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Survey of EPA Facilities for Solar Thermal Energy Applications

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SURVEY OF EPA FACILITIES FOR SOLAR THERMAL ENERGY APPLICATIONS

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16. ABSTRACT

A study was done to assess the feasibility of applying solar thermal energy systems to EPA facilities. A survey was conducted to determine those EPA facilities where solar energy could best be used. The most broadly applicable solar thermal energy could best be used. The most broadly These systems were optimized for each specific application and the system/facility combinations were ranked on the basis of greatest cost effectiveness. The Robert S. Kerr Environmental Research Laboratory in Ada, Oklahoma was selected as a preferred first site for the U.S. Department of Energy's Federal Solar Buildings Program. This report was submitted in fulfillment of Contract No. 68-03-2567 by Acurex Corporation under the sponsorship of the U.S. Environmental Protection Agency. This report covers a period from November 1, 1977 to June 30, 1978, and work was completed as of December 21, 1979.

The reader is cautioned that advances in solar technology and increased energy costs since the writing of this report (June 1978) have negated some of the assumptions made herein. The analysis approach described here is still valid, however.

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FOREWORD

When energy and material resources are extracted, processed, converted, and used, the related pollutional impacts on our environment and even on our health often require that new and increasingly more efficient pollution control methods be used. The Industrial Environmental Research Laboratory-Cincinnati (IERL-Ci) assists in developing and demonstrating new and improved methodologies that will meet these needs both efficiently and economically.

Clean, renewable energy sources such as solar energy are being investigated for their potential contributions in pollution control and energy conservation. Several federal agencies, including the Environmental Protection Agency (EPA), are involved in various aspects of this work.

This report presents the results of a survey of EPA facilities, the objective of which was to identify those facilities where solar energy systems might best be installed. A preferred first site and system are also recommended for the U.S. Department of Energy's Federal Solar Buildings Program. The Office of Research and Development will use the results of this report to help identify other facilities for solar system installation. The Alternate Energy Sources Branch of the Energy Pollution Control Division should be contacted for further information.

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Director
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ABSTRACT

A study was done to assess the feasibility of applying solar thermal energy systems to EPA facilities. A survey was conducted to determine those EPA facilities where solar thermal energy could best be used. The most broadly applicable solar energy systems were identified for these facilities. These systems were optimized for each specific application and the system/facility combinations were ranked on the basis of greatest cost effectiveness. The Robert S. Kerr Environmental Research Laboratory in Ada, Oklahoma (OK) was selected as a preferred first site for the U.S. Department of Energy's Federal Solar Buildings Program. This report was submitted in fulfillment of Contract No. 68-03-2567 by Acurex Corporation under the sponsorship of the U.S. Environmental Protection Agency. This report covers a period from November 1, 1977 to June 30, 1978, and work was completed as of December 21, 1979.

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SECTION 1

SUMMARY

This report presents the results of a study carried out by Acurex for the U.S. Environmental Protection Agency (EPA) to assess the feasibility of applying solar thermal energy systems to existing EPA facilities and to select an EPA facility suitable for the Department of Energy's Federal Solar Buildings program. A survey of potential sites included all facilities which were owned or leased by EPA. Various solar heating and cooling system concepts for the facilities were considered and compared, so that the most broadly applicable system type could be identified. A more detailed system design analysis also was carried out for the EPA facility which has the best potential for near-term applications.

The reader is cautioned that advances in solar technology and increased energy costs since the writing of this report (June 1978) have negated some of the assumptions made herein. The analysis approach described here is still valid, however.

The study was undertaken in three phases. In Phase I, EPA facilities were surveyed to determine their relative suitability in terms of:

- Building thermal energy loads
- Local costs of conventional energy sources
- Weather patterns
- Availability of adequate area for solar collectors
- Energy conservation potential
- Other constraints

In Phase II, the facility survey information was combined with an assessment of current solar technology to determine which systems were most appropriate for each facility. Each system/facility pair was analyzed for performance, cost, energy displacement and environmental impact; a ranking of the pairs was made on the basis of these parameters. In Phase III, a detailed design analysis was performed for a space heating system at the Ada, Oklahoma (OK) facility. The system was optimized, sized, and costed, and a determination was made of the fossil fuel saved by operating the solar energy system. The approach used and the results obtained in each phase are summarized below. The summary is followed by a description of the organization of this final report and the content of each section.

PHASE I -- SURVEY OF EPA FACILITIES

The facility survey was carried out in four steps:

1. Identification of candidate facilities
2. Development of initial screening criteria

3. Initial screening
4. Development of final selection criteria and collection of relevant data

In the first step of Phase I, 176 EPA facilities were identified; most of these were immediately eliminated because they were not owned by EPA. Eleven laboratory facilities were selected as candidates for the initial screening. The second step, development of the initial screening criteria, provided the following bases for selection:

- Ample area for the solar collectors (free from shading throughout the year)
- Good insolation (solar radiation), with an arbitrary minimum set at 300 langleys/day annual average
- Heating Ventilating and Air Conditioning (HVAC) and hot water systems suitable for easy interface with solar energy system

In the initial screening process, these criteria were applied to information gathered directly from facility personnel for each candidate site and from standard weather and insolation data for the United States. The results were weighed and totaled to give a relative ranking, and the top six facilities were selected for the next step of the survey. These six were:

- Robert S. Kerr Environmental Research Laboratory (ERL), Ada, OK
- South East ERL, Athens, Georgia (GA)
- ERL, Narragansett, Rhode Island (RI)
- National Water Quality Laboratory, Duluth, Minnesota (MN)
- Large Lakes Research Station, Grosse Ile, Michigan (MI)
- Central Regional Laboratory, Manchester, Washington (WA)

Further data were collected at these facilities while final selection criteria were being developed. The final criteria used in ranking system/facility combinations were, in order of their relative importance:

- Sufficient collector area to supply a significant portion of the facility energy load
- Relative cost effectiveness
- Significant fossil fuel displacement
- Minimal environmental effects

Data were collected in this step primarily from detailed questionnaires sent to the facilities and through visits to the three most promising sites. The questionnaires covered all salient features of the HVAC systems, building energy loads, and available space at each site. After receipt of this information, the facilities at Ada, OK, Athens, GA and Narragansett, RI were visited by Acurex personnel to clarify the data and evaluate system interfaces in detail. The facility survey phase was completed with reduction of the gathered data to obtain building load profiles, and evaluation of annual insolation profiles for each site. In Phase II, the load and insolation profiles were used in cost analyses of various system concepts.

PHASE II -- SOLAR ENERGY SYSTEM CONCEPT IDENTIFICATION

In this phase, specific solar energy systems were matched with each facility to construct a system/facility matrix. The candidate solar heating and cooling system types identified in the matrix were:

- Domestic Hot Water (DHW)
- Space heating and DHW
- Space cooling
- Space heating, space cooling and DHW

Two state-of-the-art solar collector types were considered for the cost/performance analyses: a single-glazed flat-plate collector, and a line-focusing parabolic trough concentrating collector. Both single- and dual-stage absorption chillers were considered for systems using space cooling. Conceptual system schematics were developed for each system type. The preliminary system/facility matrix was developed by applying the above components to the six final candidate facilities and eliminating combinations which had obvious cost/performance disadvantages.

A benefit analysis was then performed for each appropriate system/facility pair to permit ranking of the options. The benefit analysis consisted of three parts:

- Cost/Benefit Analysis
- Energy displacement analysis
- Environmental effects assessment

Heavy emphasis was placed on the Cost/Benefit Analysis, since economics are the most important factor in virtually any solar energy system selection.

The Cost/Benefit Analysis was directed at determining the optimum system size (e.g., minimum life-cycle cost) for each system/facility pair. This was done by calculating, for various collector field sizes:

- Annual collected energy
- Portion of load supplied
- Annualized cost
- Specific cost (cost per unit energy delivered)

The first step in the Cost/Benefit Analysis was a compilation of candidate system costs. This was done by defining component costs using currently available hardware and installation costs based on recent Acurex experience with solar energy systems. The installed cost for each component was separated into fixed costs and collector area-dependent costs, which grow linearly with the collector field size. Sample system costs are given in Table 1.

The annualized cost for each system and collector field size was calculated in the analysis using conventional economic modeling. The

TABLE 1. SAMPLE SYSTEM COSTS

System	Component costs		System cost/collector area, $\$/m^2$ ($\$/ft_c^2$)	
	Area dependent $\$/m^2$ ($\$/ft_c^2$)	Fixed \$	for 930 m^2 field (10,000 ft_c^2)	for 3720 m^2 field (40,000 ft_c^2)
DHW-flat-plate	341 (31.66)	66,000	412 (38.26)	358 (33.31)
Space heating-concentrator (steam loop)	354 (32.89)	86,000	446 (41.49)	377 (35.04)
Space cooling-concentrator (steam loop, 1-stage chiller)	354 (32.89)	132,000 150,000	496 (46.09)	394 (36.64)
Space cooling-concentrator (steam loop, 2-stage chiller)	394 (36.65)	244,000	657 (61.05)	460 (42.75)

annual performance (net energy delivered) of each system/field size was determined on the basis of the load and insolation profiles developed in Phase I. The specific cost (annualized cost divided by annual energy delivered) for each system/field size was then plotted as a function of the solar substitution (portion of load provided by solar energy) to yield a curve with a minimum. Figure 1 gives an example of the curves thus determined for the Ada, OK site. The minimum cost system (i.e., optimum system size and solar substitution) for each system type and facility is given in Table 2. The conclusions drawn from these results are:

- DHW generally yields the highest solar substitution at the lowest specific cost for a given site
- The lowest specific cost for systems other than DHW was achieved by the Ada, OK heating system
- The highest optimum solar substitutions for systems other than DHW were achieved by the Ada, OK combined heating and cooling system

At this point a final system/facility matrix was constructed, with the most cost-effective system or systems selected for each facility. This matrix (Table 3) provided the basis for the remaining selections and ranking.

The optimum systems defined in the Cost/Benefit Analysis were further analyzed, in the second part of the benefit analysis, to determine the amount of fossil fuel which would be displaced each year due to the operation of a given system. This analysis was made on the basis of oil and natural gas displaced directly at the site and at central electric power generating plants.

In the third part of the benefit analysis, an assessment was made of the environmental effects associated with construction and operation of the various systems. The environmental impact areas addressed here were:

- Topography
- Geology and soils
- Hydrology and water quality
- Climate and air quality
- Flora and fauna
- Resource use
- Aesthetics
- Safety

The energy displacement analysis and environmental assessment were used along with the minimum cost analysis to rank the system/facility pairs in the final matrix. The ranking, which emphasizes cost effectiveness, is shown in Table 4. The DHW systems are the most attractive in terms of the final selection criteria. Thus, DHW was identified as the best generic system with broad applicability to the EPA facilities. For the purposes of Phase III, however, a combined DHW and space heating system at the Ada, OK facility was selected for further analysis. Such a combined system has broad application throughout the

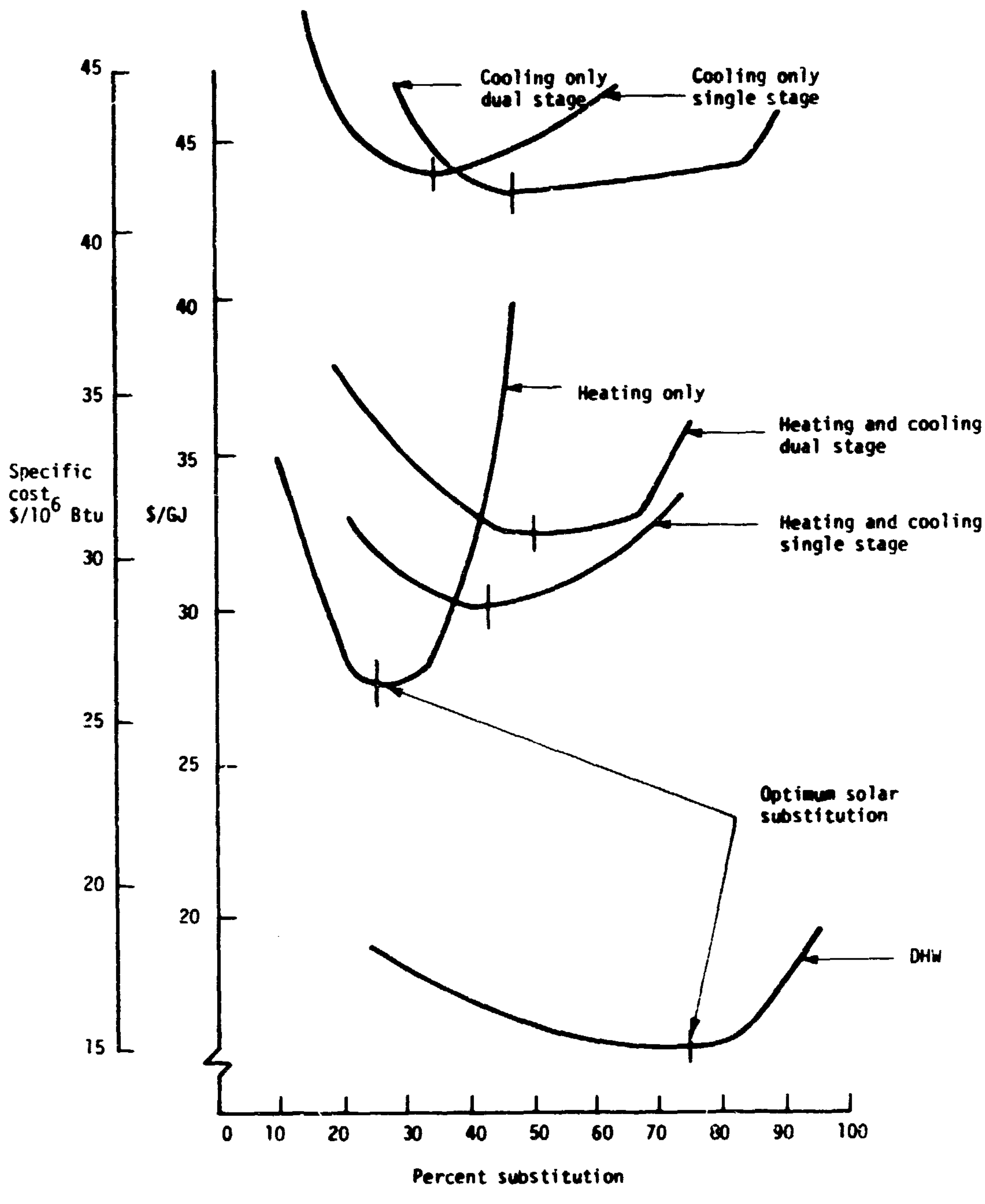

Figure 1. Results of Cost/Benefit Analysis for Ada, OK facility.

TABLE 2. SUMMARY OF RESULTS OF COST/BENEFIT ANALYSIS

Site	System	Optimum solar fraction (percent)	Specific cost $/GJ ($/10^6 Btu)	Collector area m^2 (ft^2)	Total system cost ($1000)
Ada, OK	Heating and cooling	44	27.1 (28.6)	4,650 (50,000)	1,790
	Heating only	28	24.9 (26.3)	1,400 (15,000)	579
	DHW	74	14.3 (15.1)	1,670 (18,000)	636
Narragansett, RI	Heating only	42	27.4 (28.9)	3,720 (40,000)	1,410
	DHW	63	22.7 (23.9)	1,300 (14,000)	509
Athens, GA	DHW	79	16.1 (17.0)	2,050 (22,000)	762
Duluth, MN	DHW	60	22.8 (24.0)	1,490 (16,000)	572
Grosse Ile, MI	DHW	54	24.7 (26.1)	1,300 (14,000)	509
Manchester, WA	DHW	51	27.7 (29.2)	1,300 (14,000)	509

TABLE 3. FINAL SYSTEM/FACILITY MATRIX

System / Facility	DHW flat-plate	Space heating and DHW		Space cooling		Heating, cooling and DHW	
		Flat-plate	Concentrator	Flat-plate	Concentrator	Flat-plate	Concentrator
Ada, OK	Yes • Separate DHW system		Yes • 184 kPa (12 psig) steam to heat DHW		Yes • Hot water to drive chiller		Yes • 184 kPa (12 psig) steam • Interface at 2 points for space heating, 1 point for space cooling
Athens, GA	Yes • Separate DHW system			No • Limited unobstructed space • HVAC modification most cost-effective first step			
Narragansett, RI	Yes • Separate DHW system			No • Low insolation • Limited unobstructed space			
Duluth, MN	Yes • Separate DHW system	No • Poor load match • Low insolation				No • Low insolation • Small cooling load	
Grosse Ile, MI	Yes • Separate DHW system	N/A				No • Low insolation • Small cooling load • No chilled water loop	
Manchester, WA	Yes • Separate DHW system	N/A				No • Low insolation	

TABLE 4. SYSTEM/FACILITY RANKING

		Criteria						
		Cost effectiveness		Fossil fuel displacement				
Site	System	($/GJ)	Points[a] (75 Possible)	(GJ)	Points[a] (15 possible)	Environmental effects points (10 possible)	Total points (100 possible)	Rank
Ada, OK	Space heating and cooling	27.1	3	7,415	15	0	18	7
	Space heating	24.9	15.5	3,080	2	5	22.5	6
	DHW	14.3	75	5,890	10.5	10	95.5	1
Narragansett, RI	DHW	22.7	28.5	2,980	1.5	10	40	4
Athens, GA	DHW	16.0	65	6,280	11.5	10	86.5	2
Duluth, MN	DHW	22.8	31	3,340	2.5	10	43.5	3
Grosse Ile, MI	DHW	24.7	16.5	2,730	1	10	27.5	5
Manchester, MA	DHW	27.7	0	2,440	0	10	10	8

[a]Rounded to the nearest 1/2.

United States and yet has unique characteristics relative to existing solar energy systems.

PHASE III -- DETAILED ANALYSIS

In Phase III, a design analysis was performed for a combined heating and DHW system for the Ada, OK facility. The system selected in Phase II was a concentrating collector field operating with pressurized water and using a flash boiler/storage tank to provide steam to the existing HVAC and DHW system. This steam-generating system was selected because the cost of modifying the existing 12 psig steam system at the Ada, OK facility (to interface with a pressurized water solar system) would be prohibitive. A schematic of the selected system is shown in Figure 2.

The optimization procedure utilized computer performance simulations to determine the best collector field size, storage volume, maximum operating pressure and field flow rate. Hourly simulations for seven-day periods, every other month, were run in order to fully assess the effects of weather variations on system performance. Weather data tapes were used for insolation data, and building energy load calculations were used to specify the system performance requirements. A sequential variation of key parameters led to an optimum selection for each one. The characteristics of the optimized design are summarized in Table 5.

SUMMARY OF REPORT ORGANIZATION

The following sections present the results of this study in detail. In Section 2, the mechanics of the facility survey and selection process are described. The benefit analysis carried out in Phase II is presented in Section 3, along with the rationale for ranking system/facility pairs and for selecting the best generic system and the best system for detailed analysis. The design analysis for the Ada, OK DHW and space heating system is summarized in Section 4. Appendix A presents a detailed description of the design cost-optimization computer procedure. Appendix B describes current technology of the various components of solar energy systems. Appendix C discusses typical systems applicable to building heating, cooling, and DHW.

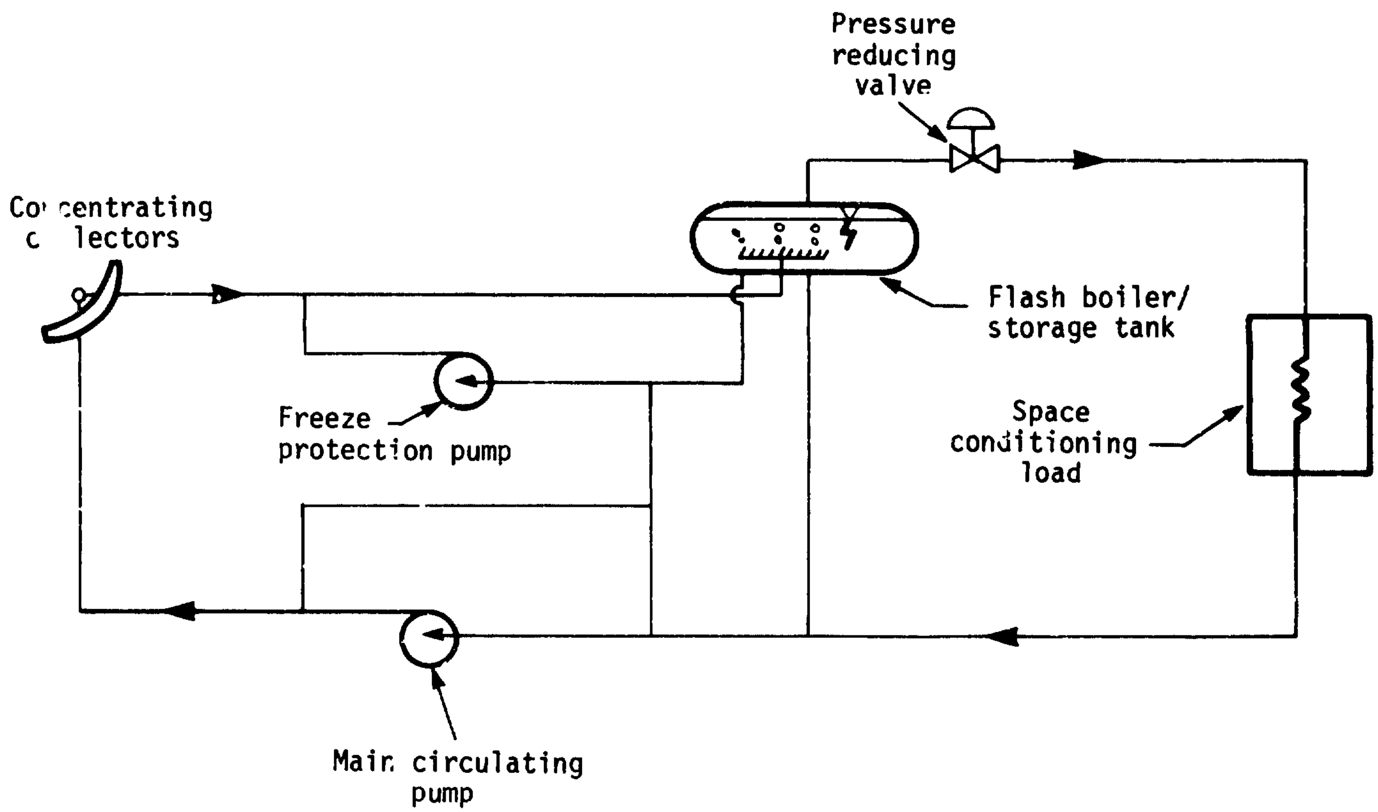

Figure 2. Schematic of solar steam space heating system for Ada, OK.

TABLE 5. COLLECTOR AND STORAGE SUBSYSTEM SUMMARY

Collector field	
Type	Parabolic trough
Orientation	North-South
Row spacing	4.57 m (15 ft.) center to center
Total aperture area	1561 m^2 (16,794 ft^2)
Collector modules	36
Fluid flow rate	363 l/min (96 gpm)
Maximum temperature	452 K (354° F)
Maximum pressure	979 kPa (142 psia)
Boiler storage	
Type	Varying pressure steam accumulator
Tank volume	139,000 l (36,800 gal)
Maximum temperature	441 K (334° F)
Maximum pressure	751 kPa (109 psia)
Energy storage capacity	18.7 GJ (17.7 x 10^6 Btu)

SECTION 2

FACILITY SURVEY

A primary objective of this study is to determine which of the existing EPA facilities is best suited for the application of solar energy. A survey of the facilities was carried out in four steps:

1. Identification of candidate facilities
2. Development of initial screening criteria
3. Initial screening
4. Final criteria and data collection

The mechanics of the selection process, including development of the screening criteria, are described in this section.

IDENTIFICATION OF CANDIDATE FACILITIES

A total of 176 EPA facilities were originally identified with the aid of the EPA Facilities Branch in Washington, D.C. Most of the facilities were eliminated from consideration because they were leased by EPA. This was used as an elimination factor because of the potential problems involved with installing government-owned equipment in leased buildings according to the appropriate federal solar program. Only government-owned facilities meet the legal requirements to provide fee-simple title to the building under the terms of the federal program, "Demonstration of Solar Heating and Cooling in Federal Buildings", federal bill H.R. 8444, Sec. 721. This restriction is imposed, in part, to satisfy current requirements of the Economy Act (Sec. 322 of the Act, 40 U.S.C. 728a), which limits the scope of improvements or additions to property leased to the government.

Several of the remaining facilities also were eliminated due to special circumstances. For example, a special case concerning compliance with H.R. 8444 was presented by the new EPA Test and Evaluation Pilot Plant in Cincinnati. This facility construction is legally classified as a temporary building, and is on city-owned land under a cooperative use agreement with the city of Cincinnati. The agreement prevents erection of a permanent building on the leased land, and the interpretation was made that the requirements of H.R. 8444 could not be met.

Another facility, the Bear's Bluff Field Station at St. John's Island, South Carolina (SC), was removed from consideration when the EPA Facilities Branch decided that the facility is too small. A list of 11

EPA-owned laboratory facilities, given in Table 6, was identified for the preliminary screening.

TABLE 6. EPA LABORATORY FACILITIES INITIAL SCREENING MATRIX

Laboratory	Location
ERL	Narragansett, RI
Central Regional Laboratory	Edison, NJ
South East ERL	Athens, GA
Gulf Breeze ERL	Gulf Breeze, FL
National Water Quality Laboratory	Duluth, MN
Large Lakes Research Station	Grosse Ile, MI
Fish Toxicology Station	Newtown, OH
National Environmental Research Center	Cincinnati, OH
Robert S. Kerr ERL	Ada, OK
ERL	Corvallis, OR
Central Regional Laboratory	Manchester, WA

INITIAL SCREENING CRITERIA

Initial screening criteria were developed to determine the best facilities for solar energy system installations. The criteria, in order of relative importance, were:

1. Ample area for collector site
2. Good insolation; minimum of 300 langleys/day annual average (1 langley equals 1 cal/cm^2 or 3.69 Btu/ft^2)
3. HVAC components suitable for interface

The following subsections discuss these criteria.

Collector Site Area

In screening candidate sites for possible solar installations the availability of adequate space for the collector site is essential. Ample, unobstructed space, close to the facility, is needed to facilitate hook-up to the HVAC and water heating systems. A roof cluttered with exhaust fans, vents and/or gas-fired package units will probably not provide an adequate collector site area. The area also must be virtually free from shading all year round.

Insolation

Selection of an attractive solar energy demonstration site depends on the availability of adequate insolation. Insolation must be sufficient for the solar system to contribute a substantial portion of energy during the year and therefore, be relatively cost-effective. A value of 300 langleys/day (on a horizontal surface) was arbitrarily selected as the minimum annual mean daily insolation level for a site. More than 90 percent of the United States receives over 300 langleys/day, and portions such as the "Sun Belt" receive over 450 langleys/day. Only portions of the Northeast and Northwest average less than 300 langleys/day.

Ease of Interface With Solar

The final criterion in the preliminary screening is ease of interface with a solar energy system. For the loads being considered, the interfacing is greatly simplified if the facility has centralized systems, such as a central DHW system, a hot water or steam loop for space heating, and a chilled water loop for space cooling.

Central DHW--

To tie into a DHW system, it is far less costly to choose a facility which has a central system with hot water piping already distributed throughout the building, rather then several individual hot water heaters at different locations. The greatest cost savings is in the elimination of additional piping and/or heat exchangers.

Hot Water/Steam Loop--

Any system with hot water or steam distributed throughout the building is preferable, again because all distribution piping and air-to-water (or steam) heat exchangers are already in place. If a building is heated by a number of roof-mounted, gas-fired air heaters, then an entire new heating system (piping, heat exchangers, and storage tank) would have to be installed. Hot water loops are generally preferable to steam loops for space heating because it is easier and less costly to supply hot water directly from the collector to the loop than it is to generate steam.

Chilled Water Loop--

For interface with an existing space cooling system, chilled water loops are preferable to other systems (e.g., direct refrigerant-to-air units), again because the distribution system is already in place.

INITIAL SCREENING

Insolation data from References 1 and 2 and telephone conversations with individual laboratory facility managers were used to collect the information for the initial screening.

Collector Siting Area

Information on each facility's available area for collectors was obtained from plot plans of the facility and from the telephone conversations with the facility managers. According to these sources, it was determined that the only facility which definitely does not have sufficient area to site collectors is the Cincinnati, OH site. That facility is a nine-story building with approximately 175,000 square feet of floor area. In order to supply a minimal portion of the building's load, say 20 percent, it was estimated that over 1 acre of unobstructed area, to the south and directly adjacent to the building, would be required for the collectors.

Insolation

Information on available insolation for each site was obtained from References 1 and 2. Figure 3 shows the geographical location and annual mean daily insolation on a horizontal surface for each facility. The facility locations are indicated by a star (*). Where the reference did not give insolation values for a specific site, the value from the closest city was taken. For example, Ada, OK, was assumed to receive 436 langleys/day, the value for Oklahoma City. It should be noted that, due to local weather patterns, the lines of constant insolation do not always correctly represent local conditions. For example, although Chicago and New York City both lie very close to the 350 insolation line, their measured values are less than 300 langleys/day. This is often the case for locations which lie near a large body of water or have significant air pollution.

Table 7 presents the approximate annual mean daily insolation values taken for each site for use in the preliminary screening. For sites for which no insolation data were given in the references, the city whose data were used is listed, as well as the approximate distance from that city to the actual site.

As can be seen from the table, of the candidate sites, only Edison, NJ receives less than 300 langleys/day, but seven of the sites receive less than an annual average of 350 langleys/per day. These are: Duluth, MN, Cincinnati, OH, Corvallis, OR, Narragansett, RI, Grosse Ile, MI, Newtown, OH, and Manchester, WA. Ada, OK, Gulf Breeze, FL, and Athens, GA, receive the greatest insolation.

In addition to the annual daily insolation, the fraction of the total incident radiation which is direct is also important. This can have a major impact on the choice of the collector type. Since all but non-concentrating and low-concentration collectors depend almost entirely

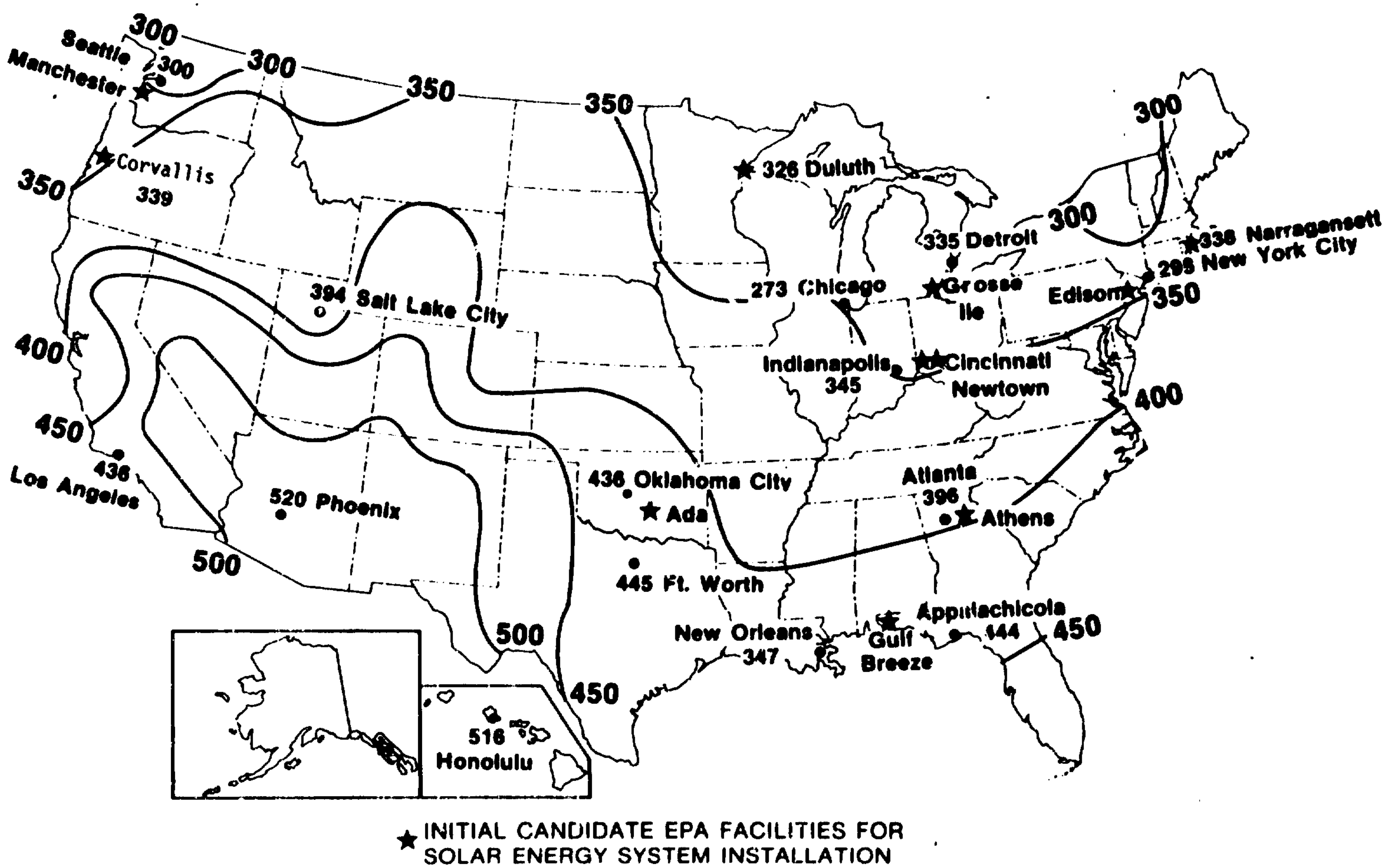

Figure 3. Geographic location and annual mean daily insolation of initial candidate EPA facilities.

TABLE 7. APPROXIMATE ANNUAL MEAN DAILY INSOLATION ON A HORIZONTAL SURFACE AND SITE LOCATIONS

Site (in order of decreasing insolation)	Nearest data source/distance from site (km)	Insolation band of site (langleys/day)	Insolation of nearest data location (langleys/day)
Gulf Breeze, FL	Apalachicola/193 New Orleans/274	400 to 450	444 347
Ada, OK	Oklahoma City/129	400 to 450	436
Athens, GA	Atlanta/96	350 to 400	396
Newtown, OH	Indianapolis/153	300 to 350	345
Cincinnati, OH	Indianapolis/153	300 to 350	345
Corvallis, OR	--	300 to 350	339
Narragansett, RI	Newport/40	300 to 350	338
Grosse Ile, MI	Detroit/16	300 to 350	335
Duluth, MN	--	300 to 350	326
Manchester, WA	Seattle/16	300 to 350	300
Edison, NJ	New York City/40	300 to 350	298

upon direct radiation, a greater direct fraction yields greater flexibility in selecting collectors. Areas such as Manchester, WA, would have to rely heavily upon diffuse radiation during most of the year and, therefore, would not be as attractive for the higher temperature applications.

Ease of Interface With Solar

The only facilities which do not have central DHW systems are Gulf Breeze, FL and Edison, NJ. From the telephone survey it was learned that all sites except Gulf Breeze, FL, Ada, OK, Corvallis, OR and Edison, NJ have hot water loops for space heating. Gulf Breeze, FL does not have a hydronic loop for space heating. The other three sites have steam loops, although Edison's steam is piped in by General Services Administration (GSA). Finally, it was learned that only Newtown, OH does not have a chilled water loop for space cooling.

Results of Initial Screening

The actual screening was carried out using a point system. For each of the three main criteria mentioned, a maximum of 10 points was awarded. However, a weighting factor of 2 was then used to give the most important criterion, collector site area, more weight.

Since ample collector site area must be available, the full 20 points were assigned to those facilities which had enough area; zero points were awarded to thos which did not. As mentioned in Section 3, Cincinnati, OH was the only facility which was judged to have an insufficient site area.

Points were awarded for annual insolation on the following basis:

- Insolation <300 langleys/day -- 0 points
- 300$\leq$ insolation <350 langleys/day -- 5 points
- 350$\leq$ insolation <400 langleys/day -- 8 points
- Insolation >400 langleys/day -- 10 points

For the initial screening, it was decided that a facility should have all three mechanical systems (central DHW, hydronic loop for space heating, chilled water loop for cooling) in order to satisfy the interface criterion. This would yield the most flexibility in designing a solar system for the facility. Therefore, no points were awarded to a facility if it did not have all three systems. Since a hot water loop would be slightly preferable to a steam loop for a space heating interface, the maximum of 10 points was assigned to facilities with hot water loops, and 5 points were assigned to facilities with steam distribution loops.

Table 8 reviews the information gathered in the preliminary survey. Table 9 presents the screening results and gives the point totals for the facilities. Facilities which received more than 75 percent of the possible 40 total points were selected for the next phase of the survey. These facilities were:

TABLE 8. SURVEY RESULTS

Site	Adequate collector site area	Approximate mean daily insolation (langleys/day)	Ease of interface			
			Central DHW	Hydronic loop for space heating		Chilled water loop
				Hot water	Steam	
Narragansett, RI	Yes	338	Yes	Yes		Yes
Edison, NJ	Yes	298	No		Yes	Yes
Athens, GA	Yes	396	Yes	Yes		Yes
Gulf Breeze, FL	Yes	444	No			Yes
Duluth, MN	Yes	326	Yes	Yes		Yes
Grosse Ile, MI	Yes	335	Yes	Yes		Yes
Newtown, OH	Yes	345	Yes	Yes		No
Cincinnati, OH	No	345	Yes	Yes		Yes
Ada, OK	Yes	436	Yes		Yes	Yes
Corvallis, OR	Yes	339	Yes		Yes	Yes
Manchester, WA	Yes	300	Yes	Yes		Yes

TABLE 9. SCREENING RESULTS

Criterion / Site	Collector site area (weighted) Adequate - 20 pts Inadequate - 0 pts	Insulation $x \geq 400$ langleys/day - 10 pts $350 \leq x < 400$ - 8 pts $300 \leq x < 350$ - 5 pts $x < 300$ - 0 pts	Ease of interface All three systems - hot water - 10 pts All three systems - steam - 5 pts Two or fewer systems - 0 pts	Total (40 possible)
Narragansett, RI	20	5	10	35
Edison, NJ	20	0	0	20
Athens, GA	20	8	10	38
Gulf Breeze, FL	20	10	0	30
Duluth, MN	20	5	10	35
Grosse Ile, MI	20	5	10	35
Newtown, OH	20	5	0	25
Cincinnati, OH	0	5	10	15
Ada, OK	20	10	5	35
Corvallis, OR	20	5	5	30
Manchester, WA	20	5	10	35

- Ada, OK
- Athens, GA
- Narragansett, RI
- Duluth, MN
- Grosse Ile, MI
- Manchester, WA

FINAL CRITERIA AND DATA COLLECTION

This section presents the final screening criteria and describes the procedure used to collect the facility and insolation data necessary for input to the Benefit Analysis (Section 3).

Final Criteria

The final screening criteria used in ranking system/facility combinations were, in order of their relative importance:

1. Adequate collector site area to supply a significant portion of the load
2. Relative cost effectiveness
3. Significant fossil fuel displacement
4. Minimal environmental effects

Of these criteria, the first two are by far the most important from an engineering viewpoint. The remaining criteria, although significant in terms of environmental impact were given less weight in ranking the remaining six facilities, as indicated by the weighting system described above.

Site Area--

The importance of collector site area as a criterion for evaluating candidate sites for installation of solar systems was discussed above. It remains the most important criterion for establishing the Final System/Facility Matrix.

Cost Effectiveness--

Relative cost effectiveness was the second most important criterion. For a given group of systems, the one that delivers the most energy per dollar is the most attractive.

Fossil Fuel Displacement--

One of the benefits of installing a solar energy system at a facility is the decrease in the facility's dependence upon fossil fuel. It is important to note that when the energy form being displaced is electricity, the amount of fossil fuel actually displaced at the power plant must be determined.

Environmental Effects--

The effects upon the environment of the solar energy systems considered here are expected to be very small. Furthermore, the variation of environmental effects from system to system is also small.

Consequently, this criterion was given the least weight of the four final criteria.

Data Collection

To obtain the specific facility and load information required to apply the above criteria, a detailed questionnaire and a copy of the facility plot plan were sent to each of the six remaining facilities. Detailed weather data, required for the benefit analysis, were gathered from References 2 and 3.

Information requested in the questionnaire included:

- Available ground and roof area indicated on plot plan
- Building codes which might restrict construction
- Space in mechanical (or similar) room
- Load characteristics
- HVAC system details

The questionnaires were answered and returned for all facilities except Manchester, WA.

Available Area--

The facility manager was requested to indicate on an enclosed plot plan if his facility had an area which was clear (or could be cleared) of obstructions, and which could be used for the collector site. He was asked to indicate only those areas which were free from shading during the entire year.

Building Codes--

Detailed information on any building codes which might restrict the installation of a solar energy system at the facility was requested in the questionnaire.

Space in Mechanical Room--

The facility manager was asked to indicate how much space was available for the placement of solar-related equipment, such as chillers, pumps, instrumentation and storage vessels.

Load Characteristics--

In order to adequately model solar energy system performance, detailed information on annual building load profiles was needed. The facilities managers were asked to supply copies or summaries of their monthly fuel and utility bills for the past 2 years. They were also asked to estimate the monthly variation of each building load (i.e., DHW, process and space heating, and space cooling) for each fuel source.

HVAC System Details--

Information requested on HVAC Systems included the following:

- Boiler design capacities
- Steam or hot water loop temperatures and pressures

- Chiller design capacities
- Chilled water supply and return temperatures

The facility managers were also asked to supply information on system operations. This information was particularly useful in determining the three most attractive sites for solar energy installations from an interface standpoint.

Based upon the above information and preliminary calculations for the performance of potential systems, the three most promising facilities for applications other than DHW were selected for site visits.

Site Visits

The purpose of the site visits was to determine the possible interface points at each facility, HVAC system condition and HVAC system operation. The three facilities selected for site visits, on the basis of questionnaire responses, were Ada, OK, Athens, GA, and Narragansett, RI. All three were shown to have significant heating and cooling loads, as well as DHW demands. Duluth, MN and Grosse Ile, MI had very low cooling loads, and construction on the Manchester, WA site had not yet begun. The activities at each site included:

- Verification of questionnaire responses
- Location of key HVAC components
- Determination of interface points with solar
- Determination of potential collector sites
- Determination of local EPA support

Verification of questionnaire responses was important, especially on those questions dealing with loads and HVAC system characteristics. The evaluation team met with the facility managers and all other key personnel involved in the maintenance and operation of the mechanical systems in order to gain a thorough understanding of each system.

The team was also taken on extensive tours of the facilities in order to determine the location of the key HVAC components and potential solar energy system interface points. The tours also afforded the opportunity to evaluate the overall mechanical systems for efficiency and maintenance.

The potential collector site area was also best evaluated in person. In many cases, areas thought by the facility managers to be suitable for siting turned out to be unacceptable due to excessive shade or poor location relative to the interface.

Finally, local support was considered to be important from the standpoint of public visibility. Members of the evaluation team met with the key administrative personnel at each laboratory to determine local support of a potential solar energy installation.

