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COMPARATIVE REPORT

PERFORMANCE OF ACTIVE SOLAR SPACE COOLING SYSTEMS

1982 COOLING SEASON

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EXECUTIVE SUMMARY

This report provides a detailed analysis of solar absorption cooling and solar Rankine cooling processes as represented by the National Solar Data Network (NSDN) systems.

Five solar cooling systems were monitored in 1982; four of these have absorption chillers and one has a Rankine engine. Of the four absorption chillers, two are directly solar fired and two are boiler fired using solar energy as the preheat to the boiler.

The composite data for the five sites covers the period from September 1981 through December 1982. There are 36 site months of of data covered in the report. These are all commercial systems with buildings ranging in size from 5,000 to 84,000 square feet There are three evacuated-tube, one flat-plate, and one linear concentrating collector systems.

The following conclusions and lessons learned were obtained from the comparative analysis:

- The best conversion efficiency was 10%. With better controllers, this can be increased to 17%, and, with improved technology, perhaps a 28% conversion efficiency is possible with absorption chillers.
- Presently, the Rankine engine outperforms the absorption cooling systems, based on energy savings and cooling Coefficient of Performance (COP).
- Solar cooling systems are not cost-effective now.
- Costs to operate solar cooling systems ranged from 149 to 560 W/ft^2 of collector per month. An operating energy cost of 250 W/ft^2-month is achievable at most installations in the United States.
- System losses ranged from 10% to 48%. On one system, bypassing storage reduced losses to 12% during the summer.
- Maintenance of collectors, controls, and chillers is necessary for continued high performance. Several of these systems performed more poorly during the 1982 cooling season than during the 1981 cooling season.
- All of the buildings were of an energy efficient design and required nine to 12 BTU/ft^2 floor area-cooling degree-day.
- Most of the cooling systems were oversized for the average load by three times. This reduces chiller performance and adds to initial cost. Perhaps designing with computer models will allow designers to more nearly match the load with the equipment.

- Flat-plate collectors perform as well as the evacuated-tube collectors at temperatures below 200°F.

- More use of ventilating air should be made for cooling since the average seasonal outdoor temperatures are lower than the average building temperatures at three of the sites.

- Solar cooling systems require performance improvements of 10 times to reach the energy savings of solar heating systems and 25 times to reach the energy savings of solar domestic hot water systems.

The NSDN is a primary vehicle for the Federal Government to track the performance of the representative space cooling systems selected for demonstration. The purpose of this report is to present the most recent composite performance results for selected active solar space cooling sites in the NSDN. Results presented have been developed on the basis of analysis of instrumented sites monitored during the 1982 cooling season. Sites analyzed include a cross section of major types of active solar cooling systems distributed throughout the United States.

Millions of individual measurements from these sites provide a large reservoir of data for operational and comparative analysis. The detailed measurement data for these systems has been analyzed and is presented on the basis of monthly and seasonal performance factors. The data points recorded by on-site instruments are accumulated, reduced, and analyzed in accordance with a hierarchical structure which leads to an understanding of overall system performance. For the NSDN, this hierarchy consists of the following:

Scan Level [five minute and 20 second (320 second) interval on-site]

Conversion to Engineering Units

Hourly Averages and Sums

Daily Averages and Sums

Monthly Averages and Sums

Seasonal Averages and Sums

In addition to this hierarchy which addresses single-site data, analyses are conducted which combine the performance results of multiple sites and allow comparative analyses to be accomplished.

Parameters and performance indices presented include overall system delivered loads, solar fraction of the load, coefficient of performance, energy collected and stored, and various subsystem efficiencies. The comparison of these factors has allowed evaluation of the relative performance of various systems.

Analyses performed for which comparative data is provided include:

- Energy savings and operating costs in terms of BTU
- Overall solar cooling efficiency and coefficient of performance
- Hourly building cooling loads
- Actual and long-term weather conditions
- Collector performance
- Chiller performance
- Normalized building cooling loads per cooling degree-day and building area
- Cooling solar fractions, design and measured

The NSDN was established by authorization and appropriations of the U.S. Congress and is administered through the Department of Energy by the Argonne National Laboratory. The availability of these results of the NSDN is in large part due to the continuing support of these and other organizations including several professional societies, grantees and owners of buildings who have participated, as well as the many analysts, engineers, and field people of Vitro Laboratories and other staff.

Information related to manufacturers and system designers has been included in the site descriptions for reference purposes. Inclusion of this information and analysis data pertaining to any specific design or product in no way represents an endorsement of that design or product by either the Federal Government or Vitro Laboratories.

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Section I

INTRODUCTION

The purpose of this document is to present the most recent composite results of analysis performed by Vitro Laboratories of solar space cooling data for active space cooling sites in the National Solar Data Network (NSDN). Results presented have been developed on the basis of analysis of instrumented sites monitored through the 1982 cooling season.

NATIONAL SOLAR DATA NETWORK (NSDN)

The United States has set a goal of achieving 20% of its energy needs through solar energy technologies by the year 2000. The National Solar Heating and Cooling Demonstration Act of 1974 represents a major mechanism for implementation of solar energy goals.

The National Solar Heating and Cooling Demonstration Program was established by this act for the collection and evaluation of solar information, and its dissemination to all potential users. To ensure that all related activities are conducted uniformly, the National Solar Data Program, including the National Solar Data Network, was established.

Approximately 5,000 residential and commercial solar sites were established since the inception of the National Solar Heating and Cooling Demonstration Program. In 1982, forty of these sites were instrumented and included in the NSDN.

Recent changes in the Department of Energy (DOE) solar energy program have focused the National Solar Data Network on research and development activities of solar cooling sites. Privately funded systems are included in the NSDN, although all of the cooling systems analyzed in this report had some government funding. This report describes the performance of five cooling systems, and compares the performance of the various subsystems at these space cooling sites.

The NSDN sites selected by DOE include a broad range of solar system types and geographical locations within the United States. Figure 1 shows the location of NSDN sites with solar cooling systems having measured performance during the 1982 cooling season. Sensors are sampled automatically, and the data is stored at each site for one or more days (Figure 2). Since December 1979, the data has been transmitted over telephone lines to a central computer at Vitro Laboratories in Silver Spring, Maryland, where data reduction and analysis take place. Thermal performance of each site is analyzed and the results are recorded on a monthly basis. Performance over longer time periods is presented in Solar Energy System Performance Evaluation reports.

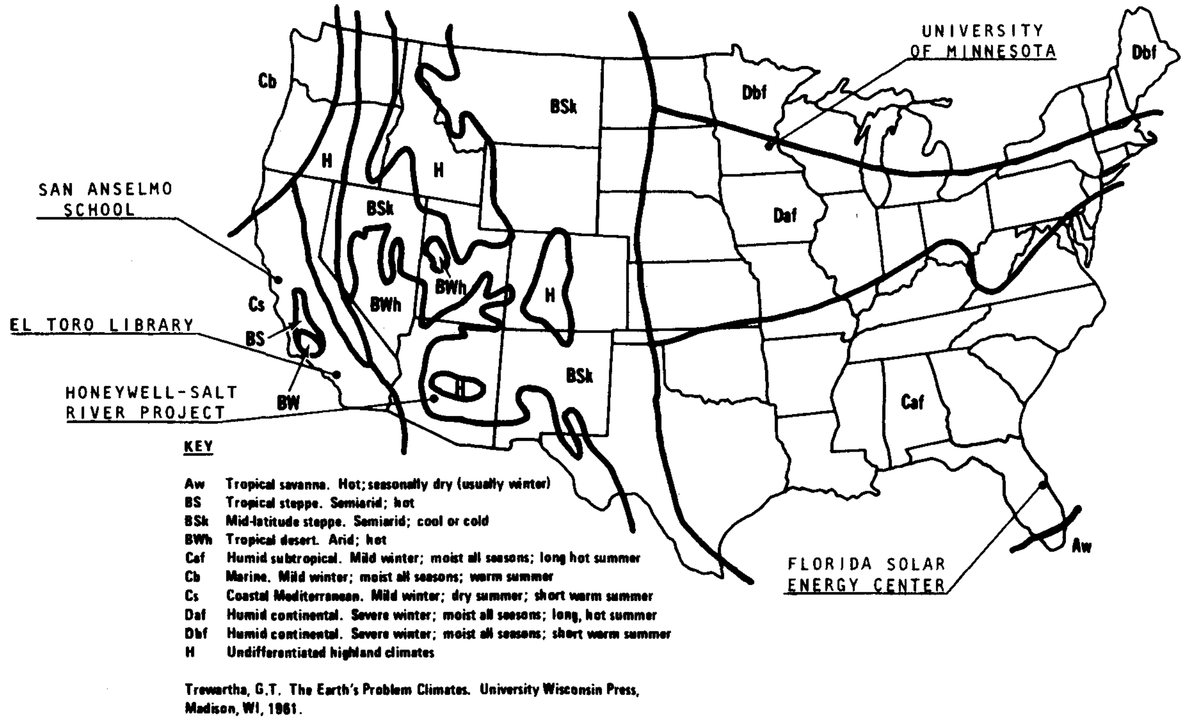

Figure 1. Climatological Map of the United States Showing Sites Discussed in this Report

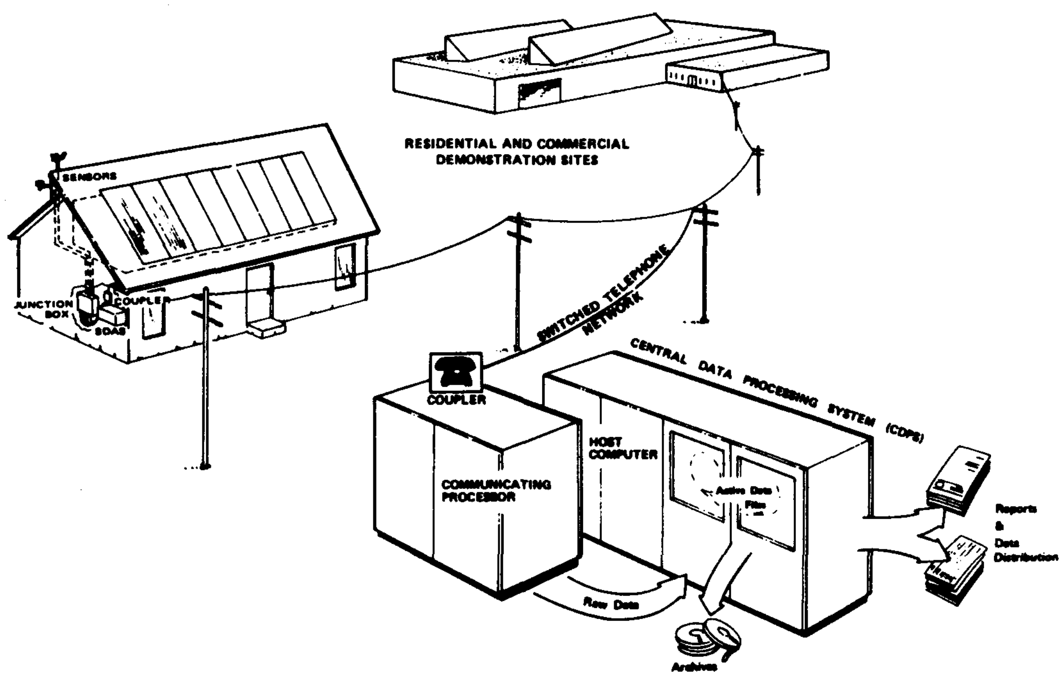

Figure 2. The National Solar Data Network

REPORT OBJECTIVES

Solar cooling would appear to be the ideal way to use solar energy because both cooling loads and solar insolation peak at about the same time of day. This report will present data to compare five solar cooling systems within the National Solar Data Network (NSDN). The report will define the critical parameters which affect solar cooling systems. The effect of climate will also be investigated. Finally, the current energy savings will be shown.

The NSDN is a primary vehicle for the Federal Government to track the performance of solar cooling systems in the program. The detailed measurement for these systems was analyzed in accordance with standardized procedures and is presented on the basis of hourly, daily, monthly, and seasonal performance factors. Millions of individual measurements were collected and reduced, providing a large reservoir of data for operational and comparative analysis.

Parameters and performance indices presented include overall system delivered loads, solar fraction of the load, energy savings, coefficient of performance, energy collected, and various subsystem efficiencies. The comparison of these factors has allowed evaluation of the relative performance of various systems. A matrix of performance indices has been constructed to facilitate comparison of the representative solar cooling installations.

OVERVIEW OF SPACE COOLING ANALYSIS CONCEPTS

Analysis of space cooling requires a general philosophy which can be applied to all systems to assure commonality and comparability of results. Within the NSDN, such a philosophy with attendant methodology has been developed consistent with National Bureau of Standards documentation, NBSIR 76-1137 (Reference 1), and the results presented reflect that philosophy.

Initial NSDN analysis concentrated on analysis of energy gains and losses associated with individual equipment and subsystems. This technique has been extended to analysis of the interfaces between subsystems to permit better understanding of overall system operation, energy flow, and energy uses. Further analysis of the entire building envelope is required to quantify the internal thermal "losses" and the passive solar component. However, the measured cooling load does contain the sum of all building heat gains.

Embodied in the NSDN methodology employed during the 1982 cooling season are the concepts of both equipment load (energy gains) and thermal energy flow analysis.

EQUIPMENT LOAD (ENERGY GAINS). The equipment load or energy gains method is characterized by the measurement of gains from the space cooling equipment which are a function of the building cooling demands not satisfied by other sources. One other source could be passive cooling, which results from building losses during cool nights or cool cloudy days. Figure 3 diagrams the major energy flows for a typical space cooling system.

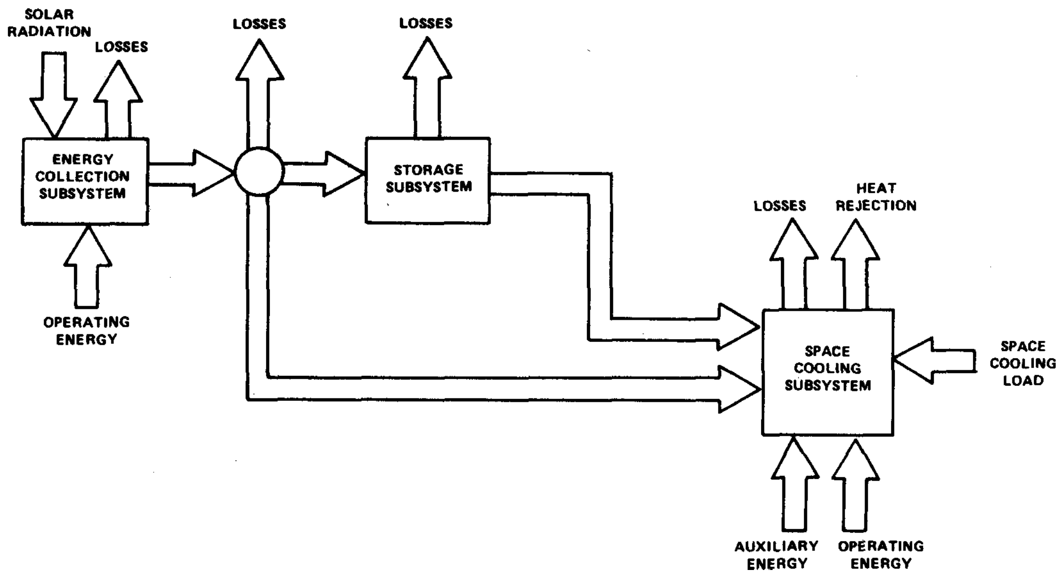

Figure 3. Typical Space Cooling Energy Flow Diagram

ENERGY FLOW. Thermal energy flow analysis requires definition of the boundary surrounding the physical structure of the system to be analyzed and the major components within that boundary. Energy flows across the boundary and between components are measured and analyzed. Performance factors are constructed from the energy flows to assess the solar system's thermal effectiveness. This type of analysis depends upon the amount of solar radiation, energy flow between subsystems, auxiliary and operating energies, load requirements and losses as shown in the flow diagram Figure 3. Appendix A contains actual energy flow diagrams for the cooling season for the five sites.

Monthly performance factors calculated for NSDN sites include:

- System level performance:
  - Thermal performance of the system
  - Solar fraction
  - Total energy consumed
  - Total energy saved
- Subsystem level performance:
  - Thermal performance of each subsystem
  - Energy Collection and Storage Subsystem (ECSS) solar conversion efficiency

- Solar fractions
- Energy consumed, energy saved

Solar system measured data consists of:

- Temperatures in and out of each subsystem
- Flow in each subsystem
- Auxiliary power used via wattmeters or flow meters
- System state (i.e., on/off, etc.)

Weather data consists of:

- Insolation, in the plane of collector (all sites)
- Ambient temperature (outdoor, all sites)
- Wind speed and direction (some sites)
- Relative humidity (some sites)

SOLAR COOLING WITH ABSORPTION CHILLERS

At the present time, most commercially available solar cooling technology utilizes a system of solar collectors which collect solar energy to be used in an absorption chiller. The absorption chiller can produce cooling using physical and chemical reactions driven by heat, in this instance, hot water from solar collectors.

Absorption chillers utilize a partial vacuum to permit concentration of a refrigerant and regeneration of an absorbent having a high affinity for the refrigerant. Most absorption chillers deployed in solar applications use a lithium bromide solution as the absorbent and water as the refrigerant. Figure 4 and the following explanation of an absorption chiller have been provided by the courtesy of the Arkla Company[1]. Figure 5 shows one of these machines installed at the El Toro Library.

The Arkla SOLAIRE WFB-300 Water Chiller operates on the absorption principle. Solar heated water is the energy source, circulating in a closed loop between the unit's generator and the solar collectors. In a second closed loop, the refrigeration tonnage is delivered by chilled water which circulates between the unit's evaporator coil and a refrigeration load. In a third water circuit, condensing water flows through the unit's absorber and condenser coils, carrying away the waste heat.

To begin the cooling cycle, solar heated water enters the generator tubes. Its heat vaporizes part of the water refrigerant in solution, separating it from the lithium bromide absorbent.

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[1] The use of this material does not constitute an endorsement of Arkla equipment by either Vitro Laboratories or the Federal Government.

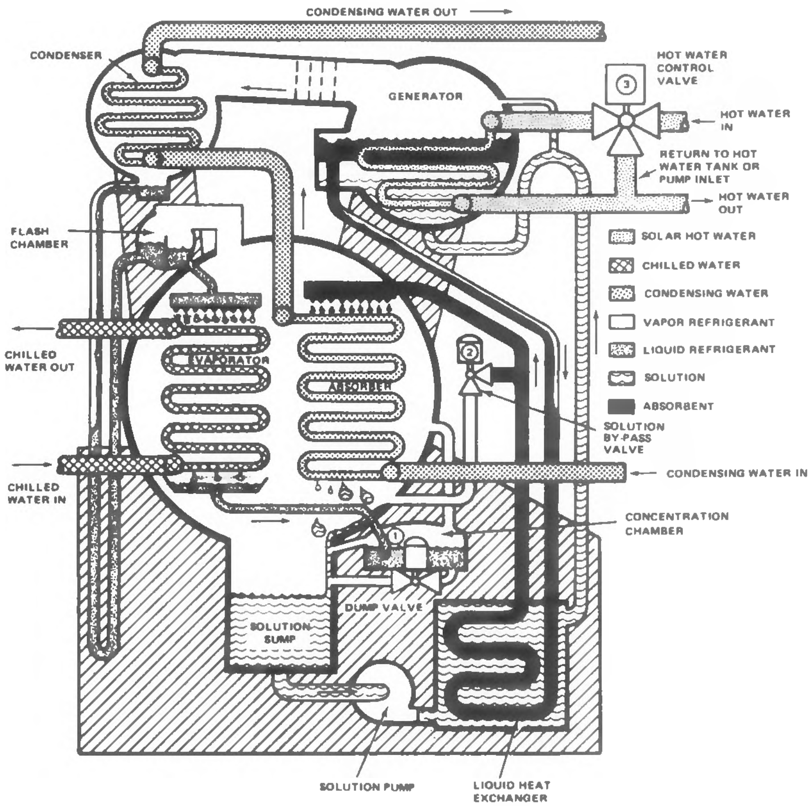

Figure 4. Arkla SOLAIRE WFB-300 Water Chiller Diagram

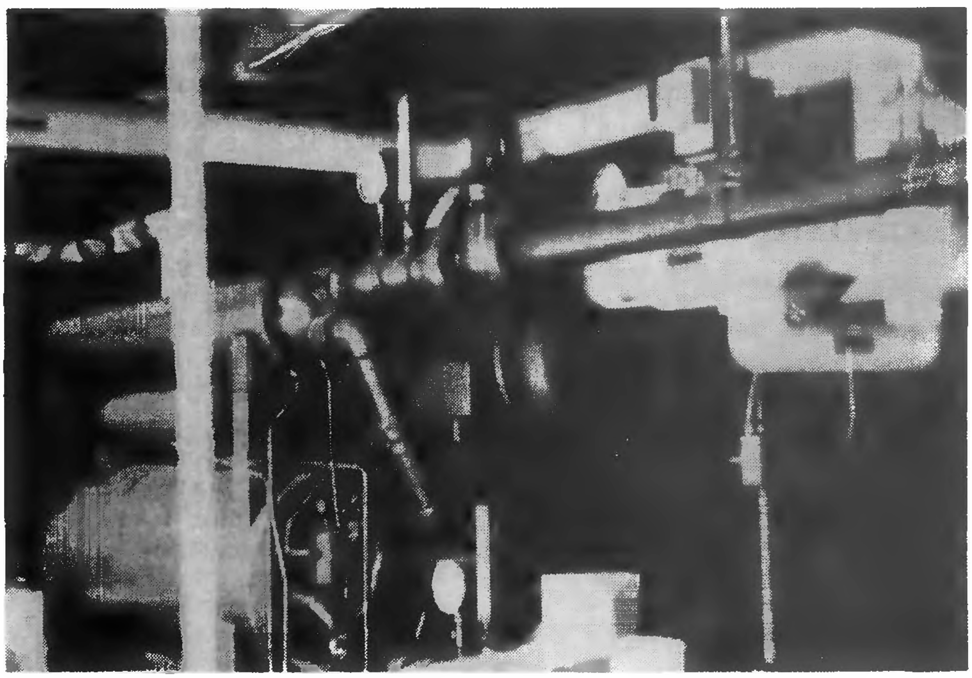

Figure 5. Arkla WFB-300 Water Chiller Installed at the El Toro Library

The vaporized refrigerant passes to the condenser where it gives up its latent heat to the condensing water and is liquefied. The refrigerant then flows through a "U" tube and through the flash chamber into the evaporator, wetting the outer surfaces of the evaporator coil tubes. There, the refrigerant is again vaporized as it absorbs the heat of the refrigeration load from the chilled water. The vapor then flows to the absorber where it is again liquefied as it combines with the absorbent in the process that gives this thermodynamic cycle its name.

The hot absorbent passes from the generator to the liquid heat exchanger where it gives up some of its heat. The precooled absorbent then enters the absorber where it wets the outer surfaces of the absorber coil tubes and combines with the vapor refrigerant. The absorbent then gives up the remainder of its heat to the condensing water flowing inside the absorber coil tubes.

After the absorption process, the reunited refrigerant absorbent solution drains into the solution sump. From there, the solution flows through the liquid heat exchanger, is preheated, and then flows to the generator to repeat the cycle.

Figure 6 shows a generalized solar energy cooling system schematic. Solar cooling sites in the NSDN use either concentrating collectors, flat-plate collectors, or evacuated-tube collectors. Storage tanks are optional at the discretion of the designer and the particular requirements of the site. Some form of auxiliary backup cooling system is usually needed. This can be a boiler to run the absorption chiller when solar energy is not available, auxiliary chilled water from a central plant, or auxiliary air conditioners connected directly to the load.

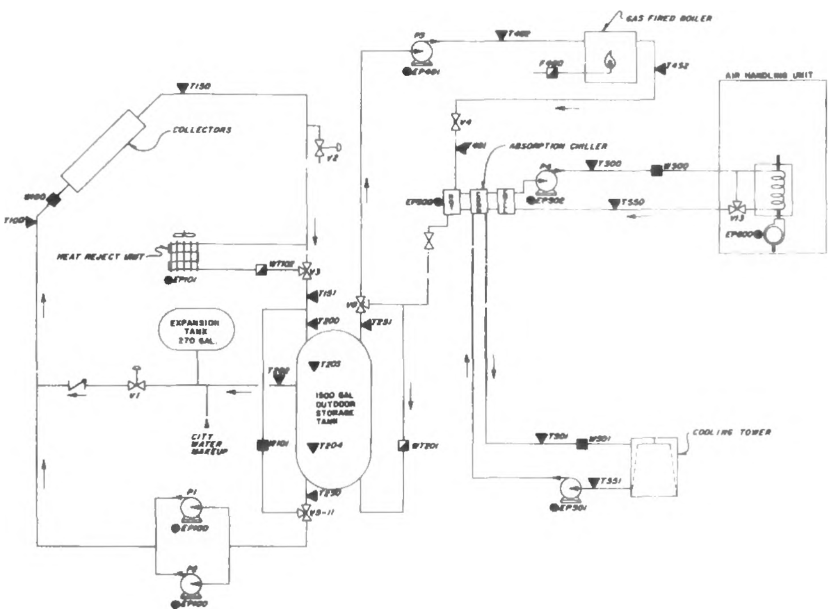

Figure 6. Generalized Solar Energy Cooling System Schematic

Schematics, which identify the system components and interconnections, are included in Appendix A.

SOLAR COOLING WITH RANKINE/CHILLERS

A solar Rankine cooling system presents an alternative to solar cooling using an absorption chiller. A Rankine cooling system incorporates a solar-driven Rankine engine to provide mechanical power to a conventional vapor compression chiller, see Figure 7. Coupled between the engine and compressor is a motor/generator. This is used to drive the vapor compressor when there is insufficient solar energy. The Rankine engine output and auxiliary power from the motor are summed to provide sufficient power to the vapor compressor. The generator may also generate electricity when the Rankine engine provides more power than required to drive the vapor compressor.

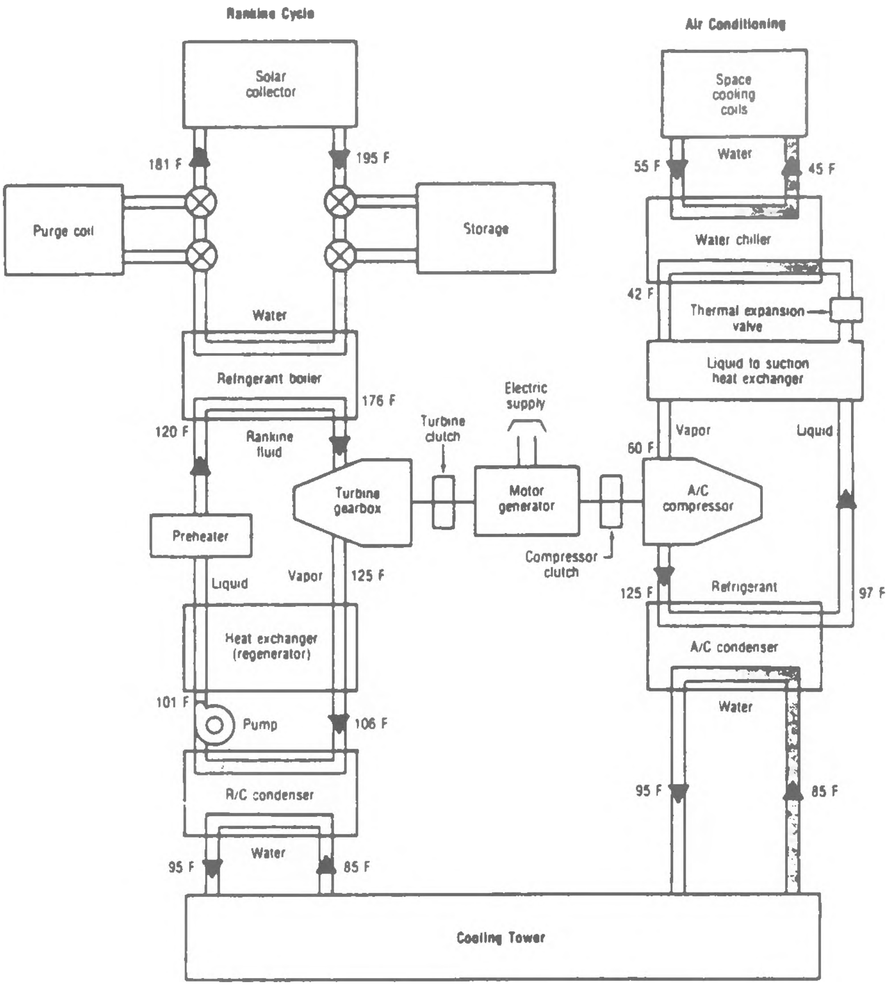

Figure 7. Solar-Powered 25-Ton AC - 20-HP RC Cooling Unit

The Rankine engine operation is similar to that of the steam turbines that have been used for years by the power companies to generate electricity.

Hot water from the solar collectors is pumped through the refrigerant boiler to create vapor. This vapor then passes through turbine nozzles where its speed is increased to drive a turbine rotor.

The refrigerant vapor is exhausted from the turbine through a regenerator (an efficiency improving heat exchanger) to a water cooled condenser where it changes to liquid and is returned by a pump to the boiler via the regenerator.

The heat transfer from vapor to liquid in the regenerator enables the vapor to reach a point just below its condensing temperature. At the same time, the liquid temperature is raised to just below its boiling point.

Cooling is provided through a standard, vapor compression, air conditioning cycle which uses R-12 refrigerant, an "open" type compressor, a water chilled evaporator, and a water cooled condenser.

A single cooling tower that serves as both the Rankine cycle and air conditioning cycle heat rejector and a solar storage tank completes the system. A Rankine engine with the motor generator and compressor is shown in Figure 8.

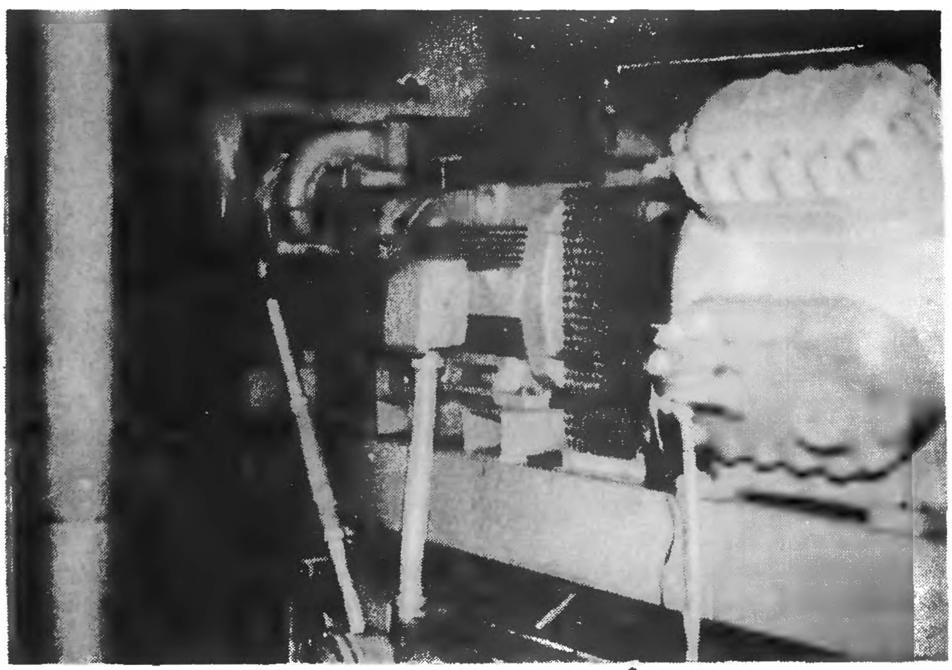

Figure 8. Rankine Engine - Motor/ Generator-Compressor

REPORT ORGANIZATION

Sections of this report have been organized to permit the reader to examine areas of special interest as well as to highlight general results. Section II contains tables and discussions of individual site parameters such as collector areas, storage tank sizes, manufacturers, building dimensions, etc. In addition, brief site descriptions are provided in this section. Summaries of the various types of sites are also provided.

Section III provides a summary of comparative data and analysis of NSDN solar cooling systems operational for the 1982 cooling season, with discussions of specific performance results. In addition, analysis results are presented in tables and graphic form to highlight key information.

Section IV presents comparative results and predictions based on the data and analysis from the previous section.

Section V provides a list of references used.

Specific detailed data and information necessary to support the development of results presented are contained in appendices to this document.

Section II

OVERVIEW OF SITES

A prerequisite to understanding the comparative results of solar systems discussed in this report is some knowledge of the solar systems upon which the report is based. This section presents summary-level descriptions of the sites for which monthly performance analysis was done. Tables 1A and 1B present a summary of site information which identifies the key equipment at each site.

In-depth system descriptions, energy flow diagrams, and seasonal performance data are provided in Appendix A. Hourly profiles of cooling loads are shown in Appendix C.

There are five solar systems monitored by the NSDN and included in this analysis which produced data sufficient for performance analysis.

Table 1A. SITE CHARACTERISTICS (DATA)

SITE NAME SITE LOCATION	COLLECTOR TYPE STORAGE TYPE	BUILDING TYPE	COOLING EQUIPMENT
El Toro Library El Toro, California	1,427 Ft^2 Evacuated-Tube Collectors 1,500-Gallon Hot Storage Tank	10,000 Ft^2 Library	Arkla 25-Ton Solar Absorption Chiller
Florida Solar Energy Center Cape Canaveral, Florida	2,089 Ft^2 Evacuated-Tube Collectors 3,000-Gallon Hot Storage Tank	5,000 Ft^2 Brick Research Facility Office Building	Arkla 25-Ton Solar Absorption Chiller Trane Vapor Compression Unit 6,000-Gallon Solar Cold Storage Tank
Honeywell-Salt River Proj. Phoenix, Arizona	8,208 Ft^2 Flat-Plate Collectors 2,500-Gallon Hot Storage Tank	55,000 Ft^2 Conditioned Cooling Space	Two 25-Ton Rankine Chillers 228-Ton Centrifugal Chiller
San Anselmo School San Jose, California	3,740 Ft^2 Evacuated-Tube Collectors 2,175-Gallon Hot Storage Tank	34,000 Ft^2 School Building	Arkla 25-Ton Solar Absorption Chiller Four Arkla 25-Ton Gas-Fired Absorption Chillers
University of Minnesota Minneapolis, Minnesota	6,350 Ft^2 Slat-Type Concentrating Collectors 8,000-Gallon Hot Storage Tank	84,000 Ft^2 Underground University Facilities	Trane 147-Ton Absorption Chiller

Table 1B. SITE CHARACTERISTICS (DESCRIPTIONS)

SITE NAME SITE LOCATION	COMMENTS	KNOWN SYSTEM ANOMALIES
El Toro Library El Toro, California	Arkla chiller operates by a combination of solar and auxiliary energy piped in series.	Collector area undersized to meet design performance. Low chiller COP through July.
Florida Solar Energy Center Cape Canaveral, Florida	Arkla chiller runs from hot storage. Cold storage utilized.	Low chiller COP. Poor cold storage utilization. Costs more to operate than cooling produced
Honeywell-Salt River Proj. Phoenix, Arizona	Employs two 25-ton Rankine chillers. Rankine engines can operate directly from collector loop. Rankine engine can supply electrical power regeneration when there is no space cooling. Exhibits low solar-specific operating energy	Lubrication problem with Rankine #1.
San Anselmo School San Anselmo, California	Arkla chiller runs from hot storage. Building has no exposed windows, and was designed for low heat loss/gain.	Severe control problems with the collector to storage control. Control logic on chillers prevents individual chiller use. Poor auxiliary and solar chiller COP.
University of Minnesota Minneapolis, Minnesota	Trane chiller operates by a combination of solar and auxiliary energy.	Poor collector focusing. Poor chiller COP.

A. SYSTEM DESCRIPTIONS

The El Toro Library solar energy system is a 10,000-square-foot library that incorporates 1,427 square feet (82 panels) of evacuated-tube collectors to provide hot water to a 1,500-gallon, insulated, outdoor storage tank. Solar energy is utilized from storage when the storage tank is warmer than the load return loop temperature. A control valve regulates the flow of solar energy from storage to the loads. A gas-fired boiler was installed in series with the storage tank to provide constant supply temperatures to the generator inlet of the 25-ton Arkla absorption chiller or to the space heating coils in the Air-Handling Unit (AHU). The chiller provides chilled water to the air-handling unit coils to satisfy the space cooling demand for the library.

This type of design, with the boiler in series with the solar storage tank, minimizes the solar-specific operating energy and preheats the return water prior to the boiler.

The solar energy system at the Florida Solar Energy Center is composed of 2,089 square feet of evacuated-tube collectors, a 3,000-gallon hot storage tank, a 6,000-gallon cold storage tank, a 25-ton absorption chiller, pumps, connecting pipe lines, and controls to operate the various system components. The existing cooling tower, electric chiller, oil-fired burner, and air-handling unit were unaltered except for control configuration.

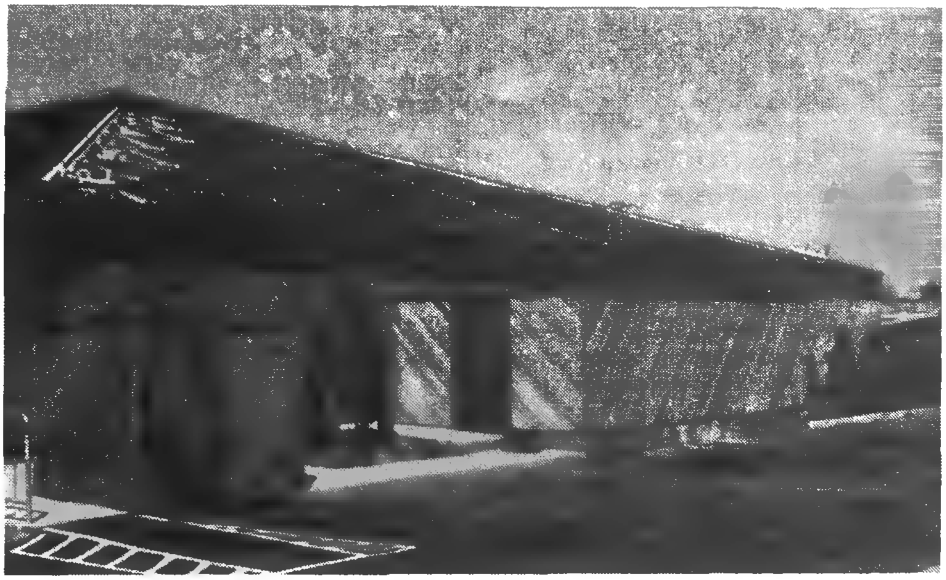

Figure 9. The El Toro Library Collector Array

In the space cooling mode, solar energy is delivered from the storage tank to the absorption chiller, where chilled water is produced via the absorption process. Chilled water can be used directly to meet the space cooling load, or stored in an outdoor cold storage tank for later use. An auxiliary electric chiller will meet the space cooling load when solar energy is depleted. During operation of the space heating mode, solar energy is delivered directly from the storage tank to help meet the space heating load. An auxiliary oil-fired boiler is used for a backup.

Figure 10. The Florida Solar Energy Center Collector Array

The Honeywell-Salt River Project solar system is a 55,000-square-foot building that utilizes 8,208 square feet (456 panels) of flat-plate collectors. A 20% ethylene glycol/water solution is used as a heat transfer fluid. This heat transfer fluid is pumped to a 2,500-gallon, insulated, indoor storage tank or directly to the Rankine engines. In the space heating mode, solar energy is delivered from the storage tank to the space heating coils in the conditioned space. If solar energy is unable to meet the space heating demand, then the auxiliary electric radiant heaters will satisfy the heating load. In the cooling mode, solar energy is delivered directly from the collectors to the Rankine engines where solar energy can be utilized for space cooling or electrical power generation. If solar energy is insufficient, then two 25-ton vapor compression chillers are supplied with auxiliary energy or a 228-ton centrifugal chiller will satisfy the space cooling load.

This solar system minimizes the solar-specific operating energy costs and provides solar energy directly to the Rankine engines from the solar collectors.

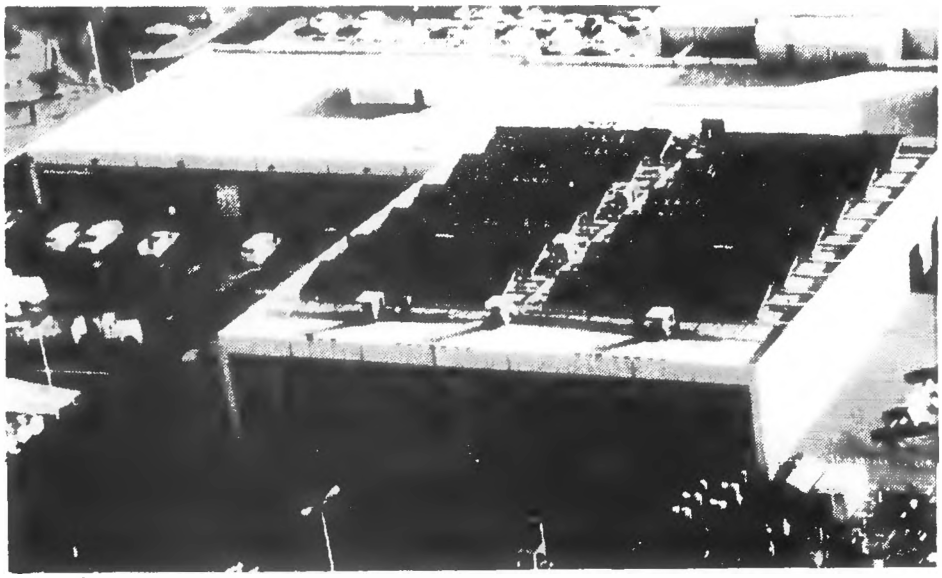

Figure 11. The Honeywell-Salt River Project Collector Array

The San Anselmo School solar system is a 34,000-square-foot school building. There are 3,740 square feet (204 panels) of evacuated-tube collectors which deliver hot water to a 2,175-gallon, insulated, outdoor storage tank. A control valve regulates the flow of collected energy to the storage tank. During the heating season, solar energy from storage is delivered directly to the coils in the air-handling units. If solar energy is unable to meet the demand, then two auxiliary gas-fired chiller/heaters will satisfy the remaining load. In the space cooling mode, hot water from storage is supplied to the generator inlet of a 25-ton Arkla WFB-300 absorption chiller. If the absorption chiller cannot supply the load, then four auxiliary gas-fired absorption chillers will meet the space cooling load.

This solar system uses a solar-unique absorption chiller to supply part of the space cooling load. There is no direct connection from the collector loop to the chiller in this design.

