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SOLAR EVACUATED TUBE COLLECTOR —ABSORPTION CHILLER SYSTEMS SIMULATION

By
James A. Leflar
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December 1977

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MASTER

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Solar Energy

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REA

ABSTRACT OF THESIS

SOLAR EVACUATED TUBE COLLECTOR -- ABSORPTION CHILLER SYSTEMS SIMULATION

A lithium-bromide absorption air conditioner is being commercially produced by Arkla Industries in a three-ton size for use with residential solar systems. The widespread incorporation of residential solar cooling systems could substantially reduce the load on the electrical power generation network and possibly reduce the cost of residential air conditioning in many areas.

Previous research indicates that evacuated tube solar collectors are much more efficient at collecting solar energy in the temperature range of absorption chiller operation. Therefore, a residential air conditioning system incorporating an Arkla Solaire absorption chiller and Corning Glass Works evacuated tube collectors is simulated and the design parameters studied. Mathematical models of the evacuated tube collector and Arkla absorption chiller based on experimental results of the components have been created and incorporated into a complete system simulation. The chiller model includes transient start-up effects and the evacuated tube collector model includes numerous optical effects.

A standard Arkla chiller in a humid climate (Washington, D.C.) and an Arkla unit with a modified charge for dry climates (Fort Collins, Colorado) are studied. Design parameters considered include the use of chilled water storage to reduce transient start-up effects of the absorption unit, the effects of removing heat from the solar system for preheating service hot water, the use of a tempering valve to prevent

over-firing of the absorption unit in dry climates, and solar storage sizing considerations. The study results and conclusions are used to specify a cooling system design.

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Fall, 1977

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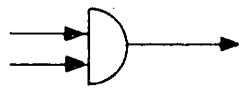
NOMENCLATURE

Binary Logic



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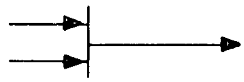


"OR"

English Symbols

A_a	Collector absorber area
C	Constant for determining the building cooling load directly attributable to solar radiation
EFF_n	Efficiency of a single collector tube at normal solar incidence
EFF_{TOTAL}	Efficiency of a collector module accounting for both beam and diffuse radiation components
$EFF_{\psi, \theta}$	Efficiency of a collector module accounting for beam radiation at any incidence angle
FACTOR	The ratio $(\tau\alpha)_e/(\tau\alpha)_n$
H	Enthalpy of the atmospheric air
I_B	Beam radiation received on the collector surface per unit area
I_D	Diffuse radiation received on the collector surface per unit area
$I_{f,n}$	Fraction of the solar radiation at normal incidence absorbed by the collector absorber plate
$I_{f;\psi,\theta}$	Fraction of the solar radiation at any incident angle absorbed by the collector absorber plate
I_H	Total solar insolation received on a unit area of horizontal surface
I_T	Total solar insolation received on a unit area of the collector surface
M_1	Initial chilled water storage tank mass during a simulation time step

Nomenclature (continued)

English Symbols

M_2	Final chilled water storage tank mass during a simulation time step
M_+	Mass of water added to a chilled water storage tank
P_W	Partial pressure of the water vapor in the atmosphere
$\dot{Q}_{ENV}$	Rate at which heat is lost from the solar storage tank
$\dot{Q}_{GEN}$	Rate at which heat is generated inside the conditioned space by people and energy consuming devices
$\dot{Q}_{LATENT}$	Latent heat load rate of the building
$\dot{Q}_u$	Rate of useful solar heat gain by the collector
S	Collector slope (measured from the horizontal)
T_{AMB}	Ambient outdoor dry bulb temperature
T_{cw}	Temperature of the condensor water entering the chiller
T_{IN}	Collector inlet temperature
T_{DP}	Outdoor dewpoint temperature
T_R	Room dry bulb temperature
T_{WB}	Atmospheric wet bulb temperature
T_1	Initial chilled water storage tank temperature during a simulation time step
T_2	Final chilled water storage tank temperature during a simulation time step
T_+	Temperature of water added to a chilled water storage tank
UA	Building heat loss coefficient
W	Atmospheric absolute humidity ratio

Greek Symbols

δ	Declination (the angular position of the sun at solar noon with respect to the plane of the equator)
ϕ	Latitude (north positive)
ϕ'	Artificial latitude (ϕ - S - 90°)
$(\tau\alpha)_e$	Effective collector tau alpha product at any sun angle
$(\tau\alpha)_n$	Effective collector tau alpha product for normal solar incidence
θ	Collector sun angle defined by Figure 2.2.4
ω	Hour angle, solar noon being zero, and each hour equaling 15 degrees of longitude with mornings positive and afternoons negative
ψ	Collector sun angle defined by Figure 2.2.4

Simulation Variables

Q_{SHW} ENV	Heat lost by the service hot water solar preheat tank
Q_{SHW} SOLAR	Heat supplied to service hot water by solar energy
Q_{SHW} AUX	Heat supplied to service hot water by an auxiliary fuel
QU	Useful energy gain by the solar collectors
Q_{OVER}	Energy lost from the solar system by storage boiling
A_{COL}	Collector area (based on absorber plate)
Q_{TANK}	Heat removed from solar storage for service hot water and cooling purposes
$Q_{A/C}$ AUX	Heat energy supplied to the chiller by auxiliary fuel
Q_{COOL}	Heat removed from the residence by the chiller
Q_{SOLAR}ENV	Heat losses from the solar storage tank
COP	Coefficient of performance of the absorption chiller

Nomenclature (continued)

Simulation Variables

η_{COL}	Collector efficiency (based on absorber area)
f_{SHW}	Fraction of the chiller generator energy supplied by the solar system
TOTAL AUX	Total auxiliary fuel requirements for cooling and service hot water over the simulation period
C.W.S. cycles	The number of times chilled water storage is used during the cooling season

Subscripts

a	Based on collector absorber area

CHAPTER 1

BACKGROUND AND REVIEW OF LITERATURE

Absorption air conditioning and refrigeration systems have been used in residential, industrial, and ocean craft systems for several decades -- wherever cooling has been needed and an inexpensive heat source has been available. In more recent years, absorption units remain in many industrial and naval installations, but have been replaced by vapor-compression air conditioning units in residential use. This change has been due in large part to the convenience and widespread availability of electricity, the comparable operating costs of the vapor-compression air conditioner, and its lower capital investment.

Solar systems are increasingly being used in residences to provide energy for space heating and domestic hot water needs. During the months when space cooling is required, however, the solar system is capable of supplying inexpensive hot water to operate an absorption air conditioning system. Although the coefficient of performance of an absorption air conditioner is one-fourth to one-third that of a vapor-compression air conditioner, the heat energy provided by the solar system may be so inexpensive that a savings can accrue from use of the absorption system over the cost of operating a vapor-compression system. Such savings might then repay the initial investment in the absorption air conditioning system and make it more economically attractive than a vapor-compression unit.

To this end there has been much research with absorption systems using solar heated hot water instead of the high temperature products of a combustion process [1,2]. As a result of this research, Arkla Industries of Evansville, Indiana is commercially producing a three-ton residential lithium-bromide absorption air conditioning system which uses solar heated hot water as an energy source. The unit is a variation of an absorption chiller which has been used extensively for over twenty years with a record of high reliability. With one year of tests completed on the solar powered unit at the Colorado State University Solar House I, the reliability aspect of the Arkla Solaire chiller appears to be excellent.

The Arkla solar powered absorption chiller is the only solar operated air conditioner commercially produced in the United States. Because of its availability, it was chosen as the object of analysis for the simulation and design of solar residential air conditioning systems.

Flat-plate solar collectors have been the type most often used for residential heating systems. However, the operation of an absorption chiller requires higher operating temperatures than does a solar heating system. Previous research [4,5,6] indicates that evacuated tube collectors are much more efficient at collecting solar energy in the temperature range required for operating an absorption chiller than flat-plate collectors. Experiments have been conducted at the Colorado State University Solar House I with a prototype evacuated tube collector manufactured by Corning Glass Works, Corning, New York and therefore an accurate model of this particular collector's performance has been developed.

The simulation of a residential air conditioning system using an Arkla chiller and Corning evacuated tube collectors has been undertaken to learn more about the performance and design of such a system. These components or similar ones could be incorporated in a large number of residences in the very near future and might provide a practical energy saving technology [3]. In this hope the following work is presented.

CHAPTER 2
SIMULATION MODELS

2.1 SYSTEM DESCRIPTION

The simulations model a residential building containing a complete liquid type solar system with an Arkla Solaire WF-36 absorption chiller and Corning evacuated tube collectors. The Colorado State University Solar House I in Fort Collins, Colorado is used as a building model as it contains such a system and simulation output can therefore be checked against actual data. CSU Solar House I is a two-story frame residential structure with 140 square meters (1500 ft^2) of living space on each floor. The upper story has double and triple glazed windows in the amount of fourteen percent of the floor area to provide natural lighting while not allowing an unreasonable amount of heat gain or loss. All windows are shaded from the sun during the summer months by reveals and overhangs which also allow the sun to penetrate the south-facing windows during the winter months. The house is insulated with 8.9 centimeters (3.5 in) of fiberglass in the walls (R = 6.85 °Cm^2/w; 12 Hr-°F-Ft^2/Btu) and 14 centimeters (5.5 in) of fiberglass in the ceiling (R = 10.85 °Cm^2/w; 19 Hr-°F-Ft^2/Btu), giving the house a design heat load of 387 w/°C (17,600 Btu/°F-day). The design cooling load is 10.5 kilowatts (3 tons or 36,000 Btu/hr). The roof is south-facing at a slope of 45 degrees and contains 71.3 square meters (768 ft^2) of collector surface.

A complete solar cooling system including chilled water storage is shown in Figure 2.1.1 with accompanying component descriptions in Table 2.1.1. Four basic subsystems can be identified in the schematic:

I. Components (1) through (5) -- Solar Collection and Heat Storage System

II. Components (6) through (11) -- Service Hot Water Solar Preheat and Auxiliary System

III. Component (15) -- Chiller Auxiliary Boiler

IV. Components (12) through (14) and (16) through (26) -- Arkla Chiller and Chilled Water Storage System

The collectors are Corning evacuated tube collectors using a coolant mixture of 50/50 antifreeze and water by weight under .07 pascals (15 psi) pressure, which induces a boiling point of approximately 118°C (245°F). So that a minimum amount of antifreeze is required, solar heat captured by the collectors is transferred to a 4160 liter (1100 gallon) storage tank through a counterflow heat exchanger. (Neither Corning nor any other prototype liquid evacuated tube solar collector made at this time can be protected by drain down so antifreeze must be used in any climate subject to freezing conditions.) The storage tank is foam insulated to R = 17.14 $°Cm^2/w$ (R = 30 Hr-°F-Ft^2/Btu) and nearly all tank losses are to the conditioned space in the house, thereby adding to the total cooling load. The collector/storage system control logic is represented in binary logic in Figure 2.1.2.

Service hot water is preheated by use of a preheat tank and heat exchanger connected to the solar storage tank. As the minimum usable hot water temperature for the Arkla chiller is 68°C (155°F) (for the modified charge unit) and the service hot water set temperature is generally 60°C (140°F) or less, most of the service hot water required

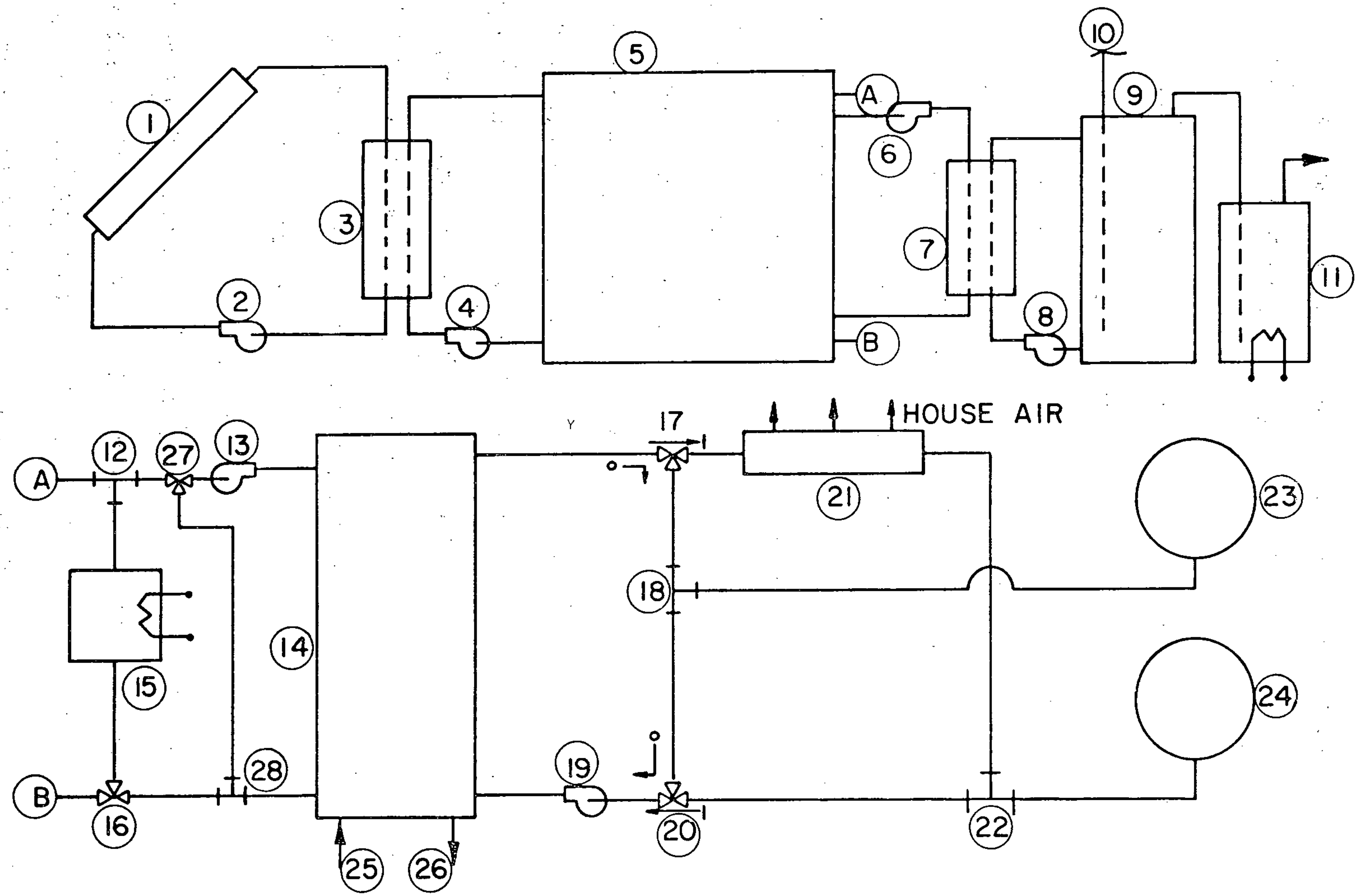


Figure 2.1.1. Solar Cooling System with Chilled Water Storage

Table 2.1.1. Solar System Component Descriptions

Component Number	Description
1	Corning evacuated tube collectors
2	Collector fluid (antifreeze solution) pump
3	Collector/storage heat exchanger
4	Collector/storage heat exchanger pump
5	Heat storage tank (water)
6	Solar hot water preheat heat exchanger pump (shell side)
7	Solar hot water preheat heat exchanger
8	Solar hot water preheat heat exchanger pump (tube side)
9	Solar hot water preheat tank
10	Potable water supply
11	Auxiliary fueled hot water heater
12	Tee
13	Arkla generator pump
14	Arkla 3 ton lithium bromide chiller
15	Auxiliary boiler for Arkla generator
16	Three-way control valve
17	Three-way control valve *
18	Tee
19	Chilled water pump
20	Three-way control valve *
21	Duct heat exchange coil (chilled water/house air)
22	Tee
23	"Cool" storage
24	"Warm" storage
25	Condenser water flow from cooling tower
26	Condenser water flow to cooling tower
27	Tempering valve (used only in arid climates)
28	Tee (used only in arid climates

* Operational Mode	Control Valve Position	
	17	20
Chilling directly to load	1	1
Cold storage to load	1	0 (chiller off)
Chilling to cold storage	0	1

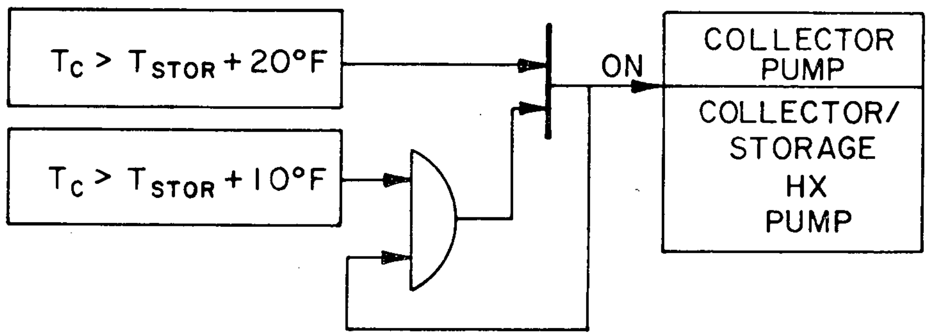
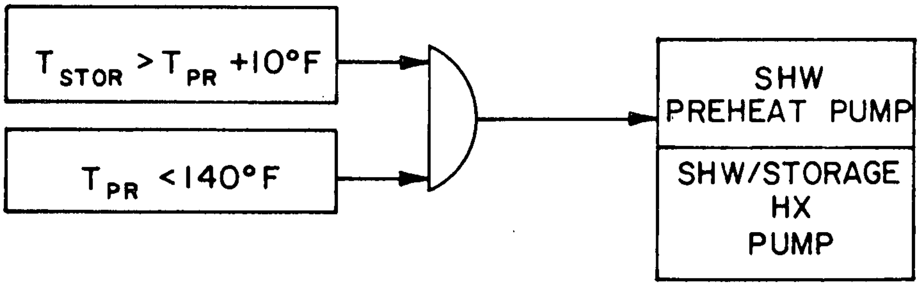


Figure 2.1.2 Collector/Storage Control Logic

during the summer months can be provided by solar energy. Concern has existed that heat removal from storage for preheating service hot water may cause more auxiliary energy to be used to operate the chiller than would be used heating service hot water. Such concern has prompted investigations at Colorado State University into other possible arrangements of the solar energy system for preheating service hot water. However, the arrangement shown in Figure 2.1.1 is indicated to be the best of all possibilities considered. The service hot water system control logic is represented in binary form in Figure 2.1.3.

The auxiliary boiler may be fueled by gas or oil, and provides hot water to the chiller when the solar energy system is not able to do so. Electricity should not be used to provide an auxiliary heat source for an absorption unit. As the coefficient of performance of the absorption unit is approximately one-fourth to one-third that of a vapor-compression air conditioner, use of electricity to provide even a



Service Hot Water Preheat System Control Logic

portion of the heat energy used by the absorption system is not economical. The absorption chiller must have an inexpensive heat source, such as one hundred percent solar or solar with a fossil fuel auxiliary, to compare economically with a vapor-compression air conditioner. The selection of solar heated or auxiliary heated hot water is made by a three-way valve which is controlled by the binary logic represented in Figure 2.1.4.

