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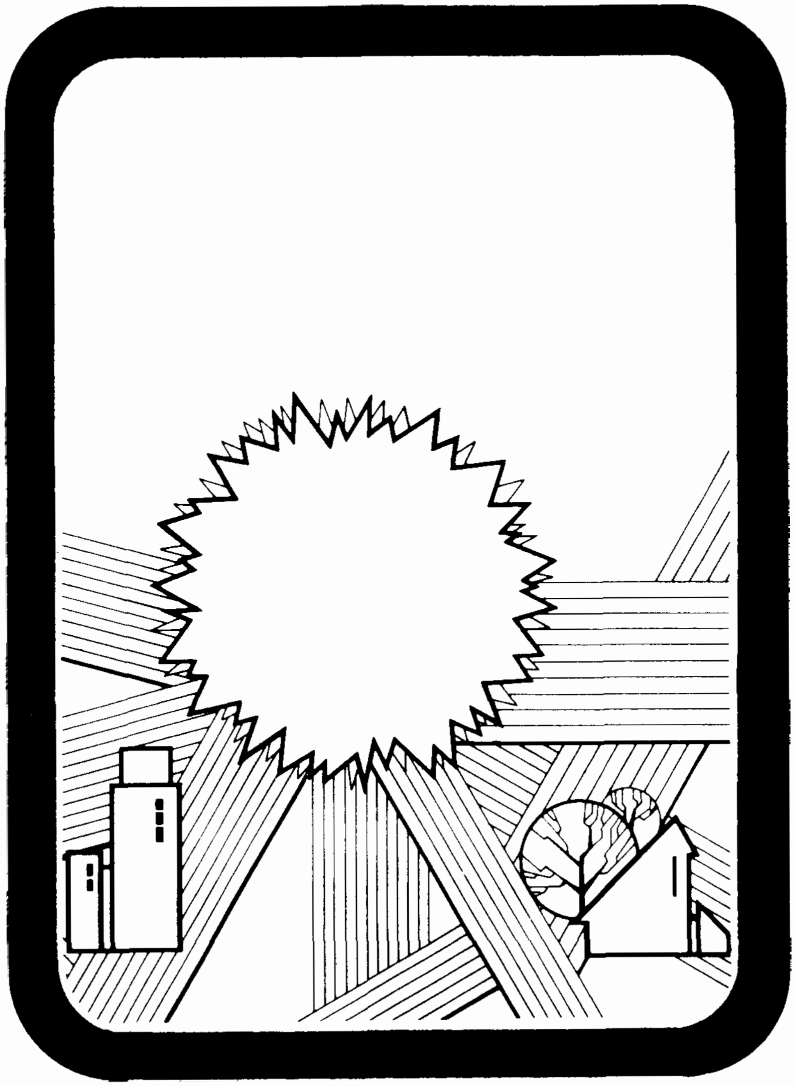
(DE82016025)

COMPARATIVE REPORT: PERFORMANCE OF ACTIVE SOLAR SPACE COOLING SYSTEMS

1981 COOLING SEASON



U.S. DEPARTMENT OF ENERGY

NATIONAL SOLAR DATA PROGRAM

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COMPARATIVE REPORT

PERFORMANCE OF ACTIVE SOLAR SPACE COOLING SYSTEMS

1981 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. There is comprehensive data on four absorption chiller cooling systems and one Rankine cooling system. Three of these systems, including the Rankine system, demonstrated that solar cooling can be operated efficiently and provide energy savings. Good designs and operating procedures are discussed. Problems which reduce savings are identified. There is also a comparison of solar cooling by absorption, Rankine, and photovoltaic processes.

Several general statements may be made concerning the seasonal performance of these sites with respect to each other, based on the data and observations from the 1981 cooling season:

- o Solar cooling systems are able to provide energy savings but are not cost-effective in terms of reasonable pay-back periods.
- o Solar cooling will not provide savings everywhere. Low insolation levels and cooling loads do not warrant use of solar cooling in some locations.
- o Solar Rankine cooling systems are able to provide solar cooling as efficiently as solar absorption cooling.
- o Solar cooling systems which were built with a storage bypass had better solar utilization.
- o Use of solar storage can be avoided with appropriate sizing of collector area or use of series boilers.
- o Flat-plate collectors can be used efficiently for solar cooling.

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 1981 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 have been analyzed and are 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 are provided include:

- Energy savings and operating costs in terms of BTU
- Energy savings in terms of dollars
- Overall solar cooling efficiency and coefficient of performance
- Hourly building cooling loads
- Actual and long-term weather conditions
- Collector performance
- Collector area to tons of chiller cooling capacity
- 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 with primary assistance from the Department of Housing and Urban Development. The availability of these results of the NSDN

are in large part due to the continuing support of these and other organizations, including the Boeing Aerospace Corporation, National Bureau of Standards, National Aeronautics and Space Administration, 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 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 selected active sites in the National Solar Data Network (NSDN). Results presented have been developed on the basis of analysis of instrumented sites monitored through the 1981 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 have been established since the inception of the National Solar Heating and Cooling Demonstration Program. As of November 1981 via planned Program Opportunity Notices or Requests for Proposals (RFPs), 45 of these sites were instrumented and included in the NSDN.

The Department of Energy (DOE) has the responsibility for the solar energy program; however, other government agencies are significantly involved. Those agencies include the National Aeronautics and Space Administration, the Department of Defense, the National Bureau of Standards, Argonne National Laboratory, and Boeing Aerospace Corporation. State and local government, portions of the private sector, and other groups within DOE are also active participants.

The NSDN sites selected by DOE include a broad range of solar system types and geographical locations with the United States. Figure 1 shows the location of NSDN sites with solar cooling systems having measured performance during the 1981 cooling season. Sensors are sampled automatically, and the data are stored at each site for one or more days (Figure 2). Since December 1979, the data have 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.

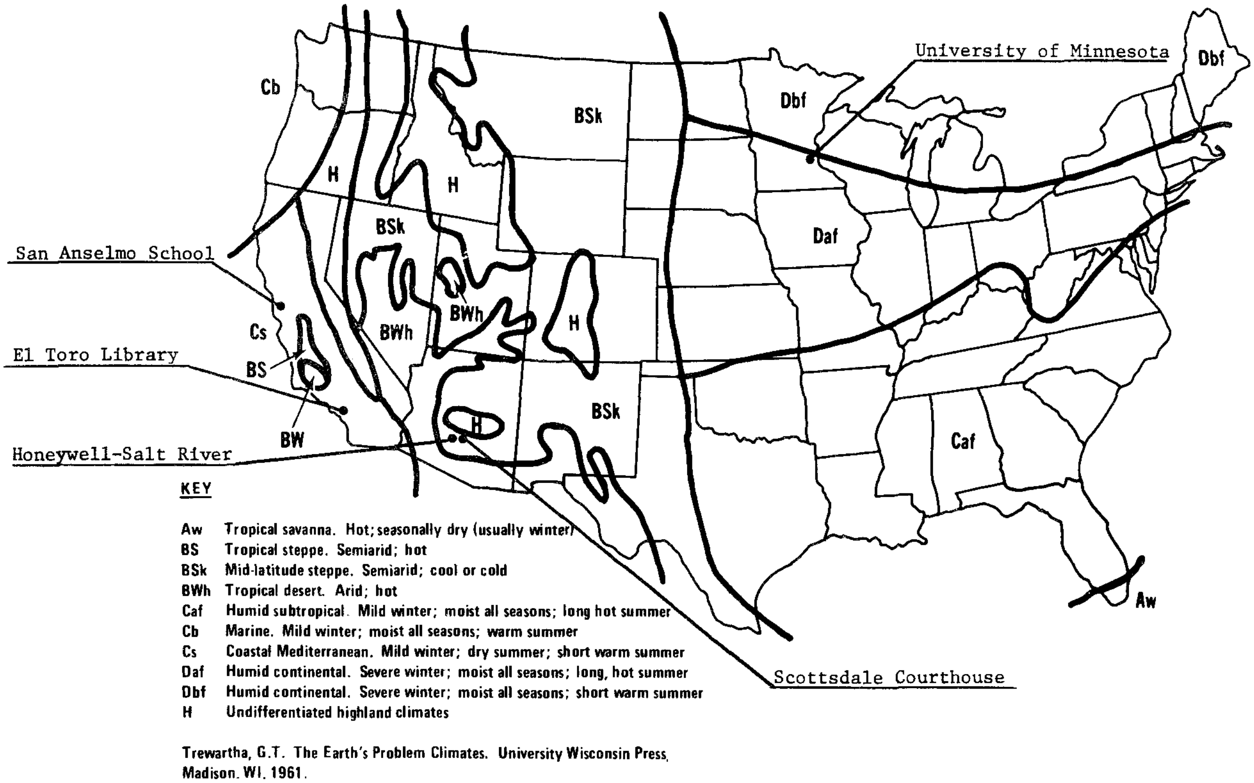

Figure 1. Climatological Map of the United States
Showing Sites Discussed in this Report

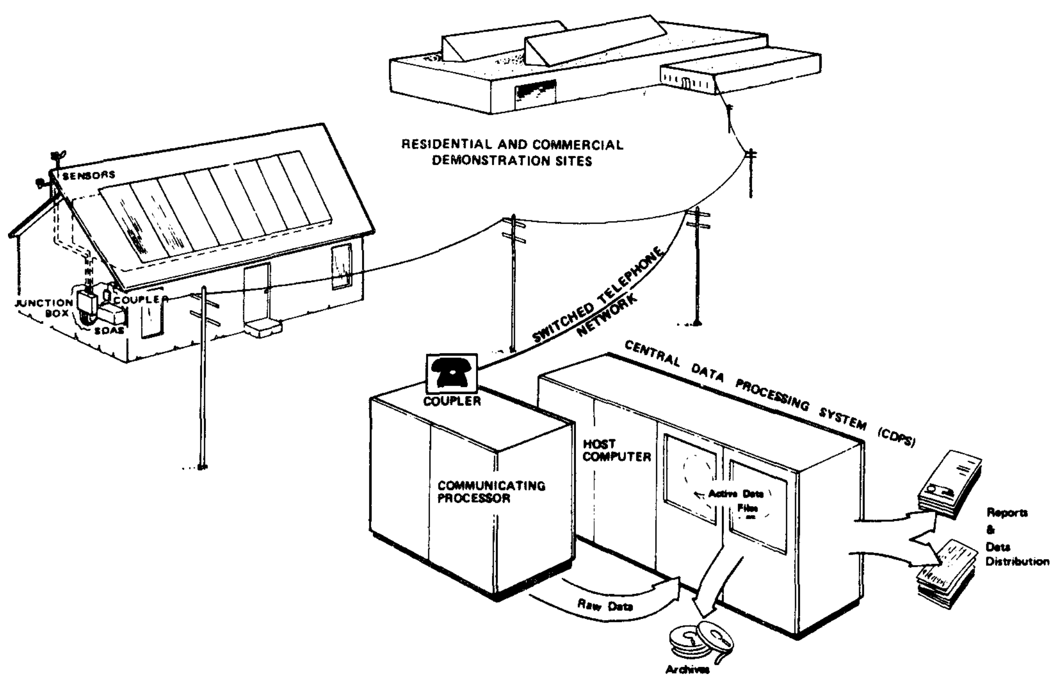

Figure 2. The National Solar Data Network

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 attempt to 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 representative cooling systems selected into the program. Cooling systems have been included in the NSDN because a history of performance is needed to act as a basis for recommendations concerning various options available.

The NSDN detailed measurement data for these systems were analyzed in accordance with standarized procedures and are 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 fully extended over the past year to analysis of the interfaces between subsystems to permit better understanding of overall system operation, energy flow, and energy uses. More recently it has been recognized that further analysis of the entire building envelope is required to fully account for the uses of internal thermal "losses" and the passive solar component.

Embodied in the NSDN methodology employed during the 1981 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.

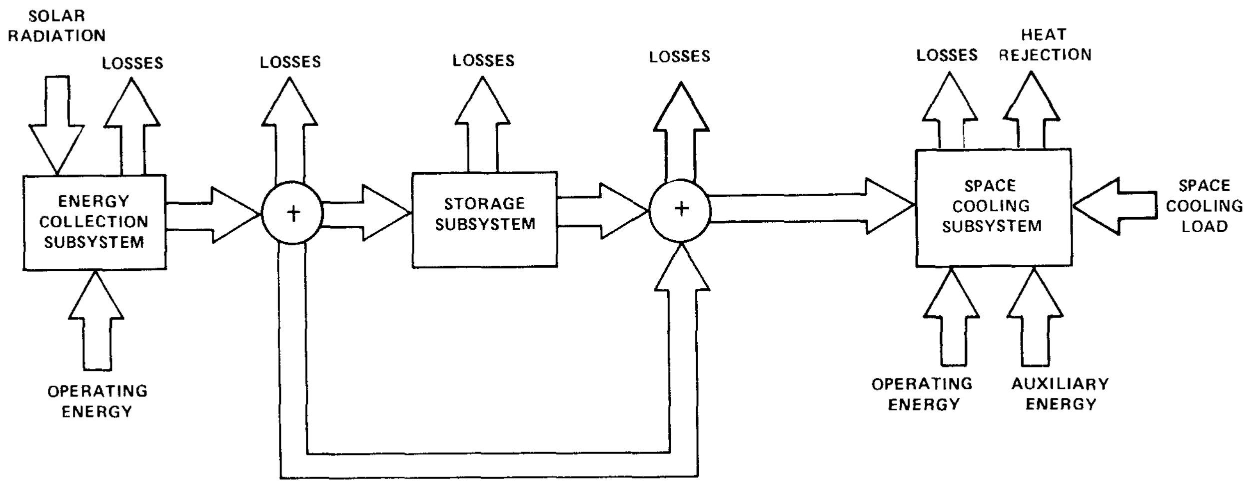

Figure 3. Typical Space Cooling Energy Flow Diagram

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 an understanding of solar radiation, flows between subsystems, auxiliary and operating energies, load requirements and losses as shown in the hypothetical flow diagram Figure 3. Appendix C contains actual flow diagrams for the cooling season for selected 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
  - ECSS solar conversion efficiency

- Solar fractions
- Energy consumed, energy saved

Solar system sensor data consists of:

- o Temperature sensors in each subsystem
- o Flow meters in each subsystem
- o Auxiliary power used via wattmeters, flow meters
- o State sensors (i.e., on/off, etc.)

Weather data consists of:

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

A more detailed discussion of space cooling analysis methodology is contained in Appendix K.

SOLAR COOLING WITH ABSORPTION CHILLERS

At the present time, most commercially available solar cooling technology utilizes a system of solar collectors and an absorption chiller. The absorption chiller can produce cooling from heat; in this instance, hot water from solar collectors.

Absorption chillers utilize a partial vacuum to permit concentration of the refrigerant and regeneration of the absorbent, which has a high affinity for the refrigerant. Most absorption chillers used 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 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 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.

The vaporized refrigerant passes to the condenser where it gives up its latent heat to the condensing water and is liquefied. The refrigerant then

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

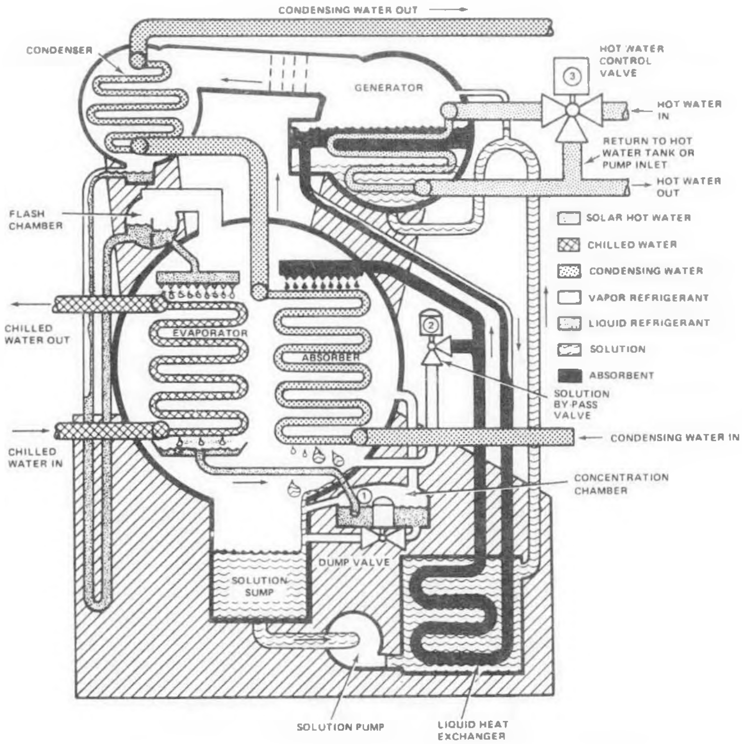

Figure 4. Arkla SOLAIRE WFB-300 Water Chiller Diagram

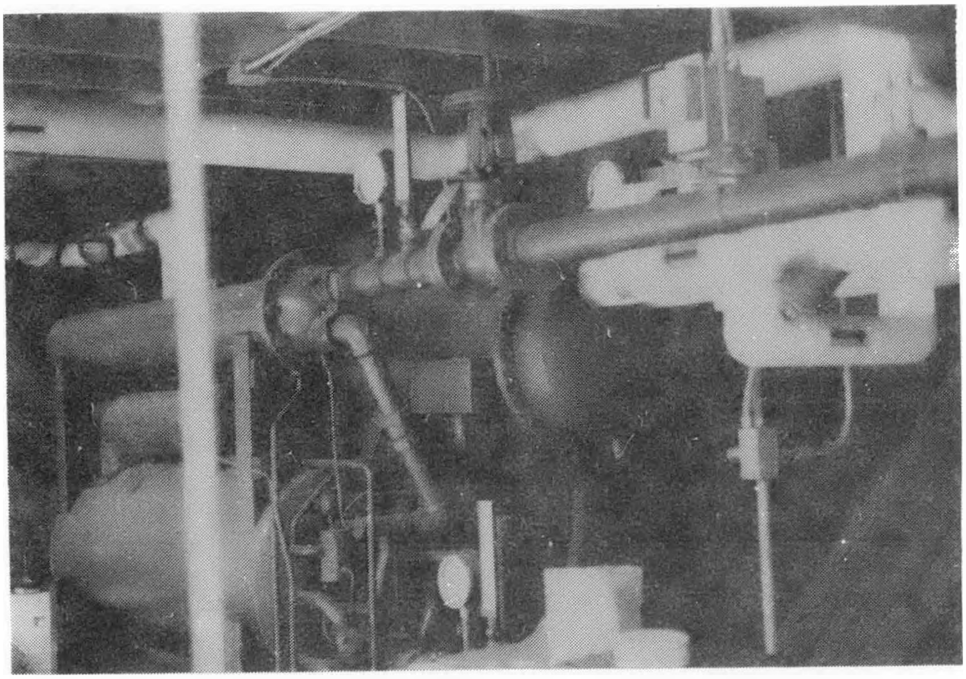

Figure 5. Arkla WFB-300 Water Chiller Installed at El Toro Library

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 the 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. Most solar cooling systems in the NSDN employ solar collectors and an absorption chiller. Solar cooling sites in the NSDN use either concentrating collectors, flat-plate collectors or evacuated-tube collectors. The hot and cold 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 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.

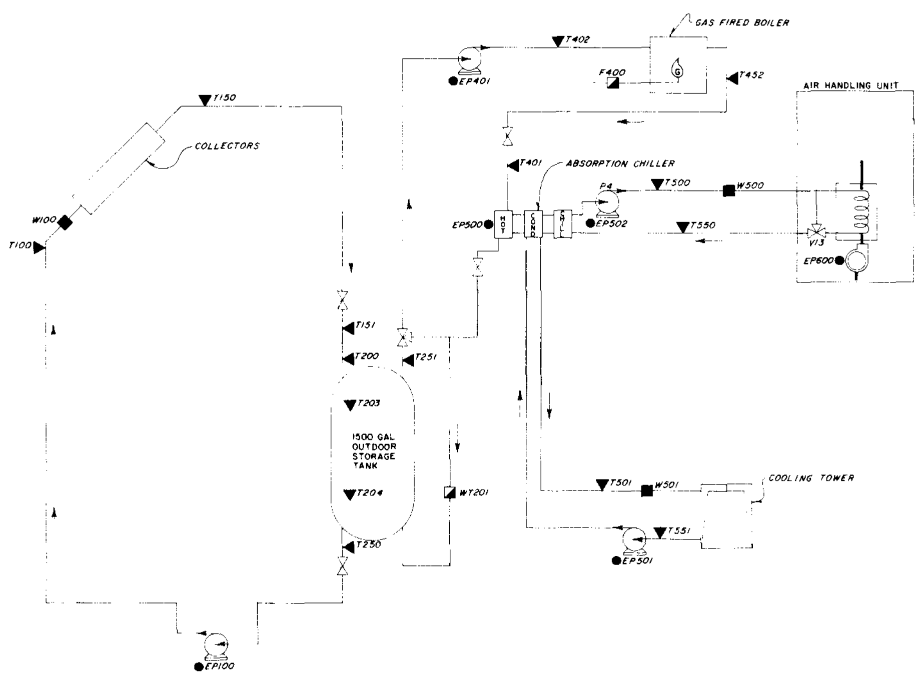

Figure 6. Generalized Solar Energy Cooling System Schematic

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

SOLAR COOLING WITH RANKINE/CHILLERS

A solar Rankine cooling system, though not yet available commercially, presents an alternative to solar cooling using an absorption chiller. A Rankine cooling system incorporates a solar-driven Rankine engine to power a conventional vapor compressor 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 mixed to provide sufficient power to the vapor compressor. The generator is also used to generate electricity when the Rankine engine provides more power than required to drive the vapor compressor.

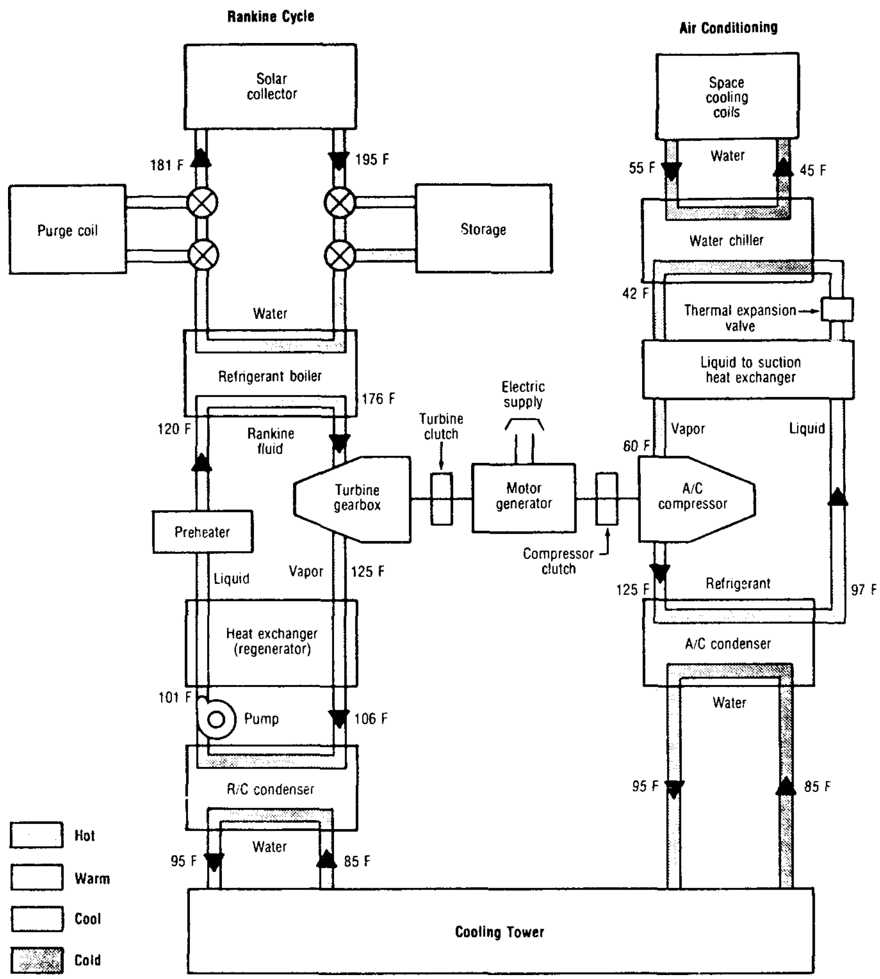

Figure 7. Solar-Powered 25-Ton AC - 20-HP RC Cooling Unit

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. Alternately, 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.

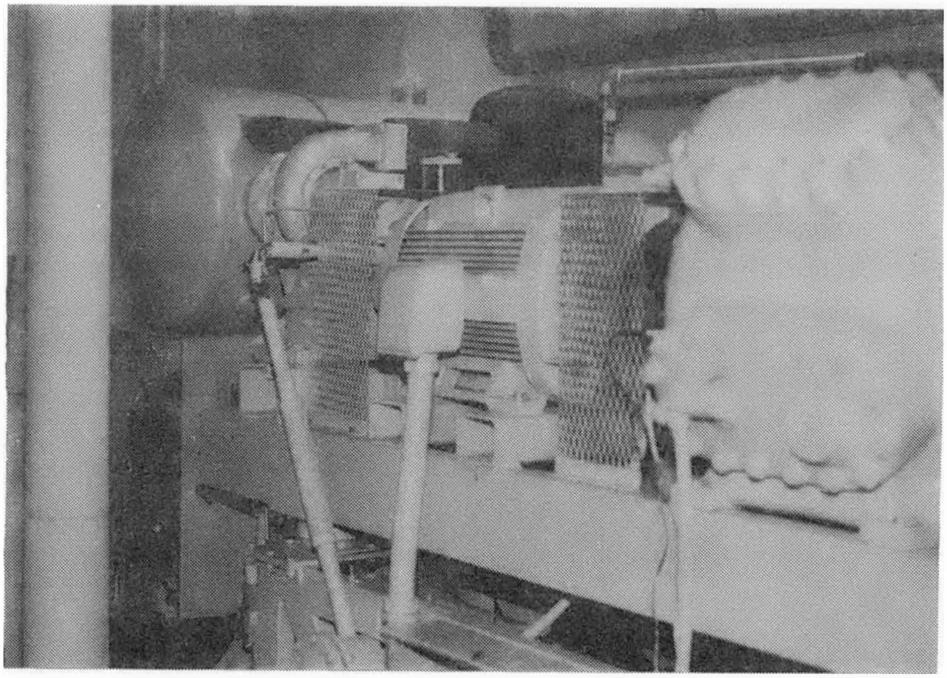

Figure 8. SRP Rankine Engine - Motor/Generator-Compressor

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 results of NSDN systems operational for the 1981 cooling season, with discussions of specific cases and conclusions which may be drawn from the data. In addition, analysis results are presented in tables and graphic form to highlight key information.

Section IV 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 are provided in Appendix A. Schematics which identify the configuration of each site are contained in Appendix B. Flow diagrams which show major sources and uses of solar and auxiliary heat for these sites are contained 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 & Location	Collector & Storage Type	Building Type	Cooling Equipment
El Toro Library El Toro, California	1,427 Sq. Ft. Evacuated Tube Collectors 1,500-Gallon Hot Storage Tank	10,000 Sq. Ft. Library	Arkla 25-Ton Absorption Chiller
Honeywell-Salt River Phoenix, Arizona	8,200 Sq. Ft. Flat-Plate Collectors 2,500-Gallon Hot Storage Tank	55,000 Sq. Ft. Conditioned Cooling Space	Two 25-Ton Rankine Chillers 228-Ton Centrifugal Chiller
San Anselmo School San Jose, California	3,740 Sq. Ft. Evacuated Tube Collectors 2,175-Gallon Hot Storage Tank	34,000 Sq. Ft. School Building	Arkla 25-Ton Solar Absorption Chiller Four Arkla 25-Ton Gas-Fired Absorption Chillers
Scottsdale Courthouse Scottsdale, Arizona	2,723 Tracking-Type Concentrating Collectors 5,000-Gallon Hot Storage Tank	6,850 Sq. Ft. Office Building	Arkla 25-Ton Absorption Chiller 25-Ton Reciprocating Chiller
University of Minnesota Minneapolis, Minnesota	6,350 Slat-Type Concentrating Collectors 8,000-Gallon Hot Storage Tank	84,000 Sq. Ft. Underground University Facilities	Trane 148-Ton Absorption Chiller

Table 1B. SITE CHARACTERISTICS (DESCRIPTIONS)

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.	Valve V8 controls the solar energy from storage to the load subsystems. Improper operation of valve V8 did not allow full use of available solar energy and allowed boiler energy into storage tank. Collector area undersized. Low chiller COP through August.
Honeywell-Salt River 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. Loss of refrigerant in vapor compressor #2 resulting in low COPs.
San Anselmo School San Jose, 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. Poor collection of solar energy. Control logic on chillers prevents individual chiller use. Poor auxiliary chiller COP.
Scottsdale Courthouse Scottsdale, Arizona	Arkla chiller can operate from storage or directly from collectors.	Tracking problems. Collector array undersized for cooling capacity. Poor control operation which resulted in poor utilization of solar energy. Low generator flow rates during part of the season.
University of Minnesota Minneapolis, Minnesota	Trane chiller operates by a combination of solar and auxiliary energy.	Poor solar utilization. High losses to storage. Minor focusing problems.