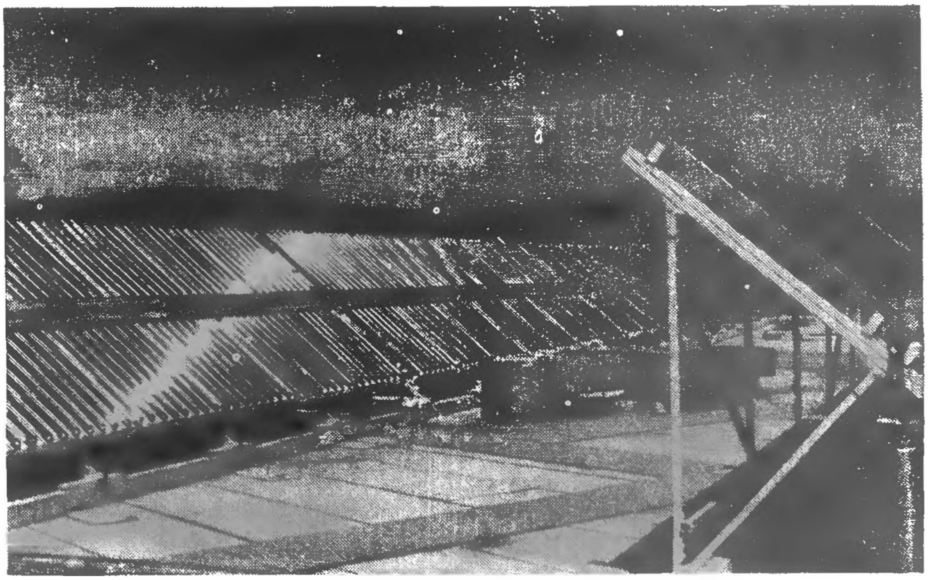

Figure 12. The San Anselmo School Collector Array

The University of Minnesota solar system supplies space heating and cooling to an 84,000-square-foot underground building. There are 6,350 square feet of concentrating slat-type collectors which concentrate solar energy onto a fixed receiver. A 38% ethylene glycol/water solution (by weight) is used for the heat transfer fluid. A heat exchanger in the collector loop separates the ethylene glycol solution from the water used in the other subsystems. Solar energy can be delivered from storage to space heating and to space cooling. In the space heating mode, solar energy from the collector loop heat exchanger or from storage can be utilized for the space heating demand. Auxiliary steam supplied through a heat exchanger will meet the load if solar energy is unable to satisfy the demand. In the cooling mode, solar energy can be utilized from an 8,000-gallon insulated buried storage tank or from the collector loop heat exchangers to the generator inlet of a 147-ton Trane absorption chiller. The auxiliary steam is provided by the heat exchanger in series with the solar energy supply if solar cannot meet the space cooling demand.

This solar system is similar to that at the El Toro Library since the auxiliary system is connected in series with the solar supply. Several more pumps and three-way control valves are used at the University of Minnesota, which complicates the control configuration.

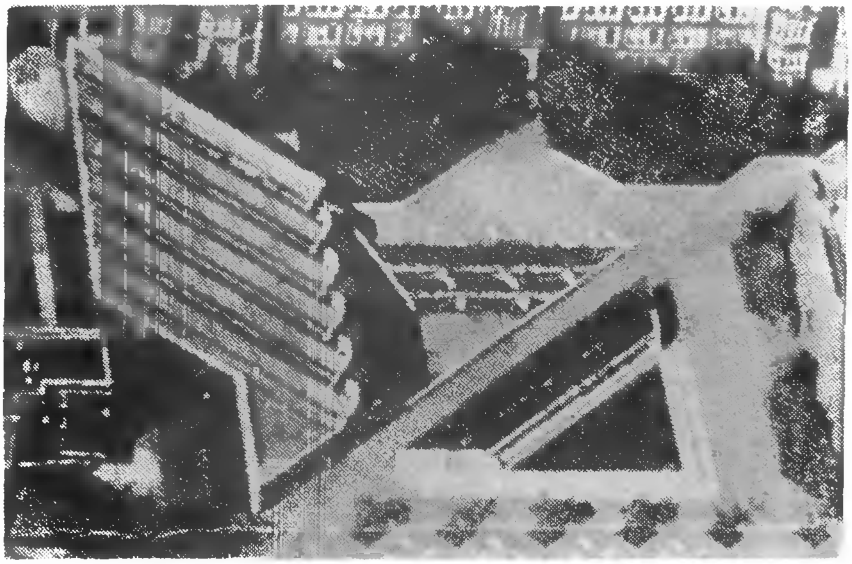

Figure 13. The University of Minnesota Collector Array

For a more in-depth understanding of each solar system, refer to Appendix A. Appendix A also contains a listing of the manufacturers of various components of the space cooling systems described in this report.

B. SUMMARY

All the space cooling solar systems discussed in this report are commercial installations. Various collection devices are used to collect solar energy. The El Toro Library, the Florida Solar Energy Center, and the San Anselmo School use evacuated-tube collectors while the Honeywell-Salt River Project has flat-plate collectors. The University of Minnesota has a tracking slat-type reflecting concentrator.

All solar sites utilize a hot storage tank, but at the Honeywell-Salt River Project the storage tank is used for space heating only, whereas all the other sites use the hot storage tank for both space heating and space cooling. The Florida Solar Energy Center incorporates a cold storage tank.

The El Toro Library and the University of Minnesota solar sites are similar in that both systems incorporate an auxiliary boiler in series with the solar heated generator inlet water. In effect, solar energy acts as a preheater to the boiler which raises the temperature to the set point of the generator inlet to the chillers. The University of Minnesota design permits the use of solar energy directly from the collectors, whereas the El Toro Library must use solar energy from storage.

The San Anselmo School and the Florida Solar Energy Center supply the generator inlet of the absorption chiller from the hot storage tank. However, the Florida Solar Energy Center also utilizes a cold storage tank, and has the ability to reject excess collected energy from the hot tank to the cold tank.

The Honeywell-Salt River Project solar site is the only space cooling site that incorporates a Rankine engine. Solar energy is utilized to drive the Rankine engine which powers the vapor compression chiller and an electrical generator. Auxiliary electrical energy is used to operate the vapor compressor if the Rankine engine is unable to power the compressor.

All solar sites supply an absorption chiller except for the Honeywell-Salt River Project which uses a Rankine engine to run a vapor compression chiller. Also, all sites are located in a moderate climate except for the University of Minnesota.

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SECTION III

COMPARATIVE DATA AND PERFORMANCE ANALYSIS

This section contains the comparative results and analysis of thermal performance data collected from the five cooling sites studied during 1982. Each subsection describes a specific topic related to the analysis and comparison of the solar cooling system performance at the five sites.

A. SOLAR COOLING PERFORMANCE

There are numerous ways of comparing the overall performance of solar cooling sites. In this section, we will discuss the overall levels of thermal performance based on design vs. actual performance, system Coefficient of Performance (COP), percentage of incident radiation utilized, and overall system losses.

The 1982 solar cooling system performance is shown in Table 2. This table is arranged to show the system total in the left column and the cooling subsystem value in the right column. In the columns marked "Load," the reader can easily verify that the cooling load was the major load at all five sites. The remainder of the system load was due to space heating. This same pattern follows for each of the subheadings in Table 2. Note that on several of these sites the sum of Solar Energy Used and Auxiliary Thermal Energy exceeds the cooling load. This excess is due to the thermodynamic conversion process which converts heat energy into cooling. Under the subheading of Solar Energy Used, the value for cooling at the El Toro Library is larger than the total. This value was due to some energy from the auxiliary space heating subsystem being returned to solar storage. Cooling solar fractions ranged from four percent to 24%.

Table 2. SPACE COOLING SYSTEM PERFORMANCE

(All values in million BTU, unless otherwise indicated)

SITE	LOAD		SOLAR ENERGY USED		AUXILIARY THERMAL ENERGY		OPERATING ENERGY		SOLAR FRACTION (%)	
	TOTAL (SYSL)	COOLING (CL)	TOTAL (SEL)	COOLING (CSE)	TOTAL (AXT)	COOLING (CAT)	TOTAL (SYSOPE)	COOLING (COPE)	TOTAL (SFR)	COOLING (CSFR)
El Toro Library	220	208	122	126	413	389	93.2	85.6	24	24
Florida Solar Energy Center	244	234	286	277	62.3	61.2	44.3	35.8	. 17	14
Honeywell-Salt River Project	2,670	2,590	805	735	825	792	418	406	8	7
San Anselmo School	117	104	67.9	58.1	322	314	64.1	60.6	19	12
University of Minnesota	825	693	92.7	82.0	2,010	1,890	387	286	5	4

For a description of acronyms in parentheses, refer to Appendix E.

During 1982, none of the five systems achieved design levels of thermal performance. Figure 14 shows a bar chart of design and actual solar cooling fractions. The best sites were the El Toro Library and the Honeywell-Salt River Project which had solar fractions roughly one-half of the design levels.

The rest of the systems showed performances which were much less than expected. The University of Minnesota showed only one-tenth of the expected solar contribution, at four percent vs. an expected cooling solar contribution of 40%.

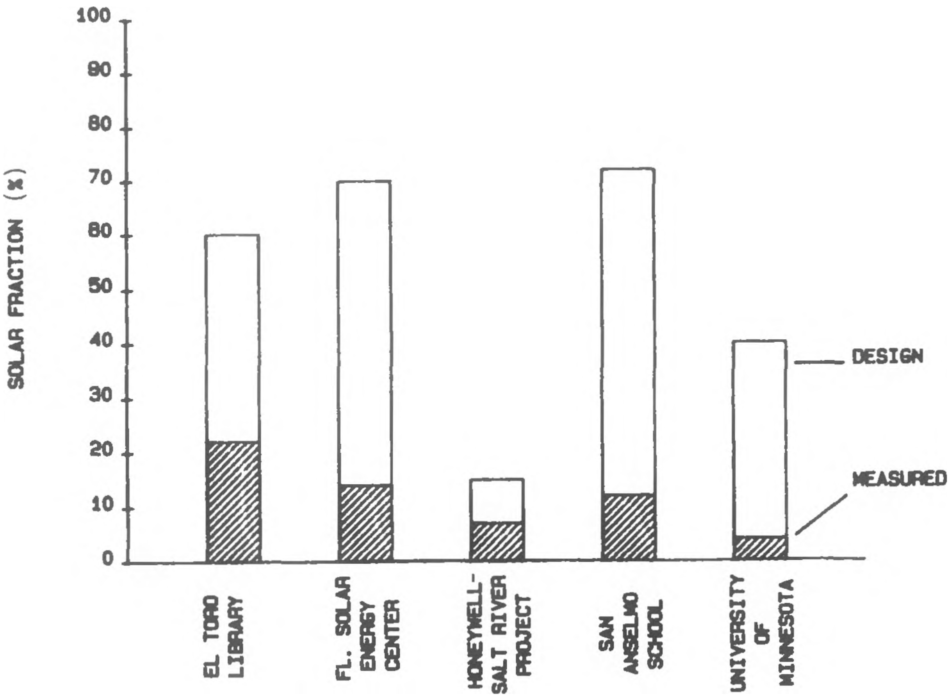

Figure 14. Measured Solar Cooling Fractions and Design Annual Cooling Solar Fractions

Part of the failure to meet design cooling solar fractions is due to poor performance of subsystem components. Poor performance is particularly true of the absorption chillers, since the design COP is often 0.55 to 0.6 and the COP of the absorption chillers in these systems ranged from 0.22 to 0.45. Also, solar insolation was less than the long-term solar insolation, which solar designers frequently use. For instance, at the El Toro Library, the measured insolation was 17% less than the long-term. Design solar fraction is also influenced by the size of cooling load. During the 1982 cooling season, there were more cooling degree-days than expected, thus loads were larger than normal.

One way to determine the reasonableness of the design cooling solar fraction is to take the measured cooling load and determine the collector efficiency required to meet design. Table 3 shows the collector efficiencies that would be required to meet the design solar fractions.

Table 3. ESTIMATED AND ACTUAL COLLECTOR ARRAY EFFICIENCIES

(All values in percent)

SITE	ESTIMATED DESIGN COLLECTOR EFFICIENCY	ACTUAL COLLECTOR EFFICIENCY	HIGHEST SINGLE-MONTH EFFICIENCY
El Toro Library	44	31*	36
Florida Solar Energy Center	23	29	32
Honeywell-Salt River Project	50	31	37
San Anselmo School	27	21	23
University of Minnesota	37	11	15

* Eight months of data

From the table the reader can see that only the Florida Solar Energy Center was designed with enough collector area to provide the design cooling solar fraction. Of the other systems, only the San Anselmo School has a low enough estimated design collector efficiency to permit a more efficient collector array to provide the design cooling solar fraction. The other three systems are undersized for the design cooling loads. These systems would require 1.4 times the area for the El Toro Library, 2.5 times the area for the University of Minnesota, and 1.6 times the area for the Honeywell-Salt River Project.

The estimated design collector efficiencies in Table 3 were calculated from the measured cooling loads, assuming 10% system losses and an absorption chiller COP of 0.55. For the Rankine system, the assumptions were 10% system losses, a COP of four on the vapor compressor, and eight percent thermal efficiency for the Rankine engine. These assumptions may not represent design values, but do represent at least a single-month average performance level of the system components.

Data on the solar cooling fraction as a function of time of day (see Appendix C) is plotted in Figure 15. Months with larger cooling loads and good chiller COPs were chosen. The data indicates how well the systems performed.

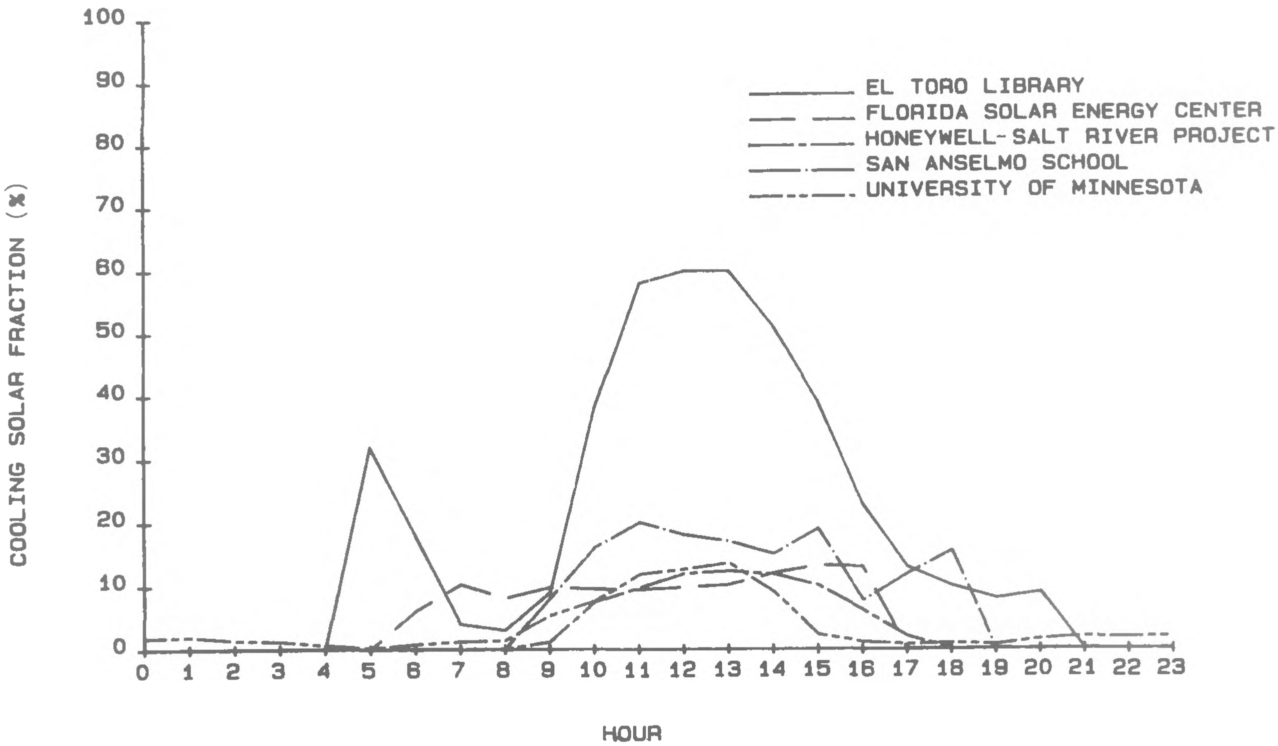

Figure 15. Cooling Solar Fraction
vs. Time of Day

The cooling subsystem at the El Toro Library ran between 5:00 a.m. and 8:00 p.m. during July. The range of cooling solar fraction was three percent to 60%, and from 10:00 a.m. until 4:00 p.m. the solar contribution was above 35%. Between 11:00 a.m. and 2:00 p.m., the solar system carried more than 50% of the load. Chiller COP during these hours was nearly constant, ranging from 0.46 to 0.48. The maximum COP of 0.51 occurred between 4:00 p.m. and 5:00 p.m. when the solar contribution had decreased to 23%. Apparently, the larger fraction of solar energy reduced the chiller COP, perhaps due to lower generator temperatures. The difference in COP between the maximum solar contribution and the best boiler-fired chiller COP was 0.03. This difference would amount to about one-half ton of cooling.

At the Florida Solar Energy Center, the solar cooling subsystem ran from 6:00 a.m. until 5:00 p.m. during July. The cooling solar fraction ranged from six percent to 13% with the maximum at 3:00 p.m. There were four hours of operation when the cooling solar fraction exceeded 10%. The average monthly cooling solar fraction was 10%. Chiller COP was a maximum of 0.27 at 2:00 p.m.

At the Honeywell-Salt River Project, there was a cooling load the entire day during August. However, the solar system only provided cooling between 9:00 a.m. and 5:00 p.m. This was also the time of the most significant building loads. Cooling solar fraction ranged from one percent to 12%, with the 12% solar fraction being maintained from noon to 3:00 p.m. during this time. The Rankine engine efficiency was seven percent. This is typical thermal efficiency for these Rankine engines.

At the San Anselmo School, the cooling solar fraction ranged from eight percent to 20% between 9:00 a.m. and 6:00 p.m. during May. For reference, the significant building cooling loads occurred between 9:00 a.m. and 3:00 p.m., and during this time the cooling solar fraction did not drop below 15%. The solar chiller COP was about 0.3 during the highest cooling loads.

Williamson Hall at the University of Minnesota had a cooling load throughout the day during July. The peak loads occurred between 8:00 a.m. and 5:00 p.m. The solar fraction ranged from one percent to 14% between 8:00 a.m. and 5:00 p.m. The chiller COP peaked to 0.4 at noon.

The dominant load at each of the solar sites is the cooling load. The University of Minnesota site is the only one of the five which showed significant heat loads, primarily due to the northern location. The remainder of the sites are located in areas of the United States which may be considered "the Sun Belt," and cooling loads are the main energy consumers in these buildings. These cooling loads will be discussed in Section III.E.

Another way of looking at the performance of these solar systems is the overall system efficiency, defined as the percentage of the incident solar radiation utilized, which is shown in Figure 16.

The bars in Figure 16 represent the percentages of total insolation transferred through common points in the systems, from incident energy, collected energy, energy to storage, energy from storage, cooling solar energy supplied to the load, and, finally, equivalent cooling load supplied by solar energy.

The energy drop at each stage of the energy transfer through the systems represents losses and inefficiencies, from incident energy to actual cooling production resulting from application of solar energy. Solar energy used for space heating and hot water heating has been subtracted out to permit comparison of the cooling subsystems only.

The El Toro Library showed the highest overall cooling efficiency, at 10%. The El Toro Library showed good solar collection efficiency (31%), reasonable storage and transport losses (although still higher than other sites), and a fair chiller COP, resulting in the highest overall thermal efficiency. If the storage/transport losses were improved, along with improvements in the chiller COP, then overall performance measured by cooling efficiency would have

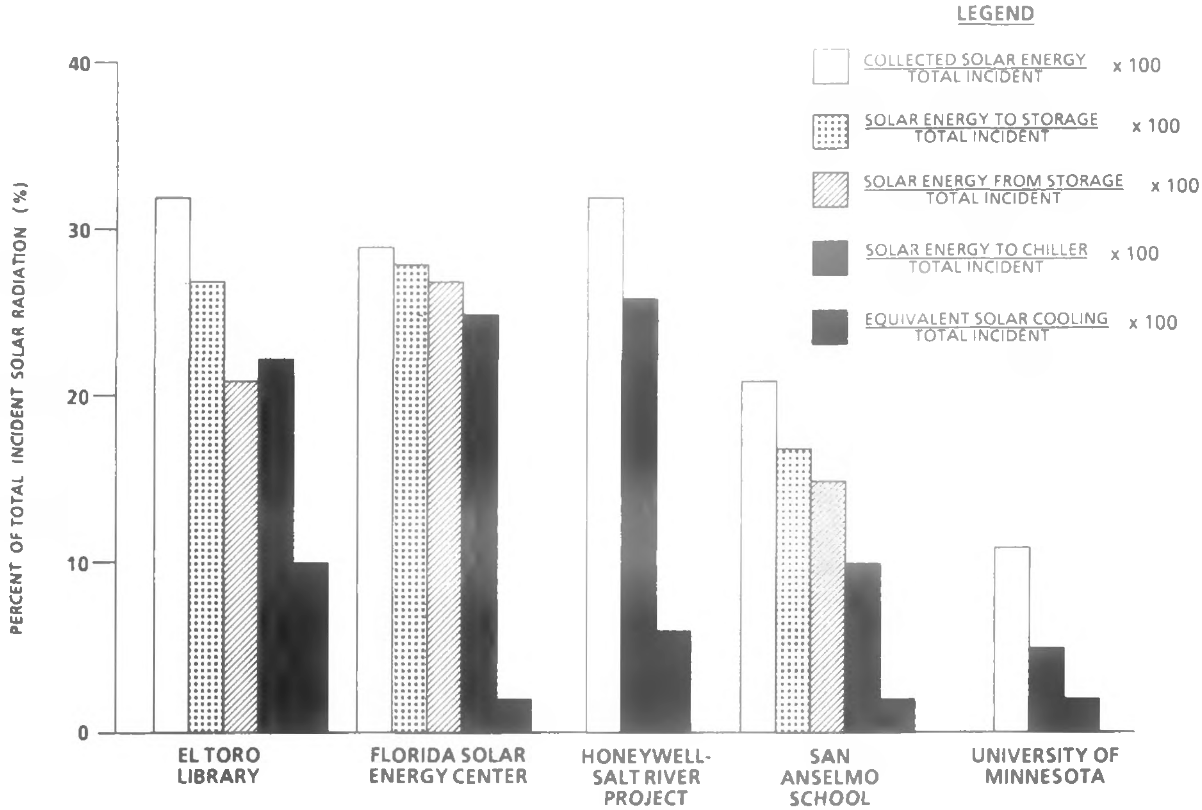

Figure 16. Solar Cooling Efficiency

increased, even if no additional energy were collected by the system. The El Toro Library efficiency improved from five percent the previous year to the 10% value this year, primarily due to better utilization of stored solar energy.

The Honeywell-Salt River Project had the next highest solar cooling efficiency of six percent (one percent less than the previous year's seven percent). Transport losses were fair, and the Rankine system showed good performance. Collection efficiency was as high as the sites having evacuated-tube solar collectors, but the Honeywell-Salt River Project used conventional flat-plate collectors directly connected to the Rankine heat engines. Thus, there were no storage losses.

The remaining sites had two percent cooling efficiencies, which were due to a number of factors, acting in combination to really reduce the potential solar cooling efficiencies.

The San Anselmo School collection efficiency was low at 21%. This was combined with a poor chiller COP of 0.22 to result in a two percent solar cooling efficiency.

The Florida Solar Energy Center showed good collection efficiency and good storage efficiency, yet the overall cooling efficiency was two percent. This was due to a poor chiller COP of

0.27, and poor utilization of chilled water from the cold storage tank.

The University of Minnesota suffered from the lowest collection efficiency, relatively high transport and storage losses, and a low chiller COP of 0.35. The resultant performance was very poor.

Table 4 presents values for solar system and subsystem Coefficients of Performance (COPs). Solar system COP is a measure of the effectiveness of the system in transferring solar energy to the load (it is not a chiller COP), and is calculated by dividing parasitic energy use into the solar energy cooling equivalent.

Table 4. SOLAR SYSTEM COEFFICIENT OF PERFORMANCE

SITE	SOLAR ENERGY SYSTEM COP (SEL/SYSOPE1)	COLLECTION SUBSYSTEM COP (SECA/CSOPE)	SOLAR COOLING SUBSYSTEM COP (CSE/COPE1)	SOLAR COOLING COP (CLS/COPE1 + CSOPE)
El Toro Library	31	45	5.4*	2.3*
Florida Solar Energy Center	6.5	39	8	1.7
Honeywell-Salt River Project	28	89	43	6.4
San Anselmo School	11	51	16	2.1
University of Minnesota	5	8.3	6.9*	1.0*

* Solar cooling operating energy was estimated by multiplying the solar fraction by the cooling subsystem operating energy.

For definitions of acronyms in parentheses, refer to Appendix E.

The sites' COPs for solar cooling ranged from 1.0 for the University of Minnesota to 6.4 for the Honeywell-Salt River Project. In order to compete with electric chillers, a system should achieve a system COP of at least 2.5. The Honeywell-Salt River Project had an 8.4 COP the previous year; this year's drop to 6.4 was due to poorer collector efficiency.

The San Anselmo School COP also degraded from 3.9 to 2.1. This was due to a 280% loss of performance on the solar chiller.

There does appear to be a consistent reduction over the previous year's COP values for the four sites which were involved in both seasons' comparative reports. This reduction may indicate a requirement for greater maintenance, particularly of the absorption chillers.

B. SOLAR COLLECTOR PERFORMANCE

Many forms of solar collectors could be adapted for use in a solar cooling system. This report is not intended to determine which type of collector should be used in a solar cooling system. However, the designer should choose collectors that operate efficiently at temperatures of 170°F and greater. Concentrating and evacuated-tube collectors would seem to be the best choices for this application. However, of the five sites compared, the flat-plate collectors were among the best performers.

Performance of the solar collectors at the five solar cooling sites is presented in Table 5, and shown graphically in Figure 17.

Operational collector efficiency is defined as the percentage of incident radiation collected during operation of the collector pump(s), while overall efficiency represents the percentage of total incident energy collected.

Table 5. COLLECTION SUBSYSTEM PERFORMANCE

(All values in million BTU, unless otherwise indicated)

SITE	INCIDENT SOLAR RADIATION	COLLECTED SOLAR ENERGY	COLLECTION EFFICIENCY (%)	OPERATIONAL INCIDENT ENERGY	OPERATIONAL COLLECTION EFFICIENCY (%)	ECSS REJECTED ENERGY	DAYTIME AMBIENT TEMPERATURE (°F)	ECSS OPERATING ENERGY
	(SEA)	(SECA)	(CLEF)	(SEOP)	(CLEFOP)	(CSRJE)	(TDA)	(CSOPE)
El Toro Library	580	178	31	516	35	10.2	73	3.98
Florida Solar Energy Center*	776	222	29	524	42	0.0	81	8.35
Honeywell-Salt River Project	2,900	897	31	2,300	39	34.7	81	10.1
San Anselmo School	608	130	21	580	22	18.1 E	72	2.53
University of Minnesota	1,500	160	11	959	17	0.0	76	19.2

* Eight months of data available.
E Denotes estimated value.

For a description of acronyms in parentheses, refer to Appendix E.

The Honeywell-Salt River Project and the El Toro Library showed the highest overall collector efficiencies, at 31% of total incident radiation. The Honeywell-Salt River Project was seven percent below the 38% value of the previous year. This collector efficiency is a 22% decrease in performance, mainly due to the lower efficiencies during the three winter months which were not included last year.

It is interesting to note that the flat-plate collectors at the Honeywell-Salt River Project performed as well as the "more technologically advanced" evacuated-tube collectors, based on overall percentage of incident energy collected.

The Florida Solar Energy Center had the highest operational collector efficiency studied, at 42% (efficiency during operation of the pump).

The University of Minnesota has a concentrating SLATS collector which uses linear mirrors to focus direct (beam) insolation onto a linear receiver. As the previous year's performance indicated, there was a focusing problem at the site which reduced collection efficiency to 17% operational and 11% overall. In addition, a heavy snowstorm occurred in December 1981 which damaged 17 of the 60 movable focusing SLATS. The damage was so severe that the system was not operational until March 1982. In April, representatives from the collector manufacturer readjusted the mirrors and consolidated them to form functional arrays. During May and June, the University of Minnesota collector array exhibited poor tracking performance. This problem was attributed to weak batteries in the tracking control system. The two problems caused very poor collector performance over the cooling season.

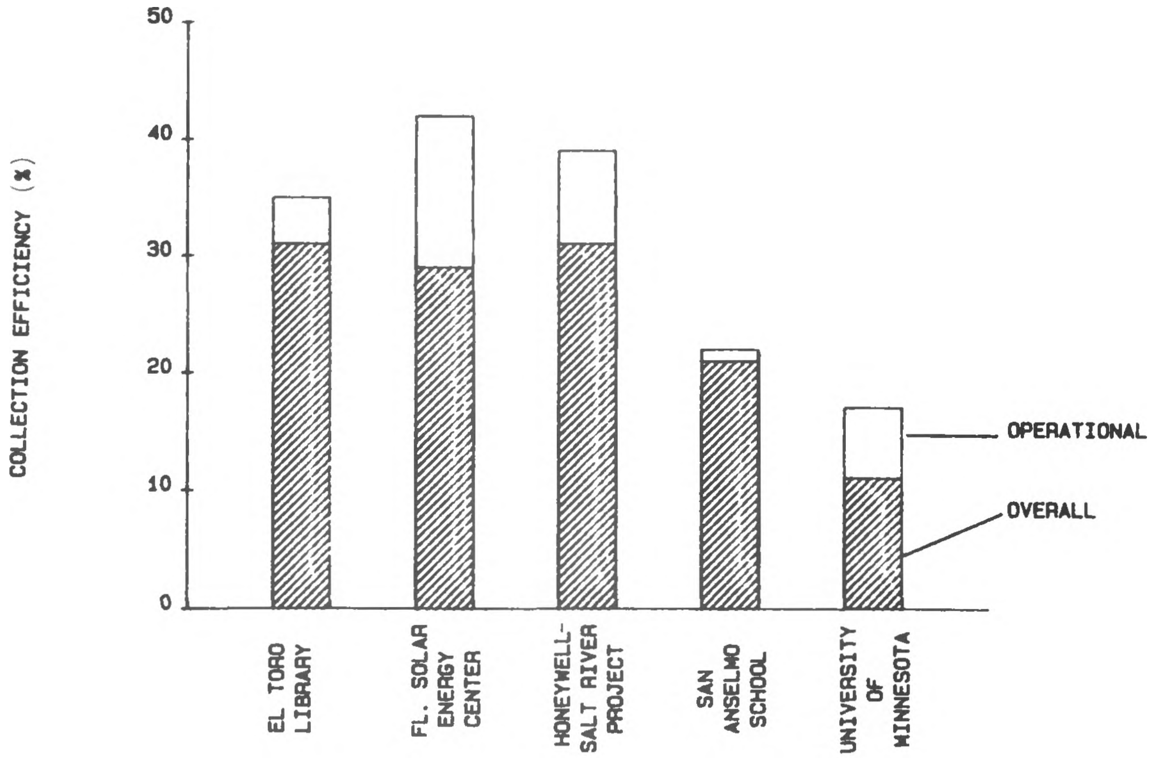

Figure 17. Solar Collection Efficiency

Curves derived from field measurements of the collector operating efficiency are presented in Figures 18 through 22. Average hourly efficiency values are plotted against collector operating points (see Footnote 1) to compare actual operating characteristics against the instantaneous single-panel ASHRAE test results.

The El Toro Library data for May 1982 is presented in Figure 18. The single-panel test result (denoted "Manufacturer's Curve") was quite similar in slope (an indication of the collector heat loss rate); however, the intercept with the y-axis was six to eight percent below the single-panel test curve, with outlying points achieving test result efficiency levels. Compared to other similar solar systems, the array performance at the El Toro Library was excellent.

The solar collector array at the El Toro Library consists of 82 solar panels (gross area of 1,427 square feet), manufactured by the General Electric Company. The collectors are evacuated-tube units designed to operate at high inlet temperatures.

Rejected energy at the El Toro Library totaled 10.2 million BTU, which represents an increase in the level of energy rejection as compared to the previous year. Review of the data indicated that the controller which allows the solar collection subsystem to transfer energy to the storage tank allowed some of this energy to be rejected from storage. Additionally, the insolation sensor (which activates the solar collector pump when insolation increases above a predetermined level) was set too low at times and allowed energy rejection to occur in the morning and afternoon. Additional energy was intentionally rejected to prevent collection subsystem overheating, primarily in June, July, and August.

The collector efficiency plot for the Florida Solar Energy Center is shown in Figure 19. The intercept is 52% and the slope ($F_R U_L$) or heat loss rate is -0.24. This slope is typical of evacuated-tube collectors; the collector array is quite efficient. During August 1982, the operational efficiency was 44%. This is the

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[1] Operating point, (TI-TA)/I, is the inlet temperature to the collectors minus the outdoor temperature divided by the insolation while the collectors are operating. The plot is for hours during which there was continuous flow through the collector array. The first hour of continuous operation for each day is not considered. Transient effects related to startup of operation often result in higher and/or lower efficiencies than subsequent hours at the same operating point. Outlying points which are greater than three standard deviations from the first order fit of the data are also filtered. Note that the first order curve fit information in the upper left of the plot is valid for the range of values of (TI-TA)/I available. Each plot is representative of the performance of the particular collector array for a particular month.

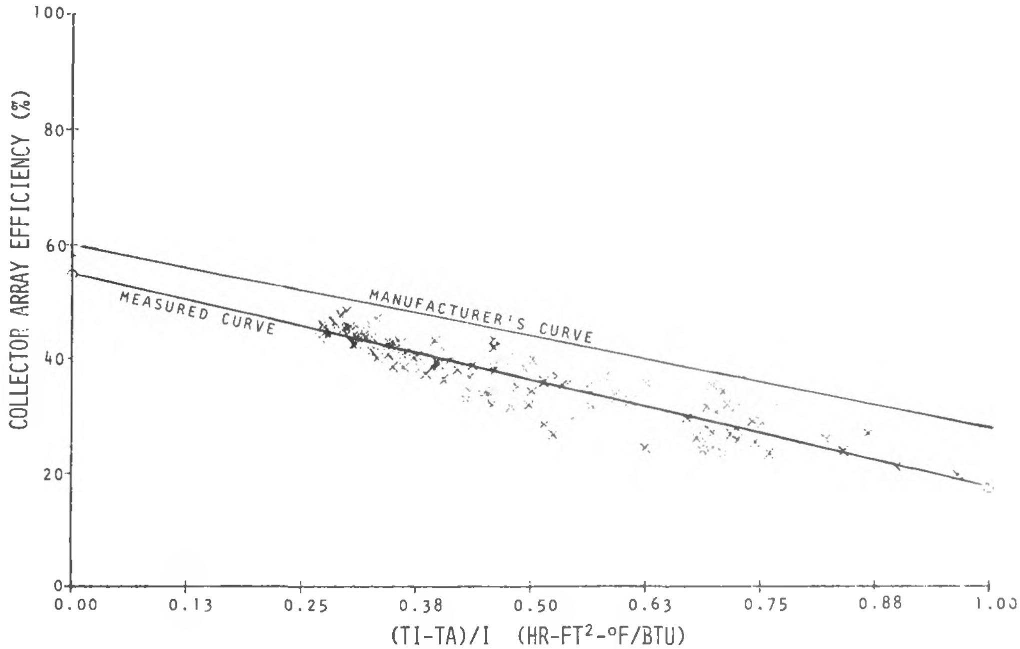

Figure 18. Average Collector Efficiency
El Toro Library
May 1982

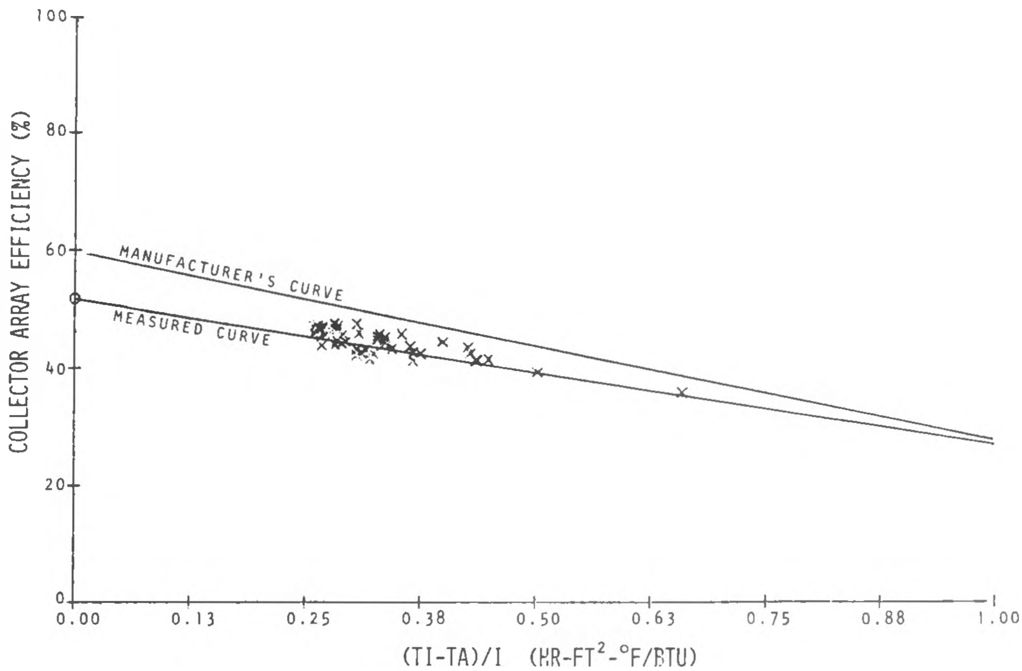

Figure 19. Average Collector Efficiency
Florida Solar Energy Center
August 1982

collector efficiency during the time the pump is on. The manufacturer's single-panel test curve shows about eight percent higher efficiency for the range of measured values. This is good agreement for an array of 2,089 square feet. The slope or loss of the measured curve is less than the manufacturer's reported value of -0.4 over the range of measured values.

The Honeywell-Salt River Project collector performance for July 1982 is presented in Figure 20.

Rankine cycle engines are more efficient at high inlet temperatures. This phenomenon would seem to put flat-plate collectors at a disadvantage compared to evacuated-tube and concentrating collectors, which have lower heat loss coefficients. Nevertheless, the efficiency of the Honeywell-Salt River Project collectors (31% overall, 39% operational) was higher than any of the other cooling system collector arrays in the NSDN, and the Honeywell-Salt River Project has the only flat-plate cooling array.

The operational collector efficiency of the Honeywell-Salt River Project array is usually 40% or higher in the cooling season (March through November). The efficiency is lower in the winter because lower ambient temperatures mean greater heat loss (the inlet fluid temperature is high year-round), and because lower winter insolation results in less energy collected, and losses make up a greater fraction of the collected energy.

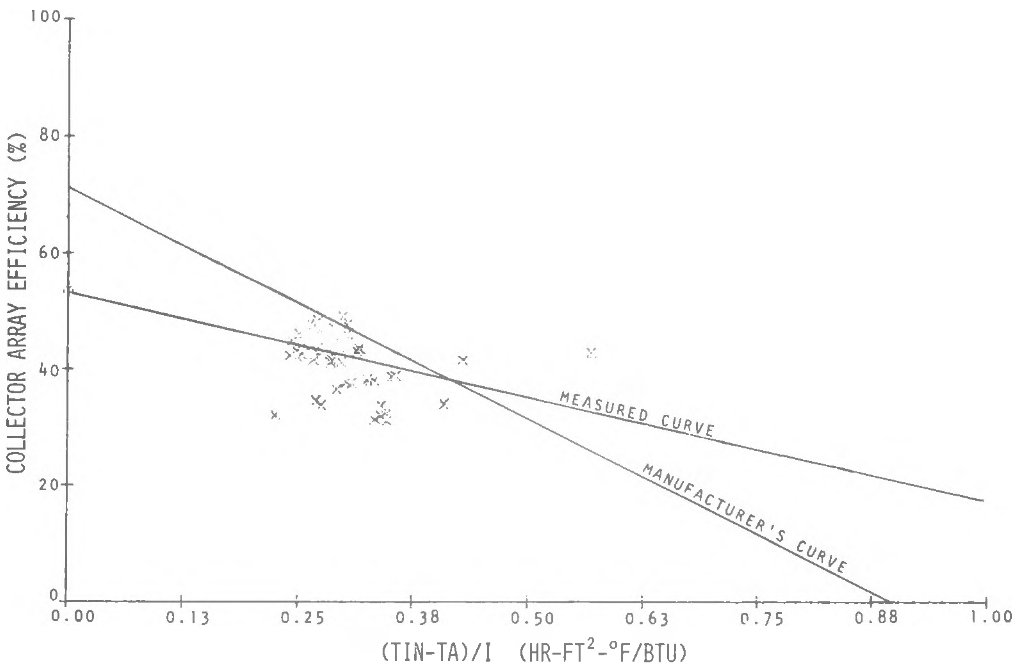

Figure 20. Average Collector Efficiency
Honeywell-Salt River Project
July 1982

The San Anselmo School field data for June 1982 is shown in Figure 21.

The solar collector array at the San Anselmo School is located on the roof of the structure, and consists of 3,740 square feet of General Electric evacuated-tube solar collectors, Model TC-100. Performance of the array during the three-month 1982 reporting period was very similar to the overall performance during the previous reporting periods.

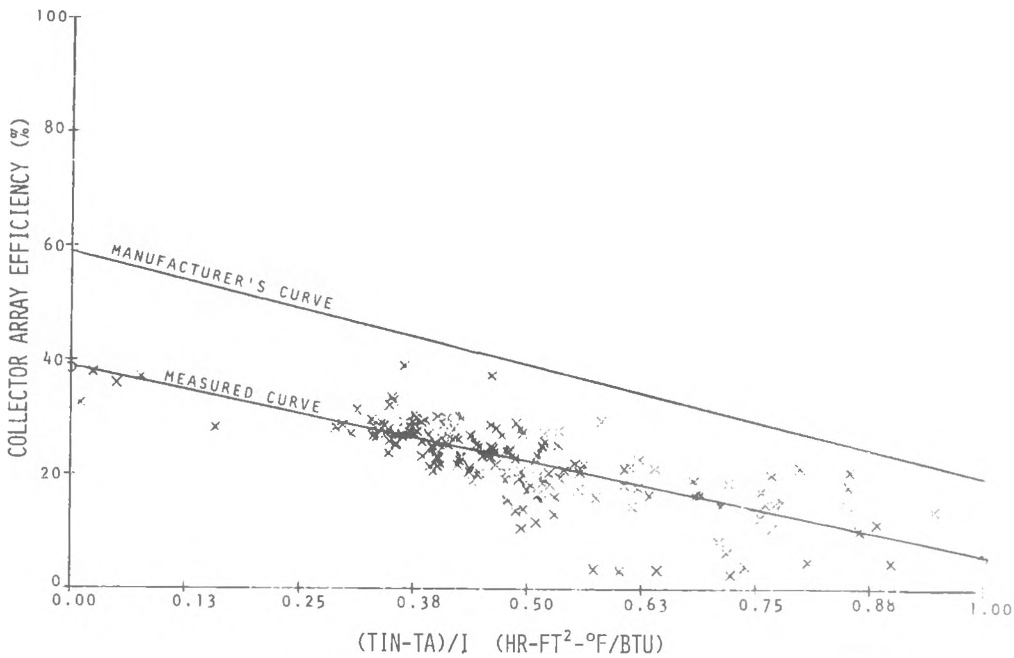

Figure 21. Average Collector Efficiency
San Anselmo School
June 1982

During an inspection and test of the energy collection subsystem in April 1982, several problems were corrected. As many as 25 of the solar collector tubes were broken. The reduction in collector area further penalized the solar system, and reduced overall system performance. According to Vitro field service engineers, rocks and other debris had been thrown at the array.

