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UC CATEGORY: UC-59

COMPARISON OF SOLAR HEAT PUMP SYSTEMS TO CONVENTIONAL METHODS FOR RESIDENTIAL HEATING, COOLING, AND WATER HEATING

VOLUME II: FINAL REPORT

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FOREWORD

This report summarizes a study of residential-size solar-assisted heat pump systems. The objectives of the study were (1) to develop a consistent framework for comparing the performance of parallel and series configuration systems against competing conventional systems, and (2) to perform thermal and economic analyses.

The analysis framework described in the report forms the starting point of a standard methodology for comparative systems studies that SERI is developing in conjunction with Science Applications, Inc.

The comparative analysis of series and parallel solar-assisted heat pump systems that was performed within this framework indicates that these system configurations are neither economically competitive with conventional stand-alone heat pump systems now, nor will they be so in the future. The conclusion indicates that solar-assisted heat pumps are not an economically viable alternative for residential heating and cooling applications.

This study was begun under a general Systems Analysis Contract (No. DE-8C04-78CS-34261) with the U.S. Department of Energy, and was completed under a Basic Ordering Agreement (BP-9-8150) with SERI as part of the Systems Analysis and Testing Program of the Systems Development Division, Office of Solar Applications, U.S. Department of Energy.

The Executive Summary (Vol. I) and Appendices (Vol. III) to this report are available from the National Technical Information Service.

Lyle Groome
SERI Program Manager

ACKNOWLEDGEMENT

The results contained in this report represent the combined efforts of many people. Specifically, the authors appreciatively acknowledge the following individuals: Frank Varcolik of SAI[1] and Marshall Duvall of JRB[2] who participated in the development of, and exercised, the computer models; Ted Swanson and Ron Overton of MAI[3] who defined the residential structures, sized the conventional equipment, and performed a thorough cost analysis; Bill Kennish and Allen Stolarz of TPI[4] who compiled heat pump performance data and predicted improvements expected for next-generation heat pumps; Tom Freeman of Altas Corporation [5] who validated the usefulness of TMY data for systems studies; Andy Parker of MAI, Dave Anand of TPI, and Pete Versteegen of SAI who provided guidance and criticism; and Jerry Lowery of SAI who directed the overall effort of the contract of which this report is part.

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ABSTRACT

This study performed an analysis of series and parallel configured solar heat pump systems for residences. The year-round thermal performance for all the heating, cooling, and hot water system configurations were determined by simulation and compared against conventional heating and cooling systems in several geographic locations.

The series and parallel combined solar heat pump systems investigated are at best marginally competitive, on a 20-year life-cycle cost basis, with conventional oil and electric furnace systems. The combined solar heat pump systems are not economically competitive with conventional gas furnace or stand-alone heat pump systems for residential space heating, cooling and water heating.

The combined solar heat pump systems do offer the potential for significant energy savings as compared to conventional furnace systems and the stand-alone heat pump. The cost of that savings, however, is beyond that which the average consumer can be expected to pay. It would seem that the same energy savings could be obtained for less cost using a combination of conventional technologies, passive techniques and conservation measures.

It appears unlikely that during the next five years any of the combined solar heat pump systems studied here will be installed for purely economic reasons. It remains to be determined what, if anything, can be done to make these systems competitive.

Barring unforeseen manufacturing process or materials breakthroughs, parallel systems prices are firm. The prices listed for series systems already include low-cost site-built collectors and an optimistic estimate of the liquid-to-air heat pump costs, and prices on other series system components are firm. A collector cost sensitivity analysis did not offer any encouraging directions towards significant systems cost reduction.

Further development of parallel and series combined solar heat pump systems should no longer be pursued, unless justified by policy level or other non-economic factors.

An attempt should be made to identify other applications more economically suitable for combined solar heat pump systems than in residential heating and cooling (e.g., medium and high temperature process heat applications).

Advanced storage concepts and hybrid PV/thermal configurations should be analyzed for solar heat pump system applications. These concepts were not included in the present study, and their cost/benefit is unknown.

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1.0 INTRODUCTION

The purpose of this study was to establish the thermal/economic framework in which residential combined solar heat pump systems will have to compete if they are to become widely used. To accomplish this task, conventional alternatives for space heating, space cooling, and domestic water heating have been included in the study. The combined solar heat pump systems evaluated are those systems most strongly advocated by researchers and manufacturers familiar with the applications and hardware. The eight systems studied and compared are listed below.

- Conventional electric resistance heating, vapor compression cooling, and electric hot water heating
- Conventional gas heating, vapor compression cooling, and gas hot water heating
- Conventional oil heating, vapor compression cooling, and electric hot water heating
- Conventional heat pump for heating and cooling, electric resistance backup heating, and electric hot water heating
- A liquid collector, series-connected solar heat pump system for heating and cooling, electric resistance backup heating, and a separate solar domestic hot water heating system with electric backup
- An air collector, series-connected solar heat pump system for heating and cooling, electric resistance backup heating, and a separate solar domestic hot water heating system with electric backup
- A liquid-based solar space and water heating system, parallel-connected heat pump for backup space heating and for cooling, and electric resistance backup space and water heating
- An air-based solar space and water heating system, parallel-connected heat pump for backup space heating and for cooling, and electric resistance backup space and water heating.

Throughout this study, emphasis has been placed on standardizing the systems analysis process so that the systems included in this study can be compared to other systems in a meaningful way. By standardizing the systems analysis process, new system concepts can be analyzed and compared to competing systems as they are devised simply by adding them into the same comparative framework. A systems analysis capability of this type allows direct comparison of new system concepts at an early stage so that only the most promising concepts proceed to the hardware stage. In addition, for those systems which do proceed to the hardware stage, the analysis provides design input from the systems viewpoint.

1.1 BACKGROUND

Numerous studies have been made comparing combined solar heat pump systems with each other, with stand-alone heat pumps, and with electric resistance heating. In general, the scopes of these studies have been limited and procedures inconsistent, making it difficult to draw meaningful conclusions. A review of these studies uncovered three positions which researchers in the area have taken, and it is believed that most workers in the field either generally agree with one of the three conclusions (perhaps for different reasons) or do not have a strong opinion on the matter. The three positions are outlined below.

1.1.1 Series Configuration

The first position advocates that the heat pump be used in a series configuration as a transfer device between the solar system and the space conditioning load (1, 2, 3, 4). By allowing the heat pump to upgrade the temperature at which solar energy is delivered to the load, the solar collectors are freed from high temperature restraints and can operate at low collection temperatures for maximum collection efficiency. Since the performance of inexpensive collectors can equal or exceed that of more costly designs at low inlet temperatures, larger collector areas may be economically justifiable. In this way, the heat pump may remove the foremost barrier to commercialization of solar space heating systems, i.e., the cost of collecting large amounts of energy at a temperature high enough for the application.

Advocates of this position also contend that, by providing the heat pump with a liquid source controlled within a reasonable temperature range, the solar system removes the major barriers to the commercial acceptance and technical improvement of residential heat pumps. Reliability is a major barrier to acceptance of stand-alone heat pumps and the defrost cycle is the major cause of reliability problems. By providing a liquid source, the solar system removes the need for a defrost cycle. Also, by providing the heat pump with a source which varies in temperature less than ambient air, the design of the heat pump can presumably be optimized for that temperature range, thus yielding much improved coefficients of performance (COPs).

Previous system studies have identified several problems with the foregoing logic. In winter, the liquid source can drop to its lowest allowable temperature, effectively starving the heat pump and causing the system to meet the load with electric resistance heat. Advocates contend that the combination of large storage volumes, large collector arrays (inexpensive collectors), and ground-coupled storage will prevent heat pump starvation. Ground-coupled storage refers to a storage vessel design whereby the liquid is in thermal contact with the ground.

Cooling mode operation of the series system involves rejecting energy extracted from the load to the liquid storage vessel. If storage is insulated, the energy must be further rejected to ambient by some means. If storage is ground coupled, mechanical rejection to ambient may not be necessary.

The series system is difficult to integrate with the domestic water system to obtain solar water heating. During winter, the heat pump condenser is the only energy source at a consistently high temperature. In summer, the collector array may be able to deliver energy at a high enough temperature despite the low-cost construction. Rather than attempt to scavenge energy from two locations, it may be more practical to simply use a conventional solar water heating system which is not connected to the space conditioning system.

1.1.2 Parallel Configuration

The second position advocates that the heat pump use ambient air as its source (and sink) and operate strictly as a backup device to conventional direct solar heating systems (5, 6, 7, 8). This arrangement is referred to as the parallel configuration. Although the heat pump and solar system do not apparently compensate for each others' limitations in this configuration, currently available hardware can be utilized. The advocates of this system freely admit that a higher percentage of heat pump run time is spent at lower ambient temperatures than with a stand-alone heat pump, causing the parallel system heat pump seasonal COP to decrease. However, total operating hours in the defrost region may remain the same or decrease so as not to magnify the defrost reliability problem.

Conceptually, advocates of the parallel system argue that, in a series configuration, solar energy must be delivered twice: First by the solar system and then by the heat pump. Since the solar system has a much higher COP (energy delivered over electrical input) than the heat pump, parallel advocates contend that the solar system should interact with the load directly. Although this requires an upgraded solar system (better collectors), solar energy can be delivered to the load without the added penalty of running the heat pump compressor.

Parallel advocates strengthen their case by pointing to several apparent weaknesses in the series concept which may prevent it from performing as envisioned. The contention that large storage volumes, large collector arrays, and ground-coupled storage will prevent heat pump starvation in the series system is questioned. For series systems using insulated storage, unrealistically large collector arrays may be required to guarantee no heat pump starvation. Ground-coupled storage may prevent starvation, but it will also cause the heat pump to operate most of the time at the low end of its source temperature range. In this source temperature regime, current heat pumps already operate at close to their maximum practical COP and redesign for the series configuration would yield little improvement in seasonal COP. Large storage volumes serve only to lower the rate of temperature fluctuation of storage. Beyond a certain size (which depends on the quality of the collector), increasing storage has no effect on the amount of solar energy collected, and thus cannot prevent heat pump starvation. The assertion that large, ground-coupled storage can provide seasonal storage (summer-to-winter) is unverified, and, even if true, would still require the heat pump to operate at a low-source temperature.

Cooling mode operation is more straightforward, and probably more efficient, in the parallel configuration as ambient air is the sink, and energy does not have to be rejected twice (first to storage, then to ambient). Parallel systems also allow the solar space heating/cooling system to be integrated with the solar water heating system, thus avoiding duplicate plumbing and hardware costs, as is postulated for the series system.

1.1.3 Neither Series Nor Parallel

Those who fall into this category either prefer the dual-source configuration (9) or do not feel that combined solar heat pump systems will ever be competitive (10, 11). It should be mentioned that many series and parallel system advocates also have reservations about the competitiveness of those systems today, but feel that, if further development is pursued, it should be devoted to the configuration which they favor.

The dual-source configuration depends on the development of a new heat pump with two heating mode evaporator coils, using both the solar storage and ambient air as sources. Controls can be optimized to choose the source which yields best thermal performance or, if time of day electrical rates exist, lowest operation cost. It is not clear that hardware problems associated with the wide variety of possible operating sequences and temperature splits can be overcome. Apparently, compressor lubrication is a problem. Even if these problems were overcome, it is not clear that significant performance improvement would be obtained. Using a simple control which chooses the source with the highest temperature, it has been shown that dual-source systems offer no performance advantage over parallel systems (5, 7). This conclusion could change if hardware advancements radically improved COP during liquid source operation, but not during air source operation.

1.2 METHODOLOGY/STANDARDIZATION OF STUDY

A methodological approach which realistically addresses the above issues has been identified. This general approach, to be produced as a separate SAI document, was used in investigating the potential of solar systems for any application. An outline of this approach is presented below for this combined solar heat pump (CSHP) system study. The limitations imposed on the scope of the study are:

- Detached single family residential sector only
- Near-term market potential analysis only (i.e., from today to five years in the future).

1.2.1 Outline of Study Approach/Methodology

The methodological approach is outlined in Table 1-1 and consists of the following basic items:

- Regional Conventional Systems' Market Conditions
- Regional Solar Thermal Analysis
- Regional Solar Economic Analysis
- Sensitivity Analysis.

Each analysis of a generic system type will proceed according to this general methodological approach. The work-items shown in Table 1-1 are not required to be performed in the order listed; rather, it should be recognized that this methodological approach allows an overview of the system analyses tasks and permits the generic CSHP systems analyses to be based on a common foundation.

1.2.2 Standardization of Simulation Studies

In order to clarify and simplify the comparisons between various heating and cooling system simulations, it is necessary to establish standard performance and reporting measures for the simulation studies. In analyzing and comparing various system types, the ideal situation would be to have identical forcing-function input data and a standard output data reporting format. The thermal and economic performance of a given system could then be easily related to the performance of other system types reported in other studies. The inputs for system simulations which was standardized for this CSHP study include geographic locations, weather data, building load determination, and economic factors. A uniform output data reporting format was also used in writing this report.

Table 1-1

Outline of Methodological Approach
to Solar System Analyses

REGIONAL CONVENTIONAL SYSTEMS' MARKET CONDITIONS

- Fix Conventional Data Base
- Identify Predominant Conventional Systems
- Determine Base Conventional Systems Economics

REGIONAL THERMAL ANALYSIS

- Model Climatic Conditions
- Model Load Conditions
- Model Solar System Components
- Simulate Candidate Solar Systems

REGIONAL SOLAR ECONOMIC ANALYSIS

- Determine Solar System Costs
- Determine Conventional System Costs
- Compare Solar Versus Conventional System Costs

SENSITIVITY ANALYSIS

- Determine Thermal/Economic/Component Relationships
- Identify Solar System Development Areas

Standardized weather data were approached through the use of SOLMET TMY tapes. A modified tape format is presented which is directed specifically to solar system simulations. The building load determination is based on standard building structures at each location. Loads can be computed using any desired control strategy since there is a coupling of the system energy delivery and the environmentally/internally generated energy loads, where both building temperature and humidity are considered primal parameters.

2.0 DESCRIPTION OF ENVIRONMENTS AND SYSTEMS

This section describes the environmental conditions, the heating/cooling and hot water loads, and the systems which were analyzed in this study. The descriptions are of a general nature since detailed descriptions are presented in later sections of this report or in separate documents.

The environmental conditions are presented in terms of three chosen geographical locations. The rationale for choosing these locations and the meteorological specifications at these locations are presented below. The space heating and cooling loads are described through the use of "typical" detached, single family residences at each of the three locations. The domestic water heating load is a standard volume-use profile combined with the local supply water temperature. The heating/cooling/hot water systems are generally described using simplified system schematic drawings. A detailed discussion of the various system operating modes is presented in Section 3.4.

2.1 ENVIRONMENTS

A subtask of the SAI systems analysis contract was to establish "standard" locations for performing solar heating and cooling system simulations. Additionally, the attempt to standardize input weather data for simulations has led to the recently developed Typical Meteorological Year (TMY) weather tapes. Thus, for consistency, the locations had to be determined from the 26 TMY cities. An outline of the location selection rationale is presented below, followed by a description of the modified TMY weather tapes.

2.1.1 Selection of Locations

The initial criterion used to determine the locations for simulating heat pump systems was based on grouping the 26 TMY cities by heating and cooling load. The heating load was assumed to be related to the number of heating degree days (12) while the cooling load was assumed to be related to the number of seasonal cooling hours (13). Figure 2-1 presents the SOLMET locations on a heating versus cooling "scatter plot." The cities were grouped into three

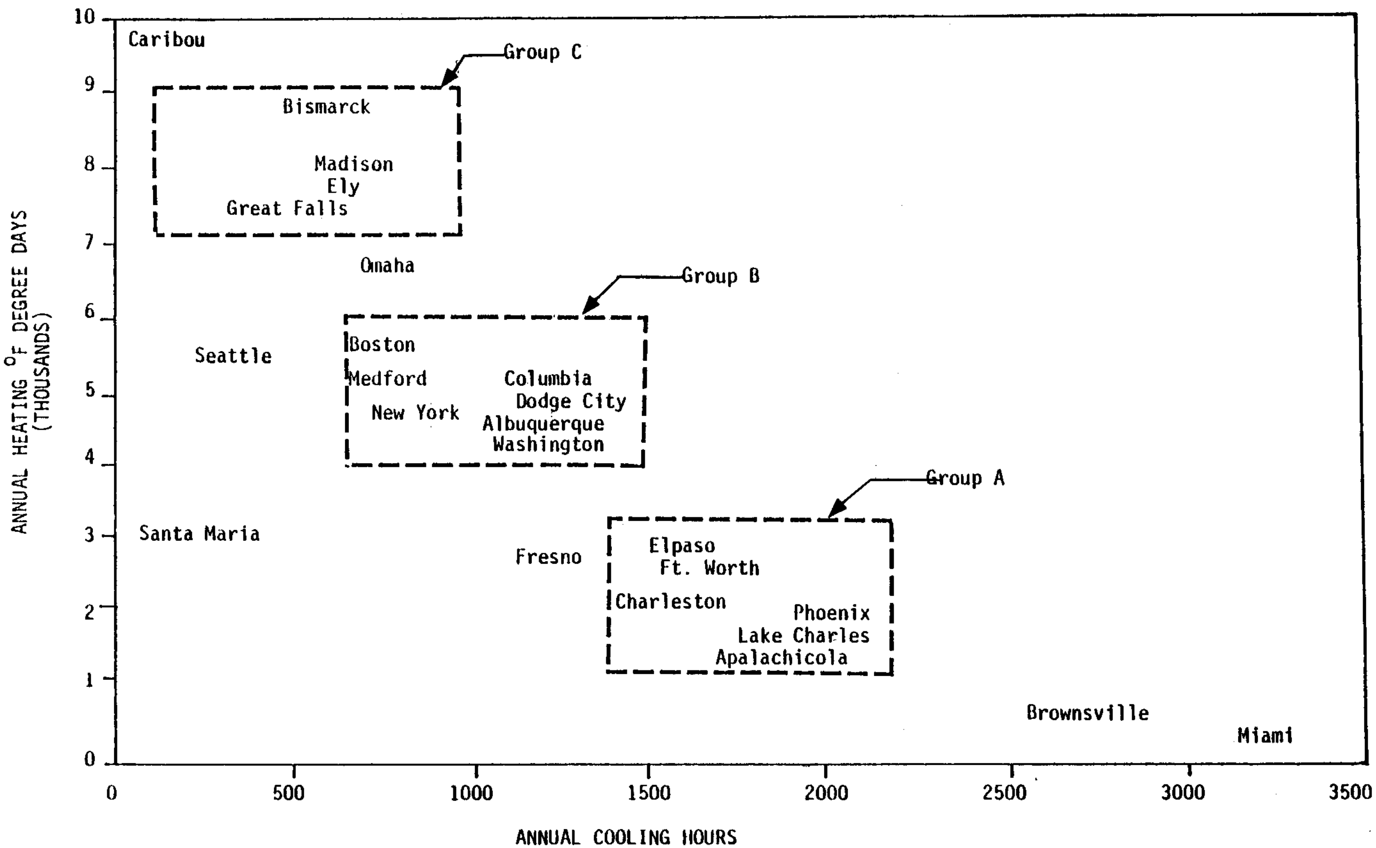


Figure 2-1. Heating Load/Cooling Load Grouping

general load types: Group A (low heating/high cooling), Group B (medium heating/medium cooling) and Group C (high heating/low cooling) as shown in Figure 2-1.

Since the computer simulations were to be for combined solar heat pump (CSHP) systems, the cities of the three groups were then examined for a winter-sunshine heating-load relationship. Group A appeared to split into two parts; 60 percent sun (Apalachicola, Charleston, Fort Worth) and 75 percent sun (El Paso, Phoenix). Lake Charles had only 48 percent sun and did not fit well; thus, it was eliminated from further consideration.

Group B also appeared to divide itself into two parts; 50 percent sun (Boston, Columbia, New York, Washington) and 70 percent sun (Albuquerque, Dodge City). Group C had no distinct sun division as there was a range of from 44 percent (Madison) to 62 percent (Ely). Thus, the grouping and elimination of locations resulted in identification of 15 candidate cities.

After the above three groups were identified, each was examined for its existing and potential market for solar-assisted heat pump systems. The market was assumed to be represented by a large regional population and/or a large number of air conditioner and heat pump sales (14). Each remaining location was examined as a region, with the surrounding states included in its region.

In Group A, Fort Worth and Apalachicola represented the largest potential markets. In Group B cities, New York represented the largest population, with Washington, D.C., representing the highest regional heat pump sales. Madison was selected as the Group C location since its region has the largest population and the greatest heat pump sales.

Final Selections

Table 2-1 lists specific climatic data for the final candidate locations. In Group A, Fort Worth was chosen over Apalachicola because it was felt that Apalachicola represented too mild a winter climate. Fort Worth had a relatively colder, more intense winter, and probably would be more likely to economically utilize CSHP systems.

In Group B, Washington was chosen over New York. Both New York and Washington were very similar in degree days, design temperature, 32°F days per year, and winter relative humidity but the present heat pump sales and the

Table 2-1

Final Location Candidates Climatic Data

	Heating DD (°F)	Cooling Hours	Mean % Sun		Days Per Year With 32°F or Below	Jan. Avg. Relative Humidity		Design Day Temp. (15) (97½%)
			Winter	Summer		7AM	1PM	
GROUP A								
Apalachicola	1308	2000	58	66	4	87	69	30
Fort Worth	2405	1700	57	77	32	78	61	22
GROUP B								
New York	4871	850	52	65	77	72	61	15
Washington*	4224	1080	48	64	73	73	56	17
GROUP C								
Madison*	7863	640	44	67	143	80	71	-7
SPECIAL								
Albuquerque **	4348	1120	71	78	107	70	49	16
Dodge City	4986	1070	67	77	127	79	56	5

*Chosen Locations for Simulations

**Special "Solar" Location

larger use of electrical heating, along with a high residential construction rate, biased the choice to Washington.

At this point, it was decided that a special "solar" location might be appropriate. The three chosen locations all had from 44 to 57 percent winter sun, which is not particularly favorable for solar systems. Albuquerque and Dodge City were reexamined (Table 2-1) and Albuquerque was chosen on the basis of lighter load, higher sun percentage, and lower relative humidity (reducing defrost conditions for the heat pump).

Thus, the final recommended heat pump simulation locations were:

- Albuquerque, NM (special "solar" conditions)
- Fort Worth, TX
- Madison, WI
- Washington, D.C.

2.1.2 Meteorological Specifications

One of the obstacles in the solar heating and cooling systems studies area has been the lack of a commonly accepted year of hourly meteorological data for use in computer simulations. Consequently, it has been nearly impossible to compare the performance of different component and system designs as simulated by different investigators. As a result, the U.S. Department of Energy (DOE) contracted with the National Oceanic and Atmospheric Administration (NOAA) of the U.S. Department of Commerce to combine available hourly solar radiation data with hourly surface meteorological observations and create a new common FORTRAN-compatible tape format with quality controlled data. This tape format is called SOLMET and was produced for 27 stations in the United States covering approximately the period of 1953-1975 (16).

Since the simulations of solar systems typically cover a year or less, it was found desirable to identify a smaller data base than the SOLMET data. SANDIA Laboratories was contracted by DOE to develop a TMY for each (less Stephenville, TX) of the stations using statistical techniques. The 26 TMY tapes generated by Sandia Laboratories are now available through the National Climatic Center in Ashville, North Carolina, for the 26 locations shown in Figure 2-2. Table 2-2 shows a typical portion of the type of data contained on this TMY tape; the data are in the format described in Table 2-3. By choosing a series of typical months, one year was constructed which is representative of

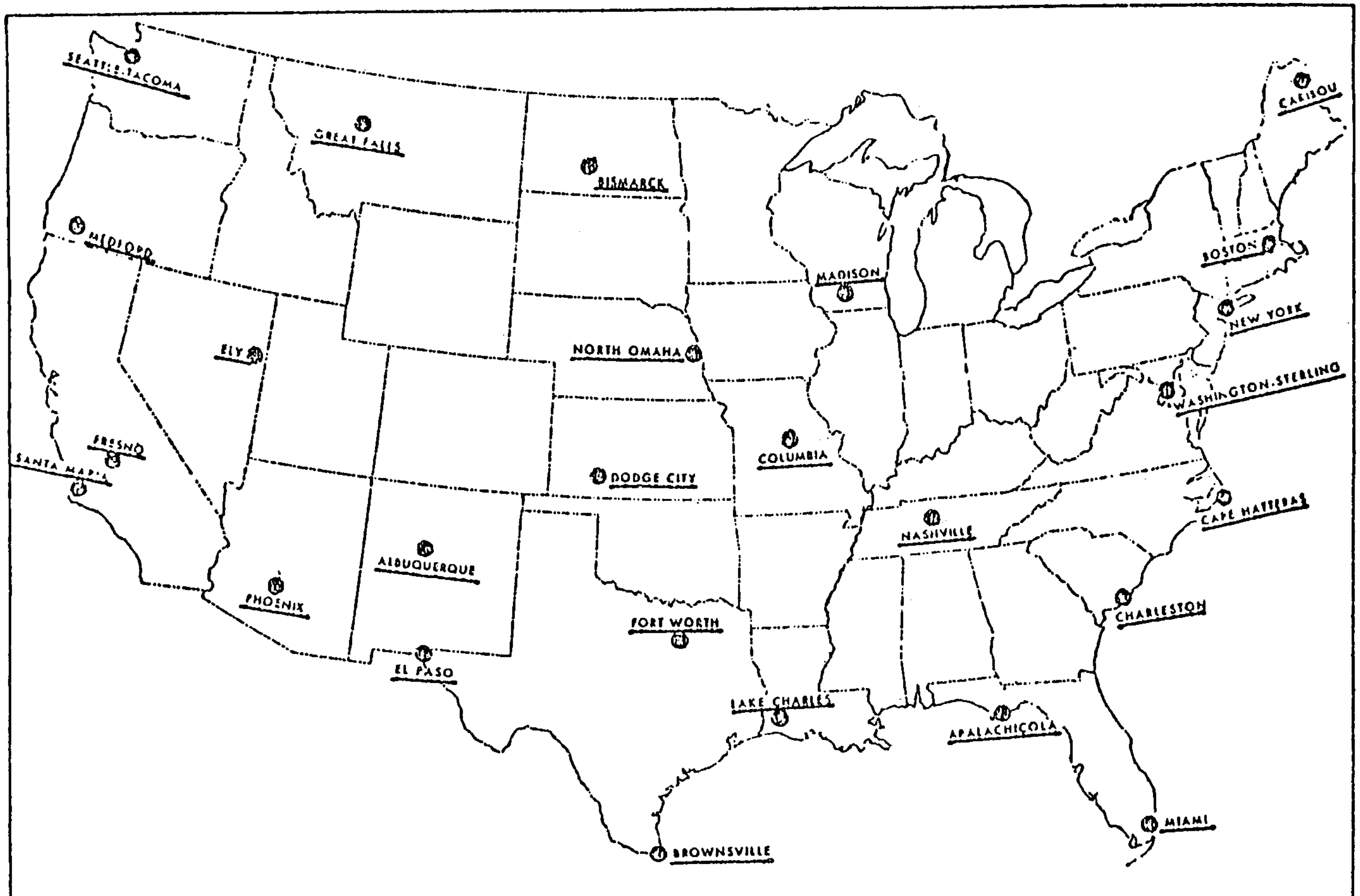


Figure 2-2. Solmet Typical Meteorological Year (TMY) Locations

Figure 2-2. Solmet Typical Meteorological Year (TMY) Locations

Table 2-2

Example of TMY Data Format

```
392772 1 1 100 132999999999999999999999999999999999999999999999999999999 199999999999999999999999910178 9970   89  83320   3199999
392772 1 1 200 232999999999999999999999999999999999999999999999999999999 299999999999999999999999910178 9970   89  83320   3199999
392772 1 1 300 332999999999999999999999999999999999999999999999999999999 3   46 4600  113 100000010178 9970   89  83320   3110109
392772 1 1 400 432999999999999999999999999999999999999999999999999999999 499999999999999999999999910171 9963   89  85333   2199999
392772 1 1 500 532999999999999999999999999999999999999999999999999999999 599999999999999999999999910164 9956   89  87347   1099999
392772 1 1 600 632999999999999999999999999999999999999999999999999999999 6   12 6000   48  100010101 57 9949   89  89   0    010109
392772 1 1 700 732      099999999999999999999999999999999800009999999999999 799999999999999999999999910157 9949   91  91  53    999990
392772 1 1 800 832 4237015399999999999999999999999999995009199999999999999 899999999999999999999999910156 9949   92  92107   1799990
392772 1 1 900 932131670057999999999999999999999999999202980999999999999999 9    9 6000   97         1010156 9949   94  94160   2610100
392772 1 11000103220457007299999999999999999999999999504979999999999999991099999999999999999999999910152 9945   98  96167   2499990
392772 1 11100113225617008999999999999999999999999999506499999999999999991199999999999999999999999910147 9940  102  98173   2399990
392772 1 112001232282870107999999999999999999999999992072999999999999999912   18 6000  129         010143 9936  106 100180   2110100
392772 1 11300133224287001299999999999999   326   318   321   3189999999139999999999999999999999999910133 9927  108 102187   2399990
392772 1 11400143225617000499999999999999   197   193   195   1939999999149999999999999999999999999910124 9918  109 104193   2499990
392772 1 115001532204570015999999999999999999999999999205009999999999999915   24 6000  129         010114 9909  111 106200   2610100
392772 1 116001632131670004999999999999999999999999999502229999999999999916999999999999999999999999910116 9911  113 106253   1799990
392772 1 11700173242370002999999999999999    42    41    41    4199999991799999999999999999999999999910118 9913  115 106307    999990
392772 1 11800183 2      0499999999999999999    0     0     0     099999991 8 213 4000  129         010120 9915  117 106   0    0 9 90
392772 1 119001932999999999999999999999999999999999999999999999999999999991999999999999999999999999910125 9920  111 102337    799999
392772 1 120002032999999999999999999999999999999999999999999999999999999992099999999999999999999999910129 9924  106  98313   1499999
392772 1 1210021329999999999999999999999999999999999999999999999999999999921   88 6000   80        1010134 9929  100  94290   2110109
392772 1 122002232999999999999999999999999999999999999999999999999999999992299999999999999999999999910140 9933   94  83313   3399999
392772 1 123002332999999999999999999999999999999999999999999999999999999992399999999999999999999999910145 9938   89  72337   4599999
392772 1 12400   32999999999999999999999999999999999999999999999999999999 07777       0 402        010151 9942   83  61   0   57 0 09
392772 1 2 100 132999999999999999999999999999999999999999999999999999999 199999999999999999999999910156 9947   70  54343   5299999
392772 1 2 200 232999999999999999999999999999999999999999999999999999999 299999999999999999999999910160 9951   57  46327   4699999
392772 1 2 300 332999999999999999999999999999999999999999999999999999999 37777       0 322        010165 9956   44  39310   41 0 09
392772 1 2 400 432999999999999999999999999999999999999999999999999999999 499999999999999999999999910164 9955   40  33303   3899999
392772 1 2 500 532999999999999999999999999999999999999999999999999999999 599999999999999999999999910162 9954   37  28297   3499999
392772 1 2 600 632999999999999999999999999999999999999999999999999999999 67777       0 241        010161 9953   33  22290   31 0 09
```

Table 2-3

TMY Format (Typical Meteorological Year) Description

TAPE FIELD #	POSITION	ELEMENT *
001	001-005	WBAN Station Number
002	006-015	Solar Time (Yr., Mo., Day, Hour, Minute)
003	016-019	Local Standard Time (Hr., Minutes)
101	020-023	Extraterrestrial Radiation
102	024-028	Direct Radiation
103	029-033	Diffuse Radiation
104	034-038	Net Radiation
105	039-043	Tilt Radiation
106	044-048	Observed Radiation
107	049-053	Engineering Corrected Radiation
108	054-058	Standard Year Corrected Radiation
109,110	059-068	Additional Radiation (A and B)
111	069-070	Minutes of Sunshine
201	071-072	Time of Surface Observation
202	073-076	Ceiling Height
203	077-081	Sky Condition
204	082-085	Visibility
205	086-093	Weather
206	094-103	Pressure
207	104-111	Temperature
208	112-118	Wind (Direction and Speed)
209	119-122	Cloud (Total Amount, Total Opaque)
210	123	Snow Cover Indicator
211	124-132	Blank

*Description of elements as contained in SOLMET User's Manual, Vol. 1.[16]

A logical record length is 132 bytes with physical records 3168 bytes (blocked 24).

long-term weather patterns at each of the 26 locations. Therefore, long-term average system performance can presumably be predicted with one year-long simulation.

Under the general systems contract, SAI identified two subtasks which were deemed necessary concerning the existing TMY tapes:

- Further processing to make the TMY data truly convenient and compatible for simulation purposes
- Assessment of how well the TMYs represent data in regard to thermal performance of solar heating and cooling systems.

Each of these items is addressed separately below.

<u>SAI - Modified TMY Tapes</u>

In order to make the TMY tapes convenient and compatible to solar simulation programs, SAI has rewritten these tapes in the format of Table 2-4. Table 2-5 contains an interval of rewritten SAI TMY data corresponding to the original TMY segment in Table 2-2. The improvements which were incorporated are listed below:

- Changed nighttime solar radiation fields from 9999 to 0.
- Calculated wet bulb temperature, absolute humidity, and relative humidity from dry bulb and dew point using modified ASHRAE relations (17). All five psychrometric indices were then included on the tape.
- Combined data for all 26 locations onto one tape, rather than the current one tape per location.
- Discontinuities between months in continuous fields on the original TMY tapes were smoothed with a cubic spline technique. Dry bulb and dew point temperatures were smoothed independently resulting in occasional instances where dew point exceeded dry bulb. These have been corrected on the SAI TMY tape.
- TMY months chosen from SOLMET years after 1964 originally had meteorological data every third hour rather than hourly. The pressure (sea level and station), temperature (dp and db), and wind (direction and speed) fields on the TMY tapes were linearly interpolated to yield continuous fields. On the rewritten SAI tape, visibility and total opaque sky cover were also interpolated so that all continuous data fields are serially complete. The snow cover indicator was always set to the last good data value so that the field contains either 0 or 1.

Table 2-4

Format and Content of the SAI TMY Data

RECORD NO. 1

Column	Abbrev.	Description	Units
1-2	IfILE	File Number	--
6-10	WBAN	Weather Bureau Station No.	--
11-13	LATDEG	Latitude - degrees	Degrees
14-15	LATMIN	Latitude - minutes	Minutes
16-18	LONDEG	Longitude - degrees	Degrees
19-20	LONMIN	Longitude - minutes	Minutes
49-52	N	Number of records	--

RECORD NO. 2 - 8761

Item	Column	Abbrev.	Description	Conv. Factor	Units
1	1-2	IFILE	File Number	--	--
2	3-4	MO	Month (1-12)	--	--
3	5-6	DY	Day (1-31)	--	--
4	7-8	HR	Hour (1-24) (solar time)	--	--
5	9-13	SOLD*	Direct normal solar radiation (SOLMET 102)	--	KJ/m^2
6	14-13	SOLG*	Global solar radiation on a horizontal surface (SOLMET 108)	--	KJ/m^2
7	19-22	DBT	Dry bulb temperature	x10	^oK
8	23-26	WBT**	Wet bulb temperature	$x10^{-1}$	^oK
9	27-30	DPT	Dew point temperature	$x10^{-1}$	^oK
10	31-34	W**	Humidity ratio	$x10^{-6}$	--
11	35-36	RH**	Relative humidity	--	%
12	37-39	WS	Wind speed	$x10^{-1}$	m/s
13	40-41	WO***	Wind direction	x10	Degrees
14	42-46	STPR	Barometric pressure	$x10^{12}$	Kilopascals
15	47	ISNOW	Snow Index	--	--
16	48-50	VIS	Visibility	$x10^{-1}$	km
17	51-52	SO	Total opaque sky cover	--	tenths

* Radiation integrated over the previous hour.
** Items 8, 10, and 11 are calculated. All other came from TMY.
*** 0 is wind from the north, 9 from east, 13 from south, etc.

Table 2-5

Example of SAI TMY Data Format
(Corresponding to the Original Shown in Figure 2-2)

```
1     3927 3250 9707                                                8760
1 1 1 1     0     02836282128O8 66283 25  8 99690136 6
1 1 1 2     0     0283328212811 67486 20  6 99650118 7
1 1 1 3     0     0283028212813 68689 15  4 99610101 8
1 1 1 4     0     0282728212816 69792 10  3 99570  83 8
1 1 1 5     0     0282428212818 70995  5  1 99530  66 9
1 1 1 6     0     0282128212821 72198  0  0 99490  48 9
1 1 1 7     0     0282328232823 73199  9  5 99490 6410
1 1 1 8   153   9128242824282 4 73699 1710 99490 8010
1 1 1 9    57   298282628262826 74699 2616 99490 9710
1 1 110    72   497283028292828 75698 2416 994501O710
1 1 111    89   649283428322830 76797 2317 9940011810
1 1 112   107   729283828352832 77896 2118 9936012910
1 1 113    12   321284028372834 78996 2318 9927012910
1 1 114     4   195284128382836 80196 2419 9918012910
1 1 115    15   500284328402838 81296 2620 9909012910
1 1 116     4   222284528412838 81295 1725 99110129 9
1 1 117     2    41284728422838 81294  930 99130129 9
1 1 118     0     0284928432838 81292  0 0 99150129 9
1 1 119     0     0284328382834 79094  733 99200112 9
1 1 120     0     0283828332830 76894 1431 99240  96 9
1 1 121     0     0283228292826 74796 2129 99290 8010
1 1 122     0     0282628202815 69392 3331 99330187 6
1 1 123     0     0282128122804 64289 4533 99380294 3
1 1 124     0     0281528042793 59486 57  0 99420402 0
1 1 2 1     0     0280227942786 56689 5234 99470375 0
1 1 2 2     0     0278927842778 53592 4632 99510348 0
1 1 2 3     0     0277627742771 50896 4131 99560322 0
1 1 2 4     0     0277227692765 48795 3830 99550295 0
1 1 2 5     0     0276927652760 47093 3429 99540268 0
1 1 2 6     0     0276527602754 45092 3129 99530241 0
1 1 2 7     0     0276927642758 46392 3130 99550193 0
```

- Only information useful to those studying the performance of heating and cooling systems was retained (see Table 2-4). Note that both horizontal global (SOLMET field 108) and direct normal (SOLMET field 102) data were retained so that empirical relationships are not necessary for estimating the diffuse component of radiation.

- The cubic spline technique described above was only applied at the January through December month-to-month intersections, smoothing over six hours from each month. This smoothing has now been applied to the SAI TMY between the last six hours of December and the first six hours of January so that heating season simulations (e.g., September - April) can be performed without a discontinuity.

In addition to the above changes, SAI recommends that certain algorithms be employed to make the weather data forcing function completely consistent among DOE contractors and other investigators using TMY data. If solar radiation on nonhorizontal surfaces is required, the algorithm outlined in Table 2-6 should be employed (18, 19). The algorithm recommended for determining atmospheric radiation (20) is presented in Table 2-7.

Evaluation of TMY for Solar Heating and Cooling Systems

The purpose of the TMY evaluation subtask was to assess how well the TMYs represent the 23 SOLMET years with regard to the thermal performance of solar heating and cooling systems. Results of this work are being produced as a separate document (22). The discussion below is intended to provide solely an overview of the evaluation study through excerpts from the study's results and conclusions.

Ideally, several different kinds of representative solar heating and cooling systems should have been simulated for both the TMY and for the long-term SOLMET in each of the 26 sites. As a practical alternative, a single solar heating and cooling system was simulated in six test sites chosen to represent a broad range of climate types across the United States (Albuquerque, Fort Worth, Madison, Miami, New York, Washington, D.C.). The base system is an active residential space heating and cooling system consisting of an array of two-cover, selective surface, flat-plate collectors which heat a solution of anti-freeze and water, a fully mixed water-filled storage tank, a liquid-to-liquid heat exchanger to isolate the collector and tank fluids, a water-to-air heat exchanger in the house air supply duct, a typical residential heating and

Table 2-6

Algorithm for Solar Radiation on a Nonhorizontal Surface (18, 19)

CALCULATIONS

CSI	=	A - B + C + D + E
CSH	=	$\cos\phi \cos\delta \cos\omega + \sin\phi \sin\delta$
QDIF	=	QH - QDIR * CSH
QBEAM	=	QDIR * CSI
QSKY	=	QDIF * (1 - cosS)/2
QGRND	=	QH * RHO * (1 - cosS)/2
QSURF	=	QBEAM + QSKY + QGRND

A	=	$\sin\delta \; \mathrm{si}\phi \; \cos S$
B	=	$\sin\delta \; \mathrm{si}\phi \; \cos S \cos\alpha$
C	=	$\cos\delta \cos\phi \cos S \cos\omega$
D	=	$\cos\delta \sin\phi \sin S \cos\alpha \cos\omega$
E	=	$\cos\delta \sin S \sin\alpha \sin\omega$

INPUTS

α	collector aximuth angle (radians, south = 0, east (-), west (+))
S	collector slope from horizontal (radians)
δ	declination of earth (radians, north positive)
ω	hour angle (solar noon = 0, $\omega = \frac{15\pi}{180}$, for 1 PM, etc.) (east (-), west (+))
ϕ	latitude (radians, north positive)
QH	total radiation on a horizontal surface (KJ/hr-m^2, from tape)
QDIR	direct (BEAM) normal radiation (KJ/hr-m^2, from tape)
RHO	ground reflectance

NOMENCLATURE

CSI	cosine of the incidence angle
CSH	cosine of the zenith angle
QDIF	diffuse radiation on a horizontal surface
QBEAM	direct radiation on the surface
QSKY	diffuse radiation on the surface
QGRND	radiation reflected onto surface from ground
QSURF	total radiation incident on surface (KJ/hr-m2)

Table 2-7

Algorithm for Atmospheric Radiation (20)

CALCULATIONS

ESKY = .7871 + .7636 * LN(TDP/273)

CA = 1 + .0224 * CC - .0035 * (CC) ** 2

QSKY = SIG * CA * ESKY * (TDB) ** 4

TEFF = (QSKY/SIG) ** .25

DEFINITIONS

SIG	=	$5.67*(10)^{-8}$ W/m^2-K^4 (Stefan-Boltzman constant)
ESKY	=	effective emissivity of the sky
TDB	=	local ambient dry bulb temperature (^oK, from weather tape)
TDP	=	local ambient dew point temperature (^oK, from weather tape)
CC	=	total opaque sky cover (tenths, 1,2,...,10, from weather tapes)
CA	=	cloud correction factor for atmospheric radiation
QSKY	=	atmospheric radiation (W/m^2)
TEFF	=	effective sky temperature (^oK, see Reference (21) for usage)

cooling load, a hot water fired lithium-bromide absorption chiller with a cooling tower for space cooling, auxiliary heat supplies for both heating and cooling, and associated piping, pumps, and controls. This base system has been sized to meet between 1/2 to 3/4 of the combined heating and cooling load in each of the test sites.

The differences between the long-term and the TMY heating performance predictions were found to be strongly correlated to differences in the gross monthly climate statistics, especially in the high-load months. The TMY selection methodology probably yielded as near "typical" a year as possible using twelve concatenated real months. The basic shortcoming of the method was that there is too small a population of months from which to pick a typical one. Typicalness in important statistics is frequently sacrificed for typicalness in others, and, often, no month in the period of record adequately represents the long-term in more than one or two statistics. Table 2-8 illustrates the impossibility of attaining a "completely" typical year by comparing the standard deviations of yearly performance factors to the long-term means. Due to the near random nature of the month-to-month atypicalnesses, the yearly and seasonal results were acceptably close to the long-term for most practical purposes. Table 2-9 is a summary of selected performance factors for both TMY data and the interim standard "Hedstrom Years" versus long-term SOLMET data.

Several observations have been made concerning typicalness of the months in the TMYs. The TMY months are consistantly among the five most representative from the 23 or so available, in terms of the insolation and temperature. This is enough to ensure that the solar system performance measures are also typical in months of high load. However, in months of low load, especially low cooling load, the performance of the solar system in the TMY month is likely to be considerably different from the long term. The reason is that performance in months of low load is sensitive to persistance and covariance of meteorological data not adequately considered in the TMY selection process. In fact, the method has a built-in bias to select low-load months which will over-predict the long-term solar contribution both in the heating and the cooling season. Fortunately, these errors are usually not critical because they contribute little to the yearly or seasonal performance measures. However, for systems that are over-sized for a given climate and load, the seasonal TMY predictions can become significantly larger than the total long-term predictions.

Table 2-8

Standard Deviation of Yearly Performance Measures as Percent of Long-Term Means

	ALBUQUERQUE	FORT WORTH	MADISON	MIAMI	NEW YORK	WASHINGTON
HDIF	2.5	2.3	2.3	2.7	2.7	1.3
HHOR	2.4	4.5	4.0	2.8	4.2	3.7
HTILT	2.6	4.8	4.4	3.0	4.4	4.1
QU	6.0	7.3	5.2	7.1	5.4	5.4
QAUXH	18.5	35.8	9.1	-	15.2	17.3
QLH	8.3	10.5	5.1	60.5	7.2	9.3
QAC	25.1	11.3	33.9	8.2	26.3	30.6
QLAT	36.3	8.2	32.4	7.8	27.3	29.4
QLC	22.5	10.2	32.9	7.9	24.9	29.8
QAUXC	35.4	10.1	84.9	9.7	56.3	63.7
FDIF	4.0	6.0	4.2	5.1	4.1	3.8
RBAR	0.7	0.9	0.8	0.6	2.0	1.0
FCOL	5.1	4.6	4.4	5.7	5.0	4.9
F-HTG	6.9	10.2	6.6	-	6.9	6.0
F-CLG	8.5	7.0	5.8	6.6	10.2	9.6
COP	3.1	1.5	1.3	0.6	2.0	1.1

Measure	Definition	Units
HDIF	Total diffuse radiation on a horizontal surface	GJ/m^2
HHOR	Total radiation on a horizontal surface	GJ/m^2
HTILT	Total radiation on the collector surface	GJ/m^2
QU	Total energy gained per unit collector area	GJ/m^2
QAUXH	Space heating auxiliary energy	GJ
QLH	Total space heating requirement	GJ
QAC	Total heat removed from room by chiller	GJ
QLAT	Latent heat removed from room by chiller	GJ
QLC	Total heat input to chiller's generator	GJ
QAUXC	Auxiliary heat input to chiller's generator	GJ
TBAR	Average tank temperature	^oC
FDIF	Fraction of horizontal radiation that is diffuse (HDIF/HHOR)	-
RBAR	Ratio or radiation on collector to horizontal (HTILT/HHOR)	-
FCOL	Long-term collection efficiency (QU/HTILT)	-
F-HTG	Fraction of heating load met by solar [(QLH-QAUXH)/QLH]	-
F-CLG	Fraction of cooling load met by solar [(QLC-QAUXC)/QLC]	-
COP	Chiller Coefficient of Performance (QAC/QLC)	-

Table 2-9

Comparison of Selected Annual Performance Measures for the Long Term, the TMY, and the "Hedstrom Years"

SITE		HHOR(GJ/m^2)	QLH(GJ)	QAC(GJ)	QLAT(GJ)	FDIF	F-HTG	F-CLG
ALBUQUERQUE	LONG TERM	7.56	56.1	22.3	8.0	0.253	0.624	0.624
	TMY	7.63	56.3	19.9	6.7	0.256	0.644	0.642
	HEDSTROM YR.	7.72	54.9	20.5	7.5	0.241	0.621	0.663
FORT WORTH	LONG TERM	6.10	32.2	54.5	21.4	0.365	0.724	0.470
	TMY	6.16	31.4	49.5	19.0	0.350	0.764	0.495
	HEDSTROM YR.	6.17	37.8	57.1	22.7	0.354	0.569	0.437
MADISON	LONG TERM	4.93	107.7	9.3	3.9	0.425	0.488	0.894
	TMY	4.93	107.9	8.0	3.3	0.435	0.486	0.958
	HEDSTROM YR.	5.03	108.1	9.6	4.4	0.410	0.476	0.918
MIAMI	LONG TERM	6.10	1.3	90.3	43.3	0.452	1.0	0.393
	TMY	6.14	1.4	90.1	43.0	0.449	1.0	0.409
	HEDSTROM YR.	5.97	1.3	90.9	44.9	0.462	1.0	0.395
NEW YORK	LONG TERM	4.55	72.9	13.9	6.0	0.469	0.590	0.775
	TMY	4.53	70.5	11.2	4.9	0.466	0.571	0.800
	HEDSTROM YR.	4.39	78.7	11.7	5.1	0.472	0.563	0.826
WASHINGTON DC	LONG TERM	4.99	68.2	20.4	8.9	0.444	0.646	0.782
	TMY	5.04	67.9	18.2	8.2	0.437	0.657	0.790

Cooling system performance, and, for that matter, most high temperature solar applications, are more sensitive to weather data structure and, hence, more likely to be over-predicted by the TMY. Smaller storage sizes further increase a solar system's sensitivity to weather data structure but no serious seasonal performance differences between the TMY and the long term arise as a result.

The long-term monthly diffuse radiation was found to be poorly represented by the TMY months due to lack of consideration in the selection process. The high weighting given to the total horizontal radiation did assure typicalness of the TMY horizontal radiation but not the diffuse component. This fact may have more important implications for solar systems which can utilize only beam radiation.

One of the valuable by-products of this study has been the extensive spot-checking of the integrity of the TMY and SOLMET data. It has been found that the long-term average data published on microfiche (23) agree with the values calculated in the TRNSYS simulations. The "surface observations" in these data also agree closely with the long-term data published annually by NOAA in the Local Climatological Data publications (24). Furthermore, the near equivalence of each of the months in each TMY with the respective "source" month in the SOLMET format has been confirmed. The only differences are slight changes in the surface observations to smooth the discontinuities a few hours on either side of each monthly interface. Table 2-10 presents the TMY versus original SOLMET comparison for system performance in Washington, D.C.

The main value of the TMYs, as far as most researchers and funding agencies are concerned, is that they offer standardized hourly meteorological forcing functions for a wide variety of climates, enabling direct comparison of the results from different simulation studies. Since the selection of a simulation test location for a given study is often arbitrary within certain broad climatological constraints, it does not matter if the TMY is perfectly representative of a specific site. It is certain that the Albuquerque TMY is representative of an "arid continental" climate, that the Miami TMY is representative of a "subtropical marine" climate, etc. That is sufficient assurance for most needs. The existing TMYs are adequate for developing design and sizing procedures since the ultimate input to the procedure is climate statistics, not the site location. It is difficult to conceive of any real need for a more "accurate" typical meterological year.

Table 2-10

Performance of TMY Months in Context of TMY and in Context of Original Data for Washington

	TMY CONTEXT					ORIGINAL CONTEXT				
	HHOR	QLH	QAC	F-HTG	F-CLG	HHOR	QLH	QAC	F-HTG	F-CLG
JAN	0.2141	15.530	0	0.461	1.0	0.2136	15.620	0	0.475	1.0
FEB	0.2585	13.340	0	0.553	1.0	0.2579	13.280	0	0.564	1.0
MAR	0.4154	9.921	0	0.908	1.0	0.4147	9.967	0	0.877	1.0
APR	0.5048	4.395	0.10	1.0	1.0	0.5042	4.321	0.11	1.0	1.0
MAY	0.6021	1.163	0.68	1.0	0.954	0.6014	1.058	0.71	1.0	0.942
JUN	0.6442	0	2.42	1.0	0.889	0.6434	0.029	2.40	1.0	0.885
JUL	0.6041	0	6.73	1.0	0.729	0.6033	0	6.73	1.0	0.737
AUG	0.5984	0	5.83	1.0	0.801	0.5977	0	5.80	1.0	0.796
SEP	0.4470	0.113	2.41	1.0	0.778	0.4464	0.114	2.50	1.0	0.755
OCT	0.3458	2.967	0	1.0	1.0	0.3452	2.939	0	1.0	1.0
NOV	0.2303	7.850	0	0.801	1.0	0.2298	7.794	0	0.798	1.0
DEC	0.1722	12.620	0	0.487	1.0	0.1717	12.640	0	0.481	1.0
YEAR	5.037	67.90	18.18	0.657	0.790	5.029	67.76	18.25	0.655	0.787

2.2 LOAD DESCRIPTIONS

The heating and cooling loads were fixed by specifying a set of building thermal characteristics. The selection of building characteristics is presented below for the "typical" housing types for each location while the detailed load model equations are presented later in subsection 3.2. The conventional equipment sizing and hot water loads are also discussed in this section.

2.2.1 Selection of Houses

A "typical" single-family residence for each of the three locations was selected for analysis in this study. The standards set forth for the regional housing types and their building characteristics were established primarily from information supplied by the U.S. Department of Commerce, Bureau of the Census (25). The only city not located well within one of the four Bureau of the Census sections is Washington, D.C., which is located at the northeastern tip of the South. However, it was felt that the house types found in the Washington, D.C., area are more typical of what is common in the Northeast section. Accordingly, the house characteristics of the Northeast section were used to define a typical house in Washington, D.C.

The gross information available from the Bureau of Census was refined to a more local (SMSA or state) level through communications with the National Association of Home Builders (26). This led to a definition of a "typical" residence for each of the three locations that was utilized in this study. The "typical" house in each location was defined by a set of building characteristics wherein each characteristic is the dominate one for that locale. The building characteristics are generally described in Table 2-11 for the type of house chosen to represent each location. The detailed housing descriptions are presented in Appendix A.

Thermal insulation characteristics were obtained by utilizing ASHRAE 90-75 and reference (26). The values developed were based on an economic analysis which states that a payback period of seven years will result if these insulating guidelines are followed. This procedure specifies houses that are well designed and exceed the minimum guidelines of ASHRAE 90-75. For the purposes of this study, it was assumed that the major axis of the house is on an

Table 2-11

General Description of Housing Types Chosen for Each Location

Characteristics	Washington, D.C./ N.Y. City Corridor	Madison, Wisconsin Area	Fort Worth, Texas Area
House Type:			
House style	Colonial	Rambler	Rancher
Type of construction	Wood frame	Wood frame	Frame, brick veneer
Foundation	Basement	Basement	Slab
Number of stories	Two	One	One
Heating system	Warm air-oil	Warm air-gas	Warm air-electric
Cooling system	Central A/C	Central A/C	Central A/C
House Size:			
Living area	161 m^2	158 m^2	167 m^2
Outside wall area	162 m^2	109 m^2	112 m^2
Window area	28 m^2	19 m^2	20 m^2
Thermal Characteristics:			
Roof insulation	Batt, 9 in.	Batt, 9 in.	Batt, 6 in.
Wall insulation	Batt, 6 in.	Batt, 6 in.	Batt, 3 1/2 in.
Floor insulation	Batt, 3 1/2 in.	Batt, 3 1/2 in.	N/A
Window type	Double pane	Triple pane	Double pane

east-west orientation. The structure is thus assured to lend itself to active solar systems. No assumptions were made as to whether the front or rear of the house has the desired southern exposure.

Equipment Sizing and Design Loads

The simulation and cost analysis aspects of this study require that the various types of heating and cooling equipment be sized. This has been accomplished for these three residences by applying the above sets of characteristics to standard ASHRAE procedures (15). The design heating and cooling loads for each residence are presented in Table 2-12. The required capacities for the cooling equipment turned out to be determined by an equipment limitation on the amount of area cooled per unit capacity rather than by actual design load. The resulting equipment sizes are presented in Table 2-13.

2.2.2 Domestic Hot Water Profile

The domestic hot water daily use profile was based on a four-person consumption of 300 liters (80 gallons) per day. The hourly consumption was generated by using the Rand (27) profile, which distributes the hourly hot water consumption as shown in Table 2-14.

Since the hot water load is directly dependent on the temperature rise of the water from inlet to delivery, the monthly average water supply temperatures were used in each location. The supply water temperatures used in this study are presented in Table 2-15.

2.3 SYSTEMS DESCRIPTIONS

The rationale for the selection of the systems to be analyzed is presented below. Additionally, a brief description of each system is presented with a schematic of the major components comprising that system. The details concerning systems operating modes are presented later in subsection 3.4, with the pertinent energy flows being documented in Section 5.

2.3.1 Selection of System Types

The range of system types which must be analyzed was fixed by the purpose of this study, i.e., the establishment of the economic framework in

Table 2-12

Design Heating and Cooling Loads (Simplified Method)

Location	Design Cooling Load (BTU/hr)	Design Heating Load (BTU/hr)
Madison, Wisconsin	18,309	35,460
Washington, D.C.	21,836	32,250
Ft. Worth, Texas	21,772	23,210

Table 2-13

Conventional Equipment Sizes Chosen

Equipment (in 10^3 KJ/HR)	Location		
	Wash. D.C./N.Y. City Corridor	Madison Wisconsin Area	Ft. Worth Texas Area
Electric Furnace (E = 0.90)	48	50	36
Gas Furnace (E = 0.60)	72	75	56
Oil Furnace (E = 0.50)	87	90	67
Vapor Compression A/C	32	32	33
Heat Pump	32	32	33

Note: 1) A heat pump is generally sized for cooling.
2) E refers to system efficiency in delivering conditioned air to the space.

Table 2-14

Hourly Profile of Domestic Hot Water Consumption

Time	Consumption (Liters)	Time	Consumption (Liters)
12-1 a.m.	6.4	12-1 p.m.	10.8
1-2 a.m.	0	1-2 p.m.	15.2
2-3 a.m.	0	2-3 p.m.	8.0
3-4 a.m.	0	3-4 p.m.	7.2
4-5 a.m.	0	4-5 p.m.	6.4
5-6 a.m.	0	5-6 p.m.	11.2
6-7 a.m.	4.4	6-7 p.m.	20.4
7-8 a.m.	14.0	7-8 p.m.	34.8
8-9 a.m.	21.6	8-9 p.m.	28.8
9-10 a.m.	25.6	9-10 p.m.	20.8
10-11 a.m.	20.8	10-11 p.m.	16.4
11-12 a.m.	13.6	11-12 p.m.	13.6

Table 2-15

Monthly Average Water Supply Temperatures

	J	F	M	A	M	J	J	A	S	O	N	D
Fort Worth, TX (28)												
°F	42	49	58	65	73	80	82	83	78	63	53	49
°C	6	9	14	18	23	27	28	28	26	17	12	9
Washington, D.C. (29)												
°F	42	42	52	56	63	67	67	78	79	68	55	46
°C	6	6	11	13	17	19	19	26	26	20	13	8
Madison, WI (30)												
°F	52	Constant Year-Round										
°C	11	Constant Year-Round										

which residential combined solar heat pump systems will have to compete. Thus, the basic conventional system alternatives for space heating, space cooling, and domestic water heating have been included in this analysis. The conventional systems selected include:

- Conventional electric resistance heating, vapor compression cooling, and electric hot water heating
- Conventional gas heating, vapor compression cooling, and gas hot water heating
- Conventional oil heating, vapor compression cooling, and electric hot water heating
- Conventional heat pump for heating and cooling, electric resistance backup heating, and electric hot water heating.

The combined solar heat pump systems analyzed were the series and parallel configurations. Each configuration employed both liquid and air collector system designs. Additionally, heat pumps typical of both today's technology or of next generation's (future) technology were modeled. Since the dual-source configuration has been shown to have little or no performance advantage over the standard parallel systems (5, 7), it was not studied.

The combined solar heat pump systems selected include:

- A liquid collector, series-connected solar heat pump system for heating and cooling, electric resistance backup heating, and a separate solar domestic hot water heating system with electric backup.
- An air collector, series-connected solar heat pump system for heating and cooling, electric resistance backup heating, and a separate solar domestic hot water heating system with electric backup.
- A liquid-based solar space and water heating system, parallel-connected heat pump for backup space heating and for cooling, and electric resistance backup space and water heating.
- An air-based solar space and water heating system, parallel-connected heat pump for backup space heating and for cooling, and electric resistance backup space and water heating.

2.3.2 Brief Systems Descriptions

In the following paragraphs, each system is described in enough detail to provide a basic understanding of its operation. To avoid lengthy titles, hereafter the systems will be referenced only by the underlined portions of the above descriptions.

Conventional Electric, Oil, and Gas Systems

The conventional systems differ only in the energy source used to meet the space and water heating loads. They are shown schematically in Figure 2-3. Heated air is delivered to the conditioned space in response to a single-stage heating thermostat, with the air being heated in the furnace via electric resistance or the combustion of gas or oil. Likewise, a cooling thermostat controls the delivery of air during the cooling season. All cooling systems utilize a conventional split vapor compression air conditioner. The water heater, whether gas or electric, simply maintains a sufficient volume of water at a given set temperature to meet normal demand.

Conventional Heat Pump

The conventional heat pump system is shown schematically in Figure 2-4. An air-to-air split system heat pump is used with staged auxiliary electric resistance heaters built into the indoor unit. A two-stage heating thermostat controls delivery of heated air to the conditioned space as described later in the controls descriptions (subsection 3.4). A single-stage cooling thermostat causes the heat pump to operate in reverse so as to chill and dehumidify air being delivered to the space. The electric water heater maintains a sufficient volume of water at a given set temperature to meet normal demand.

Liquid Series System

The liquid series system employs a liquid-to-air heat pump which uses the solar storage tank as a source (heating mode) or sink (cooling mode). When used as a source, the solar collector system attempts to replenish the energy extracted by the heat pump. When used as a sink, a fan coil is used to reject to ambient that energy which accumulated in the tank. The system schematic for the liquid series configuration is given in Figure 2-5.