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. Control valve V8 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 Honeywell-Salt River Project solar system is a 55,000-square-foot building that utilizes 8,200 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

Figure 9. El Toro Library Collector Array

cooling or electrical power generation. If solar energy is insufficient, then two 25-ton vapor chiller compressors 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.

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. Control valve V2 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 (AHU). 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.

The Scottsdale Courthouse solar energy system incorporates 2,723 square feet of tracking-type, concentrating collectors for a 6,850-square-foot office building. Collected solar energy can be delivered to a 5,000-gallon insulated

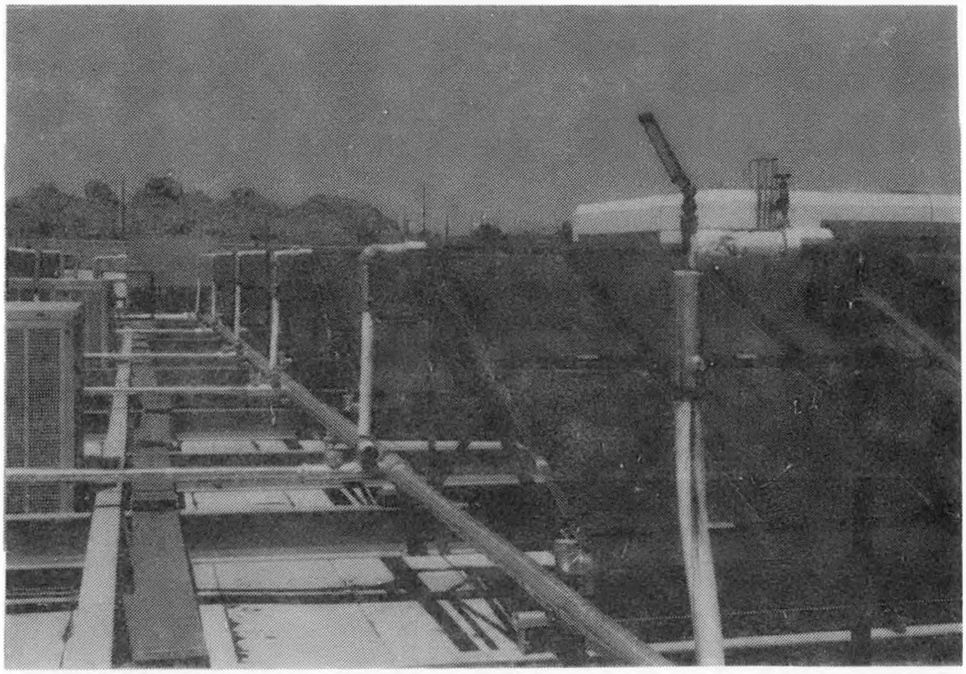

Figure 10. Honeywell-Salt River Collector Array

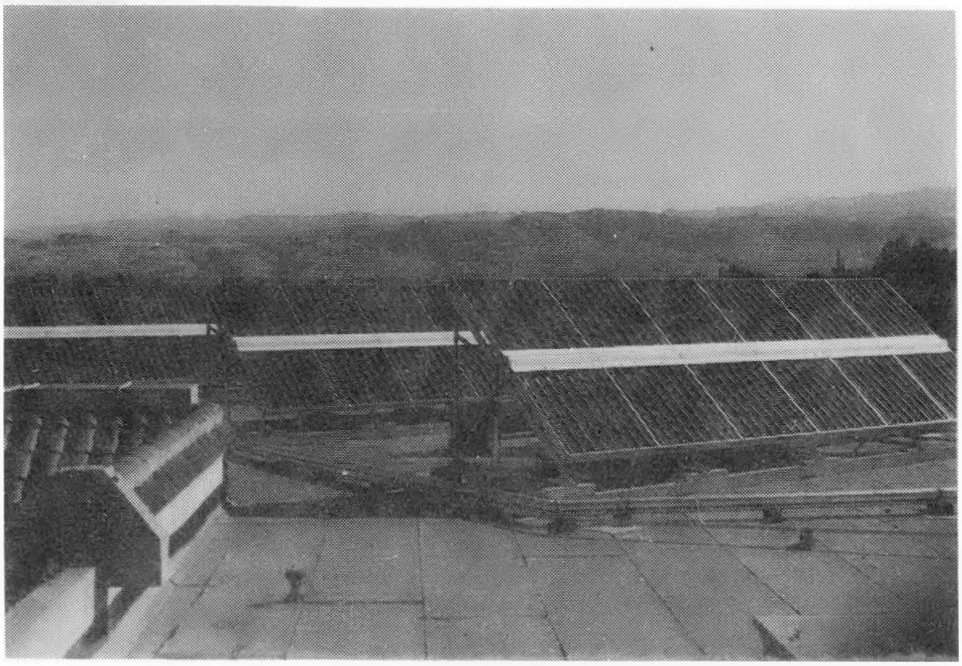

Figure 11. San Anselmo School Collector Array

buried storage tank or delivered directly to the space heating and cooling subsystems. During the heating mode, solar energy can be utilized from the storage tank or directly from the collectors. If solar energy is insufficient, then auxiliary electrical strip heaters will supply the heating demand. In the cooling mode, solar energy is supplied to the generator inlet of a WFB-300 Arkla 25-ton absorption chiller from the storage tank or directly from the collector subsystem. An auxiliary reciprocating chiller will supply the space cooling load if solar energy is insufficient to meet the demand.

This solar system can utilize energy from storage or energy directly from the collectors to a solar-operated absorption chiller.

Figure 12. Scottsdale Courthouse 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 148-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.

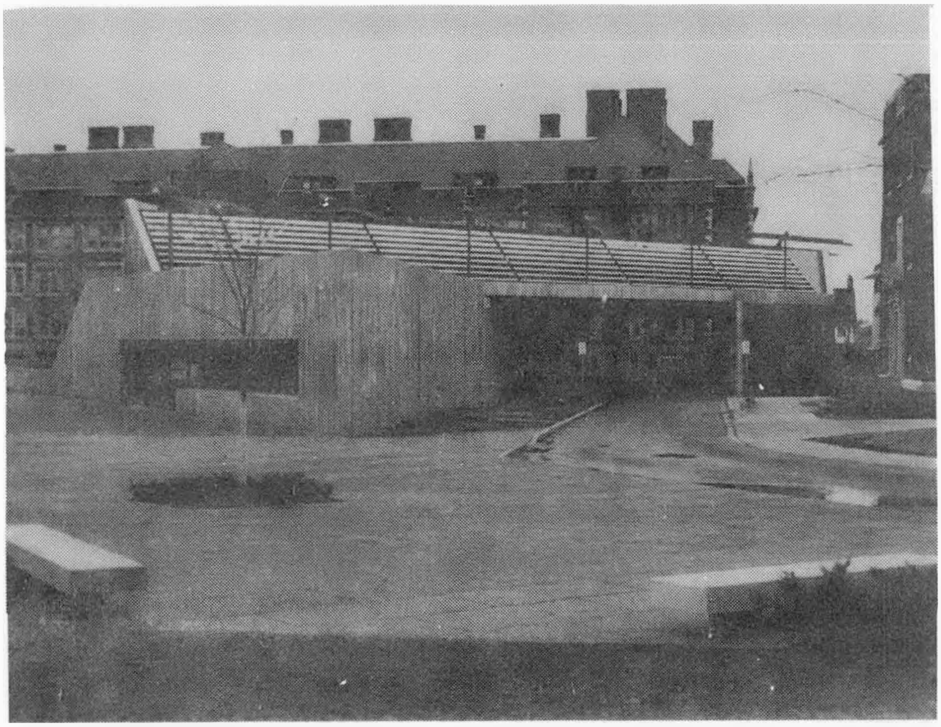

Figure 13. University of Minnesota Collector Array

This solar system is similar to that at 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.

For a more in-depth understanding of each solar system, refer to Appendix A, Site Descriptions, and Appendix B, Site Schematics. Also, Appendix A 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. El Toro Library and San Anselmo School use evacuated-tube collectors while Honeywell-Salt River has flat-plate collectors. Scottsdale Courthouse and University of Minnesota both have concentrating collectors, but Scottsdale Courthouse employs a tracking-type, parabolic concentrator while University of Minnesota has a tracking slat-type, reflecting concentrator.

All solar sites utilize a hot storage tank, but, at Honeywell-Salt River 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. None of the sites incorporate a cold storage tank.

El Toro Library and 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 El Toro Library must use solar energy from storage.

San Anselmo School and Scottsdale Courthouse solar sites utilize solar energy only to supply the generator inlet of the absorption chiller. However, Scottsdale Courthouse can use solar energy directly from the collector outlet, while San Anselmo School must draw solar energy from the storage tank.

The Honeywell-Salt River 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 compressor 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 Honeywell-Salt River which uses a Rankine engine to run a vapor compressor chiller. Also, all sites are located in a moderate climate except for the University of Minnesota.

Section III

COMPARATIVE DATA, PERFORMANCE ANALYSIS AND RESULTS

This section contains data tables and graphs detailing the performance of the major components and subsystems of the solar cooling systems. The data were accumulated from NSDN monthly performance reports for the 1981 cooling season. Seasonal data for the sites are contained in Appendix D:

Because the length of the cooling season is variable, the number of data months for each site is different. Data months available for this comparative are:

Site	Month (1981)
El Toro Library	March through June, August through November
Honeywell-Salt River	June through October
San Anselmo School	March through November
Scottsdale Courthouse	March, April, July, August, October, and November
University of Minnesota	July through September

A. WEATHER CONDITIONS

One of the most significant factors affecting the performance of a solar collector subsystem is the weather. The amount of available insolation and the ambient temperature strongly affect the collector efficiency. The cooling loads are also strongly affected by the weather conditions. Thus, for a better understanding of space cooling and solar collection, it is necessary to have a knowledge of the weather conditions.

Table 2 portrays the measured and long-term weather conditions for each site. It should be noted that there is a significant difference between the measured and long-term values for most of the weather parameters.

In Figure 14, the actual insolation is compared to the long-term insolation as a percentage of the long-term. All the sites received below the long-term average insolation values. Scottsdale Courthouse received the smallest percentage of insolation, 85% of the long-term, and El Toro Library the highest, 91% of the long-term.

A comparison of the ambient temperatures shows much warmer temperatures at the sites than the long-term temperatures. The average ambient temperature for the eight-month period at El Toro Library was 7°F warmer than the long-term average for the same period; 71°F versus 64°F. This increase resulted in 1,690 cooling degree-days for the season. The long-term average was 571 cooling degree-days.

For all the sites, there was a combined total of 9,299 cooling degree-days measured but there were only 6,383 long-term average cooling degree-days.

As designers use long-term weather information to size the solar energy system, large deviations from long-term patterns will affect the performance

Table 2. WEATHER CONDITIONS

SITE		DAILY INCIDENT SOLAR ENERGY AT COLLECTOR TILT (BTU/FT² -DAY) MEASURED	LONG-TERM AVERAGE	AMBIENT TEMPERATURE (°F) MEASURED	LONG-TERM AVERAGE	HEATING DEGREE-DAYS MEASURED	LONG-TERM AVERAGE	COOLING DEGREE-DAYS MEASURED	LONG-TERM AVERAGE
El Toro	(Total)	-	-	-	-	250	856	1,690	571
Library	(Average)	1,695	1,861	71	64	31	107	211	71
Honeywell-	(Total)	-	-	-	-	5	17	3,479	2,951
Salt River	(Average)	1,977	2,278	88	84	1	3	696	590
San Anselmo	(Total)	-	-	-	-	909	1,129	914	444
School	(Average)	1,702(1)	1,893(1)	65	63	101	125	102	49
Scottsdale	(Total)	-	-	-	-	189	444	2,499	1,987
Courthouse	(Average)	1,646	1,946	78	73	32	74	417	331
University of	(Total)	-	-	-	-	102	205	717	430
Minnesota	(Average)	1,404 451(2)	1,627	71	70	34	68	239	143

(1)Value excludes November data.
(2)Measured diffuse component of solar radiation.

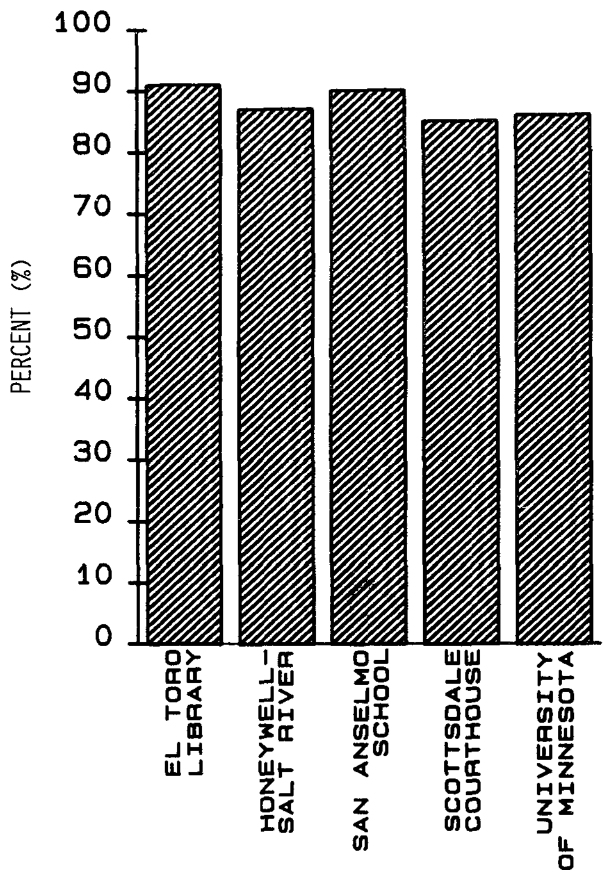

Figure 14. Actual vs. Long-Term Insolation

of the system. The lower levels of insolation would mean less solar energy is available for collection. It would also reduce collector efficiency and, thus, reduce the solar energy delivered to the load and the solar fraction would subsequently drop below design predictions.

The higher than long-term ambient temperatures or cooling degree-days would have several effects. The cooling load for the site would be higher than design, further lowering the cooling solar fraction. The ability to reject waste heat from the cooling equipment to the environment would be reduced. Electric consumption for heat removal equipment would be increased, thus, increasing solar-specific operating energy and lowering system savings.

One slight benefit of the higher ambient temperature would be an increased collector efficiency. This would result from the decreased temperature difference between the ambient and the collector plate, or absorber tubes for the sites with evacuated tubes or concentrating collectors.

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

Collector efficiency can be approximated by the equation:

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

$$\text{where } OPPNT = \frac{Tin-Tamb \ (^\circ F)}{I \quad (BTU/ft^2\text{-hr})}$$

At Honeywell-Salt River: $FrU_L = -0.51$ Tin = Collector Inlet Temperature
$FrT_\alpha = 0.67$ Tamb = Collector Ambient Temperature

Case (1)	Case (2)	Case (3)	Case (4)
Tin = 175°F	Tin = 175°F	Tin = 175°F	Tin = 175°F
Tamb = 84°F	Tamb = 88°F	Tamb = 84°F	Tamb = 88°F
I=300 BTU/ft^2-hr	I=300 BTU/ft^2	I=0.9 x 300 BTU/ft^2-hr	I=0.9 x 300 BTU/ft^2-hr
OPPNT(1) = 0.303	OPPNT(2) = 0.29	OPPNT(3) = 0.337	OPPNT(4) = 0.322
η(1) = 51.5%	η(2) = 52.2%	η(3) = 49.8%	η(4) = 50.5%

Thus, during these typical operating conditions, increasing the ambient temperature increased the collector efficiency by 0.7%. Lowering the level of insolation decreased the efficiency by 1.7%. The combined result of both changes would decrease the collector efficiency by one percent. The resulting solar energy collected at this reduced collector efficiency would be:

$$SECA_f = \eta \times I = 50.5\% \times (0.9 \times 300 \ BTU/ft^2\text{-hr}) = 136.4 \ BTU/ft^2\text{-month}$$

The solar energy that would have been collected with the initial conditions is:

$$SECA_i = \eta \times I = 51.5\% \times (300 \ BTU/ft^2\text{-hr}) = 154.5 \ BTU/ft^2\text{-hr.}$$

$SECA_f$ is approximately 12% less than $SECA_i$.

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 type collectors would seem to be the best choice for this application. However, of the five sites compared, the flat-plate collectors provided the best performance.

Collector array efficiencies, the percentage of total available insolation collected by the array, are provided in Table 3. These efficiencies range from seven percent at Scottsdale Courthouse to 38% at Honeywell-Salt River. The operational collector efficiency is the percentage of available insolation during collector operation that is collected by the array. These values are also provided in Table 3. Again, the poorest performance was recorded by Scottsdale Courthouse, eight percent efficiency. Honeywell-Salt River had the highest operational collector efficiency, 44%.

Table 3. COLLECTOR PERFORMANCE

SITE	TYPE COLLECTOR	TILT	AZIMUTH	INSOLATION AT COLLECTOR TILT (BTU/FT^2-DAY)	ARRAY EFFICIENCY (%)	OPERATIONAL EFFICIENCY (%)	AMBIENT TEMPERATURE
El Toro Library	Evacuated Tube	19°	30°W	1,695	29	31	71
Honeywell-Salt River	Flat-Plate	20°	34°W	1,977	38	44	88
San Anselmo School	Evacuated-Tube	40°	0°	1,702(1)	20(1)	23(1)	66(1)
Scottsdale Courthouse	Concentrator	0°	0°	1,476(2)	7(2)	8(2)	74(2)
University of Minnesota	Concentrator	45°	0°	1,404/451(3)	25	38	71

(1)Excludes November data.
(2)Excludes April and August data.
(3)Diffuse insolation.

Honeywell-Salt River, using single-glazed Lennox flat-plate collectors, was able to collect solar energy more efficiently than the four other collector arrays. Two of these arrays were evacuated-tube and the remaining two were concentrators. The lead Rankine engine at Honeywell-Salt River was able to use solar fluid temperatures as low as 150°F, but the average fluid temperature during operation was 176°F. This means that the collector system at Honeywell-Salt River was operating at approximately the same temperatures as the solar absorption cooling systems. The average operating temperature of these systems was approximately 175°F.

The concentrating collector array at Scottsdale Courthouse experienced several problems which contributed to the poor performance during the season. The collector system is activated by a light-level sensor. If the incident light was strong enough to activate the controls but the light was largely diffuse, the collector array would operate at a loss since the diffuse component of the incident light is not usable by the concentrators. The collectors did not focus properly on the absorber tube due to loose connections in the linkage. One of the four banks of collectors sometimes jammed in the stowed position during collector pump operation. All these problems contributed to the extremely poor collector performance.

The University of Minnesota concentrating collector array also experienced focusing problems which contributed to reduced efficiency, though the collector array performed much better than the Scottsdale Courthouse array. Large fluxuations in collector temperatures would occur under steady-state load and insolation. These fluctuations were indicative of a collector focusing problem. The University of Minnesota system collector efficiency was 25% versus seven percent at Scottsdale Courthouse.

The diffuse component of the insolation is listed in Table 3 for University of Minnesota. This was 451 BTU/ft^2-day or 32% of the total insolation. No accurate measurement could be made for Scottsdale Courthouse but the diffuse component of the insolation in the Scottsdale Courthouse climate is considerably smaller. Thus, concentrating collectors are more suitable in Arizona than in Minnesota.

The collector arrays at El Toro Library and San Anselmo School are made up of General Electric TC-100 evacuated tubes. Both systems are very similar in construction, yet performance during the season was very different. El Toro Library had a seasonal collector efficiency of 29% and an operational efficiency of 31%. San Anselmo School had a seasonal collector efficiency of 20% and an operational efficiency of 23%.

Figure 15 and Figure 16 show the average hourly operating efficiencies plotted against operating point for El Toro Library and San Anselmo School during the month of August. A least-squares fit of the plots yields for El Toro Library a y-intercept of 0.42 and a slope of -0.30, and for San Anselmo School a y-intercept of 0.40 and a slope of -0.39. Also shown on Figures 15 and 16 is the manufacturer's curve[1] for a single panel. This curve has been corrected for gross collector area. It can be seen that both collector arrays performed well below the manufacturer's expectations. There are many possibilities for the causes of these poor performances but further discussion is beyond the focus of this report.

Both El Toro Library and San Anselmo School use the same type of collector controller, but the controller at El Toro Library operated more effectively than the one at San Anselmo School. The storage bypass controller was

[1]Manufacturer's curve generated from testing by DSET Laboratories, Inc. Report #7851204-1, December 5, 1978 - March 4, 1979.

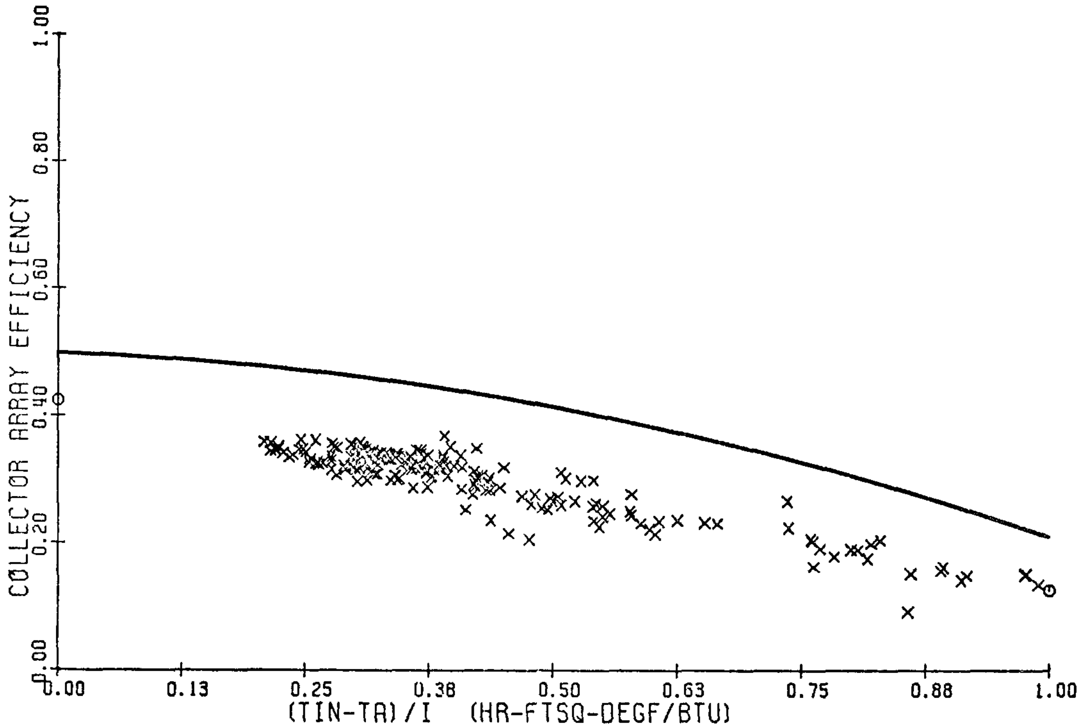

Figure 15. Average Collector Efficiency
El Toro Library
August 1981

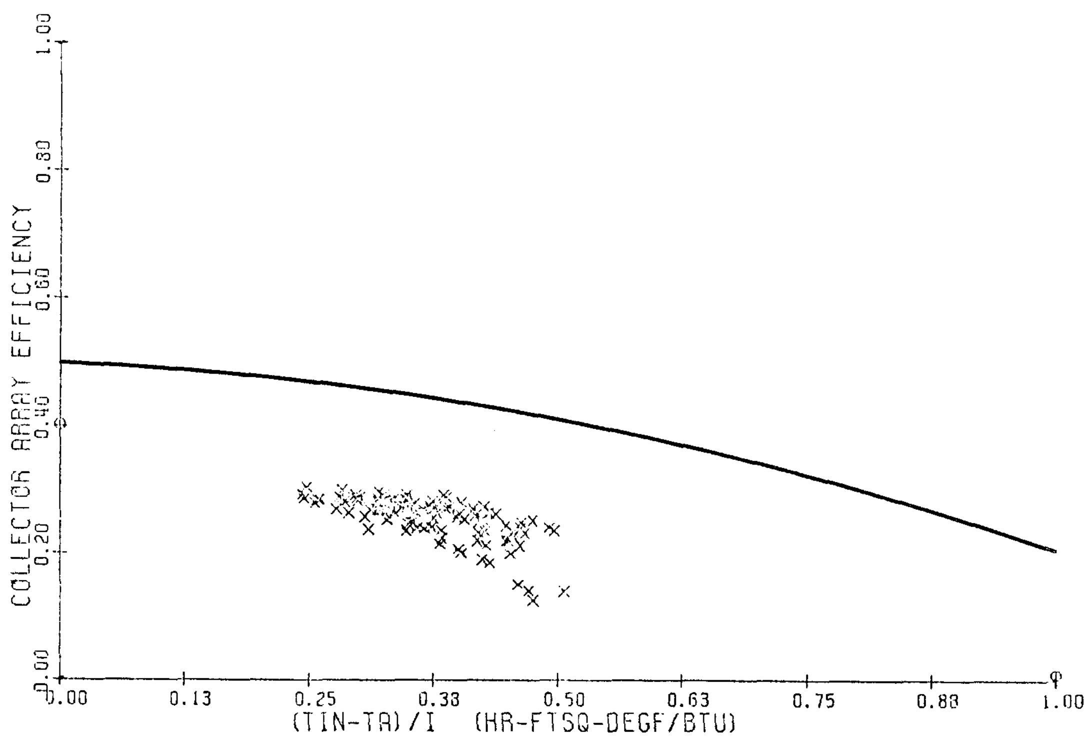

Figure 16. Average Collector Efficiency
San Anselmo School
August 1981

designed differently at the two sites. At El Toro Library, the storage tank is bypassed when the collector outlet temperature falls below the storage tank temperature. At San Anselmo School, the storage tank is bypassed when the collector outlet temperature drops below 175°F. If the storage tank at San Anselmo School was hotter than the collector array, there was negative energy to storage, negative solar energy collected, and reduced collector efficiency. The San Anselmo School collector array was exposed to an average solar insolation of 1,702 BTU/ft^2-day while the El Toro Library array was exposed to an average of 1,695 BTU/ft^2-day.

C. STORAGE PERFORMANCE AND OVERALL SOLAR UTILIZATION

Each of the sites, with the exception of Honeywell-Salt River, uses a solar storage tank to hold and supply solar heated water for the space cooling subsystem. The Honeywell-Salt River solar energy system contains a solar storage tank but this is used for solar heating only. None of the sites use cold storage tanks.

No storage data is included for the University of Minnesota as the subsystem was not instrumented well enough to determine all energy flows. However, solar energy delivered to the absorption chiller was 19% of the collected solar energy. The remaining solar energy was lost from piping and from storage. The storage tank is buried in the earth. This would cause high storage losses during the summer months when the earth is relatively cool and the storage fluid is maintained at high temperatures.

The systems which made best use of the collected solar energy were Honeywell-Salt River and Scottsdale Courthouse. These systems were able to deliver 90% and 91%, respectively, of the collected solar energy to the load (see Table 4). Each of these systems incorporates a storage bypass which allows solar energy to be delivered directly from the collector. Each of these had poor storage utilization contributed, in part, by the storage bypass loop. Honeywell-Salt River used only two percent of the solar energy delivered to storage and Scottsdale Courthouse used only 30% of the solar energy delivered to storage. But, as seen from Table 4, much less energy was delivered to these storage tanks.