Summary of Results

This section summarizes the data collected for input to the Cost/Benefit Analysis. This information consists primarily of insolation and load data.

Insolation Data--

Figures 4 through 9 present annual insolation profiles for each of the six sites. For each site, both monthly average daily total, $\bar{H}$, and direct, $\bar{H}_b$, radiation are given. The data were taken from Reference 2 which lists insolation data for a large number of cities. Although it does not list data for all six sites considered here, insolation data is available for nearby cities. Reference 2 does have data for Duluth, MN. The cities whose data were used for the remaining five facilities in the above figures are:

- Oklahoma City, for Ada, OK
- Atlanta, for Athens, GA
- Newport, for Narragansett, RI
- Detroit, for Grosse Ile, MI
- Seattle, for Manchester, WA

For applications requiring average collector fluid operating temperatures above 339 K (150° F), concentrating collectors have the cost/performance advantage over flat-plates. Since concentrating collectors use primarily the beam component of solar radiation, Figures 10 and 11 show a comparison of the variation of monthly average daily direct radiation over the year for the six sites. The six insolation profiles are separated into two plots for readability. The three sites visited are represented in Figure 10, with the remaining three sites represented in Figure 11.

As expected, Ada, OK shows the highest direct insolation throughout the year. This is especially important for a site when considering a solar cooling system, or combined heating and cooling system. Duluth, MN, Grosse Ile, MI, and Manchester, WA, have the lowest winter direct insolation, while Narragansett, RI, and Athens, GA, have the lowest summer direct.

Also of interest in determining collector and, therefore, system performance is the local ambient temperature while the collector is operating. Figures 12 and 13 present monthly average daytime ambient temperature profiles for the six sites (Reference 3). As expected, Ada, OK, and Athens, GA, show the highest ambient temperatures throughout the year. Duluth, MN, and Grosse Ile, MI, have the lowest winter ambient temperatures, and Manchester, WA, has the lowest summer temperatures.

Building Loads--

Building load information was difficult to obtain. In most areas, only total natural gas, fuel oil or electricity consumption data were available on a monthly basis. In order to determine the actual portion of a particular fuel which was consumed for a particular load, the facility

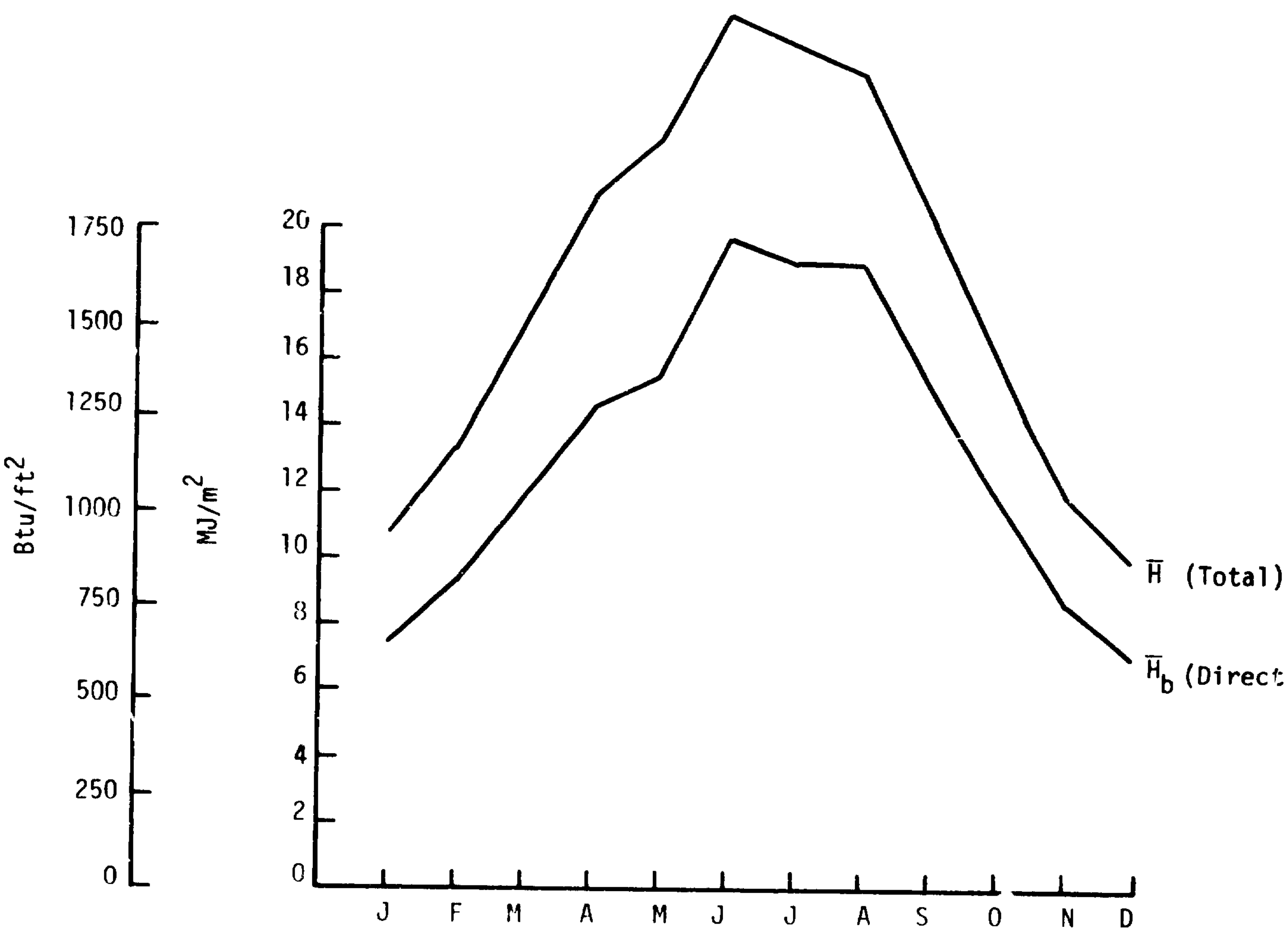

Figure 4. Monthly average daily insolation, Ada, OK.

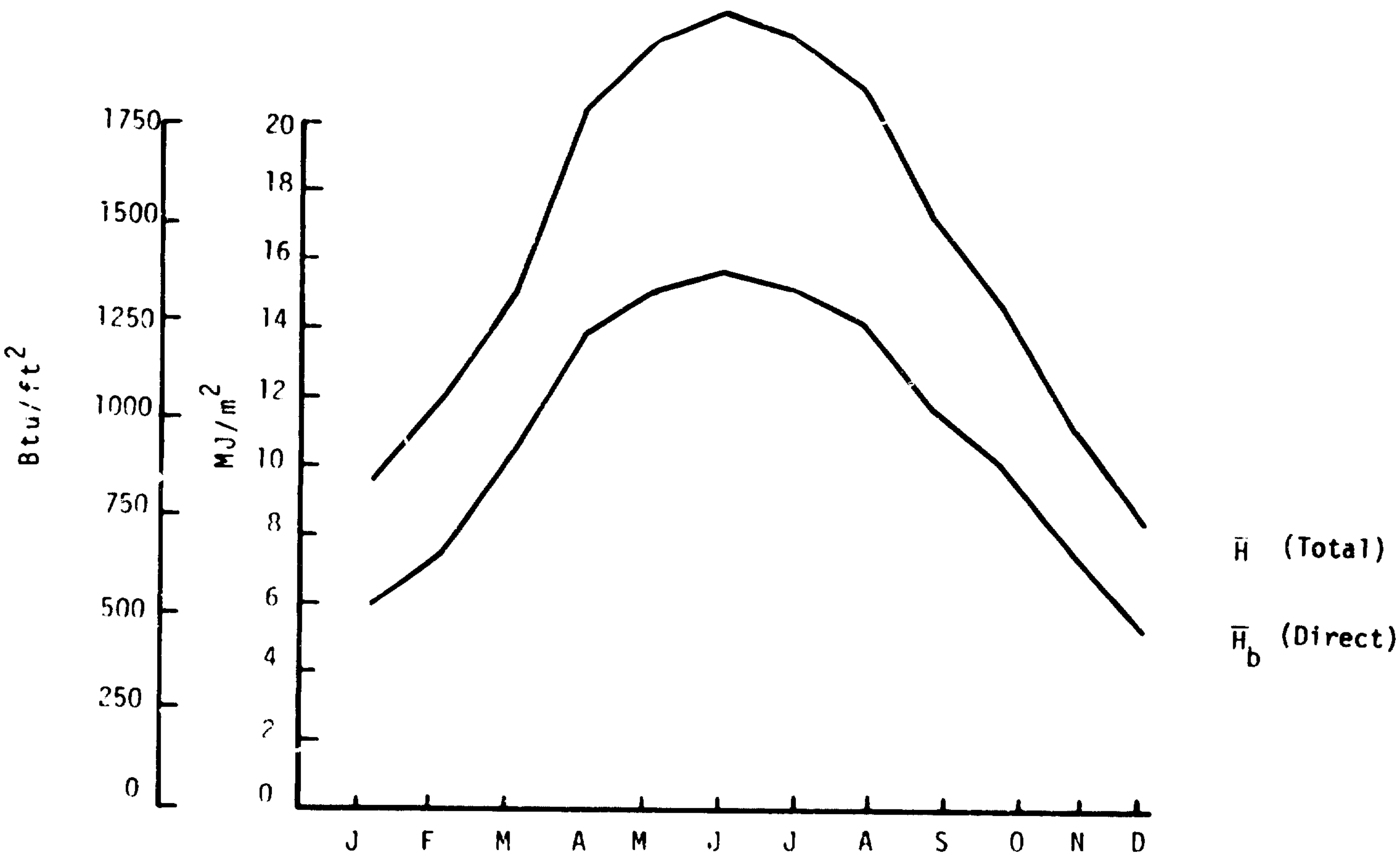

Figure 5. Monthly average daily insolation, Athens, GA.

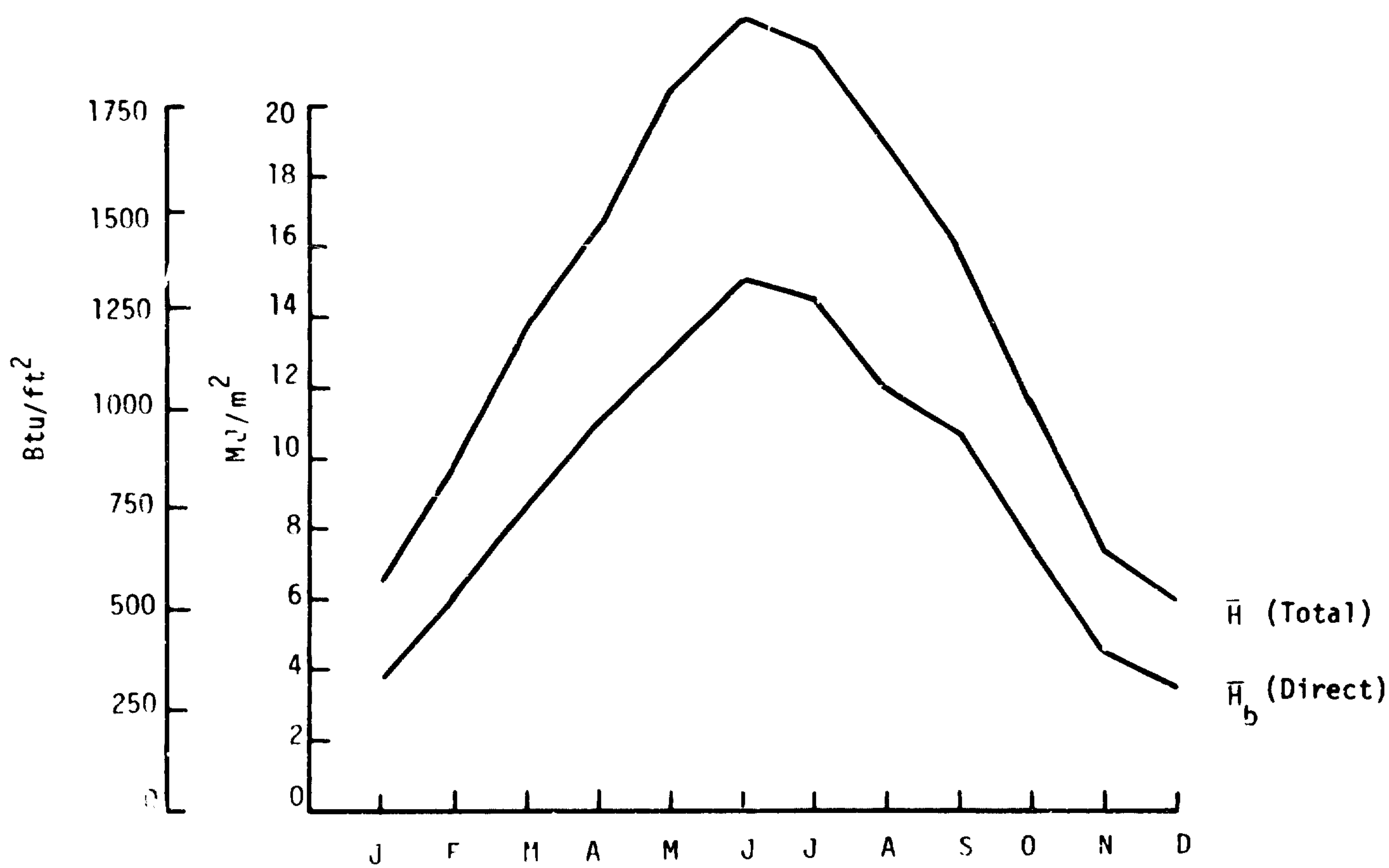

Figure 6. Monthly average daily insolation, Narragansett, RI.

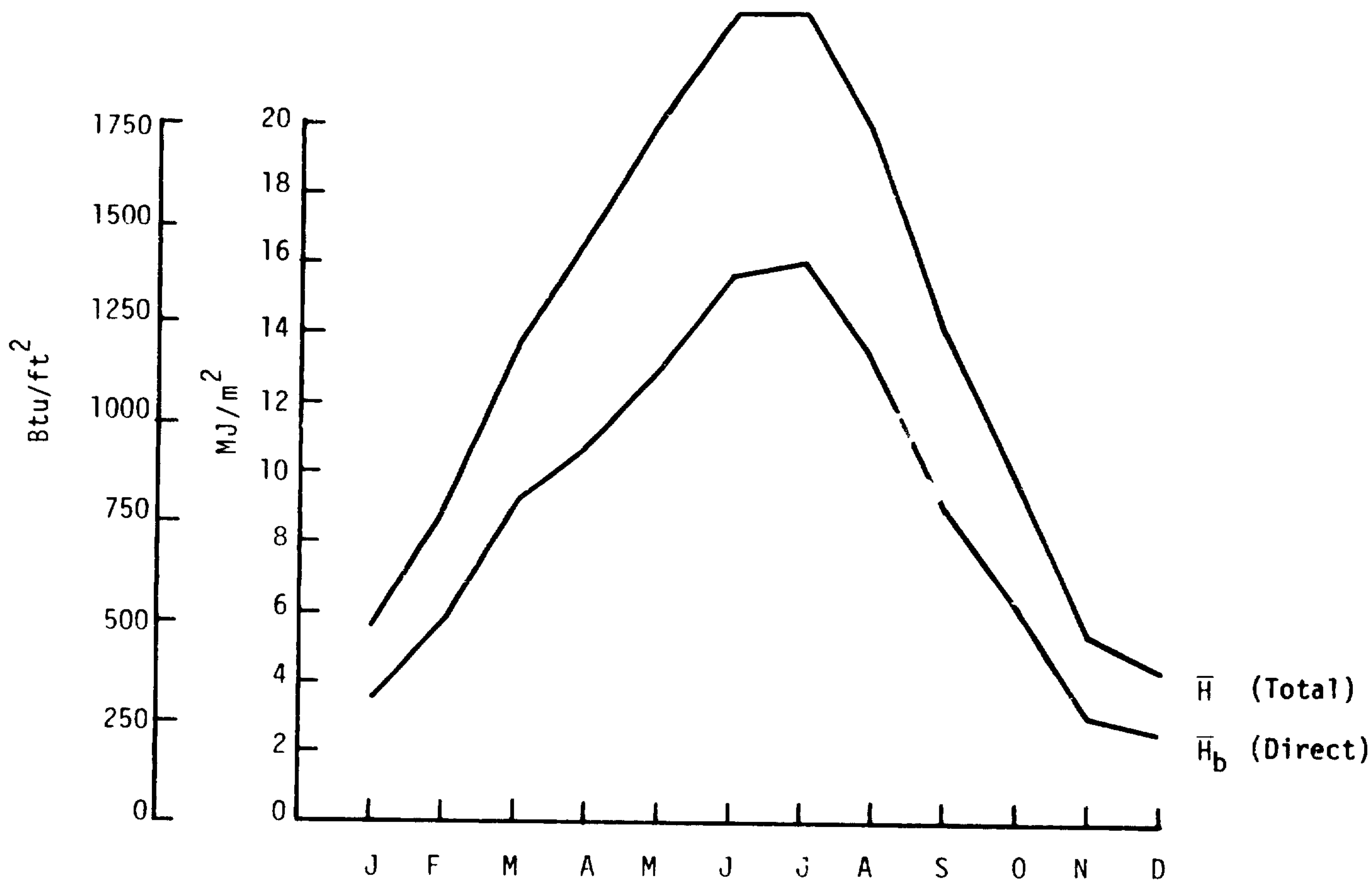

Figure 7. Monthly average daily insolation, Duluth, MN.

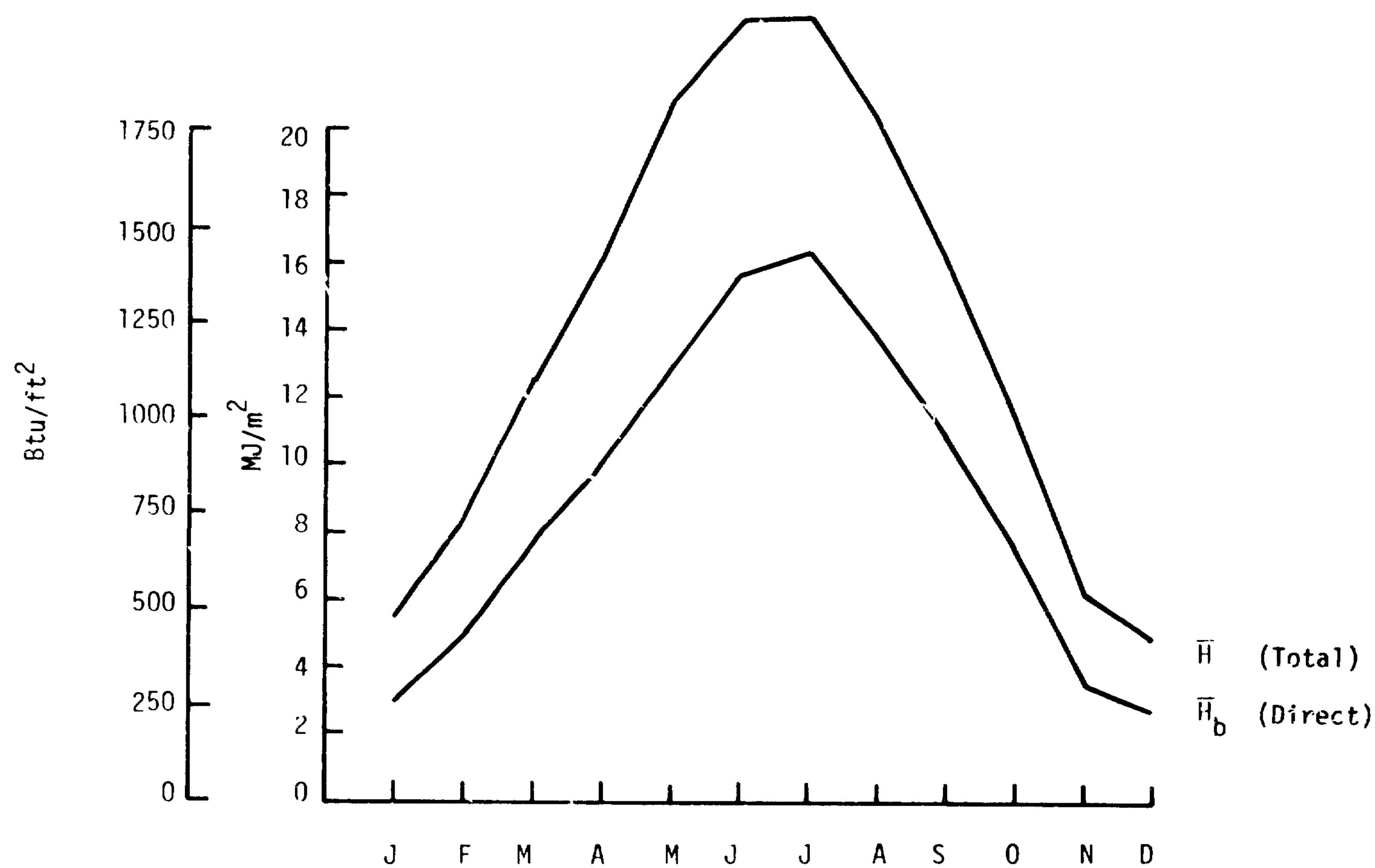

Figure 8. Monthly average daily insolation, Grosse Ile, MI.

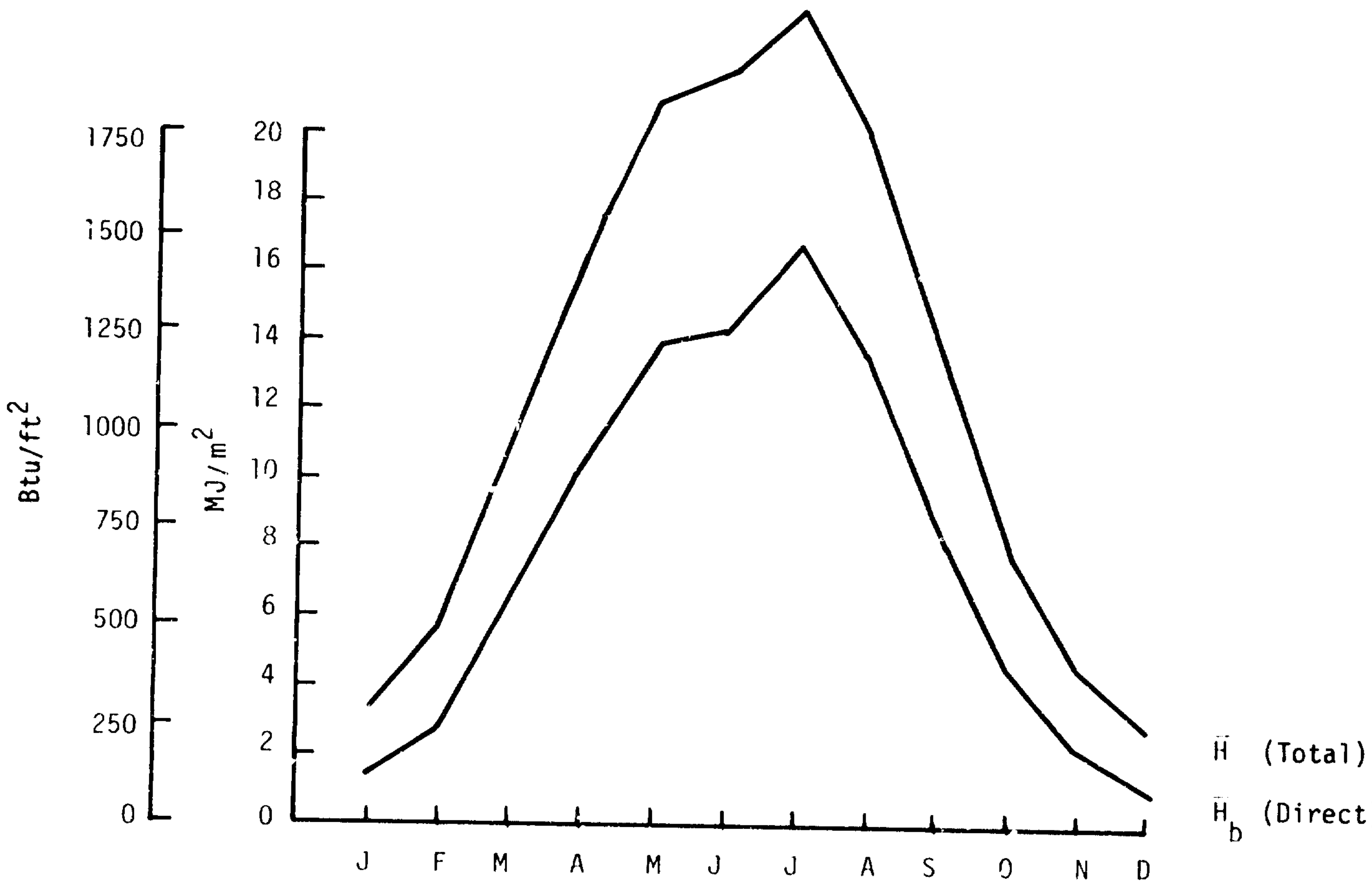

Figure 9. Monthly average daily insolation, Manchester, WA.

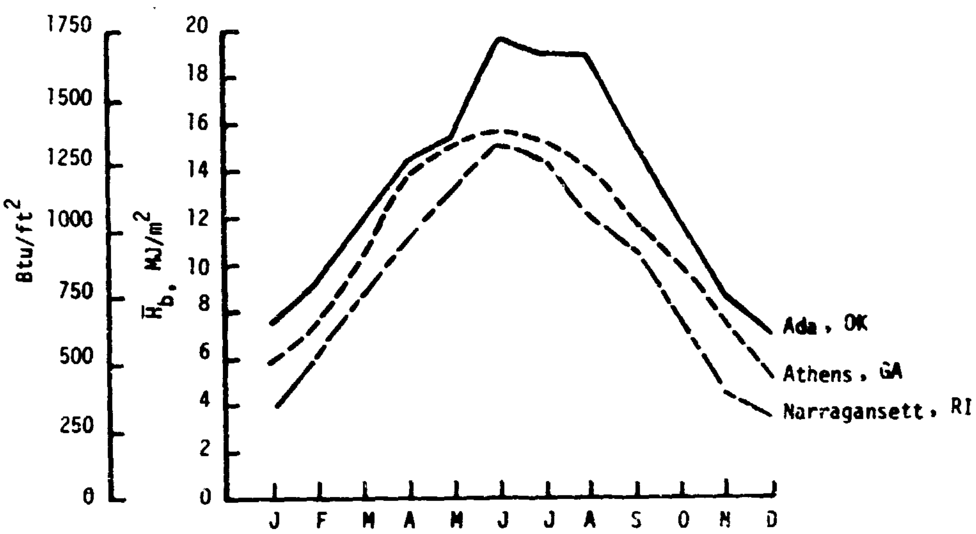

Figure 10. Comparison of monthly average daily direct radiation for Ada, OK, Athens, GA, and Narragansett, RI.

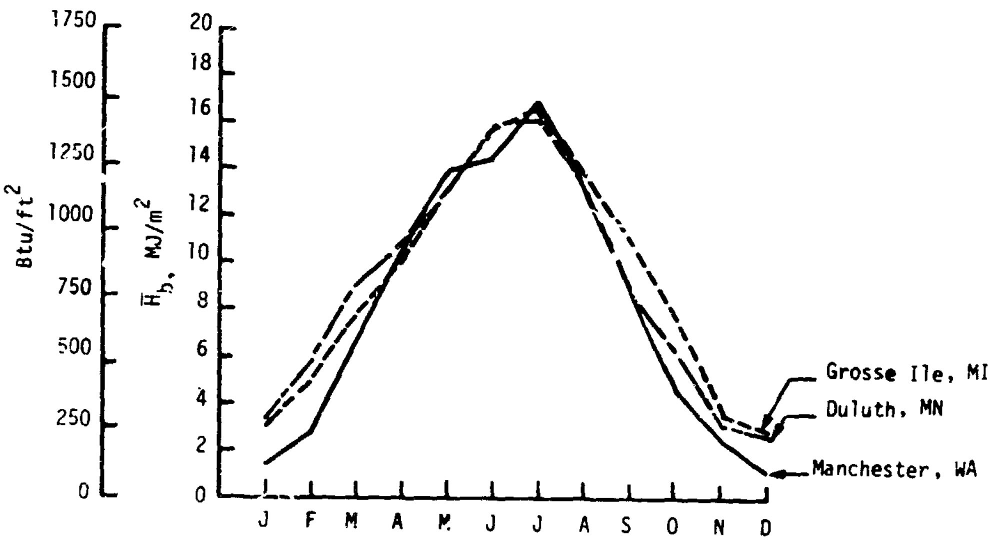

Figure 11. Comparison of monthly average daily direct radiation for Duluth, MN, Grosse Ile, MI, and Manchester, WA.

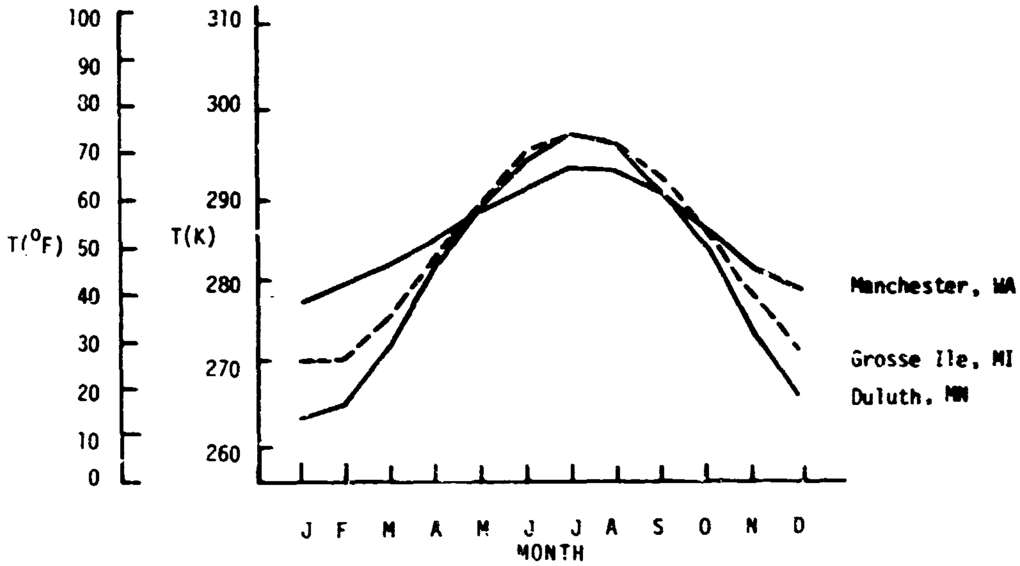

Figure 12. Monthly average daytime ambient temperature for Duluth, MN, Grosse Ile, MI, and Manchester, WA.

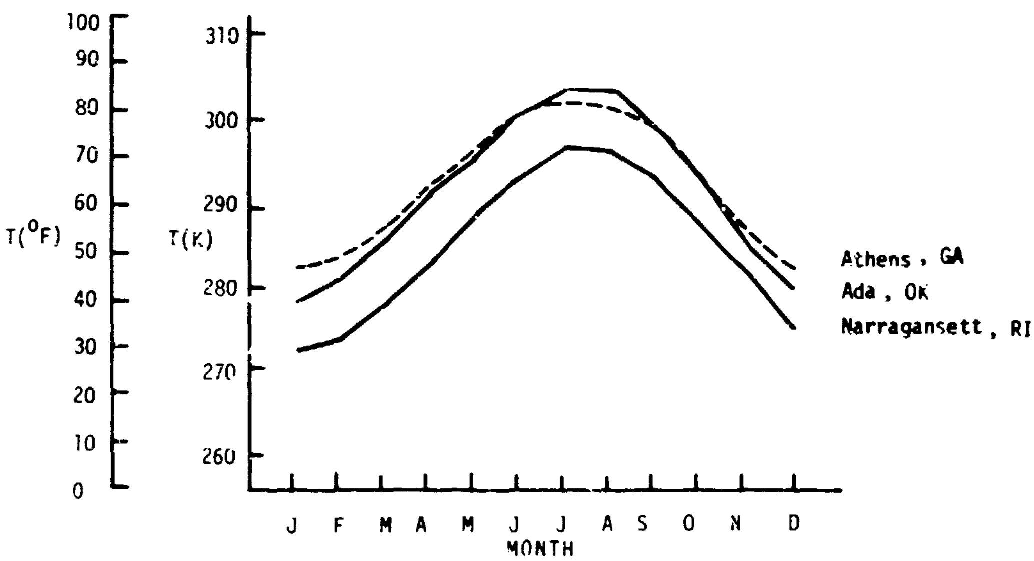

Figure 13. Monthly average daytime ambient temperature for Ada, OK, Athens, GA, and Narragansett, RI.

managers were asked to give estimates. These values were checked whenever possible with some or all of the following techniques:

- Bin-hour method for determining heating load, as outlined in Reference 4
- Ambient temperature profiles
- Flow rates and other design operating conditions
- HVAC "Rules of Thumb" relating demands such as electricity consumption to total floor area

A boiler efficiency of 80 percent and a Coefficient of Performance (COP) of 3.0 were assumed in order to convert fuel consumption to heating and cooling loads, respectively.

Ada, OK--Figure 14 presents the building loads for Ada, OK. The bin-hour method of Reference 4 was employed to confirm the heating load based upon the U-values supplied by the facility manager. Since the facility manager reported a negligible cooling load in the winter, base-line (noncooling) electrical demand was estimated from the winter electrical consumption and subtracted from the total consumption to determine the cooling electrical demand for each month. An effective COP of 3 was calculated for the American Standard chillers and used to determine the cooling load. Domestic hot water demand was reported to be negligible. Because the site visit revealed ample collector site area, the combined heating, cooling and DHW load profile was also determined.

Athens, GA--Building load profiles for Athens, GA are shown in Figure 15. Because of limited unobstructed collector site area at this facility and the size of the loads, detailed confirmation of the loads was not attempted.

Narragansett, RI--Figure 16 presents the building load profile for Narragansett, RI. The process load, heating sea water from ambient to 293 K (68° F), was calculated using 6.3 l/sec (100 gpm) process water entering the heat exchanger at 355 K (180° F) and leaving at 333 K (140° F). The other loads were supplied by the system designer.

Duluth, MN--Figure 17 presents Duluth's heating, DHW and process water load profile. The profile represents the average of 1976 and 1977 data. An estimated component breakdown was not available. Electrical consumption taken from utility bills for the 2 years showed a relatively flat profile, indicating a negligible cooling load throughout the year. This was confirmed by the facility manager. Domestic hot water information was not available.

Grosse Ile, MI--Figure 18 presents the heating load profile for Grosse Ile, MI. The reason for the December load being much higher than the previous January load is that only the second floor of the building was occupied until October 1977. As with Duluth, MN, the cooling load for Grosse Ile, MI was assumed to be negligible.

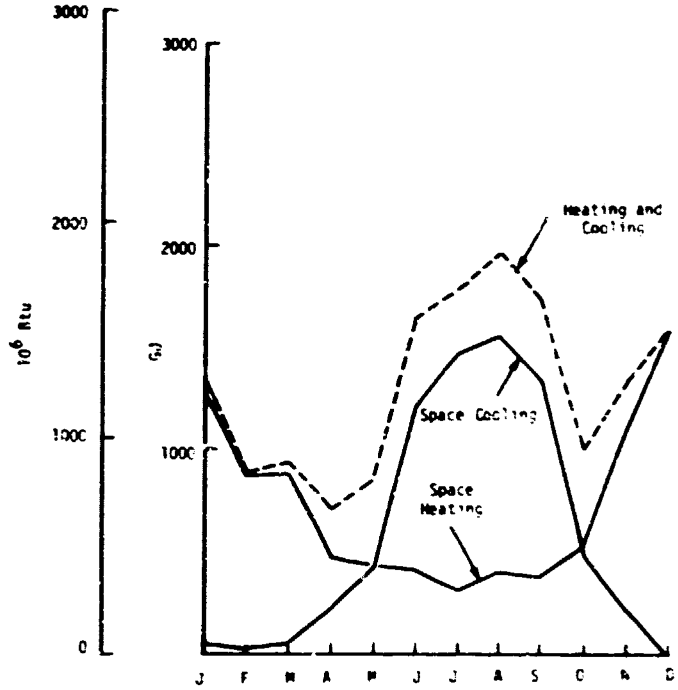

Figure 14. Ada, OK building loads.

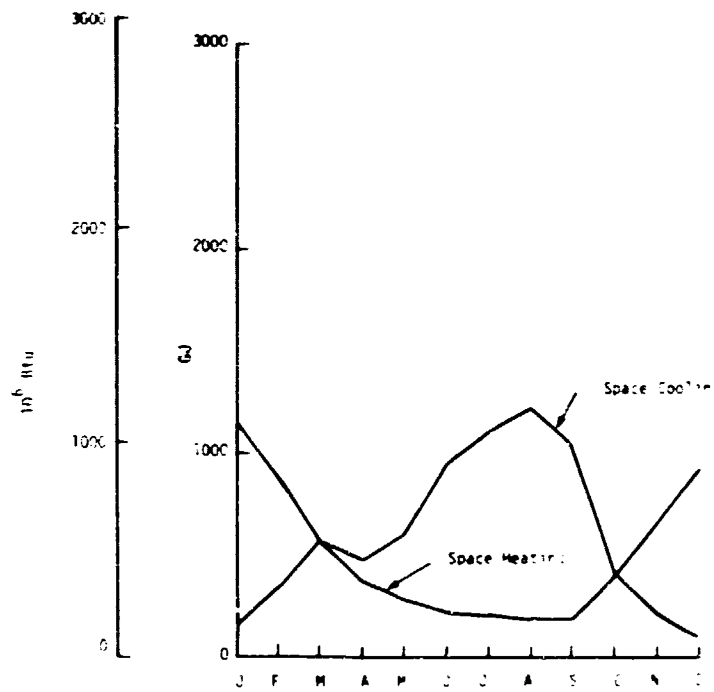

Figure 15. Athens, GA building loads.

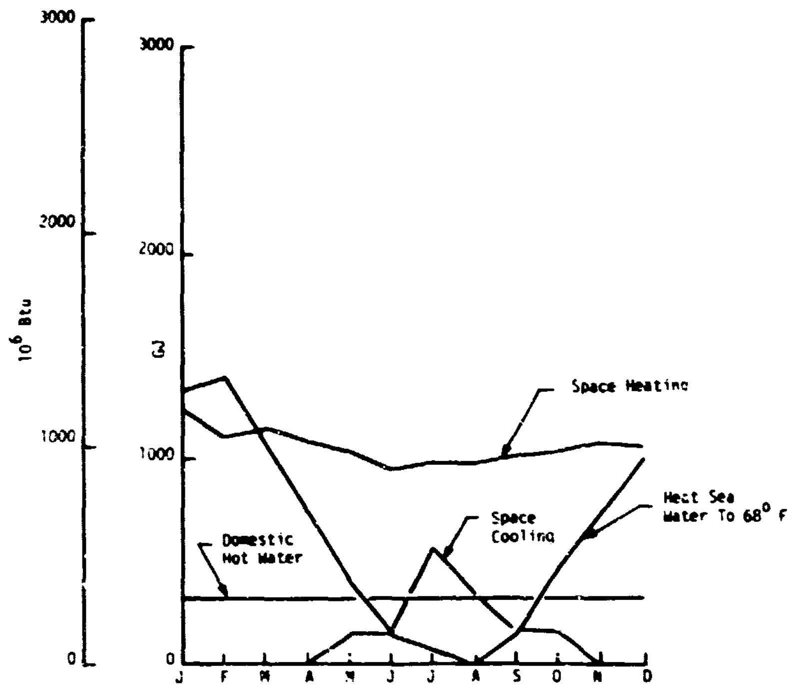

Figure 16. Narragansett, RI building loads.

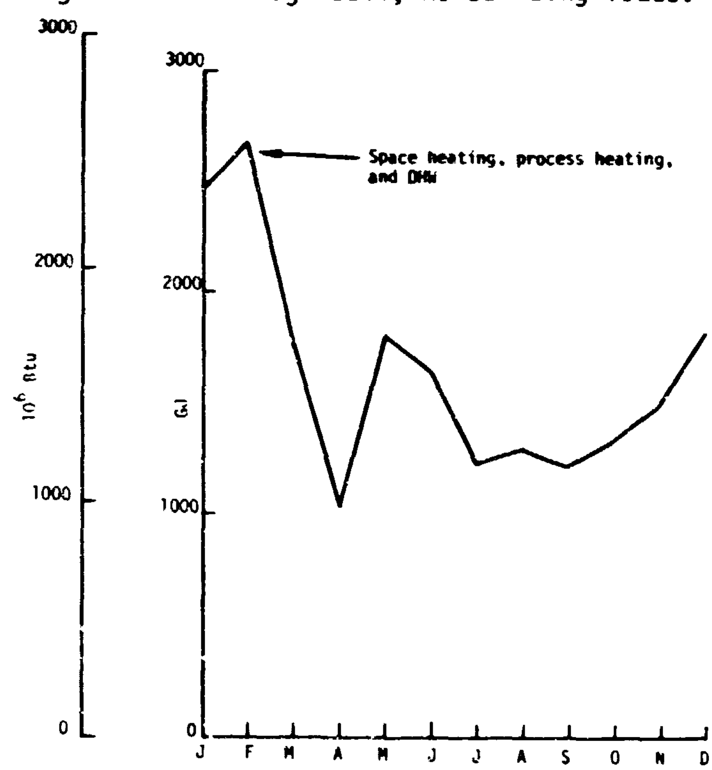

Figure 17. Duluth, MN building loads.

Manchester, WA--Because the Manchester, WA facility was not yet under construction, no fuel consumption data was available. Load projections were not available from the mechanical system designer.

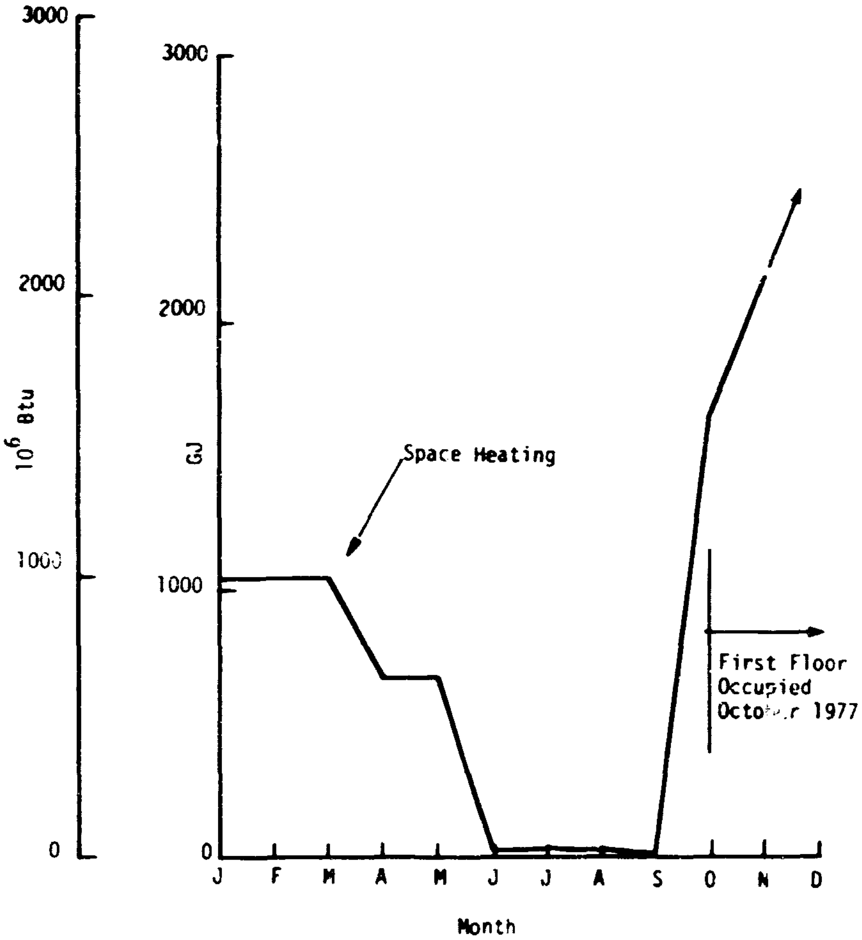

Figure 18. Grosse Ile, MI building loads.