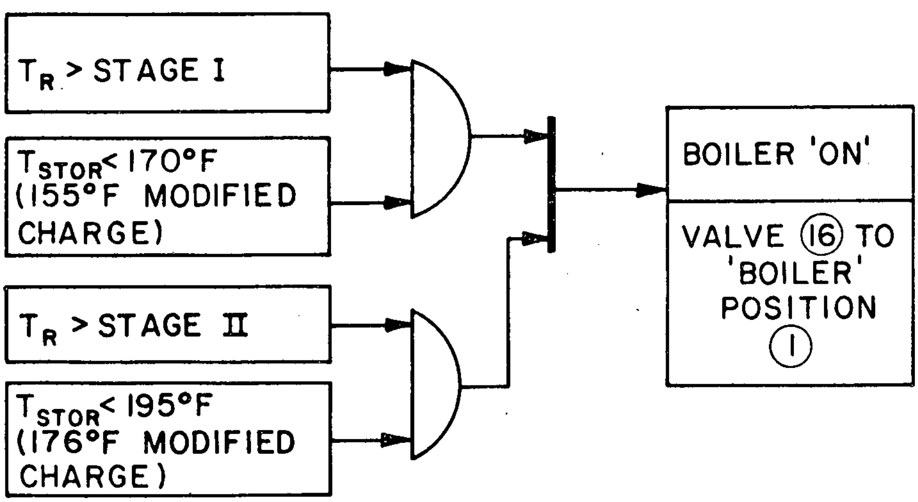


Figure 2.1.4. Auxiliary Boiler Control Logic

The Arkla lithium bromide absorption machine is termed a chiller rather than air conditioner as it directly chills only water, not air. This chilled water may then be stored for later use or passed directly through a duct coil to cool the house. Condenser water is supplied to the Arkla chiller to remove generator and house heat from the unit and reject it outside the house through a cooling tower. Wet cooling towers are required to provide cool enough condenser temperatures for proper chiller operation with solar heated generator water. In arid regions where wet bulb temperatures are much cooler than their corresponding dry bulb temperatures, the charge of lithium bromide in the refrigerant solution can be reduced from 52 percent to approximately 50 percent. The modified charge allows use of cooler condenser water and also reduces the minimum allowable hot water generator temperature from 77°C (170°F) to 68°C (155°F).

A diagram showing internal operation of the Arkla Solaire chiller is included in Appendix D. The control strategy of the Arkla chiller is represented in binary logic in Figure 2.1.5.

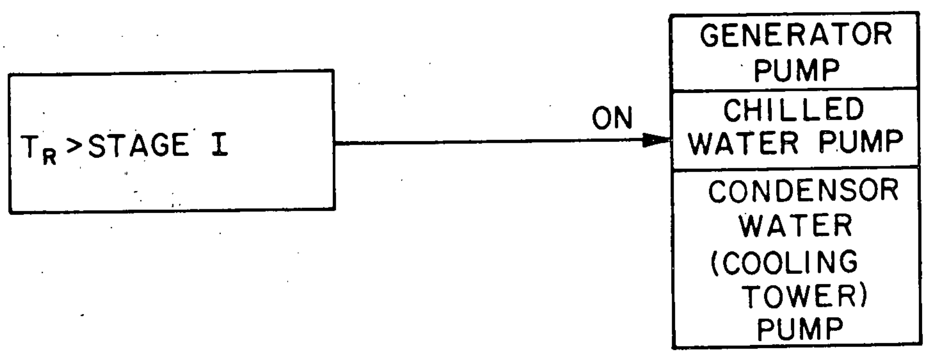


Figure 2.1.5. Chiller Control Logic

Chilled water storage was provided in the solar air conditioning subsystem of CSU Solar House I to decrease the chiller "on/off" cycling and to reduce thermal losses from the solar storage tank. Research done with a prototype chiller [7], one considerably different than the Arkla Solaire WF-36 model, indicated a substantial reduction in the seasonal coefficient of performance of the absorption unit due to numerous start-up and transient performance losses.

The simplest design of a chilled water storage system would utilize a single storage tank, but such a design is not workable with an absorption chiller. Chilled water delivered by the Arkla unit would be re-mixed in the chilled water storage tank with the "warm" water being supplied to the chiller. Unfortunately, the absorption chiller process is very sensitive to chilled water temperature and should the storage tank temperature fall below 13°C (55°F), the chiller coefficient of performance (COP) would fall drastically. Since chilled water is to be cooled and stored at 7°C (45°F), an impossible paradox is created by the single tank design. A two tank storage system is employed instead, with the volume of water used in the storage system equal to the volume of only one of the tanks. Water is transferred from one tank to another for purposes of being chilled or for cooling the house. Ideally, any water in the "chilled water" tank will be at 7°C (45°F) and any water in the "warm" storage tank will be at 13°C (55°F). To help retain these conditions each of the storage tanks is insulated to R = 17.14 $°Cm^2/w$ (R = 30 $Hr\text{-}°F\text{-}Ft^2/Btu$).

Heat gain in the storage units will not result in a loss of cooling capacity if the storage is placed in the conditioned space of the house. Such heat gains will simply be uncontrolled cooling of the house. Such an arrangement is assumed in the simulations presented here.

The chilled water storage system has three modes of operation:

I. Direct cooling of the house by the Arkla chiller. (No chilled water storage is involved.)

II. "Warm" storage water being chilled by the Arkla unit and transferred to "chilled water" storage. (No house cooling is provided.)

III. Chilled water from storage used directly to cool the house. (No chiller operation.)

These modes of operation are shown schematically in Figures 2.1.6, 2.1.7, and 2.1.8, respectively.

The "chilled water" storage system can be operated by either of two control strategies. Both strategies transfer fluid from "warm" to "chilled water" storage when solar heat is available for chiller operation and the house requires no cooling. They have different priorities, however, when the house needs to be cooled. The first strategy uses solar heated water to operate the chiller when it is available regardless of whether chilled water is available in storage. This strategy keeps heat losses from solar storage to the conditioned space to a minimum because heat is removed from solar storage at every opportunity. The second strategy uses chilled water from storage to cool the house when it is available, regardless of whether solar heated hot water is available. This strategy provides maximum use of the chilled water storage system. Neither strategy uses the auxiliary boiler to place chilled water in storage.

Control strategies I and II are presented in program logic in Figures 2.1.9 and 2.1.10, respectively.

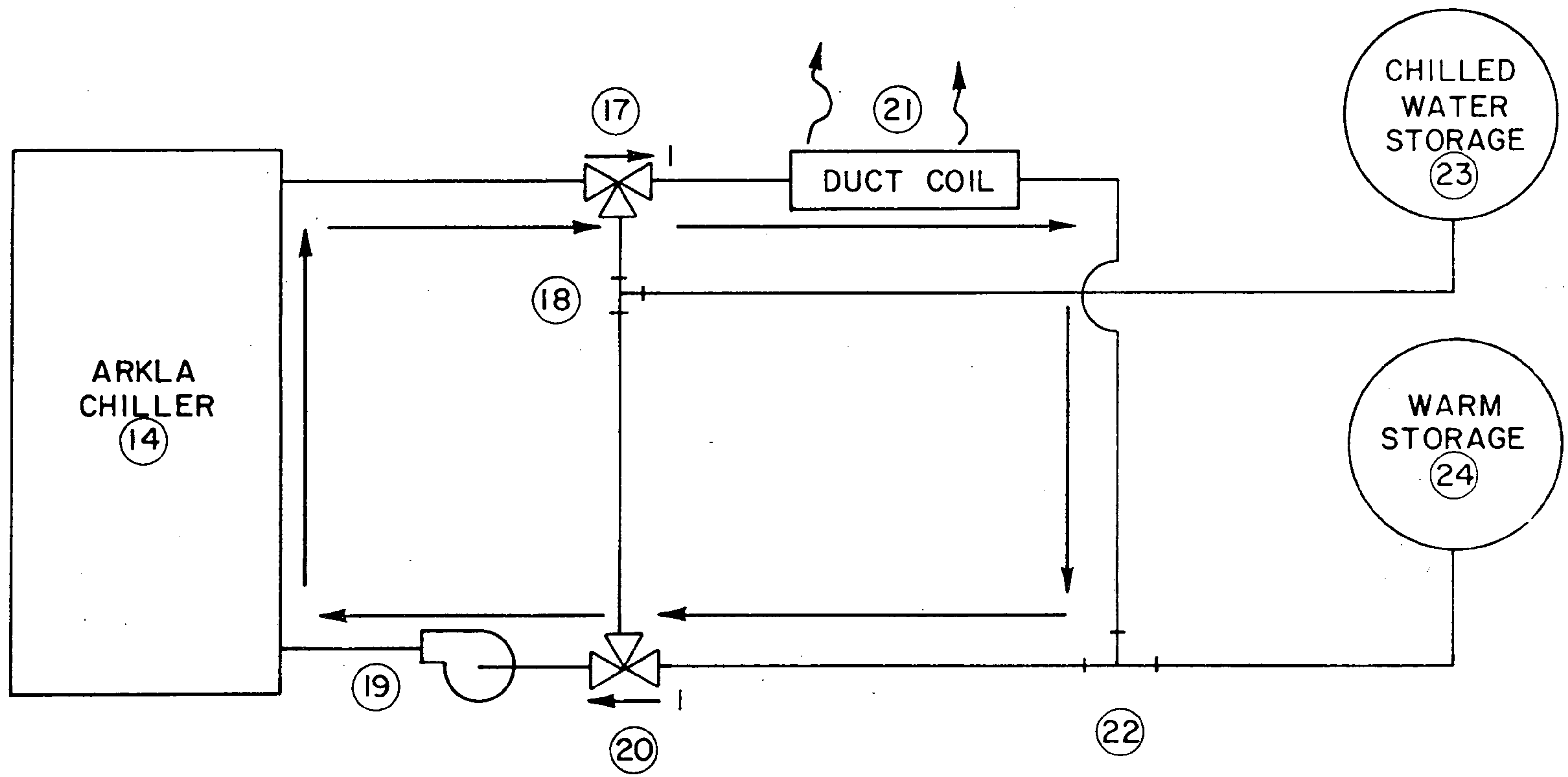
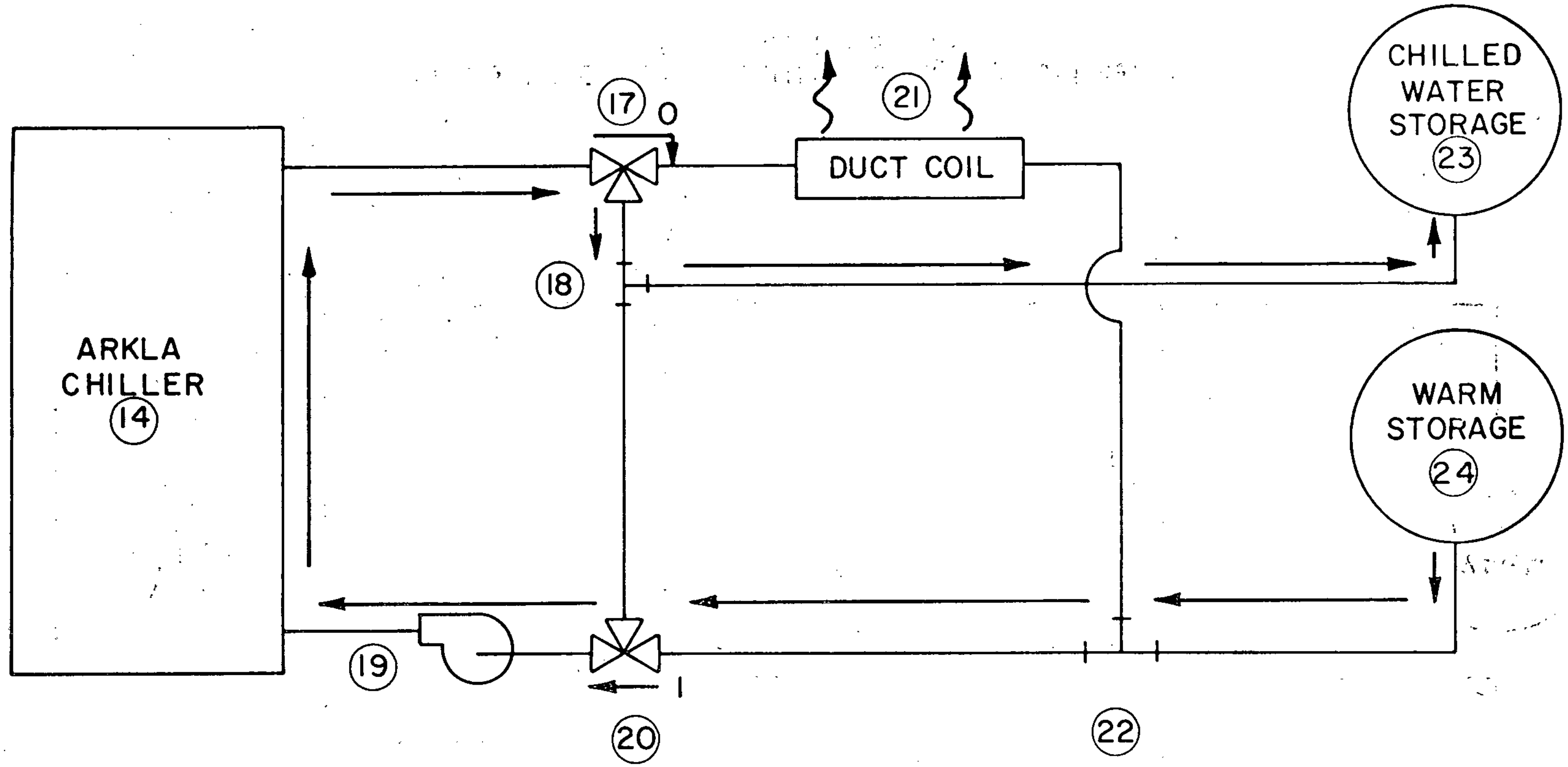
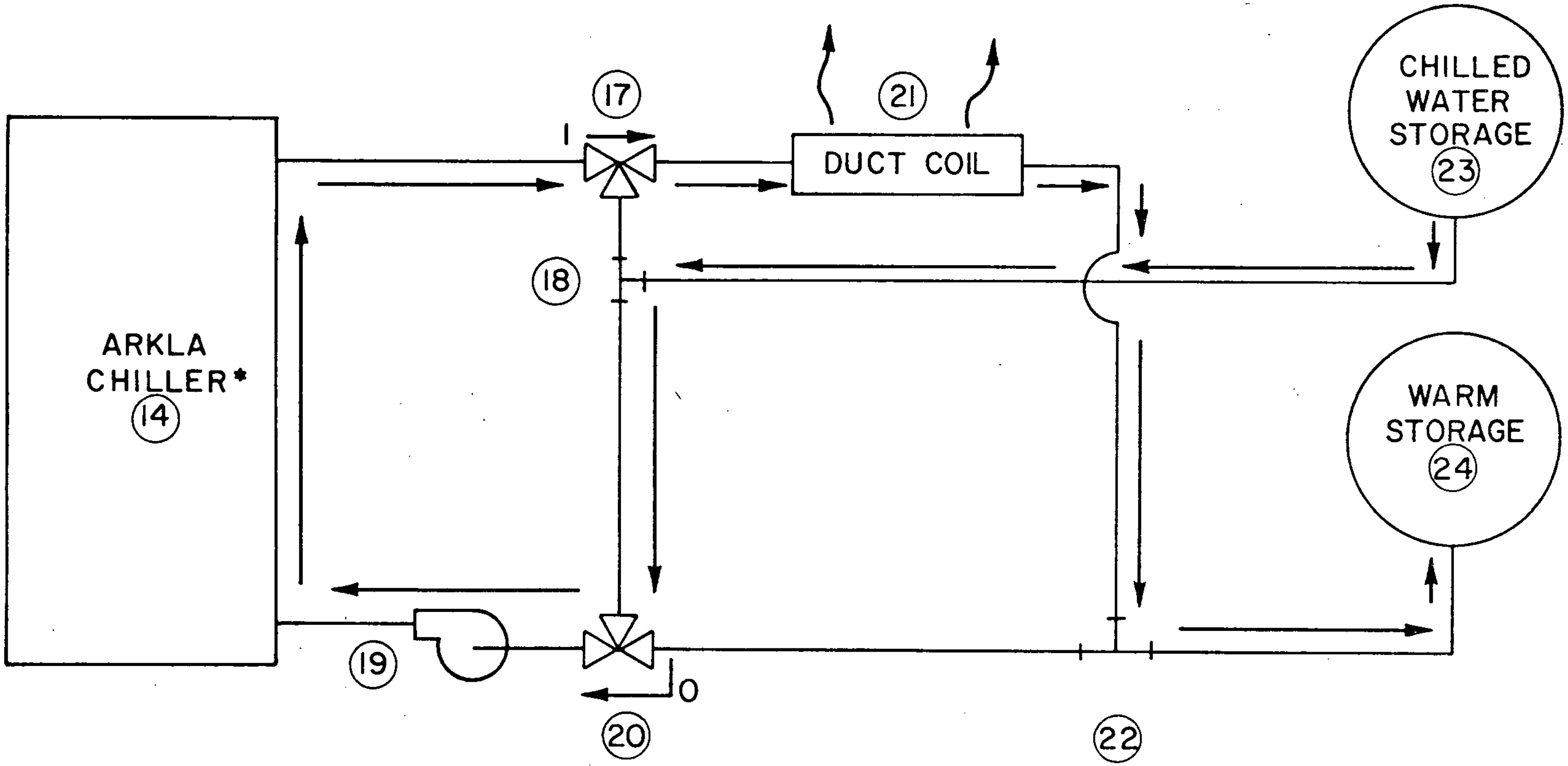


Figure 2.1.6. Chilled WAter Storage Operational Modes,
Direct Cooling by Chiller



igure 2.1.7. Chilled Water Storage Operational Modes, Cooling to Chilled Water Storage



*ARKLA CHILLER IS "OFF"

Figure 2.1.8. Chilled Water Storage Operational Modes, Cooling from Chilled Water Storage