The solar utilization at Honeywell-Salt River would have been better except that the lead Rankine engine would not start up occasionally due to a lubrication problem. As a result, collected solar energy was intentionally rejected to the environment in order to prevent overheating.

The El Toro Library and San Anselmo School solar energy systems are both constructed so that solar energy is delivered to the load from storage only. These systems had somewhat poorer utilization of collected solar energy than Scottsdale Courthouse and Honeywell-Salt River. El Toro Library was able to deliver 51% of the collected solar energy to the loads and San Anselmo School delivered 46% of the collected solar energy to the loads.

The San Anselmo School solar storage tank appears to be utilized much better than the other systems. Seventy-three percent of the solar energy delivered to storage was later removed for delivery to the load, but the operating procedure mentioned in the collector discussion should be

Table 4. STORAGE PERFORMANCE AND SOLAR UTILIZATION

SITE	STORAGE CAPACITY (GAL/FT²)	ENERGY IN (BTU/FT²-MONTH)	ENERGY OUT (BTU/FT²-MONTH)	STORAGE LOSSES (BTU/FT²-MONTH)	STORAGE TEMPERATURE (°F)	ENERGY OUT / ENERGY IN (%)	STORAGE LOSSES / ENERGY IN (%)	ENERGY TO LOADS / COLLECTED ENERGY (%)
El Toro Library	1.05	13,663	7,527	6,133	170	55	45	51
Honeywell-Salt River(1)	0.30	47	1	6	102	2	13	90
San Anselmo School	0.58	7,287	5,310	1,980	164	73	27	46
Scottsdale Courthouse	1.84	1,520	451	1,080	175	30	71	91(2)
University of Minnesota	1.26	*	*	*	*	*	*	19

(1)Storage used for heating only.
(2)Excludes April and August data.
* Denotes unavailable data.

considered. At San Anselmo School, collector fluid is circulated through storage if the collector fluid outlet temperature is greater than 175°F regardless of the storage fluid temperature. This often resulted in energy being removed from storage and rejected back to the environment through the collector subsystem. The storage tank is effectively cooled; thus, less energy is lost to the environment from the storage tank. The storage tank efficiency is higher but the solar utilization is reduced.

The El Toro Library system had the second most efficient storage subsystem. Fifty-five percent of the energy delivered to storage was later removed for load consumption.

D. SOLAR CHILLER OPERATION

The ratio of collector area per ton of cooling capacity is shown in Table 5. The systems in which the solar chillers must operate using only solar energy have notably higher ratios. This is done to assure more steady operation of the chiller. Honeywell-Salt River, San Anselmo School, and Scottsdale Courthouse have an average collector area to ton cooling ratio of 140 ft^2/ton. El Toro Library and the University of Minnesota mix both solar and auxiliary energy on the generator side of the absorption chiller and large collector areas are not required. The average collector area to ton cooling ratio for El Toro Library and the University of Minnesota is 50 ft^2/ton.

The chiller which performed the best throughout the cooling season was at San Anselmo School. This chiller had an average COP of 0.60. COP is the ratio of energy removed from the space to energy delivered to the generator of

Table 5. SOLAR CHILLER COOLING CAPACITY AND PERFORMANCE

SITE	CHILLER SIZE (TONS)	COLLECTOR AREA/ TON OF COOLING (FT^2/TON)	SEASONAL CHILLER EFFICIENCY (COP)
El Toro Library	25	57	0.43
Honeywell-Salt River (two)	25	164	0.24
San Anselmo School	25	150	0.60
Scottsdale Courthouse	25	107	0.34
University of Minnesota	147.5	43	0.48

the chiller. The monthly COP values are shown in Figure 17. The COP of the chiller at San Anselmo School was excellent throughout the season, ranging between 0.53 and 0.61. The chiller performed very well in July and August. It delivered 23.71 million BTU of space cooling using 38.85 million BTU of solar energy. This was a COP of 0.61.

At El Toro Library, the absorption chiller performed somewhat more poorly. During the first four months of operation, the chiller COP ranged from 0.35 to 0.45. Changes were then made on the chiller and performance improved substantially. During the final four months of operation, the chiller COP ranged between 0.46 and 0.50. The average COP for the season was 0.43.

The University of Minnesota chiller operated well during the season. The average COP was 0.48. The chiller is the only source of space cooling and is run with both auxiliary and solar energy, much the same as El Toro Library. Because it is the only source of cooling, the chiller is operated more constantly which is a key factor to good chiller efficiencies.

Scottsdale Courthouse operated poorly throughout the cooling season. The average COP was 0.34 and the highest monthly COP was 0.38. This chiller's only source of energy was from the solar collectors which performed very poorly. Thus, the absorption chiller could not maintain steady, efficient operation.

The Honeywell-Salt River Rankine/chiller efficiency or COP is defined as the ratio of the cooling provided to the energy delivered to the Rankine engines for cooling. During the five-month season, 822.23 million BTU of solar energy were delivered to the Rankine engines for cooling. With this energy, the Rankine/chiller system was able to provide 200.41 million BTU of space cooling. This is a COP of 0.24.

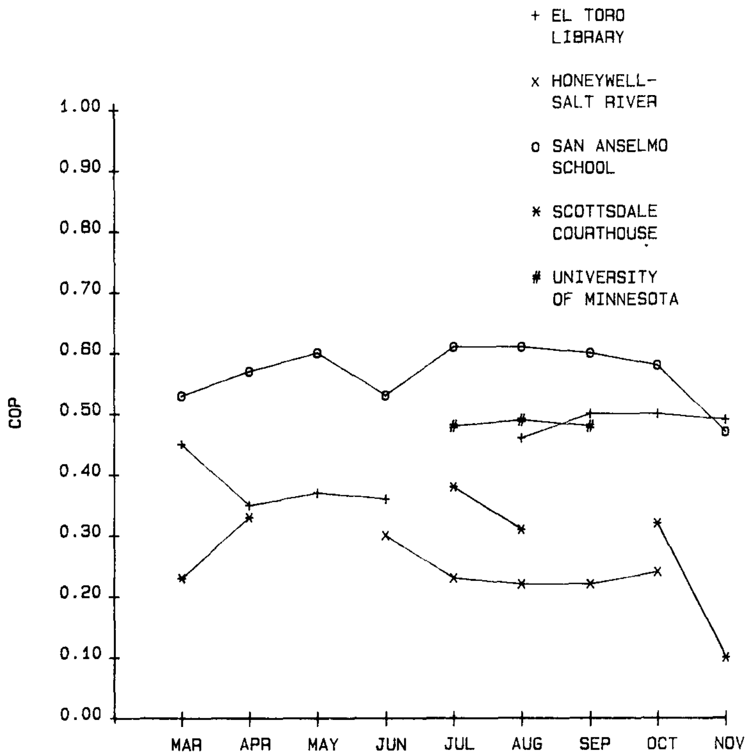

Figure 17. Monthly COP Values

To better understand this Rankine chiller COP, consider the following illustration: the thermal efficiency of a Rankine engine, approximately 0.08, multiplied by a vapor compressor thermal COP of 3.00 yields a Rankine/chiller COP of 0.24.

The highest monthly COP at Honeywell-Salt River was 0.30 in June. As seen from Figure 17, this COP value degraded slowly through the following months to 0.22 and then increased to 0.24 in October. During this period, one of the vapor compressors was losing refrigerant and performance dropped off. This problem was corrected and performance improved slightly in October.

E. COOLING LOADS

The space cooling load is the measure of energy removed from the space by the evaporator of the cooling equipment. Table 6 indicates the equipment cooling capacity installed at each site. This value has also been normalized to the floor area of the space for comparison purposes. For a more complete description of installed equipment, refer to the site descriptions in Appendix A. Also contained in Table 6 are the normalized cooling loads.

Honeywell-Salt River had the largest cooling capacity with a 278-ton total. But, when normalized to unit floor area, Scottsdale Courthouse had the largest normalized capacity, 7.30 tons of installed cooling capacity per 1,000 ft^2 of floor area. Honeywell-Salt River had the next highest normalized installed capacity with 5.05 $ton/10^3ft^2$. This high value is consistent with climate. The University of Minnesota, for example, is in a much cooler

Table 6. SPACE COOLING LOADS

SITE	TOTAL INSTALLED COOLING CAPACITY (TONS)	INSTALLED COOLING CAPACITY PER UNIT AREA ($TON/10^3 FT^2$)	COOLING LOAD (BTU/FT^2-MONTH)	NORMALIZED COOLING LOAD (BTU/FT^2-CDD)	COOLING DEGREE-DAYS (CDD)
El Toro Library	25	2.50	2,721	12.88	1,690
Honeywell-Salt River	278	5.05	11,924	17.12	3,479
San Anselmo School	125	3.68	1,287	12.67	914
Scottsdale Courthouse	50	7.30	8,871	21.30	2,499
University of Minnesota	147.5	1.76	1,757	7.35	717

climate and has an installed cooling capacity of 1.76 $ton/10^3ft^2$. El Toro Library and San Anselmo School fall between the above systems with 2.50 and 3.68 $ton/10^3ft^2$, respectively.

The amount of installed cooling capacity is not only climate dependent but also depends on building heat gains and usage patterns. This is reflected in the actual loads at the individual sites. Table 6 indicates the actual cooling loads at the sites. The first load column indicates the cooling load normalized to square foot of building area per month. The next load column indicates the cooling load normalized to square feet per Cooling Degree-Day (CDD). This last load can be considered an empirical heat gain coefficient for the building. This heat gain coefficient not only includes energy transfer through the building surfaces but gains to the building due to usage patterns, body heat from occupants, and internal heat gains from equipment (lighting, etc.).

Scottsdale Courthouse had the highest heat gain coefficient, 21.30 BTU/ft^2-CDD. Honeywell-Salt River was the next highest with 17.12 BTU/ft^2-CDD. Honeywell-Salt River had the highest load per unit floor area, 11,924 BTU/ft^2-month. The University of Minnesota had the lowest heat gain coefficient, 7.35 BTU/ft^2-CDD. This low value can be partially attributed to the design of the building. Much of the building, 95%, is below ground. Thus, the surrounding earth naturally cools the building and internal heat gains are the principal load.

The normalized cooling load for El Toro Library and San Anselmo School fell between the other sites. El Toro Library had a heat gain of 12.88 BTU/ft^2-CDD and San Anselmo School had a heat gain of 12.67 BTU/ft^2-CDD.

Both buildings have low heat transfer coefficients due to minimized window area.

Appendix F contains the hourly cooling load profile for each site for each month of data that is used in this report. This will allow a much better understanding of usage and control patterns on the cooling systems.

F. SOLAR COOLING PERFORMANCE

One method of judging the performance of a solar energy system is to compare the solar fraction to the system design solar fraction. For a solar cooling system, the cooling solar fraction is defined as the ratio of the space cooling load which is supplied by solar energy to the total space cooling load. Figure 18 shows the measured seasonal cooling solar fractions and the design annual cooling solar fractions.

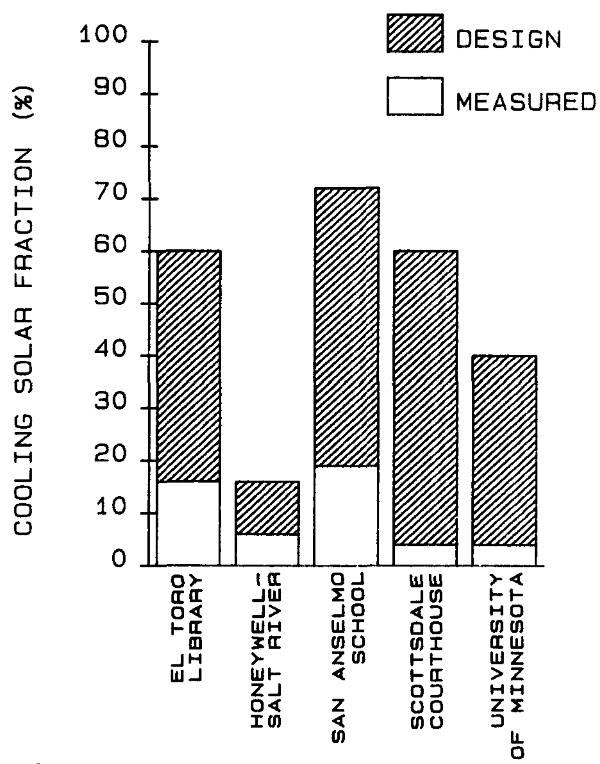

Figure 18. Measured Seasonal and Design Annual Cooling Solar Fractions

It is evident from Figure 18 that none of these systems performed as well as expected. The system which provided the closest to design solar fraction was Honeywell-Salt River. The system provided a cooling solar fraction of six percent versus the design cooling solar fraction of 16%. The systems which performed furthest from the design values were San Anselmo School and Scottsdale Courthouse. The differences between design and actual solar fractions were 55% and 56%, respectively. El Toro Library had a difference of 44% and the University of Minnesota had a difference of 36%.

The design values for these systems are not unreasonably high. There were lower insolation and higher cooling loads than one would predict from long-term weather conditions. Each site had some component that failed or was not functioning up to the specifications of the manufacturer and each site had high solar heat losses. With the exception of Honeywell-Salt River, the collector subsystems performed rather poorly. Each of the chillers, at some point in the season, had poor chiller efficiencies. The highest average chiller efficiency was 0.60 at San Anselmo School. A properly set up absorption chiller should provide chiller efficiencies between approximately 0.50 and 0.70 when used for solar application. If each system's components had been functioning as specified and the actual weather conditions had approximated long-term values, then the design solar fraction could have been attained.

A simplified extrapolation is performed below for El Toro Library in order to show that the system could provide the designed cooling solar fraction during long-term weather conditions and proper equipment performance.

During actual system operation, 92% of the total available radiation was incident on the array when the collector pump was operational. This 92% value will be used for available solar radiation. The total insolation during the eight-month period was 590 million BTU[1] which was 91% of the long-term average. So the available long-term solar radiation (Ia) would have been:

$$Ia = 590^{1} \times 10^{6} \text{ BTU} \times 0.92/0.91 = 597 \text{ million BTU}$$

The actual operational collector efficiency was 31%. If the collector array had been performing as predicted by the manufacturer's single-panel test data, the collector efficiency would have been approximately 44%.[2] The collected solar energy (SECA) would then have been:

$$SECA = \eta \times Ia = 0.44 \times 597 \times 10^{6} \text{ BTU} = 263 \text{ million BTU}$$

Transport losses to storage of five percent, 13 million BTU, will be assumed based on the actual losses. Energy to storage (STEI) would be:

$$STEI = 263 \times 10^{6} \text{ BTU} - 13 \times 10^{6} \text{ BTU} = 250 \text{ million BTU}$$

Actual losses from storage were 45% of the input solar energy. Thus, usable solar energy from storage would be:

$$STEO = STEI \times 0.55 = 250 \times 10^{6} \text{ BTU} \times 0.55 = 138 \text{ million BTU}$$

The cooling load was 218 million BTU for the season, but there were 1,690 CDD versus the long-term average of 571 CDD. Estimating that the long-term cooling load would be approximately one-half of this season's cooling load, the long-term Cooling Load (CL) would have been 109 million BTU.

[1] See Appendix D.

[2] Refer to previous collector efficiency plots in collector performance discussion.

In order to provide 109 million BTU of cooling, the absorption chiller would require 218 million BTU of thermal energy. This is computed using an assumed chiller COP of 0.50 which the chiller was able to provide during September and October.

The solar fraction is the ratio of the solar energy supplied to the thermal requirements of the chiller, or:

$$\frac{138 \times 10^6 \text{ BTU}}{218 \times 10^6 \text{ BTU}} = 0.63 \text{ or } 63\%$$

The design cooling solar fraction at El Toro Library was 60%. Thus, it is possible for the solar cooling system to provide the design cooling solar fraction if the equipment performs properly and the weather conditions approximate long-term values.

Another measure of performance is the coefficient of performance (COP). The seasonal solar cooling coefficient of performance (COP) is shown in Figure 19. The solar cooling coefficient of performance is defined as the ratio of solar cooling to the solar-specific operating energy required to provide it. The higher the COP, the more solar energy delivered per unit of operating energy. Included in this operating energy is a portion of the energy collection operating energy. This portion is equal to the solar energy used for cooling divided by the total solar energy used.

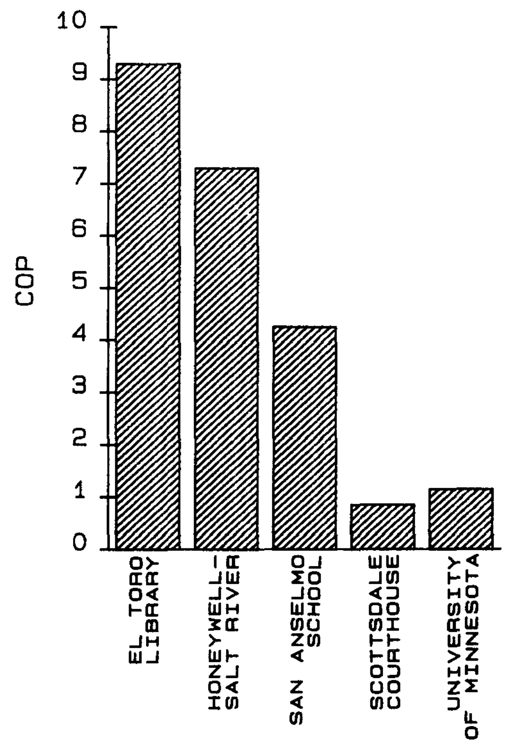

Figure 19. Seasonal Solar Cooling Coefficient of Performance (COP)

El Toro Library had the highest COP, 9.29. El Toro Library required very little solar-specific operating energy because the absorption chiller was used for both solar and auxiliary cooling. The collector pumps and the collector heat rejection unit constituted the only equipment charged to solar-specific operating energy.

Scottsdale Courthouse had the worst COP, 0.85. This indicates that the solar cooling system was operating at a net electrical expense. The Scottsdale Courthouse system has a separate auxiliary electric cooling system so the solar-specific operating energy was greater.

The University of Minnesota's COP was 1.15, also very poor. San Anselmo School had a COP of 4.25 and Honeywell-Salt River had a COP of 7.29.

Unless a system can operate with a COP better than approximately 2.00, the system is providing cooling less efficiently than a standard electric vapor compressor. Thus, only El Toro Library, Honeywell-Salt River, and San Anselmo School operated efficiently enough to outperform an electric chiller.

Another measure of performance is the solar cooling efficiency. This performance factor indicates how much of the incident solar energy appears as cooling. This performance factor is defined as the ratio of solar cooling minus solar-specific operating energy to the incident solar energy. The incident solar energy has been apportioned by multiplying the ratio of cooling solar energy to total solar energy used so that a site with other subsystems which use solar energy will not be penalized. These efficiencies can be seen in Figure 20.

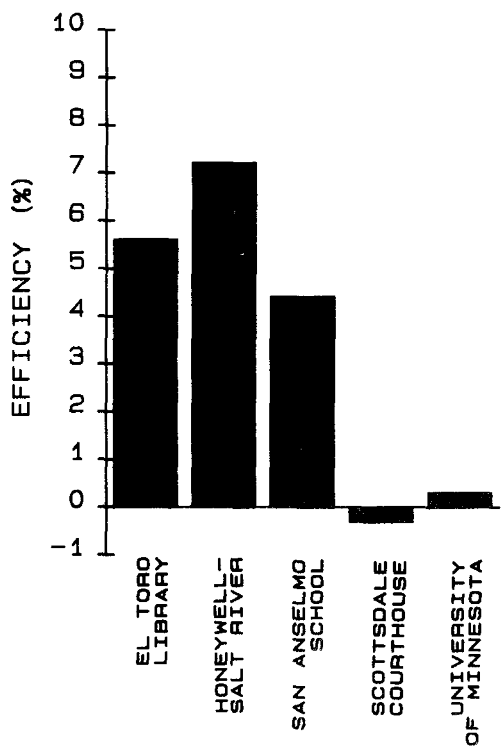

Figure 20. Solar Cooling Efficiency

The most efficient system was Honeywell-Salt River. This system was able to convert 7.2% of the incident solar energy into useful cooling. El Toro

Library and San Anselmo School also had good efficiencies, 5.6% and 4.4%, respectively. The worst performing site was Scottsdale Courthouse. This system had a negative solar cooling efficiency. This was a result of very large solar-specific operating energy in comparison to the small amount of solar cooling provided. The University of Minnesota was also a poor performer, as it had a 0.3% solar cooling efficiency.

G. PHOTOVOLTAIC COMPARISON

Although the NSDN solar cooling data base does not include any photovoltaic powered systems, a model for a photovoltaic powered system was examined for comparison (Reference 29). The specifications for the model are given below:

o Cell packing density in array	0.763
o Efficiency of unencapsulated cells	12.3%
o Losses due to using an inverter	15.0%
o COP of standard vapor comp. A/C	2.0
o Losses from cell glazing material	8.0%

These data are applied to a photovoltaic cooling system with an assumed solar insolation of 1,000 BTU (See Table 7). The output to the air conditioning unit is multiplied by two to arrive at a delivered output of 146.8 BTU/1,000 BTU of solar insolation input. This can be compared to the hypothetical absorption cooling efficiency of 95 BTU/1,000 BTU of solar insolation. Clearly, there is a substantial gain when photovoltaic systems are used for cooling.

Table 7. ESTIMATED PHOTOVOLTAIC EFFICIENCY

Incident Solar Insolation	1,000.00 BTU
Glazing Losses (8%)	80.00 BTU
Usable Insolation	920.00 BTU
Packing Density	0.76
Net Useful Insolation	702.00 BTU
Conversion Efficiency (@ 12.3%, Solar Cell Output)	86.35 BTU
Input to Inverter	86.35 BTU
Inverter Losses (@ 15% losses)	12.96 BTU
Output to Air Conditioning Unit	73.39 BTU
COP (@ 2.0)	146.78

Total cooling provided is about 15% of incident solar energy as compared to the Rankine and absorption chiller cooling systems in which the best systems in this report converted 7.2% and 5.6% of the incident solar energy into solar cooling.

H. COOLING SAVINGS

The savings per square foot of collector area provided by the solar cooling subsystems are listed in Table 8. These include electric savings, fossil fuel savings, and two forms of dollar savings. The first energy savings value is the savings that would be provided based on energy that would be required by the auxiliary cooling equipment in order to deliver the equivalent space cooling that was provided by the solar equipment less solar-specific operating energy. The second energy savings value is the savings that would be provided based on the energy that would be required to operate an electric vapor compressor chiller with a COP of 2.0 in order to deliver the equivalent space cooling that was provided by the solar equipment less solar-specific operating energy.

Table 8. COOLING ENERGY SAVINGS

SITE	SOLAR COOLING PROVIDED (BTU/FT^2-MONTH)	SOLAR OPERATING ENERGY (BTU/FT^2-MONTH)	ELECTRIC SAVINGS (BTU/FT^2-MONTH)	FOSSIL SAVINGS (BTU/FT^2-MONTH)	ENERGY SAVINGS SYSTEM-SPECIFIC AUXILIARY (CENTS/FT^2-MONTH)	ENERGY SAVINGS SIMILAR AUXILIARY (CENTS/FT^2-MONTH)
El Toro Library	3,231	348	-348	10,753	4.15	2.44
Honeywell-Salt River	4,888	670	1,051	N.A.	2.02	3.42
San Anselmo School	2,243	528	-528	6,235	1.77	1.14
Scottsdale Courthouse	802	951	-293	N.A.	-0.56	-1.06
University of Minnesota	1,006	878	-878	3,456	-1.23	-0.72

N.A. Denotes not applicable.

The energy savings are based on the following national long-term averages:

$4.57 per thousand cubic feet of natural gas
6.58 cents per kwh of electricity

One system, the University of Minnesota, used a coal boiler for an auxiliary heat source for the absorption chiller. The actual site coal cost was used for the savings calculation. This was:

$23.00 per ton of coal
(For fuel conversion factors, refer to Appendix J.)

The system which provided the best savings based on in situ auxiliary equipment was El Toro Library. This system provided a savings of 4.15 cents/ft^2-month. The savings that would have been provided if the system had used an auxiliary electric chiller would have been 2.44 cents/ft^2-month. These good savings can be attributed to relatively good operational collector and chiller efficiencies, 0.31 and 0.43, respectively, and the small amount of solar-specific operating energy consumed.

Honeywell-Salt River and San Anselmo School also provided good savings. Honeywell-Salt River saved 2.02 cents/ft^2-month and San Anselmo School saved 1.77 cents/ft^2-month. The savings for Honeywell-Salt River are calculated based on replacing cooling provided by auxiliary electric chillers with average COPs higher than 2.0. So the savings value is lower than that of El Toro Library even though the system provided more solar cooling per square foot of collector area. If a chiller with a 2.0 COP were used for auxiliary cooling, the Honeywell-Salt River savings would have been 3.42 cents/ft^2-month and the San Anselmo School savings would have been 1.14 cents/ft^2-month.

Both Scottsdale Courthouse and the University of Minnesota failed to provide a savings, but, instead, penalized the system with an electrical expense. Scottsdale Courthouse had an expense of 0.56 cents/ft^2-month and the University of Minnesota had an expense of 1.23 cents/ft^2-month. The University of Minnesota provided more solar cooling and expended less operating energy than Scottsdale Courthouse, but the cost of the auxiliary fuel, coal, was so little, $23/ton, that the operating expense overshadowed the fossil energy savings.

The second energy savings value shows that Scottsdale Courthouse would have lost 1.06 cents/ft^2-month and the University of Minnesota would have lost 0.72 cents/ft^2-month if these systems had used an electric chiller (2.0 COP) for auxiliary cooling. The Scottsdale Courthouse energy savings value decreased when compared to the first savings value because the actual auxiliary electric chiller had a COP of only 1.50 causing the actual savings to be computed higher.

None of the systems were able to demonstrate cost-effective solar cooling though three of the sites had energy savings. El Toro Library is an example of this, as this system provided the best savings per unit of collector area. An approximate pay-back period for the collector panels is calculated based on an approximate panel cost of $375 per panel.

$$\text{Pay-Back Period} = \frac{ft^2\text{-month}}{\$0.0415} \times \frac{1 \text{ year}}{12 \text{ months}} \times \frac{1 \text{ collector}}{17.4 \text{ ft}^2} \times \frac{\$375}{1 \text{ collector}} = 43 \text{ years}$$

This pay-back period includes only the collector panel cost which is only a fraction of the total cost of designing and constructing the solar cooling system. Thus, the actual pay-back period would be much higher than the 43 years. This is not a reasonable time period for pay-back as the life expectancy of the equipment is generally rated for 20 years. Thus, the system would not be cost-effective.

I. CONCLUSIONS

Solar cooling systems cannot be practically applied everywhere. One must carefully evaluate the length of the cooling season and the availability of solar energy during the cooling season. A good example of a poor application would be the University of Minnesota. Long-term weather data indicates only three months with 100 or more cooling degree-days: June, July, and August. During these three months, the average long-term horizontal insolation is 1,862 BTU/ft^2-day.

Examples of good applications are the sites in Arizona where there are seven months of 100 or more cooling degree-days and average horizontal insolation of 2,306 BTU/ft^2-day during this seven-month period.

Choosing a collector which will operate efficiently at high temperatures is critical in designing a solar cooling system. The flat-plate collectors by Lennox at Honeywell-Salt River provided the best performance and showed no signs of deterioration from the stress of high temperatures. The evacuated-tubes and concentrating collectors which are designed to operate more efficiently at higher temperatures did not perform as well as the flat-plates. Each of the concentrating arrays experienced focusing problems; this problem is designed out of the flat-plate or evacuated-tube systems. Since better performance can be attained with a good flat-plate collector, the flat-plate collectors seem a more cost-effective choice.

Proper proportioning of collector area to chiller operation is necessary for systems which deliver solar energy directly from collector to chiller in order to maintain adequate generator temperatures. The Rankine chiller system performed well using 164 square feet of collector area per ton of cooling capacity. A ratio for absorption chillers could not be determined from the limited data in this report but 200 to 250 square feet per ton of cooling capacity is regarded as an appropriate value.[1] The solar absorption systems in this report had much smaller ratios than this but used auxiliary boilers or storage to maintain adequate generator temperatures.