In the past, heat rejection has occurred due to overheating of the loop when modulating valve V-2 was closed. Several collector headers were leaking water onto the roof. Additionally, much of the collector piping has not been insulated.

Previous to April 1982, the collector control system only allowed heated solar fluid to enter the storage tank when greater than 160°F. The Barber Colman (BC) CP-8102 temperature controller, which controls switchover from circulation in the collector bank to storage, was replaced with a BC CC-8811 Differential Controller The new control strategy follows.

When solar energy is above a nominal value (~80 BTU/ft^2-hr), the collector pump (P-8) activates, and circulates fluid in the collector loop.

When the collector fluid temperature is 10°F greater than a sensor located in the storage tank, fluid is routed to the tank via operation of modulating valve V-2. Valve V-2 will be fully open when the collector (outlet) fluid temperature is 15°F greater than the tank.

Examination of data shows that these adjustments to the controls were not entirely successful. Observation of data during the week of April 15, 1982 indicated that the solar collector control was set too low (40 BTU/ft^2-hr vs. 80 BTU/ft^2-hr) while the differential temperature controller was set too high. Considerable amounts of energy were rejected due to these adjustments. These problems continued through May and June, causing excessive energy rejection.

The actual array efficiency was lower than the single-panel efficiency, primarily due to the problems mentioned above. Some broken evacuated tubes and collector degradation also contributed to the drop in collector efficiency. Observations from other NSDN systems indicate similar deviations from single-panel efficiencies vs. actual collector array efficiencies.

The curves were very similar from month to month as well. The overall heat loss rate (indicated in part by the slope of the operating curve) is nearly parallel to a single-panel test result. This similarity is typical of multiple-panel arrays. The y-intercept (a function of absorptivity, transmission of the glass, etc.) was around 20% lower than single-panel results. Again, this effect is typical, particularly when panels are out of commission, dusty, etc.

Repair of broken tubes, replacement of poor/weathered insulation, and insulation of bare pipes should be priority action items at the school, in order to improve collector array performance.

The University of Minnesota collector performance curve (Figure 22) shows a very erratic operational sequence. June 1982 depicts typical collector performance curves for this array. The large scattering of points is believed to be due to poor focusing of the mirrors. The University of Minnesota collector array performance was poor during 1982 due to the previously mentioned snow damage and control problems.

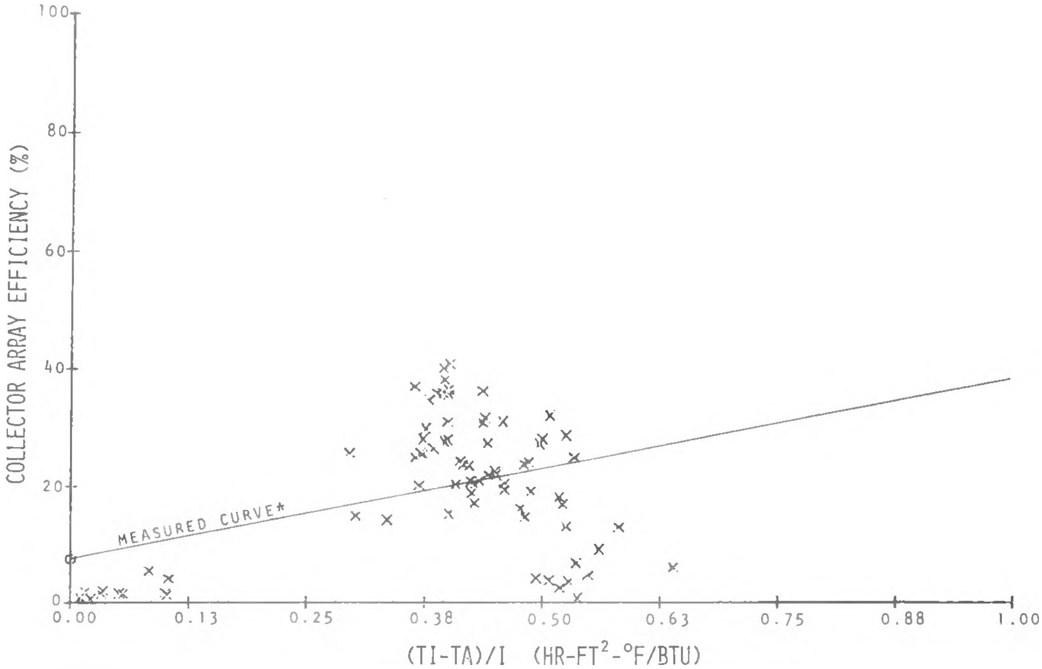

Figure 22. Average Collector Efficiency
University of Minnesota
June 1982

C. HOT STORAGE SUBSYSTEMS

All of the five sites had hot water storage tanks. Storage efficiency is a convenient and relatively simple index of comparison among the five sites. Table 6 is a compilation of storage energy flows over the season. The average effective heat loss coefficient is also calculated from the measured energy flows and surrounding temperatures. The University of Minnesota instrumentation did not measure storage performance. The Honeywell-Salt River Project used the hot storage tank for heating only; the Rankine engines are driven directly from the collector array.

Note that the heat loss coefficient is much lower at the Florida Solar Energy Center than at the other sites. Low heat loss rates contribute to good storage performance. In addition, when solar energy is in the storage tank for only a short time, then storage efficiency is improved. Storage efficiency is defined as the ratio of the output energy divided by the input energy.

The Florida Solar Energy Center had the highest storage efficiency, 96%. Storage energy output was estimated at this site, due to sensor failure.

None of the tanks operated at average seasonal temperatures over 170°F.

Table 6. HOT STORAGE SUBSYSTEM PERFORMANCE

(All values in million BTU, unless otherwise indicated)

SITE	SOLAR ENERGY TO STORAGE (STEI)	ENERGY FROM STORAGE (STEO)	CHANGE IN STORED ENERGY (STECH)	STORAGE EFFICIENCY (%) (STEFF)	AVERAGE STORAGE TEMPERATURE (°F) (TST)	HEAT LOSS COEFFICIENT (BTU/ft^2-°F-hr)	LOSS FROM STORAGE (STLOSS)
El Toro Library	155	122 E	0.70	79 E	168	0.2 E	37.7 E
Florida Solar Energy Center	314 E	300 E	0.01	96 E	163	0.01E	14.2 E
Honeywell-Salt River Project	75.7	46.5	1.8	64	130	0.31	27.4
San Anselmo School	99.4	84.6	0.16	85	167	0.37	14.7

E Denotes estimated value.

For a description of acronyms in parentheses, refer to Appendix E.

One of the sites (Florida Solar Energy Center) uses a cold storage tank to store chilled water for use by the Heating, Ventilation, and Air-Conditioning (HVAC) system. Chilled water was often available in the tank, yet, due to control problems, the stored chilled water was not used to its full potential. This problem reduced the overall solar contribution, allowed the chiller to cycle frequently, increased cold storage losses, and lowered performance.

Figure 23 presents the storage efficiency for the four NSDN sites for which storage energy flow was available.

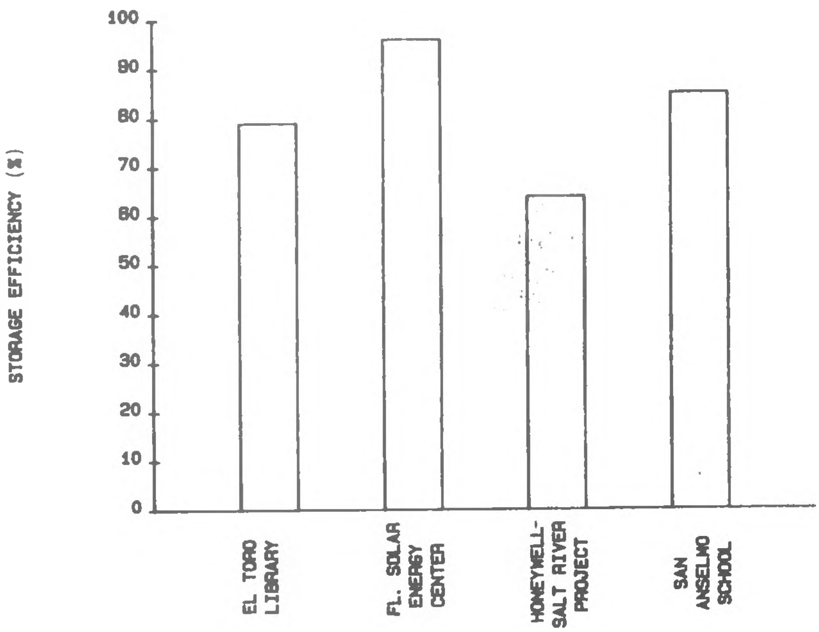

Figure 23. Storage Efficiency

D. SOLAR CHILLER OPERATION

Table 7 describes the solar-fired chillers studied during the 1982 cooling season at the five sites. Table 8 presents the measured performance at four of these sites.

Table 7. SOLAR CHILLER DESCRIPTIONS

SITE	SOLAR CHILLER TYPE	MANUFACTURER MODEL NO.	REFRIGERANT & WORKING FLUID	CAPACITY (TONS)
EL Toro Library	Absorption	Arkla Air Conditioning Co. WFB-300	Lithium-Bromide Water solution	25 tons
Florida Solar Energy Center	Absorption	Arkla Air Conditioning Co. WFB-300	Lithium-Bromide Water solution	25 tons
Honeywell-Salt River Project	Rankine-powered vapor compression	Barber-Nichols	Freon R-113	50 (total)
San Anselmo School	Absorption	Arkla Air Conditioning Co. WFB-300	Lithium-Bromide Water solution	25 tons
University of Minnesota	Absorption	Trane, Inc. C2J-W-5	Lithium-Bromide Water solution	147 tons

Table 8. ABSORPTION CHILLER PERFORMANCE

(All values in million BTU, unless otherwise indicated)

SITE	EQUIPMENT LOAD (TCEL)	THERMAL ENERGY INPUT (TCEI)	OPERATING ENERGY (TCEOPE)	REJECTED ENERGY (TCERJE)	COEFFICIENT OF PERFORMANCE (COP) (TCEI/TCEL)
El Toro Library	208	462	53.8	778	0.45
Florida Solar Energy Center	75.6	277	35.8	311	0.27
San Anselmo School	12.9	58.1	3.56	84.9	0.22
University of Minnesota	693	1,980	175	2,630	0.35

For a description of acronyms in parentheses, refer to Appendix E.

Three of the five sites use the Arkla Model WFB-300 lithium-bromide type absorption chiller fired with solar energy. The University of Minnesota uses a larger 147-ton Trane absorption chiller, while the Honeywell-Salt River Project uses two Barber-Nichols chillers driven by solar heated Rankine cycle turbines. The Rankine engine is connected via a gearbox/clutch system to a motor generator, which is in turn connected to the chiller. The Honeywell-Salt River Project site has the capability of generating electrical power with excess shaft horsepower.

Monthly average COP values (thermal cooling production divided by thermal energy input) for the four absorption chiller sites are shown in Figure 24. None of the sites achieved the design COPs of 0.6 to 0.8.

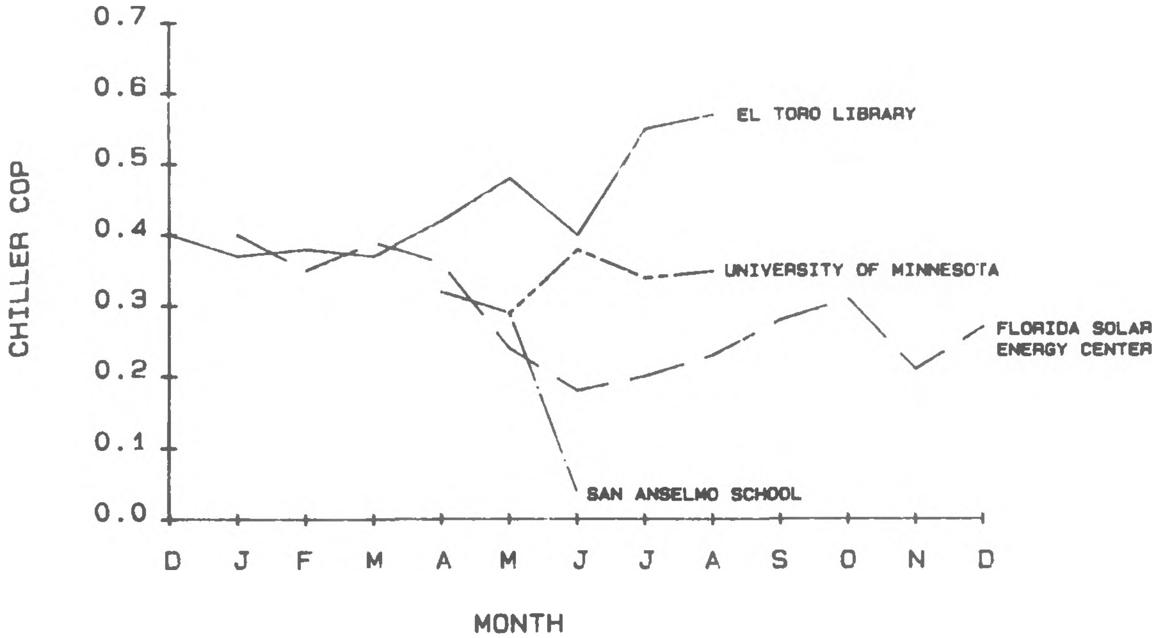

Figure 24. Monthly Average Chiller COP

Note that all of the four sites showed COP values which were quite variable over the test periods. The El Toro Library seemed to exhibit a general trend of improvement in COP, with the exception of June. The El Toro Library system was operating at a COP of 0.40 to 0.38 at the beginning of the season, and rose to a COP of 0.55 after a slight drop in June.

The solar chiller at the El Toro Library, while showing improvement over the cooling season, operated with some control problems during the nine-month monitoring period. The control problems were not directly due to chiller mechanical problems, but were a result of rapid cycling between heating and cooling at the library, particularly during transitional months. There is no provision for anticipation of loads based on external or internal conditions, and the thermostats which cause the system to switch between cooling and heating have little or no "deadband." The result was that space heating and cooling occurred simultaneously, and the effect was to increase the net cooling load significantly over the previous year's value (see Section 3.E, Building Loads and Cooling Subsystem Performance.)

The Florida Solar Energy Center had a COP of 0.38 to 0.40 during the first four months of 1982, then the value fell to 0.2 to 0.25 during June, July, and August. The Florida Solar Energy Center also exhibited a control problem during the cooling season. The pump which circulates cooling water from the solar chiller operated continuously during May and June, due to a control failure on the switchover between auxiliary and solar cooling. The auxiliary chiller thus provided most of the cooling, although solar energy was available.

Absorption chillers require steady-state operating conditions to provide maximum COPs, and both inlet hot water temperatures and condenser (heat rejection loop) temperatures should be within narrow limits. At the Florida Solar Energy Center, high humidity conditions seemed to limit the energy rejection loop output, and cycling on and off resulted in a low seasonal COP.

Several items could be changed to improve cooling performance at the Florida Solar Energy Center, including:

- Repair the solar/auxiliary changeover mode control.

- Prioritize the use of stored chilled water.

- Maintain chillers to improve performance.

The chiller at the San Anselmo School showed stable performance at a COP of 0.3 during the first two months of its three-month analysis period, yet the COP dropped off very severely during June.

Space cooling for the San Anselmo School is provided by four auxiliary gas-fired absorption chillers and a solar-supplied absorption chiller. Space cooling is initiated and terminated by a time-clock control which typically operates Monday through Friday from 7:00 a.m. to 3:30 p.m. During system operation, the interior thermostats determine the space cooling needs of the building.

The solar chiller performed poorly in comparison to the previous analysis period at an average COP of 0.22 vs. a 0.57 COP the previous period. The auxiliary chiller performance was nearly the same at an average COP of 0.29 vs. 0.32 the previous year. The overall solar contribution of 12% was low compared to the design solar cooling fraction of 72%.

There are basic equipment problems affecting space cooling performance at the San Anselmo School. Most significant of these has been the inability of the chiller units to hold a vacuum and thus function properly. Additionally, the chillers are designed to stage operation during variable cooling loads, and apparently the units are not operating in sequence as designed.

In April 1982, all of the chiller units were vacuum-pumped, checked out, and performance tests made. Soon after this work was done, however, due to the rapid vacuum loss and continuing control problems, cooling performance returned to the more typical (poor) performance. Combined with the inadequacy of chiller/heater unit #5, overall space cooling performance was very poor, and significant problems still remain after the numerous changes made to the system in April 1982.

The chillers had lost vacuum by May 25, and site personnel vacuum-pumped them on that day. It appears that service intervals for these units increased to a one-to-two month period. Arkla representatives indicated that units should operate for up to three years without service. Seals, gaskets, and other components may require replacement on these chiller units.

Improper staging of the auxiliary chillers required a proportionally greater energy input for a small increase in chilled water production. The improper staging activates all the auxiliary chillers even though the demand is very low. This activation resulted in large quantities of auxiliary energy usage and low chiller COP. The staging control is supposed to allow a small time delay during startup conditions between chillers, and then stage the chillers according to return temperature from the loads.

The flow rate through the various chillers also varied with the number of units which were on-line at any given time, thus affecting performance. There was a variation of 15 to 20 gpm per chiller from peak flow to low flow. Constant flow will help overall COP by allowing more steady operation of all units. The staging thermostat should be replaced as well.

Some control valves on the individual zone air-handling units were apparently stuck open, which allowed full heating/cooling of a zone regardless of the load in the area. The school personnel were aware of the problem and were planning to fix these units.

E. BUILDING LOADS AND COOLING SUBSYSTEM PERFORMANCE

This section presents comparative data on building loads and cooling subsystem performance. For site-specific data, the reader is directed to Appendix A.

Data on the space cooling subsystems is shown in Table 9. The values in the table represent totals for the measurement period. Note that the solar cooling loads at the El Toro Library and the University of Minnesota are not directly measured from chiller output but are proportional to chiller input. This proportion is necessary because solar and auxiliary energy are mixed at the input to the chiller. In addition, the solar energy used and auxiliary thermal energy do not sum up to the cooling load because of the thermodynamic conversion occurring in the chiller. The building temperature is not measured at the Honeywell-Salt River Project.

Table 9. SPACE COOLING SUBSYSTEM PERFORMANCE

(All values in million BTU, unless otherwise indicated)

SITE	COOLING LOAD (CL)	SOLAR COOLING LOAD (CLS)	SOLAR FRACTION OF LOAD (%) (CSFR)	SOLAR ENERGY USED (CSE)	OPERATING ENERGY (COPE)	AUXILIARY THERMAL (CAT)	AUXILIARY FOSSIL (CAF)	AUXILIARY ELECT. (CAE)	BUILDING TEMPERATURE (°F) (TB)
El Toro Library	208	56.7	24 E	126	85.6	389	572	-	74
Florida Solar Energy Center	234	32.0 (17.5)*	14 E	277	67.7	61.2	-	71.9	77
Honeywell-Salt River Project	2,590	166	7	735	406	792	-	931	-
San Anselmo School	104	12.9	12	58.1	60.6	314	522	-	75
University of Minnesota	693	28.7	4	82.0	286	1,890	3,160	-	77

E Denotes estimated value.
* Solar cooling load minus the August through November months.

For a description of acronyms in parentheses, refer to Appendix E.

Table 10 shows the space cooling loads normalized to building area and number of months of data.

The normalized cooling load indicates the energy efficiency of a building. This parameter varies from 9.4 to 22.9 BTU/ft^2-cooling degree-day. These values are larger this year than the previous year.

Table 10. SPACE COOLING LOADS

SITE	INSTALLED COOLING CAPACITY PER UNIT FLOOR AREA (TON/10^3FT^2)	COOLING LOAD (BTU/FT^2-MO)	NORMALIZED COOLING LOAD (BTU/FT^2-CDD)	TEMPERATURE COMPENSATED NORMALIZED COOLING LOAD (BTU/FT^2-CDDT)*
El Toro Library	2.5	2,310	22.9	12.0
Florida Solar Energy Center	9	3,900	10.0	10.0
Honeywell-Salt River Project	5.05	6,740	18.3	-
San Anselmo School	3.68	1,020	19.5	9.0
University of Minnesota	1.75	1,650	9.4	9.4

* CDDT is the temperature compensated cooling degree-day and represents the difference between a reference temperature and measured building temperature times the number of days that the difference existed.

The Florida Solar Energy Center and the University of Minnesota have the lowest normalized cooling loads, 10.0 and 9.4 BTU/ft^2-cooling degree-day respectively. The low value for the Florida Solar Energy Center indicates a well-insulated building, and, since Williamson Hall at the University of Minnesota is 95% below ground, a low cooling load is expected.

The El Toro Library showed the largest increase in normalized cooling load, from 12.9 to 22.9 BTU/ft^2-cooling degree-day. Some of this increase may be due to the lack of a deadband in the heating to cooling changeover thermostats. The San Anselmo School also had a large increase in the normalized cooling load, from 12.7 to 19.5 BTU/ft^2-cooling degree-day.

There are two sites with warmer building temperatures: the Florida Solar Energy Center and the University of Minnesota (see Appendix A). These buildings also have the lowest normalized cooling loads. Since overcooling a building will result in a cooler building temperature, it seemed that there might be some correlation between the temperature difference from the warmer buildings to the cooler buildings and the loads. Therefore, the temperature compensated cooling degree-day (CDDT) is defined as the temperature difference between some reference and the desired building times the number of hours in the period of study divided by 24 hours. The temperature compensated cooling degree-day and the regular cooling degree-days are added together.

Note that in the column marked Temperature Compensated Normalized Cooling Load, the values now range from nine to 12 BTU/ft^2-CDDT. The Honeywell-Salt River Project value was not obtained because it has no building temperature sensor. The El Toro Library still has the largest load, but it is the least insulated building of the four. The other three buildings have remarkably similar normalized cooling loads. It is also interesting to note that a 2°F to 3°F reduction in building temperature equaled 10 BTU/ft^2-CDDT. This is also roughly equivalent to the internal and environmental gains at these buildings.

The normalized cooling load at the Honeywell-Salt River Project was just slightly larger this year than the previous year.

Comparing the cooling loads (BTU/ft^2-month) to the installed cooling capacity, two systems seem to be oversized, the Florida Solar Energy Center and the San Anselmo School.

At the Florida Solar Energy Center, although the system appears to be oversized, the solar and auxiliary systems are actually quite separate. Therefore, the installed capacity per 100 ft^2 is really only four to five tons/1,000 ft^2 of collector.

The oversized system at the San Anselmo School may have contributed to the system inefficiency because cycling and part load operation reduce absorption chiller efficiency. The effect of oversizing was also more severe because the auxiliary chiller did not stage properly. This improper staging meant that more chillers were operating, thus reducing the load per chiller.

If the cooling load in BTU/ft^2 per month is divided by the installed capacity per 1,000 ft^2, the result is the cooling load per ton per month (million BTU/ton-month). This parameter yields a value of about one million BTU/ton-month for the University of Minnesota and the El Toro Library. The Florida Solar Energy Center loading is also about one million BTU/ton-month if one considers either the solar or auxiliary chiller alone. The Honeywell-Salt River Project has a higher loading of 1.33 million BTU/ton-month. However, the San Anselmo School has only about 0.28 million BTU/ton-month. This is certainly an indication of an oversized system.

It is interesting to note that a loading of one million BTU/ton-month would require full load operation for only 83⅓ hours in a month. Obviously, these chillers are running at part load most of the time. In fact, if we assume 21⅔ working days per month and 12 hours of operation per day, then there would be 260 hours of cooling system operation per month. This implies that these chillers will operate at full load for 32% of the time, or at an average of 32% of full load for the month. Better equipment utilization and performance would be possible with higher loadings. The Honeywell-Salt River Project operates for an average of 111 hours in a month, or an average loading of 43%.

The preceding observations are best illustrated by Figures 25 and 26, Hourly Building Cooling Load vs. Time of Day. These graphs show the average cooling load for each hour of the day during one of the hottest months. Hourly cooling loads for the other cooling season months are shown in Appendix C.

Figure 25 shows the hourly cooling loads for the El Toro Library, the Florida Solar Energy Center, and the San Anselmo School. Note that both the Florida Solar Energy Center and the El Toro Library have similar loads and loading patterns. At these two sites, the loads are fairly constant at about eight and one-half tons during the day. However, at the San Anselmo School, the load increases during midday to about 23 tons, suggesting that this building is more weather-dominated. Remembering that the installed capacity at the El Toro Library is 25 tons (see Appendix B) and that the available maximum solar capacity at the Florida Solar Energy Center is 25 tons, one can see that the systems are oversized by three to one. This is the same conclusion reached above, where one million BTU/ton-month would require full load operation for only 32% of the time. Likewise, at the San Anselmo School the installed capacity is 125 tons (see Appendix B), which is about five times larger than the midday loads.

Figure 26 shows the hourly cooling loads for the Honeywell-Salt River Project and the University of Minnesota. Note that the cooling loads at these sites are continuous; there is no off time. However, solar energy is utilized only during the day at the Honeywell-Salt River Project because there is no storage on the solar cooling system. At the University of Minnesota there is some solar cooling throughout the day.

Comparing the hourly building cooling loads to the installed cooling capacity shown in Appendix B yields a load-to-capacity ratio

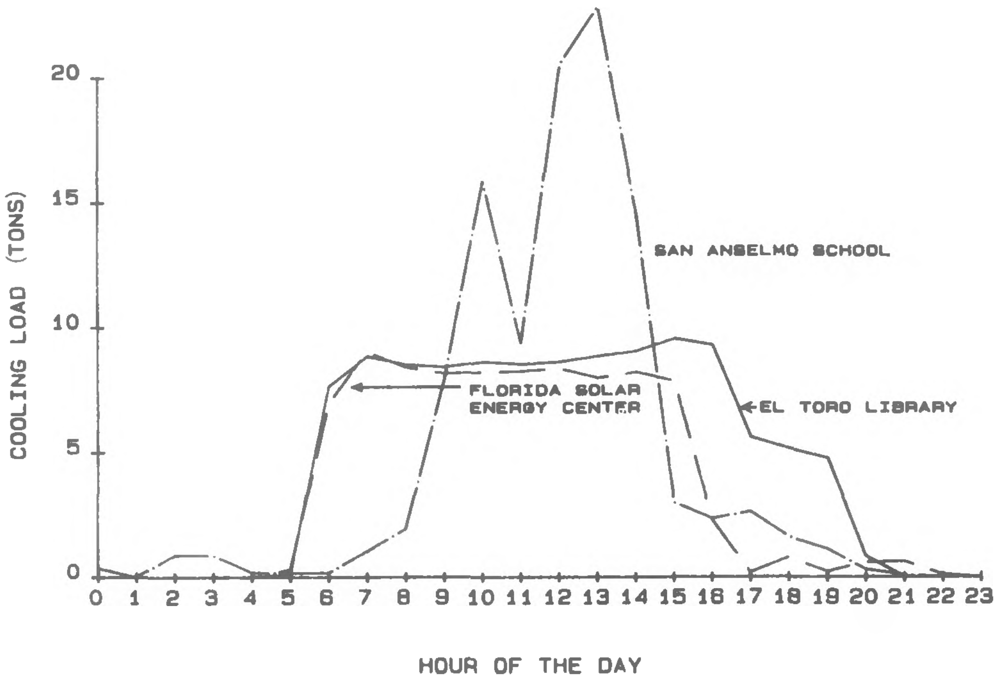

Figure 25. Hourly Building Cooling Load
vs. Time of Day
El Toro Library, Florida Solar Energy Center,
and San Anselmo School

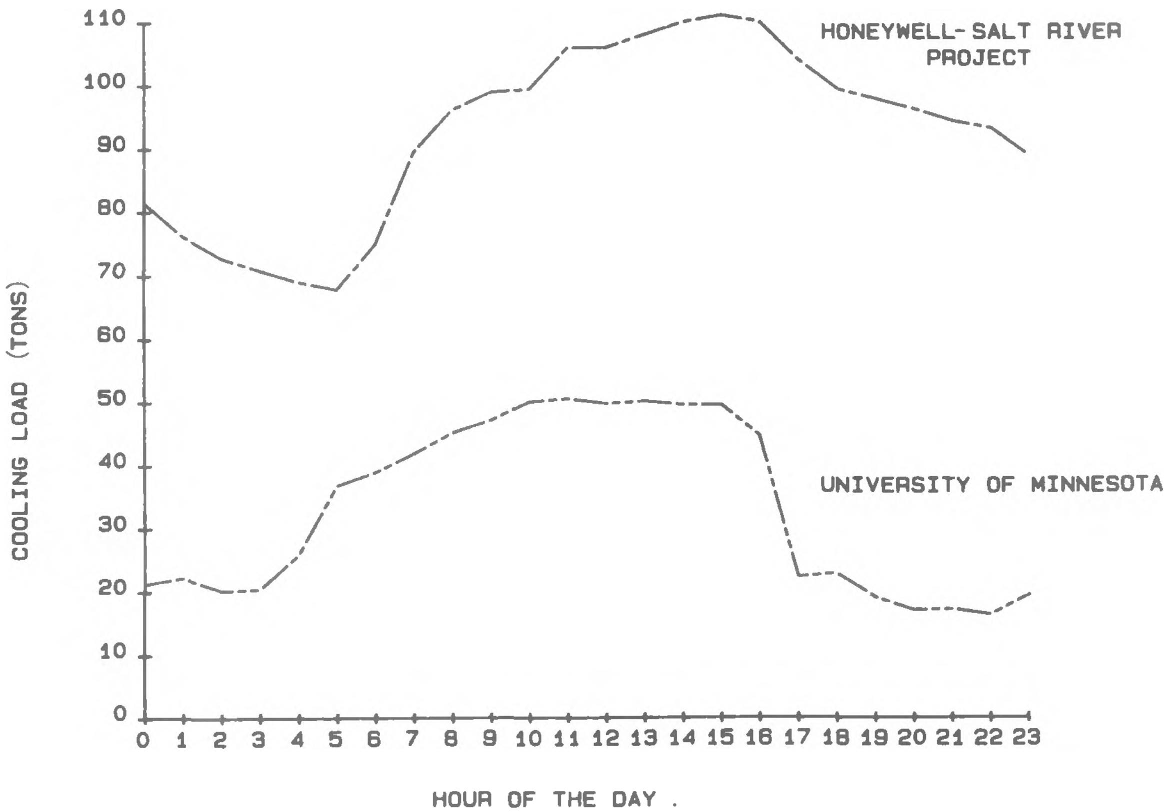

Figure 26. Hourly Building Cooling Load
vs. Time of Day
Honeywell-Salt River Project and University of Minnesota

of 0.4 for the Honeywell-Salt River Project. The same comparison for the University of Minnesota site yields a load-to-capacity ratio of 0.33. Part of the reason that the Honeywell-Salt River Project has a higher loading is that the second auxiliary chiller is only used between 10:00 a.m. and 6:00 p.m. (apparently a time clock turns it on). However, it would appear that this form of peaking chiller operation is quite inefficient because very little load exists for it. This observation is readily seen in Figure 26 by looking at the eighth and ninth hours and noting that there is no change in load in the tenth hour when the second auxiliary chiller is on. Fortunately, the solar chiller is used as a precooler to the auxiliary, so it is under full load at all times.

The effect of higher loadings during midday at the University of Minnesota was to increase the chiller COP from about 0.25 in the early morning to about 0.39 during the midday hours. The change in COP represents over a 50% improvement in efficiency. If the chiller had operated at a monthly COP of 0.39 rather than the 0.34 measured, then 162 million BTU of energy would have been saved at a boiler efficiency of 60% during July.

F. WEATHER

The weather conditions have been discussed with respect to design vs. actual cooling solar fraction. From Table 11, the reader can see that insolation was lower than the long-term average and temperatures were higher than the long-term average.

Table 11. WEATHER CONDITIONS

SITE	AVERAGE DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY)		AMBIENT TEMPERATURE (°F)		HEATING DEGREE-DAYS		COOLING DEGREE-DAYS	
	MEASURED (SE)	LONG-TERM AVERAGE	MEASURED (TA)	LONG-TERM AVERAGE	MEASURED (HDD)	LONG-TERM AVERAGE	MEASURED (CDD)	LONG-TERM AVERAGE
El Toro Library	1,481	1,786	65	61	935	1,599	907	512
Florida Solar Energy Center	I	1,550	77	75	153	206	4,660	4,038
Honeywell-Salt River Project	1,660	1,870	73	71	707	1,122	2,574	2,374
San Anselmo School	1,780	1,947	63	62	323	401	157	103
University of Minnesota	1,553	1,622	67	62	480	965	878	555

I Denotes invalid data.

For a description of acronyms in parentheses, refer to Appendix E.

The daily incident solar energy per unit area was 17% below the long-term for the El Toro Library, 11% below the long-term at the Honeywell-Salt River Project, nine percent below the long-term at the San Anselmo School, and four percent below the long-term at the University of Minnesota.

To see whether the reduction in solar energy changed collector performance, we must compare the previous year's collector efficiencies and solar insolation to this year's data. Curiously, at the El Toro Library the level of solar insolation in 1982 was less but collector performance improved from 29% to 31%. Collector performance at the San Anselmo School improved from 20% to 21%; however, solar insolation also improved slightly. The level of solar insolation dropped considerably at the Honeywell-Salt River Project and so did collector efficiency. Insolation at the University of Minnesota improved somewhat but collector performance did not; however, collector performance was degraded due to poor tracking. It is interesting to note that the two collector system efficiencies which were only slightly changed have evacuated-tube solar collectors.

Measured temperatures were warmer than the long-term average during both 1981 and 1982. Warmer temperatures tend to improve solar collector performance, but the effect is quite small. Primarily, warmer temperatures increase the cooling loads. The average space cooling loads in BTU/ft^2-month (see Table 10) are lower during 1982 because the data season includes more winter months.

The following simplified example shows the effect of lower insolation levels and higher than ambient temperatures on collector efficiency and on collected solar energy.

Collector efficiency can be approximated by the equation:

$$\eta = FrU_L \times (OPPNT) + FrT\alpha$$

$$\text{where } OPPNT = \frac{\text{Tinlet} - \text{Tambient } (^{o}F)}{\text{Insolation } (BTU/ft^2\text{-hr})}$$

Tinlet = the collector inlet temperature
Tambient = the temperature of the air around the collectors

At the Honeywell-Salt River Project:

$FrU_L = -0.51$

$FrT\alpha = 0.67$

Case (1)

Tinlet = 175°F
Tambient = 73°F
Insolation = 300 BTU/ft^2-hr

OPPNT = 0.34
η = 49.7%

Case (2)

T_{inlet} = 175°F
$T_{ambient}$ = 71°F
Insolation = 300 BTU/ft^2-hr

OPPNT = 0.347
η = 49.3%

Cases (1) and (2) represent the change in design efficiency due to the ambient temperature. Obviously, this effect is quite small, 0.4%, and probably could not be measured under NSDN conditions.

Case (3)

T_{inlet} = 175°F
$T_{ambient}$ = 73°F
Insolation = 0.89 x 300 BTU/ft^2-hr

OPPNT = 0.382
η = 47.5%

Case (4)

T_{inlet} = 175°F
$T_{ambient}$ = 71°F
Insolation = 0.89 x 300 BTU/ft^2-hr

OPPNT = 0.39
η = 47.1%

Cases (3) and (4) represent the change in collector efficiency which would be due to the difference in measured and long-term insolation at the Honeywell-Salt River Project site. Comparing Cases (2) and (4), one can see that a difference of 11% in insolation causes a 2.2% drop in collector efficiency. The estimated overall change in design collector efficiency is the difference between Cases (2) and (3), or a 1.8% decrease. The slight increase in the actual temperature above the long-term average temperature only partially compensates for the decrease in insolation from the long-term average insolation.

Section IV

COMPARATIVE RESULTS AND PREDICTIONS

A. ENERGY SAVINGS

With respect to economic viability and commercialization, energy savings are the most important performance parameter. Several energy savings parameters are shown in Table 12. These parameters are presented in terms of kilowatt hours (kwh) for easier conversion to dollars.

Table 12. SOLAR COOLING ENERGY SAVINGS

SITE	SOLAR COOLING LOAD (MILLION BTU) (CLS)	ELECTRIC EQUIVALENT* (KWH)	COOLING SUBSYSTEM OPERATING ENERGY (KWH) (COPEI)	COLLECTOR OPERATING ENERGY (KWH) (CSOPE)	NET** SAVINGS (KWH)	NET SAVINGS PER COLLECTOR AREA (W/FT^2-MONTH)
El Toro Library	56.7	6,650	6,020 (20.5)	1,170 (3.98)	-540 (-1.84)	-42.0 (-143)
Florida Solar Energy Center	32.0	3,750	10,500 (35.8)	2,450 (8.35)	-9,190 (-31.4)	-366 (-1,250)
Honeywell-Salt River Project	166	19,500	5,600 (19.1)	2,960 (10.1)	10,900 (37.2)	190 (648)
San Anselmo School	12.9	1,510	1,040 (3.55)	741 (2.53)	-272 (-0.93)	-24.2 (-82.6)
University of Minnesota	28.7	3,360	3,350 (11.4)	5,630 (19.2)	-5,610 (-19.1)	-177 (-604)

* Solar cooling load divided by 2.5 times 293.
** Electric equivalent minus solar-specific operating energy minus collector operating energy.

For a description of acronyms in parentheses, refer to Appendix E. Numbers in parentheses are in million BTU.

The net savings value is for the number of months for which data was available in this report. Note that four of the five systems have negative net savings. This means that they cost more to operate than the value of cooling produced.

The systems with negative net savings were all absorption cooling systems. The reasons for such poor performances are high parasitic power requirements (operating costs) on the cooling subsystems and generally poor chiller performance.

The Rankine system at the Honeywell-Salt River Project has a large positive net energy savings because of low operating costs.

The values of net savings per unit collector area per month provide a way to compare these systems on an equivalent basis. Note that the Honeywell-Salt River Project system would produce only 6.33 W/ft^2-day compared to 50 to 65 W/ft^2-day for space heating systems and 140 to 155 W/ft^2-day for domestic hot water systems. Obviously, much greater savings would be necessary to make solar cooling systems cost-effective.

There are several other parameters in Table 12 which will be discussed. The solar cooling load represents the equivalent solar cooling which would have been produced if only solar energy were used to power the chiller. This definition is used because the El Toro Library and the University of Minnesota systems have an auxiliary boiler in series with the solar energy systems. The solar cooling load is used to derive the equivalent electrical savings parameter which is an estimate of the amount of power a standard centrifugal chiller system would require. This is calculated as solar cooling load divided by 2.5 (the COP of a standard chiller) times 293 (the conversion of a million BTU to kwh).

The equivalent electrical savings are the total savings for these systems. The cooling subsystem and collector operating energy are parasitic energies needed for system operation. The parasitic energy must be subtracted from the total energy saved to obtain the net energy savings.

Four of these solar systems were also monitored during the 1981 cooling season. Comparing the 1981 and 1982 performance, the Honeywell-Salt River Project, the San Anselmo School, and the University of Minnesota systems all performed better in 1981. The El Toro Library performed only slightly better in 1982, but still provided negative energy savings.

In 1981, the San Anselmo School had a net savings per collector square foot per month of 108 watts. The large decrease in performance from 1981 to 1982 was primarily due to the decrease in chiller performance from a COP of 0.6 to 0.22. There was also a slight reduction in the proportional amount of operating energy used in 1982 compared to 1981, but this did not compensate for the decreased chiller performance.

At the University of Minnesota, the poorer performance in 1982 was due to a significant reduction in collector efficiency and an increase in operating energy used to deliver solar energy. The decrease in performance at the Honeywell-Salt River Project was caused by poorer collector performance.

B. EXTRAPOLATED PERFORMANCE AND SAVINGS

Table 13 shows a sensitivity analysis of the percent of incident solar energy used to several system improvements at the El Toro Library solar system. Basically, the improvements would include increasing absorption chiller COP to 0.6, reducing storage and transport losses to 10% or bypassing the storage tank and using the solar energy directly with five percent transport losses. These improvements represent achievable levels of performance. The last improvement represents an advanced technology absorption chiller with a COP of 1.0. This piece of equipment represents what may be possible in the future.

Table 13. PREDICTED SOLAR ABSORPTION COOLING EFFICIENCY

EL TORO LIBRARY

CASE NUMBER	SYSTEM IMPROVEMENT	PERCENT OF INCIDENT INSOLATION: COLLECTED SOLAR ENERGY	STORAGE INPUT	STORAGE OUTPUT	SOLAR COOLING	PERCENT IMPROVEMENT
1.	Chiller COP to 0.6	31	27	22	13	33
2.	5% storage losses	31	27	25	11	17
3.	Storage efficiency to 0.95 and chiller COP to 0.6	31	27	25	15	56
4.	Transport losses to 5% plus Case 3	31	29	28	17	71
5.	5% losses and chiller COP of 0.45	31	-	29	13	35
6.	5% transport losses and chiller COP of 0.6	31	-	29	18	80
7.	10% transport and storage losses and chiller COP of 1.0	31	29	28	28	185

Case 1 shows that if the chiller is operated at a COP of 0.6, there will be a 33% improvement in performance. Contrast that increase in performance to Case 2, where storage losses are reduced to five percent (perhaps by more effective solar utilization) and performance is only increased 17%. Case 3 represents the case where both chiller COP is improved to 0.6 and storage losses are reduced to five percent. The total improvement is 56%; several percent greater than the sum of the two improvements separately. Case 4 would represent the best performance available from this system without design changes. An improvement of 71% is projected.

Cases 5 and 6 represent the El Toro Library system with the storage valved off. These improvements may not be representative of the seasonal performance because the winter loads may not use all the solar energy collected. However, it does seem possible to bypass storage in the summer during library hours. This design change could improve performance by 35% without chiller maintenance. With good chiller maintenance, this change could result in an 80% performance improvement; somewhat better than the best performance without system changes. Perhaps the incremental improvement of nine percent gained by bypassing storage would not merit a change from the present system configuration.

Case 7 represents the El Toro Library system with a double-effect absorption chiller installed. Assuming a COP of 1.0 (1.15

may be possible) for this chiller and 10% transport and storage losses, there would be a 185% performance improvement possible.

The extrapolations of energy savings shown in Table 14 indicate that present systems could perform much better if they were well maintained. This improvement is particularly true of the absorption chiller. At the El Toro Library, most of the storage losses could be avoided with better control of the motor-driven valves. Although most of the control system installation problems were corrected in April 1982, a recent site visit by NSDN personnel disclosed that a control sensor had fallen out of the sensor well. Greater need for careful inspection and maintenance of the system is warranted.