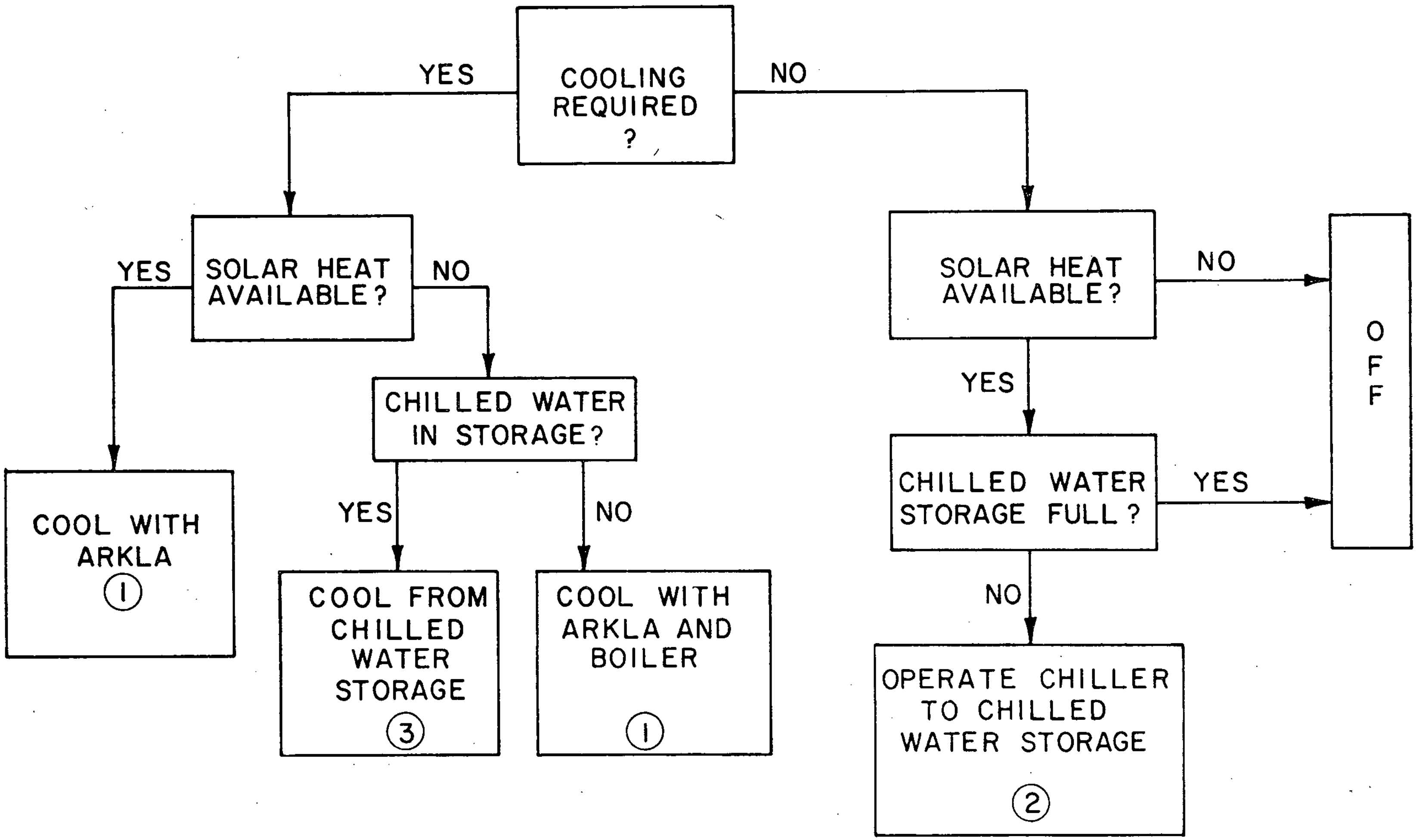


Figure 2.1.9. Chilled Water Storage Control Strategy I

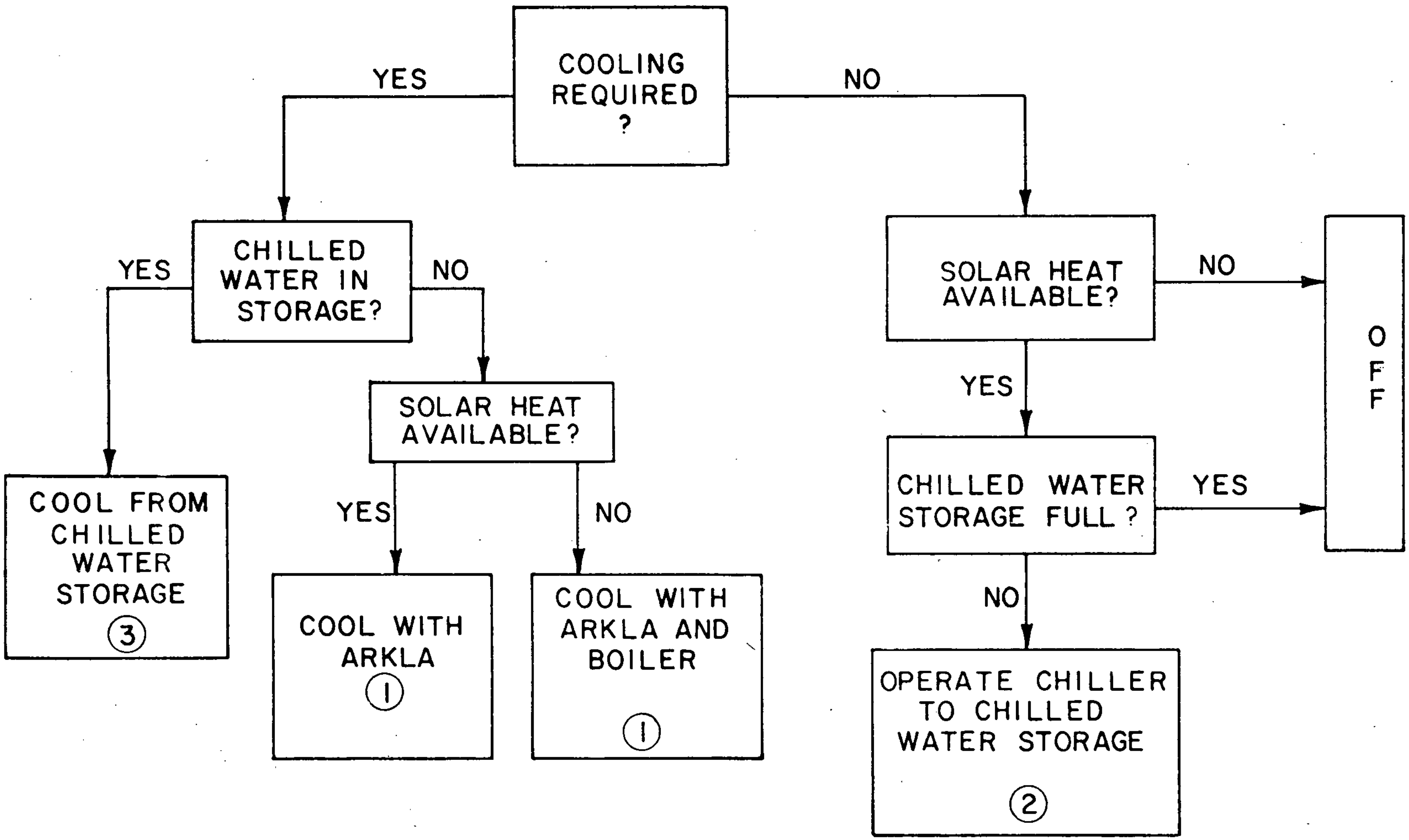


Figure. 2.1.10. Chilled Water Storage Control Strategy II

2.2 CORNING EVACUATED TUBE COLLECTOR

The Corning evacuated tube collectors at CSU Solar House I are shown in Figure 2.2.1. Each pyrex glass tube is 10.2 centimeters (4 in) in diameter, approximately 2.25 meters (7.5 ft) in length, and contains .186 square meters (2 ft^2) of absorber plate. The ratio of roof area covered by the collectors (less manifolds) to the absorber area is 1.46. The smallest collection unit which can be installed consists of a module of six collector tubes connected in series. All modules are then connected in parallel flow. The flow rate through each module is approximately 100 liters/hour (.45 gpm).

Experimental measurements on a single evacuated tube collector have been performed by Corning in a solar simulator with normal incidence of the beam radiation. Results of these tests are plotted in Figure 2.2.2. To simplify simulation calculations, information in Figure 2.2.2 was normalized to the parameter $(T_{in} - T_{amb})/I_T$, as shown in Figure 2.2.3. A least squares linear fit was found. The resulting model for performance of a single collector tube at normal solar incidence of beam radiation is:

$$\text{Efficiency} = .788 - .517 \left(\frac{T_{in} - T_{amb}}{I_T}\right) \quad [°Cm^2/w] \qquad (1)$$

$$\text{Efficiency} = .788 - .295 \left(\frac{T_{in} - T_{amb}}{I_T}\right) \quad [Hr\text{-}°F\text{-}Ft^2/Btu]$$

Research by Frick [4] indicated that evacuated tube collector performance is affected by non-normal sun angles and by interactive effects with adjacent tubes, none of which are accounted for in the above model. These effects include changes in the transmissivity of the glass tube, internal reflections in the tube, reflections from

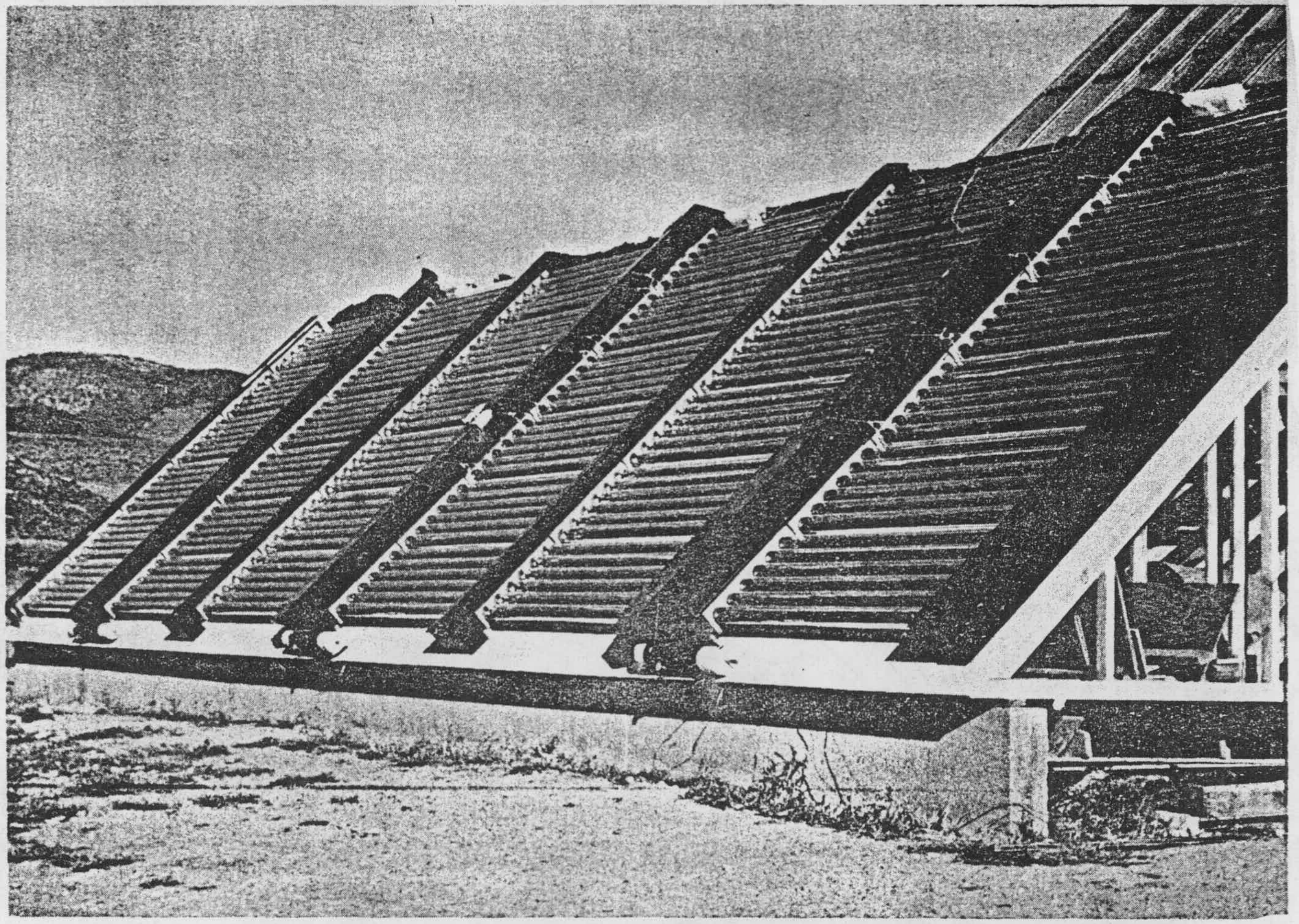


Figure 2.2.1. Corning Evacuated Tube Collectors

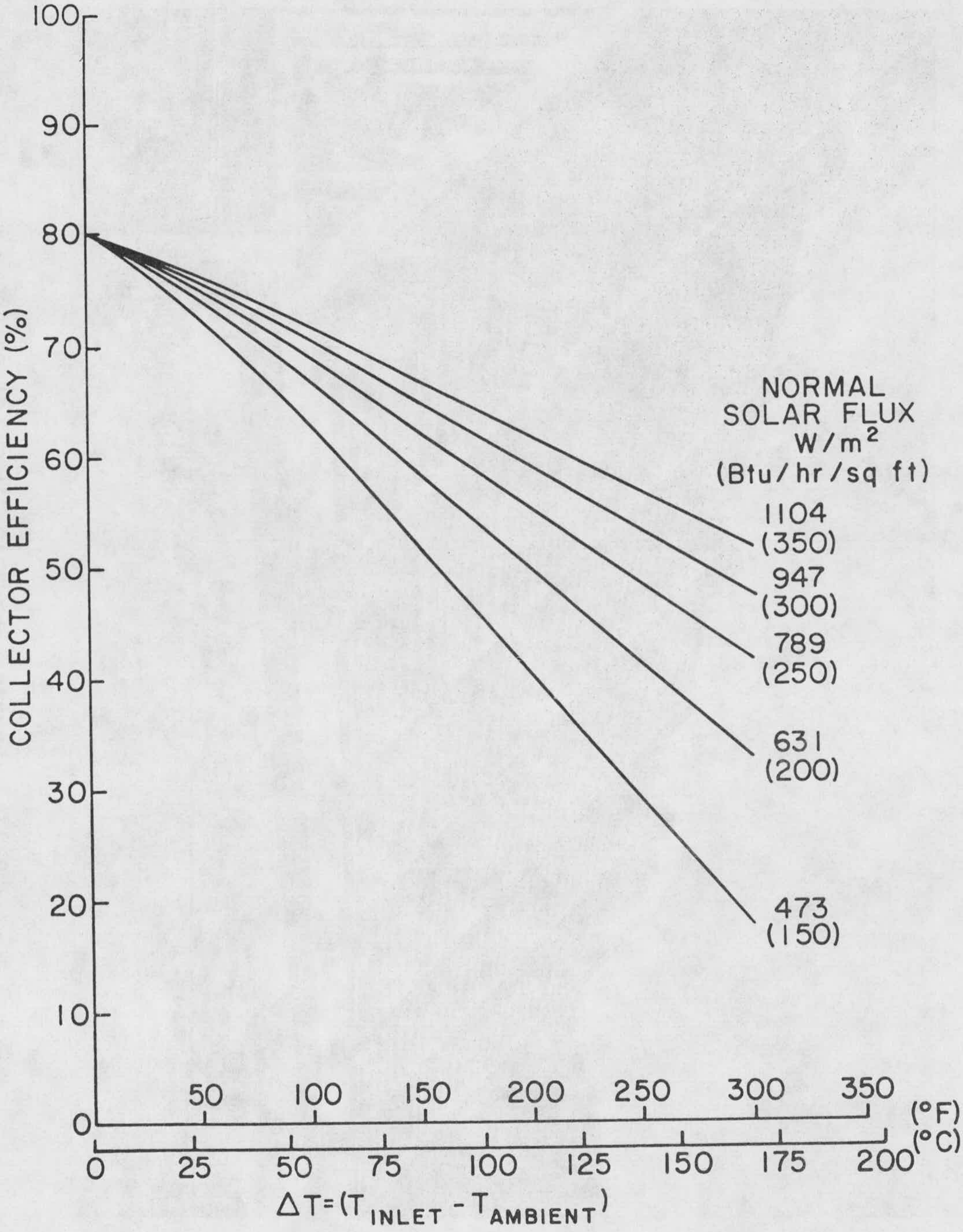


Figure 2.2.2. Collector Simulator Test Performance

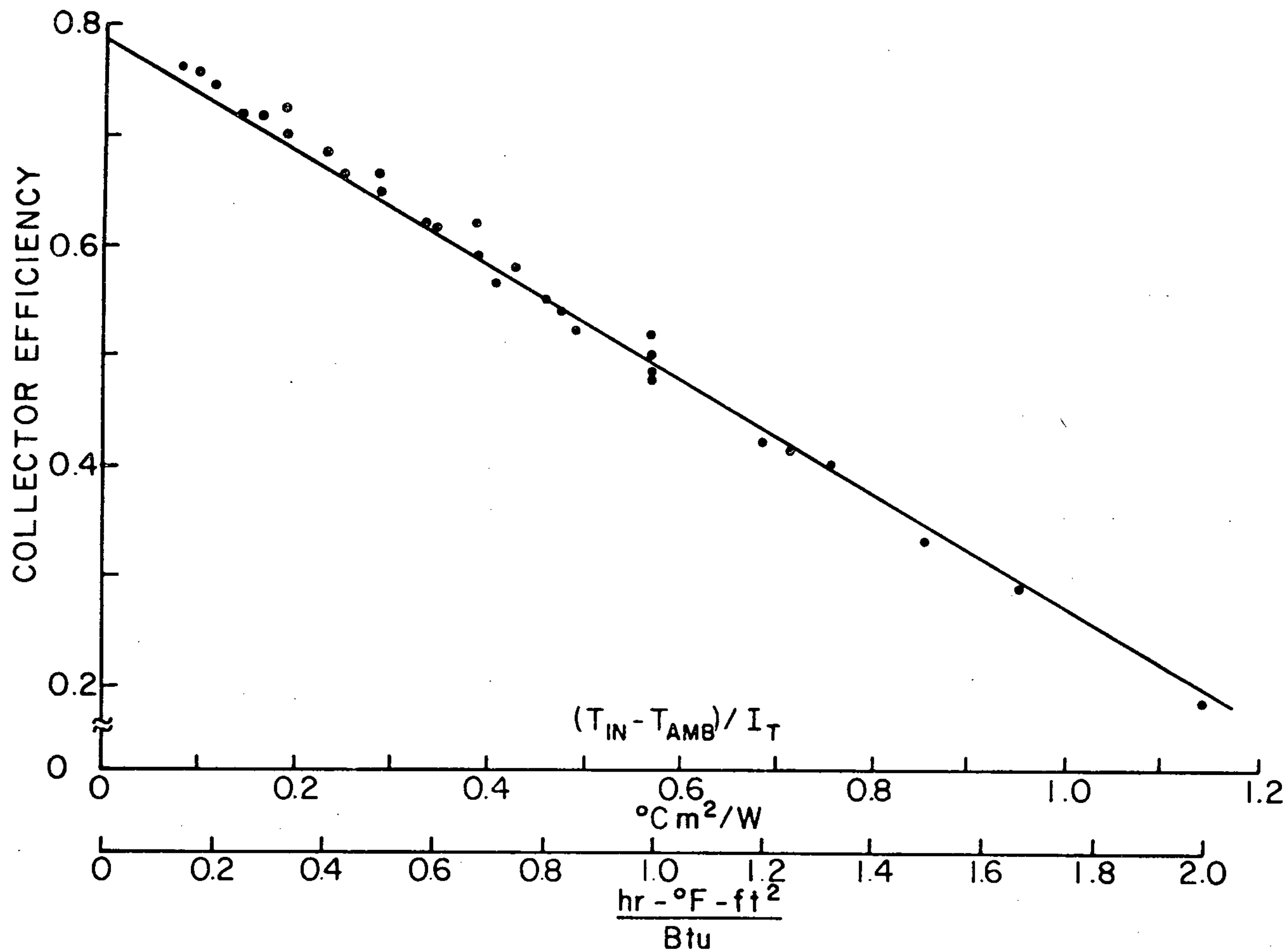


Figure 2.2.3. Normalized Collector Simulator Test Performance

adjacent tubes, shading by adjacent tubes, and reflections from the surface behind the collector tubes. All of these phenomena cause the effective $(\tau\alpha)$ product of the collector to vary with sun angle. Frick has calculated the combined results of all these effects for beam radiation on the $(\tau\alpha)$ product as a function of the angles ψ and φ, which are defined in Figure 2.2.4. The results of this work can be used to complete the performance model of the evacuated tube collector for any sun angle.