An obvious key component in any cooling system is the chiller. In a solar cooling system, the performance of this chiller is critical if the system is to provide a solar savings. For good performance or efficiencies, the temperatures and flows to the chiller must be carefully controlled. Large fluxuations from design values can cause the efficiencies of the chiller to drop significantly. Modern absorption chillers are capable of operating with generator temperatures of 160°F and lower but with reduced capacity. The absorption chillers analyzed generally operated between 170°F and 180°F. The Rankine engines are able to operate with solar water temperatures as low as 150°F. However, the average operating temperatures of the Rankine system also fell between 170°F and 180°F.

[1]Comparative Report: Performance of Active Solar Space Cooling Systems, 1980 Cooling Season, SOLAR/0023-81/40, Vitro Laboratories, Silver Spring, Maryland.

Storage tanks may be needed for systems which have limited collector areas. High performance from an absorption chiller requires steady-state operation.[1] To economize a collector area, a storage tank may be required in order to provide a steady energy flow to the chiller.

Another method of providing steady-state chiller operation, without large collector areas or storage tanks, is to place an auxiliary energy source in series with the collector and chiller. El Toro Library and the University of Minnesota both used a series boiler system. The El Toro Library design still required solar energy to be delivered to the chiller from storage. Both chillers had fair to good chiller efficiencies, El Toro Library was 0.43 and the University of Minnesota was 0.48, but the auxiliary boilers did not significantly improve chiller efficiency.

The systems which provided the best utilization of solar energy were those which were able to bypass the solar storage tank. Because of the extremely high temperatures needed for solar cooling, storage losses can be excessive. Systems which were designed to deliver solar energy to the chiller from storage showed very poor utilization of collected solar energy. El Toro Library and San Anselmo School, which must use storage, used 51% and 46%, respectively, of the collected solar energy, whereas Honeywell-Salt River and Scottsdale Courthouse were able to bypass storage and used 90% and 91%, respectively, of the collected solar energy.

Clearly, losses of energy throughout the systems were excessive. More concern must be given to better insulate and minimize pipe runs. If storage tanks are used, they must be well insulated and located to minimize transfer losses. The University of Minnesota uses a buried storage tank which is good during the heating season but results in higher losses during the cooling season.

Overall solar cooling efficiencies were higher this year than in previous years for sites analyzed in the NSDN. This year, there were three sites which had average solar cooling efficiencies greater than five percent. Honeywell-Salt River had a 7.2% efficiency. The highest system solar cooling efficiency last season was about four percent. This season's performance still falls below that of a photovoltaic cooling system which, if properly designed, could perform at an approximate 15% efficiency.

Cooling solar fractions for the solar cooling systems were generally low due to poor solar utilization, inefficient energy delivery devices, and higher than usual cooling loads. Cooling solar fractions were below design cooling solar fractions in all areas. Design values were not unreasonable if system components had operated properly and efficiently and losses had been minimized. Honeywell-Salt River performed closest to design, providing six percent of the cooling load versus the design value of 16%.

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[1]Guentin,J.M. and B.D. Wogd, Transient Effects on the Performance of a Residential Solar Absorption Chiller.

SECTION IV

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APPENDIX A

SITE DESCRIPTIONS

EL TORO LIBRARY

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, located at the building entrances. The library is functional year-round and is occupied Monday 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° west of due south at a tilt of 19 degrees from horizontal. The collector 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 will provide 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 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/Components	Manufacturer	Model No.
Evacuated tube collectors	General Electric	TC-100
Heat Rejector	Young Radiator Co.	22D20
Solar Storage Tank	Sante 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 Inc. Products	

The system, shown schematically in Appendix B, has nine modes of operation.

Mode 1 - Solar Energy Collection - Solar energy collection occurs when insolation levels are sufficient (as controlled 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 method, since all the flow bypasses the storage tank. Pumps P1 and 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. (Collector 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 the collector pump P1 or P2 when the ambient temperature falls below 38°F. All the collector flow will bypass storage and 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.

HONEYWELL-SALT RIVER

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 cooled conditioned space. The system contains an 8,200-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% ethelyne glycol/water solution.

Space cooling is provided by two 25-ton vapor compressors and a duel 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 in Appendix B, has the following operating modes:

Mode 1 - Collector-to-Storage - 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.

Mode 2 - Storage-to-Space Heating - When the storage fluid temperature is above 110°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 #1 is started, and when the fluid temperature reaches 170°F, Rankine engine #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 #2 is deactivated and, at 150°F, Rankine engine #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 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 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.

SAN ANSELMO SCHOOL

The San Anselmc 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 collector 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 collection is excessive, then solar energy is dissipated to the environment via a water-to-air heat rejector. When a sufficiently 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 in meeting 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-Perflex, 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 in Appendix B, 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 P8 is activated and valve V3 is opened to allow flow through the heat rejector and collector panels. Energy from the storage tank maintains the water in the collector loop at 38°F via modulating valve V2. 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 V4 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 P8 is turned on and all the flow bypasses the storage tank and returns to the collectors to complete the cycle. Pump P8 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 V2 and allows all water to flow through storage. When the temperature falls below 175°F, valve V2 reverses position and allows all water to bypass the storage tank. This assures a positive energy storage into the 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 P7 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 P7 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 V3. 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 V4 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 V4 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.

SCOTTSDALE COURTHOUSE

The Scottsdale County Courts Building is a 6,850-square-foot public office building located in Scottsdale, Arizona. The building is a single-story office building constructed of concrete block. The solar energy system is a retrofit for space cooling and heating, and is designed to provide 60% of the total cooling load and 100% of the space heating requirement. Tracking-type, concentrating collectors (2,723 square feet) are mounted at a shallow angle (0.5 degrees) from the horizontal in a field adjacent to the building.

The collector array is utilized to heat water which is used directly by load subsystems or stored in a 5,000-gallon, below-grade, storage tank. Hot water from the storage tank is used to provide space heating to the buiding through eight separate duct heating coils. The hot water also may be used to drive a 25.5-ton absorption chiller. Backup auxiliary heat for space heating is provided by a series of in-duct electric resistance strip heaters. A conventional reciprocating vapor-compression chiller provides auxiliary cooling capacity.

The system is designed to provide hot water at temperatures of up to 260°F at 12 psi. Excess thermal energy is rejected through a shell-tube heat exchanger to a swimming pool adjacent to the courthouse. The collectors are controlled by photocell-based tracking units, which also defocus the concentrating array when the collector outlet temperatures are above 260°F. A photocell system is also utilized to activate the collector subsystem, when a predetermined amount of total solar radiation is present. Freeze protection is by circulation from storage or draindown during electric failure. The system has several modes of operation.

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

Equipment/Components	Manufacturer	Model or Description
Collectors	Sun Power Systems, Inc.	Axial concentrating
Absorption Chiller	Arkla Solaire Arkla Industries, Inc.	WFB-300
Controls	Barber-Colman, Inc.	
Pumps	Thrush Products, Inc.	

The system, shown schematically in Appendix B, has seven modes of operation.

Mode 1 - Collector-to-Storage - This mode is entered when 75 BTU/ft^2-hr of solar radiation are available, storage temperature is less than 250°F, and no solar load subsystems are activated. Pump P1 is activated, valve V1 is opened and the collector tracking control units focus the collectors toward the sun. This mode continues until collector outlet temperatures reach 260°F, a demand subsystem activates, or solar radiation becomes unavailable. A demand subsystem may activate and thus switch the mode. The collectors may still operate during this mode switch.

Mode 2 - Collector-to-Cooling - This mode is entered when the collector subsystem is activated and a demand for mechanical refrigeration exists. The absorption chiller is energized, pump P2 is activated to draw a percentage of the collector hot water return flow to the heated side of the chiller. Pumps P3, P4, the air handling unit, and the chilled water recirculation pump are activated. Modulating valve V4 regulates the hot water flow rate through the absorption chiller to maintain a 45°F outlet temperature. Temperature control of the inlet stream to the chiller is achieved by modulation of valve V2, which limits the inlet temperature to 200°F. The cooling tower loop flow is modulated through the action of valve V5, to maintain an 85°F condensor temperature. This mode ceases when cooling demand is satisfied, or when the inlet temperature to the chiller falls below 160°F.

Mode 3 - Collector-to-Heating - This mode is activated when the collector subsystem is activated, no cooling demand exists, and there is a demand for heating in any of the building's control zones. The hot water circulation pump, P5, is activated whenever the outside ambient temperature is below 65°F, and collected energy is available. The air handling unit is also activated. Modulating valve V3 varies the proportion of hot water supply and return flows from the heating coils and thus varies the hot water temperature according to the outside temperature. The hot water is circulated to a series of eight hydronic heating coils mounted in the multizone air handling unit. Modulating valves at each of the eight coils vary the hot water flow according to one of eight zone thermostats, each of which controls a heating coil. If the hydronic coils do not meet the zone thermostatic control requirements, a limit switch at the individual control valves activates one or more of the resistance heaters located in the air handling system. Auxiliary heating thus may occur simultaneously with this mode. If the auxiliary heating only is active, this is considered to be mode 5A.

Mode 4 - Storage-to-Cooling - This mode is entered when temperature levels in the building require cooling, no collection is occurring and the storage temperature is greater than 180°F. The control sequences for the chiller operating in mode 4 are similar to mode 2, except that the hot water supply is from storage only, rather than directly from the collector field. This mode ceases when cooling demand is satisfied, or when the chiller inlet temperature falls below 160°F.

Mode 5A, 5B - Storage-to-Heating - Mode 5A occurs when a heating demand exists in one or more zones, collection is not occurring, and the storage temperature averages 100°F or more. Mode 5B occurs when storage is below 100°F, and auxiliary electric heating only occurs.

The control logic of this mode is similar to that of mode 3, except that the hot water is supplied directly from storage and not from the collection loop. Auxiliary electric heating also occurs simultaneously with this mode.

Mode 6 - Auxiliary Cooling - If there is no collected solar energy available either directly or from storage, and a cooling demand exists in any of the control zones, then the auxiliary cooling mode is entered. The 25-ton reciprocating chiller is activated, the chilled water circulation pump is activated, and the air handling unit is activated. Auxiliary cooling proceeds until cooling demand is satisfied, or solar cooling becomes available. Modes

2 and 4 are interlocked so that no solar cooling occurs when the reciprocating unit is operating, and vice versa.

<u>Modes 7A, 7B - Energy Rejection</u> - Mode 7A is entered when the pool temperature is below 85°F and excess temperature (above 250°F) is available from the collector subsystem. Mode 7B is similar except flow is from storage only. The pool pump and pump P6 are activated until energy is no longer available from collector or storage.

UNIVERSITY OF MINNESOTA

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 purpose of the unusual underground construction is conservation of energy. Natural light is admitted to the building through terraced south- and west-facing windows.

The solar energy system retrofitted to this building was designed to provide 60% of its heating needs and 40% of its cooling needs. Solar energy collection is accomplished using a concentrating slat-type 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 ten individual, movable slat 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. Sufficiently intense insolation signals focusing by the reflectors. This causes reflected sunlight to be concentrated on the receivers. 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 manufacturers of the major solar system equipment and components are listed below.

Equipment/Components	Manufacturer	Model or Description
Collectors	Suntec Systems, Inc.	Suntec concentrating slats
Chiller	Trane, Inc.	Model C2J-W-5 Absorption Chiller (147.5 tons)
Storage	Wheeler Tank Manufacturing Co.	8-foot-diameter x 21-foot-long 8,000-gallon, steel, insulated tank

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 energizes upon successful tracking and the reflectors are rotated to focus sunlight on the receivers.

The system, shown schematically in Appendix B, 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 energizes. 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.

APPENDIX B

SITE SCHEMATICS[1]

[1]Reader should refer to Appendix I for better designation and numbering sequence of the sensors.

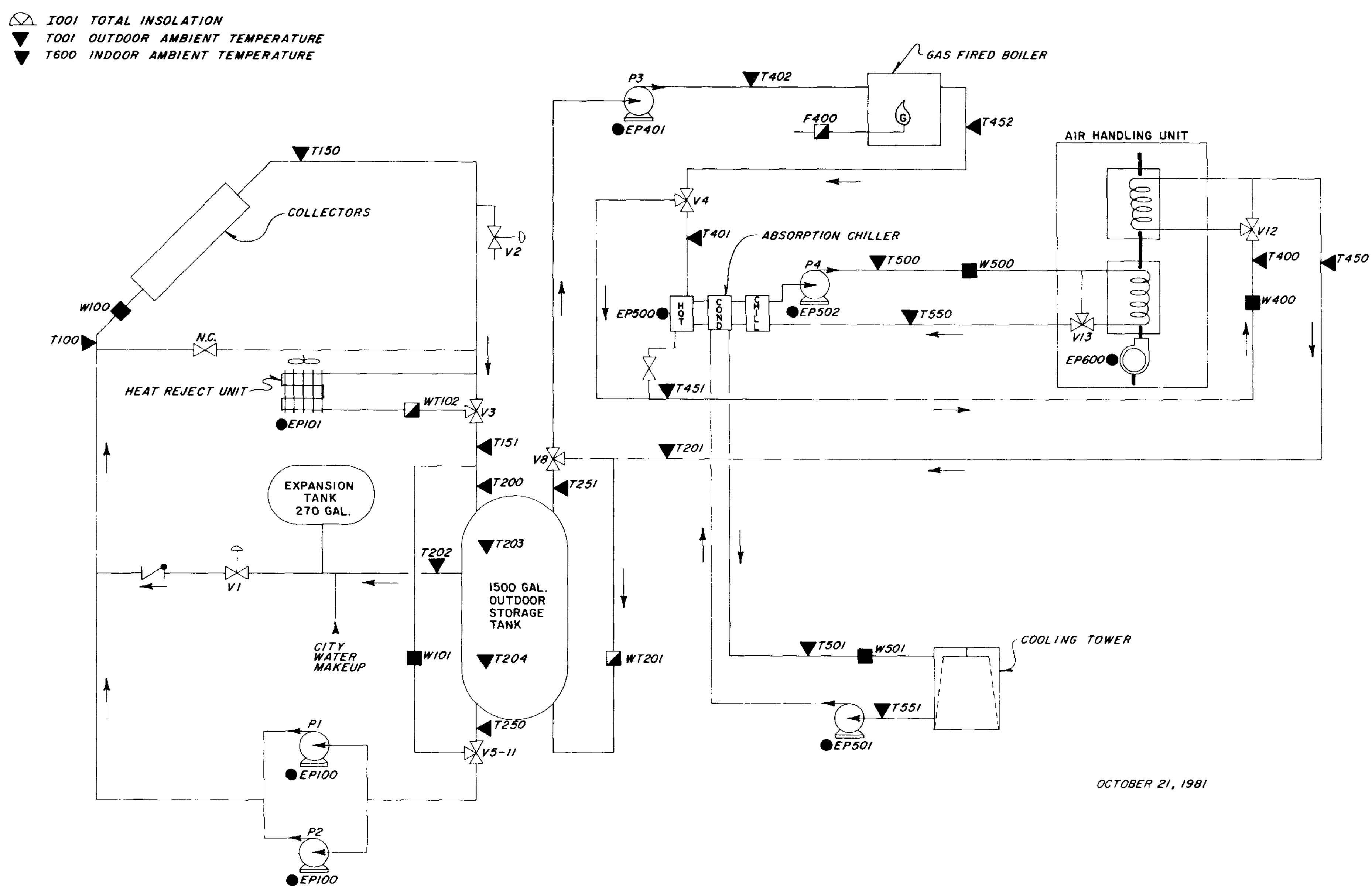


Figure B-1. El Toro Library Solar Energy System Schematic

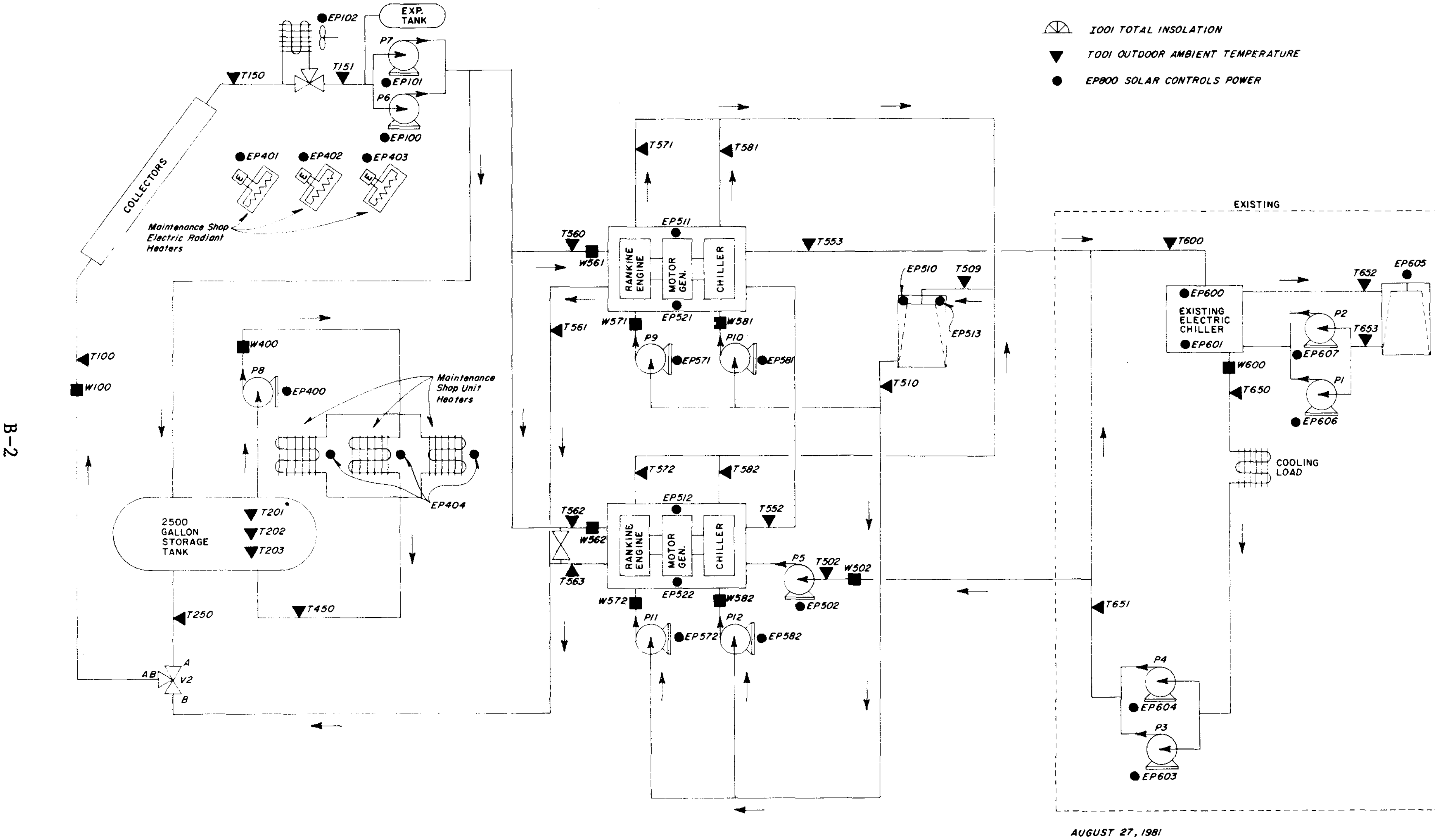


Figure B-2. Honeywell-Salt River Solar Energy System Schematic

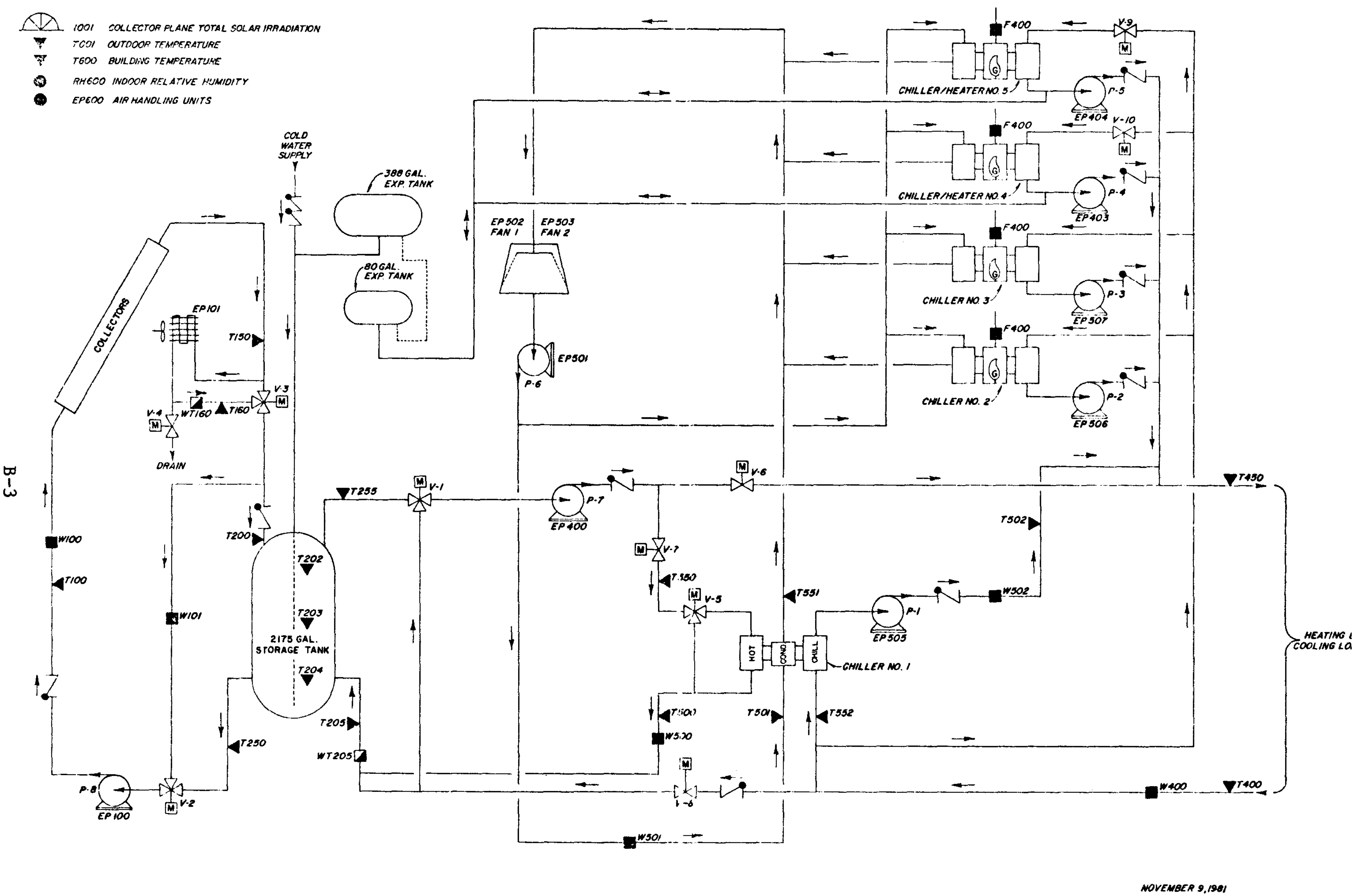


Figure B-3. San Anselmo School Solar Energy System Schematic

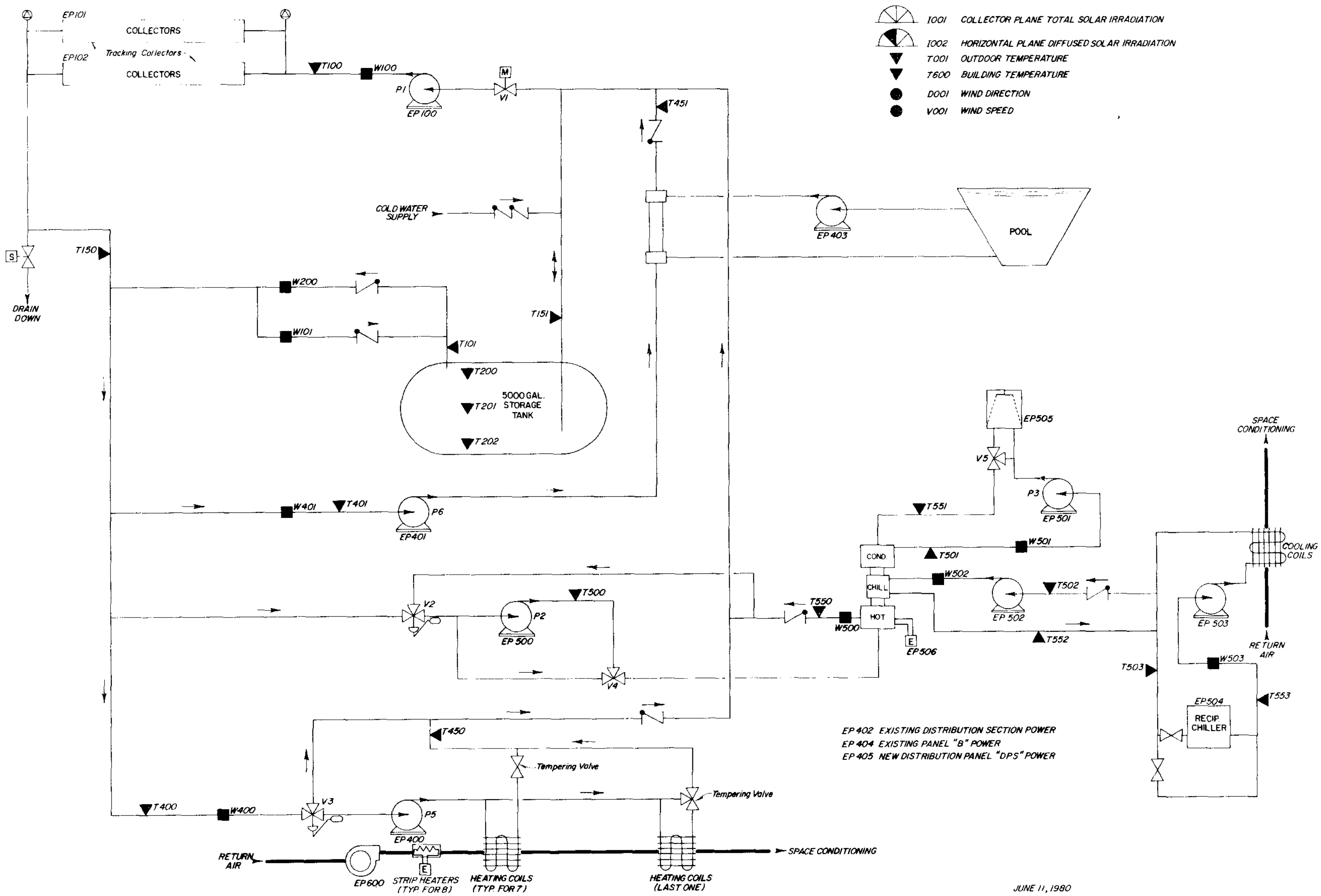


Figure B-4. Scottsdale Courthouse Solar Energy System Schematic

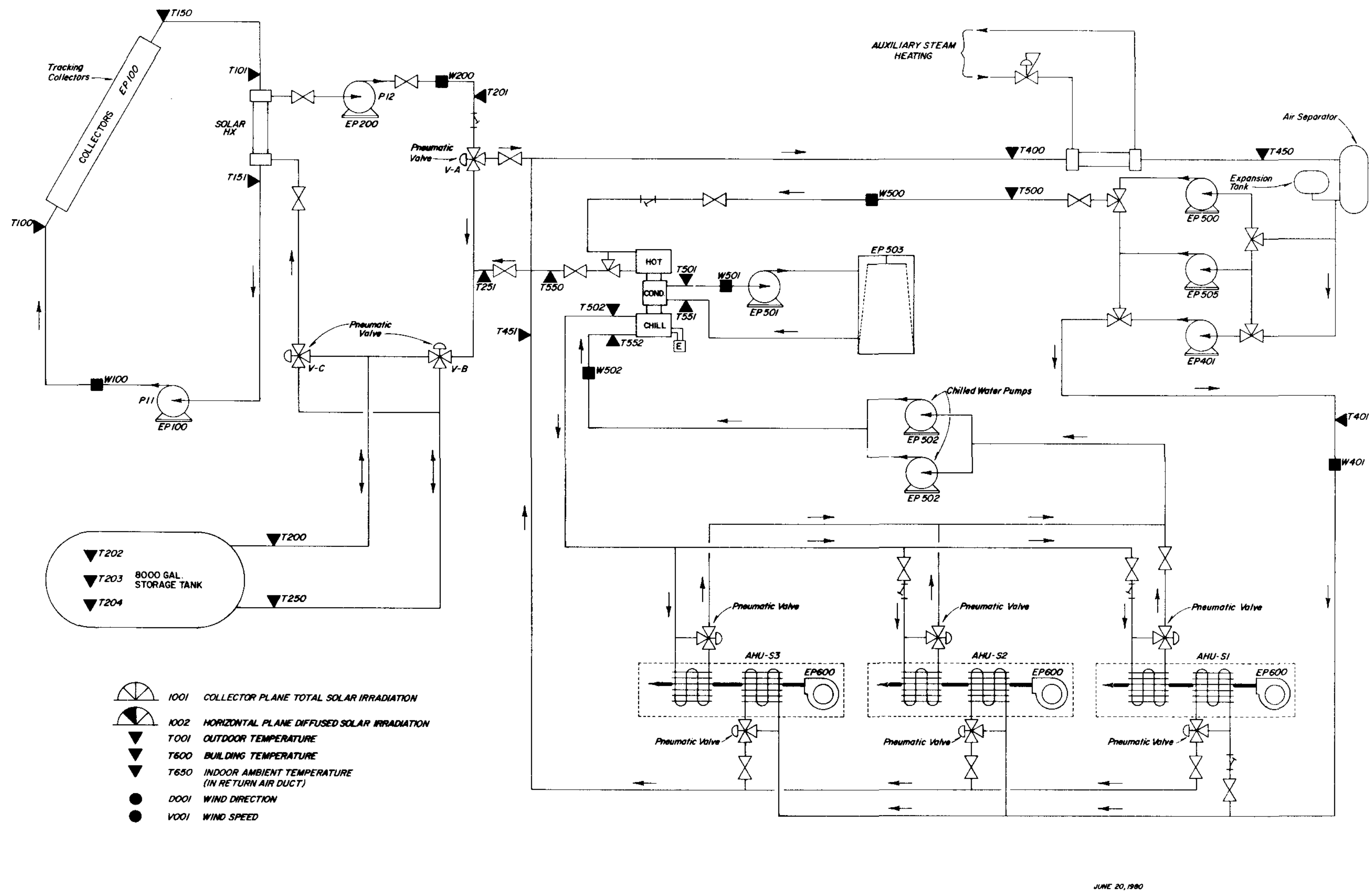


Figure B-5. University of Minnesota Solar Energy System Schematic

APPENDIX C

ENERGY FLOW DIAGRAMS

Energy flow diagrams are presented in Appendix C. These flow diagrams illustrate the pathway and magnitude of energy flows in each system, and thus serve to illustrate the overall performance of each subsystem. Subsystems are represented by rectangular blocks, into and out of which arrows are drawn. Within these arrows, the magnitude of energy flow is placed.