One question still remains to be answered. "Will solar cooling provide cost-effective energy savings with maintenance and new technology?" Authors Warren and Wahlig report in Reference 8 that "Four steps are required to develop a viable solar cooling technology: 1) thermal performance improvement, 2) electric performance improvement, 3) decrease of component costs, and 4) system integration." Using the El Toro Library system as a typical candidate will give the reader some insight to the amount of performance improvement required.

Table 14. EXTRAPOLATED ENERGY SAVINGS

EL TORO LIBRARY

CASE NUMBER	SYSTEM IMPROVEMENT	EQUIVALENT ELECTRICAL SAVINGS (KWH)	OPERATING ENERGY (KWH)	NET SAVINGS (KWH)	COOLING SYSTEM COP
1.	Chiller COP to 0.6	8,580	7,190	1,400	3.0
2.	Reduce storage losses to 5%, chiller COP to 0.6	10,400	7,190	3,200	3.6
3.	Reduce transport and storage losses to 10%, chiller COP to 0.6	11,300	7,190	4,140	3.9
4.	Reduce chiller operating energy by 12% plus Case 3	11,300	6,500	4,820	4.4
5.	Reduce chiller and collector operating energy by 20% plus Case 3	11,300	6,080	5,240	4.7
6.	Double-effect chiller, COP 1.0 plus reduction of transport and storage losses to 10%	18,900	7,190	11,700	6.6
7.	Reduce chiller and collector operating energy by 50% plus Case 6	18,900	3,590	15,300	13

Table 14 shows the extrapolated energy savings and the cooling system COP. Cases 1 through 3 represent the improvement in energy savings which could be expected with the present system with good maintenance. Note that in each case the savings are positive, and in Case 3 amount to 320 W/ft^2-month. These savings are better than those at the Honeywell-Salt River Project site (see Table 12). Case 4 assumes the conditions of Case 3 plus a reduction in chiller operating energy of 12%. The energy savings amount to 375 W/ft^2-month. The reduction in operating energy is justified by the improvement in chiller COP; i.e., there will be less energy rejected and, therefore, less cooling tower operating energy. Case 5 assumes an additional reduction in operating energy by better pump sizing and piping layout. The energy savings now amount to 408 W/ft^2-month. The reduction in collector pumping power has been achieved at other sites, so it is considered reasonable.

Cases 6 and 7 represent the improvement due to an advanced technology absorption chiller. Case 6 shows only the savings from thermal improvements, while Case 7 includes a 50% reduction in operating energy also. For Case 7, the savings would amount to 1,190 W/ft^2-month. This would equal about 39.6 W/ft^2-day and would still be less than the 50 to 65 W/ft^2-day savings presently available from solar space heating systems. Obviously, additional performance improvements and cost reductions will be required before solar cooling becomes cost-effective.

Table 14 also shows the cooling system COP. This is calculated similarly to the COP of a conventional air conditioning system; i.e., cooling produced divided by the operating energy. It is interesting to note that the COP at the El Toro Library could be substantially increased by better maintenance. Case 4, a COP of 4.4, is an estimate of what this increase could amount to without any design or equipment changes. This COP is much greater than the COP of 2.5 to 3.0 available with conventional cooling equipment. However, comparing the Case 4 COP of 4.4 to the Honeywell-Salt River Project COP of 6.4 shows that the Rankine-powered chiller is obviously more efficient in the use of electrical operating energy. Comparing the COP of 13 in Case 7 to the Honeywell-Salt River Project COP of 6.4 makes the absorption chiller system appear much better and the absorption system remains better than the Rankine system compared to the 8.4 COP from the Honeywell-Salt River Project during 1981.

From the preceding comparative analyses of five solar cooling systems, there are several observations on system performance which are summarized below. Conclusions on these and other observations follow these lessons learned.

C. OBSERVATIONS

1. Present system conversion efficiencies ranged from two percent to 10%. That is, 10% of incident solar radiation appeared as cooling delivered to the load. With proper

maintenance of the absorption chiller and control systems, conversion efficiency can be increased to 17% on present systems. Improving chiller technology to a COP of 1.0 could increase the conversion efficiency to 28%.

Following the same logic with the Rankine cooling systems yields a conversion efficiency of 12% with the present system. The assumptions are eight percent Rankine thermal efficiency, 38% collector efficiency, and a 4.2 COP of the vapor compressor. Conversion efficiency can be improved by increasing Rankine inlet temperature to 300°F. At that temperature, the Rankine efficiency could be 13%, and, with 38% collector efficiency and a 4.2 COP of the vapor compressor, a 19% conversion efficiency is possible.

The degradation of system efficiency at the Florida Solar Energy Center also illustrates that a cold storage tank should not be used. Table 8, Absorption Chiller Performance, shows the absorption chiller output for the Florida Solar Energy Center at 75.6 million BTU. If all of this output had been utilized, system efficiency at the Florida Solar Energy Center would have increased from four percent to 10%.

2. Operating energy costs for the collector and cooling subsystems ranged from 149 to 560 W/ft^2 of collector per month. It is not known how much of the operating energy on the cooling subsystem may be weather dependent. However, the Florida Solar Energy Center would have the most humid climate, and it used 516 W/ft^2-month operating energy compared to 560 W/ft^2-month operating energy used at the El Toro Library. Perhaps a figure of 250 W/ft^2-month operating energy is achievable at most installations in the United States.

   Parasitic operating energy is a critical factor in cooling system performance. The impact of the size of pumps and cooling tower fans is illustrated by the range of COPs in Table 4, Solar System Coefficient of Performance. The Rankine system, at a COP of 6.4, is indicative of a well-designed pumping system. Perhaps staging the cooling tower and using a smaller safety factor on pump sizing would help to reduce parasitic energy.

3. System losses of collected energy ranged from 10% to 48%. One absorption cooling system had solar energy losses of 12% compared to the 10% losses at the Rankine site. Storage losses were the major losses at two of the sites, while collector losses were significant at one site. At the University of Minnesota, losses dropped from 64% to 12% during July and August when the storage tank was bypassed. The storage bypass also improved the system efficiency or percentage of incident delivered to the load. This percentage increased from four percent to 10% when the storage was bypassed.

4. The control of a solar absorption cooling system can affect performance significantly. This is illustrated by comparing the results in Figure 16, Solar Cooling Efficiency. THe Honeywell-Salt River Project and the Florida Solar Energy Center have similar efficiencies when comparing the percentage of incident energy to the chiller. However, partially because of a control problem on the cold storage tank, the Florida Solar Energy Center has a final efficiency of less than half of the final efficiency of the Honeywell-Salt River Project. Other control problems occurred at the San Anselmo School and the El Toro Library, but the impacts were not easily quantified. These control problems concerned the collector controller and automatic valve controllers.

5. The El Toro Library achieved a cooling solar fraction of 24% compared to the design cooling solar fraction of 60%. Under present operational conditions, none of the systems could meet the design cooling solar fractions. However, based on better chiller COP and low losses, two of the systems could meet design cooling solar fractions at presently achievable collector efficiencies.

6. Table 5, Collection Subsystem Performance, shows that the flat-plate and evacuated-tube efficiencies were very good compared to the efficiency of the concentrating collector at the University of Minnesota. The level of performance of either the flat-plate or evacuated-tube collectors is easily able to operate the absorption chillers and Rankine engines connected to them. Since collector performance is better on the flat-plate or evacuated-tube collectors, those types of collector systems are recommended for use on absorption chillers over a concentrating, tracking collector system.

7. Space cooling loads ranged from 1,020 to 6,740 BTU/ft2 of floor area-month. Space cooling loads compared on a cooling degree-day basis ranged from 9.4 to 22.9 BTU/ft2-cooling degree-day. As expected, the University of Minnesota had the smallest load on a cooling degree-day basis because it is underground.

   If the building load is normalized and compensated for building temperature above or below 77°F, then the building loads per square foot of floor area per cooling degree-day range from nine to 12 BTU. This result indicates the effect of raising the building temperature 3°F in the case of the El Toro Library.

8. The load thermostat should have a deadband between heating and cooling changeover. This problem occurred at the El Toro Library, but the impact on performance could not be quantified. Although there was a considerable difference between the cooling load per square foot per cooling degree-day from the 1981 cooling season to the 1982

cooling season, some of the difference is due to the greater number of cooling degree-days in 1981.

9. Average monthly cooling system loadings ranged from 0.28 to 1.33 million BTU/ton-month. By this criteria, three of the systems with one million BTU/ton-month have only a 32% load. The Honeywell-Salt River Project has a 43% average loading at 1.33 million BTU/ton-month. Perhaps designers can increase the loadings to get more full load operation and better performance.

10. The Rankine system produced energy savings of 6.3 W/ft^2-day. This is equal to 1/10 of the savings of the best solar heating site, or 1/25 of the savings of the best domestic hot water sites. With operational improvements and maintenance, the present system at the El Toro Library could reach energy savings of 13.8 W/ft^2-day. However, even with an advanced double-effect chiller, energy savings would only approach 40 W/ft^2-day.

11. Prior NSDN data has not supported the practice of placing a boiler in series with the collector on solar absorption cooling systems because chiller COPs were not improved and collector efficiencies appeared to be degraded. However, the data in Table 8, Absorption Chiller Performance, indicates a higher COP for the University of Minnesota and the El Toro Library systems where solar energy is a preheat to the boiler. Also, the collector efficiency for the El Toro Library shown in Table 5, Collection Subsystem Performance, is equal to the other systems. This data indicates that, with evacuated-tube or concentrating collectors, the solar energy system can be used as a boiler preheat to an absorption cooling system with no loss in performance. However, the boiler outlet temperatures on both of these systems were still kept below 195°F.

12. Referring to the chiller COPs shown in Table 8, Absorption Chiller Performance, note that they are all below 0.5. This is poor performance for absorption chillers. The San Anselmo School has had long-term COPs of 0.58 and 0.60 during 1980 and 1981, respectively. However, in 1982, the chiller COP at the San Anselmo School dropped to 0.22 due to a loss of vacuum. Two other NSDN chillers also lost vacuum this year. The loss of performance after two years of operation is an indication that chiller reliability may be low.

13. The average outdoor ambient temperatures are lower than building temperatures at three of the sites. This temperature difference indicates that some portion of the building cooling could be met by ventilating with outside air, particularly in dry climates or during dry periods.

D. CONCLUSIONS

The Rankine engine solar cooling system is better than the solar absorption cooling systems on the basis of energy savings. The Rankine system is also superior on the basis of solar cooling COP. This result is due to low thermal losses and very low operating costs.

Although the Rankine system outperformed the absorption cooling systems, there appears to be more potential for energy savings with the absorption cooling systems. These systems have higher thermal conversion efficiencies than the present Rankine system.

Solar cooling systems are not cost-effective now and do not appear likely to become so in the near future. The large increase in energy savings required for cost-effectiveness would require significant advances in solar cooling technology. For example, to achieve energy savings with absorption cooling comparable to those being obtained in solar domestic hot water systems would require 50% collector efficiencies and an absorption chiller COP of 2.0. Likewise, energy savings for the Rankine systems that would be comparable to those obtained in domestic hot water systems would require 50% collector efficiencies at 300°F temperatures, 13% thermal efficiency, and a vapor compressor COP of 5.0.

Maintenance of control systems, collectors, and absorption chillers is necessary for high performance. The increased maintenance requirement is probably due to aging of these solar cooling systems.

Storage and transport losses can be held to 10% or less. High rates of solar energy utilization and good system maintenance would permit this goal to be achieved.

Flat-plate and evacuated-tube solar collector arrays performed similarly at temperatures below 200°F.

Building loads normalized to account for differences in climate and building temperature required cooling levels of 10 BTU per square foot of floor area per building temperature compensated cooling degree-day. The effect of lowering building temperature 3°F is about 10 BTU/ft^2-cooling degree-day, or equivalent to the internal and environmental gains.

Monthly average loadings on the cooling systems at these sites were only 32% to 43%. This implies oversizing of cooling equipment by a factor of nearly three. Designers should consider sizing equipment to more nearly fit the load since this equipment performs more efficiently at full load.

In summary, solar cooling systems are not cost-effective and require an improvement in performance of 10 times to reach the energy savings of solar heating systems. The Rankine system outperformed the absorption systems for the second year due to better controls and lower operating costs. Maintenance of absorption chillers, control systems, and solar collection systems is essential to keep performance at high levels.

Section V

REFERENCES

1. Thermal Data Requirements and Performance Evaluation Procedures for the National Solar Heating and Cooling Demonstration Program, NBSIR 76-1137, National Bureau of Standards, August 1976.

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*7. Solar Energy System Performance Evaluation, University of Minnesota, April 1982 through August 1982, SOLAR/2032-83/14, Vitro Laboratories, Silver Spring, Maryland.

8. Warren, M.L. and M. Wahlig, "Methodology to Determine Cost and Performance Goals for Active Solar Cooling Systems, in J. of Solar Energy Engineering, May 1983, Vol. 105/217-223.

* Copies of these reports may be obtained from Technical Information Center, P.O. Box 62, Oak Ridge, Tennessee 37830.

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APPENDIX A

SITE DESCRIPTIONS, SITE HISTORY AND PROBLEMS, SCHEMATIC DIAGRAMS, ENERGY FLOW DIAGRAMS, SEASONAL PERFORMANCE DATA, AND LONG-TERM WEATHER CONDITIONS

Each site in this report is described in the following sections, and data is presented indicating the performance of each system over the 1982 cooling season.

SECTION A-1

EL TORO LIBRARY

SITE DESCRIPTION

The El Toro Library is a one-story facility of modern design, located in El Toro, California. The building contains 10,000 square feet of floor area with very few windows, which are located near the building entrances. The library is functional year-round and is occupied Tuesday through Saturday.

The building was designed to incorporate a solar energy system on the south-facing roof. The solar energy system is interconnected to the building space heating and cooling equipment. The solar energy system was designed to provide 97% of the space heating load and 60% of the space cooling load.

The solar energy system incorporates 82 panels with a gross area of 1,427 square feet of evacuated tubular glass collectors (TC-100) manufactured by General Electric. The collectors are oriented 30 degrees west of due south at a tilt of 19 degrees from the horizontal. The collection subsystem utilizes treated city water as a transfer medium from collector to storage tank. The storage tank is a 1,500-gallon insulated steel tank which is located outside, above ground level. The storage tank provides thermal storage for the collected solar energy before delivery to the building load.

The space heating subsystem uses solar energy from storage and/or thermal energy from the natural-gas-fired boiler. The thermal energy is delivered to the air-handling unit, which distributes the energy to the conditioned space.

The space cooling subsystem uses an absorption chiller to provide chilled water to the air-handling unit. The generator portion of the absorption chiller unit uses hot water from the solar storage and/or hot water supplied by the natural-gas-fired boiler.

The manufacturers of the major solar system equipment and components are listed below.

Equipment/Component	Manufacturer	Model No.
Evacuated-Tube Collectors	General Electric	TC-100
Heat Rejector	Young Radiator Co.	22D20
Solar Storage Tank	Santa Fe Tank & Heater Co.	18333
Gas-Fired Boiler	Ray Pak	E602-T
Absorption Chiller	Arkla Corp.	WFB-300
Cooling Tower	Baltimore Aircoil of CA	VXT-45C
Air-Handling Unit (AHU)	Air Dynamics, Inc.	MTW-90
Pumps P1, P2, P3, P4, P5	Frederick Pump Engineering	
3-Way Valves V3, V4, V5-11, V8, V12, V13	Barber Colman	
Expansion Tanks	Wood Products, Inc.	

The system, shown schematically on Page A-4, has nine modes of operation.

Mode 1 - Solar Energy Collection - Solar energy collection occurs when insolation levels are sufficient (as measured by a Barber Colman comparator). When the insolation levels exceed the predetermined set point, collector pump P1 or P2 will activate flow for solar energy collection. This mode behaves like a collector loop warm-up, since all the flow bypasses the storage tank. Pump P1 or P2 will deactivate when insolation levels fall below the set point.

Mode 2 - Collector-to-Storage Flow - Solar energy is delivered to the storage tank when the collector outlet temperature exceeds the temperature in the storage tank. Three-way control valve V5-11 will change position to allow full flow into the storage tank. When the collector outlet temperature falls below the storage tank temperature, valve V5-11 will reverse its position and flow will again bypass the storage tank. (Collection pump P1 or P2 must be operating.) Valve V5-11 has complete control of this mode.

Mode 3 - Solar Storage-to-Space Heating/Cooling Load - This mode occurs when there is a cooling or heating demand and the storage tank temperature is greater than the load loop return temperature. Control valve V8 will allow flow from the load loop return into storage and provide solar heated water to the loads. Valve V8 will continue to deliver stored energy until the load loop return temperature exceeds the storage temperature. Valve V8 will then change position and all flow will bypass the storage tank. Valve V8 has complete control of solar energy delivered to the loads.

Mode 4 - Auxiliary Energy for Heating/Cooling - When the boiler set point is greater than the storage tank temperature, then the auxiliary natural-gas-fired boiler will turn on to meet the energy needs of the building. The boiler will provide energy for the space heating coils or to the generator inlet of the absorption chiller.

Mode 5 - Solar Energy Heat Rejection - This mode will activate when the storage tank temperature exceeds 210°F. Control valve V3 will allow flow to the heat rejector and the fan will dissipate excess collected energy to the environment. The heat rejection mode is for equipment protection from high temperatures.

Mode 6 - Freeze Protection - Stage 1 - This mode will activate collector pump P1 or P2 when the ambient temperature falls below 38°F. All the collector flow will bypass storage. This is the first stage of freeze protection.

Mode 7 - Freeze Protection - Stage 2 - This second stage of freeze protection follows the first stage of freeze protection. The second stage will allow modulation valve V5-11 to use stored energy into the collector loop.

Mode 8 - Freeze Protection - Stage 3 - The third stage of freeze protection will allow flow of city water to the collector loop when the collector outlet temperature falls below 35°F. Valves V1 and V2 will purge city water and discharge flushing water to drain.

Mode 9 - Collector Over-Temperature - If the collector array experiences temperatures greater than 320°F, then the control sensor will lock out solar pumps P1 and P2 and retain valves V1 and V2 in their closed position. This will prevent thermal shock in the collector array.

SITE HISTORY AND PROBLEMS

During December 1981, valve V8 was operating poorly. Upon inspection, it was discovered that the control sensor was mislocated. The control sensor was fixed by installing a larger thermowell and sensor so that it would reach the storage tank. After the new sensor was installed, the valve required adjustment because the return water was warmer than the storage tank. Further adjustments to valve V8 were required in January before the valve worked properly. In February, the valve again malfunctioned due to a short circuit in a sensor cable. Again in April, the valve required adjustment. After this date, valve V8 has worked properly.

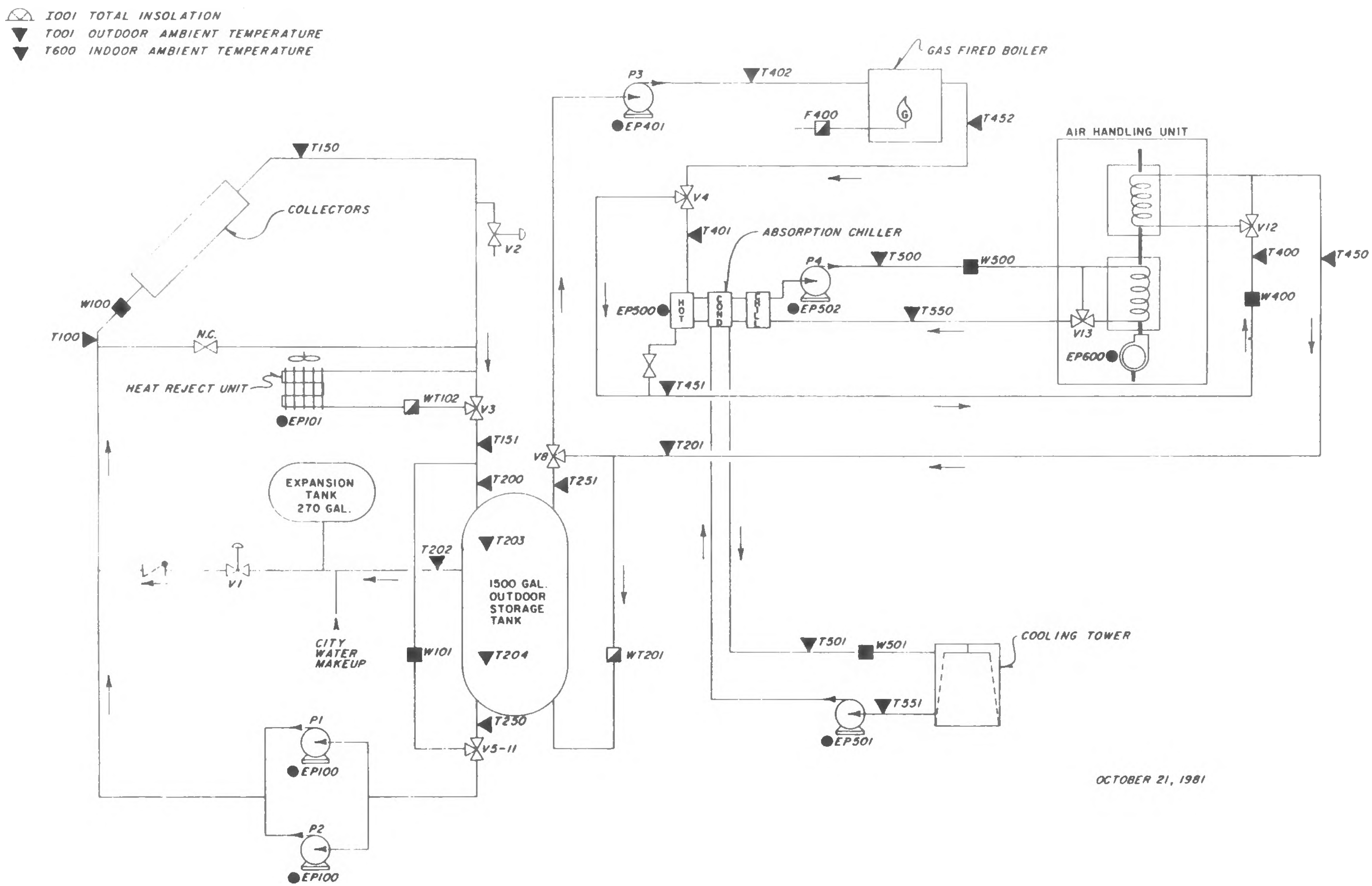
Valve V5-11 stuck open in January 1982. This condition allowed energy to be rejected from storage during collector startup. The control for valve V5-11 was repaired during February 1982.

One of the long-standing problems with the conventional Heating, Ventilating, and Air-Conditioning (HVAC) system has been the lack of a deadband between the heating and cooling thermostats. During February 1982, the thermostats were adjusted in an attempt to prevent simultaneous heating and cooling.

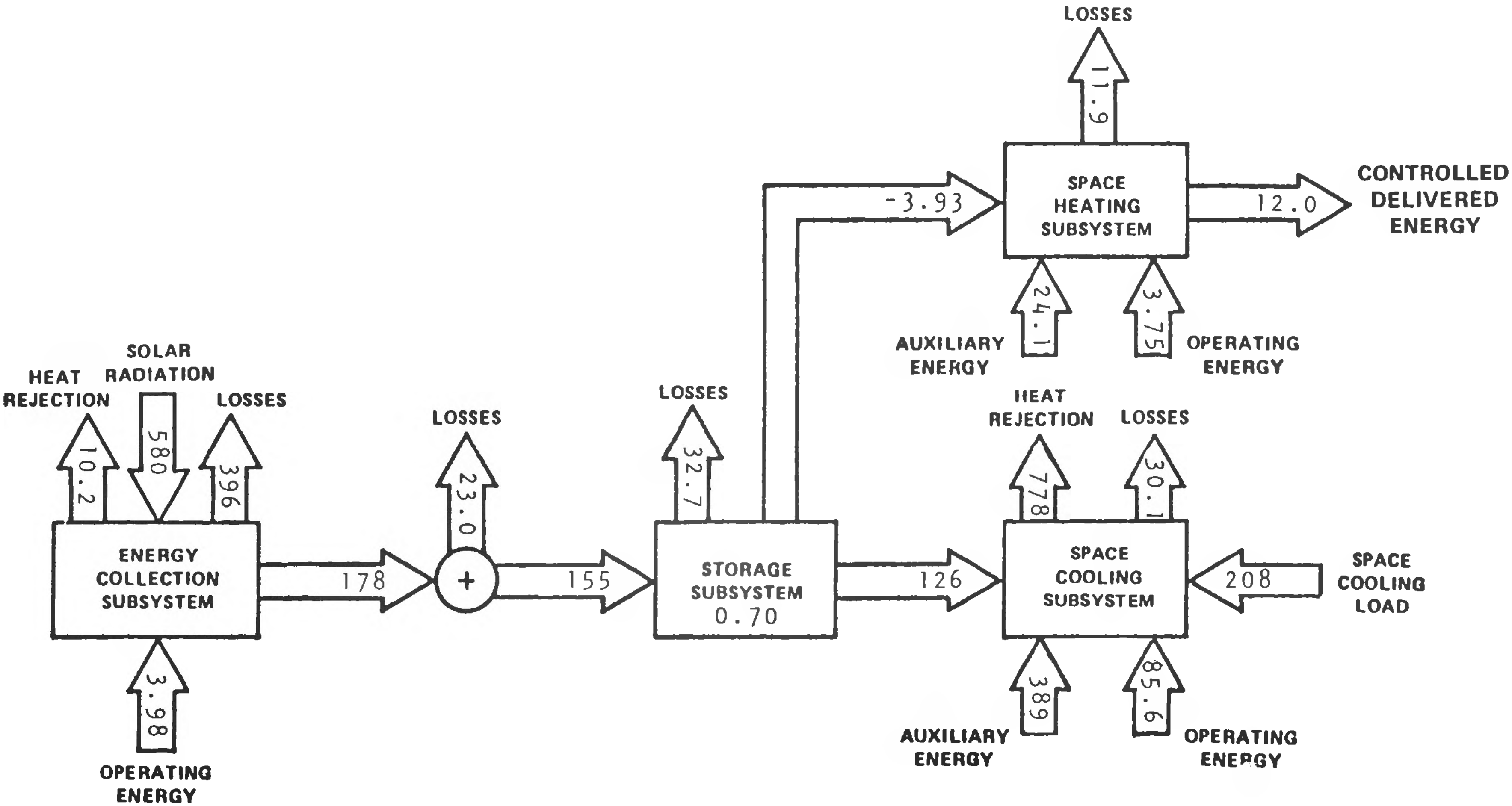
There was some trouble with the collector control set points during March and April 1982. The collector was running too late into the evening and rejecting energy from storage. The collector was running until solar insolation dropped below 39 BTU/ft^2-hr.

In May 1982, the solar chiller operated sometimes when the air-handling units were off. Also, boiler operation was erratic.

The cooling tower pump failed in June 1982, so the solar cooling system was down for four days. Later in the month, there was more trouble with the cooling tower pump. Apparently the heaters on the motor starter were too small.



El Toro Library Solar Energy System Schematic



Energy Flow Diagram for the El Toro Library
December 1981 through August 1982
(Figures in million BTU)

Table 1. SOLAR SYSTEM THERMAL PERFORMANCE

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED (SECA)	SYSTEM LOAD (SYSL)	SOLAR ENERGY USED (SEL)	AUXILIARY ENERGY FOSSIL (AXF)	AUXILIARY ENERGY THERMAL (AXT)	OPERATING ENERGY (SYSOPE)	ENERGY SAVINGS FOSSIL (TSVF)	ENERGY SAVINGS ELECTRICAL (TSVE)	SOLAR FRACTION (%) (SFR)
DEC	12.5	20.9	6.56 E	55.2	44.0	9.99	9.38 E	-0.35	12 E
JAN	14.2	23.3	7.88 E	80.9	52.8	10.4	11.3 E	-0.36	12 E
FEB	13.1	17.5	6.44 E	59.0	38.6	8.37	9.20 E	-0.30	9 E
MAR	17.1	19.0	15.4 E	65.0	42.6	9.63	22.1 E	-0.40	28 E
APR	24.6	21.7	16.1 E	61.1	40.2	10.3	27.2 E	-0.56	28 E
MAY	19.1	21.4	10.5 E	58.0	38.2	9.37	15.0 E	-0.42	22 E
JUN	19.3	17.4	11.5 E	57.6	38.8	9.58	16.5 E	-0.44	23 E
JUL	33.4	41.9	26.4	90.2	61.8	12.9	37.7	-0.65	30
AUG	25.1	37.0	21.2	81.3	55.7	12.7	30.3	-0.50	25
TOTAL	178	220	122 E	608	413	93.2	179 E	-3.98	-
AVERAGE	19.8	24.4	13.6 E	67.6	45.9	10.4	19.9 E	-0.44	22 E

E Denotes estimated value.

Table 2. COLLECTION SUBSYSTEM PERFORMANCE

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION (SEA)	COLLECTED SOLAR ENERGY (SECA)	COLLECTION SUBSYSTEM EFFICIENCY (%) (CLEF)	OPERATIONAL INCIDENT ENERGY (SEOP)	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%) (CLEFOP)	ECSS REJECTED ENERGY (CSRJE)	ECSS OPERATING ENERGY (CSOPE)	SOLAR ENERGY TO LOADS (CSEO)	SOLAR ENERGY TO STORAGE (STEI)	DAYTIME AMBIENT TEMPERATURE (°F) (TA)
DEC	46.9	12.5	27	43.7	29	0.00	0.35	6.56	11.4	69
JAN	51.3	14.2	28	48.3	29	0.00	0.36	7.88	12.7	64
FEB	46.0	13.1	28	41.5	31	0.02	0.30	6.44E	11.5	68
MAR	59.8	17.1	29	51.7	33	0.69	0.40	15.4 E	15.2	67
APR	75.1	24.6	33	70.7	35	1.64	0.56	16.1 E	21.8	72
MAY	61.5	19.1	31	52.0	37	1.25	0.42	10.5 E	16.4	71
JUN	63.4	19.3	31	50.8	38	2.58	0.44	11.5	14.3	72
JUL	93.5	33.4	36	88.8	38	2.59	0.65	26.4	29.5	87
AUG	82.7	25.1	30	68.6	37	1.38	0.50	21.2	22.6	86
TOTAL	580	178	-	516	-	10.2	3.98	122 E	155	-
AVERAGE	64.5	19.8	31	57.3	35	1.13	0.44	13.6 E	17.3	73

E Denotes estimated value.

For a description of acronyms in parentheses, refer to Appendix E.

Table 3. STORAGE PERFORMANCE

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE (STEI)	ENERGY FROM STORAGE (STEO)	CHANGE IN STORED ENERGY (STECH)	STORAGE EFFICIENCY (%) (STEFF)	AVERAGE STORAGE TEMPERATURE (°F) (TST)	EFFECTIVE HEAT LOSS COEFFICIENT (BTU/hr-ft^2-°F)	LOSS FROM STORAGE (STLOSS)
DEC	11.4	6.56	-0.12	57	159	0.25	4.96
JAN	12.7	7.88	0.44	66	159	0.23	4.38
FEB	11.5	6.44E	0.00	64 E	164	0.23 E	5.06 E
MAR	15.2	15.4 E	-0.33	100 E	169	0.31 E	0.13 E
APR	21.8	16.1 E	-0.11	84 E	169	I	5.81 E
MAY	16.4	10.5 E	0.81	69 E	171	0.60 E	5.09 E
JUN	14.3	11.5	-0.57	76	176	0.21	3.37
JUL	29.5	26.4	-0.04	89	176	0.14	3.14
AUG	22.6	21.2	0.62	97	172	0.19	0.78
TOTAL	155	122 E	0.70	-	-	-	32.7 E
AVERAGE	17.3	13.6 E	0.08	79 E	168	0.2 E	3.64 E

E Denotes estimated value.
I Denotes invalid data.

Table 4. SPACE HEATING SUBSYSTEM

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD (EHL)	CONTROLLED DELIVERED ENERGY (CDE)	TOTAL SOLAR ENERGY USED (HSE)	TOTAL AUXILIARY THERMAL USED (HAT)	SOLAR FRACTION OF LOAD (%) (HSFR)	BUILDING TEMPERATURE (°F) (TB)	AMBIENT TEMPERATURE (°F) (TA)
DEC	2.66	2.66	-0.10 E	3.32	-3 E	72	60
JAN	3.76	3.76	-0.57 E	4.71	-14 E	71	56
FEB	2.15	2.15	-2.08 E	5.13	-68 E	72	61
MAR	1.73	1.73	0.22 E	6.25	3 E	72	59
APR	1.20	1.20	-1.40 E	3.93	-55 E	74	63
MAY	0.45	0.45	0.00	0.63	0	74	65
JUN	0.04	0.04	0.00	0.10	0	77	67
JUL	0.00	0.00	0.00	0.00	-	78	76
AUG	0.00	0.00	0.00	0.03	-	78	74
TOTAL	12.0	12.0	-3.93 E	24.1	-	-	-
AVERAGE	1.33	1.33	-0.44 E	2.68	-20 E	74	65

E Denotes estimated value.

For a description of acronyms in parentheses, refer to Appendix E.

Table 4a. SPACE HEATING SUBSYSTEM (Continued)

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD (EHL)	TOTAL SOLAR ENERGY USED (HSE)	TOTAL OPERATING ENERGY (HOPE)	FOSSIL ENERGY SAVINGS (HSVF)	AUXILIARY FOSSIL FUEL (HAF)	HEATING DEGREE-DAYS (#) (HDD)
DEC	2.66	-0.10 E	0.54	-0.14 E	4.93	163
JAN	3.76	-0.57 E	0.69	-0.81 E	7.16	282
FEB	2.15	-2.08 E	0.69	-2.97 E	7.87	131
MAR	1.73	0.22 E	0.83	0.31 E	9.39	196
APR	1.20	-1.40 E	0.51	-2.00 E	6.03	106
MAY	0.45	0.00	0.33	0.00	0.91	42
JUN	0.04	0.00	0.11	0.00	0.13	10
JUL	0.00	0.00	0.00	0.00	0.00	5
AUG	0.00	0.00	0.05	0.00	0.00	0
TOTAL	12.0	-3.93 E	3.75	-5.61 E	36.4	935
AVERAGE	1.33	-0.44 E	0.42	-0.62 E	4.05	104

E Denotes estimated value.

Table 5. SPACE COOLING SUBSYSTEM

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD (CL)	SOLAR FRACTION OF LOAD (CSFR)	SOLAR ENERGY USED (CSE)	OPERATING ENERGY (COPE)	AUXILIARY THERMAL USED (CAT)	FOSSIL ENERGY SAVINGS (CSVF)	AUXILIARY FOSSIL FUEL (CAF)	BUILDING TEMPERATURE (°F) (TB)
DEC	18.2	14	6.66	9.10	40.6	9.51	50.3	72
JAN	19.5	15	8.45	9.39	48.0	12.1	73.8	71
FEB	15.4	20 E	8.52E	7.38	33.5	12.2 E	51.1	72
MAR	17.3	30 E	15.2 E	8.40	36.4	21.7 E	55.6	72
APR	20.5	33 E	17.5 E	9.22	36.3	25.0 E	55.0	74
MAY	20.9	22 E	10.5 E	8.62	37.6	15.0 E	57.1	74
JUN	17.4	23	11.5	9.03	38.7	16.4	57.5	77
JUL	41.9	30	26.4	12.3	61.8	37.7	90.2	78
AUG	37.0	25	21.2	12.2	55.6	30.3	81.2	78
TOTAL	208	-	126 E	85.6	389	180 E	572	-
AVERAGE	23.1	24 E	14.0 E	9.51	43.2	20.0 E	63.5	74

E Denotes estimated value.

For a description of acronyms in parentheses, refer to Appendix E.

Table 6. ABSORPTION CHILLER PERFORMANCE

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD (TCEL)	THERMAL ENERGY INPUT (TCEI)	OPERATING ENERGY (TCEOPE)	REJECTED ENERGY (TCERJE)	COEFFICIENT OF PERFORMANCE (COP) (TCEI/TCEL)
DEC	18.2	45.3	5.70	72.5	0.40
JAN	19.5	52.8	5.72	86.1	0.37
FEB	15.4	40.6	4.43	66.2	0.38
MAR	17.3	46.1	5.06	75.5	0.37
APR	20.5	49.4	5.71	81.5	0.42
MAY	20.9	43.4	5.34	74.3	0.48
JUN	17.4	43.4	5.44	68.3	0.40
JUL	41.9	76.6	8.28	136	0.55
AUG	37.0	64.8	8.08	118	0.57
TOTAL	208	462	53.8	778	-
AVERAGE	23.1	51.4	5.97	86.5	0.45

Table 7. SOLAR OPERATING ENERGY

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

(All values in million BTU)

MONTH	ECSS OPERATING ENERGY (SOLAR-UNIQUE) (CSOPEI)	TOTAL SOLAR OPERATING ENERGY (SYSOPEI)
DEC	0.35	0.35
JAN	0.36	0.36
FEB	0.30	0.30
MAR	0.40	0.40
APR	0.56	0.56
MAY	0.42	0.42
JUN	0.44	0.44
JUL	0.65	0.65
AUG	0.50	0.50
TOTAL	3.98	3.98
AVERAGE	0.44	0.44

For a description of acronyms in parentheses, refer to Appendix E.

Table 8. SOLAR COEFFICIENT OF PERFORMANCE

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

MONTH	SOLAR ENERGY SYSTEM $\frac{SEL}{SYSOPE1}$	COLLECTION SUBSYSTEM $\frac{SECA}{CSOPE}$
DEC	19	36
JAN	22	39
FEB	21	44
MAR	39	43
APR	29	44
MAY	25	45
JUN	26	44
JUL	41	51
AUG	42	51
WEIGHTED AVERAGE*	31	45

* Weighted using $\Sigma(SEL_{month})/\Sigma(SYSOPE1_{month})$ and $\Sigma(SECA_{month})/\Sigma(CSOPE_{month})$

Table 9. ENERGY SAVINGS

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982
(All values in million BTU)

MONTH	SOLAR ENERGY USED (SEL)	SPACE HEATING FOSSIL FUEL (HSVF)	SPACE COOLING FOSSIL FUEL (CSVF)	NET OPERATING ENERGY (CSOPE)	NET ENERGY SAVINGS ELECTRICAL (TSVE)	NET ENERGY SAVINGS FOSSIL FUEL (TSVF)
DEC	6.56 E	-0.14 E	9.51	-0.35	-0.35	9.37 E
JAN	7.88 E	-0.81 E	12.1	-0.36	-0.36	11.3 E
FEB	6.44 E	-2.97 E	12.2 E	-0.30	-0.30	9.23 E
MAR	15.4 E	0.31 E	21.7 E	-0.40	-0.40	22.0 E
APR	16.1 E	-2.00 E	25.0 E	-0.56	-0.56	23.0 E
MAY	10.5 E	0.00	15.0 E	-0.42	-0.42	15.0 E
JUN	11.5	0.00	16.4	-0.44	-0.44	16.4
JUL	26.4	0.00	37.7	-0.65	-0.65	37.7
AUG	21.2	0.00	30.3	-0.50	-0.50	30.3
TOTAL	122 E	-5.61 E	180 E	-3.98	-3.98	174 E
AVERAGE	13.6 E	-0.62 E	20.0 E	-0.44	-0.44	19.4 E

E Denotes estimated value.

For a description of acronyms in parentheses, refer to Appendix E.

Table 10. WEATHER CONDITIONS

EL TORO LIBRARY
DECEMBER 1981 THROUGH AUGUST 1982

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY)		AMBIENT TEMPERATURE (^oF)		HEATING DEGREE-DAYS		COOLING DEGREE-DAYS	
	MEASURED (SE)	LONG-TERM AVERAGE	MEASURED (TA)	LONG-TERM AVERAGE	MEASURED (HDD)	LONG-TERM AVERAGE	MEASURED (CDD)	LONG-TERM AVERAGE
DEC	1,060	1,167	60	54	163	341	0	0
JAN	1,158	1,240	56	53	282	372	0	0
FEB	1,151	1,498	61	55	131	298	13	7
MAR	1,351	1,611	59	56	196	279	5	0
APR	1,755	1,993	63	59	106	177	32	9
MAY	1,390	2,024	65	63	42	94	57	29
JUN	1,480	2,090	67	66	10	38	73	77
JUL	2,114	2,274	76	71	5	0	348	181
AUG	1,870	2,178	74	72	0	0	379	209
TOTAL	-	-	-	-	935	1,599	907	512
AVERAGE	1,481	1,786	65	61	104	178	101	57

For a description of acronyms in parentheses, refer to Appendix E.

EL TORO LIBRARY LONG-TERM WEATHER DATA

COLLECTOR TILT: 19 DEGREES
LATITUDE: 33.7 DEGREES

LOCATION: EL TORO, CALIFORNIA
COLLECTOR AZIMUTH: -30 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1663.	948.	0.56985	1.309	1240.	372	0	53.
FEB	2096.	1235.	0.58915	1.212	1498.	298	7	55.
MAR	2630.	1611.	0.61262	1.118	1802.	279	0	56.
APR	3150.	1928.	0.61208	1.034	1993.	177	9	59.
MAY	3489.	2072.	0.59393	0.977	2024.	94	29	63.
JUN	3616.	2194.	0.60671	0.953	2090.	38	77	66.
JUL	3545.	2363.	0.66676	0.962	2274.	0	181	71.
AUG	3273.	2157.	0.65896	1.010	2178.	0	209	72.
SEP	2812.	1737.	0.61747	1.083	1881.	9	165	70.
OCT	2249.	1357.	0.60338	1.181	1602.	64	70	65.
NOV	1762.	1025.	0.58169	1.284	1316.	195	12	59.
DEC	1540.	870.	0.56504	1.342	1167.	341	0	54.

LEGEND:

HOBAR ==> MONTHLY AVERAGE DAILY EXTRATERRESTRIAL RADIATION (IDEAL) IN BTU/DAY-FT2.
HBAR ==> MONTHLY AVERAGE DAILY RADIATION (ACTUAL) IN BTU/DAY-FT2.
KBAR ==> RATIO OF HBAR TO HOBAR.
RBAR ==> RATIO OF MONTHLY AVERAGE DAILY RADIATION ON TILTED SURFACE TO THAT ON A HORIZONTAL SURFACE FOR EACH MONTH (I.E., MULTIPLIER OBTAINED BY TILTING).
SBAR ==> MONTHLY AVERAGE DAILY RADIATION ON A TILTED SURFACE (I.E., RBAR * HBAR) IN BTU/DAY-FT2.
HDD ==> NUMBER OF HEATING DEGREE DAYS PER MONTH.
CDD ==> NUMBER OF COOLING DEGREE DAYS PER MONTH.
TBAR ==> AVERAGE AMBIENT TEMPERATURE IN DEGREES FAHRENHEIT.

SECTION A-2

FLORIDA SOLAR ENERGY CENTER

SITE DESCRIPTION

The Florida Solar Energy Center is a one-story brick building located in Cape Canaveral, Florida. The building contains approximately 5,000 square feet of floor area, and is almost entirely bounded by brick walls. Only a very small window area was used to prevent excessive passive solar energy gain. The building is usually occupied between the hours of 8:00 a.m. and 5:00 p.m. on weekdays, and is usually unoccupied on Saturdays and Sundays.