The efficiency of a single collector tube at normal solar incidence is:

$$EFF_n = f\left(\frac{T_{in} - T_{amb}}{I_T}\right) \qquad (2)$$

where f is defined as the function defined by Equation (1). If $(I_{f,n})$ is the fraction of the total radiation (I_T) that is transmitted through the glass tube at normal incidence and is absorbed by the absorber plate, then:

$$I_{f,n} = (I_T) \times (\tau\alpha)_n \qquad (3)$$

where $(\tau\alpha)_n$ is the effective $(\tau\alpha)$ product at normal incidence. Therefor the efficiency of a single collector tube operating at normal solar incidence may be rewritten as:

$$EFF_n = f\left[\frac{(T_{in} - T_{amb})(\tau\alpha)_n}{I_{f,n}}\right] \qquad (4)$$

Let FACTOR be defined as:

$$FACTOR = \frac{(\tau\alpha)_e}{(\tau\alpha)_n} \qquad (5)$$

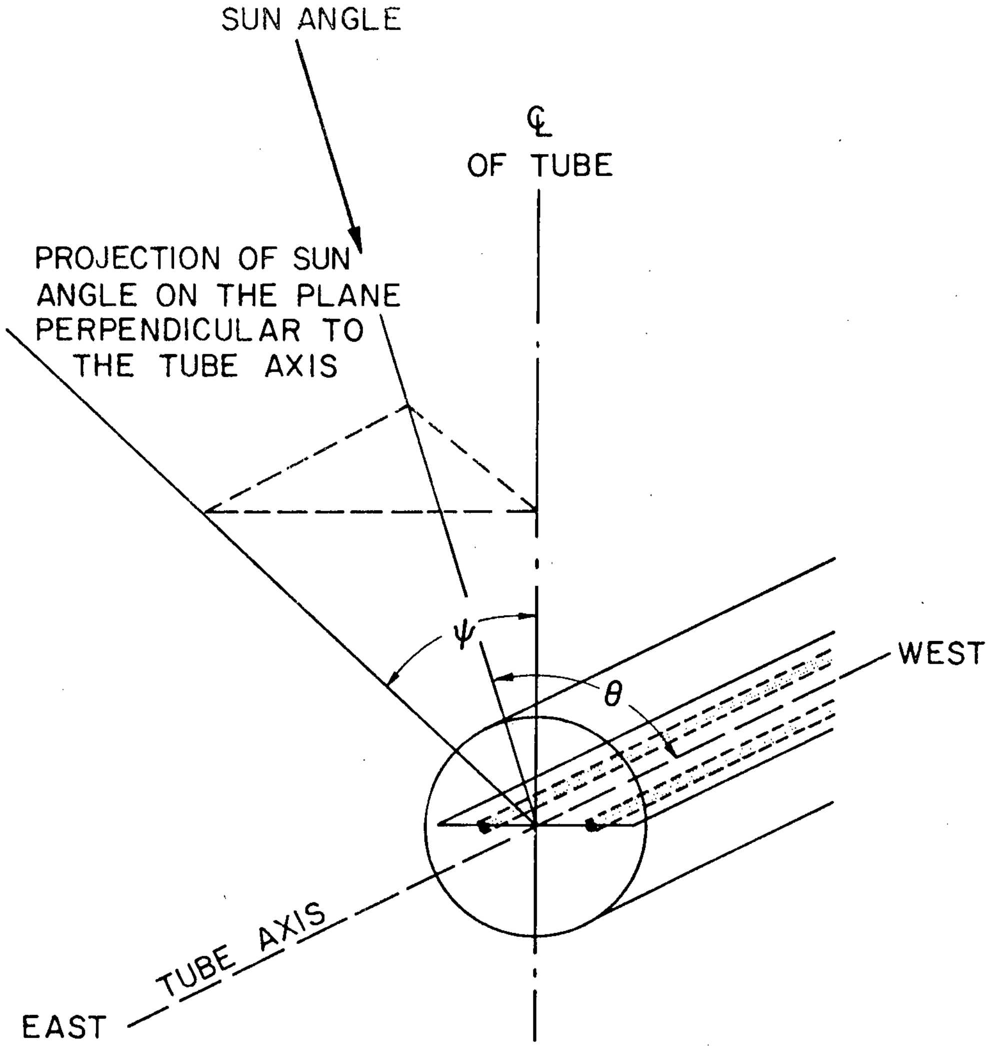


Figure 2.2.4. Collector Sun Angles

where $(\tau\alpha)_e$ is the effective $(\tau\alpha)$ product for any number of collection tubes at any sun angle. Then the fraction of the total radiation absorbed by the collector absorber plate at any sun angle is:

$$I_{f;\psi,\theta} = (I_T) \times (\tau\alpha)_e \tag{6}$$

The efficiency of the collector at any sun angle may be found by rewriting Equation (4) as:

$$EFF_{\psi,\theta} = f\left[\frac{(T_{in} - T_{amb})(\tau\alpha)_n}{I_{f;\psi,\theta}}\right] \tag{7}$$

Combining Equations (5), (6) and (7) yields:

$$EFF_{\psi,\theta} = f\left[\frac{(T_{in} - T_{amb})}{(I_T)(FACTOR)}\right] \tag{8}$$

Equation (8) completes the collector performance model for any angle (ψ,θ) of beam radiation. However, that portion of the solar insolation which is diffuse will not strike the collector at the same angle as the beam component. Frick has calculated the effective $(\tau\alpha)$ product for diffuse radiation as 0.774. Assuming that the radiation in the collector simulator experiments for a single collector tube was nearly all beam radiation, diffuse radiation may be added to the collector model with little error as:

$$EFF_{TOTAL} = f\left[\frac{(T_{in} - T_{amb})}{(I_B)(FACTOR) + (I_D)(.774)}\right] \tag{9}$$

where

$$I_B + I_D = I_T \tag{10}$$

Equation (10) is in the form of the linear regression model obtained from the solar simulator tests but includes the term FACTOR from Equation (5). FACTOR is calculated from the effective $(\tau\alpha)$ products in Frick [4], the results of which are shown in Table 2.2.1. The completed collector model is then:

$$EFF_{TOTAL} = .788 - .517 \left[\frac{(T_{in} - T_{amb})}{(I_B)(FACTOR) + (I_D)(.774)}\right]\left[\frac{°Cm^2}{w}\right] \quad (11)$$

$$EFF_{TOTAL} = .788 - .295 \left[\frac{(T_{in} - T_{amb})}{(I_B)(FACTOR) + (I_D)(.774)}\right]\left[\frac{Hr\text{-}°F\text{-}Ft^2}{Btu}\right]$$

where efficiency is based on absorber area. Collector efficiency may also be rewritten as:

$$EFF_{TOTAL} = Qu/(I_T \times A_a) \quad (12)$$

where A_a is the collector absorber area.

The sun angles, ψ and θ, needed to calculate the appropriate FACTOR for collector performance may be found from Threlkeld [10].

For east-west orientation of the Corning collector tubes:

$$\tan\psi = \frac{\tan(90 - \beta)}{\cos\alpha} \quad (13)$$

where

$$\cos\beta = \cos\delta \ \cos\phi' \ \cos\omega + \sin\phi' \ \sin\delta \quad (14)$$

and

$$\tan\alpha = \frac{-\cos\delta \ \sin\omega}{\cos\beta} \quad (15)$$

Table 2.2.1. Corning Evacuated Tube Collector
Sun Angle FACTOR

± ψ \ ± θ	20°	30°	40°	50°	60°	70°	80°	90°
0°	.741	.890	.956	.981	.998	.995	.995	1.00*
10°	.755	.901	.959	.998	1.00	1.00	1.01	1.01
20°	.760	.907	.967	1.00	1.01	1.01	1.01	1.02
30°	.793	.931	.989	1.03	1.04	1.03	1.03	1.05
40°	.807	.951	1.01	1.04	1.06	1.05	1.05	1.05
50°	.726	.788	.849	.886	.894	.901	.903	.903
60°	.525	.641	.726	.782	.797	.817	.880	.880
70°	.212	.390	.512	.647	.639	.666	.676	.676

* Normal Incidence

and

$$\cos\theta = \sin\psi \ \sin\alpha \tag{16}$$

2.3 ARKLA SOLAIRE WF-36 ABSORPTION CHILLER

The Arkla Solaire Model WF-36 Chiller used in CSU Solar House I is shown in Figure 2.3.1. The steady-state performance, i.e., capacity and coefficient of performance, is a function of the generator and condenser water temperatures. Characteristic performance of the standard Arkla chiller and the modified charge system for arid climates is shown in Figures 2.3.2 and 2.3.3, respectively. Data for these characteristic performance curves was generated at Arkla Industries and Colorado State University by operation of the chillers at steady-state conditions (Appendix A).

Condenser water for the chillers is provided by a cooling tower modeled as a 4.4°C (8°F) approach to the ambient wet bulb temperature or a minimum set temperature, whichever is greatest. The minimum set temperatures are 23.9°C (75°F) for the standard chiller and 21.7°C (71°F) for the modified charge chiller. As most weather tapes record dew point rather than wet bulb temperature, a routine to calculate the wet bulb temperature from the dew point and ambient temperature is used in the simulations (Appendix B).

Operation of an absorption chiller requires many heat exchange processes at various temperature levels within the chiller. When a chiller first becomes operational, the fluids and heat exchange surfaces approach their normal operating temperatures but, until steady-state conditions are achieved, cooling capacity is reduced. When a chiller is turned off, the fluids and material temperatures decay approximately

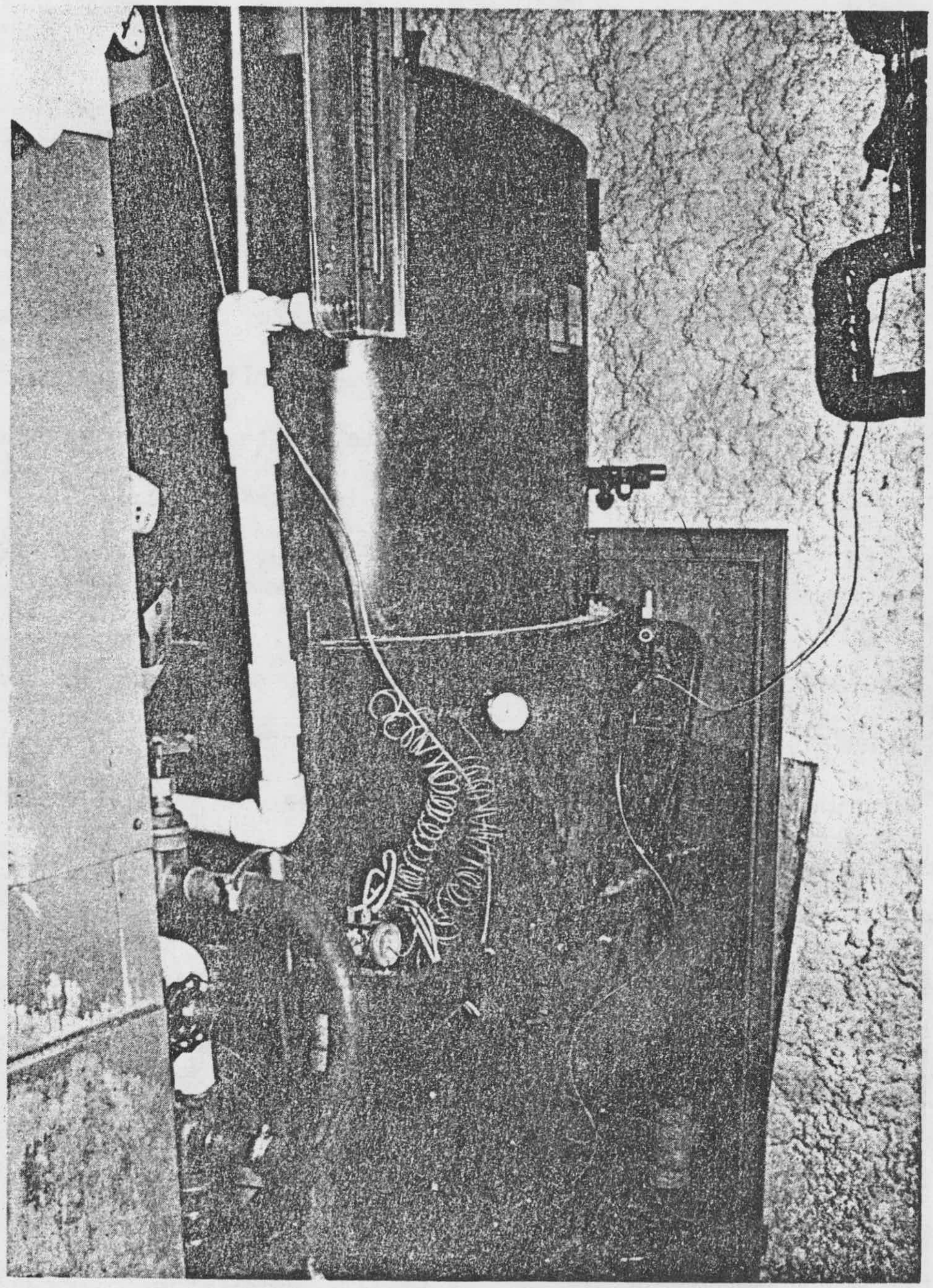


Figure 2.3.1. Arkla Solaire WF-36 Absorption Chiller

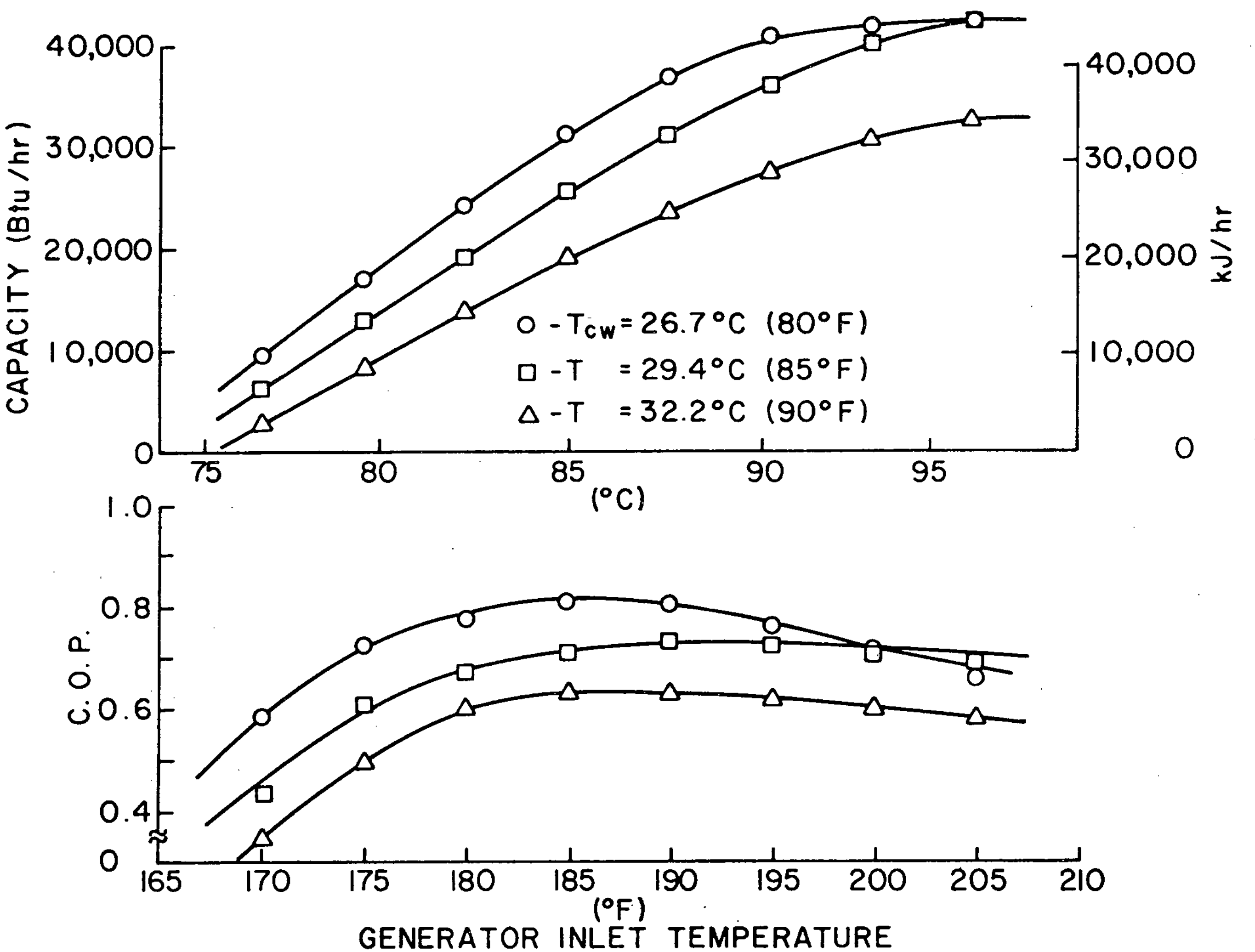


Figure 2.3.2. Standard Arkla Chiller Performance Characteristics

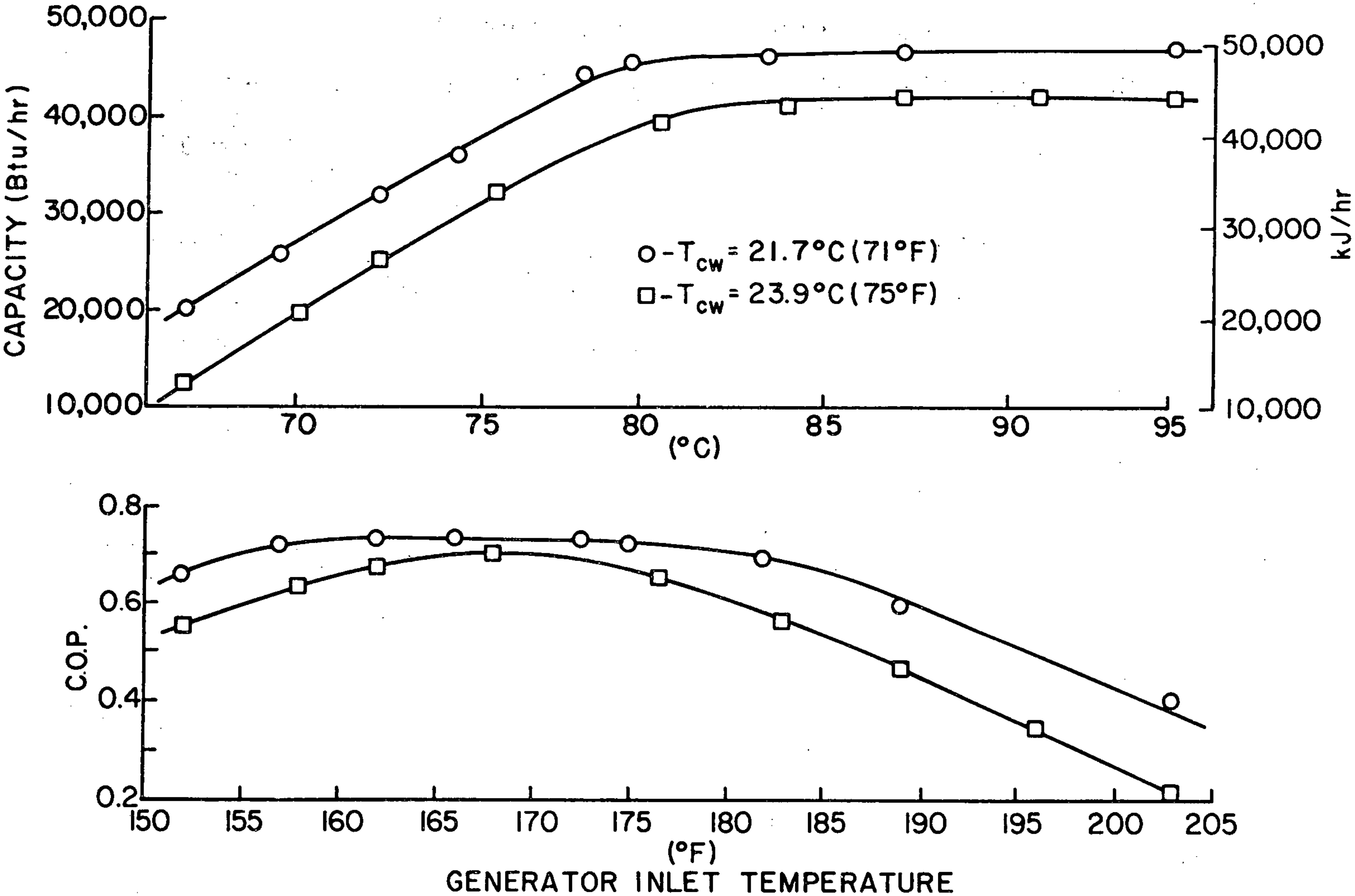


Figure 2.3.3. Modified Charge Arkla Chiller Performance Characteristics

exponentially to the ambient temperature of the chiller unit. Therefore, the performance of an absorption chiller in a transient state depends on how long the unit has been off as well as how long it has been on. Transient response experiments with the modified charge Arkla unit were performed at CSU with "off" times between chiller operation of one-half, two, and eight hours (Appendix A). Transient response for both Arkla units is assumed to be identical.