SECTION 3

FINAL SELECTION

This section describes the procedure followed to select the primary facility and solar energy system. Also, the method used to determine the best generic solar energy system concept which could be applied to a number of facilities is discussed.

The development of the Preliminary System/Facility Matrix and the benefit analysis used to rank the various systems for each facility are discussed below. Based on the benefit analysis, the Robert S. Kerr ERL in Ada, OK was selected as the primary site. A combined solar space heating and DHW system was selected for detailed evaluation at the Ada, OK site. The single most promising and broadly applicable system, however, was found to be DHW alone, based on site-specific characteristics of the six facilities considered in detail. These conclusions and the final System/Facility Matrix are also presented in this section.

PRELIMINARY SYSTEM/FACILITY MATRIX

As discussed in Section 2, six facilities were selected for the second screening, and detailed information was sought from each to allow a comparison of solar energy system feasibility. These six facilities considered in the System/Facility Matrix were:

- Ada, OK
- Athens, GA
- Narragansett, RI
- Duluth, MN
- Grosse Ile, MI
- Manchester, WA

The system types originally considered (based on the discussions presented in Section 1) were:

- Combined heating, cooling and water heating
  - -- Parabolic trough/2-stage absorption chiller
  - -- Parabolic trough; low concentration; flat-plate/1-stage chiller
- Space cooling (same)

- Space heating and water heating
  - -- Parabolic trough; low concentration; flat-plate/heating
  - -- Flat-plate/heat pump
- Water heating
  - -- Flat-plate

Formulation of the preliminary matrix was carried out by taking into consideration the information gathered from the data collection effort and site visits discussed in Section 2.

Insolation/Load Matching

Insolation and load profiles for the various sites were presented in Section 2. This subsection presents a brief discussion of the importance of matching solar energy availability with building loads. Figure 19 presents plots of typical solar energy availability and building load profiles.

Solar availability profiles (Figure 19(a)) typically show a peak in the summer. Building load profiles have varying characteristics, as shown in Figure 19(b). Space heating loads typically are greatest in the winter months and lowest in the summer. They are, therefore, out of phase with solar availability. Cooling loads, on the other hand, are more closely in phase with solar availability. Domestic hot water and many process loads are often nearly flat (constant throughout the year). For solar energy systems, the advantage of achieving a good insolation/load match is that one can supply a larger percentage of the load while minimizing wasted collected energy.

Candidate Systems

The systems considered for the benefit analysis and included in the Preliminary Matrix are discussed in this section. Table 10 summarizes the key elements and operating temperatures for each system considered for the benefit analysis.

Ada, OK--

Ada, OK has a large collector site area, good insolation and well-maintained mechanical system; thus, water heating, space heating, space cooling and combined heating and cooling systems were all considered for the benefit analysis.

Domestic Hot Water--Since DHW load information was not available, a nominal DHW load was assumed, scaled in proportion to floor area from the load given for Narragansett, RI. The assumption that the DHW load was proportional to building floor area was also applied to all remaining sites. Collector operating temperature for DHW for all sites was assumed to be 316 to 344 K (110^{o} to 160^{o} F), delivering hot water at 333 K (140^{o} F). Solargenics single-glazed, selective-surface collectors were used. A schematic of the hot water system is shown in Appendix C. Storage was accomplished in an unpressurized vessel.

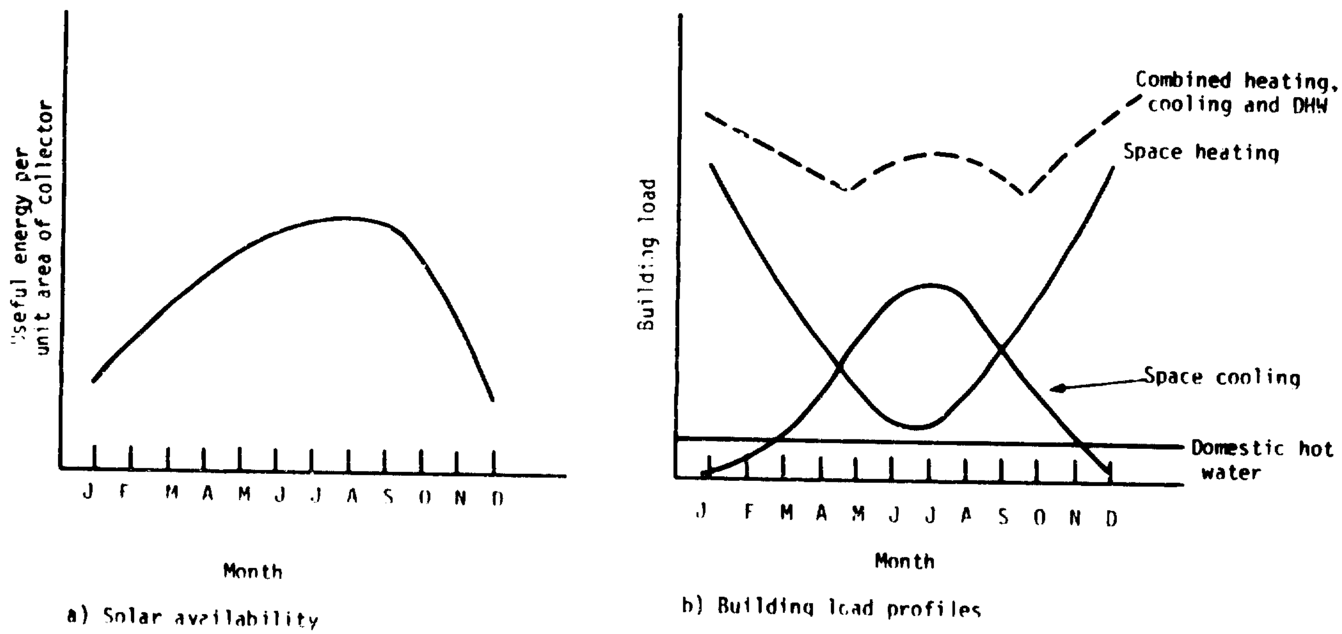

Figure 19. Typical solar availability and building load profiles.

TABLE 10. SUMMARY OF SYSTEMS CONSIDERED FOR EACH SITE

Site	Load		Collector operating temperature K (° F)	Storage	
	Type	Supply temperature K (° F)		Vessel type	ΔT K (° F)
Ada, OK	DHW	333 (140)	316 to 344 (110 to 160)	Unpressurized vessel	28 (50)
	Space heating	391 (244)	436 to 463 (325 to 375)	1136 kPa (150 psig) pressure vessel	58 (122)
	Space cooling and combined heating and cooling -- single-stage				
	Space cooling and combined heating and cooling -- dual-stage	451 (352)	463 to 491 (375 to 425)	1825 kPa (250 psig) pressure vessel	28 (51)
Narragansett, RI	DHW	333 (140)	316 to 344 (110 to 160)	Unpressurized vessel	28 (50)
	Space heating	355 (180)	333 to 366 (140 to 200)		22 (40)
	Heat sea water				
Athens, GA Duluth, MN Grosse Ile, MI Manchester, WA	DHW	333 (140)	316 to 344 (110 to 160)	Unpressurized vessel	28 (50)

Space Heating--Because the Ada, OK facility uses 184 kPa (12 psig) steam for distribution throughout the building, the solar heating system considered would generate low pressure steam for interface with the existing system. This would avoid having to re-pipe the entire building and install new coils to distribute hot water. The Acurex 3001 collector was used to operate from 436 to 463 K (325° to 375° F) in order to produce a 68 K (122° F) storage ΔT (391 to 459 K, 244° to 366° F). Storage was accomplished in a 1136 kPa (150 psig) pressure vessel. Heat pumps were eliminated from consideration for Ada, OK because the optimum operating temperature for commercially available units is still around 355 K (180° F). (Personal communication with R. C. Niess, Westinghouse Electric Corporation, February 1978.)

Space Cooling--Space cooling systems using both single- and dual-stage absorption chillers were considered. The Acurex 3001 collector was used to operate from 436 to 463 K (325° to 375° F) in order to produce 184 kPa (12 psig) steam to drive the single-stage chillers. A 1136 kPa (150 psig) pressure vessel was used.

The Acurex collector was also used to operate from 463 to 491 K (375° to 425° F) to generate 949 kPa (123 psig) steam to drive the dual-stage chiller. In this case, a 1825 kPa (250 psig) pressure vessel was used for storage.

Heating and Cooling--Greater utilization of solar energy can be realized when a combined heating and cooling system is used, because of the benefits of load matching. Even with the inclusion of space heating, the two combined systems were mechanically identical to the two cooling systems considered. The single-stage combined system was operated on 184 kPa (12 psig) steam year-round. The dual-stage system was operated on 184 kPa (12 psig) steam in the heating season (December through March) and on 949 kPa (123 psig) steam in the cooling season (April through November). The negligible cooling load in the winter months resulted in the opportunity to operate at the lower temperatures and increase collector performance.

Narragansett, RI--

The Cost/Benefit Analysis for Ada, OK revealed that cooling systems alone have a very high specific cost (cost per unit energy delivered) compared to heating alone or combined heating and cooling. Because of this and the fact that Narragansett, RI has limited collector site area and low insolation, the cooling and combined heating and cooling systems were not considered further. Since Narragansett's process heat requirement was considered to be a unique application of solar energy for an EPA laboratory, it was considered along with space heating and DHW.

Domestic Hot Water--A solar DHW system, identical to that considered for Ada, OK, was evaluated for Narragansett, RI. Narragansett's load was calculated by the facility's mechanical system designer.

Space Heating--The space heating system considered for Narragansett, RI consisted of Acurex concentrators operating in the 333 to 366 K (140° to 200° F) range and delivering 355 K (180° F) hot water to interface with the existing system. An unpressurized vessel was used for storage.

Process Water--Narragansett's process load requirement is for 6.3 to 12.7 l/sec (100 to 200 gal/min) of 293 K (68° F) sea water for experimental purposes. The existing process heating loop delivers 355 K (180° F) hot water to the load and returns it at 333 K (140° F). Therefore, the Acurex concentrator was used and operated in the 333 to 366 K (140° to 200° F) range. An unpressurized vessel was used for storage.

Other Sites--

A solar DHW system identical to that considered for Ada, OK and Narragansett, RI was evaluated for each of the remaining six sites. No other systems were considered for these facilities. The reasons for this decision follow.

Space Heating--Because of the poor load match and low insolation for Manchester, WA, Grosse Ile, MI and Duluth, MN, space heating was eliminated from further consideration for these sites. The low winter ambient temperature for the Great Lakes facilities also contributed to the decision. For Athens, GA it was felt that, rather than investing capital in any solar system other than DHW, the most cost-effective first step would be to upgrade the efficiency of the existing HVAC system.

Space Cooling and Combined Heating and Cooling--Because of the very low cooling loads for Duluth, MN and Grosse Ile, MI, and the low summer insolation for Manchester, WA cooling, and therefore combined heating and cooling, were eliminated from further consideration. These two systems were also eliminated from consideration for Athens, GA for the reason mentioned above.

Preliminary Matrix

The Preliminary System/Facility Matrix is presented in Table 11. Because of limitations in existing mechanical system characteristics, certain candidate system components were eliminated from consideration in the preliminary matrix.

In Ada, OK, for example, the existing heating system uses 184 kPa (12 psig) steam which is distributed to air handlers and heat exchangers throughout the building. In order to interface with this system at the temperature required, a concentrating collector would be required to generate steam for either space heating or combined space heating, cooling and DHW systems. Currently available flat-plate collectors could not compete with concentrators at the required temperatures. A concentrator would also be used to heat hot water to drive an absorption chiller if cooling alone were chosen.

TABLE 11. PRELIMINARY SYSTEM/FACILITY MATRIX

Facility \ System	Service hot water flat-plate	Space heating and DHW		Space cooling		Heating, cooling and DHW	
		Flat-plate	Concentrator	Flat-plate	Concentrator	Flat-plate	Concentrator
Ada, OK	Yes • Separate DHW system	No	Yes • 184 kPa (12 psig) steam to heat hot water	No	Yes • Hot water to drive chiller	No	Yes • 184 kPa (12 psig) steam • Interface at 2 Points
Athens, GA	Yes • Separate DHW system	Yes • Hot water	Yes • Hot water	No	Yes • Hot water to drive chiller	No	Yes • Hot water
Narragansett, RI	Yes • Separate DHW system • Heat sea water to 293 K (68° F)*	Yes • Hot water	Yes • Hot water	No	Yes • Hot water to drive chiller	No	Yes • Hot water to drive chiller
Duluth, MN	Yes • Separate DHW System	No	Yes • Low pressure steam to heat hot water	No • Low insolation • Small cooling load			
Grosse Ile, MI	Yes • Separate DHW system	N/A		No • Low insolation • Small cooling load • No chilled water loop			
Manchester, WA	Yes • Separate DHW system	N/A		No • Low insolation			

* Concentrator

For all sites, flat-plate collectors would be used to heat DHW (water heating only systems), because of their slight cost/performance advantage at the lower operating temperatures of 316 to 344 K (110° to 160° F). It should be noted that some facilities in the matrix currently supply DHW at temperatures which would require higher collector operating temperatures, which, in turn, might suggest the use of concentrating collectors. However, since energy efficiency must have high priority when designing a solar energy system, high DHW supply temperatures should be lowered, whenever practicable, to the 322 to 333 K (120° to 140° F) range (even lower for preheat), where the flat-plate collectors would again have the advantage. Low concentration collectors were eliminated from consideration, because, in the temperature ranges considered, either the flat-plate or concentrator has the cost/performance advantage.

Due to the higher collector operating temperatures required to drive a thermal chiller, flat-plates were not considered for any cooling applications. For Duluth, MN, Grosse Ile, MI and Manchester, WA, space cooling was eliminated from consideration, since Duluth, MN and Grosse Ile, MI have very low cooling loads and all three have low insolation. Also, Grosse Ile, MI has no existing chilled water loop, making interface with their cooling system difficult and expensive. Narragansett's "process load" (heating sea water taken from Narragansett Bay from ambient to 293 K (68° F) was thought to be a unique solar application. Therefore, it was included in the preliminary matrix.

BENEFIT ANALYSIS

A benefit analysis was performed for each of the systems included in the Preliminary System/Facility Matrix. The purpose of the analysis was to provide a basis for selecting a primary system for a more detailed analysis and also to select the best generic system.

Cost/Benefit Analysis

A minimum Cost/Benefit Analysis was performed for each system in the Preliminary System/Facility Matrix. For each system, the optimum system size was determined by calculating, for various collector field sizes:

- Useful collected energy
- Portion of load supplied
- Total annualized system cost
- Specific cost ($/unit energy delivered)

Methodology--

The amount of useful collected energy was calculated from the incident insolation using the appropriate collector performance curve and system operating and ambient temperatures. The total and direct insolation were determined using the correlations of Reference 5. The amount of energy supplied to the load was determined from the useful collected energy for various collector field sizes.

The annualized system cost, as outlined in Reference 4, was determined as follows:

$$COST = CRF\ (C_A A_C + C_E) + M + I + O + T \tag{1}$$

where C_A = solar energy system first costs that are directly proportional to collector area ($\$/m_c^2$)

A_C = collector area (m_c^2)

C_E = solar energy system fixed costs ($)

M = annual maintenance costs ($)

I = annual insurance costs ($)

O = annual solar system operating costs (pumps, fans, etc.) ($)

T = net annual taxes ($)

CRF = capital recovery factor

Since the solar system would be installed on a federally owned building, the tax term was eliminated. Annual insurance, maintenance and operating costs were taken to be fixed percentages of the initial capital costs. The values taken were:

I = 0.25 percent ($.25/$100)

M = 0.5 percent

O = 0.5 percent

A system life of 25 years was assumed, yielding a capital recovery factor, CRF = 0.09368, for the 8 percent discount rate suggested by Reference 4. Equation (1) then became

$$COST = 0.106\ (C_A A_C + C_E) \tag{2}$$

The component costs of C_A were:

- Collectors (installed)
- Storage (insulated)
- Controls; area-dependent portion

Collector costs included installation, piping, foundation, support structure, supervision and shipping. The component costs of C_E were:

- Storage foundation and instrumentation
- Heat exchangers
- Controls; fixed portion
- Pumps

- Power
- Cooling tower

Power costs were only applicable when concentrating collectors were used, and tower costs were only included for cooling or combined heating and cooling systems.

The cost component that did not readily fall into either of the above categories was the chiller cost. Details of the sizing and costing of chillers are given below.

The annualized total system cost for each collector field size was then divided by the amount of energy actually delivered to the load by that system. This yielded the system specific cost ($/unit energy), which was then used to determine the optimum field size (lowest specific cost) for that system and the cost/benefits of the system relative to the others. This figure of merit was then used to point out certain characteristics of each system.

It should be noted here that cost effectiveness, not payback period, is the figure of merit used in this study to optimize and compare systems. Current economic analysis indicates that the payback period for medium temperature solar energy systems is typically greater than system lifetime. Furthermore, estimates of payback period for solar energy systems are highly variable, depending on such uncertain parameters as future conventional energy prices. Therefore, although payback period is a useful figure of merit for evaluating conventional energy systems, it is presently less meaningful for optimizing and comparing solar energy systems.

The collector field size, total system cost and solar fraction were noted for each system at the minimum specific cost. This was necessary to determine whether a given system, although characterized by a relatively attractive specific cost, was really feasible for a given facility. For example, if the minimum specific cost called for a total system cost of $10 million, yielded only 5 percent solar substitution, or required two acres of collector site area when only one was available, that system would not be feasible.

Costs--

Components--Table 12 summarizes the cost assumed for each component for the Cost/Benefit Analysis. The costs used for the Cost/Benefit Analysis and listed in Table 12 were conservative, and therefore, represented the upper range expected for each component. It should be pointed out that these costs represent today's values. No attempt was made to project future component costs.

Flat-Plate Collectors--Current flat-plate collector installed cost estimates were obtained from manufacturers and distributors. Table 13 lists the information gathered. In a few cases, installed costs were not available. The Solargenics water white, single-glazed, selective surface flat-plate collector was chosen for DHW applications and its performance

TABLE 12. COMPONENT COSTS USED IN COST/BENEFIT ANALYSIS

Site \ System	Area dependent costs $/m² ($/ft²): Collector Type	Collector Installed	Storage Type	Storage Insulated	Controls	Fixed costs ($): Storage foundation + installation	Heat exchangers	Controls	Power	Cooling tower	Absorption chiller, installed: Single-stage <352 kW $	Single-stage >879 kW $/kW ($/ton)	Dual-stage ($)
Ada, OK													
DHW	Solargenics flat-plate	279 (25.84)	Unpressurized vessel	21.6 (2.00)	41.2 (3.82)	4,000	4,000	58,000	--	--	--	--	--
Space heating	Acurex parabolic trough	296 (27.40)	1136 kPa (150 psig) pressure vessel	18.0 (1.67)					20,000	--	--	--	--
Space cooling/ combined heating and cooling single-stage										6,000	40,000	74.0 (260)	--
Space cooling/ combined heating and cooling dual-stage			1825 kPa (250 psig) pressure vessel	58.6 (5.43)							--	--	154,000
Narragansett, RI													
DHW	Solargenics flat-plate	279 (25.84)	Unpressurized vessel	21.6 (2.00)					--	--	--	--	--
Space heating/ process heating	Acurex parabolic trough	296 (27.40)							20,000	--	--	--	--
Other Sites													
DHW	Solargenics flat-plate	279 (25.84)							--	--	--	--	--

TABLE 13. CURRENT COLLECTOR COST ESTIMATES

Type	Manufacturer	Model	Cost* $/m^2 ($/ft^2)	Installed cost† $/m^2 ($/ft^2)
Flat-Plate	Solargenics	Single-glazed selective water-white	116.0 (10.80)	278.0 (25.84)
Flat-Plate	Solar Corporation of America	Single-glazed selective	114.0 (10.64)	323-430 (30-40)
Flat-Plate	Revere	Double-glazed non-selective	119.0 (11.09)	N/A
Flat-Plate	PPG	Double-glazed non-selective	190.0 (17.66)	N/A
CPC	Great Natural Structures	--	161.0 (15.00)	N/A
Evacuated tube	Owens-Illinois	--	215.0 (20.00)	N/A
Parabolic trough	Acurex	3001	159.0 (14.78)	295.0 (27.40)

*For quantities of 2,790 m^2 (30,000 ft^2)
†Includes piping, support, installation, shipping, supervision, footings

was judged to be typical of good flat-plate collectors operating in the required 316 to 344 K (110° to 160° F) range. The current collector cost for quantities of 2790 m^2 (30,000 ft^2) was \$115/m^2 (\$10.80/ft^2). Solargenics representatives indicated the collector could be completely installed, including shipping, for between \$220 and \$278/m^2 (\$20.44 to \$25.84/ft^2). The higher, more conservative, number was used for all DHW analyses.

Concentrating Collectors--The Acurex 3001 parabolic trough concentrating collector was chosen for all system applications other than DHW (see Appendix B). The installed cost used was \$295/m^2 (\$27.40/ft^2).

Storage--To design storage tanks, detailed system optimizations must be performed. For a given storage requirement, decreasing cost, and therefore decreasing tank size, requires a larger storage ΔT which, in turn, leads to higher collector operating temperatures. This leads to decreased collector performance and, therefore, larger collector fields.

Pressure vessels are generally more expensive than unpressurized storage tanks. However, unpressurized vessels are considerably larger, which results in increased weight and costs. Also, larger vessels require more insulation, which runs at least \$43/m^2 (\$4/ft^2) of tank surface area.

Needless to say, for the requirements of the benefit analysis, detailed optimizations were not performed. For all DHW systems and Narragansett's space heating and process loads, an unpressurized vessel was used. Reference 4 suggests a \$21.60 to \$54.00/m^2 (\$2.00 to \$5.00/ft^2) range for insulated storage tanks. Because the tanks considered for these applications are simple unpressurized storage vessels, a \$21.60/m^2 (\$2.00/ft^2) cost was chosen.

Because of Ada's unique steam requirement, a flash boiler was selected as the storage tank. Because of the increased cost/unit volume incurred by a pressure vessel, a rough sizing was done. Vessels were sized to be able to store as much energy as could be collected by a 3720 m^2 (40,000 ft^2) field on a peak June day for the operating temperatures required for the particular application. The ullage requirement of a flash boiler was estimated to increase this volume by 10 percent. The American Society of Mechanical Engineers (ASME) pressure vessel design code was consulted to determine the wall thickness for each requirement. Given the volume and wall thickness, and therefore the weight of the tank, the cost was then computed at \$2.2/kg (\$1.00/lb). Insulation costs mentioned above were also calculated. A tank cost per unit collector area value was then calculated (using the above 3720 m^2 field) to be applied to other field sizes for the system.

For Ada's space heating, one-stage cooling and combined systems, a 1136 kPa (150 psig) pressure vessel was chosen. The tank cost derived was \$17.97/m^2 (\$1.67/ft^2). For Ada's two-stage cooling and combined system, a 1825 kPa (250 psig) vessel was chosen. The cost derived for this tank was \$58.64/m^2 (\$5.43/ft^2). The tank foundation and installation cost for each system was estimated to be \$4,000.

Controls--The control and data acquisition costs were estimated, based upon Acurex's past experiences with similar solar energy systems. A fixed cost of \$58,000 was determined. Increasing field size was found to have a significant impact on the total control system cost. Beyond the fixed cost, a figure of \$41.26/m^2 (\$3.82/ft^2) was used.

Other Fixed Costs--Other component costs used were \$4,000 for heat exchangers, \$20,000 for electrical power and \$6,000 for a cooling tower. The power cost was applied only to systems using the concentrator. The cooling tower cost was applied only to cooling systems.

Absorption Chillers--Chiller costs can not be treated as either fixed or collector area-dependent costs. Instead, chiller costs are dependent on tonnage capacity requirements.

The single-stage chiller cost/ton was found to decrease for increasing capacity between 352 and 879 kW (100 and 250 tons). From the cost data of a chiller manufacturer, a relationship was formulated that adequately described this capacity dependence:*

$$\text{Chiller cost} = (257 - 0.55\ C)C$$

where C = chiller capacity in tons. For capacities less than 352 kW (100 tons), a flat cost of \$20,000 per unit is assumed. For capacities greater than 879 kW (250 tons), the cost was \$37/kW (\$130/ton). The installed chiller cost was estimated by multiplying the equipment cost by a factor of two. The required chiller capacity for a given system and collector field size was determined directly from the peak cooling delivered by the system.

The smallest dual-stage chiller commercially available is the Trane 1354-kW (385-ton) chiller. Dual-stage chillers cost about \$57.90/kW (\$200/ton), or \$77,000 for the 1354-kW (385-ton) unit. Installed, that would be \$154,000. This unit was found to be adequate for all reasonable collector field sizes.

Systems--Sample system costs/unit collector area are presented in Table 14. Costs are given for each system for both 930 m^2 (10,000 ft^2) and 3720 m^2 (40,000 ft^2) collector fields. The collector area-dependent and fixed costs components are the sums of the appropriate component costs listed in Table 12. These costs lie within the ranges given for the various systems in Reference 4. It should be noted that combined heating and cooling system costs would be essentially the same as the cooling only costs given in Table 14, because all system components

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*Personal communication with Tyler Clemmer, The Trane Company, Cupertino, California, February 10, 1977.

TABLE 14. SAMPLE SYSTEM COSTS

System	Component costs: Area Dependent $/m^2$ ($\$/ft_c^2$)	Component costs: Fixed $	System cost/collector area, $/m^2$ ($\$/ft_c^2$): for 930 m^2 field (10,000 ft_c^2)	System cost/collector area, $/m^2$ ($\$/ft_c^2$): for 3720 m^2 field (40,000 ft_c^2)
DHW-flat-plate	341 (31.66)	66,000	412 (38.26)	358 (33.31)
Space heating-concentrator (steam loop)	354 (32.89)	86,000	446 (41.49)	377 (35.04)
Space cooling-concentrator (steam loop, 1-stage chiller)	354 (32.89)	132,000 150,000*	496 (46.09)	394 (36.64)
Space cooling-concentrator (steam loop, 2-stage chiller)	394 (36.65)	244,000	657 (61.05)	460 (42.75)

*Chiller size (and cost) increases with field size (although not directly) for single-stage chillers. See text, this section.

required for the combined system are already included in the cooling system.

Results--

As described above, the specific cost of a system is calculated by dividing the annualized total solar system cost by the amount of energy delivered to the building load. This number gives a measure of the cost effectiveness of that system.

Specific cost ($/unit energy) is primarily a function of collector area. If specific cost is plotted against collector area, a curve results which resembles Figure 20. The point of lowest specific cost, (B), represents the most cost-effective system size. Decreasing system size below B results in increased specific cost, because of the increasingly dominant effect of fixed costs. Increasing system size above B results in smaller energy gains for each dollar invested because energy is beginning to be wasted. That is, there is a point during the year for which the collector field is capable of supplying more energy than is required by the load.

Figure 20 illustrates the relationship between collector area and energy supplied. As can be seen from the figure, point B represents the "critical" point discussed above. Points A, B, and C represent increasing collector field sizes for a given system supplying a given load.

The remainder of this section presents the calculated system costs for all systems included in the Preliminary System/Facility Matrix. The components and costs used were summarized in Table 12.

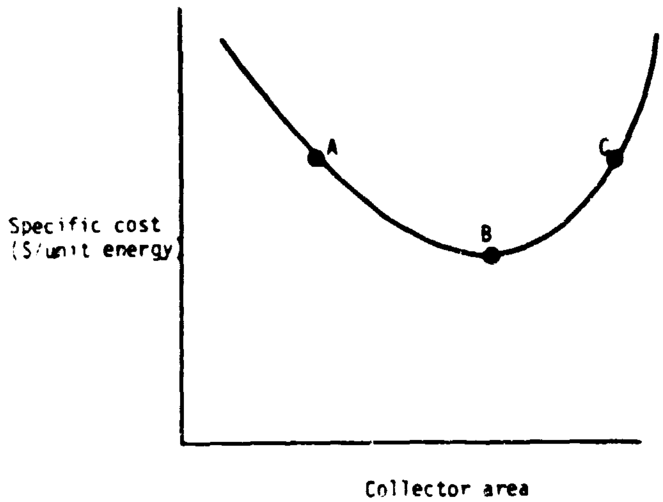
Ada, OK--Since Ada's combination of insolation, facility maintenance, collector site area and load match was judged exceptional, the most detailed cost/benefit was performed upon that facility. Determinations of optimum collector field size were made, as described above, for all candidate systems in the Preliminary System/Facility Matrix.

Figure 21 presents a composite plot of specific cost versus the percent of solar substitution (of the total building load) for the candidate systems:

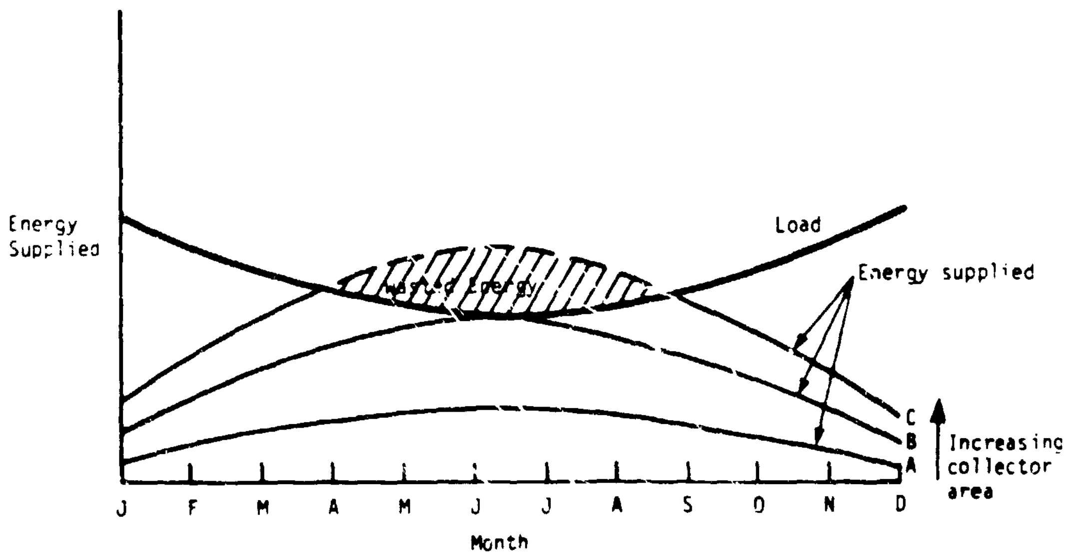
- DHW
- Space heating
- Space cooling, single- and dual-stage chillers
- Space heating and cooling, single- and dual-stage chillers

It should be emphasized that these costs should be used for comparison purposes only. A more detailed cost analysis of the primary system is presented in Section 4.

As might be expected, the DHW system shows the lowest specific cost at optimum design, about \$14.20/GJ (\$15/10^6 Btu) delivered. This is due largely to the lower-cost collectors made possible by the lower temperature requirements of a DHW system. All of the other systems for



a. Specific cost as a function of collector area



b. Energy supplied for various collector field sizes

Figure 20. Explanation of specific cost variation with collector field size.

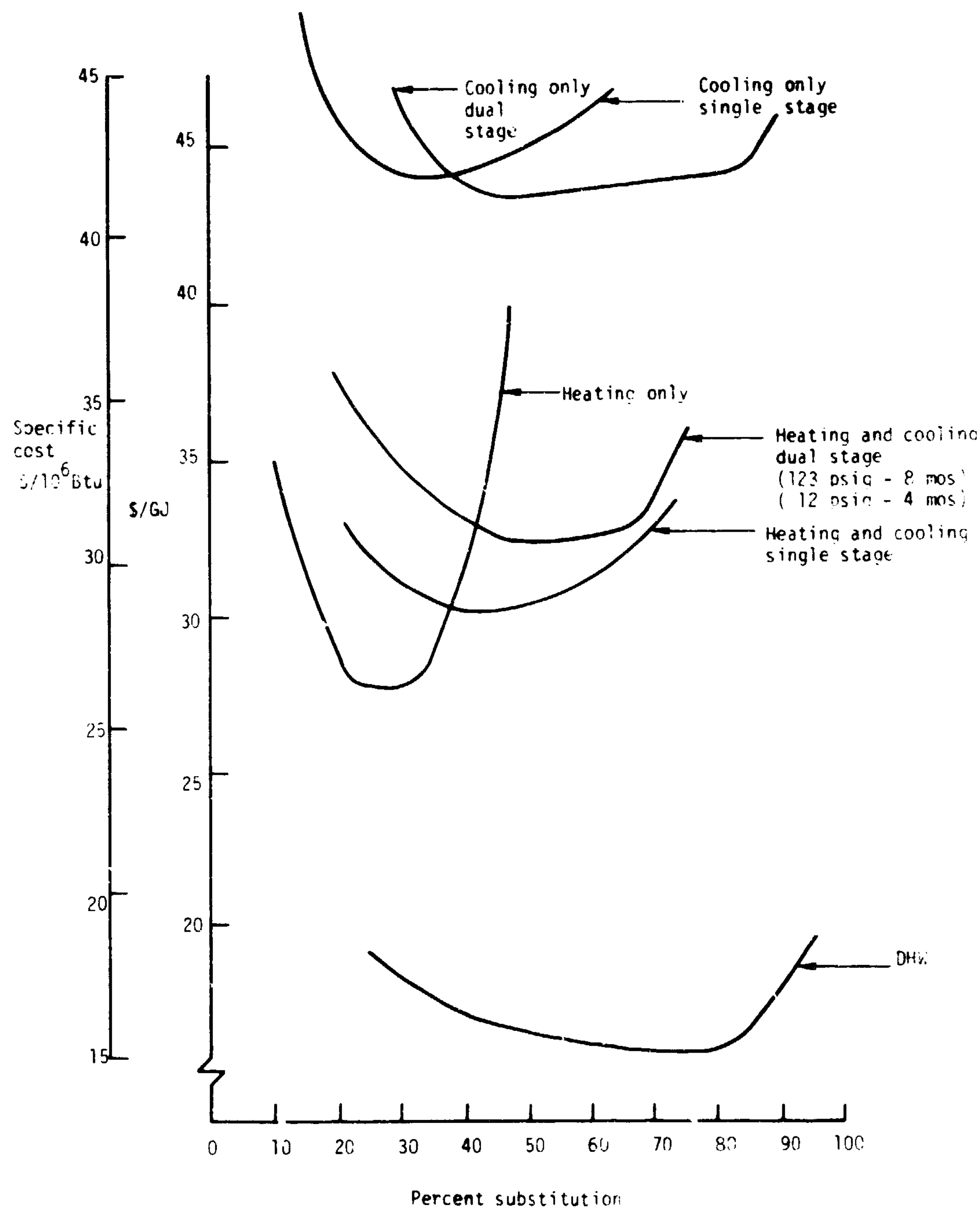


Figure 21. System specific cost vs. percent of solar substitution, Ada, OK.

Ada, OK use the concentrator, which is more cost-effective at the higher temperatures.

Space heating is the next most cost-effective system at \$24.9/GJ (\$26.3/10^6 Btu). The percent of substitution is low, however, at only 28 percent; this is because of the relatively poor load match.

Systems doing only cooling had considerably higher specific costs than any other system mainly because of the high capital costs and low COP's associated with absorption chillers. Single-stage chillers yielded an optimum specific cost of \$39.5/GJ (\$41.7/10^6 Btu) while dual-stage chillers yielded \$39.1/GJ (\$41.2/10^6 Btu).

The combined heating and cooling systems are much more cost-competitive with the heating-only system. This is because, for the same capital cost of a cooling-only system, a substantial portion of the heating load can be met by the collector/storage part of the system. Energy is not thrown away when cooling is not required. The single-stage combined system gives a specific cost of \$27.11/GJ (\$28.6/10^6 Btu). This is 8 percent more than the optimum specific cost of heating only.

The optimized combined system yields a 44 percent solar substitution, while the optimized heating system yields a 28 percent solar substitution. In order to achieve 44 percent solar substitution for the latter system, the specific cost would be about \$32.4/GJ (\$34.2/10^6 Btu), due to the poor load match.

Figure 22 presents plots of specific cost versus collector area for all systems, except for cooling-only. DHW would require 1670 m^2 (18,000 ft^2) while space heating would require 1400 m^2 (15,000 ft^2). Single-stage and dual-stage combined systems would each require 4650 m^2 (50,000 ft^2) for optimum design. Figure 23 shows specific cost versus total system cost for these systems. Both heating and hot water would cost around \$500,000, while the single-stage, combined system would cost approximately \$1,790,000.

<u>Narragansett, RI</u>--The systems considered for Narragansett, RI were:

- DHW
- Space heating
- Process heating

Figure 24 presents specific cost versus percent solar substitution for these systems. Again, because of the low collector operating temperatures required, DHW gives the lowest specific cost. Space heating, because of a substantial summer heating load, gives a 42 percent substitution. On the other hand, the process load, with a poor load match and relatively high collector operating temperature, gives very low substitution and high specific cost. Figures 25 and 26 present specific cost versus collector area and total system cost, respectively, for hot water and space heating. The important points to note here are the large collector area and total system cost required for the space heating system. At this

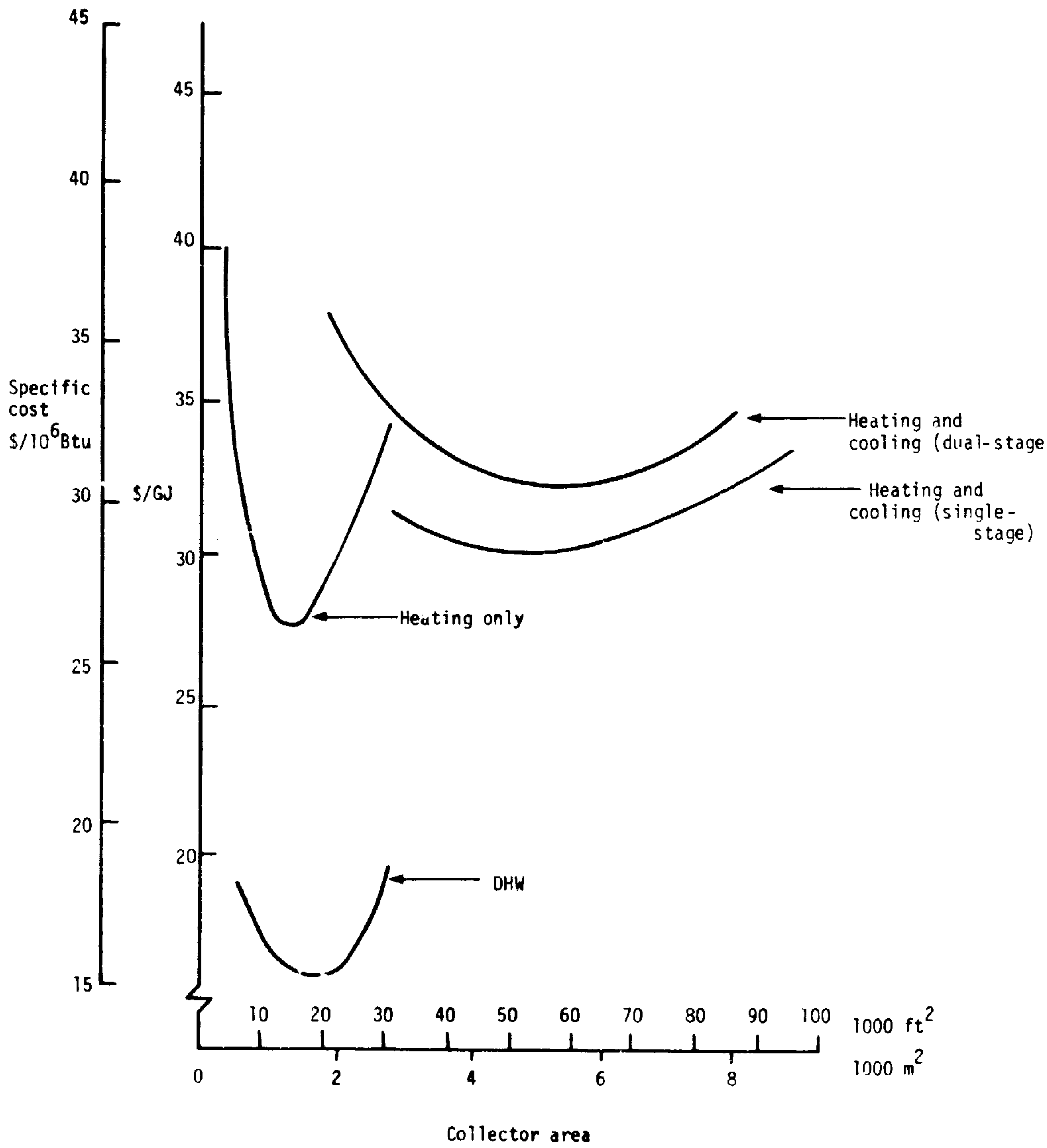


Figure 22. System specific cost vs. collector area, Ada, OK.

