.75	8546.97	5722.08	26225.37	.67
0.00	4.50	161.48	5651.63	10430.84	6983.31	29370.47	.75
0.00	4.50	215.30	7535.50	12314.72	8244.54	32515.58	.83
0.00	4.50	269.13	9419.37	14198.59	9505.77	35660.68	.91
0.00	4.50	322.95	11303.25	16082.47	10766.99	38805.79	.99
0.00	5.00	107.65	3767.75	8546.97	6357.87	26861.16	.68
0.00	5.00	161.48	5651.63	10430.84	7759.23	30146.40	.77
0.00	5.00	215.30	7535.50	12314.72	9160.60	33431.64	.85
0.00	5.00	269.13	9419.37	14198.59	10561.96	36716.88	.94
0.00	5.00	322.95	11303.25	16082.47	11963.33	40002.12	1.02
25.00	.50	107.65	3767.75	6410.23	635.79	19002.34	.48
25.00	.50	161.48	5651.63	7823.13	775.92	20555.38	.52
25.00	.50	215.30	7535.50	9236.04	916.06	22108.42	.56
25.00	.50	269.13	9419.37	10648.95	1056.20	23661.46	.60
25.00	.50	322.95	11303.25	12061.85	1196.33	25214.51	.64
25.00	1.00	107.65	3767.75	6410.23	1271.57	19638.12	.50
25.00	1.00	161.48	5651.63	7823.13	1551.85	21331.30	.54
25.00	1.00	215.30	7535.50	9236.04	1832.12	23024.48	.59
25.00	1.00	269.13	9419.37	10648.95	2112.39	24717.66	.63
25.00	1.00	322.95	11303.25	12061.85	2392.67	26410.84	.67
25.00	1.50	107.65	3767.75	6410.23	1907.36	20273.91	.52
25.00	1.50	161.48	5651.63	7823.13	2327.77	22107.23	.56
25.00	1.50	215.30	7535.50	9236.04	2748.19	23940.54	.61
25.00	1.50	269.13	9419.37	10648.95	3168.59	25773.86	.66
25.00	1.50	322.95	11303.25	12061.85	3589.00	27607.17	.70
25.00	2.00	107.65	3767.75	6410.23	2543.15	20909.70	.53
25.00	2.00	161.48	5651.63	7823.13	3103.69	22883.15	.58
25.00	2.00	215.30	7535.50	9236.04	3664.24	24856.60	.63
25.00	2.00	269.13	9419.37	10648.95	4224.78	26830.05	.68
25.00	2.00	322.95	11303.25	12061.85	4785.33	28803.51	.73
25.00	2.50	107.65	3767.75	6410.23	3178.93	21545.48	.55
25.00	2.50	161.48	5651.63	7823.13	3879.62	23659.07	.60
25.00	2.50	215.30	7535.50	9236.04	4580.30	25772.66	.66
25.00	2.50	269.13	9419.37	10648.95	5280.98	27886.25	.71
25.00	2.50	322.95	11303.25	12061.85	5981.66	29999.84	.76
25.00	3.00	107.65	3767.75	6410.23	3814.72	22181.27	.56
25.00	3.00	161.48	5651.63	7823.13	4655.54	24434.99	.62
25.00	3.00	215.30	7535.50	9236.04	5496.36	26688.72	.68
25.00	3.00	269.13	9419.37	10648.95	6337.18	28942.45	.74
25.00	3.00	322.95	11303.25	12061.85	7178.00	31196.17	.79
25.00	3.50	107.65	3767.75	6410.23	4450.51	22817.06	.58
25.00	3.50	161.48	5651.63	7823.13	5431.46	25210.92	.64
25.00	3.50	215.30	7535.50	9236.04	6412.42	27604.78	.70
25.00	3.50	269.13	9419.37	10648.95	7393.37	29998.64	.76
25.00	3.50	322.95	11303.25	12061.85	8374.33	32392.50	.82
25.00	4.00	107.65	3767.75	6410.23	5086.29	23452.84	.60
25.00	4.00	161.48	5651.63	7823.13	6207.38	25996.84	.66
25.00	4.00	215.30	7535.50	9236.04	7328.48	28520.84	.73
25.00	4.00	269.13	9419.37	10648.95	8449.57	31054.84	.79
25.00	4.00	322.95	11303.25	12061.85	9570.66	33588.84	.86
25.00	4.50	107.65	3767.75	6410.23	5722.08	24088.63	.61

25.00	4.50	161.48	5651.63	7823.13	6983.31	26762.76	.68
25.00	4.50	215.30	7535.50	9236.04	8244.54	29436.90	.75
25.00	4.50	269.13	9419.37	10648.95	9505.77	32111.03	.82
25.00	4.50	322.95	11303.25	12061.85	10766.99	34785.17	.89
25.00	5.00	107.65	3767.75	6410.23	6357.87	24724.42	.63
25.00	5.00	161.48	5651.63	7823.13	7759.23	27538.69	.70
25.00	5.00	215.30	7535.50	9236.04	9160.60	30352.96	.77
25.00	5.00	269.13	9419.37	10648.95	10561.96	33167.23	.84
25.00	5.00	322.95	11303.25	12061.85	11963.33	35981.50	.92