Analysis of the experimental data did not yield a usable model which predicts transitory behavior of the chiller as a function of both the "on" and "off" times. The data does indicate that such a model would probably be somewhat complex, requiring additional computer time and money for performance simulations. However, the simulations are conducted with fifteen minute time steps and such a detailed transient model is therefore not warranted. Observation of the experimental data reveal that transient characteristics do not exist for more than thirty minutes after start-up and that during this transient period, at least 50 percent of the steady-state cooling capacity of the chiller is delivered to the load. Accordingly, all of the transitory cooling capacity losses of the chiller may be attributed to the system during the first time step of simulation in which the chiller is turned on and steady-state response assumed each time step thereafter. Because the temperature of the environment, house, and solar system usually changes little in one time step, the simplification causes negligible error in tabulating system performance. The transient response model of chiller capacity, then, is a function only of chiller "off" time and is computed as a fraction of the steady-state performance. The transient response factors to be multiplied by the steady-state chiller capacity during the first simulation time step after the chiller is turned on are shown in Figure 2.3.4. Exponential response, as predicted by the temperature decay theory, is assumed.

2.4 CHILLED WATER STORAGE

The operation of chilled water storage and its control logic have been previously described. Storage tank volume is modeled in steps equal to the volume of chilled water moved by the chilled water pump in fifteen minutes (one time step of the simulation). The chilled water flow rate is 27.25 liters per minute (7.2 gpm) and the storage tank incremental size is therefore 409 liters (108 gallons). Initial simulations with chilled water storage assume tank sizes of 1227 liters (324 gallons).

Water may be either added or removed from the "chilled" or "warm" storage tanks but not both in one simulation time step. When water is removed from a storage tank, the temperature of the tank remains unchanged. When water is added to a storage tank, the resulting tank temperature is calculated as a result of the mixing process:

$$T_2 = \frac{(T_1)(M_1) + (T_+)(M_+)}{M_1 + M_+} \tag{17}$$

where T_2 is the final tank temperature, T_1 and M_1 the initial tank temperature and mass, respectively, and T_+ and M_+ the temperature and mass, respectively of the fluid added to the tank during the simulation time step.

2.5 BUILDING COOLING LOAD

The instantaneous cooling load on CSU Solar House I is computed for simulation purposes as:

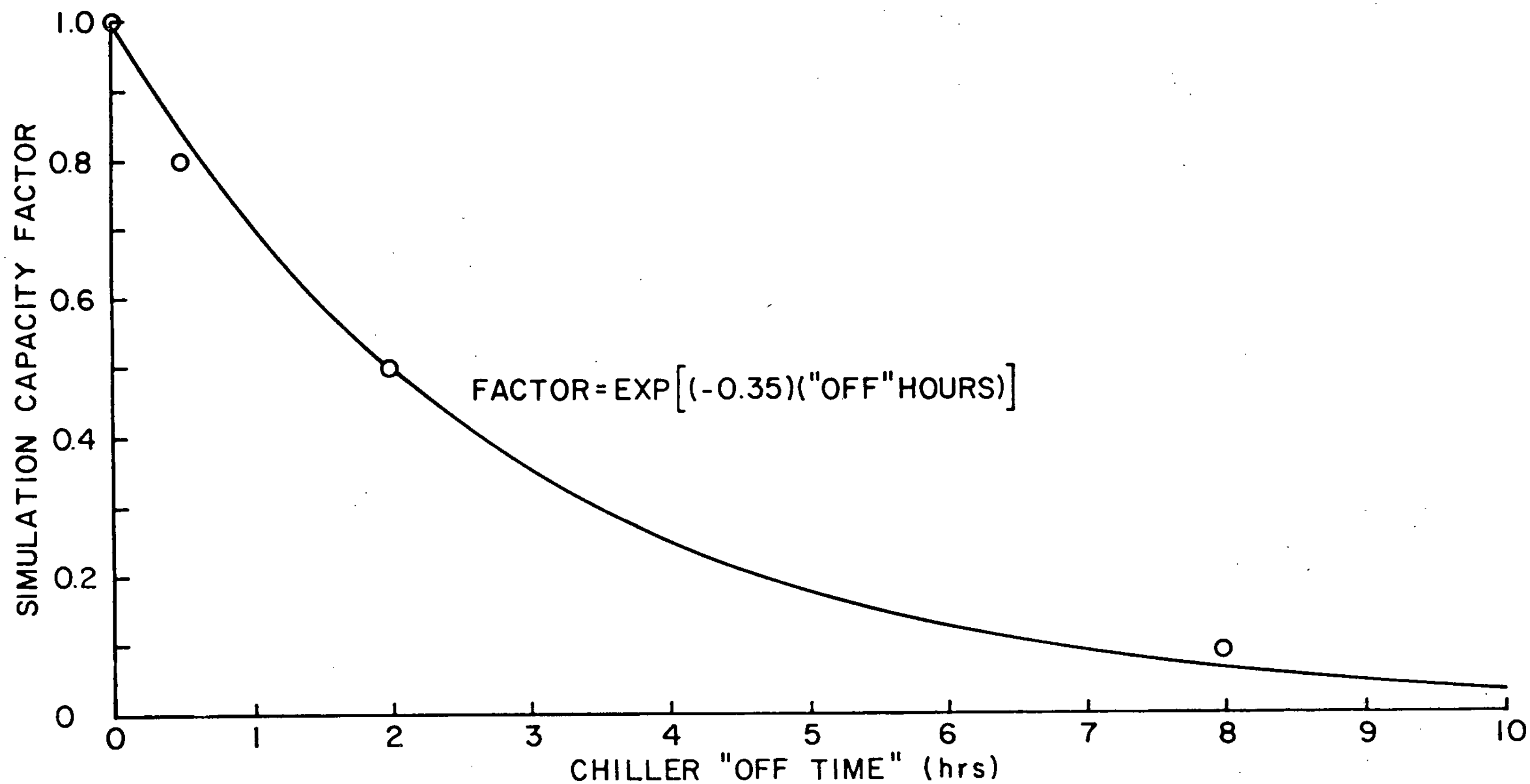


Figure 2.3.4. Chilled Transient Response Characteristics

$$\dot{Q}_{ENV} + \dot{Q}_{GEN} + C(I_H) + UA(T_{AMB} - T_R) + \dot{Q}_{LATENT} \qquad (18)$$

where $\dot{Q}_{ENV}$ is the heat loss rate from the solar storage tank to the conditioned space, $\dot{Q}_{GEN}$ is the rate of internal heat generation by people and energy consuming devices, $C(I_H)$ accounts for the cooling load created by solar radiation on the house, $UA(T_{AMB} - T_R)$ is the rate of heat gain through the house structure, and Q_{LATENT} is the latent heat load rate of the conditioned space. The function used for $\dot{Q}_{GEN}$ in the simulations is shown in Figure 2.5.1.

Sunlight on a building increases the cooling load in two ways. The first is the "greenhouse effect" of windows. Sunlight passes through windows and is absorbed by objects in the building which converts the visible light to heat. This effect does not add to the cooling load immediately, however, because the solar radiation must first raise the temperature of the masses in the house before they can radiate and convect heat to the house interior. The duration of the delay in the increased cooling load depends on the type of construction and the thermal mass of the house. The thermal mass of CSU Solar House I is calculated for air conditioning design purposes as 6331 w-hr/°C (12,000 Btu/°F). The second way solar insolation increases the cooling load is by increasing the temperature of the outside walls of the house, thus increasing the equivalent temperature difference between the ambient and the building interior conditions.

The total effect of solar insolation on a building cooling load can be calculated for a specific instance in time by techniques presented in The Carrier Cooling Load Analysis [8]. However, the calculations are quite extensive and to repeat them each time step in computer simulations would be quite expensive. Accordingly, a single design

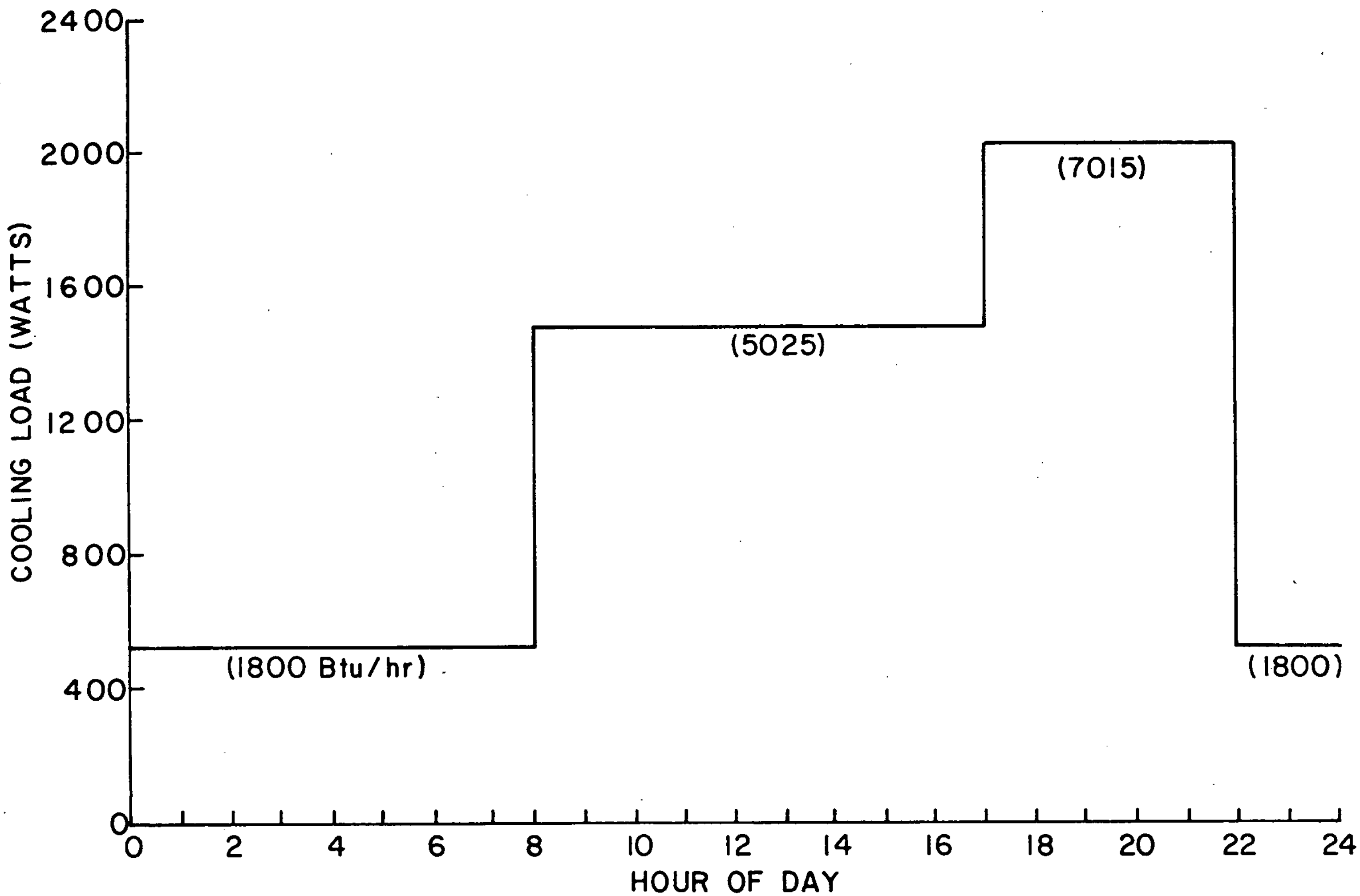


Figure 2.5.1. Building Cooling Load: Internal Heat Generation

condition is analyzed for CSU Solar House I. On August 1, at 2:00 P.M. with a solar insolation of 834 W/m^2 (264 $Btu/Hr\text{-}Ft^2$), the instantaneous cooling load directly attributable to the solar insolation on the building in Fort Collins, Colorado is 3,600 watts (12,270 Btu/Hr). A proportioning of the instantaneous load with solar insolation can then be used to estimate the instantaneous solar gain at any other time:

$$\underset{\text{Solar Gain}}{\text{Instantaneous}} = (\text{Proportional Factor}) \times \underset{\text{for horizontal}}{(\text{Solar Insolation})}$$

$$3600\ W = (C) \times (834\ W/m^2)$$

$$C = 4.32$$

Therefore the instantaneous cooling load caused by solar insolation on the house is approximated as:

$$\underset{\text{Solar Gain}}{\text{Instantaneous}} = (4.32) \times \underset{\text{on horizontal}}{(\text{Solar Insolation})}$$

The "UA" portion of the cooling load consists of heat gain conducted through walls and windows and by infiltrating outside air. The "UA" of CSU Solar House I has been calculated by conventional means as 554 W/°C (1050 Btu/Hr-°F), including infiltration.

The final contribution to the cooling load is the latent heat of condensation of water vapor from the conditioned space. Latent load has been simulated according to ASHRAE recommendations of 30 percent of the total load in Washington, D.C. and 10 percent of the total load in Fort Collins, Colorado.

2.6 TEMPERING VALVE

The reduction in the lithium bromide charge in the Arkla chiller and the cooler condenser water available in arid climates allow operation of the chiller with reduced generator water temperatures. However, the solar energy system operating the chiller may still reach boiling temperatures and "overfire" the chiller generator. The coefficient of performance characteristics of the modified charge Arkla chiller shown in Figure 2.3.3 shows a decrease in coefficient of performance when generator water reaches temperatures over 76.7°C (170°F). Operation above this temperature causes stored solar energy to be wasted with a corresponding decrease in the seasonal solar contribution to the cooling system. Use of a tempering valve, as shown in Figure 2.1.1 would mix water returning from the generator with high temperature solar heated water to supply the generator with water not exceeding 82°C (180°F). This temperature was chosen because it is the minimum chiller generator temperature which will provide the maximum cooling capacity of the machine. However, should the nominal cooling capacity of the chiller be much greater than the design load, i.e., a three ton chiller installed for a two ton load, a 76.9°C (170°F) tempering valve and auxiliary boiler set temperature might be chosen. Under these circumstances, full capacity is not needed and the lower generator temperature will increase the average chiller coefficient of performance.

The operational characteristics of the modified charge Arkla chiller with a tempering valve set at 82°C (180°F) appear as in Figure 2.6.1. This performance model is used for simulating performance of the chiller when such a valve is used with the unit.

All other solar system components indicated in Figure 2.1.1 are modeled by TRNSYS component models [9] consistent with the operational mode and control logic previously described. Other important design parameters of the system are listed in Table 2.7.1.

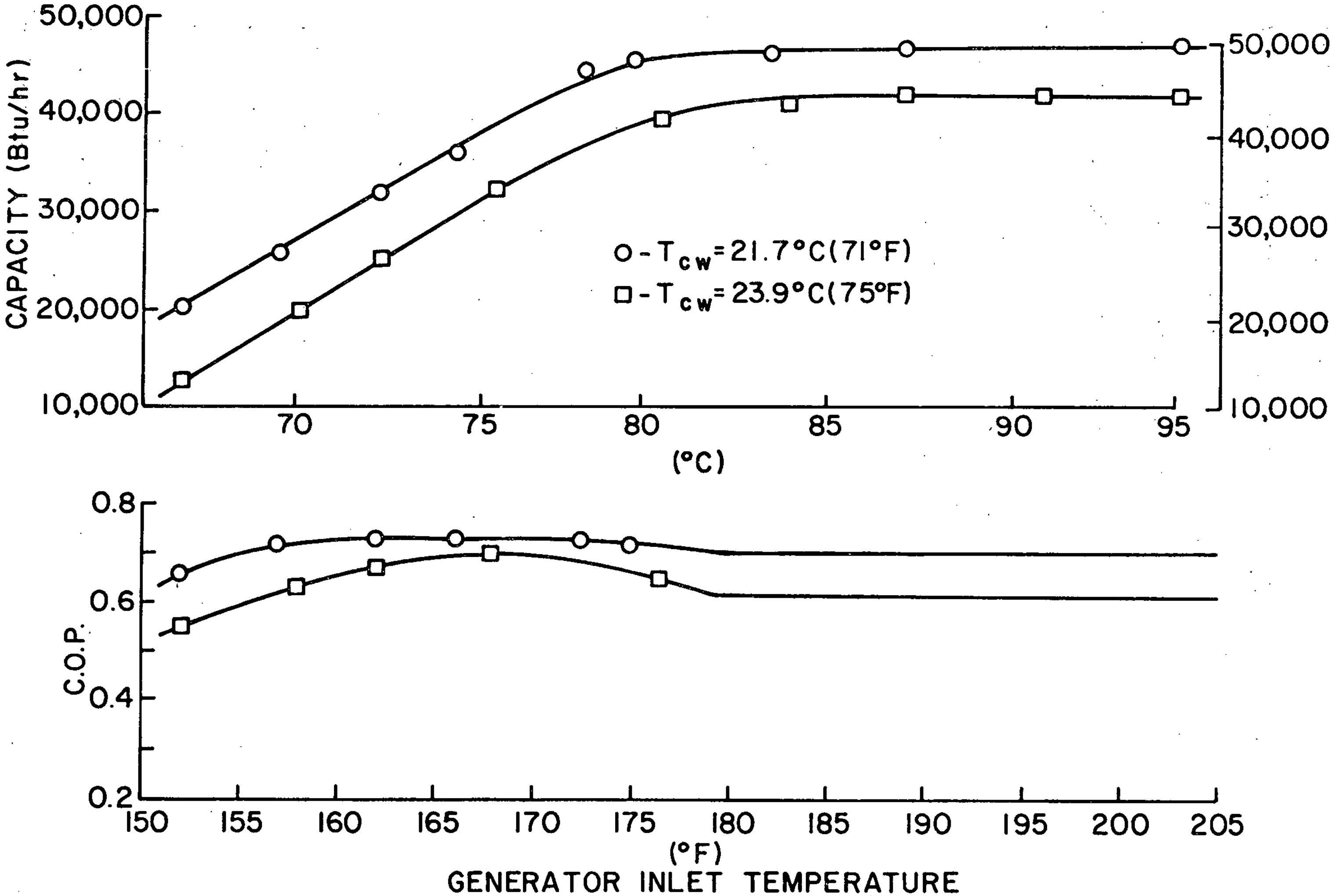


Figure 2.6.1. Performance Characteristics of Modified Charge Arkla Chiller with Tempering Valve

Table 2.7.1. Other Component Specifications

Collector/storage heat exchanger effectiveness	.74
Solar storage boiling point: Washington, D.C. Fort Collins, Colorado	 100°C (212°F) 95°C (203°F)
Service hot water heat exchanger effectiveness	.56
Service hot water load	63,300 kJ/day (60,000 Btu/day)
Service hot water preheat tank volume	303 liters (80 gallons)
House thermostat settings Stage I Stage II	 24.5°C (76.1°F) 25.6°C (78.1°F)
Auxiliary boiler temperature Washington, D.C. Fort Collins, Colorado	 90.6°C (195°F) 80.0°C (176°F)

CHAPTER 3
SIMULATION RESULTS

Simulations of various configurations of the solar air conditioning system were made using TRNSYS with fifteen minute time steps. All configurations (except the use of a tempering valve on the chiller generator) were simulated in both Washington, D.C. and Fort Collins, Colorado for the period June 1 through September 30. Reduction of the simulation output was done according to the following definitions.

$$\eta_{COL} = \frac{Qu}{(I_T) \times (A_{COL})}$$

$$COP = \frac{Q_{COOL}}{Q_{TANK} - Q_{SHW}Solar + Q_{A/C}Aux}$$

$$f_{SHW} = \frac{Q_{SHW}Solar - Q_{SHW}Aux}{Q_{SHW}Solar - Q_{SHW}ENV + Q_{SHW}Aux}$$

$$f_{COOL} = \frac{Q_{TANK} - Q_{SHW}Solar}{Q_{TANK} - Q_{SHW}Solar + Q_{A/C}Aux}$$

$$\text{Total Aux} = Q_{SHW}Aux + Q_{A/C}Aux$$

3.1 TRANSIENT RESPONSE AND CHILLED WATER STORAGE

Four simulation runs were made to investigate the effects of the Arkla chiller transient response on the seasonal performance. They are:

1. Chilled water storage used in Control Mode I, i.e., with priority of chiller operation given to solar storage.
2. Chilled water storage used in Control Mode II, i.e., with priority of chiller operation given to chilled water storage.
3. No chilled water storage used.
4. No chilled water storage and no transient effects considered in the Arkla chiller model.