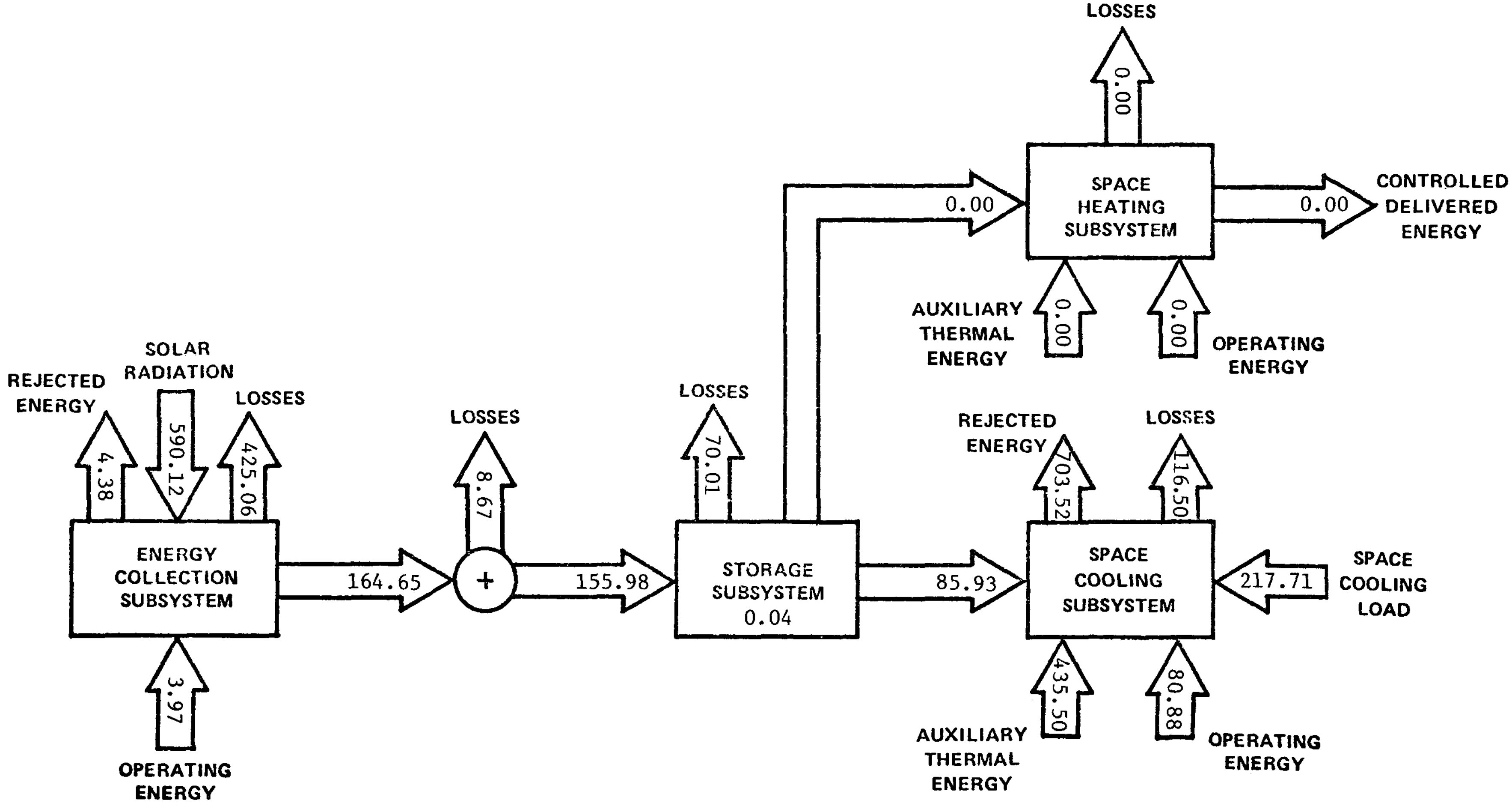


Figure C-1. Energy Flow Diagram for El Toro Library
March 1981 through June 1981,
August 1981 through November 1981
(Figures in million BTU)

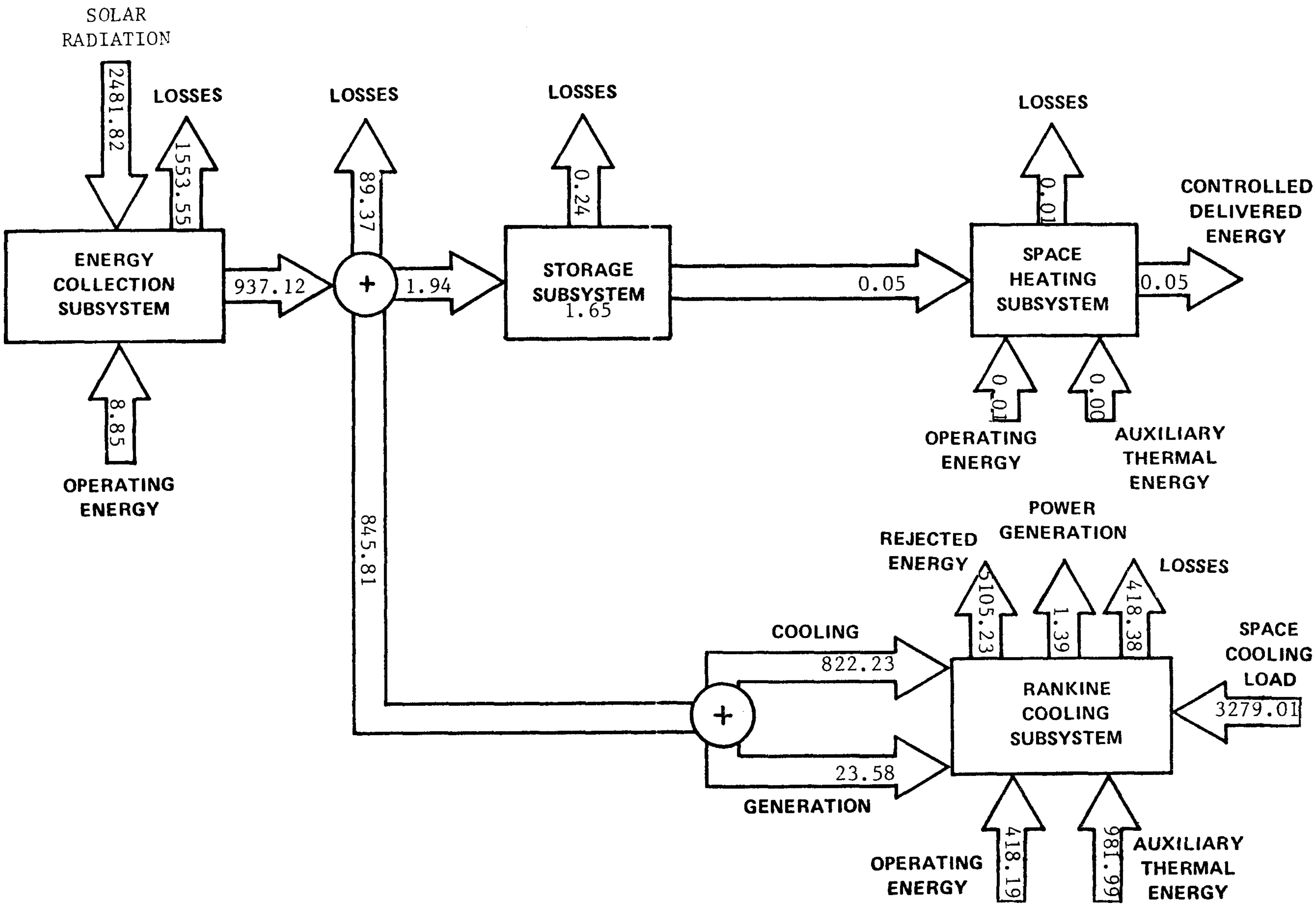
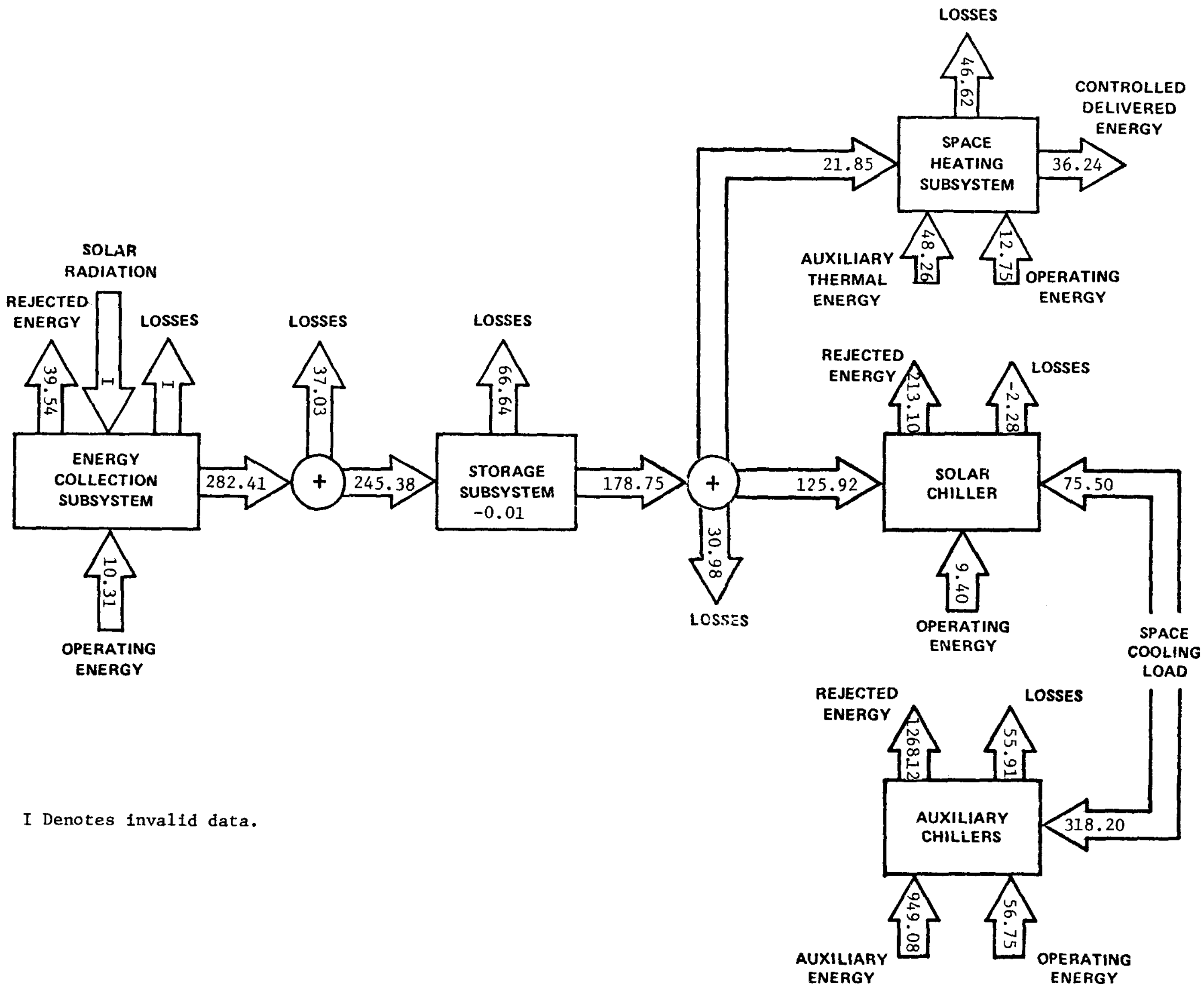


Figure C-2. Energy Flow Diagram for Honeywell-Salt River
June 1981 through October 1981
(Figures in million BTU)



I Denotes invalid data.

Figure C-3. Energy Flow Diagram for San Anselmo School
March 1981 through November 1981
(Figures in millin BTU)

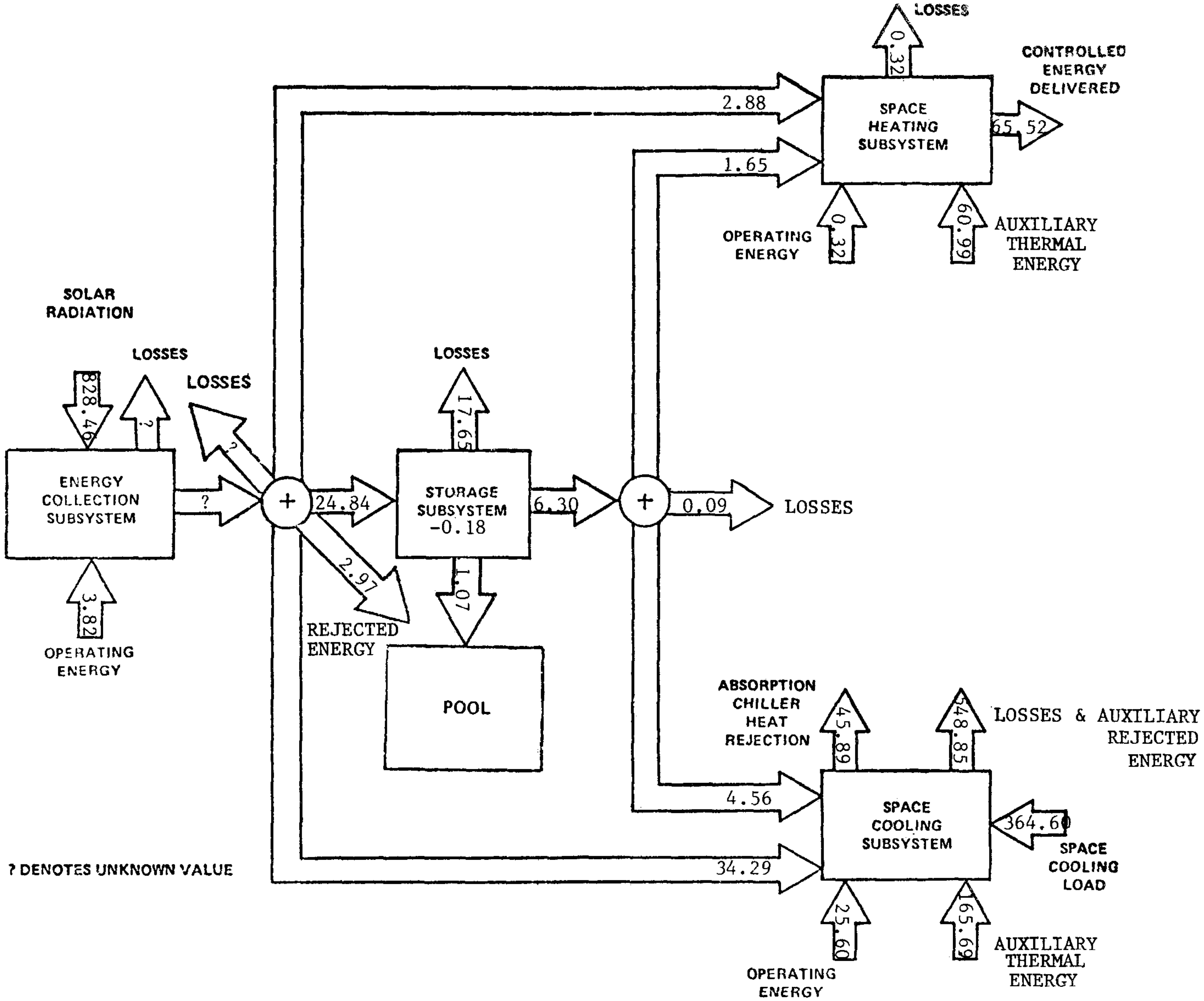


Figure C-4. Energy Flow Diagram for Scottsdale Courthouse
March, April, July, August, October and November 1981
(Figures in million BTU)

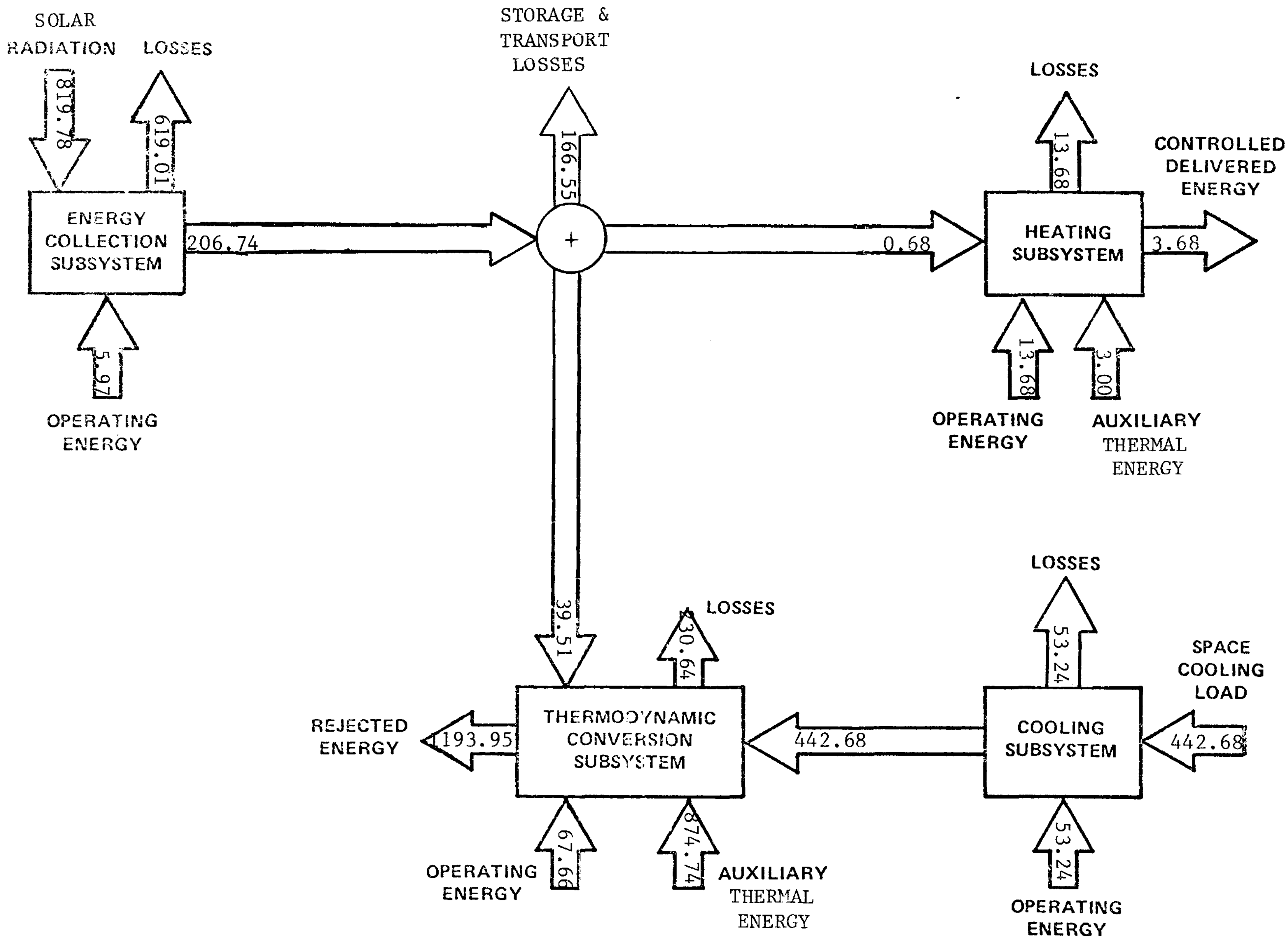


Figure C-5. Energy Flow Diagram for the University of Minnesota
July 1981 through September 1981
(Figures in million BTU)

APPENDIX D

SEASONAL PERFORMANCE DATA

Table D-1. SOLAR SYSTEM THERMAL PERFORMANCE

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981,
AUGUST 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED	SYSTEM LOAD	SOLAR ENERGY USED	AUXILIARY ENERGY FOSSIL	OPERATING ENERGY	ENERGY SAVINGS FOSSIL	ENERGY SAVINGS ELECTRICAL	SOLAR FRACTION (%)	SOLAR COEFFICIENT OF PERFORMANCE(1) (COP)
MAR	20.34	14.76	7.99 E	51.38	7.89	11.41	-0.48	20	16.65
APR	24.19	22.14	9.48 E	84.07	9.36	13.54	-0.54	13	17.56
MAY	22.83	26.64	8.14 E	96.29	10.03	11.63	-0.59	10	13.80
JUN	23.38	30.56	10.29 E	93.11	10.59	14.70	-0.59	13	17.44
AUG	20.58	34.91	14.84 E	91.88	12.42	21.20	-0.49	19	30.29
SEP	24.13	39.96	13.60	84.72	13.03	19.43	-0.50	20	27.20
OCT	20.24	28.18	13.94	68.39	11.85	19.92	-0.43	24	32.42
NOV	13.34	20.56	7.65	55.19	9.68	10.93	-0.35	16	21.86
TOTAL	169.03	217.71	85.93	625.03	84.85	122.76	-3.97	-	-
AVERAGE	21.13	27.21	10.74	78.13	10.61	15.35	-0.50	16	21.59

(1)Solar energy use/solar-specific operating energy.
E Denotes estimated value.

Table D-2. COLLECTOR SUBSYSTEM PERFORMANCE

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981,
AUGUST 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION	COLLECTED SOLAR ENERGY	COLLECTOR SUBSYSTEM EFFICIENCY (%)	OPERATIONAL INCIDENT ENERGY	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%)	ECSS OPERATING ENERGY	DAYTIME AMBIENT TEMPERATURE (°F)
MAR	67.31	20.34	30	63.29	32	0.48	71
APR	76.58	24.19	32	74.03	33	0.54	74
MAY	76.43	22.83	30	72.21	32	0.59	75
JUN	89.80	23.38	26	80.20	29	0.59	87
AUG	88.44	20.58	23	73.48	28	0.49	91
SEP	77.59	24.13	31	71.90	34	0.50	86
OCT	63.95	20.24	32	61.72	33	0.43	76
NOV	50.02	13.34	27	49.92	28	0.35	73
TOTAL	590.12	169.03	-	543.75	-	3.97	-
AVERAGE	73.77	21.13	29	67.97	31	0.50	79

Table D-3. STORAGE PERFORMANCE

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981,
AUGUST 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE	ENERGY FROM STORAGE	CHANGE IN STORED ENERGY	AVERAGE STORAGE TEMPERATURE (°F)
MAR	18.61	7.99 E	-0.04	167
APR	21.67	9.48 E	0.18	176
MAY	19.95	8.14 E	0.71	178
JUN	20.93	10.29 E	-0.63	177
AUG	20.86	14.84 E	-0.29	175
SEP	22.13	13.60	0.03	171
OCT	19.10	13.94	-0.03	159
NOV	12.73	7.65	0.11	158
TOTAL	155.98	85.93	0.04	-
AVERAGE	19.50	10.74	0.01	170

E Denotes estimated value.

Table D-4. SPACE COOLING SUBSYSTEM

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981
AUGUST 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD	SOLAR FRACTION OF LOAD (%)	SOLAR ENERGY USED	OPERATING ENERGY	AUX THERMAL USED	AUX FOSSIL FUEL	BUILDING TEMP (°F)	AMB TEMP (°F)	COOLING DEGREE-DAYS
MAR	14.76	21	7.99 E	7.41	30.91	51.88	72	63	28
APR	22.14	13	9.48 E	8.82	61.74	84.07	73	67	100
MAY	26.64	10	8.14 E	9.44	70.89	96.29	74	69	138
JUN	30.56	13	10.29 E	10.00	66.88	93.11	78	78	394
AUG	34.91	19	14.84 E	11.93	65.07	91.88	79	82	519
SEP	39.96	20	13.60	12.53	58.26	84.72	76	77	373
OCT	28.18	24	13.94	11.42	45.20	68.39	75	67	96
NOV	20.56	16	7.65	9.33	36.55	55.19	72	64	42
TOTAL	217.71	-	85.93	80.88	435.50	625.03	-	-	1,690
AVERAGE	27.21	16	10.74	10.11	54.44	78.13	75	71	211

E Denotes estimated value.