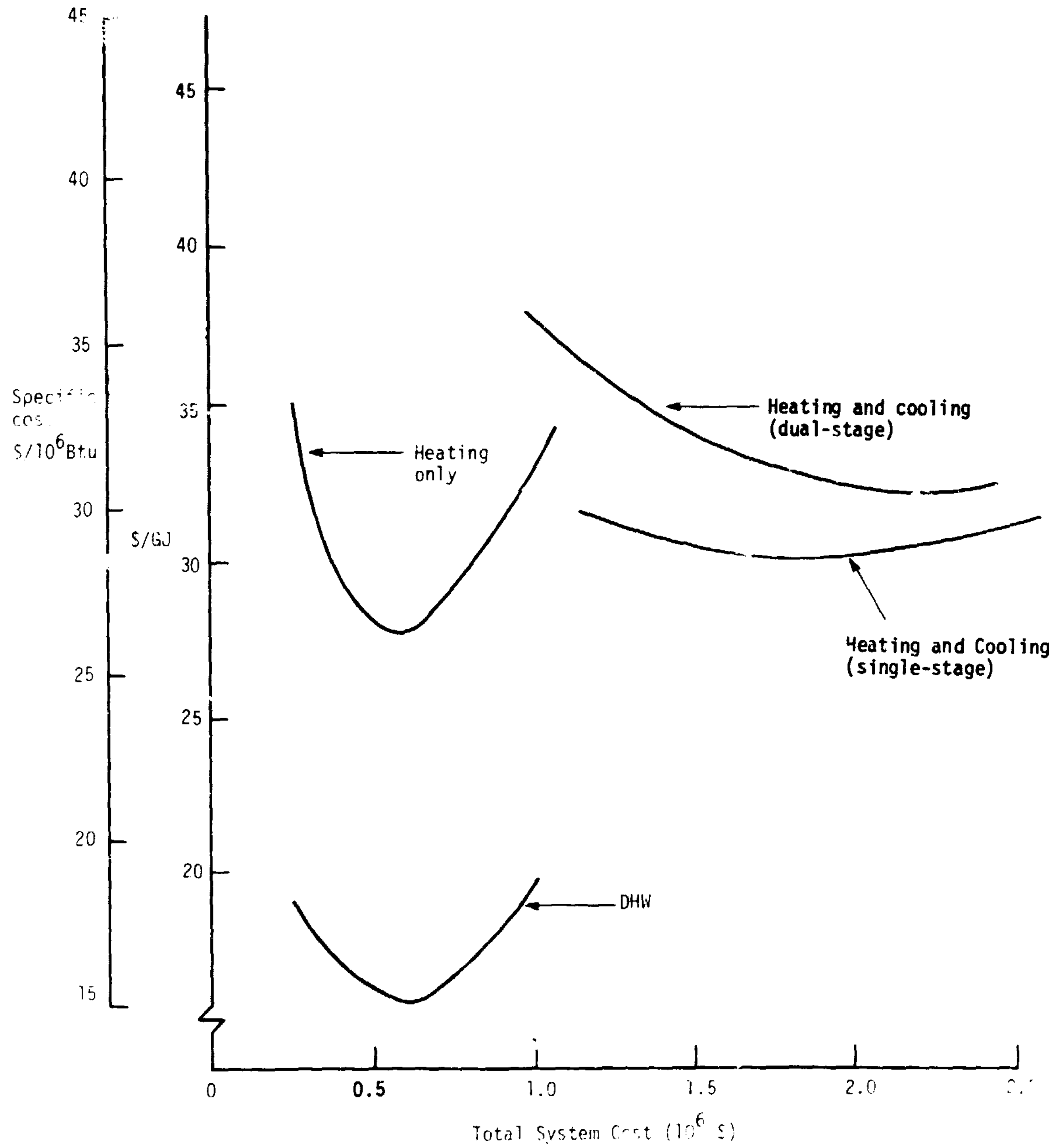


Figure 23. System specific cost vs. total system cost, Ada, OK.

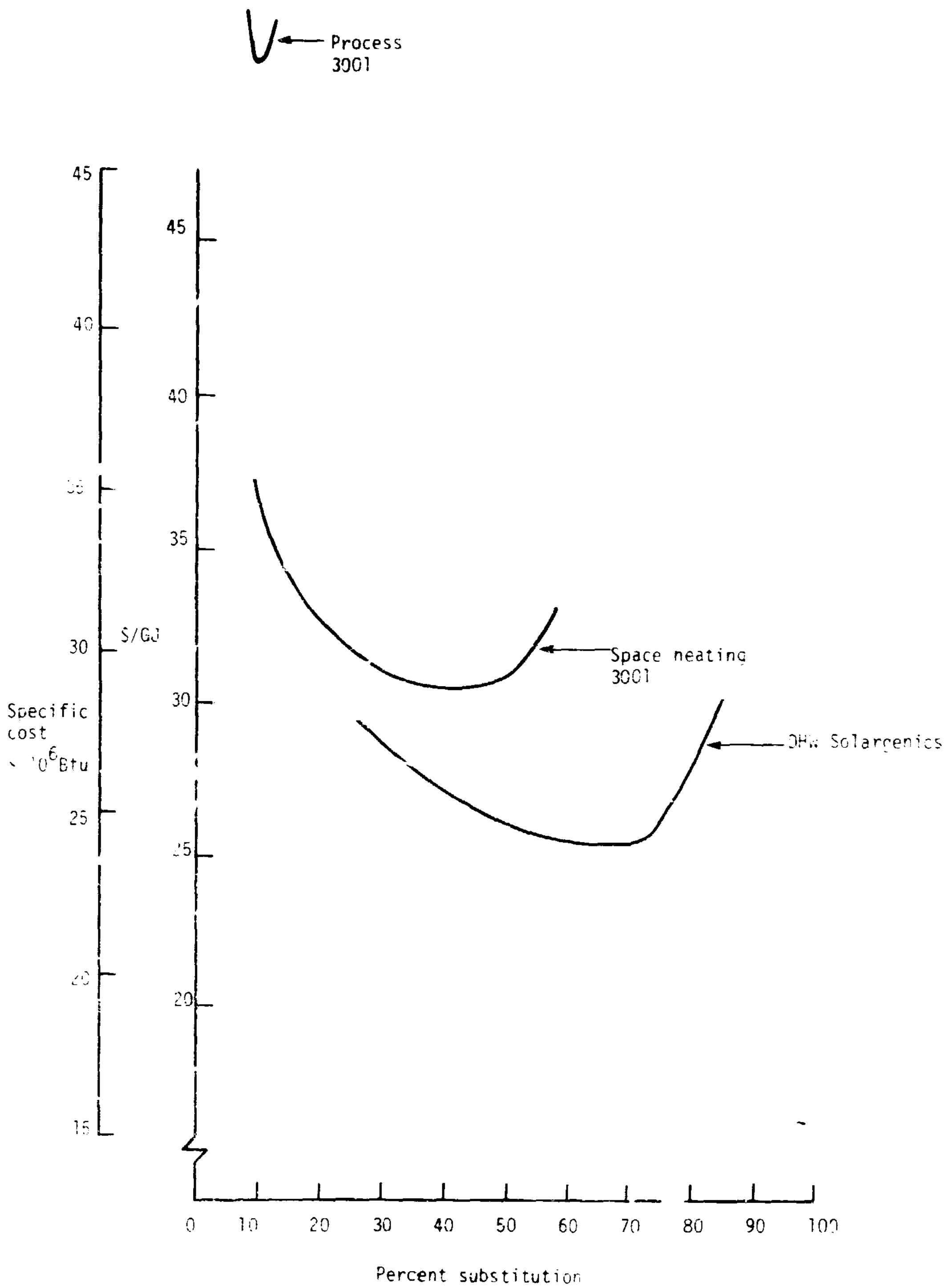


Figure 24. System specific cost vs. percent substitution, Narragansett, RI.

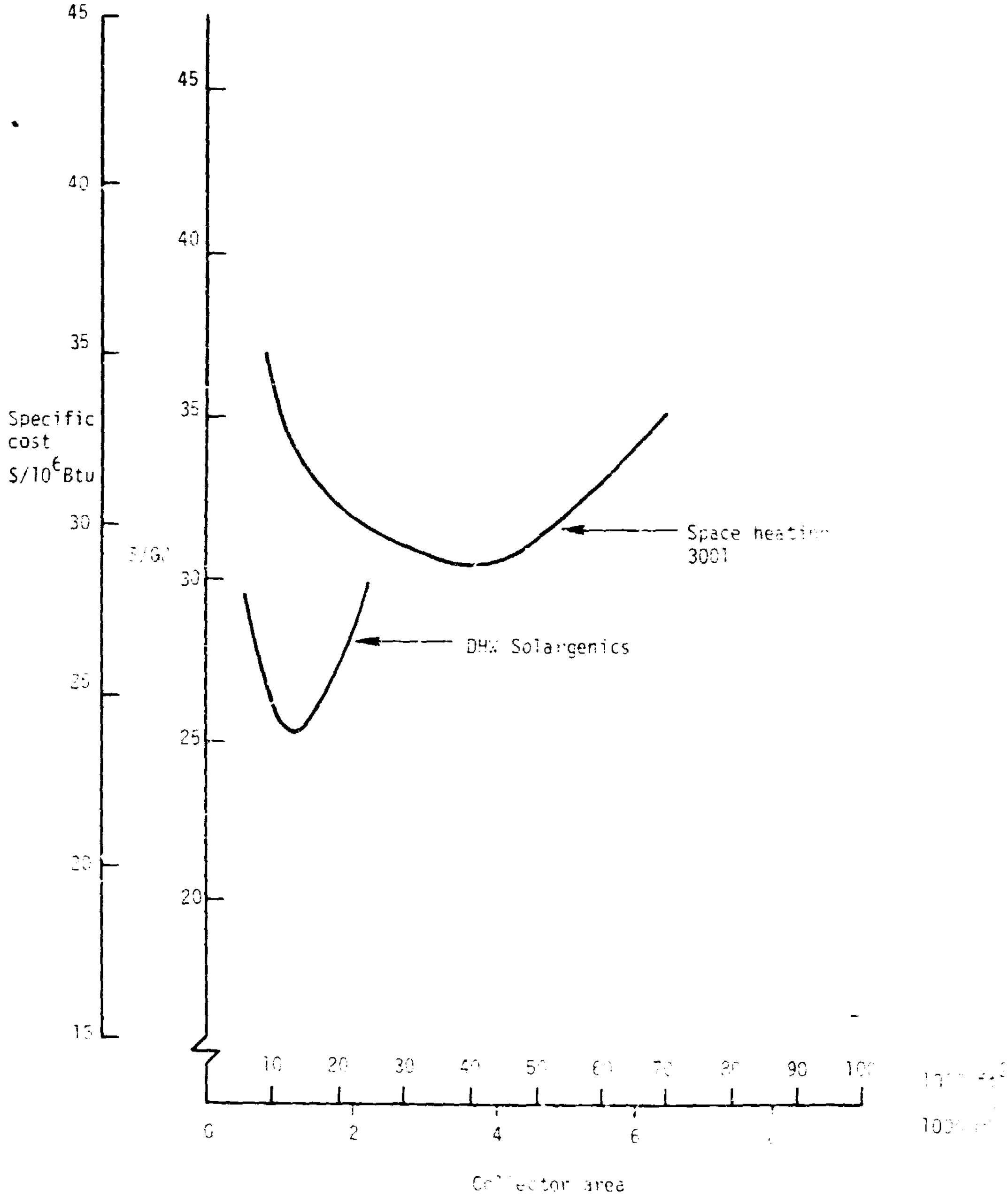


Figure 25. System specific cost vs. collector area, Narragansett, RI.

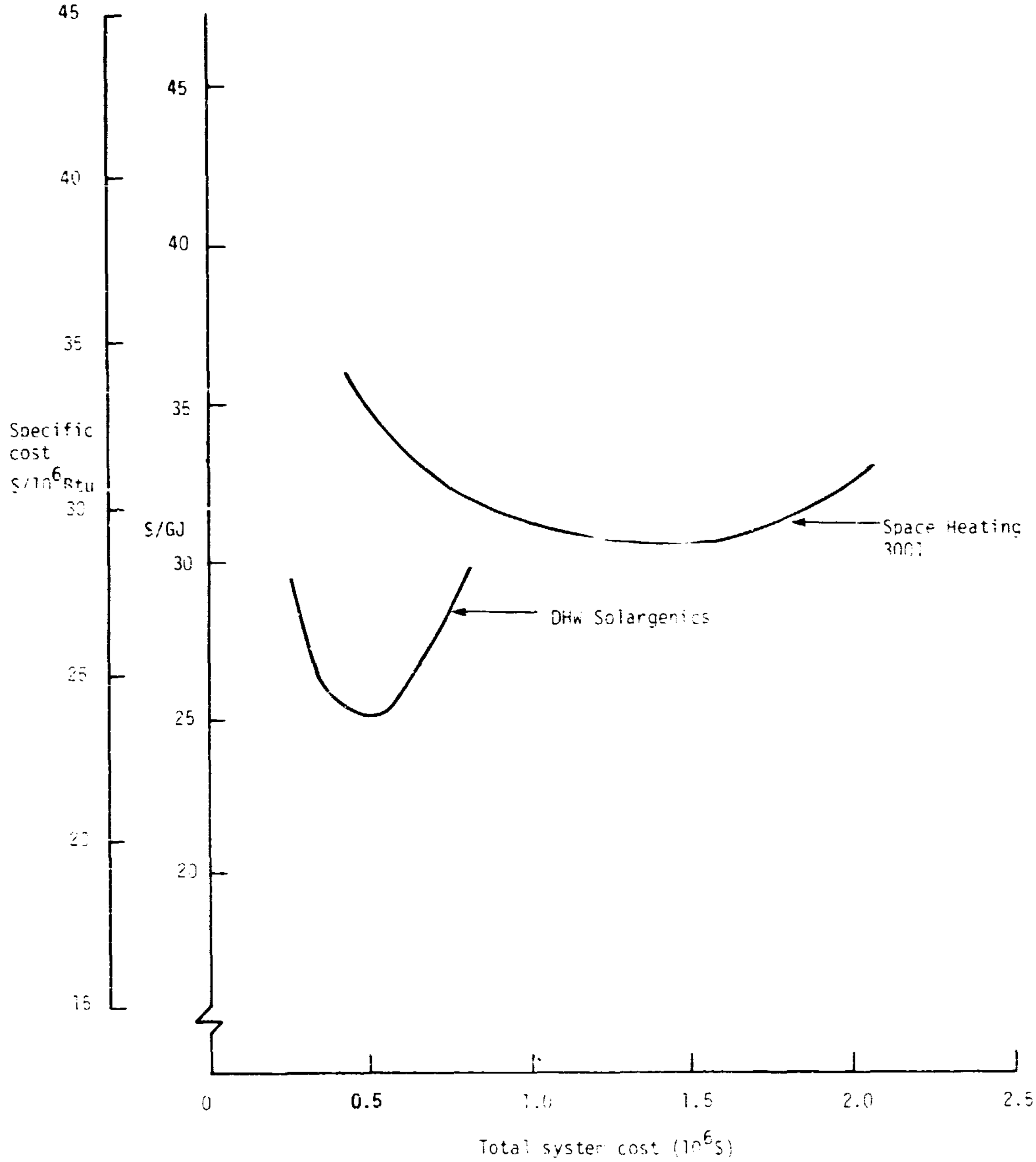


Figure 26. System specific cost vs. total system cost, Narragansett, RI.

time, the Narragansett, RI site does not have access to the two to three acres of suitable land required to locate 3,717 m^2 (40,000 ft^2) of collectors.

Other Sites--Figure 27 shows plots of DHW specific cost versus percent of solar substitution for Athens, GA, Duluth, MN, Grosse Ile, MI and Manchester, WA, as well as the plots already shown for Ada, OK and Narragansett, RI. All sites used the same system components and component costs. Therefore, the site with the greatest insolation -- Ada, OK -- had the most cost-effective system, and the site with the lowest insolation -- Manchester, WA -- had the least cost-effective system.

Summary of Results--Table 15 presents a summary of the results of the Cost/Benefit Analysis. Excluded from the table, because of high relative costs, are Ada's dual-stage and cooling-only systems. Also excluded is Narragansett's process heat system. The following points are readily seen from the table:

- DHW generally yields the highest substitution at the lowest specific cost for a given site
- Lowest specific cost for systems other than hot water was achieved by Ada's heating-only system
- Highest optimum solar fractions for systems other than hot water were achieved by Ada's combined system

Energy Displacement

For the heating systems considered, the amount of gas or oil displaced annually was calculated by dividing the energy supplied by the solar energy system by an assumed boiler efficiency of 0.8. For the cooling systems considered, the amount of electrical energy required to run a vapor compression chiller (COP = 3) was calculated based upon the cooling effect produced by the absorption chiller (COP = 0.7). The amount of fossil fuel displaced at the power plant was then calculated assuming an overall plant efficiency of 40 percent.

For each optimized system the amount of fossil fuel displaced annually was calculated. Table 16 presents a summary of the results. From the table, it is seen that the combined system at Ada, OK would displace the most natural gas, 199,000 m^3 (7,035 mcf), when the gas displaced at the facility and that at the control power plant are summed. Also, when comparing the cost of fuel displaced on an energy basis, the combined system has the advantage because a larger amount of electrical energy is displaced. The figures for fuel displacement in the DHW system should not be strictly compared with each other, because of the nature of the load information. Domestic hot water load information was generally not available, and loads were estimated as being proportional to building floor area, based upon the information for Narragansett, RI. The hot water system displacement figures could be high by as much as 50 percent.

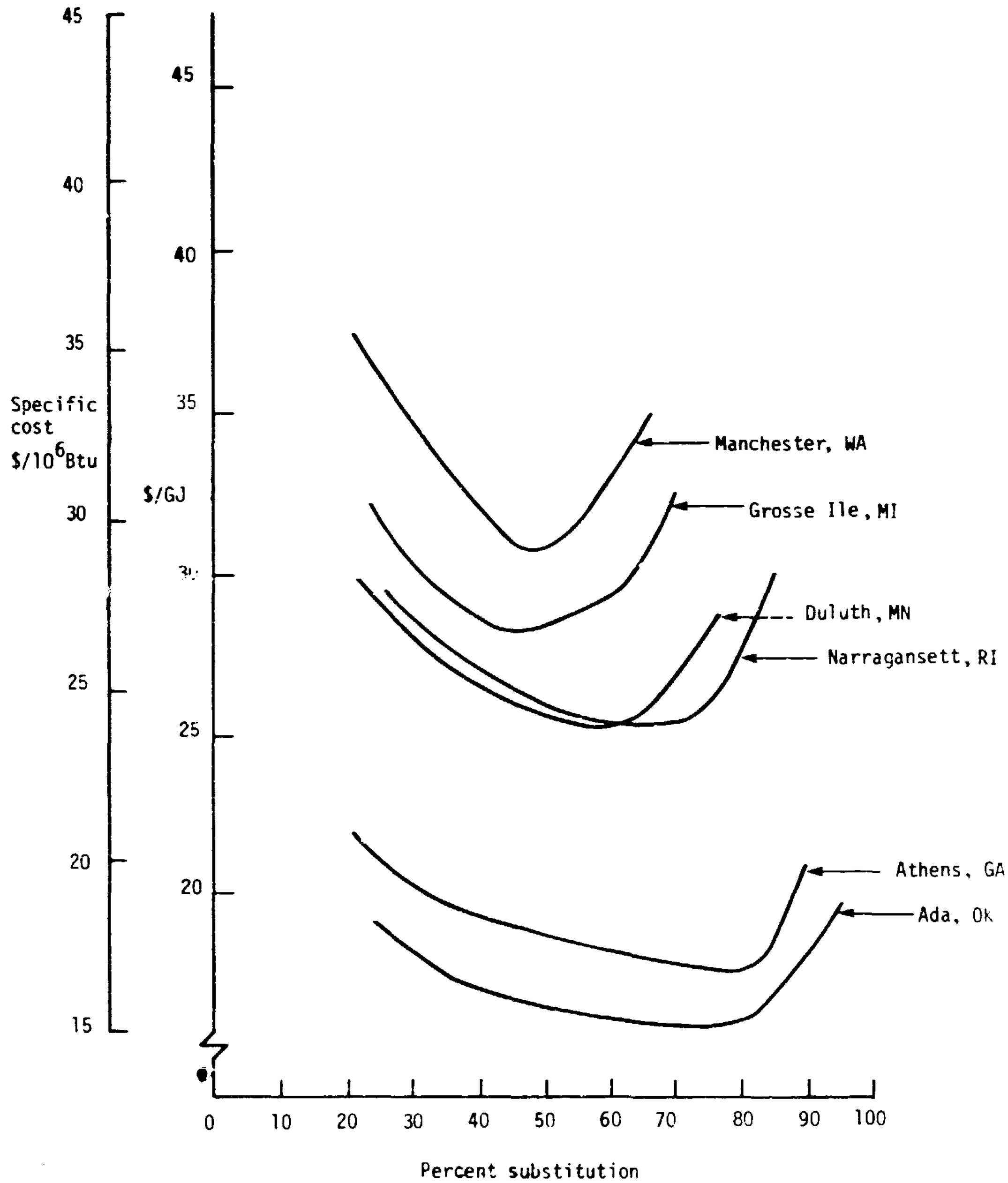


Figure 27. Comparison of system specific cost vs. percent substitution for all sites for DHW systems.

TABLE 15. SUMMARY OF RESULTS OF COST/BENEFIT ANALYSIS

Site	System	Optimum solar fraction (percent)	Specific cost $/GJ ($/10^6 Btu)	Collector area m^2 (ft^2)	Total system cost ($1000)
Ada, OK	• Heating and cooling	44	27.1 (28.6)	4,650 (50,000)	1,790
	• Heating-only	28	24.9 (26.3)	1,400 (15,000)	579
	• DHW	74	14.3 (15.1)	1,670 (18,000)	636
Narragansett, RI	• Heating-only	42	27.4 (28.9)	3,720 (40,000)	1,410
	• DHW	63	22.7 (23.9)	1,300 (14,000)	509
Athens, GA	• DHW	79	16.1 (17.0)	2,050 (22,000)	762
Duluth, MN	• DHW	60	22.8 (24.0)	1,490 (16,000)	572
Grosse Ile, MI	• DHW	54	24.7 (26.1)	1,300 (14,000)	509
Manchester, WA	• DHW	51	27.7 (29.2)	1,300 (14,000)	509

TABLE 16. ANNUAL FOSSIL FUEL DISPLACEMENT OF OPTIMIZED SYSTEMS

Site	System	Gas GJ (10^6 Btu)	Gas 10^3 m^3 (1000 CF)	Oil GJ (10^6 Btu)	Oil 10^3 l (bbl)	Electric GJ (10^6 Btu)	Electric 1000 kWh	Electric Gas at plant 10^3 m^3 (1000 CF)	Total* GJ (10^6 Btu)
Ada, OK	• Heating and cooling	4690 (4450)	126 (4450)			1090 (1034)	303	73.2 (2585)	7415 (7028)
	• Heating-only	3080 (2918)	82.6 (2918)						3080 (2918)
	• DHW	5890 (5585)	158 (5585)						5890 (5585)
Narragansett, RI	• Heating-only			6840 (6480)	169 (1064)				6840 (6480)
	• DHW			2980 (2825)	73.8 (464)				[illegible] (2825)
Athens, GA	• DHW	6280 (5955)	169 (5955)						6280 (5955)
Duluth, MN	• DHW	3340 (3162)	89.4 (3162)						3340 (3162)
Grosse Ile, MI	• DHW	2730 (2585)	73.2 (2585)						2730 (2585)
Manchester, WA	• DHW			2440 (2312)	60.4 (380)				2440 (2312)

*Electric component displaced at plant.

Environmental Effects

The possible environmental components which could be affected by the construction and operation of a solar energy system include:

- Topography
- Geology and soils
- Hydrology and water quality
- Climate and air quality
- Flora and fauna
- Resource use
- Aesthetics
- Safety

Table 17 lists the environmental impacts which might be expected as a result of the construction and operation of the main components of the systems. Since we are concerned with the impact of only one system at a time, none of these effects would appear to have serious environmental consequences.

As a part of the benefit analysis, it does not appear as though the environmental impacts would vary much from site to site. All systems would have the impacts due to construction. All systems would have the impacts caused by collector field operation, with the following exceptions: (1) glare could be greatest for systems using concentrators, and (2) higher temperatures and pressures would be attained with Ada's system generating 184 kPa (12 psig) steam. Energy storage impacts would be the same for all systems except Ada's 184 kPa (12 psig) steam system where 184 to 1136 kPa (12 to 150 psig) hot water/steam storage could present a greater safety hazzard. Finally, space conditioning operation would have the greatest impacts for cooling systems. Possible lithium bromide leakage could only occur with cooling systems.

Table 18 presents systems which could be used to monitor some environmental impacts of solar systems. Possible emissions which could affect local air quality are:

- Ni and Cr from absorber coatings
- Ethylene glycol or hydrocarbons from collector fluid
- An assortment of materials from cooling tower drift
- Li and Br from absorption chillers

All materials can be monitored using modifications of standard EPA and National Institute for Occupational Safety and Health (NIOSH) methods. Again, however, it is doubtful that any of these materials would be given off in quantities which would have a measurable environmental impact.

Summary of Benefit Analysis

Table 19 presents a summary of the results of the benefit analysis. The table is essentially a condensation of Tables 15 and 16, showing the results of the cost/benefit and energy displacement analyses. The results

TABLE 17. ENVIRONMENTAL IMPACT SUMMARY MATRIX

Impact \ Action	Construction	Collector field operation	Energy storage operation	Space conditioning operation
Topography	Moderate -- Short term* disturbance of 1/2 to 3 acres			
Geology and soils	Moderate -- Short term site grading			
Hydrology and water quality	Moderate -- Short term local siltation and erosion from clearing 1/2 to 3 acres	Moderate -- Short term increased runoff and decreased infiltration off denuded acreage: Chemical treatment for vegetation control		
Climate and air quality	Minor -- Temporary† local increase in air temperature, road and field dust, and vehicle emissions	Minor -- Short term local climatorical effects such as increased temp. and wind speed due to removal of vegetation and presence of collectors. Outgassing of collector insulation and coating, venting of fluid		Moderate -- Short term cooling tower spray of dissolved solids and additives; Localized fogging; LiBr leakage
Flora and fauna	Major -- Short term removal of whole segment of local ecosystem	Moderate -- Short term vegetation will be chemically controlled, perimeter fencing will limit access		Minor -- Short term humidity and salt fallout from cooling tower
Resource use	Minor -- Short term commitment of capital goods	Minor -- Short term collector site in dedicated industrial park		Moderate -- Short term cooling water requirement
Aesthetics	Major -- Short term human activity where little existed	Major -- Short term visual impact on visitors and staff; Glare		Moderate -- Short term cooling tower plume
Safety	Moderate -- Temporary working fluid handling; noise; waste disposal	Moderate -- Short term high temp. piping and organic fluid handling; glare; system overheat	Moderate -- Short term working fluid toxicity risk handling, and flammability; high temperature piping	Minor -- Short term toxicity of cooling tower additives in spray; LiBr leakage from chiller

*Short term is defined in terms of duration of the project. That is, the impacts would not be felt if the system were dismantled.
†Temporary is defined in terms of the construction period.

TABLE 18. ENVIRONMENTAL IMPACT MONITORING SYSTEMS

Equipment	Possible emissions	Monitor technique
Collector coating	• Ni, Cr	• Impingers with absorbers
Collector fluid	• Ethylene glycol • Hydrocarbon	• Three main methods (1) GC/MS -- All organics quantity and quality (2) GC -- Many organics quantity and quality (3) Gravimetric -- quantity only
Cooling tower drift	• Fe, Pb, As, Cl, Fl, salts	• Impingers
Absorption chiller	• Li, Br	• Impingers

TABLE 19. SUMMARY OF BENEFIT ANALYSIS

Site	System	Optimum solar fraction percent	Collector area m^2 (ft^2)	Total Fossil fuel displaced* GJ (10^6 Btu)	Total system cost (1000$)	Specific cost $/GJ ($/10^6 Btu)
Ada, OK	• Heating and cooling	44	4,650 (50,000)	7415 (7028)	1,790	27.1 (28.6)
	• Heating only	28	1,400 (15,000)	3080 (2918)	579	24.9 (26.3)
	• DHW	74	1,670 (18,000)	5890 (5585)	636	14.3 (15.1)
Narragansett, RI	• Heating only	42	3,720 (40,000)	6840 (6480)	1,410	27.4 (28.9)
	• DHW	63	1,300 (14,000)	2980 (2825)	509	22.7 (23.9)
Athens, GA	• DHW	79	2,050 (22,000)	6280 (5955)	762	16.0 (17.0)
Duluth, MN	• DHW	60	1,490 (16,000)	3340 (3162)	572	22.8 (24.0)
Grosse Ile, MI	• DHW	54	1,300 (14,000)	2730 (3162)	509	24.7 (26.1)
Manchester, WA	• DHW	51	1,300 (14,000)	2440 (2312)	509	27.7 (29.2)

*Electric component displaced at plant.

of the environmental impact analysis are not included in the table, because of the minor differences among systems and relatively small anticipated impacts of the systems.

Domestic hot water systems yield the highest solar substitution at the lowest specific cost, because of the good load match and lower collector operating temperatures. The most cost-effective DHW systems could be installed at Ada, OK and Athens, GA because of their superior insolation.

Space heating at Ada, OK is the next most cost-effective system. At solar substitutions higher than about 37 percent, combined heating and cooling is the most cost-effective solar system.

FINAL SELECTION

This subsection presents the final selection of a primary system for a more detailed analysis. Included are the Final System/Facility Matrix, the ranking of the eight systems included in the matrix according to the Final Criteria, the best generic solar system for general application to EPA facilities and the final system selection for a more detailed analysis.

Final Matrix

The Final System/Facility Matrix is presented in Table 3. The matrix is the product of the combined results of the facility survey and the Cost/Benefit Analysis. Existing mechanical system characteristics, available insolation, and building load characteristics were considered in arriving at the final configuration. Available collector site area was a prime consideration in determining the feasibility of certain system/facility combinations.

The reasons for not including certain system/facility combinations are shown in the table. For example, the most cost-effective space heating system, based upon the Cost/Benefit Analysis described above, for Narragansett, RI would require about 3720 m^2 (40,000 ft^2) of collector area. A typical packing density requires about three times as much siting area as collector area. That indicates that Narragansett, RI would need almost three acres for the optimized heating system. For this reason, the otherwise relatively attractive heating system was eliminated from the Final System/Facility Matrix.

System/Facility Ranking

Only the system/facility combinations presented in the Final System/Facility Matrix were considered for ranking. These systems were ranked on the basis of the Final Criteria discussed in Section 2 and a criteria weighting scheme. Note that the first criterion, that of sufficient collector site area, has already been applied in developing the Final System/Facility Matrix. The remaining criteria and their relative weighting were:

- Relative cost effectiveness (75 percent)
- Fossil fuel displacement (15 percent)
- Environmental effects (10 percent)

Cost effectiveness was the single most important criterion for ranking the systems (provided that there was sufficient collector site area).

A total of 100 points were possible in the ranking process. These were alloted to each criterion according to the weighting scheme above. For example, 75 points were available for cost effectiveness. For cost effectiveness and fossil fuel displacement, the system which best satisfied each criterion was assigned the maximum number of points available, and the system which trailed all others for a given criterion was assigned zero points. The remaining systems were assigned points, between zero and the maximum for the criterion, proportional to their cost effectiveness (cost per unit energy) or fossil fuel displacement.

Since quantitative information was not available concerning the environmental effects of various systems, points for that criterion were assigned by a different procedure. The minimal effects, summarized in Table 17, vary little from system to system. There is, however, a slightly higher risk associated with Ada's pressurized steam system. Also, for Ada's combined heating and cooling system, there is the added possibility, however remote, of refrigerant leakage. Points for this criterion were therefore awarded as follows:

- All DHW systems -- 10 points
- Ada, OK space heating -- 5 points
- Ada, OK combined heating and cooling -- 0 points

Table 4 summarizes the results of the ranking process. From this table, it is seen that the systems can be ranked in the following order:

1. Ada, OK DHW
2. Athens, GA DHW
3. Duluth, MN DHW
4. Narragansett, RI DHW
5. Grosse Ile, MI DHW
6. Ada, OK space heating
7. Ada, OK combined heating and cooling
8. Manchester, WA DHW

Domestic hot water, in general, was the top-ranked system. This is primarily due to the greater cost effectiveness, resulting from lower collector operating temperatures, and to the greater weight given to cost effectiveness. DHW for Manchester, WA was the exception, due in great part to the very low insolation at that site.

Best Generic Concept

Based upon the results of the Benefit Analysis, as summarized in Table 19, and the System/Facility Ranking it is clear that the solar application with the widest applicability is DHW. The lower collector fluid temperature required presents the operating regime in which some flat-plate collectors operate at greater efficiencies than concentrating collectors. Furthermore, the lower operating temperatures permit cost-effective system operation over a greater geographical region.

Another reason for the overall attractiveness of solar DHW systems is that, because of the smaller loads and lower temperatures, they generally require significantly less collector site area than either space-heating or space-cooling solar systems. Adequate collector site area is a basic requirement of any solar installation, and many of the EPA facilities in the initial screening have less than a half acre of clear, unobstructed and unshaded available space.

Primary System

The primary system selected for a more detailed benefit analysis was the solar steam space heating system for the Robert S. Kerr ERL in Ada, OK. Although solar DHW systems are generally the most cost-effective systems at this time, they are also the most commonly installed and operating systems in this country. Ada's solar steam space heating system proved to be the most cost-effective system after DHW, and also offers a novel approach to solar space heating in a large facility.

SECTION 4

ANALYSIS OF PRIMARY SYSTEM DESIGN

This section describes the detailed optimization procedure used on the primary system configuration and presents the results of a 12-month performance simulation and a benefit analysis for this optimized system.

DESIGN SUMMARY AND ANNUAL PERFORMANCE

The solar steam generation facility consists of four major subsystems: (1) the collector field, (2) the boiler/storage tank, (3) the steam distribution plant interface, and (4) the control/instrumentation center.

As indicated in figure 2, pressurized water is circulated through a distributed collector field of parabolic trough concentrators where it is heated and throttled back into a pressurized storage vessel. Pressure in the field is maintained above the saturation pressure of the fluid by means of a differential pressure regulating valve. The regulating valve is set so that the potential temperature rise in the field during peak insolation conditions is always less than the temperature rise that would allow boiling in the field.

Steam and water coexist in the 791 kPa (100 psig) pressure vessel which serves as a combination flash boiler/storage tank. Steam is delivered to the load through a pressure-reducing valve to maintain proper conditions for the plant interface. Condensate from the plant is returned to the boiler/storage tank to maintain a relatively constant level in the tank. Table 20 lists the characteristics of the collector and storage subsystem as determined by the optimization/performance procedure detailed in Appendix A. The costing of the system for the benefit analysis was based upon these characteristics and Acurex's previous experience in designing and building solar energy systems of this scale. The two primary factors which determine system cost, found in Table 20, are the collector field area, 1,561 m^2 (16,794 ft_c^2), and flash boiler/storage tank volume of 139,000 l (36,800 gallons).

The performance simulations, detailed in Appendix A, led to the selection of the optimum configuration (described above) based on annual performance. System performance was determined for 6 months (for alternating months, starting with February) using the computerized code described in Appendix A. The monthly distribution of the fraction of the load met by the solar steam system (solar substitution) is shown in

TABLE 20. COLLECTOR AND STORAGE SUBSYSTEM SUMMARY

Collector field	
-- Type	Parabolic trough
-- Orientation	North-south
-- Row spacing	4.57 m (15 ft) center to center
-- Total aperture area	1561 m^2 (16,794 ft^2)
-- Collector modules	36
-- Fluid flow rate	363 l/min (96 gpm)
-- Maximum temperature	452 K (354° F)
-- Maximum pressure	979 kPa (142 psia)
Boiler storage	
-- Type	Varying pressure steam accumulator
-- Tank volume	139,000 l (36,800 gal)
-- Maximum temperature	441 K (334° F)
-- Maximum pressure	751 kPa (109 psia)
-- Energy storage capacity	18.7 GJ (17.7 x 10^6 Btu)

Figure 28. The distribution resulted in a solar substitution of 35 percent annually.

OPTIMIZATION PROCEDURE

This section presents the methodology used to optimize the solar steam space heating system, shown schematically in Figure 2. The detailed procedure is presented in Appendix A.

The procedure considers the effects of component performance on overall system performance in selecting the most cost-effective system design. Classical engineering economic analyses, employing the concept of minimum life cycle cost, were used to determine the optimum or most economical system to supply the required energy.

The procedure is more detailed than that employed in Section 3. As with the simple analysis, the figure of merit by which alternative configurations were compared, was cost per unit energy supplied to the building load. The primary advantages over the simpler approach are more detailed:

- Component costing
- Load analysis
- Insolation data
- Performance analysis

For a detailed description of the performance simulation and optimization methodology, see Appendix A.

BENEFIT ANALYSIS

This section describes the benefit analysis performed on the optimized primary system with respect to life cycle cost and energy displacement.

Life Cycle Cost

The after tax Life Cycle Cost (LCC) for the proposed system was based on the formula:

$$LCC = CRF\ (1-ITC)\ (C) + (1-t)\ OC - t(D) \qquad (3)$$

where CRF = capital recovery factor

ITC = investment tax credit

C = capital costs

t = weighted federal and state tax rate

OC = operating costs

D = depreciation, straight line

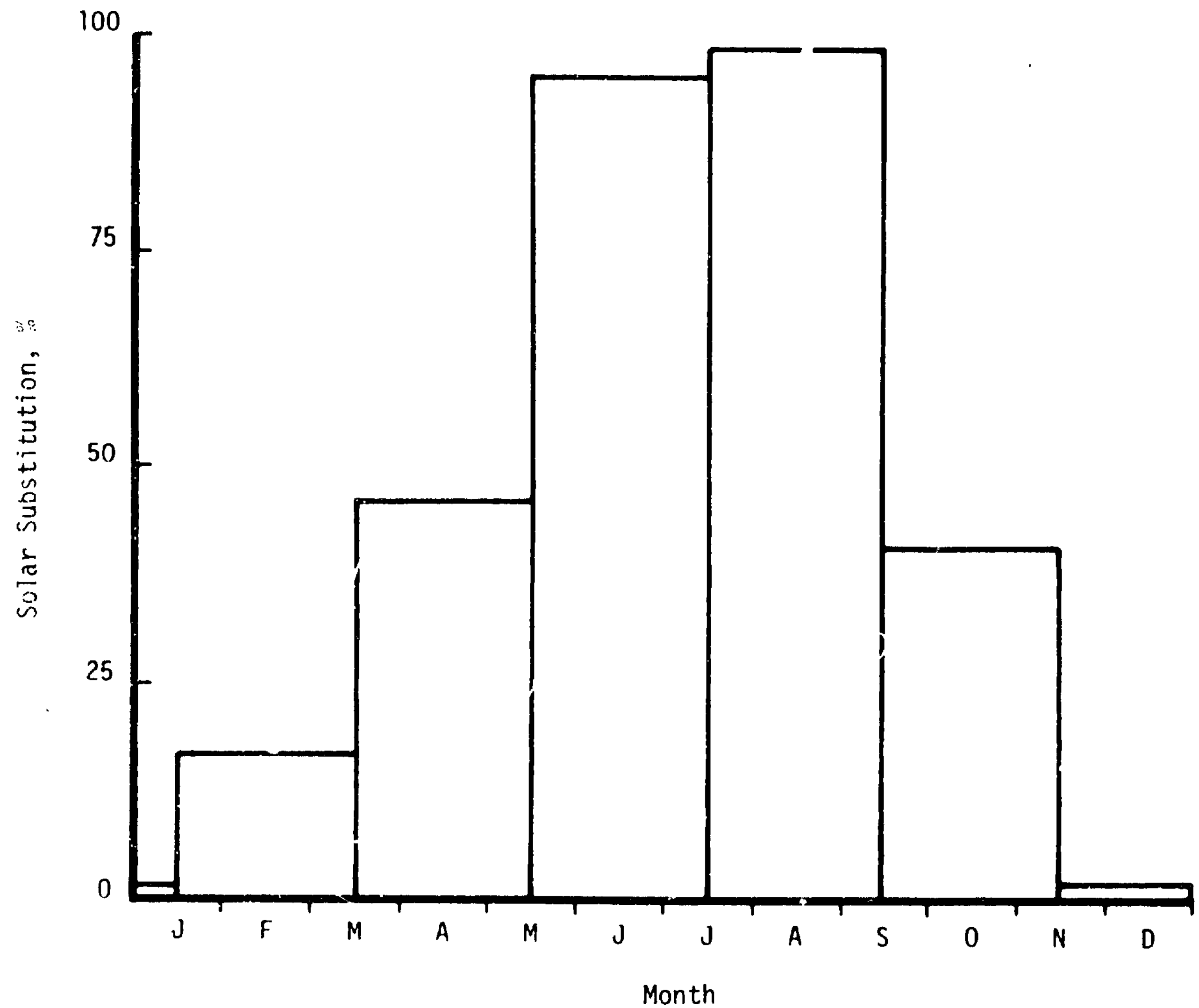


Figure 28. Monthly distribution of the load supplied by the solar steam heating system.

Since the facility for which the solar system is designed is government owned, the terms related to tax benefits (i.e., ITC, t, D) must be excluded, resulting in the simplified equation:

$$LCC = CRF\ (C) + OC \tag{4}$$

Based upon a 20-year lifetime and a 10 percent discount rate, CRF = 0.11746.

Operating costs are calculated from the annual parasitic consumption (calculated in the performance simulation) multiplied by the Levelized Fuel Cost (LFC). The LFC is the product of the Discount Escalation Factor (DEF), the CRF and the current cost of electricity at the Ada, OK facility.

Based upon a DEF of 20 (20-year lifetime, 10 percent projected real growth rate, 10 percent discount rate), a current cost of electricity of 3.05¢/kWh and a CRF of 0.11746, the LFC is 7.17¢/kWh. Based on the calculated annual parasitic consumption of 23,138 kWh for the optimized system, the operating costs are then $1,658/yr.

Table 21 summarizes the capital cost of materials and installation of the optimized system. The total cost was calculated to be $617,800. This cost does not include design and other program costs, only materials and installation.

TABLE 21. SUMMARY OF SYSTEM COSTS

Description	Cost (1000 $)
Collectors (including foundations, shipping, installation)	357.9
Boiler/storage tank (including foundation, shipping, installation and insulation)	69.0
Piping, pumps, valves	71.3
Instrumentation	60.0
Electrical power	36.1
Site preparation (including clearing and grading)	20.0
Temporary construction building	3.5
Total	617.8

With the above information, Equation (4) yields an annual LCC of $74,225/yr.

Energy Displacement

One of the major benefits of solar energy systems is the displacement of fossil fuel as an energy source. Figure 28 presents a monthly breakdown of the portion of the space heating load at Ada, OK that could be supplied by the proposed solar system. Furthermore, on an annual basis, it was determined that approximately 35 percent of the load could be provided by this system. This was calculated by dividing the annual energy suppled to the building load by solar by the total annual space heating load.

In order to determine the amount of fossil fuel displaced by the solar system, however, the thermal equivalent of the parasitic requirements for the system must be subtracted from the gross thermal output of the system divided by the facility's boiler efficiency. The electricity consumption for the pumps is expressed as thermal energy after accounting for power plant conversion efficiencies (1 J electric = 2.5 J Thermal). A boiler efficiency of 70 percent is assumed for the facility. Therefore, the net energy displaced can be expressed by

$$E_{NET}\ (J) = (1.43)\ E_{GROSS} - (2.5)\ (3.6 \times 10^6) P \tag{5}$$

where E_{GROSS} = gross thermal output of solar system (J)

P = electrical consumption by solar-related parasitics (kWh)

It was determined from the performance simulation that about 23,000 kWh/year would be required by the solar system parasitics. The gross thermal energy supplied by the system was calculated to be 2,616 GJ (2,480 10^6 Btu).

According to the above equation, the net energy displaced by the proposed system would therefore be 3,530 GJ (3,350 million Btu). This is the equivalent of approximately 96,200 cubic meters (3.4 million cubic feet) of natural gas which would be saved every year.