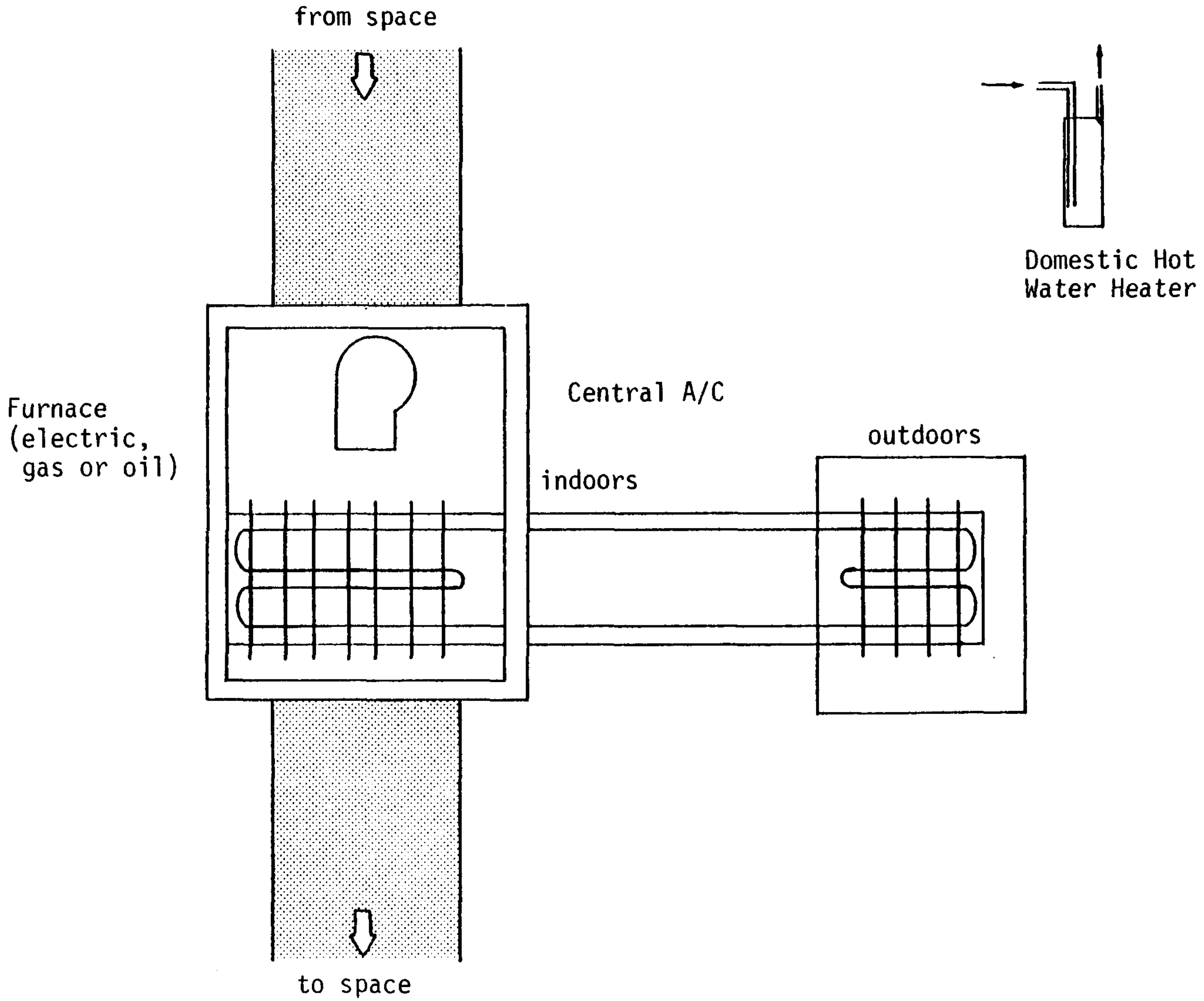


Figure 2-3. Conventional Electric, Gas or Oil System

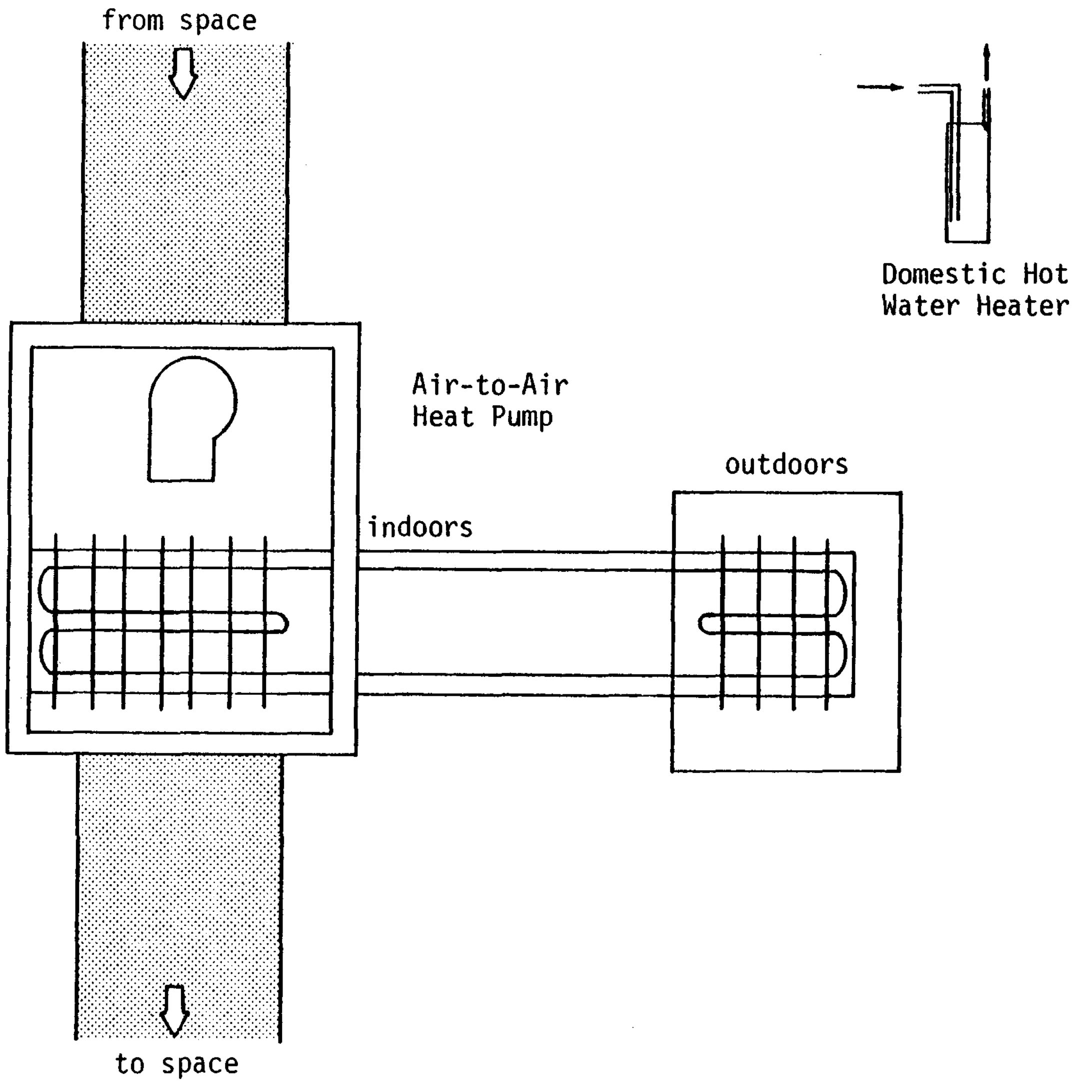


Figure 2-4. Conventional (Stand-Alone) Heat Pump System

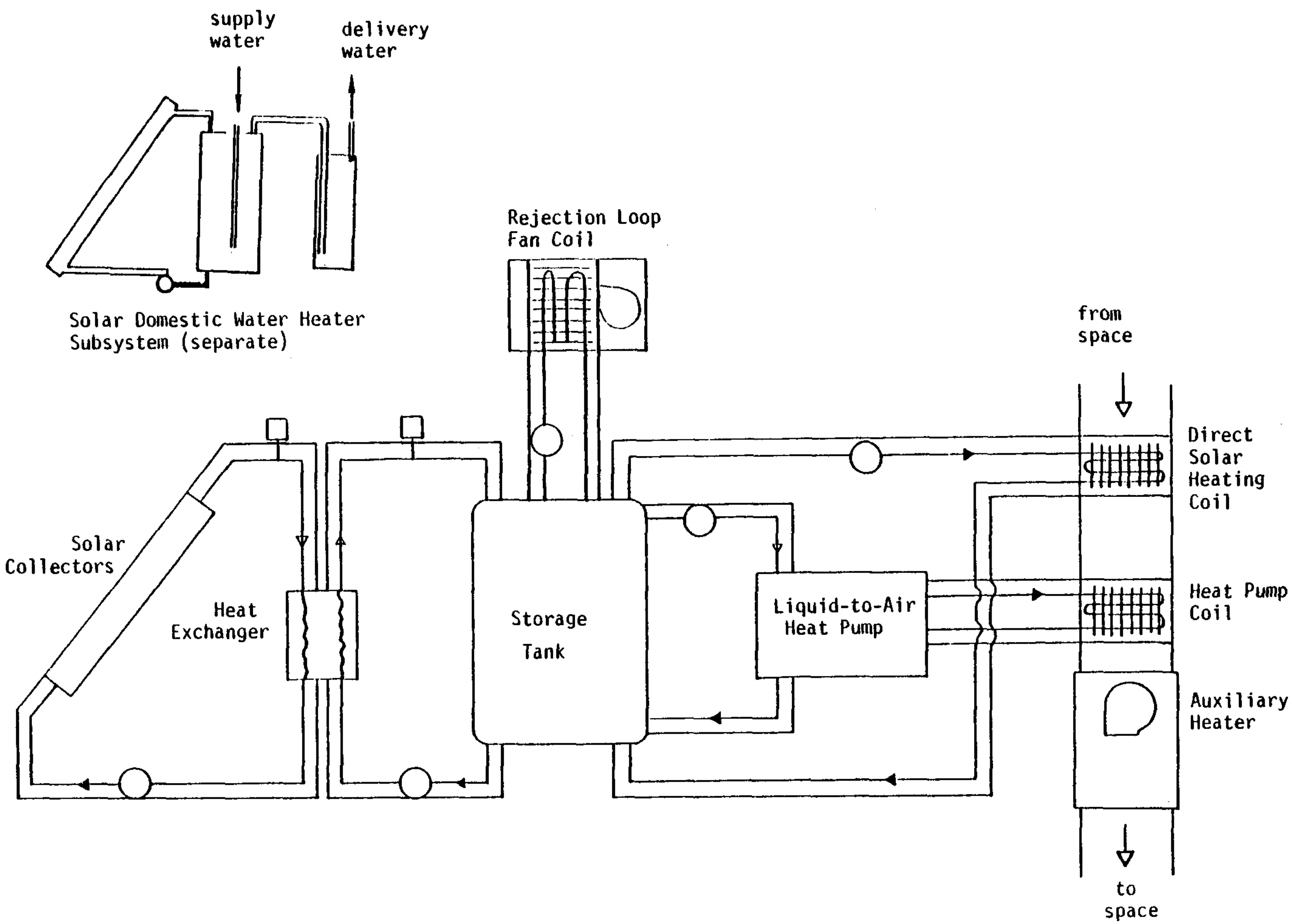


Figure 2-5. Liquid Series Combined Solar Heat Pump System

Solar energy collection can occur simultaneously with either heat pump heating or direct solar heating. However, heat pump and direct solar heating never occur simultaneously since today's residential liquid-to-air heat pumps cannot, in general, operate at a source temperature high enough to allow direct solar heating to occur simultaneously. Liquid-to-air heat pumps currently under development will be able to utilize a source up to approximately 37.8^{o}C (100^{o}F) which approaches the lower temperature limit for direct solar heating. Also, heat pump cooling and rejection to ambient can occur independently or simultaneously.

The domestic water heating system operates completely independently of the space heating and cooling system. The collector operates whenever it can contribute energy to the preheater.

Air Series Systems

The air series system is identical to the liquid series system except that air collectors are used and the fan coil serves to both supply energy to and reject energy from the tank, as illustrated in Figure 2-6. As in the liquid series system, solar energy collection can occur simultaneously with heat pump heating or direct solar heating but heat pump and direct solar heating can never occur simultaneously. Heat pump cooling and heat rejection to ambient can also occur simultaneously. As in the liquid series system, domestic water heating is achieved with a completely independent solar system.

Liquid Parallel System

The liquid parallel system, shown in Figure 2-7, combines a typical liquid-based active solar space and water heating system with a conventional air-to-air heat pump. The two subsystems operate independently, with the solar system being the preferred heat source for space heating. When neither the solar system nor the heat pump can meet the heating load, electric resistance backup is provided.

Solar energy collection can occur simultaneously with heat pump heating, direct solar heating, heat pump cooling, or domestic water heating. The solar energy collection and domestic water heating subsystems operate independently. Heat pump heating and direct solar heating are not allowed to occur simultaneously since air-to-air heat pumps are not designed for the high indoor-unit entering air temperatures which would occur during simultaneous

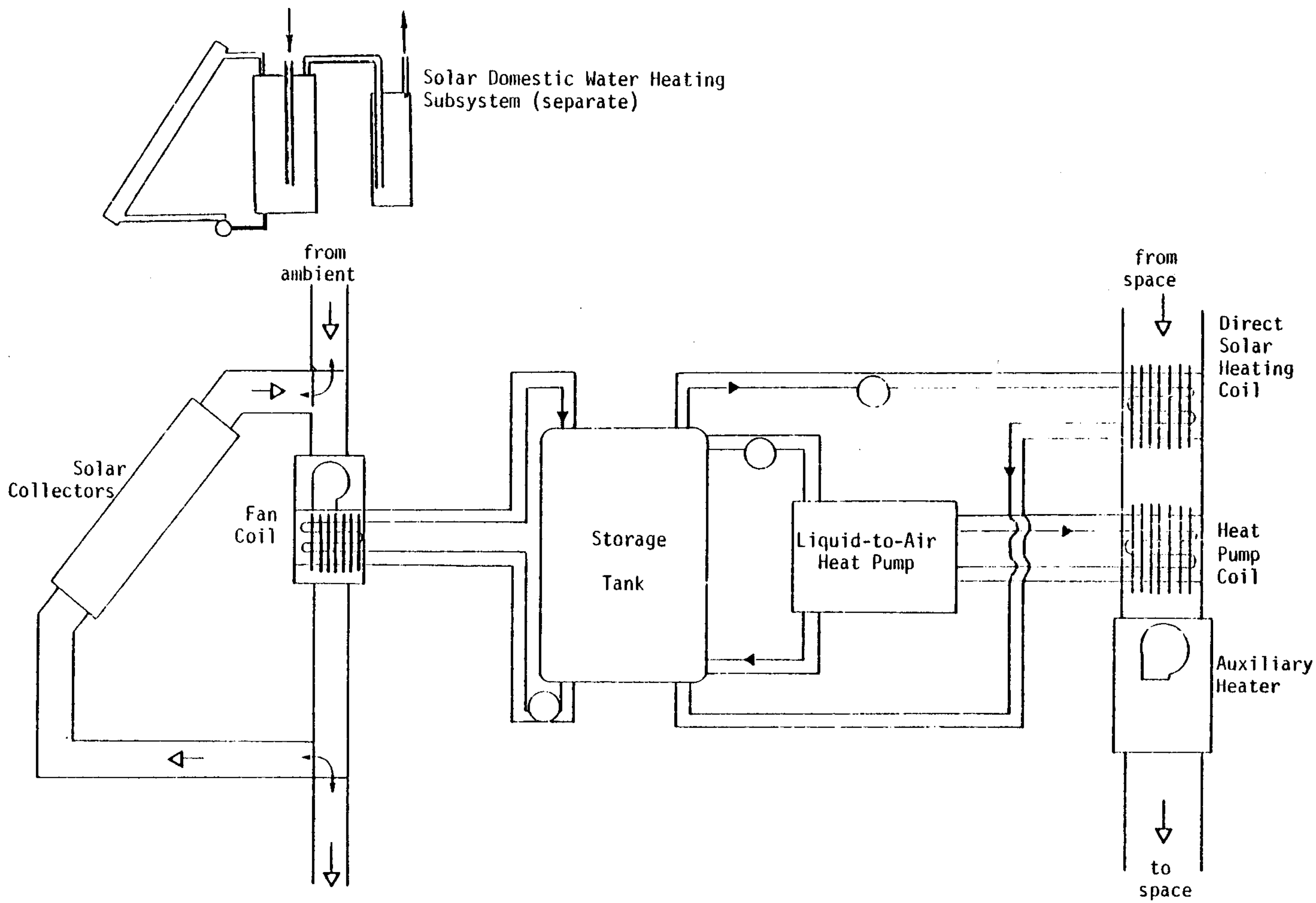


Figure 2-6. Air Series Combined Solar Heat Pump System

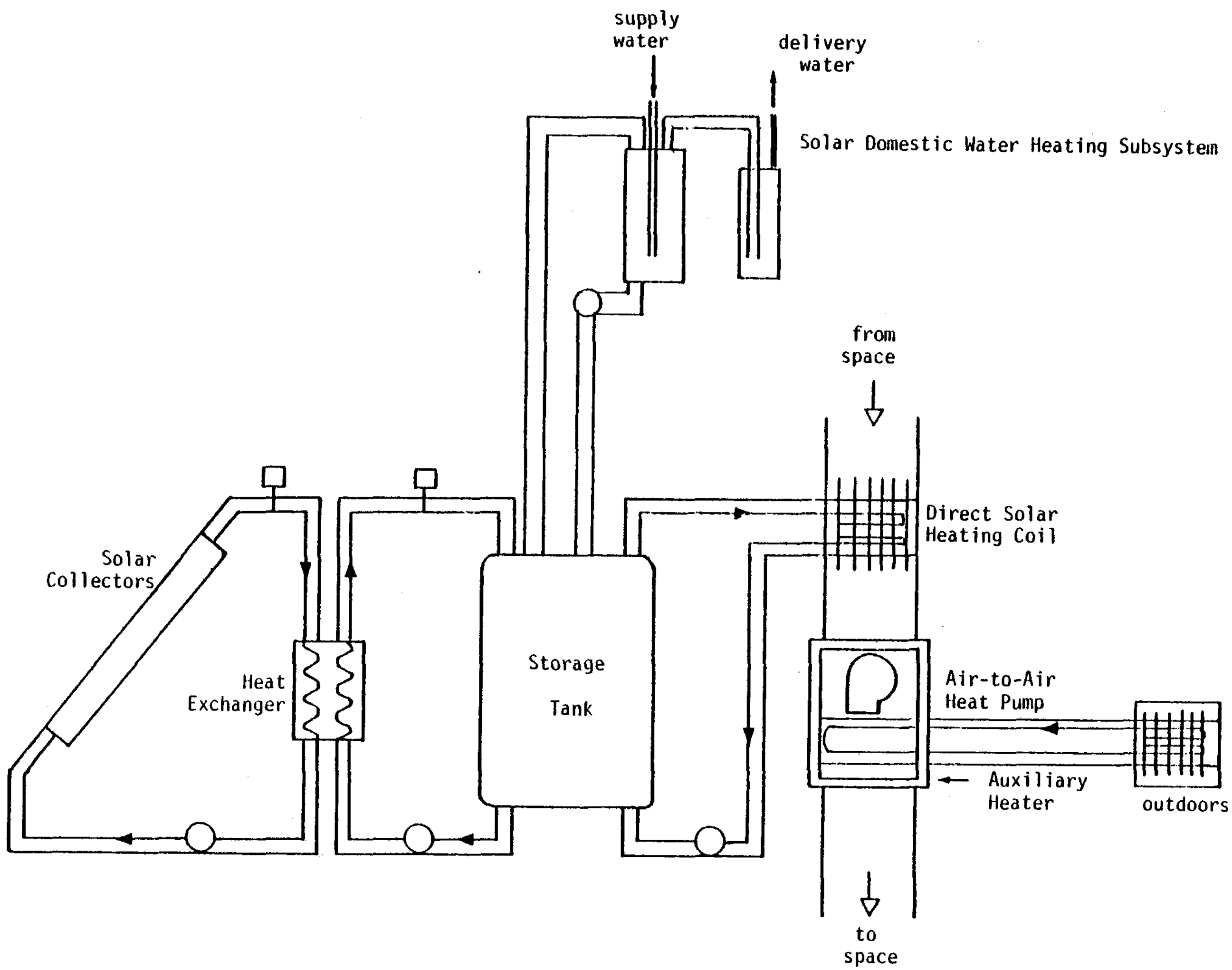


Figure 2-7. Liquid Parallel Combined Solar Heat Pump System

heating mode operation with the direct solar coil upstream of the heat pump in the conditioning air street. Alternately, operating with the heat pump coil upstream of the solar coil would degrade the amount of heat transfer from the solar tank and could even cause reverse heat transfer (into the tank).

In the two series systems described previously, the main collector array is disabled in the summer and energy delivered to the tank by the heat pump is rejected via a fan coil. In the parallel system, the main bank of collectors could be active year round. This allows a larger portion of the year-round domestic water load to be met by solar, but also can result in larger cooling loads if storage losses transfer into the conditioned space.

Air Parallel System

The air parallel system is configured as shown in Figure 2-8. The position of motorized dampers and on/off state of the two blowers control the air flow for each operating mode.

In the heating mode, the collector can deliver energy directly to the load or to storage. In order for wintertime domestic solar water heating to occur, one of these two heating modes must be active. During times when there is no solar energy to collect, the storage bin or the heat pump can deliver energy to the load. As in the liquid parallel system, these heating modes are not allowed simultaneously because, in one configuration (storage bin upstream from indoor unit), the air temperature entering the indoor unit is too high while, in the other configuration (indoor unit upstream from the storage bin), the heat pump would deposit energy in storage.

In summer, the heat pump operates as a central air conditioner in response to the cooling thermostat. In order to continue heating water all summer, the rock bed is bypassed during collector operation. If this were not done, storage would stagnate at a high temperature, reducing the number of hours which the collector would operate and, hence, reduce the number of hours of water preheating. In addition, losses from the storage bin would increase the cooling load if the bin were in thermal contact with the conditioned space.

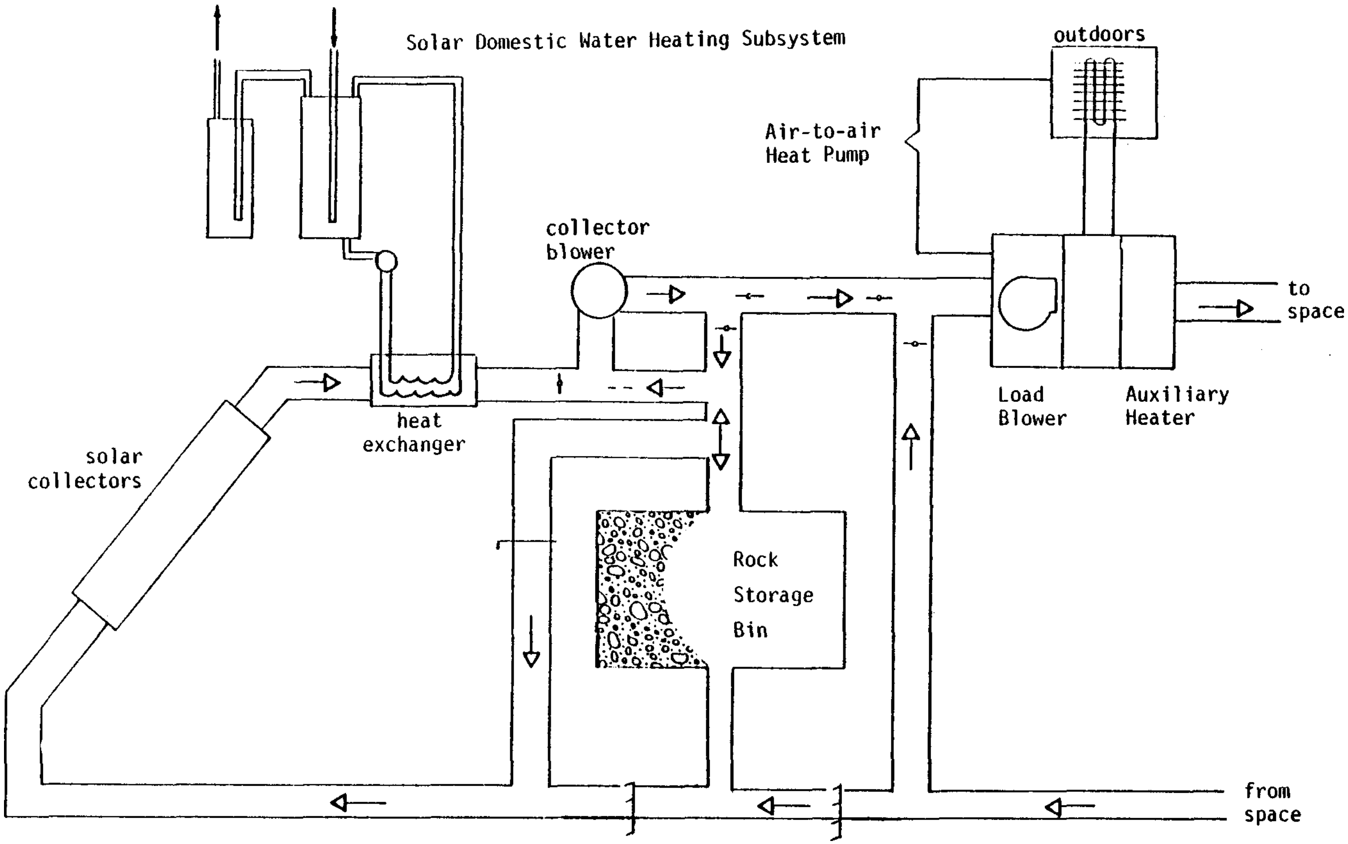


Figure 2-8. Air Parallel Combined Solar Heat Pump System

3.0 DESCRIPTION OF SIMULATION PROGRAM

This section describes the simulation methods which were used in the study. A brief introduction to the TRNSYS (31) program is presented in subsection 3.1 along with the basic approach taken to system simulations. The development of new, TRNSYS-compatible load and heat pump component models is detailed in subsections 3.2 and 3.3, respectively. The control schemes and operation modes for the various systems are detailed in subsection 3.4. The simulation run procedure is described in terms of the system sizing techniques employed and the output data requirements in subsection 3.5.

3.1 TRNSYS PROGRAM BACKGROUND

The modeling in this study was performed using TRNSYS (31, 32, 33), the generalized simulation program developed at the University of Wisconsin with the support from the National Science Foundation (NSF), Energy Research and Development Administration (ERDA), and DOE. The program consists of a central differential and algebraic equation solver, a library of component models, and front-end software which facilitates the building of system models and the interfacing with system forcing functions (weather data, etc.). The technique is to iteratively solve the set of simultaneous equations which describe the system at discrete intervals of time, and thereby mimic the operation of the system on the computer. Output devices such as printers, summarizers, and histogram plotters allow the user to "probe" system dynamics by tracking key state variables and energy flows.

TRNSYS defines a system as a set of components, interconnected in such a fashion as to accomplish a specified task. Judgment is required to decide where component boundaries are appropriate and how interfaces should be modeled. The space heating load/space heating system interface in particular has been handled in two different ways, described in the TRNSYS Users Manual (31) as temperature level control and energy rate control. Most systems studies performed with TRNSYS have used energy rate control because of its simplicity and lower cost. However, this energy rate technique is not detailed enough for control system studies, nor can it impress thermal comfort standards on the systems being modeled. Furthermore, energy rate control cannot be used for simulations in which the system and load models dynamically interact. For these reasons, temperature level control has been used in this study.

3.2 LOAD MODEL

Maximum accuracy is obtained when building load calculations and HVAC system performance calculations proceed simultaneously in an hourly system simulation since there can be an accounting for the effect of the varying interior conditions on envelope load. However, hourly HVAC system and load simulations are prohibitively expensive for system studies where many sensitivities are of interest unless a simplified load model is available. The ASHRAE transfer function technique (15, 34) is a compromise between very detailed methods (17, 35) and simple UA load calculations (36), with the method approximating load time lags due to building capacitance with the use of algebraic equations. The ASHRAE method has also been compared to a finite difference model which produced nearly identical results.

The load model is presented below in three subsections: a description of the simple load model (subsection 3.2.1), the validation of the simple load model (subsection 3.2.2), and a discussion of the accuracy of precalculated loads (subsection 3.2.3). A more detailed treatment of the load modeling issue is under preparation as a separate document which will include the TRNSYS-compatible program listing of the new simple load model.

3.2.1 Description of Simple Load Model

The ASHRAE transfer function technique has been utilized in Version 9.2 of TRNSYS (31) where separate roof, wall, and room modules are provided. These three modules have been combined into one TRNSYS component module in this new simple load model. Additionally, most of the original flexibility has been retained and several new features have been added. The new load module conserves computer time and is easier to use than the existing TRNSYS load package. A description of the model capabilities follows.

<u>Building Modeled</u>

The following assumptions are made concerning the geometry and orientation of the building:

- N-S-E-W orientation
- Simple rectangle
- Pitched roof with single gable (E-W axis)
- Fraction of wall area which is window can be specified individually for N, S walls, but must be the same for E, W walls
- All walls are of identical construction
- South walls can have arbitrary window-overhang geometries (overhangs on other walls are not modeled).

Conduction Terms

Heat flow through the walls and ceiling is calculated with the time series transfer function relation shown in Table 3-1, equation 1. In the case of the ceiling, an effective sol-air temperature ($t_{sa,n}$) accounts for the presence of the attic space. For further information, see Pawelski (34). Heat flow between the conditioned envelope and the basement is calculated with the steady-state relation given in Table 3-1, equation 2, and heat flux through windows is estimated also with a steady-state equation (equation 3).

The equations cited above calculate the energy flows penetrating the envelope via conduction at any given time. This flow may not immediately contribute to an energy load on the air in the conditioned space due to radiative exchange and storage effects. ASHRAE's transfer function technique (15) for distributing loads in time has been used and is described in the paragraph below on time distribution.

Solar Heat Gains

Using the sol-air temperature as described above allows the modeling of how solar radiation incident on opaque surfaces influences the conductive energy flow through those surfaces. Solar radiation entering the space through windows is a completely different problem.

The load model accepts as inputs the total solar radiation incident on each wall and the angle of incidence of beam radiation on each wall. To estimate the beam and diffuse components of total solar radiation, it is assumed that the total solar flux on the north wall is diffuse and equal to the diffuse flux on all other walls. This approximation limits use of the model to the northern hemisphere.

Table 3-1

Model Transfer Function Relations

EQUATIONS

Conduction Terms

- Walls and Ceiling

$$q = \left[\sum_{n=o} b_n \, (t_{sa,n} - t_r) - \sum_{n=o} d_n q''_n - t_r C_n \right] A \qquad (1)$$

- Conditioned Space/Basement

$$q_F = UA_F(T_b - T_r) \quad \text{if } T_r \leq T_a \qquad (2)$$
$$= 0 \quad \text{if } T_r > T_a$$

- Windows

$$q_w = UA_w(T_a - T_r) \qquad (3)$$

Time Distribution

$$Q_o = \sum_{j=1}^{4} \left[\sum_{i=o} v_i q_i - \sum_{i=o} w_i Q_i \right]_j \qquad (6)$$

Infiltration

$$\text{Sensible} = \dot{m} \; C_p \, (T_a - T_r) \qquad (7)$$

$$\text{Latent} = \dot{m} \; (W_a - W_r) \qquad (8)$$

Energy/Mass Balances

$$(MC)_S \frac{d(Tr)}{d\,\Theta} = S1(T_r, \Theta) + S2\,(T_r, \Theta) + S3\,(T_r, \Theta) \qquad (9)$$

$$(MC)_L \frac{d(W_r)}{d\,\Theta} = L1\,(W_r, \Theta) + L2\,(\Theta) + L3(W_r, \Theta) \qquad (10)$$

Table 3-1 (continued)

DEFINITIONS

q	=	current heat flow entering or leaving the envelope
b_n	=	transfer function coefficients of temperature terms
d_n	=	transfer function coefficients of heat flux terms
$t_{sa,n}$	=	sol-air temperature of the exterior surface at time n
t_r	=	the current value of room temperature
q''_n	=	values of heat fluxes entering or leaving the envelope at time n
C_n	=	the sum of all b_n
A	=	area of the surface
n	=	time (0 = now, 1 = previous hour, etc.)
T_a	=	ambient temperature
UA_F	=	the overall conductance between the basement and room
T_b	=	basement air temperature (assumed constant)
UA_W	=	overall conductance of the window system
Q_i	=	load on room air at time i
Q_o	=	current load on room air
q_i	=	energy flow at time I due to source j (conduction, solar heat gain, etc.)
v_i	=	transfer function coefficients for energy flows
w_i	=	transfer function coefficients for loads
$\dot{m}$	=	infiltration mass flow rate
W_a	=	ambient humidity ratio
W_r	=	room humidity ratio
C_p	=	specific heat of air
Θ	=	time
$(MC)_S$	=	effective thermal capacity of room
$(MC)_L$	=	effective dry air mass of room
S1	=	distributed sensible load from equation 6
S2	=	infiltration sensible load from equation 7
S3	=	conditioning air stream sensible term from equation 7
L1	=	infiltration latent load from equation 8
L2	=	generation latent load from 24 hour schedule
L3	=	conditioning air stream latent term from equation 8

To estimate the overall transmittance of the glazing system, the fresnel relationships (21), are used to calculate separate transmittance values for beam and diffuse components. The incidence angle for diffuse radiation is assumed to be 60^{o}. The two radiative components are weighted by their respective transmittances to estimate the solar heat gain.

The model also estimates the effect of a roof overhang on solar heat gain through windows on the south wall. As an overhang only effects the beam component of radiation, it would not influence solar heat gain through north-facing windows. It is assumed that east and west windows do not have overhangs.

The overhang model is shown schematically in Figure 3-1. VOH, HOH, and VHSW are use-specified parameters. H, the vertical height of the shadow, is calculated with the following equation:

$$H = \frac{HOH \tan(ALT)}{\cos(ASM)} \qquad [4]$$

where,

HOH = length of horizontal overhang

ASM = solar azimuth angle

ALT = solar altitude angle.

The fraction of the window shaded from beam radiation is then

$$FSHAD = \frac{H-VOH}{VHSW}, \qquad [5]$$

where,

VOH = vertical distance from top of window to plane of overhang

VHSW = vertical height of south windows.

As with the conduction terms, solar heat gains are distributed in time using the AHSRAE transfer function method.

Heat Generation Within the Space

Heat generation within the space is handled simply by reading in a 24-hour schedule of values from a data file. It is assumed that the daily distribution and hourly magnitudes of heat generation remain constant throughout the year. Three, 24-hour schedules must be provided: one each for sensible

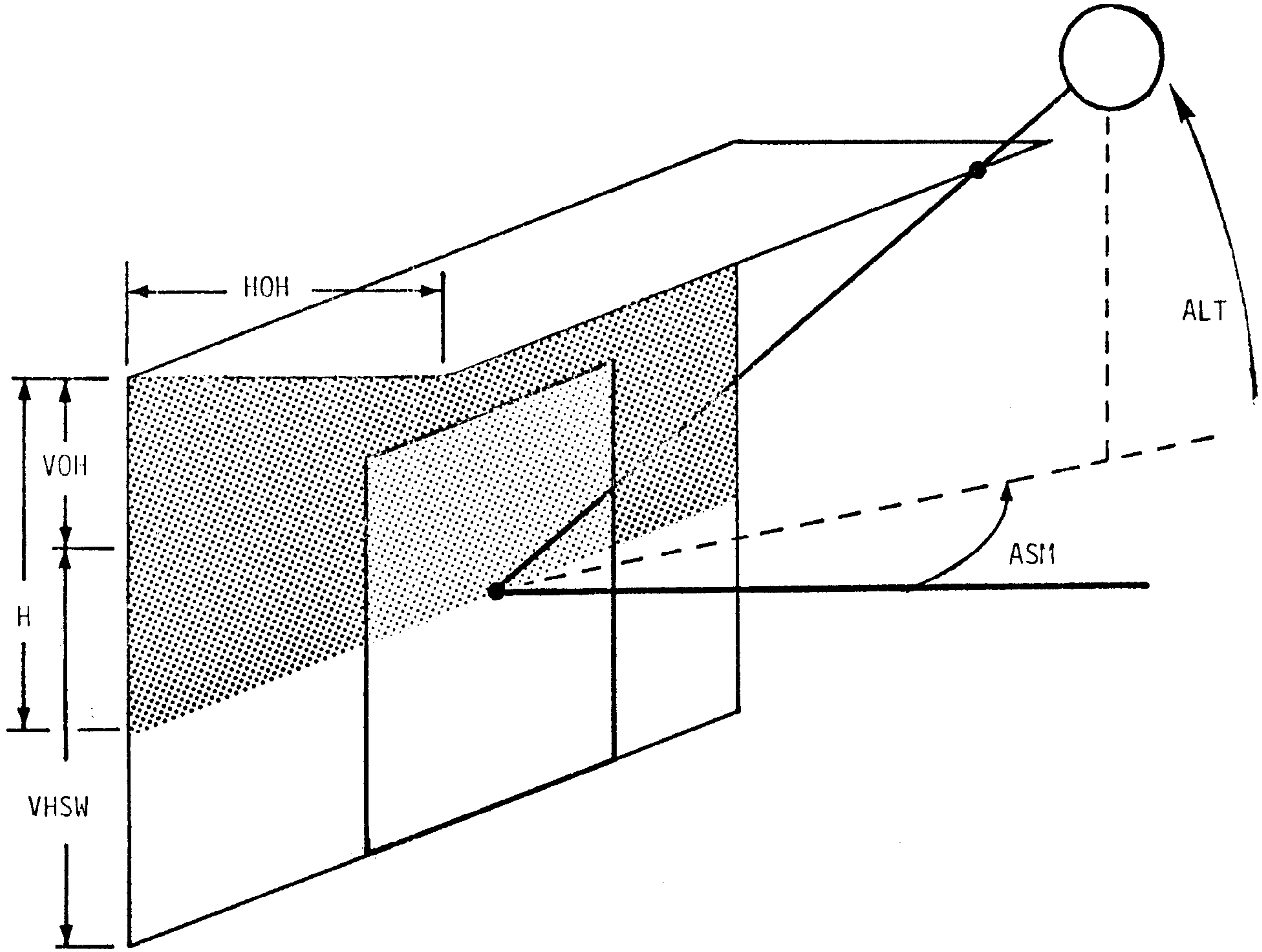


Figure 3-1. Window Overhang Geometry

heat generation from appliances and lighting, sensible heat generation from people, and water vapor generation. Sensible heat generation is split into appliance and people terms since the ASHRAE method of distributing loads distinguishes between the two.

ASHRAE Transfer Function Time Distribution

The ASHRAE transfer function method approximates the current energy load (heating or cooling) on the air in the conditioned space by distributing in time loads due to conduction, solar heat gain, appliance sensible heat generation, and occupant sensible heat generation. The method distinguishes among these four load sources since the source determines the fraction which is immediately seen as a load on room air. The algorithm used is listed in Table 3-1 as equation 6. Equation 6 only applies for sensible heat load distribution. Water vapor generation is assumed to immediately enter the room air.

Infiltration

Infiltration is assumed to occur at a constant air mass flow rate. The infiltration loads (sensible and latent) are assumed to instantaneously affect room air conditions. The equations are given in Table 3-1 as equations 7 and 8.

Conditioning Air Stream

The HVAC equipment interfaces with the building via the conditioning air stream. The sensible and latent components can be described with equations 7 and 8. Here, $\dot{m}$ represents the mass flow rate of air from the conditioning equipment; $\dot{m}$ is on or off in response to the room temperature thermostat.

Overall Energy/Mass Balances: Capacitance Determination

The energy and mass balance relationships are given by equations 9 and 10, respectively, in Table 3-1.

In the energy balance (equation 9), $(MC)_s$ is chosen to yield the desired HVAC equipment cycle characteristic. A typical cycle characteristic for heating mode operation is shown in Figure 3-2. The curve spans an ambient temperature interval from slightly below the design outdoor temperature to room temperature. At both extremes, the cycle rate is zero; at lower ambient

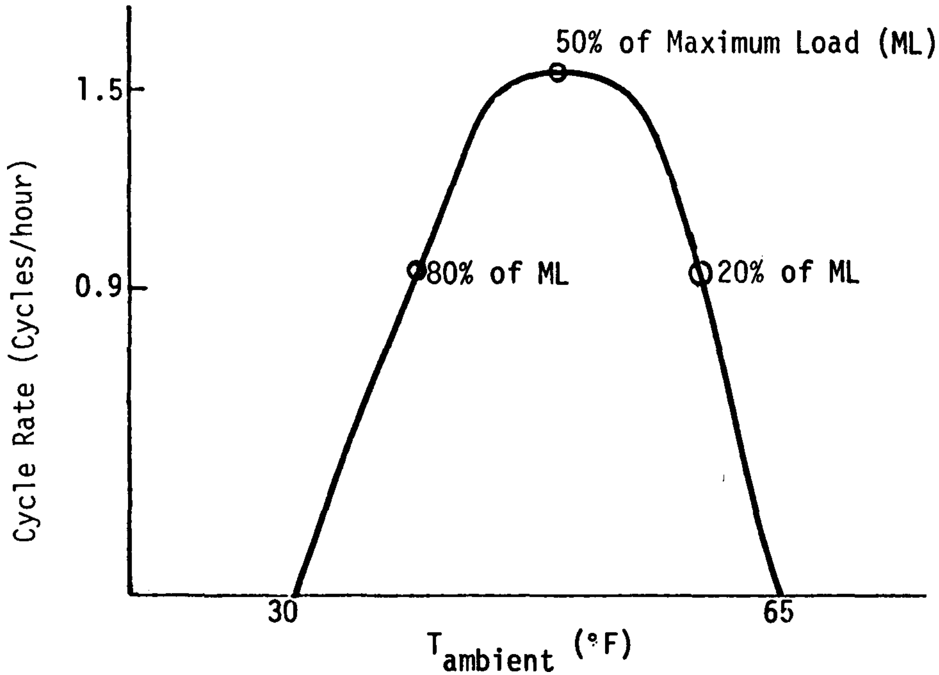


Figure 3-2. Typical Residential Heating Cycle Characteristics

tempeatures, the HVAC equipment runs continuously; and, at higher temperatures, heat is not required. The shape of the characteristic contains information concerning the ratio of HVAC equipment capacity to design load, thermostat characteristics (deadband, anticipation, etc.), $(MC)_s$, and functional relationship between ambient temperature and load.

By assuming the load is linear with ambient temperature (i.e., $UA\Delta T$) and the HVAC equipment capacity is equal to the maximum load, a room thermal capacitance, $(MC)_s$, can be calculated which, when used in a simulation, reproduces the desired characteristics.

In general,

$$(MC)_s \frac{dTr}{d\theta} = CAP\text{-}LOAD, \qquad [11]$$

and, with the heater off,

$$(MC)_s \frac{dTr}{d\theta} = -\ LOAD \qquad [12]$$

where,

CAP = heater capacity, and LOAD = current heating load.

Example: Determining effective thermal capacity of space during heating mode.

Referring to Figure 3-2, when the heating load is one-half the maximum load, ML (i.e., design load or machine maximum capacity), the cycle rate is desired to be 1.5 cycles/hour. Over any one complete cycle, CAP = LOAD; so

$$CAP = ML\ (\Delta\theta_{on}) \quad \text{and} \quad LOAD = .5\ ML\ (\Delta\theta_{cycle}).$$

Thus,

$$ML\ (\Delta\theta_{on}) = .5\ ML\ (\Delta\theta_{cycle})$$

$$(\Delta\theta_{on}) = .5\ (\Delta\theta_{cycle}) = .5\ (1\ \text{hour}/1.5\ \text{cycles})$$

$$(\Delta\theta_{on}) = 1/3\ \text{hour}$$

And, since $(\Delta\theta_{cycle}) = 2/3$ hour, then $(\Delta\theta_{off}) = 1/3$ hour. During the off-cycle, equation 12 becomes (with thermostat deadband of 2.5°F either side of set point),

$$(MC)_s = (1/3 \text{ hour})(-0.5ML)/(-5^oF)$$

$$(MC)_s = 0.033ML.$$

If a different cycle characteristic exists for the cooling season (e.g., 3.0 cycles/hour at 50 percent of maximum load), a separate $(MC)_s$ can be determined for cooling.

A similarly simple procedure for calculating $(MC)_L$ is not available. Room humidity is not used as a control variable; therefore, no control stability criterion affects the choice of $(MC)_L$. $(MC)_L$ represents the effective dry air mass which is available for storing water vapor. As partitions and furnishings add to this effect, $(MC)_L$ should be larger than the dry air mass within the building. A currently used rule of thumb is 10 times the dry air mass (37).

Stability Criteria

It remains to verify that the simulation timestep used is such that the solution to equation 9 is always stable. The most extreme condition (i.e., largest net energy flow into or out of the building) is when the HVAC equipment is on at full capacity while no load is being incurred. In this circumstance, $\Delta\theta$ (the timestep) must be small enough so that room temperature does not change more than the thermostat bandwidth (2x deadband), which would cause controller oscillation. In other words:

$$(MC)_s \frac{\Delta Tr}{\Delta\theta} = ML$$

$$\Delta\theta_{max} = (MC)_s \ (2x \text{ deadband})/ML$$

$$= .033ML\ (5^oF)/ML$$

$$\Delta\theta_{max} = .165 \text{ hr.}$$

In addition to controllor stability, numerical integration stability must be ensured. The maximum stable timestep cannot be determined explicitly from equation 9 because S1 (see equation 6) is dependent upon past conditions. An upper limit is obtained by ignoring S1. For Euler integration, the procedure would be as follows:

$$(MC)_s \frac{dTr}{d\theta} = \dot{m}Cp(T_i - T_r) + \dot{m}_{inf}Cp(T_2-T_r);$$

therefore,

$$T_r^+ = T_r\left[1 - \Delta\theta\left(\frac{\dot{m}Cp+\dot{m}_{inf}Cp}{(MC)_s}\right)\right] + \text{Constant.}$$

For stability,

$$\Delta\theta\left(\frac{\dot{m}Cp+\dot{m}_{inf}Cp}{(MC)_s}\right) \leq 1$$

$$\Delta\theta_{max} = \frac{(MC)_s}{\dot{m}Cp+\dot{m}_{inf}Cp} \quad .$$

Simple Load Model Representation

This simple load model has been constructed to be TRNSYS-compatible. The parameters, inputs, and outputs used with this model are listed in Appendix B. Additionally, Appendix B describes the data file format for internal heat generation schedules.

3.2.2 Validation of Simple Load Model

The simple load model described in the previous subsection has been validated by comparing its output to that from the load modules existing within TRNSYS (31). Since the TRNSYS load models are based on a generally accepted ASHRAE procedure (15) and compare favorably with more detailed models and with experiment (34, 38, 39, 40), further validation was not considered necessary.

In the following subsections, plots are presented which compare predicted energy flows from the existing TRNSYS components and from the new simple model. Typical Meteorological Year (TMY) (41) weather data for the first week of January in Washington, D.C., have been used. Temperature level control (31) has been used to interface furnaces with the load models and thus maintain room comfort conditions.

Conduction Heat Flows

Equation 1 from the previous subsection is used to calculate conduction heat flows through the walls and pitched roof. Figure 3-3 shows plots of these heat flows calculated during the first two days of the simulation by the two models. Whenever variables have the same value, the variable represented by the largest numeric symbol is plotted. Hence, the simple load model indicators 2 and 4 are shown wherever they superimpose the TRNSYS indicators 1 and 3. The agreement is shown to be excellent. Note that the ASHRAE response factor technique requires roughly 18 hours at the beginning of a simulation to initialize itself.

Room Air Loads

Equation 3 from the previous subsection is used to convert envelope heat flows due to conduction, solar heat gain, appliance sensible heat generation, and occupation sensible heat generation into the current load on the air mass of the building. Figure 3-4 illustrates the room air loads estimated by the current TRNSYS models and by the simplified model. The heat flow terms included for this comparison were wall, ceiling, and floor conduction; infiltration, and generation. The solar heat gain term was not included because the new model utilizes more accurate approximations to estimate this term, causing it to disagree with the current TRNSYS models.

Figure 3-4 illustrates that the room air load estimates of the two models agree very well. The calculations illustrated in Figures 3-3 and 3-4 are only performed on the first iteration of simulation timesteps which fall at one-hour intervals (hour 0, hour 1, hour 2, etc.); thus, slight differences in room temperature on this iteration cause slight load estimate differences. But, since both models were controlled to maintain room temperature within a narrow band, the difference in integrated energy flows is very small.

Total Instantaneous Loads

In addition to the room air load, the total instantaneous load includes contributions from infiltration, ventilation, or any other direct interaction between room air and an energy source or sink. Figure 3-5 illustrates how the two estimates of total instantaneous load compare. Figure 3-5 includes an infiltration term of one air change per hour in addition to the

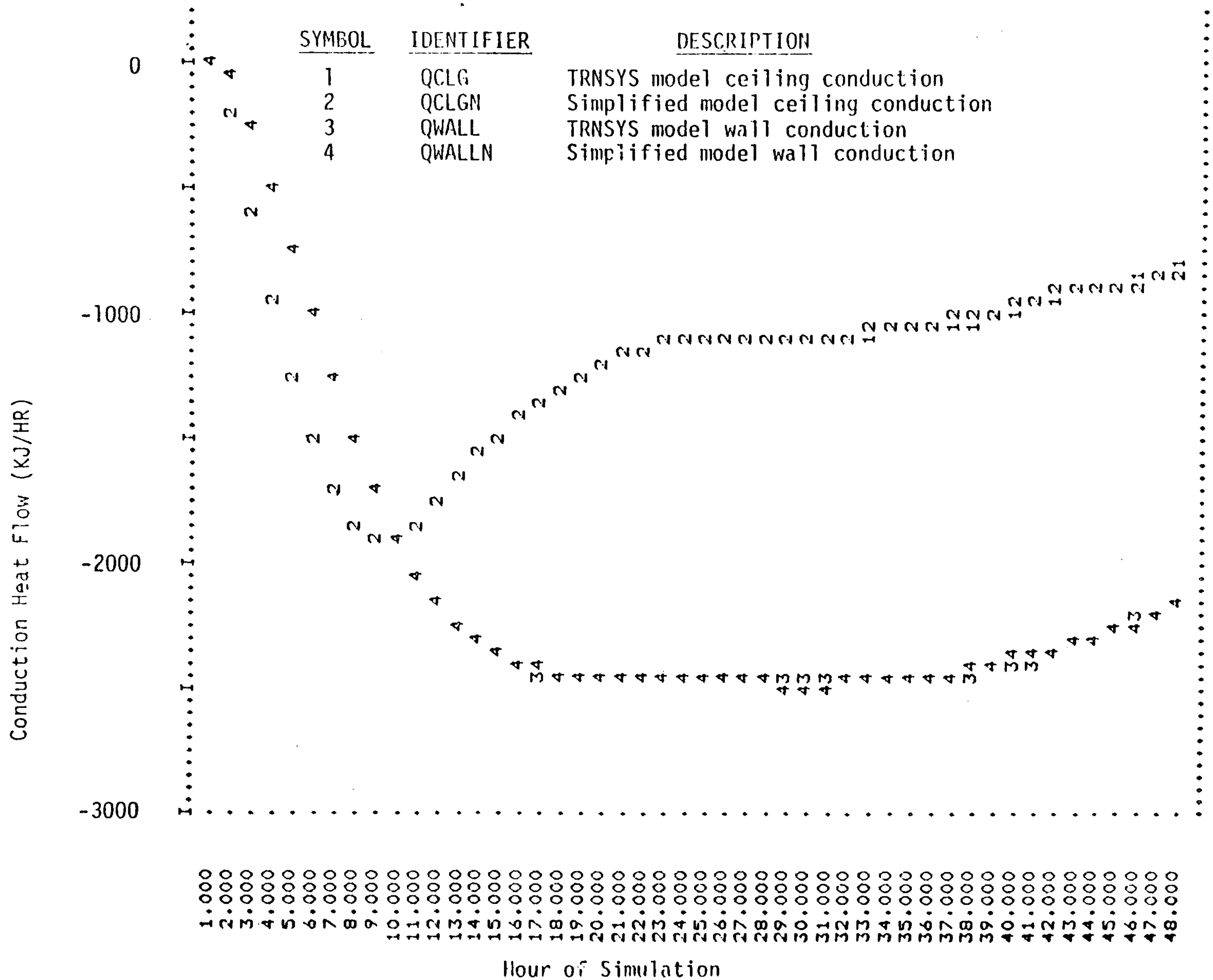


Figure 3-3. Comparison of TRNSYS and Simplified Load Model Estimates of Wall and Ceiling Conduction Heat Flows

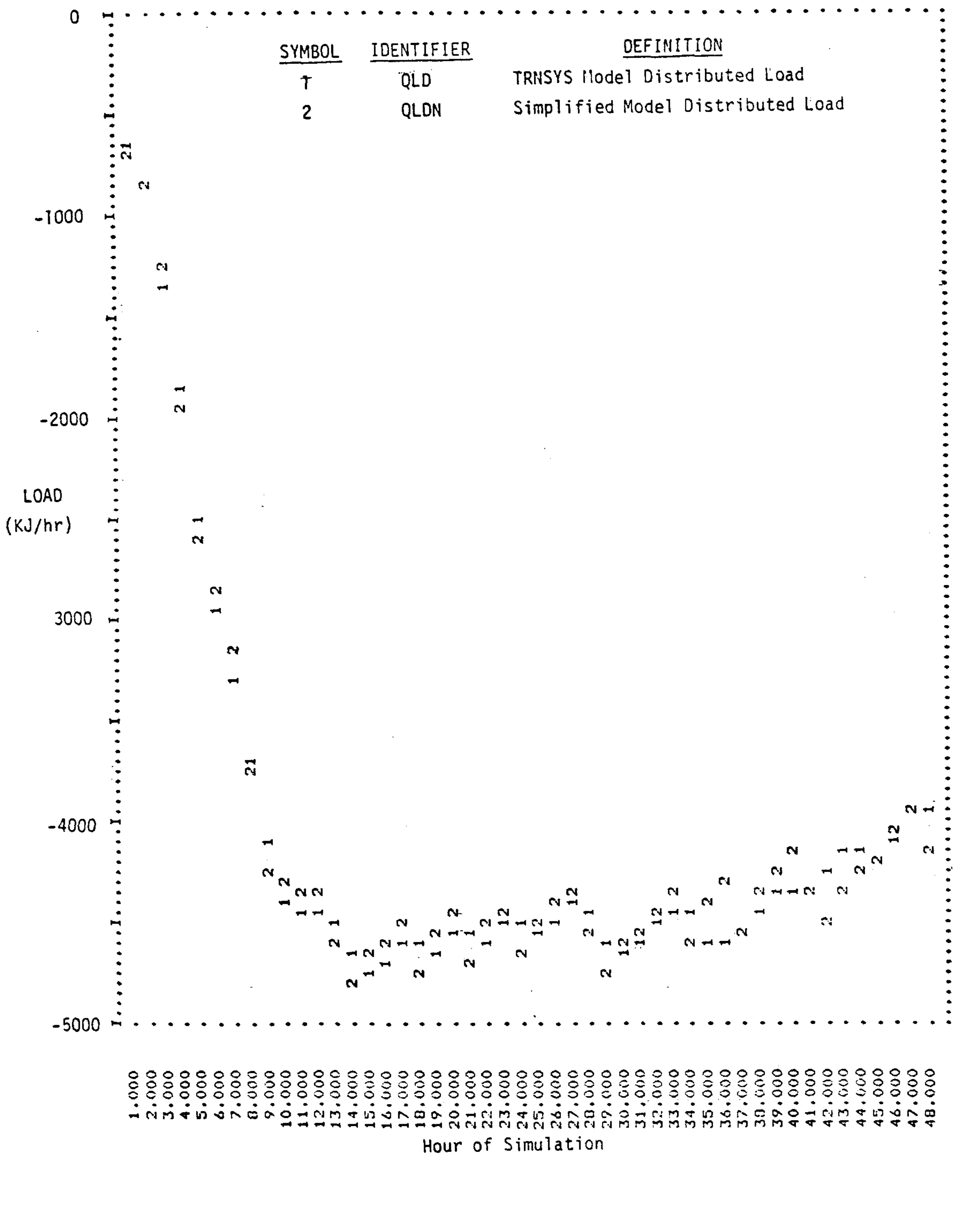


Figure 3-4. Comparison of TRNSYS and Simplified Load Model Estimates of Room Air Load

SYMBOL	IDENTIFIER	DEFINITION
1	QLOAD	TRNSYS Model Total Instantaneous Load
2	QLOADN	Simplified Model Total Instantaneous Load

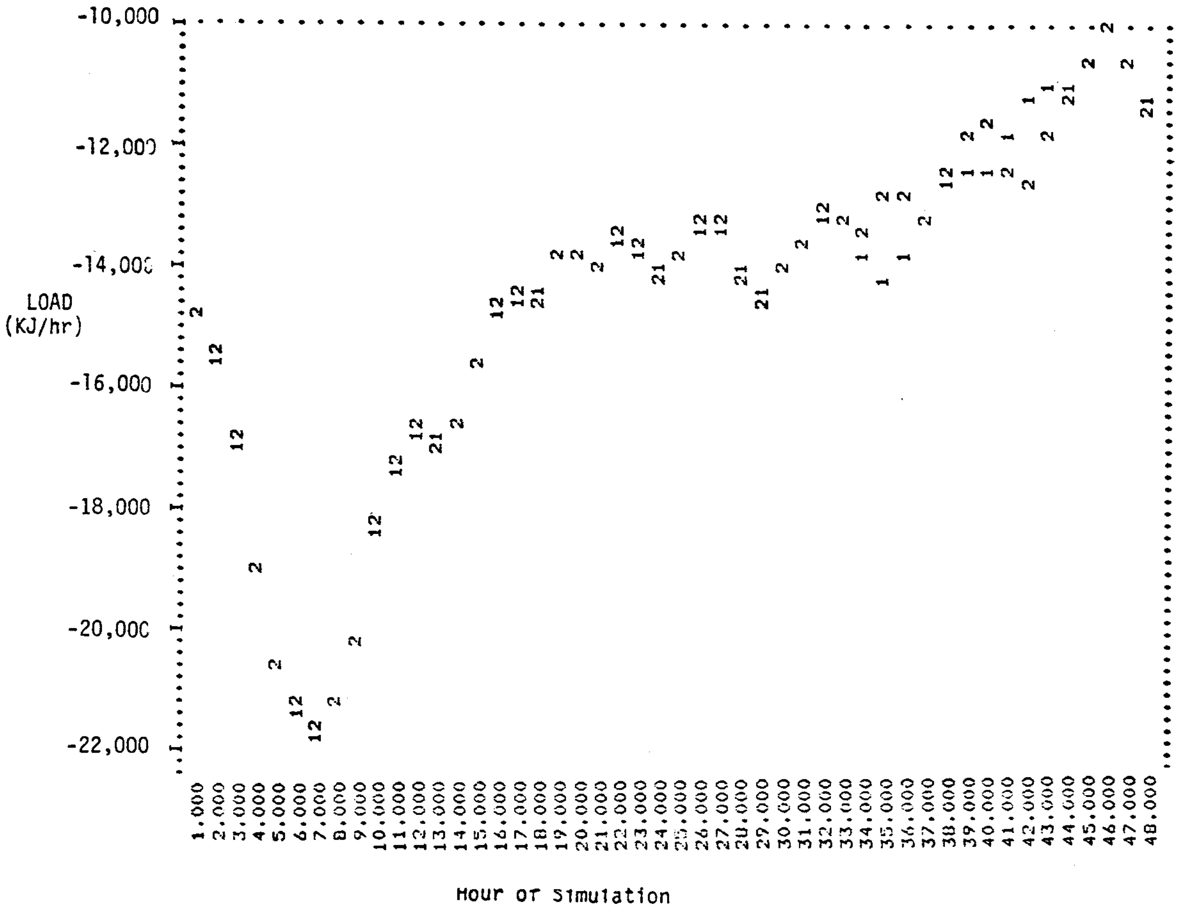


Figure 3-5. Comparison of TRNSYS and Simplified Load Model Estimates of Total Instantaneous Load

room air load of Figure 3-4. Agreement between the two models is shown to be very good.

Solar Heat Gains

The solar heat gain calculations performed by the TRNSYS and simple load models differ in two respects. The TRNSYS model assumes that the transmittance of glazing is constant regardless of the angle of incidence, while the new model divides radiation into beam and diffuse components and calculates transmittance based on the actual incident angles for each. The TRNSYS model is also incapable of estimating the effect of overhangs, whereas the new model can account for horizontal overhangs on the south wall. The effect of these differences is illustrated in Figures 3-6 and 3-7. The third and fourth days of January have been used because both were clear days and maximum differences can be seen.

Figure 3-6 shows solar heat gain estimates for the east and west walls. A transmittance value for double glazing at 60° incident angle was used for the TRNSYS model. Since the new model assumes the incident angle of diffuse radiation is 60°, the models should agree whenever all radiation incident on the surface is diffuse. As expected, this occurs in the afternoon on east surfaces and in the morning on west surfaces.

When the surface receives both beam and diffuse radiation, the magnitudes of the two components and the incident angle for beam radiation determines which of the two models predicts higher. In the late morning hours, the incident angle of beam radiation on the east wall is larger than 60°, so the new model predicts less solar heat gain than the TRNSYS model. However, earlier in the morning, the incident angle is smaller than 60°, so the new model predicts a higher solar heat gain; in the afternoon, solar heat gain through west-facing windows shows the same behavior.

Figure 3-7 illustrates the effect of the horizontal overhang on the south windows. The new model estimates higher solar heat gain through south windows when no overhang is specified because the incident angle for beam radiation is less than 60° throughout the day. By applying a three-foot horizontal overhang, the base of which is one foot above the top of the window, the beam component is substantially shaded and the model predicts a lower solar heat gain. A four-foot overhang is also shown to illustrate the progressive

SYMBOL	IDENTIFIER	DESCRIPTION
1	QSHGE	TRNSYS Model solar heat gain through east windows.
2	QSHGEN	Simplified Model solar heat gain through east windows.
3	QSHGW	TRNSYS Model solar heat gain through west windows.
4	QSHGWN	Simplified Model solar heat gain through west windows.