Table D-5. CHILLER PERFORMANCE

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981,
AUGUST 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	EQUIPMENT LOAD	THERMAL ENERGY INPUT	COEFFICIENT OF PERFORMANCE (COP)
MAR	14.76	32.57	0.45
APR	22.14	62.83	0.35
MAY	26.64	71.04	0.37
JUN	30.56	84.01	0.36
AUG	34.91	75.25	0.46
SEP	39.96	79.14	0.50
OCT	28.18	56.29	0.50
NOV	20.56	42.35	0.49
TOTAL	217.71	503.48	-
AVERAGE	27.21	62.94	0.43

Table D-6. ENERGY SAVINGS

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981,
AUGUST 1981 THROUGH NOVEMBER 1981

(All values in million BTU)

MONTH	SOLAR ENERGY USED	SPACE HEATING FOSSIL FUEL	SPACE HEATING FOSSIL FUEL	ECSS OPERATING ENERGY SOLAR-UNIQUE	NET ENERGY SAVINGS ELECTRICAL	NET ENERGY SAVINGS FOSSIL FUEL
MAR	7.99	0.00	11.41	0.48	-0.48	11.41
APR	9.48	0.00	13.54	0.54	-0.54	13.54
MAY	8.14	0.00	11.63	0.59	-0.59	11.63
JUN	10.29	0.00	14.70	0.59	-0.59	14.70
AUG	14.84	0.00	21.20	0.49	-0.49	21.20
SEP	13.60	0.00	19.43	0.50	-0.50	19.43
OCT	13.94	0.00	19.92	0.43	-0.43	19.92
NOV	7.65	0.00	10.93	0.35	-0.35	10.93
TOTAL	85.93	0.00	122.76	3.97	-3.97	122.76
AVERAGE	10.74	0.00	15.35	0.50	-0.50	15.35

Table D-7. WEATHER CONDITIONS

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981, AUGUST 1981 THROUGH NOVEMBER 1981

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY)		AMBIENT TEMPERATURE (°F)		HEATING DEGREE-DAYS		COOLING DEGREE-DAYS	
	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE
MAR	1,522	1,802	63	56	96	279	28	0
APR	1,789	1,993	67	59	45	177	100	9
MAY	1,728	2,024	69	63	2	94	138	29
JUN	2,098	2,090	78	66	0	38	394	77
AUG	1,999	2,178	82	72	0	0	519	209
SEP	1,812	1,881	77	70	0	9	373	165
OCT	1,446	1,602	67	65	33	64	96	70
NOV	1,168	1,316	64	59	74	195	42	12
TOTAL	-	-	-	-	250	856	1,690	571
AVERAGE	1,695	1,861	71	64	31	107	211	71

Table D-8. SOLAR SYSTEM THERMAL PERFORMANCE

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED	SYSTEM LOAD	SOLAR ENERGY USED	AUXILIARY ENERGY ELECTRICAL	OPERATING ENERGY	ENERGY SAVINGS ELECTRICAL	SOLAR FRACTION (%)	SOLAR COEFFICIENT OF PERFORMANCE[1] (COP)
JUN	242.96	700.34	201.85	213.66	78.40	10.85	9	32.19
JUL	218.65	850.31	202.16	319.58	101.62	10.27	5	35.40
AUG	205.55	773.88	184.56	283.79	101.57	9.81	5	25.14
SEP	151.61	625.46	141.79	228.40	89.01	7.36	5	27.59
OCT	118.35	329.08	115.50	109.84	56.52	5.34	8	28.17
TOTAL	937.12	3,279.07	845.86	1,155.27	427.12	43.63	-	-
AVERAGE	187.42	655.81	169.17	231.05	85.42	8.73	6	29.62

(1)Solar energy used/solar-specific operating energy.

Table D-9. COLLECTOR SUBSYSTEM PERFORMANCE

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION	COLLECTED SOLAR ENERGY	COLLECTOR SUBSYSTEM EFFICIENCY (%)	OPERATIONAL INCIDENT ENERGY	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%)	ECSS OPERATING ENERGY
JUN	546.15	242.96	45	476.68	51	2.17
JUL	523.90	218.65	42	454.95	48	1.49
AUG	521.04	205.55	40	457.92	45	2.14
SEP	467.41	151.61	32	399.26	38	1.73
OCT	423.32	118.35	28	332.13	36	1.32
TOTAL	2,481.82	937.12	-	2,120.94	-	8.85
AVERAGE	496.36	187.42	38	424.19	44	1.77

Table D-10. STORAGE PERFORMANCE

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE	ENERGY FROM STORAGE	CHANGE IN STORED ENERGY	AVERAGE STORAGE TEMPERATURE (°F)
JUN	0.00	0.00	0.31	124
JUL	0.00	0.00	-0.20	101
AUG	0.00	0.00	0.31	95
SEP	0.00	0.00	-0.28	99
OCT	1.94	0.05	1.51	92
TOTAL	1.95	0.05	1.65	-
AVERAGE	0.39	0.01	0.33	102

Table D-11. SPACE HEATING SUBSYSTEM

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(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 ELECT FUEL	TOTAL OPERATING ENERGY	AMB TEMP (°F)	HEATING DEGREE-DAYS
JUN	0.00	0.00	100	0.00	0.00	0.00	92	0
JUL	0.00	0.00	100	0.00	0.00	0.00	93	0
AUG	0.00	0.00	100	0.00	0.00	0.00	94	0
SEP	0.00	0.00	100	0.00	0.00	0.00	87	0
OCT	0.05	0.05	100	0.00	0.00	0.01	73	5
TOTAL	0.05	0.05	-	0.00	0.00	0.01	-	5
AVERAGE	0.01	0.01	100	0.00	0.00	0.00	88	1

Table D-12. SPACE COOLING SUBSYSTEM

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD	SOLAR FRACTION OF LOAD (%)	SOLAR ENERGY USED	OPERATING ENERGY	AUX THERMAL USED	AUX ELECT FUEL	AMB TEMP (°F)	COOLING DEGREE-DAYS
JUN	700.34	9	198.87	76.06	181.62	213.66	92	810
JUL	850.31	5	195.34	99.94	271.64	319.58	93	868
AUG	773.88	5	184.45	99.24	241.23	283.79	94	899
SEP	625.46	5	136.66	87.10	194.14	228.40	87	657
OCT	329.03	8	106.91	55.02	93.36	109.84	73	245
TOTAL	3,279.02	-	822.23	417.36	981.99	1,155.27	-	3,479
AVERAGE	655.80	6	164.45	83.47	196.40	231.05	88	696

Table D-13. CHILLER PERFORMANCE

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU, unless otherwise indicated)

	AUXILIARY CHILLER		
MONTH	EQUIPMENT LOAD	THERMAL ENERGY INPUT	COEFFICIENT OF PERFORMANCE (COP)
JUN	478.40	141.08	2.9
JUL	612.28	214.24	2.4
AUG	595.34	185.99	2.7
SEP	394.51	136.44	2.5
OCT	147.15	49.95	2.5
TOTAL	2,227.68	727.70	-
AVERAGE	445.54	145.54	2.6

Table D-14. RANKINE 1 PERFORMANCE

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY TO RANKINE	POWER OUT OF RANKINE	RANKINE EFFICIENCY (%)	ELECTRIC POWER GENERATED	RANKINE POWER TO COMPRESSOR	AUXILIARY POWER TO COMPRESSOR	COOLING PRODUCED	THERMAL COP OF VAPOR COMPRESSOR
JUN	93.36	6.59	7.1	0.15	6.40	18.32	120.38	4.9
JUL	97.36	6.50	6.7	0.22	6.23	27.66	139.17	4.1
AUG	71.60	4.90	6.8	0.01	4.89	25.81	123.49	4.0
SEP	63.85	4.67	7.3	0.16	4.47	27.89	131.77	4.1
OCT	51.52	3.41	6.6	0.01	3.40	22.79	106.33	4.1
TOTAL	377.69	26.07	-	0.55	25.39	122.47	621.14	-
AVERAGE	75.54	5.21	6.9	0.11	5.08	24.49	124.23	4.2

Table D-15. RANKINE 2 PERFORMANCE

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY TO RANKINE	POWER OUT OF RANKINE	RANKINE EFFICIENCY (%)	ELECTRIC POWER GENERATED	RANKINE POWER TO COMPRESSOR	AUXILIARY POWER TO COMPRESSOR	COOLING PRODUCED	THERMAL COP OF VAPOR COMPRESSOR
JUN	108.35	8.04	7.4	0.03	8.00	22.21	106.78	3.5
JUL	104.80	7.25	6.9	0.01	7.24	29.75	100.23	2.7
AUG	112.96	8.42	7.5	0.00	8.42	29.42	85.62	2.3
SEP	77.94	6.13	7.9	0.27	5.80	29.81	107.57	3.0
OCT	63.94	4.78	7.5	0.53	4.1	20.62	85.00	3.4
TOTAL	467.99	34.62	-	0.84	33.57	131.81	485.20	-
AVERAGE	93.60	6.92	7.4	0.17	6.71	26.36	97.04	2.9

Table D-16. SOLAR-SPECIFIC OPERATING ENERGY

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU)

MONTH	ECSS OPERATING ENERGY	POWER GENERATION OPERATING ENERGY	SHS OPERATING ENERGY	SCS OPERATING ENERGY	TOTAL SOLAR OPERATING ENERGY
JUN	2.17	0.07	0.00	4.03	6.27
JUL	1.49	0.21	0.00	4.01	5.71
AUG	2.14	0.00	0.00	5.20	7.34
SEP	1.73	0.30	0.00	3.11	5.14
OCT	1.32	0.25	0.01	2.52	4.10
TOTAL	8.85	0.83	0.01	18.87	28.56
AVERAGE	1.77	0.17	0.00	3.77	5.71

Table D-17. ENERGY SAVINGS

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in million BTU)

MONTH	SOLAR ENERGY USED	SPACE HEATING ELECTRICAL	POWER GENERATION ELECTRICAL	SPACE COOLING ELECTRICAL	ECSS OPERATING ENERGY EXPENSE SOLAR-UNIQUE	NET ENERGY SAVINGS ELECTRICAL
JUN	201.85	0.00	0.11	12.91	-2.17	10.85
JUL	202.16	0.00	0.20	11.56	-1.49	10.27
AUG	184.56	0.00	0.01	11.94	-2.14	9.81
SEP	141.79	0.00	0.12	8.97	-1.73	7.36
OCT	115.50	0.04	0.29	6.33	-1.32	5.34
TOTAL	845.86	0.04	0.73	51.71	-8.85	43.63
AVERAGE	169.17	0.01	0.15	10.34	1.77	8.73

Table D-18. WEATHER CONDITIONS

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY)		AMBIENT TEMPERATURE (°F)		HEATING DEGREE-DAYS		COOLING DEGREE-DAYS	
	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE
JUN	2,218	2,592	92	85	0	0	810	588
JUL	2,059	2,388	93	91	0	0	868	812
AUG	2,048	2,314	94	89	0	0	899	747
SEP	1,898	2,203	87	84	0	0	657	564
OCT	1,664	1,893	73	72	5	17	245	240
TOTAL	-	-	-	-	5	17	3,479	2,951
AVERAGE	1,977	2,278	88	84	1	3	696	590

Table D-19. SOLAR SYSTEM THERMAL PERFORMANCE

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED	SYSTEM LOAD	SOLAR ENERGY USED	AUXILIARY ENERGY FOSSIL	OPERATING ENERGY	ENERGY SAVINGS FOSSIL	ENERGY SAVINGS ELECTRICAL	SOLAR FRACTION (%)	SOLAR COEFFICIENT OF PERFORMANCE(1) (COP)
MAR	32.69	27.65	13.20	104.67	16.63	22.00	-1.61	16	0.79
APR	47.63	34.12	22.68	172.34	24.15	37.80	-2.87	32	7.90
MAY	41.26	41.58	14.16	200.61	27.19	23.60	-1.91	15	7.41
JUN	44.56	69.56	24.43	239.36	27.44	40.72	-2.53	20	9.66
JUL	38.47	67.37	21.75	226.37	26.58	36.25	-2.08	20	10.46
AUG	35.29	56.31	17.10	203.20	27.74	28.49	-1.85	19	9.24
SEP	32.05	61.07	11.81	237.73	30.05	19.69	-2.18	7	5.42
OCT	33.01	45.82	17.91	210.68	21.23	29.84	-2.34	12	7.65
NOV	16.99	26.45	4.73	83.92	8.27	7.88	-2.19	5	2.16
TOTAL	321.95	429.93	147.77	1,678.88	209.28	246.27	-19.56	-	-
AVERAGE	35.77	47.77	16.42	186.54	23.25	27.36	-2.17	23	7.55

(1)Solar energy used/solar-specific operating energy.

Table D-20. COLLECTOR SUBSYSTEM PERFORMANCE

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION	COLLECTED SOLAR ENERGY	COLLECTOR SUBSYSTEM EFFICIENCY (%)	OPERATIONAL INCIDENT ENERGY	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%)	ECSS OPERATING ENERGY	DAYTIME AMBIENT TEMPERATURE (°F)
MAR	150.69	32.69	22	147.66	22	1.27	62
APR	209.64	47.63	23	208.61	23	1.36	71
MAY	198.47	41.26	21	197.01	21	1.15	74
JUN	204.88	44.56	22	177.54	25	0.74	84
JUL	209.93	38.47	18	158.45	24	0.55	85
AUG	212.22	35.29	17	151.84	23	0.49	83
SEP	197.25	32.05	16	155.25	21	1.47	79
OCT	174.72	33.01	19	141.83	23	1.34	71
NOV	I	16.99	I	I	I	1.94	64
TOTAL	*	321.95	*	*	*	10.31	-
AVERAGE	*	35.77	*	*	*	1.15	75

I Denotes invalid data.
* Denotes unavailable data.

Table D-21. STORAGE PERFORMANCE

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE	ENERGY FROM STORAGE	CHANGE IN STORED ENERGY	AVERAGE STORAGE TEMPERATURE (°F)
MAR	23.86	13.92	-0.28	141
APR	35.70	25.92	0.48	166
MAY	25.97	19.77	0.64	171
JUN	35.64	29.67	-0.64	169
JUL	31.68	25.52	0.01	168
AUG	29.91	20.89	0.66	168
SEP	19.81	14.42	-0.67	161
OCT	29.54	21.53	0.46	163
NOV	13.27	6.11	-0.67	165
TOTAL	245.38	178.75	-0.01	-
AVERAGE	27.26	19.86	0.00	164

Table D-22. SPACE HEATING SUBSYSTEM

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(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 FOSSIL FUEL	TOTAL OPERATING ENERGY	BLDG TEMP (°F)	AMB TEMP (°F)	HEATING DEGREE-DAYS
MAR	15.85	9.77	35	18.51	30.86	7.78	71	56	295
APR	1.64	6.68	85	1.37	2.29	1.99	73	60	197
MAY	0.43	4.12	100	0.00	0.00	0.01	76	64	81
JUN	0.75	1.28	100	0.00	0.05	0.00	79	73	5
JUL	0.00	0.00	0	0.00	0.00	0.00	77	72	0
AUG	0.00	0.00	0	0.00	0.00	0.00	76	72	0
SEP	0.57	0.00	0	1.66	2.77	0.83	76	70	4
OCT	3.95	0.00	0	9.35	15.59	0.95	73	62	112
NOV	13.05	0.00	0	19.03	31.71	1.19	71	58	215
TOTAL	36.24	21.85	-	48.26	83.27	12.75	-	-	909
AVERAGE	4.03	2.43	38	5.36	9.25	1.42	75	65	101

Table D-23. SPACE COOLING SUBSYSTEM

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD	SOLAR FRACTION OF LOAD (%)	SOLAR ENERGY USED	OPERATING ENERGY	AUX THERMAL USED	AUX FOSSIL FUEL	BUILDING TEMP (°F)	AMB TEMP (°F)	COOLING DEGREE-DAYS
MAR	11.80	15	3.43	7.59	35.99	59.99	71	56	0
APR	32.48	29	16.00	20.80	102.03	170.05	73	60	34
MAY	41.15	15	10.04	26.03	120.37	200.61	76	64	41
JUN	68.81	20	23.15	26.68	143.59	239.31	79	73	236
JUL	67.37	20	21.75	26.03	135.82	226.37	77	72	230
AUG	56.31	19	17.10	27.25	121.92	203.20	76	72	212
SEP	60.50	12	11.81	27.75	140.98	234.96	76	70	146
OCT	41.87	25	17.91	18.94	117.06	195.09	73	62	14
NOV	13.40	17	4.73	5.14	31.32	52.21	71	59	1
TOTAL	393.69	-	125.92	186.21	949.08	1,581.79	-	-	914
AVERAGE	43.74	19	13.99	20.69	105.45	175.75	75	65	102

Table D-24. CHILLER PERFORMANCE

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR			AUXILIARY		
	EQUIPMENT LOAD	THERMAL ENERGY INPUT	COEFFICIENT OF PERFORMANCE (COP)	EQUIPMENT LOAD	THERMAL ENERGY INPUT	COEFFICIENT OF PERFORMANCE (COP)
MAR	1.84	3.43	0.53	9.96	35.99	0.27
APR	9.65	16.00	0.57	22.84	102.03	0.24
MAY	5.98	10.04	0.60	35.17	120.37	0.30
JUN	14.49	23.15	0.53	54.32	143.59	0.32
JUL	13.23	21.75	0.61	54.14	135.82	0.40
AUG	10.48	17.10	0.61	45.83	121.92	0.38
SEP	7.14 E	11.81	0.60	53.36	140.98	0.38
OCT	10.46 E	17.91	0.58	31.41	117.06	0.27
NOV	2.23 E	4.73	0.47	11.17	31.32	0.36
TOTAL	75.50	125.92	-	318.20	949.08	-
AVERAGE	8.39	13.99	0.60	35.36	105.45	0.34

E Denotes estimated value.

Table D-25. SOLAR-SPECIFIC OPERATING ENERGY

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in million BTU)

MONTH	ECSS OPERATING ENERGY	SHS OPERATING ENERGY	SCS OPERATING ENERGY	TOTAL SOLAR OPERATING ENERGY
MAR	1.27	0.05	0.29	1.61
APR	1.36	0.03	1.48	2.87
MAY	1.15	0.01	0.75	1.91
JUN	0.74	0.00	1.79	2.53
JUL	0.55	0.00	1.53	2.08
AUG	0.49	0.00	1.36	1.85
SEP	1.47	0.00	0.71	2.18
OCT	1.34	0.00	1.00	2.34
NOV	1.94	0.00	0.25	2.19
TOTAL	10.31	0.09	9.16	19.56
AVERAGE	1.15	0.01	1.02	2.17

Table D-26. ENERGY SAVINGS

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in million BTU)

MONTH	SOLAR ENERGY USED	SPACE HEATING		SPACE COOLING		ECSS OPERATING ENERGY SOLAR-UNIQUE	NET ENERGY SAVINGS	
		ELECTRICAL	FOSSIL FUEL	ELECTRICAL	FOSSIL FUEL		ELECTRICAL	FOSSIL FUEL
MAR	13.20	-0.05	16.28	-0.29	5.72	-1.27	-1.61	22.00
APR	22.68	-0.03	11.13	-1.48	26.67	-1.36	-2.87	37.80
MAY	14.16	-0.01	6.86	-0.75	16.74	-1.15	-1.91	23.60
JUN	24.43	0.00	2.13	-1.79	38.59	-0.74	-2.53	40.72
JUL	21.75	0.00	0.00	-1.53	36.25	-0.55	-2.08	36.25
AUG	17.10	0.00	0.00	-1.36	28.49	-0.49	-1.85	28.49
SEP	11.81	0.00	0.00	-0.71	19.69	-1.47	-2.18	19.69
OCT	17.91	0.00	0.00	-1.00	29.84	-1.34	-2.34	29.84
NOV	4.73	0.00	0.00	-0.25	7.88	-1.94	-2.19	7.88
TOTAL	147.77	-0.09	36.40	-9.16	209.87	-10.31	-19.56	246.27
AVERAGE	16.42	-0.01	4.04	-1.02	23.32	-1.15	-2.17	27.36

Table D-27. WEATHER CONDITIONS

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY)		AMBIENT TEMPERATURE (°F)		HEATING DEGREE-DAYS		COOLING DEGREE-DAYS	
	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE
MAR	1,300	1,768	56	55	295	322	0	0
APR	1,868	1,944	60	58	197	228	34	12
MAY	1,712	1,952	64	62	81	123	41	20
JUN	1,826	1,947	73	66	5	50	236	71
JUL	1,811	1,978	72	68	0	12	230	117
AUG	1,830	1,958	72	68	0	15	212	111
SEP	1,758	1,929	70	68	4	13	146	94
OCT	1,507	1,671	62	63	112	90	14	19
NOV	I	1,332	58	56	215	276	1	0
TOTAL	-	-	-	-	909	1,129	914	444
AVERAGE	*	1,831	65	63	101	125	102	49

I Denotes invalid data.
* Denotes unavailable data.

Table D-28. SOLAR SYSTEM THERMAL PERFORMANCE

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED	SYSTEM LOAD	SOLAR ENERGY USED	AUXILIARY ENERGY ELECTRICAL	OPERATING ENERGY	ENERGY SAVINGS ELECTRICAL	SOLAR FRACTION (%)	SOLAR COEFFICIENT OF PERFORMANCE[1] (COP)
MAR	10.03	42.04	5.65	27.59	3.43	3.61	11	7.74
APR	I	41.29	7.09	41.59	5.04	-0.05	6	2.58
JUL	19.47	103.17	18.71	67.92	4.93	3.25	6	3.80
AUG	I	96.77	5.85	64.50	7.93	-4.14	2	0.74
OCT	5.93	77.07	3.93	49.20	5.96	-3.20	2	0.66
NOV	-2.05	69.78	2.15	46.88	2.60	-0.54	1	0.83
TOTAL	*	430.12	43.38	297.68	29.89	-1.07	-	-
AVERAGE	*	71.69	7.23	49.61	4.98	-0.18	4	1.75

(1)Solar energy used/solar-specific operating energy.
I Denotes invalid data.
* Denotes unavailable data.

Table D-29. COLLECTOR SUBSYSTEM PERFORMANCE

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION	COLLECTED SOLAR ENERGY	COLLECTOR SUBSYSTEM EFFICIENCY (%)	OPERATIONAL INCIDENT ENERGY	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%)	ECSS OPERATING ENERGY	DAYTIME AMBIENT TEMPERATURE (°F)
MAR	127.84	10.03	8	97.28	10	0.60	71
APR	164.86	I	I	157.32	I	1.01	73
JUL	178.98	19.48	11	132.86	15	0.72	103
AUG	165.97	I	I	I	I	0.50	107
OCT	111.62	5.93	5	94.29	6	0.60	82
NOV	79.19	-2.05	-3	71.64	-3	0.39	76
TOTAL	828.46	*	*	*	*	3.82	-
AVERAGE	138.08	*	*	*	*	0.64	85

I Denotes invalid data.
* Denotes unavailable data.

Table D-30. STORAGE PERFORMANCE

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	ENERGY TO STORAGE	ENERGY FROM STORAGE	CHANGE IN STORED ENERGY	AVERAGE STORAGE TEMPERATURE (°F)
MAR	6.07 E	1.67 E	4.00	130
APR	7.58	2.99	-0.26	210
JUL	4.77	0.24	0.72	168
AUG	3.81	0.06	-1.46	203
OCT	2.55	2.32	-0.96	182
NOV	0.06	0.09	-1.86	156
TOTAL	24.84	7.37	0.18	-
AVERAGE	4.14	1.23	0.03	175

E Denotes estimated value.

Table D-31. SPACE HEATING SUBSYSTEM

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(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 ELECT FUEL	TOTAL OPERATING ENERGY	BLDG TEMP (°F)	AMB TEMP (°F)	HEATING DEGREE-DAYS
MAR	20.34	4.39	21	15.95	15.95	0.13	74	63	103
APR	5.38	0.00	0	5.38	5.38	0.00	79	74	15
JUL	0.76	0.00	0	0.76	0.76	0.00	85	94	0
AUG	1.14	0.00	0	1.14	1.14	0.00	86	96	0
OCT	14.94	0.00	0	14.94	14.94	0.00	79	73	3
NOV	22.96	0.14	1	22.82	22.82	0.19	75	65	68
TOTAL	65.52	4.53	-	60.99	60.99	0.32	-	-	189
AVERAGE	10.92	0.76	7	10.17	10.17	0.05	80	76	32

Table D-32. SPACE COOLING SUBSYSTEM

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD	SOLAR FRACTION OF LOAD (%)	SOLAR ENERGY USED	OPERATING ENERGY	AUX THERMAL USED	AUX ELECT FUEL	BUILDING TEMP (°F)	AMB TEMP (°F)	COOLING DEGREE-DAYS
MAR	21.70	1	1.26	2.70	8.15	11.64	74	63	27
APR	35.91	6	7.09	4.03	25.35	36.21	79	74	281
JUL	102.41	6	18.71	4.21	47.02	67.16	85	94	910
AUG	95.63	2	5.85	7.38	44.35	63.36	86	96	961
OCT	62.13	6	3.93	5.27	23.98	34.26	79	73	236
NOV	46.82	1	2.01	2.01	16.84	24.06	75	65	84
TOTAL	364.60	-	38.85	25.60	165.69	236.69	-	-	2,499
AVERAGE	60.77	4	6.48	4.27	27.62	39.45	80	76	417

Table D-33. CHILLER PERFORMANCE

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in million BTU, unless otherwise indicated)

	SOLAR			AUXILIARY		
MONTH	EQUIPMENT LOAD	THERMAL ENERGY INPUT	COEFFICIENT OF PERFORMANCE (COP)	EQUIPMENT LOAD	THERMAL ENERGY INPUT	COEFFICIENT OF PERFORMANCE (COP)
MAR	0.29	1.26	0.23	21.41	8.15	1.84
APR	2.36	7.09	0.33	33.55	25.35	0.93
JUL	7.18	18.71	0.38	95.23	47.02	1.42
AUG	1.82	5.85	0.38	93.81	44.35	1.48
OCT	1.24	3.93	0.32	60.89	23.98	1.78
NOV	0.21	2.01	0.10	46.61	16.84	1.94
TOTAL	13.10	38.85	-	351.50	165.69	-
AVERAGE	2.18	6.48	0.34	58.58	27.62	1.49

Table D-34. SOLAR-SPECIFIC OPERATING ENERGY

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in million BTU)

MONTH	ECSS OPERATING ENERGY	POOL PUMP OPERATING ENERGY	SHS OPERATING ENERGY	SCS OPERATING ENERGY	TOTAL SOLAR OPERATING ENERGY
MAR	0.60	0.00	0.13	0.21	0.94
APR	1.01	0.00	0.00	1.61	2.62
JUL	0.72	0.00	0.00	1.77	2.49
AUG	0.50	0.00	0.00	5.13	5.63
OCT	0.60	0.09	0.00	3.20	3.89
NOV	0.39	0.00	0.19	0.20	0.78
TOTAL	3.82	0.09	0.32	12.12	16.35
AVERAGE	0.64	0.02	0.05	2.02	2.73

Table D-35. ENERGY SAVINGS

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in million BTU)

MONTH	SOLAR ENERGY USED	SPACE HEATING ELECTRICAL	SPACE COOLING ELECTRICAL	ECSS OPERATING ENERGY SOLAR-UNIQUE	NET ENERGY SAVINGS ELECTRICAL
MAR	5.65	4.26	-0.05	-0.60	3.61
APR	7.09	0.00	0.96	-1.01	-0.05
JUL	18.71	0.00	3.97	-0.72	3.25
AUG	5.85	0.00	-3.64	-0.50	-4.14
OCT	3.93	0.00	-2.51	-0.60 (+0.09)(1)	-3.20
NOV	2.15	-0.06	-0.09	-0.39	-0.54
TOTAL	43.38	4.20	-1.36	-3.82	-1.07
AVERAGE	7.23	0.70	-0.23	-0.64	-0.18

(1)Pool pump operating energy.

Table D-36. WEATHER CONDITIONS

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY)		AMBIENT TEMPERATURE (°F)		HEATING DEGREE-DAYS		COOLING DEGREE-DAYS	
	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE	MEASURED	LONG-TERM AVERAGE
MAR	1,509	1,814	63	60	103	185	27	21
APR	2,011	2,356	74	68	15	60	281	141
JUL	2,113	2,485	94	91	0	0	910	812
AUG	1,959	2,293	96	89	0	0	961	747
OCT	1,317	1,578	73	72	3	17	236	240
NOV	966	1,150	65	60	68	182	84	26
TOTAL	-	-	-	-	189	444	2,499	1,987
AVERAGE	1,646	1,946	78	73	32	74	417	331

Table D-37. SOLAR SYSTEM THERMAL PERFORMANCE

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	SOLAR ENERGY COLLECTED	SYSTEM LOAD	SOLAR ENERGY USED	AUXILIARY ENERGY FOSSIL	OPERATING ENERGY	ENERGY SAVINGS FOSSIL	ENERGY SAVINGS ELECTRICAL	SOLAR FRACTION (%)	SOLAR COEFFICIENT OF PERFORMANCE[1] (COP)
JUL	66.84	211.83	7.24	719.51	58.58	12.06	-7.33	2	0.99
AUG	71.11	190.58	18.91	622.24	53.62	31.52	-6.45	5	2.93
SEP	68.79	43.95	14.04	121.18	28.36	22.37	-3.04	16	4.62
TOTAL	206.74	446.36	40.19	1,462.93	140.56	65.95	-16.82	-	-
AVERAGE	68.91	148.79	13.39	487.64	46.85	21.90	-5.61	4	2.39

(1)Solar energy used/solar-specific operating energy.