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APPENDIX A

DESIGN METHODOLOGY

This appendix describes the process used to determine system performance for the flash boiler/storage configuration described in Section 4, and to establish the cost optimized solar steam system. Included are the optimization methodology adopted for this analysis, a description of the system performance model and computer code, and the optimization results.

COST OPTIMIZATION METHODOLOGY AND APPROACH

An optimization procedure is essential in designing a solar energy system that consists of interrelated subsystems. Optimization defines an integrated set of design parameters for which a cost-effective and reliable energy system can be constructed. This procedure considers the effects of component performance on overall system performance. The result is a "best" system designed by the optimization criteria. The following section describes the optimization criteria and constraints established for this program.

Optimization Criteria and Constraints

In this section, criteria are established that will lead to selecting an economical design for the configuration described in Section 4. An energy system design should be selected for construction based on an economic analysis. Classical engineering economic analyses employ the concept of minimum LCC to define the optimum or most economical system to supply the required energy. This is the system whose ownership and operating costs over the project lifetime are smallest for those alternatives that can satisfy the total energy demand. To apply this approach to the design of solar energy systems, the total LCC (including conventional fuel makeup) for the energy system is calculated and minimized by selecting appropriate system parameters such as collector area and storage volume. The optimization objective is to find the proportion of conventional and solar energy that minimizes the overall cost of meeting the energy demand. Figure A-1 illustrates the results of such an analysis. For this example, an optimum system configuration is clearly defined. This example assumed the cost of conventional fuels (levelized for price escalation over the project lifetime) exceeds the cost of the optimum energy system. Therefore, the optimum design combines that proportion of conventional and solar energy sources that minimizes total LCC for the output required.

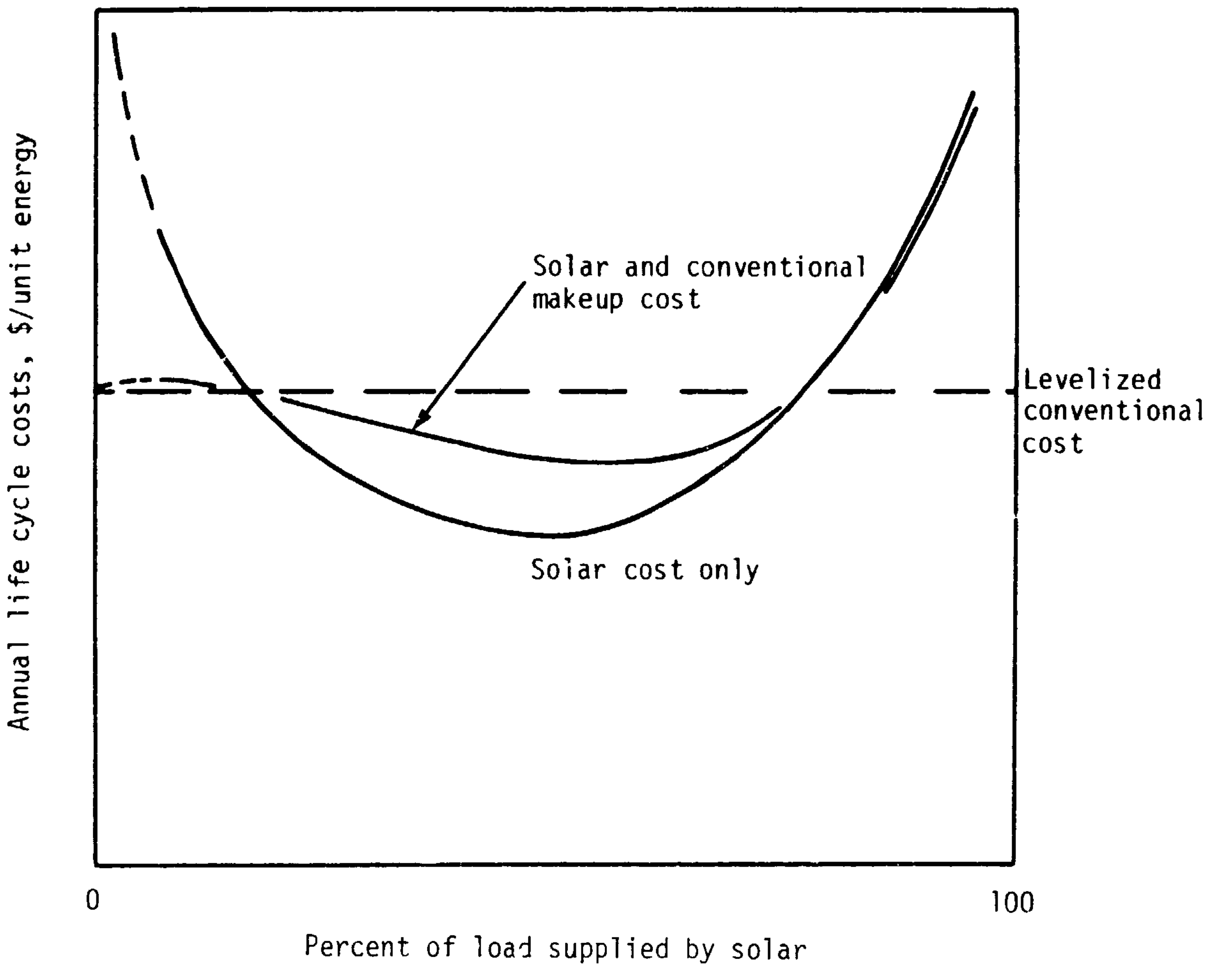


Figure A-1. Life cycle costs: solar more economical than conventional energy system.

However, based on current installed solar equipment costs, existing tax laws, and projected conventional fuel costs, economic analyses typically give the result shown in Figure A-2. As indicated, the levelized cost of conventional fuels is less than the total cost of the solar energy system (solar and conventional costs); therefore, the optimum energy system, based on total system costs, consists of a 100 percent conventional fuel system. As conventional fuel prices continue to escalate, solar equipment costs decrease, and solar tax incentives are implemented, the economics of solar energy in relation to conventional systems will improve.

Since a solar energy system is to be built, the goal of the system designer should be to design the most cost-effective solar system. System optimization, therefore, should be based on minimizing solar costs only, excluding conventional fuel makeup costs. This criterion will define the solar energy system that provides maximum output for minimum cost (see Figure A-2). On the basis of solar costs and output alone, an optimum design is defined. Above this optimum, a proportionate increase in system size (cost) does not produce a proportionate increase in output (energy delivered). Similarly, a proportionate decrease in system output from the optimum does not result in a proportionate decrease in cost, because system costs include fixed costs not sensitive to system size.

For this demonstration, the optimum system design has the minimum LCC per unit of energy supplied by the solar energy system, excluding conventional energy makeup costs. This energy output will be defined as the gross thermal output of the solar energy system minus the thermal equivalent of parasitic power requirements for the system. For example, electricity for pumps is expressed as thermal energy after accounting for conversion efficiencies, 1 J electric = 2.5 J thermal. This definition will ensure selecting a design that maximizes the displacement of conventional fuels and should prevent misleading optimization results based on gross solar output. Misleading optimization results occur because cost tradeoffs for some system parameters, such as collector field flow rate, are sensitive to parasitic power requirements. Since the marginal cost of parasitic (conventional) energy is lower than the overall system cost, optimizations of flow rate based on gross, rather than net, energy output will favor a system with a higher parasitic power (higher flow rates). Figure A-3 illustrates that an optimum flow rate is clearly defined when system output is defined as net thermal energy supplied to the load by the solar energy system. This definition of system output was used throughout the optimization task and was important in selecting system parameters affecting parasitic power requirements. Although these parameters are of secondary importance in the optimization, since collector area and storage volume predominate, they should not be ignored because parasitic power requirements are roughly 10 percent (expressed as thermal energy) of the total output of the system.

Optimization Procedure

The optimization procedure for this analysis provided a rational approach for sizing major system components and determining important

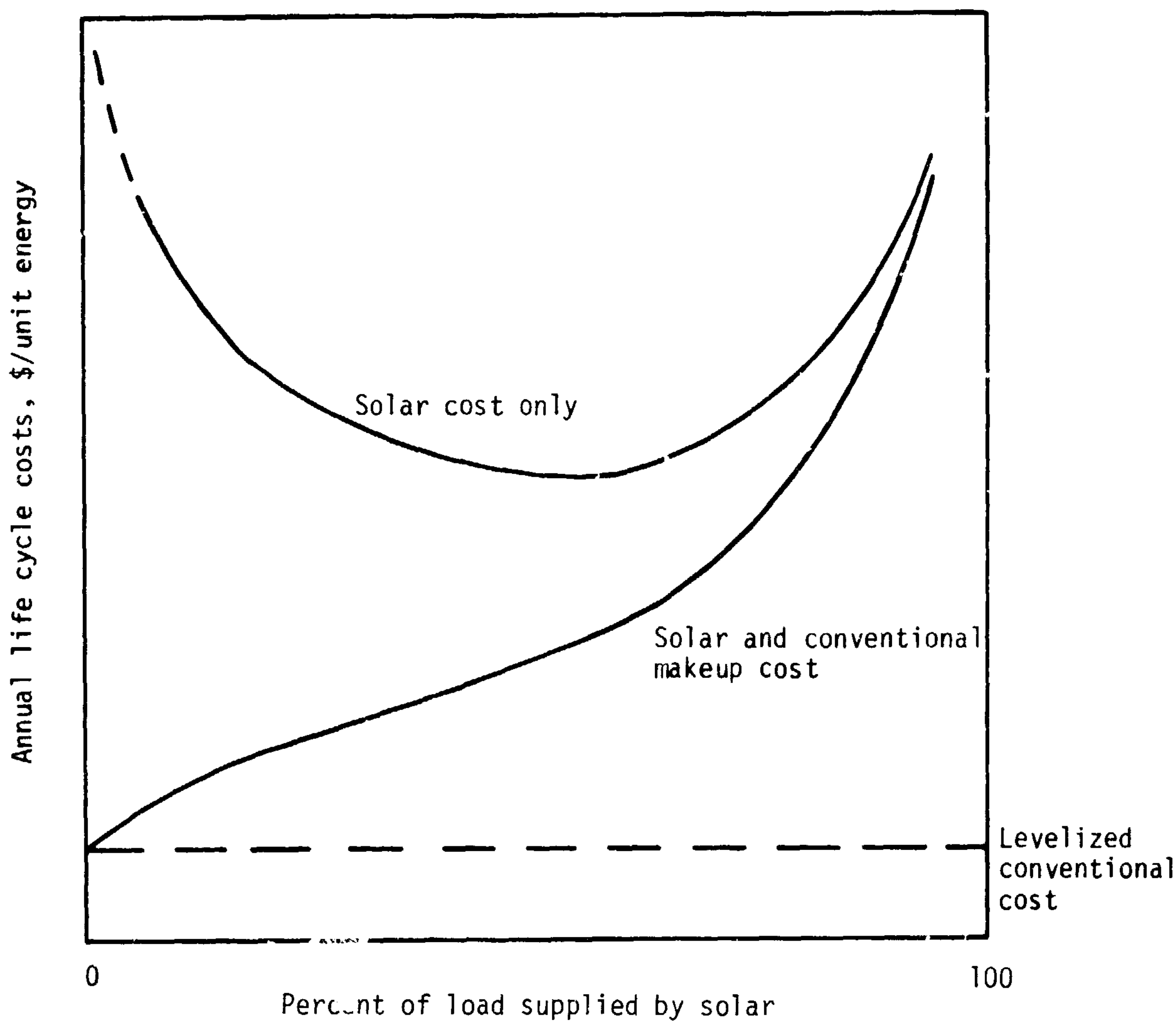


Figure A-2. Life cycle costs: conventional more economical than solar energy system.

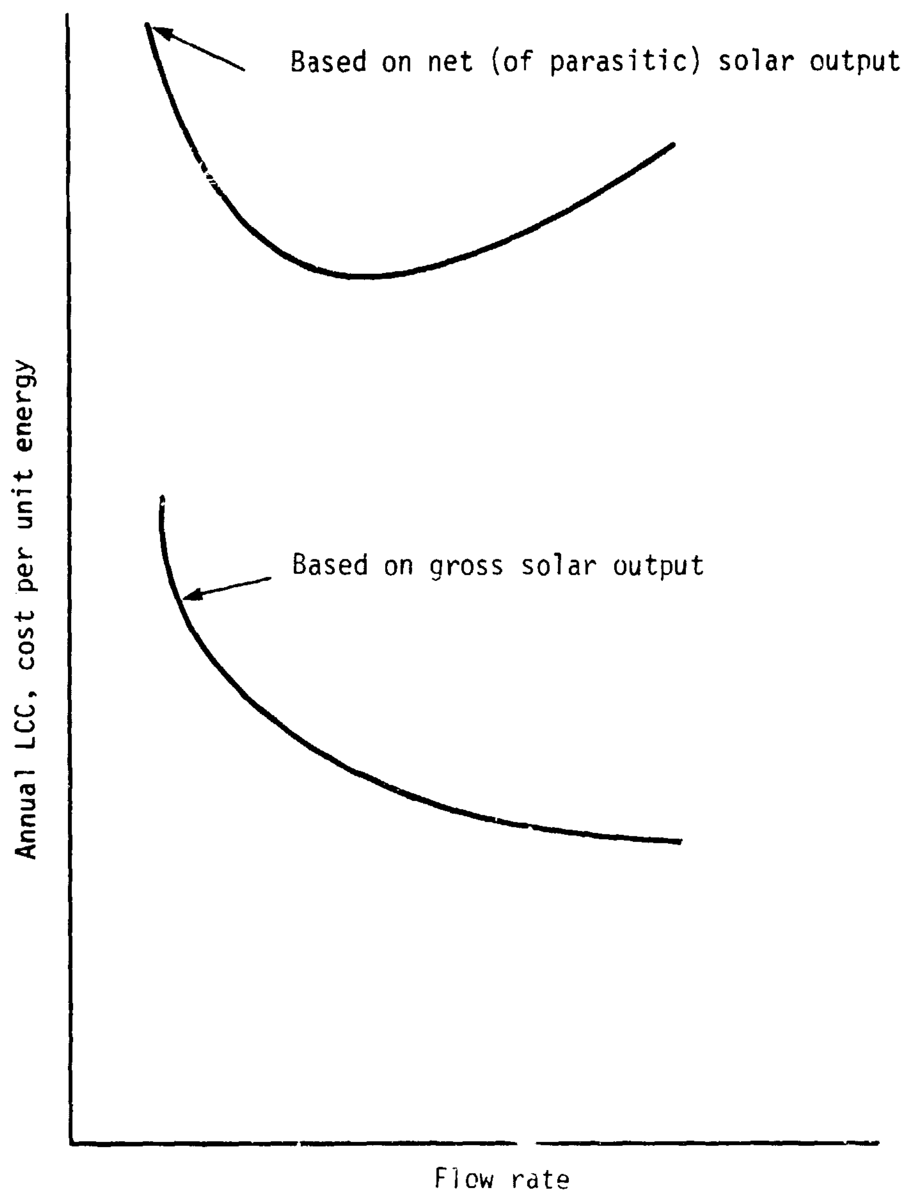


Figure A-3. Comparison of flow rate optimization based on gross and net solar output.

system parameters. The procedure combines engineering judgment with the results of computer performance simulations to determine a cost-effective design in a practical manner. System components and parameters that most affect annual performance were determined first, then secondary parameters were investigated.

Optimization consisted of determining the best:

- Collector field size and storage volume combination
- Maximum operating pressure
- Field flow rate

As each component value was varied, less sensitive parameters were held constant. Less sensitive parameters, such as flow rate and collector row spacing, were nominally optimized based on previous experience. By proceeding from the most sensitive to least sensitive components, we minimized the impact of errors in nominal sizing.

SYSTEM MODELING AND DATA INPUT

This subsection describes the system model and data requirements that were used to evaluate the performance of the flash boiler/storage configuration described in Section 4.

System Model

The solar steam heating system consists of a number of components which are interconnected to provide steam to the space conditioning heat exchanger. The system model, therefore, consists of several mathematical models that perform mass and energy balance calculations for the various components. For example, the transient response of the system (due to variations in insolation, ambient conditions, and steam load demand) was determined by using a simple numerical technique that solved the ordinary differential equations for mass and energy balance of the flash boiler/storage tank. The system performance simulation also accounted for the energy losses due to collector inefficiencies, collector field warm-up and freeze protection, and storage tank and pipe wall losses.

The numerical integration scheme used a 1-hour time step during which steady-state component performance was assumed. This time interval was small enough to adequately model the variations in heating load, and warm-up and freeze protection transients, but did not incur excessive computer operating costs. For each time increment, we computed the rates of energy collected or lost by the collector field, manifold piping, and storage tank, and the energy withdrawn from the tank to meet the heating load. At the end of the time step, the net change in storage tank energy was computed, the new fluid state in the tank was determined, and the next time step was intitiated with a new set of constant tank fluid state, load, insolation, and weather values. This cycle was continued until the full simulation period had been completed.

The system control functions were used by the model to properly simulate system responses to insolation, weather, and load variations. The operation of the collection and load loops are basically independent, coupled only by the state of the storage tank. As stated above, one of the design parameters was maximum storage pressure. In all cases, the minimum storage pressure allowed for solar system operation was 274 kPa (25 psig) in order to effect efficient displacement of the 184 kPa (12 psig) steam in the existing system. The nominal maximum allowable pressure was 1074 kPa (141 psig), allowing a 6 percent safety margin for a 1,136 kPa (150 psig tank). The control functions included in the system model are described briefly for the nominal case in the following paragraphs.

Whenever the storage tank temperature is above 404 K (268° F), the steam load is met by the solar energy system. When tank temperatures fall below this value, the load is supplied by the existing plant steam system.

When the tank temperature is less than 441 K (334° F) and the insolation level exceeds 315 W/m^2 (100 Btu/ft^2-hr) (tracking is difficult at lower levels), the collector field circulation pump is activated, and fluid is circulated through the collector field at a constant flow rate and returned to the tank, thereby adding energy to the tank. Whenever the outlet temperature from a loop of collectors exceeds the set point of 452 K (354° F), collectors are desteered in groups of eight (eight collectors are mounted on a common tracking drive) to reduce the temperature. When the storage tank temperature reaches 441 K (334° F), the collector field is stowed and the field circulation pump is shut off.

Whenever the collector or manifold fluid temperature drops below 275 K (35° F), the freeze protection pump is started to increase the temperature. Some of the cold water is withdrawn from the pipes, and replaced with hot water from the tank, until the temperature of the water is increased to 283 K (50° F).

During morning start-up, the cold fluid in the field piping is initially flushed out with hotter tank fluid, causing only a slight drop in storage tank temperature, and greatly simplifying warm-up control requirements.

Computer Simulation of System Performance

Annual system performance was simulated with a computer code based on the system model previously described. The simulation scheme is illustrated in Figure A-4. The code consists of a central driver and 14 subroutines which model an individual subsystem or component. A basic flow chart of the calculation procedure is presented in Figure A-5. The central driver and major subroutines are described in the following paragraphs.

Driver--

Figure A-4 illustrates that the performance simulation is controlled by the central driver. As indicated in Figure A-5, the driver controls

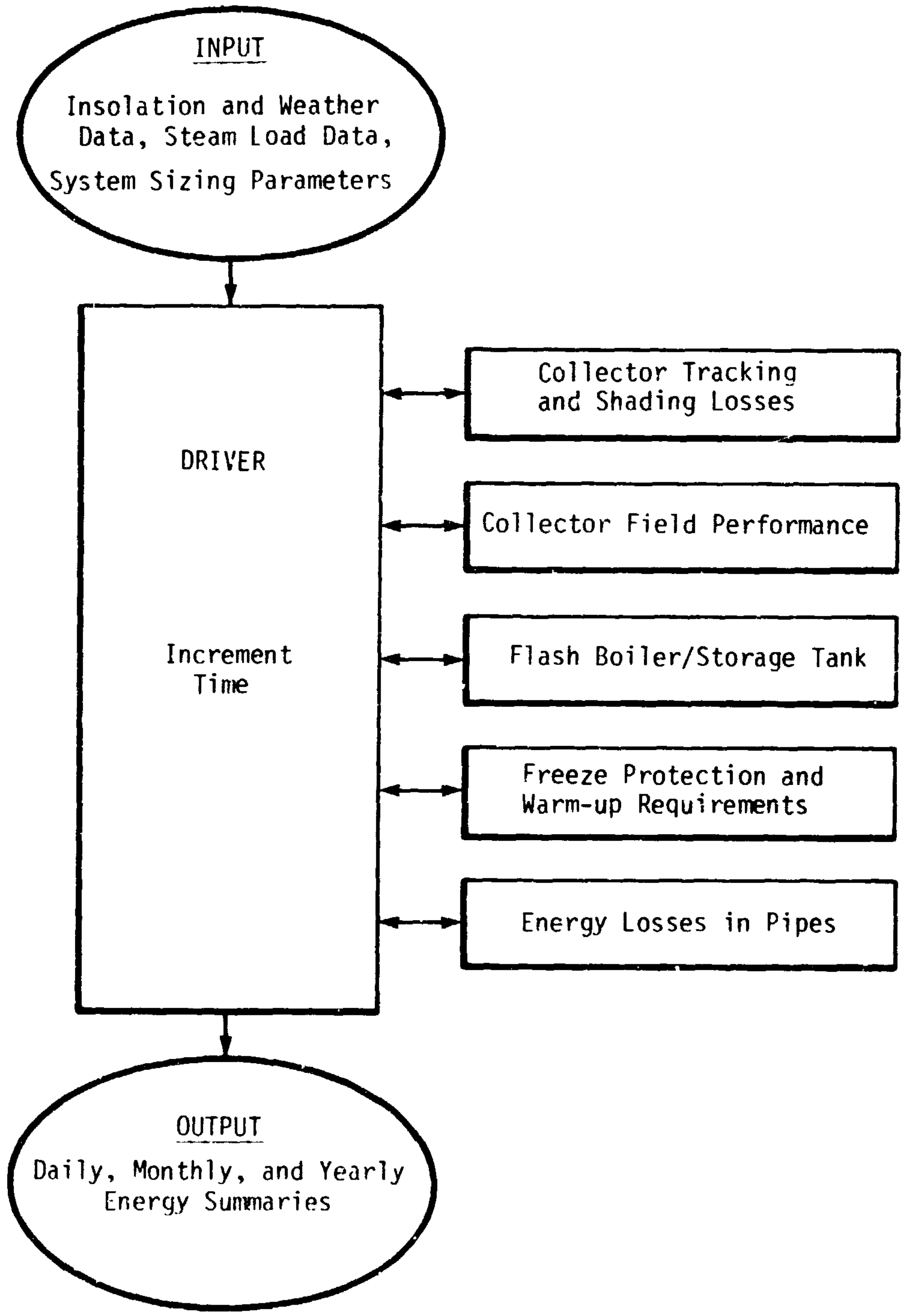


Figure A-4. Schematic of computer peformance simulation.

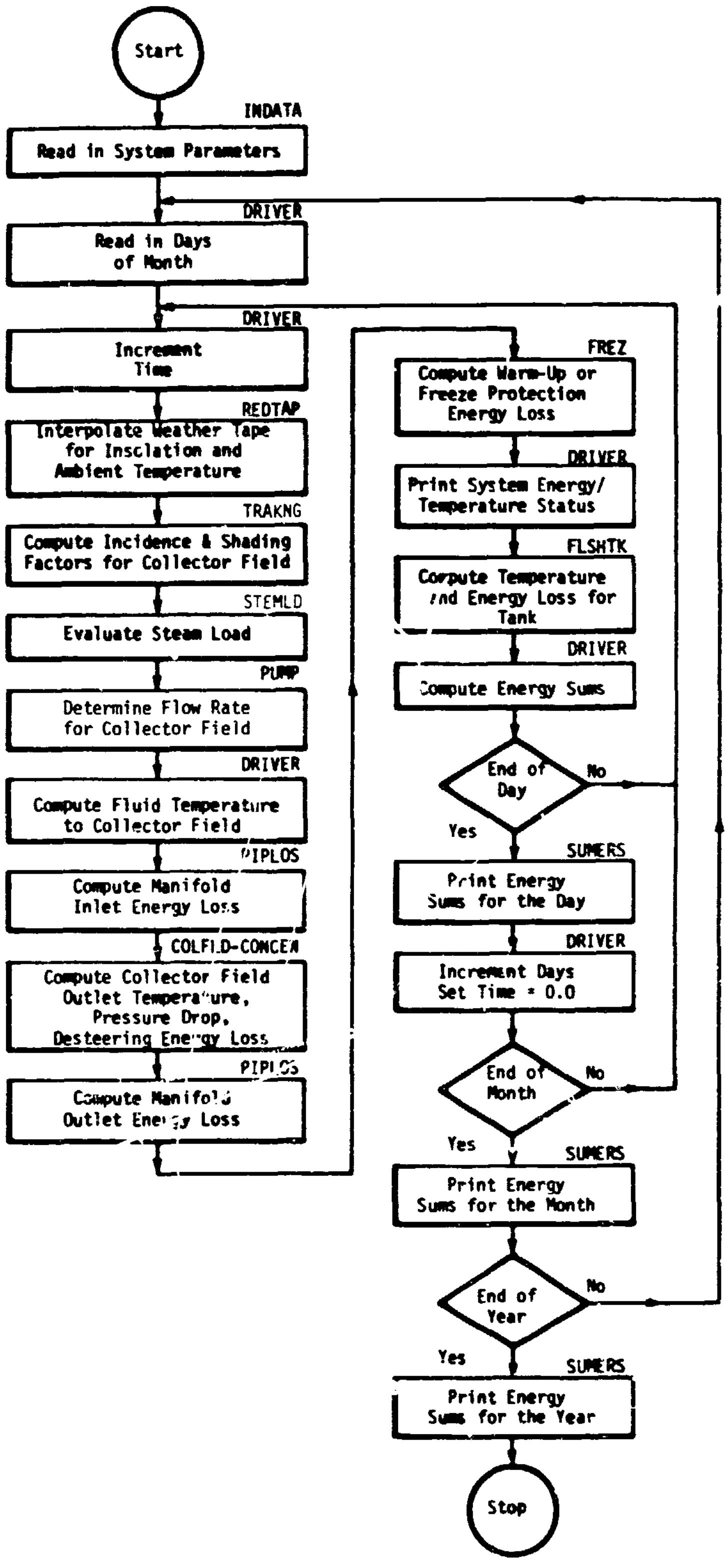


Figure A-5. Flow chart for simulation computer code.

the sequencing of subroutine operations, performs minor fluid flow balance calculations, increments time, and controls the program output. The following paragraphs describe the various subroutines controlled by this driver.

Input Parameters--

The subroutine INDATA reads input parameters which specify the layout of the field and sizing of system components for a given simulation run. In addition, it prints out the system parameters for reference.

The subroutine REDTAP reads insolation and ambient temperature values from the weather tape. Since the hourly values of insolation on the data tape are actually integrated values of incident radiation over the previous 60 minutes, the time associated with each value in the interpolation scheme was midway in the integration period, i.e., 30 minutes before the hour. Using this interpolation, an insolation value can be assigned to any given time of day. This provides a more accurate simulation of the actual insolation profile for the computer model. The subroutine STEMLD evaluates the heating demand on the solar system for a given time of day.

Tracking and Shading Losses--

The subroutine TRAKNG computes the incidence and shading factors for the collectors. The incidence factor (the cosine of the angle between the normal to the collector aperature and the sun beam) and the shading factor (that fraction of the collector aperture that is shaded by the adjacent row of collectors) are computed for a one-axis tracking collector for any given orientation of the collector field.

Field Flow Rate--

Mass flow rate through the collector field is determined in the subroutine called PUMP. In addition, this routine contains the decision logic to determine whether or not the field pump is operated.

Collector Field Performance--

The overall performance of the collector field is computed in the subroutine COLFLD. This subroutine serves as the driver for the CONCEN subroutine which sequentially computes the performance of each collector in a flow loop. Fluid outlet temperature (for a given inlet temperature, flow rate, ambient temperature, and insolation value) is computed from a collector efficiency correlation based on a combination of experimental data and analytical computations. Pressure drop through each collector is also computed for either of two receiver configurations (a simple tube was assumed herein). The COLFLD subroutine relates collector outlet parameters to the inlet of the next collector, and desteers collector modules (groups of eight collectors on a common tracking drive) when outlet temperatures exceed 452 K (354° F).

Flash Boiler/Storage Tank--

The mass and energy balances for the flash boiler/storage tank are computed in the FLSHTK subroutine. The temperature of the water in the tank at the end of the time step and the energy loss through the tank walls during the time step are computed using the simple Euler method for the solution of the transient mass and energy equations.

These equations are as follows:

$$\frac{(PV)_{t+\Delta t} - (PV)_t}{\Delta t} = \Sigma\, \dot{m}_{in} - \Sigma\, \dot{m}_{out}$$

and

$$\frac{(PVE)_{t+\Delta t} - (PVE)_t}{\Delta t} = \Sigma\, (\dot{m}\, h)_{in} - \Sigma\, (\dot{m}\, h)_{out} - \Sigma Q_{loss}$$

where P is the density of the water, V the volume, E the internal energy, h the enthalpy, m the mass flow rate, t the time, and Q_{loss} the miscellaneous energy withdrawn for wall losses, warmup losses, etc.

This routine requires the following input: current water temperature and fluid volume in the tank, temperature and flow rate of water from the collector field, flow rate of liquid water and steam from the tank, enthalpy of condensate return, ambient temperature, and the time step. In addition, any miscellaneous, instantaneous energy requirement, such as that required for warmup or freeze protection, is determined. The new temperature of the water is determined based on the energy remaining in the tank.

Freeze Protection and Warmup--

The subroutine FREZ computes the energy required for both the freeze protection of the system and for morning startup (or warmup) of the collector field. When the collector field pump is not operating, the temperature of the pipes (receiver tubes and manifold) is computed using a simple transient energy equation. The pipes are permitted to cool to as low as 275 K (35° F), at which time, energy is withdrawn from the tank by circulating tank fluid through the collector field until a field fluid temperature of 283 K (50° F) is reached.

Energy Losses in Pipes--

The subroutine PIPLOS computes the temperature drop and energy loss between the inlet and outlet of a pipe. The input parameters used in the calculation are as follows: inlet water temperature, mass flow rate, ambient temperature, length of pipe, product of overall heat transfer coefficient and surface area per unit length, and time step. This routine is used to evaluate the energy loss for both collector inlet and outlet manifold pipes.

Output--

All energy accounting is performed in the SUMERS subroutine. This routine computes the daily, monthly, and annual totals of such items as incident radiation, all energy loss terms, and steam load met by solar energy. This routine also contains the print statements for the code output.

Life Cycle Cost Model

The after tax annual LCC of the solar system alone was computed for each simulation run. The optimization criteria of net Btu/LCC was then computed to permit comparison of different component sizes or field layouts. The after tax LCC was calculated based on the following formula:

$$LCC = CRF\ (1\text{-}ITC)\ (C) + (1\text{-}t)\ OC - t\ (D)$$

where CRF = capital recovery factor, i = 8 percent, n = 25 years

ITC = investment tax credit

C = capital costs

t = weighted federal and state tax rate

OC = tax deductible operating costs

D = depreciation, straight line

Since the proposed system is for a government agency, those items related to tax benefits cannot be included in the calculation of LCC. Therefore, the above equation becomes

$$\text{Annualized cost} = CRF(C) + OC$$

where the operating costs equal the levelized fuel cost multiplied by the parasitic consumption. In addition to these parameters, the levelized cost of electricity (a parasitic operating cost) was computed at 7¢/kWh assuming 8 percent real (net of inflation, assumed to be 6 percent per year) escalation and an 8 percent discount rate. The CRF was calculated based upon this discount rate and a 25-year system life.

Data Input

The proper sizing and optimization of the system design depends heavily on the formulation of an accurate simulation mode. Therefore the representation of the insolation and steam load must be accurate.

Insolation Model--

The insolation for a given site varies over a broad range, not only from winter to summer and day to night, but also hourly, since cloud cover can cause rapid and drastic changes in the incident solar radiation during daylight hours. Therefore, a comprehensive study was conducted to select representative values of insolation for the simulation period.

The selection criteria were based on the objective of the analysis to design a solar system that most economically supplies steam for space heating for the 25-year life of the system. Since the variation in the annual total insolation for the 25 years is small (i.e., not greater than 10 percent), a single year of simulation is satisfactory. Of course, the typical month-to-month variation as well as day-to-day variation is significant and, therefore, is modeled in the calculations. The technique used to select the insolation period was similar to the scheme utilized in Reference 6. The primary criterion is that the insolation selected for system simulation should produce an average value close to the long term average for each month. In addition, it is also desirable to model the actual hourly variation that would be expected for the system.

Therefore, a data tape, which contained a complete history of weather for Ft. Worth, Texas, approximately 225 km (140 miles) from Ada, OK, for the years 1952 to 1974, was obtained from the Aerospace Corporation. The tape contains the direct normal and total hemispheric insolation values as well as relevant meteorological data for every hour of each year. A comparison of monthly average daily total insolation for Ft. Worth and Oklahoma City indicated that Ft. Worth data would be sufficient, since Ada, OK lies between the two cities. The simulation for the year was approximated by six 7-day periods of insolation, which were selected to model system operation when energy was collected and stored during weekends. The criteria used to select these periods were: (1) that the mean daily total insolation should closely agree with the long-term value for that month, and (2) the day-to-day variation in daily total insolation should also be modeled. This was accomplished by requiring the interquartile range (defined as the difference between the 75th and 25th percentiles) for each period to also agree with the long term value for the month.

Therefore, long term values of both the mean daily total insolation and the interquartile range for each month was first formulated. It was found that the years from 1958 to 1962 had representative values of insolation; therefore, the hourly insolation values were summed to form the daily totals for these 5 years. The average and interquartile range for these daily totals were computed for each of the 12 months for the 5 years.

The seven consecutive day periods whose average and interquartile range agreed closest with the long term values were then determined. All possible combinations of seven consecutive day periods in the years 1958 to 1962 were examined simply and quickly by computer. It was desired to select data within 2 percent of the long term average and 5 percent of the interquartile range. These tolerances were relaxed, however, and in all cases the average daily total insolation for the selected period was within 3 percent of the long term average and the interquartile range was within 20 percent of the long term average. The limit for the average was made more stringent than the interquartile range, since the primary purpose of the simulation is to accurately represent long term (25-year) performance.

The results of the selection process are shown in Figure A-6. The histogram shows the six monthly values of the average daily total

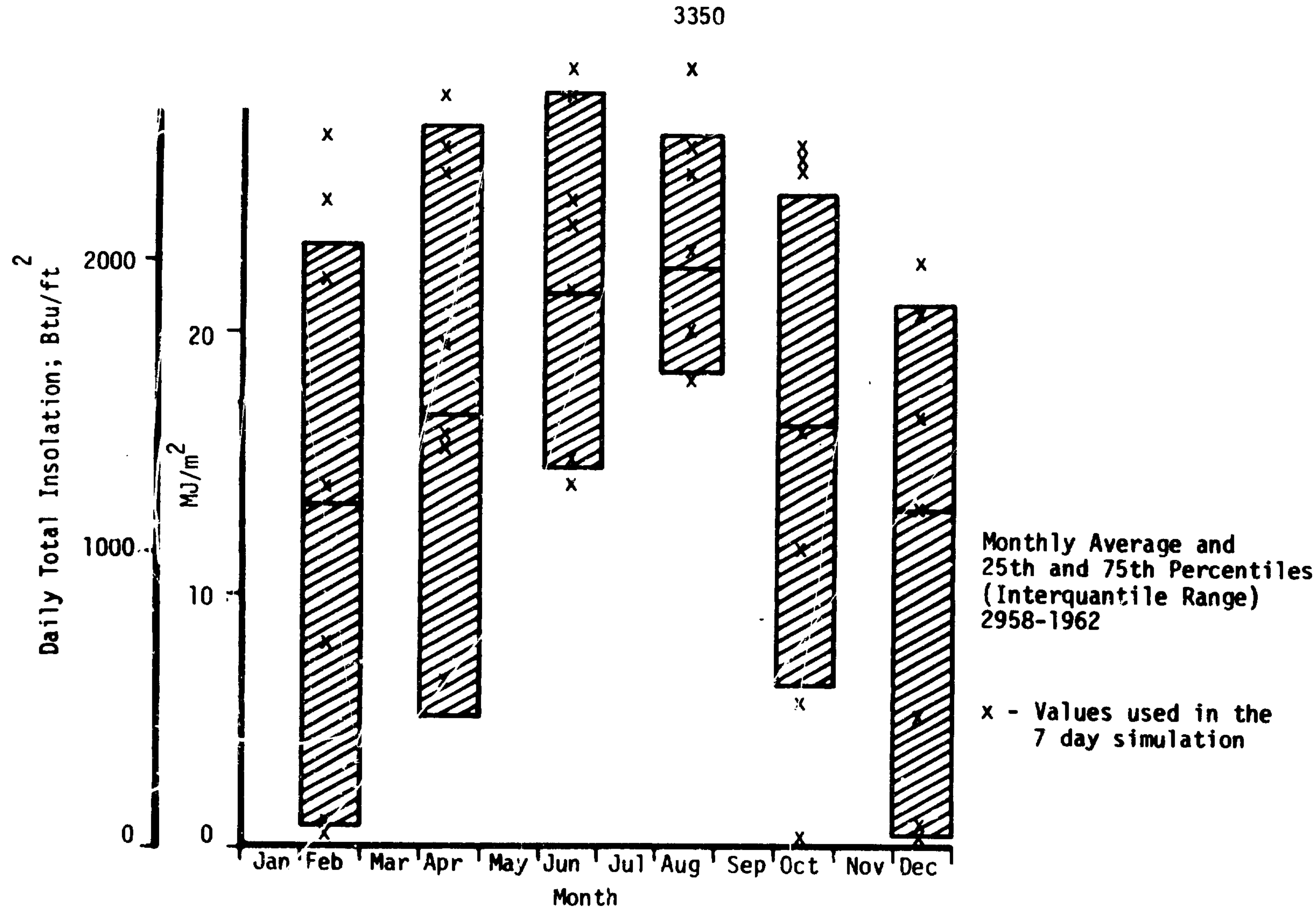


Figure A-6. Daily total insolation for 7-day weeks and 5-year results.

insolation and the corresponding 25th and 75th percentiles for the years 1958 to 1962 for the Ft. Worth location. The average value for the summer months is just below 22.7 MJ/m^2-day (2000 Btu/ft^2-day); for the winter months the average is somewhat above 11.4 MJ/m^2-day (1000 Btu/ft^2-day). In addition, the percentiles shown in Figure A-6 indicate that the variation of daily total insolation from day-to-day is much smaller in the summer months, particularly the month of August, than in the winter months. This is expected, given the more intermittent winter weather. Since the daily total insolation for the 25th percentile for the winter months was less than 2.27 MJ/m^2-day (200 Btu/ft^2-day), the collectors would not operate much during one-fourth of the days in those months, since hourly insolation would frequently be below the minimum 1.14 MJ/m^2 (100 Btu/ft^2) required for collector tracking.

In addition, X's are noted in Figure A-6 for the values of the daily total insolation for the seven-day periods which were selected by the process just described. It can be seen that in each case the chosen data agree very well with both the long-term average values and the interquartile range.

Load Analysis--

The simulation model also required a specification of the steam load according to the demand of the heating coils. In order to adequately model system performance, hourly load information was required. Direct and seasonal load profile were generated as input for the transient analysis and for solar substitution calculation of conceptual system configurations. The space conditioning load was calculated according to the time averaging method presented in the <u>ASHRAE Handbook of Fundamentals</u> (Reference 7), again using the weather data tape. Fuel consumption data for the years 1976 and 1977 for the Ada, OK facility were used to check the results.

It should be noted that the facility has been analyzed by Philip R. Jones and Associates, Consulting Engineers, of Pensacola, Florida in the past year. Certain recommendations have been made by them in order to increase building efficiency, but it is not clear at this time which actions, if any, would be taken. Therefore, current U-values, occupancy and ventilation schedules were used in the load analysis.

The following subsections describe in detail the methodology used in calculating the load profiles and present more detailed results. The first subsection describes the facility load characteristics and general assumptions. The following subsections describe the space conditioning calculations. Finally, a representative load profile is presented.

<u>Facility characteristics and assumptions</u>--The size, schedule and employee data for the Ada, OK facility are summarized in Table A-1. These data were used for the final load analysis.

TABLE A-1. ADA, OK FACILITY CHARACTERISTICS

Size: 4,721 m^2 (50,800 ft^2)

Work schedule:

- 6:00 AM to 6:00 PM, 5 days/week
- Infrequent off-hours occupancy by regular staff

Employees:

- 82 persons on regular schedule
- Three persons in off-hours (security, janitorial)

The majority of the assumptions used in the analysis, detailed below, are based on information obtained from Philip R. Jones and Associates, mentioned above. The only two assumptions made by Acurex concerned the space conditioning load. First, the building was treated as a single zone. The approximate load analysis required for a conceptual design effort did not warrant the added time and expense of a multizone calculation.

The second assumption concerned the building set points. The desired indoor conditions were set at 299 K (78° F), 50 percent relative humidity in summer and 293 K (68° F) in winter. Although these conditions might not be in effect at present, these are realistic, energy-efficient goals for the immediate future.

Exterior heat gain--The calculation procedure was initiated by computing the heat transfer due to the various external sources. Included are gains (losses) through the walls, roof, windows, and floor perimeter. The heat gain through the windows is both radiant and convective. The radiant component was computed as the area of the glass multiplied by the solar heat gain factor for the appropriate time and window orientation at 37.7 degrees north latitude (Reference 7). The building parameters used in this analysis are listed in Table A-2. The convective portion is the product of the overall coefficient of heat transmission, U, the indoor/outdoor temperature difference, and the area.

Ventilation heat transfer--Ventilation is the next significant heat transfer mechanism. This is convective heat transfer that has both sensible and latent components. The sensible heat transfer is simply the product of indoor/outdoor temperature difference, the mass flow rate of air, and the specific heat of air. The latent heat transfer is the product of the mass flow of air, the indoor/outdoor humidity ratio

TABLE A-2. BUILDING PARAMETERS USED IN SPACE CONDITIONING LOAD ANALYSIS

Building component	Parameters
Windows	Single-glazed, clear
North	Area = 73 m^2 (783 ft^2)
East	30 m^2 (324 ft^2)
South	83 m^2 (891 ft^2)
West	50 m^2 (540 ft^2)
Roof	Concrete deck with insulation
Insulation	$U = 2.86 \frac{KJ}{hr\ m^2\ {}^oC}$ $0.14 \frac{Btu}{hr\ ft^2\ {}^oF}$
Area	1162 m^2 (12,500 ft^2)
Walls	7.6 cm (3") precast concrete panel, exterior surface, 15.2 cm (6") concrete block, interior surface, 30.5 cm (12") thickness overall
North	Area = 427 m^2 (4598 ft^2)
East	364 m^2 (3914 ft^2)
South	427 m^2 (4598 ft^2)
West	364 m^2 (3914 ft^2)
Insulation	$U = 6.74 \frac{KJ}{hr\ m^2\ {}^oC}$ $0.33 \frac{Btu}{hr\ ft^2\ {}^oF}$
Ventilation	
Minimum (summer)	1339 m^3/min (47,330 cfm)
Maximum (winter)	1654 m^3/min (58,455 cfm)
Electric lighting	
6:00 AM to 6:00 PM, Monday through Friday	450 MJ/hr (427,000 Btu/hr)
Off-hours	45.4 MJ/hr (43,000 Btu/hr)
Occupancy	82 persons daily, 3 persons off-hours
6:00 AM to 6:00 PM, Monday through Friday	sensible 21.6 MJ/hr (20,500 Btu/hr)
	latent 17.3 MJ/hr (16.400 Btu/hr)
Off-hours	sensible 0.79 MJ/hr (750 Btu/hr)
	latent 0.63 MJ/hr (600 Btu/hr)

difference and the heat of vaporization of water. Dehumidification (latent heat removal) is only required when the indoor RH rises above 65 percent at 299 K (78° F). As noted in Table A-2, the ventilation requirement is 315 m^3/min (11,125 cfm) greater in the heating season than in the cooling season. This is due to the fact that the ventilation supplied to all fume hoods (315 m^3/min) is conditioned (heated make-up) in the winter only.