Data collected from the simulations is presented in Table 3.1.1 for Washington, D.C. and Table 3.1.2 for Fort Collins, Colorado. Also included is the reduced data representing the average COP of the Arkla chiller, the seasonal collector efficiency based on absorber area, the fraction of the cooling and service hot water loads provided by solar energy, and the total auxiliary energy needed for cooling and service hot water requirements.

In both locations considered no discernible difference in system performance exists between Control Modes I and II of chilled water storage. In Mode II the chilled water storage is used more often than in Mode I, but without any significant advantage. However, chilled water storage does improve seasonal performance of the chiller system. Purchased auxiliary energy is decreased by .37 GJ in Washington, D.C. and by .61 GJ in Fort Collins. This trend is as expected, but the magnitude of the energy savings is quite small. Auxiliary energy savings as a result of chilled water storage installation would not pay back the $750 to $1000 estimated investment in the system.

The probable reason for the small savings with chilled water storage is evidenced by the small seasonal improvement of the chiller

Table 3.1.1. Transient Response and Chilled Water Storage Performance - Washington, D.C.

Seasonal Value	Chilled Water Storage Control Mode I	Chilled Water Storage Control Mode II	No Chilled Water Storage	No Chilled Water Storage/ Transients
Q_{SHW} Aux	.5130	.5103	.5022	.4968
Q_{OVER}	4.571	4.775	4.623	4.722
$Q_{A/C}$Aux	22.52	22.48	22.86	21.50
Q_{COOL}	30.21	30.16	30.27	30.44
Qu	41.58	41.71	41.55	41.50
Q_{SHW}Solar	7.957	7.960	7.960	7.974
I_T (GJ/m^2)	2.106	2.106	2.106	2.106
Q_{SOLAR}ENV	7.647	7.605	7.677	7.725
Q_{SHW}ENV	.3873	.3872	.3874	.3894
Q_{TANK}	31.17	31.06	31.02	30.77
A_{COL} (m^2)	40.1	40.1	40.1	40.1
Chilled Water Storage Cycles	21	72	--	--
COP	0.66	0.66	0.66	0.69
η_{COL}	0.49	0.49	0.49	0.49
f_{SHW}	0.94	0.94	0.94	0.94
f_{COOL}	0.51	0.51	0.50	0.51
Total Aux	23.03	22.99	23.36	22.00

All units in GJ unless otherwise noted

Table 3.1.2. Transient Response and Chilled Water Storage Performance - Fort Collins, Colorado

Seasonal Value	Chilled Water Storage Control Mode I	Chilled Water Storage Control Mode II	No Chilled Water Storage	No Chilled Water Storage/ Transients
$Q_{SHW}Aux$	.7722	.7857	.7560	.7290
Q_{OVER}	4.267	4.592	4.308	4.515
$Q_{A/C}Aux$	5.528	5.700	6.146	4.771
Q_{COOL}	19.82	19.71	19.96	20.23
Qu	46.69	46.72	46.59	46.30
$Q_{SHW}Solar$	7.802	7.797	7.812	7.829
I_T (GJ/m^2)	2.164	2.164	2.164	2.164
$Q_{SOLAR}ENV$	6.333	6.289	6.398	6.562
$Q_{SHW}ENV$	.3664	.3657	.3683	.3706
Q_{TANK}	38.82	38.42	38.57	37.76
A_{COL} (m^2)	40.1	40.1	40.1	40.1
Chilled Water Storage Cycles	32	121	--	--
COP	0.54	0.54	0.54	0.58
η_{COL}	0.54	0.54	0.54	0.53
f_{SHW}	0.91	0.90	0.91	0.91
f_{COOL}	0.85	0.86	0.83	0.86
Total Aux	6.29	6.49	6.90	5.50

All units in GJ unless otherwise noted

system when no transient response characteristics are considered. As can be seen from the data in Tables 3.1.1 and 3.1.2, the seasonal COP of the chiller is reduced only three to five percent by transient response effects. This small decrease in seasonal performance is due to the rapid start-up nature of the chiller with quick attainment of steady-state operation and the good matching of the chiller to the house load, which prevents extensive chiller cycling.

3.2 SERVICE HOT WATER PREHEAT SYSTEM

The simulation results concerning the use of solar storage for preheating service hot water is shown in Table 3.2.1 for both locations considered. No chilled water storage system is used in these simulations.

In the first simulation, the service hot water system is removed from the solar storage tank and one hundred percent auxiliary energy is used. This arrangement allows all of the available high temperature solar energy to be used for chiller operation. In both cities' simulations, the percent of solar cooling naturally increases significantly when no service hot water preheating is used. However, the auxiliary energy required for the service hot water in this case increases more than the savings in auxiliary chiller energy. Therefore total auxiliary energy purchased is minimized when solar energy is used for service hot water preheating.

3.3 CHILLER TEMPERING VALVE

Simulation results for the use of a tempering valve to prevent overfiring of a modified charge Arkla chiller properly matched to the cooling load are presented in Table 3.3.1. The tempering valve causes

Table 3.2.1. Service Hot Water Preheat System Performance

Seasonal Value	Fort Collins, Colorado		Washington, D.C.	
	Solar Preheat	100 Percent Auxiliary	Solar Preheat	100 Percent Auxiliary
$Q_{SHW}Aux$	.7560	7.633	.5022	7.633
Q_{OVER}	4.308	5.429	4.623	5.658
$Q_{A/C}Aux$	6.146	3.444	22.86	18.92
Q_{COOL}	19.96	20.39	30.27	30.72
Qu	46.59	45.44	41.55	40.88
$Q_{SHW}Solar$	7.812	0	7.960	0
I_T (GJ/m^2)	2.164	2.164	2.106	2.106
$Q_{SOLAR}ENV$	6.398	7.101	7.677	8.345
$Q_{SHW}ENV$	.3683	0	.3874	0
Q_{TANK}	38.57	34.82	31.02	27.98
A_{COL} (m^2)	40.1	40.1	40.1	40.1
COP	0.54	0.53	0.66	0.66
η_{COL}	0.54	0.52	0.49	0.48
f_{SHW}	0.91	0	0.94	0
f_{COOL}	0.83	0.91	0.50	0.60
Total Aux	6.902	11.077	23.36	26.55

All units in GJ unless otherwise noted

Table 3.3.1. Tempering Valve Performance with Normal Chiller/Load Matching

Seasonal Value	Basic Chiller Design	Chiller System with Tempering Valve
$Q_{SHW}Aux$	.7650	.7398
Q_{OVER}	4.308	5.040
$Q_{A/C}Aux$	6.146	5.793
Q_{COOL}	19.96	20.06
Qu	46.59	46.38
$Q_{SHW}Solar$	7.812	7.838
I_T (GJ/m^2)	2.164	2.164
$Q_{SOLAR}ENV$	6.398	6.550
$Q_{SHW}ENV$	.3683	.3700
Q_{TANK}	38.57	37.08
A_{COL} (m^2)	40.1	40.1
COP	0.54	0.57
η_{COL}	0.54	0.53
f_{SHW}	0.91	0.91
f_{COOL}	0.83	0.84
Total Aux	6.902	6.533

All units in GJ unless otherwise noted

only a slight improvement in the chiller seasonal COP and cooling fraction provided by solar energy.

Table 3.3.2 presents simulation results for a largely oversized chiller with a reduced auxiliary boiler temperature to maximize the system coefficient of performance. Table 3.3.2 presents two system simulations. One uses a tempering valve to prevent overfiring of the chiller generator by solar heated water and the other does not. Little improvement in the average system COP is gained by use of the tempering valve.

The largest savings generated by use of a tempering valve is in the case of the properly sized chiller presented in Table 3.1.1. Using the assumption that an investment of ten times the annual savings is justified for a tempering valve, an investment of $18 can be made if fuel costs are to be $4.75/GJ ($5/MMBtu). However, a minimum investment of $300 in a tempering valve or equivalent equipment is necessary for proper generator temperautre control. Experiments with a less expensive valve at the Colorado State University Solar House I indicate that the cheaper temperating valve's performance is erratic and that it will not deliver the desired temperature to the chiller generator under all conditions. Also, since the valve must be located on the suction side of the generator pump (Figure 2.1.1), cavitation problems are experienced at solar water temperatures near boiling. Use of an additional "primer" pump near the solar storage tank to keep a positive gauge pressure on the tempering valve is therefore required with this valve.

Table 3.3.2. Tempering Valve Performance with a Fifty Percent Oversized Chiller

Seasonal Value	Basic System	System with Tempering Valve
$Q_{SHW}Aux$	.9099	.8937
Q_{OVER}	3.021	3.567
$Q_{A/C}Aux$	9.221	9.178
Q_{COOL}	21.03	21.11
Qu	47.18	46.99
$Q_{SHW}Solar$	7.771	7.789
I_T (GJ/m^2)	2.164	2.164
$Q_{SOLAR}ENV$	7.820	7.951
$Q_{SHW}ENV$	.4110	.4129
Q_{TANK}	38.22	37.32
A_{COL} (m^2)	40.1	40.1
COP	0.53	0.55
η_{COL}	0.54	0.54
f_{SHW}	0.89	0.89
f_{COOL}	0.76	0.76
Total Aux	10.13	10.07

All units in GJ unless otherwise noted

3.4 CHILLER/LOAD MATCHING

Proper matching of an air conditioner's size to the required cooling load is important for efficient operation of any air conditioning system. Proper sizing is even more important for an absorption chiller because of its transient start-up characteristics. However, Arkla only supplies the chiller for residential use in three ton sizes. A design load of five tons would have to be met with two three ton units and a design load of three and one-half tons would likely be met with one three ton unit. Therefore, solar powered absorption chillers are often likely to be either oversized or undersized for the load requirements. To investigate these conditions, two simulation periods are examined. Simulation data is drawn from Fort Collins during the month of September as an example of a chiller oversized for the load requirements. Simulation data is also drawn from a Washington, D.C. "heat wave" in July as an example of a chiller undersized for the load requirements. Data for these situations is shown in Table 3.4.1 and 3.4.2, respectively.

The chiller oversized for the load naturally keeps the house comfortable but cycles frequently. This cycling causes a decrease in the overall coefficient of performance of the unit due to the transient start-up characteristics of the absorption chiller. The simulated data shows a COP that is lower than the seasonal average because of this cycling and also because the absorption unit's generator is being overfired. To examine only the effects of the cycling, data from this period is compared to simulation data for the same period where no transient response is accounted for in the simulation models. Cycling transients and overfiring of the chiller reduced the average chiller COP to 0.40 but overfiring alone reduced the average to 0.53 (Table 3.4.1). Therefore transient characteristics account for a 25 percent reduction in

Table 3.4.1. Oversized Chiller Performance

Period Value	Basic System	Chiller System with no Transients
$Q_{SHW}Aux$	.1053	.0972
Q_{OVER}	4.398	5.510
$Q_{A/C}Aux$	.1020	.0680
Q_{COOL}	2.140	2.220
Qu	13.29	13.28
$Q_{SHW}Solar$	1.973	1.980
I_T (GJ/m^2)	.5900	.5910
$Q_{SOLAR}ENV$	2.034	2.082
$Q_{SHW}ENV$	7.302	6.080
Q_{TANK}	.0983	.0990
A_{COL} (m^2)	40.1	40.1
COP	0.40	0.53
η_{COL}	0.56	0.56
f_{SHW}	0.95	0.95
f_{COOL}	0.98	0.98
Total Aux	.2073	.1652

All units in GJ unless otherwise noted

Table 3.4.2. Undersized Chiller Performance

Period Value	Basic System	Chiller System with no Transients
$Q_{SHW}Aux$	3.24	3.24
Q_{OVER}	14.01	15.31
$Q_{A/C}Aux$	542.00	527.20
Q_{COOL}	540.00	542.00
Qu	330.00	336.00
$Q_{SHW}Solar$	65.00	65.00
I_T (MJ/m^2-day)	17.46	17.26
$Q_{SOLAR}ENV$	64.00	62.93
$Q_{SHW}ENV$	3.24	3.18
Q_{TANK}	268.00	274.00
A_{COL} (m^2)	40.10	40.10
COP	0.73	0.74
η_{COL}	0.47	0.49
f_{SHW}	0.95	0.95
f_{COOL}	0.27	0.28
Total Aux	545.20	530.40

All units in MJ/day unless otherwise noted the average chiller COP or a corresponding 25 percent increase in generator energy for the case of this oversized chiller.

The undersized chiller, as shown in Table 3.4.2, operates with a relatively high COP because the chiller is neither cycling nor being overfired. However, the house temperature rises well above the desired temperature level, causing the two-stage thermostat to operate the chiller on the auxiliary boiler whenever the solar storage temperature is less than the boiler operating temperature. The boiler operates the chiller at maximum cooling capacity but still provides neither adequate cooling of the house nor a reasonable solar operating fraction.

3.5 SOLAR STORAGE VOLUME

Solar heated water storage volume has an effect on the performance of any solar system. In a heating mode, increased storage volume tends to increase the thermal performance of a solar heating system although with a strongly diminishing return past a certain optimal size. The performance of a solar air conditioning system, however, can be diminished by too large a solar storage volume. Because a solar operated absorption chiller has a relatively high minimum operating temperature, an oversized storage system can often prevent storage temperatures from reaching the minimum required temperature when needed, causing more auxiliary fuel to be burned. Too small a storage system, however, allows storage boiling and the accompanying loss of collected solar heat.

Sensitivity studies for a range of collector and storage sizes were made for both the standard and modified charge Arkla chillers. Results of these studies are presented in Figure 3.5.1 and 3.5.2. The standard charge chiller requires higher generator temperatures to

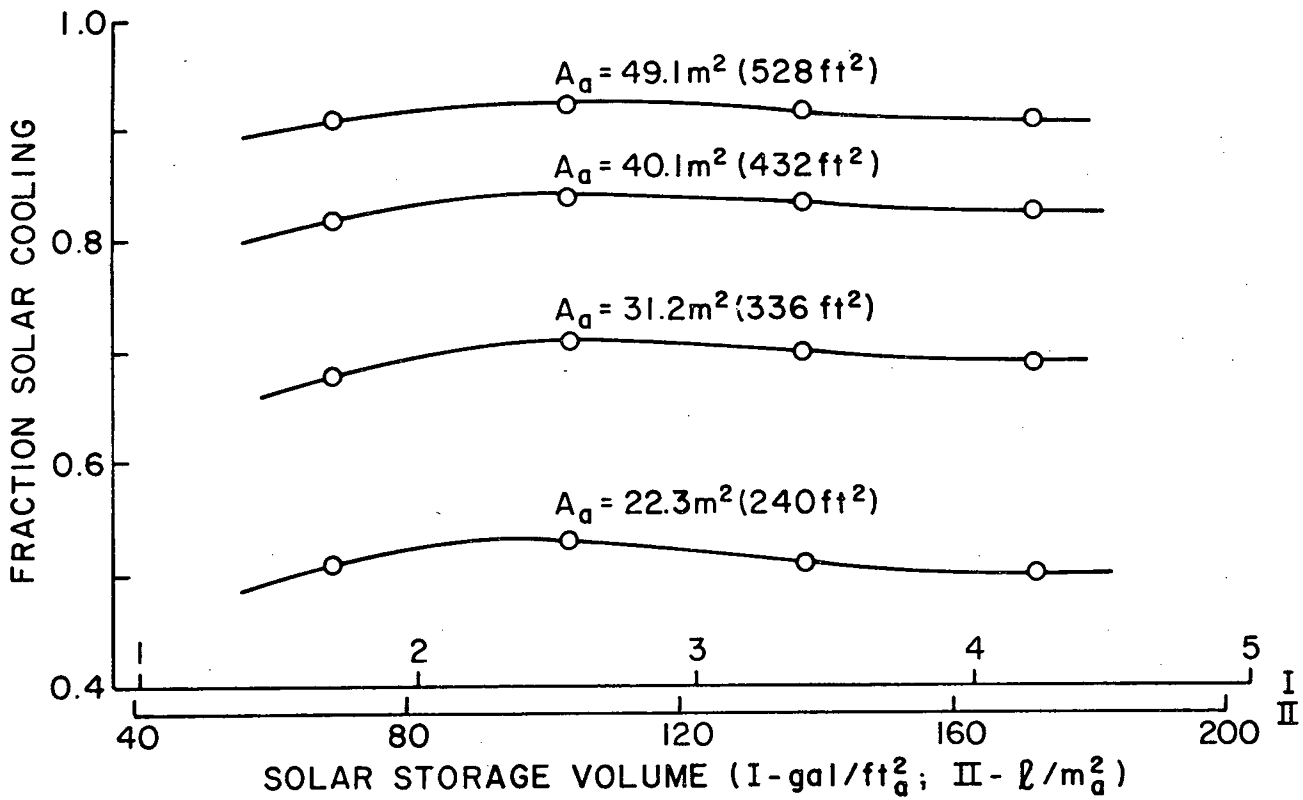


Figure 3.5.1. Solar Storage Volume Effects -- Fort Collins, Colorado System

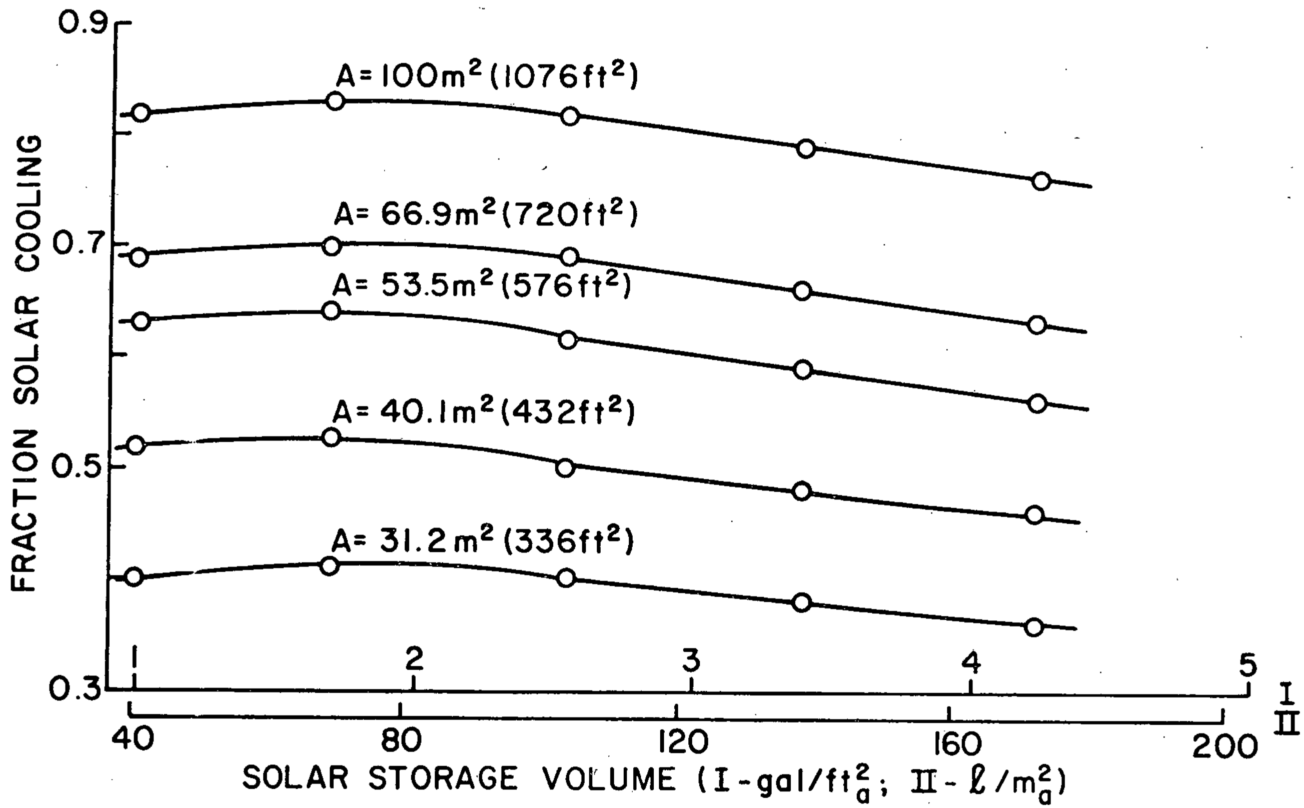


Figure 3.5.2. Solar Storage Volume Effects -- Washington, D.C. System

operate than the modified charge unit and requires a corresponding decrease in the relative solar storage volume. The optimal storage volume for the modified charge chiller in Fort Collins, Colorado is 90 to 115 ℓ/m^2 (2.2 to 2.8 gal/ft^2_a). The optimal storage volume for the standard charge chiller in Washington, D.C. is between 60 and 80 ℓ/m^2 (1.5 and 2.0 gal/ft^2_a).