Table D-38. COLLECTOR SUBSYSTEM PERFORMANCE

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	INCIDENT SOLAR RADIATION	COLLECTED SOLAR ENERGY	COLLECTOR SUBSYSTEM EFFICIENCY (%)	OPERATIONAL INCIDENT ENERGY	COLLECTOR ARRAY OPERATIONAL EFFICIENCY (%)	ECSS OPERATING ENERGY	DAYTIME AMBIENT TEMPERATURE (°F)
JUL	280.96	66.84	24	160.16	42	2.30	85
AUG	263.03	71.11	27	184.33	39	1.79	80
SEP	275.79	68.79	25	194.95	35	1.88	72
TOTAL	819.78	206.74	-	539.44	-	5.97	-
AVERAGE	273.26	68.91	25	179.81	39	1.99	79

Table D-39. SPACE HEATING SUBSYSTEM

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(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 FOSSIL FUEL	TOTAL OPERATING ENERGY	BLDG TEMP (°F)	AMB TEMP (°F)	HEATING DEGREE-DAYS
JUL	0.00	0.00	0	0.00	0.00	0.00	78	76	0
AUG	0.00	0.00	0	0.00	0.00	0.00	79	72	2
SEP	3.68	0.68	18	3.00	5.00	13.68	79	63	100
TOTAL	3.68	0.68	-	3.00	5.00	13.68	-	-	102
AVERAGE	1.23	0.23	18	1.00	1.67	4.56	79	70	34

Table D-40. SPACE COOLING SUBSYSTEM

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(All values in million BTU, unless otherwise indicated)

MONTH	COOLING LOAD	SOLAR FRACTION OF LOAD (%)	SOLAR ENERGY USED	OPERATING ENERGY	AUX THERMAL USED	AUX FOSSIL FUEL	BUILDING TEMP (°F)	AMB TEMP (°F)	COOLING DEGREE-DAYS
JUL	211.83	2	7.24	56.27	431.70	719.51	78	76	343
AUG	190.58	5	18.91	51.83	373.34	622.24	79	72	289
SEP	40.27	16	13.36	12.80	69.70	116.17	79	63	85
TOTAL	442.68	-	39.51	120.90	874.74	1,457.92	-	-	717
AVERAGE	147.56	4	13.17	40.30	291.58	485.97	79	70	239

Table D-41. CHILLER PERFORMANCE

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(All values in million BTU, unless otherwise indicated)

		SOLAR AND AUXILLIARY	
MONTH	EQUIPMENT LOAD	THERMAL ENERGY INPUT	COEFFICIENT OF PERFORMANCE (COP)
JUL	211.83	438.94	0.48
AUG	190.58	392.25	0.49
SEP	40.27	83.06	0.48
TOTAL	442.68	914.25	-
AVERAGE	147.56	304.75	0.48

Table D-42. SOLAR-SPECIFIC OPERATING ENERGY

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(All values in million BTU)

MONTH	ECSS OPERATING ENERGY	SCS OPERATING ENERGY	TOTAL SOLAR OPERATING ENERGY
JUL	2.30	5.03	7.33
AUG	1.79	4.66	6.45
SEP	1.88	1.16	3.04
TOTAL	5.97	10.85	16.82
AVERAGE	1.99	3.62	5.61

Table D-43. ENERGY SAVINGS

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(All values in million BTU)

MONTH	SOLAR ENERGY USED	SPACE HEATING FOSSIL FUEL	SPACE COOLING ELECTRICAL	SPACE COOLING FOSSIL FUEL	ECSS OPERATING ENERGY SOLAR-UNIQUE	NET ENERGY SAVINGS ELECTRICAL	NET ENERGY SAVINGS FOSSIL FUEL
JUL	7.24	0.00	-5.03	12.06	-2.30	-7.33	12.06
AUG	18.91	0.00	-4.66	31.52	-1.79	-6.45	31.52
SEP	14.04	0.11	-1.16	22.26	-1.88	-3.04	22.37
TOTAL	40.19	0.11	-10.85	65.84	-5.97	-16.82	65.95
AVERAGE	13.39	0.04	-3.62	21.95	-1.99	-5.61	21.98

Table D-44. WEATHER CONDITIONS

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

MONTH	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY) MEASURED	DAILY INCIDENT SOLAR ENERGY PER UNIT AREA (BTU/FT^2-DAY) LONG-TERM AVERAGE	AMBIENT TEMPERATURE (°F) MEASURED	AMBIENT TEMPERATURE (°F) LONG-TERM AVERAGE	HEATING DEGREE-DAYS MEASURED	HEATING DEGREE-DAYS LONG-TERM AVERAGE	COOLING DEGREE-DAYS MEASURED	COOLING DEGREE-DAYS LONG-TERM AVERAGE
JUL	1,427	1,721	76	72	0	11	343	225
AUG	1,336	1,665	72	70	2	21	289	182
SEP	1,448	1,494	64	69	100	173	85	23
TOTAL	-	-	-	-	102	205	717	430
AVERAGE	1,404	1,627	71	70	34	68	239	143

APPENDIX E

LONG-TERM WEATHER DATA

Table E-1. EL TORO LIBRARY LONG-TERM WEATHER DATA

COLLECTOR TILT: 19.00 DEGREES
LATITUDE: 33.68 DEGREES

LOCATION: EL TORO, CALIFORNIA
COLLECTOR AZIMUTH: -30.00 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1,663	948	0.56985	1.309	1,240	372	0	53
FEB	2,096	1,235	0.58915	1.212	1,498	298	7	55
MAR	2,630	1,611	0.61262	1.118	1,802	279	0	56
APR	3,150	1,928	0.61208	1.034	1,993	177	9	59
MAY	3,489	2,072	0.59393	0.977	2,024	94	29	63
JUN	3,616	2,194	0.60671	0.953	2,090	38	77	66
JUL	3,545	2,363	0.66676	0.962	2,274	0	181	71
AUG	3,273	2,157	0.65896	1.010	2,178	0	209	72
SEP	2,812	1,737	0.61747	1.083	1,881	9	165	70
OCT	2,249	1,357	0.60338	1.181	1,602	64	70	65
NOV	1,762	1,025	0.58169	1.284	1,316	195	12	59
DEC	1,540	870	0.56504	1.342	1,167	341	0	54

LEGEND:

HOBAR - Monthly average daily extraterrestrial radiation (ideal) in BTU/day-ft^2

HBAR - Monthly average daily radiation (actual) in BTU/day-ft^2.

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 x HBAR) in BTU/day-ft^2.

HDD - Number heating degree-days per month.

CDD - Number of cooling degree-days per month.

TBAR - Average ambient temperature in degrees Fahrenheit.

Table E-2. HONEYWELL-SALT RIVER LONG-TERM WEATHER DATA

COLLECTOR TILT: 20.00 DEGREES
LATITUDE: 33.50 DEGREES

LOCATION: PHOENIX, ARIZONA
COLLECTOR AZIMUTH: -34.00 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1,672	1,021	0.61072	1.320	1,348	428	0	51
FEB	2,105	1,375	0.65337	1.226	1,686	292	14	55
MAR	2,636	1,814	0.68807	1.127	2,045	185	21	60
APR	3,154	2,356	0.74707	1.040	2,450	60	141	68
MAY	3,489	2,677	0.76719	0.975	2,609	0	355	76
JUN	3,615	2,739	0.75787	0.946	2,592	0	588	85
JUL	3,544	2,489	0.70223	0.960	2,388	0	812	91
AUG	3,275	2,293	0.70022	1.009	2,314	0	747	89
SEP	2,818	2,017	0.71580	1.092	2,203	0	564	84
OCT	2,256	1,578	0.69935	1.199	1,893	17	240	72
NOV	1,771	1,150	0.64945	1.303	1,499	182	26	60
DEC	1,550	933	0.60198	1.352	1,261	388	0	53

LEGEND:

HOBAR - Monthly average daily extraterrestrial radiation (ideal) in BTU/day-ft^2

HBAR - Monthly average daily radiation (actual) in BTU/day-ft^2.

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 x HBAR) in BTU/day-ft^2.

HDD - Number heating degree-days per month.

CDD - Number of cooling degree-days per month.

TBAR - Average ambient temperature in degrees Fahrenheit.

Table E-3. SAN ANSELMO SCHOOL LONG-TERM WEATHER DATA

COLLECTOR TILT: 40.00 DEGREES
LATITUDE: 37.34 DEGREES

LOCATION: SAN JOSE, CALIFORNIA
COLLECTOR AZIMUTH: 0.00 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1,469	708	0.48195	1.656	1,172	481	0	50
FEB	1,922	1,018	0.52947	1.438	1,463	350	0	53
MAR	2,496	1,456	0.58341	1.214	1,768	322	0	55
APR	3,079	1,921	0.62389	1.012	1,944	228	12	58
MAY	3,477	2,212	0.63622	0.882	1,952	123	20	62
JUN	3,634	2,349	0.64623	0.829	1,947	50	71	66
JUL	3,549	2,323	0.65442	0.851	1,978	12	117	68
AUG	3,227	2,054	0.63643	0.953	1,958	15	111	68
SEP	2,702	1,700	0.62895	1.135	1,929	13	94	68
OCT	2,087	1,213	0.58118	1.378	1,671	90	19	63
NOV	1,573	822	0.52263	1.620	1,332	276	0	56
DEC	1,343	645	0.48036	1.740	1,123	456	0	50

LEGEND:

HOBAR - Monthly average daily extraterrestrial radiation (ideal) in BTU/day-ft^2

HBAR - Monthly average daily radiation (actual) in BTU/day-ft^2.

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 x HBAR) in BTU/day-ft^2.

HDD - Number heating degree-days per month.

CDD - Number of cooling degree-days per month.

TBAR - Average ambient temperature in degrees Fahrenheit.

Table E-4. SCOTTSDALE COURTHOUSE LONG-TERM WEATHER DATA

COLLECTOR TILT: 0.00 DEGREES
LATITUDE: 34.39 DEGREES

LOCATION: SCOTTSDALE, ARIZONA
COLLECTOR AZIMUTH: 0.00 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1,625	1,021	0.62833	1.000	1,021	428	0	51
FEB	2,063	1,372	0.66479	1.000	1,372	292	14	55
MAR	2,605	1,814	0.69637	1.000	1,814	185	21	60
APR	3,138	2,356	0.75091	1.000	2,356	60	141	68
MAY	3,487	2,677	0.76755	1.000	2,677	0	355	76
JUN	3,620	2,739	0.75669	1.000	2,739	0	588	85
JUL	3,546	2,485	0.70073	1.000	2,485	0	812	91
AUG	3,265	2,293	0.70237	1.000	2,293	0	747	89
SEP	2,792	2,017	0.72236	1.000	2,017	0	564	84
OCT	2,218	1,578	0.71146	1.000	1,578	17	240	72
NOV	1,726	1,150	0.66658	1.000	1,150	182	26	60
DEC	1,502	933	0.62109	1.000	933	388	0	53

LEGEND:

HOBAR - Monthly average daily extraterrestrial radiation (ideal) in BTU/day-ft^2

HBAR - Monthly average daily radiation (actual) in BTU/day-ft^2.

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 x HBAR) in BTU/day-ft^2.

HDD - Number heating degree-days per month.

CDD - Number of cooling degree-days per month.

TBAR - Average ambient temperature in degrees Fahrenheit.

Table E-5. UNIVERSITY OF MINNESOTA LONG-TERM WEATHER DATA

COLLECTOR TILT: 45.00 DEGREES
LATITUDE: 45.12 DEGREES

LOCATION: MINNEAPOLIS, MINNESOTA
COLLECTOR AZIMUTH: 0.00 DEGREES

MONTH	HOBAR	HBAR	KBAR	RBAR	SBAR	HDD	CDD	TBAR
JAN	1,052	465	0.44179	1.990	924	1.637	0	12
FEB	1,531	763	0.49853	1.650	1,259	1,358	0	17
MAR	2,179	1,102	0.50590	1.302	1,436	1,138	0	28
APR	2,888	1,442	0.49913	1.045	1,507	597	0	45
MAY	3,416	1,737	0.50840	0.905	1,572	271	26	57
JUN	3,641	1,928	0.52966	0.850	1,640	65	122	67
JUL	3,525	1,969	0.55850	0.874	1,721	11	225	72
AUG	3,090	1,689	0.54646	0.986	1,665	21	182	70
SEP	2,433	1,254	0.51528	1.192	1,494	173	23	69
OCT	1,719	859	0.49972	1.520	1,306	472	7	50
NOV	1,163	479	0.41231	1,817	871	978	0	32
DEC	925	354	0.38260	2.007	710	1,438	0	19

LEGEND:

HOBAR - Monthly average daily extraterrestrial radiation (ideal) in BTU/day-ft^2

HBAR - Monthly average daily radiation (actual) in BTU/day-ft^2.

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 x HBAR) in BTU/day-ft^2.

HDD - Number heating degree-days per month.

CDD - Number of cooling degree-days per month.

TBAR - Average ambient temperature in degrees Fahrenheit.

APPENDIX F

HOURLY BUILDING COOLING LOADS

This appendix contains averaged hourly building cooling loads for the five sites. The tables contain the monthly average hourly building cooling loads. The graphs show the peak and lowest cooling months. These months indicate the range of variation of building cooling loads over the seasons.

Table F-1. COOLING LOAD VS. TIME OF DAY

EL TORO LIBRARY
MARCH 1981 THROUGH JUNE 1981,
AUGUST 1981 THROUGH NOVEMBER 1981

(All values in tons)

HOUR OF DAY	MAR	APR	MAY	JUN	AUG	SEP	OCT	NOV
0	0.00	0.00	0.08	0.00	0.00	0.00	0.00	0.00
1	0.00	0.00	0.08	0.00	0.00	0.00	0.00	0.00
2	0.00	0.00	0.06	0.00	0.00	0.00	0.00	0.00
3	0.00	0.00	0.06	0.00	0.00	0.00	0.00	0.00
4	0.00	0.00	0.05	0.00	0.00	0.00	0.00	0.00
5	0.00	0.00	0.73	0.86	0.63	0.92	0.25	0.00
6	0.37	1.28	3.29	4.48	5.16	8.00	3.08	0.08
7	1.24	2.24	4.01	5.35	6.18	8.20	3.90	2.90
8	2.42	3.37	4.52	5.90	7.20	7.91	4.56	4.06
9	3.11	3.84	5.00	6.40	7.73	7.84	5.12	3.99
10	3.58	4.37	5.46	6.80	8.01	8.07	5.71	4.00
11	3.96	4.89	5.81	6.81	8.01	8.69	6.50	4.37
12	4.26	5.19	6.10	6.64	8.47	9.33	7.29	4.75
13	4.45	5.53	6.35	6.99	8.28	9.87	7.16	4.69
14	4.69	5.77	6.25	6.90	8.41	9.85	6.95	4.89
15	4.72	5.73	6.28	6.96	7.61	8.90	6.39	4.71
16	4.43	5.79	6.01	6.67	6.67	7.53	5.68	4.86
17	2.41	4.40	3.88	4.71	3.78	5.17	4.26	4.34
18	0.04	3.49	3.58	4.75	3.80	4.99	3.88	3.12
19	0.00	3.13	2.98	4.13	3.48	4.87	3.64	2.62
20	0.00	2.22	0.75	0.52	0.43	0.84	1.11	2.63
21	0.00	0.28	0.16	0.00	0.00	0.00	0.28	1.10
22	0.00	0.00	0.04	0.00	0.00	0.00	0.00	0.00
23	0.00	0.00	0.08	0.00	0.00	0.00	0.00	0.00

Table F-2. COOLING LOAD VS. TIME OF DAY

HONEYWELL-SALT RIVER
JUNE 1981 THROUGH OCTOBER 1981

(All values in tons)

HOUR OF DAY	JUN	JUL	AUG	SEP	OCT
0	24	90	*	54	14
1	23	82	*	51	13
2	23	75	*	48	12
3	22	73	*	46	11
4	23	69	*	44	10
5	84	68	*	49	13
6	93	83	*	58	21
7	96	92	*	66	31
8	101	97	*	75	35
9	102	100	*	79	43
10	104	106	*	83	50
11	105	107	*	88	55
12	103	108	*	89	59
13	109	110	*	92	63
14	110	114	*	95	67
15	112	114	*	96	67
16	108	109	*	91	64
17	105	106	*	88	59
18	103	104	*	85	52
19	99	102	*	82	47
20	95	97	*	78	39
21	93	96	*	74	27
22	83	93	*	70	20
23	28	93	*	59	14

* Denotes unavailable data.

Table F-3. COOLING LOAD VS. TIME OF DAY

SAN ANSELMO SCHOOL
MARCH 1981 THROUGH NOVEMBER 1981

(All values in tons)

HOUR OF DAY	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV
0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
1	0.00	0.00	0.00	0.00	0.00	0.00	0.01	0.00	0.00
2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00	0.00	0.00	0.43	0.00	0.00
4	0.00	0.00	0.00	1.28	4.31	2.88	0.96	0.00	0.00
5	0.00	0.25	0.00	4.12	4.68	3.84	2.97	0.75	0.00
6	0.00	0.60	1.47	3.55	2.50	4.63	3.21	0.30	0.00
7	0.00	2.26	1.05	6.93	4.89	5.55	4.85	1.82	0.30
8	0.00	4.10	4.17	13.11	9.44	7.95	8.64	2.82	0.84
9	0.05	6.46	8.67	15.69	14.82	10.64	13.22	7.22	1.18
10	0.00	9.71	12.28	18.43	18.44	13.86	16.69	10.75	3.17
11	3.95	12.25	16.28	22.95	23.57	19.13	20.02	16.34	4.12
12	6.97	12.41	18.47	26.96	27.77	22.08	21.94	18.71	7.38
13	7.38	14.05	18.31	26.69	26.21	20.86	23.78	18.06	7.55
14	8.86	12.95	15.42	23.13	21.58	16.72	22.86	16.78	6.65
15	3.64	9.41	9.66	17.43	14.30	12.77	15.45	12.52	4.32
16	1.12	3.44	4.26	8.50	4.95	6.46	7.47	5.92	1.50
17	0.37	1.01	2.17	1.91	2.55	3.56	2.73	0.54	0.05
18	0.00	0.70	0.72	0.46	1.09	0.46	1.31	0.00	0.15
19	0.00	0.61	0.31	0.00	0.00	0.00	1.01	0.00	0.00
20	0.00	0.02	0.00	0.00	0.00	0.00	0.58	0.00	0.00
21	0.00	0.00	0.00	0.00	0.00	0.00	0.05	0.00	0.00
22	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
23	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00

Table F-4. COOLING LOAD VS. TIME OF DAY

SCOTTSDALE COURTHOUSE
MARCH, APRIL, JULY, AUGUST, OCTOBER, AND NOVEMBER 1981

(All values in tons)

HOUR OF DAY	MAR	APR	JUL	AUG	OCT	NOV
0	0.00	0.00	0.00	*	0.00	0.00
1	0.00	0.00	0.00	*	0.00	0.00
2	0.24	0.66	1.88	*	0.51	0.00
3	0.56	0.66	2.72	*	0.84	0.00
4	1.09	3.57	11.83	*	2.31	0.00
5	2.50	4.03	16.00	*	3.04	1.39
6	2.73	4.12	16.58	*	9.17	6.68
7	2.88	5.04	16.92	*	11.11	9.14
8	3.22	5.90	16.33	*	11.21	9.67
9	3.62	6.33	16.75	*	12.73	10.15
10	3.62	6.24	16.83	*	13.74	10.62
11	3.82	6.69	17.75	*	13.40	10.82
12	3.92	6.57	18.33	*	13.05	10.33
13	4.04	6.83	18.08	*	12.89	10.67
14	4.16	7.18	18.08	*	13.51	10.98
15	4.20	7.27	18.33	*	13.55	11.22
16	4.30	7.55	17.83	*	13.41	11.33
17	4.05	7.27	16.75	*	10.97	9.17
18	3.76	6.47	16.58	*	4.66	2.46
19	3.23	5.48	15.42	*	3.35	1.80
20	2.40	1.86	4.23	*	2.03	1.74
21	0.04	0.00	0.00	*	0.92	1.01
22	0.00	0.00	0.00	*	0.00	0.00
23	0.00	0.00	0.00	*	0.00	0.00

* Denotes unavailable data.

Table F-5. COOLING LOAD VS. TIME OF DAY

UNIVERSITY OF MINNESOTA
JULY 1981 THROUGH SEPTEMBER 1981

(All values in tons)

HOUR OF DAY	JUL	AUG	SEP
0	10.28	0.00	0.25
1	10.38	0.00	0.24
2	10.29	0.00	0.00
3	10.17	0.00	0.00
4	12.76	1.48	0.00
5	23.69	20.05	1.48
6	32.55	40.35	4.13
7	33.91	42.32	5.29
8	36.48	42.85	6.78
9	38.22	48.72	8.74
10	41.05	47.04	8.45
11	42.09	45.75	14.17
12	44.88	47.26	14.68
13	43.73	47.60	14.01
14	42.03	47.38	13.66
15	41.45	44.59	13.29
16	30.58	36.94	8.37
17	7.69	0.00	0.05
18	7.70	0.00	0.25
19	7.75	0.00	0.00
20	10.12	0.00	0.00
21	10.46	0.00	0.00
22	10.56	0.00	0.00
23	10.60	0.00	0.00

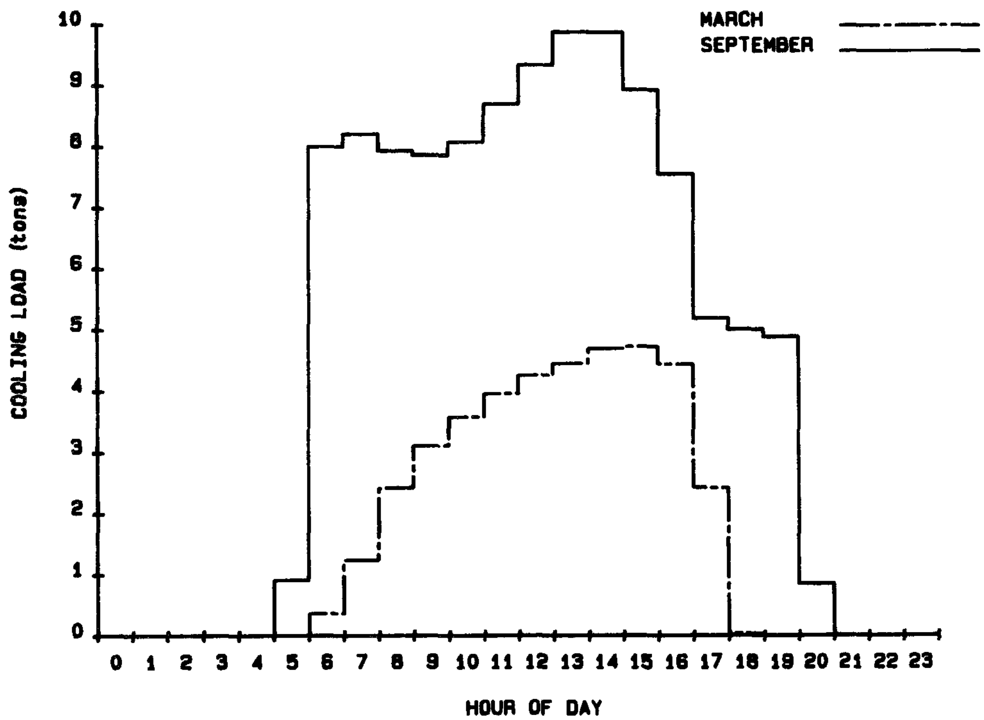


Figure F-1. Hourly Cooling Load Profile
El Toro Library
March and September 1981

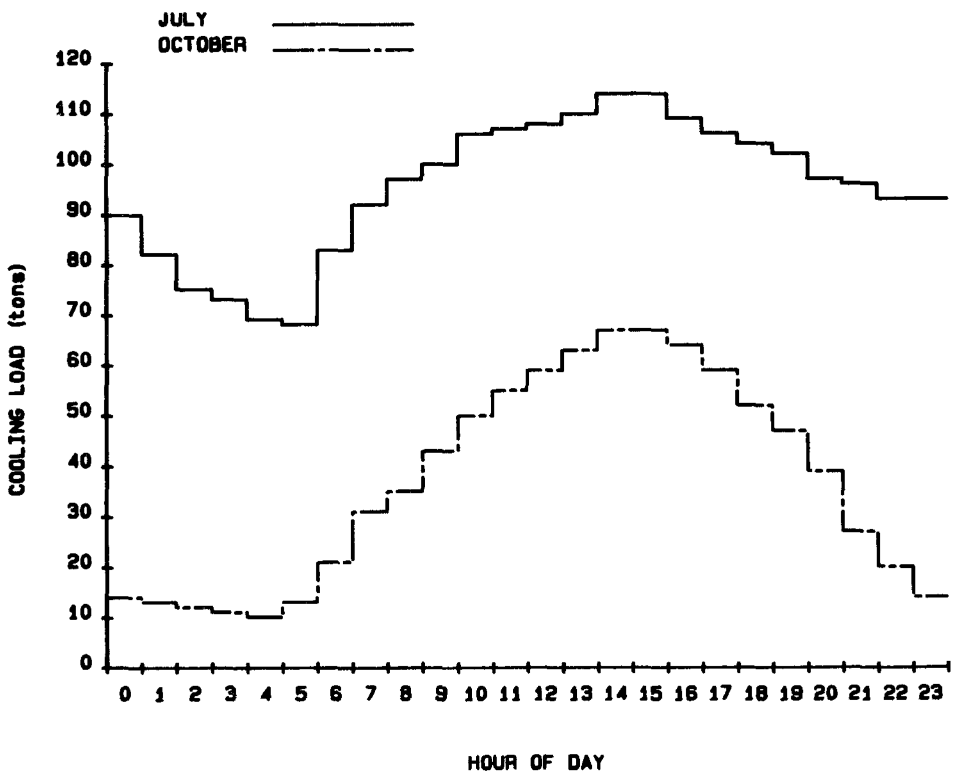


Figure F-2. Hourly Cooling Load Profile
Honeywell-Salt River
July and October 1981