Employee heat gain--The heat gain from the employees and visitors has a sensible and latent component. The rates of heat gain are 253 KJ/hr (240 Btu/hr) per person, sensible and 211 KJ/hr (200 Btu/hr) per person, latent. The regular occupancy schedule assumed was 82 persons daily from 6:00 AM to 6:00 PM, Monday through Friday. Off-hours personnel consisted of three people.

Electrical heat gain--The electrical loads contribute significantly to the total heat gain of a building. Nearly all of the electrical energy fed into a building to operate lights and machinery ends up as a sensible heat gain. It was assumed that the Ada, OK facility will have adequate canopy hood ventilation to handle the heat gain contributed by all autoclaves and dishwashers. A nominal heat gain of 26.9 W/m^2 (2.5 W/ft^2) of floor area was attributed to electric lighting.

Building load- simplified method for relating the instantaneous heat gain (or loss) to instantaneous building load was used (Reference 7). The convective portion of the instantaneous heat transfer is considered as an instantaneous building load. The radiant portion is considered as reduced or averaged over a period of time by the thermal storage of the building. Based on a medium weight construction for the Ada, OK facility, 5 hours was used for the time averaging period. The total instantaneous building load is the sum of the convective portion and the time-integrated average of the radiant portion. Once the hourly load is the sum of the convective portion and the time-integrated average of the radiant portion. Once the hourly load characteristics were determined using the above method, the monthly totals were compared with the facility's fuel consumption data for 1976 and 1977. Figure A-7 presents a plot of the monthly variation in heating load based upon the average of 1976 and 1977 natural gas consumption and an assumed boiler efficiency of 70 percent. Also shown in the figure is the cooling load based on electrical consumption, an assumed COP of three for the existing vapor compression chillers and a baseline consumption yielding zero cooling load in December. The heating load which occurs during the summer is due to building reheat and is approximately 25 percent of the average summer cooling load.

For the months during the heating season in which there were discrepancies between the heating load characteristics derived from the ASHRAE time averaging method and those shown in Figure A-7, scaling factors were determined and applied to the hourly values. For the cooling season, where the actual heating load is due to reheat and, therefore, not modeled in the ASHRAE method, an hourly heating load was assigned equal to 25 percent of the calculated cooling load.

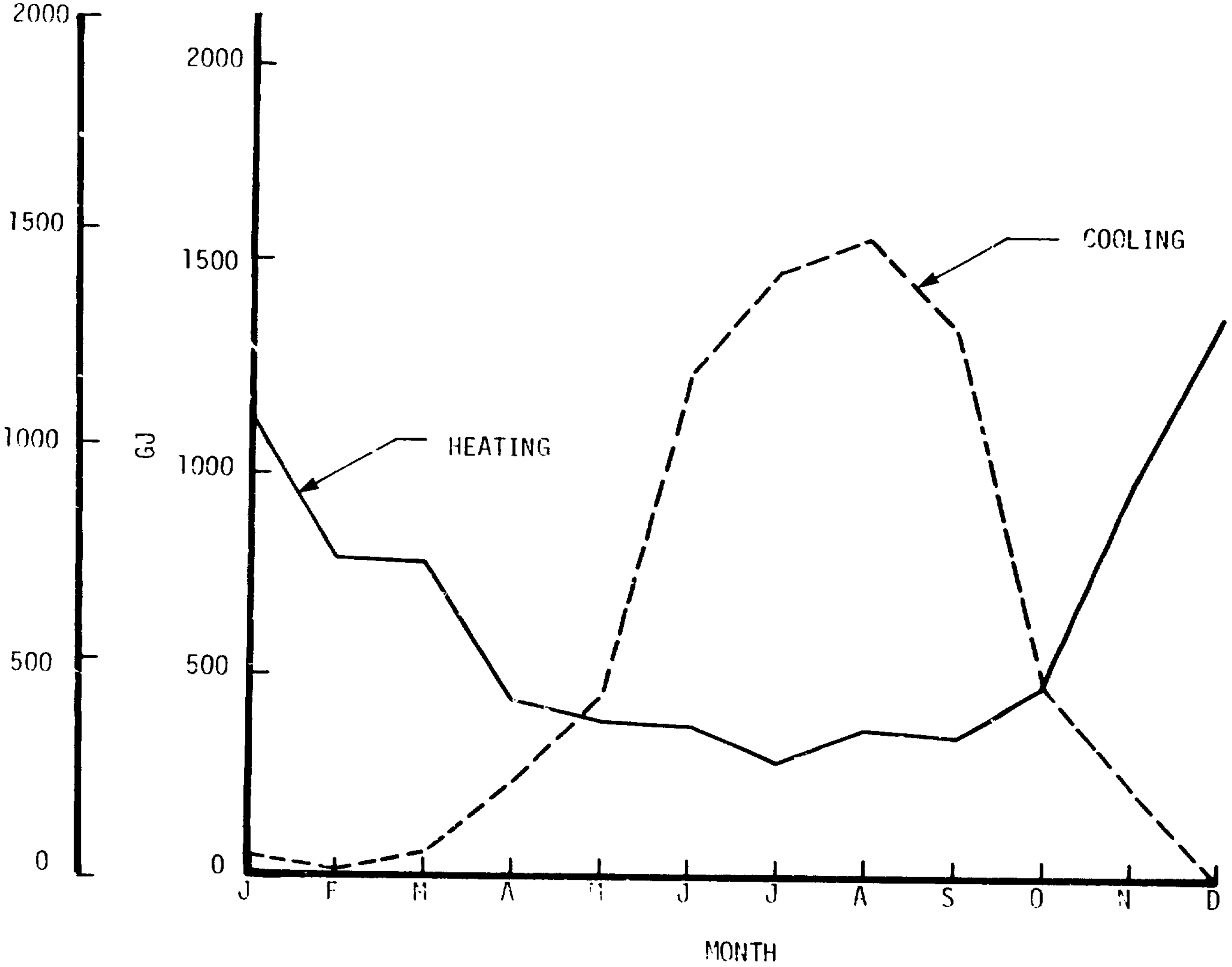


Figure A-7. Monthly building loads for Ada, OK facility based on 1976 to 1977 fuel consumption.

Figure A-8 shows a typical result of the load analysis. It shows the hourly load distribution for one selected 7-day period in the month of June, based upon the above procedure and insolation and temperature data from the weather tape. Both heating and cooling loads were calculated for six alternating months in the year starting with February.

OPTIMIZATION RESULTS

This section presents the results of the optimization. As discussed above, our basic approach was to optimize first those parameters which had the greatest effect on system performance (i.e., collector field size and storage volume), while holding other, secondary parameters (e.g., orientation, flow rate, insulation, thickness, collector row spacing, etc.) at nominally optimized values. With these major system parameters established, the secondary parameters were then systematically optimized. The following sections describe the sequence of results that led to the selection of a cost-optimized solar steam configuration.

Nominal Sizing

Based on the results of the simplified modeling performed during the Cost/Benefit Analysis and additional economic tradeoffs, nominally optimized values for collector orientation and spacing, fluid flow rate per square foot of collector area, and pipe and tank insulation values were determined. These values, shown below, were held constant while collector area and storage volume were optimized.

Nominally Optimized System Parameters

Field orientation	North-South
Row separation	15 ft
Fluid flow rate (four modules/loop)	2.39 lbm/hr-ft^2
Pipe loss coefficient (UA/L)	0.24 Btu/hr-ft-^{o} F
Tank loss coefficient (U)	0.05 Btu/hr-ft^2-^{o} F

Collector/Storage Tank Sizing

Collector area and storage volume were investigated first, since system performance is most sensitive to these parameters (60 to 70 percent of system LCC is due to collectors and storage). Several collector field sizes were first selected. Then, for each field size, system performance and LCC's were computed for each of several storage tank volumes. The objective of these calculations was to determine the cost optimum combination of collector area and storage volume.

To reduce system complexity, simplify construction, minimize manifolding, and simplify fluid flow balancing and temperature control, all collector flow loops were laid out identically. These practical

JUNE HEATING LOAD

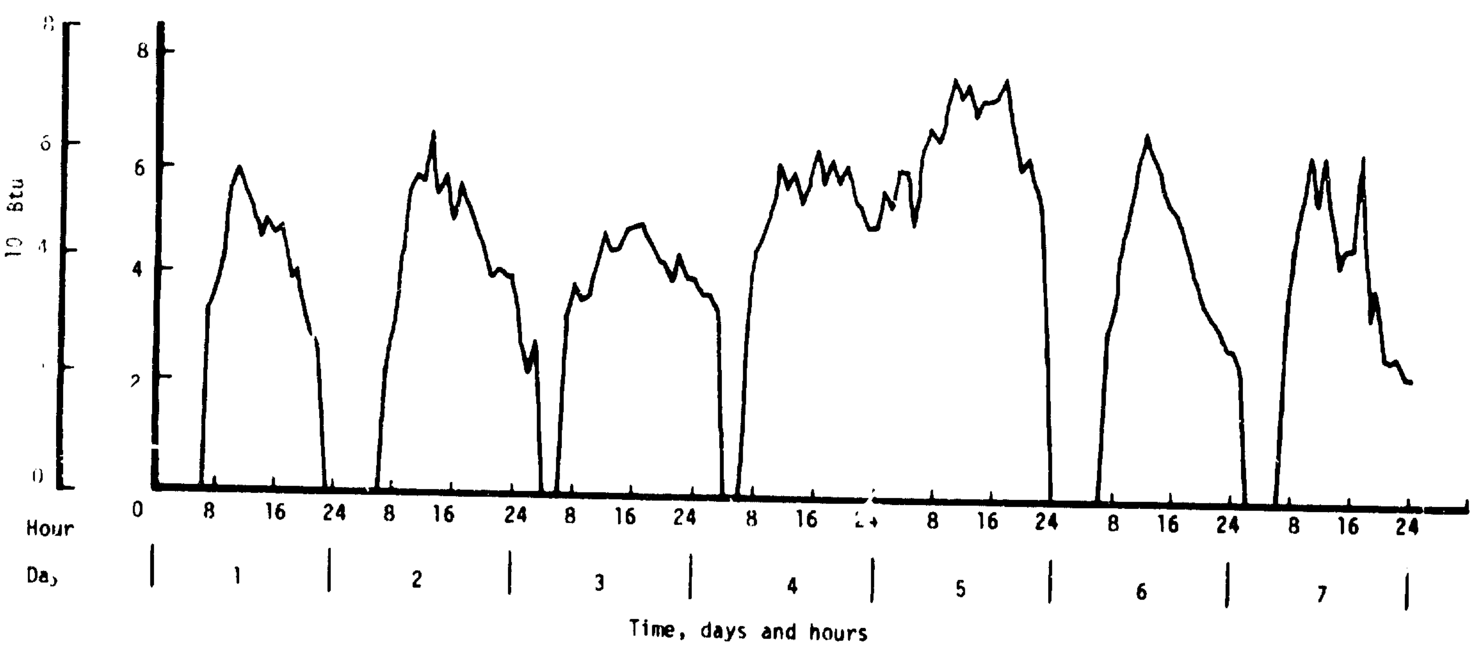


Figure A-8. Hourly heating load distribution for 1 week in June.

factors and collector siting constraints led to the consideration of one basic loop design. It was comprised of four collector modules (a module consists of eight collectors that use the same tracking motor) connected in series to form a flow loop (two rows to a loop; two modules out and two back). This arrangement is shown in Figure A-9. The collector field consists of an integral number of these flow loops. Thus the minimum incremental change in field size was 173 m^2 (1866 ft^2) for each four-module/loop.

The results of the cost/performance calculations for this field layout are shown on Figure A-10. Each curve represents the change in LCC's with the variation in storage volume for a given collector area and row design. The minimum point on a particular curve therefore represents the optimum storage volume for that collector field size. The results of Figure A-10 show that the 1651 m^2 (16,794 ft^2) collector field composed of nine loops of four modules is the overall economic optimum configuration, with a storage volume of 139,000 l (36,800 gallons).

Optimum Maximum Operating Pressure

In the previous subsection, the system optimization was based upon a maximum operating pressure of 1074 kPa (141 psig) for the 1136 kPa (150 psig) rated boiler/storage tank. For a given energy storage capability, increasing pressure has the effect of decreasing storage volume and increasing tank wall thickness. Parasitic requirements increase in order to produce the higher ΔP. Also, since tank volume decreases, so do insulation costs and heat loss as a result of decreased surface area.

Based upon the 1561 m^2 (16,794 ft^2) collector field size selected above, the maximum operating pressure was varied to determine the optimum storage volume and pressure rating. Figure A-11 presents the results of this optimization. Decreasing the tank design pressure from 1480 kPa (200 psig) to 791 kPa (100 psig) results in a significant decrease in LCC primarily because of decreasing vessel wall thickness and parasitic consumption. Decreasing the pressure beyond 791 kPa (100 psig), however, results in a marked rise in system cost, as the insulation costs, increased storage volume and heat losses due to increased surface area become the dominant cost factors. Therefore, the optimum operating pressure was selected for the 791 kPa (100 psig) pressure vessel.

Row Spacing

Having determined collector row design, field size, and approximate storage volume, and operating pressure range, row spacing was evaluated. As rows are spaced closer together, manifold costs decrease, but performance is also decreased due to mutual shading. As rows are spaced more widely apart, field losses from shading decrease at the expense of longer manifolds (more pumping power and greater heat loss).

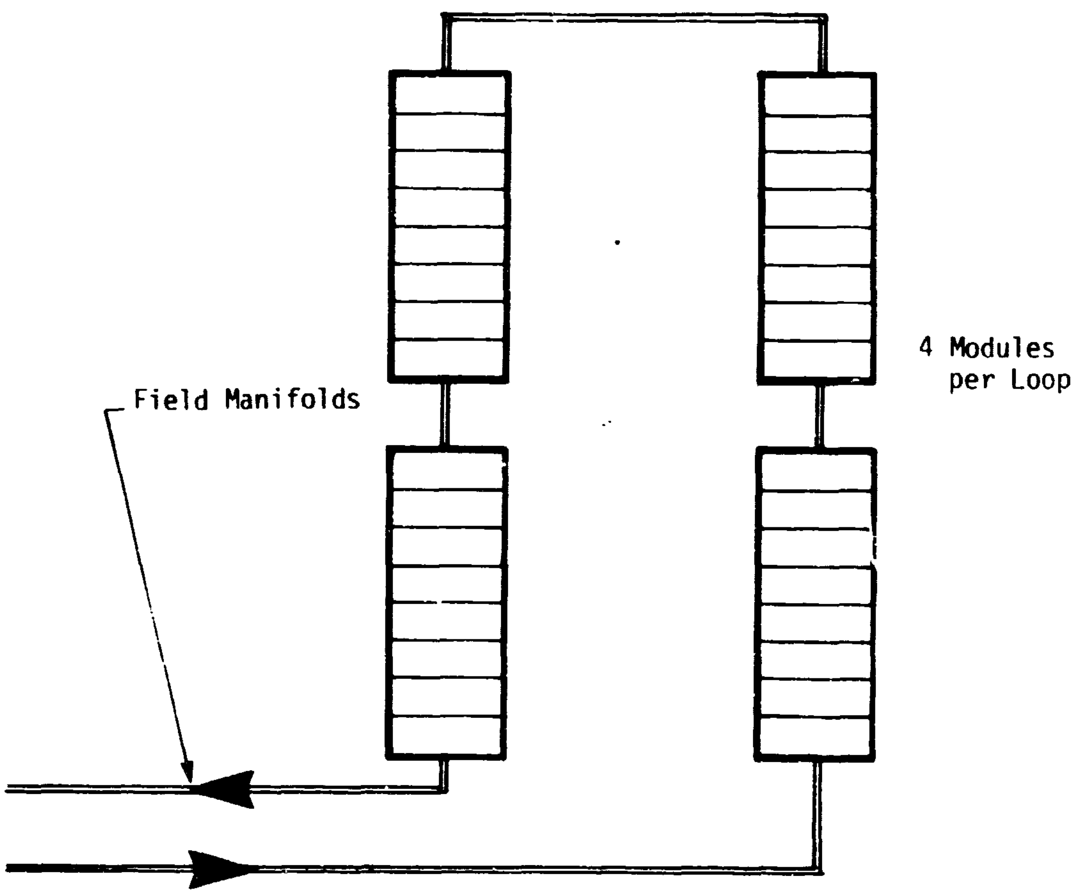


Figure A-9. Collector field flow loop.

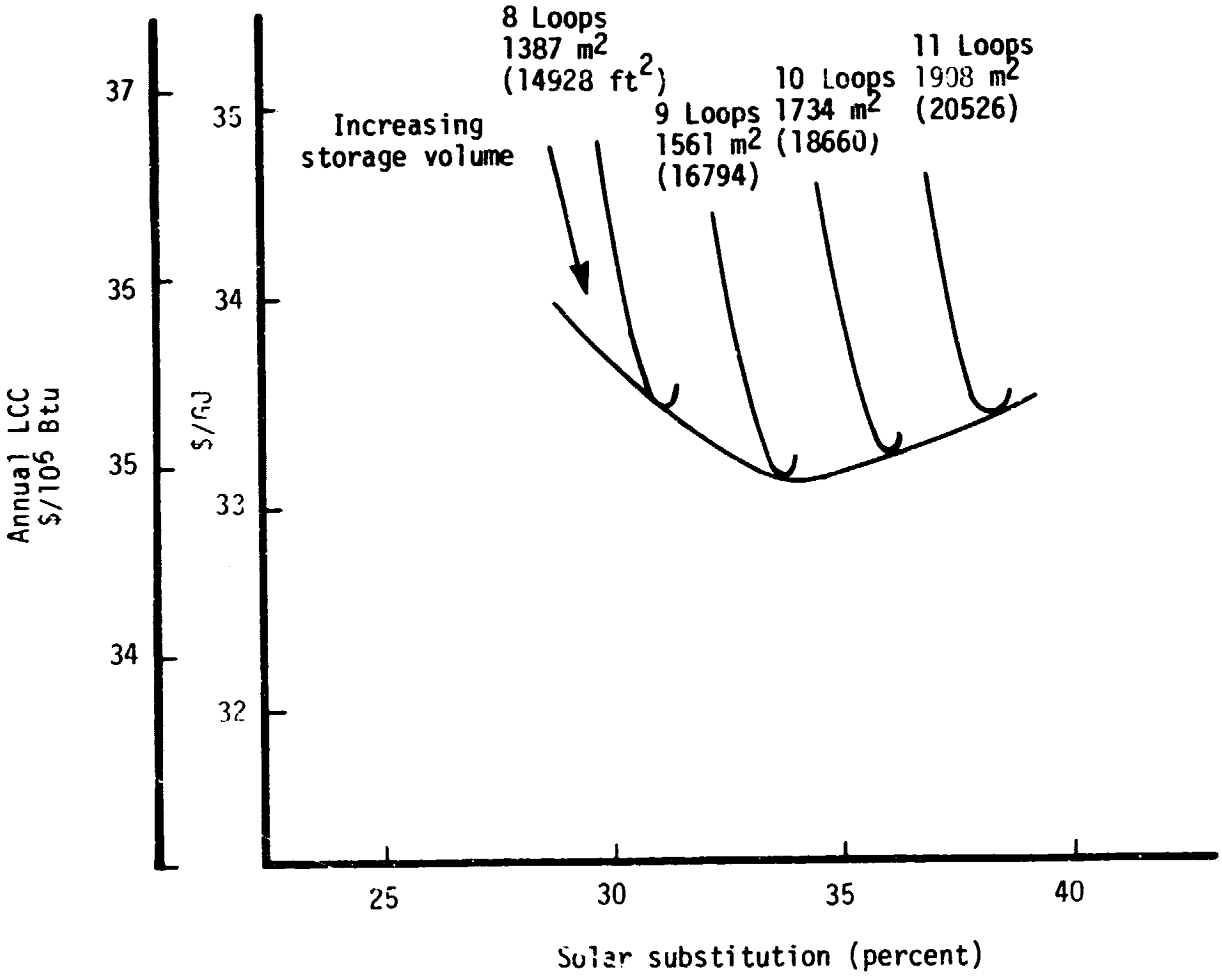


Figure A-10. Effect of collector area and storage volume on LCC.

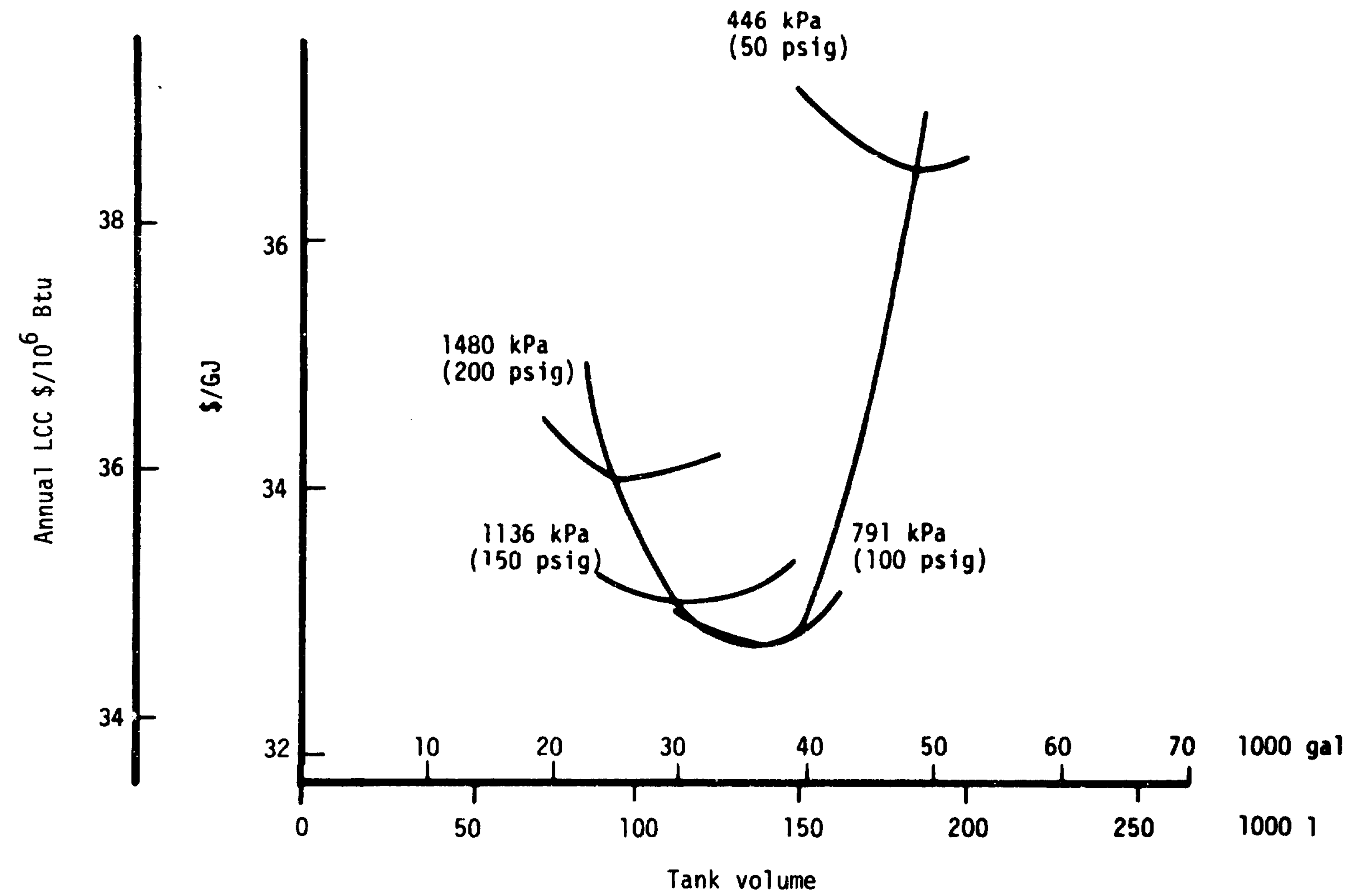


Figure A-11. Effect of tank design pressure on LCC's.

The effect of row spacing is shown in Figure A-12. A center-to-center collector row spacing of 4.6 m (15 feet) gives the minimum cost as well as adequate access to the field.

Collector Field Flow Rate

To prevent boiling in the field, collector outlet temperatures must be kept below the saturation temperature corresponding to the field fluid pressure. This is accomplished by desteering a collector module of a flow loop if the fluid temperature exceeds its set-point. By increasing fluid flow rate, the temperature rise in the field is reduced, allowing a greater amount of energy to be collected with less desteering. Increasing flow rate, however, increases parasitic energy consumption and, consequently, reduces net system output. Therefore a cost trade-off was performed to determine the optimum flow rate for the four-module/nine-loop configuration. This trade-off indicated that a flow rate of 6.05 l/sec (96 gpm) is optimum for nine loops. These results concluded the optimization study and a configuration was selected.

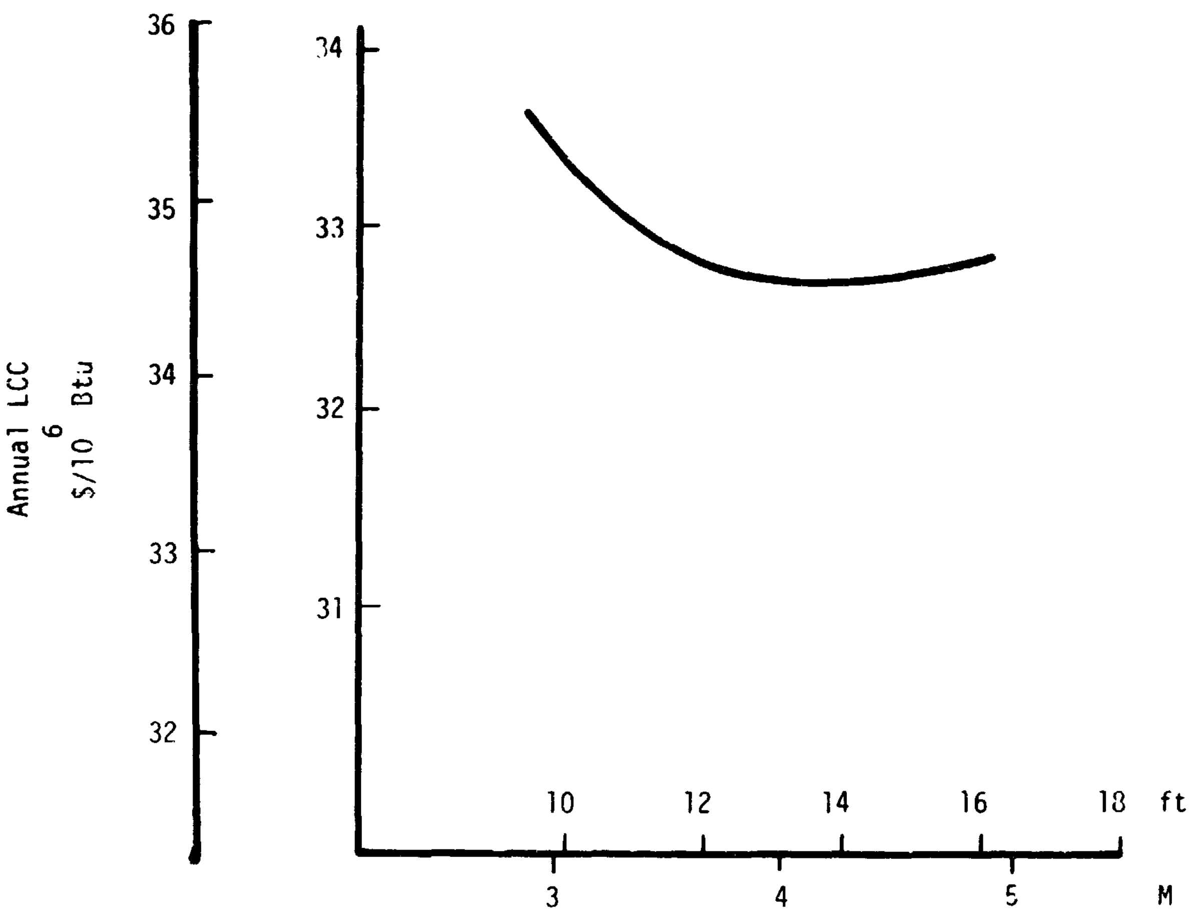


Figure A-12. Effect of row spacing on system performance.

APPENDIX B

SUMMARY OF RELEVANT SOLAR TECHNOLOGY

Solar energy can most easily be applied to typical building energy loads (space heating and cooling, DHW, and process hot water) by directly converting radiative solar energy to thermal energy at the end-use temperature required. For space cooling loads, the end-use temperature is typically the heat source temperature of a thermal chiller. Indirect methods also could be used to meet the loads. The most common indirect methods are photovoltaic or solar-thermal-electric conversion, combined with electric resistance heat pump heating and/or vapor compression cooling. However, since these indirect methods are further from economic feasibility, only direct conversion to thermal energy is discussed in detail below. This section briefly describes the basic principles of solar energy collection and thermal conversion, gives a general overview of current technology in solar thermal collectors, summarizes energy storage techniques, discusses heating and cooling components, and summarizes solar electrical power generation.

BASIC CONSIDERATIONS

Since the technology now exists for applying solar energy to any typical building energy load, the question of how to best meet a given load becomes one of economics, that is, how cost-effective is a solar energy system compared to the alternatives. The economics, however, depend strongly on the characteristics of solar radiation at the Earth's surface, the characteristics of the load, and the projected cost of conventional fuels. Because of factors including location and seasonal variation, some loads are more appropriate than others for solar applications. Characteristics of terrestrial solar energy that affect the applications and system designs are as follows:

- Solar energy is dilute, reaching only about 945 W/m^2 (300 $Btu/hr\text{-}ft^2$) under favorable conditions. Applications which tend to use large amounts of energy therefore will require large areas for solar collection; and space limitation is likely to become a design constraint.
- Solar energy varies on a seasonal and a daily basis. Matching the energy demand curve with the availability of the solar energy will have a major impact on the economics of the system and may require a relatively large thermal storage capacity.

- Terrestrial solar radiation depends strongly on geographic location, particularly latitude and dominant climatic conditions, for the location. The more attractive applications will be for facilities located in sunny climates at lower latitudes.
- Solar energy is unpredictable and instantly interrupted by local weather. A reliable backup energy source and sufficient storage capacity to allow uninterrupted operation during switchover to the backup system is therefore needed.

In addition to these characteristics, other factors affect system design. For example, the highly directional nature of solar radiation, and directional changes throughout diurnal and annual cycles, require that collectors be optimally oriented, either in fixed positions or through active tracking of the sun. This restricts the structural design of collector installations. Shading problems limit the ability to extract maximum energy from a given area.

In addition to its highly directional nature, solar radiation at the Earth's surface always has a diffuse (nondirectional) component, varying from about 10 percent of the total radiation under clear conditions to 100 percent on overcast days. The diffuse component results from cloudiness, humidity and air quality, and thus can vary widely according to location, season and time. Since some collector types cannot utilize diffuse solar energy, this factor may have a significant impact on the system design.

All solar thermal collectors also function more efficiently when the difference between the collector and ambient temperature is reduced. Thus, a warm climate is generally more suitable than a cold one, if the same insolation is available.

These examples illustrate some of the factors which must be considered in the design of a system for a particular facility. The solar technology aspects of the problem are discussed in the following subsections.

COLLECTORS

In a solar thermal energy system, the collector is the key element. It absorbs the solar radiation, converts it to thermal energy, and then transfers the thermal energy to the working fluid. This fluid, in turn, delivers the energy to the point of application.

Solar collectors can be categorized according to their means of collection as follows:

- Non-concentrating (e.g., flat-plate)
- Concentrating
  - -- Low-concentration (e.g., augmented flat-plate)
  - -- Line-focusing (e.g., parabolic trough)
  - -- Point-focusing (e.g., paraboloid dish)

They can also be categorized on the basis of operational temperature level, the heat transfer fluid (gas or liquid), the application of the converted energy, or the tracking requirements. Operating temperatures increase with the degree of concentration, with point-focusing collectors reaching the highest temperatures and non-concentrating collectors addressing the lowest temperature levels. In general, non-concentrating and low-concentration collectors are not used with tracking systems. These collectors are applied primarily to DHW production, and space heating and cooling. Line-focusing collectors require single-axis tracking of the sun; their primary applications are for higher temperature space heating, space cooling, process heat, and electrical power generation. Point-focusing collectors, used primarily for power generation, require two-axis sun tracking.

In addition to these considerations, the nature of solar radiation at the Earth's surface influences the type, size, and materials of the collector. Thus, the selection of the optimum collector for a given purpose depends entirely on the application's characteristics (e.g., temperature level, energy input rates, periodicity, and location). No single collector can meet the widely varying requirements of all potential applications. Of the state-of-the-art collectors, all but the point-focusing types are commercially available and can be practically applied to the loads that are discussed in this document.

Non-concentrating Collectors

Non-concentrating collectors are those that do not use reflective or refractive devices to concentrate solar energy on the absorber surface. Since no optical concentration or focusing is required, these collectors can make full use of both the direct and diffuse components of the available solar energy.

Non-concentrating collectors can be grouped roughly into conventional flat-plate and alternate types. As shown in Figure B-1, flat-plate collectors commonly consist of: a metal plate with a black absorbing surface, metal tubes for carrying the heat transfer fluid, glazing (glass or plastic) to reduce losses by convection and reradiation to the environment, insulation to reduce conduction losses through the back and sides, and a casing for protection and support of the other components. However, other flat-plate collectors differing slightly from this basic "sheet-and-tube" type are also available, as shown in Figure B-2. These sketches show some of the alternate absorber plate designs (i.e. configurations and flow passages). With the exception of Configuration G, all of these collectors are functionally similar.

In contrast, the alternate group of non-concentrating collectors consists of configurations with distinct differences. For example, the Hay "passive" system (Configuration G) differs from conventional flat-plate models because it incorporates collection, thermal storage and heat transport in the same unit. Another alternate design includes shallow solar ponds, which are simple non-concentrating collectors

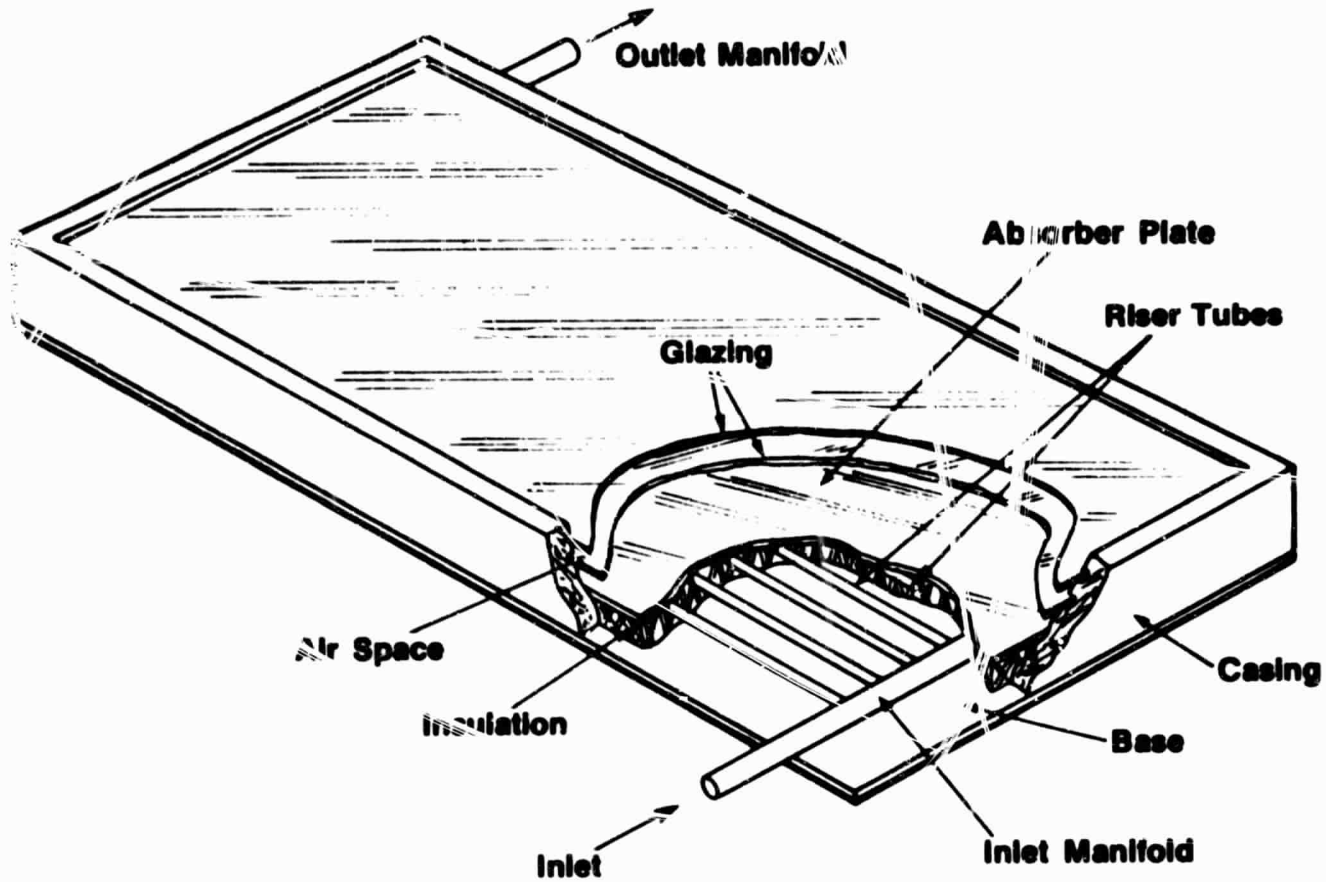


Figure B-1. Typical construction of a "sheet-and-tube" flat-plate collector.

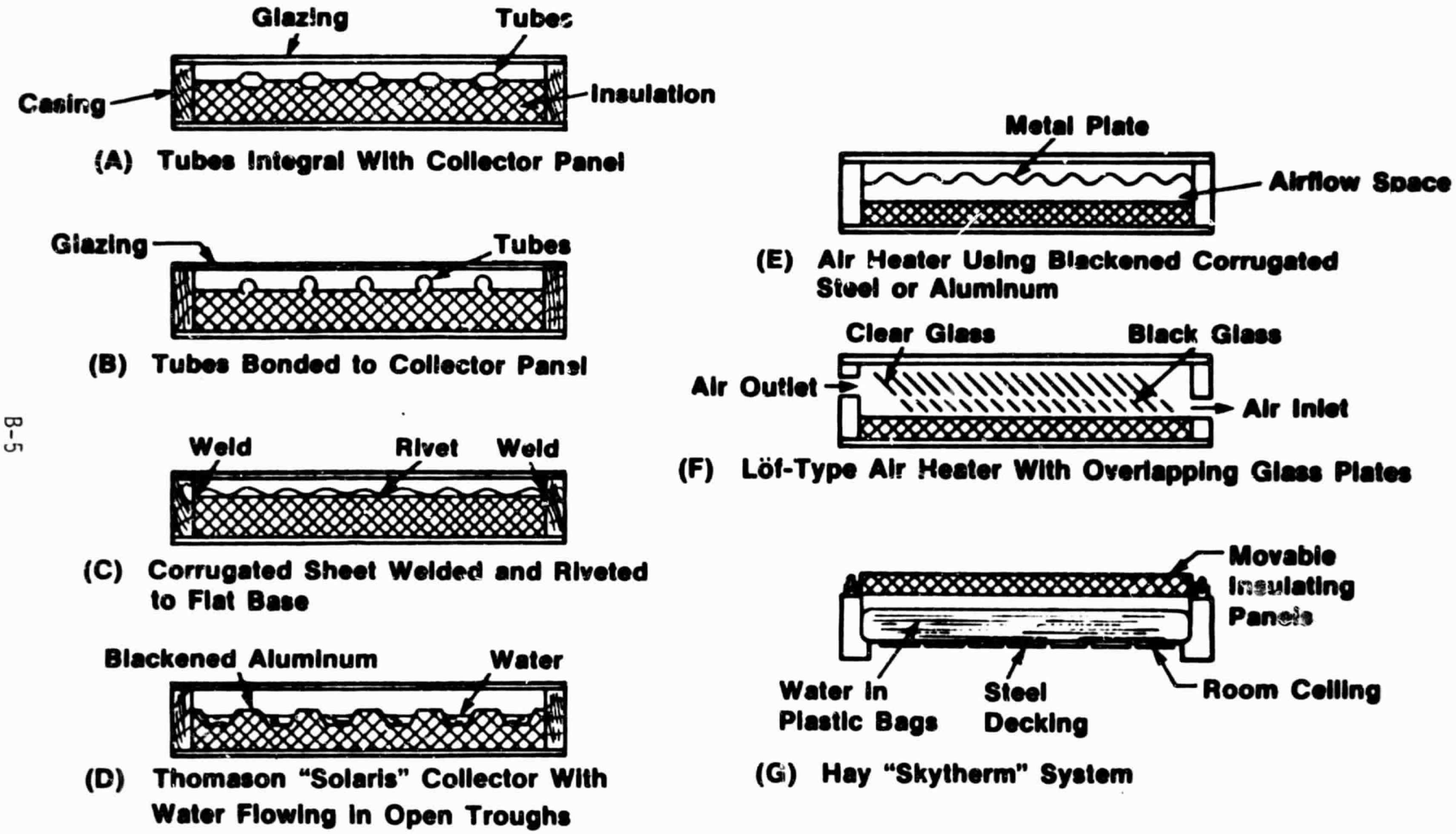


Figure B-2. Representative flat-plate solar collectors (Reference 8).

which operate on the same principles as glazed flat-plate collectors. They are long, narrow plastic bags, through which water flows slowly at a depth of a few centimeters. Typical bags may be 30 meters in length and 3 meters in width (Reference 9). In general, non-concentrating collectors provide working fluid temperatures of less than 366 K (200° F) and operate at collection efficiencies of 30 to 50 percent. Efficiencies of 70 to 80 percent are possible at lower temperatures (i.e., close to ambient temperature). Collection efficiency is defined as the ratio of collected usable energy to the total incident energy. The cost per unit area, which is the most common basis for comparison at present, varies from about $4 to over $40 per square foot for commercial versions. The cost per unit of energy delivered -- a more meaningful number for comparison -- is seldom reported on a common basis, because it is so difficult to determine.