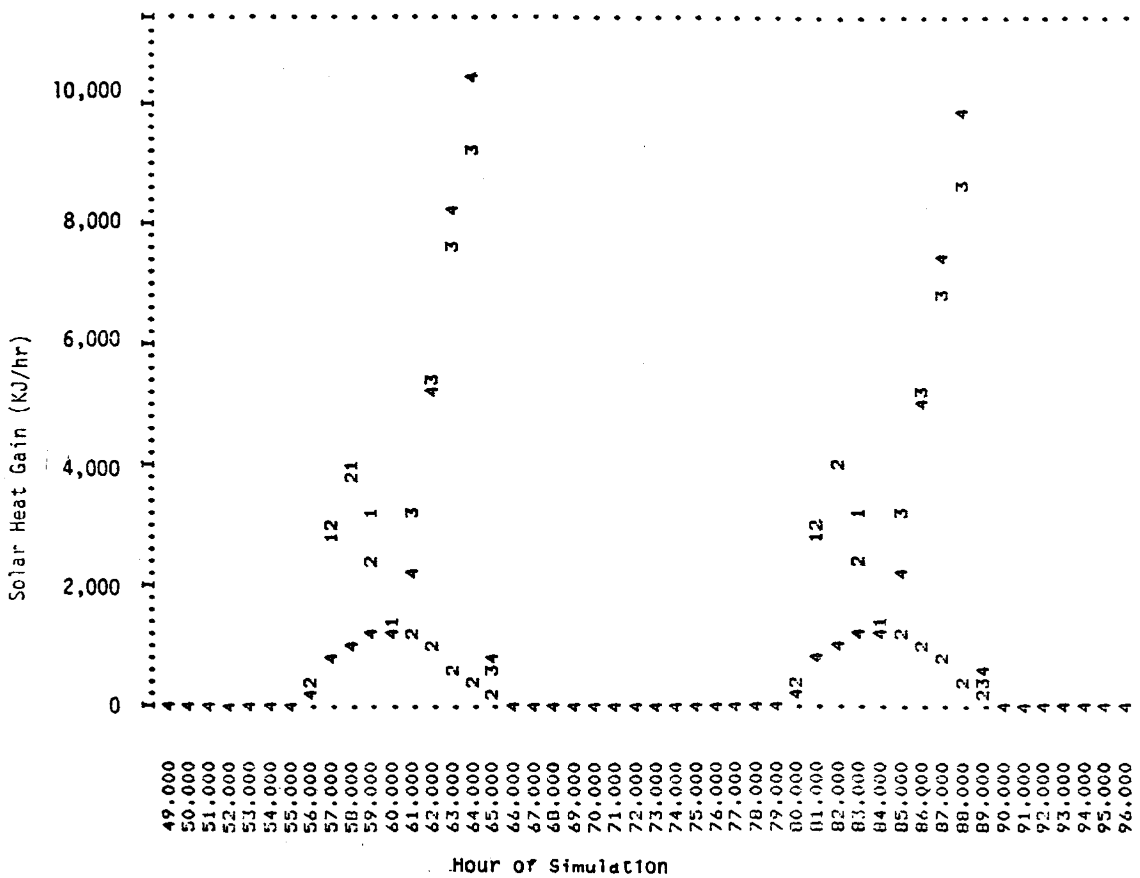


Figure 3-6. Comparison of TRNSYS and Simplified Load Model Estimates of Solar Heat Gains Through East and West Windows

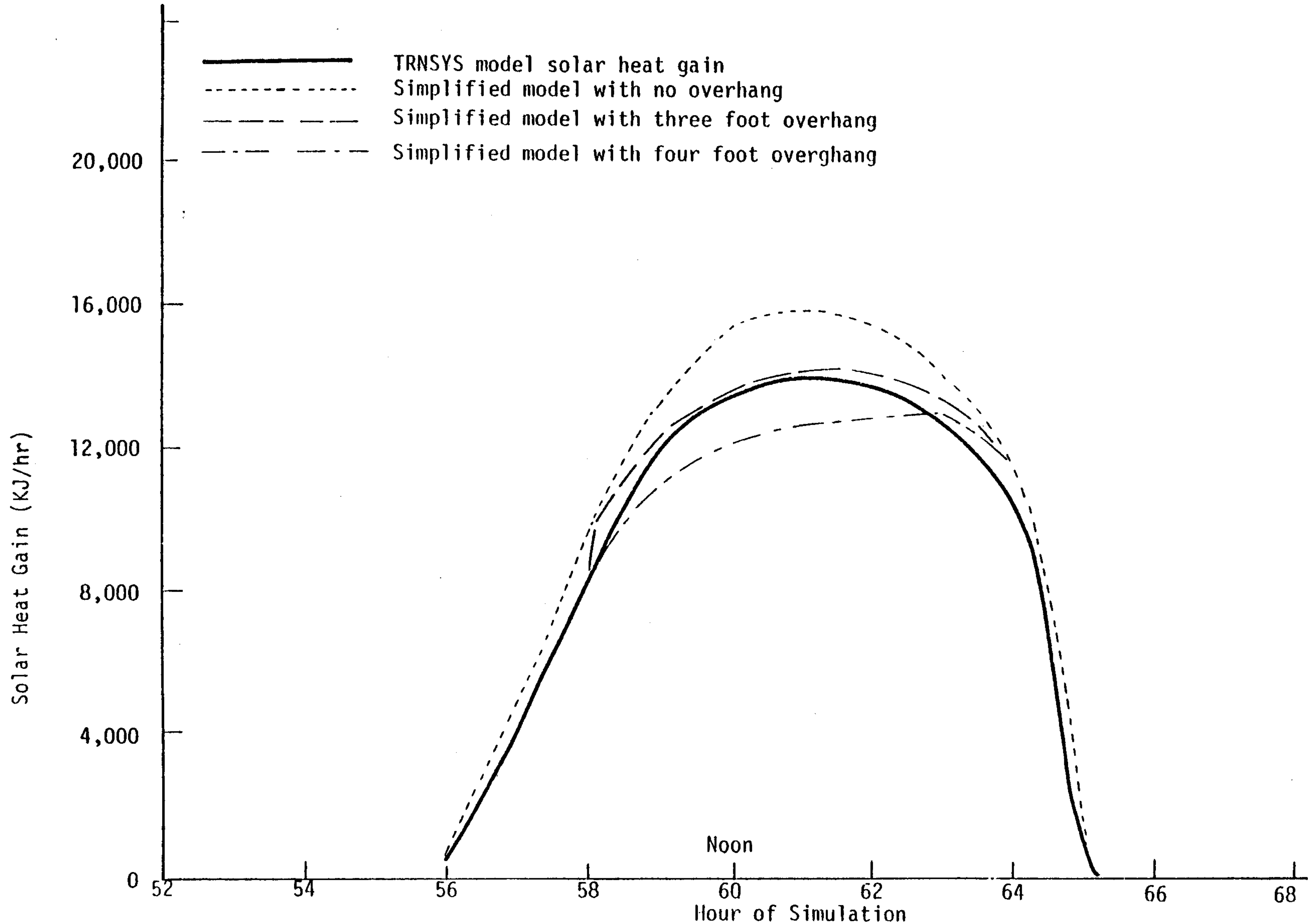


Figure 3-7. Comparison of TRNSYS and Simplified Load Model Estimates of Solar Heat Gain Through South Windows

effect of larger overhangs. Note that incident solar radiation was not symmetric on this particular day (hour 60 = noon) due to morning cloudiness. The effect of the overhang is also not symmetrical because the ratio of beam to total radiation incident on the south wall varies from morning to afternoon.

3.2.3 Accuracy of the Precalculated Load Approximation

In general, sophisticated load analyses are too costly to perform simultaneously with system simulations if a parametric study requiring many simulations is intended. An alternative to the above approach is to calculate loads in advance using a sophisticated analysis and then to use the precalculated loads as input to the system simulation model. However, the precalculated load approach is criticized for not properly accounting for the load/HVAC-system interaction.

The major criticism of the simple load approach is that the transient responses of a building to changes in solar heat gain, ambient temperature, etc., are not properly accounted for, resulting in load predictions which are incorrect in magnitude and time distribution. In the previous section, a semi-sophisticated, simple load model was described which can be used to precalculate loads or interact dynamically with a system simulation. This simple model is sufficiently detailed to account for the transient characteristics of a typical residence and thus satisfies most of the simple load critics. It is also a stand-alone TRNSYS component model and, as such, is less expensive to use than the load package currently in the TRNSYS program.

Surprisingly little information has been reported to support either of the above currently used approaches or to support criticism of them. In this subsection, the simple-load model is used to compare the accuracy of precalculated load versus dynamically interacting loads. Since a separate document is under preparation which presents the detailed comparison of the load approaches, the discussion below merely outlines the major issues.

Load/System Interface: Energy Rate and Temperature Level Control

When modeling HVAC systems, a decision must be made as to where the conditioning system ends and the load model begins. After the boundary is defined, there must be agreement upon a method of interfacing the system and load. The originators of TRNSYS have identified two interface techniques which they refer to as "energy rate control" and "temperature level control." The building load component is represented as a black box with inputs and outputs corresponding to the conditioning air stream parameters. The conditioning air stream is used to link the system simulation and the load model.

In "energy rate control," the load/system boundary is defined with the coil, radiator, or heat transfer device inside of the load model. The inlet and outlet heat transfer device fluid system is then a liquid, implying that the system (central plant, solar system, etc.) is liquid based. In an "energy rate control" simulation, the long-term energy balance between the building and system is maintained by forcing the energy balance to close every timestep. This is done by allowing the building load model to control the flow rate from the system, and also to calculate the temperature of the fluid returning from the load so that an energy balance is guaranteed.

In "temperature level control," the long-term energy balance is not guaranteed, but results only if the conditioning system properly responds to a controller monitoring room temperature (and possibly humidity), thus keeping room temperature within the desired bounds. Although not a requirement, the load/system boundary is usually chosen so that the fluid stream is the forced air circuit used to condition the space. In this way, the entering air temperature can be forced to meet a minimum delivery requirement (heating) and the outlet temperature is room temperature and can be used for control purposes.

Precalculated Load Approximation

With either energy-rate control or temperature-level control, the sensible load may be calculated simultaneously with the system simulation or precalculated and read in with the meteorological data. Latent load calculations are trivial; thus, precalculation offers no advantage. When energy rate control is used, precalculated loads and simultaneous loads yield identical system simulation results. Since energy rate control implies constant room conditions, no additional approximation is made when loads are precalculated. The methods for precalculating loads for energy rate control simulations are thoroughly explained in the TRNSYS User's Manual (15). The following describes the approximation required to precalculate loads for temperature level control simulations.

Temperature level control has the feature that dynamic room conditions (temperature and humidity) are calculated during the system simulation. It cannot be hoped to precalculate loads for such a simulation using the actual room conditions at any given time. The best that can be done is to precalculate loads with room temperature constant at a value equal to the midpoint of the control band. It is reasonable to expect that room temperature will hover above and below the control midpoint, and that, over a year, the time spent above and below will be nearly equal. Thus, errors in the precalculated loads will tend to be offsetting. Furthermore, if the temperature control band is narrow (typical of conventional HVAC systems and active solar systems), the magnitude of errors will be small to begin with since the minor deviation of room temperature (difference between value used for precalculating loads and instantaneous value during simulation) is small compared to the heat transfer forcing function (room-to-ambient-temperature difference).

Comparative Simulations: Precalculated vs. Interactive Loads

A TRNSYS simulation deck was constructed to read TMY data for January in Washington, D.C., and write a file containing the precalculated load, as well as all meteorological data needed to drive subsequent simulations. Two load files were created, one including and one excluding infiltration, so that effects due to this term could be studied. The loads were calculated with the simple load model described previously.

Next, a TRNSYS deck was constructed to simulate two simple heating systems simultaneously. One system was driven with the precalculated load while the other calculated loads simultaneously with the simple load model, this latter configuration representing a load calculation which was fully interactive with the system simulation. The heating systems for both loads consisted of a single-stage thermostat controlling a fully-on or fully-off furnace. The loads were precalculated with room temperature equal to the midpoint of the thermostat control band. The solar heat gain and generation terms are realistic estimates for a residence in Washington, D.C., with insulation values typical of new construction being used. In this way, the comparison was not biased for or against precalculated loads by weighting heavily or lightly the terms which are not dependent on room temperature.

The first test run used a building air load consisting of conduction, solar heat gain, and generation terms only; no infiltration term was included.

As expected, the precalculated and interactive approaches tracked very closely. The interacting load oscillated above and below the precalculated load as room temperature oscillated around the midpoint of the control region.

Figure 3-8 corresponds to the first test run except that infiltration was included in the precalculated load file and included dynamically in the interactive load calculation. From Figure 3-8, it can be concluded that the room temperature differences are minor in their effect on loads which consider conduction, solar heat gain, generation, and infiltration.

Precalculated loads are apparently adequate for temperature level control simulations so long as the loads are calculated with a constant room temperature equal to the center of the control deadband used in subsequent simulations. This control midpoint is well defined for systems with singlestage controllers. Changing the control midpoint from summer to winter presents no problem, provided the constant room temperature used to generate the precalculated loads is changed on the correct calendar date. A further condition required, if precalculated loads are to be accurate, is that the HVAC system must have adequate capacity to meet the load because, if room temperature is not maintained within the control band, load calculation errors are no longer counteracting and the errors will accrue.

Two-Stage Controller

It is unclear how accurate precalculated loads are when used to simulate systems with multi-stage controllers. Here, the "time-averaged" midpoint room temperature to use for precalculating the load is unknown since the relative frequency of first- and second-stage heating experienced during the subsequent simulation is not known a priori. To investigate the use of precalculated loads for multi-stage control systems, the TRNSYS deck was modified; the single-stage controller and furnace combinations were replaced with two-stage controllers and a heat pump. First-stage heating operates the heat pump using ambient air as the source and second-stage heating adds auxiliary heat in addition to that produced by the heat pump. The controllers were set so that the temperature used to precalculate the loads fell in the center of the total control region.

The result of this simulation is shown in Figure 3-9. The substantial differences shown are partly exaggerated due to the elimination of infiltration

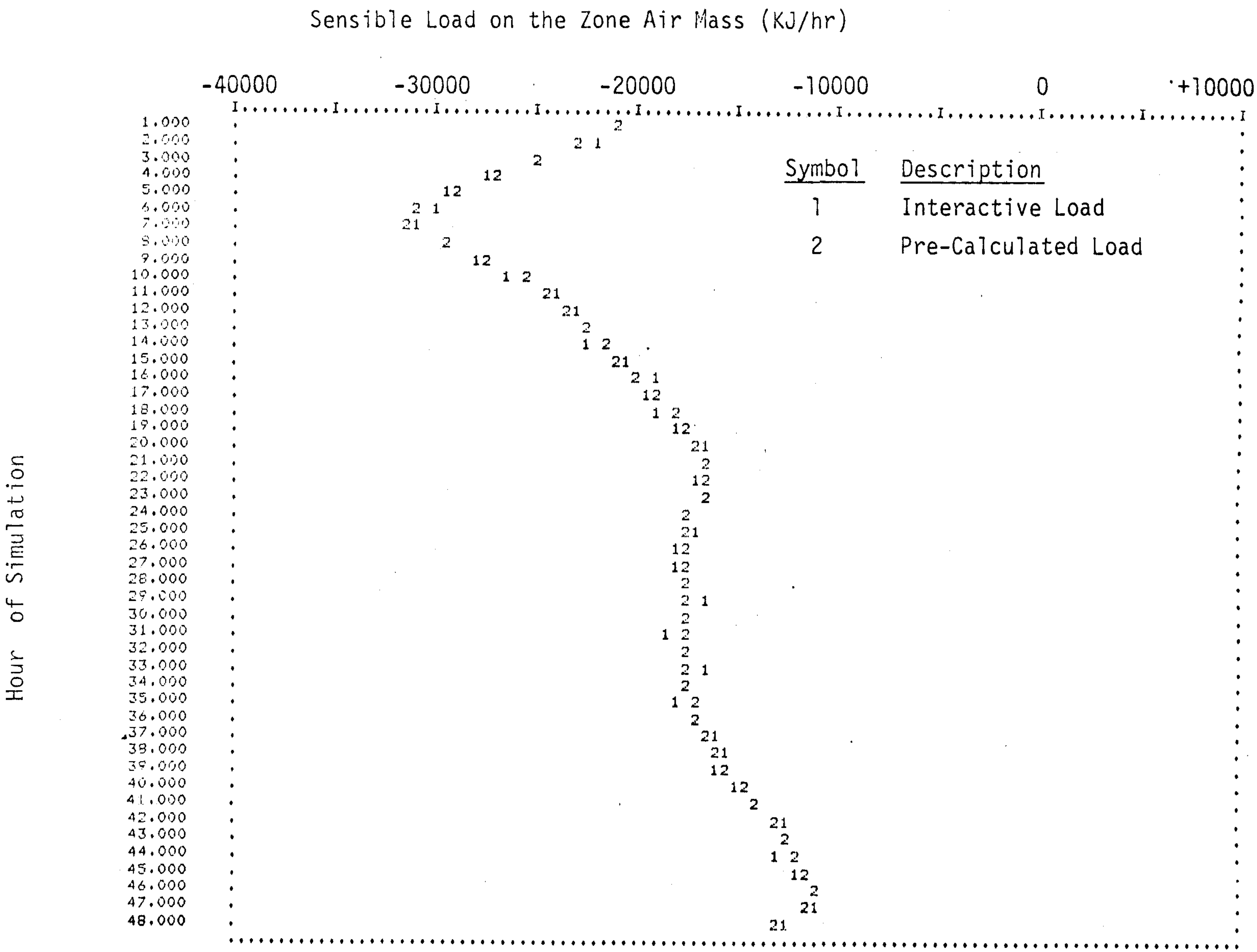


Figure 3-8. Pre-Calculated vs. Interactively Calculated Loads for a Furnace with Single Stage Controller (with infiltration).

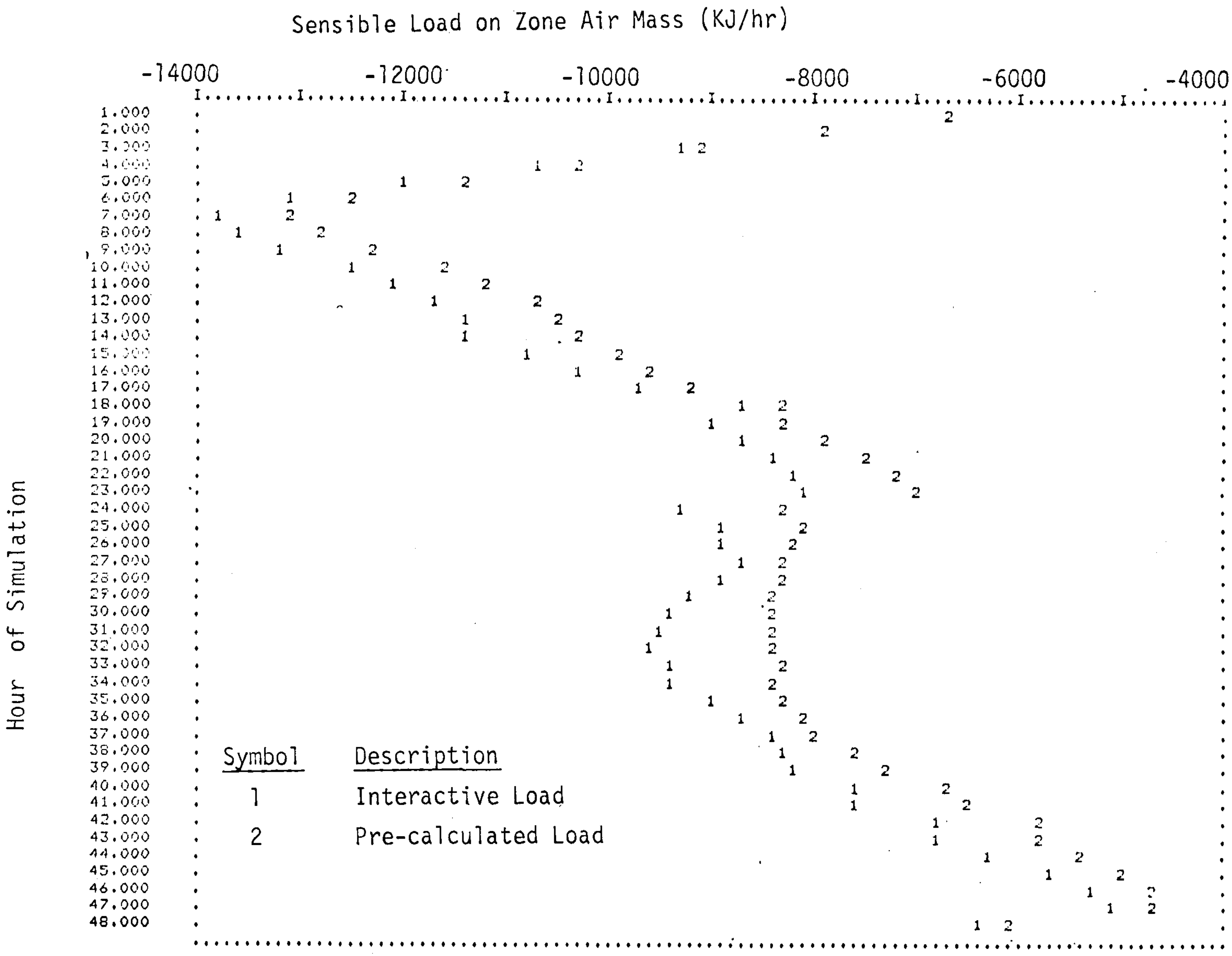


Figure 3-9. Pre-Calculated vs. Interactive Loads for an Air-to-Air Heat Pump with 2 Stage Controller (No Infiltration, 1st Stage Centered at 21^{O}C)

from the load terms; however, the errors are biased and will accumulate. The substantial differences are due to the fact that the system operated primarily in first stage during this two-day period.

To investigate this effect over a longer time period, the heat pump simulations were rerun for the complete month of January. The TRNSYS timestep and error tolerances were adjusted to obtain maximum accuracy. The temperature used to precalculate loads was moved up to be centered on first stage, anticipating that first-stage operation would predominate. As seen in Table 3-2, heat pump operation in the two simulations was very similar. The interactive load was slightly less because room temperature spent more time below the center temperature than above. It is disturbing that the extra load is completely provided by auxiliary heat. Under environmental conditions where room temperature apporaches the second-stage turn-on temperature, the interactive load was smaller than the precalculated load. Apparently it was smaller by an amount which allowed the first-stage heat source (heat pump) to prevent room temperature from reaching the control band bottom on several occasions where it was reached using precalculated loads. Consequently, more auxiliary was used in the precalculated load simulation while fewer hours of heating were required; this situation arose since the energy delivery rate was higher in second stage.

Conclusions

Several conclusions can be drawn from the above. Precalculated loads are acceptable for temperature level control simulations if the system using these loads has a single-stage room temperature control, a reasonably narrow room temperature control band, and adequate capacity so that the room temperature can be maintained in the control band. Systems which require two-stage controllers cannot, in general, be accurately modeled with precalculated loads. A cumulative load bias will occur, high in some months and low in others, no matter how the control band and load precalculation temperature are matched.

3.3 DESCRIPTION OF HEAT PUMP MODEL

A new heat pump component model has been developed for this project. The major differences from past public-domain models are that cooling mode performance is calculated as a function of indoor entering wet bulb temperature and that the model interfaces with the rest of TRNSYS via temperature level control so that delivered air temperature and control dynamics can be studied.

Table 3-2

Precalculated vs. Interactive Loads for an Air-to-Air Heat Pump with 2-Stage Controller (Infiltration Included, 1st Stage Centered at 20°C, Results for the Month of January, Washington, D.C., TMY Data

	QHP-L (GJ)	WHP (GJ)	COP	QLOAD (GJ)	QA-L (GJ)	HRHP (hr)	HRHTG (hr)
Pre-Calculated	11.91	5.24	2.27	13.00	1.08	46.24	46.24
Interactive	11.94	5.27	2.27	12.77	0.83	46.51	46.51

Definitions:

QHP-L - energy delivered to the load by the heat pump

WHP - total work input to run heat pump (compressor, indoor and outdoor fans)

COP - coefficient of performance of heat pump

QLOAD - energy delivered to the load

QA-L - auxiliary energy required to meet the load

HRHP - compressor run time

HRHTG - total hours of heating system operation

The use of this new heat pump model with performance data is given in subsection 3.3.1. The model also has the capability of modifying capacity for start-up and shut-down transients (subsection 3.3.2). Subsection 3.3.3 describes the system interface refinements included with this model. The heat pump performance characteristics of present-day and future-generation heat pumps used in this study are presented in Subsection 3.3.4.

A detailed description of this heat pump model is being prepared as a separate document. Appendix C contains a TRNSYS-module listing of model inputs, outputs, and parameters.

3.3.1 Performance Data Preparation

The new heat pump model accepts user-specified performance data tables for both the heating and cooling modes. Performance data must be provided for the range of source temperatures (heating mode) or sink temperatures (cooling mode) that will be encountered during the simulation. In this way, off-design performance is specified rather than extrapolated.

Cooling Mode

The heat pump cooling mode operation was constructed to use manufacturers' data which report total and sensible capacities as a function of entering indoor wet bulb temperature. Thus, this model is able to deliver latent and sensible cooling capacities as specified for the particular heat pump. When the model input data are carefully constructed as described below, the original manufacturer's data can be reproduced adequately. However, the model is still only applicable for the manufacturer's specified indoor air flow rate (FLOWR) and entering dry bulb temperature (EDB).

A table of cooling mode performance characteristics is required in the form shown in Table 3-3. The data should be for a specific indoor air mass flow rate and entering indoor dry bulb temperature. The performance data table is accepted in English units to minimize data preparation time since all manufacturers report data in these units. In addition to the performance data table, several constants (parameters) are needed, as listed in Table 3-3.

To illustrate how the required input data are obtained from manufacturers' data, an example is given here for a Carrier 38RQ034/40AQ036-1275/.13 air-to-air heat pump. The original cooling mode performance data for this heat

Table 3-3

Cooling Mode Heat Pump Performance Data
and Associated Parameters Required

Performance Data Format

$TSINK_1$	$DCCC_1$	BPT_1	W_1
.	.	.	.
.	.	.	.
.	.	.	.
$TSINK_n$	$DCCC_n$	BPT_n	W_n

where

TSINK = sink (outdoor or tank) dry bulb temperature (C)

DCCC = dry coil cooling capacity (KJ/hr)

BPT = breakpoint entering indoor wet bulb temperature (C)

W = work input (compressor, indoor and outdoor fans) at the breakpoint entering wet bulb temperature (KJ/hr)

Required Parameters

Air-to-air or liquid-to-air heat pumps:

FLOWR = indoor air mass flow rate corresponding to performance data table (Kg/hr)

EDB = indoor entering dry bulb temperature corresponding to performance data table (C)

TCS = slope of the total capacity vs. indoor entering wet bulb temperature (EWB) line in the wet coil region. (KJ/hr-C)

SHCS = slope of the sensible heat capacity vs. EWB line in the wet coil region. (KJ/hr-C)

WS = slope of the work vs. EWB line. (KJ/hr-C)

Liquid-to-air heat pumps only:

FLOWS = sink liquid mass flow rate (Kg/hr)

CPS = sink fluid specific heat (KJ/kg-C) pump are presented in Table 3-4 where total and sensible cooling capacities are given for several sink and entering indoor wet bulb (EWB) temperatures. If these data are plotted as shown in Figure 3-10, a series of y-curves is obtained. The "break point" indoor entering wet bulb temperature (BPT) corresponds to the boundary between wet coil and dry coil operation. The machine operates at the dry coil cooling capacity (DCCC) whenever EWB is less than BPT. In the wet coil region, total capacity increases and sensible capacity decreases with increasing EWB. The difference, of course, is latent cooling capacity. Note that the total cooling capacity slope (TCS) and sensible cooling capacity slope (SHCS) are independent of sink temperature. The model simply uses TCS and SHCS to obtain the total and sensible cooling capacities for the actual EWB condition. In this example, BPT also appears to be independent of TSINK. It is unclear whether this is true in general however.

If the work input from Table 3-4 is plotted versus the indoor entering wet bulb temperature (EWB), Figure 3-11 is obtained. Note that the slopes of the lines (WS) are again independent of TSINK. The model accepts W at the break point and calculates work input for the actual EWB condition using the slope WS. Carrier reports total work input on their data sheets and the cooling capacities reported include the slight degradation due to the heat input from the indoor fan. If only compressor work is reported, care must be taken to add electrical consumption of the indoor fan and sink-side fan (air-to-air) or pump (liquid-to-air). The cooling capacity provided to the model should include any degradation due to the indoor fan.

The indoor air mass flow rate (FLOWR) and entering dry bulb temperature (EDB) corresponding to the data in Table 3-4 were obtained from the text of the Carrier data sheet. The final cooling performance data format and required parameters to be input into this new heat pump model for the cooling mode are shown in Table 3-5, which corresponds to the format of Table 3-3.

Heating Mode

The modeled heat pump heating mode operation is specified in a manner similar to cooling mode. A table of performance characteristics and required parameters is needed as listed in Table 3-6. The data again should be for a specified entering indoor dry bulb temperature and indoor unit air mass flow rate.

Table 3-4

Cooling Mode Performance Data for the Carrier 38RQ034/40AQ036 Air-to-Air Heat Pump (EDB = 80 F, FLOWR = 1275 CFM)

38RQ034/40AQ036 or 40FS160 with 28MQ036										
Temp (F) Air Ent Outdoor Unit		Air Ent Indoor Unit – Cfm/BF								
		1115/.11			1275/.13			1400/.14		
		Indoor Unit Ent Air Temp – Ewb (F)								
		72	67	62	72	67	62	72	67	62
85	TC	37.4	34.7	31.4	37.8	35.2	32.2	37.8	35.4	32.7
	SHC	19.1	24.8	29.4	19.8	26.4	31.1	20.2	27.3	32.2
	KW	4.13	3.99	3.84	4.22	4.09	3.95	4.27	4.15	4.03
95	TC	35.6	33.0	30.0	35.7	33.5	30.8	35.6	33.5	31.4
	SHC	18.4	24.1	28.6	19.0	25.5	30.1	19.3	26.2	31.1
	KW	4.34	4.21	4.06	4.42	4.30	4.17	4.47	4.37	4.25
100	TC	34.7	32.1	29.2	34.9	32.5	30.0	34.9	32.6	30.6
	SHC	18.1	23.8	28.0	18.7	25.1	29.5	19.1	26.0	30.4
	KW	4.46	4.32	4.16	4.55	4.42	4.28	4.60	4.48	4.37
105	TC	33.8	31.1	28.3	34.2	31.5	29.1	34.1	31.6	29.7
	SHC	17.8	23.4	27.5	18.5	24.8	28.9	18.9	25.8	29.7
	KW	4.58	4.43	4.27	4.68	4.53	4.39	4.73	4.59	4.48
115	TC	32.0	29.2	26.6	32.6	29.6	27.5	32.6	29.8	28.1
	SHC	17.3	22.6	26.4	18.1	24.1	27.5	18.6	25.3	28.1
	KW	4.82	4.65	4.48	4.93	4.75	4.61	4.99	4.82	4.71

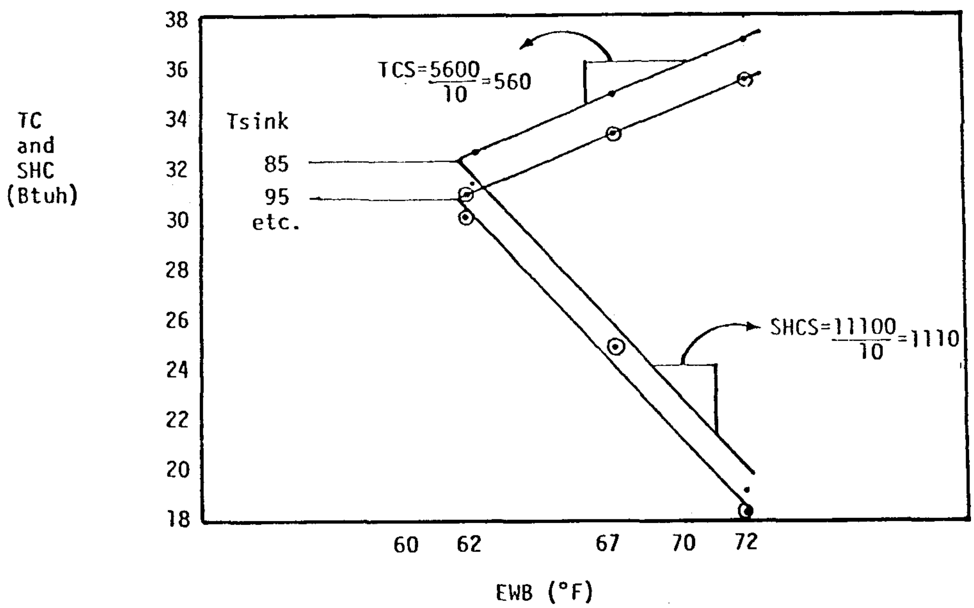


Figure 3-10. Plot of Cooling Performance Data for the Carrier 38RQ034/40AQ036 Air-to-Air Heat Pump (EDB = 80 F, FLOWR = 1275 CFM)

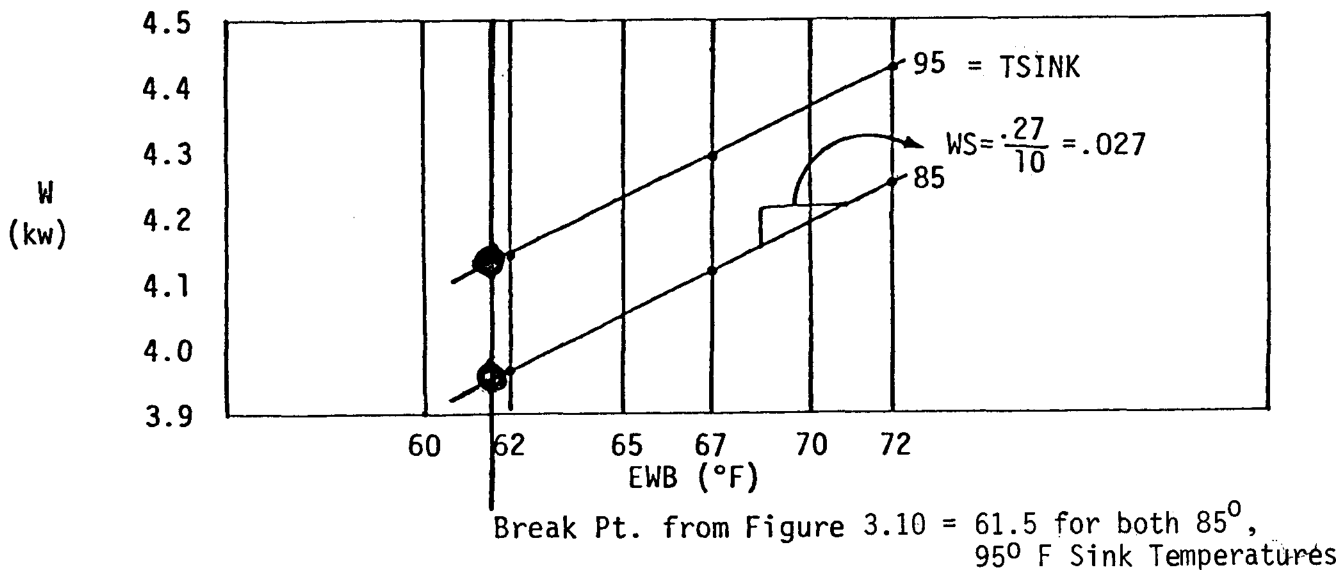


Figure 3-11. Plot of Cooling Mode Total Work Input for the Carrier 38RQ034/40AQ036 Air-to-Air Heat Pump (EDB = 80 F, FLOWR = 1275 CFM)

Table 3-5

Example Cooling Performance Data and
Required Parameters (Carrier Air-to-Air)

From Figures 3-10 and 3-11:

TSINK	DCCC	BPT	W
85	32000	61.5	3.94
95	30800	61.5	4.15
.	.	.	.
.	.	.	.
$TSINK_n$	$DCCC_n$	BPT_n	W_n

From the text of manufacturer's literature, and from Figures 3-10 and 3-11:

TCS	=	560 Btuh/F	=	1063. KJ/hr-C
SHCS	=	1110 Btuh/F	=	2108. KJ/hr-C
WS	=	.027 KW/F	=	175 KJ/hr-C
FLOWR	=	1275 CFM	=	2602 Kg/hr
EDB	=	80^{o}F	=	26.7 ^{o}C

Table 3-6

Heating Mode Heat Pump Performance Data
Format and Associated Parameters Required

Performance Data Format

TS_1	HC_1	W_1
.	.	.
.	.	.
.	.	.
TS_n	HC_n	W_n

where

TS = source (outdoor or tank) dry bulb temperature (C)

HC = dry coil (no frost) heating capacity (KJ/hr)

W = work input (compressor, indoor and outdoor fans)(KJ/hr)

Required Parameters

Air-to-air or liquid-to-air heat pumps:

FLOWR = indoor air mass flow rate corresponding to performance data table (Kg/hr)

EDB = indoor entering dry bulb temperature corresponding to performance data table (C)

THPMIN = minimum source temperature for heat pump operation (should equal TS_1) (C)

THPMAX = maximum source temperature for heat pump operation (should equal TS_n) (C)

Liquid-to-air heat pumps only:

FLOWS = source liquid mass flow rate (Kg/hr)

CPS = source fluid specific heat (KJ/kg-C)

To use manufacturers' data, simply plot heating capacity as a function of source temperature as shown in Figure 3-12. For air-to-air heat pumps, the reported performance within the defrost region (17-47^{o}F) is an average integrated performance which includes the effect of the defrost cycle. In this case, Carrier reports average integrated performance measurements taken with an 85 percent relative humidity (RH) condition for air entering the outdoor coil. Other manufacturers report defrost region data taken at other RH conditions. To avoid confusion and ensure uniform treatment of defrost degradation for all manufacturers, the model accepts dry coil (no frost) heating capacity data. The dry coil capacity is represented on Figure 3-12 by the dotted line-drawn through the defrost region with a smooth curve. For air-to-air heat pumps, the model will then modify dry coil capacity in the defrost region using an expression suggested by Carrier (42) and representative of a 90-minute cycle, time/temperature controlled defrost.

3.3.2 Transient Operation

The performance characteristics described in the previous subsection are for steady-state operation. The model has the capability to modify capacity for start-up/shut-down transients if measurements of start-up and shut-down time constants are available. Table 3-7 summarizes the time constant measurements available in the literature for residential heat pumps (43). The model assumes the heat pump is on or off each timestep so that the length of an on-cycle is an integer multiple of the simulation timestep.

During start-up, it is assumed that capacity exponentially approaches the steady-state value as shown in Figure 3-13. As the start-up time constants of Table 3-7 are small compared to the simulation timestep (7.5 - 15 minutes), it is assumed that the transient only affects the first timestep of an on-cycle. This would avoid numerical complications if the steady-state capacity were to change from one timestep to the next due to a change in source or sink temperature. The energy transferred to the load during start-up equals the integral of the exponential curve over the timestep.

The model assumes indoor autofan operation so that shut-down transient capacity is lost. Although this capacity could be salvaged with continuous fan operation, Parken, et al. (44), conclude that increased fan power makes this undesirable. The electrical input transient is not modeled as Groff, et al. (45), suggest and steady-state is reached very quickly.

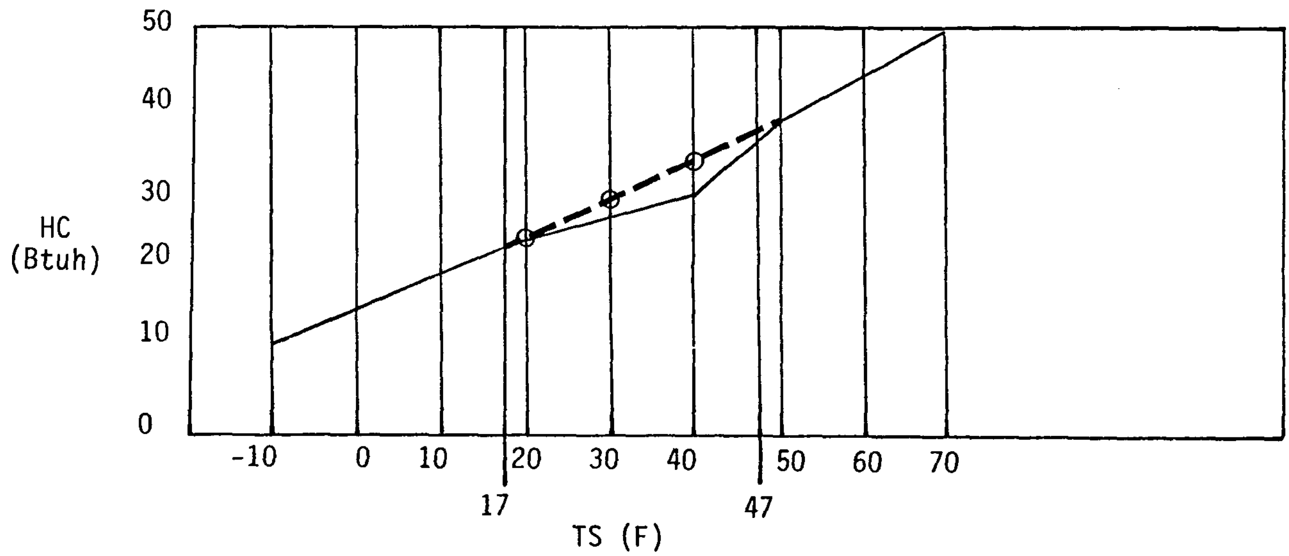
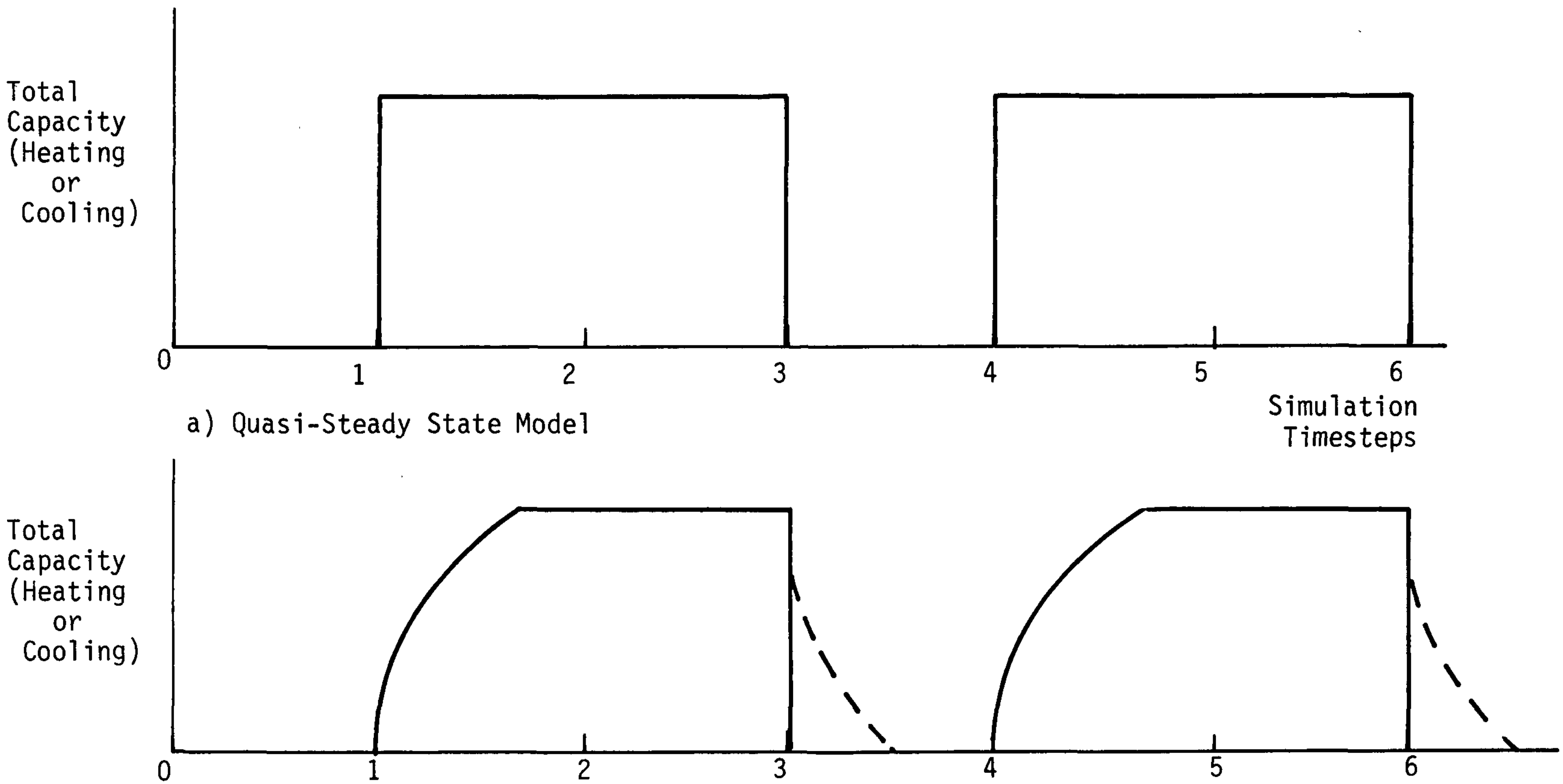


Figure 3-12. Heating Capacity vs. Source Temperature for the Carrier 38Q$034/40AQ036 Air-to-Air Heat Pump. (EDB = 70 F, FLOWR = 1275 CFM)

Table 3-7

Start-Up/Shut-Down Time Constants Minutes for Four Residential Heat Pumps (From Reference 43)

	COOLING MODE		HEATING MODE	
	τ_{on}	τ_{off}	τ_{on}	τ_{off}
AMANA CRH4-1/AFCH40	3.98	2.13	2.40	1.65
CARRIER 38CQ027/40AQ030	3.72	3.58	4.30	1.61
GE BWB936A/BWV936G	3.49	2.39	2.25	1.92
LENNOX HP8-261/CB11-41	3.84	3.26	2.71	1.90
AVERAGE	3.76	2.84	2.92	1.77



a) Quasi-Steady State Model

b) Time Constant Degraded Performance

Figure 3-13. Ideal vs. Time Constant Degraded Heat Pump Capacity

3.3.3 Systems Interface

The heat pump model described in previous section is general in the sense that performance of any air-to-air or liquid-to-air heat pump can be estimated given the operating conditions. In order to make the model convenient for this system study, however, further refinements have been added. These refinements include the addition of system controls, auxiliary strip heaters, and a mode of operation whereby the load is directly heated with source water (liquid-to-air heat pump only). The refinements allow the heat pump model to be easily interfaced with system models for stand-alone heat pump systems, liquid storage solar systems with a series-connected heat pump, and liquid- or air-based direct solar heating systems with a parallel-connected heat pump.

3.3.4 Today's and Next-Generation Heat Pump Characteristics

The heat pump system simulations were performed utilizing two different levels of heat pump technology: today's technology, and projected next-generation technology. This section presents the heat pump performance characteristics for both technology levels in the format compatible with subsection 3.3.1 on data preparation. Performance data for air-to-air and liquid-to-air heat pumps are given since the series and parallel system configurations were simulated. A separate document is being prepared to detail the choice of both today's and next-generation heat pump performance characteristics.

Today's Technology

The Air Conditioning and Refrigeration Institute (ARI) Directory of Certified Manufacturers was reviewed for manufacturers and model numbers of air-to-air heat pumps with a cooling capacity of 30,000 Btu, and information was obtained from seven manufacturers. In accord with data preparation procedures (subsection 3.3.1), the cooling and heating performance characteristics were recorded for each of the manufacturers. The air-to-air heat pump which was chosen for the system simulations is characterized by the data given in Table 3-8.

The ARI directory lists only one liquid-to-air heat pump of 30,000 Btu capacity, and contact with the manufacturer was attempted several times with no

Table 3-8

Heat Pump Performance Data:
Air-to-Air, Today's Technology

HEATING MODE

Source Temperature (^{o} F)	Dry Coil Capacity (Btu/hr)	Work (kw)
-3	7000	2.08
7	11500	2.39
17	16000	2.65
27	21000	2.79
37	25500	2.93
47	30200	3.15
57	35000	3.38
67	39600	3.48

COOLING MODE

Sink Temperature (^{o} F)	Dry Coil Cooling Capacity (Btu/hr)	Break Point Temperature (^{o} F)	Work (kw)
85	28800	61.0	3.68
95	27300	61.5	3.90
105	25900	61.8	4.10
115	24300	61.8	4.27

success. Contact was attempted with five manufacturing companies which produce 32,500 and 33,000 Btu/hr models. Although these sizes are greater than the design loads, they should be well within the design estimate uncertainties. The cooling and heating performance data were graphed and tabulated in the same manner as was the air-to-air heat pump data, and the liquid-to-air, today's technology heat pump characteristics are listed in Table 3-9.

Next-Generation Technology

The improvement of heat pump performance has been hypothesized for air-to-air heating and cooling and liquid-to-air heating and cooling. In all but the liquid-to-air heating mode, fixed percentage improvements were assumed for all source temperatures. A 35 percent improvement in COP was assumed for air-to-air heating and a 25 percent reduction in work was assumed for both liquid-to-air and air-to-air cooling modes. The resulting characteristics for air-to-air and liquid-to-air future heat pumps are presented in Tables 3-10 and 3-11, respectively.

It was assumed that the improvement in performance of the liquid-to-air heating mode would not be so simply tied to the improvement of current components as it was for the other modes and heat pumps described above. Instead, the performance for two-speed machines operating at higher source temperatures was predicted (see Table 3-11). This was done by calculating theoretical work requirements for each speed and adding a constant inefficient work. The details of this analysis are being prepared as a separate document.

3.4 CONTROLS AND OPERATING MODES

This subsection describes in detail how each system was controlled during the simulation. Locations of sensors, control logic, and the rationale for choosing these controls are discussed.

3.4.1 Rationale for Choosing Controls

As discussed later in the section on thermal comfort (subsection 5.3), systems must provide the same degree of comfort before they can be directly compared. It has been established that each system will maintain similar comfort conditions if it is subject to the same control restraints. These restraints are:

Table 3-9

Heat Pump Performance Data:
Liquid-to-Air, Today's Technology

HEATING MODE

Source Temperature (^oF)	Dry Coil Capacity (Btu/hr)	Work (KW)
58	39500	4.03
63	41519	4.15
68	43980	4.41
73	46315	4.75
78	47703	4.94
83	48649	5.11
88	49785	5.31
93	50858	5.56

COOLING MODE

Sink Temperature (^oF)	Dry Coil Cooling Capacity (Btu/hr)	Break Point Temperature (^oF)	Work (KW)
72	39752	58.5	4.19
77	38642	58.5	4.31
82	37228	58.5	4.42
87	35966	58.5	4.56
92	35588	58.5	4.70
97	33064	58.5	4.84

Table 3-10

Heat Pump Performance Data:
Air-to-Air, Future Technology

HEATING MODE (2-SPEED COMPRESSOR)

Source Temperature (°F)	Dry Coil Capacity (Btu/hr)	Work (KW)
-3	23000	4.93
7	27750	5.14
17	32500	5.35
27	37200	5.56
37	25500	2.17
47	30200	2.33
57	35000	2.50
67	39600	2.58

COOLING MODE

Sink Temperature (°F)	Dry Coil Cooling Capacity (Btu/hr)	Break Point Temperature (°F)	Work (KW)
85	28800	61.0	2.72
95	27300	61.5	2.89
105	25900	61.8	3.03
115	24300	61.8	3.16

Table 3-11

Heat Pump Performance Data:
Liquid-to-Air, Future Technology

HEATING MODE (2-SPEED COMPRESSOR)		
Source Temperature (^oF)	Dry Coil Capacity (Btu/hr)	Work (KW)
35	39500	3.82
45	49374	4.10
55	60347	4.32
65	71319	4.55
75	82290	4.78
85	49374	2.96
95	60347	3.02
105	71319	3.08

COOLING MODE			
Sink Temperature (^oF)	Dry Coil Cooling Capacity (Btu/hr)	Break Point Temperature (^oF)	Work (KW)
72	39752	58.5	3.11
77	38642	58.5	3.18
82	37228	58.5	3.27
87	35966	58.5	3.38
92	35588	58.5	3.49
97	33064	58.5	3.58

- Each system must deliver energy to the conditioned space at a temperature higher than a specified minimum (during heating mode)
- Each system operates with the same load flow rate
- Each system is controlled by thermostats with the same set points and deadbands.

This being the case, each system will provide roughly the same comfort level. Differences will occur because of two factors: some systems will deliver air to the load above the minimum temperature more often than others; and some of the systems respond differently to thermostat control. For instance, solar systems with parallel heat pumps are idle during first-stage heating unless the solar system can deliver energy at or above the minimum temperature. The conventional furnace, however, always responds to first-stage heating.

3.4.2 Individual Systems' Operating Modes

The system models have been formulated so that each can be controlled to deliver energy to the load at or above a specified minimum temperature. This allows the systems to be compared on a uniform basis since each is providing comparable thermal comfort. In order to accomplish this uniformity, however, some of the control strategies differ from those now commonly used in practice. The stand-alone heat pump, for instance, is normally allowed to run alone during first-stage heating regardless of delivered air temperature. Wherever possible, standard controls have been used.

Uniformity in one respect can lead to nonuniformity in another. For instance, in the two parallel systems, the two stages of the heating thermostat are used to give solar energy first priority. But, if solar cannot supply energy at a high enough temperature, the systems remain idle during first-stage heating. Although the stand-alone heat pump and series systems can also remain idle during first stage, the frequency of the occurrence is much less. Consequently, the parallel systems do not achieve the same level of room temperature control.

The control systems described below are associated with the following system types:

- Stand-Alone Heat Pump System
- Liquid Series System
- Air Series System
- Liquid Parallel System
- Air Parallel System
- Conventional Furnaces with Central Air Conditioning.

Each of these systems is controlled differently and the descriptions are aided by system schematics showing the fluid flow paths associated with a particular control scheme.

3.4.3 Stand-Alone Heat Pump System

Heating Mode - Conventional Heat Pump Controls

In the heating mode, the stand-alone heat pump/load interface is controlled by a two-stage thermostat. First stage actuates the indoor and outdoor fans and the compressor, unless the source temperature is outside of the allowable range. In this event, the heat pump remains idle and no response is made to the thermostat.

Second stage also actuates the heat pump if the source temperature is in the allowable range. In addition, auxiliary strip heaters are staged-in to obtain a specified outlet temperature. The maximum energy input from the auxiliary heaters cannot exceed a specified maximum. It is assumed that the modeler has chosen the outlet temperature, auxiliary capacity, and the room air flow rate such that the design heating load can be met. Set points and deadbands are adjustable on both stages of the thermostat.

Heating Mode - Controlled Delivered Air Temperature

Preliminary system studies with the conventional heat pump controls indicated that delivered air temperature was often too low during first-stage heating. This occurred under low source (ambient) temperature conditions. To force the heat pump to meet the same comfort criteria as the other systems, an option was included to cause the heat pump to respond to both first- and second-stage heating with the second-stage algorithm described above.

Cooling Mode

In the cooling mode, the heat pump/load interface is controlled by the cooling thermostat. Control is open ended in that room temperature will float if the machine is undersized, and room humidity is not explicitly controlled.

3.4.4 Liquid Series System

Heating Mode - Conventional Controls

Again, the heat pump/load interface is controlled by a two-stage thermostat. In the first stage, direct heating from the liquid source occurs if the liquid source temperature is greater than a specified minimum. Otherwise, the heat pump is actuated unless the source temperature is outside the allowable range, in which case the system remains idle.

In second-stage heating, direct heating from the liquid source occurs if the liquid source temperature is greater than the minimum. Otherwise, the heat pump is actuated unless the source temperature is outside of the allowable range. In addition to the above, auxiliary strip heaters are staged in to maintain a minimum outlet air temperature. The maximum energy input from the auxiliary heaters cannot exceed a specified maximum.

Heating Mode - Controlled Delivered Air Temperature

Preliminary system studies with the conventional liquid series system controls indicated that, at low source (water tank) temperatures, delivered air from the series heat pump was too cool. One approach to solving the problem is to use multi-speed compressor heating mode performance data. In the limit of infinitely variable speed, heat pump capacity can exactly match that required to maintain a constant delivered air temperature regardless of source temperature.

A more practical approach is to design the series heat pump with, perhaps, two compressor speeds and to add auxiliary as necessary to maintain a minimum delivered air temperature in first-stage as well as second-stage heating. An option has been included in the heat pump model for this purpose. It causes the heat pump to respond to both first- and second-stage heating with second-stage algorithm described above.

Cooling Mode

Cooling mode operation is nearly identical to that for the stand-alone heat pump. The difference is that heat is rejected to the storage tank rather than to ambient air. Energy from the liquid tank must be further rejected to ambient air via a crossflow heat exchanger. The ambient rejection loop control is shown in Figure 3-14. Rejection occurs whenever tank temperature exceeds ambient by a specified deadband. No attempt was made to limit this operation to off-peak hours or to hours when compressor is idle.

Collector Loop Control

Control of the collector loop, shown in Figure 3-15, is independent of the rest of the system. A differential controller compares the collector outlet and tank temperatures, initiating fluid flow whenever the temperature difference exceeds the deadband.

Solar Domestic Water Heating Control

The domestic water heating portion of the series systems is completely separate from the space conditioning system, both physically and in its control. It consists of a preheat tank which supplies solar preheated water to a conventional water heater. The conventional water heater supplies water to the load at the required temperature. A mixing valve between the cold and hot water lines prevents the delivery of over-heated water. The domestic water heating collector loop operates whenever the temperature rise through the collector exceeds a specified deadband. This control is shown in Figure 3-16.

Summer/Winter Switchover

In order to run a continuous, 12-month simulation of the liquid series system, it is necessary to decide when system operation should switch from summer to winter. In summer operation, the space heating collector array is disabled and the rejection loop is allowed to run. In winter, the reverse is true. Heat pump operation is unaffected by season. It is controlled by a two-stage heating thermostat and one-stage cooling thermostat year round. The domestic water subsystem is also unaffected by season. For lack of a better alternative, the spring and fall switchover dates which yielded maximum solar contribution were determined through trial and error.

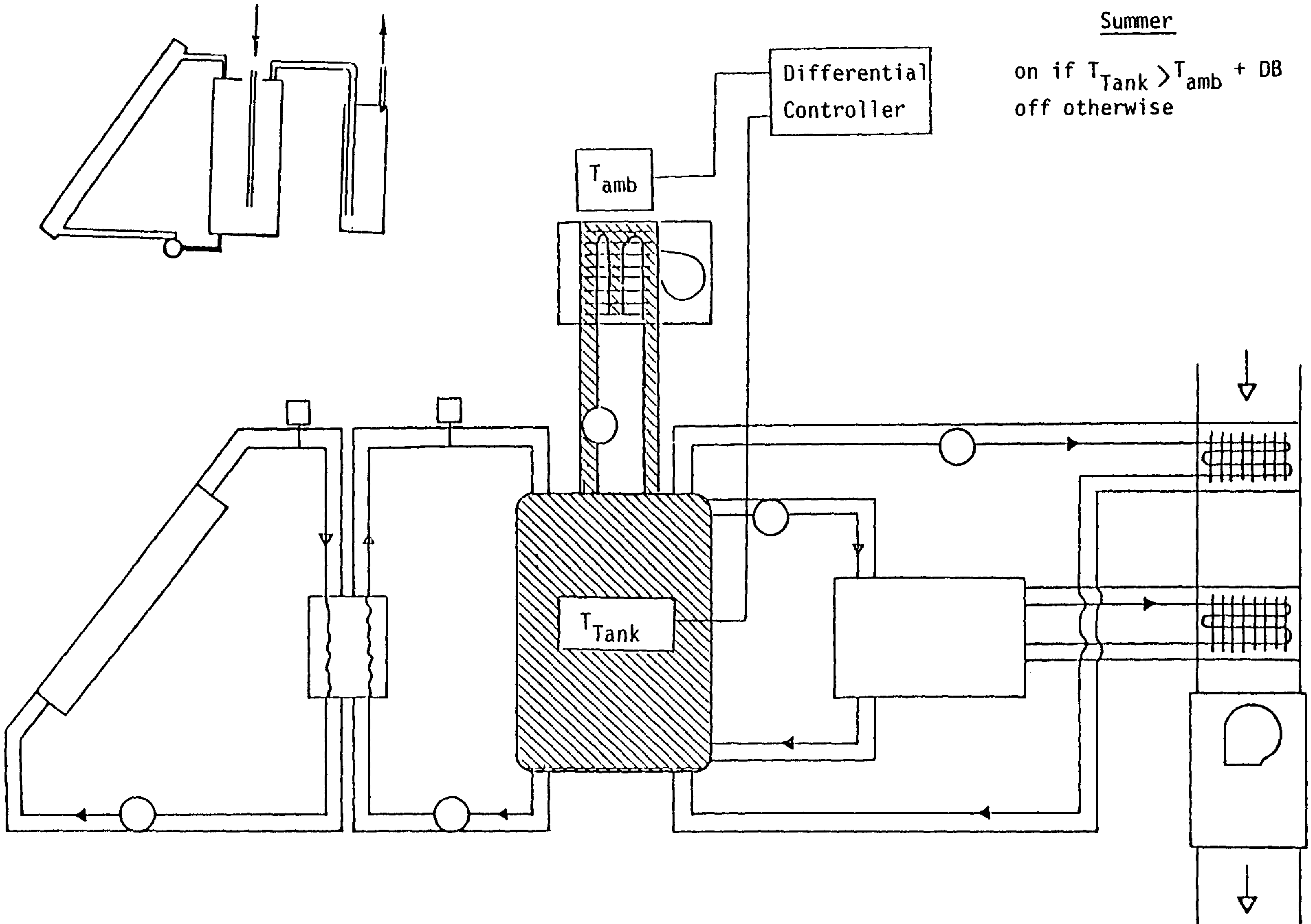


Figure 3-14. Liquid Series System: Control of Rejection Loop

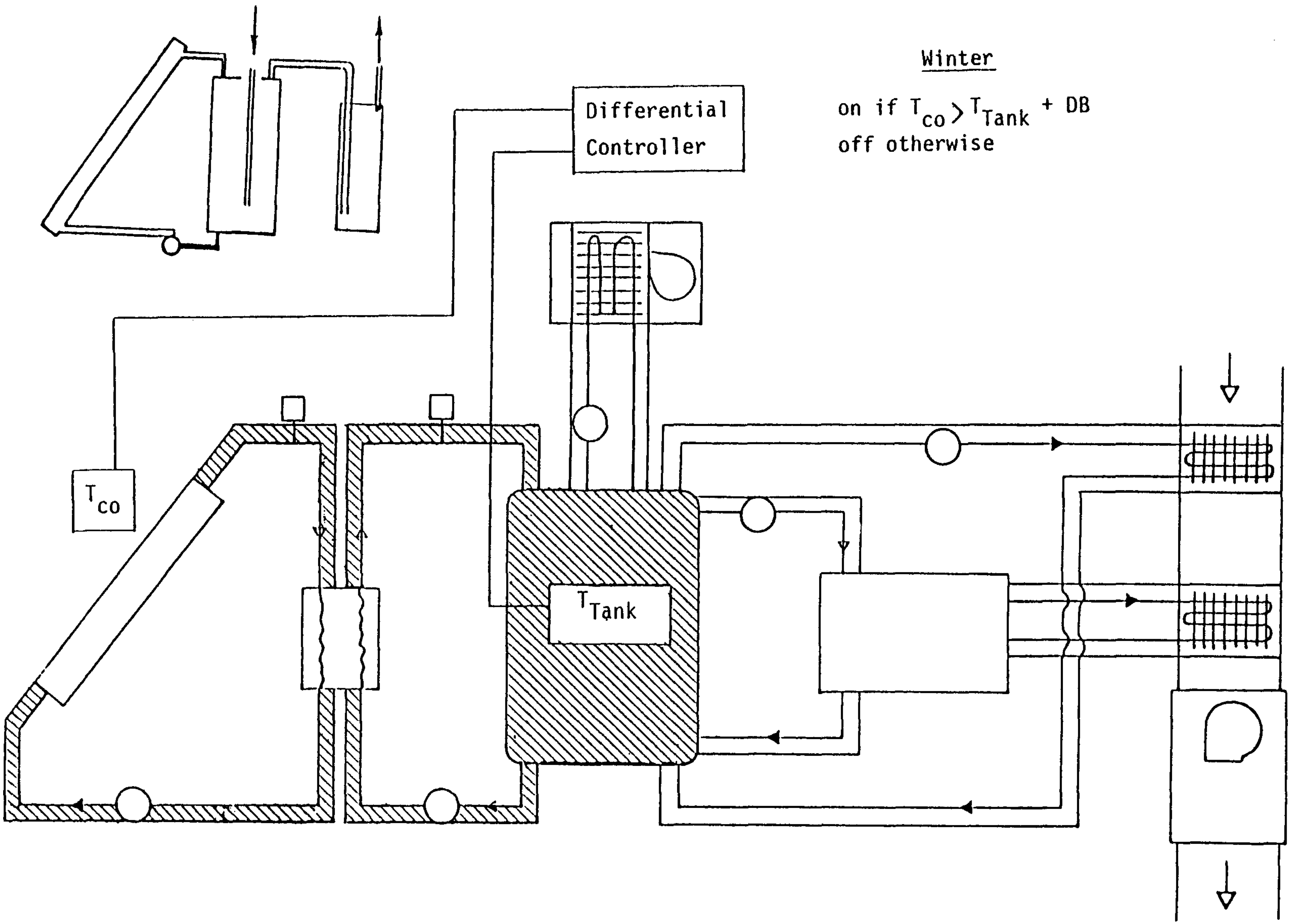


Figure 3-15. Liquid Series System: Collector Loop Control

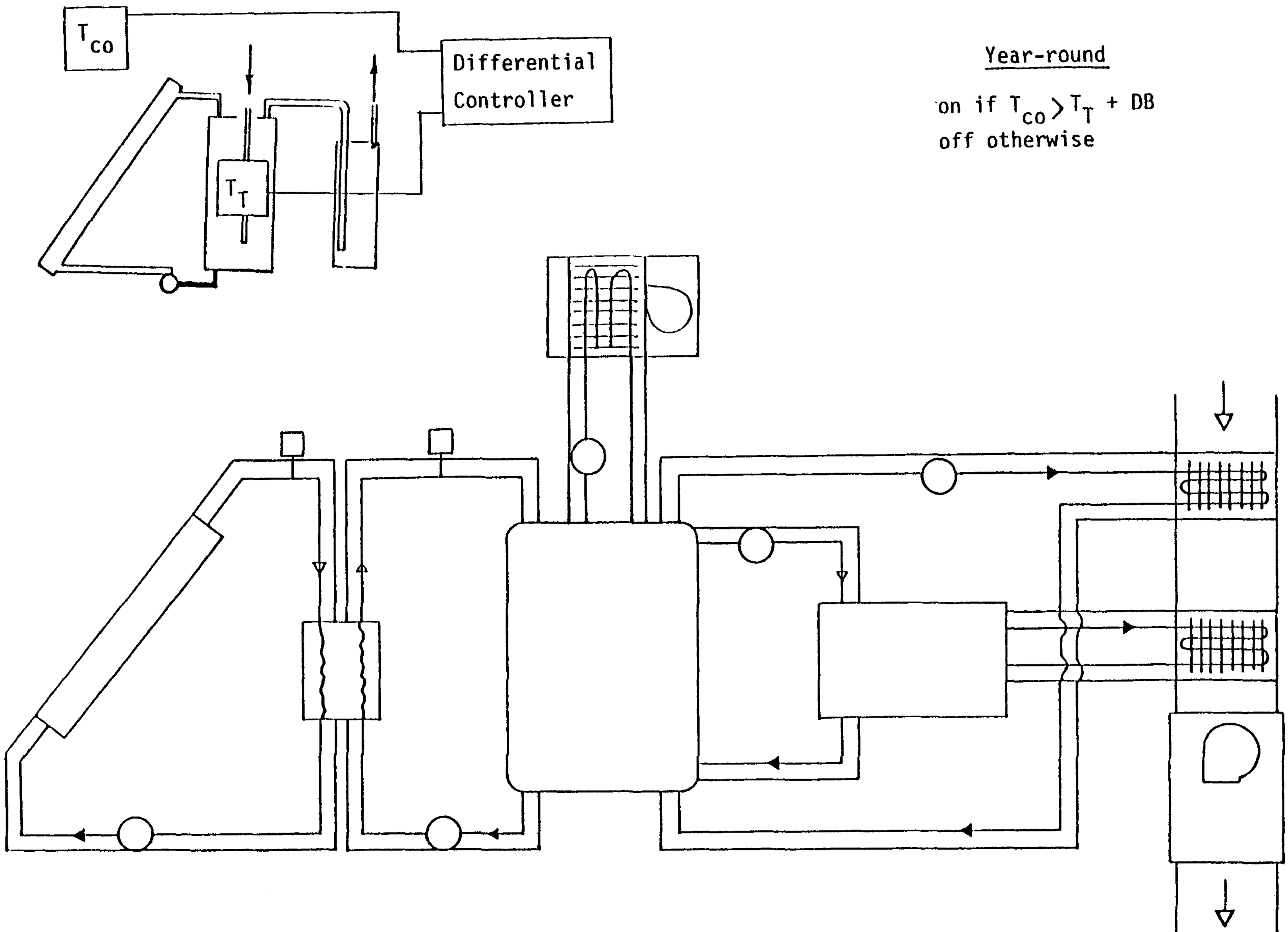


Figure 3-16. Liquid Series System: Domestic Water Subsystem Control

Air Series System

Control of the air series system is identical to that for the liquid series system in all respects. Physically, the only difference between the two systems is in the design of the collector and rejection loops. Air heating collectors are used and energy is transferred to the liquid storage tank via a crossflow heat exchanger. As shown in Figures 3-17 and 3-18, the same heat exchanger can be used for heat rejection by incorporating two air dampers into the design.