The solar energy system is a retrofit design which is connected to the existing space cooling and heating equipment. The solar energy system was designed to supply 70% of the annual space cooling load and 100% of the annual space heating load.

The solar energy system is composed of 2,089 square feet of evacuated-tube collectors, a 3,000-gallon hot storage tank, a 6,000-gallon cold storage tank, a 25-ton absorption chiller, pumps, connecting pipe lines, and controls to operate the various system components. The existing cooling tower, electric chiller, oil-fired burner, and Air-Handling Unit (AHU) were unaltered except for control configuration.

The collector array is located in a field near the building. The array is oriented due south at a tilt of 15 degrees to the horizontal. The collection subsystem uses treated city water as a transfer medium between the collector array and the hot storage tank. In the heating mode, solar energy from storage is delivered to the space-heating coils in the air-handling unit. In the cooling mode, solar energy is delivered to the absorption chiller which provides chilled water for cold storage or directly to the space cooling coils in the air-handling unit. If solar energy is unable to meet the cooling demand, an auxiliary electric chiller will provide the remaining space cooling load. An auxiliary oil-fired burner will satisfy the space heating load if there is not enough solar energy.

The manufacturers of the major solar energy system equipment and components are listed below.

Equipment/Component	Manufacturer	Model No.
Evacuated-Tube Collectors	General Electric	TC-100
Hot Storage Tank	Modern Welding Co.	Custom-made
Cold Storage Tank	Modern Welding Co.	Custom-made
25-Ton Absorption Chiller	Arkla Corporation	WFB-300
Electric Chiller	Trane	SCMZ204C
Cooling Tower	Goodfellow	P-60
Pumps P1, P2, P3, P4, P5, P6	Taco Mfg.	-
Electric Chiller Condensing Pump	Franklin Electric	1113940400
Oil-Fired Burner	National	10-47
Controllers	Robertshaw	-

The system, shown schematically on Page A-17, has seven modes of operation.

Mode 1 - Solar Collection and Collector-to-Storage - These two operations occur simultaneously when the collector pump P1 is activated. Pump P1 operates when the collector outlet temperature exceeds the storage tank temperature by 5°F. During collector pump operation, solar energy is collected and delivered to storage at the same time. When the temperature differential falls below 5°F, then the collector pump is shut off and this mode of operation is terminated.

Mode 2 - Heat Rejection - When the hot storage tank temperature exceeds 235°F, pump P6 will activate and transfer energy between the hot and cold storage tanks. The pump will deactivate when the hot storage tank falls below 230°F and terminates the heat rejection mode.

Mode 3 - Storage-to-Space Heating - When there is a space heating demand and solar energy is available in storage, then pump P2 is activated to deliver solar energy directly to the heating coils in the Air-Handling Unit (AHU). Pump P2 will continue to operate until the heating demand is satisfied or solar energy is depleted from storage.

Mode 4 - Auxiliary Space Heating - When there is a space heating demand and solar energy is insufficient, then an auxiliary oil-fired burner will supply the remaining space heating load.

Mode 5 - Solar-to-Cold Storage and/or Space Cooling - When the hot storage tank exceeds 185°F, then pump P2, the solar chiller and related equipment become operational. Chilled water is pumped to the cold storage tank or to the cooling coils in the AHU if there is a cooling demand. This mode will continue to operate until the hot storage tank temperature falls below 170°F.

Mode 6 - Cold Storage-to-Space Cooling - If the cold storage tank temperature is below 58°F and there is a space cooling demand, then pump P4 will deliver chilled water to the cooling coils in the AHU. This mode will terminate when the cold storage tank temperature rises above 60°F.

Mode 7 - Auxiliary Space Cooling - This mode activates when there is a space cooling demand and solar chilled water is insufficient. The auxiliary electric chiller will supply the remaining space cooling load.

SITE HISTORY AND PROBLEMS

The solar system at the Florida Solar Energy Center became operational in May 1979. Initial problems included collector tube breakage and some problems with the electronic controls. Some portions of the construction and installation labor were performed by the Florida Solar Energy Center personnel to reduce costs. Also, the evacuated-tube panels were installed with the evacuated tubes lined up in the horizontal (east-west) position rather than the manufacturer's recommended vertical (north-south) position. However, the typical due south orientation was maintained.

The instrumentation suite installation and the operational checkout were completed July 1981. However, there were initial problems with the site data acquisition subsystem and the flow meters. Upon overcoming the initial problems, the first monthly performance report was prepared for the December 1981 data month. Performance evaluation has continued to the present date.

The solar system operated well from January 1982 through April 1982. However, in May several problems were encountered and system performance deteriorated from then to the end of the year. These problems are summarized below.

- Solar/auxiliary changeover mode control problem.

  In April 1982, the solar/auxiliary control switchover failed, causing the solar cooling water pump to operate continuously unless controlled manually. The control is designed to permit either solar or auxiliary cooling on demand, depending on the availability of solar chiller output. Due to the control switchover failure, auxiliary and solar cooling occurred simultaneously.

  The control failure caused the auxiliary chiller to provide most of the space cooling even when the solar space cooling equipment was operating. The impact of this problem resulted in a low solar contribution, higher auxiliary energy usage, higher losses, and larger quantities of operating energy usage without a substantial reduction in the auxiliary load.

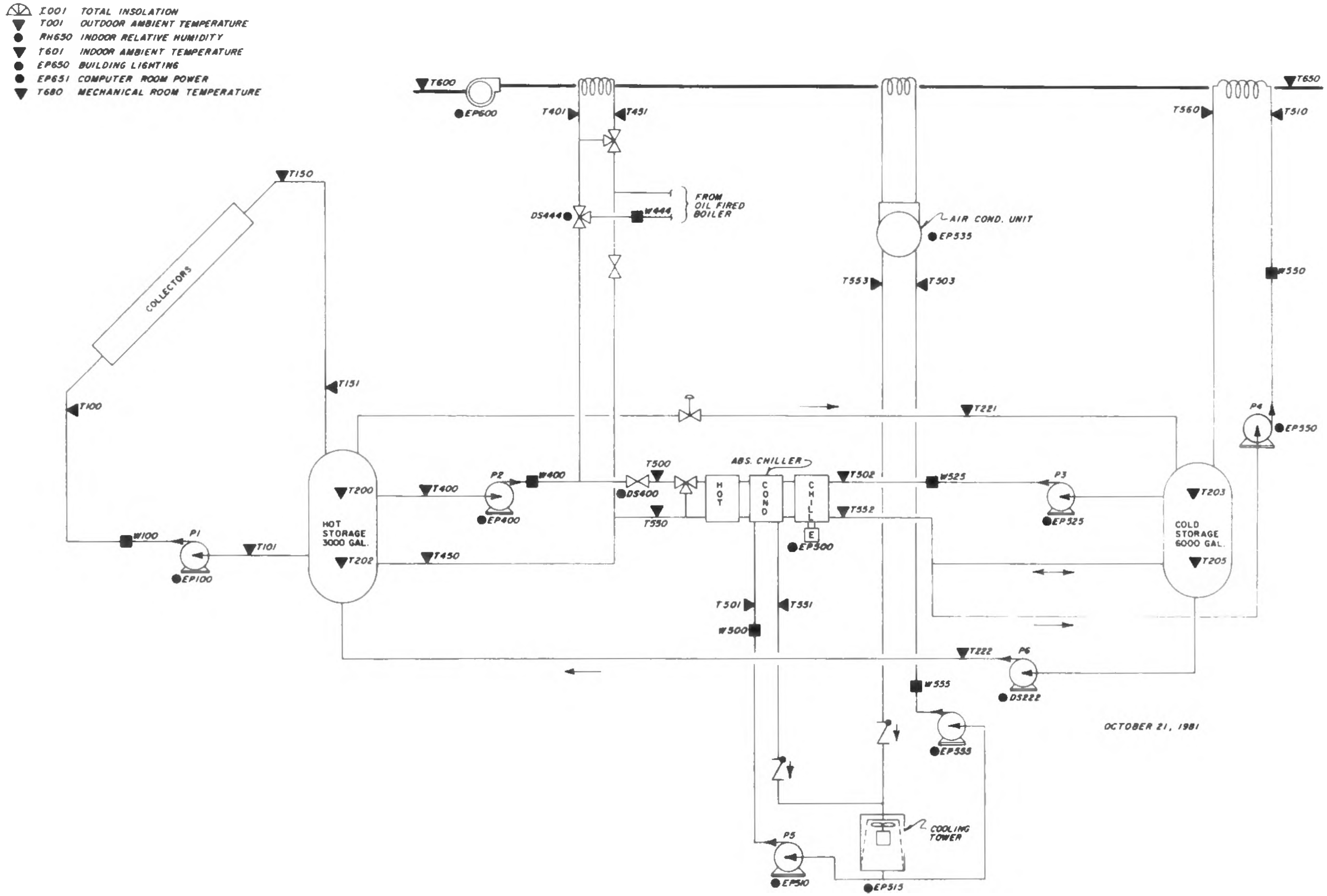
- Low absorption chiller COP.

  The absorption chiller COP was very low, at 0.27, compared to the manufacturer's performance specification of approximately 0.60. A problem with the solar chiller cycling on and off had an influence upon the chiller COP. Also, due to the humidity experienced in Florida, the evaporative cooling tower seemed to be limited as to the amount of energy which could be rejected. This also lowered the chiller COP. The solar chiller seems to be very sensitive to proper operating parameters, and this is another possible reason that the chiller COP is not up to the manufacturer's performance specifications.

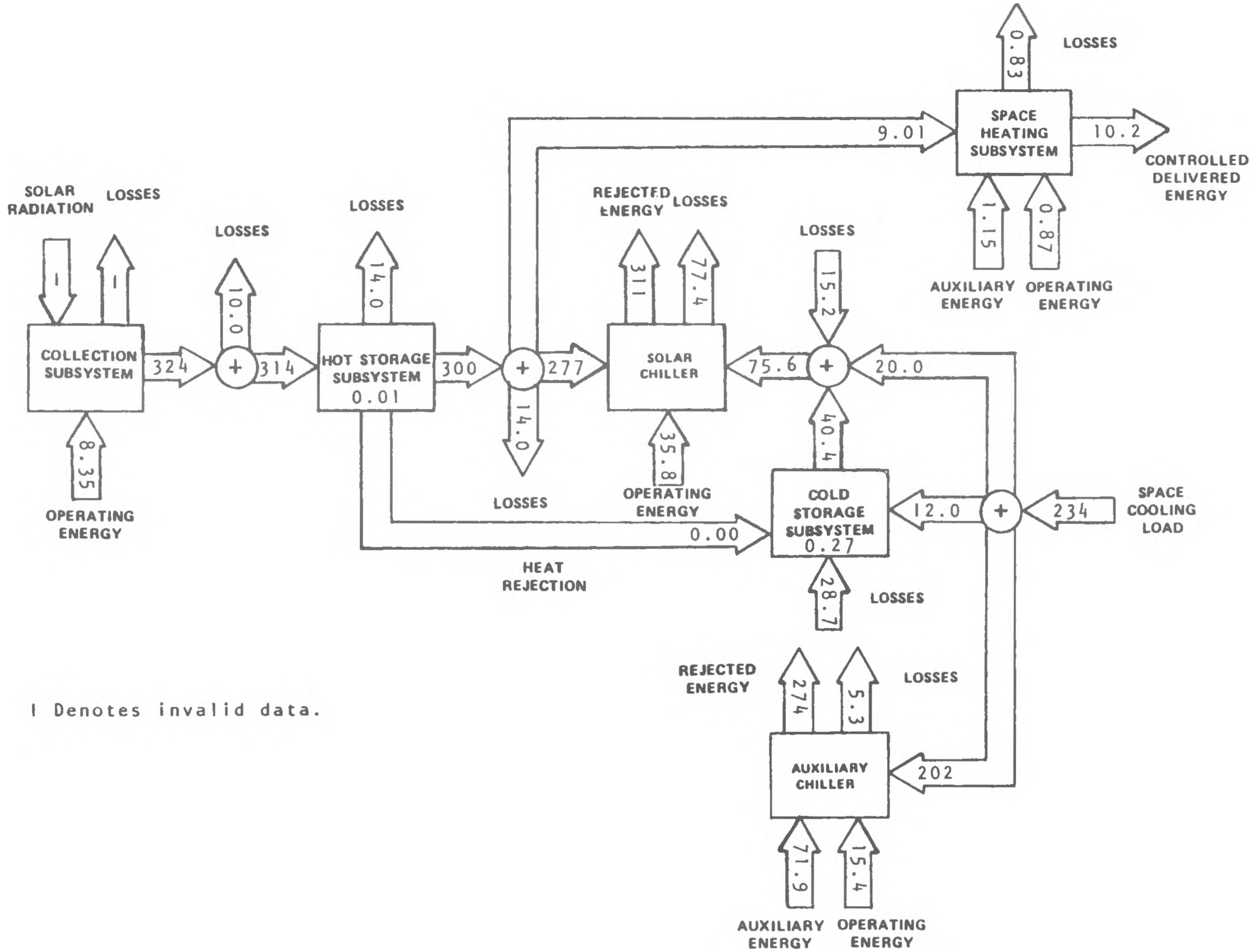
- Chilled water storage utilization.

  Chilled water in the cold storage tank was available for usage, but, due to the control problem, was not utilized to its full potential. This reduced the solar contribution, allowed the solar chiller to cycle more frequently, and caused higher cold storage losses. Solar chilled water in the cold storage tank should be utilized when available.

There is a piping connection between the hot and cold storage tanks. This connection was designed to be used for heat rejection, but was never activated during the reporting period. This connection allowed heat conduction between the tanks, which increased the storage losses for the tanks. This piping connection should be eliminated and, if necessary, a heat rejector should be installed upstream of the collector pump for overheat protection.



Florida Solar Energy Center Solar Energy System Schematic



Energe Flow Diagram for the Florida Solar Energy Center
January 1982 through December 1982
(Figures in million BTU)

Table 1. SOLAR SYSTEM THERMAL PERFORMANCE

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED (SECA)	SYSTEM LOAD (SYSL)	SOLAR ENERGY USED (SEL)	AUXILIARY ENERGY THERMAL (AXT)	AUXILIARY ENERGY FOSSIL (AXF)	AUXILIARY ENERGY ELECTRICAL (AXE)	SOLAR OPERATING ENERGY (SYSOPE1)	ENERGY SAVINGS FOSSIL (TSVF)	ENERGY SAVINGS ELECTRICAL (TSVE)	SOLAR FRACTION (%) (SFR)
JAN	17.3	13.8	16.2	1.61	0.98	1.19	1.69	14.0	-1.19	70
FEB	20.2	10.3	16.0	2.14	0.00	2.51	2.36	0.00	-1.20	28
MAR	30.7	14.7	29.2	2.28	0.00	2.68	3.50	1.01	-1.12	44
APR	32.1 E	15.1	29.3	2.84	0.00	3.34	3.68	0.00	-1.61	34
MAY	34.3 E	15.4	31.0	3.99	0.00	4.69	4.76	0.00	-3.28	24
JUN	33.9 E	32.0	30.7	8.34	0.00	9.81	5.00	0.00	-4.49	4
JUL	44.1 E	32.4	40.1	8.70	0.00	10.2	6.57	0.00	-5.28	10
AUG	40.0	31.4	35.6	8.96	0.00	10.5	5.73	0.00	-4.58	9
SEP	28.3 E	26.6	25.3	7.42	0.00	8.73	4.09	0.00	-2.97	11
OCT	22.4 E	24.1	19.3	5.92	0.00	6.97	3.25	0.00	-2.51	8
NOV	12.0	16.4	7.20	6.29	0.00	7.40	2.02	0.00	-1.68	5
DEC	9.02	11.8	6.04	3.82	0.94	3.83	1.68	0.00	-1.65	1
TOTAL	324	244	286	62.3	1.92	71.9	44.3	15.0	-31.6	-
AVERAGE	27.0	20.3	23.8	5.19	0.16	5.99	3.69	1.25	-2.63	17

E Denotes estimated value.

Table 2. COLLECTION SUBSYSTEM PERFORMANCE

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION (SEA)	COLLECTED SOLAR ENERGY (SECA)	COLLECTION SUBSYSTEM EFFICIENCY (%) (CLEF)	OPERATIONAL INCIDENT ENERGY (SEOP)	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%) (CLEFOP)	ECSS OPERATING ENERGY (CSOPE)	SOLAR ENERGY TO LOADS (CSEO)	SOLAR ENERGY TO STORAGE (STEI)	DAYTIME AMBIENT TEMPERATURE (°F) (TDA)
JAN	80.8	17.3	21	52.3	33	0.61	16.2	16.9 E	69
FEB	78.2	20.2	26	49.6	41	0.54	16.0	19.3 E	73
MAR	10.6	30.7	29	75.0	41	0.79	29.2	29.9 E	77
APR	101	32.1 E	32 E	71.5	45 E	0.75	29.3	31.3 E	79
MAY	116	34.3 E	30 E	78.5	44 E	0.82	31.0	33.6 E	83
JUN	105	33.9 E	32 E	76.3	44 E	0.84	30.7	32.6 E	87
JUL	121	44.1 E	36 E	96.9	46 E	1.03	40.1	42.9 E	89
AUG	I	40.0	I	I	I	0.91	35.6	39.1	89
SEP	I	28.3 E	I	I	I	0.72	25.3	27.8 E	87
OCT	I	22.4 E	I	I	I	0.61	19.3	21.8 E	83
NOV	I	12.0	I	I	I	0.43	7.20	10.6	78
DEC	67.9	9.02	13	24.1	37	0.30	6.04	8.33	75
TOTAL	776*	222*	-	524*	-	8.35	286	314	-
AVERAGE	97.0*	27.8*	29*	65.5*	42*	0.70	23.8	26.2	81

E Denotes estimated value.
I Denotes invalid data.
* Summary statistics based upon eight months of data, January through July, and December.

For a description of acronyms in parentheses, refer to Appendix E.

Table 3. HOT STORAGE PERFORMANCE

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE (STEI)	ENERGY FROM STORAGE (STEO)	CHANGE IN STORED ENERGY (STECH)	STORAGE EFFICIENCY (%) (STEFF)	AVERAGE STORAGE TEMPERATURE (°F) (TST)	EFFECTIVE HEAT LOSS COEFFICIENT (BTU/hr-ft^2-°F) (STPER)	LOSS FROM STORAGE (STLOSS)
JAN	16.9 E	16.5 E	0.06	98 E	154	0.01	0.34 E
FEB	19.3 E	18.4 E	0.07	96 E	158	0.00	0.83 E
MAR	29.9 E	29.6 E	0.04	99 E	167	0.01	0.26 E
APR	31.3 E	30.4 E	-0.16	97 E	167	0.01	1.06 E
MAY	33.6 E	31.7 E	-0.01	94 E	163	0.01	1.91 E
JUN	32.6 E	31.5 E	-0.04	97 E	160	0.01	1.14 E
JUL	42.9 E	41.3 E	0.17	97 E	161	0.01	1.43 E
AUG	39.1	37.9 E	0.01	97 E	163	0.01	1.19 E
SEP	27.8 E	26.2 E	0.09	95 E	161	0.01	1.51 E
OCT	21.8 E	20.5 E	-0.19	93 E	163	0.01	1.49 E
NOV	10.6	8.28	1.32	91	175	0.01	1.00
DEC	8.33	7.69	-1.35	76	165	0.01	1.99
TOTAL	314	300	0.01	-	-	-	14.2
AVERAGE	26.2	25.0	0.00	96 (1)	163	0.01	1.18

E Denotes estimated value.

(1) Weighted average = $[\Sigma(STEO_{month}) + \Sigma(STECH_{month})]/\Sigma(STEI_{month})$

Table 4. COLD STORAGE PERFORMANCE

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE (ASTEI)	ENERGY FROM STORAGE (ASTEO)	CHANGE IN STORED ENERGY (ASTECH)	STORAGE EFFICIENCY (%) (ASTEFF)	AVERAGE STORAGE TEMPERATURE (°F) (ATST)	LOSS FROM STORAGE (ASTLOSS)
JAN	0.69	2.18	-0.32	46	58	1.17
FEB	1.38	3.16	-0.12	48	53	1.66
MAR	2.17	5.41	0.01	40	53	3.25
APR	1.04	4.06	-0.28	33	53	2.74
MAY	1.69	4.06	0.74	23	49	3.11
JUN	0.57	4.32	-0.65	28	48	3.10
JUL	0.94	4.27	-0.11	25	49	3.22
AUG	1.14	4.06	0.14	25	49	3.06
SEP	0.96	4.16	0.19	19	52	3.39
OCT	0.88	2.50	-0.06	38	53	1.56
NOV	0.51	0.70	0.66	-21	58	0.85
DEC	0.00	1.48	0.07	-5	60	1.55
TOTAL	12.0	40.4	0.27	-	-	28.7
AVERAGE	1.0	3.36	0.02	29 (1)	53	2.39

(1) Weighted average = $[\Sigma(ASTEI_{month}) - \Sigma(ASTECH_{month})]/\Sigma(ASTEO_{month})$

For a description of acronyms in parentheses, refer to Appendix E.

Table 5. SPACE HEATING SUBSYSTEM

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR HEATING LOAD (EHL)	CONTROLLED DELIVERED ENERGY (CDE)	TOTAL SOLAR ENERGY USED (HSE)	TOTAL AUXILIARY THERMAL USED (HAT)	SOLAR FRACTION OF LOAD (%) (HSFR)	BUILDING TEMPERATURE (°F) (TB)	AMBIENT TEMPERATURE (°F) (TAVE)
JAN	8.99	8.99	8.40	0.59	93	74	65
FEB	0.00	0.00	0.00	0.00	0	74	70
MAR	0.61	0.61	0.61	0.00	100	76	73
APR	0.00	0.00	0.00	0.00	0	77	76
MAY	0.00	0.00	0.00	0.00	0	78	79
JUN	0.00	0.00	0.00	0.00	0	78	84
JUL	0.00	0.00	0.00	0.00	0	79	84
AUG	0.00	0.00	0.00	0.00	0	79	86
SEP	0.00	0.00	0.00	0.00	0	79	84
OCT	0.00	0.00	0.00	0.00	0	77	79
NOV	0.00	0.00	0.00	0.00	0	76	76
DEC	0.56	0.56	0.00	0.56	0	76	72
TOTAL	10.2	10.2	9.01	1.15	-	-	-
AVERAGE	0.85	0.85	0.75	0.10	88 (1)	77	77

(1) Weighted average = $\Sigma(HSE_{month})/\Sigma(EHL_{month})$

Table 5a. SPACE HEATING SUBSYSTEM (Continued)

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD (EHL)	MEASURED SOLAR ENERGY USED (HSEM)	TOTAL OPERATING ENERGY (HOPE)	SOLAR-SPECIFIC OPERATING ENERGY (HOPEI)	FOSSIL ENERGY SAVINGS (HSVF)	AUXILIARY FOSSIL FUEL (HAF)	HEATING DEGREE DAYS (#) (HDD)
JAN	8.99	8.40	0.77	0.21	14.0	0.98	109
FEB	0.00	0.00	0.00	0.00	0.00	0.00	3
MAR	0.61	0.61	0.04	0.01	1.01	0.00	13
APR	0.00	0.00	0.00	0.00	0.00	0.00	0
MAY	0.00	0.00	0.00	0.00	0.00	0.00	0
JUN	0.00	0.00	0.00	0.00	0.00	0.00	0
JUL	0.00	0.00	0.00	0.00	0.00	0.00	0
AUG	0.00	0.00	0.00	0.00	0.00	0.00	0
SEP	0.00	0.00	0.00	0.00	0.00	0.00	0
OCT	0.00	0.00	0.00	0.00	0.00	0.00	0
NOV	0.00	0.00	0.00	0.00	0.00	0.00	0
DEC	0.56	0.00	0.06	0.00	0.00	0.94	28
TOTAL	10.2	9.01	0.87	0.22	15.0	1.92	153
AVERAGE	0.85	0.75	0.07	0.02	1.25	0.16	13

For a description of acronyms in parentheses, refer to Appendix E.

Table 6. SPACE COOLING SUBSYSTEM

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD (CL)	SOLAR FRACTION OF LOAD (%) (CSFR)	SOLAR ENERGY USED (CSE)	OPERATING ENERGY (COPE)	SOLAR-SPECIFIC OPERATING ENERGY (COPEI)	AUXILIARY THERMAL USED (CAT)	AUXILIARY ELECTRICAL FUEL (CAE)	BUILDING TEMPERATURE (°F) (TB)
JAN	4.76	26	7.84	1.74	0.87	1.02	1.19	74
FEB	10.3	28	16.0	3.84	1.82	2.14	2.51	74
MAR	14.1	42	28.6	5.06	2.70	2.28	2.68	76
APR	15.1	34	29.3	5.30	2.93	2.84	3.34	77
MAY	15.4	24	31.0	6.27	3.94	3.99	4.69	78
JUN	32.0	4	30.7	7.60	4.16	8.34	9.81	78
JUL	32.4	10	40.1	8.89	5.54	8.70	10.2	79
AUG	31.4	9	35.6	8.34	4.82	8.96	10.5	79
SEP	26.6	11	25.3	6.85	3.37	7.42	8.73	79
OCT	24.1	8	19.3	5.64	2.64	5.92	6.97	77
NOV	16.4	5	7.20	4.58	1.59	6.29	7.40	76
DEC	11.3	1	6.04	3.57	1.38	3.26	3.83	76
TOTAL	234	-	277	67.7	35.8	61.2	71.9	-
AVERAGE	19.5	14 (1)	23.1	5.64	2.98	5.10	5.99	77

(1) Weighted average = $\Sigma(CLS_{month})/\Sigma(CL_{month})$

Table 7. SOLAR CHILLER PERFORMANCE

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD (TCEL)	THERMAL ENERGY INPUT (TCEI)	OPERATING ENERGY (TCEOPE)	REJECTED ENERGY (TCERJE)	COEFFICIENT OF PERFORMANCE (DIMENSIONLESS) (COP) (TCECOP)
JAN	3.13	7.84	0.87	9.48	0.40
FEB	5.66	16.0	1.82	18.4	0.35
MAR	11.2	28.6	2.70	34.7	0.39
APR	10.5	29.3	2.93	35.2	0.36
MAY	7.30	31.0	3.94	33.7	0.24
JUN	5.51	30.7	4.16	31.8	0.18
JUL	8.17	40.1	5.54	43.6	0.20
AUG	8.09	35.6	4.82	40.4	0.23
SEP	7.07	25.3	3.37	28.9	0.28
OCT	5.80	19.3	2.64	22.7	0.31
NOV	1.51	7.20	1.59	6.26	0.21
DEC	1.61	6.04	1.38	6.22	0.27
TOTAL	75.6	277	35.8	311	-
AVERAGE	6.30	23.1	2.98	25.9	0.27 (1)

(1) Weighted average = $\Sigma(TCEL_{month})/\Sigma(TCEI_{month})$

For a description of acronyms in parentheses, refer to Appendix E.

Table 8. AUXILIARY CHILLER PERFORMANCE

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD (ATCEL)	THERMAL ENERGY INPUT (ATCEI)	OPERATING ENERGY (ATCEOPE)	REJECTED ENERGY (ATCERJE)	COEFFICIENT OF PERFORMANCE (DIMENSIONLESS) (COP) (ATCECOP)
JAN	3.52	1.19	0.17	4.71	2.96
FEB	7.37	2.51	0.37	9.88	2.94
MAR	8.10	2.68	0.38	10.8	3.02
APR	9.93	3.34	0.51	13.3	2.97
MAY	11.7	4.69	0.68	16.4	2.50
JUN	30.7	9.81	1.77	40.5	3.13
JUL	29.2	10.2	1.74	39.4	2.85
AUG	28.5	10.5	3.45	39.1	2.70
SEP	23.8	8.73	3.41	32.5	2.72
OCT	22.3	6.97	0.99	29.3	3.20
NOV	15.5	7.40	0.93	22.9	2.10
DEC	11.2	3.83	0.98	15.0	2.92
TOTAL	202	71.9	15.4	274	-
AVERAGE	16.8	5.99	1.28	22.8	2.81 (1)

(1) Weighted average = $\Sigma(ATCEL_{month})/\Sigma(ATCEI_{month})$

Table 9. SOLAR OPERATING ENERGY

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU, unless otherwise indicated)

MONTH	ECSS OPERATING ENERGY (CSOPE)	SHS OPERATING ENERGY (HOPEI)	SCS OPERATING ENERGY (COPEI)	TOTAL SOLAR OPERATING ENERGY (SYSOPEI)
JAN	0.61	0.21	0.87	1.69
FEB	0.54	0.00	1.82	2.36
MAR	0.79	0.01	2.70	3.50
APR	0.75	0.00	2.93	3.68
MAY	0.82	0.00	3.94	4.76
JUN	0.84	0.00	4.16	5.00
JUL	1.03	0.00	5.54	6.57
AUG	0.91	0.00	4.82	5.73
SEP	0.72	0.00	3.37	4.09
OCT	0.61	0.00	2.64	3.25
NOV	0.43	0.00	1.59	2.02
DEC	0.30	0.00	1.38	1.68
TOTAL	8.35	0.22	35.8	44.3
AVERAGE	0.70	0.02	2.98	3.69

For a description of acronyms in parentheses, refer to Appendix E.

Table 10. SOLAR COEFFICIENT OF PERFORMANCE

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

MONTH	SOLAR ENERGY SYSTEM $\left(\frac{SEL}{SYSOPE}\right)$	COLLECTION SUBSYSTEM $\left(\frac{SECA}{CSOPE}\right)$	SPACE HEATING SUBSYSTEM $\left(\frac{HSEM}{HOPET}\right)$	SPACE COOLING SUBSYSTEM $\left(\frac{CSE}{COPET}\right)$
JAN	9.61	28.4	40.0	9.01
FEB	6.76	37.4	0.00	8.77
MAR	8.33	38.9	61.0	10.6
APR	7.96	42.8 E	0.00	10.0
MAY	6.51	41.8 E	0.00	7.87
JUN	6.14	40.4 E	0.00	7.38
JUL	6.11	42.8 E	0.00	7.24
AUG	6.21	44.0	0.00	7.38
SEP	6.18	39.3 E	0.00	7.50
OCT	5.95	36.7 E	0.00	7.32
NOV	3.56	27.9	0.00	4.53
DEC	3.60	30.1	0.00	4.38
WEIGHTED AVERAGE (1)	6.46	38.8	41.0	7.74

E Denotes estimated value.
(1) Summation of numerator for the month divided by summation of denominator for the month.

Table 11. ENERGY SAVINGS

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

(All values in million BTU)

MONTH	SOLAR ENERGY USED (SEL)	SPACE HEATING ELECTRICAL (HSVE)	SPACE HEATING FOSSIL FUEL (HSVF)	SPACE COOLING ELECTRICAL (CSVE)	ECSS OPERATING ENERGY SOLAR-UNIQUE (CSOPE)	NET ENERGY SAVINGS ELECTRICAL (TSVE)	NET ENERGY SAVINGS FOSSIL FUEL (TSVF)
JAN	16.2	-0.21	14.0	-0.37	-0.61	-1.19	14.0
FEB	16.0	0.00	0.00	-0.66	-0.54	-1.20	0.00
MAR	29.2	-0.01	1.01	-0.32	-0.79	-1.12	1.01
APR	29.3	0.00	0.00	-0.86	-0.75	-1.61	0.00
MAY	31.0	0.00	0.00	-2.46	-0.82	-3.28	0.00
JUN	30.7	0.00	0.00	-3.65	-0.84	-4.49	0.00
JUL	40.1	0.00	0.00	-4.25	-1.03	-5.28	0.00
AUG	35.6	0.00	0.00	-3.67	-0.91	-4.58	0.00
SEP	25.3	0.00	0.00	-2.25	-0.72	-2.97	0.00
OCT	19.3	0.00	0.00	-1.90	-0.61	-2.51	0.00
NOV	7.20	0.00	0.00	-1.25	-0.43	-1.68	0.00
DEC	6.04	0.00	0.00	-1.35	-0.30	-1.65	0.00
TOTAL	286	-0.22	15.0	-23.0	-8.35	-31.6	15.0
AVERAGE	23.8	-0.02	1.25	-1.92	-0.70	-2.63	1.25

For a description of acronyms in parentheses, refer to Appendix E.

Table 12. WEATHER CONDITIONS

FLORIDA SOLAR ENERGY CENTER
JANUARY 1982 THROUGH DECEMBER 1982

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY) MEASURED (SEC)	LONG-TERM AVERAGE	TEMPERATURE (°F) MEASURED (TAVE)	LONG-TERM AVERAGE	HEATING DEGREE-DAYS MEASURED (HDD)	LONG-TERM AVERAGE	COOLING DEGREE-DAYS MEASURED (CDD)	LONG-TERM AVERAGE
JAN	1,250	1,185	65	67	109	53	102	121
FEB	1,340	1,405	70	68	3	67	150	145
MAR	1,630	1,684	73	71	13	17	248	212
APR	1,620	1,897	76	75	0	0	322	300
MAY	1,790	1,880	79	78	0	0	436	403
JUN	1,680	1,709	84	81	0	0	560	480
JUL	1,860	1,716	84	82	0	0	631	536
AUG	I	1,669	86	83	0	0	642	555
SEP	I	1,539	84	82	0	0	570	501
OCT	I	1,399	79	78	0	0	438	397
NOV	I	1,279	76	72	0	13	328	229
DEC	1,050	1,108	72	68	28	56	233	159
TOTAL	-	-	-	-	153	206	4,660	4,038
AVERAGE	1,530*	1,539(1,573*)	77	75	13	17	388	337

I Denotes Invalid data.
* Based on eight months, January through July, and December.

For a description of acronyms in parentheses, refer to Appendix E.

FLORIDA SOLAR ENERGY CENTER LONG-TERM WEATHER DATA

COLLECTOR TILT: 15 DEGREES
LATITUDE: 28.5 DEGREES

LOCATION: CAPE CANAVERAL, FLORIDA
COLLECTOR AZIMUTH: 0 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1931.	970.	0.50208	1.222	1185.	53	121	67.
FEB	2331.	1217.	0.52202	1.155	1405.	67	145	68.
MAR	2801.	1552.	0.55418	1.085	1684.	17	212	71.
APR	3231.	1866.	0.57748	1.017	1897.	0	300	75.
MAY	3485.	1939.	0.55644	0.969	1880.	0	403	78.
JUN	3571.	1796.	0.50280	0.952	1709.	0	480	81.
JUL	3518.	1788.	0.50824	0.960	1716.	0	536	82.
AUG	3318.	1678.	0.50559	0.995	1669.	0	555	83.
SEP	2948.	1464.	0.49645	1.051	1539.	0	501	82.
OCT	2463.	1243.	0.50455	1.126	1399.	0	397	78.
NOV	2022.	1058.	0.52346	1.208	1279.	13	229	72.
DEC	1814.	892.	0.49175	1.242	1108.	56	159	68.

LEGEND:

HOBAR ==> MONTHLY AVERAGE DAILY EXTRATERRESTRIAL RADIATION (IDEAL) IN BTU/DAY-FT2.
HBAR ==> MONTHLY AVERAGE DAILY RADIATION (ACTUAL) IN BTU/DAY-FT2.
KBAR ==> RATIO OF HBAR TO HOBAR.
RBAR ==> RATIO OF MONTHLY AVERAGE DAILY RADIATION ON TILTED SURFACE TO THAT ON A HORIZONTAL SURFACE FOR EACH MONTH (I.E., MULTIPLIER OBTAINED BY TILTING).
SBAR ==> MONTHLY AVERAGE DAILY RADIATION ON A TILTED SURFACE (I.E., RBAR * HBAR) IN BTU/DAY-F
HDD ==> NUMBER OF HEATING DEGREE DAYS PER MONTH.
CDD ==> NUMBER OF COOLING DEGREE DAYS PER MONTH.
TBAR ==> AVERAGE AMBIENT TEMPERATURE IN DEGREES FAHRENHEIT.

SECTION A-3

HONEYWELL-SALT RIVER PROJECT

SITE DESCRIPTION

The Honeywell-Salt River Project site is the Crosscut Operation and Maintenance Building at the Salt River Project in Phoenix, Arizona. The solar energy system provides energy for space cooling, space heating, and electrical power generation. The system is designed to provide 16% of the seasonal cooling load and 89% of the seasonal heating load. There are 55,000 square feet of conditioned space. The system contains an 8,208-square-foot collector array composed of 456 Lennox flat-plate collectors. The array is mounted on the roof at a tilt of 20 degrees and is facing 34 degrees west of south. The collector fluid is a 20% ethylene glycol/water solution.

Space cooling is provided by two 25-ton vapor compressors and a dual compressor, 228-ton Westinghouse centrifugal chiller. The two 25-ton compressors are each coupled to a solar-driven Rankine engine. In the absence of a space cooling load, the Rankine engines are used to drive generators to produce electrical energy.

Solar space heating is provided by circulating solar heated water from a 2,500-gallon storage tank to three wall-mounted unit heaters. Auxiliary heating is provided by manually controlled electric radiant heaters.

The manufacturers of the major solar equipment and components are listed below:

Equipment/Components	Manufacturer	Model No.
Flat-plate Collectors	Lennox	LSC-18-1
Rankine/Vapor Compressor	Lennox/Barber-Nichols	
Centrifugal Chiller	Westinghouse	TS240-B

The solar system, shown schematically on Page A-30, has the following operating modes:

<u>Mode 1 - Collector-to-Storage</u> - The system enters the collector-to-storage mode if the collector plate temperature rises 5°F above the storage fluid temperature and the system is in the winter operation mode (a manual switchover). Pump P6 or P7 is activated and valve V2 is positioned to A-AB. This mode is continued until the plate temperature drops below the storage fluid temperature or the storage fluid temperature rises above 190°F.

<u>Mode 2 - Storage-to-Space Heating</u> - When the storage fluid temperature is above 140°F and there is a call for heating, pump P8 is activated, pumping solar heated water from storage to the three unit space heaters.

Mode 3 - Auxiliary Heating - When the solar heating subsystem is unable to meet the space heating requirements, manually controlled electric radiant heaters are activated.

Mode 4 - Solar Cooling - Solar cooling can be provided during both the summer and winter modes of operation. During the summer mode of operation, the collector pumps are activated when the collector plate temperature reaches 165°F. Valve V2 is positioned to allow full collector flow to the Rankine engines. When the collector fluid temperature can be maintained at 160°F, Rankine engine Number 1 is started, and when the fluid temperature reaches 170°F, Rankine engine Number 2 is started. Each Rankine engine is mechanically coupled to a 25-ton vapor compressor. When the collector fluid temperature drops below 160°F, Rankine engine Number 2 is deactivated and, at 150°F, Rankine engine Number 1 is deactivated.

During the winter mode of operation, the collector-to-Rankine loop can be activated when the collector plate temperature is lower than the storage fluid temperature or when the storage fluid temperature is higher than 190°F. In these cases, the Rankine start-up and turn-off logic remains the same as mentioned above.

Pumps P4 and P3 are activated to deliver chilled water to the conditioned space whenever the vapor compressors are operating.

Mode 5 - Generation - Electric power is generated using the solar-powered Rankine engine to drive the auxiliary electric motor as a generator during the heating or cooling season.

Mode 6 - Auxiliary Cooling Mode - If the Rankine engines are unable to provide the required power to the vapor compressors, an auxiliary motor coupled between each Rankine engine and the compressor is used to provide the balance of the required power. The 228-ton centrifugal chiller is activated whenever the two 25-ton vapor compressors are unable to satisfy the cooling load.

Mode 7 - Heat Rejection Mode - When the collector fluid rises above 212°F, the purge fan, monitored by sensor EP102, is activated. Valve V1 is positioned to allow partial flow from the collector to the purge coils at 212°F and full flow at 220°F.

SITE HISTORY AND PROBLEMS

This section is reproduced verbatim from the Honeywell Final Report (see Footnote 1).

"Most of the components in the solar energy system performed well. . . . The Rankine-cycle air conditioner, being a prototype

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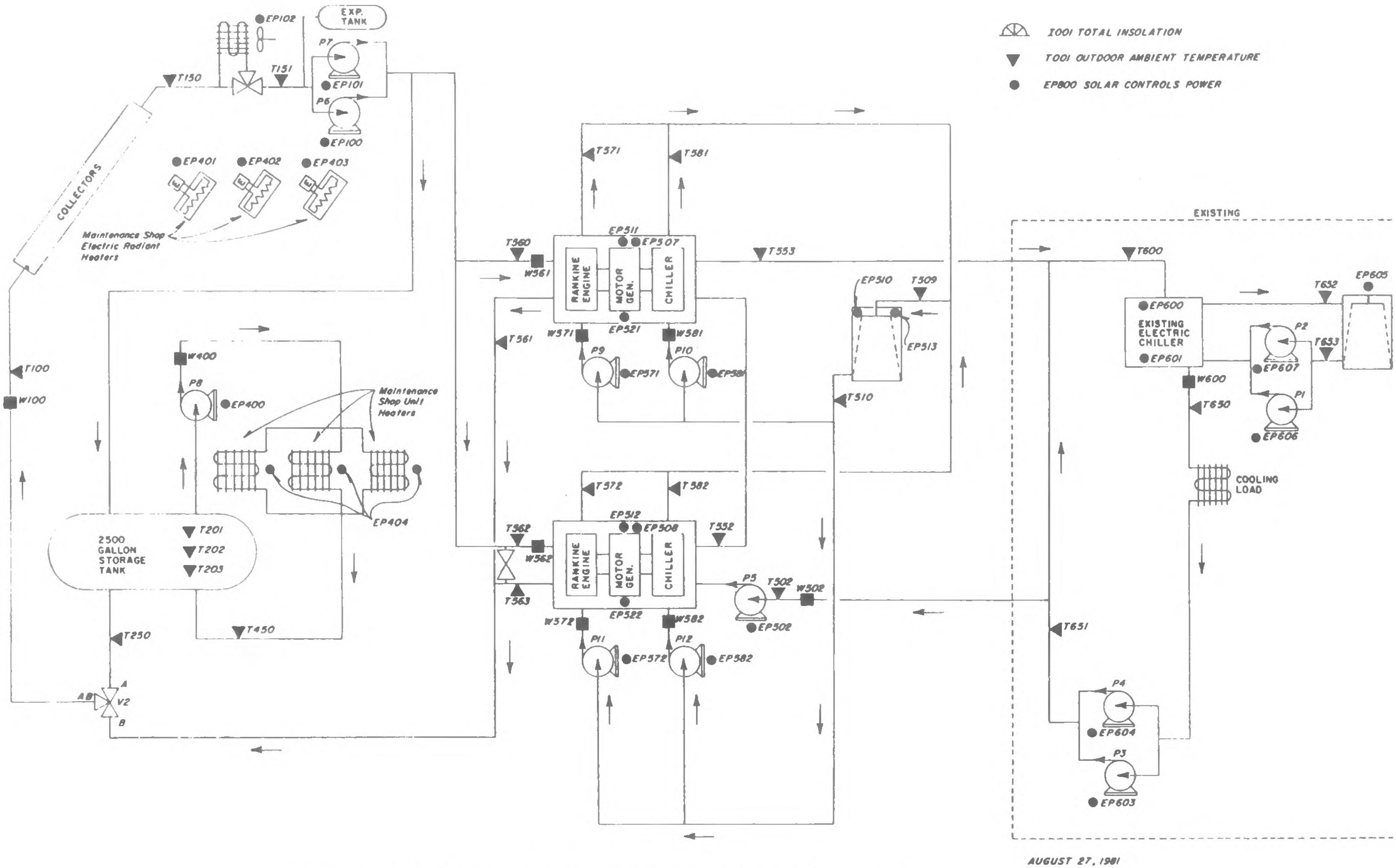
[1] Solar Energy System Performance Evaluation - Final Report for Honeywell OTS 45 Salt River Project, Phoenix, Arizona, DOE/NASA CR-, Honeywell Inc., Roseville, Minnesota, June 1981.