Recent improvements in non-concentrating collectors have come principally from changes in materials and configurations. Some of the most significant improvements have been in selective absorber surface coatings (which minimize the reradiation of energy), nonreflective surfaces for glazing, and plastics for glazing. Further developments that may be significant in the near term are:

- Low-cost structures made with plastics
- Honeycomb materials for convection suppression
- "Heat mirrors" (high infrared reflectance materials) for glazing
- Vacuum "insulation" between absorber plate and glazing
- Black liquids for direct absorption of solar radiation

Commercial Models--

Flat-plate collectors are manufactured in a variety of configurations. The primary design differences include variations in the number and type of glazings, optical characteristics of the absorber plate, configurations of fluid passages, and fabrication and packaging techniques. The two factors which have the greatest impact on collector performance are the number and type of glazings, and the type of absorber coating. Figure B-3 compares the collection efficiency for the four most common commercially available combinations of these design parameters. Since for a given collector this value depends primarily on the temperature difference between the absorber plate and ambient, and the instantaneous insolation, I, it has become common to present collector performance curves as efficiency plotted versus the collector fluid parameter

$$\frac{T_F - T_a}{I}$$

where T_F is the collector fluid temperature and T_a is the ambient temperature.

As shown in Figure B-3, the thermal performance of a double-glazed selective-absorber collector is significantly better than the other three alternative flat-plate configurations when operating at higher fluid

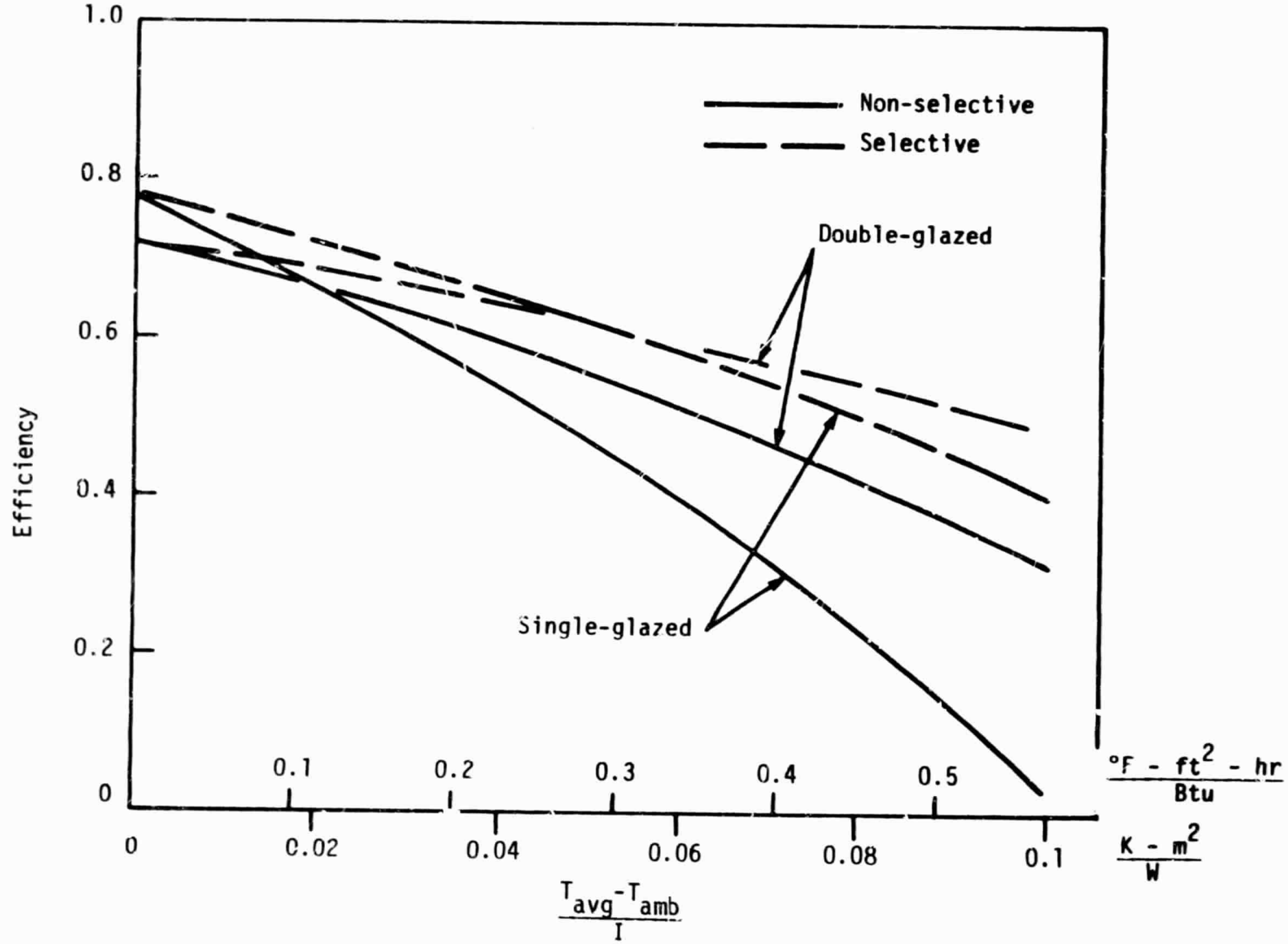


Figure B-3. Flat-plate collector performance characteristics.

parameter values (which can be thought of as higher temperatures). However, at low values, corresponding to low temperature differentials between the absorber and ambient, the less costly single-glazed collectors are superior. The variation in relative performance as a function of fluid parameter value is a result of the interaction of the glazing and absorber coating characteristics.

Each layer of glazing serves to reduce convective and radiative energy losses from the absorber plate. Because of its optical transmittance characteristics, it also reduces the amount of radiant energy incident upon the absorber. At lower temperatures, where heat loss is smaller, adding a second glazing only hinders performance by reducing the incident energy less than it reduces the thermal losses.

Similarly, using a selective surface coating (which has a relatively high absorptance in the high-energy visible spectrum, but has a low emittance in the long-wave infrared range), can significantly improve collector performance in the higher temperature range. This is because heat loss by radiation is a significant factor at higher temperatures. However, at lower temperatures (because the absorptance is typically less than that for a quality flat-back non-selective coating) the selective surface has the same effect as the additional glazing. Four selective coatings typically used in commercially available collectors are listed along with their optical properties in Table B-1.

TABLE B-1. OPTICAL PROPERTIES OF FOUR COMMERCIALLY AVAILABLE COATINGS

Coating	Absorptance	Emittance
Nickel black	0.89 to 0.94	0.07 to 0.11
Black chrome	0.93	0.12
Copper oxide	0.81 to 0.93	0.11 to 0.17
Lead oxide	0.99	0.25

The collector performance curves in Figure B-3 are analytic comparisons that demonstrate the effects of major collector design parameters on efficiency. Actual efficiencies for commercially available models vary considerably due to differences in materials and construction. Uniform performance data on commercially available flat-plate collectors are unavailable, although some comparative evaluations have been made. In one of these, NASA-Lewis Research Center (LRC) conducted a number of collector performance evaluations in a solar simulator under closely controlled environmental conditions

(Reference 10). The collectors examined by LRC included commercially available flat-plates (both single- and double-glazed, selective and non-selective) and evacuated glass tubes. These evaluations showed that variations in case construction, type of selective coating, and panel insulation significantly affected the overall performance of the panel.

Medium-temperature flat-plate collectors are currently manufactured by a number of firms, including:

- American Solar Heat Corporation
- Energex Corporation
- General Electric
- Grumman Aerospace
- Owens-Illinois
- PPG
- Raypak
- Revere Copper and Brass
- Reynolds Aluminum
- Solar Energy Systems
- Solargenics
- Sun Power Systems
- Sun Works

A more complete listing of solar collector manufacturers has been published by the Solar Energy Industries Association (Reference 11).

Concentrating Collectors

If the solar energy incident on a collector is concentrated (by reflection or refraction) onto a smaller absorber surface, the collector can operate with much higher efficiency at high temperatures than a non-concentrating collector. Reducing the absorber (or receiver) area, and therefore reducing the area-dependent convective and radiative heat losses, increases the efficiency.

Concentrating collectors can be divided into three generic types according to the degree of concentration achieved (and thus on the attainable temperature of the delivered energy). In increasing order of outlet temperature, the three types of concentrating collectors are low-concentration, line-focusing, and point-focusing systems. Both low-concentration and line-focusing systems can be produced at costs competitive with flat-plate collectors, and thus are viable options for the loads being considered. Although point-focusing collectors can deliver energy efficiently at much higher temperatures, the lack of commercially available, cost-competitive units makes them impractical for the loads addressed in this program. Therefore, point-focusing collectors will not be discussed again in this document.

Description--

Low-concentration collectors are usually simple, in that they consist of flat-plate receivers augmented with either planar or curved reflectors. These collectors have higher efficiencies than comparable flat-plate collectors when operating at higher temperatures (up to about 422 K or 300° F). Furthermore, they may be able to extend the usefulness of the solar energy system into periods of marginal insolation (morning, evening, or partial obscuration). Such collectors utilize both the direct component and a portion of the diffuse component of terrestrial solar radiation and require only seasonal adjustments of position, if any. While low-concentration collectors may prove to be of substantial value in solar thermal energy systems from a performance/cost standpoint, there is relatively little operational experience with them to date.

A subgroup of low-concentration collectors, which utilize a linear receiver and are generally termed stationary concentrators, also exists. The most well-known is the Winston Compound Parabolic Concentrator (CPC). This collector gives some increased energy concentration without the need for tracking mechanisms. Owens-Illinois makes a concentric glass tube collector which is arranged in banks and uses reflectors to achieve low concentration ratios.

Line-focusing collectors are distinguished by reflective parabolic troughs (Figure B-4) and "linear" Fresnel lenses and reflectors, although several other configurations are also under consideration. All of these systems require linear receivers (e.g., metal tubes) to capture a finite width beam of concentrated solar energy. Current line-focusing collectors typically have geometric concentration ratios (aperture to receiver area) of less than 50, although values up to 200 are theoretically possible. The actual flux concentration achieved by a collector is dependent on the geometric concentration ratio and the optical efficiency of the unit. Optical efficiency can be defined as the ratio of energy impinging on the receiver to energy incident upon the collector aperture area. This term accounts for reflective losses and surface imperfections which scatter a portion of the incident radiation. At present, well-designed, line-focusing collectors operate efficiently (in the range of 40 to 60 percent) up to about 589 K (600° F). There are three disadvantages of the higher concentration line-focusing collector. They are:
(1) optical losses are generally higher than for a flat-plate,
(2) tracking is required to keep the sun's image on the absorber, and
(3) only the direct or beam component of solar radiation can be collected.

In addition to the simple parabolic trough, other variations of the line-focusing concept include the Fixed Faceted Mirror Concentrator (FFMC) and the movable slat reflector concentrator. Current models of these collectors attain concentration ratios similar to parabolic troughs. On the FFMC, the reflectors are fixed and the receiver tracks the sun's image. On the movable slat collector, the receiver is fixed, and each reflector segment rotates individually for tracking.

The receivers used in line-focusing collectors are typically metal tubes. High-performance designs employ selective surface coatings and concentric glass envelopes (glazing) for convection and re-radiation suppression. The receiver cross-sectional shape need not be circular; "pancake" shapes or linear cavities have some performance advantages over circular shapes.

Figure B-4. Acurex Model 3001 concentrating collector.

Line-focusing collectors require a means of following the sun, either by active sun-seeking systems such as photoelectric trackers, or by programmable or clock-drive systems. Tracking is not a difficult problem with current technology.

Manufacturers of line-focusing concentrating collectors include:

- Acurex
- Scientific Atlanta
- Sheldahl
- Jacobs-Del
- Hexcel
- Soltrax
- Solar Kinetics

Performance curves for six of these concentrating collectors are presented in Figure B-5. All six are line-focusing concentrators with one-axis tracking (except the McDonnell-Douglas, which has two-axis tracking). All but the Scientific Atlanta FFMC have actual experimental performance data. The predicted performance curve for the FFMC is based on an experimentally measured optical efficiency, but an analytically derived receiver-loss coefficient.

Collector Comparisons

As previously described, line-focusing and other concentrating collectors concentrate the solar radiation so that the absorber receives solar energy from an area larger than the absorber itself. This offers the advantage of reducing re-radiation by decreasing the area losing energy to the environment. As a result, efficiencies are better at high temperatures compared to a flat-plate collector. This observation is generally true for collectors of different concentration ratios; that is, the collector with the higher concentration ratio will generally show better performance at higher temperatures. Figure B-6 shows a performance comparison for several collectors typifying the flat-plate, low-concentration, and line-focusing categories. The two flat-plate collector efficiencies drop off rapidly with increasing fluid parameter (i.e., increasing temperature). The two low-concentration collectors (CPC and Owens-Illinois) have poorer performance than the flat-plates at very low temperatures, but have significantly better performance than the flat-plates at higher temperatures. The line-focusing parabolic trough also has poorer performance than the flat-plates at very low temperatures, but has much better performance than any of the others at higher temperatures, due primarily to its higher concentration ratio.

As discussed in Section 3, the building loads addressed in this study, other than DHW, require collector fluid temperatures in the range from 343 K (160° F) to 473 K (400° F). These temperatures correspond roughly to fluid parameter values of 0.07 K-m^2/W (0.4° F-hr-ft^2/Btu) and higher, so in general the concentrating collectors will operate more efficiently for the loads in question. This conclusion is not a sufficient basis for selecting a collector, however, since the cost of

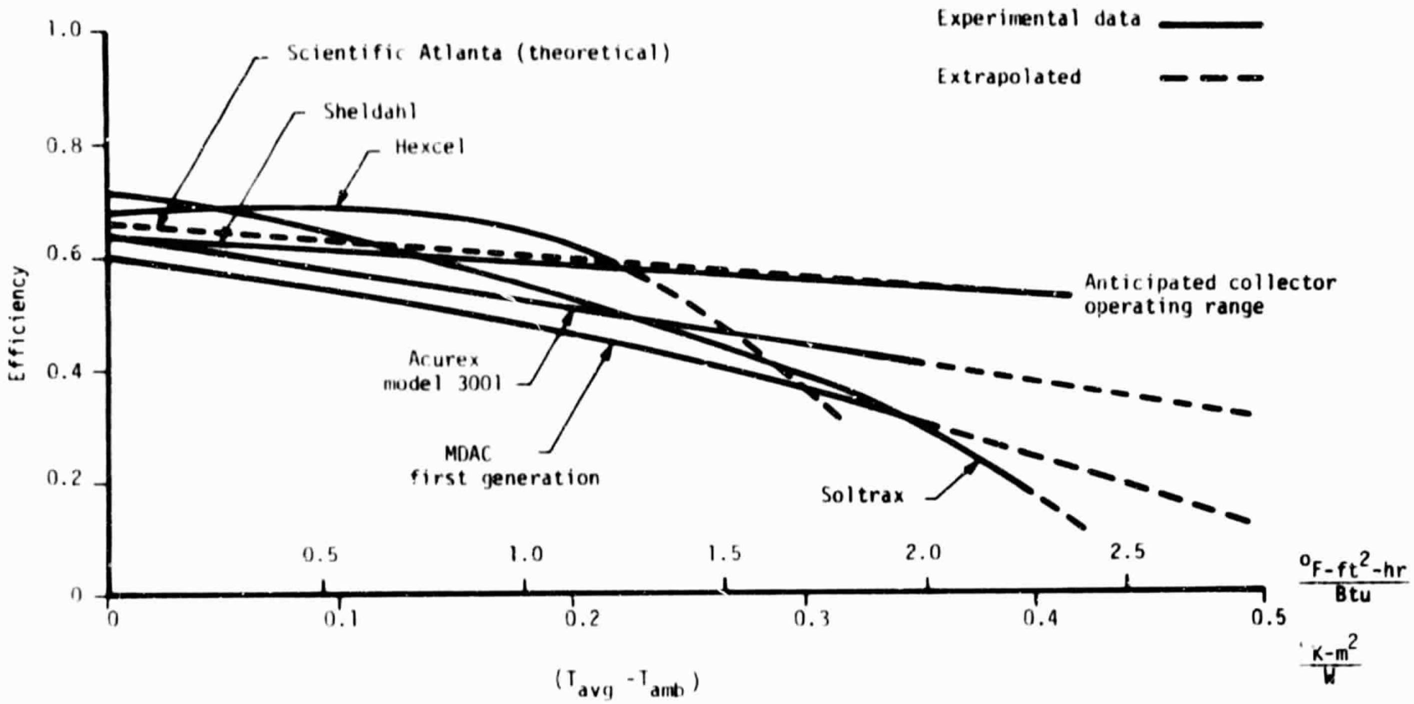


Figure B-5. Line-focusing collector efficiency comparison.

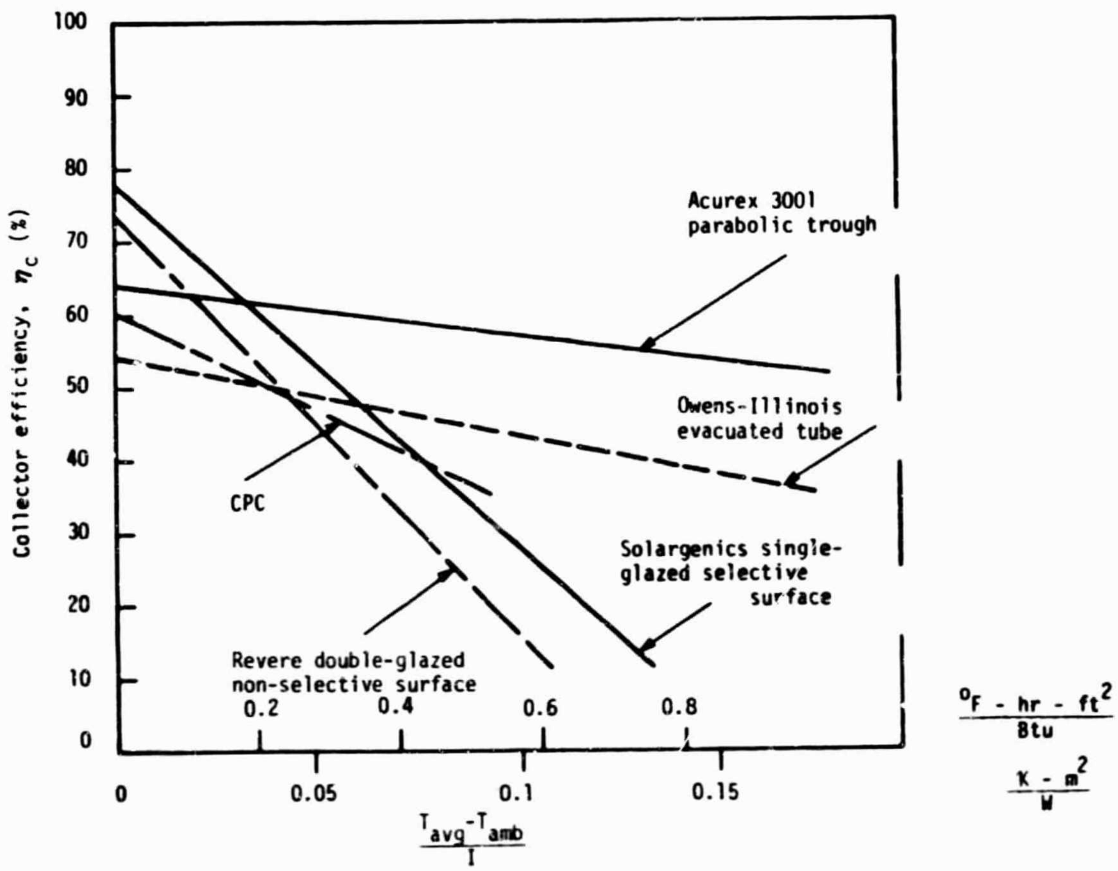


Figure B-6. Comparison of performance of non-concentrating and concentrating collectors.

delivered energy (in \$/MJ or \$/10^6 Btu), which is a function of several other collector characteristics, is the critical factor for comparison of collectors. These include initial and operating costs, collector orientation and tracking, and utilization of diffuse solar energy, in addition to the fundamental collection efficiency shown in Figure B-6. The analysis in Section 3 includes all of these factors in comparing systems and collectors, so that the results truly allow selection based on system LCC's.

Heat Transport

An integral part of any solar thermal energy conversion system is a means of transporting the thermal energy efficiently and economically from the point of collection to the point of application. This is important because solar systems differ from conventional systems in that they are very capital-intensive; it often is more economical to transport the thermal energy efficiently than to compensate for losses with additional collector area. Heat lost in transit can be a significant fraction of the total energy flow in solar thermal systems. The complexity of the energy transport process varies considerably; it may involve two or more heat transport fluids and one or more heat exchangers.

Most solar collectors use a liquid heat transfer medium, although some use a gas or vapor. Some of the common possibilities for transport fluids are:

- Liquids (water, water and glycols, hydrocarbon oils, liquid metals)
- Gases (air, helium, carbon dioxide, ammonia)
- Vaporizing liquids (water/steam, organics)

The only fluids which have seen substantial application to date are water, water and glycols, water/steam, hydrocarbon oils, and air. Several of the others (e.g., helium, liquid metals) have seen limited usage and have received serious consideration. Heat pipes, using a vaporizing/condensing fluid in an enclosed space, are potentially useful and have been applied to solar thermal systems in the past. If collector operating temperatures in excess of 373 K (212° F) are required, then pressurized water, water and glycol additives, steam, or common organic heat transfer fluids (e.g., Dowtherm, Therminol) are the most logical choices. While no significant technical problems exist in this area (since the technology of heat transport with these common media is well understood from years of experience in conventional systems), one must pay particular attention to the economic trade-offs between system first cost and operating costs as reflected in parasitic pumping power, transport heat loss, and collector performance.

ENERGY STORAGE

Virtually all solar thermal energy systems require some minimal storage capability to smooth out transients in the solar input and to take better advantage of available solar energy. While energy may be stored in a wide variety of forms (potential, kinetic, chemical, or thermal), thermal energy storage is the most common. Energy storage is a necessity for any solar system which supplies energy to a demand source on a continuous basis (i.e., day and night, under various weather conditions). In reality, economic considerations generally dictate a functional upper limit to storage capability in terms of consecutive operating days without solar input. Thermal Energy Storage (TES) can be subdivided into sensible heat, latent heat, and thermochemical storage. Since the last type is still far from commercial availability, it will not be included in this discussion. The remaining two types are basically similar in terms of their storage medium, containment unit, insulation, transfer fluid, heat exchangers and associated piping, valving and control components. While the storage medium is of primary interest in a cost/performance optimization, the containment unit and insulation can also have major cost impacts on storage design (especially at high pressures and temperatures). Whenever possible, it is desirable to use the same fluid for the transfer fluid and the storage medium to avoid the need for a separate heat exchanger.

The most significant parameters relative to TES are:

- Temperature level
- Costs per unit mass ($/kg)
- Volumetric heat capacity (J/m^3)
- Net storage efficiency

A summary of storage media and applicable temperatures is given in Table B-2.

Sensible Heat TES

Sensible heat storage, by definition, utilizes the energy associated with the temperature change of a solid, liquid or gaseous material. In this type of energy storage, the storage medium undergoes no phase change as heat is transferred to or from the TES subsystem.

Gases are undesirable as storage media, since they require large, expensive containment devices, due to their low density and specific heat, and the large volumetric or pressure changes involved during a heat-transfer process.

Solid media (e.g., rocks used in conjunction with air-heating collectors) are practical for some applications, but are generally not competitive with liquid media for larger systems, except in the packed-bed concept discussed later in this section.

Selection of a suitable liquid for sensible heat storage should be based on the following criteria:

- Low cost
- Availability in required quantities
- High specific heat

TABLE B-2. THERMAL ENERGY STORAGE MEDIA

Temperature range	Sensible heat				Latent heat		
	Water	Hydro-carbon oil	Packed bed	Molten salts	Salt hydrates and eutectics	Organic compounds and eutectics	Combined phase-change and sensible heat
High (477 to 644 K) (400° to 700° F)	X	X	X	X	X		X
Medium (366 to 477 K) (200° to 400° F)	X	X	X		X		X
Low (294 to 366 K) (70° to 200° F)	X				X	X	
Cold (277 to 289 K) (40° to 60° F)	X				X	X	

- High density
- Low vapor pressure
- High thermal conductivity
- Non-corrosibility
- Stability under thermal cycling
- Nontoxic properties and other safety characteristics

There are two principal advantages of the sensible heat storage concept. First, its single-phase heat transfer mechanisms are well understood. Secondly, liquids used for storage may also be circulated through the solar collector/receiver subsystem. This can eliminate the cost and temperature penalties incurred when transferring heat from the collector fluid to the storage medium through a heat exchanger.

The principal disadvantage of the sensible heat storage concept is that the storage temperature declines as thermal energy is transferred from the storage subsystem to the load. This disadvantage can be avoided by properly designing the storage device and the method of extracting energy from it.

Figures B-7 and B-8 illustrate two design concepts which allow a relatively constant-temperature energy extraction. In Figure B-7, a two-tank sensible heat storage concept is presented. The hot thermal storage fluid is stored in a high-temperature tank. To withdraw energy from the tank, this hot fluid is circulated through a heat exchanger. The cooled thermal storage fluid emerging from the heat exchanger is then transported to the low-temperature tank. Fluid is therefore cyclically pumped from one tank to the other as the storage is charged and discharged. Using this mode of operation, the temperature conditions in the heat exchanger are maintained relatively constant as thermal energy is progressively extracted from the high temperature tank.

The primary disadvantage of this approach is the cost of two tanks. However, instead of two separate tanks, a single tank equipped with a movable partition to separate the hot and the cold fluid reservoirs might be used. While this would minimize tank cost, it is a difficult system to build.

The second approach, as illustrated in Figure B-8, is the stratified thermal storage concept. By employing a tall, slender tank, it is possible to establish a temperature gradient in the storage fluid along the vertical axis of the tank. This gradient is maintained by the density difference between the hot and cold fluids and a thermocline separates the high-temperature fluid at the top from the lower-temperature fluid below. The hot fluid can be withdrawn from the upper part of the tank, circulated through a heat exchanger to extract the energy, and then returned to the tank through the bottom. In doing so, the thermal boundary in the tank shifts toward the top. This allows a relatively constant-temperature energy extraction from the tank over an appreciable range of thermal storage capacity.

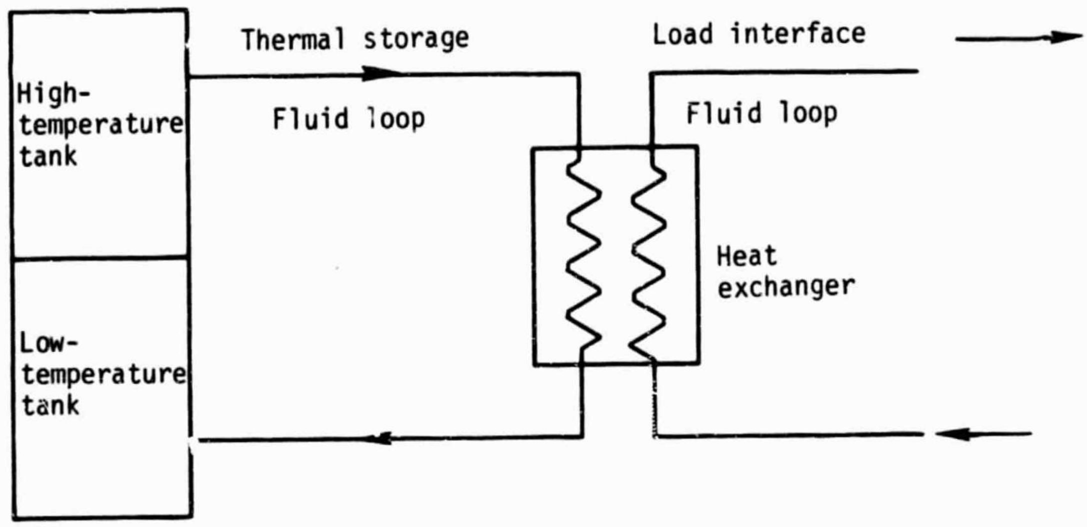


Figure B-7. Two-tank sensible heat storage.

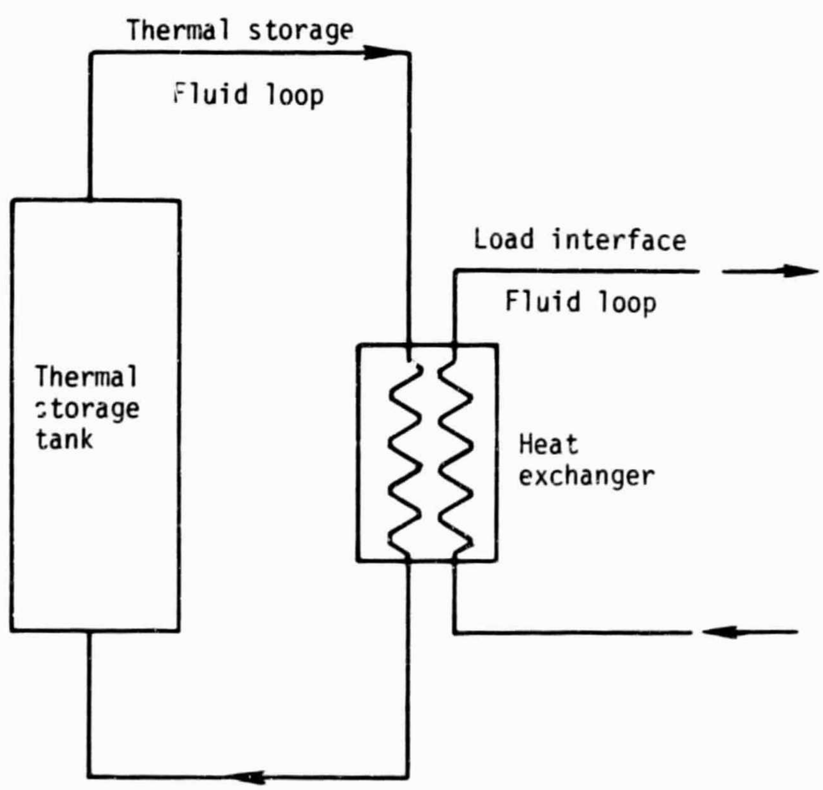


Figure B-8. Stratified sensible heat storage concept.

While liquids are the most commonly used sensible heat storage media, solids also have potential. Possible solid storage media include rocks, cast iron, refractory blocks, and chemical compounds. One of the more promising sensible heat storage media for high temperature applications is a packed bed, combining solid and liquid media. The primary advantage of the packed-bed concept is that it can provide a stable thermocline with a reasonably high energy storage density, while allowing a significant reduction in the mass of fluid storage media required.

Due to its high volumetric heat capacity and extremely low cost, water is the most likely storage candidate for low-temperature or cold TES. Over the temperature range of interest in solar thermal energy systems, water is actually superior to any other medium, but its high vapor pressure at high temperatures requires relatively expensive containment vessels. Therefore, for medium- and high-temperature storage, hydrocarbon oils (and packed beds using hydrocarbon oils) have the advantage of low-pressure operation, although materials costs are higher and heat capacity is lower.

Phase-Change Thermal Storage

Phase-change storage avoids the temperature degradation of sensible heat storage by isothermally transferring thermal energy to and from a storage material at the material's melting or boiling point temperature. Solid/liquid phase change is preferable to liquid gas phase change, because of the added cost incurred for storage containment vessels needed to handle the larger variations in volume of a liquid/gas phase change. In a few cases, solid/solid phase change can provide significant additional storage capacity.

Selection of a suitable latent heat storage material should be based on the following criteria:

- Low cost
- Melting point temperature compatible with subsystem requirements
- High latent heat of fusion
- High density and specific heat
- High thermal conductivit
- Low vapor pressure
- Low volume change on melting
- Congruent melting characteristics
- High rate of crystallization on cooling
- Stability under thermal cycling
- Nontoxicity and other safety characteristics
- Non-corrosibility

Phase-change storage has two principal advantages: first, heat is transferred to and from storage isothermally; secondly, the volume of storage material is generally less than with sensible heat storage -- allowing a less costly storage containment vessel to be used.

The primary disadvantage of phase-change media involves the difficulty of transferring heat from the bulk of the storage material as the contact surface freezes. By mechanically scraping the freezing surface, or microencapsulating the medium to increase the surface/volume ratio, this problem could be solved.

Two other potential disadvantages of phase-change media deserve mention. First, the volume change of the storage material upon phase change can leave voids or exert excess pressure between the material and the heat transfer surfaces. Secondly, because the material properties degrade after repeated cycling, melting point drift or a decrease in the material's latent heat of fusion can occur.

Phase-change storage media can be classified into three groups:

- Inorganic salt hydrates and eutectics
- Organic compounds and eutectics
- Combined phase-change and sensible heat materials

In the first two groups, the storage function would be considered to occur at or near the phase-change temperature. In the third group (formed from members of the first two groups) the specific heat of the material is also important. Combined phase-change and sensible heat materials are best applied where high temperature swings are required. For that reason, the third group is primarily made up of candidates for high-temperature storage. The first group, salt hydrates and eutectics, is applicable to all four temperature ranges and includes a large number of media with identified potential for solar application. Organic compounds and eutectics have been considered mainly for low temperature storage.

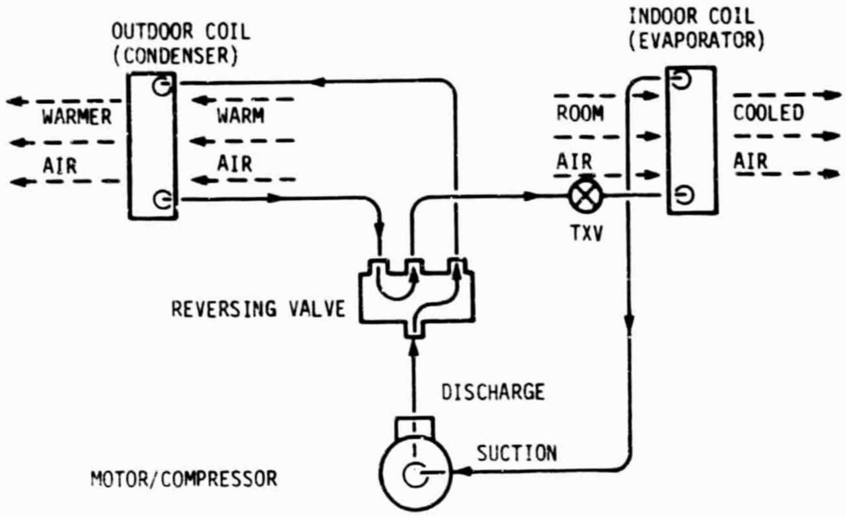
HEATING AND COOLING COMPONENTS

Major space heating and cooling components are summarized below. Common components such as air handlers, coils, pumps, heat exchangers, boilers and standard vapor compression chillers are not discussed. The components which interface most readily with solar thermal energy input are heat pumps, absorption chillers and Rankine/mechanical chillers.

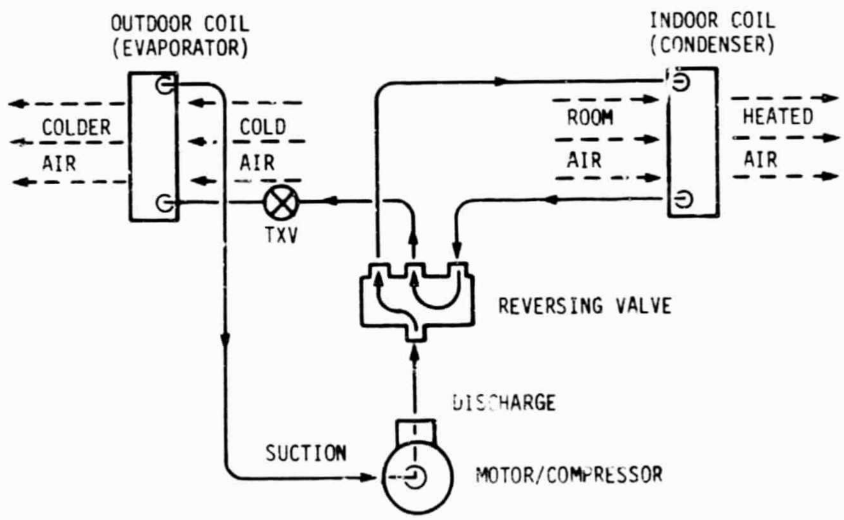
Heat Pumps

Heat pumps typically heat a building by removing thermal energy from a low-temperature source such as the outdoors and discharging it at a higher temperature sink such as working or living space. Although the term normally refers to the desired heating effect, technically a refrigerator also can be considered a heat pump operating in a reverse mode. Since all of the major components for a cooling system are included in a heat pump, the most cost-effective use of one would be as a reversible heating/cooling unit.

Figure B-9 shows a schematic of a typical reversible air-to-air heat pump. Other types include water-to-air and water-to-water heat pumps. In the first two types, a valve reverses the flow of refrigerant in the unit



a. Cooling and defrost cycle.



b. Heating cycle.

Figure B-9. Basic air-to air heat pump system.

so that the indoor coil acts as an evaporator on the cooling cycle and a condenser on the heating cycle. Water-to-water heat pumps can be optimized for peak efficiency of condenser and evaporator, since these components serve only one function. Instead of reversing refrigerant flow, the water-to-water heat pump reverses the water flow, directing warm water leaving the condenser to the air handler for heating or sending chilled water leaving the evaporator to the air handler for cooling.

The COP of a typical heat pump ranges from three to six in the heating function and from two to five in the cooling function. The major factors which affect COP are the difference between evaporating and condensing temperatures (temperature lift) and compressor volumetric efficiency. The greater the lift, the poorer the performance. The control of lift is limited, because the source and sink temperature are fixed by application. Increased heat exhange surface can often lower the temperature lift, but increase the cost of the heat pump.

Heat pumps have good potential for application in solar energy systems because low-temperature flat-plat collectors (unglazed or single-glazed) can be used for thermal input to water-to-air or water-to-water heat pumps. This sort of system, termed a solar-assisted heat pump, allows operation of the heat pump at a higher COP in the heating mode due to the higher source temperature. At the same time, the collectors are operated at lower temperatures than required for direct heating systems, and thus have higher collection efficiency. When cooling is required, the heat pump is reversed and operates as a chiller, with no input from the collectors. Solar-assisted heat pumps are available commercially in residential sizes, but large-capacity units for the buildings considered in this study are unavailable.

Absorption Chillers

Absorption chillers normally use the proven lithium bromide and water refrigeration cycle (Figure B-10). The difference between an absorption chiller and its vapor-compression counterpart is that the former requires only thermal energy (and a small electric pump to circulate the refrigerant-absorbant solution) to drive the cycle. Vapor is compressed by converting low-pressure refrigerant vapor to liquid by absorption by a secondary fluid. The refrigerant-absorbant mixture is then pressurized in the circulating pump. The mechanical work required to pressurize this solution is much less than would be required to compress the same mass of refrigerant vapor, because the volume of the mixture is much less than that of the vapor. The refrigerant vapor, driven out of solution by the addition of thermal energy in the generator, is then condensed and expanded back to the low-pressure state to achieve refrigeration.

Absorption chillers come in one- or two-stage models, with rated COP's varying between 0.65 and 1.2. A two-stage absorption chiller differs from a single-stage model in that it includes two generators: one, at a high temperature and pressure, which is driven by an external heat source, and a second generator, at a lower temperature and pressure,

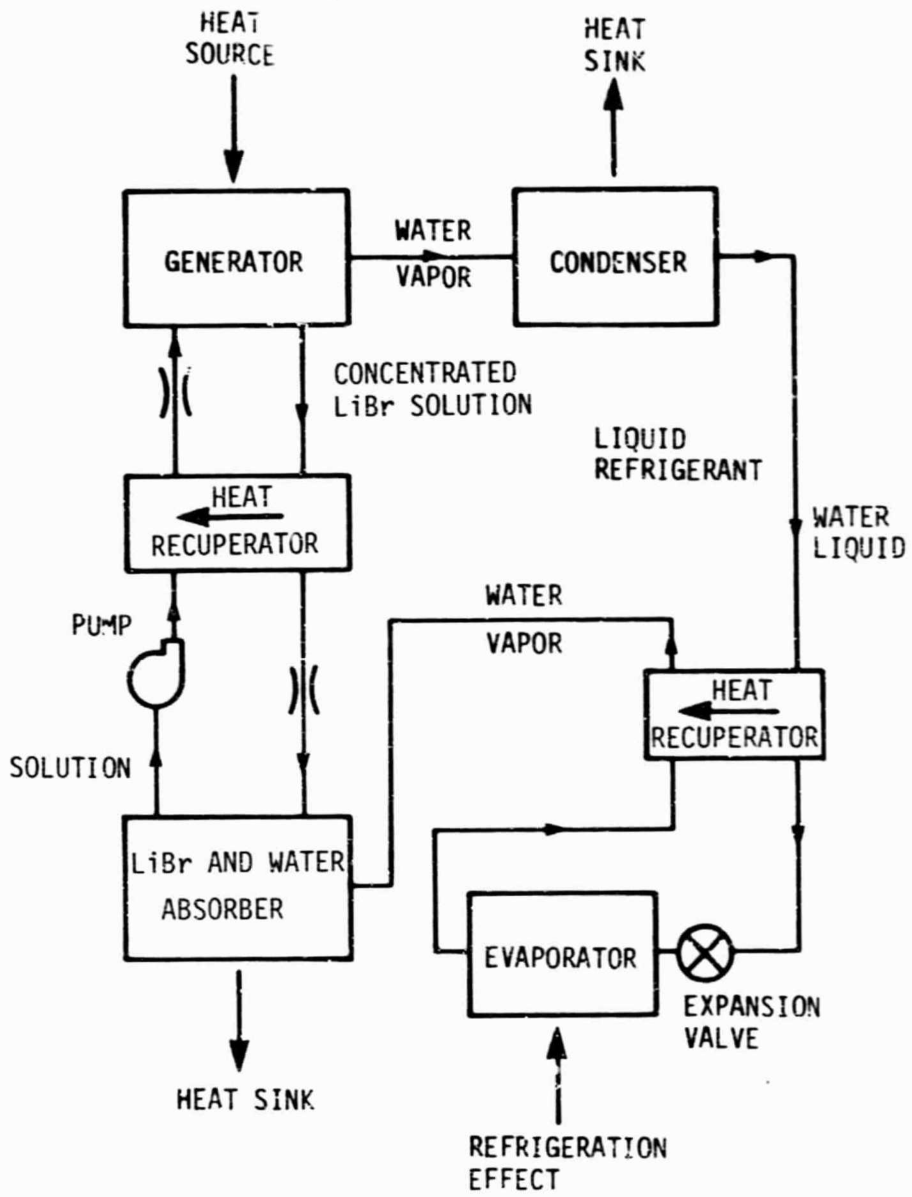


Figure B-10. LiBr/water absorption system schematic.

which is driven by the heat of condensation of the vapor from the first generator.