3.4.5 Liquid Parallel System

Heating Mode

The heating system/load interface is controlled by a two-stage thermostat. Since solar energy is the preferred heat source, the heat pump is idle in first stage and air flow to the building is initiated only if the solar system cannot provide energy at a temperature above a specified minimum; otherwise, the system remains idle. In second stage, the solar system is idle while the heat pump operates, unless the source (ambient) temperature is outside of the allowable range. If necessary, auxiliary heat is staged in to ensure that the delivered air temperature equals the specified minimum.

Cooling Mode

Cooling mode operation is identical to that for the stand-alone heat pump.

Collector Loop

Collector loop control is identical to that for the liquid series system.

Solar Domestic Water Heating

The domestic water heating system consists of a preheat tank which supplies solar preheated water to a conventional water heater. The conventional

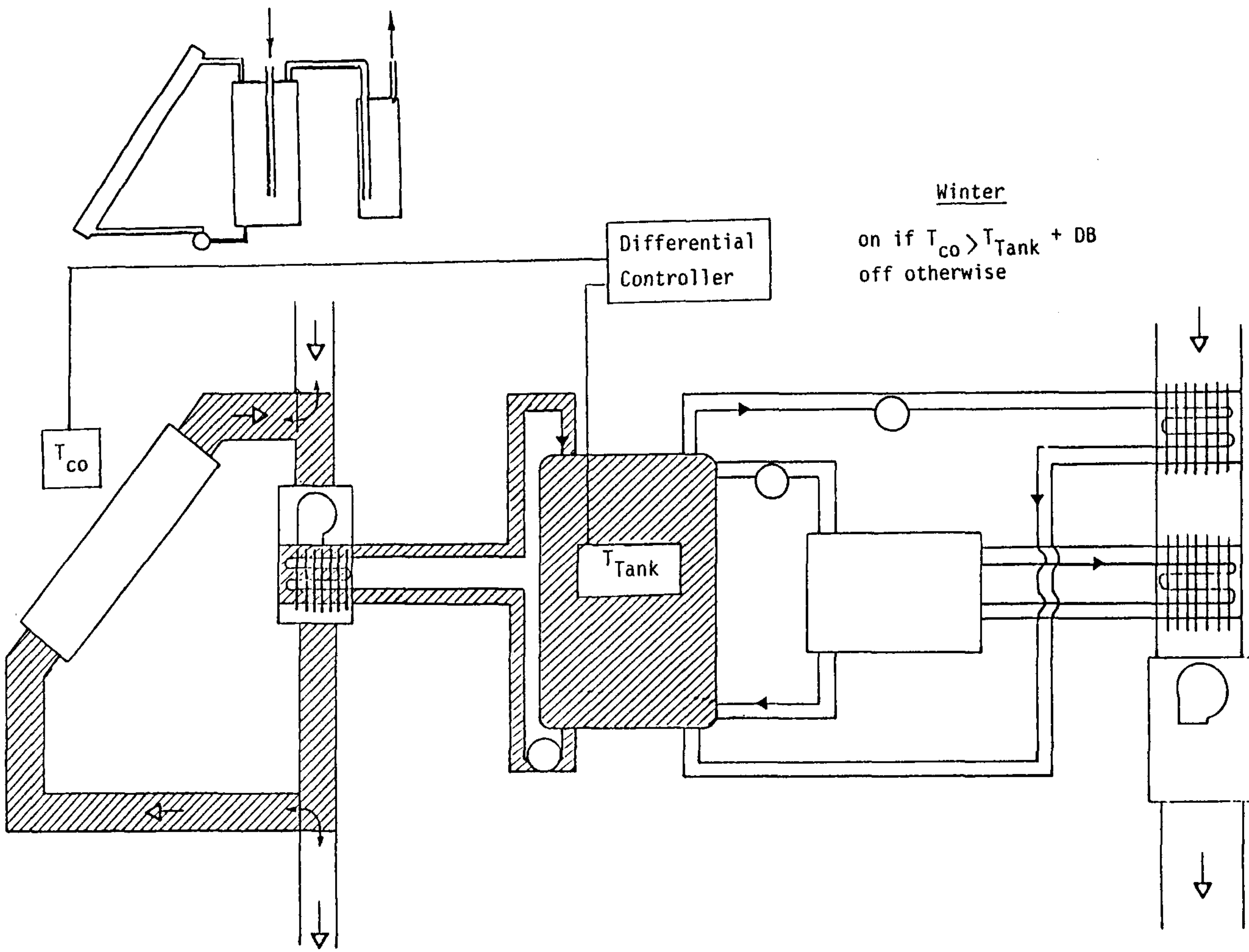


Figure 3-17. Air Series System: Collector Loop Control

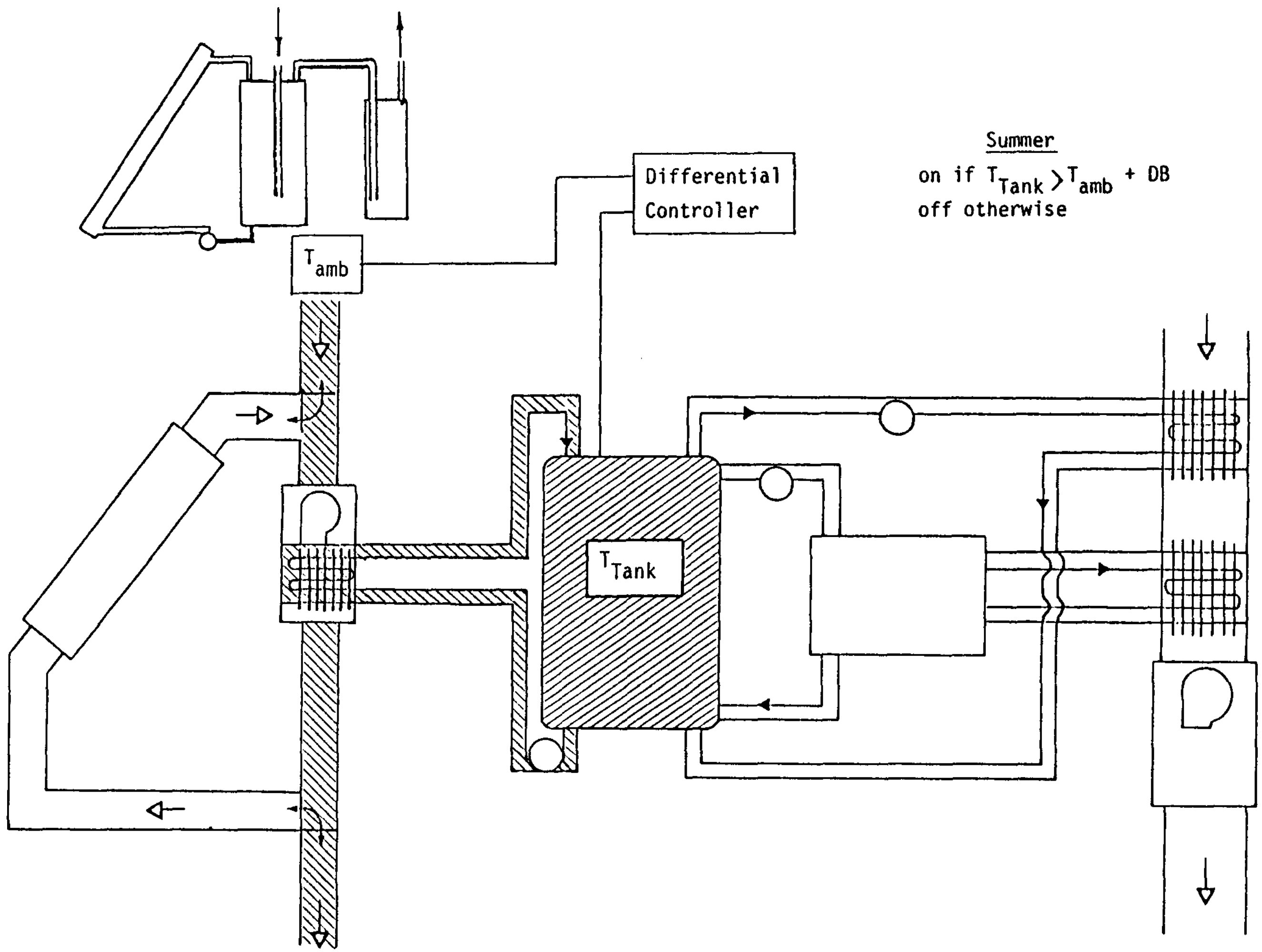


Figure 3-18. Air Series System: Rejection Loop Control

heater supplies water to the load at the required temperature. A mixing valve between the cold and hot water lines prevents the delivery of overheated water.

The preheat tank obtains energy from the main storage tank as shown in Figure 3-19. Whenever the main tank temperature exceeds the preheat tank temperature by a specified deadband, the water heating loop is actuated.

Summer/Winter Switchover

The liquid parallel system operates with the same control year round. The collector array continues to operate in the summer, heating the main tank which, in turn, supplies energy to the domestic water subsystem.

3.4.6 Air Parallel System

Heating Mode

The heating system/load interface is controlled by a two-stage thermostat. Since solar energy is the preferred heat source, first stage draws air through the solar system if it is delivered at a temperature greater than the minimum. The path may be through the collectors or through storage depending on the state of the collector controller; these two cases are illustrated in Figures 3-20 and 3-21. In Figure 3-20, the collector is off, causing air to be drawn through storage. If the collector outlet temperature is greater than the inlet by a specified deadband, flow is through the collector as shown in Figure 3-21. If the temperature of air delivered by the solar system is less than the minimum, the load air-flow rate remains zero and no response is made to first-stage heating.

Whenever air circulates directly from the collector to the load (Figure 3-21), two fans operate simultaneously in the flow loop and the modeler specifies the flow rate of each fan independently. If the flow rates differ, the model assumes that the differential flow rate passes through the pebble bed (downward if collector flow rate is higher, upward otherwise). This approximates actual operation.

During second-stage heating, the solar system is bypassed as shown in Figure 3-22. The collector may or may not run, depending on the state of the collector controller, while the heat pump will operate if the source (ambient) temperature is within the required limits. In addition, auxiliary heat is

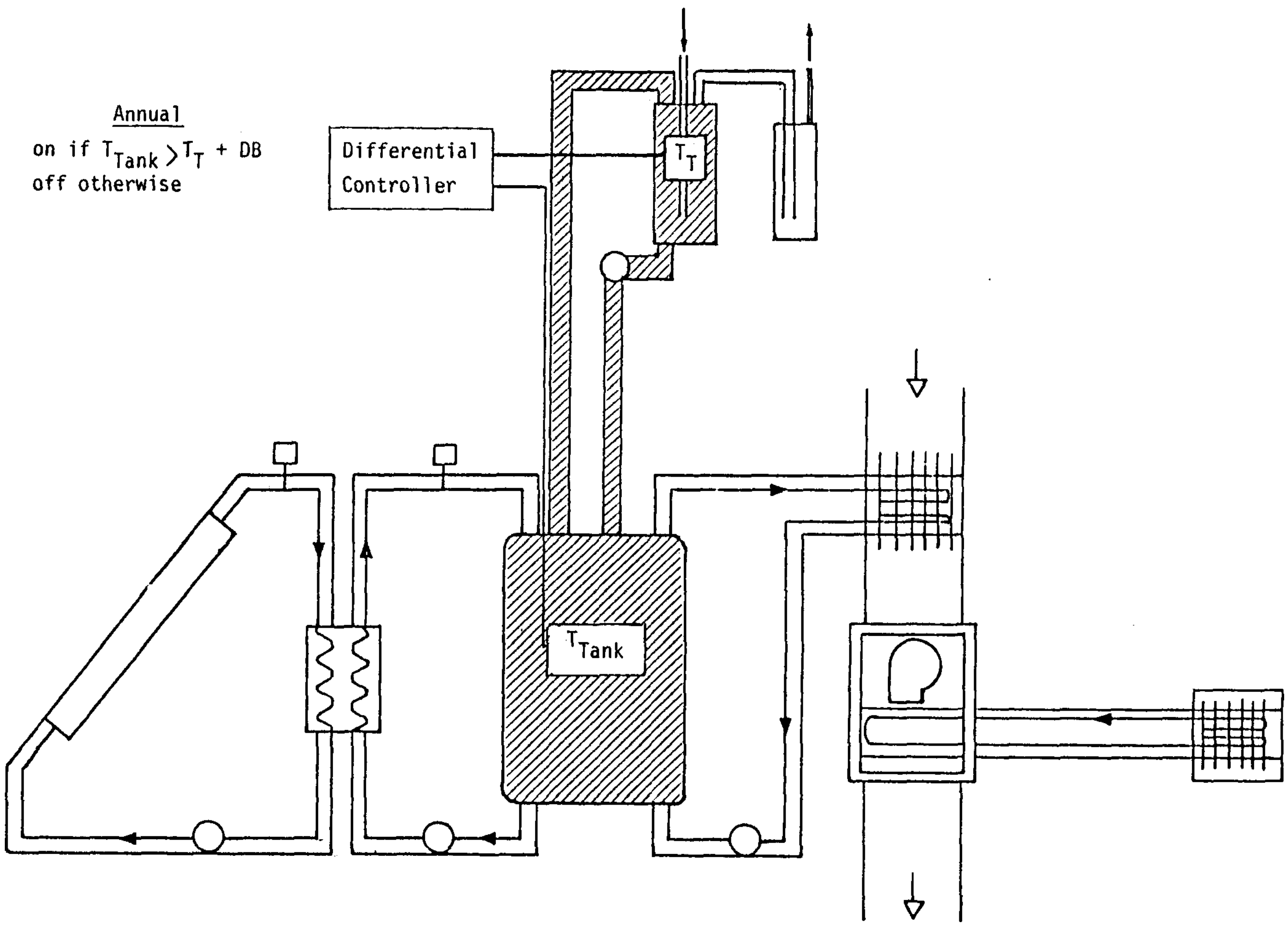


Figure 3-19. Liquid Parallel System: Domestic Water Heating Control

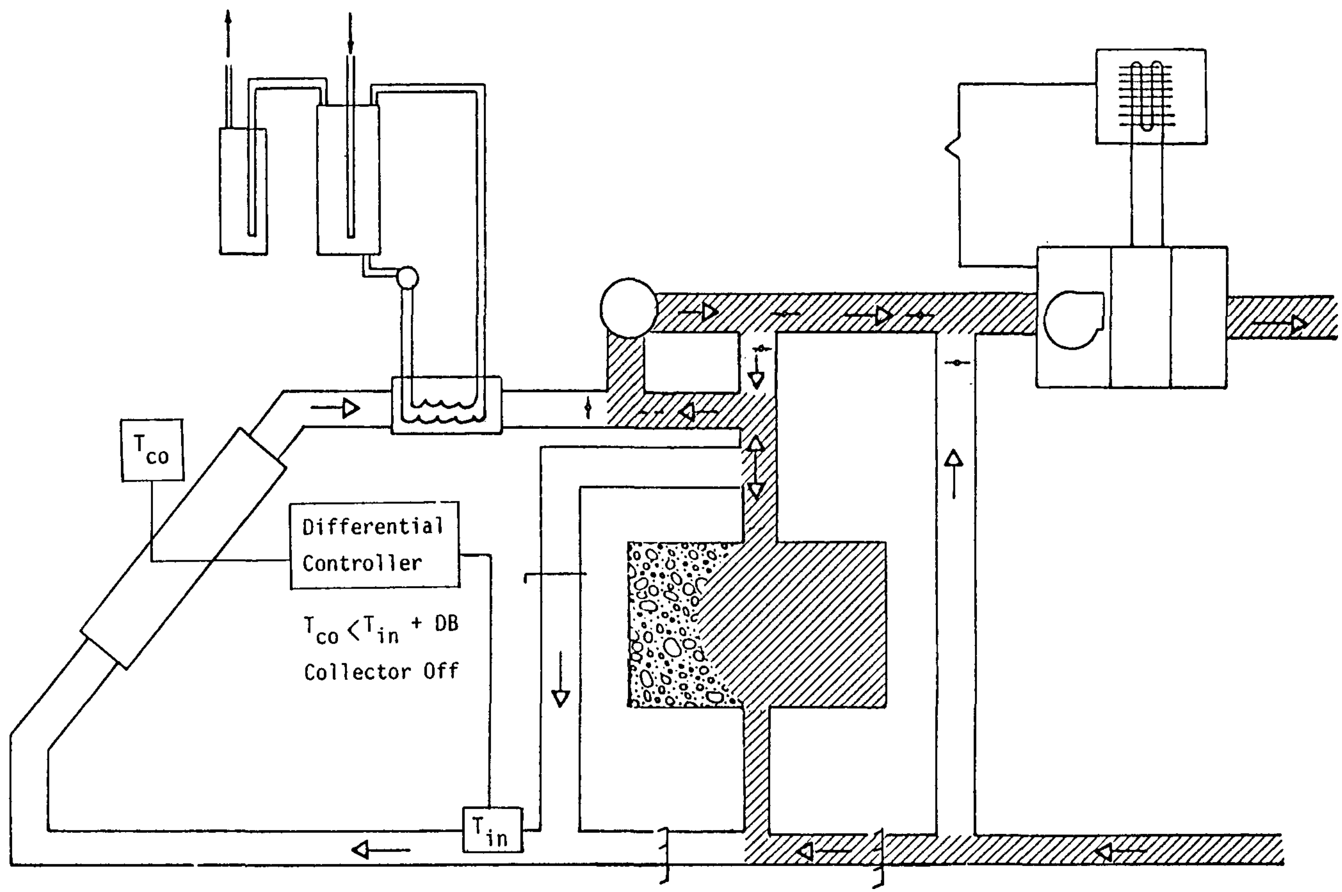


Figure 3-20. Air Parallel System: 1st Stage Heating, Collector Off

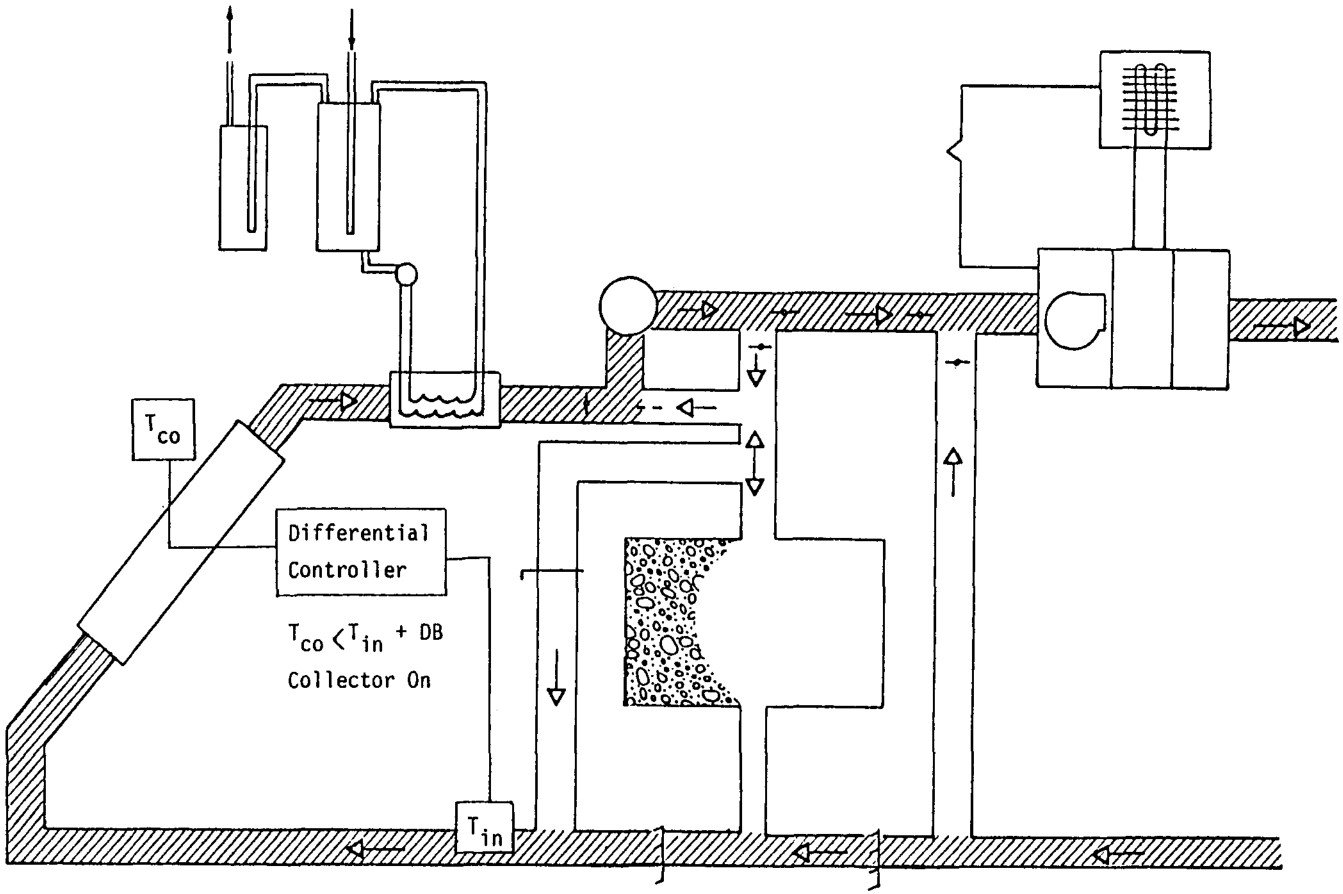


Figure 3-21. Air Parallel System: 1st Stage Heating, Collector On

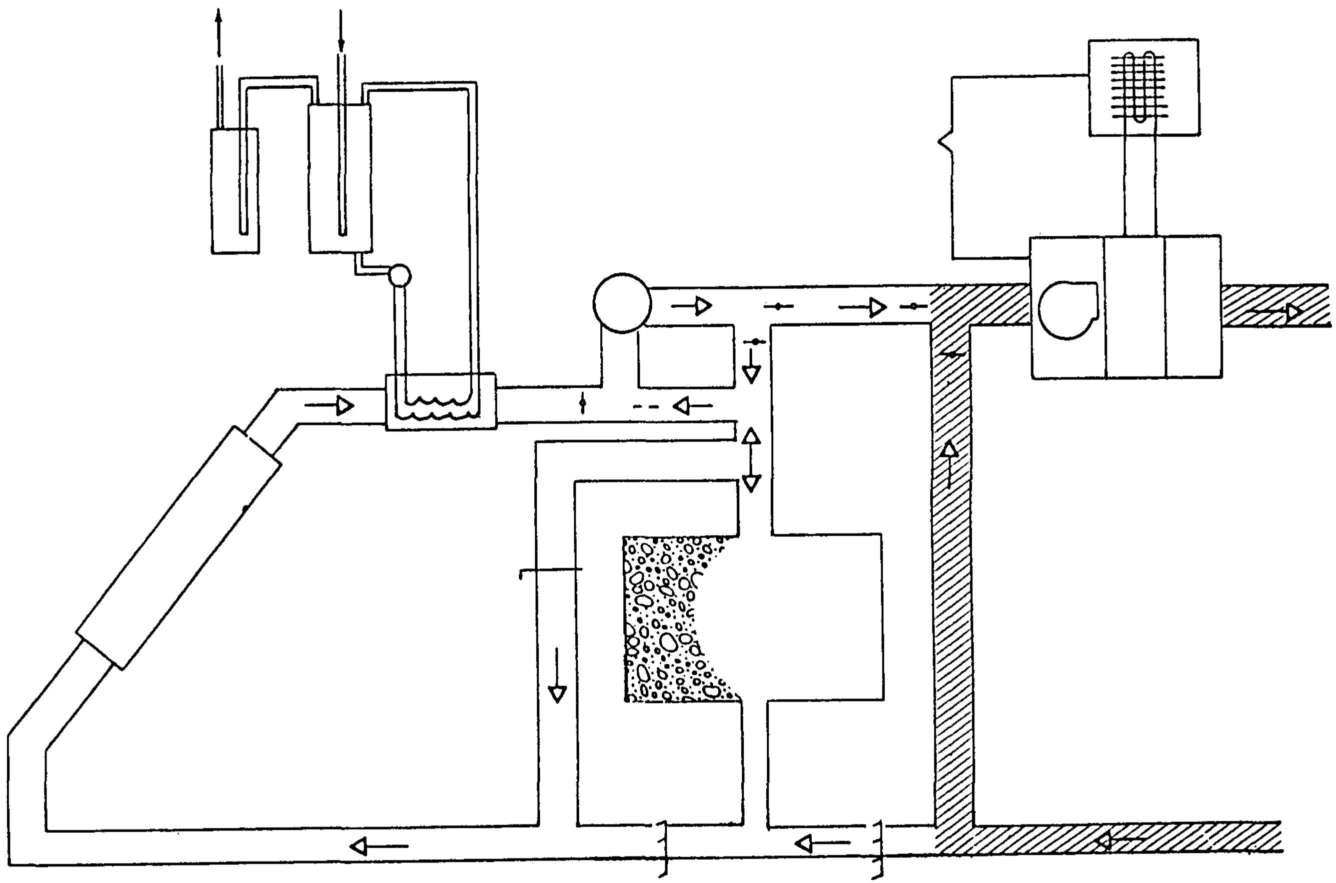


Figure 3-22. Air Parallel System: 2nd Stage Heating

staged-in, if necessary, to ensure that the delivered air temperature is equal to the required minimum.

Cooling Mode

Cooling mode operation is identical to that for the stand-alone heat pump. Air flow is the same as that shown in Figure 3-22.

Solar Domestic Water Heating

Energy is transferred to the domestic water heating subsystem via a crossflow heat exchanger at the collector outlet. During the heating season, energy can be transferred only if the collectors happen to be operating for some other purpose (heating storage or the load). The water loop between the preheat tank and crossflow heat exchanger is on whenever the collector is on, as long as the collector outlet temperature is greater than the preheat tank temperature.

During the cooling season, a manual damper is turned so that collector flow always bypasses storage and the load, resulting in a closed loop between the collectors and crossflow heat exchanger. This configuration is shown in Figure 3-23. The collector and water loops are on whenever the collector outlet temperature exceeds the preheat tank temperature by a specified deadband.

3.4.7 Conventional Furnaces with Central Air Conditioning

Heating Mode

All of the conventional systems (gas, oil, electric) operate in a simple on/off mode in response to a single-stage thermostat.

Cooling Mode

Cooling is provided by a split system, central air conditioner which cycles on or off in response to a single-stage cooling thermostat.

Domestic Water Heating

The water heating load is met with a conventional water heater which maintains a volume of water at the required temperature and supplies it to the load upon demand.

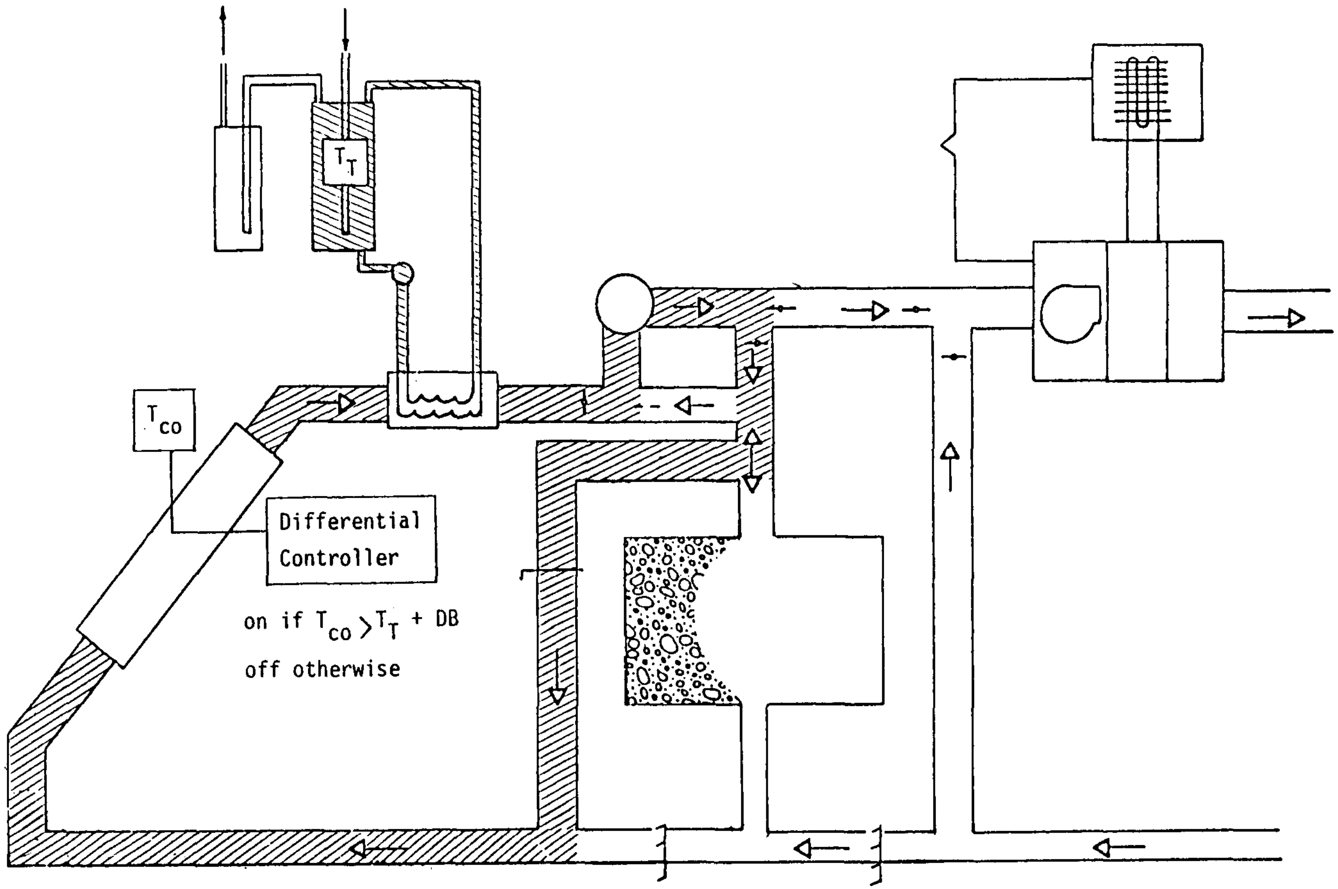


Figure 3-23. Air Parallel System: Summertime Domestic Water Heating

3.5 SYSTEM LEVEL SIMULATION PARAMETERS

The simulation model parameters used in this study are listed in Appendix D. Care was taken to use the thermal characteristics which corresponded with the actual hardware as priced in Appendix E. Only the collector parameters will be further justified here.

Rules of thumb for choosing the quality of collector required for series systems are not yet available. The work of Andrews, et al. (3), at Brookhaven National Laboratory is the only recent paper found which addresses the issue. For one location (Washington, D.C.), one storage size (400 ft^3 water) and one space heating load (UA = 750 KJ/hr-C) graph were constructed, illustrating that, beyond a certain quality, improved collector performance did not improve overall system performance. Collector characteristics for both the liquid and air series systems were chosen from these graphs and used in Fort Worth, Texas, and Madison, Wisconsin, as well as Washington, D.C.

The liquid series collector had the following recommended characteristics: $F'\tau\alpha = .75$, $\tau\alpha/U_L = 0.10$ C - m^2/W. The analysis by Andrews assumed a drain-down system with no heat exchanger between collectors and storage. The above characteristics were improved using the method of deWinter (67) so that, with a heat exchanger (effectiveness = 0.7) and ethylene glycol solution in the collectors, the rest of the system would experience the same thermal input as it did with no heat exchanger and the original characteristics. The modified characteristics are: $\tau\alpha = .88$, $F'\tau\alpha = .90$, $U_L = 8.52$ W/m^2-C. The costs itemized in Appendix E are for this improved collector.

The air series collector had the following recommended characteristics: $F'\tau\alpha = .75$ and $\tau\alpha/U_L = 0.12$ C-m^2/W. In this case, the analysis by Andrews corresponded to our collector arrangement and no further modifications were necessary.

The parallel system flat plate collector characteristics were chosen by simply using those of commercially available products which are representative of the air and liquid generic types. The costs in Appendix E correspond to the particular collectors chosen. The air parallel system collector characteristics were: $F_R\tau\alpha = .522$ and $F_RU_L = 17.3$ KJ/hr-m^2-C. The liquid parallel system collector had $F_R\tau\alpha = .70$ and $F_RU_L = 15.9$ KJ/hr-m^2-C.

4.0 ECONOMICS

The economic analysis of the various systems is based on a life-cycle cost evaluation. The relevant costs incurred by the system through acquisition and operation over the period of analysis are referenced to present worth values. The assumptions used for establishing the economic parameters are discussed and fixed for the system analyses. In subsection 4.2.2, the fixed and variable costs for each system are presented.

4.1 LIFE-CYCLE COST FORMULATION

The life-cycle cost formulation is approached in as general a manner as possible. The formulation is to be used for both residential and commercial investors; thus, the general formulation is reduced to a specific case (residential or commercial) for a given analysis.

Two approaches exist to life-cycle cost comparisons between alternative systems: the total cost of each alternative, or the cost differences between alternatives. The total cost approach is the simplest to formulate, but the determination of all the system costs is often a difficult task. The difference-in-costs approach involves determining only those costs which are not common between the compared systems. The difference approach usually has fewer items which must be cost analyzed, thus allowing a simpler statement and formulation. However, total costs are usually preferable for historical data files to be used in other analyses. Additionally, total costs are sometimes the only approach if the systems have no common cost elements.

The approach taken herein allows both total and difference costing to be employed, depending solely on which is considered more useful. This approach is the net benefits (NB) method in which the difference in the present values of the alternative systems' life-cycle costs (PVC) are calculated:

$$NB = PVC_{\substack{\text{System}\\ \text{A}}} - PVC_{\substack{\text{System.}\\ \text{B}}}$$

The net benefits method is appropriate for a number of reasons. First, solar systems are almost always compared directly with a given conventional energy system, and the benefits method illustrates either a positive or negative benefit in present dollars between the compared systems. Second, the net benefits method is suitable for using either total or difference costing, noting that the benefits format is especially suited for difference costing. Third, the present value total costs can be tabulated for each system independently and the net benefits determined for a series of comparisons. Last, the presentation of net benefit (sometimes referred to as "solar savings") itself is easily understood and accepted by a wide variety of solar system investigators (50, 51, 52).

It should be pointed out that the present value life-cycle cost evaluation is also compatible with other familiar methods of economic analysis. Both annual cost and cost per unit energy figures are readily calculated from present value life-cycle formulations.

4.1.1 Relevant Cost Elements

The system cost elements included in the present value cost formulation are:

- System acquisition costs

  Initial investment costs including design, delivery, installation, building modification, value of system, occupied space, and tax credits (negative)

- System repair and replacement costs

  Cost of repairing or replacing system parts, exclusive of routine maintenance, and net of insurance reimbursement and parts salvage

- Maintenance costs

  Cost of routine up-keep, maintenance, labor, and parts

- Operating costs

  Cost of all fuels used in operating the system including primary and auxiliary equipment

- Insurance costs

  Cost of insuring the system

- Tax costs

  Incremental property tax costs, Federal/state income tax reductions due to interest paid and depreciation, and tax reduction due to fuel expenses

- Salvage

  The salvage value of the system at the end of the period of analysis net of removal and disposal costs.

Each of the above cost elements is described in more detail in a separate document under preparation, including a mathematical description of each element. The input parameters to the SAI life-cycle economics program (ECON) and its capabilities are also presented in this separate document.

4.1.2 General Economic Factors

In order to calculate the above cost elements, a number of general economic factors must be determined and/or assumed. These factors include the following general rates:

- Annual discount rate
- General inflation rate
- Fuel inflation rate

Additionally, there are several specific factors which could be site/user specific. These factors will depend primarily on the site location and the economic circumstances of the systems owner:

- Mortgage interest rate
- Fuel cost
- Federal/state income tax rate
- Property tax rate
- Depreciation rate and salvage value (commercial only)

The economic factors which were used at all locations and for all systems are presented in Table 4-1. These factors correspond to those presently being used in the commercialization readiness assessments for solar heating and cooling systems conducted by DOE (53), except for the economic analysis period

Table 4-1

Economic Factors Used at all Three Locations for all Systems (Residential) (53)

FACTOR	VALUE
Repair and maintenance including replacement	5% annually of initial investment
General inflation rate	7.5% annually
Discount rate	8.5% annually
Interest rate	11% annually
System life (economic analysis period)	20 years
Down payment	20%
Period of loan	20 years
Fuel inflation rates:	
• Electricity	10% annually
• Oil	11% annually
• Natural gas	12% annually
Income tax rate	30% Federal
Solar system property tax rate	0%

which was chosen as 20 years. Note that the only site-specific economic factor used in this study is the fuel cost factor, and this is included in the system-related economics below.

4.2 SYSTEM-RELATED ECONOMIC PARAMETERS

The system-related economic factors are of two types: fuel costs and equipment/installation costs. The fuel costs used are location dependent and are presented in subsection 4.2.1. The equipment/installation costs are both location and system dependent and are presented in subsection 4.2.2 below.

4.2.1 Fuel Costs

The fuel costs used in this study were determined by contacting the various utilities at each of the locations. Since more than one utility was usually involved at a location, the rates were averaged for that location. The resulting rates for electricity, natural gas, and fuel oil are listed in Table 4-2 in dollars per million Btu.

4.2.2 Equipment/Installation Costs

Site-specific cost estimates were developed for each of the eight residential heating systems under study. These cost estimates, together with their justifications and limitations, are discussed in detail below.

Development of Costs

Due to the rapidly evolving nature of the solar industry and competitive aspects of the construction business, the development of precise costs for hypothetical solar systems is almost impossible. Wide cost variations can occur due to differing construction techniques, materials selection, and market considerations. Standardized specifications for solar systems are only now beginning to crystallize while new concepts and materials are constantly being introduced. Such considerations are particularly true for series solar heat pump systems. However, the authors feel that, for the purposes of this study, it is possible to develop reasonably accurate cost estimates. The cost figures presented herein reflect the above considerations.

The cost estimates for the four solar and four nonsolar systems were derived from the synthesis of a wide variety of sources. These sources include:

Table 4-2

Location-Specific Fuel Costs for Residential Users
($/$10^6$ BTU)

MADISON*		
GAS:	3.04 year round rate 2.44 space heating only rate	
ELECTRIC:	11.05 year round rate	
OIL:	6.50+	
WASHINGTON AREA**		
GAS:	3.90 year round rate	
ELECTRIC:	9.03 winter rate (7 months) 16.12 summer rate (5 months)	11.98 year round rate
OIL:	6.50+	
FT. WORTH/DALLAS***		
GAS:	2.23 year round rate	
ELECTRIC:	11.34 average year round rate	
OIL:	6.50+	

*Public Service Commission of Wisconsin (gas, electric), Nov. 1978

**PEPCO, VEPCO (gas, electric), Feb. 1979

***Texas A&M University, CEMR Survey, 1978

+Assumed National Value

- Cost data from actual solar projects (54, 55, 56)
- Standard construction cost estimating guides (57, 58)
- Cost estimates by other study groups (59, 60, 61, 62)
- First-hand experience in design of actual solar systems.

The cost information resulting from actual solar projects has been given the greatest weight in developing estimates.

Presented below are cost estimates which are representative for complete, reasonably well-designed systems, up to, but not including, the duct work. Duct work costs were excluded since each alternative had an identical air distribution system.

The complete system costs estimates were established for heating systems appropriate to the "typical residences" defined earlier in this report. Cost estimates for the four solar systems have been separated into a fixed cost and a variable cost. The fixed cost is approximately constant for an appropriately sized solar system. The variable cost is a direct function of unit collector area. Each physical component of a given system design has been given either a fixed or variable cost. Since labor costs are much more sensitive to geographic location than material costs, these system component costs have been further broken down into separate material and labor categories. In developing the site-specific costs, the labor category of each cost estimate has been multiplied by an adjustment factor to account for this geographic sensitivity. For Washington, D.C., this factor was 1.00; for Madison, it was 0.93; and for Fort Worth, it was 0.84 (57, 58).

The cost estimates for these solar systems assume that these designs are common, that they utilize off-the-shelf components, and that their installation is well within the technical capabilities of most residential contractors. In this respect, the costs developed herein may be somewhat optimistic. Furthermore, the actual average cost for some components may vary significantly from what is presented in the following breakdowns. For example, the collector and heat pump costs for the series type solar-assisted heat pump system may be different (probably higher) from what is assumed here. However, actual cost data indicate that such collector costs are obtainable for site-built systems. The series heat pump costs are representative of present costs of current technology equipment.

In the development of any cost estimate for a hypothetical solar system, certain critical assumptions concerning the design are necessary. It has already been mentioned that a reasonably good technical design has been assumed. In addition to this, other significant assumptions include:

- Internally manifolded collectors with a copper absorber plate
- For the parallel systems, good quality collectors with a black chrome absorber surface and single glazing
- High temperature plastic tubing (CPVC) for the reverse return type collector loop
- Unpressurized wooden storage tanks with fiberglass insulation
- For the series systems, a storage volume of 415 kg (of H_2O)/m^2 of collector area); for the parallel systems, a storage volume of 75 kg (of H_2O)/m^2 or 244 kg (of rock)/m^2
- A two-story house for the Washington, D.C. location and a single-story house for the Madison and Fort Worth locations
- Availability of sophisticated, but inexpensive, solid-state controls
- No tax credits of any kind are considered.

It should be noted that, were some of these assumptions to be different, such as the storage volume to collector area ratio, the projected cost and/or performance of the system could be significantly different. This is discussed in subsection 8.4.

Appendix E lists system-specific cost estimates broken down into a component level. Each cost element is separated into a materials and labor category. Contractor profit and overhead are not included in the Appendix E listing.

Site-Specific Cost Summary

Based on the best available cost data and the assumptions discussed above, the figures presented in Table 4-3 are considered to be reasonable fixed and variable cost estimates for the various system types as a function of geographic location. These costs are in 1979 dollars, with a 20 percent overhead and profit factor included.

Table 4-3

Conventional and Solar Heat Pump Systems' Costs 1/
Including Overheat and Profit

SYSTEM / LOCATION	Gas + A/C	Oil + A/C	Elec. + A/C	Heat Pump	CSHP: Series/ Liq.	CSHP: Parallel/ Liq.	CSHP: Series/ Air	CSHP: Parallel/ Air
Washington, D.C.	$2,120	$2,395	$1,999	$2,413	$\$142.00/\text{m}^2 + 9{,}598$	$\$222.00/\text{m}^2 + 8{,}142$	$\$136.00/\text{m}^2 + 8{,}161$	$\$297.00/\text{m}^2 + 5{,}852$
Madison, Wisconsin	$2,083	$2,359	$1,964	$2,376	$\$138.00/\text{m}^2 + 9{,}529$	$\$220.00/\text{m}^2 + 8{,}126$	$\$132.00/\text{m}^2 + 8{,}251$	$\$293.00/\text{m}^2 + 6{,}024$
Ft. Worth, Texas	$2,047	$2,312	$1,907	$2,329	$\$134.00/\text{m}^2 + 9{,}200$	$\$217.00/\text{m}^2 + 7{,}865$	$\$127.00/\text{m}^2 + 7{,}985$	$\$289.00/\text{m}^2 + 5{,}861$

NOTE: m^2 refers to square meters of collection area.

1/ Add $500 to fixed costs for "future" systems which include an improved heat pump.

5.0 THERMAL PERFORMANCE FACTORS

The models for this study have been formulated so that sufficient outputs are received to perform energy balances, track important state variables, account for parasitic energy usage, and determine important time-of-day and frequency distributions. This section is designed to define the important energy quantities used in this study (subsection 5.1), present and define the system and subsystem performance indicators (subsection 5.2), and discuss the thermal comfort factors involved with these systems (subsection 5.3). A detailed definition of all system quantities is presented with appropriate sketches in Appendices F through J.

5.1 ENERGY QUANTITIES

The energy quantities tracked during the system simulations are defined in Table 5-1. The energy quantities associated with a given system are described in the aforementioned appendices.

5.2 THERMAL PERFORMANCE INDICATORS

The thermal performance indicators are presented in two groups: those associated with subsystem characterization and those indicating overall system thermal performance. A listing of subsystem performance indicator definitions is presented in Table 5-2, while Table 5-3 lists the definitions of the overall system performance indicators. Again, the system-specific definitions and sketches are presented in Appendices F through J.

5.3 THERMAL COMFORT FACTORS

The purpose of a heating or cooling system is to provide thermal comfort to the occupants of the conditioned space. This study attempts to compare alternative residential heating and cooling systems using simulation to predict thermal performance and life-cycle economics as the basis of comparison. In order for this approach to be successful, the thermal simulation models for each system must be constructed in such a way that the modeler can force each system to provide the same level of thermal comfort. Before this can be

Table 5-1

Energy Quantities Tracked During the System Simulations

HEAT PUMP		
QHP-L	-	heating energy transferred to the load by the heat pump (KJ)
WHPH	-	work input to the heat pump in the heating mode (KJ)
QSCC	-	sensible cooling provided by the heat pump (KJ)
QLCC	-	latent cooling provided by the heat pump (KJ)
QTCC	-	total cooling provided by the heat pump (KJ)
WHPC	-	work input to the heat pump in the cooling mode (KJ)
COLLECTOR		
QU	-	useful energy obtained by the solar collector (KJ)
QDUMP	-	energy dumped through the collector loop relief valve (KJ)
LOAD		
QDIR-L	-	solar energy transferred directly to the load bypassing the heat pump (KJ): series configuration
QCOL-L	-	solar energy delivered to the space heating load directly from the collectors (KJ): parallel configuration
QPB-L	-	solar energy delivered to the space heating load from the pebble bed (KJ)
QA-L	-	auxiliary energy transferred to the space heating load (KJ)
QLOAD	-	energy delivered to the load for space heating (KJ)
STORAGE		
QTNK	-	energy removed from the tank via mechanical flow circuits (KJ)
QENV	-	heat losses from the tank to surrounding environment (KJ)
QREJ	-	energy rejected from storage through the fan coil unit (KJ)
DOMESTIC HOT WATER		
DHWLD	-	domestic hot water load (KJ)
DHWIN	-	solar energy input to the domestic water heating subsystem (KJ)
DHWTNK	-	energy transferred from the preheat tank to the domestic water heating load (KJ)
DHWENV	-	heat losses from the preheat tank to the environment (KJ)
DHWAUX	-	auxiliary energy required to meet the domestic water heating load (KJ)

Table 5-2

Subsystem Performance Measures Tracked
During the System Simulations

HEAT PUMP		
COPH	-	heat pump COP in the heating mode
COPC	-	heat pump COP in the cooling mode
HRHPH	-	hours which the heat pump runs in the heating mode
HRIFAN	-	hours which the heat pump indoor unit fan runs while the compressor is idle
HRHPC	-	hours which the heat pump runs in the cooling mode
COLLECTORS		
COLEFF	-	overall collector efficiency, QU divided by incident solar radiation on the collector
CEFFON	-	operating collector efficiency, QU divided by incident solar radiation on the collector when the collector is on
HRCOLL	-	hours which the collector loop runs
STORAGE		
HRRL	-	hours which the cooling mode tank heat rejection loop runs
ATPB	-	average pebble bed temperature (C)
ATTANK	-	average main tank temperature (C)
DOMESTIC HOT WATER		
HRDWH	-	hours during which solar heat is supplied to the domestic water preheat tank
ATPHT	-	average temperature of domestic hot water preheat tank (C)

Table 5-3

System Performance Measures Tracked
During the System Simulations

FREE ENERGY FRACTION		
FSSH	-	fraction of the space heating and cooling load met by non-purchased energy (parasitic energy usage is considered)
FSDHW	-	fraction of the domestic hot water load met by solar energy (parasitic energy usage is considered)
SYSTEM OPERATING HOURS		
HRDSL	-	hours which the direct solar heating loop pump runs
HRHTG	-	hours which the system runs to meet the heating load
SPACE AND AMBIENT CONDITIONS		
ATR	-	average room temperature (C)
AWR	-	average room humidity ratio
ATAMB	-	average ambient temperature (C)
AWAMB	-	average ambient humidity ratio
ATDELH	-	average delivered air temperature during heating mode operation (C)
ATDELC	-	average delivered air temperature during cooling mode operation (C)

accomplished, an acceptable analytic characterization of thermal comfort is required.

5.3.1 Practical Application of a Thermal Comfort Index

Fanger (63) describes a method of relating environmental conditions to the predicted percentage of dissatisfied occupants (PPD). The variables required to calculate PPD are:

- Activity level
- Type of clothing (clo value)
- Air temperature experienced by the occupant(t_a)
- Mean radiant temperature experienced by the occupant (t_{mrt})
- Humidity experienced by the occupant (RH)
- Relative air velocity experienced by the occupant (V).

Activity level is normally classified as sedentary (50 kcal/hr-m^2). Clothing is classified as nude (0.0 clo), light (0.5 clo), medium (1.0 clo), and heavy (1.5 clo), where 1 clo = .18 C-m$_2$-hr/kcal. Air temperature refers to the dry bulb temperature in the vicinity of the occupant. The mean radiant temperature is defined as the uniform temperature of black surroundings which will give the same radiant heat loss from a person as the actual case under study. The relative air velocity refers to the velocity of air relative to the body surface of the occupant.

Given that these six environmental factors allow the rating of the environment with the PPD index, the remaining problem is to relate the six environmental factors to outputs of a system simulation. State-of-the-art simulation is not yet able to relate system operation to the comfort index on a dynamic basis. The direction which model development must take to reach this goal is, however, understood. For the present study, several assumptions were made allowing currently available models to be utilized.

5.3.2 Assumptions and Approach

The type of clothing can be specified monthly or seasonally at a typical value for the geographic location and type of activity which takes place in the building. Since the current study deals exclusively with residential heating and cooling systems, it is reasonable to assume that occupants have a sedentary (50 kcal/hr-m^2) activity level. For this study, regional clothing preferences will be ignored and occupants in all regions of the country are assumed to have medium clothing (1.0 clo). The other four variables are directly affected by the mechanical system.

In conventional residental construction, where walls are well insulated and window areas are not a major fraction of the building envelope, it is reasonable to assume that all surfaces adjoining the conditioned space are at the same temperature as the space. Therefore, assuming mean radiant temperature is equal to air temperature is reasonable. The current models are able to predict a fully mixed bulk air temperature and humidity for the space. Thus, it was assumed that temperature and humidity conditions are uniform throughout the space.

The most difficult assumption concerns how to determine the relative velocity experienced by an occupant of the space. If solar and conventional systems are allowed to deliver energy to the conditioned space at different flow rates and temperatures, the air distribution systems required to maintain equivalent velocity fields in the space must, by necessity, have different sizes and costs. A thorough analysis would have to include the incremental cost of larger air distribution systems for those designs which require them. Even if correct sizing of the air distribution systems allowed each alternative to maintain equivalent velocity fields, intuitively, the alternatives with lower delivered air temperatures would still not provide the same level of comfort while air flow is on. Current analysis is unable to link the delivered air temperature to an average comfort index for the space.

As a practical matter, to ensure that all systems provide the same comfort, all that can be done with current simulations techniques is to force each system to deliver the same air flow to the load. Furthermore, each system must be expected to deliver energy at a temperature greater than or equal to a specified minimum. Under this restraint, all air distribution systems will be the same size and, therefore, do not enter into the comparative cost analysis. Furthermore, temperature and velocity fields will be similar for each system when air flow is on. Any dissimilarity will be due to higher delivered air temperature (heating mode), which normally is not a source of discomfort.

One further assumption is that the air distribution systems are well designed, following the guidelines of references (15, 64, 65, 66). Under this condition, it can be assumed that an occupant in the space will be exposed to an air velocity of approximately 30 fpm (still air), regardless of whether the air distribution system is on or off.

With the above assumptions, room temperature and humidity can be used to calculate the PPD comfort index dynamically. Assuming the temporal response of each system to the thermostat is similar, PPD for each system will be similar. In this study, the room temperature and humidity, plus the delivered air temperature during heating mode operation, were tracked during the simulations. The data were recorded via frequency histograms and are presented for each system in Appendices F through J. Later work will be directed toward establishing a dynamic thermal comfort index coupled with the thermal simulation program.

6.0 PRESENTATION OF RESULTS

This section presents the results of the system simulations. Due to the extraordinary amount of data generated, only selected and summarized presentations are made; the complete simulation results are presented in Appendices F through J. The summarized presentations are given in three parts: Monthly-Annual System Thermal results; Subsystem Characteristics results; and the results related to Systems' Economic, Comfort, Reliability and Utility Impact Characteristics.

6.1 MONTHLY AND ANNUAL SYSTEMS THERMAL SUMMARIES

The results concerning the liquid series configuration are presented in some depth since this system has previously been "less studied." The monthly and annual energy quantities associated with all system configurations are presented in both graphical and tabular form for heating, cooling, and hot water.

6.1.1 Series System Thermal Performance

Since rules of thumb for sizing the liquid series system were not available, thermal performance maps were generated over a range of storage and collector sizes. These maps were then used to optimally size the liquid series system in each study location using the method described in Section 7. The thermal performance maps are presented and discussed below.

Liquid Series, Today's Technology

Figure 6-1 displays the performance map for the liquid series system in Washington, D.C., when a liquid-to-air heat pump typical of today's technology is utilized. The heat pump could operate in the heating mode when the liquid source was between 14.4 and 33.9°C. Simultaneous heating of the load with the direct solar loop and the heat pump was not allowed since the heat pump cannot tolerate high entering air temperatures and placing the heat pump upstream of the solar coil caused low or reversed heat transfer across the coil. Consequently, if the source temperature is above 33.9°C, direct solar heating and auxiliary heating are the only options available.

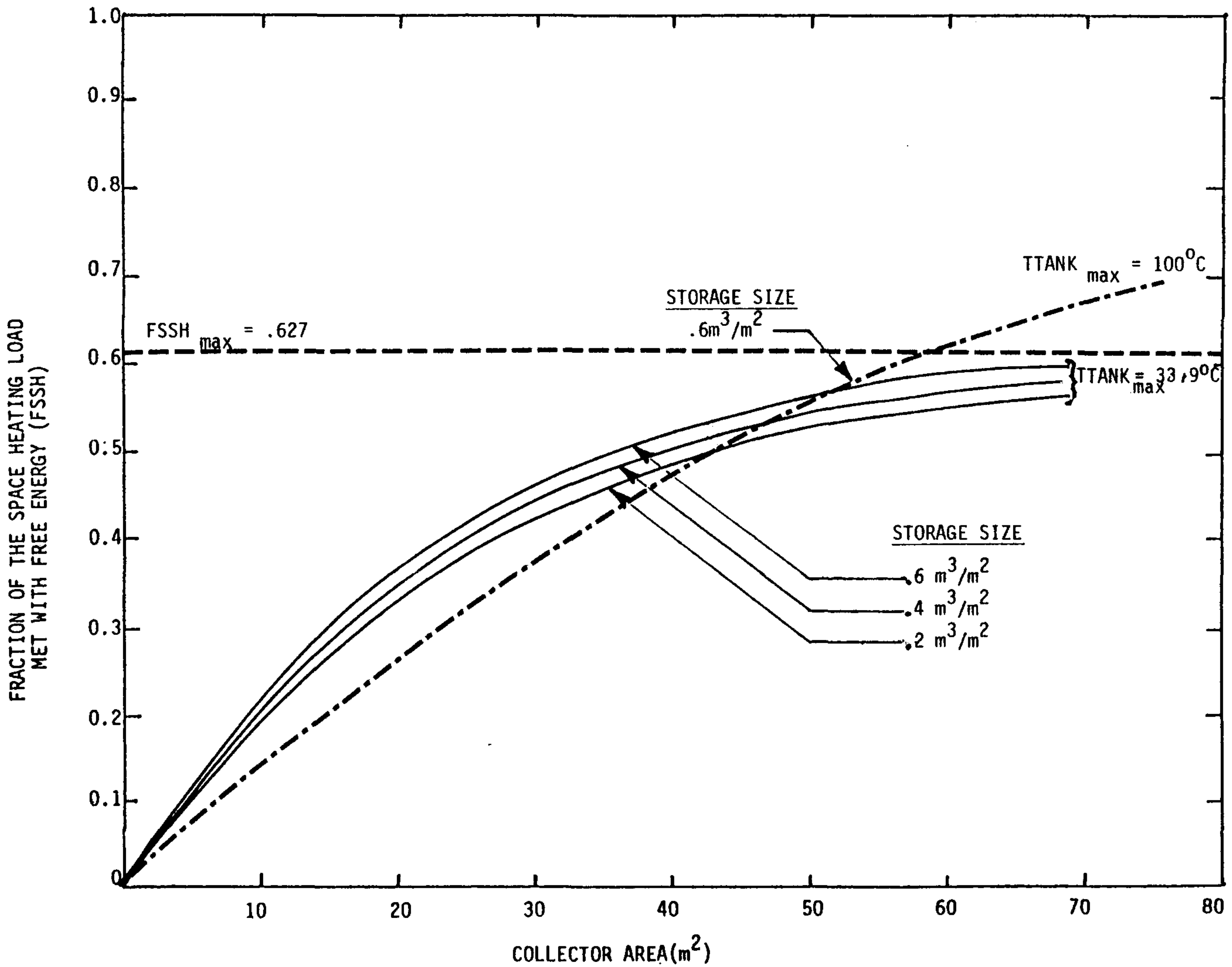


Figure 6-1. Liquid Series System Performance Map: Washington, D.C., Today's Technology Heat Pump

It was not initially clear how the system should be controlled to minimize the use of purchased energy. One alternative was to control the collector array so that storage never exceeded 33.9°C; then the heat pump could always operate in the heating mode but direct solar heating could never occur. The solid lines on Figure 6-1 were generated using this strategy. An alternative was to allow solar collection until storage reached the boiling point. When storage exceeded 33.9°C, the system was controlled as follows: in first stage, direct solar heating occurred if storage was above 40.2°C, the required source temperature, if the temperature of air delivered to the space was to equal or exceed 35°C; otherwise, the system was idle in first stage and met the load with auxiliary in second stage. The dashed line on Figure 6-1 was generated with this strategy.

At large collector areas, the second strategy is superior, even though it results in a "dead region" between 33.9 and 40.2°C where solar storage cannot be utilized. Apparently, direct solar heating occurs often enough to compensate for this shortcoming. At low collector areas, the first strategy is superior. The first strategy was selected for this study since optimum collector areas fell in the region where it was superior. More sophisticated control strategies using direct solar heat and auxiliary simultaneously when storage exceeds 33.9°C were not investigated. Nor was the possibility of using a mixing valve to temper the heat pump inlet temperature (when storage is above 33.9°C) investigated.

Figures 6-2 and 6-3 present this same information, but for Fort Worth and Madison. In Figure 6-2, note that FSSH actually drops at large collector areas. This occurs because the maximum COP of the heat pump did not occur at its maximum source temperature (33.9°C). This is a feature of the particular liquid-to-air heat pump used and probably results from the manufacturer attempting to design the machine to operate over a broad source temperature range (14.4 - 33.9°C) without using a two-speed compressor or other advanced techniques. At large collector areas, the same phenomena will occur in Washington and Madison.