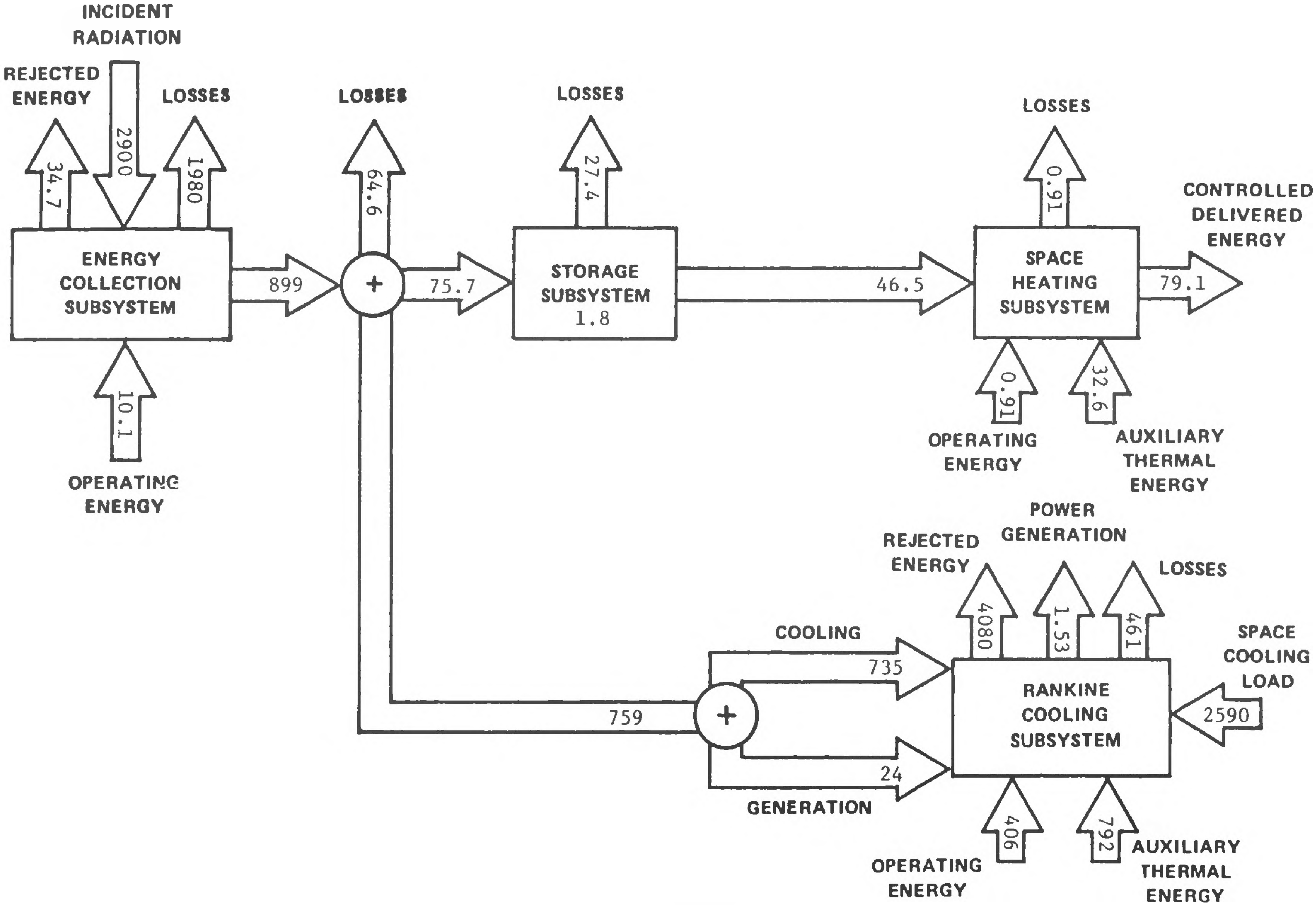
machine, experienced some unintentional shutdowns. Unit No. 1 occasionally did not start due to low lubrication pressure. Most of the system maintenance on this unit was performed to rectify this problem. These efforts have not been completely successful and, at the end of the test period, the unit was still experiencing this problem.

"Since the Rankine condensers operate at subatmospheric pressures, there is a possibility of air leaking into the system. An automatic air purge system is provided with each unit to discharge this air. Occasional problems with these automatic systems were experienced. Therefore, the system was periodically purged manually.

"The heating units, located about 200 feet from the thermal storage tank, initially blew cold air for a few minutes. This was because a warm-up period was needed to provide the heat loss in piping. To rectify this, a 5-minute time delay relay was installed on the fan motors. In addition, when the supply temperature dropped below 140°F, the air temperature from the unit heaters was at an uncomfortable level. Thus, the minimum temperature set points for discharging from storage to heating and for charging storage were raised to 140°F."



Honeywell-Salt River Project Solar Energy System Schematic



Energy Flow Diagram for the Honeywell-Salt River Project
September, October, December 1981
January, February, July, August 1982
(Figures in million BTU)

Table 1. SOLAR SYSTEM THERMAL PERFORMANCE

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED (SECA)	SYSTEM LOAD (SYSL)	SOLAR ENERGY USED (SEL)	AUXILIARY ENERGY ELECTRICAL (AXE)	OPERATING ENERGY (SYSOPE1)	ENERGY SAVINGS ELECTRICAL (TSVE)	SOLAR FRACTION (%) (SFR)
SEP	152	625	142	228	5.14	7.4	6
OCT	118	329	116	110	4.10	5.4	9
DEC	59	79[1]	41	*	2.32	16.0	*
JAN	64	48	45	26	2.33	13.0	33
FEB	106	35[2]	75	39	3.98	15.5[3]	*
JUL	208	723	196	255	6.25	9.9	6
AUG	190	826	190	284	5.07	9.7	4
TOTAL	897	2,670	805	942	29.2	76.9	-
AVERAGE	128	381	115	157	4.17	11.0	8

* Denotes unavailable data.
1 Includes only solar portion of heating load.
2 Includes only heating load, cooling load not available.
3 Cooling savings not included.

Table 2. COLLECTION SUBSYSTEM PERFORMANCE

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION (SEA)	COLLECTED SOLAR ENERGY (SECA)	COLLECTION SUBSYSTEM EFFICIENCY (%) (CLEF)	OPERATIONAL INCIDENT ENERGY (SEOP)	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%) (CLEFOP)	ECSS OPERATING ENERGY (CSOPE)	SOLAR ENERGY DIRECTLY TO LOADS (RSE)	SOLAR ENERGY TO STORAGE (STEI)	DAYTIME AMBIENT TEMPERATURE (^{o}F) (TDA)
SEP	467	152	32	399	38	1.73	142	0.0	94
OCT	423	118	28	332	36	1.32	116	1.9	82
DEC	315	59	19	198	30	1.40	25	28.6	67
JAN	299	64	21	179	36	0.91	33	18.0	58
FEB	330	106	32	254	42	1.72	57	26.5	67
JUL	560	208	37	497	42	2.04	196	0.7	101
AUG	510	190	37	442	43	0.97	190	0.0	99
TOTAL	2,900	897	-	2,300	-	10.1	759	75.7	-
AVERAGE	415	128	31	329	39	1.44	108	10.8	81

For a description of acronyms in parentheses, refer to Appendix E.

Table 3. STORAGE PERFORMANCE

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE	ENERGY FROM STORAGE	CHANGE IN STORED ENERGY	STORAGE EFFICIENCY (%)	AVERAGE STORAGE TEMPERATURE (°F)	EFFECTIVE HEAT LOSS COEFFICIENT (BTU/hr-°F-ft²)	LOSS FROM STORAGE
	(STEI)	(STEO)	(STECH)	(STEFF)	(TST)	(STPER)	(STLOSS)
SEP	0.0	0.0	-0.3	N.A.	99	0.12	0.3
OCT	1.9	0.1	1.5	80	92	0.06	0.3
DEC	28.6	16.2	0.8	59	172	0.46	11.6
JAN	18.0	12.5	0.3	71	173	0.27	5.2
FEB	26.5	17.7	-0.1	66	175	0.37	8.9
JUL	0.7	0.0	-0.3	-50	105	0.18	1.0
AUG	0.0	0.0	-0.1	N.A.	93	0.08	0.1
TOTAL	75.7	46.5	1.8	-	-	-	27.4
AVERAGE	10.8	6.6	0.3	64	130	0.31	3.9

N.A. Denotes not applicable.

Table 4. SPACE HEATING SUBSYSTEM

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD	TOTAL SOLAR ENERGY USED	SOLAR FRACTION OF LOAD (%)	TOTAL AUXILIARY THERMAL USED	AUXILIARY ELECTRIC FUEL	TOTAL OPERATING ENERGY	AMBIENT TEMPERATURE (°F)	HEATING DEGREE-DAYS (#)
	(EHL)	(HSE)	(HSFR)	(HAT)	(HAE)	(HOPE)	(TA)	(HDD)
SEP	0.0	0.0	N.A.	0.0	0.0	0.00	87	0
OCT	0.1	0.1	100	0.0	0.0	0.01	72	5
DEC	16.2[1]	16.2	*	*	*	0.30	57	184
JAN	28.0	12.5	45	15.5	15.5	0.33	52	354
FEB	34.8	17.7	51	17.1	17.1	0.27	59	164
JUL	0.0	0.0	N.A.	0.0	0.0	0.00	92	0
AUG	0.0	0.0	N.A.	0.0	0.0	0.00	91	0
TOTAL	79.1	46.5	-	32.6	32.6	0.91	-	707
AVERAGE	11.3	6.6	48[2]	5.4	5.4	0.13	73	101

N.A. Denotes not applicable.
* Denotes unavailable data.
[1] Solar portion of load only, auxiliary contribution not available.
[2] Average solar fraction for October, January, and February.

For a description of acronyms in parentheses, refer to Appendix E.

Table 5. SPACE COOLING SUBSYSTEM

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD (CL)	SOLAR FRACTION OF LOAD (%) (CSFR)	SOLAR ENERGY USED (CSE)	OPERATING ENERGY (COPE)	AUXILIARY THERMAL USED (CAT)	AUXILIARY ELECTRIC FUEL (CAE)	AMBIENT TEMPERATURE (°F) (TA)	COOLING DEGREE-DAYS (#) (CDD)
SEP	625	6	137	87.1	194	228	87	657
OCT	329	9	107	55.0	93	110	72	245
DEC	63	10	24	24.5	19	22	57	3
JAN	20	20	30	21.5	9	10	52	0
FEB	*	*	51	24.2	19	22	59	14
JUL	723	6	196	94.3	217	255	92	819
AUG	826	4	190	99.2	241	284	91	823
TOTAL	2,590	-	735	406	792	931	-	2,560
AVERAGE	431	7	105	58.0	113	133	73	366

* Denotes unavailable data.

Table 6. AUXILIARY CHILLER PERFORMANCE

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD (ATCEL)	THERMAL ENERGY INPUT (ATCEI)	COEFFICIENT OF PERFORMANCE (COP) (ATCECOP)
SEP	395	136	2.5
OCT	147	50	2.5
DEC	20	8	2.1
JAN	1	0.4	3.1
FEB	12	4	2.5
JUL	479	154	2.6
AUG	635	185	2.9
TOTAL	1,690	537	-
AVERAGE	241	77	2.7

For a description of acronyms in parentheses, refer to Appendix E.

Table 7. RANKINE NUMBER 1 SUMMARY

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY USED (RSE)	TURBINE OUTPUT (RANKOUT)	RANKINE THERMAL EFF (%) (REFF)	AUXILIARY ELECTRIC USED (RSCAE)	PARA-SITIC POWER (PARA)	OPERATING ENERGY (ROPE)	ENERGY REJECTED (RSRJE)	POWER GENERATED (PWRGEN)	COOLING PRODUCED (OUTVC)	COEFFICIENT OF PERFORMANCE (COP) (RSCOP)	SOLAR WATER TEMP (°F) (TRANKS)	COND WATER TEMP (°F) (TRANKC)
SEP	64	4.7	7	32.8	0.49	9.7	223	0.16	132	3.10	181	82
OCT	52	3.4	7	26.8	1.49	7.8	181	0.01	106	3.08	175	79
DEC	25	1.9	8	12.5	0.41	3.9	78	0.06	43	2.61	173	74
JAN	21	1.5	7	7.1	0.40	3.2	36	0.06	9	0.90	181	74
FEB	22	1.6	7	5.9	0.37	4.5	*	0.16	*	*	169	76
JUL	119	8.2	7	33.5	0.55	10.3	255	0.01	107	2.45	174	84
AUG	116	7.6	7	34.5	0.56	10.3	247	0.00	102	2.27	177	86
TOTAL	419	28.9	-	153	3.19	49.7	1,020	0.46	499	-	-	-
AVERAGE	60	4.1	7	21.9	0.46	7.1	146	0.07	83	2.60	176	79

* Denotes unavailable data.

Table 8. RANKINE NUMBER 2 SUMMARY

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY USED (RSE)	TURBINE OUTPUT (RANKOUT)	RANKINE THERMAL EFF (%) (REFF)	AUXILIARY ELECTRIC USED (RSCAE)	PARA-SITIC POWER (PARA)	OPERATING ENERGY (ROPE)	ENERGY REJECTED (RSRJE)	POWER GENERATED (PWRGEN)	COOLING PRODUCED (OUTVC)	COEFFICIENT OF PERFORMANCE (COP) (RSCOP)	SOLAR WATER TEMP (°F) (TRANKS)	COND WATER TEMP (°F) (TRANKC)
SEP	78	6.1	8	35.1	0.53	9.8	215	0.27	108	2.40	181	82
OCT	64	4.8	8	24.3	0.50	8.3	169	0.53	85	2.63	175	79
DEC	0	0	N.A.	0.0	0.36	5.0	0	0.00	0	N.A.	N.A.	N.A.
JAN	12	0.9	8	2.8	0.38	5.0	24	0.08	9	1.19	181	76
FEB	35	2.7	8	11.5	0.41	5.5	92	0.18	47	2.64	168	76
JUL	77	5.5	7	39.9	0.56	10.3	248	0.00	137	2.73	174	85
AUG	74	5.0	7	31.7	0.55	10.3	191	0.01	89	2.12	177	88
TOTAL	340	25.0	-	145	3.29	54.2	939	1.07	475	-	-	-
AVERAGE	49	3.6	8	20.8	0.47	7.7	134	0.15	68	2.45	176	81

N.A. Denotes not applicable.

For a description of acronyms in parentheses, refer to Appendix E.

Table 9. SOLAR OPERATING ENERGY

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU)

MONTH	ECSS OPERATING ENERGY (CSOPE)	POWER GENERATION OPERATING ENERGY (GENOPE)	SHS OPERATING ENERGY (HOPE1)	SCS OPERATING ENERGY (COPE1)	TOTAL SOLAR OPERATING ENERGY (SYSOPE1)
SEP	1.73	0.30	0.00	3.11	5.14
OCT	1.32	0.25	0.01	2.52	4.10
DEC	1.40	0.02	0.30	0.60	2.32
JAN	0.91	0.06	0.33	1.03	2.33
FEB	1.72	0.55	0.27	1.44	3.98
JUL	2.04	0.00	0.00	4.21	6.25
AUG	0.97	0.00	0.00	4.10	5.07
TOTAL	10.1	1.18	0.91	17.0	29.2
AVERAGE	1.44	0.17	0.13	2.43	4.17

Table 10. SOLAR COEFFICIENT OF PERFORMANCE

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

MONTH	SOLAR ENERGY SYSTEM $\left(\frac{SEL}{SYSOPE1}\right)$	COLLECTION SUBSYSTEM $\left(\frac{SECA}{CSOPE}\right)$	SOLAR POWER GENERATION $\left(\frac{PWRGEN}{GENOPE}\right)$	SPACE HEATING SOLAR $\left(\frac{HSE}{HOPE1}\right)$	SPACE COOLING SOLAR $\left(\frac{CSE}{COPE1}\right)$
SEP	28	88	1.4	N.A.	44
OCT	28	90	2.2	5	42
DEC	18	42	3.0	54	40
JAN	19	70	2.3	38	29
FEB	19	62	0.6	66	35
JUL	31	100	N.A.	N.A.	47
AUG	37	200	N.A.	N.A.	46
AVERAGE	28	89	1.3	51	43

N.A. Denotes not applicable.

For a description of acronyms in parentheses, refer to Appendix E.

Table 11. ENERGY SAVINGS

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

(All values in million BTU)

MONTH	SOLAR ENERGY USED (SEL)	SPACE HEATING ELECTRICAL (HSVE)	POWER GENERATION ELECTRICAL (PWRSVE)	SPACE COOLING ELECTRICAL (CSVE)	ECSS OPERATING ENERGY SOLAR UNIQUE (CSOPE)	NET ENERGY SAVINGS ELECTRICAL (TSVE)
SEP	142	0.0	0.13	9.0	1.73	7.4
OCT	116	0.1	0.29	6.3	1.32	5.4
DEC	41	15.9	0.04	1.5	1.40	16.0
JAN	45	12.2	0.08	1.6	0.91	13.0
FEB	75	17.4	-0.21	*	1.72	15.5[1]
JUL	196	0.0	0.01	11.9	2.04	9.9
AUG	190	0.0	0.01	10.7	0.97	9.7
TOTAL	805	45.6	0.35	41.0	10.1	76.9
AVERAGE	115	6.5	0.05	6.8	1.4	11.0

* Denotes unavailable data.
1 Cooling savings not included.

Table 12. WEATHER CONDITIONS

HONEYWELL-SALT RIVER PROJECT
SEPTEMBER, OCTOBER, DECEMBER 1981
JANUARY, FEBRUARY, JULY, AUGUST 1982

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY)		AMBIENT TEMPERATURE (^{o}F)		HEATING DEGREE-DAYS		COOLING DEGREE-DAYS	
	MEASURED (SE)	LONG-TERM AVERAGE	MEASURED (TA)	LONG-TERM AVERAGE	MEASURED (HDD)	LONG-TERM AVERAGE	MEASURED (CDD)	LONG-TERM AVERAGE
SEP	1,898	2,203	87	84	0	0	657	563
OCT	1,664	1,893	72	72	5	16	245	239
DEC	1,239	1,261	57	53	184	387	6	0
JAN	1,174	1,348	52	51	354	427	0	0
FEB	1,437	1,686	59	55	164	292	24	13
JUL	2,201	2,388	92	91	0	0	819	812
AUG	2,004	2,314	91	89	0	0	823	747
TOTAL	-	-	-	-	707	1,122	2,574	2,374
AVERAGE	1,660	1,870	73	71	101	160	368	339

For a description of acronyms in parentheses, refer to Appendix E.

HONEYWELL-SALT RIVER PROJECT LONG-TERM WEATHER DATA

COLLECTOR TILT: 20 DEGREES
LATITUDE: 33.5 DEGREES

LOCATION: PHOENIX, ARIZONA
COLLECTOR AZIMUTH: -34 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1672.	1021.	0.61072	1.320	1348.	427	0	51.
FEB	2105.	1375.	0.65337	1.226	1686.	292	13	55.
MAR	2636.	1814.	0.68807	1.127	2045.	184	20	60.
APR	3154.	2356.	0.74707	1.040	2450.	59	140	68.
MAY	3489.	2677.	0.76719	0.975	2609.	0	355	76.
JUN	3615.	2736.	0.75685	0.946	2588.	0	587	85.
JUL	3544.	2489.	0.70223	0.960	2388.	0	812	91.
AUG	3275.	2293.	0.70022	1.009	2314.	0	747	89.
SEP	2818.	2017.	0.71580	1.092	2203.	0	563	84.
OCT	2256.	1578.	0.69935	1.199	1893.	16	239	72.
NOV	1771.	1150.	0.64945	1.303	1499.	182	25	60.
DEC	1550.	933.	0.60198	1.352	1261.	387	0	53.

LEGEND:

HOBAR ==> MONTHLY AVERAGE DAILY EXTRATERRESTRIAL RADIATION (IDEAL) IN BTU/DAY-FT2.
HBAR ==> MONTHLY AVERAGE DAILY RADIATION (ACTUAL) IN BTU/DAY-FT2.
KBAR ==> RATIO OF HBAR TO HOBAR.
RBAR ==> RATIO OF MONTHLY AVERAGE DAILY RADIATION ON TILTED SURFACE TO THAT ON A HORIZONTAL SURFACE FOR EACH MONTH (I.E., MULTIPLIER OBTAINED BY TILTING).
SBAR ==> MONTHLY AVERAGE DAILY RADIATION ON A TILTED SURFACE (I.E., RBAR * HBAR) IN BTU/DAY-FT2.
HDD ==> NUMBER OF HEATING DEGREE DAYS PER MONTH.
CDD ==> NUMBER OF COOLING DEGREE DAYS PER MONTH.
TBAR ==> AVERAGE AMBIENT TEMPERATURE IN DEGREES FAHRENHEIT.

SECTION A-4

SAN ANSELMO SCHOOL

SITE DESCRIPTION

The San Anselmo School is a one-story, brick elementary school, located in San Jose, California. The building contains approximately 34,000 square feet of floor area, and is entirely bound by brick walls except for a small portion of window area. The school is functional all year-round and typically operates between the hours of 8:00 a.m. and 3:00 p.m. on weekdays. The school is usually unoccupied on the weekends.

The solar energy system was added to the existing building and is interconnected to the original cooling and heating equipment. The system was designed to supply 70% of the annual space heating requirements and 72% of the annual space cooling needs for the school.

The solar energy system incorporates 3,740 square feet of evacuated tubular glass collectors, a heat rejector, an expansion tank, a storage tank, a solar-operated absorption chiller, electronic controls, and interconnecting pipelines and hardware between the solar system and original heating and cooling equipment. Existing equipment was unaltered except for controls. These components include two gas-fired absorption chillers, two gas-fired absorption chiller/heaters, a cooling tower, 33 air-handling units, heating/cooling coils, and five pumps.

The collector array faces due south at a tilt of 40 degrees to the horizontal for collecting solar energy. The collection subsystem utilizes city water as a transfer medium from collector to storage and back to the collector again to complete the cycle. If solar energy is excessive, then solar energy is dissipated to the environment via a water-to-air heat rejector. When sufficient high temperature is reached in the storage tank, hot water is either transferred to the solar chiller during the cooling mode, or is transferred directly to the heating coils during the heating mode. If solar energy is insufficient to meet the space cooling and heating requirements, then two auxiliary gas-fired absorption chillers and two auxiliary gas-fired absorption chiller/heaters will satisfy the energy demand for the school.

The manufacturers of the major solar system equipment and components are listed below.

Equipment/Components	Manufacturer	Model No.
Evacuated-Tube Collectors	General Electric	TC-100
Heat Rejector	McQuay-Perfex, Inc.	LHD-217 CH
Outdoor Storage Tank	Ace Buehler, Inc.	VS72-9A
Auxiliary Absorption Chiller and Chiller Heaters	Arkla Corporation	DFE300-600
Solar Absorption Chiller	Arkla Corporation	WFB-300
Valves	Barber Colman	
Controllers	Barber Colman	

The system, shown schematically on Page A-43, has nine modes of solar operation.

Mode 1 - Collector Freeze Protection - This mode occurs when the outside ambient temperature is below 43°F and the level of insolation is not sufficient for energy collection. Solar pump P-8 is activated and valve V-3 is opened to allow flow through the heat rejector. Energy from the storage tank maintains the water in the collector loop at 38°F via modulating valve V-2. This prevents all equipment from being damaged by freezing.

Mode 2 - Auxiliary Collector Freeze Protection - This is a safety backup freeze protection mode. If the temperature exiting the collectors drops below 34°F, then dump valve V-4 directs city water through the collector loop to prevent the collectors from freezing.

Mode 3 - Solar Energy Collection - Solar energy collection is activated whenever insolation levels are sufficient. Pump P-8 is turned on and all the flow bypasses the storage tank and returns to the collectors to complete the cycle. Pump P-8 is deactivated when insolation levels fall below the set point.

Mode 4 - Collector-to-Storage - This mode occurs when the temperature exiting the collectors is 175°F or above. This closes the bypass port on valve V-2 and allows all water to flow through storage. When the temperature falls below 175°F, valve V-2 reverses position and allows all water to bypass the storage tank. This assures a positive energy flow into the storage tank.

Mode 5 - Storage-to-Space Cooling - Whenever space cooling is required and the temperature in the storage tank is above 175°F, then pump P-7 is activated, allowing flow from storage to the solar-operated absorption chiller. If solar energy is insufficient to meet the cooling demand, then two auxiliary gas-fired absorption chillers and two auxiliary gas-fired absorption chiller/heaters will supply the space cooling requirements.

Mode 6 - Storage-to-Space Heating - Whenever space heating is required and there is sufficient energy in the storage tank, then pump P-7 is activated, allowing hot water to flow to the heating coils for distribution to the heating zones via the air-handling units. If solar energy is insufficient, then two auxiliary gas-fired absorption chiller/heaters will supply the remaining heating requirements.

Mode 7 - Solar Heat Rejection - This mode occurs when excess solar energy is diverted from the collectors to the heat rejector unit via valve V-3. This mode operates when the temperature exiting the collectors is 220°F or above to reject excess energy to the environment. This deactivates when the temperature exiting the collectors falls below 220°F.

Mode 8 - Auxiliary Heat Protection - This is a safety backup protection to prevent collector damage. This mode activates when the temperature leaving the collectors exceeds 240°F and opens dump valve V-4 to allow city water to cool the collectors. This mode deactivates when the water leaving the collectors falls below 232°F.

Mode 9 - Power Failure Protection - This mode activates at any time during a power failure. Dump valve V-4 opens to allow city water to the collector loop and remains open until power is restored.

NOTE: An absorption chiller/heater is an absorption chiller which can be utilized for space heating by deactivating the cooling tower flow.

SITE HISTORY AND PROBLEMS

During April 1982, valve V-2 malfunctioned and allowed less than full flow from collectors to storage. This valve also was opening when the temperature in storage was greater than the collector loop temperature. Therefore, some energy was rejected from storage.

The collector startup set point was set too low during April. Collection began at 14 BTU/ft^2-hr rather than the 80 BTU/ft^2-hr called for in the design.

During the acceptance test in April, the following actions were taken:

- Established liquid levels in the expansion tanks and isolated the tanks from each other. Tried to eliminate excessive pressure during pump startup and the mixing of hot and cold fluids.

- Valves V-6 and V-8 were found to be disconnected and were reconnected to the system. This allowed solar heat to reach the building. Also, a normally open shutoff valve in the line to the space heating subsystem was found to be incorrectly closed, and it was opened to allow solar energy to reach the loads.

- A differential controller was installed on the storage bypass switchover valve (V-1) instead of a temperature controller.

- Pump P-5 was activated with the air-handling units to maintain flow over the control temperature sensor when the air-handlers were off.

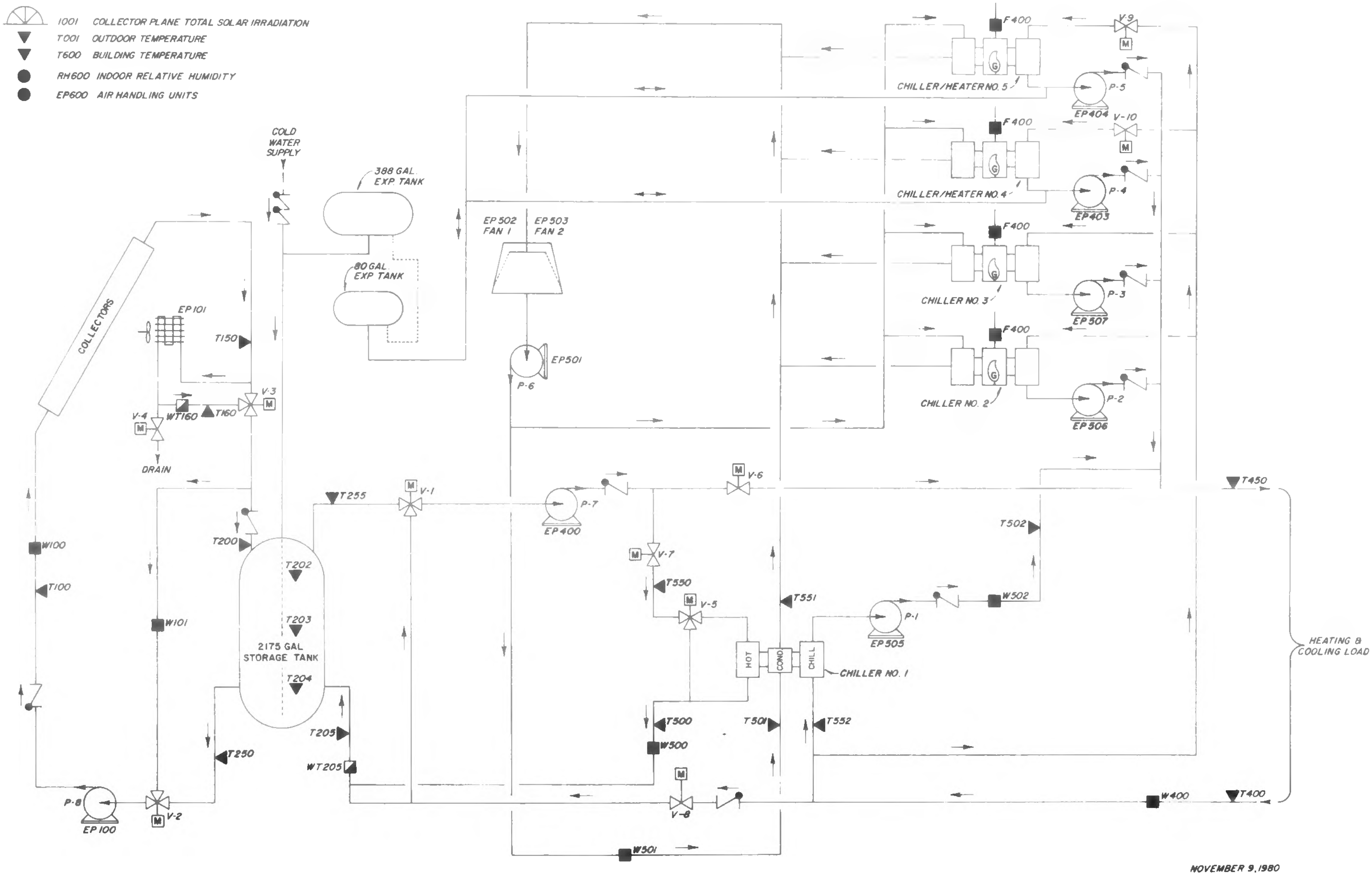
- Chiller/heater CH5 was the only chiller operational, based on temperature drop across the unit. All units were vacuum pumped and noncondensibles removed.

- The auxiliary staging control thermostat was found to be defective.

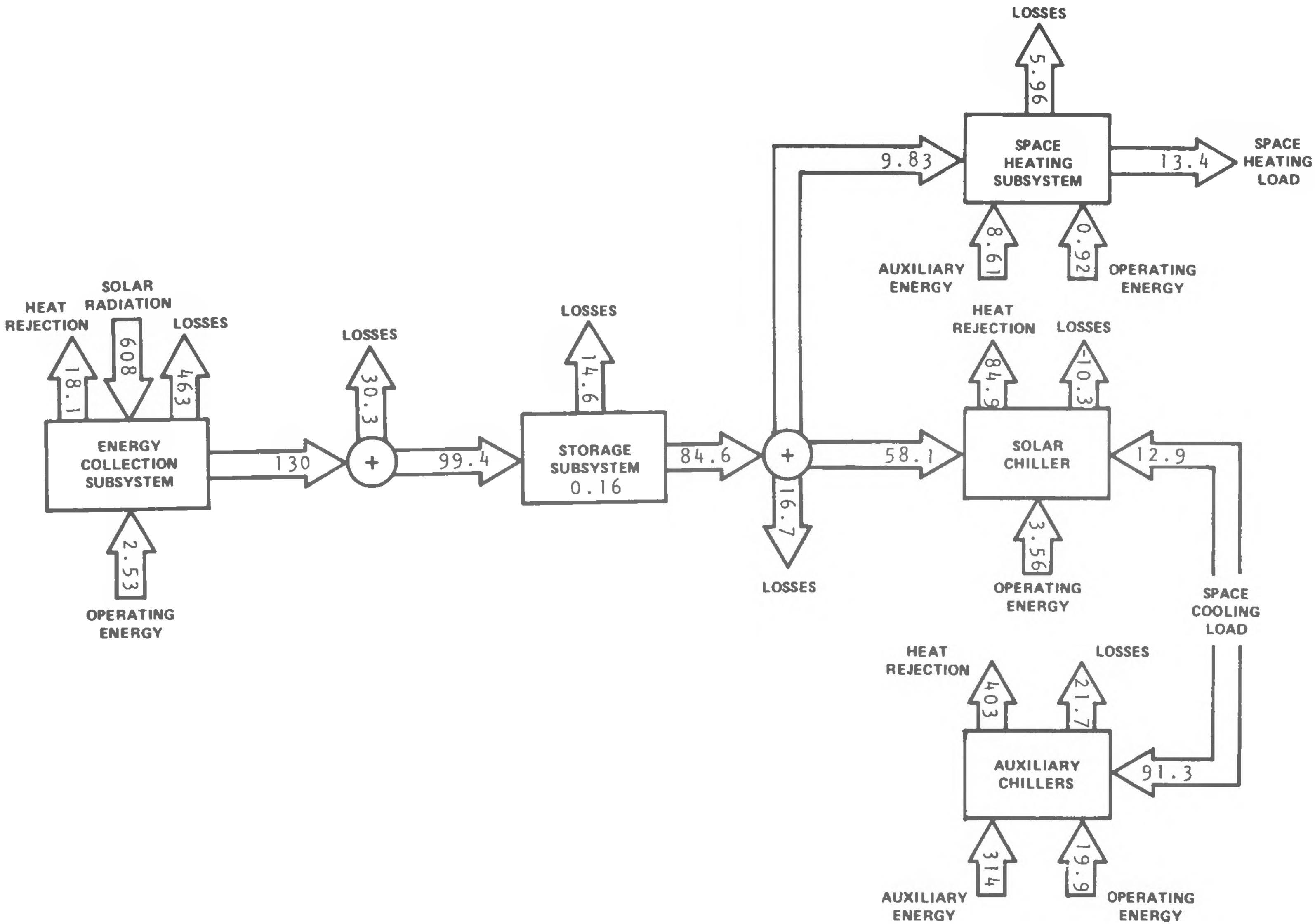
- Identified 25 broken collector tubes, leakage in some collector headers, and poor insulation.

In May, the absorption chillers were vacuum pumped again after performance had degraded. All of the chillers were operational but performance did not improve very much after several maintenance visits by site personnel.

The staging control was still performing improperly during May and the heating/cooling changeover switch was cycling frequently.



San Anselmo School Solar Energy System Schematic



Energy Flow Diagram for the San Anselmo School
April 1982 through June 1982
(Figures in million BTU)

Table 1. SOLAR SYSTEM THERMAL PERFORMANCE

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED (SECA)	SYSTEM LOAD (SYSL)	SOLAR ENERGY USED* (SEL)	AUXILIARY ENERGY FOSSIL (AXF)	AUXILIARY ENERGY THERMAL (AXT)	OPERATING ENERGY (SYSOPE)	ENERGY SAVINGS FOSSIL (TSVF)	ENERGY SAVINGS ELECTRICAL (TSVE)	SOLAR FRACTION (%) (SFR)
APR	43.1	40.1	22.8	150	90.0	15.3	38.0	-1.91	31
MAY	50.1	45.4	27.5	175	105	21.3	45.8	-2.50	22
JUN	36.5	31.9	17.6	212	127	27.5	29.4	-1.74	2
TOTAL	130	117	67.9	537	322	64.1	113	-6.15	-
AVERAGE	43.2	39.1	22.6	179	107	21.4	37.7	-2.05	19

* Input to cooling subsystem + heating subsystem

Table 2. COLLECTION SUBSYSTEM PERFORMANCE

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION (SEA)	COLLECTED SOLAR ENERGY (SECA)	COLLECTION SUBSYSTEM EFFICIENCY (%) (CLEF)	OPERATIONAL INCIDENT ENERGY (SEOP)	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%) (CLEFOP)	ECSS REJECTED ENERGY (CSRJE)	ECSS OPERATING ENERGY (CSOPE)	SOLAR ENERGY TO LOADS (CSEO)	SOLAR ENERGY TO STORAGE (STEI)	DAYTIME AMBIENT TEMPERATURE (°F) (TA)
APR	190	43.1	23	179	24	6.04E	0.79	22.8	35.0	67
MAY	226	50.1	22	220	23	4.97E	0.95	27.5	37.0	75
JUN	192	36.5	19	181	20	7.08E	0.79	17.6	27.4	75
TOTAL	608	130	-	580	-	18.1 E	2.53	67.9	99.4	-
AVERAGE	203	43.2	21	193	22	6.03E	0.84	22.6	33.1	72

E Denotes estimated value.

For a description of acronyms in parentheses, refer to Appendix E.

Table 3. STORAGE PERFORMANCE

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE (STEI)	ENERGY FROM STORAGE (STEO)	CHANGE IN STORED ENERGY (STECH)	STORAGE EFFICIENCY (%) (STEFF)	AVERAGE STORAGE TEMPERATURE (°F) (TST)	EFFECTIVE HEAT LOSS COEFFICIENT ($BTU/hr\text{-}ft^2\text{-}^\circ F$)	LOSS FROM STORAGE (STLOSS)
APR	35.0	26.8	0.16	77	161	0.55	8.04
MAY	37.0	34.6	0.53	95	173	0.21	1.87
JUN	27.4	23.2	-0.53	83	167	0.36	4.73
TOTAL	99.4	84.6	0.16	-	-	-	14.6
AVERAGE	33.1	28.2	0.05	85	167	0.37	4.88

Table 4. SPACE HEATING SUBSYSTEM

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD (EHL)	CONTROLLED DELIVERED ENERGY (CDE)	TOTAL SOLAR ENERGY USED (HSE)	TOTAL AUXILIARY THERMAL USED (HAT)	SOLAR FRACTION OF LOAD (%) (HSFR)	BUILDING TEMPERATURE (°F) (TB)	AMBIENT TEMPERATURE (°F) (TA)
APR	11.8	11.8	7.30	8.48	46	73	58
MAY	1.58	1.58	2.53	0.13	94	76	65
JUN	0.00	0.00	0.00	0.00	0	76	67
TOTAL	13.4	13.4	9.83	8.61	-	-	-
AVERAGE	4.46	4.46	3.28	2.87	73*	75	63

* Weighted average = $\Sigma(HSE_{month})/\Sigma(EHL_{month})$

For a description of acronyms in parentheses, refer to Appendix E.

Table 4a. SPACE HEATING SUBSYSTEM (Continued)

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD (EHL)	MEASURED SOLAR ENERGY USED (HSEM)	TOTAL OPERATING ENERGY (HOPE)	SOLAR SPECIFIC OPERATING ENERGY (HOPEI)	FOSSIL ENERGY SAVINGS (HSVF)	AUXILIARY FOSSIL FUEL (HAF)	HEATING DEGREE-DAYS (HDD)
APR	11.8	7.30	0.87	0.05	12.2	14.1	215
MAY	1.58	2.53	0.05	0.01	4.21	0.22	65
JUN	0.00	0.00	0.00	0.00	0.00	0.00	43
TOTAL	13.4	9.83	0.92	0.06	16.4	14.3	323
AVERAGE	4.46	3.28	0.31	0.02	5.47	4.77	108

Table 5. SPACE COOLING SUBSYSTEM

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD (CL)	SOLAR FRACTION OF LOAD (%) (CSFR)	SOLAR ENERGY USED (CSE)	OPERATING ENERGY (COPE)	AUXILIARY THERMAL USED (CAT)	AUXILIARY FOSSIL FUEL (CAF)	FOSSIL ENERGY SAVINGS (CSVF)	BUILDING TEMPERATURE (°F) (TB)
APR	28.3	18	15.5	13.6	81.5	136	25.8	73
MAY	43.8	17	25.0	20.3	105	174	41.6	76
JUN	31.9	2	17.6	26.7	127	212	29.4	76
TOTAL	104	-	58.1	60.6	314	522	96.8	-
AVERAGE	34.7	12*	19.4	20.2	105	174	32.3	75

* Based on $\Sigma(TCEL_{month})/\Sigma(CL_{month})$

For a description of acronyms in parentheses, refer to Appendix E.

Table 6. SOLAR CHILLER PERFORMANCE

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD (TCEL)	THERMAL ENERGY INPUT (TCEI)	OPERATING ENERGY (TCEOPE)	ENERGY REJECTED (TCERJE)	COEFFICIENT OF PERFORMANCE (RATIO) (TCECOP)
APR	4.96	15.5	1.07	26.3	0.32
MAY	7.26	25.0	1.54	38.3	0.29
JUN	0.66	17.6	0.95	20.3	0.04
TOTAL	12.9	58.1	3.56	84.9	-
AVERAGE	4.29	19.4	1.19	28.3	0.22*

* Weighted average = $\Sigma(TCEL_{month})/\Sigma(TCEI_{month})$

Table 7. AUXILIARY CHILLER PERFORMANCE

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD (ATCEL)	THERMAL ENERGY INPUT (ATCEI)	OPERATING ENERGY (ATCEOPE)	ENERGY REJECTED (ATCRJE)	COEFFICIENT OF PERFORMANCE (RATIO) (ATCECOP)
APR	23.4	81.5	4.22	103	0.29
MAY	36.6	105	6.21	141	0.35
JUN	31.3	127	9.51	159	0.25
TOTAL	91.3	314	19.9	403	-
AVERAGE	30.4	105	6.65	134	0.29*

* Weighted average = $\Sigma(ATCEL_{month})/\Sigma(ATCEI_{month})$

For a description of acronyms in parentheses, refer to Appendix E.