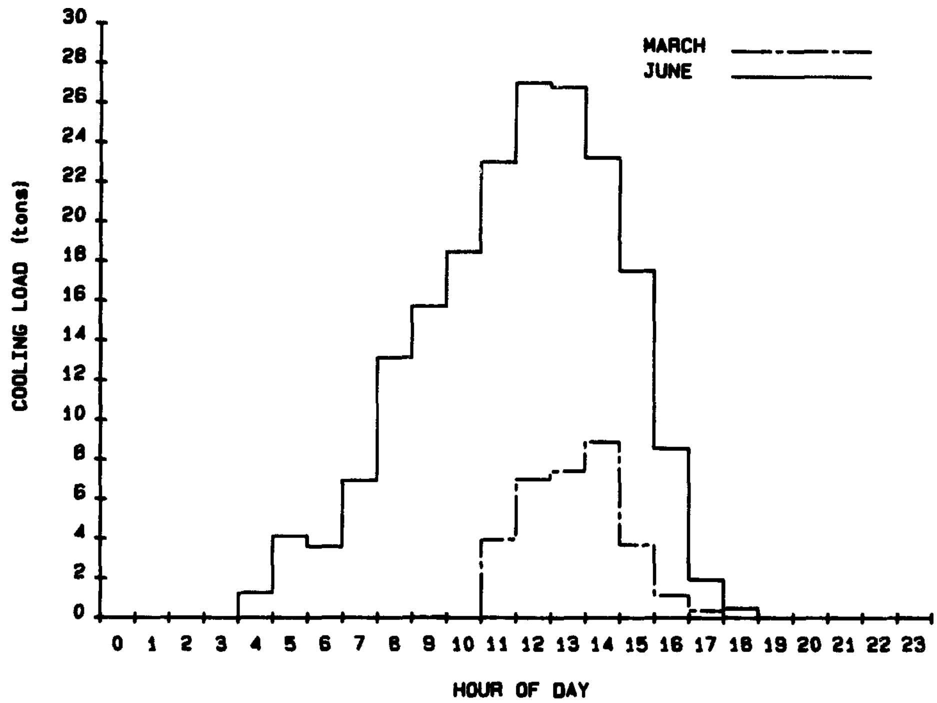


Figure F-3. Hourly Cooling Load Profile
San Anselmo School
March and June 1981

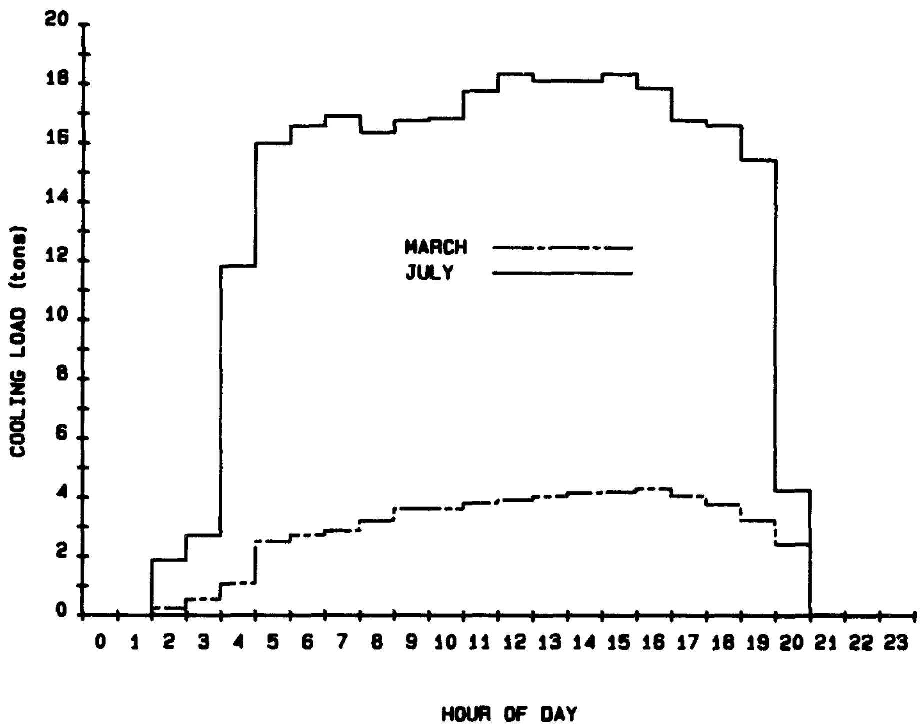


Figure F-4. Hourly Cooling Load Profile
Scottsdale Courthouse
March and July 1981

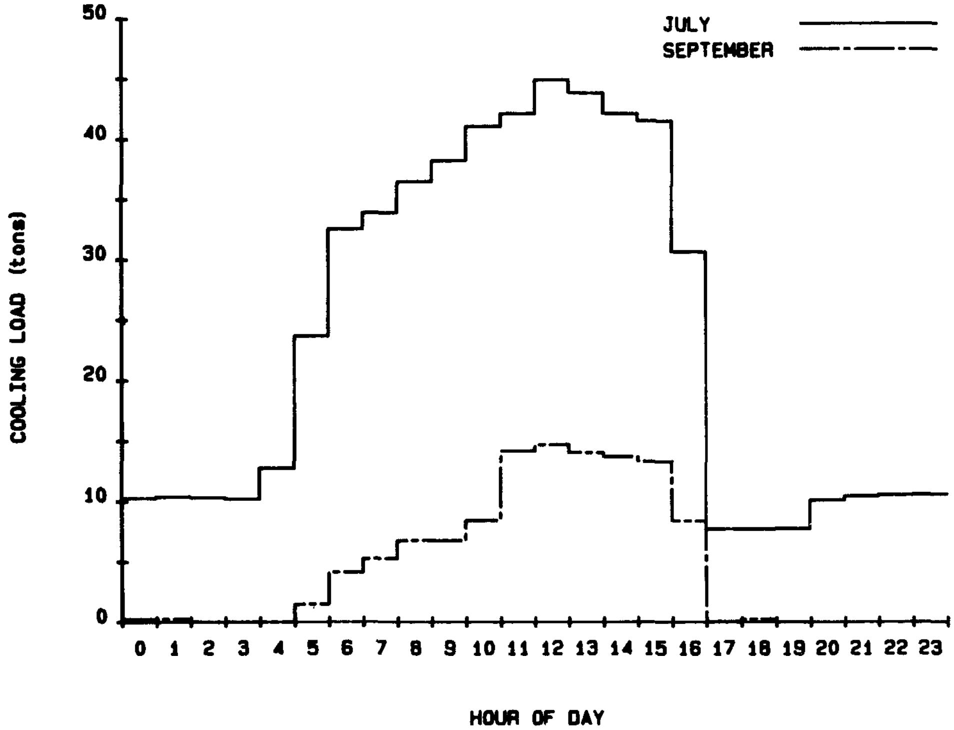


Figure F-5. Hourly Cooling Load Profile
University of Minnesota
July and September 1981

APPENDIX G

PERFORMANCE EVALUATION TECHNIQUES

APPENDIX G

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 G-1.

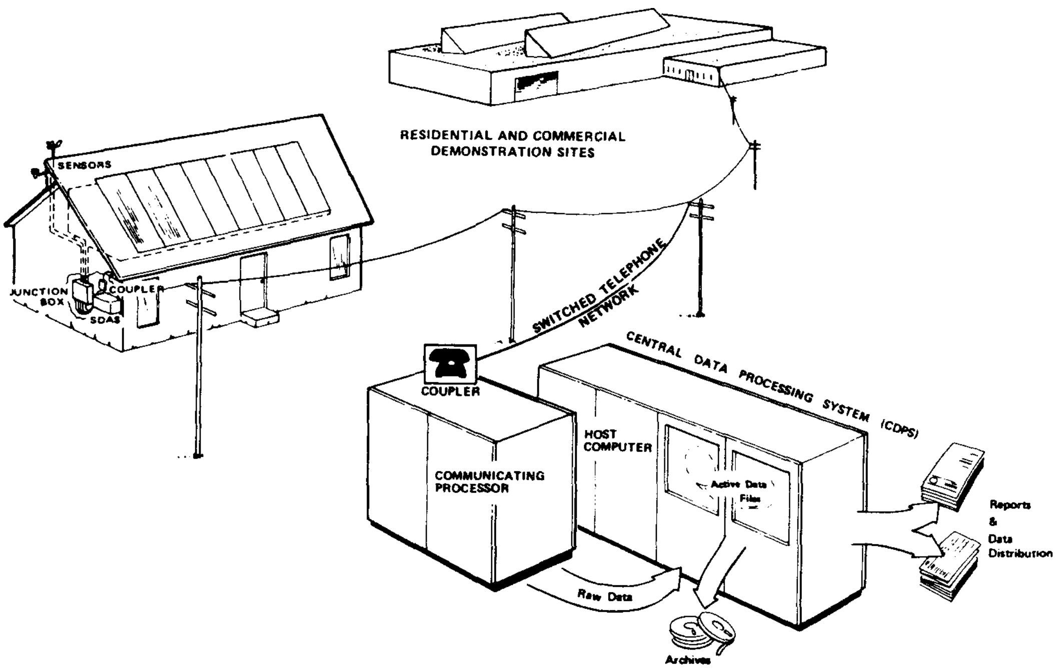


Figure G-1. The National Solar Data Network

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 in 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, an IBM 370/145, and an IBM 3033. The System 7 periodically calls up each SDAS in System 7. 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 are 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 are then tabulated on a daily 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" (see Appendix I) 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 H) 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 H), 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.

There are irregularly occurring cases of missing data as is normal for any real time data collection from mechanical equipment. When data for individual scans or whole hours are missing, values of performance factors are assigned which are interpolated from measured data. If no valid measured data are available for interpolation, a zero value is assigned. If data are missing for a whole day, each hour is interpolated separately. Data are interpolated in order to provide solar system performance factors on a whole hour, whole day, and whole month basis for use by architects and designers.

APPENDIX H

PERFORMANCE FACTORS AND SOLAR TERMS

APPENDIX H

PERFORMANCE FACTORS AND SOLAR TERMS

The performance factors identified in the site equations (Appendix I) by the use of acronyms or symbols are defined in this Appendix in Section 1. Section 1 includes the acronym, the actual name of the performance factor, and a short definition.

Section 2 contains a glossary of solar terminology, in alphabetical order. These terms are included for quick reference by the reader.

Section 3 describes general acronyms used in this report.

Section 1.	Performance Factor Definitions and Acronyms
Section 2.	Solar Terminology
Section 3.	General Acronyms

SECTION 1. PERFORMANCE FACTOR DEFINITIONS AND ACRONYMS

ACRONYM	NAME	DEFINITION
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.
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.
CAT	SCS Auxiliary Thermal Energy	Amount of energy provided to the SCS by a BTU heat transfer fluid from an auxiliary source.
* 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.
COPE	SCS Operating Energy	Amount of energy required to support the SCS operation which is not intended to be applied directly to the SCS load.
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.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
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	Portion of the SCS load which is supported by solar energy.
CSOPE	ECSS Operating Energy	Amount of energy used to support the ECSS operation (which is not intended to be supplied to the ECSS thermal state).
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 loads.
HAE	SHS Auxiliary Electrical Fuel Energy	Amount of electrical energy provided to the SHS to be converted and applied to the SHS load.
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 energy provided to the SHS by a heat transfer fluid from an auxiliary source.
* HL	Space Heating Subsystem Load	Energy required to satisfy the temperature control demands of the space heating subsystem.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
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).
HOURCT	Record Time	Count of hours elapsed from the start of 1977.
* HSFR	SHS Solar Fraction	Portion of the SHS load which is supported by solar energy.
HSE	Solar Energy to SHS	Amount of solar energy delivered to the SHS.
* 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 hot water subsystem.
* HWL	Hot Water Subsystem Load	Amount of energy supplied to the HWS.
* HWDM	Hot Water Demand	Energy required to satisfy the temperature control demands of the building service hot water system.

* 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.
HWSE	Solar Energy to HWS	Amount of solar energy delivered to the HWS.
* HWSFR	HWS Solar Fraction	Portion 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.
RELH	Relative Humidity	Average outdoor relative humidity at the site.
* 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.
SEOP	Operational Incident Solar Energy	Amount of incident solar energy upon the collector array whenever the collector loop is active.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
* SEL	Solar Energy to Load Subsystems	Amount of solar energy supplied by the ECSS to all load subsystems.
* SFR	Solar Fraction of System Load	Portion of the system load which was supported by solar energy.
STECH	Change in ECSS Stored Energy	Change in ECSS stored energy during reference 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.
* SYSL	System Load	Energy required to satisfy all desired temperature control demands at the output of all subsystems.
* 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.
* 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.
* TB	Building Temperature	Average temperature of the controlled space of the building.
TCECOP	TCE Coefficient of Performance	Coefficient of performance of the thermodynamic conversion equipment.
TCEI	TCE Thermal Input Energy	Equivalent thermal energy which is supplied as a fuel source to thermodynamic conversion equipment.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
TCEL	Thermodynamic Conversion Equipment Load	Controlled energy output of thermodynamic conversion equipment.
TCEOPE	TCE Operating Energy	Amount of energy required to support the operation of thermodynamic conversion equipment which is not intended to appear directly in the load.
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.
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.
* 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.

* Primary Performance Factors

ACRONYM	NAME	DEFINITION
WDIR	Wind Direction	Average wind direction at the site.
WIND	Wind Velocity	Average wind velocity at the site.

* Primary Performance Factors

SECTION 2. SOLAR TERMINOLOGY

Absorptivity	The ratio of absorbed radiation by a surface to the total incident radiated energy on that surface.
Active Solar System	A system in which a transfer fluid (liquid or air) is circulated through a solar collector where the collected energy is converted, or transferred, to energy in the medium.
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.
Auxiliary Energy	In solar energy technology, the energy supplied to the heat 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 supplemented in nature but does not have the auxiliary system as an origin, i.e., energy supplied to the space heating load from the external ambient environment by a heat pump. The electric energy input to a heat pump is defined as operating energy.
Auxiliary Energy Subsystem	In solar energy technology the Auxiliary Energy System is the conventional heating and/or cooling equipment used as supplemental or backup to the solar system.
Array	An assembly of a number of collector elements, or panels, into the solar collector for a solar energy system.
Backflow	Reverse flow.
Backflow Preventer	A valve or damper installed to prevent reverse flow.
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.

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 Subsystem	The assembly of components that absorbs incident solar energy and transfers the absorbed thermal energy to a heat transfer fluid.
Concentrating Solar Collector	A solar collector that concentrates the energy from a larger area onto an absorbing element of smaller area.
Conversion Efficiency	Ratio of thermal energy output to solar energy incident on the collector array.
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 average 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.
Drain Down	An arrangement of sensors, valves and actuators to automatically 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-Curve	The collector instantaneous efficiency curve. Used in the "F-curve" procedure for collector analysis (see Instantaneous Efficiency).
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 re-radiated from the panel is trapped within the collector 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 technology, 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 average daily temperature is <u>below</u> 65°F.
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).
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.
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 required heating or cooling.
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 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 (T), and outside ambient temperature (Ta). The operating point is defined as:

$$\frac{Ti-Ta}{I} \left(\frac{°F \times hr. \times sq.\ ft.}{BTU}\right)$$

Operational Collector Efficiency — Ratio of collected solar energy to incident solar energy only during the time the collector fluid is being circulated with the intention of delivering solar-source energy to the system.

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 re-radiates little of it as thermal radiation.

Sensor — A device used to monitor a physical parameter in 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, N_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. 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.

SECTION 3. GENERAL ACRONYMS

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 Fahrenheit degree. One BTU is equivalent to 2.932×10^{-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.

APPENDIX I

EXAMPLE OF PERFORMANCE EQUATIONS

APPENDIX I

EXAMPLE OF PERFORMANCE EQUATIONS
(FOR SAN ANSELMO SCHOOL)

INTRODUCTION

Solar energy system performance is evaluated by performing energy balance computations on the system and its major subsystems. These calculations are based on physical measurement data taken from each sensor every 320 seconds.* This data is then mathematically combined to determine the hourly, daily, and monthly performance of the system. This appendix describes the general computational methods and the specific energy balance equations used for this site.

Data samples from the system measurements are integrated to provide discrete approximations of the continuous functions which characterize the system's dynamic behavior. This integration is performed by summation of the product of the measured rate of the appropriate performance parameters and the sampling interval over the total time period of interest.

There are several general forms of integration equations which are applied to each site. These general forms are exemplified as follows: the total solar energy available to the collector array is given by

$$\text{SOLAR ENERGY AVAILABLE} = (1/60)\ \Sigma\ [\text{I001} \times \text{CLAREA}] \times \Delta\tau$$

where I001 is the solar radiation measurement provided by the pyranometer in BTU per square foot per hour, CLAREA is the area of the collector array in square feet, $\Delta\tau$ is the sampling interval in minutes, and the factor (1/60) is included to convert the solar radiation "rate" to the proper units of time.

Similarly, the energy flow within a system is given typically by

$$\text{COLLECTED SOLAR ENERGY} = \Sigma\ [\text{M100} \times \Delta H] \times \Delta\tau$$

where M100 is the mass flow rate of the heat transfer fluid in lb_m/min and ΔH is the enthalpy change, in BTU/lb_m, of the fluid as it passes through the heat exchanging component.

For a liquid system ΔH is generally given by

$$\Delta H = \bar{C}_p\ \Delta T$$

where $\bar{C}_p$ is the average specific heat, in BTU/lb_m-°F, of the heat transfer fluid and ΔT, in °F, is the temperature differential across the heat exchanging component.

* See Appendix G.

For an air system ΔH is generally given by

$$\Delta H = H_a(T_{out}) - H_a(T_{in})$$

where $H_a(T)$ is the enthalpy, in BTU/lb_m, of the transport air evaluated at the inlet and outlet temperatures of the heat exchanging component.

$H_a(T)$ can have various forms, depending on whether or not the humidity ratio of the transport air remains constant as it passes through the heat exchanging component.

For electrical power, a general example is

$$\text{ECSS OPERATING ENERGY} = (3413/60)\ \Sigma\ [EP100] \times \Delta\tau$$

where EP100 is the power required by electrical equipment in kilowatts and the two factors (1/60) and 3413 correct the data to BTU/min.

Letter Designations

C or CP	=	Specific Heat
D	=	Direction or Position
EE	=	Electric Energy
EP	=	Electric Power
F	=	Fuel Flow Rate
HWD	=	Functional procedure to calculate the enthalpy change of water at the average of the inlet and outlet temperatures
H	=	Enthalpy
HR	=	Humidity Ratio
I	=	Incident Solar Flux (Insolation)
M	=	Mass Flow Rate
N	=	Performance Parameter
P	=	Pressure
PD	=	Differential Pressure
Q	=	Thermal Energy
RHO	=	Density
T	=	Temperature
TD	=	Differential Temperature
V	=	Velocity
W	=	Heat Transport Medium Volume Flow Rate
TI	=	Time
_P	=	Appended to a function designator to signify the value of the function during the previous iteration

Subsystem Designations Number Sequence	Subsystem/Data Group
001 to 099	Climatological
100 to 199	Collector and Heat Transport
200 to 299	Thermal Storage
300 to 399	Hot Water
400 to 499	Space Heating
500 to 599	Space Cooling
600 to 699	Building/Load

EQUATIONS USED TO GENERATE MONTHLY PERFORMANCE VALUES

WEATHER DATA

AVERAGE AMBIENT TEMPERATURE (^{o}F)

$$TA = (1/60) \times \Sigma\ T001 \times \Delta\tau$$

AVERAGE BUILDING TEMPERATURE (^{o}F)

$$TB = (1/60) \times \Sigma\ T600 \times \Delta\tau$$

DAYTIME AVERAGE AMBIENT TEMPERATURE (^{o}F)

$$TDA = (1/360) \times \Sigma\ T001 \times \Delta\tau$$

for ± three hours from solar noon

BUILDING RELATIVE HUMIDITY (%)

$$RELH = (1/60) \times \Sigma\ RH600 \times \Delta\tau$$

COLLECTOR SUBSYSTEM

INCIDENT SOLAR ENERGY PER SQUARE FOOT (BTU/FT^2)

$$SE = (1/60) \times \Sigma\ I001 \times \Delta\tau$$

OPERATIONAL INCIDENT SOLAR ENERGY (BTU)

$$SEOP = (1/60) \times \Sigma\ [I001 \times CLAREA] \times \Delta\tau$$

when the collector loop is activated

SOLAR ENERGY COLLECTED BY THE ARRAY (BTU)

$$SECA = \Sigma[M100 \times CP\ (T150 - T100)] \times \Delta\tau$$

REJECTED SOLAR ENERGY (BTU)

$$CSRJE = \Sigma\ [M160 \times CP \times (T150 - T160)] \times \Delta\tau$$

when rejector fan is activated

INCIDENT SOLAR ENERGY ON COLLECTOR ARRAY (BTU)

$$SEA = CLAREA \times SE$$

COLLECTED SOLAR ENERGY (BTU/ft^2)

$$SEC = SECA/CLAREA$$

COLLECTOR ARRAY EFFICIENCY

$$CLEF = SECA/SEA$$

COLLECTOR ARRAY OPERATIONAL EFFICIENCY

$$CLEFOP = SECA/SEOP$$

ECSS OPERATING ENERGY (BTU)

$$CSOPE = 56.8833 \times \Sigma\ (EP100 + EP101) \times \Delta\tau$$

STORAGE SUBSYSTEM

AVERAGE TEMPERATURE OF STORAGE (^oF)

$$TST = (1/60) \times \Sigma\ [(T202 + T203 + T204)/3] \times \Delta\tau$$

SOLAR ENERGY TO STORAGE (BTU)

$$STEI = \Sigma\ [(M100 - M101) \times CP \times (T200 - T250)] \times \Delta\tau$$

SOLAR ENERGY FROM STORAGE (BTU)

$$STEO = \Sigma\ [M205 \times CP \times (T255 - T205)] \times \Delta\tau$$

CHANGE IN STORED ENERGY (BTU)

$$STECH1 = STOCAP \times CP(TST1) \times RHO\ (TST1) \times TST1$$

$$STECH = STECH1 - STECH1_p$$

where the subscript $_p$ refers to a prior reference value

TST1 = last hourly storage temperature

STORAGE EFFICIENCY (%)

$$STEFF = (STECH + STEO)/STEI \times 100$$

EFFECTIVE HEAT TRANSFER COEFFICIENT ($BTU/^{o}F\text{-}FT^{2}\text{-}HR$)

$$STPER = (1/60) \times \Sigma\ [SUR\_AREA \times (TST - AMB)] \times \Delta\tau$$

SUR_AREA = storage tank surface area

ABM = temperature surrounding storage tank

SPACE HEATING SUBSYSTEM

SPACE HEATING SOLAR-UNIQUE OPERATING ENERGY (BTU)

$$HOPE1 = [56.8833 \times EP400] \times \Delta\tau$$

in heating mode

SPACE HEATING SUBSYSTEM OPERATING ENERGY (BTU)

$$HOPE = [56.8833 \times \Sigma\ (EP400 + EP403 + EP404 + EP600 + AUXP6 + AUXP7)] \times \Delta\tau$$

in heating mode

AUXP6 = Chiller #4 internal power

AUXP7 = Chiller #5 internal power

SOLAR ENERGY TO SPACE HEATING SUBSYSTEM (BTU)

$$HSE = \Sigma\ [M205 \times CP \times (T255 - T205)] \times \Delta\tau$$

in heating mode

SPACE HEATING AUXILIARY FOSSIL ENERGY (BTU)

$$HAF = \Sigma\ F400 \times NGC$$

$$NGC = 1021\ BTU/FT^3$$

in heating mode

SPACE HEATING AUXILIARY THERMAL ENERGY (BTU)

$$HAT = HAF \times 0.6$$

SPACE HEATING LOAD (BTU)

$$CDE = \Sigma\ [M400 \times CP \times (T450 - T400)] \times \Delta\tau$$

$$EHL = CDE$$

in heating mode

SPACE HEATING SOLAR FRACTION (PERCENT)

$$HSFR = 100 \times HSE/(HSE + HAT)$$

SPACE HEATING FOSSIL SAVINGS (BTU)

$$HSVF = HSE/0.6$$

SPACE HEATING ELECTRICAL SAVINGS

$$HSVE = -HOPE1$$

SPACE COOLING SUBSYSTEM

SPACE COOLING OPERATING ENERGY (BTU)

$$COPE = [56.8833 \times \Sigma\ (EP400 + EP403 + EP404 + EP501 + EP502 + EP503 + EP505 + EP506 + EP507 + EP600 + AUXP1 + AUXP2 + AUXP3 + AUXP4 + AUXP5)] \times \Delta\tau$$

in cooling mode

AUXP1 = Chiller #1 internal power (solar chiller)

AUXP2 = Chiller #2 internal power

AUXP3 = Chiller #3 internal power

AUXP4 = Chiller #4 internal power

AUXP5 = Chiller #5 internal power

SPACE COOLING - SOLAR UNIQUE OPERATING ENERGY (BTU)

$$COPE1 = [56.8833 \times TCEL/CL \times \Sigma\ (EP501 + EP502 + EP503)] \times \Delta\tau + 56.8833 \times \Sigma(EP400 + EP505 + AUXP1) \times \Delta\tau$$

SPACE COOLING AUX FOSSIL ENERGY (BTU)

$$CAF = \Sigma F400 \times NGC$$

in cooling mode

SPACE COOLING AUXILIARY THERMAL ENERGY (BTU)

$$CAT = CAF \times 0.6$$

SOLAR ENERGY TO SPACE COOLING SUBSYSTEM (BTU)

$$CSE = \Sigma\ [M500 \times CP \times (T550 - T500)] \times \Delta\tau$$

SPACE COOLING LOAD (BTU)

$$CL = \Sigma\ [M400 \times CP \times (T400 - T450)] \times \Delta\tau$$

in cooling mode

SPACE COOLING SOLAR FRACTION (%)

$$CSFR = 100 \times TCEL/CL$$

SPACE COOLING FOSSIL SAVINGS (BTU)

$$CSVF = CSE/0.6$$

SPACE COOLING ELECTRICAL SAVINGS (BTU)

$$CSVE = -COPE1$$

<u>THERMODYNAMIC CONVERSION EQUIPMENT</u> (SOLAR-UNIQUE CHILLER)

TCE EQUIPMENT LOAD (BTU)

$$TCEL = \Sigma\ [M502 \times CP \times (T552 - T502)] \times \Delta\tau$$

TCE INPUT ENERGY (BTU)

$$TCEI = CSE$$

TCE REJECTED ENERGY (BTU)

$$TCERJE = \Sigma\ [M501 \times CP \times (T551 - T501)] \times \Delta\tau$$

TCE OPERATING ENERGY (BTU)

$$TCEOPE = COPE1$$

TCE CHILLER COP

$$TCECOP = TCEL/TCEI$$

<u>AUXILIARY THERMODYNAMIC CONVERSION EQUIPMENT (ATCE)</u> (AUXILIARY CHILLERS)

ATCE EQUIPMENT LOAD (BTU)

$$ATCEL = CL - TCEL$$

ATCE INPUT ENERGY (BTU)

$$ATCEI = CAT$$

ATCE REJECTED ENERGY (BTU)

$$ATCERJE = ATCEI + ATCEL$$

ATCE OPERATING ENERGY (BTU)

$$ATCEOPE = [56.8833 \times ATCEL/CL \times \Sigma\ (EP501 + EP502 + EP503)] \times \Delta\tau +$$
$$[56.8833 \times \Sigma\ (EP403 + EP404 + EP506 + EP507 + AUXP2 + AUXP3 + AUXP4 +$$
$$AUXP5)] \times \Delta\tau$$

ATCE CHILLERS COP

$$ATCECOP = ATCEL/ATCEI$$

SYSTEM FACTORS

ENERGY TO LOADS

$$CSEO = CSE + HSE$$

SOLAR ENERGY USED

$$SEL = CSEO$$

ECSS SOLAR CONVERSION EFFICIENCY

$$CSCEF = SEL/SEA$$

SYSTEM LOAD

$$SYSL = CL + EHL$$

SYSTEM SOLAR FRACTION

$$SFR = (CSFR \times CL + HSFR \times EHL)/SYSL$$

SYSTEM OPERATING ENERGY

$$SYSOPE = CSOPE + COPE + HOPE$$

SYSTEM AUXILIARY FOSSIL ENERGY

$$AXF = \Sigma\ F400 \times NGC$$

SYSTEM AUXILIARY THERMAL ENERGY

AXT = HAT + CAT

SYSTEM ELECTRICAL SAVINGS

TSVE = HSVE + CSVE - CSOPE

SYSTEM FOSSIL SAVINGS

TSVF = HSVF + CSVF

TOTAL ENERGY CONSUMED

TECSM = SECA + SYSOPE + AXF

SYSTEM PERFORMANCE FACTOR

SYSPF = SYSL/(AXF + 3.33 x SYSOPE)

APPENDIX J

CONVERSION FACTORS

APPENDIX J

CONVERSION FACTORS

Energy Conversion Factors

Fuel Type	Energy Content	Fuel Source Conversion Factor
Distillate fuel oil[1]	138,690 BTU/gallon	7.21×10^{-6} gallon/BTU
Residual fuel oil[2]	149,690 BTU/gallon	6.68×10^{-6} gallon/BTU
Kerosene	135,000 BTU/gallon	7.41×10^{-6} gallon/BTU
Propane	91,500 BTU/gallon	10.93×10^{-6} gallon/BTU
Natural gas	1,021 BTU/cubic feet	979.4×10^{-6} cubic feet/BTU
Electricity	3,413 BTU/kilowatt-hour	292.8×10^{-6} kwh/BTU

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

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

APPENDIX K

SENSOR TECHNOLOGY

APPENDIX K

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, measurement 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 and rotates with the upper body assembly.

Size:	29-3/4"L X 30"H
Starting Speed:	1 mph
Complete Tracking:	3 mph
Maximum Speed:	200 mph
Distance Constant (30 mph):	6.2'
Accuracy:	± 1% below 25 mph
	± 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:	± 3% from 0-80% R.H. ± 5-6% 80-100% 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 multijunction 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:	± 1% over ambient temperature range -20°C to 40°C
Linearity:	0.5% from 0 to 2,800 W/M^2
Response Time:	1 second
Cosine Error:	± 1% 0-70° zenith angle ± 3% 70-80° 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°F to 250°F
Calibration Accuracy:	± 1% (½" to 3½" line size) ± 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:	± 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°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 cubic ft/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° 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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