Figure 6-4 plots the performance curve for all three locations at the largest storage size. FSSH increases with collector area most quickly in warm, sunny climates (low load, high incident radiation) since the heat pump is "starved" less often. The black dots indicate the optimum collector areas at each location, as determined by using the procedure of subsection 7.3.

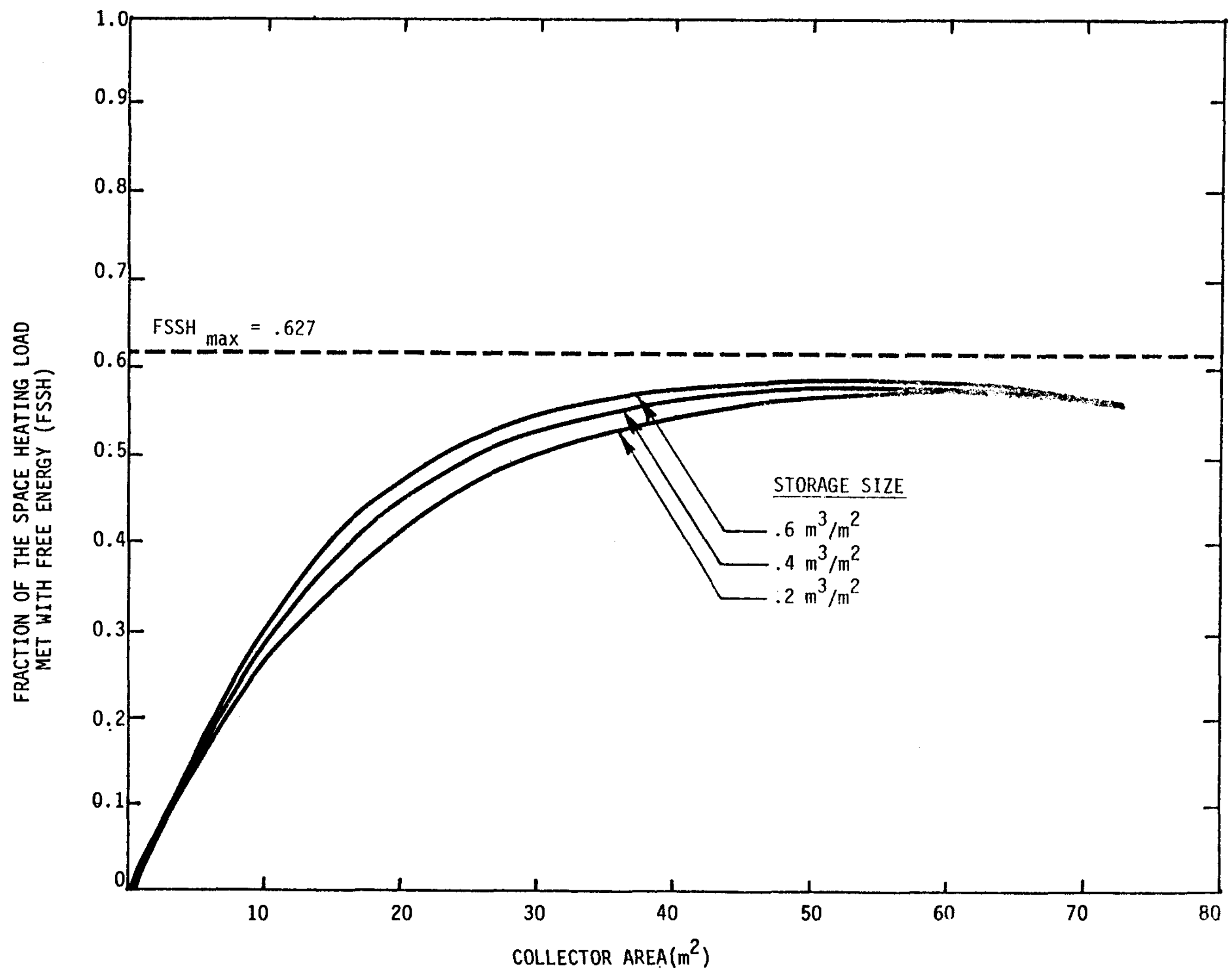


Figure 6-2. Liquid Series System Performance Map: Ft. Worth, Today's Technology Heat Pump

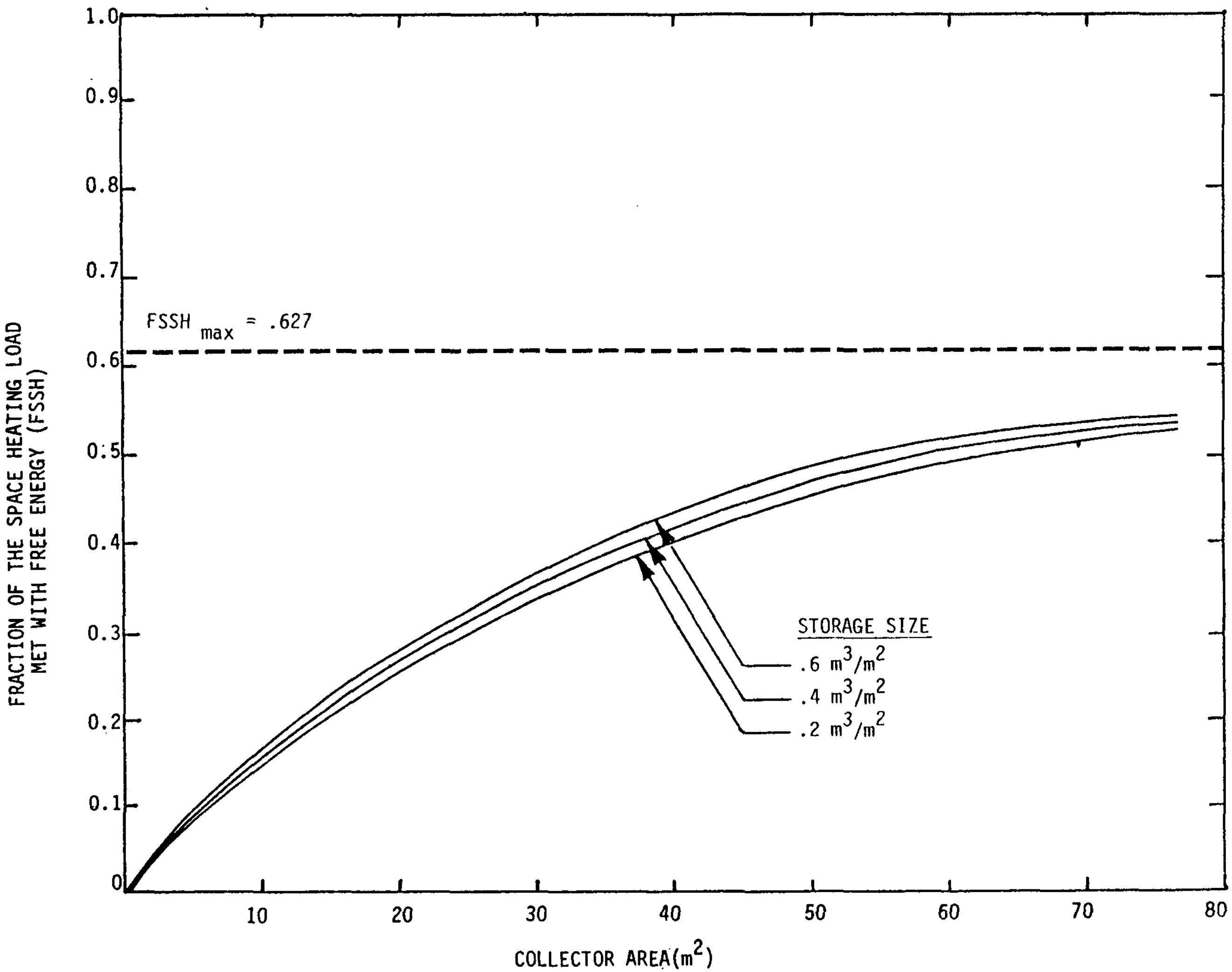


Figure 6-3. Liquid Series System Performance Map: Madison
Today's Technology

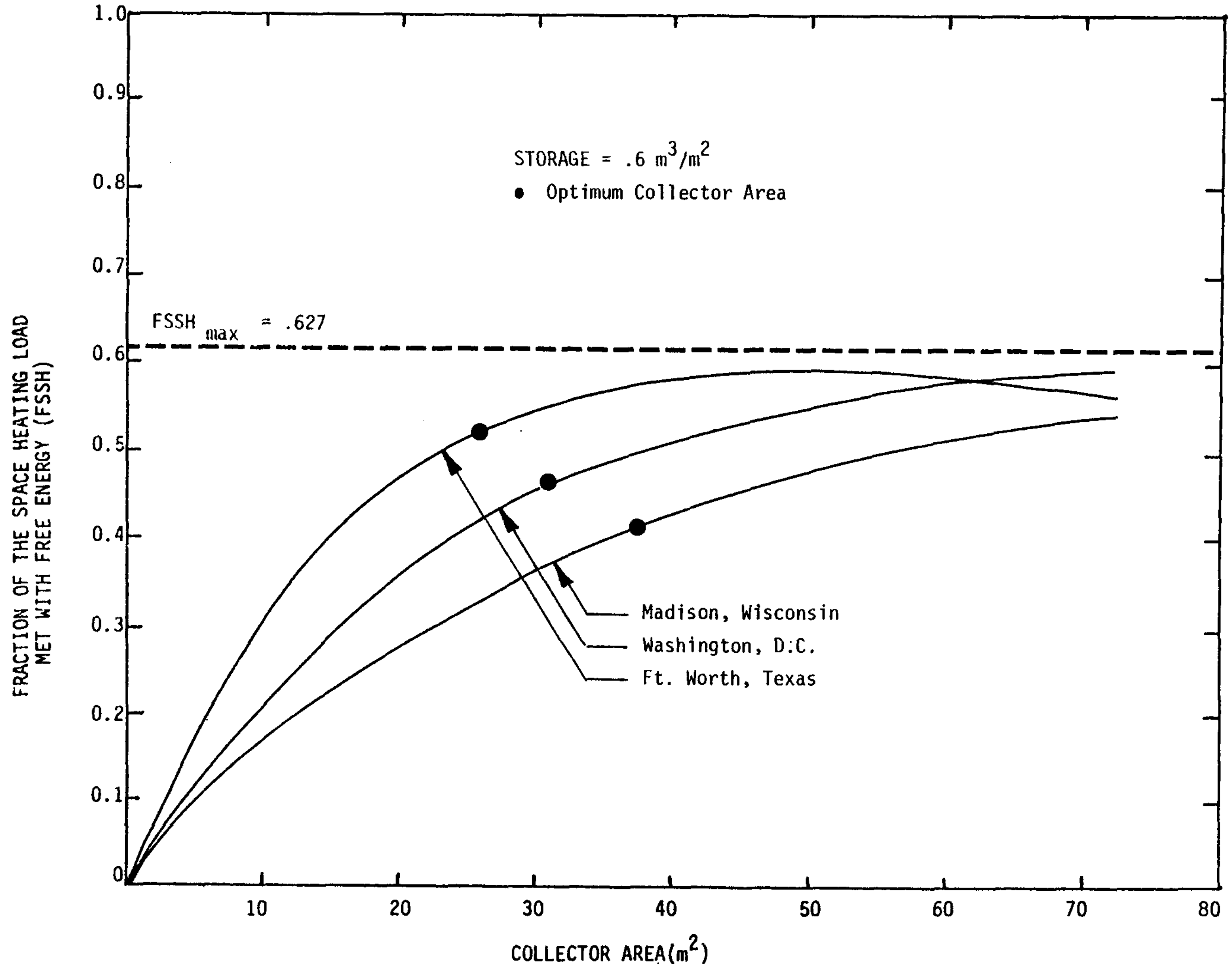


Figure 6-4. Three Locations' Liquid Series Performance Maps at $0.6 m^3$ Storage Volume/m^2 Collector Area: Today's Technology Heat Pump

Liquid Series, Future Technology

The liquid series system with a future technology heat pump does not have the control problem investigated in the last section so long as the heat pump can use the tank as a source all the way up to 40.2°C. Beyond 40.2°C, direct solar heating can occur. Consequently, FSSH approaches 1.0 at infinite collector area (100 percent direct solar heating).

Figures 6-5 through 6-7 display the thermal performance of this system in Washington, Fort Worth, and Madison. Note that the curves are not true exponentials. FSSH rapidly increases as collector area rises from zero because the heat pump is starved less often. FSSH rises more gradually at intermediate collector areas since starvation during long cloudy periods is less easily prevented. At large collector areas, the rise in FSSH remains gradual as the average heating COP rises with average source temperature and the proportion of direct solar heating increases. Note that, at large collector areas, storage size has no effect on system performance over the range of storage sizes investigated here. In this region, a large proportion of the load is met with direct solar heating and it appears that direct solar systems characteristically require smaller storage sizes that those used in this series study.

Figure 6-8 illustrates performance curves for all three locations at the largest storage size. The optimum collector areas, determined by using the procedure of subsection 7.3, are indicated.

6.1.2 Parallel System Thermal Performance

Thermal performance maps for parallel systems are available in the literature (7). These maps, however, were not generated with imposed thermal comfort requirements and did not account for parasitic energy consumption. Consequently, new maps were generated for the three study locations.

Liquid Parallel System

Figures 6-9 through 6-11 display liquid parallel system performance maps for the three study locations. Note that the ordinate includes water heating subsystem performance since it is integrated into the system and, therefore, should impact the sizing of the main collector array.

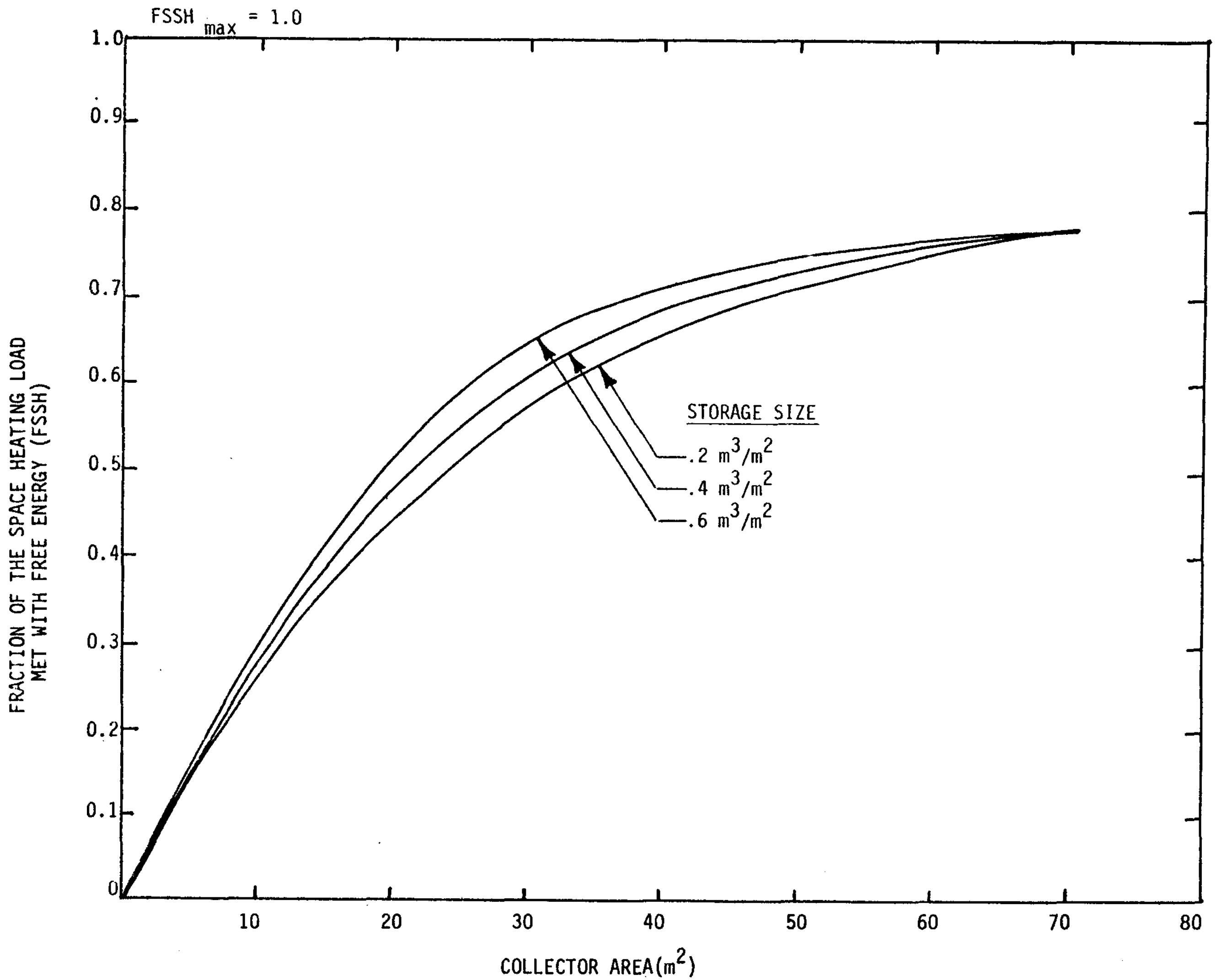


Figure 6-5. Liquid Series System Performance Map: Washington, D.C., Future Technology Heat Pump

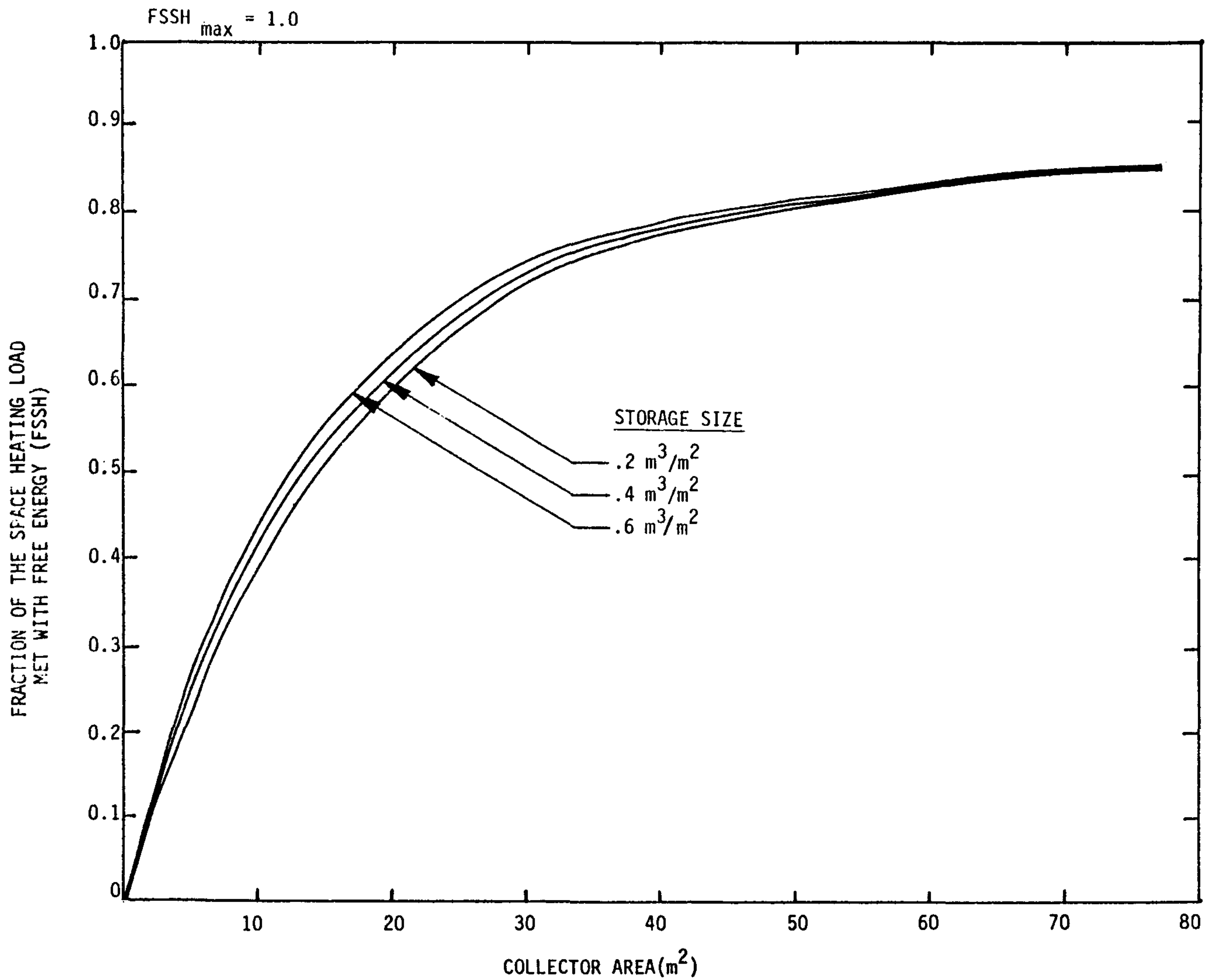


Figure 6-6. Liquid Series System Performance Map: Ft. Worth, Future Technology Heat Pump

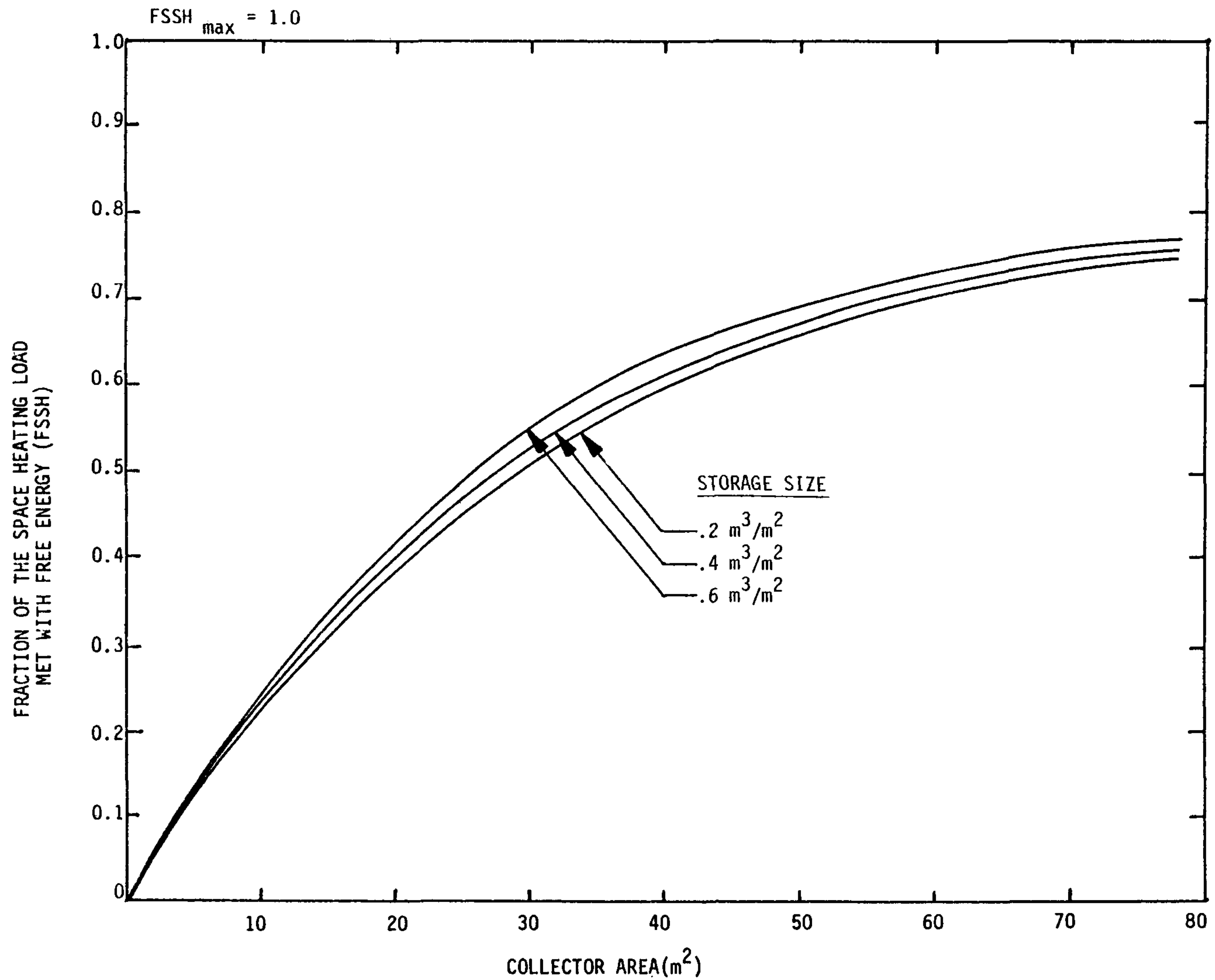


Figure 6-7. Liquid Series System Performance Map: Madison, Future Technology Heat Pump

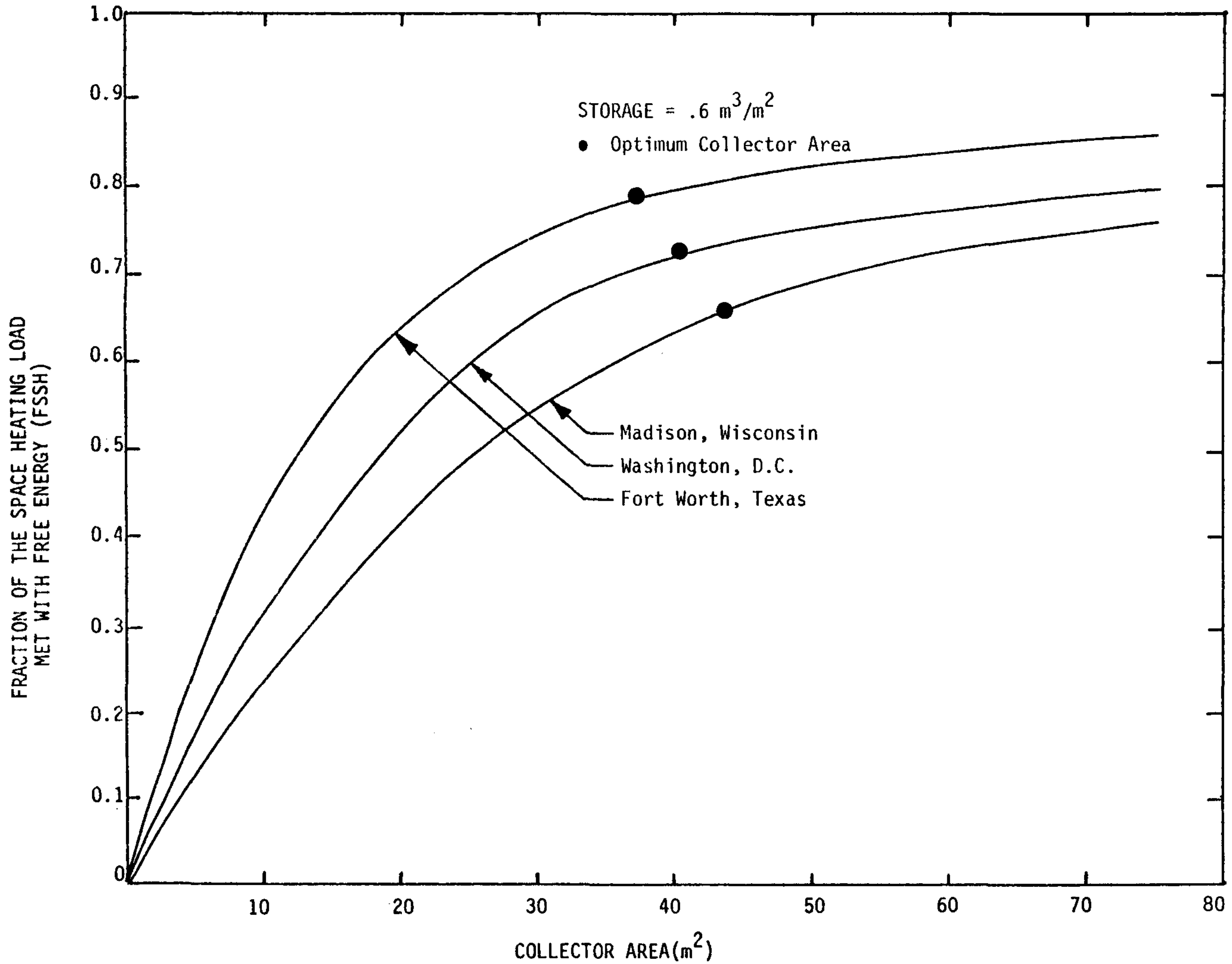


Figure 6-8. Three Locations' Liquid Series Performance Maps at $0.6m^3$ Storage Volume/m^2 Collector Area

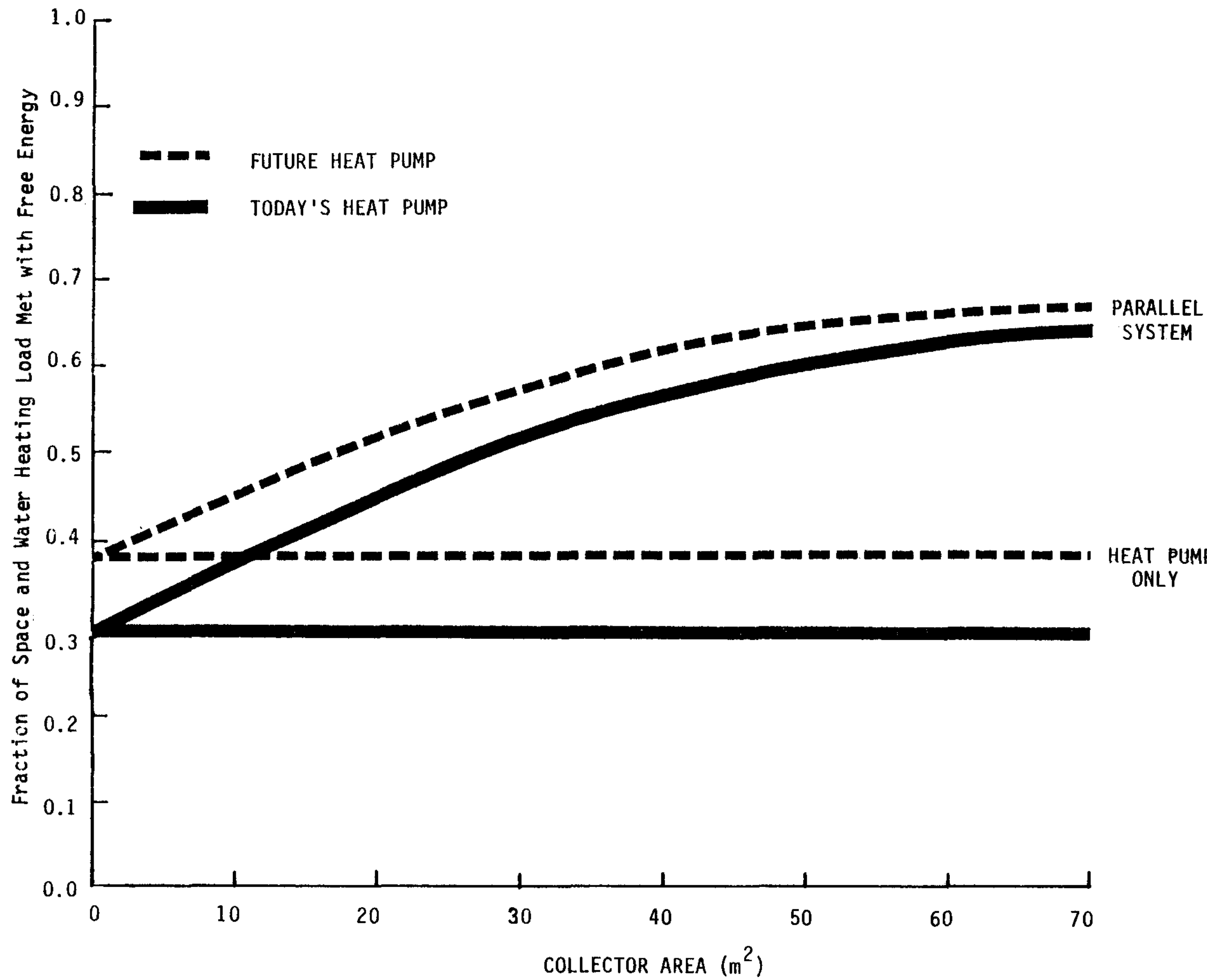


Figure 6-9. Liquid Parallel System Performance Map: Washington, D.C.

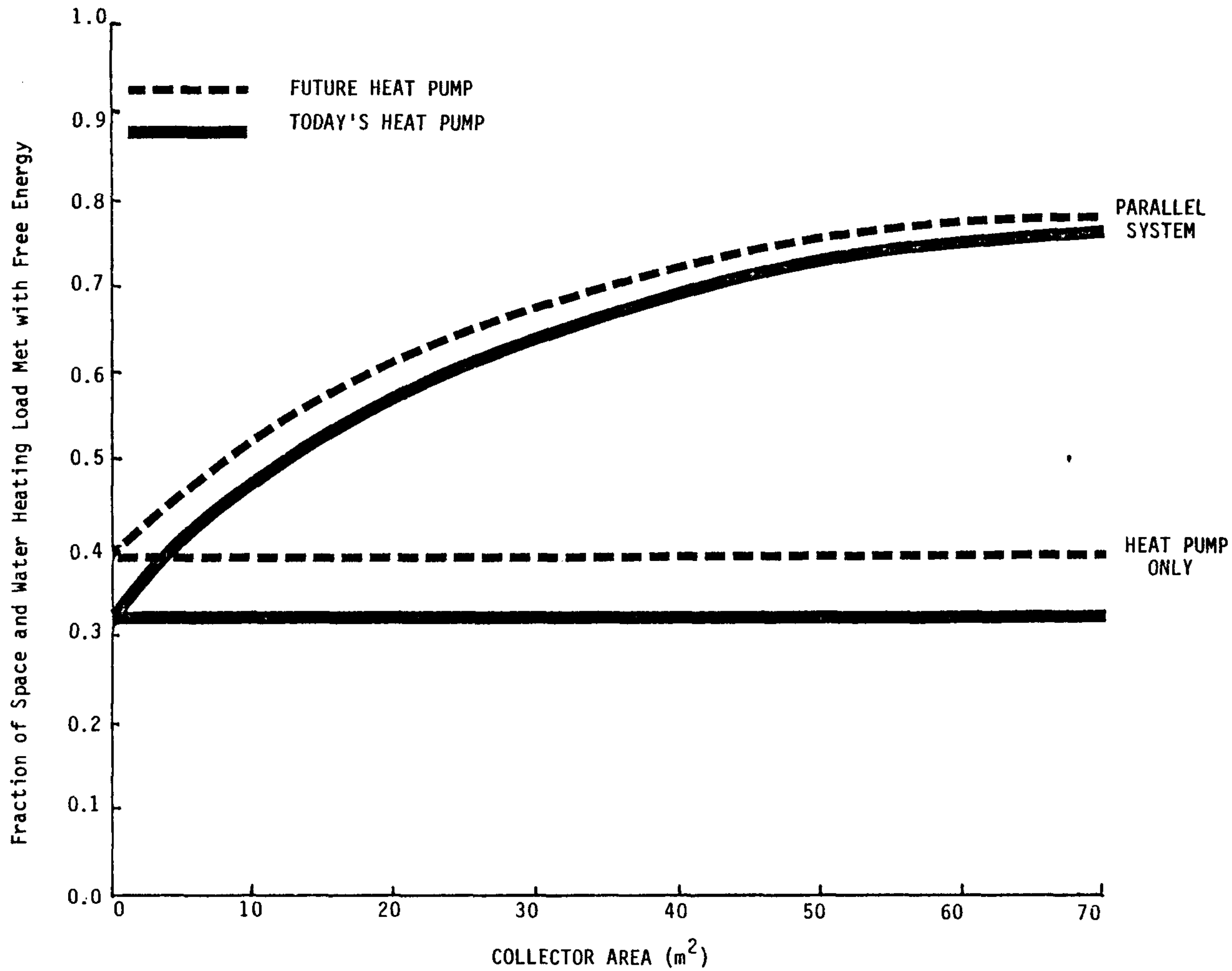


Figure 6-10. Liquid Parallel System Performance Map: Fort Worth, Texas

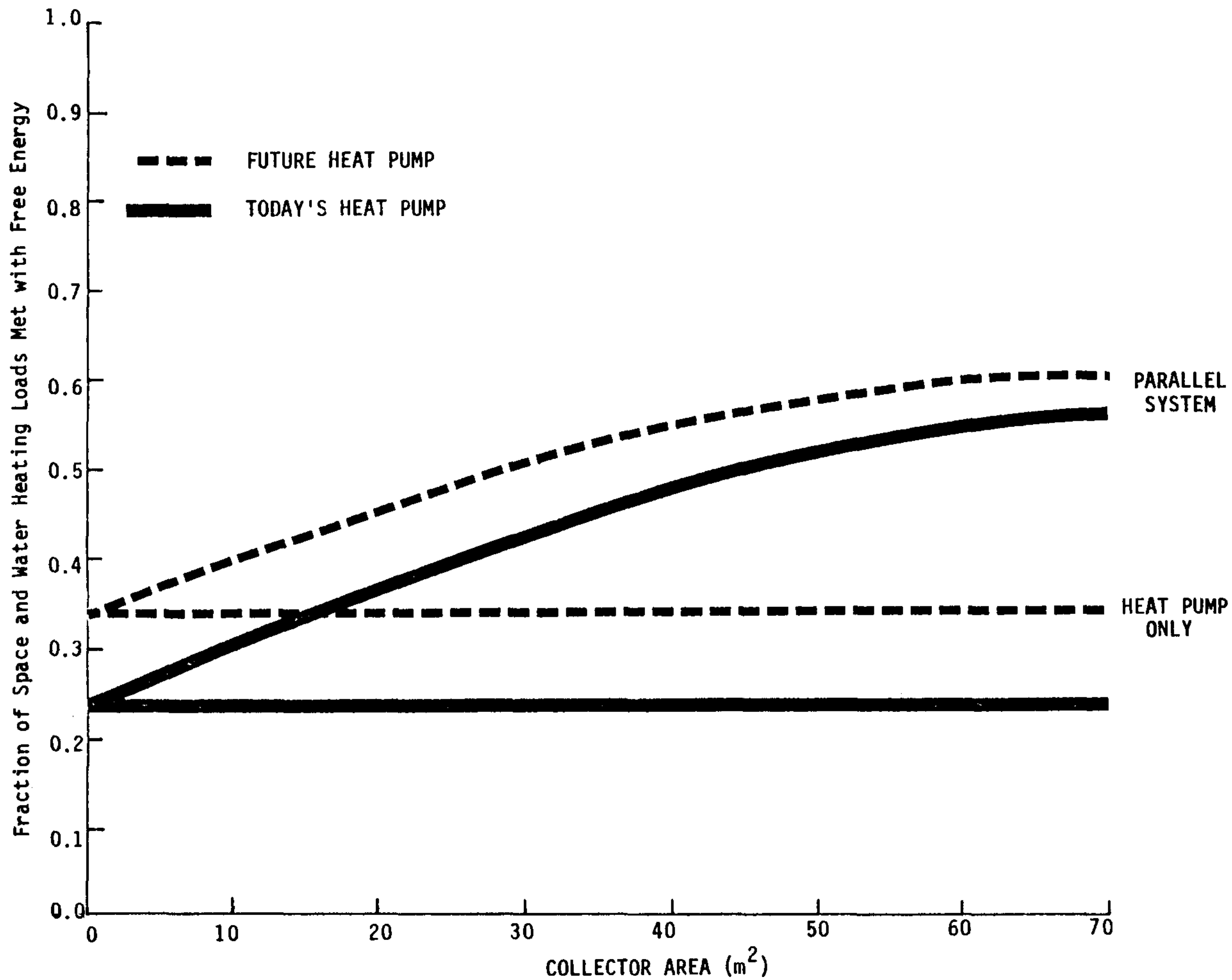


Figure 6-11. Liquid Parallel System Performance Map: Madison, Wisconsin

Air Parallel System

Figures 6-12 through 6-14 display air parallel system performance maps for the three study locations. Note that parasitic energy consumption causes reverse curvature at low collector areas.

6.1.3 Monthly Heating and Cooling Thermal Performance

The monthly thermal performance of each system in each location is presented in Figures 6-15 through 6-26. The solar heat pump systems have optimized collector areas as per subsection 7.3; these figures demonstrate the effect of improving the heat pump. Space heating or cooling energy is plotted versus month of the year, where the dark black outline represents the heating or cooling load. For heating, this corresponds to the amount of electricity which must be purchased if the load were met with electric resistance heat (divide by the furnace efficiency to obtain the energy content of gas or oil which must be purchased). The dotted area represents the free energy applied to the load by the system using a state-of-the-art heat pump. The cross-hatched area represents the additional free energy provided by the system when using the next-generation heat pump and the optimum collector area.

Note that cooling mode performance is identical for all systems except the liquid series system where energy is rejected to a tank and then to ambient rather than to ambient directly. Note also that, in the parallel systems, no heating mode performance improvement is seen in the spring and fall with the improved heat pump since the heat pump is never used due to direct heating.

6.1.4 Monthly Domestic Water Heating Thermal Performance

This study involved three different types of solar domestic water heating systems. The air and liquid parallel systems each have domestic water subsystems integrated into the space heating and cooling system. The series systems utilize a stand-alone solar domestic water system with its own collectors, plumbing, and storage. The performance of these three types of subsystems is given in Figures 6-27 through 6-29 where domestic water heating energy is plotted versus month of the year. The dark black outline represents the domestic water load. Each month, three columns are drawn which represent the portion of the load met with solar energy by the three subsystem types.

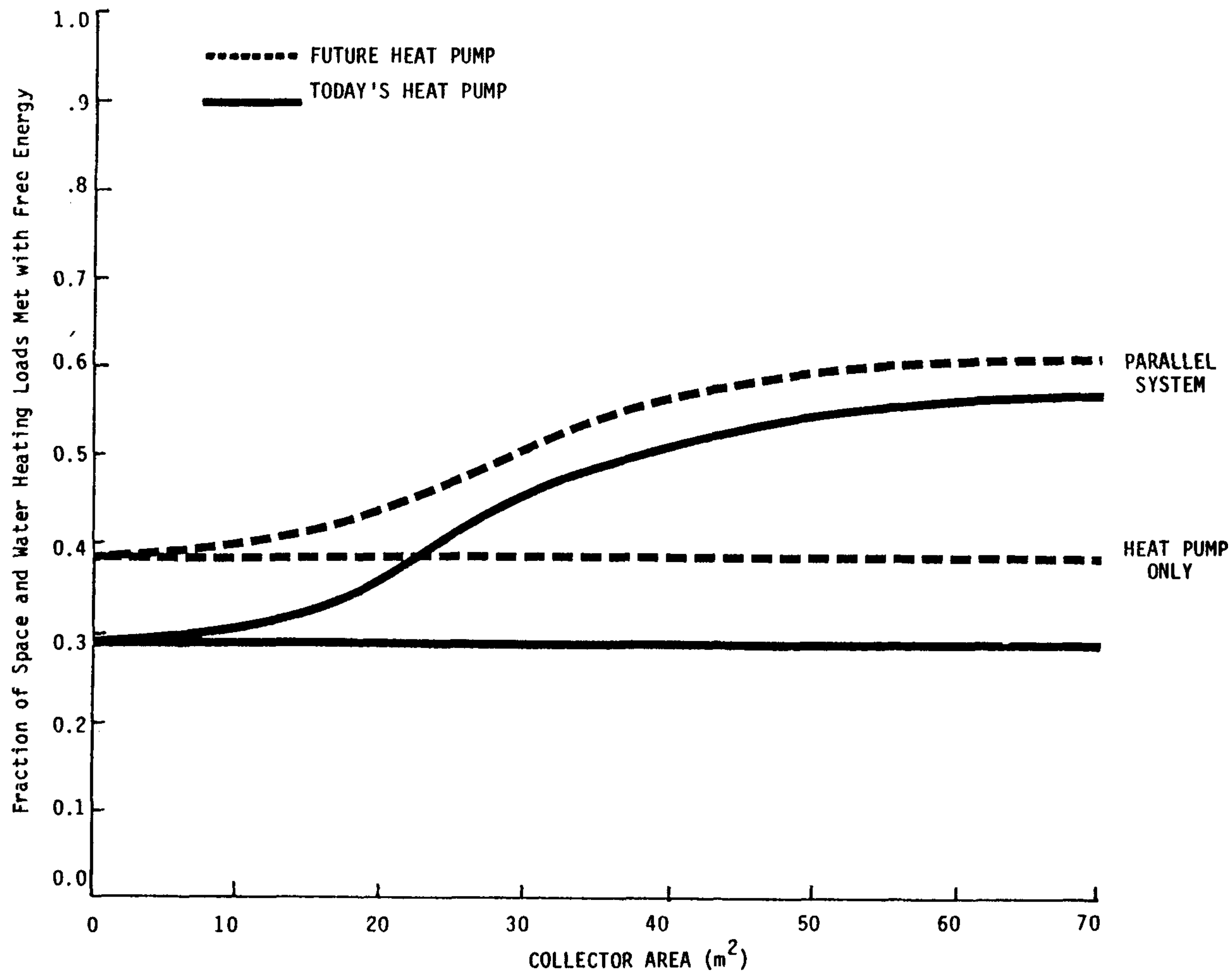


Figure 6-12. Air Parallel System Performance Map: Washington, D.C.

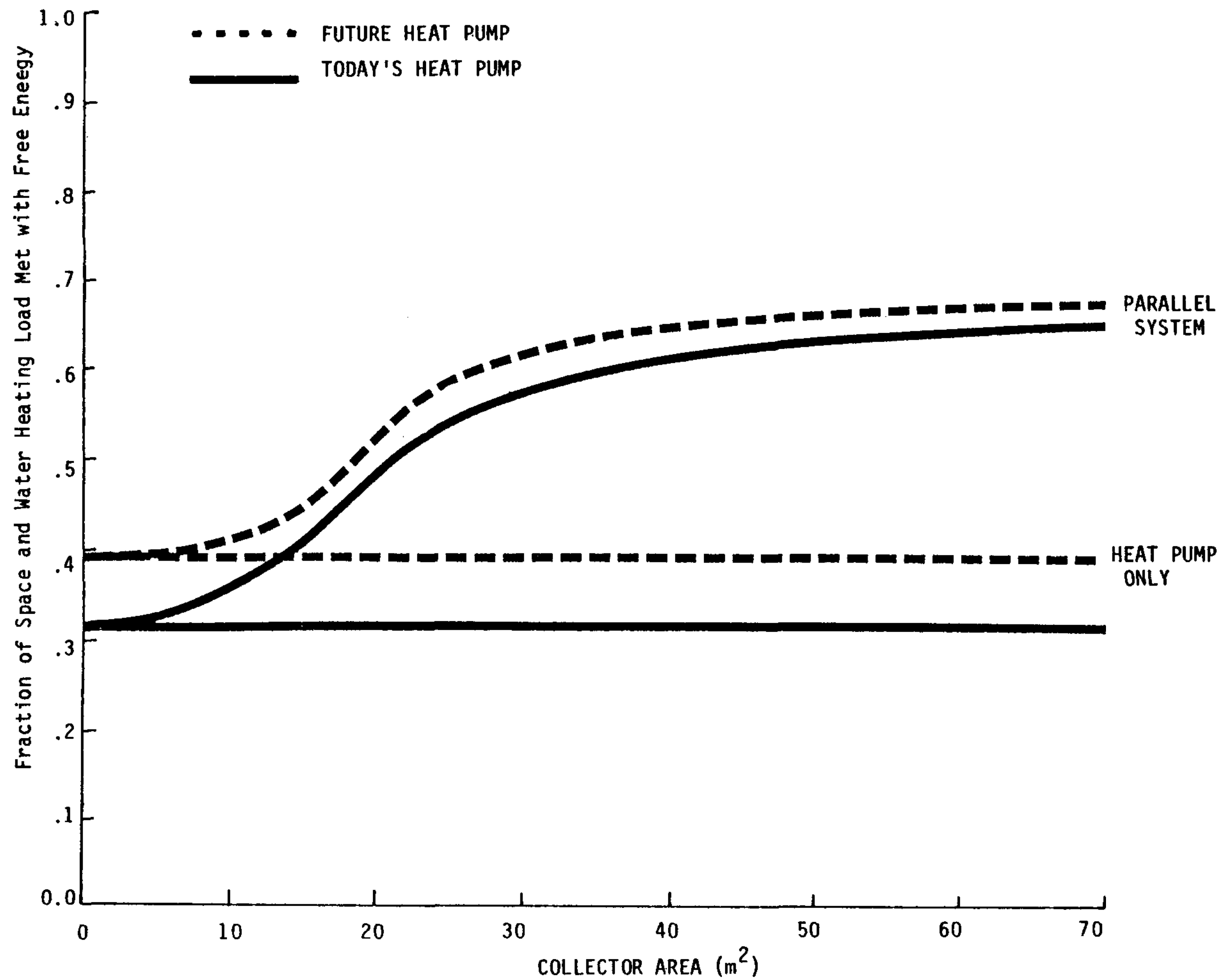


Figure 6-13. Air Parallel System Performance Map: Fort Worth, Texas

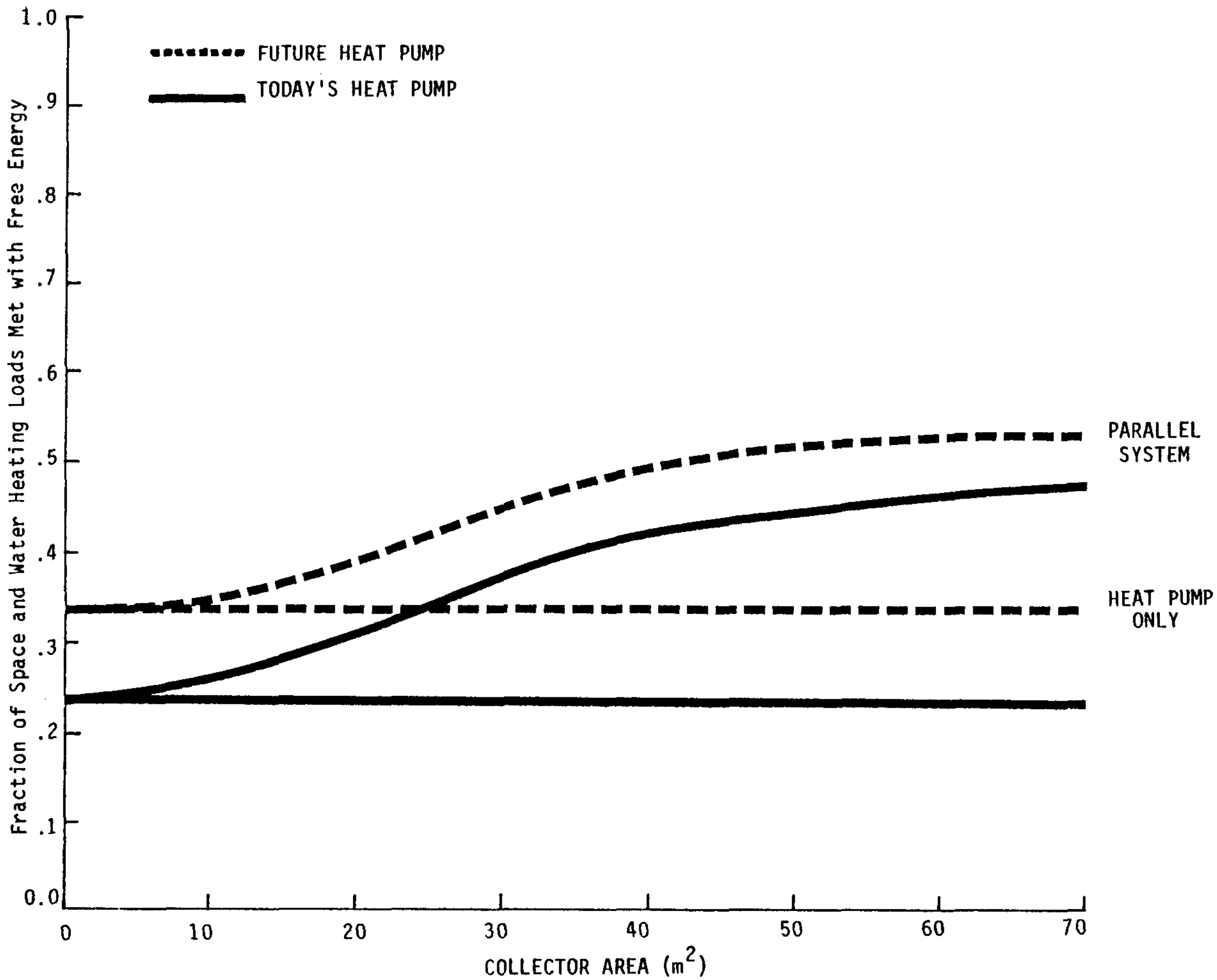


Figure 6-14. Air Parallel System Performance Map: Madison, Wisconsin

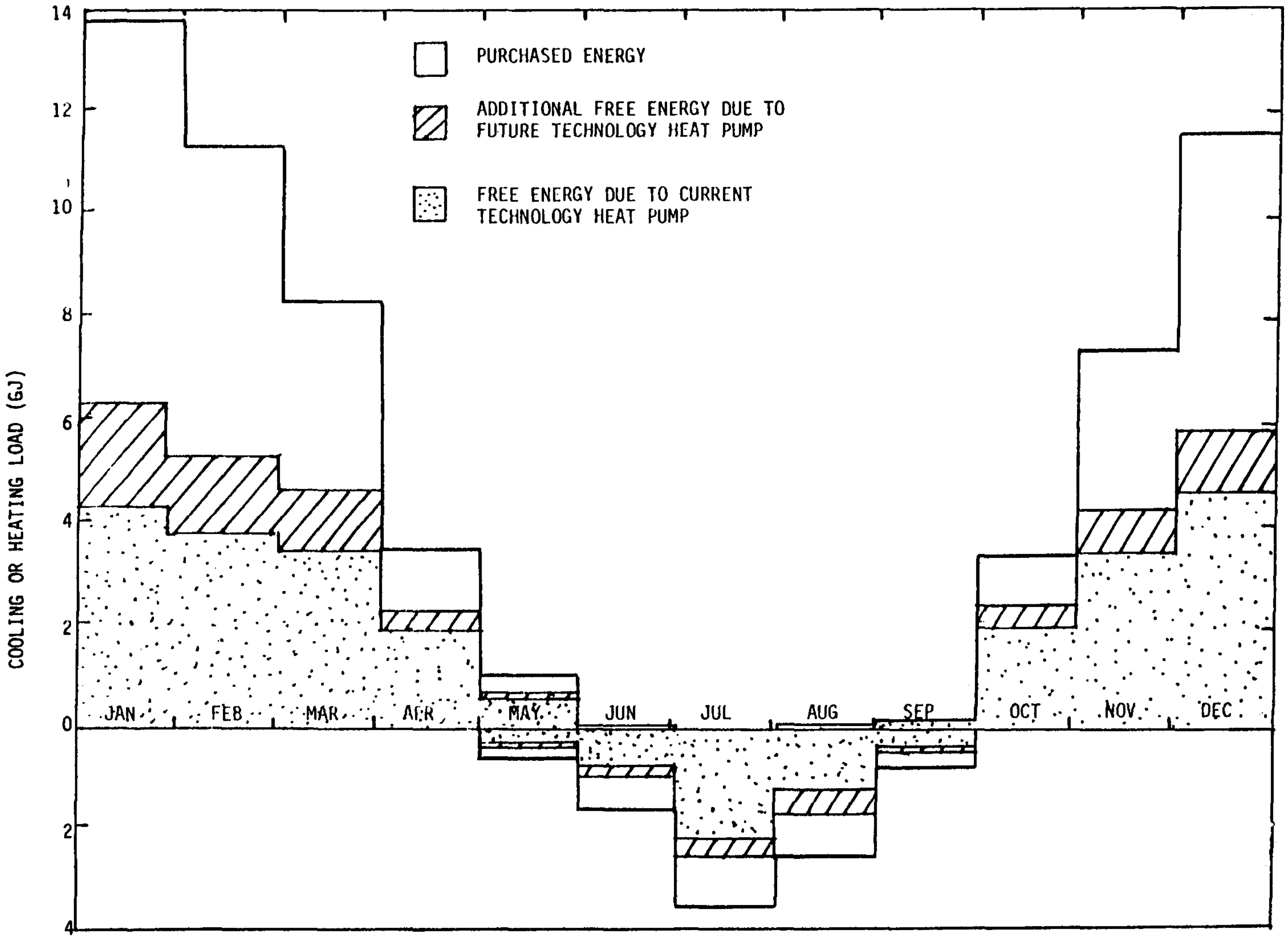


Figure 6-15. Monthly Stand-Alone Heat Pump Performance: Washington, D.C.

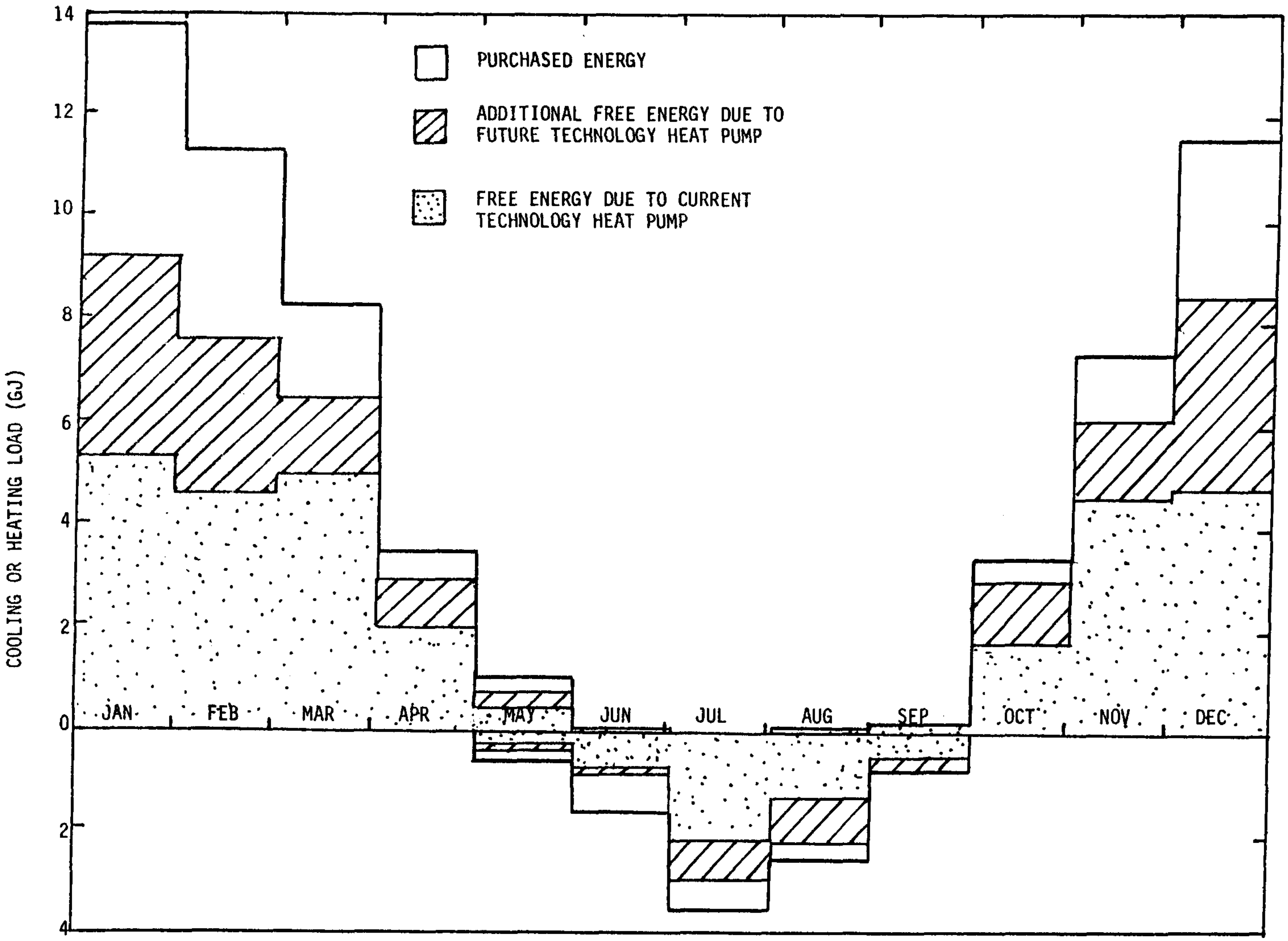


Figure 6-16. Monthly Liquid Series System Performance: Washington, D.C.

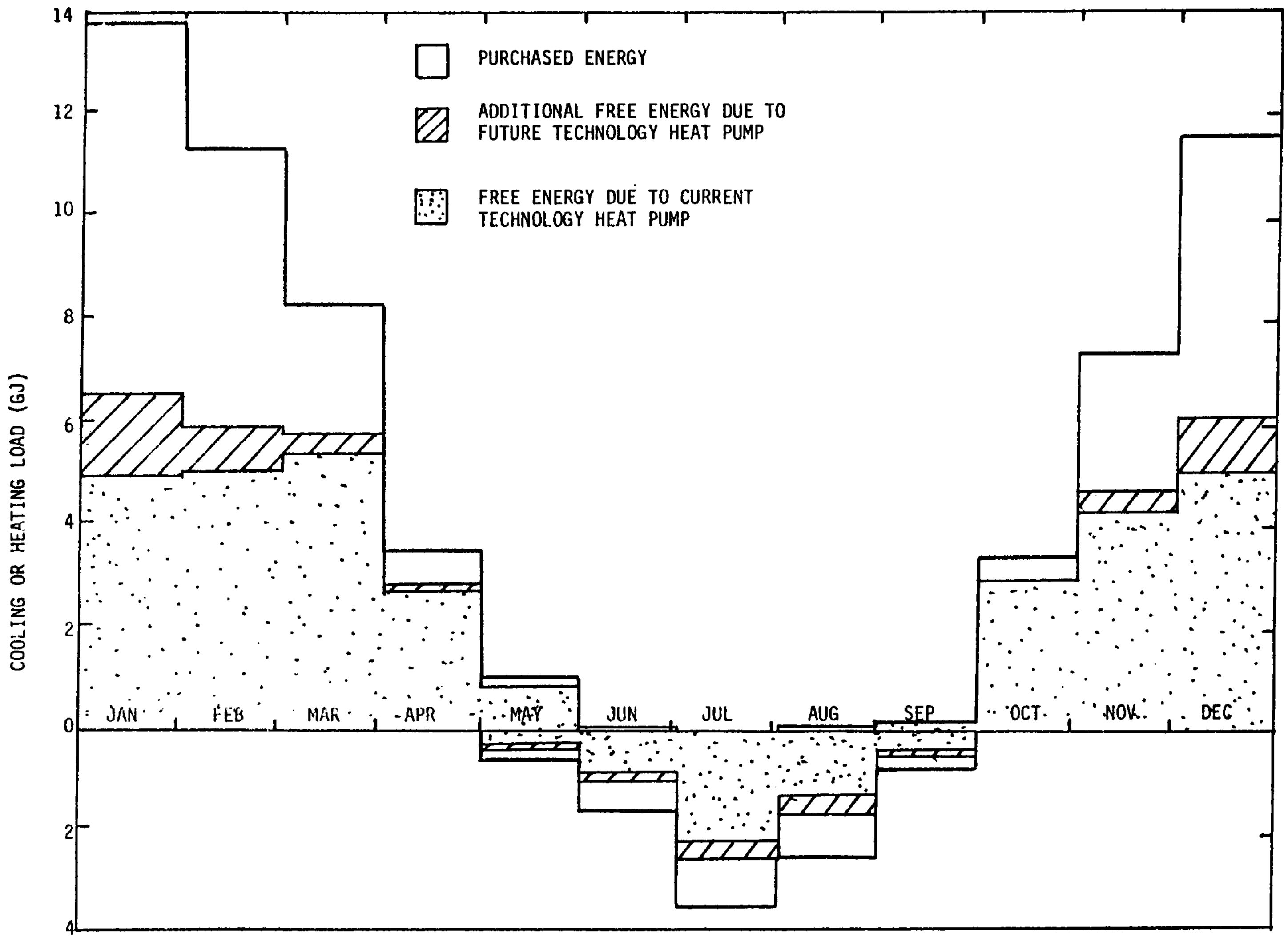


Figure 6-17. Monthly Liquid Parallel System Performance: Washington, D.C.