Table 8. SOLAR-UNIQUE OPERATING ENERGY

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU)

MONTH	ECSS OPERATING ENERGY (CSOPE)	SHS OPERATING ENERGY (HOPE1)	SCS OPERATING ENERGY (COPE1)	TOTAL SOLAR-UNIQUE OPERATING ENERGY (SYSOPE1)
APR	0.79	0.05	1.07	1.91
MAY	0.95	0.01	1.54	2.50
JUN	0.79	0.00	0.95	1.74
TOTAL	2.53	0.06	3.56	6.15
AVERAGE	0.84	0.02	1.19	2.05

Table 9. SOLAR COEFFICIENT OF PERFORMANCE

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

MONTH	SOLAR ENERGY SYSTEM $\left(\frac{SEL}{SYSOPE1}\right)$	COLLECTION SUBSYSTEM $\left(\frac{SECA}{CSOPE}\right)$	SPACE HEATING SOLAR $\left(\frac{HSEM}{HOPE1}\right)$	SPACE COOLING SOLAR $\left(\frac{CSE}{COPE1}\right)$
APR	12	55	150	15
MAY	11	53	250	16
JUN	10	46	0	19
WEIGHTED AVERAGE*	11	51	160	16

* Weighted average = $\Sigma(\text{Solar Input Performance Factor}_{month})/$
$\Sigma(\text{Operating Energy Performance Factor}_{month})$,
e.g., $\Sigma(SECA_{month})/\Sigma(CSOPE_{month})$

For a description of acronyms in parentheses, refer to Appendix E.

Table 10. ENERGY SAVINGS

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

(All values in million BTU)

MONTH	SOLAR ENERGY USED (SEL)	SPACE HEATING ELECTRICAL (HSVE)	SPACE HEATING FOSSIL FUEL (HSVF)	SPACE COOLING ELECTRICAL (CSVE)	SPACE COOLING FOSSIL FUEL (CSVF)	ECSS OPERATING ENERGY (CSOPE)	NET ENERGY SAVINGS ELECTRICAL (TSVE)	NET ENERGY SAVINGS FOSSIL FUEL (TSVF)
APR	22.8	-0.05	12.2	-1.07	25.8	-0.79	-1.91	38.0
MAY	27.5	-0.01	4.21	-1.54	41.6	-0.95	-2.50	45.8
JUN	17.6	0.00	0.0	-0.95	29.4	-0.79	-1.74	29.4
TOTAL	67.9	-0.06	16.4	-3.56	96.8	-2.53	-6.15	113
AVERAGE	22.6	-0.02	5.47	-1.19	32.3	-0.84	-2.05	37.7

Table 11. WEATHER CONDITIONS

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT2-DAY) MEASURED (SE)	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT2-DAY) LONG-TERM AVERAGE	AMBIENT TEMPERATURE (OF) MEASURED (TA)	AMBIENT TEMPERATURE (OF) LONG-TERM AVERAGE	HEATING DEGREE-DAYS MEASURED (HDD)	HEATING DEGREE-DAYS LONG-TERM AVERAGE	COOLING DEGREE-DAYS MEASURED (CDD)	COOLING DEGREE-DAYS LONG-TERM AVERAGE
APR	1,690	1,944	58	58	215	228	3	12
MAY	1,950	1,952	65	62	65	123	58	20
JUN	1,710	1,947	67	66	43	50	96	71
TOTAL	-	-	-	-	323	401	157	103
AVERAGE	1,780	1,948	63	62	108	134	52	34

For a description of acronyms in parentheses, refer to Appendix E.

SAN ANSELMO SCHOOL LONG-TERM WEATHER DATA

COLLECTOR TILT: 40 DEGREES
LATITUDE: 37 DEGREES

LOCATION: SAN JOSE, CALIFORNIA
COLLECTOR AZIMUTH: 0 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1469.	708.	0.48195	1.656	1172.	481	0	50.
FEB	1922.	1018.	0.52947	1.438	1463.	350	0	53.
MAR	2496.	1456.	0.58341	1.214	1768.	322	0	55.
APR	3079.	1921.	0.62389	1.012	1944.	228	12	58.
MAY	3477.	2212.	0.63622	0.882	1952.	123	20	62.
JUN	3634.	2349.	0.64623	0.829	1947.	50	71	66.
JUL	3549.	2323.	0.65442	0.851	1978.	12	117	68.
AUG	3227.	2054.	0.63643	0.953	1958.	15	111	68.
SEP	2702.	1700.	0.62895	1.135	1929.	13	94	68.
OCT	2087.	1213.	0.58118	1.378	1671.	90	19	63.
NOV	1573.	822.	0.52263	1.620	1332.	276	0	56.
DEC	1343.	645.	0.48036	1.740	1123.	456	0	50.

LEGEND:

HOBAR ==> MONTHLY AVERAGE DAILY EXTRATERRESTRIAL RADIATION (IDEAL) IN BTU/DAY-FT2.
HBAR ==> MONTHLY AVERAGE DAILY RADIATION (ACTUAL) IN BTU/DAY-FT2.
KBAR ==> RATIO OF HBAR TO HOBAR.
RBAR ==> RATIO OF MONTHLY AVERAGE DAILY RADIATION ON TILTED SURFACE TO THAT ON A HORIZONTAL SURFACE FOR EACH MONTH (I.E., MULTIPLIER OBTAINED BY TILTING).
SBAR ==> MONTHLY AVERAGE DAILY RADIATION ON A TILTED SURFACE (I.E., RBAR * HBAR) IN BTU/DAY-FT2.
HDD ==> NUMBER OF HEATING DEGREE DAYS PER MONTH.
CDD ==> NUMBER OF COOLING DEGREE DAYS PER MONTH.
TBAR ==> AVERAGE AMBIENT TEMPERATURE IN DEGREES FAHRENHEIT.

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SECTION A-5

UNIVERSITY OF MINNESOTA

SITE DESCRIPTION

The University of Minnesota site in Minneapolis, Minnesota is an 84,000-square-foot underground building, housing a bookstore and other university-related facilities. The building is 95% below ground with its lowest floor 45 feet below the surface. The building was constructed underground to conserve 60% of the energy normally required by a building of this size. Natural light is admitted to the building through terraced south and west windows.

The solar energy system retrofitted to this building was designed to provide 63% of its heating needs and 40% of its cooling needs. Solar energy collection is accomplished using a concentrating collector array. The array, which consists of six stationary units mounted in a row on the surface over the building, faces 15 degrees east of south. Each of the six units consists of 10 individual, movable reflectors (each 110-feet-long by one-foot-wide) mounted along a 45-degree slope, and a fixed receiver supported over them. The 10 movable reflectors within each stationary unit track the sun by pivoting in north-south arcs around their long axes, in a coordinated motion. The reflectors focus sunlight on a stationary receiver at a concentration ratio of 35 to one. A water/glycol solution absorbs heat as it circulates through the copper absorber tubes in the receivers. The total effective collecting area is 6,350 square feet. Solar energy storage is provided by a buried 21-foot-long, eight-foot-diameter, insulated steel tank with an 8,000-gallon capacity.

The system provides both heating and cooling of the building's conditioned space. Cooling is accomplished using solar energy to power an absorption-cycle chiller. Auxiliary energy for both heating and cooling is provided by a central steam system which is fueled by coal. Interface with the conditioned air takes place at three large fan-coil units.

The concentrating collectors, which operate on the direct component of the total insolation, are set to track the sun when they receive 127 BTU/ft^2-hr (400 W/M^2) total insolation, and according to a timer. Pump P11 is energized by a timer with a seasonally dependent set point.

The manufacturers of the major solar energy system components include:

Collectors	Suntec concentrating SLATS	Suntec Systems, Inc.
Chiller	Model C2J-W-5 absorption chiller (147 tons)	Trane, Inc.
Storage	Eight-foot-diameter x 21-foot-long 8,000-gallon steel, insulated tank.	Wheeler Tank Manufacturing Co.

The system, shown schematically on Page A-56, can be set in either its winter (space heating) or summer (space cooling) configuration. There are three modes of operation each for the space heating and space cooling configurations.

WINTER SPACE HEATING OPERATION

Mode 1 - Collector-to-Storage - When the incident solar energy is sufficient to raise the collector outlet temperature to 135°F and there is no space heating demand, this mode is activated. Pumps P11 and P12 are energized. Collected solar energy is delivered directly into the storage tank.

Mode 2 - Collector-to-Space Heating, Excess-to-Storage - This mode activates when incident solar energy is sufficient to raise the collector outlet temperature to 135°F and there is a space heating demand. Pumps P11 and P12 are energized to collect solar energy, and the load pump turns on. Heated water is delivered from the collector loop heat exchanger, past the auxiliary steam heating unit, to the three fan-coil units for space heating. If the temperature of the water leaving the coils is higher than the temperature of the water in the center of the tank, then this excess heat is delivered to the storage tank.

Mode 3 - Storage-to-Space Heating - When no incident solar energy is available and there is a space heating demand, then, if the storage tank temperature is above 120°F, the storage-to-space heating mode activates. Pump P12 and the load pump energize. Heated water is pumped from storage, past the auxiliary steam heating unit, to the three fan-coil units for space heating.

SUMMER SPACE COOLING OPERATION

Mode 1 - Collector-to-Storage - When the incident solar energy is sufficient to raise the collector outlet temperature to 180°F and there is no space cooling demand, this mode is activated. Pumps P11 and P12 are energized. Collected solar energy is delivered directly into the storage tank.

Mode 2 - Collector-to-Chiller, Excess-to-Storage - This mode activates when the incident solar energy is sufficient to raise the collector outlet temperature to 180°F and there is a space cooling demand. Pumps P11 and P12 are energized to collect solar energy, and one of the two cooling load pumps turns on. Heated water is delivered from the collector loop heat exchanger, past the auxiliary steam heating unit, to the chiller. On its return through the storage loop, any excess heat is delivered to the storage tank, to maintain the tank at 185°F. (Returning water is delivered to storage in this manner only if it is hotter than the water in the center of the tank.) Cold water from the chiller output is pumped to the three fan-coil units for space cooling.

Mode 3 - Storage-to-Chiller - When no incident solar energy is available and there is a space cooling demand, then, if the storage tank is at least 185°F, the storage-to-chiller mode activates. Pump P12 and one of the two cooling load pumps energize. Heated water is pumped from storage, past the auxiliary steam heating unit, to the chiller. Cold water from the chiller output is pumped to the three fan-coil units for space cooling.

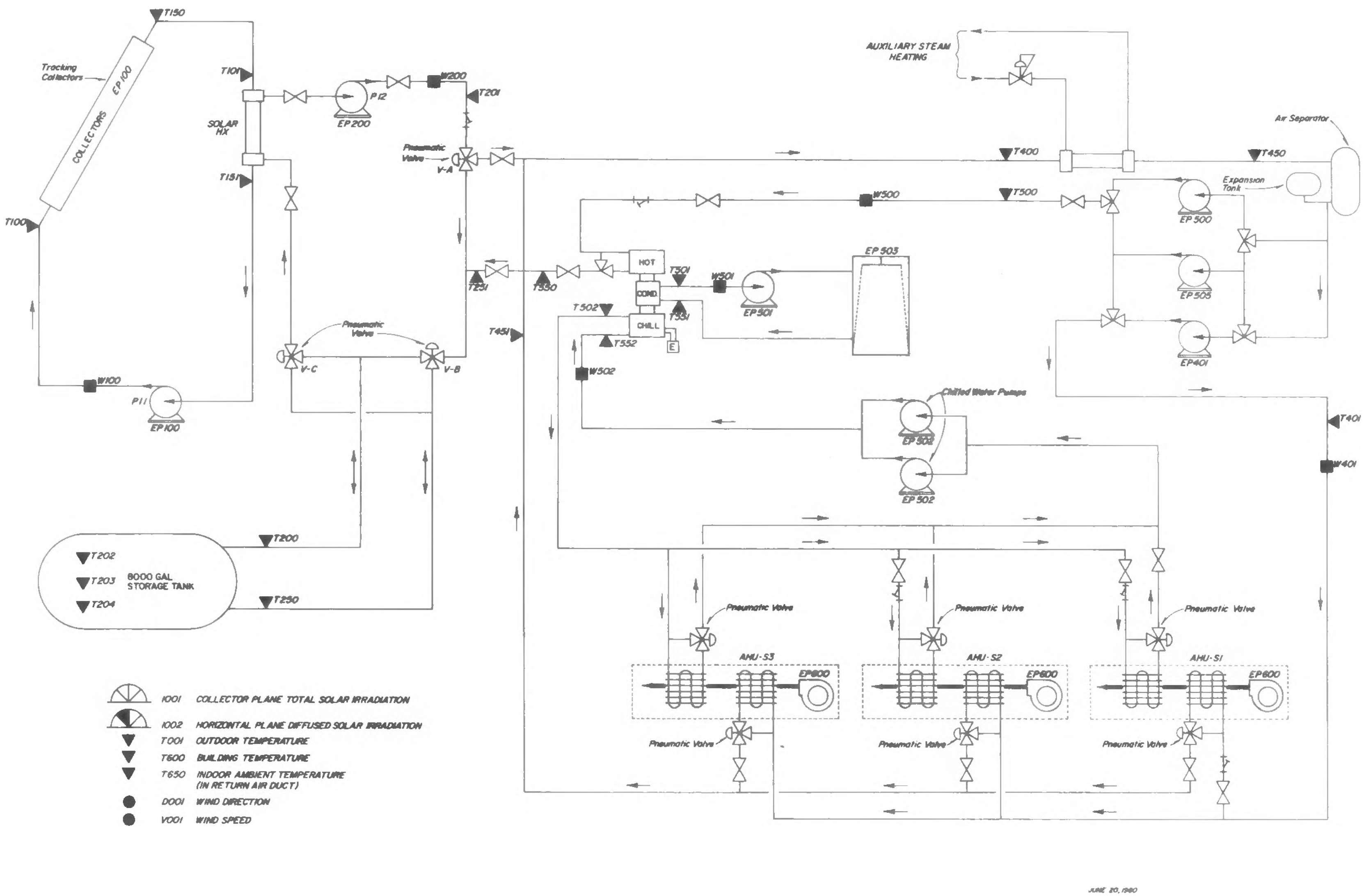
SITE HISTORY AND PROBLEMS

During a heavy snowstorm on December 2, 1981, seventeen of the 60 movable slats were damaged. Damage was so severe that the system was not operational until March 1982. In April 1982, representatives from the collector manufacturer, Suntec, spent considerable time focusing the collector slats. The remaining good mirrors were also consolidated to make completely functional subarrays. The area without mirrors was valved off until new mirrors could be installed.

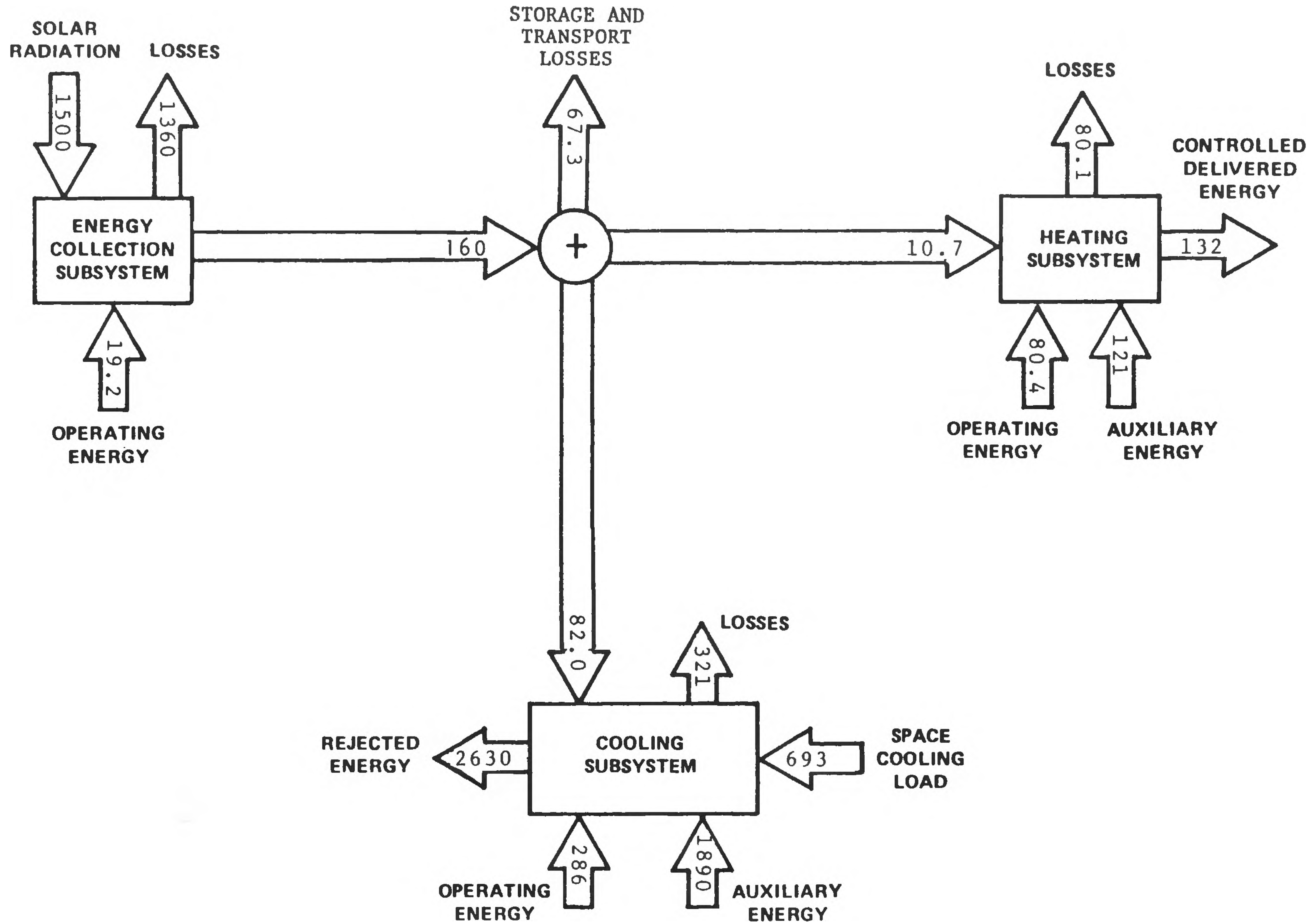
During May and June 1982, solar tracking was very poor. This was attributed to weak batteries in the tracking control system. On several days with very good insolation, the solar collectors did not even operate. There was also a malfunction of the building Heating, Ventilating, and Air Conditioning (HVAC) controller during May and June. This problem resulted in the HVAC system cycling between cooling and heating. Consequently, these loads were higher than normal.

In July 1982, the storage tank was valved off to prevent auxiliary energy from maintaining storage temperatures. The storage tank remained valved off until mid-August, but the problem of auxiliary energy maintaining storage temperatures was not solved. The grantee is continuing to investigate the cause of this anomaly.

The erratic collector operation did not improve during July or August 1982. The tracking controller was still malfunctioning.



University of Minnesota Bookstore Solar Energy System Schematic



Energy Flow Diagram for the University of Minnesota
April 1982 through August 1982
(Figures in million BTU)

Table 1. SOLAR SYSTEM THERMAL PERFORMANCE

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED (SECA)	SYSTEM LOAD (SYSL)	SOLAR ENERGY USED (SEL)	AUXILIARY ENERGY FOSSIL (AXF)	SYSTEM OPERATING ENERGY (SYSOPE)	ENERGY SAVINGS FOSSIL (TSVF)	ENERGY SAVINGS ELECTRICAL (TSVE)	SOLAR FRACTION (%) (SFR)
APR	46.2	26.9	9.40	29.2	24.8	15.7	-5.97	30
MAY	18.2	110	7.13	256	69.5	11.9	-5.73	2
JUN	27.1	185	16.3	619	88.1	27.2	-3.44	3
JUL	42.3	297	36.6	1,370	122	61.0	-2.49	5
AUG	26.1	206	23.3	914	82.1	38.8	-1.52	4
TOTAL	160	825	92.7	3,190	387	155	-19.2	-
AVERAGE	32.0	165	18.5	638	77.3	30.9	-3.83	5

Table 2. COLLECTION SUBSYSTEM PERFORMANCE

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION (SEA)	COLLECTED SOLAR ENERGY (SECA)	COLLECTOR SUBSYSTEM EFFICIENCY (%) (CLEF)	OPERATIONAL INCIDENT ENERGY (SEOP)	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%) (CLEFOP)	ECSS OPERATING ENERGY (CSOPE)	DAYTIME AMBIENT TEMPERATURE (°F) (TA)
APR	309	46.2	15	256	18	5.97	59
MAY	260	18.2	7	134	14	5.73	72
JUN	316	27.1	9	152	18	3.44	77
JUL	317	42.3	13	245	17	2.49	88
AUG	302	26.1	8	172	15	1.52	82
TOTAL	1,500	160	-	959	-	19.2	-
AVERAGE	300	32.0	11	192	17	3.83	76

For a description of acronyms in parentheses, refer to Appendix E.

Table 3. SPACE HEATING SUBSYSTEM

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD (EHL)	TOTAL SOLAR ENERGY USED (HSE)	TOTAL AUXILIARY THERMAL USED (HAT)	SOLAR FRACTION OF LOAD (%) (HSFR)	BUILDING TEMPERATURE (°F) (TB)	AMBIENT TEMPERATURE (°F) (TA)
APR	26.9	9.40	17.5	30	73	52
MAY	43.1	0.85	42.2	2	77	65
JUN	61.0	0.46	60.5	1	78	68
JUL	0.0	0.00	0.0	0	77	78
AUG	0.81	0.00	0.81	0	78	74
TOTAL	132	10.7	121	-	-	-
AVERAGE	26.4	2.14	24.2	8	77	67

Table 3a. SPACE HEATING SUBSYSTEM (Continued)

UNIVERSIITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	SPACE HEATING LOAD (EHL)	MEASURED SOLAR ENERGY USED (HSEM)	TOTAL OPERATING ENERGY (HOPE)	AUXILIARY FOSSIL FUEL (HAF)	HEATING DEGREE-DAYS (#) (HDD)
APR	26.9	9.40	18.8	29.2	394
MAY	43.1	0.85	26.5	70.4	68
JUN	61.0	0.46	34.8	100.9	15
JUL	0.0	0.00	0.0	0.0	0
AUG	0.81	0.00	0.34	1.35	3
TOTAL	132	10.7	80.4	202	480
AVERAGE	26.4	2.14	16.1	40.4	96

For a description of acronyms in parentheses, refer to Appendix E.

Table 4. SPACE COOLING SUBSYSTEM

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD (CL)	SOLAR FRACTION OF LOAD (%) (CSFR)	SOLAR ENERGY USED (CSE)	OPERATING ENERGY (COPE)	AUXILIARY THERMAL USED (CAT)	AUXILIARY FOSSIL FUEL (CAF)	BUILDING TEMPERATURE (°F) (TB)
APR	0.0	0	0.0	0.0	0	0	73
MAY	67.3	2	6.27	37.3	214	357	77
JUN	124	5	15.8	49.9	311	518	78
JUL	297	5	36.6	119	822	1,370	77
AUG	205	4	23.3	80.2	547	912	78
TOTAL	693	-	82.0	286	1,890	3,160	-
AVERAGE	139	4	16.4	57.2	378	632	77

Table 5. THERMODYNAMIC CONVERSION EQUIPMENT

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD (TCEL)	THERMAL ENERGY INPUT (TCEI)	OPERATING ENERGY (TCEOPE)	ENERGY REJECTED (TCERJE)	COEFFICIENT OF PERFORMANCE (TCECOP)
APR	0.0	0	0.0	0	0.00
MAY	67.3	220	21.0	311	0.29
JUN	124	327	27.8	487	0.38
JUL	297	859	77.3	1,160	0.34
AUG	205	571	49.2	669	0.35
TOTAL	693	1,980	175	2,630	-
AVERAGE	139	396	35	526	0.35*

* Weighted average = $\Sigma(TCEL_{month})/\Sigma(TCEI_{month})$

For a description of acronyms in parentheses, refer to Appendix E.

Table 6. SOLAR OPERATING ENERGY

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU)

MONTH	ECSS (CSOPE)	SCS (COPEI)	TOTAL SOLAR (SYSOPEI)
APR	5.97	0.00	5.97
MAY	5.73	0.00	5.73
JUN	3.44	0.00	3.44
JUL	2.49	0.00	2.49
AUG	1.52	0.00	1.52
TOTAL	19.2	0.00	19.2
AVERAGE	3.83	0.00	3.83

Table 7. SOLAR COEFFICIENT OF PERFORMANCE

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

MONTH	SOLAR ENERGY SYSTEM $\left(\frac{SEL}{SYSOPEI}\right)$	COLLECTION SUBSYSTEM $\left(\frac{SECA}{CSOPE}\right)$
APR	1.6	7.7
MAY	1.2	3.2
JUN	4.7	7.9
JUL	15	17
AUG	15	17
WEIGHTED AVERAGE	4.8	8.3

For a description of acronyms in parentheses, refer to Appendix E.

Table 8. ENERGY SAVINGS

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

(All values in million BTU)

MONTH	SOLAR ENERGY USED (SEL)	SPACE HEATING FOSSIL FUEL (HSVF)	SPACE COOLING FOSSIL FUEL (CSVF)	ECSS OPERATING ENERGY SOLAR-UNIQUE (CSOPE)	NET ENERGY SAVINGS ELECTRICAL (TSVE)	NET ENERGY SAVINGS FOSSIL FUEL (TSVF)
APR	9.40	15.7	0.0	-5,97	-5.97	15.7
MAY	7.13	1.42	10.5	-5.73	-5.73	11.9
JUN	16.3	0.77	26.4	-3.44	-3.44	27.2
JUL	36.6	0.00	61.0	-2.49	-2.49	61.0
AUG	23.3	0.00	38.8	-1.52	-1.52	38.8
TOTAL	92.7	17.9	137	-19.2	-19.2	155
AVERAGE	18.5	3.58	27.4	-3.83	-3.83	30.9

Table 9. WEATHER CONDITIONS

UNIVERSITY OF MINNESOTA
APRIL 1982 THROUGH AUGUST 1982

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY) MEASURED (SE)	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY) LONG-TERM AVERAGE	AMBIENT TEMPERATURE (°F) MEASURED (TA)	AMBIENT TEMPERATURE (°F) LONG-TERM AVERAGE	HEATING DEGREE-DAYS MEASURED (HDD)	HEATING DEGREE-DAYS LONG-TERM AVERAGE	COOLING DEGREE-DAYS MEASURED (CDD)	COOLING DEGREE-DAYS LONG-TERM AVERAGE
APR	1,624	1,507	52	45	394	597	2	0
MAY	1,321	1,573	65	57	68	271	60	26
JUN	1,659	1,641	68	67	15	65	105	122
JUL	1,624	1,722	78	72	0	11	418	225
AUG	1,536	1,666	74	70	3	21	293	182
TOTAL	-	-	-	-	480	965	878	555
AVERAGE	1,553	1,622	67	62	96	193	176	111

For a description of acronyms in parentheses, refer to Appendix E.

UNIVERSITY OF MINNESOTA LONG-TERM WEATHER DATA

COLLECTOR TILT: 45 DEGREES
LATITUDE: 45.12 DEGREES

LOCATION: MINNEAPOLIS, MINNESOTA
COLLECTOR AZIMUTH: -5 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1052.	465.	0.44179	1.985	922.	1637	0	12.
FEB	1531.	763.	0.49853	1.647	1257.	1358	0	17.
MAR	2179.	1102.	0.50590	1.300	1434.	1138	0	28.
APR	2888.	1442.	0.49913	1.045	1507.	597	0	45.
MAY	3416.	1737.	0.50840	0.906	1573.	271	26	57.
JUN	3641.	1928.	0.52966	0.851	1641.	65	122	67.
JUL	3525.	1969.	0.55850	0.875	1722.	11	225	72.
AUG	3090.	1689.	0.54646	0.986	1666.	21	182	70.
SEP	2433.	1254.	0.51528	1.192	1494.	173	23	60.
OCT	1719.	859.	0.49972	1.517	1303.	472	7	50.
NOV	1163.	479.	0.41231	1.813	869.	978	0	32.
DEC	925.	354.	0.38260	2.002	709.	1438	0	19.

LEGEND:

- HOBAR ==> MONTHLY AVERAGE DAILY EXTRATERRESTRIAL RADIATION (IDEAL) IN BTU/DAY-FT2.
- HBAR ==> MONTHLY AVERAGE DAILY RADIATION (ACTUAL) IN BTU/DAY-FT2.
- KBAR ==> RATIO OF HBAR TO HOBAR.
- RBAR ==> RATIO OF MONTHLY AVERAGE DAILY RADIATION ON TILTED SURFACE TO THAT ON A HORIZONTAL SURFACE FOR EACH MONTH (I.E., MULTIPLIER OBTAINED BY TILTING).
- SBAR ==> MONTHLY AVERAGE DAILY RADIATION ON A TILTED SURFACE (I.E., RBAR * HBAR) IN BTU/DAY-FT2.
- HDD ==> NUMBER OF HEATING DEGREE DAYS PER MONTH.
- CDD ==> NUMBER OF COOLING DEGREE DAYS PER MONTH.
- TBAR ==> AVERAGE AMBIENT TEMPERATURE IN DEGREES FAHRENHEIT.

APPENDIX B

COOLING COMPARATIVE SITE CHARACTERISTICS

APPENDIX B

COOLING COMPARATIVE SITE CHARACTERISTICS

SITE	COLLECTOR AREA (FT^2)	BUILDING AREA (FT^2)	NUMBER OF MONTHS OF DATA	TOTAL INSTALLED COOLING CAPACITY (TONS)	
El Toro Library	1,427	10,000	9	25	
Florida Solar Energy Center	2,089	5,000	12	45	25 Solar 20 Auxiliary
Honeywell-Salt River Project	8,200	55,000	7	278	50 Solar 228 Auxiliary
San Anselmo School	3,740	34,000	3	125	25 Solar 100 Auxiliary
University of Minnesota	6,350	84,000	5	147	

APPENDIX C

COOLING LOAD AND COOLING SOLAR FRACTION VS. TIME OF DAY

COOLING LOAD AND COOLING SOLAR FRACTION VS. TIME OF DAY

EL TORO LIBRARY
MAY 1982 THROUGH AUGUST 1982

HOUR OF DAY	MAY 1982		JUNE 1982		JULY 1982		AUGUST 1982	
	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)
0	0	0	0	0	0	0	0.12	24
1	0	0	0	0	0	0	0.14	6.4
2	0	0	0	0	0	0	0.10	7.8
3	0.26	0	0	0	0	0	0.16	23.1
4	0.34	0	0	0	0	0	0	0
5	1.07	9.0	0.17	39.7	0.11	32	0.04	20.4
6	2.62	24.4	2.27	21.5	7.64	18	6.00	16.1
7	3.68	11.5	2.80	9.0	8.86	4	7.17	2.3
8	3.79	10.5	3.05	1.4	8.52	3	6.74	2.4
9	3.25	24.9	3.71	1.1	8.43	9	6.65	6.2
10	3.98	34.5	4.37	16.3	8.63	38	6.48	36.1
11	4.38	49.2	4.11	25.4	8.54	58	7.66	52.6
12	4.36	53.7	3.76	31.6	8.64	60	8.26	56.9
13	4.84	55.2	4.15	40.7	8.87	60	8.48	56.8
14	5.04	52.4	4.29	39.7	9.07	51	8.45	54.2
15	4.96	49.6	4.41	35.4	9.57	39	8.12	38.2
16	4.60	43.0	3.69	30.0	9.31	23	8.11	24.3
17	3.05	37.3	2.75	23.2	5.62	13	5.46	19.4
18	2.67	30.7	2.35	12.5	5.16	10	4.83	10.2
19	2.68	31.0	2.16	37.7	4.78	8	4.77	11.6
20	0.66	31.2	0.20	100	0.83	9	1.31	15.2
21	0	0	0	0	0	0	0.21	55.7
22	0	0	0	0	0	0	0.15	80
23	0	0	0	0	0	0	0.15	11.4

COOLING LOAD AND COOLING SOLAR FRACTION VS. TIME OF DAY

FLORIDA SOLAR ENERGY CENTER
MAY 1982 THROUGH SEPTEMBER 1982

HOUR OF DAY	MAY 1982 COOLING LOAD (TONS) (CL)	MAY 1982 COOLING SOLAR FRACTION (%) (CSFR)	JUNE 1982 COOLING LOAD (TONS) (CL)	JUNE 1982 COOLING SOLAR FRACTION (%) (CSFR)	JULY 1982 COOLING LOAD (TONS) (CL)	JULY 1982 COOLING SOLAR FRACTION (%) (CSFR)	AUGUST 1982 COOLING LOAD (TONS) (CL)	AUGUST 1982 COOLING SOLAR FRACTION (%) (CSFR)	SEPTEMBER 1982 COOLING LOAD (TONS) (CL)	SEPTEMBER 1982 COOLING SOLAR FRACTION (%) (CSFR)
0	0	0	0	0	0.37	0	0	0	0	0
1	0	0	0	0	0	0	0	0	0	0
2	0	0	0	0	0	0	0	0	0	0
3	0	0	0	0	0	0	0	0	0	0
4	0	0	0	0	0	0	0	0	0	0
5	0	0	0	0	0.31	0	0	0	0	0
6	2.54	15.4	7.72	3.4	6.99	6.1	6.34	0	5.62	0
7	4.04	29.7	9.20	5.1	9.05	10.3	8.58	15.9	7.42	8.7
8	4.36	26.8	8.59	4.7	8.42	8.1	8.23	7.7	7.27	11.4
9	4.11	51.6	8.77	7.0	8.18	9.8	8.11	9.1	6.82	11.6
10	3.65	32.1	8.56	7.1	8.22	9.5	8.21	10.4	6.93	14.2
11	3.95	30.9	8.63	3.8	8.26	9.4	8.04	10.9	7.11	14.5
12	4.15	31.4	8.67	6.5	8.38	9.8	7.94	11.2	6.81	15.1
13	4.4	28.1	8.26	10.6	8.00	10.2	8.22	9.5	6.91	17.8
14	4.66	24.5	8.91	7.7	8.23	12.0	8.18	10.1	7.17	13.2
15	4.20	26.4	8.59	7.1	7.87	13.2	8.20	10.2	7.02	11.6
16	1.38	26.7	2.45	7.1	2.31	13.0	2.52	8.8	2.68	11.0
17	0	0	0.15	0	0.19	0	0.05	0	0.93	3.8
18	0	0	0	0	0.79	0	0.42	0	0.35	0
19	0	0	0.40	0	0.22	0	0.39	0	0	0
20	0	0	0	0	0.59	0	0.40	0	0.50	0
21	0	0	0	0	0.63	0	0	0	0.26	0
22	0	0	0	0	0.10	0	0.02	0	0	0
23	0	0	0	0	0	0	0.36	0	0	0

COOLING LOAD AND COOLING SOLAR FRACTION VS. TIME OF DAY

HONEYWELL-SALT RIVER PROJECT
JULY AND AUGUST 1982, SEPTEMBER AND OCTOBER 1981

HOUR OF DAY	JULY 1982		AUGUST 1982		SEPTEMBER 1981		OCTOBER 1981	
	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)
0	6.80	0	81.5	0	54.4	0	13.7	0
1	5.59	0	76.2	0	51.2	0	12.8	0
2	6.12	0	72.7	0	47.6	0	11.7	0
3	5.30	0	70.8	0	45.9	0	10.5	0
4	4.86	0	69.0	0	44.2	0	10.1	0
5	5.89	0	67.8	0	49.1	0	13.1	0
6	7.01	0	75.0	0	58.2	0	21.2	0
7	7.71	0	89.5	0	66.2	0	30.7	0
8	83.6	0	96.1	0	74.7	0	34.7	0
9	87.3	3.8	98.9	1.1	79.1	0.9	43.0	0.09
10	89.4	11.5	99.4	7.4	83.2	10.7	50.0	10.6
11	91.5	14.2	106	9.7	87.1	15.6	55.4	19.3
12	94.5	17.2	106	11.9	89.0	18	59.0	22.4
13	98.2	17.6	108	12.3	91.3	13.9	62.5	23.2
14	99.7	16.6	110	11.8	94.5	13.1	66.5	20.2
15	101	13.7	111	10.0	95.5	12.7	67.0	13.7
16	98.4	9.3	110	6.1	91.0	6.9	64.1	4.3
17	96.7	4.2	104	2.0	87.8	1.3	59.1	0
18	92.7	0.3	99.3	0	84.6	0	52.3	0
19	89.0	0	97.7	0	81.8	0	47.0	0
20	84.7	0	96.1	0	77.6	0	38.8	0
21	83.9	0	94.2	0	74.5	0	27.3	0
22	81.0	0	93.1	0	70.2	0	19.6	0
23	77.9	0	88.7	0	59.0	0	14.1	0

COOLING LOAD AND COOLING SOLAR FRACTION VS. TIME OF DAY

SAN ANSELMO SCHOOL
APRIL 1982 THROUGH JUNE 1982

HOUR OF DAY	APRIL 1982 COOLING LOAD (TONS) (CL)	APRIL 1982 COOLING SOLAR FRACTION (%) (CSFR)	MAY 1982 COOLING LOAD (TONS) (CL)	MAY 1982 COOLING SOLAR FRACTION (%) (CSFR)	JUNE 1982 COOLING LOAD (TONS) (CL)	JUNE 1982 COOLING SOLAR FRACTION (%) (CSFR)
0	0	0	0	0	0.3	0
1	0	0	0.04	0	3.37	0
2	0	0	0.89	0	1.63	0
3	0	0	0.89	0	1.32	0
4	0	0	0.19	0	2.05	0
5	0.28	0	0.16	0	3.16	0
6	0	0	0.18	0	5.38	0
7	0.07	100	1.05	0	8.49	0
8	0.19	100	1.95	0	11.6	0.3
9	2.25	50.6	7.92	8	13.2	0.3
10	8.81	15.6	15.9	16	14.5	0.8
11	10.9	10.3	19.4	20	15.4	2.2
12	12.7	14.6	20.6	18	7.17	3.0
13	14.6	15.8	22.9	17	0.79	2.9
14	13.0	17.6	14.5	15	0.36	2.3
15	9.82	19.4	3.01	19	0	7.2
16	32.6	26.6	2.36	7.7	0	1.7
17	1.28	10.4	2.64	11.9	0	0
18	1.07	0	1.62	15.6	0	0
19	0.38	0	1.13	0	0	0
20	0.12	0.02	0.30	0	0	0
21	0	0	0.08	0	0	0
22	0	0	0	0	0	0
23	0	0	0	0	0	0

COOLING LOAD AND COOLING SOLAR FRACTION VS. TIME OF DAY

UNIVERSITY OF MINNESOTA
JUNE 1982 THROUGH AUGUST 1982

	JUNE 1982		JULY 1982		AUGUST 1982	
HOUR OF DAY	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)	COOLING LOAD (TONS) (CL)	COOLING SOLAR FRACTION (%) (CSFR)
0	2.06	0	21.2	1.8	18.0	0.5
1	1.96	0	22.2	2	17.4	0.9
2	1.90	0	20.1	1.4	15.8	1.0
3	1.85	0	20.4	1.2	16.6	0.7
4	3.24	0	25.9	0.6	19.8	1.1
5	16.2	1.4	36.8	0.2	23.8	1.9
6	20.3	1.6	38.9	0.8	26.9	1.6
7	23.9	1.8	41.9	1.2	25.4	1.4
8	27.6	2.0	45.2	1.4	27.3	1.9
9	28.7	2.4	47.2	5.3	30.0	-1.5
10	30.4	3.2	50.1	7.6	30.0	4.9
11	31.3	3.9	50.6	11.8	32.9	5.3
12	32.5	4.3	49.8	12.6	32.9	6.2
13	33.5	4.0	50.2	13.6	36.5	5.6
14	31.5	3.5	49.6	9	37.1	4.7
15	31.0	2.3	49.6	2.2	36.5	1.8
16	19.8	1.8	44.7	1.1	31.3	0.5
17	0	0	22.3	0.7	11.8	0.4
18	0	0	22.8	0.9	14.7	2.1
19	0	0	19.0	0.6	13.3	2.9
20	1.25	0	17.0	1.5	12.1	0.9
21	2.46	0	17.2	1.9	10.7	4.6
22	2.04	0	16.3	1.8	10.8	0.4
23	1.89	0	19.4	1.9	20.2	0.6

APPENDIX D

PERFORMANCE EVALUATION TECHNIQUES AND DATA ACCURACY ESTIMATES

APPENDIX D

PERFORMANCE EVALUATION TECHNIQUES AND DATA ACCURACY ESTIMATES

PERFORMANCE EVALUATION TECHNIQUES

The performance of a solar energy system is evaluated by calculating a set of primary performance factors which are based on those in the intergovernmental agency report Thermal Data Requirements and Performance Evaluation Procedures for the National Solar Heating and Cooling Demonstration Program (NBSIR 76-1137).

An overview of the NSDN data collection and dissemination process is shown in Figure D-1.

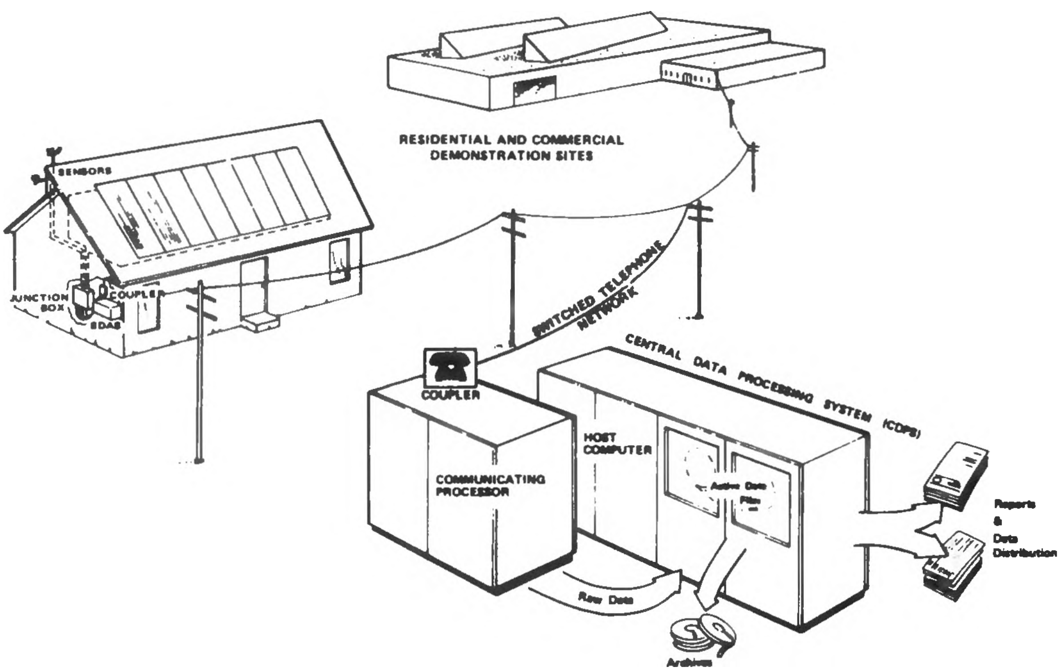


Figure D-1. The National Solar Data Network

DATA COLLECTION AND PROCESSING

Each site contains standard industrial instrumentation modified for the particular site. Sensors measure temperatures, flows, insolation, electric power, fossil fuel usage, and other parameters. These sensors are all wired into a junction box (J-box), which is in turn connected to a microprocessor data logger called the Site Data Acquisition Subsystem (SDAS). The SDAS can read up to 96 different channels, one channel for each sensor. The SDAS takes the analog voltage input to each channel and converts it to a 10-bit word. At intervals of every 320 seconds, the SDAS samples each channel and records the values on a cassette tape. Some of the channels can be sampled 10 times in each 320 second interval, and the average value is recorded on the tape.