CHAPTER 4

SUMMARY, CONCLUSIONS, AND RECOMMENDED DESIGN SPECIFICATIONS

4.1 CHILLED WATER STORAGE

The transient response characteristics of the Arkla Solaire WF-36 Absorption Chiller are not a severe enough system inefficiency to warrant improvement by the addition of a chilled water storage system. Transient performance does deteriorate the seasonal coefficient of performance of the chiller, especially if the unit is oversized for the load during a large portion of the cooling season. However, the incremental improvement in coefficient of performance due to the installation of chilled water storage cannot return the investment required for the storage system. Chilled water storage savings were a maximum of .61 GJ (.58 MMBtu) for the cooling season. Allowing an investment of ten times the annual savings, an expenditure of only thirty dollars is justified if fuel costs are $4.75 GJ ($5.00/MMBtu).

4.2 SERVICE HOT WATER PREHEAT SYSTEM

The service hot water heating requirements of a typical residence can be a substantial portion of the residential energy consumption. The use of a service hot water solar preheating system, as shown in Figure 2.1.1, can reduce the overall auxiliary energy requirements of the residence. Because of the high minimum operating temperature of the absorption chiller, the service hot water preheating system can be sized to deliver approximately 90 percent or more of the service hot water energy requirements during the cooling season.

It should be noted that double wall protection between potable water and a toxic material such as the collector antifreeze solution is required by national building codes. The two walls are the collector/storage heat exchanger and the storage/service hot water preheat tank heat exchanger. Therefore, the water in the solar storage tank must not contain toxic or polluting corrosion inhibitors. To prevent the need for such inhibitors, it is recommended that the solar storage tank be manufactured of fiberglass with high temperature resins.

4.3 CHILLER TEMPERING VALVE

A modified charge absorption chiller operating in an arid climate can often be "overfired" by high temperature solar heated water, causing a reduction in the average COP of the chiller. However, for a properly sized solar system, overfiring does not occur often enough nor with severe enough consequences to warrant correction by a generator tempering valve. Most "overfiring" occurs in the early and late portions of the cooling season when excess solar heat is frequently being boiled from the storage tank. Under these circumstances the tempering valve is not able to substantially improve the average COP of the chiller.

In a situation where the chiller is largely oversized for the load, the tempering valve also does little to increase the seasonal performance of the system and is not an economically justified addition to the chiller system.

4.4 SOLAR STORAGE VOLUME

Proper design of the solar storage volume relative to the collector area is important for optimal performance of the solar operated absorption chilling system. A frequent misconception that more solar storage yields a greater fraction of solar contribution to the cooling load is shown to be incorrect by the simulation results. In fact, as the minimum chiller operating temperature increases, the data tends to indicate a reduction in the optimal storage volume/collector area ratio. The optimal storage volume for the modified chiller in Fort Collins, Colorado is 90 to 115 ℓ/m_a^2 (2.2 to 2.8 gal/ft_a^2). The optimal storage volume for the standard charge chiller in Washington, D.C. is between 60 and 80 ℓ/m_a^2 (1.5 to 2.0 gal/ft_a^2).

4.5 RECOMMENDED DESIGN SPECIFICATIONS OF A RESIDENTIAL SOLAR ABSORPTION CHILLER SYSTEM

The simulation studies examining the performance of the solar operated absorption chiller in various operating configurations reveal no "add on" equipment which is economically viable in improving the performance in residential operation. The simplest design, and therefore the least capital intensive, fortunately is the most economically viable. A complete design schematic for a residential solar operated air conditioning system and its specifications are shown in Figures 4.6.1, 4.6.2, 4.6.3, 4.6.4, and Table 4.6.1. This design specification may be used to build a technically sound solar operated air conditioning system. The economic characteristics of the system, however, must be determined by other means.

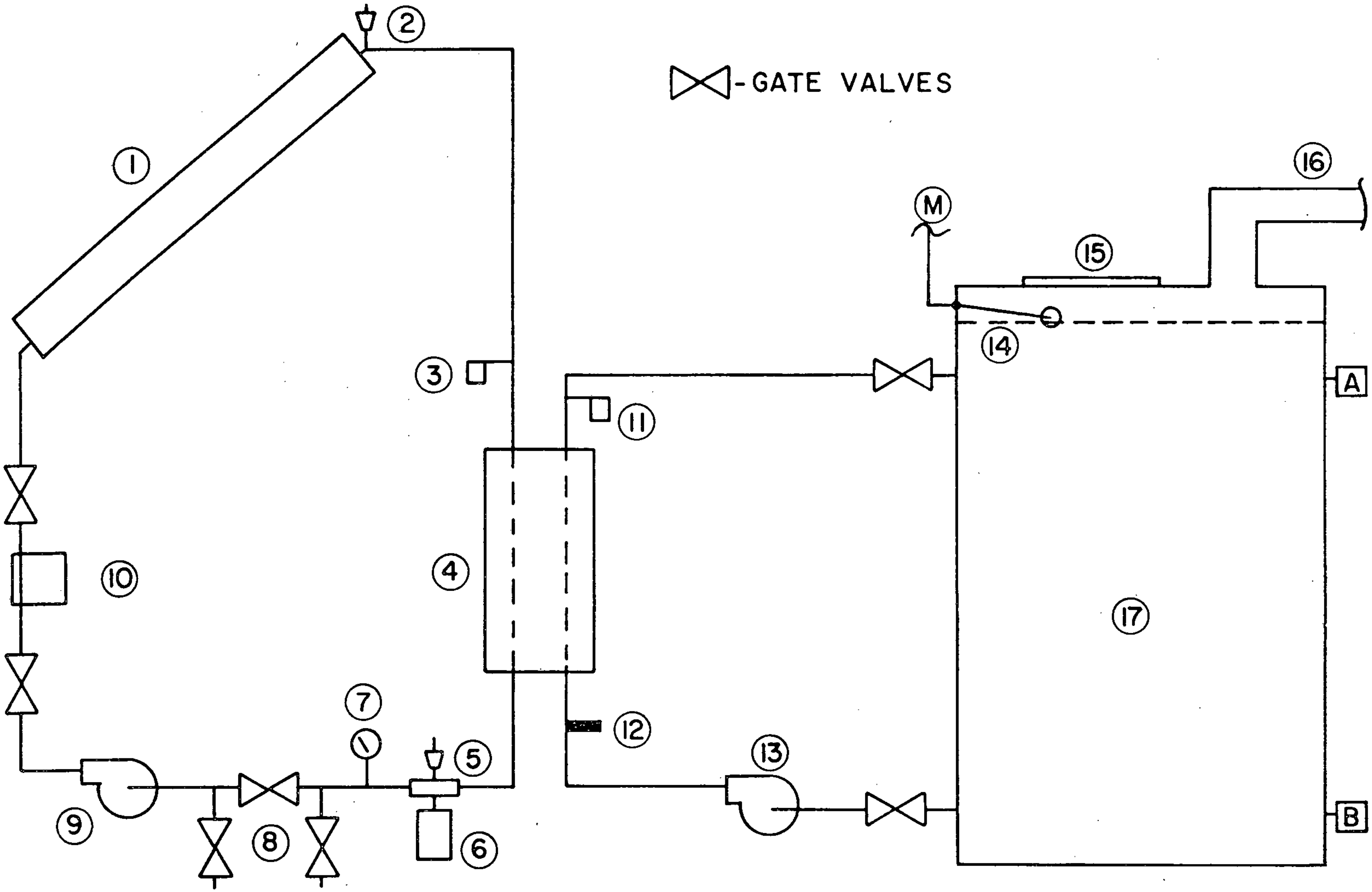


Figure 4.6.1. Recommended Design Schematics of a Residential Solar Operating Air Conditioning System, Collector/Storage System

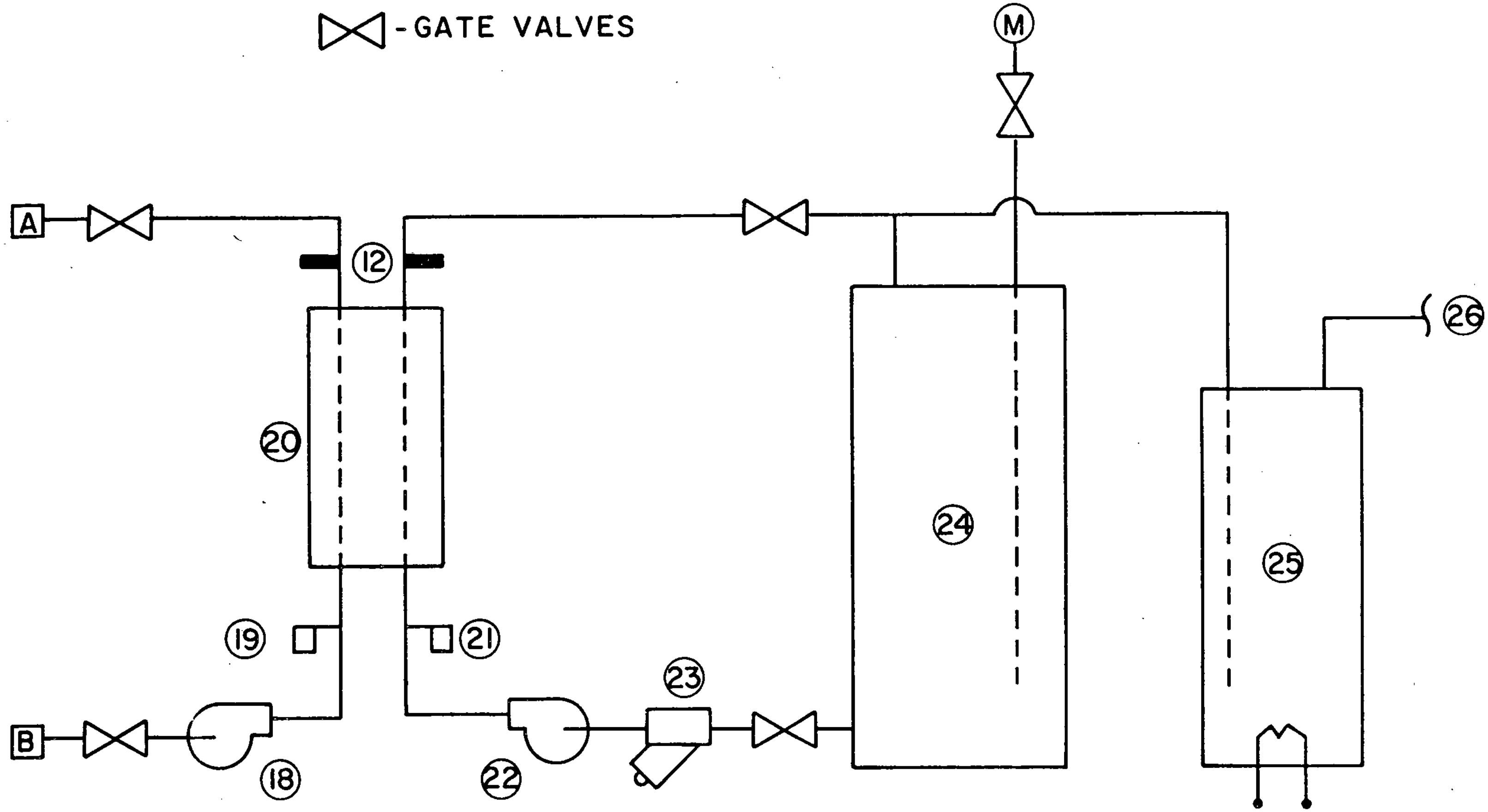


Figure 4.6.2. Recommended Design Schematics of a Residential Solar Operated Air Conditioning System, Service Hot Water Preheat System

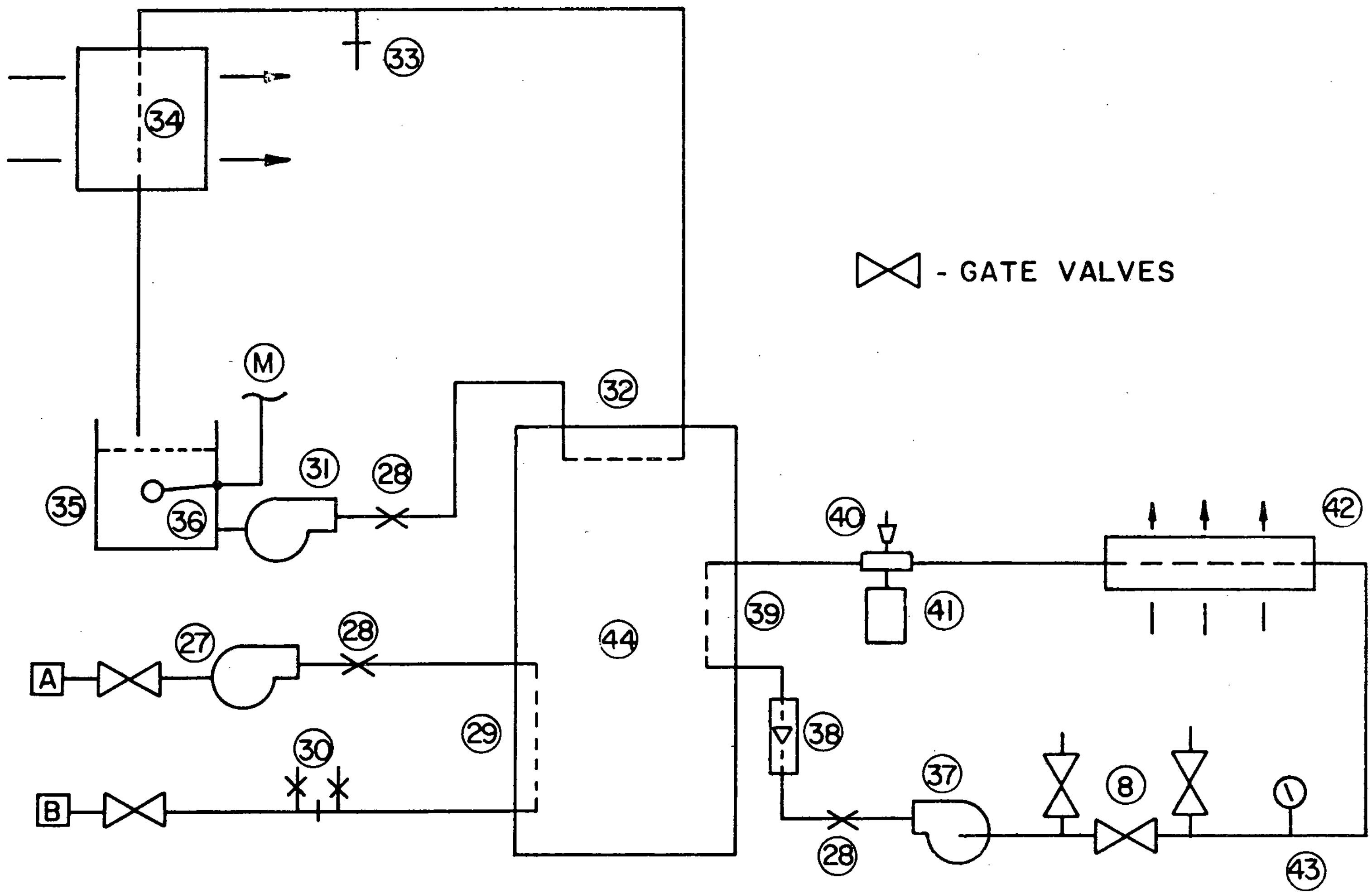


Figure 4.6.3. Recommended Design Schematics of a Residential Solar Operated Air Conditioning System, Absorption Chiller System

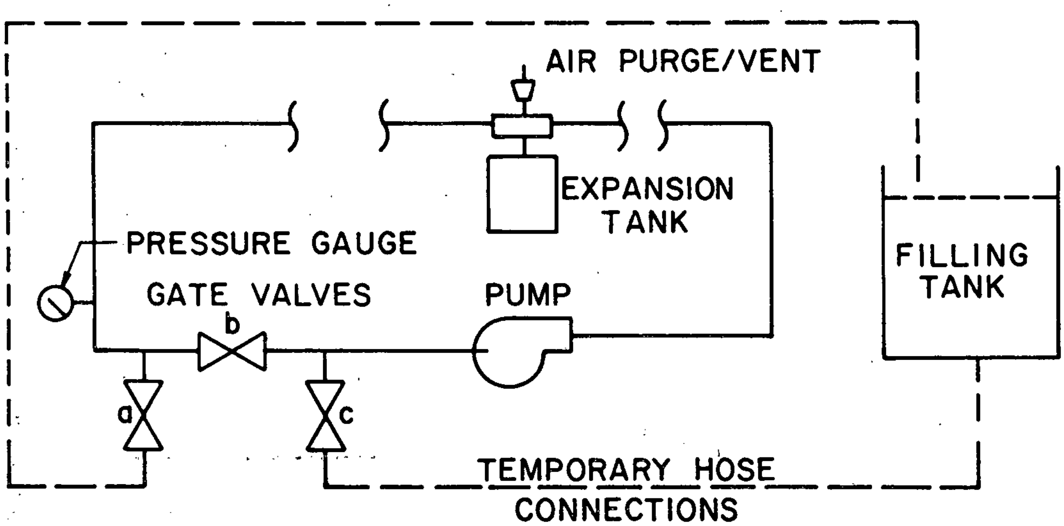


Figure 4.6.4. Closed Hydronic System Fill/Drain

Table 4.6.1. Recommended Design Specifications of a Residential Solar Operated Air Conditioning System (Reference Figures 4.6.1, 4.6.2, 4.6.3)