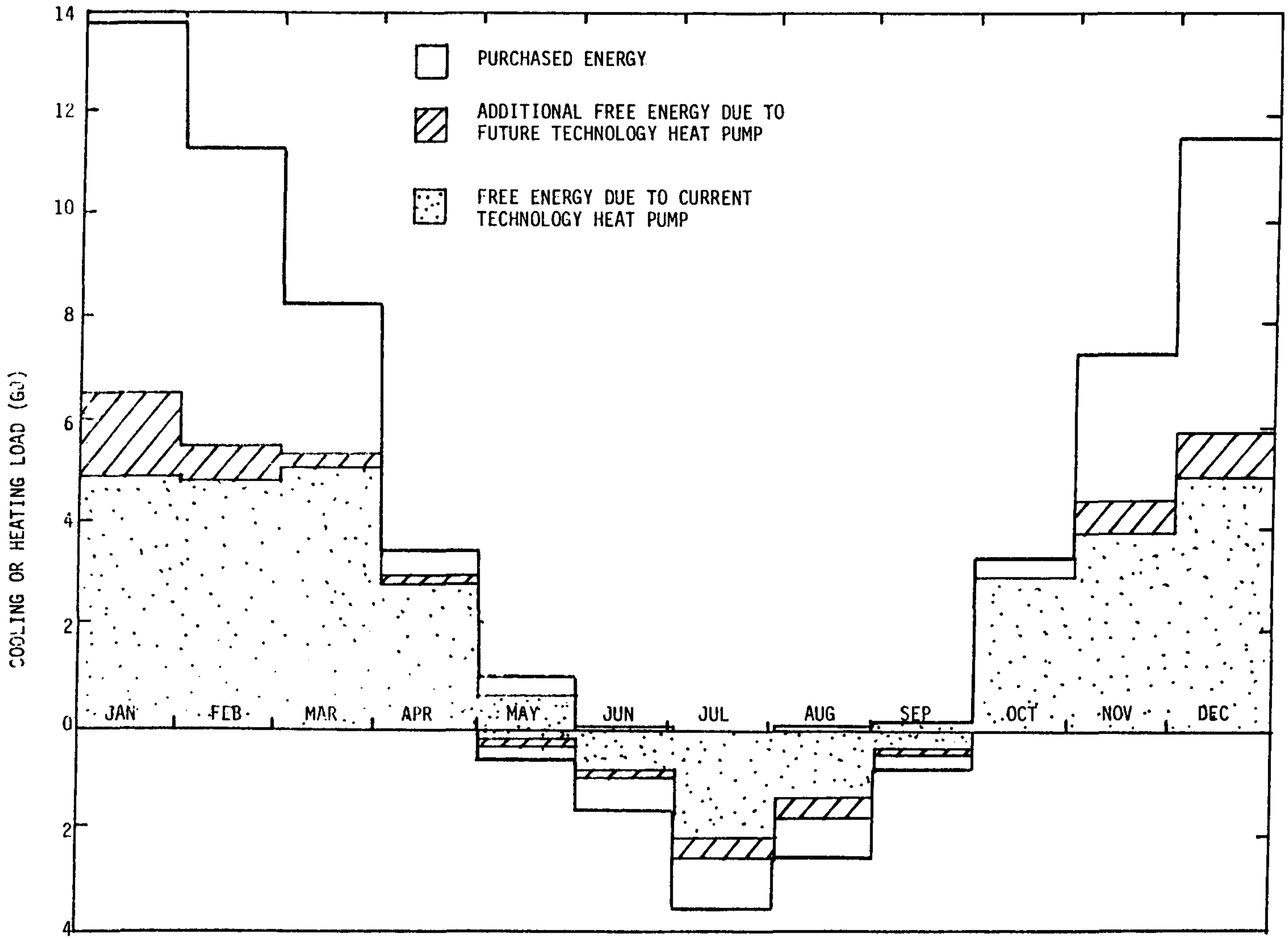


Figure 6-18. Monthly Air Parallel System Performance: Washington, D.C.

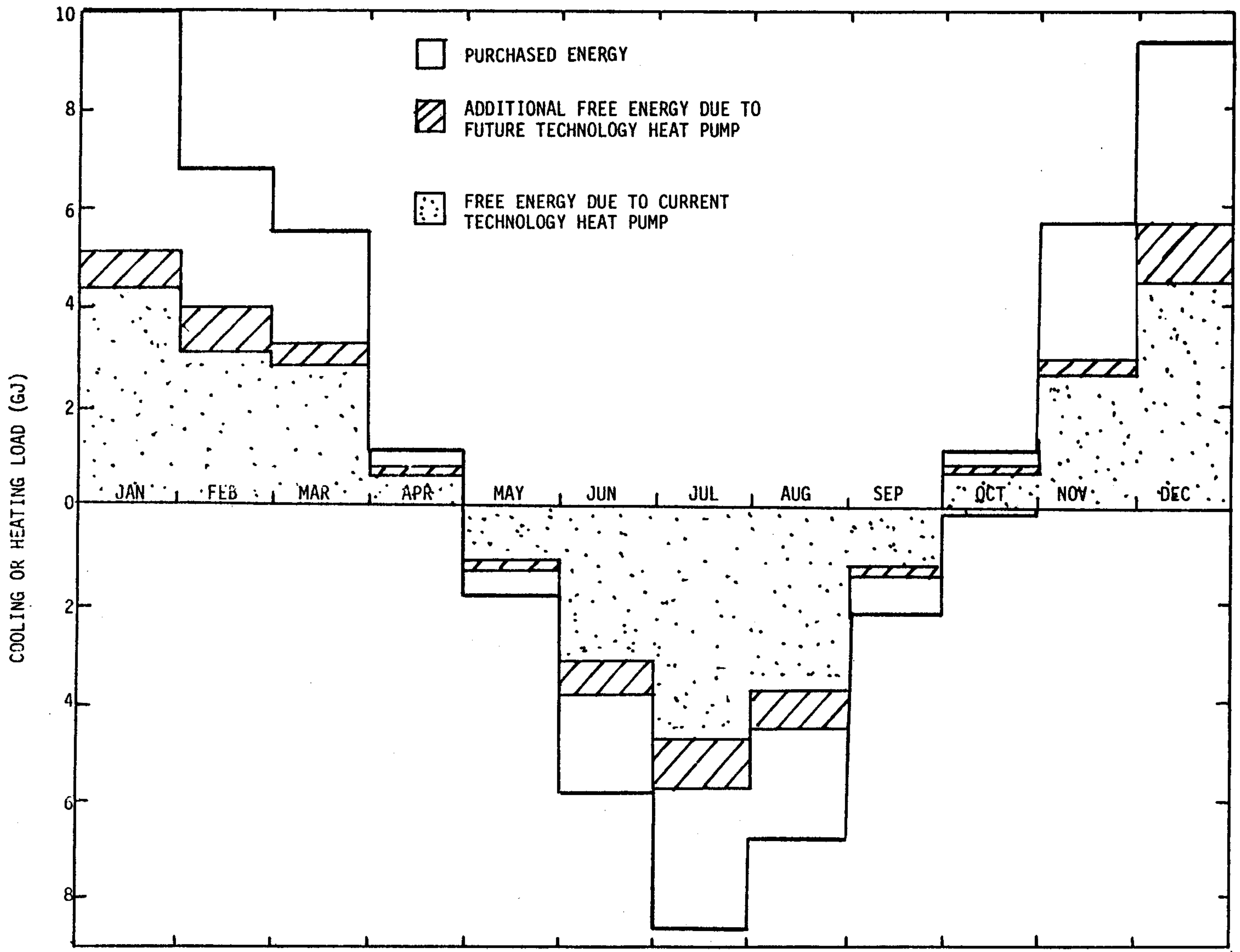


Figure 6-19. Monthly Stand-Alone Heat Pump Performance: Fort Worth, Texas

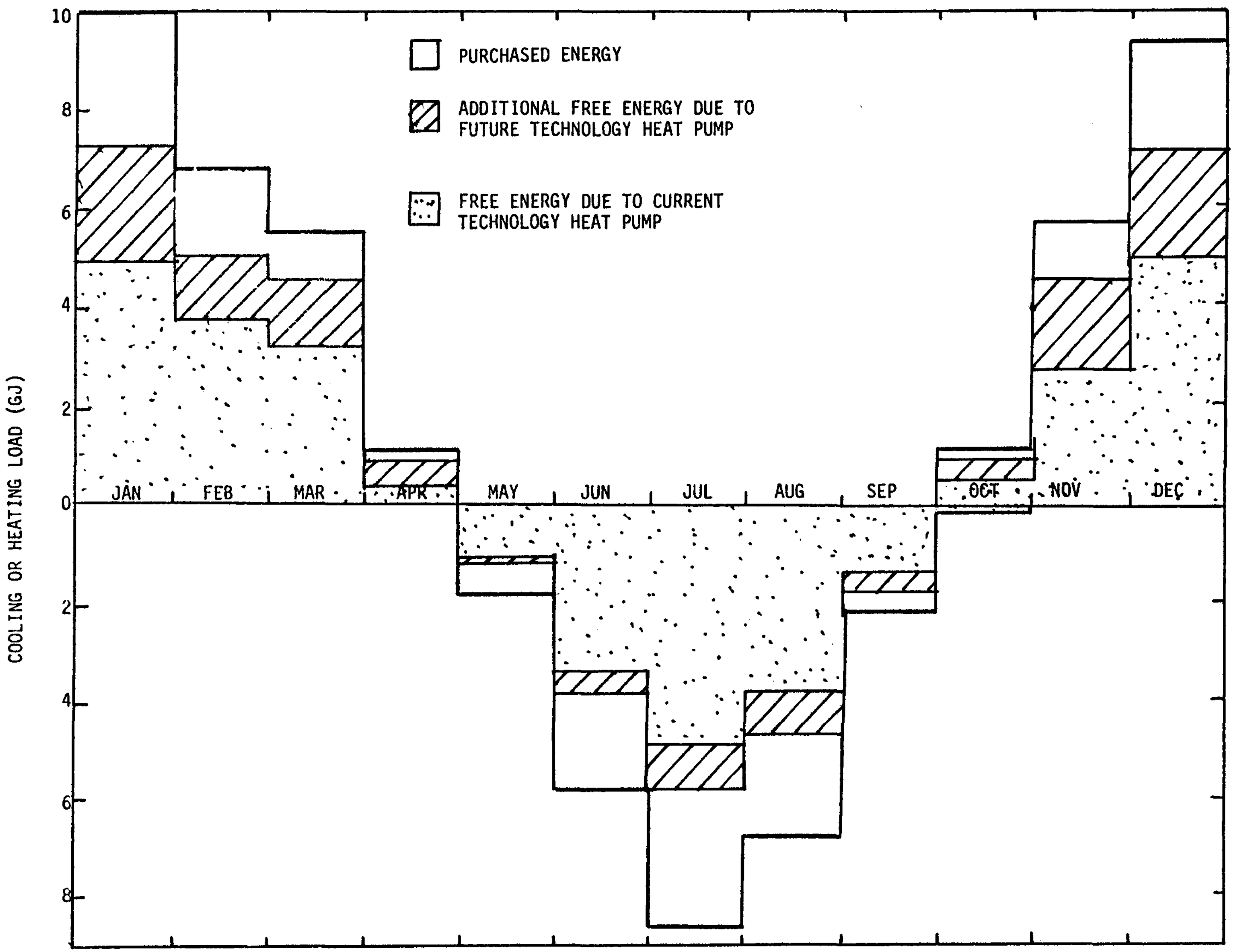


Figure 6-20. Monthly Liquid Series System Performance: Fort Worth, Texas

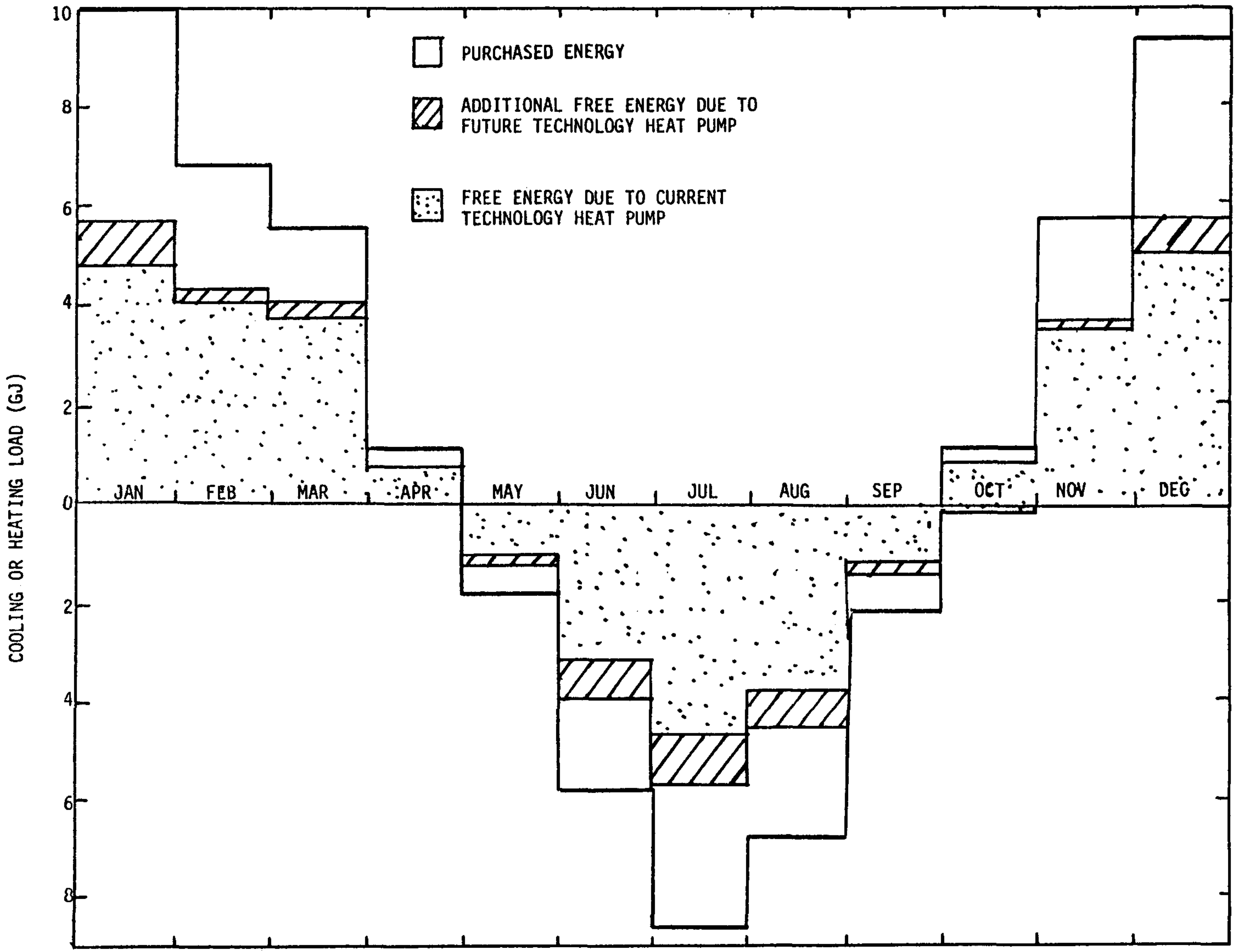


Figure 6-21. Monthly Liquid Parallel System Performance: Fort Worth, Texas

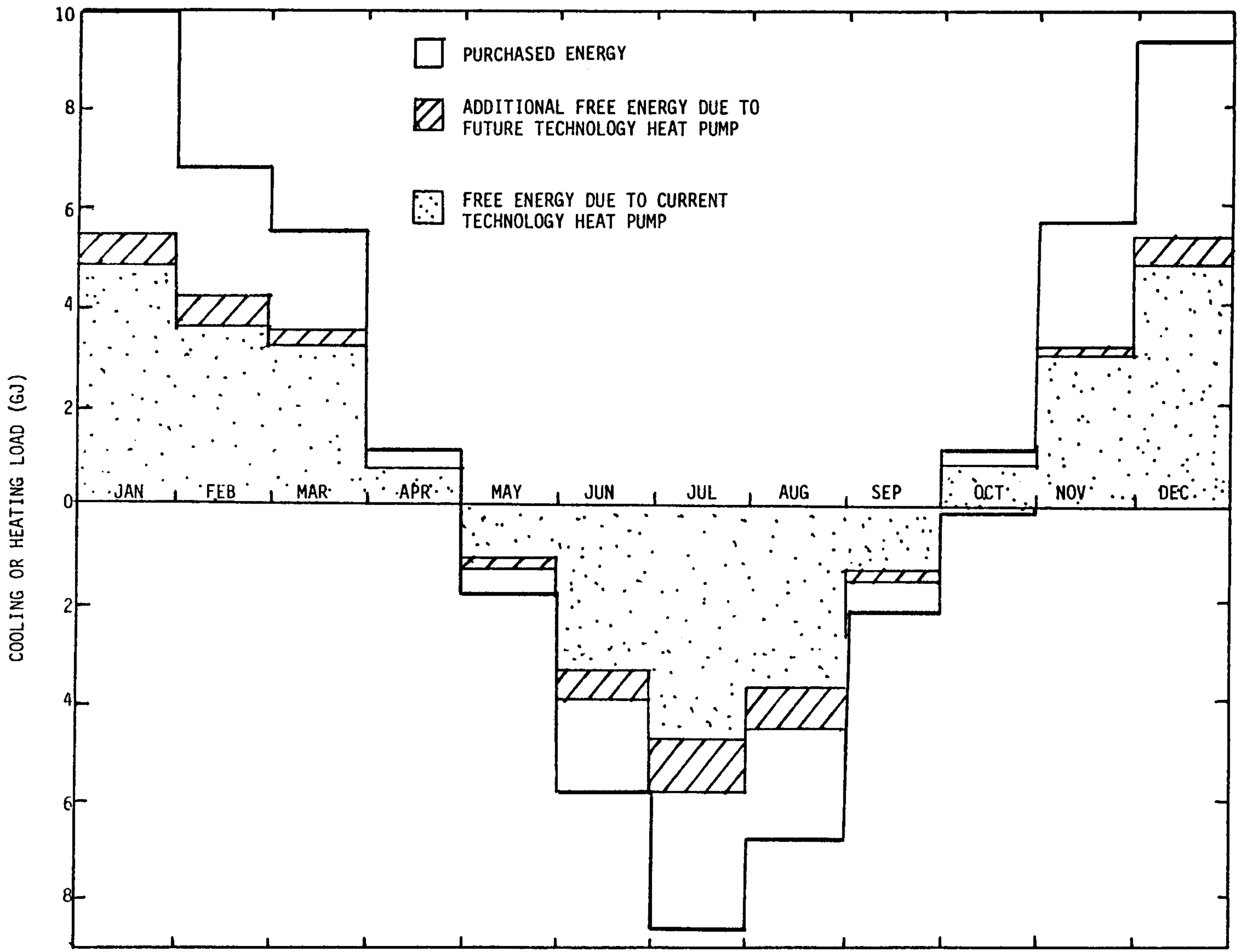


Figure 6-22. Monthly Air Parallel System Performance: Fort Worth, Texas

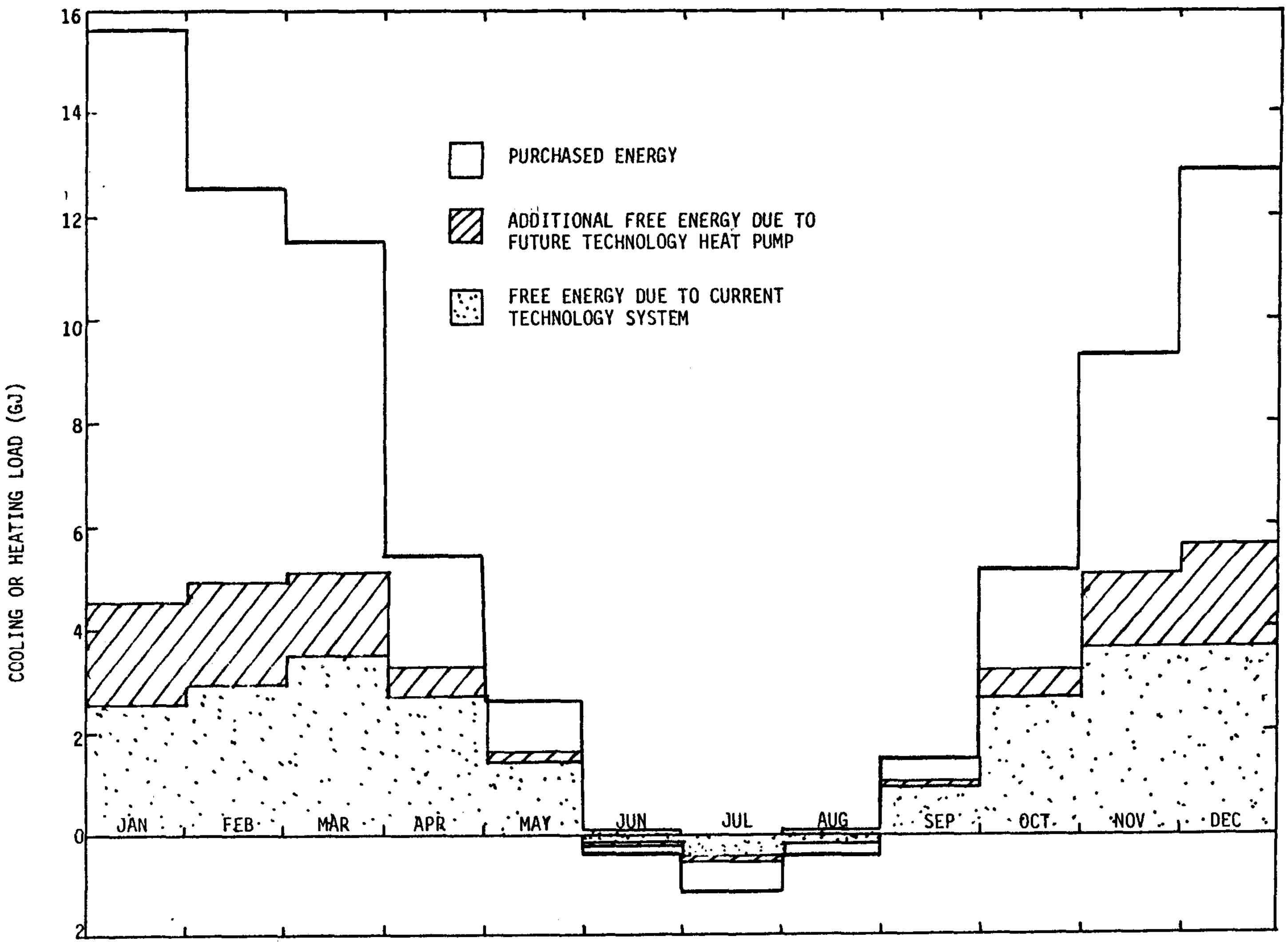


Figure 6-23. Monthly Stand-Alone Heat Pump Performance: Madison, Wisconsin

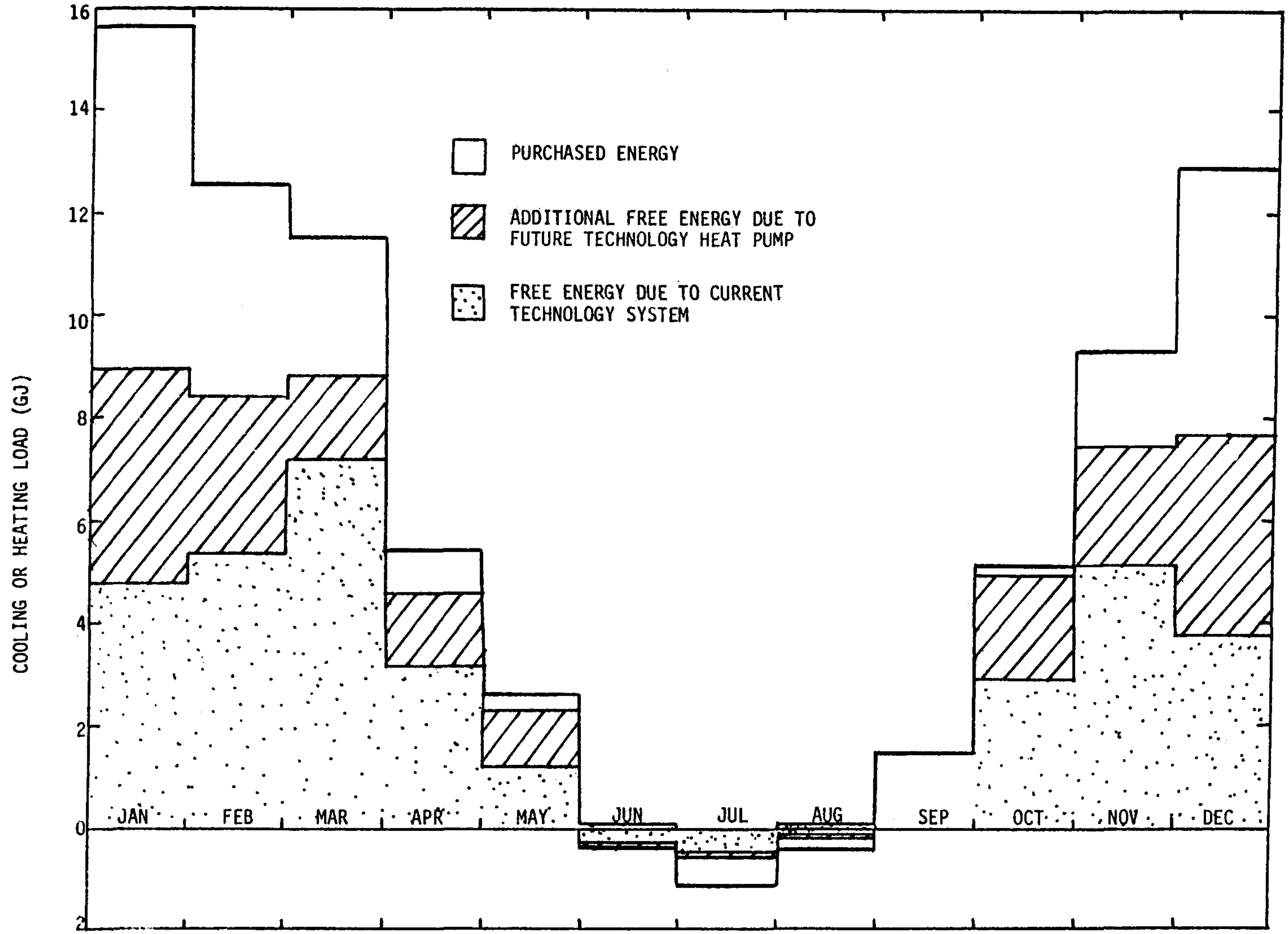


Figure 6-24. Monthly Liquid Series System Performance: Madison, Wisconsin

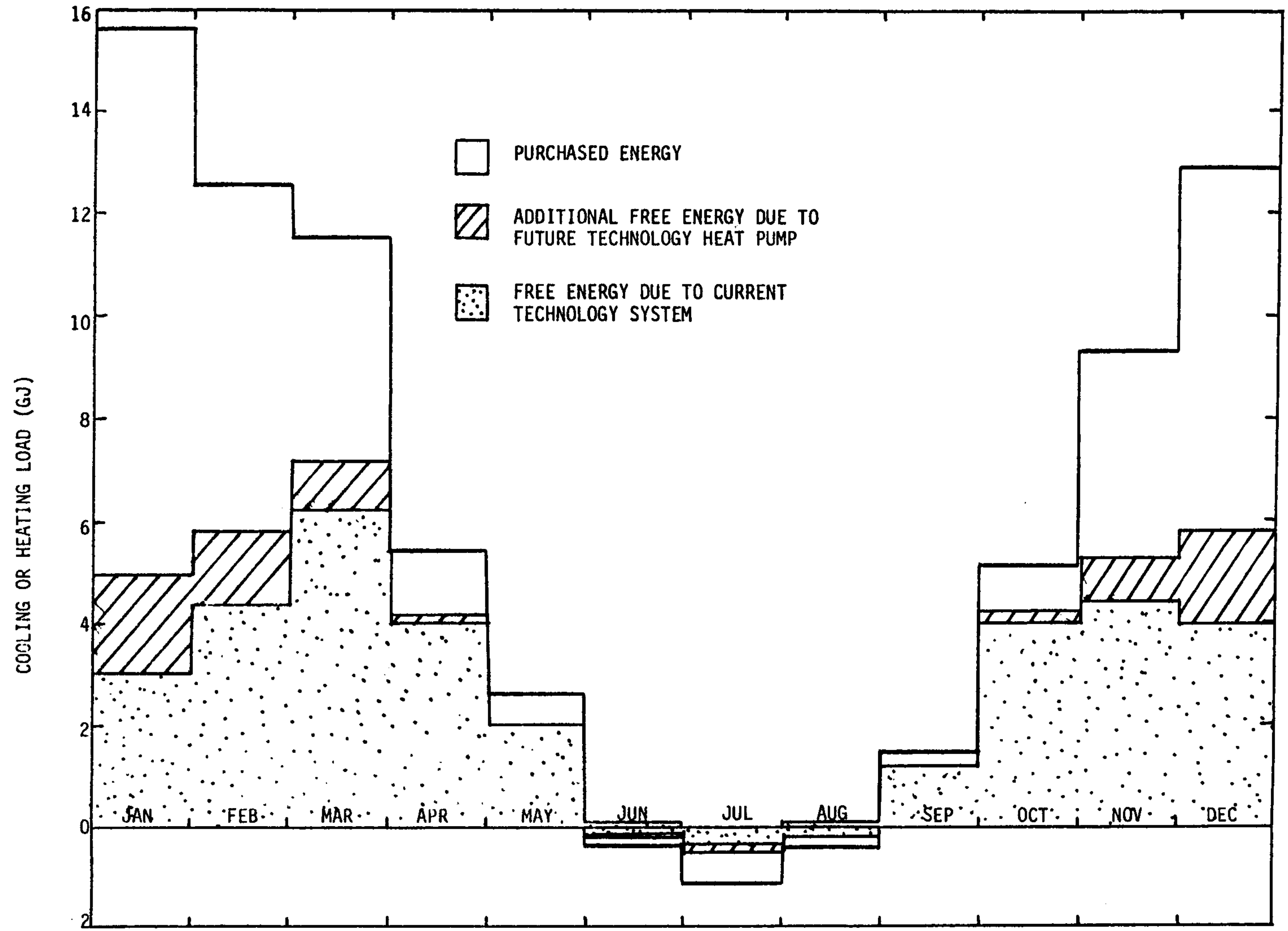


Figure 6-25. Monthly Liquid Parallel System Performance: Madison, Wisconsin

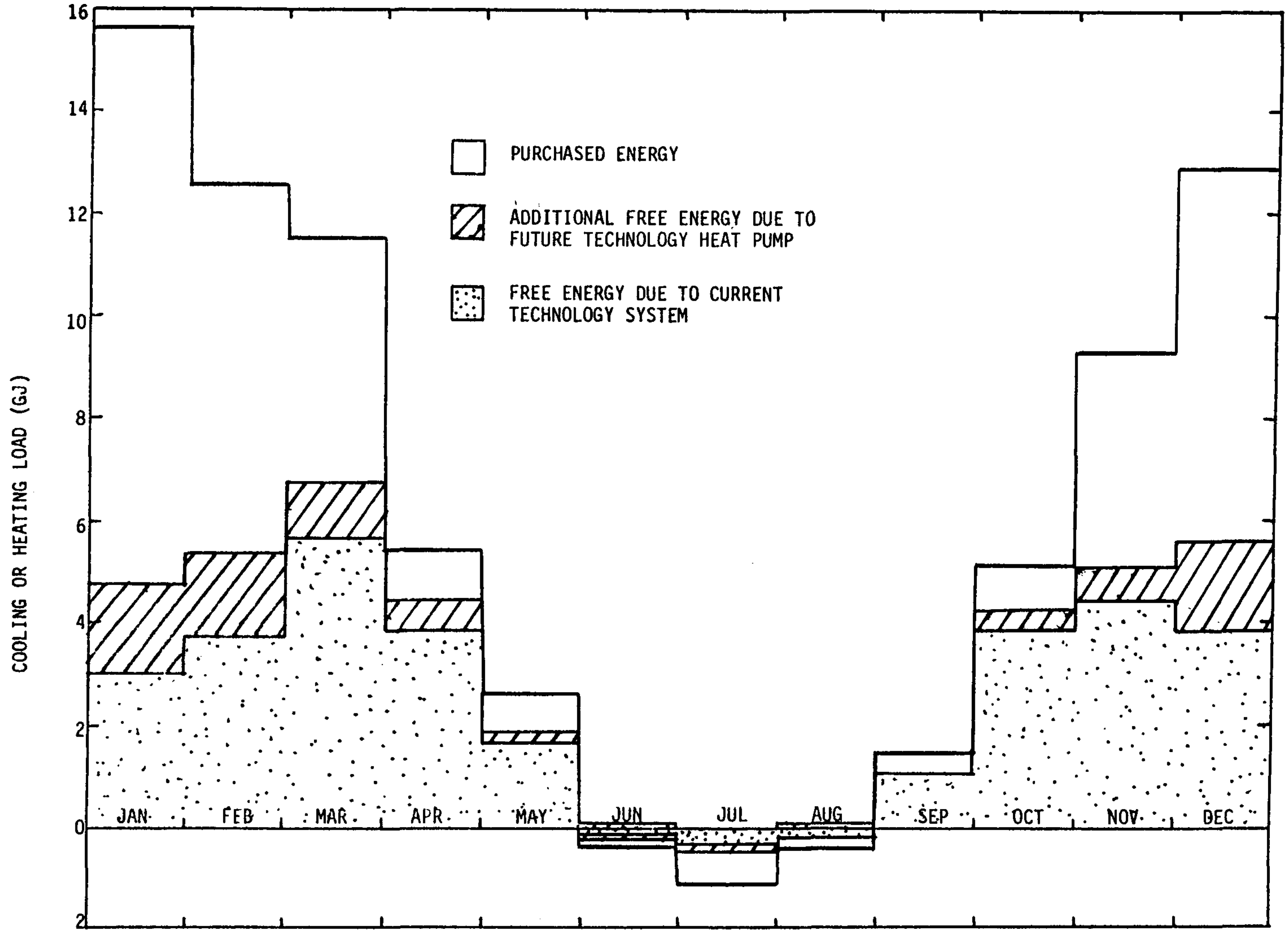


Figure 6-26. Monthly Air Parallel System Performance: Madison, Wisconsin

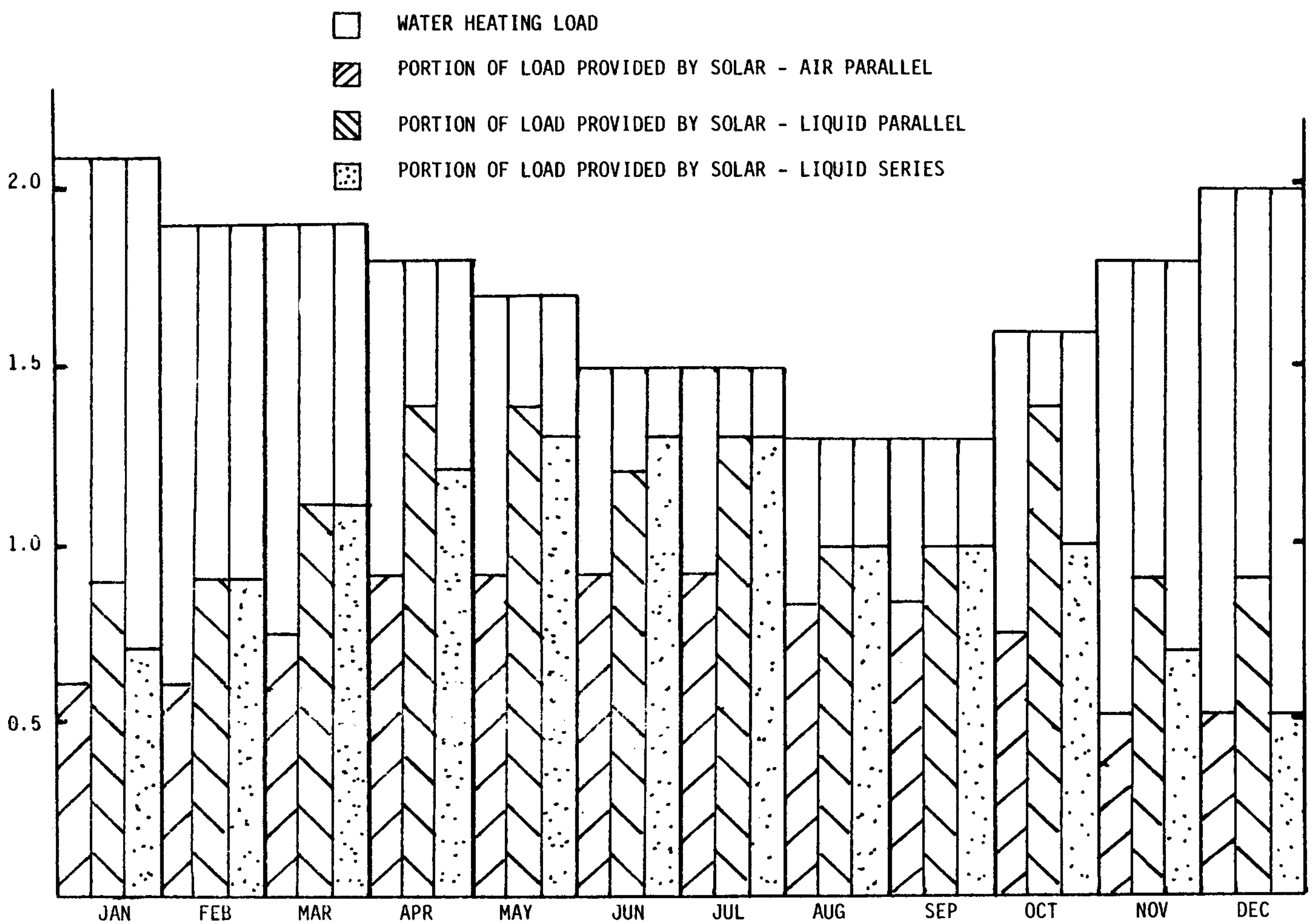


Figure 6-27. Monthly Solar Domestic Water Heating Performance: Washington, D.C.

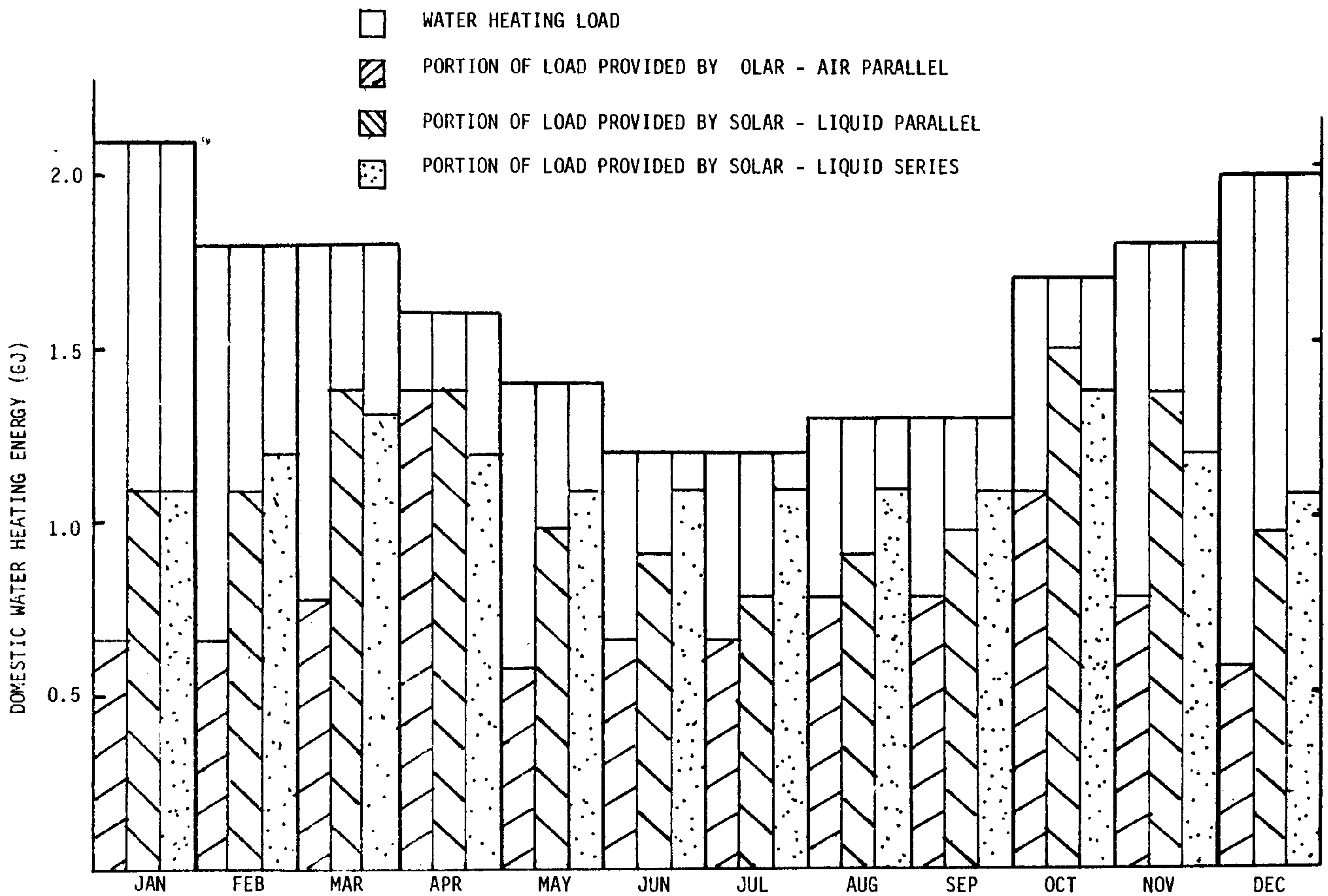


Figure 6-28. Monthly Solar Domestic Water Heating Performance: Fort Worth, Texas

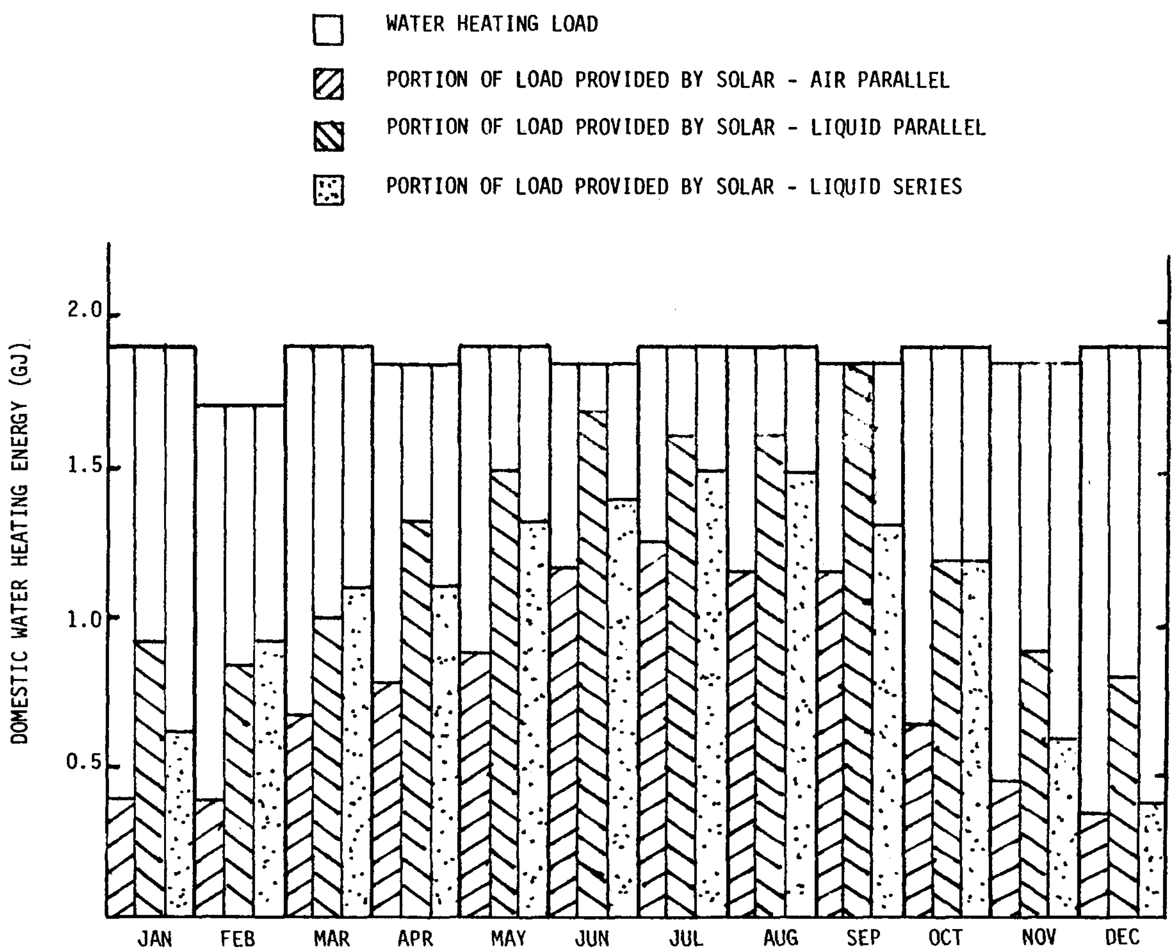


Figure 6-29. Monthly Solar Domestic Water Heating Performance: Madison, Wisconsin

6.1.5 Summary of Thermal Performance

Table 6-1 summarizes the thermal performance of each system. The total annual conventional energy requirement is given for each system. The portion of this requirement which must be electricity is given in brackets. The remainder (total minus portion which must be electrical) represents thermal energy which must be delivered to the load. For systems which use electric auxiliary, this equals the amount of electrical energy which must be purchased. For gas or oil burning systems, divide by the heater efficiency to obtain the energy quantity which must be purchased.

Note that Table 6-1 summarizes the performance of optimally sized solar heat pump systems. The collector arrays for the various systems are not, in general, the same size and the initial costs are not equal.

6.2 SUBSYSTEM SUMMARIES

This section displays several key characteristics of the combined solar heat pump systems in order to aid in understanding their operation. The stand-alone heat pump and conventional systems are not included as they are well understood.

6.2.1 Liquid Parallel System

Figures 6-30 through 6-32 display several characteristics of the liquid parallel system on a monthly basis. Since the heat pump is merely a backup device to the solar system, the future technology heat pump does not affect the performance of the solar system; therefore, only one plot is presented for each location.

In all locations, the operating collector efficiency hovers between 20 and 40 percent throughout the year. In winter, the tank temperatures remain near 40.2°C, below which the tank cannot be used for space heating. The tank reaches the 90°C level in summer when water heating is the only load.

6.2.2 Air Parallel System

Figures 6-33 through 6-35 display several characteristics of the air parallel system. As with liquid parallel systems, the quality of the heat pump does not affect solar system performance, so one figure is sufficient for each location. Note that the pebble bed is bypassed in the summer while the

Table 6-1

Conventional Energy which must be Delivered to the System and Load for Heating (Compressor Input Plus Thermal Auxiliary Requirement), Cooling (Compressor Input), Water Heating (Thermal Auxiliary Requirement), and Parasitics (Electrical).

Systems / Locations	Conventional Systems	Stand Alone Heat Pump	Solar/Heat Pump Systems			
			Liquid Series	Liquid Parallel	Air Series	Air Parallel
Washington, D.C. Today	85.0(4.9)	60.4(23.2)	42.5	34.8	-	42.5
Future	-	54.4(25.8)	21.2	31.4	26.1	39.1
Madison, WI Today	101.2(2.3)	78.2(24.4)	52.6	45.5	-	56.7
Future	-	67.7(36.0)	27.3	41.5	-	50.6
Fort Worth, TX Today	69.1(11.9)	50.1(23.8)	32.5	24.2	-	30.4
Future	-	43.6(20.2)	13.8	22.1	-	27.6

Note: The amounts in parentheses represent energy inputs which must be electrical (units are GJ).

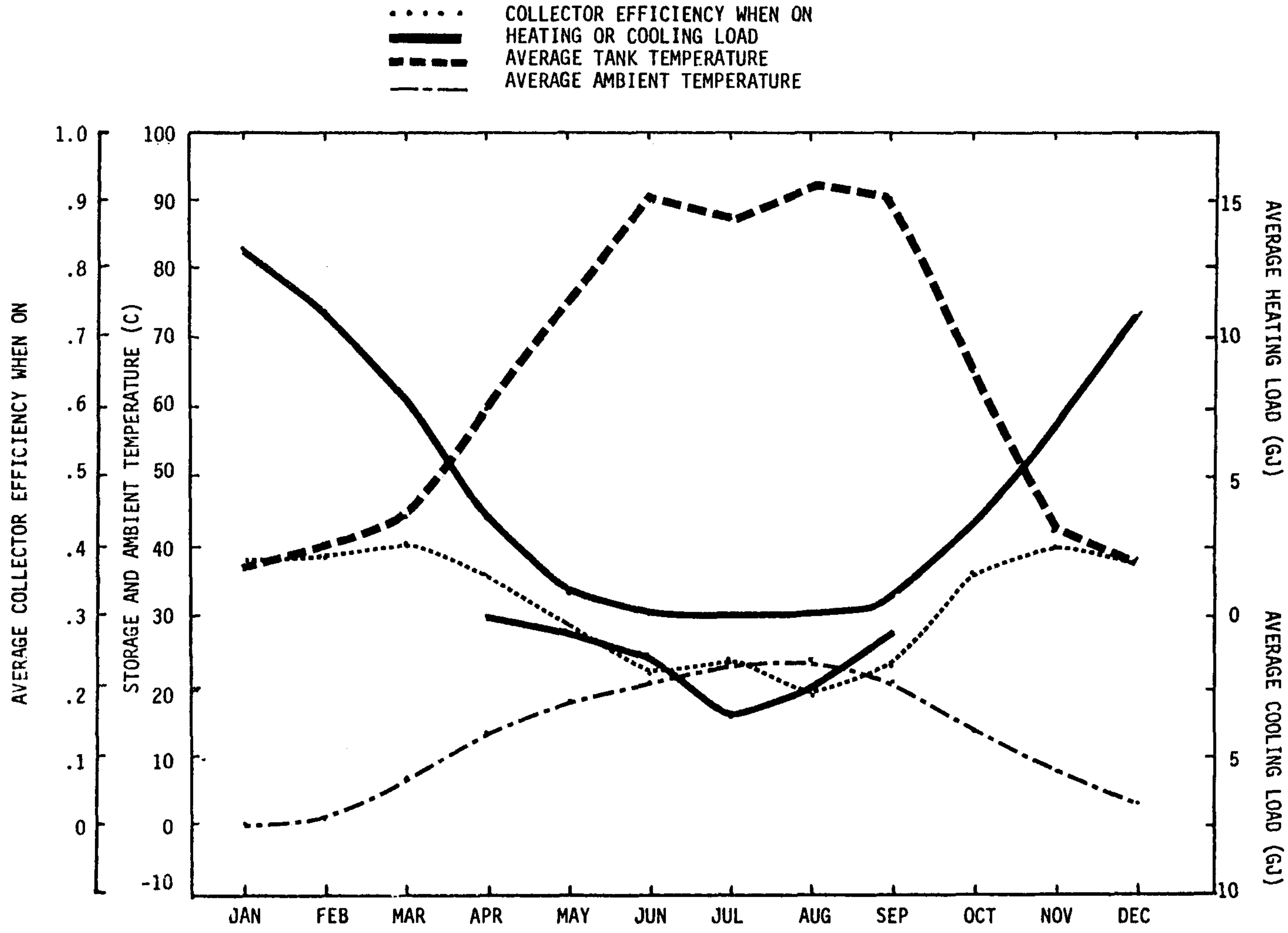


Figure 6-30. Subsystem Parameters: Liquid Parallel (Today's Tech): Washington, D.C.

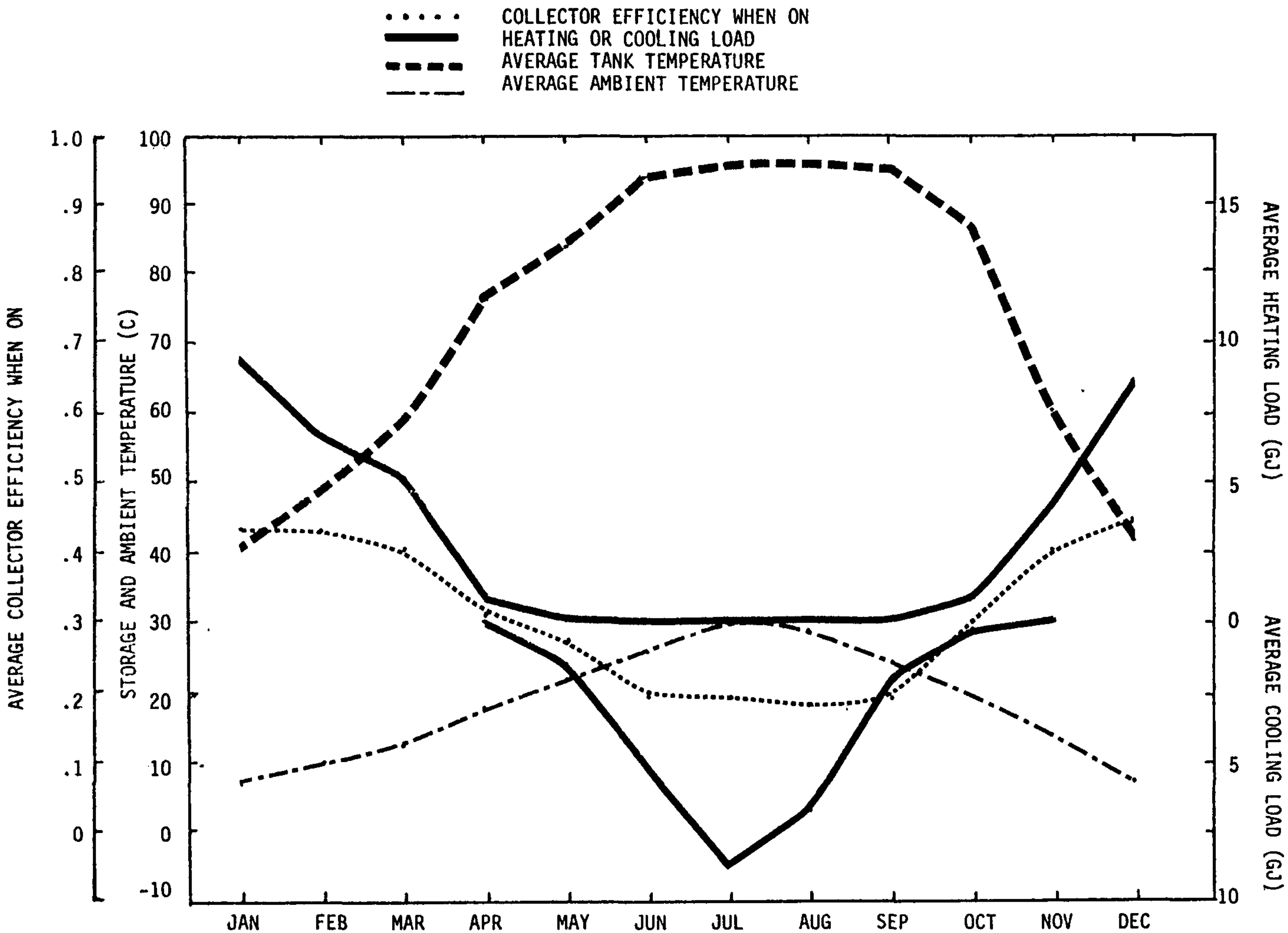


Figure 6-31. Subsystem Parameters: Liquid Parallel (Today's Tech): Fort Worth, Texas

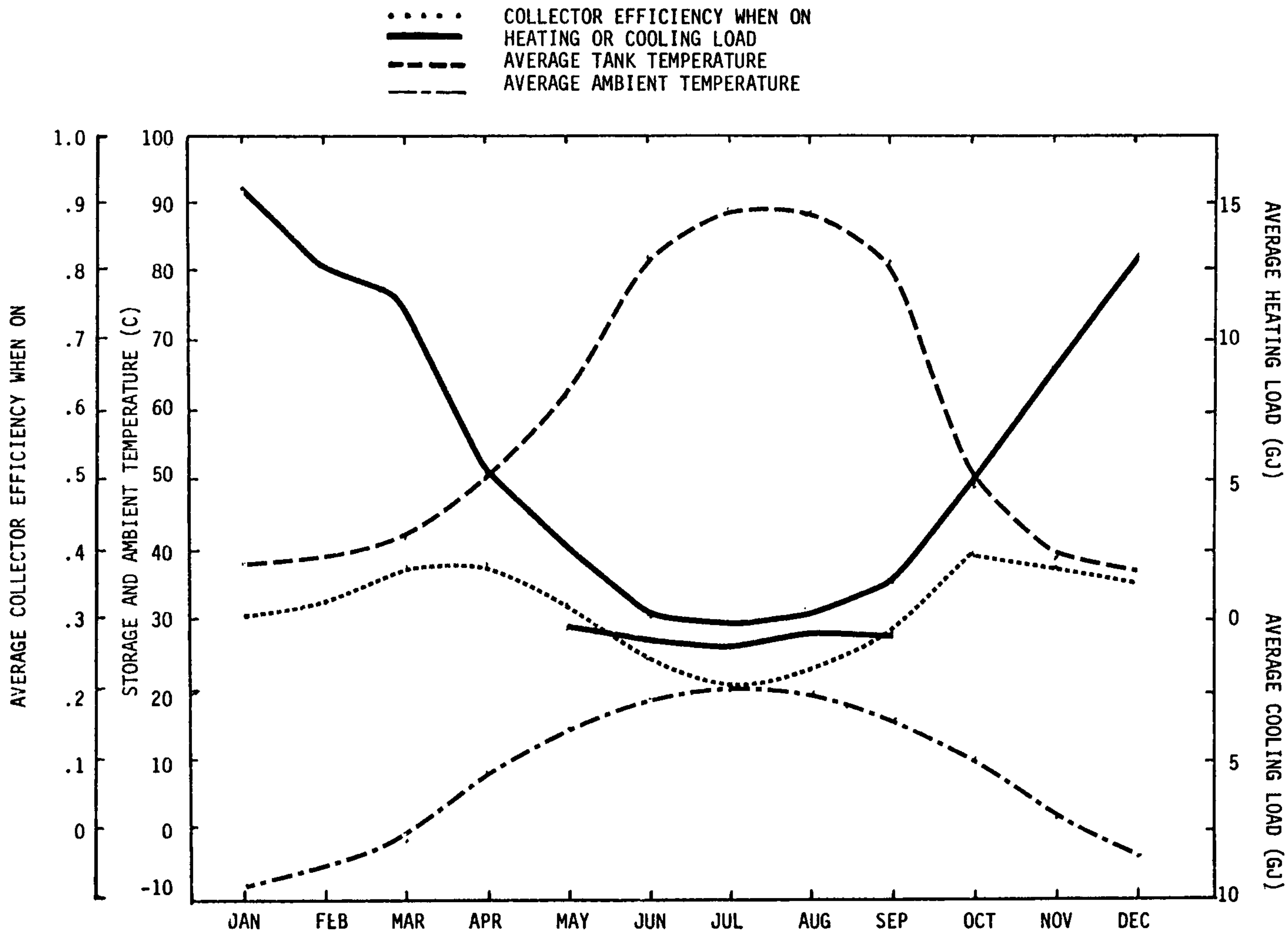


Figure 6-32. Subsystem Parameters: Liquid Parallel (Today's Tech): Madison , Wisconsin

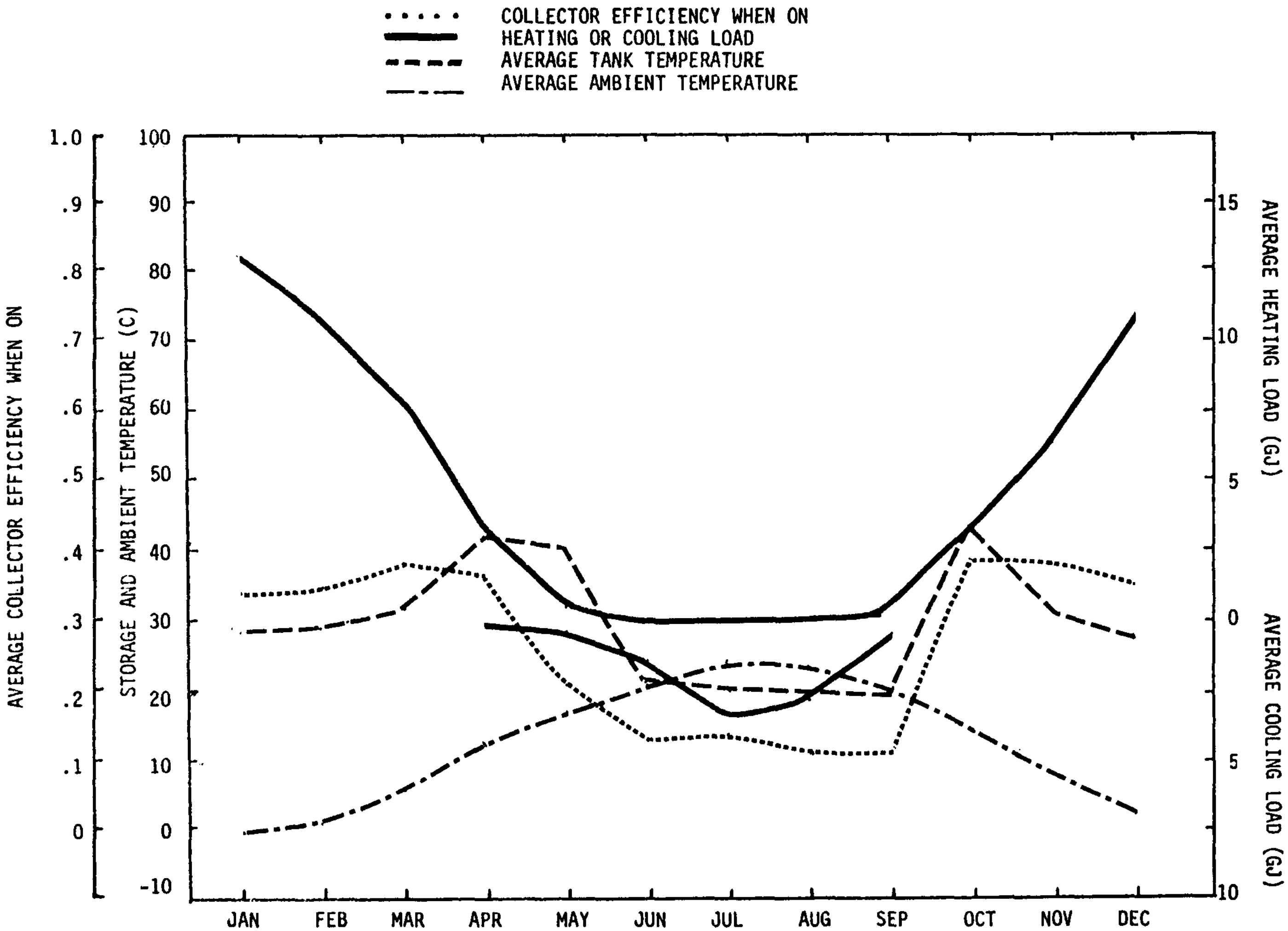


Figure 6-33. Subsystem Parameters: Air Parallel (Today's Tech): Washington, D.C.