Each SDAS is connected through a modem to voice-grade telephone lines which are used to transmit the data to a central computer facility. This facility is the Central Data Processing System (CDPS), located at Vitro Laboratories in Silver Spring, Maryland. The CDPS hardware consists of an IBM System 7 and dual IBM 3033 computers. Typically, the System 7 collects data from each SDAS six times a week, although the tape can hold three to five days of data, depending on the number of channels.

The data received by the System 7 is in the form of digital counts in the range of 0-1,023. These counts are then processed by software in the CDPS, where they are converted from counts to Engineering Units (EU) by applying appropriate calibration constants. The engineering unit data called "detailed measurements" in the software is then tabulated on a weekly basis for the site analyst. The CDPS is also capable of transforming this data into plots, graphs, and processed reports.

Solar system performance reports present system parameters as monthly values. If some of the data during the month is not collected due to solar system instrumentation system or data acquisition problems, or if some of the collected data is invalid, then the collected valid data is extrapolated to provide the monthly performance estimates. Researchers and other users who require unextrapolated, "raw" data may obtain data by contacting Vitro Laboratories.

DATA ANALYSIS

The analyst develops a unique set of "site equations" for each site in the NSDN, following the guidelines presented herein.

The equations calculate the flow of energy through the system, including solar energy, auxiliary energy, and losses. These equations are programmed in PL/1 and become part of the Central Data Processing System. The PL/1 program for each site is termed the site software. The site software processes the detailed data, using as input a "measurement record" containing the data for each scan interval. The site software produces as output a set of performance factors, on an hourly, daily, and monthly basis.

These performance factors (Appendix E) quantify the thermal performance of the system by computing energy flows throughout the various subsystems. The system performance may then be evaluated based on the efficiency of the system in transferring these energies.

Performance factors which are considered to be of primary importance are those which are essential for system evaluation. Without these primary performance factors (which are denoted by an asterisk in Appendix E), comparative evaluation of the wide variety of solar energy systems would be impossible. An example of a primary performance factor is "Solar Energy Collected by the Array". This is quite obviously a key parameter in system analysis.

Secondary performance factors are data deemed important and useful in comparison and evaluation of solar systems, particularly with respect to component interactions and simulation. In most cases these secondary performance factors are computed as functions of primary performance factors.

DATA ACCURACY ESTIMATES

The primary tool used to determine the data requirements and the selection of instrumentation is the analytical heat balance. Sufficient heat balance calculations are required to equate the total energy input to the total energy output for the subsystem or component under study to provide an energy balance closure of less than 10%. As a general rule, a six percent accuracy is assumed for NSDN performance results, based on the requirements described in Reference D-1 and other theoretical calculations and tests from Reference D-2.

Errors greater than approximately 10% for active systems and 15% for passive systems will not permit useful comparison between different systems. Error analysis of most performance evaluation factors for active NSDN solar energy systems has shown that the experimental data is obtained with accuracy of about ± six percent using the sensors shown in Table D-1. (Reference D-2)

The data accuracy conclusions were based on a composite of all available information sources, including:

- Field data from selected sample sites (Reference D-2)
- Manufacturer's accuracy data (Reference D-2)
- Internal laboratory calibration data (Reference D-2)

Table D-1. SENSOR ACCURACY

THE NATIONAL SOLAR DATA NETWORK

PARAMETER	SENSOR TYPE	MANUFACTURER	ACCURACY (% of Full Scale unless indicated)
Temperature	3-wire Platinum Resistance Thermometer (RDT)	Minco	± 0.5°F
Insolation	Precision Spectral Pyranometer	Eppley	± 3% 0-70° Angle ± 6% 70-80° Angle
Wind	Propeller-type Anemometer	WeatherMeasure	± 1% <25 mph ± 3% >25 mph
Humidity	Solid State	WeatherMeasure	± 3% <80% RH ± 6% >80% RH
Liquid Flow (Rate)	Impact-type Target Flow Meter	Ramapo	± 1% ½" to 3½" Pipe ± 2% 4" Pipe
Liquid Flow (Total)	Nutating Disk Flow Meter	Hersey	± 1.5% Total Flow
Air Flow	Thermal Anemometer	Kurz	± 2% -68 - 140°F
Fuel Flow	Oscillating Piston Flow Meter	Kent	± 1% Full Scale
Gas Flow	Bellows Type-4 Chamber	American	± 1% Full Scale
Electric Power	Hall Effect Transducer	Ohio Semitronics	± 0.5% Full Scale
Heat Flux	Thermoelectric Junction	Hy-Cal Engineering	± 2% Linearity ± 0.5% Repeatability

- Site verification from special accuracy tests (Reference D-2)

- Special tests required to verify system accuracy

The error elements of the NSDN data system are categorized into three major groups. These are the sensor error sources, the Site Data Acquisition Subsystem (SDAS) error sources, and the computational error sources. Each of these areas is briefly discussed below. Additional detail is available in Reference D-2.

Sensor errors are defined as all error sources arising between the point of measurement and the input to the SDAS. Sensor errors are of two types. The first type is inherent sensor error. These errors are independent of the installation of a sensor at a particular location. The sources for quantifying these errors are manufacturers' references and laboratory tests conducted at the manufacturers' facilities. Estimates of these errors are given in Table D-1. Sensor descriptions are given in Appendix G.

The second type of sensor error is "in-situ" or location error. These errors are specific to the sensor location, sensor wiring, installation technique, and to the state of the system where the measurement is made. In general, sensors for all sites have been installed in accordance with manufacturers' and National Bureau of Standards (NBS) standards, in order to minimize errors due to sensor location. (See Reference D-2.)

SDAS errors are defined as all errors propagated in the Site Data Acquisition Subsystem.

Two sources of SDAS accuracy data are available. An unpublished report details the results of testing performed at Argonne National Laboratory (ANL). Error numbers related to the variation of regulated voltages within the SDAS from several sources were established and found to be less than 0.05% in most cases. A significant area of concern was long-term drift of readings at many sites. Line voltage variation, temperature regime of the SDAS, and repair/replacement were found to have less significance. Secondly, side-by-side testing of a fully deployed sensor/SDAS system resulted in performance factor accuracy within ± six percent of reference measurement.

Computational errors are propagated from application of analytical techniques to the data stream, and include rounding errors, data gap errors, and sampling rate errors.

Estimation of actual computational errors was accomplished using computer simulation to determine round-off and sampling rate errors, the effect of data gap bridging, and the effect of errors in the measurement of certain constants and auxiliary parameters that affect performance factor computations. The effects of these errors were established by actual measurement at the test sites, data acquired from other sources, and from analytical techniques.

Results of these tests are available in Reference D-2. In general, the results showed no significant introduction of error in computations at most sites.

Data is occasionally lost at NSDN sites for a variety of reasons. Values for missing data elements are created by a data bridging routine. There will always be some error associated with the estimation process.

For data losses of 10% or less, the performance factor accuracy is not significantly affected. Most errors are less than three percent. All but one are four percent or less. The significant exception is change in stored energy, which is very sensitive to data loss.

Some performance factors are stable with relatively large data loss. Calculation of overall system performance generally remains stable with less than 20% data loss. (Reference D-2)

The results of several related studies indicate that the measurement of the performance of typical active solar systems can be accomplished with a relatively high degree of accuracy. Performance factor accuracy is within the National Bureau of Standards (NBS) criteria of six percent accuracy. (Reference D-1) Exceptions are those performance factors which depend directly on the estimation of burner efficiency or estimates due to known sensor failures.

APPENDIX D

REFERENCES

D-1 Streed, E., et al. Thermal Requirements and Performance Evaluation Procedures for the National Heating and Cooling Demonstration Program, NBSIR 76-1137, National Bureau of Standards, Washington, D.C., 1976.

D-2 Seropian, A. Data Accuracy Study (Two Parts), Technical Memo #03200.8, Vitro Laboratories, Silver Spring, Maryland, March 13, 1981.

APPENDIX E

PERFORMANCE FACTORS AND SOLAR TERMS

APPENDIX E

PERFROMANCE FACTORS AND SOLAR TERMS

The performance factors identified by the use of acronyms or symbols are defined in this appendix. Section 1 describes general acronyms used in this report. Section 2 includes the acronym, the actual name of the performance factor, and a short definition.

Section 3 contains a glossary of solar terminology, in alphabetical order. These terms are included for quick reference by the reader.

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SECTION 1

GENERAL ACRONYMS

ATCE	Auxiliary Thermodynamic Conversion Equipment.
ASHRAE	American Society of Heating, Refrigeration, and Air-Conditioning Engineering.
BTU	British Thermal Unit, a measure of heat energy. The quantity of heat required to raise the temperature of one pound of pure water one degree Fahrenheit. One BTU is equivalent to 2.928×10^{-4} kwh of electrical energy.
COP	Coefficient of Performance. The ratio of total load to solar-source energy.
DHW	Domestic Hot Water.
ECSS	Energy Collection and Storage System.
HWS	Domestic or Service Hot Water Subsystem.
KWH	Kilowatt Hours, a measure of electrical energy. The product of kilowatts of electrical power applied to a load times the hours it is applied. One kwh is equivalent to 3,413 BTU of heat energy.
NSDN	National Solar Data Network.
SCS	Space Cooling Subsystem.
SHS	Space Heating Subsystem.
SOLMET	Solar Radiation/Meteorology Data.
TCE	Thermodynamic Conversion Equipment.

SECTION 2

PERFORMANCE FACTOR DEFINITIONS AND ACRONYMS

ACRONYM	NAME	DEFINITION
ASTECH	Change in Energy Stored in Cold Storage	Change in stored energy in cold storage during specific time period.
ASTEFF	Cold Storage Efficiency	Ratio of the sum of energy supplied to cold storage and the change in cold storage energy to the energy removed from cold storage.
ASTEI	Energy Delivered to Cold Storage	Amount of energy delivered to cold storage from the load.
ASTEO	Energy from Cold Storage	Amount of energy removed from cold storage by the chiller
ASTLOSS	Cold Storage Loss	Total energy losses from the cold storage subsystem.
ATCECOP	Auxiliary Cooling Subsystem Coefficient of Performance	The ratio of the auxiliary cooling subsystem load to thermal or electrical energy input.
ATCEI	Auxiliary Cooling Subsystem Thermal Energy Input	Equivalent thermal energy supplied as a fuel source to the auxiliary thermodynamic conversion equipment.
ATCEL	Auxiliary Cooling Load	Thermal energy removed from the air being cooled by the auxiliary thermodynamic conversion equipment.
ATCEOPE	Auxiliary Thermodynamic Conversion Equipment Operating Energy	Energy required to support the operation of the auxiliary thermodynamic conversion equipment; e.g., pumps, fans, etc.

ACRONYM	NAME	DEFINITION
ATCERJE	Auxiliary Rejected Energy	Amount of energy intentionally rejected from thermodynamic conversion equipment as a by-product of its operation.
ATST	Average Cold Storage Temperature	Average temperature of the cold storage medium.
AXE	Auxiliary Electric Fuel Energy to Load Subsystem	Amount of electrical energy required as a fuel source for all load subsystems.
AXF	Auxiliary Fossil Fuel Energy to Load Subsystem	Amount of fossil energy required as a fuel source for all load subsystems.
* AXT	Auxiliary Thermal Energy to Load Subsystems	Thermal energy delivered to all load subsystems to support a portion of the subsystem loads, from all auxiliary sources.
* BL	Building Load	Sum of heat conducted through the building walls and ceilings, and heat convected through cracks, doors, and windows as air infiltration.
CAE	SCS Auxiliary Electrical Fuel Energy	Amount of electrical energy provided to the SCS to be converted and applied to the SCS load.
CAF	SCS Auxiliary Fossil Fuel Energy	Amount of fossil energy provided to the SCS to be converted and applied to the SCS load.
CAREF	Collector Array Efficiency	Ratio of the collected solar energy to the incident solar energy.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
CAT	SCS Auxiliary Thermal Energy	Amount of thermal energy supplied to the SCS by the auxiliary equipment. For vapor compression units, it is CAE multiplied by compressor efficiency.
CDD	Cooling Degree-Days	A rough measure of the cooling requirement. This performance factor is the difference between the mean daily temperature, TAVE, and $65^{o}F$. If the mean is $65^{o}F$ or less, cooling degree-days are zero.
CDE	Controlled Delivered Energy	Space heating intentionally delivered by the space heating subsystem including solar and auxiliary. This does not include heat losses from electric motors, pipes, storage, and other equipment.
* CL	Space Cooling Subsystem Load	Energy required to satisfy the temperature control demands of the space cooling subsystem.
CLAREA	Collector Array Area	The gross area of one collector panel multiplied by the number of panels in the array.
CLEF	Collection Subsystem Efficiency	Ratio of the energy collected to the total energy incident on the collector array.
CLEFOP	Operational Collection Subsystem Efficiency	Efficiency when there is fluid in the collector loop.
CLS	Solar Energy Contribution to Cooling Load	The portion of the total cooling load which was satisfied by solar energy.

* Primary Performance Factor

ACRONYM	NAME	DEFINITION
COPE	SCS Operating Energy	Amount of electrical energy required to support the SCS operation (fans and pumps) which is not intended to directly affect the thermal state of the subsystem.
COPE1	Solar-Specific Operating Energy	The operating energy necessary to the functioning of the solar energy portions of the SCS.
CSAUX	Auxiliary Energy to ECSS	Amount of auxiliary energy supplied to the ECSS.
* CSCEF	ECSS Solar Conversion Efficiency	Ratio of the solar energy supplied from the ECSS to the load subsystems to the incident solar energy on the collector array.
CSE	Solar Energy to SCS	Amount of solar energy delivered to the SCS.
CSEO	Energy Delivered from ECSS to Load Subsystems	Amount of energy supplied from the ECSS to the load subsystems (including any auxiliary energy supplied to the ECSS).
* CSFR	SCS Solar Fraction	Percentage of the SCS load which is supported by solar energy.
CSOPE	ECSS Operating Energy	Amount of energy used to support the ECSS operation (e.g., fans, pumps, etc.) which is not intended to affect directly the thermal state of the subsystem.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
CSRJE	ECSS Rejected Energy	Amount of energy intentionally rejected or dumped from the ECSS subsystem.
* CSVE	SCS Electrical Energy Savings	Difference in the electrical energy required to support an assumed similar conventional SCS and the actual electrical energy required to support the demonstration SCS, for identical SCS loads.
* CSVF	SCS Fossil Energy Savings	Difference in the fossil energy required to support an assumed similar conventional SCS and the actual fossil energy required to support the demonstration SCS, for identical SCS loads.
EHL	Equipment Heating Load	Amount of energy supplied to the space heating subsystem equipment: solar, auxiliary thermal, operating energy converted to heat, and losses from the space heating equipment which contribute to heating (the building heating load less internal gains).
GENOPE	Power Generation Operating Energy	The electrical energy required to operate the ECSS and Rankine subsystems when they are in the power generation mode.
HAE	SHS Auxiliary Electrical Fuel Energy	Amount of electrical energy provided to the SHS to be converted and applied to the SHS load.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
HAF	SHS Auxiliary Fossil Fuel Energy	Amount of fossil energy provided to the SHS to be converted and applied to the SHS load.
HAT	SHS Auxiliary Thermal Energy	Amount of thermal energy provided to the SHS by the auxiliary SHS.
HDD	Heating Degree-Days	A rough measure of the heating requirement. This performance factor is the difference between the mean daily temperature and 65°F. The mean is the average of the minimum and maximum temperatures for a given day. If the mean is 65°F or more, heating degree-days are zero.
HOPE	SHS Operating Energy	Amount of energy required to support the SHS operation (which is not intended to be applied directly to the SHS load).
HOPE1	Solar-Specific SHS Operating Energy	Operating energy necessary to the functioning of the solar energy portions of the SHS.
HOURCT	Record Time	Count of hours elapsed from the start of 1977.
HSE	Solar Energy to SHS	Amount of solar energy delivered to the SHS, including thermal losses from solar heated fluids.
HSEL	Solar Energy Losses to Load	Solar energy losses from storage and other equipment which heat the conditioned space.
HSEM	Measured Solar Energy to SHS	Solar energy intentionally delivered to SHS by the distribution network. Does not include solar energy losses which also sometimes contribute to space heating.

ACRONYM	NAME	DEFINITION
* HSFR	SHS Solar Fraction	Percentage of the SHS load which is supported by solar energy.
* HSVE	SHS Electrical Energy Savings	Difference in the electrical energy required to support an assumed similar conventional SHS and the actual electrical energy required to support the demonstration SHS, for identical SHS loads.
* HSVF	SHS Fossil Energy Savings	Difference in the fossil energy required to support an assumed similar conventional SHS and the actual fossil energy required to support the demonstration SHS, for identical SHS loads.
HWAE	HWS Auxiliary Electrical Fuel Energy	Amount of electrical energy provided to the HWS to be converted and applied to the HWS load.
HWAF	HWS Auxiliary Fossil Fuel Energy	Amount of fossil energy provided to the HWS to be converted and applied to the HWS load.
HWAT	HWS Auxiliary Thermal Energy	Amount of energy provided to the HWS by a heat transfer fluid from an auxiliary source.
HWCSM	Service Hot Water Consumption	Amount of heated water delivered to the load from the HWS.
HWCSMA	Tempered Hot Water Consumed	Total energy required to raise the hot water used from the supply water temperature to the hot water temperature.
HWDSFR	HWS Solar Fraction of Demand	Percentage of the "hot water demand" which is supplied by solar energy.
* HWL	Hot Water Subsystem Load	Amount of energy supplied to the HWS.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
HWOPE	HWS Operating Energy	Amount of energy required to support the HWS operation which is not intended to be applied directly to the HWS load.
HWOPE1	Solar-Unique HWS Operating Energy	"Operating energy" necessary to the functioning of the solar energy portions of the HWS.
HWSE	Solar Energy to HWS	Amount of solar energy delivered to the HWS.
HWSE1	Solar Energy to Preheat Tank	The amount of solar energy input to a preheat tank.
* HWSFR	HWS Solar Fraction	Percentage of the HWS load which is supported by solar energy.
* HWSVE	HWS Electrical Energy Savings	Difference in the electrical energy required to support an assumed similar conventional HWS and the actual electrical energy required to support the demonstration HWS, for identical HWS loads.
* HWSVF	HWS Fossil Energy Savings	Difference in the fossil energy required to support an assumed similar conventional HWS and the actual fossil energy required to support the demonstration HWS, for identical loads.
LINLOS	Recirculation Loop Losses	Thermal energy losses due to recirculation of hot water in a large building loop.
OUTVC	Cooling Produced	Space cooling provided by the air conditioner; energy removed from the conditioned space.
PARA	Rankine Parasitic Power	Amount of auxiliary electrical energy supplied per unit time to the refrigerant pump and the Rankine subsystem controls.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
PRELOS	Preheat Tank Losses	The difference between the input solar energy to a pre-heat tank and the output solar energy to the HWS tank. This includes losses and changes in internal energy.
PWRGEN	Rankine Power Generated	Amount of electrical energy per unit time produced by the motor generator from the shaft power of the gas turbine.
PWRSVE	Rankine Power Generation Savings	The net output of the Rankine engines when operating in the power generation mode.
RANKOUT	Rankine Turbine Output	Mechanical energy developed at the output shaft of the Rankine engine gas turbine. Includes energy losses in the gearbox. This shaft output can drive a motor generator or an air conditioning compressor.
REFF	Rankine Thermal Efficiency	The ratio of RANKOUT to RSE. This percentage was developed from laboratory experimental data taken from gas turbines operating under typical conditions at the solar sites.
RELH	Relative Humidity	Average outdoor relative humidity at the site.
ROPE	Rankine Operating Energy	Amount of electrical energy required to support the Rankine system. Includes energy for boiler feedwater pumping, cooling tower pumping, cooling tower fan and parasitics.
RSCAE	Rankine Auxiliary Electric Used	Amount of auxiliary electrical energy supplied to the motor generator for driving the air conditioning compressor.

ACRONYM	NAME	DEFINITION
RSCOP	Rankine Coefficient of Performance	The ratio of useful energy provided by the Rankine subsystem including the associated air conditioner to the operating and auxiliary energy input to the subsystem. Specifically, it is (OUTVC + PWRGEN)/(RSCAE + ROPE).
RSE	Rankine Solar Energy Used	Amount of solar energy supplied to boil the refrigerant which drives the Rankine cycle gas turbine.
RSRJE	Rankine Energy Rejected	Amounts of energy intentionally rejected through the cooling tower from the Rankine condenser and air conditioning condenser.
* SE	Incident Solar Energy	Amount of solar energy incident upon one square foot of the collector plane.
SEA	Incident Solar Energy on Array	Amount of solar energy incident upon the collector array.
* SEC	Collector Solar Energy	Amount of thermal energy added to the heat transfer fluid for each square foot of the collector area.
* SECA	Collected Solar Energy by Array	Amount of thermal energy added to the heat transfer fluid by the collector array.
SEDF	Diffuse Insolation	Amount of diffuse solar energy incident upon one square foot of a collector plane.
* SEL	Solar energy to Load Subsystems	Amount of solar energy supplied by the ECSS to all load subsystems.
SEOP	Operational Incident Solar Energy	Amount of solar energy incident upon the collector array when the collector loop is active.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
* SFR	Solar Fraction of System Load	Percentage of the system load which was supported by solar energy.
SSSR	System Solar Savings Ratio	The ratio of the sum of the solar contributions to the system load minus the solar-specific system operating energy to the total system load.
STECH	Change in ECSS Stored Energy	Change in ECSS stored energy during specific time period.
STEFF	ECSS Storage Efficiency	Ratio of the sum of energy supplied by ECSS storage and the change in ECSS stored energy to the energy delivered to the ECSS storage.
STEI	Energy Delivered to ECSS Storage	Amount of energy delivered to ECSS storage by the collector array and from auxiliary sources.
STEO	Energy Supplied by ECSS Storage	Amount of energy supplied by ECSS storage to the load subsystems.
STLOSS	Storage Loss	Total energy losses from the storage subsystem.
STOCAP	Storage Capacity	The volumetric storage capacity of the storage subsystem.
STPER	Effective Heat Transfer Coefficient	The overall heat transfer coefficient for the hot solar storage tank as measured for the month: ratio of storage loss to product of outside tank area, temperature difference across insulation, and number of hours in the month.
SYSCOP	System Coefficient of Performance	The ratio of the total solar energy delivered to the load to the sum of the solar operating energies.
* SYSL	System Load	Energy required to satisfy all desired temperature control demands at the output of all subsystems.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
* SYSOPE	System Operating Energy	Amount of energy required to support the system operation, including all subsystems, which is not intended to be applied directly to the system load.
SYSOPE1	Solar System Operating Energy	Operating energy that is specifically used for the solar components of the system.
* SYSPF	System Performance Factor	Ratio of the system load to the total equivalent fossil energy expended or required to support the system load.
* TA	Ambient Temperature	Average temperature of the ambient air.
TANKV	HWS Heat-up Energy	The energy required to heat all the water in the HWS tank from the cold water supply temperature to the hot water outlet temperature.
TAVE	Average Daily Temperature	The average daily temperature as defined by the National Weather Service; i.e., the average of the minimum and maximum temperatures for a given day.
* TB	Building Temperature	Average temperature of the air in the controlled space of the building.
TCECOP	TCE Coefficient of Performance	Coefficient of performance of the thermodynamic conversion equipment, typically, the ratio of equipment load to thermal energy input.
TCEI	TCE Thermal Input Energy	Equivalent thermal energy which is supplied as a fuel source to thermodynamic conversion equipment.
TCEL	Thermodynamic Conversion Equipment Load	Controlled energy output of thermodynamic conversion equipment.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
TCEOPE	TCE Operating Energy	Amount of energy required to support the operation of thermodynamic conversion equipment (e.g., pumps and fans).
TCERJE	TCE Reject Energy	Amount of energy intentionally rejected or dumped from thermodynamic conversion equipment as a by-product or consequence of its principal operation.
TDA	Daytime Average Ambient Temperature	Average temperature of the ambient air during the daytime (during normal collector operation period).
* TECSM	Total Energy Consumed by System	Amount of energy demand of the system from external sources; sum of all fuels, operating energies, and collected solar energy.
THW	Service Hot Water Temperature	Average temperature of the service hot water supplied by the system.
TRANKC	Rankine Condensing Water Temperature	Temperature of the heat transfer fluid at the inlet to the condenser of the gas turbine subsystem.
TRANKS	Rankine Solar Water Temperature	Temperature of the heat transfer fluid at the inlet to the refrigerant boiler of the gas turbine.
TST	ECSS Storage Temperature	Average temperature of the ECSS storage medium.
* TSVE	Total Electrical Energy Savings	Difference in the estimated electrical energy required to support an assumed similar conventional system and the actual electrical energy required to support the system, for identical loads; sum of electrical energy savings for all subsystems.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
* TSVF	Total Fossil Energy Savings	Difference in the estimated fossil energy required to support an assumed similar conventional system and the actual fossil energy required to support the system, for identical loads; sum of fossil energy savings of all subsystems.
TSW	Supply Water Temperature	Average temperature of the supply water to the hot water subsystem.
WDIR	Wind Direction	Average wind direction at the site.
WIND	Wind Velocity	Average wind velocity at the site.

* Primary Performance Factor

SECTION 3

SOLAR TERMINOLOGY

Absorptivity	The ratio of radiation absorbed by a surface to the total radiated energy incident on that surface.
Active Solar System	A system in which a transfer fluid (liquid or air) is circulated (by pump or fan) through a solar collector.
Air Conditioning	Popularly defined as space cooling; more precisely, the process of treating indoor air by controlling the temperature, humidity, and distribution to maintain specified comfort conditions.
Ambient Temperature	The surrounding air temperature.
Array	An assembly of a number of collector elements, or panels, into the solar collector for a solar energy system.
Auxiliary Energy	In solar energy terminology, the energy supplied to the heating or cooling load from other than the solar source, usually from a conventional heating or cooling system. Excluded are operating energy, and energy which may be supplemental in nature but does not have the auxiliary system as an origin; e.g., energy supplied to the space heating load from the external environment by a heat pump.
Auxiliary Energy Subsystem	In solar energy terminology, the auxiliary energy system is the conventional heating and/or cooling equipment used as a supplement or backup to the solar system.
Backflow	Reverse flow.
Backflow Preventer	A valve or damper installed in a pipe or duct to prevent reverse flow of the fluid.
Beam Radiation	Radiated energy received directly, not from scattering or reflecting sources.

Collected Solar Energy	The thermal energy added to the heat transfer fluid by the solar collector.
Collection Subsystem	The assembly of components that absorbs incident solar energy and transfers the absorbed thermal energy to a heat transfer fluid.
Collector Array Efficiency	Same as Collector Conversion Efficiency. Ratio of the collected solar energy to the incident solar energy. (See also Operational Collector Efficiency.)
Collector Conversion Efficiency	Ratio of thermal energy output to solar energy incident on the collector array.
Concentrating Solar Collector	A solar collector that concentrates the energy from a larger area onto an absorbing element of smaller area.
Conditioned Space	The space in a building in which the air is heated or cooled to maintain a desired temperature range.
Control System or Subsystem	The assembly of electric, pneumatic, or hydraulic, sensing, and actuating devices used to control the operating equipment in a system.
Cooling Degree-Days	The sum over a specified period of time of the number of degrees the mean daily temperature is <u>above</u> 65°F.
Cooling Tower	A heat exchanger that transfers waste heat to outside ambient air.
Diffuse Radiation	Solar radiation which is scattered by air molecules, dust, or water droplets and incapable of being focused.
Drainback	Automatic draining of the collector array and piping to storage each time the collector pump shuts off.
Draindown	A system equipped with automatic or manual valves which drain the solar collectors and collector piping to prevent freezing in the event of cold weather.

Duct Heating Coil	A liquid-to-air heat exchanger in the duct distribution system.
Effective Heat Transfer Coefficient	The heat transfer coefficient, per unit plate area of a collector, which is a measure of the total heat losses per unit area from all sides, top, back, and edges.
Energy Gain	The thermal energy gained by the collector transfer fluid. The thermal energy output of the collector.
Energy Savings	The estimated difference between the fossil and/or electrical energy requirements of an assumed conventional system (carrying the full measured load) and the actual electrical and/or fossil energy requirements of the installed solar-assisted system.
Expansion Tank	A tank with a confined volume of air (or gas) whose inlet port is open to the system heat transfer fluid. The pressure and volume of the confined air varies as the system heat transfer fluid expands and contracts to prevent excessive pressure from developing and causing damage.
F-Chart	A computer program developed by the University of Wisconsin Solar Energy Laboratory, which calculates solar heating system performance and economics.
Fixed Collector	A solar collector that is fixed in position and cannot be rotated to follow the sun daily or seasonably.
Flat-Plate Collector	A solar energy collecting device consisting of a relatively thin panel of absorbing material. A container with insulated bottom and sides and covered with one or more covers transparent to visible solar energy and relatively opaque to infrared energy. Visible energy from the sun enters through the transparent cover and raises the temperature of the absorbing panel. The infrared energy reradiated from the panel is trapped within the collector

Flat-Plate Collector (Continued)	because it cannot pass through the cover. Glass is an effective cover material (see Selective Surface).
Focusing Collector	A concentrating type collector using parabolic mirrors or optical lenses to focus the energy from a large area onto a small absorbing area.
Fossil Fuel	Petroleum, coal, and natural gas derived fuels.
Glazing	In solar energy terminology, the transparent covers used to reduce energy losses from a collector panel.
Heat Exchanger	A device used to transfer energy from one heat transfer fluid to another while maintaining physical segregation of the fluids. Normally used in systems to provide an interface between two different heat transfer fluids.
Heat Transfer Fluid	The fluid circulated through a heat source (solar collector) or heat exchanger that transports the thermal energy by virtue of its temperature.
Heating Degree-Days	The sum over a specified period of time of the number of degrees the mean daily temperature is <u>below</u> 65°F.
Incidence Angle	The angle between the line to a radiating source (the sun) and a line normal to the plane of the surface being irradiated.
Incident Solar Energy	The amount of solar energy irradiating a surface taking into account the angle of incidence. The effective area receiving energy is the product of the area of the surface times the cosine of the angle of incidence.
Insolation	Incoming solar radiation
Instantaneous Efficiency	The efficiency of a solar collector at one operating point, $\frac{Ti - Ta}{I}$, under steady-state conditions (see Operating Point).

Instantaneous Efficiency Curve	A plot of solar collector efficiency against operating point, $\frac{Ti - Ta}{I}$, (see Operating Point).
Load	That to which energy is supplied, such as space heating load or cooling load. The system load is the total solar and auxiliary energy required to satisfy the heating or cooling requirements.
Manifold	The piping that distributes the transport fluid to and from the individual panels of a collector array.
Microclimate	Highly localized weather features which may differ from long-term regional values due to the interaction of the local earth's surface with the atmosphere.
Nocturnal Radiation	The loss of thermal energy by the solar collector to the night sky.
Operating Energy	The amount of energy (usually electrical energy) required to operate the solar and auxiliary equipments and to transport the thermal energy to the point of use, and which is not intended to directly affect the thermal state of the system.
Operating Point	A solar energy system has a dynamic operating range due to changes in level of insolation (I), fluid input temperature (Ti), and outside ambient temperature (Ta). The operating point is defined as: $$\frac{Ti - Ta}{I} \left(\frac{{}^{o}F \times hr \times ft^2}{BTU}\right)$$
Operational Collector Efficiency	Ratio of collected solar energy to incident solar energy <u>only during the time the collector fluid is being circulated with the intention of delivering solar-source energy to the system</u>.
Outgassing	The emission of gas by materials and components, usually during exposure to elevated temperature, or reduced pressure.

Passive Solar System	A system which uses architectural components of the building to collect, distribute, and store solar energy.
Pebble Bed (Rock Bed)	A space filled with uniform-sized pebbles to store solar-source energy by raising the temperature of the pebbles.
Reflected Radiation	Insolation reflected from a surface, such as the ground or a reflecting element, onto the solar collector.
Rejected Energy	Energy intentionally rejected, dissipated, or dumped from the solar system.
Retrofit	The addition of a solar energy system to an existing structure.
Selective Surface	A surface that has the ability to readily absorb solar radiation, but reradiates little of it as thermal radiation.
Sensor	A device used to monitor a physical parameter of a system, such as temperature or flow rate, for the purpose of measurement or control.
Solar Conditioned Space	The area in a building that depends on solar energy to provide a fraction of the heating and cooling needs.
Solar Fraction	The fraction of the total load supplied by solar energy. The ratio of solar energy supplied to loads divided by total load. Often expressed as a percentage.
Solar Savings Ratio	The ratio of the solar energy supplied to the load minus the solar system operating energy, divided by the system load.
Storage Efficiency, η_s	Measure of effectiveness of transfer of energy through the storage subsystem taking into account system losses.
Storage Subsystem	The assembly of components used to store solar-source energy for use during periods of low insolation.

Stratification	A phenomenon that causes a distinct thermal gradient in a heat transfer fluid, in contrast to a thermally homogeneous fluid which results in the layering of the heat transfer fluid, with each layer at a different temperature. In solar energy systems, stratification can occur in liquid storage tanks or rock beds, and may even occur in pipes and ducts. The temperature gradient or layering may occur in a horizontal, vertical, or radial direction.
System Performance Factor	Ratio of system load to the total equivalent fossil energy expended or required to support the system load.
Ton of Refrigeration	The heat equivalent to the melting of one ton (2,000 pounds) of ice at 32°F in 24 hours. A ton of refrigeration will absorb 12,000 BTU/hr, or 288,000 BTU/day.
Tracking Collector	A solar collector that moves to point in the direction of the sun.
Zone	A portion of a conditioned space that is controlled to meet heating or cooling requirements separately from the other space or other zones.

APPENDIX F

ENERGY CONVERSION FACTORS

APPENDIX F

ENERGY CONVERSION FACTORS

Fuel Type	Energy Content	Fuel Source Conversion Factor
Coal	8,600 BTU/pound 17.2×10^6 BTU/ton	0.05814×10^{-6} tons/BTU
Electricity	3,413 BTU/kilowatt-hour	292.8×10^{-6} kwh/BTU
Kerosene	135,000 BTU/gallon	7.41×10^{-6} gallons/BTU
Natural gas	1,021 BTU/cubic foot	979.4×10^{-6} cubic feet/BTU
Oil, distillate fuel[1]	138,690 BTU/gallon	7.21×10^{-6} gallons/BTU
Oil, residual fuel[2]	149,690 BTU/gallon	6.68×10^{-6} gallons/BTU
Propane	91,500 BTU/gallon	10.93×10^{-6} gallons/BTU

[1] No. 1 and No. 2 heating oils, diesel fuel, No. 4 fuel oils

[2] No. 5 and No. 6 fuel oils

APPENDIX G

SENSOR TECHNOLOGY

APPENDIX G

SENSOR TECHNOLOGY

TEMPERATURE SENSORS

Temperatures are measured by a Minco Products S53P platinum Resistance Temperature Detector (RTD). Because the resistance of platinum wire varies as a function of temperature, measurements of the resistance of a calibrated length of platinum wire can be used to accurately determine the temperature of the wire. This is the principle of the platinum RTD which utilizes a tiny coil of platinum wire encased in a copper-tipped probe to measure temperature.

Ambient temperature sensors are housed in a WeatherMeasure Radiation Shield in order to protect the probe from solar radiation. Care is taken to locate the sensor away from extraneous heat sources which could produce erroneous temperature readings. Temperature probes mounted in pipes are installed in stainless steel thermowells for physical protection of the sensor and to allow easy removal and replacement of the sensors. A thermally-conductive grease is used between the probe and the thermowell to assure faster temperature response.

All temperature sensors are individually calibrated at the factory. In addition, the bridge circuit is calibrated in the field using a five-point check.

Nominal Resistance @ 25°C:	100 ohms
No. of Leads:	3
Electrical Connection:	Wheatstone Bridge
Time Constant:	1.5 seconds max. in water at 3 fps
Self Heating:	27 mw/°F

WIND SENSOR

Wind speed and direction are measured by a WeatherMeasure W102-P-DC/540 or W101-P-DC/540 wind sensor. Wind speed is measured by means of a four-bladed propeller coupled to a DC generator.

Wind direction is sensed by means of a dual-wiper 1,000-ohm long-life conductive plastic potentiometer. It is attached to the stainless steel shaft which supports the vane and propeller and rotates with the upper body assembly.

Size:	$29\frac{3}{4}$"L x 30"H
Starting Speed:	1 mph
Complete Tracking:	3 mph
Maximum Speed:	200 mph
Distance Constant (30 mph):	6.2'
Accuracy:	$\pm$ 1% below 25 mph $\pm$ 3% above 25 mph
Time Constant:	0.145 second

HUMIDITY SENSORS

The WeatherMeasure HMP-14U Solid State Relative Humidity Probe is used for the measurement of relative humidity. The operation of the sensor is based upon the capacitance of the polymer thin film capacitor. A one-micron-thick dielectric polymer layer absorbs water molecules through a thin metal electrode and causes capacitance change proportional to relative humidity.

Range:	0-100% R.H.
Response Time:	1 second to 90% humidity change at 20°C
Temperature Coefficient:	0.05% R.H./°C
Accuracy:	$\pm$ 3% from 0 to 80% R.H. $\pm$ 5 to 6% from 80 to 100% R.H.
Sensitivity:	0.2% R.H.

INSOLATION SENSORS

The Eppley Model PSP pyranometer is used for the measurement of insolation. The pyranometer consists of a circular multi-junction thermopile of the plated (copper-constantan) wirewound type which is temperature compensated to render the response essentially independent of ambient temperature. The receiver is coated with Parsons' black lacquer (non-wavelength-selective absorption). The instrument is supplied with a pair of precision-ground polished concentric hemispheres of Schott optical glass transparent to light between 285 and 2800 nm of wavelength. The instrument is provided with a dessicator which may be readily inspected. Pyranometers designated as shadowband pyranometers are equipped with a shadowband which may be adjusted to block out any direct solar radiation. These instruments are used for the measurement of diffuse insolation.

Sensitivity:	9 μ V/W/m^2
Temperature Dependence:	$\pm$ 1% over ambient temperature range -20°C to 40°C
Linearity:	0.5% from 1 to 2,800 W/M^2
Response Time:	1 second
Cosine Error:	$\pm$ 1% 0 to 70° zenith angle $\pm$ 3% 70 to 80° zenith angle

LIQUID FLOW SENSORS (NON-TOTALIZING)

The Ramapo Mark V strain gauge flow meters are used for the measurement of liquid flow. The flow meters sense the flow of the liquids by measuring the force exerted by the flow on a target suspended in the flow stream. This force is transmitted to a four active arm strain gauge bridge to provide a signal proportional to flow rate squared. The flow meters are available in a screwed end configuration, a flanged configuration, and a wafer configuration. Each flow meter is calibrated for the particular fluid being used in the application.

Materials:	Target - 17-PH stainless steel Body - Brass or stainless steel Seals - Buna-N
Fluid Temperature:	$-40^{o}F$ to $250^{o}F$
Calibration Accuracy:	$\pm$ 1% (½" to 3½" line size) $\pm$ 2% (4" and greater line size)
Repeatability and Hysteresis:	0.25% of reading

LIQUID FLOW SENSORS (TOTALIZING)

Hersey Series 400 flow meters are used to measure totalized liquid flow. The meter is a nutating disk, positive displacement type meter. An R-15 register with an SPDT reed switch is used to provide an output to the data acquisition subsystem.

The output of the reed switch is input to a Martin DR-1 Digital Ramp which counts the number of pulses and produces a zero to five volt analog signal corresponding to the pulse count.

Materials:	Meter body - bronze Measuring chamber - plastic
Accuracy	$\pm$ 1.5%

AIR FLOW SENSORS

The Kurz 430 Series of thermal anemometers is used for the measurement of air flow. The basic sensing element is a probe which consists of a velocity sensor and a temperature sensor. The velocity sensor is heated and operated as a constant temperature thermal anemometer which responds to a "standard" velocity (referenced to $25^{o}C$ and 760 mm Hg) or mass flow by sensing the cooling effect of the air as it passes over the heated sensor. The temperature sensor compensates for variations in ambient temperature.

Since the probe measures air velocity at only one point in the cross section of the duct, it is necessary to perform a careful duct mapping to relate the probe reading to the amount of air flowing through the entire duct. This is done by dividing the duct into small areas and taking a reading at the center of each area using a portable probe. The readings are then averaged to determine the overall duct velocity. The reading at the permanently installed probe is then ratioed to this reading. This duct mapping is done for each mode.

Accuracy:	± 2% of full scale over temperature range -20°C to 60°C ± 5% of full scale over temperature range -60°C to 250°C
Response Time:	0.025 second
Repeatability:	0.25% full scale

FUEL OIL FLOW SENSOR

The Kent Mini-Major is used as a flow oil flow meter. The meter utilizes an oscillating piston as a positive displacement element. The oscillating piston is connected to a pulser which sends pulses to the Site Data Acquisition Subsystem for totalization.

Operating Temperature:	100 C (max)
Flow Range:	0.6 to 48 gph
Accuracy:	± 1% of full scale

FUEL GAS FLOW SENSOR

The American AC-175 gas meter is used for the measurement of totalized fuel gas flow. The drop in pressure between the inlet and outlet of the meter is responsible for the action of the meter. The principle of measurement is positive displacement. Four chambers in the meter fill and empty in sequence. The exact volume of compartments is known, so by counting the number of displacements the volume is measured. Sliding control valves control the entrance and exit of the gas to the compartments. The meter is temperature compensated to reference all volumetric readings to 60°F.

Rated Capacity:	175 ft^3/hr
Max Working Pressure:	5 psi

ELECTRIC POWER SENSORS

Ohio Semitronics Series PC5 wattmeters are used as electric power sensors. They utilize Hall effect devices as multipliers taking the product of the instantaneous voltage and current readings to determine the electrical power. This technique automatically takes power factor into consideration and produces a true power reading.

Power Factor Range:	1 to 0 (lead or lag)
Response Time:	250 ms
Temperature Effect:	1% of reading
Accuracy:	0.5% of full scale

HEAT FLUX SENSORS

The Hy-Cal Engineering Model BI-7X heat flow sensor is used for the measurement of heat flux. The sensor consists basically of an insulating wafer, with a series of thermocouples arranged such that consecutive thermoelectric junctions fall on opposite sides of the wafer. This assembly is bonded to a heat sink to assure heat flow through the sensor. Heat is received on the exposed surface of the wafer and conducted through the heat sink. A temperature drop across the wafer is thus developed and is measured directly by each junction combination embodied along the wafer. Since the differential thermocouples are connected electrically in series, the voltages produced by each set of junctions is additive, thereby amplifying the signal directly proportional to the number of junctions. The temperature drop across the wafer, and thus the output signal, is directly proportional to the heating rate.

Operation Temperature:	-50°F to 200°F
Response Time:	6 seconds
Linearity:	2%
Repeatability:	0.5%
Sensitivity:	2 mv/BTU/ft^2-hr
Size:	2" x 2"

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