Generator inlet fluid temperature requirements for single-stage absorption chillers range from 355 to 422 K (180° to 300° F) with about a 5.6 K (10° F) temperature drop across the generator. The inlet requirements for a two-stage chiller are 422 to 477 K (300° to 400° F) hot water or steam. While the instantaneous COP of an absorption chiller may meet or exceed the manufacturer's ratings, the seasonal average COP of a single-stage absorption chiller is typically closer to 0.5 than 0.65 due to cycling and warm-up losses.

Since absorption chillers are driven directly by thermal energy (except for the relatively small amount of energy required by the circulating pump), they are well suited to solar space cooling. Mechanical vapor compression cycles can be also used for solar application if they are used with a thermally driven Rankine cycle.

Rankine/Mechanical Chillers

Vapor compression chillers are characterized by their comparatively high COP's, but require mechanical energy input. One way to obtain this input is to use a solar-powered Organic Rankine Cycle (ORC) to generate mechanical energy. This energy in turn drives a vapor compression chiller.

Figure B-11 shows a schematic of a Rankine/mechanical chiller in which the ORC turbine drives the chiller compressor directly. While the COP range of the refrigeration cycle is far greater than that of an absorption chiller, the overall COP of the Rankine/mechanical chiller is reduced by the relatively low thermal efficiency of the Rankine cycle. Figure B-12 compares COP for Rankine/mechanical and absorption chillers as a function of heat source inlet temperature. For the typical design conditions indicated for summer operation, the Rankine/mechanical chillers consume 15 to 40 percent less thermal energy than absorption chillers. However, the estimated cost for one unit is about 10 times that of a single-stage absorption chiller, since the Rankine/mechanical chiller is not available commercially and must be specially built.

ELECTRICITY GENERATION

Generation of electricity using solar energy has not been considered as a viable option for the building energy loads in this study, due primarily to the very high costs associated with solar electrical generation. The discussion below is presented as a means of adding perspective to the previous discussions regarding thermal uses of solar.

Electricity can be generated from solar energy by two principle methods which are markedly different. In solar thermal electric systems, concentrating collectors are used to attain high temperatures to drive heat engines, involving turbines or pistons, which in turn drive generators that produce electricity. In solar photovoltaic electric

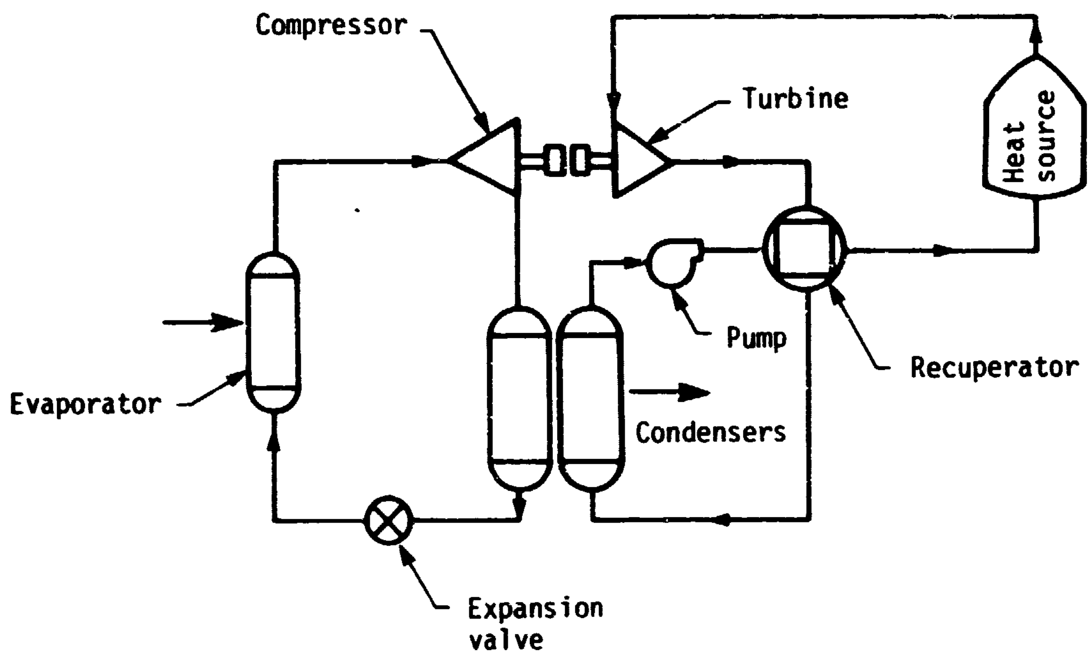


Figure B-11. Schematic of Rankine/mechanical chiller.

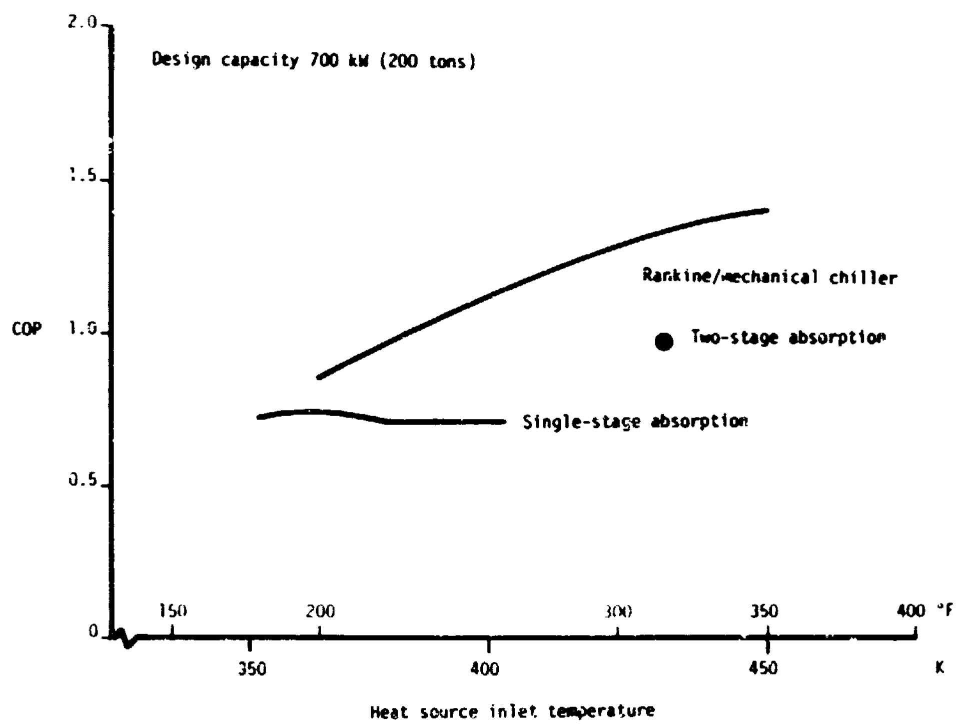


Figure B-12. A comparison of COP for three candidate thermal chillers.

systems, sunlight is converted directly to electricity (DC) using semiconductor photovoltaic cells.

Thermal Electric Systems

Solar-driven thermal electrical power generation requires the use of one of the typical heat engine cycles to convert the thermal energy to mechanical energy, which in turn is converted to electrical energy with a generator or alternator. The candidate heat engine cycles include the familiar Rankine, Brayton and Stirling cycles. The Rankine cycle (for which the best example is the common steam turbine used in large-scale power generation) involves boiling the working fluid (water or organic), which is then expanding through a turbine or reciprocator to turn a shaft. Rankine cycles can operate at temperatures from about 373 K (200° F) to over 803 K (1000° F) with thermal efficiencies ranging from under 20 percent to about 40 percent. The Brayton cycle is typified by the gas turbines used to generate power on medium to large scales, and involves heating the gas (usually air) and then expanding it through a turbine. Gas turbines require high temperatures, on the order of 973 K (1500° F) and higher, to achieve thermal efficiencies in the range of 20 percent to over 30 percent. Stirling engines are reciprocators (piston engines) which are mechanically more complex than Rankine or Brayton cycles, but offer higher efficiency at lower temperatures than gas turbine Brayton cycles. They are not competitive, however, with Rankine cycle engines at the temperatures which can be obtained with current solar collectors (up to about 588 K, 600° F).

Of the three cycles, Rankine cycles have found the widest application in solar systems due to their higher relative efficiencies at lower heat-source temperatures. Systems ranging in size from several kW to 500 kW are either currently operating or in the design phase. Most of these systems use organic working fluids (e.g., toluene and freon) in the Rankine cycle because of the higher efficiencies relative to steam at current practical temperature limits. The installed cost of these systems is typically around $20,000/kW, which is much too high to compete with conventional fossil fuel power generation.

A typical Rankine-cycle prime mover/generator subsystem is shown in Figure B-13. While the heat engine efficiency will typically improve with increasing heat-source temperatures, the collector subsystem performance will fall off at elevated operating temperatures. The combined system efficiency for a line-focusing concentrator with an ORC turbine, will be in the range of 10 percent at peak temperatures of 588 K (600° F).

Photovoltaic Electric Systems

Photovoltaics present one alternative to thermal-powered generation of electricity, providing direct conversion of sunlight to electricity. Both concentrating and non-concentrating photovoltaics can be used to generate electricity.

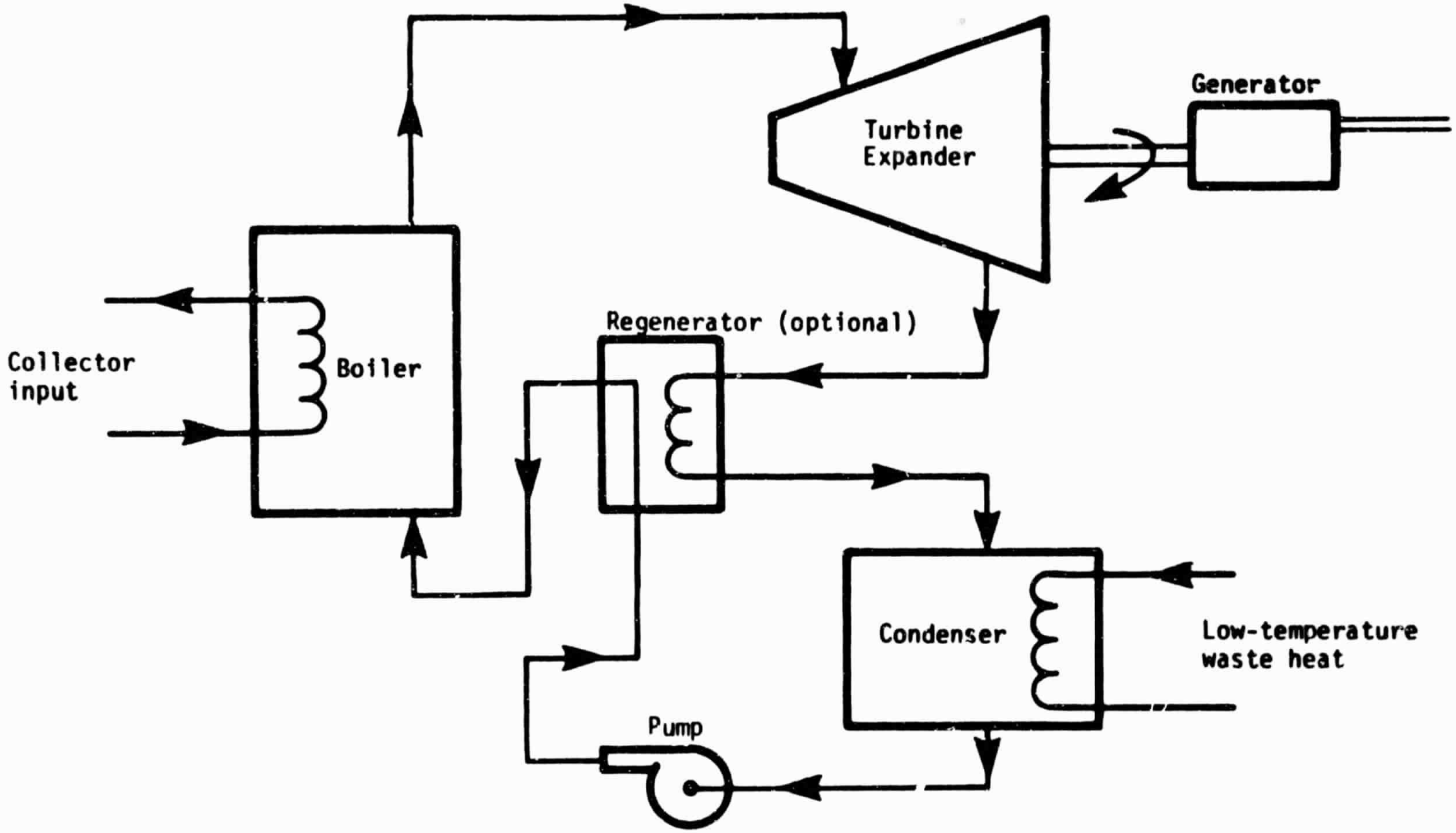


Figure B-13. Schematic of a Rankine-cycle prime mover/generator subsystem.

Gallium arsenide, single-crystal silicon, and cadmium sulfide solar cells can be used in either concentrating or non-concentrating collector applications. Single-crystal silicon cells, the most commonly available photovoltaic cells, have demonstrated efficiencies up to 15 percent (Reference 12). Typical single-crystal silicon cells presently cost about $2,000/m^2, and a flat-plate (non-concentrating) array would require 8.7 m^2 of cells to produce a peak kW of electricity. With commercially available concentrating systems approximately 10.7 m^2 of aperture and 0.30 m^2 of cells are required per peak kilowatt. This approach implies a minimum installed cost of about $2,200/peak kW for a concentrating system. Non-concentrating systems would cost roughly $17,400/peak kW. Concentrating photovoltaic systems thus offer more promise for becoming economically competitive in the near term, although future decreases in cell costs may shift the balance to non-concentrating systems.

A variety of concentrating optical systems can be used in conjunction with photovoltaic arrays to produce both electrical and thermal energy. Both point- or line-focusing concepts are being developed. Point-focusing options include paraboloid reflectors and Fresnel lenses and require two-axis sun tracking. Line-focusing concepts include parabolic troughs, Fresnel lenses, Fresnel reflectors, and movable slat reflector configurations. These configurations can be used with one-axis sun tracking or, in some cases, one-axis tracking with a seasonal tilt adjustment. In each of the above concentrating systems, the array temperature must be maintained below approximately 273 K (212^o F) for silicon cells.

APPENDIX C

CANDIDATE SOLAR HEATING AND COOLING SYSTEMS

This section describes the solar heating and cooling systems evaluated in the study. Since detailed system descriptions would vary from facility to facility, only general system schematics are presented and discussed. The systems considered are:

- Domestic Hot Water (DHW)
- Space heating and DHW
- Space cooling
- Space heating and cooling and DHW

SYSTEM FUNDAMENTALS

The proper way to design a solar energy system is to use a systems approach. The interaction of components and subsystems must be evaluated to determine the best component arrangement and optimum set of operating conditions for the whole system. Typically, the most expensive components of a solar energy system are the solar collectors and thermal storage subsystems. The interaction of these two subsystems is of primary interest.

While there are several ways in which the collectors, storage, auxiliary energy, and load can be connected for any given application, fundamental points must be kept in mind to design an efficient, cost-effective system. Some of these are:

- The operating temperature of the collectors should be kept to a minimum
- The use of heat exchangers should be minimized
- The operating temperature range for sensible heat storage should be as great as practicable
- Auxiliary energy should not be added to storage

Each of these points is discussed briefly below. Where there is a conflict, a trade-off must be performed to determine the most cost-effective compromise.

Since all solar collectors operate more efficiently at lower temperatures, less collector area (and consequently less capital investment) is required at lower operating temperatures for a given energy demand. The required collector operating temperature is determined by the end-use temperature and the temperature drops associated with heat exchangers and thermal losses within the system. It is therefore desirable to minimize the number of heat exchangers whenever possible, and to provide good insulation for piping and other components.

At present the most cost-effective, commercially available thermal energy storage means is sensible heat storage. By definition, sensible heat storage requires a significant operating temperature range to effect energy storage. The larger the range, the smaller the thermal storage mass and containment vessel volume. In general, therefore, the higher the operating temperature range, the lower the first cost of storage. The exception to this, however, is with high vapor pressure fluids (such as water above 373 K, 212^{o} F). For such a fluid, the pressure rise associated with the higher temperatures may require excessively costly containment vessels. Therefore, a trade-off between the cost of fluid and tank is required for the storage subsystem.

Because minimizing collector temperature and maximizing storage temperature range conflict, good design practice involves a trade-off between these two subsystems to provide the most cost-effective integrated system. In any case, however, the addition of heat exchangers is detrimental to performance.

Auxiliary energy is required as a backup for virtually any solar energy system in which cost is a consideration. To ensure effective system operation, several factors must be considered when integrating auxiliary energy into the system. It is undesirable to add auxiliary energy to the TES, since this would have two detrimental effects. First, no storage system has a 100 percent retrieval efficiency; the auxiliary energy is therefore best stored in its original form as fossil fuel, hydroelectric, or nuclear energy. Secondly, the storage temperature rise would cause higher collector operating temperatures and consequently lower efficiencies. Another consideration with respect to auxiliary energy integration is that it is desirable to design a system that allows solar energy to serve as a preheater. This provides part of the system load during low insolation or high demand periods.

These considerations are evident in the system configurations described below. While other variations certainly exist, these systems represent concepts which are practical from control as well as efficiency and cost standpoints.

SYSTEM TYPES

Domestic Hot Water

Solar DHW systems can be configured in a number of ways. Some of the options for components, materials, and configurations are:

- Collector fluid
  - -- Potable water
  - -- Nonpotable heat transfer fluid

- Storage
  - -- Potable water
  - -- Nonpotable heat transfer fluid
- Collector/storage interface
  - -- Direct (no heat exchanger)
  - -- Internal heat exchanger
  - -- External heat exchanger
- Auxiliary backup
  - -- At main storage
  - -- Instantaneous boost heater
  - -- Separate series storage heater

While it is generally desirable to eliminate heat exchangers, direct circulation of line pressure domestic water through the collector is often not practical. Corrosion of collectors (due to the high oxygen content of potable water) and scaling (due to the high mineral content) often require the use of a separate fluid loop for the collectors.

Large-volume storage of potable water requires specially lined storage tanks which may prove less cost-effective than mild steel or fiber glass tanks. Fiber glass tanks are only suitable for use with treated water or heat transfer fluids.

If separation of fluids is required, the designer should avoid the use of internal immersion-type heat exchangers, due to their relatively low heat transfer coefficients and large temperature differentials. Separate external shell and tube heat exchangers can be selected to provide adequate heat transfer with relatively small temperature penalties, but with the required addition of a separate circulation pump.

The location of the auxiliary energy input can also have a major impact on system performance. As discussed above, it is generally best to add auxiliary energy only on demand, and preferably only to boost the solar preheated fluid up to its required end-use temperature. Therefore, as indicated in Figure C-1, auxiliary energy supplied through an instantaneous boost heater will provide the optimum solar performance by allowing the solar system either to act as a preheater or to meet the full load whenever adequate energy is available.

No standby energy losses are incurred in the boost heater, but the peak capacity of the unit is quite high. In order to alleviate this problem, a separate series storage tank, which is maintained at its minimum set point by auxiliary energy, can be employed as illustrated in Figure C-2. Due to the storage volume of the second tank, peak auxiliary energy demands can be reduced to an acceptable level. While this system has the disadvantage of incurring standby energy losses under some conditions, it provides a practical compromise to the instantaneous boost heater approach.

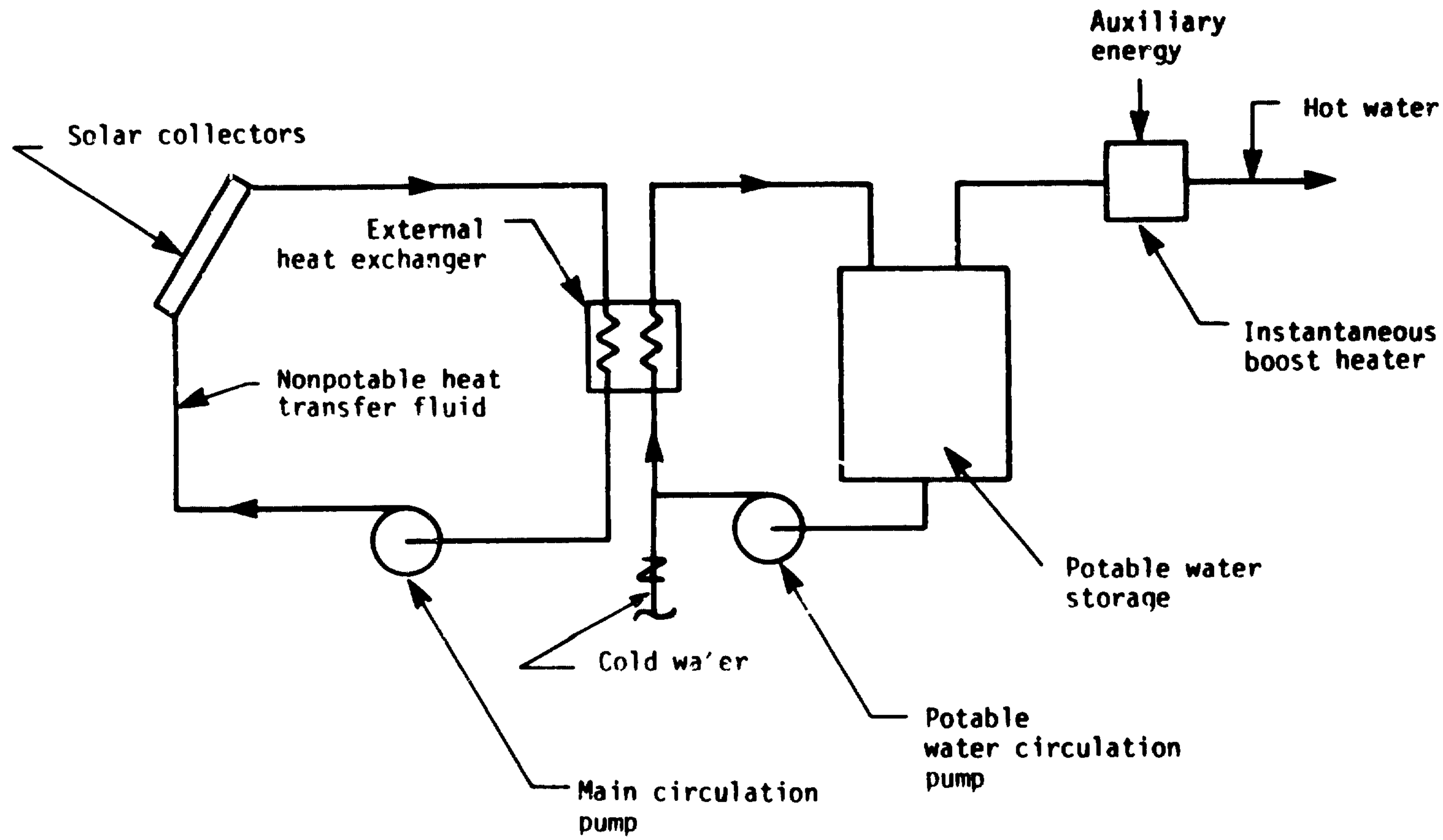


Figure C-1. DHW system with boost heater.

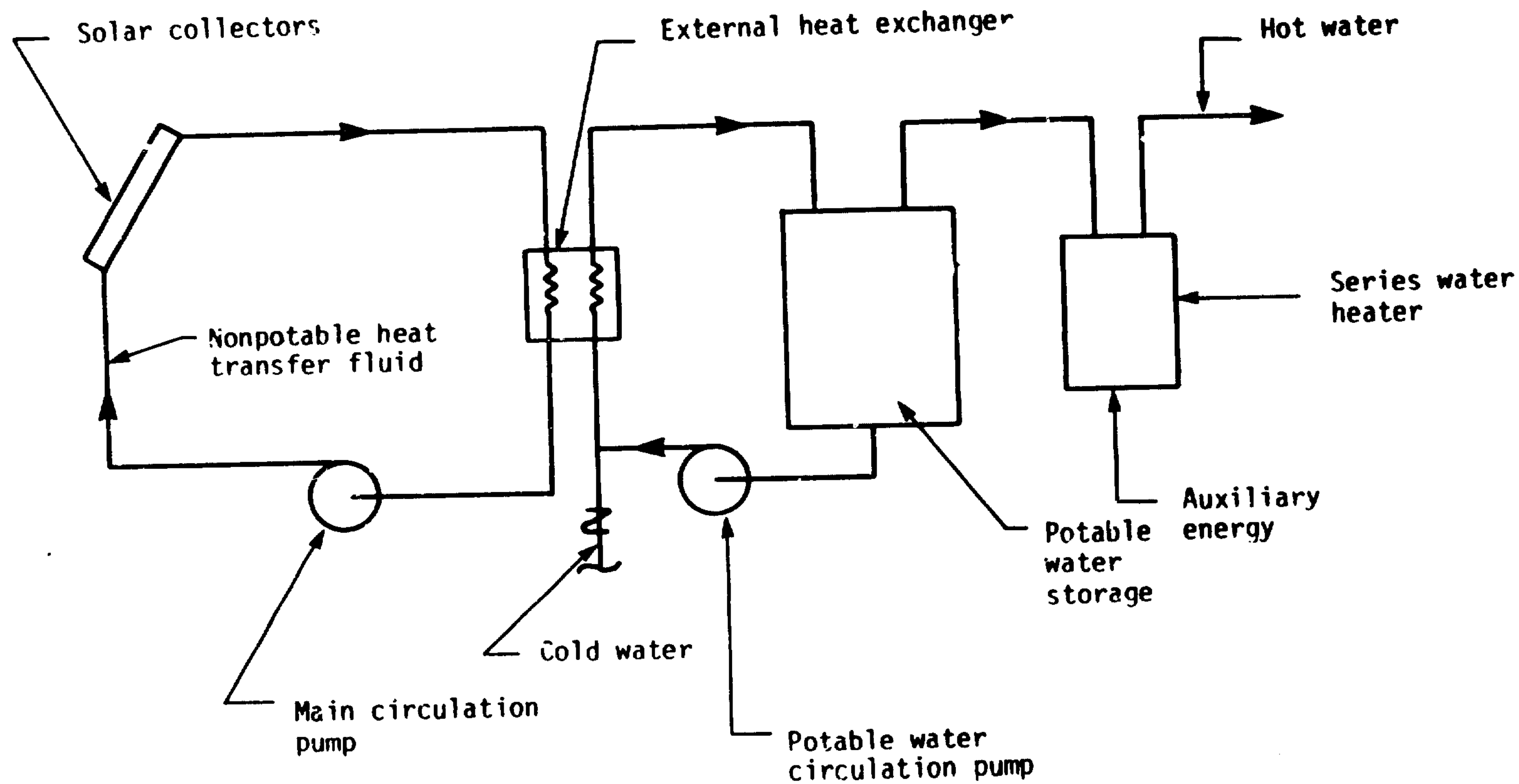


Figure C-2. DHW system with a series water heater.

Space Heating And DHW

As with DHW systems, a number of options exist for the integration of a solar space heating system. The options listed for DHW systems also apply to space heating system.

Direct connection of the collector and storage subsystems without heat exchangers is often possible with space heating systems. At times, however, freeze protection considerations may lead to the use of antifreeze or other heat transfer fluid in the collectors. Since the cost of such fluids in sufficient quantities to provide the required storage mass may be excessive, such an arrangement might necessitate the use of an external heat exchanger similar to that in Figures C-1 and C-2.

A simplified schematic of a possible space heating system is presented in Figure C-3. As indicated, a common working fluid is used for the collector loop, the storage media, and the load loop. This type of design, where practical, allows the most efficient collector operation, since no temperature penalties for heat exchangers are incurred. For liquid systems, it is desirable to stratify the sensible heat storage tank to achieve a thermocline if possible. While this may increase control complexity, it will allow a reduction in storage volume which may reduce the system cost.

The inclusion of a storage tank bypass line on the load loop side serves three functions. First, it allows isolation of the storage tank from the auxiliary energy system, thereby preventing the addition of auxiliary energy to storage. Secondly, it allows part-load solar preheat of the load loop fluid when thermal storage is at a higher temperature than the return fluid from the end-use heat exchanger. Thirdly, it provides the flexibility to optimize the collector and storage performance independent of required end-use supply temperature requirements. For example, a system which requires an end-use supply temperature of 333 K (140° F), but has only an 11 K (20° F) temperature drop through the terminal heat exchanger, may best be served by an integrated collector/storage subsystem designed to operate between 355 K (180° F) and the 322 K (120° F) base temperature. In such a case, the 310 K (60° F) temperature range for storage would reduce storage requirements by two-thirds, while penalizing collector performance by an amount dependent on collector type. Based on the relative costs of collectors and storage and the efficiency curves of the collectors, the optimum operating conditions can be determined. The control valve on the bypass line would then be controlled to mix return and supply fluids to maintain the desired 333 K (140° F) supply temperature.

As indicated in Figure C-3, DHW can be provided in a fashion similar to the space heating load by adding a parallel load loop. This loop would serve as the solar input to the DHW system (as in Figures C-1 or C-2). Controls can be incorporated to prevent the addition of auxiliary energy from the solar system to the first storage tank of the DHW system.

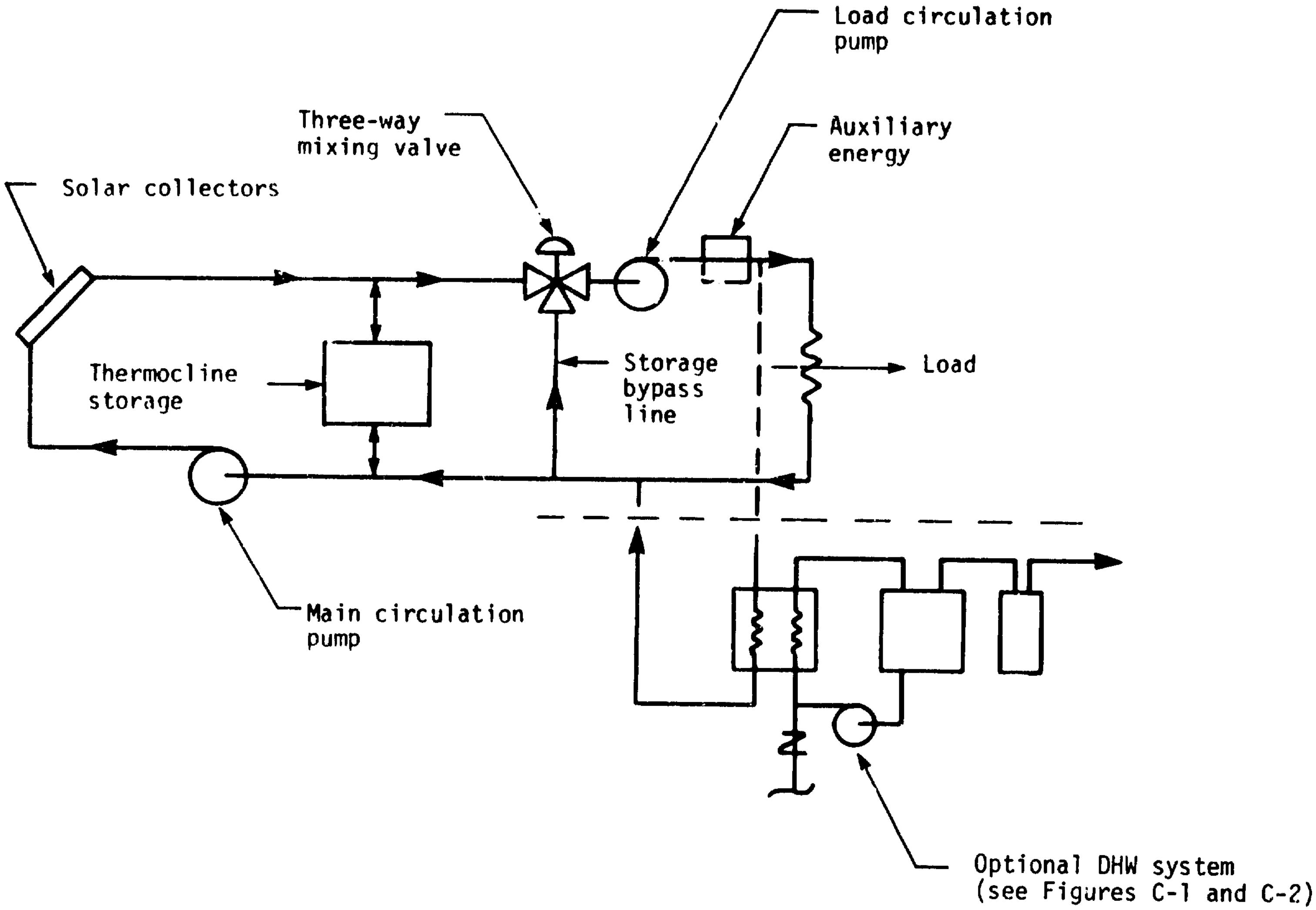


Figure C-3. Space heating and DHW system.

For solar retrofit applications, the interface with the existing system can have a major effect on the system design. It is quite common to use steam as the heat transfer medium in large space heating systems. Steam generated in a central boiler is often distributed throughout a building to supply remote heat exchangers. These heat exchangers either provide the space heating directly or heat a secondary heat transfer fluid. For integration with such a system, it may be more cost-effective to generate steam with solar energy, thereby allowing a single-point interface with the existing system. This approach is discussed in Section 4.

Space Cooling

Solar-powered cooling can be provided best by absorption or Rankine/mechanical chillers. In either case, thermal energy is supplied to the chiller, producing the refrigeration effect. It also generates waste heat which must be ejected to the atmosphere (generally through a cooling tower). Due to the relatively low COP of such thermal chillers, the heat rejection ratio, and consequently the cooling tower size, is roughly double that of a comparably sized mechanical (vapor compression) chiller.

The solar component of a space cooling system resembles a space heating system; the space heating load has merely been replaced by a thermal chiller. However, as indicated in Figure C-4, two fundamental differences often exist: chilled water storage has been incorporated in the system, and the auxiliary thermal backup has been left out.

For retrofit applications with existing vapor compression chillers, it may be more cost-effective to rely on the existing high-COP equipment to provide the cooling backup. Higher parasitic electrical requirements for absorption system cooling towers and pumps often negate the energy saved during solar operation when the absorption equipment is fired with fossil fuels for backup operation.

Cold storage, due to its relatively small range of operating temperatures, is generally less cost-effective than high-temperature storage. However, due to the relatively long warm-up period required for presently available absorption equipment, the incorporation of chilled water storage can be beneficial. A cold storage system requires less time to cycle and the number of daily cycles, with the corresponding warm-up and cool-down losses, are also reduced.

Space Heating and Cooling and DHW

The load-matching benefits of space heating and cooling systems can provide high annual utilization of available solar energy. A viable option to either the space heating or space cooling system alone is an integrated space heating and cooling system. As indicated in Figure C-5, integration of space heating and cooling and DHW into one system follows the same basic approaches set forth above. The solar side of the system resembles a separate space heating system, with auxiliary energy

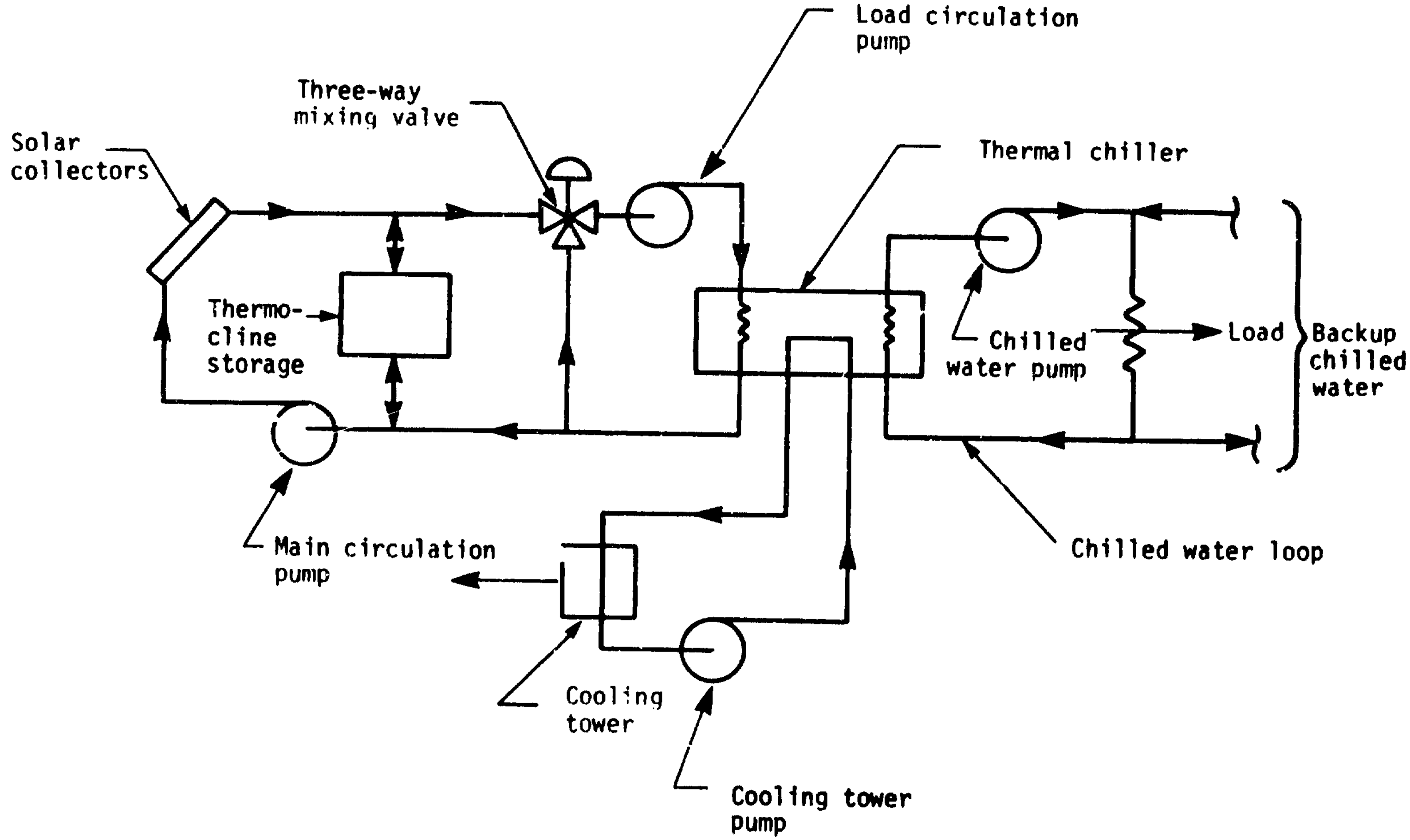


Figure C-4. Space cooling system.

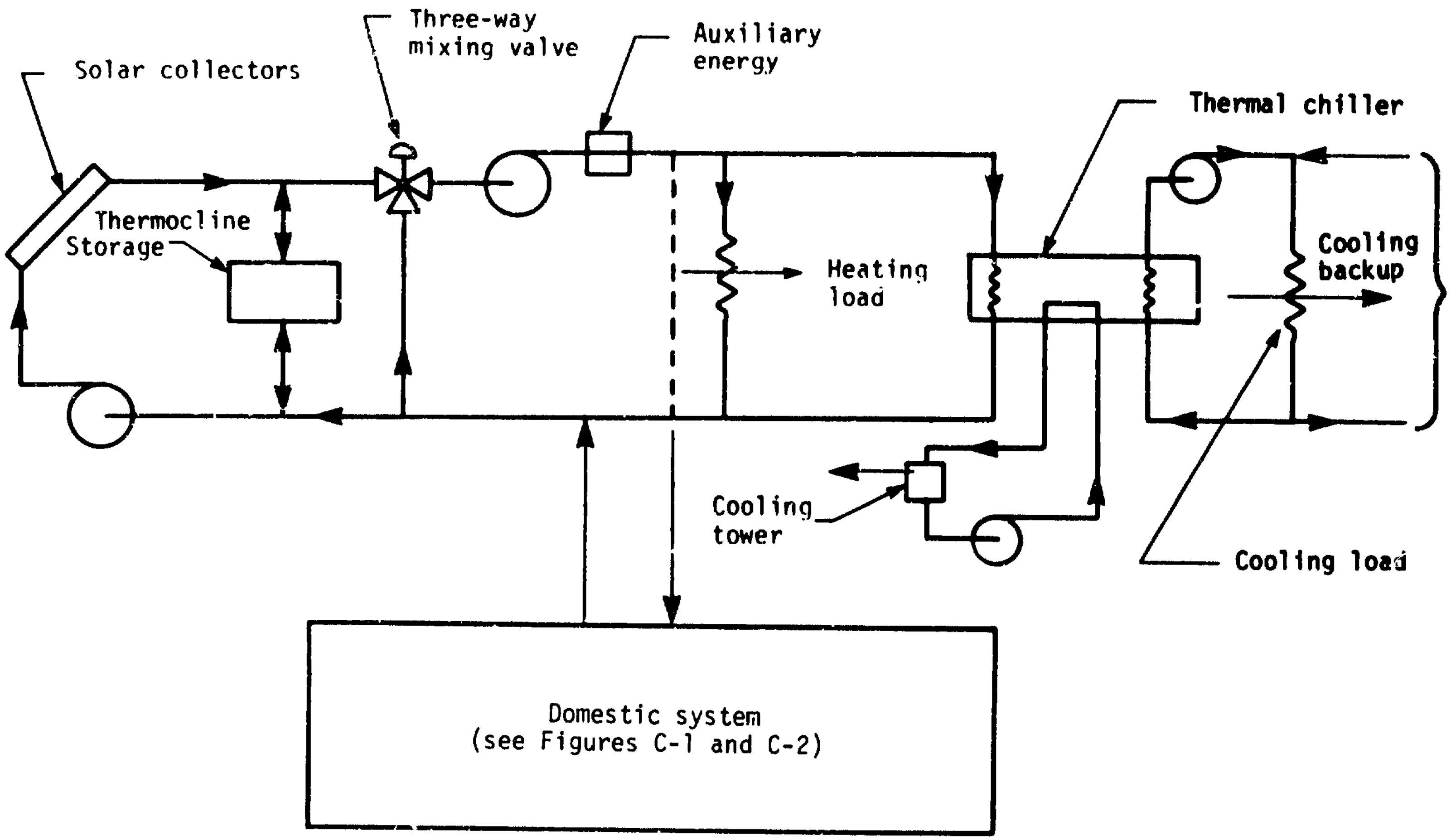


Figure C-5. Space heating and cooling and DWH.

provided. The thermal chiller is treated as a parallel load which can be isolated during auxiliary thermal backup operation to allow separate vapor compression chiller backup. Chilled water storage is provided as in the separate space cooling system to reduce cycling and enhance response.

The use of solar-generated steam to interface with existing heating systems, as discussed in Section 4, also can be incorporated effectively in an integrated heating and cooling system. Because of the higher temperature required, commercially available dual-effect absorption chillers can best be fired with steam. The cost penalties associated with solar generation of steam may be offset by the simplification of the system and ease of interface with existing equipment.