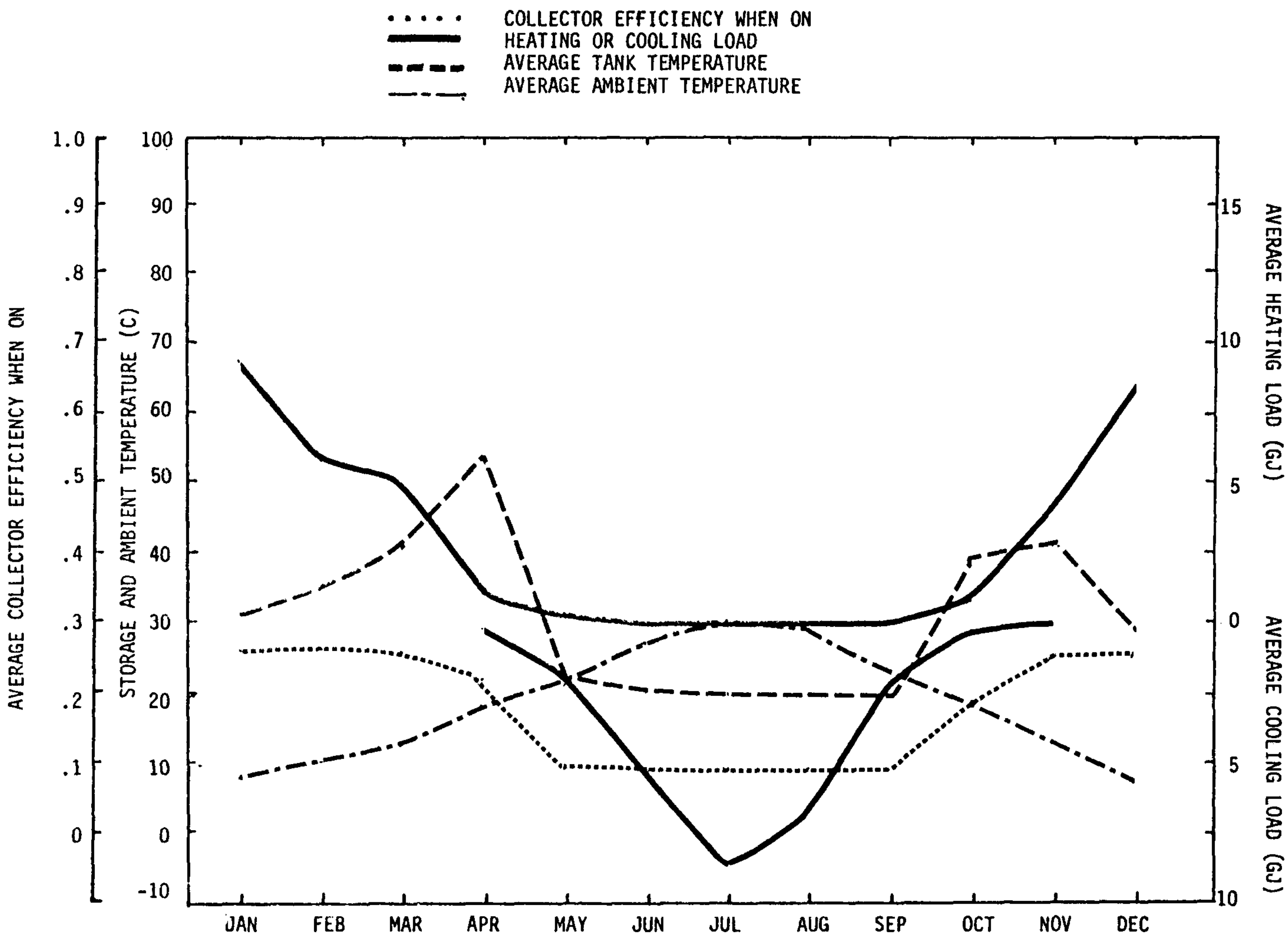


Figure 6-34. Subsystem Parameters: Air Parallel (Today's Tech): Fort Worth, Texas

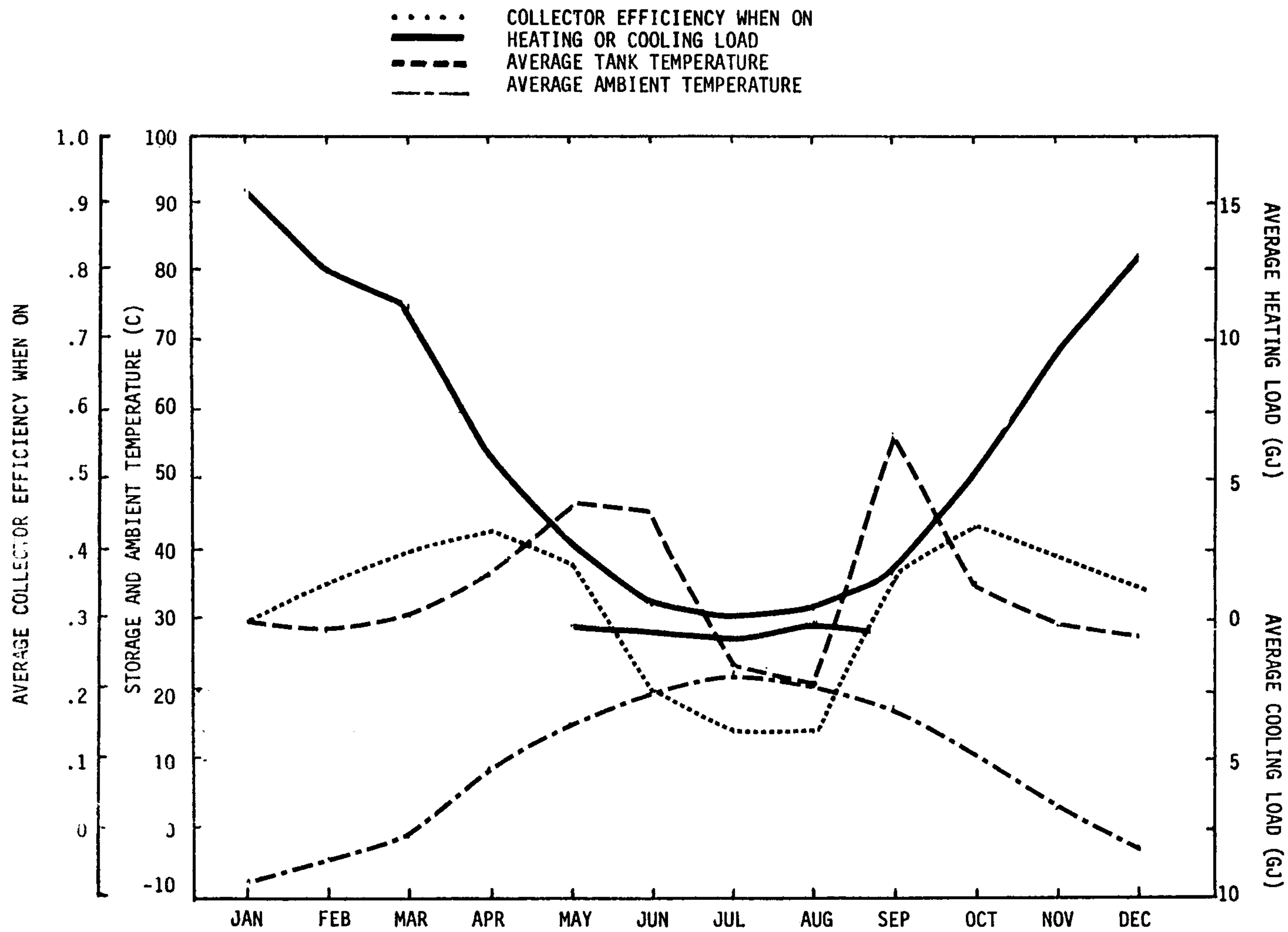


Figure 6-35. Subsystem Parameters: Air Parallel (Today's Tech): Madison, Wisconsin

collectors operate to supply domestic hot water. Consequently, storage approaches room temperature. The average pebble bed temperature seldom exceeds 35^{o}C in winter, the temperature at which energy must be supplied to the load. Collector efficiencies are very low in the summer when used exclusively for water heating since the preheat tank is maintained at a high temperature. In winter, collector efficiencies reach 25 to 40 percent, depending on storage temperature and location.

6.2.3 Liquid Series System

Figures 6-36 through 6-38 display how operating collector efficiency and average tank temperature are affected by the future technology heat pump in the liquid series system. The improved liquid-to-air heat pump has a source temperature operating range which extends both above and below that of current liquid-to-air heat pumps. The improved performance of the overall system due to the improved heat pump causes the system to optimize at a larger collector area than with the current heat pump. Depending on location, the larger collector area and higher winter collection efficiency (due to the heat pump being able to draw storage down to a lower temperature) more than counteract the fact that the improved heat pump puts a large thermal load on the solar system (since compressor work is lower). In Fort Worth, the optimally sized system never starved the heat pump. In Madison and Washington, D.C., starvation occurred but it appears that enough collectors could be installed to prevent this if an economic incentive (such as demand charge electrical rates) were provided. Care should be taken when making this judgment since it is unlikely that the TMY data contain the "worst" period of weather ever experienced in each location.

Figures 6-36 through 6-38 contain an apparent anomaly in spring and fall when the improved heat pump system reported both a higher collection efficiency and a higher storage temperature. However, it must be remembered that the current heat pump system is not allowed to elevate storage above 33.9^{o}C (see subsection 6.1.1). The collectors operate but dump energy through the relief valve. Since the definition of operating collector efficiency is the useful energy collected over the incident solar energy when the system operates, the calculated collector efficiency for the current heat pump system is artificially low.

Note that significant direct solar heating occurred in the spring and fall even through the series system used low-quality collectors. In the winter, storage temperatures were low and the collectors operated very effectively.

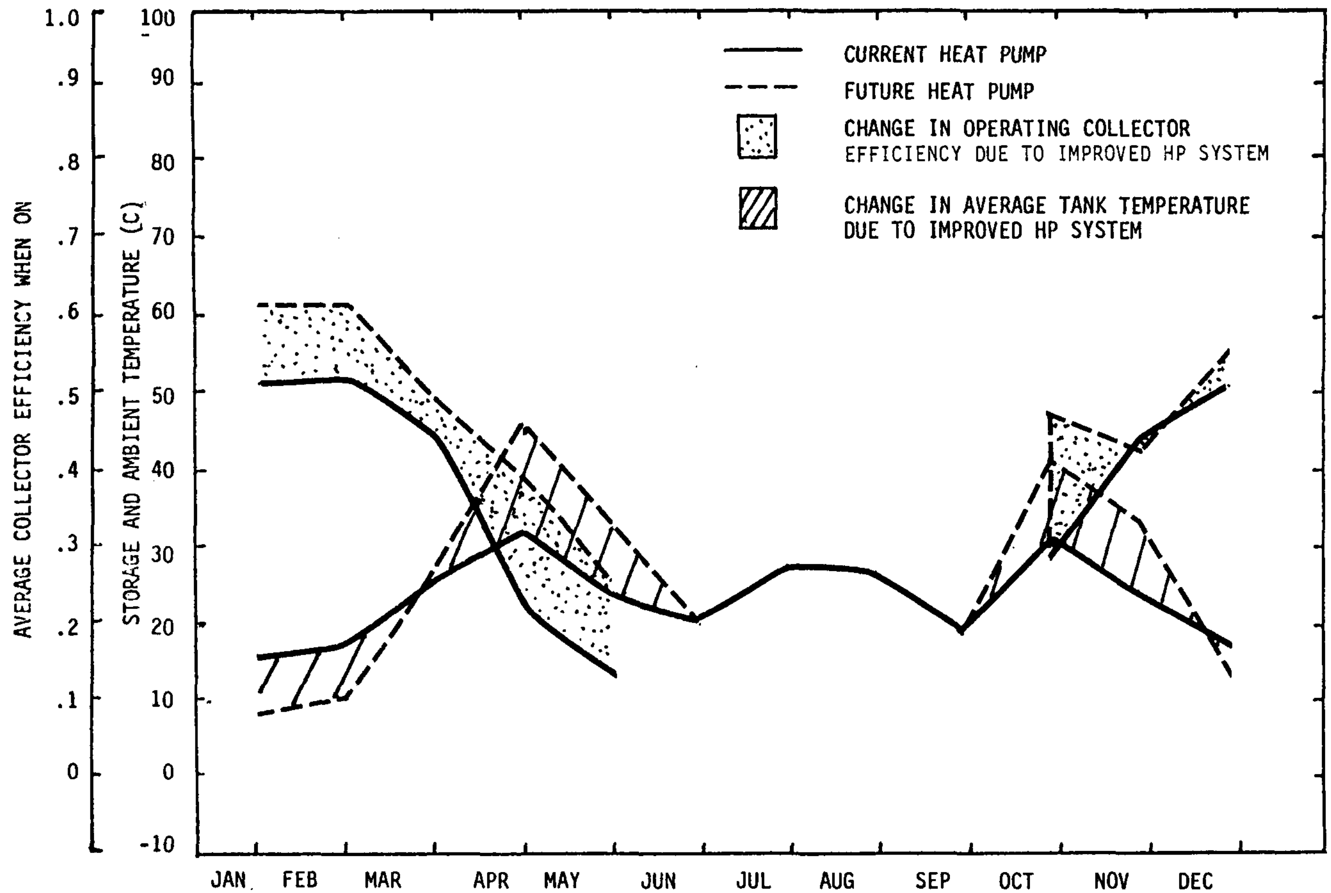


Figure 6-36. Subsystem Parameters: Liquid Series (Today/Future): Washington, D.C.

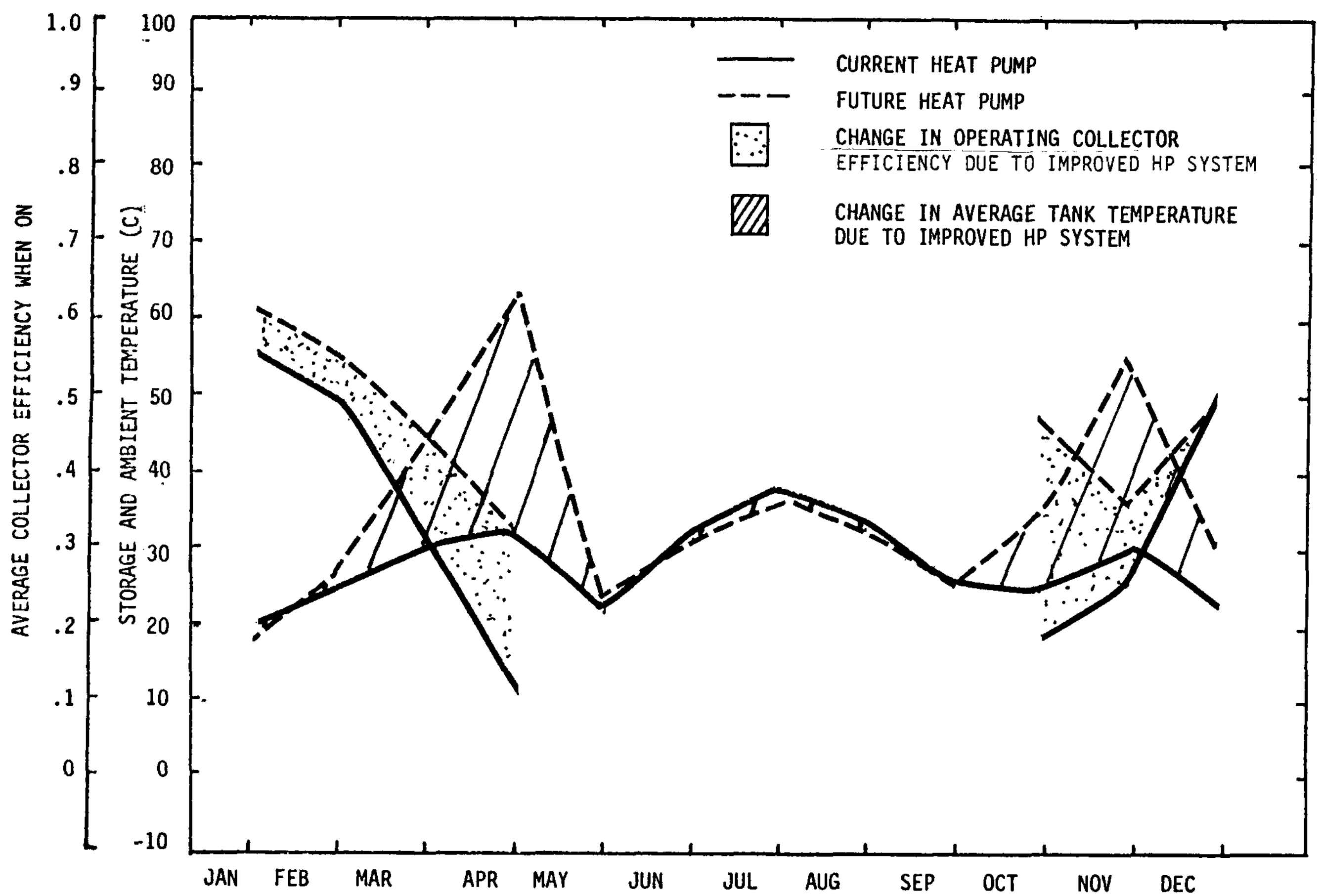


Figure 6-37. Subsystem Parameters: Liquid Series (Today/Future): Fort Worth , Texas

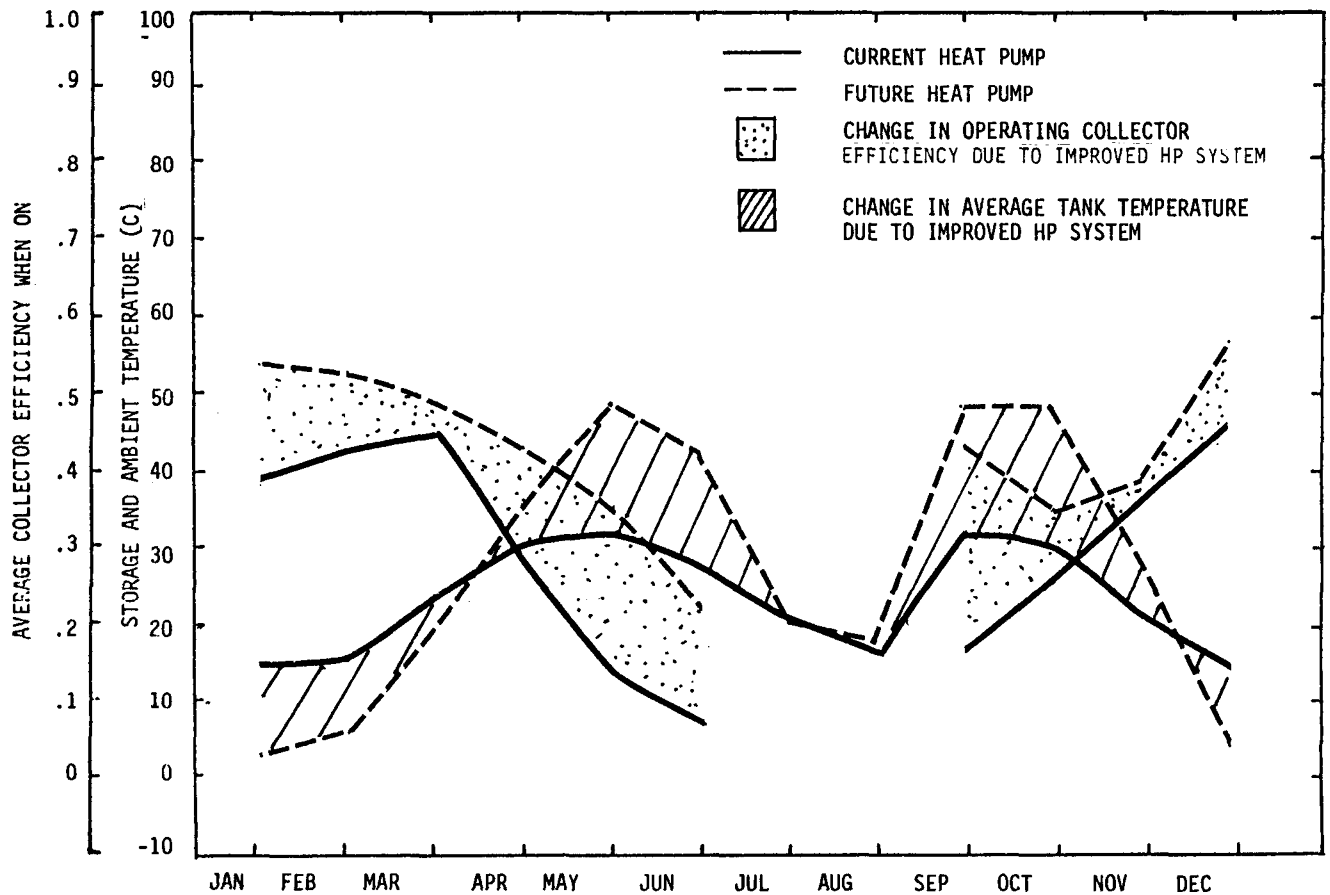


Figure 6-38. Subsystem Parameters: Liquid Series (Today/Future): Madison, Wisconsin

6.3 ECONOMICS, THERMAL COMFORT, RELIABILITY, AND UTILITY IMPACT

The section is concerned with the results of the nonthermal characteristics of the systems. While these characteristics are generally difficult to deal with, the simulation models used in this study yielded results which can be directly used for many nonthermal issues.

6.3.1 Economics

The bases for the economics of the various systems are found in Section 4. The life-cycle costing methodology is described, along with the particular system costs in terms of fixed and variable costs (see Table 4-3). The capital cost of each combined solar heat pump system was calculated using the fixed/variable costs associated with the optimum collector area, as is determined in subsection 7.3. The total acquisition (initial) cost for each of the systems under investigation is presented in Table 6-2. If a separate set of economic assumptions is to be used, the thermal performance maps for the various solar systems are presented in Figures 6-1 through 6-14, while Table 6-1 summarizes the thermal performances of the systems as they were utilized in this study.

The present value life-cycle costs of each of the conventional and solar systems was calculated with the economic parameters listed in Table 4-1. The results of these calculations are presented in Table 6-3. As can be expected, the natural gas systems have the lowest lifecycle costs among the conventional furnace-type systems, even with a gas fuel escalation rate of 12 percent per annum over a 20-year period. The stand-alone heat pump systems have the lowest life-cycle costs of any of the systems investigated and have payback periods of around eight years when compared to the oil and electric furnace-type conventional systems. In fact, in Washington, the stand-alone heat pump has a life-cycle close to natural gas furnaces.

Among today's technology solar heat pump systems, both series and parallel, the life-cycle costs at a given location were found to be within approximately 15 percent of one another for optimally-sized systems. Future-generation heat pumps will improve the series configuration economics by approximately 7.5 percent in Fort Worth and Madison, while not affecting Washington-based series systems' costs. The liquid and air parallel systems'

Table 6-2

Conventional and Solar Heat Pump Systems' Capital (Acquisition) Costs, Including Overhead and Profit ($)

Systems / Locations	Gas + A/C	Oil + A/C	Electric + A/C	Stand Alone Heat Pump	Solar/Heat Pump Systems			
					Liquid Series	Liquid Parallel	Air Series	Air Parallel
Washington, D.C. Today	2,120	2,395	1,999	2,413	14,285	18,263	12,635	16,825
Future	-	-	-	2,913	15,672	17,875	14,591	16,434
Madison, WI Today	2,083	2,359	1,964	2,376	15,481	21,024	13,926	20,371
Future	-	-	-	2,876	17,050	19,984	15,449	18,820
Fort Worth, TX Today	2,047	2,312	1,907	2,329	12,545	14,487	11,167	13,614
Future	-	-	-	2,829	14,215	14,553	12,786	13,826

Table 6-3

Life-Cycle Present Value Costs of Conventional and Combined Solar Heat Pump Systems (Using Optimized) CSHP System Sizes)

Systems / Locations	Gas + A/C	Oil + A/C	Electric + A/C	Stand Alone Heat Pump	Solar/Heat Pump Systems			
					Liquid Series	Liquid Parallel	Air Series	Air Parallel
Washington, D.C. Today	$18,710	26,970	27,800	21,260	31,570	40,100	-	39,830
Future	-	-	-	20,280	29,280	38,420	34,650	38,150
Madison, WI Today	$17,170	35,350	30,190	24,630	35,535	47,020	-	48,900
Future	-	-	-	22,580	31,810	44,120	-	44,540
Fort Worth, TX Today	$12,025	24,785	21,930	17,320	25,070	30,010	-	30,200
Future	-	-	-	16,305	23,310	29,540	-	29,790

economics are not significantly affected by future-generation heat pumps. This seeming anomaly arises since future heat pumps are assumed to cost $500 more to purchase initially and the improved heat pump does not increase the energy-savings costs by that much.

The series and parallel solar heat pump systems both have life-cycle costs greater than the oil and electric furnace systems. Versus these oil and electric furnace systems, the CSHP systems have simple payback periods between 15 and 20 years in Washington and Madison, and greater than 20 years payback in Fort Worth.

6.3.2 Comfort

As discussed in subsection 5.3, forced air systems provide equal thermal comfort and can be compared without including the conventional ductwork in the economic analysis, as long as each system has the same load air flow rate and minimum delivered air temperature requirement. As explained in subsection 3.2.3, the TRNSYS program is capable of impressing these thermal comfort restraints on a system simulation if the "temperature level control" load/system interface technique is used. In the same discussion, it was demonstrated that precalculated loads are only acceptable for temperature level control simulations if the system subsequently modeled has a single-stage room temperature control, a narrow room temperature control band, and adequate capacity so that room temperature can be maintained in the control band. Since several systems in this study require two-stage controllers, loads and system performance have been calculated simultaneously in a fully interactive manner. The purpose of this section is simply to demonstrate that each system did provide comparable thermal comfort.

Figure 6-39 displays frequency histograms of delivered air temperature for future-technology heat pump systems in Washington, D.C. Similar histograms for Forth Worth and Madison can be found in Appendices F through J. It is seen that all systems meet the delivered air temperature requirement of $35^{o}C$ ($95^{o}F$), with the solar heat pump systems providing energy at higher temperatures a significant amount of the time. The series system in particular delivers high temperatures because the heat pump is oversized to prevent auxiliary energy usage unless the heat pump is starved.

Figure 6-39. Annual Frequency Histograms of Delivered Air Temperature During Heating Mode Operation (Washington, D.C., Next-Generation Technology)

Stand-Alone Heat Pump

TEMP INTERVAL (C)	HR	
22.00	0.	I
24.00	0.	I
26.00	0.	I
28.00	0.	I
30.00	0.	I
32.00	0.	I
34.00	0.	I
36.00	1617.25	I******************************
38.00	209.62	I****
40.00	10.25	I
42.00	0.	I
44.00	0.	I
46.00	0.	I
48.00	0.	I
50.00	0.	I
52.00	0.	I
54.00	0.	I
56.00	0.	I
58.00	0.	I
60.00	0.	I
62.00	0.	I
64.00	0.	I
66.00	0.	I
68.00	0.	I

Liquid Series

TEMP INTERVAL (C)	HR	
22,00	0,	I
24,00	0,	I
26,00	0,	I
28,00	0,	I
30,00	0,	I
32,00	0,	I
34,00	0,	I
36,00	20,50	I***
38,00	28,37	I****
40,00	112,00	I****************
42,00	186,50	I**************************
44,00	224,25	I*******************************
46,00	169,50	I************************
48,00	100,00	I**************
50,00	69,87	I**********
52,00	68,50	I**********
54,00	52,12	I********
56,00	34,12	I******
58,00	25,25	I***
60,00	5,75	I*
62,00	0,	I
64,00	0,	I
66,00	0,	I
68.00	0.	I

Air Parallel

TEMP INTERVAL (C)	HR	
22.00	0.	I
24.00	0.	I
26.00	0.	I
28.00	0.	I
30.00	1.50	I
32.00	4.00	I
34.00	8.37	I
36.00	1087.87	I*******************************
38.00	130.62	I****
40.00	73.87	I**
42.00	53.37	I*
44.00	35.25	I*
46.00	31.12	I*
48.00	22.62	I*
50.00	12.50	I
52.00	13.25	I
54.00	7.00	I
56.00	4.50	I
58.00	2.62	I
60.00	3.87	I
62.00	1.88	I
64.00	1.00	I
66.00	0.	I
68.00	0.	I

Liquid Parallel

TEMP INTERVAL (C)	HR	
22.00	0.	I
24.00	0.	I
26.00	0.	I
28.00	0.	I
30.00	0.	I
32.00	0.	I
34.00	0.	I
36.00	1026.12	I*******************************
38.00	162.75	I*****
40.00	63.87	I**
42.00	52.25	I**
44.00	40.25	I*
46.00	28.00	I*
48.00	24.87	I*
50.00	15.12	I
52.00	13.12	I
54.00	7.75	I
56.00	9.00	I
58.00	7.87	I
60.00	2.87	I
62.00	2.87	I
64.00	2.50	I
66.00	2.37	I
68.00	1.13	I

Figure 6-40 displays histograms of room temperature for the future-technology heat pump systems in Washington, D.C. Note that the parallel systems operate in second-stage heating much more often since no response is made to first stage if the solar system is depleted. Series systems only reach second stage if the heat pump is starved. To meet the thermal comfort criteria, the stand-alone heat pump responds to both first and second stage as if second stage were on. Consequently, the parallel systems do not control room temperature as tightly as the others. This is reflected in the fact that annual heating loads are slightly lower for the parallel systems (see Appendices F through J).

Two-dimensional histograms mapping the temperature/humidity history of the room are provided in Appendices F through I. If a comfort zone were chosen, the fraction of the time in the comfort zone could be calculated. For now, it is sufficient to note that all systems provided comparable comfort.

As explained in subsection 3.4.2, the stand-alone heat pump controls used in this study were nonconventional in the sense that the system was forced to deliver energy to the load at or above 35^{o}C. Figure 6-41 illustrates the differences which resulted in delivered air temperature and room temperature. Table 6-4 summarizes the performance differences between the two control strategies. Note that substantially more auxiliary energy is used when the comfort criteria are met. Under this circumstance, standard air-to-air heat pump sizing techniques may undersize equipment from an economic viewpoint.

6.3.3 Reliability

Several outputs of this analysis are useful for assessing the reliability of the various systems. The purpose of this subsection is to suggest which outputs are of interest and to indicate the appendices which display the information. A full investigation of system reliability is beyond the scope of this study, but the models developed here are ideal for providing input to such an investigation.

Figures 6-42 and 6-43 illustrate the distribution of heat pump compressor on-cycle lengths in the heating and cooling modes. Both total cycles and cycle length impact the compressor lifetime.

Figures 6-44 and 6-45 illustrate the distribution of source and sink temperatures during heat pump operation in the heating and cooling modes. In

Figure 6-40. Annual Frequency Histograms of Room Temperature (Washington, D.C., Next-Generation Technology)

Stand-Alone Heat Pump

TEMP INTERVAL (C)	HR	
16.00	0.	I
17.00	0.	I
18.00	0.	I
19.00	0.	I
20.00	3.87	I
21.00	2557.12	I***************************
22.00	2840.87	I******************************
23.00	505.87	I*****
24.00	521.75	I******
25.00	968.37	I**********
26.00	1020.25	I***********
27.00	340.87	I****
28.00	0.	I
29.00	0.	I
30.00	0.	I
31.00	0.	I
32.00	0.	I
33.00	0.	I
34.00	0.	I
35.00	0.	I
36.00	0.	I
37.00	0.	I
38.00	0.	I
39.00	0.	I

Liquid Series

TEMP INTERVAL (C)	HR	
16.00	0.	I
17.00	0.	I
18.00	0.	I
19.00	12.75	I
20.00	17.12	I
21.00	2615.75	I*****************************
22.00	2708.12	I******************************
23.00	553.12	I******
24.00	515.00	I******
25.00	928.75	I**********
26.00	1060.87	I************
27.00	347.50	I****
28.00	0.	I
29.00	0.	I
30.00	0.	I
31.00	0.	I
32.00	0.	I
33.00	0.	I
34.00	0.	I
35.00	0.	I
36.00	0.	I
37.00	0.	I
38.00	0.	I
39.00	0.	I

Air Parallel

TEMP INTERVAL (C)	HR	
16.00	0.	I
17.00	0.	I
18.00	.75	I
19.00	638.87	I*********
20.00	735.75	I***********
21.00	1938.25	I****************************
22.00	2041.62	I******************************
23.00	520.62	I********
24.00	478.00	I*******
25.00	977.12	I**************
26.00	1046.37	I***************
27.00	381.62	I******
28.00	0.	I
29.00	0.	I
30.00	0.	I
31.00	0.	I
32.00	0.	I
33.00	0.	I
34.00	0.	I
35.00	0.	I
36.00	0.	I
37.00	0.	I
38.00	0.	I
39.00	0.	I

Liquid Parallel

TEMP INTERVAL (C)	HR	
16.00	0.	I
17.00	0.	I
18.00	.63	I
19.00	602.37	I*********
20.00	630.75	I*********
21.00	1992.75	I*****************************
22.00	2092.75	I******************************
23.00	523.62	I********
24.00	498.25	I*******
25.00	978.12	I**************
26.00	1059.37	I***************
27.00	380.37	I*****
28.00	0.	I
29.00	0.	I
30.00	0.	I
31.00	0.	I
32.00	0.	I
33.00	0.	I
34.00	0.	I
35.00	0.	I
36.00	0.	I
37.00	0.	I
38.00	0.	I
39.00	0.	I

Figure 6-41. Annual Frequency Histograms Illustrating the Effect of Forcing the Stand-Alone Heat Pump to Meet the Delivered Air Temperature Requirement (Washington, D.C., Today's Technology Heat Pump)

Delivered Air Temperature

Conventional Controls

TEMP INTERVAL (C)	HR	
22.00	7.62	I
24.00	3.75	I
26.00	63.25	I**
28.00	174.62	I******
30.00	449.12	I***************
32.00	906.50	I******************************
34.00	434.75	I**************
36.00	304.00	I**********
38.00	90.75	I***
40.00	11.62	I
42.00	0.	I
44.00	0.	I
46.00	0.	I
48.00	0.	I
50.00	0.	I
52.00	0.	I
54.00	0.	I
56.00	0.	I
58.00	0.	I
60.00	0.	I
62.00	0.	I
64.00	0.	I
66.00	0.	I
68.00	0.	I

Controlled Delivered Air Temperature

TEMP INTERVAL (C)	HR	
22.00	0.	I
24.00	0.	I
26.00	0.	I
28.00	0.	I
30.00	0.	I
32.00	0.	I
34.00	0.	I
36.00	1756.75	I******************************
38.00	94.50	I**
40.00	8.87	I
42.00	0.	I
44.00	0.	I
46.00	0.	I
48.00	0.	I
50.00	0.	I
52.00	0.	I
54.00	0.	I
56.00	0.	I
58.00	0.	I
60.00	0.	I
62.00	0.	I
64.00	0.	I
66.00	0.	I
68.00	0.	I

Room Temperature

Conventional Controls

TEMP INTERVAL (C)	HR	
16.00	0.	I
17.00	0.	I
18.00	.25	I
19.00	114.37	I*
20.00	239.25	I***
21.00	2491.62	I*****************************
22.00	2564.00	I******************************
23.00	479.25	I******
24.00	532.37	I******
25.00	968.12	I***********
26.00	1022.62	I************
27.00	342.12	I****
28.00	0.	I
29.00	0.	I
30.00	0.	I
31.00	0.	I
32.00	0.	I
33.00	0.	I
34.00	0.	I
35.00	0.	I
36.00	0.	I
37.00	0.	I
38.00	0.	I
39.00	0.	I

Controlled Delivered Air Temperature

TEMP INTERVAL (C)	HR	
16.00	0.	I
17.00	0.	I
18.00	0.	I
19.00	0.	I
20.00	4.62	I
21.00	2558.75	I***************************
22.00	2833.00	I******************************
23.00	510.62	I*****
24.00	521.87	I******
25.00	969.00	I**********
26.00	1020.25	I***********
27.00	340.87	I****
28.00	0.	I
29.00	0.	I
30.00	0.	I
31.00	0.	I
32.00	0.	I
33.00	0.	I
34.00	0.	I
35.00	0.	I
36.00	0.	I
37.00	0.	I
38.00	0.	I
39.00	0.	I

Table 6-4

Performance Differences Between the Stand-Alone Heat Pump System with Conventional Controls and with Controlled Delivered Air Temperature

	QHP-L (GJ)	WHPH (GJ)	COPH	QA-L (GJ)	QLOAD (GJ)	FSSH
Conventional Controls	56.7	25.5	2.2	3.2	59.9	.52
Controlled Delivered Air Temperature	44.0	19.5	2.2	16.7	60.7	.40

Where

QHP-L - heating energy transferred to the load by the heat pump

WHPH - work input to the heat pump in the heating mode

COPH - heat pump COP in the heating mode

QA-L - auxiliary energy transferred to the space heating load

QLOAD - energy delivered to the load for space heating

FSSH - fraction of the space heating and cooling load met by non-purchased energy (parasitic energy usage is considered)

Figure 6-42. Annual Frequency Histograms of Heat Pump On-Cycle Length in the Heating Mode (Washington, D.C., Future Technology Heat Pumps)

Stand Alone Heat Pump

CYCLE LENGTH INTERVAL (HR)	CYCLES	
.63	0	I
1.13	3	I
1.63	107	I*************
2.13	231	I*****************************
2.63	238	I******************************
3.13	124	I****************
3.63	29	I****
4.13	26	I***
4.63	9	I*
5.13	3	I
5.63	3	I
6.13	0	I
6.63	1	I
7.13	0	I
7.63	1	I
8.12	1	I
6.62	0	I
9.12	2	I
9.62	0	I
10.13	1	I
10.63	0	I
11.13	0	I
11.63	1	I
12.13	0	I

Liquid Series

CYCLE LENGTH INTERVAL (HR)	CYCLES	
.60	28	I**
1.10	494	I******************************
1.60	340	I*********************
2.10	61	I****
2.60	14	I*
3.10	7	I
3.60	1	I
4.10	0	I
4.60	0	I
5.10	0	I
5.60	0	I
6.10	0	I
6.60	0	I
7.10	0	I
7.60	0	I
8.10	0	I
8.60	0	I
9.10	0	I
9.60	0	I
10.10	0	I
10.60	0	I
11.10	0	I
11.60	0	I
12.10	0	I

Air Parallel

.60	0	I
1.10	0	I
1.60	0	I
2.10	0	I
2.60	5	I***
3.10	7	I****
3.60	38	I**********************
4.10	47	I***************************
4.60	55	I*******************************
5.10	36	I********************
5.60	18	I**********
6.10	9	I*****
6.60	10	I*****
7.10	2	I*
7.60	2	I*
8.10	1	I*
8.60	2	I*
9.10	1	I*
9.60	2	I*
10.10	1	I*
10.60	1	I*
11.10	0	I
11.60	0	I
12.10	1	I*

Liquid Parallel

.60	0	I
1.10	0	I
1.60	0	I
2.10	0	I
2.60	4	I**
3.10	5	I***
3.60	40	I***********************
4.10	41	I***********************
4.60	51	I*****************************
5.10	40	I***********************
5.60	11	I******
6.10	10	I******
6.60	9	I******
7.10	2	I*
7.60	4	I***
8.10	1	I*
8.60	3	I**
9.10	0	I
9.60	3	I*
10.10	0	I
10.60	0	I
11.10	1	I*
11.60	1	I*
12.10	0	I

Figure 6-43. Annual Frequency Histograms of Heat Pump On-Cycle Length in the Cooling Mode (Washington, D.C., Future-Technology Heat Pumps)

Stand-Alone Heat Pump

CYCLE LENGTH INTERVAL (HR)	CYCLES
.63	0
1.13	0
1.63	1
2.13	33
2.63	37
3.13	30
3.63	11
4.13	1
4.63	0
5.13	0
5.63	0
6.13	0
6.63	0
7.13	0
7.63	0
8.12	0
6.62	0
9.12	0
9.62	0
10.13	0
10.63	0
11.13	0
11.63	0
12.13	0

Liquid Series

CYCLE LENGTH INTERVAL (HR)	CYCLES
.60	0
1.10	0
1.60	27
2.10	86
2.60	13
3.10	0
3.60	0
4.10	0
4.60	0
5.10	0
5.60	0
6.10	0
6.60	0
7.10	0
7.60	0
8.10	0
8.60	0
9.10	0
9.60	0
10.10	0
10.60	0
11.10	0
11.60	0
12.10	0

Air Parallel

CYCLE LENGTH INTERVAL (HR)	CYCLES
.60	0
1.10	0
1.60	0
2.10	19
2.60	44
3.10	31
3.60	19
4.10	3
4.60	0
5.10	0
5.60	0
6.10	0
6.60	0
7.10	0
7.60	0
8.10	0
8.60	0
9.10	0
9.60	0
10.10	0
10.60	0
11.10	0
11.60	0
12.10	1

Liquid Parallel

CYCLE LENGTH INTERVAL (HR)	CYCLES
.60	0
1.10	0
1.60	0
2.10	20
2.60	45
3.10	29
3.60	20
4.10	3
4.60	0
5.10	0
5.60	0
6.10	0
6.60	0
7.10	0
7.60	0
8.10	0
8.60	0
9.10	0
9.60	0
10.10	0
10.60	0
11.10	0
11.60	0
12.10	0

Figure 6-44. Annual Frequency Histograms of Source Temperature During Heat Pump Operation in the Heating Mode (Washington, D.C., Future-Technology Heat Pump)

Stand-Alone Heat Pump

TEMP INTERVAL (C)	HR	
-18.00	4.25	I
-16.00	4.75	I
-14.00	8.50	I*
-12.00	30.25	I***
-10.00	25.50	I***
-8.00	56.62	I******
-6.00	99.75	I***********
-4.00	125.87	I**************
-2.00	127.12	I**************
0.	192.12	I********************
2.00	299.75	I********************************
4.00	233.87	I*************************
6.00	189.62	I********************
8.00	181.25	I*******************
10.00	116.50	I*************
12.00	76.37	I*********
14.00	48.00	I*****
16.00	26.75	I***
18.00	5.62	I*
20.00	3.[illegible]	I
22.00	0.	I
24.00	0.	I

Liquid Series

TEMP INTERVAL (C)	HR	
2.00	27.75	I******
4.00	72.50	I***************
6.00	126.12	I*************************
8.00	150.87	I*******************************
10.00	98.37	I********************
12.00	63.25	I**************
14.00	13.12	I***
16.00	32.25	I******
18.00	30.00	I******
20.00	22.62	I****
22.00	12.37	I**
24.00	13.00	I***
26.00	42.00	I********
28.00	54.87	I***********
30.00	41.75	I********
32.00	31.62	I******
34.00	18.62	I****
36.00	32.12	I******
38.00	17.25	I***
40.00	3.37	I*
42.00	.38	I
44.00	0.	I
46.00	0.	I
48.00	0.	I

Air Parallel

TEMP INTERVAL (C)	HR	
-18.00	3.87	I*
-16.00	1.88	I
-14.00	5.37	I*
-12.00	25.87	I****
-10.00	24.12	I****
-8.00	39.00	I******
-6.00	76.00	I*************
-4.00	116.50	I*******************
-2.00	95.25	I****************
0.	136.37	I***********************
2.00	183.00	I********************************
4.00	122.37	I********************
6.00	102.00	I*****************
8.00	77.75	I**************
10.00	37.62	I******
12.00	11.87	I**
14.00	6.75	I*
16.00	7.87	I*
18.00	1.38	I
20.00	0.	I

Liquid Parallel

TEMP INTERVAL (C)	HR	
-18.00	4.00	I*
-16.00	4.37	I*
-14.00	8.00	I*
-12.00	24.12	I****
-10.00	19.87	I***
-8.00	44.25	I*******
-6.00	80.75	I**************
-4.00	104.00	I*****************
-2.00	65.00	I***********
0.	125.62	I*********************
2.00	189.62	I*********************************
4.00	131.50	I***********************
6.00	100.62	I*****************
8.00	65.25	I***********
10.00	33.87	I*****
12.00	12.62	I**
14.00	9.75	I**
16.00	3.62	I*
18.00	0.	I
20.00	0.	I

Figure 6-45. Annual Frequency Histograms of Sink Temperature During Heat Pump Operation in the Cooling Mode (Washington, D.C., Future-Technology Heat Pump)

Stand-Alone Heat Pump

TEMP INTERVAL (C)	HR	
19.00	0.	I
20.00	0.	I
21.00	1.75	I*
22.00	6.12	I*****
23.00	8.62	I*******
24.00	11.00	I*********
25.00	13.12	I***********
26.00	18.25	I***************
27.00	19.75	I****************
28.00	22.37	I*******************
29.00	28.87	I************************
30.00	26.25	I**********************
31.00	34.37	I*****************************
32.00	37.37	I*******************************
33.00	24.75	I*********************
34.00	12.12	I**********
35.00	4.50	I****
36.00	1.88	I**
37.00	.63	I*
38.00	0.	I
39.00	0.	I
40.00	0.	I
41.00	0.	I
42.00	0.	I

Liquid Series

TEMP INTERVAL (C)	HR	
9,00	0,	I
11,00	0,	I
13,00	0,	I
15,00	0,	I
17,00	4,50	I**
19,00	7,37	I****
21,00	7,75	I****
23,00	10,25	I*****
25,00	26,25	I**************
27,00	53.12	I****************************
29,00	56,37	I******************************
31,00	24,12	I*************
33,00	2,62	I*
35,00	0,	I
37,00	0,	I
39,00	0,	I
41,00	0,	I
43,00	0,	I
45,00	0,	I
47,00	0,	I
49,00	1,63	I*
51,00	0,	I
53,00	0,	I
55.00	.87	I

Air Parallel

TEMP INTERVAL (C)	HR	
6.00	0.	I
10.00	0.	I
14.00	0.	I
18.00	0.	I
22.00	2.75	I*
26.00	49.25	I*************
30.00	105.87	I***************************
34,00	118.37	I******************************
38.00	7.50	I**
42.00	0.	I
46.00	0.	I
50.00	0.	I
54.00	0.	I
58.00	0.	I
62,00	0.	I
66,00	0.	I
~~70.00~~	0.	I
74.00	0.	I
78.00	0.	I
82.00	0.	I
86.00	0.	I
90.00	0.	I
94,00	0.	I
98.00	0.	I

Liquid Parallel

TEMP INTERVAL (C)	HR	
16.42	0.	I
17.83	0.	I
19.25	0.	I
20.67	.25	I
22.08	3.87	I**
23.50	9.87	I*****
24.92	19.00	I*********
26.33	28.00	I**************
27.75	34.00	I*****************
29.17	42.50	I*********************
30.58	36.37	I******************
32.00	60.62	I******************************
33.42	37.25	I******************
34.83	11.25	I******
36.25	1.63	I*
37.67	.50	I
39.08	0.	I
40.50	0.	I
41.92	0.	I
43.33	0.	I
44.75	0.	I
46.17	0.	I
47.58	0.	I
49.00	0.	I

systems with air-to-air heat pumps, the number of defrost cycles can be estimated by dividing the operating hours spent with the source temperature in the defrost region (-8 to 6C) by 1.5 (assumes a 90-minute time/temperature defrost). If there are source temperature regions where reliability problems are acute (e.g., high source temperatures which cause pressure problems), these histograms identify the systems which aggravate the problem most.

In Table 6-5, estimates of the number of heat pump on/off cycles, defrost cycles and average on-cycle length are given to illustrate the type of information available.

Figures 6-42 through 6-45 and Table 6-5 are for Washington, D.C., with the future-technology heat pumps. For other location/heat pump combinations, see Appendices F through J.

In addition to the heat pump reliability, the same models could be used to track variables which impact durability of solar system components. Temperatures, pump and blower cycles and cycle lengths, and other variables may be of interest.

6.3.4 Utility Impact

As with reliability analysis, utility impact analysis is beyond the scope of this study. However, the models assembled for this work are ideal for providing input to such studies. The purpose of this subsection is to display the type of information which can be generated.

Figure 6-46 illustrates the frequency of total electrical demand incurred by the systems for heating and cooling. Although this figure contains annual histograms, monthly or other time intervals could be specified. This information allows investigation of the effect of demand electrical rates on the cost competitiveness of alternative heating and cooling technologies.

Figure 6-47 illustrates the annual time-of-day electrical consumption required by the systems for heating and cooling. The same information could be generated monthly or for the utility's peak load day. The information is useful for the study of time-of-day rate structures and to assess the impact of various heating and cooling technologies on utility load management.

Figures 6-46 and 6-47 are for systems in Washington, D.C., with future technology heat pumps. See Appendices F through J for the same information for

Table 6-5

Summary of Heat Pump Reliability Information Available from the Current Study (Washington D.C., Future Technology Heat Pump)

	Heating Mode			Cooling Mode	
	on/off cycles	defrost cycles	average on cycle length (hr)	on/off	average on cycle length (hr)
Stand Alone Heat Pump	783	845	2.3	113	2.4
Liquid Series	968	-	0.9	126	1.6
Air Parallel	239	554	4.5	116	2.4
Liquid Parallel	228	531	4.5	117	2.4

Figure 6-46. Annual Frequency Histograms of Total Electrical Demand (Compressor + Auxiliary + Parasitics) for Space Heating and Cooling (Washington, D.C., Future-Technology Heat Pump)

Stand-Alone Heat Pump

ENERGY DEMAND INTERVAL (KJ/hr)	HR	
3000.00	0.	I
5500.00	0.	I
8000.00	0.	I
10500.00	470.37	I******************************
13000.00	174.12	I***********
15500.00	421.62	I***************************
18000.00	458.37	I*****************************
20500.00	454.87	I*****************************
23000.00	100.75	I******
25500.00	22.12	I*
28000.00	6.62	I
30500.00	0.	I
33000.00	0.	I
35500.00	0.	I
38000.00	0.	I
40500.00	0.	I
43000.00	0.	I
45500.00	0.	I
48000.00	0.	I
50500.00	0.	I
53000.00	0.	I
55500.00	0.	I
58000.00	0.	I
60500.00	0.	I

Liquid Series

ENERGY DEMAND INTERVAL (KJ/hr)	HR	
3000,00	1913.87	I******************************
5500,00	0.	I
8000,00	0.	I
10500,00	0.	I
13000,00	351,75	I******
15500,00	470.37	I*******
18000.00	275,75	I****
20500,00	10.00	I
23000,00	0.	I
25500,00	0.	I
28000,00	0.	I
30500.00	0.	I
33000,00	4,12	I
35500,00	4,25	I
38000,00	5.12	I
40500,00	5,12	I
43800.00	.13	I
45500,00	0.	I
48000,00	0.	I
50500,00	0.	I
53000,00	0,	I
55500,00	0,	I
58000,00	0.	I
60500.00	0.	I

Air Parallel

ENERGY DEMAND INTERVAL (KJ/hr)	HR	
4500.00	1053.62	I******************************
8500.00	3.62	I
12500.00	343.50	I**********
16500.00	226.62	I******
20500.00	432.50	I************
24500.00	293.50	I********
28500.00	51.12	I*
32500.00	5.25	I
36500.00	1.88	I
40500,00	0.	I
44500.00	0.	I
48500.00	0.	I
52500.00	0,	I
56500,00	0.	I
60500.00	0.	I
64500,00	0.	I
68500.00	0.	I
72500,00	0.	I
76500.00	0.	I
80500,00	0.	I
84500.00	0.	I
88500.00	0.	I
92500.00	0.	I
96500.00	0.	I

Liquid Parallel

ENERGY DEMAND INTERVAL (KJ/hr)	HR	
4500.00	1194.87	I******************************
8500.00	131.62	I***
12500.00	328.12	I********
16500.00	197.62	I*****
20500.00	415.00	I**********
24500.00	291.75	I*******
28500.00	62.75	I**
32500.00	12.00	I
36500.00	0.	I
40500.00	0.	I
44500.00	0.	I
48500.00	0.	I
52500.00	0.	I
56500.00	0.	I
60500.00	0.	I
64500.00	0.	I
68500.00	0.	I
72500.00	0.	I
76500.00	0.	I
80500.00	0.	I
84500.00	0.	I
88500.00	0.	I
92500.00	0.	I
96500.00	0.	I

Figure 6-47. Annual Time of Day Histograms of Total Electrical Demand (Compressor + Auxiliary + Parasitics) for Space Heating and Cooling (Washington, D.C.,

Stand-Alone Heat Pump

TIME OF DAY (HR)	ENERGY USAGE (KJ)
1.00	1118385.27
2.00	1408583.28
3.00	1666827.76
4.00	1664860.63
5.00	1707028.55
6.00	1623631.03
7.00	1636465.23
8.00	1674031.05
9.00	1396543.29
10.00	1028968.59
11.00	895399.33
12.00	1084379.81
13.00	1278690.11
14.00	1282167.54
15.00	1224265.82
16.00	1056699.50
17.00	1012652.22
18.00	1179971.85
19.00	1209765.39
20.00	1313335.63
21.00	1344081.74
22.00	1367644.67
23.00	1323244.74
24.00	1274764.30

Liquid Series

TIME OF DAY (HR)	ENERGY USAGE (KJ)
1.00	626372.72
2.00	797113.28
3.00	836505.62
4.00	788723.55
5.00	865413.47
6.00	747759.81
7.00	903715.81
8.00	847057.24
9.00	798014.20
10.00	848802.99
11.00	900696.84
12.00	908620.11
13.00	976840.38
14.00	982122.01
15.00	898534.08
16.00	812748.13
17.00	752679.66
18.00	748199.17
19.00	641870.53
20.00	774357.42
21.00	692136.03
22.00	602084.63
23.00	726024.56
24.00	659326.86

Air Parallel

TIME OF DAY (HR)	HR
1.00	800324.78
2.00	945142.92
3.00	1166911.92
4.00	1263820.32
5.00	1416961.01
6.00	1487723.16
7.00	1420612.14
8.00	1295872.92
9.00	1247846.54
10.00	1165795.14
11.00	981725.27
12.00	1054233.46
13.00	1121810.45
14.00	994494.31
15.00	881620.57
16.00	753733.12
17.00	753147.72
18.00	816257.21
19.00	964924.83
20.00	1006875.93
21.00	1013772.76
22.00	944198.46
23.00	876205.94
24.00	839162.22

Liquid Parallel

TEMP INTERVAL (C)	ENERGY USAGE (KJ)
1.00	843183.68
2.00	1064737.90
3.00	1212687.46
4.00	1408507.89
5.00	1429157.47
6.00	1341085.56
7.00	1325038.97
8.00	1361326.89
9.00	1400885.70
10.00	1396654.85
11.00	1396304.87
12.00	1409251.37
13.00	1373927.61
14.00	1250706.63
15.00	1095282.65
16.00	883582.04
17.00	733973.54
18.00	726887.15
19.00	852791.27
20.00	838116.38
21.00	761382.08
22.00	746731.09
23.00	768423.62
24.00	817730.57

other location/heat pump combinations. These appendices also contain time-of-day histograms of individual components of the electrical load so that the subsystems with the most potential for on-site load management are identified.

It should be noted that the models developed for this study are capable of generating a file of electrical demands for each time-step throughout the year. This load file can then be manipulated to obtain any desired statistic or histogram for use in a utility impact study.

7.0 SYSTEM SIZING

The conventional furnaces and stand-alone heat pump systems were sized to match the design loads estimated in the conventional manner from the building characteristics. Collector areas in both the parallel and series system configurations were sized by minimizing the initial system cost per unit of purchased energy displaced. Solar system storage size was referenced to collector area using the accepted rule of thumb in the parallel system while storage size was viewed as an independent variable to be optimized economically in the series system.

7.1 CONVENTIONAL SYSTEMS

The conventional furnace and central air conditioning systems were sized according to standard practices utilizing design-day loads (see subsection 2.2). The conventional domestic hot water system was chosen for a family of four.

The stand-alone heat pump system was sized by referring to the design-day cooling load, including appropriate corrections for geographic location. The heating mode capacity was thus fixed by the cooling capacity, which is the current sizing practice.

7.2 SOLAR STORAGE AND HEAT PUMP SIZING

Summaries of storage and heat pump sizing are presented in the following subsections for parallel and series systems.

7.2.1 Parallel Systems

The parallel system heat pump is sized in the conventional manner as described above. Fortunately, the solar system configurations used in parallel with the heat pump are well understood and rules of thumb exist for relating storage size to collector area. For the air parallel system, pebble bed storage sizes of from .15 to .25m^3 per m^2 of collector are generally recommended (46, 47, 48). In this study, .25m^3 per m^2 of collector has been used. For liquid parallel systems, the generally accepted rule of thumb of 75 kg of water per m^2

of collector has been used (48, 49). The problem of sizing has then been simplified to choosing a collector area based on economic criteria.

7.2.2 Series Systems

The series system heat pumps were sized so that auxiliary energy was never required unless the heat pump was starved (storage tank temperature below the minimum). Thus, the heat pump was sized to supply the design load when the storage tank was just above its minimum temperature. Additionally, the heat pump was required to meet the delivered air minimum temperature under the above conditions.

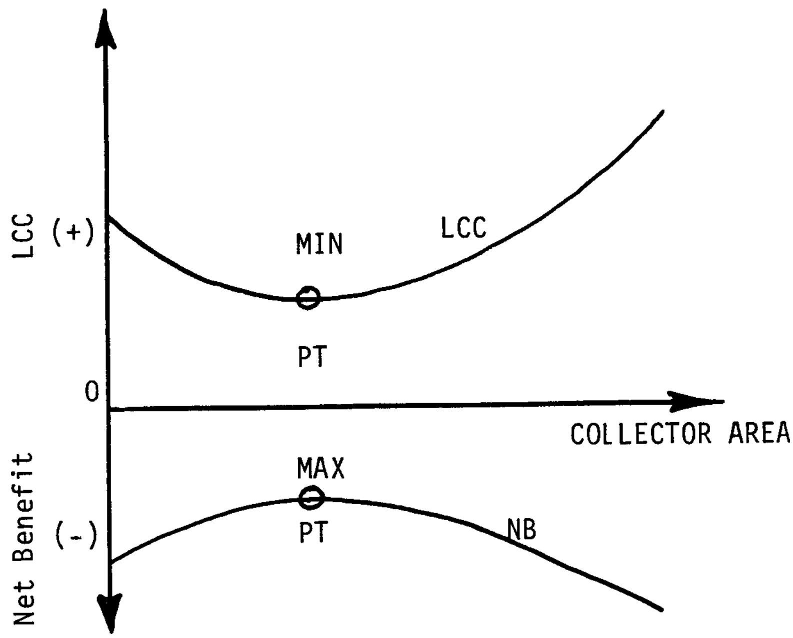
The series storage sizing was performed in conjunction with the collector area sizing since no thumb rules exist which would allow a storage/collector relationship to be determined a priori. It was found that storage size had little influence on system first cost and thermal performance since larger storage allowed smaller collector areas. As seen in Section 6, the series storage sizes investigated ranged from .2 to .6m^3 per m^2 (5 to 15 gallons per ft^2) of collector area.

7.3 COLLECTOR SIZING TECHNIQUES

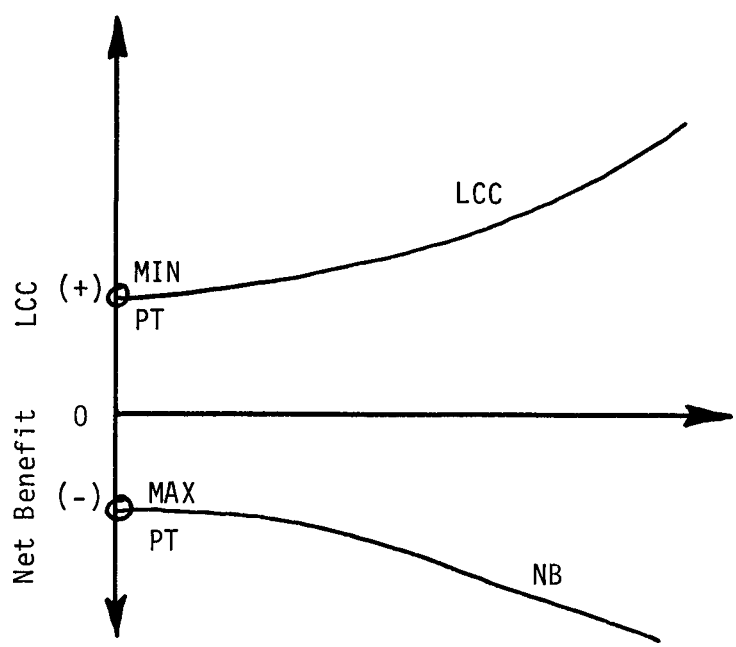
For solar system sizing, an accepted collector sizing technique is to minimize the solar system life-cycle costs (LCC) or to maximize the system's life-cycle net benefits against the collector area used in the system. Figure 7-1 illustrates the two possible situations in which LCC and net benefits analyses are employed. Figure 7-1(a) represents the case when the LCCs reach a minimum and the net benefits reach a maximum (even if the net benefit is negative, as shown).

The second case, Figure 7-1(b), arises when no maximum or minimum is present at finite collector area. The LCC continually increases with more collector area and the net benefit continually decreases. Zero collector area (no solar system) is the optimum economic size.

From the systems' cost data (Appendix E) and the systems' thermal performance data (Section 6), the LCCs and net benefits for the series and parallel systems were calculated for various collector areas. With the realistic costs used, Figure 7-2 shows that the LCC/net benefits approach cannot determine an "optimum" collector size for either system in any of the three locations. This is not an unusual circumstance for new technology.