Equipment Number	Specification
1	Evacuated tube collectors with 50/50 antifreeze solution
2	Coin type air vent(s) at system high point(s)
3	30 psi pressure relief valve. No shut off valves to be placed between relief valves and collectors
4	Single pass counterflow heat exchanger with UA and flow rates suitable for an effectiveness of 0.75 or greater
5	Air purge/vent
6	Bladder type expansion tank(s) with antifreeze make-up supply. Size expansion tanks to be at least 3 gallons for every 10 gallons of antifreeze solution
7	Pressure gauge 0 - 45 psi
8	Purge/fill/drain gate valves. See Figure 4.6.4
9	Collector pump. Size for a flow rate of .038 gpm per square foot of absorber area.
10	Course cartridge filter or #100 mesh strainer
11	30 psi relief valve
12	Vacuum breakers
13	Storage pump. Size for a flow rate one and a half times that of the collector pump or according to specifications by the heat exchanger manufacturer
14	Float valve for filling the storage tank
15	Tight fitting manhole cover to prevent steam escaping to the house
16	Vent to outdoors
17	Fiberglass storage tank with high temperature resins insulated to at least R = 30 Ft^2-Hr-°F/Btu
18	Service hot water pump. Locate and size the inlet piping to prevent cavitation. Size the flow rate according to heat exchanger specifications
19	30 psi relief valve
20	Service hot water heat exchanger. Size for an effectiveness of 0.5 or greater
21	150 psi relief valve
22	Service hot water pump. Size according to heat exchanger specifications
23	Y-strainer with a #100 mesh screen

Table 4.6.1 (continued)

Equipment Number	Specification
24	Service hot water preheat tank
25	Conventionally fueled service hot water tank
26	Service hot water supply to house
27	Chiller generator pump. Size the flow rate according to chiller specifications. Locate and size the inlet piping to prevent cavitation
28	Balancing valve
29	Chiller generator connection
30	Calibrated orifice and petcocks or equivalent apparatus for measuring the generator flow rate
31	Condenser water circulation pump. Size the flow rate according to chiller specifications
32	Chiller condenser connections
33	Automatic bleed off. Adjust the bleed off rate to approximately equal the cooling tower evaporation rate
34	Cooling tower. Size according to chiller specifications
35	Condenser water holding tank. Approximately 30 gallons. Polypropylene or equivalent material
36	Float valve to provide make-up condenser water
37	Chilled water circulating pump. Flow rate according to chiller specifications
38	Visual flow meter
39	Chiller chilled water connections
40	Air purge/vent
41	Expansion tank and chilled water make-up supply. Chilled water may be 10 percent antifreeze for extra insurance against chiller freeze-up
42	Duct coil. Size according to the chiller capacity and the load sensible heat ratio
43	Pressure gauge, 0 - 45 psi

Insulate pipes to R = 6 Ft^2-Hr-°F/Btu minimum.

Table 4.6.2. Closed Hydronic System Filling Procedure

Step	Procedure
1	With the system empty, inflate the bladder in the expansion tank to approximately six pounds per square inch
2	Place or mix the fluid to be used in the filling tank
3	Connect the temporary hoses to the filling tank as shown
4	Close valve B
5	Open valves A and C
6	Start the pump and recirculate the fluid until the system is full and well purged
7	Close valve A until the pressure gauge reads approximately fifteen pounds per square inch
8	Close valve C and open valve B
9	Remove filling apparatus if desired
10	Repeat the procedure when the pressure gauge reads less than five pounds per square inch

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APPENDIX A

ARKLA SOLAIRE ABSORPTION CHILLER PERFORMANCE MEASUREMENTS

ARKLA CHILLER PERFORMANCE DATA†

STANDARD MODEL WF-36

Hot Water Flow .... 11.0 gpm
Condensing Water Flow................ 12.0 gpm
Chilled Water Flow................... 7.2 gpm
Chilled Water Leaving Temperature..... 45°F

Hot Water Inlet Temp	Hot Water Outlet Temp	Energy Input Btu/Hr	Inlet Cond. Water Temp	Delivered Capacity Btu/Hr	Delivered Capacity Tons	Rejected Heat Btu/Hr	COP
170°F	167.0	16,400	80°F	9,700	0.81	26,100	.59
	167.4	14,500	85°F	6,400	0.53	20,900	.44
	*	*	90°F	*	*	*	*
175°F	170.7	23,800	80°F	17,300	1.44	41,400	.73
	171.1	21,600	85°F	13,100	1.09	34,700	.61
	*	*	90°F	*	*	*	*
180°F	174.3	31,200	80°F	24,400	2.03	55,600	.78
	174.8	28,800	85°F	19,400	1.62	48,200	.67
	175.7	23,800	90°F	14,200	1.18	38,000	.60
185°F	178.0	38,400	80°F	31,100	2.59	69,500	.81
	178.5	35,900	85°F	25,600	2.13	61,500	.71
	179.4	30,600	90°F	19,300	1.61	49,900	.63
190°F	181.7	45,800	80°F	36,800	3.07	82,600	.80
	182.2	42,900	85°F	31,300	2.61	74,200	.73
	183.2	37,500	90°F	23,800	1.98	61,300	.63
195°F	185.3	53,100	80°F	40,600	3.38	93,700	.76
	185.9	50,000	85°F	36,000	3.00	86,000	.72
	186.9	44,300	90°F	27,600	2.30	71,900	.62
200°F	189.3	58,800	80°F	41,800	3.48	100,600	.71
	189.8	56,000	85°F	40,200	3.35	96,200	.72
	190.7	51,000	90°F	30,500	2.54	80,500	.60
205°F	193.4	63,800	80°F	42,000	3.50	105,800	.66
	193.9	60,800	85°F	42,000	3.50	102,800	.69
	194.8	56,200	90°F	32,500	2.71	88,700	.58

*Unit operation unstable in these areas. Conditions for rated capacity

†As published by Arkla Industries, Evansville, Indiana

MODIFIED CHARGE ARKLA CHILLER STEADY-STATE
PERFORMANCE AT "OVERFIRED" CONDITIONS

Experimental Data Taken August 2, 1977 at CSU Solar House I
by James A. Leflar

$T_{C.W.} = 71°F$

Chilled Water Flow Rate = 7.2 gpm

T_{gen} (°F)		$T_{CH.W.}$ (°F)		Generator Flow (gpm)	Q_{gen} (Btu/hr)	CAP (Btu/hr)	COP
In	Out	In	Out				
175.2	163.7	55.1	42.4	11.3	64,950	45,700	.704
181.9	170.1	56.5	43.7	11.3	66,640	46,060	.691
189.4	175.0	56.1	43.1	11.4	82,050	46,780	.570
101.7	18.21	58.3	45.2	11.4	117,400	47,140	.402

$T_{C.W.} = 75°F$

Chilled Water Flow Rate = 7.2 gpm

T_{gen} (°F)		$T_{CH.W.}$ (°F)		Generator Flow (gpm)	Q_{gen} (Btu/hr)	CAP (Btu/hr)	COP
In	Out	In	Out				
183.1	169.8	54.7	43.2	11.2	74,450	41,380	.556
188.8	172.7	55.7	44.1	11.2	90,120	41,740	.463
195.9	173.8	57.2	45.5	11.2	123,700	42,100	.340
203.3	167.9	56.6	45.0	11.2	198,200	41,740	.211

ARKLA TRANSIENT RESPONSE TEST: OFF TIME = ½ HOUR

September 13, 1977 James A. Leflar

Time	Generator Temp (°F)	Chilled Water Temp (°F) In	Chilled Water Temp (°F) Out	ΔT
0	171	68	68	0
1	174	66	64	2
2	175	67	62	5
3	178	68	61	7
4	180	68	60	8
5	180	68	59	9
6	181	67	58	9
7	185	67	58	9
8	187	66	57	9
9	188	66	57	9
10	188	65	56	9
11	190	64	55	9
12	192	61	52	9
13	192	62	53	9
14	192	61	52	9
15	192	61	52	9
16	192	60	51	9
17	194	59	50	9
18	194	59	49	10
19	194	58	48	10
20	194	57	47	10
21	194	57	47	10
22	195	55	45	10
23	195	55	45	10
24	195	55	45	10
25	195	55	45	10
26	195	55	45	10
27	195	55	45	10
28	195	55	45	10
29	195	55	45	10
30	195	55	45	10

Over two (15 minute) time steps the unit delivered 269/300 = 90% of steady-state operation

Assuming all of steady-state performance in the first time step, the Factor is:

$$\frac{1+F}{2} = .9 \Rightarrow \underline{F = .80}$$

ARKLA TRANSIENT RESPONSE TEST: OFF TIME = 2 HOURS

September 13, 1977 James A. Leflar

Time	Generator Temp (°F)	Chilled Water Temp (°F) In	Chilled Water Temp (°F) Out	ΔT
0	164	91	91	0
1	172	85	82	3
2	174	81	76	5
3	175	77	71	6
4	175	68	64	4
5	176	66	58	8
6	176	62	54	8
7	177	60	52	8
8	177	59	51	8
9	178	58	51	7
10	178	59	52	7
11	179	59	52	7
12	179	59	52	7
13	178	59	51	8
14	178	58	51	7
15	178	58	50	8
16	178	57	49	8
17	178	57	49	8
18	178	57	47	10
19	178	55	46	9
20	178	55	45	10
21	178	55	44	10
22	178	54	44	10
23	178	54	44	10
24	178	53	43	10
25	178	53	43	10
26	178	53	43	10
27	178	53	43	10
28	178	53	43	10
29	178	53	43	10
30	178	53	43	11
31		53	43	11
32		53	43	11
33		53	43	11
34		53	43	11
35		53	43	11

Over two (15 minute) time steps the unit delivered 247/300 = 75% of steady-state operation

Assuming all losses from steady-state in the first timestep, the Factor is:

$$\frac{1 + F}{2} = .75 \Rightarrow \underline{F = .50}$$

ARKLA TRANSIENT RESPONSE TEST: OFF TIME = 8 HOURS

September 13, 1977 James A. Leflar

Time	Generator Temp (°F)	Chilled Water Temp (°F)		ΔT
		In	Out	
0	124	73	73	0
1	125	73	73	0
2	129	73	73	0
3	131	73	73	0
4	136	73	73	0
5	142	73	73	0
6	145	73	73	0
7	149	73	73	0
8	152	73	72	1
9	156	73	71	2
10	159	72	68	4
11	164	72	67	5
12	169	67	63	5
13	173	66	61	5
14	176	65	59	5
15	179	63	57	6
16	182	61	57	6
17	184	61	54	7
18	187	60	52	8
19	189	58	51	7
20	190	57	50	7
21	191	57	47	10
22	192	55	46	9
23	193	54	46	8
24	193	55	46	9
25	193	55	46	9
26	193	55	45	10
27	193	55	45	10
28	193	55	45	10
29	193	55	45	10
30	193	55	45	10

Over two (15 minute) time steps the unit delivered 163/300 = 54% of steady-state operation

Assuming all losses from steady-state in the first time step, the Factor is:

$$\frac{1 + F}{2} = .54 \Rightarrow \underline{F = .09}$$

APPENDIX B

CALCULATIONS OF WET BULB TEMPERATURES

CALCULATION OF WET BULB TEMPERATURES AT SEA LEVEL

Given: Ambient Dry Bulb Temperature (T_{amb})
Dew Point Temperature (T_{DP})

Assuming: Standard Barometric Pressure

$$P_w = [3.1792995 + (.011552)(T_{DP}) + (.0054763)(T_{DP})^2 - (.000030066)(T_{DP})^3 + (.00000065782)(T_{DP})^4]/100$$

$$W = (.622)(P_w)/(14.696 - P_w)$$

$$H = (.622)(P_w)/(14.696 - P_w)$$

$$T_{WB} = -4.1749430 + (3.7955)(H) - (.070084)(H)^2 + (.00079516)(H)^3 - (.0000038914)(H)^4$$

Allowable Ranges:

$32°F < T_{DP} < 85°F$

$17.65 < H < 49.42$

CALCULATION OF WET BULB TEMPERATURES AT 5000 FEET ELEVATION

Given: Ambient Dry Bulb Temperature (T_{amb})
Dew Point Temperature (T_{DP})

Assuming: Standard Barometric Pressure

$$P_w = [-.17569208 + (.30217)(T_{DP}) - (.0037797)(T_{DP})^2 + (.000098496)(T_{DP})^3]/100$$

$$W = (.622)(P_w)/(12.23 - P_w)$$

$$H = [.240 + (.45)(W)](T_{AMB}) + (1061)(W)$$

$$T_{WB} = 4.8770557 + (2.7025)(H) - (.032626)(H)^2 + (.00018112)(H)^3$$

Allowable Figures:

$32°F < T_{DP} < 65°F$

$28.85 < H < 48.71$

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APPENDIX C

CLIMATOLOGICAL DATA

CLIMATOLOGICAL DATA
(May - September)

	Washington, D.C.	Fort Collins, Colorado
Climate	Humid	Arid
Average dew point temperature	18.3°C (65°F)	8.3°C (47°F)
Average relative humidity	70 percent	30 percent
Average dry bulb temperature	22.2°C (72°F)	17.2°C (63°F)
Cooling degree days	556°C-Day/Year (1000°F-Day Year)	278°C-Day/Year (500°F-Day/Year)
Cooling design temperature	35°C (95°F)	35°C (95°F)
Average solar radiation		
kJ/m^2-Day	170.2	170.5
Btu/ft^2-Day	1737	1740

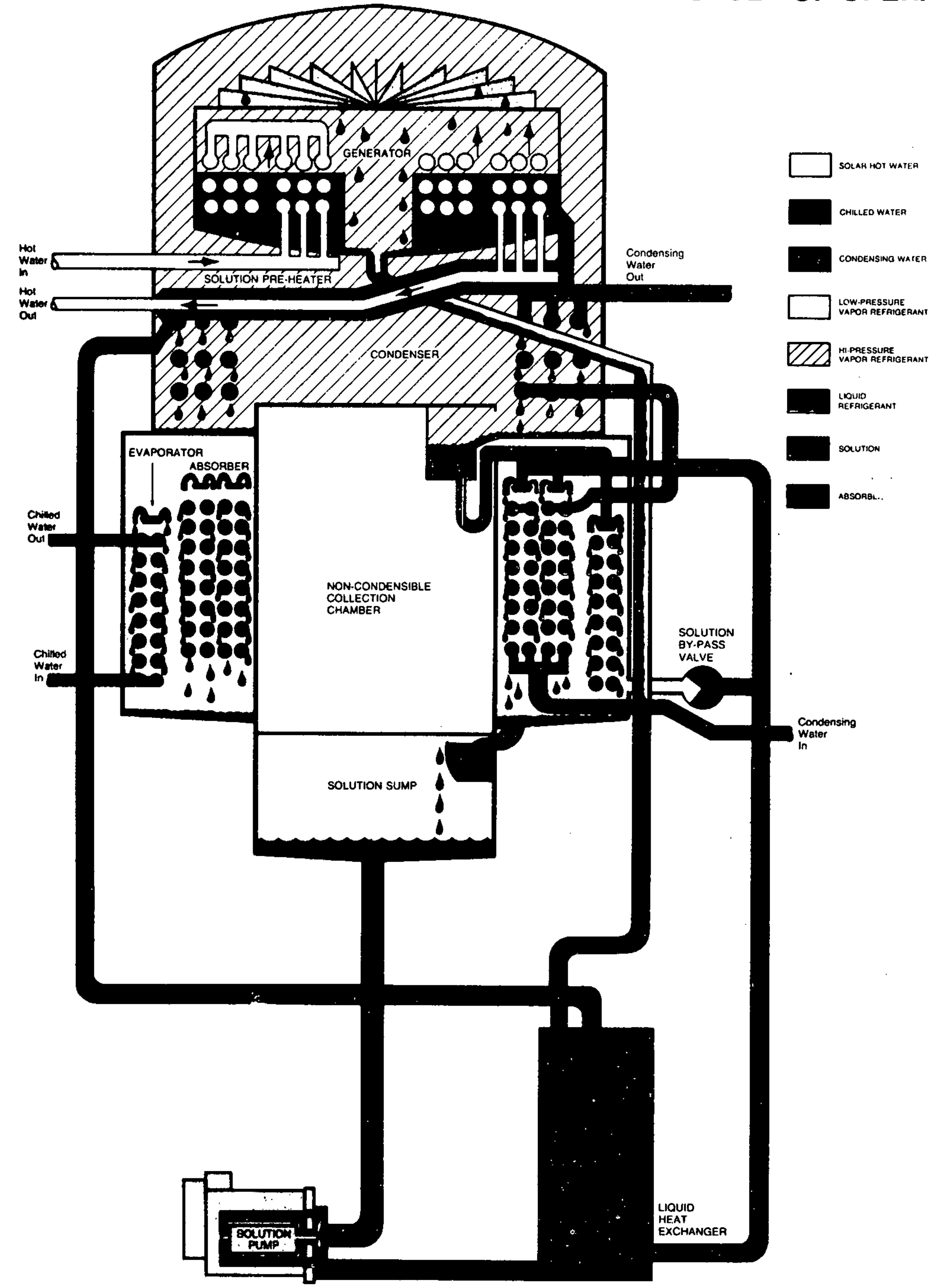
APPENDIX D

ARKLA SOLAIRE ABSORPTION CHILLER OPERATION

ARKLA solaire® SUN POWERED AIR CONDITIONING

SOLAIRE 36

CYCLE OF OPERATION



The Arkla SOLAIRE® unit operates on the absorption principle, utilizing solar heated water as the energy source, with lithium bromide and water as the absorbent-refrigerant solution. Its refrigeration tonnage is delivered by chilled water which circulates in a closed loop between the unit's evaporator coil and a standard fan-coil assembly(s) located inside the conditioned space. In another loop, condensing water is circulated through the unit's absorber and condenser coils to remove waste heat from the cycle.

The explanation of the cycle illustrated above begins with the solution in the solution sump. The hermedically sealed stainless steel solution pump moves solution through the liquid heat exchanger to the generator. Inside the generator, heat (from solar heated water) seperates the refrigerant (water) from the absorbent (lithium bromide solution). At this point the absorbent flows by gravity back through the liquid heat exchanger to the absorber. The vaporized refrigerant passes to the condenser, where it gives up its latent heat to the condensing water and is liquefied. It then flows through a metering device to the evaporator. There, the heat from the chilled water circuit is absorbed by the evaporating refrigerant. The refrigerant vapor and absorbent are reunited by the absorption process. The lithium bromide solution flows into the solution sump to begin the cycle again.

Refinements to improve the efficiency and flexibility of the cycle include;

A. The liquid heat exchanger which conserves heat by using the hot absorbent liquid to preheat the solution before it enters the generator.

B. The solution by-pass valve. This valve is opened by a thermostatic switch which senses refrigerant temperature in the evaporator. The valve by-passes solution from the absorber coil allowing the unit to maintain maximum efficiency.

71S-259

*U.S. GOVERNMENT PRINTING OFFICE: 1978-740-306/4254. Region 4.