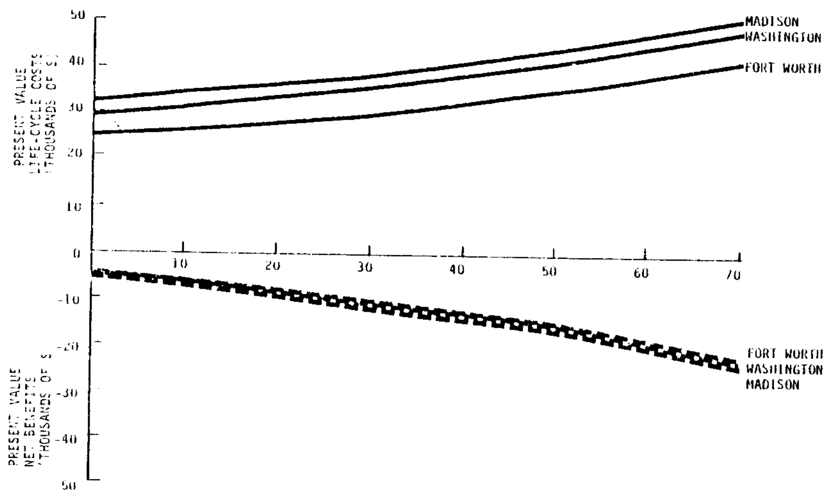


(a) LCC and NB analyses have finite collector area minimum/maximum points.

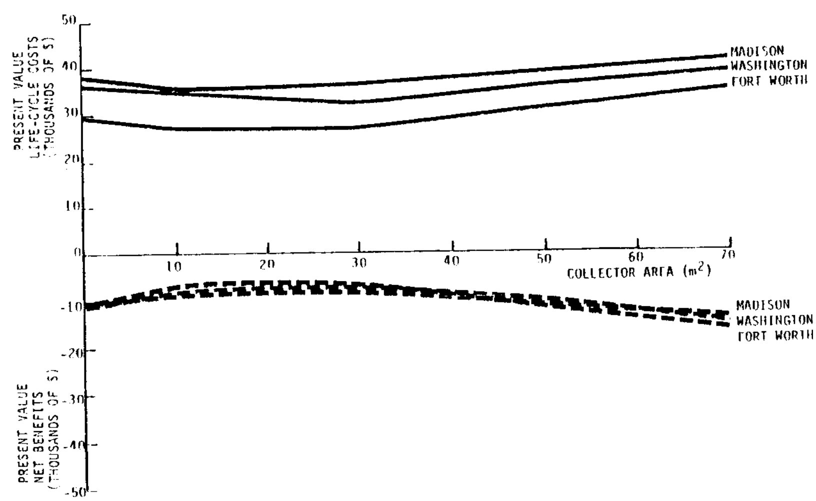


(b) LCC and NB analyses have <u>no</u> finite collector area minimum/maximum analysis.

Figure 7-1. Two Cases for Sizing Analysis By LCC or Net Benefits



a) Liquid Series CSHP Systems



b) Liquid Parallel CSHP Systems

Figure 7-2. Life Cycle Costs and Net Benefit Versus Collector Area: Series and Parallel CSHP Systems (Today)

As a practical matter, if these systems were to be installed today, it is reasonable to expect that collector area would be sized large enough to meet a substantial portion of the load. The economic sizing technique adopted in this study identifies system sizes which are reasonable for the solar industry to use as thermal design points. Other constraints, such as maximum available roof area for collectors and air flow balance between collector and load loops (air parallel system) have also been considered. By comparing system costs at these sizes to the costs of conventional alternatives, an indication of how much price reduction is required before these systems are competitive is obtained.

7.3.1 TCE Method

The collector sizing method adopted here is to minimize the installed system cost per unit of purchased energy displaced (TCE). At zero collector area, no purchased energy is displaced by the solar system, yet sunk costs are nonzero; therefore, TCE is infinite. At infinite collector area, installed cost is infinite, but purchased energy displaced cannot exceed the load; thus, TCE is again infinite. Between these extremes lies an optimum collector area for minimizing the TCE.

The TCE is a "$/Btu" form of economic analysis. As a reference to other "$/Btu" analyses, Figure 7-3 has been constructed to show the "optimum" collector areas chosen with: the TCE method; the initial investment per life-cycle solar energy delivered method; and the present value LCC per life-cycle solar energy delivered method. The TCE method sizes collector area between the other two methods.

The TCE relationships are developed in subsection 7.3.2 below and have been used to optimize collector area for the liquid and air parallel and series systems in Washington, D.C.; Fort Worth, TX; and Madison, WI. The costs itemized in subsection 4.4.4 have been used to determine the fixed and variable costs for each system. The current Federal tax credit has not been included because it is a temporary market incentive and it was not felt that thermal design point size should be chosen on that basis.

7.3.2 Derivation of TCE Relations

The collector sizing criteria are to minimize total installed system cost per unit of purchased energy displaced as defined in the following equation:

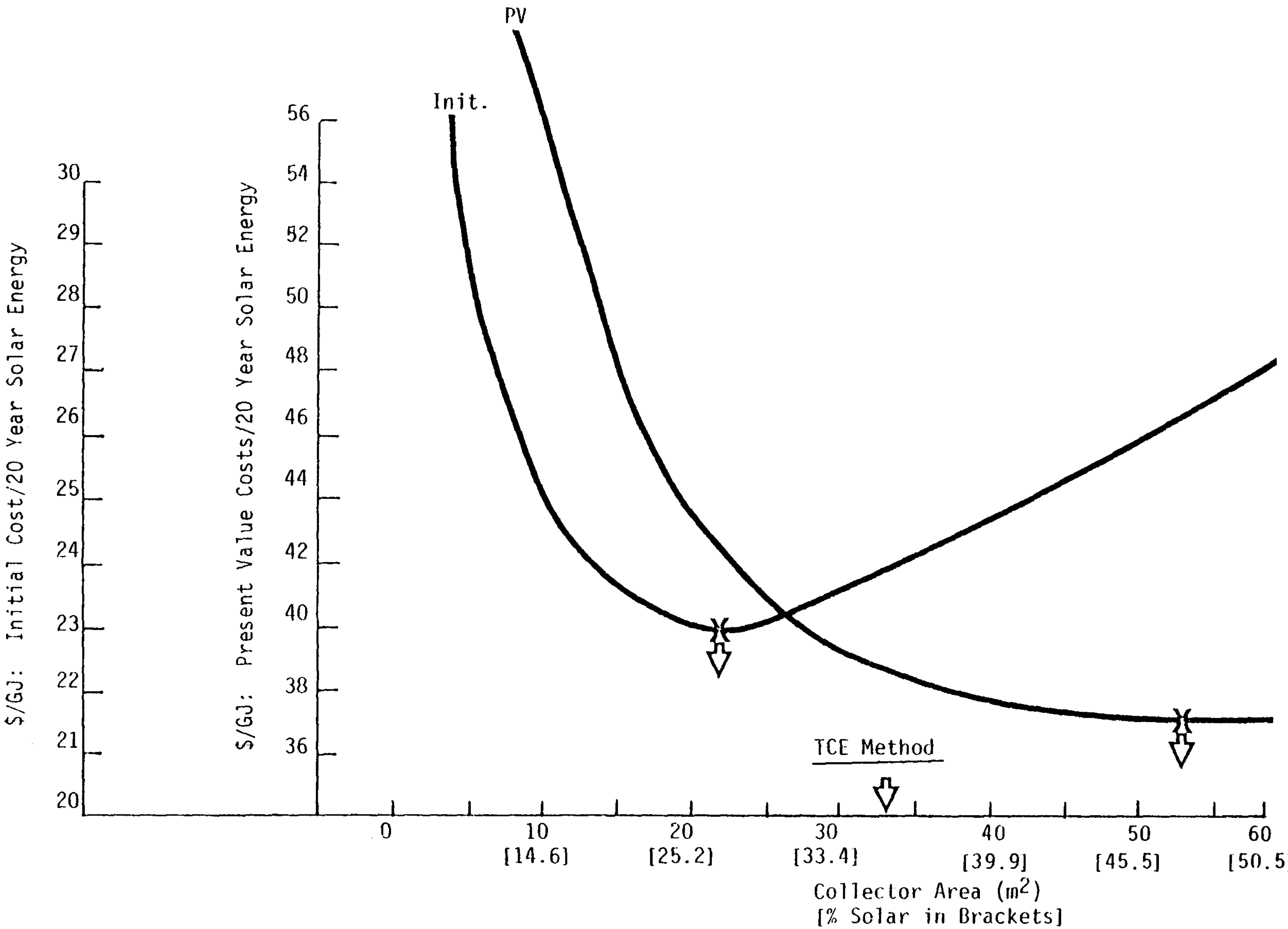


Figure 7-3. "$/BTU" Sizing Methods: An Example of Air Parallel System Sizing
(FC = $3857, VC = $297/m^2)

$$TCE = [FC + A(VC)] \quad /CE, \qquad [1]$$

where,

> TCE = total installed system cost per unit of purchased energy displaced (\$/GJ)
>
> FC = fixed (sunk) cost of system (\$)
>
> A = collector area (m^2)
>
> VC = collector area dependent system cost (\$/m^2)
>
> CE = purchased energy displaced by the solar system (GJ).

Finding the collector area for which TCE is minimum:

$$\frac{d(TCE)}{dA} = 0 = \frac{-FC}{CE^2}\frac{\partial(CE)}{\partial A} + \frac{VC}{CE} \quad \frac{A(VC)}{CE^2}\frac{\partial(CE)}{\partial A}$$

$$A_{opt} = CE_{opt} \left[\frac{\partial(CE)}{\partial A}\right]_{opt}^{-1} - \frac{FC}{VC} \qquad [2]$$

The problem now is to determine the functional relationship between the purchased energy displaced by solar energy (CE) and the collector area (A). This relationship is commonly expressed in terms of the solar fraction (f) and is exponential in nature. Thus, the assumed form is:

$$f = \frac{CE}{TL} = f_{max}\ (1 - e^{-bA}) , \qquad [3]$$

where,

> b is an arbitrary coefficient (which can be calculated after one simulation provides f and A), and f_{max} represents the maximum fraction of the load which is available for displacement (discussed below).

The optimum collector area is then given by the following expression,

$$A_{opt} = \frac{1}{b}\,(e^{bA_{opt}} - 1) - \frac{FC}{VC} \qquad [4]$$

Determining f_{max} for a given system requires an intuitive understanding of the system's thermal performance. The f_{max} will be different for differing system configurations (series, parallel, direct, etc.), for differing system control strategies (with or without direct heating mode), and for differing component efficiencies and operating energy requirements (heat pump COP and system parasitics).

Direct Heating/Hot Water System's f_{max}

For the standard direct heating and hot water solar system having a conventional energy furnace auxiliary supply, the theoretical f_{max} would be unity ($f_{max} = 1$). All the heating and hot water load is theoretically available to be supplied by solar energy. In reality, the f_{max} will be reduced from unity by the parasitic energy fraction required to operate the solar system.

Series Combined Solar Heat Pump System's f_{max}

The series CSHP systems used in this study have the hot water load being supplied by a separate solar system. Thus, the energy available for displacement by the series system is referenced only to the space heating load. As is seen in Figure 7-4, the series system utilizes solar energy in either of two ways: as source energy for the heat pump or as direct heating energy to the load.

In this study, "Today's Technology" liquid source heat pumps preclude the use of the direct heating option because of the heat pump's source temperature limitations. Thus, the solar energy can only be the source energy to the heat pump, and f_{max} is limited by the heat pump COP (SPF). Energy must always be purchased to operate the heat pump, so f_{max} is given by (minus the parasitics),

$$f_{max} = - \frac{1}{COP_{max}} \qquad [5]$$

for the heat pump in this study f_{max} (today, series) = 0.627 (see Figure 7-4).

With "Next-Generation" heat pump technology, the source temperature limitation will be raised so that direct solar heating will be possible. In this case, the series CSHP system would theoretically be able to displace the entire heating load by direct heating and ($f_{max} = 1$) minus the parasitic energy. Practically, as shown in Figure 7-4, the solar energy utilization is "split" between direct use and heat pump source use.

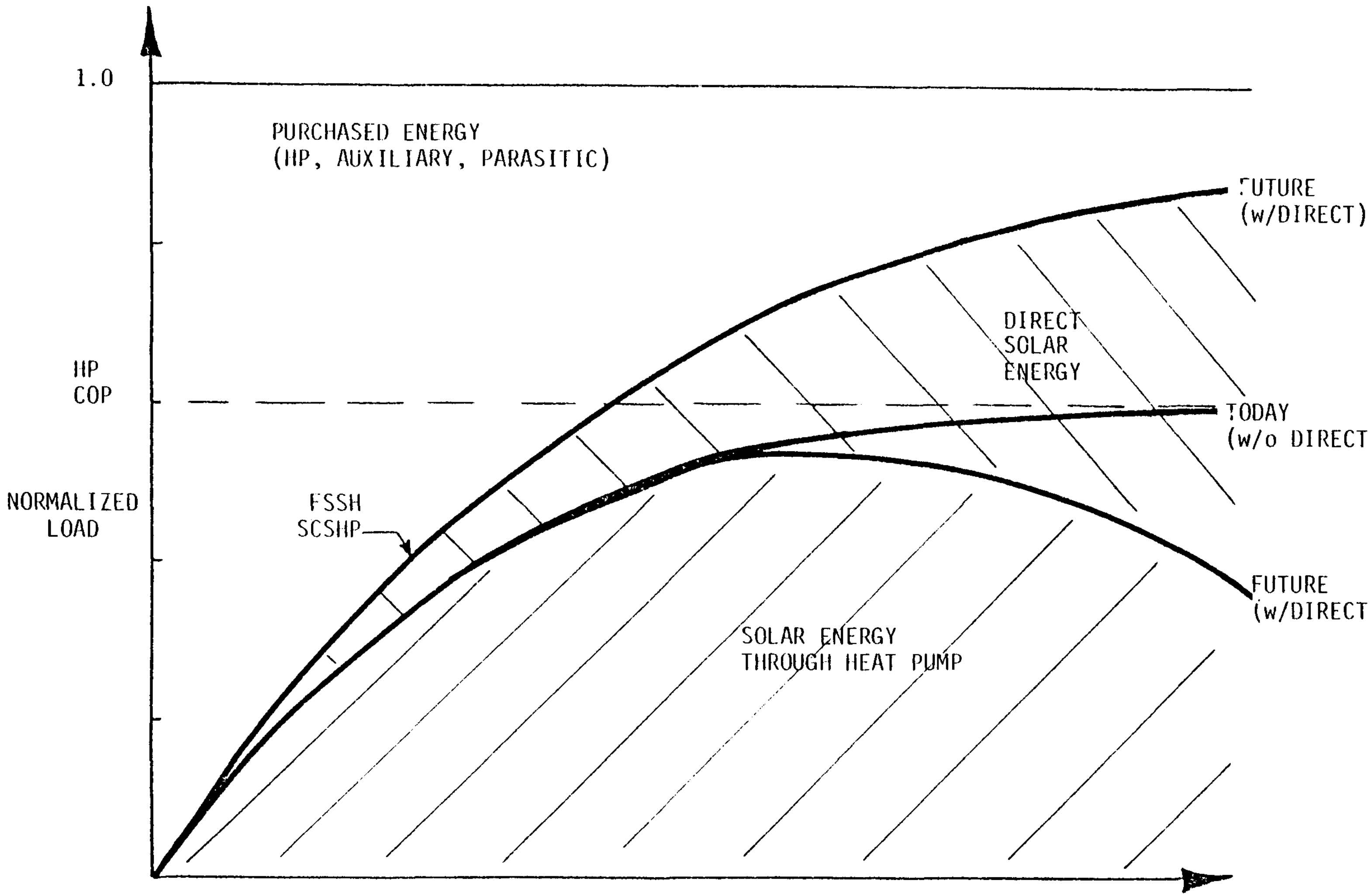


Figure 7-4. Series Combined Solar Heat Pump System Energy Sources

Parallel Combined Solar Heat Pump System's f_{max}

In the parallel configuration, the solar energy could theoretically displace the entire heating and hot water load through direct heating. Actually, the maximum displacement of conventional energy will be less than unity due to parasitic energy use. As seen in Figure 7-5, the purchased energy (mainly HP compressor work and auxiliary energy) decreases with increasing collector area. However, the ambient energy (from HP) is replaced by solar energy, and, since both of these are "free" energies, no real load displacement is being handled. Thus, the parallel systems actually use solar energy to displace ambient free energy.

7.4 TCE PROCEDURE AND SYSTEMS SUMMARY

The procedure for applying the TCE method is presented and the resulting collector area sizes for the various CSHP systems are summarized below.

7.4.1 TCE Application Procedure

In order to size the collector area with the TCE method, the thermal performance curve for the system is needed (see Section 6). Additionally, the fixed and variable costs are needed, along with f_{max} for the particular system.

The sizing procedure begins with the picking of an arbitrary f/A-pair from the thermal performance curve. When the system f_{max} and the f/A-pair are substituted into equation [3], b is solved for as shown below:

$$b = -\frac{1}{A}\left[\mathrm{LN}\ (1 - \frac{f}{f_{max}})\right] . \qquad [6]$$

With b, FC, and VC now known, A_{opt} is solved for by trial-and-error using equation [4]. If A_{opt} is different from the A originally chosen from the thermal performance curve, the procedure is iterated with a new A.

7.4.2 Systems Sizing Summary

The systems were sized using the TCE method described above. Table 7-1(a) and (b) presents a summary of the series systems' thermal and cost data used to produce the "optimum" collector areas and energy fractions. Note that various storage volumes were used (with varying VC charges) in the sizing

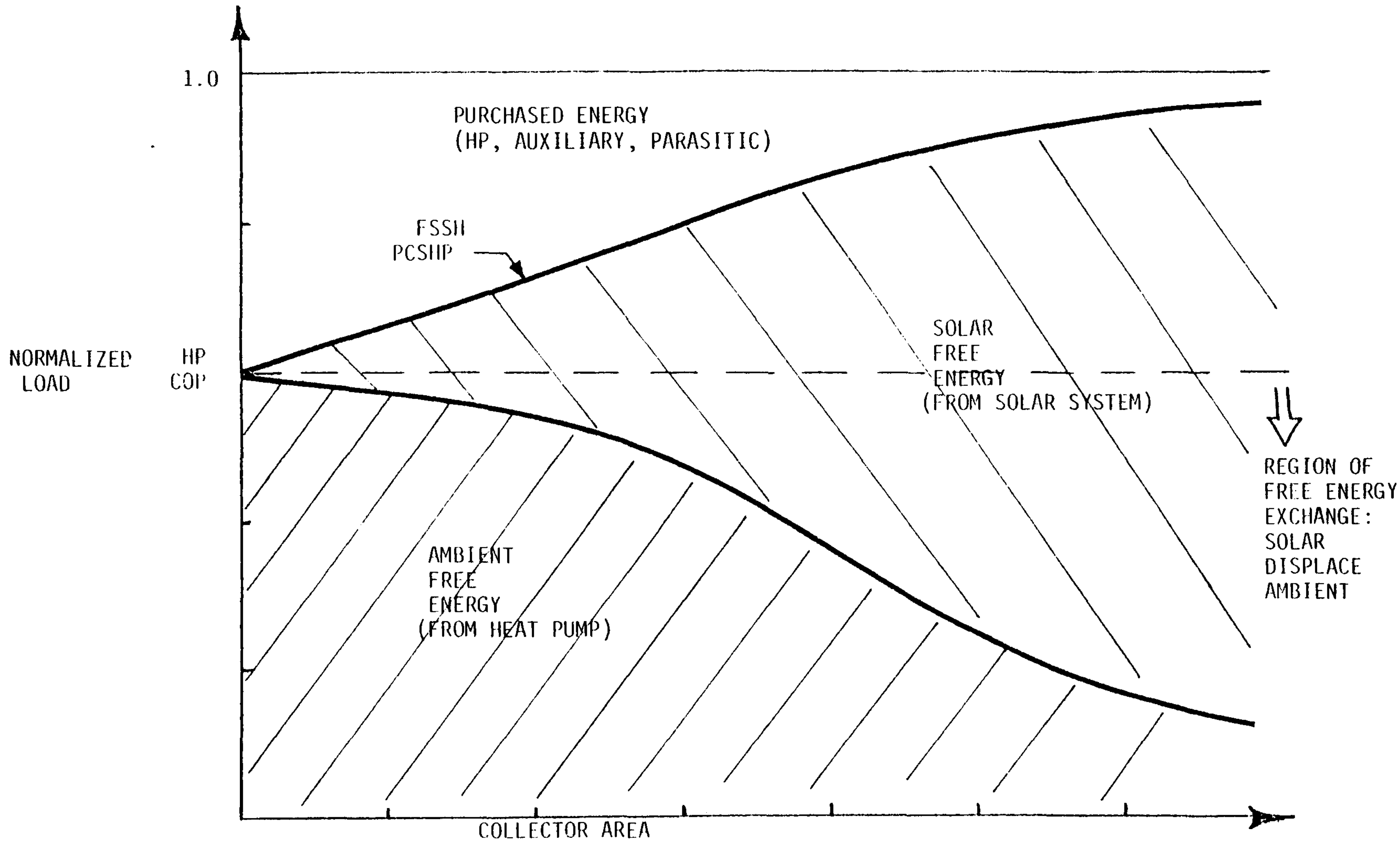


Figure 7-5. Parallel Combined Solar Heat Pump System Energy Sources

Table 7-I(a)

Summary of Liquid Series System Sizing Calculations (Today's Technology)

Thermal Data	Washington D. C.			Fort Worth Texas			Madison Wisconsin		
Storage Size (m^3/m^2)	.6	.4	.2	.6	.4	.2	.6	.4	.2
Space Heating Load (TL) (GJ)	59.6	59.6	59.6	38.4	38.4	38.4	78.3	78.3	78.3
F_{max}	.627	.627	.627	.627	.627	.627	.627	.627	.627
Cost Data									
FC ($)	7170	7170	7170	7170	7170	7170	7170	7170	7170
VC ($/m^2)	158	142	126	158	142	126	158	142	126
Optimum									
Collector Area (m^2)	33	38	43	25	26	35	43	49	56
F	.48	.49	.50	.51	.51	.53	.45	.46	.48
Initial Cost ($)	13206	12806	12100	11338	11020	10305	14694	14182	13363

Table 7-I (b)

Summary of Liquid Series System Sizing Calculations (Future Technology)

Thermal Data	Washington D. C.			Fort Worth Texas			Madison Wisconsin		
Storage Size (m^3/m^2)	.6	.4	.2	.6	.4	.2	.6	.4	.2
Space Heating Load (TL) (GJ)	59.6	59.6	59.6	38.4	38.4	38.4	78.3	78.3	78.3
F_{max}	1	1	1	1	1	1	1	1	1
Cost Data									
FC ($)	7848	7848	7848	7546	7546	7546	7834	7834	7834
VC ($\$/m^2$)	174	142	110	165	133	101	170	138	106
Optimum									
Collector Area (m^2)	43	51	60	36	42	50	50	58	69
F	.72	.73	.75	.77	.78	.80	.70	.71	.73
Initial Cost ($)	15279	15125	14353	13604	13269	12300	16311	15830	15137

procedure; the lowest initial cost system was chosen as the "sized" system. In all cases the 0.2 m^3/m^2 storage size was most economic.

Tables 7-2 and 7-3 present the data and results from the sizing of the liquid and air parallel systems, respectively. Storage size was not a sizing variable as it was fixed at rule of thumb values discussed earlier.

The areas chosen by system and location are summarized below.

	Collector Area (m^2)		
	Washington	Fort Worth	Madison
Liq. Series (Today)	43	35	56
Liq. Series (Future)	60	50	69
Liq. Parallel (Today)	47	32	60
Liq. Parallel (Future)	43	30	53
Air Parallel (Today)	38	28	50
Air Parallel (Future)	35	27	43

Table 7-2

Summary of Liquid Parallel Sizing Calculations

Data	Washington, D.C.	Fort Worth, TX	Madison, WI
Today's Technology			
f_{max}	.92	.88	.94
FC ($)	7829	7543	7824
VC ($)	222	217	220
Optimum Area (m^2)	47	32	60
Optimum f	.59	.65	.55
Optimum initial cost ($)	18263	14487	21024
Future Technology			
f_{max}	.92	.88	.94
FC ($)	8329	8043	8324
VC ($)	222	217	220
Optimum Area (m^2)	43	30	53
Optimum f	.63	.68	.59
Optimum initial cost ($)	17875	14553	19984

Table 7-3

Summary of Air Parallel Sizing Calculations

Data	Washington, D.C.	Fort Worth, TX	Madison, WI
Today's Technology			
f_{max}	.95	.91	.96
FC ($)	5539	5550	5721
VC ($)	297	288	293
Optimum Area (m^2)	38	28	50
Optimum f	.50	.56	.443
Optimum initial cost ($)	16825	13614	20371
Future Technology			
f_{max}	.95	.91	.96
FC ($)	6039	6050	6221
VC ($)	297	288	293
Optimum Area (m^2)	35	27	43
Optimum f	.54	.60	.5
Optimum initial cost ($)	16434	13826	18820

8.0 DISCUSSION OF MODEL LIMITATIONS, SYSTEMS APPROACH METHODOLOGY, AND SIMULATION RESULTS

The discussion of results falls into three groups: the discussion of the limitations of computer modeling for systems comparisons; the discussion of the methodological approach taken in this study to establish the foundation for this and future system comparisons; and the discussion of the systems simulation results concerning thermal performance, economic analysis, thermal comfort standardization, reliability issues, and utility impact factors. Finally, the LCC sensitivity of CSHP systems to retail collector cost is discussed.

8.1 LIMITATIONS OF MODELING FOR SYSTEMS COMPARISONS

Modeling is an attempt to mimic a system with a set of mathematical relationships which are practical to solve; thus, approximations are required in order to obtain a useful set of mathematical relationships. Deriving mathematical relationships which describe portions of the system from fundamental principles is a science, while simplifying the fundamental set of relations to obtain a set which is practical to solve, and which also retains the important systems-level characteristics, is an art. From a systems analysis viewpoint, the approximations required fall into two categories.

The first category consists of approximations which do not affect the ability of a model to compare systems. The approximations either do not change systems-level predicted performance or they bias the predicted performance of all systems being compared in a uniform manner. Approximations of the latter type will cause modeled performance deviations, but will not undermine the ability of the modeling tool to compare systems. Validation exercises have been undertaken to assess the deviation between actual and predicted performance (68,69).

The second category consists of approximations which do impact the ability of a model to compare systems. The major approximations of this type which impact the heat pump study are listed below:

- No liquid collector manifold and piping heat losses were considered

- No air collector manifold and duct heat losses and air leaks were considered
- Series system collectors were modeled with constant loss coefficients although windspeed is probably an important factor for low-cost collectors
- Thermal transients in collectors, piping, ductwork, heat pumps, auxiliary heaters, etc., were not modeled.

All of these factors will tend to improve the competitive basis of the combined solar heat pump systems when compared to the conventional alternatives.

8.2 DISCUSSION OF METHODOLOGICAL APPROACH

The general methodological approach used in this study is outlined in subsection 1.3 Its application to this study is described in subsections 2.1 and 2.2 Subsection 8.2.1 summarizes the rationale for developing this methodology. Subsection 8.2.2 discusses the application of this methodology for the comparison of combined solar heat pump systems to their conventional alternatives.

8.2.1 Summary of Rationale

The following paragraphs outline the elements considered in developing the study methodology.

Limitations of Past Studies

In general, the scopes of past studies have been limited and procedures inconsistent, making it difficult to draw meaningful conclusions. Studies which conclude that the series system is superior (1, 2, 3, 4) fail to compare complete heating, cooling, and water heating systems on an annual basis. The costs used for series system collectors and the performance projections for the liquid-to-air heat pump being developed for this application have been criticized as optimistic. Studies which conclude that the parallel system is superior (5, 6, 7, 8) generally draw from thermal performance predictions which show parallel out-performing series at any collector area for systems utilizing the same collectors, storage size, and heat pump characteristics. This work has been criticized because collector type, collector area, storage size, and heat pump characteristics were not optimized for each system individually before comparisons were made.

The models used in all of the past studies were not capable of forcing each system to deliver a comparable level of thermal comfort. It is not clear which system is favored by this limitation. A critical evaluation of past models was performed to uncover other limitations.

The ambiguity caused by inconsistent systems analysis methodologies and questionable modeling practices leaves the series/parallel question open to debate. A general methodology was needed which would overcome the problems of past studies. The purpose of the methodology was to establish the framework in which new residential heating, cooling, and water heating technologies must compete if they are to achieve significant market penetration. The methodology should be as general as possible so that, in addition to answering the series/parallel question, the economic framework can be used to compare future system concepts, passive measures, and conservation measures on the same basis.

Methodology Development

The methodology developed addresses selection of climatic locations (subsection 2.1.1), meteorological forcing functions (subsection 2.1.2), selections of housing types for each location (subsection 2.2.1), water heating loads (subsection 2.2.2), selection of competing systems (subsection 2.3.1), and model improvements to facilitate system comparisons (Section 3). The focus has been to create a rational framework for comparing new and existing technologies for residential heating, cooling, and water heating.

The Current Study

The current study has placed conventional systems and combined solar heat pump systems into the comparative framework. The models have been formulated on the following bases:

- Thermal
- Economic
- Comfort
- Reliability
- Utility Impact.

In addition, the models have been formulated in such a manner that any of the following energy conservation concepts could be added to the same framework:

- Direct active solar systems
- Passive-design concepts
- Energy-conserving controls (day/night setback, etc.)
- Conservation measures
- New system concepts.

8.2.2 Application of Methodology to Current Study

Application of the methodology to the current study brought several problems to the forefront which do not have generally agreed-upon solutions. These are itemized below with a description of the approach taken.

- How to force all systems to meet thermal comfort criteria:

  It was resolved that uniform thermal comfort is guaranteed if all systems have the same load air flow rate and deliver energy at or above the same specified temperature (see subsection 5.3). Temperature level control simulation was required so that the models could force the delivered air temperature requirement on each system (see subsection 3.2.2). Interactive, rather than precalculated, loads were required in order to perform accurate temperature level control simulations (see subsection 3.2.3). Nonstandard controls were required to make stand-alone heat pump meet the comfort requirements (see subsection 3.4.2). The heat pump in the series system was sized so that the comfort requirement was always met without using auxiliary unless the heat pump was starved (see subsection 3.5.4).

- How to model the heat pump:

  A new TRNSYS-compatible heat pump model was developed which calculates sensible and latent cooling effect as a function of indoor entering wet bulb, dry bulb, and sink temperatures. The heating mode model start-up transients have been added. The model is formulated to interface with the rest of the system via temperature level control (see subsection 3.3).

- How to model the load:

  Since loads calculated simultaneously with the system simulation were desired, the existing TRNSYS load package was reformulated into a single component model to make the simulations more economical. Several improvements were also made. The capability of providing an overhang on the south wall, and of calculating glazing transmittance as a function of incident angle were added (see subsection 3.2).

- How to size collectors, storage, and heat pumps for direct system comparisons:

  Stand-alone heat pumps and parallel system heat pumps were sized using conventional techniques. Series system heat pumps were sized to meet thermal comfort criteria without using auxiliary (see subsection 3.5.4). Parallel system storage size was referenced to collector area using the conventional rule of thumb. Series storage was sized on economic criteria. Both parallel and series collectors were sized on the basis of minimizing system cost per unit purchased energy displaced (see subsections 3.5.3 and 3.5.4).

- How to control each system:

  Previous studies used energy rate control which essentially assumes that optimum system/load control occurs without getting into the details of how it is done. Temperature level control requires the modeler to choose the controls. Conventional or simple control algorithms were used where possible (see subsection 3.4).

8.3 DISCUSSION OF SIMULATION RESULTS

The simulation results are discussed below in terms of thermal performance, system economics, comfort, and utility impacts.

8.3.1 Thermal Performance

The bottom-line thermal results of this study are presented in Table 6-1. This table presents the amount of conventional energy which must be delivered in order to meet the heating, cooling, and water heating loads in each location and for each system. The portion of the purchased energy which <u>must</u> be electricity is broken out and presented in brackets. This method of presentation allows the calculation of life-cycle costs for any combination of system and backup fuel.

It should be noted that each system has been sized using conventional design guidelines, or using the economic criteria explained in subsections 7.3 and 7.4. The collector areas, storage sizes, and heat pump sizes in the combined solar heat pump systems are not the same. Rather, each system has been optimally sized based on its own thermal characteristics. The initial costs of each system are also different.

Table 6-1 indicates that the liquid series system with a future technology heat pump offers the greatest potential energy savings. A single run with the air series system in Washington, D.C., indicates inferior performance due to increased parasitics and lower collection efficiencies (due to air collectors, air-to-water heat exchanger). The parallel, series, and conventional systems should not be compared directly at this point due to their different sizes and initial costs.

Heat pump starvation in the series system occurred at all locations except Fort Worth with the future-technology heat pump. It is not, however, considered practical to design the series system with no backup energy source in any location due to the variability of weather from year to year (see subsection 6.2.3). The backup could, however, be an alternate heat pump source (e.g., ground-coupled coil) rather than electric heat.

Tables 8-1 through 8-3 summarize various subsystem efficiencies. The operating collector efficiencies (CEFFON) are generally higher for the series system, with a large improvement shown for the future-technology heat pump. Heating mode heat pump COPs increase greatly for the liquid series system and slightly for the systems using air-to-air heat pumps. The exception is in Madison where the future-technology air-to-air heat pump actually has the same or lower heating mode COP. The two-speed future heat pump design has a large capacity at low source temperatures, however, so that much less auxiliary heat is required and FSSH still increases.

At the water heating subsystem level, it is seen that the stand-alone solar water heating system used in conjunction with the liquid series system outperforms the air parallel system design where water heating is integrated into the space heating system. The reason for this is the large parasitic energy loss associated with operating the main collector array in summer strictly to provide hot water. This is in general agreement with experience from the HUD demonstration program. Another reason is that, in winter, heated water can only be obtained if the collectors are operating for some other purpose (collector to storage or load).

Table 8-1

Washington, D.C.: Annual Subsystem Performance Summary

		All-Day Collector Efficiency (COLEFF)	On-Time Collector Efficiency (CEFFON)	Heating Mode COP (COPH)	Cooling Mode COP (COPC)	Heating Free Energy Fraction (FSSH)	Hot Water Free Energy Fraction (FSDHW)
Stand-Alone Heat Pump	Today	-	-	2.3	2.4	.40	-
	Future	-	-	2.4	3.2	.52	-
Liquid Series CSHP	Today	.19	.38	2.8	2.6	.48	.59
	Future	.24	.49	4.1	3.7	.74	.59
Air Parallel CSHP	Today	.18	.26	2.1	2.4	.52	.43
	Future	.18	.26	2.2	3.2	.60	.43
Liquid Parallel Series CSHP	Today	.25	.31	2.1	2.4	.55	.67
	Future	.25	.31	2.2	3.2	.62	.67

Table 8-2

Fort Worth, Texas: Annual Subsystem Performance Summary

		All-Day Collector Efficiency (COLEFF)	On-Time Collector Efficiency (CEFFON)	Heating Mode COP (COPH)	Cooling Mode COP (COPC)	Heating Free Energy Fraction (FSSH)	Hot Water Free Energy Fraction (FSDHW)
Stand-Alone Heat Pump	Today	-	-	2.5	2.3	.49	-
	Future	-	-	2.9	3.1	.60	-
Liquid Series CSHP	Today	.16	.34	2.8	2.3	.54	.72
	Future	.20	.46	4.9	3.1	.79	.72
Air Parallel CSHP	Today	.13	.17	2.3	2.3	.59	.46
	Future	.13	.17	2.6	3.1	.65	.46
Liquid Parallel Series CSHP	Today	.25	.30	2.3	2.3	.63	.70
	Future	.25	.30	2.5	3.1	.69	.70

Table 8-3

Madison, Wisconsin: Annual Subsystem Performance Summary

		All-Day Collector Efficiency (COLEFF)	On-Time Collector Efficiency (CEFFON)	Heating Mode COP (COPH)	Cooling Mode COP (COPC)	Heating Free Energy Fraction (FSSH)	Hot Water Free Energy Fraction (FSDHW)
Stand-Alone Heat Pump	Today	-	-	2.0	2.4	.31	-
	Future	-	-	1.9	3.2	.44	-
Liquid Series CSHP	Today	.19	.29	2.8	2.8	.45	.56
	Future	.27	.44	3.8	3.6	.70	.56
Air Parallel CSHP	Today	.20	.31	1.9	2.4	.42	.42
	Future	.20	.31	1.9	3.2	.52	.42
Liquid Parallel Series CSHP	Today	.25	.31	1.9	2.4	.44	.66
	Future	.25	.31	1.9	3.2	.54	.66

8.3.2 System Economics

Table 6-2 displays the initial cost of each optimized system. Using these costs and the thermal information from Table 6-1, life-cycle costs for each system were calculated and are displayed in Table 6-3. It is seen that the conventional stand-alone heat pump systems are far less expensive than the combined solar heat pump systems, both on a first-cost and life-cycle cost basis. Of the combined solar heat pump systems, the liquid series system appears to have the most potential for becoming competitive. However, this system is also furthest from commercial readiness as both the heat pump and collectors are still under development. Furthermore, the cost uncertainty is greatest for this system since site-built collector costs are highly dependent on contractor experience, which is negligible presently.

8.3.3 Comfort

As illustrated in subsection 6.3.2, all systems provided the same degree of thermal comfort. Previous studies have not dealt with the comfort issue either in the macro- or micro-time sense. The present results should at least open the discussion and investigation into system comfort factors involving solar system simulations.

8.3.4 Reliability

Table 6-5 indicates that the parallel configuration should improve air-to-air heat pump reliability due to the lower number of on/off and defrost cycles in the heating mode. In the cooling mode, the air-to-air heat pump in the liquid series system should be inherently more reliable since defrost is unnecessary. However, the machine must be able to operate with source temperatures in the 1.7 to 40.2^{o}C range, which may cause unforeseen reliability problems. Sizing the machine to meet the delivered air requirement at the lowest source temperature also causes a larger number of on/off cycles than experienced by the other systems.

Variables influencing solar system reliability, such as pump/blower cycles and cycle lengths, were not monitored during these simulation runs, although they could have been. Demonstration program experience indicates that much improvement is necessary in this area. It is perhaps fair to say that reliability of solar systems will never approach that of conventional or heat pump systems simply because of the physical size and complexity of the systems.

8.3.5 Utility Impact

As indicated in Figure 6-46, on an instantaneous demand basis, the liquid series system has the most negative impact on a utility if electric auxiliary is used when the heat pump starves. Unfortunately, heat pump starvation will most likely correspond with the utilities' winter peak. If starvation were eliminated or handled without electric backup, the system would have the best utility impact. Of the other systems, the stand-alone heat pump appears slightly better than the parallel systems on a demand/peak basis.

Figure 6-47 illustrates time-of-day energy usage of the systems on an annual basis. The parallel systems show more of a load bias than the other systems. Apparently, the storage effect carries these systems until early morning hours when most of the auxiliary consumption occurs.

It is not clear that utility impact should be a major factor in system comparison, since, in most areas, new residential construction is not expected to be a major component in electric utility loads. Even with 100 percent market penetration in a given area, the near-term impact would still be small.

8.4 COLLECTOR COST SENSITIVITY

With the unfavorable economic situation of the CSHP systems vis-a-vis the conventional systems, an investigation into CSHP component cost sensitivity was warranted. The initial component chosen for cost sensitivity analysis was the collector itself.

8.4.1 Cost Sensitivity Approach

The approach taken in the sensitivity analysis was to vary the collector cost (retail cost) over a range of values. For each cost and each system, the TCE method was used to resize the system's collector area to correspond with the new economic parameters. The LCC of the resized system was then determined and compared to both solar and conventional system LCC values.

The collector costs were assumed to vary from 0 to \$200 per square meter in the analysis. The zero cost collector was assumed to be one which acts and costs the same as roofing. The \$200/m^2 cost corresponds to many presently available "good" collectors.

Tables 8-4 through 8-6 present a summary of the thermal and cost parameters used in the TCE method of sizing; the optimum areas as a function of collector cost are presented. Figure 8-1 represents the retail collector cost and collector area relationship for today's and future series solar heat pump systems. Note that, with the improved heat pump (future), the series system is sized larger for any given collector cost. Note also that, at zero cost (free) collectors, the systems are of finite size.

8.4.2 LCC Comparisons

The LCC of each resized system was calculated for each collector cost. The change in LCC for a system over the range of collector costs yielded the system's sensitivity to collector costs. Figures 8-2 through 8-4 represent this system LCC variation with collector cost for today's technology liquid series and parallel heat pump systems in Washington, Madison, and Fort Worth, respectively. The conventional furnaces and stand-alone heat pump LCC are also shown on the above three figures to illustrate comparative relationships.

The utility of the cost sensitivity analysis is realized by examining Figures 8-2 through 8-4. Note that, in Washington, D.C. (Figure 8-2), neither series nor parallel heat pump systems are ever LCC competitive with conventional systems. However, it is also noted that series systems have a lower LCC value than parallel for any given collector cost. A similar situation exists in Madison (Figure 8-3), except that the series system LCC betters the conventional oil furnace LCC if the collector cost is below $\$40/m^2$. Figure 8-4 illustrates the parity of series and parallel systems in Fort Worth, with both bettering oil furnace systems below $\$30/m^2$ collector costs.

One issue to note is that the series system collector (low performance) is inherently more amenable to low-cost construction than the parallel system collector (high performance). Thus, the comparison should not be made between systems at the same collector cost; the particular collector should be costed for that particular system.

Another point, arising from the LCC comparison between the CSHP systems and the conventional systems, is that the differential gap between solar and conventional LCC's is readily apparent. As is seen in the above figures, even with "free" collectors, the systems are not LCC competitive with conventional alternatives. Other components of the systems should undergo cost sensitivity analysis, as well as economic factors such as "repair and maintenance."

Table 8-4

Summary of Liquid Series Sizing Calculations

Collector Cost ($\$/m^2$)	Today's Technology 0	39	100	150	200
Washington DC					
f_{max}	.63	.63	.63	.63	.63
FC (\$)	7348	7348	7348	7348	7348
VC (\$)	71	110	203	253	303
Optimum Area (m^2)	50	43	32	29	27
Optimum f	.53	.50	.46	.44	.43
Optimum initial cost (\$)	11148	12100	13525	14503	15196
Fort Worth TX					
f_{max}	.63	.63	.63	.63	.63
FC (\$)	7046	7046	7046	7046	7046
VC (\$)	62	101	226	244	294
Optimum Area (m^2)	43	35	22	23	21
Optimum f	.55	.53	.49	.48	.46
Optimum initial cost (\$)	9927	10350	11692	12633	12855
Madison WI					
f_{max}	.63	.63	.63	.63	.63
FC (\$)	7334	7334	7334	7334	7334
VC (\$)	67	106	167	217	299
Optimum Area (m^2)	65	56	47	43	35
Optimum f	.51	.48	.43	.41	.38
Optimum initial cost (\$)	12014	13363	15120	16845	17488

Collector Cost ($\$/m^2$)	Future Technology 0	39	100	150	200
Washington DC					
f_{max}	.95	.95	.95	.95	.95
FC (\$)	7848	7848	7848	7848	7848
VC (\$)	71	110	171	285	303
Optimum Area (m^2)	78	60	48	32	32
Optimum f	.79	.75	.70	.67	.67
Optimum initial cost (\$)	13389	14353	16068	16702	17756
Fort Worth TX					
f_{max}	.92	.92	.92	.92	.92
FC (\$)	7546	7546	7546	7546	7546
VC (\$)	62	101	162	212	262
Optimum Area (m^2)	65	50	38	32	29
Optimum f	.84	.80	.76	.74	.71
Optimum initial cost (\$)	11545	12300	13802	14168	15027
Madison WI					
f_{max}	.96	.96	.96	.96	.96
FC (\$)	7834	7834	7834	7834	7834
VC (\$)	67	106	167	217	267
Optimum Area (m^2)	91	69	54	48	43
Optimum f	.77	.73	.68	.65	.62
Optimum initial cost (\$)	13964	15137	16818	18426	19360

Table 8-5

Summary of Liquid Parallel Sizing Calculations

Collector Cost ($/m^2)	Today's Technology				
	0	50	100	155	200
Washington DC					
f_{max}	.92	.92	.92	.92	.92
FC ($)	7829	7829	7829	7829	7829
VC ($)	67	117	167	222	267
Optimum Area (m^2)			56	47	42
Optimum f			.62	.59	.57
Optimum initial cost ($)			17181	18263	19043
Fort Worth TX					
f_{max}	.88	.88	.88	.88	.88
FC ($)	7543	7543	7543	7543	7543
VC ($)	62	112	162	217	262
Optimum Area (m^2)	61	45	37	32	29
Optimum f	.75	.71	.68	.65	.63
Optimum initial cost ($)	11325	12583	13537	14487	15141
Madison WI					
f_{max}	.94	.94	.94	.94	.94
FC ($)	7824	7824	7824	7824	7824
VC ($)	65	115	165	220	265
Optimum Area (m^2)				60	52
Optimum f				.55	.53
Optimum initial cost ($)				21024	21604
	Future Technology				
Collector Cost ($/m^2)	0	50	100	155	200
Washington DC					
f_{max}	.92	.92	.92	.92	.92
FC ($)	8329	8329	8329	8329	8329
VC ($)	67	117	167	222	267
Optimum Area (m^2)		68	52	43	38
Optimum f		.67	.64	.63	.61
Optimum initial cost ($)		16285	17013	17875	18475
Fort Worth TX					
f_{max}	.88	.88	.88	.88	.88
FC ($)	8043	8043	8043	8043	8043
VC ($)	62	112	162	217	262
Optimum Area (m^2)	56	42	35	30	27
Optimum f	.767	.73	.70	.68	.66
Optimum initial cost ($)	11515	12747	13713	14553	15117
Madison WI					
f_{max}	.94	.94	.94	.94	.94
FC ($)	8324	8324	8324	8324	8324
VC ($)	65	115	165	220	265
Optimum Area (m^2)			68	53	47
Optimum f			.6	.59	.57
Optimum initial cost ($)			19544	19984	20779

Table 8-6

Summary of Air Parallel Sizing Calculations

	Today's Technology				
Collector Cost ($/m^2)	0	50	100	150	194
Washington DC					
f_{max}	.95	.95	.95	.95	.95
FC ($)	5539	5539	5539	5539	5539
VC ($)	103	153	203	253	297
Optimum Area (m^2)		59	49	42	38
Optimum f		.56	.54	.52	.50
Optimum initial cost ($)		14566	15486	16165	16825
Fort Worth TX					
f_{max}	.91	.91	.91	.91	.91
FC ($)	5550	5550	5550	5550	5550
VC ($)	94	144	194	244	288
Optimum Area (m^2)	61	44	36	31	28
Optimum f	.64	.62	.60	.58	.56
Optimum initial cost ($)	11284	11886	12534	13114	13614
Madison WI					
f_{max}	.96	.96	.96	.96	.96
FC ($)	5721	5721	5721	5721	5721
VC ($)	99	149	199	249	293
Optimum Area (m^2)			68	58	50
Optimum f			.47	.458	.443
Optimum initial cost ($)			19253	20163	20371
	Future Technology				
Collector Cost ($/m^2)	0	50	100	150	194
Washington DC					
f_{max}	.95	.95	.95	.95	.95
FC ($)	6039	6039	6039	6039	6039
VC ($)	103	153	203	253	297
Optimum Area (m^2)		55	45	39	35
Optimum f		.6	.58	.56	.54
Optimum initial cost ($)		14454	15174	15906	16434
Fort Worth TX					
f_{max}	.91	.91	.91	.91	.91
FC ($)	6050	6050	6050	6050	6050
VC ($)	94	144	194	244	288
Optimum Area (m^2)	60	43	35	30	27
Optimum f	.665	.646	.63	.61	.60
Optimum initial cost ($)	11690	12242	12840	13370	13826
Madison WI					
f_{max}	.96	.96	.96	.96	.96
FC ($)	6221	6221	6221	6221	6221
VC ($)	99	149	199	249	293
Optimum Area (m^2)			60	50	43
Optimum f			.53	.52	.5
Optimum initial cost ($)			18161	18671	18820

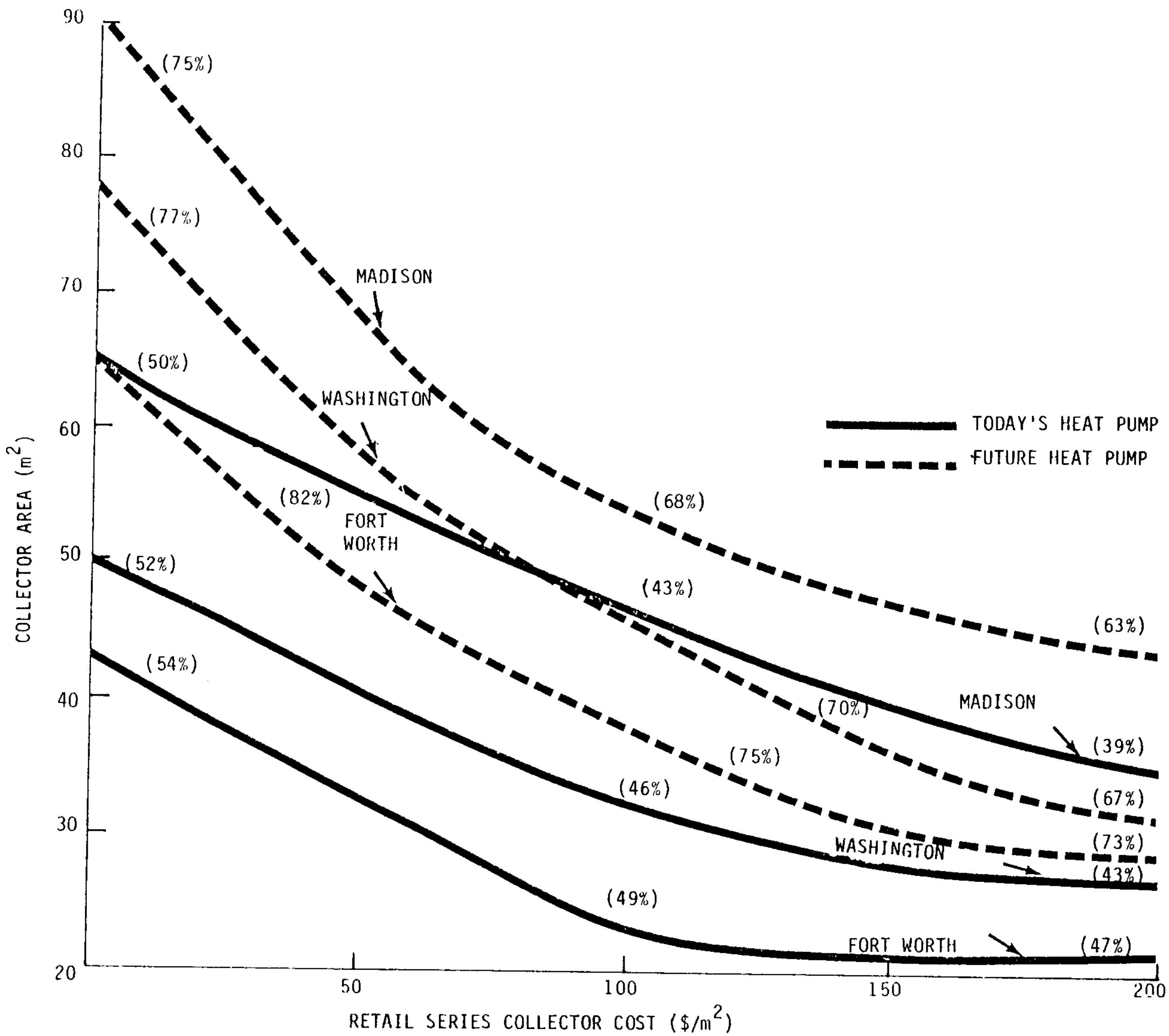


Figure 8-1. Series Collector Cost/Area Relationship

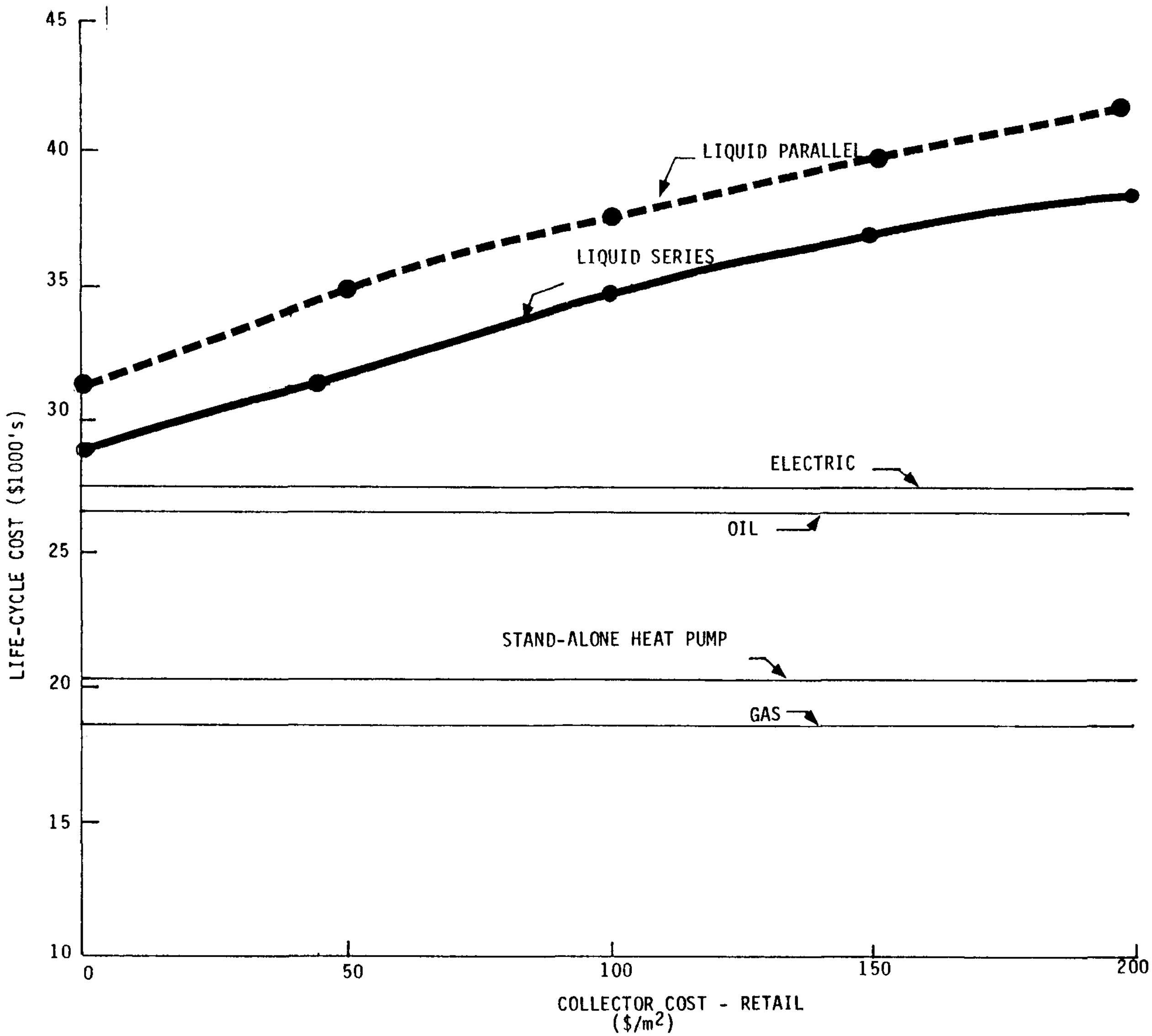


Figure 8-2. CSHP Collector Retail Cost Sensitivity on LCC: Washington, D.C.

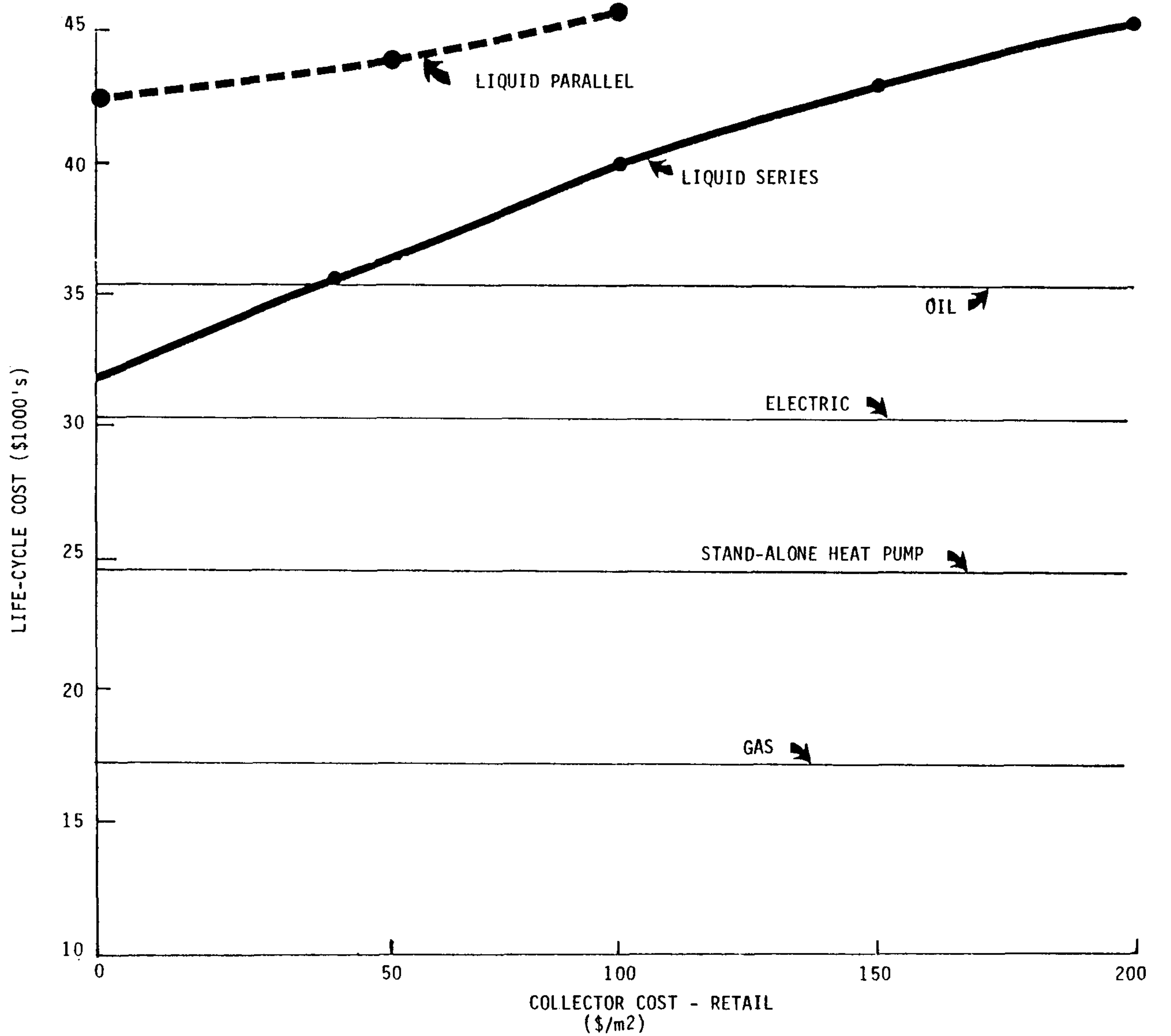


Figure 8-3. CSHP Collector Retail Cost Sensitivity on LCC: Madison, WI.

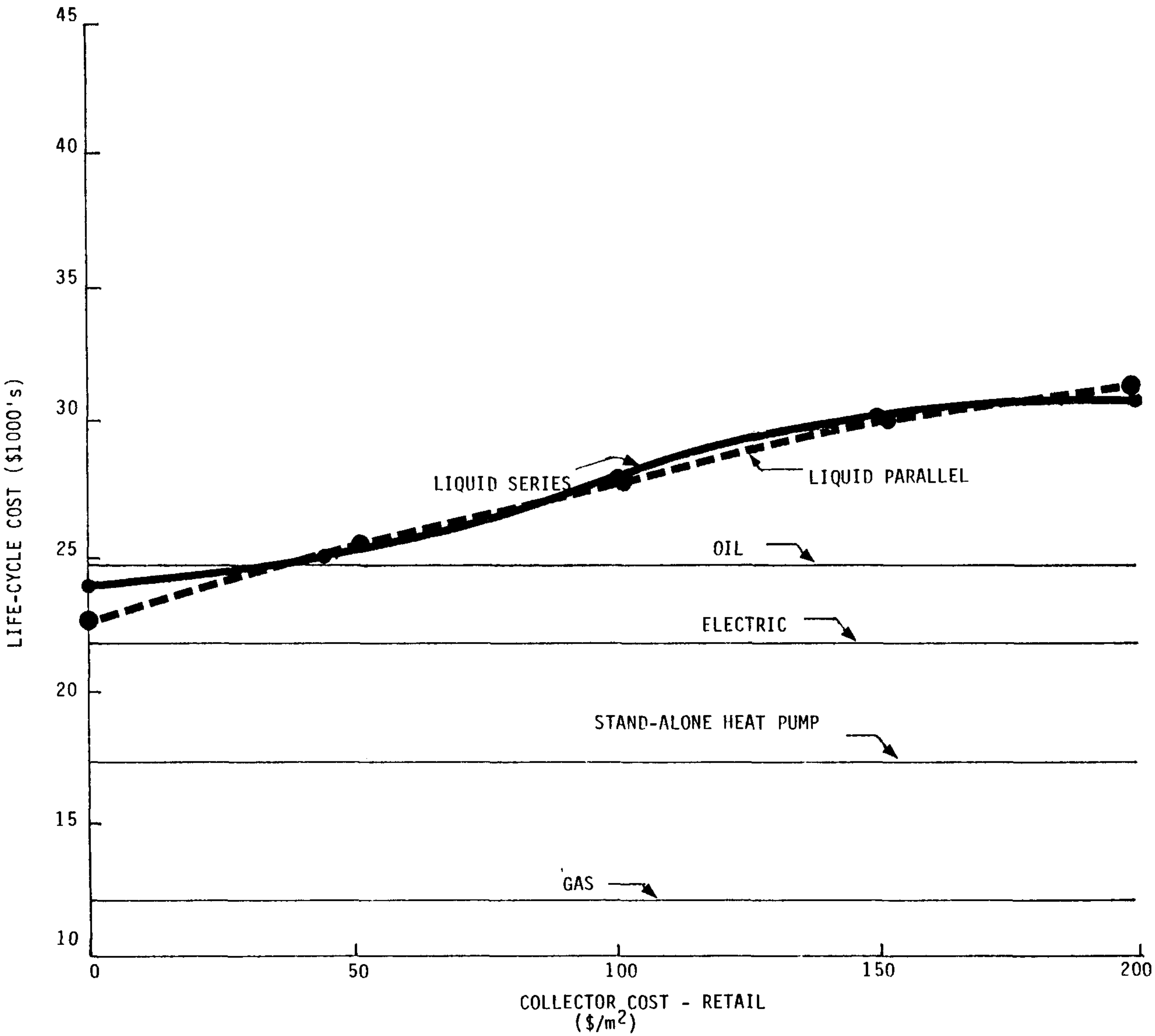


Figure 8-4. CSHP Collector Retail Cost Sensitivity on LCC: Ft. Worth, TX.

9.0 CONCLUSIONS AND RECOMMENDATIONS

The thermal/economic conclusions derived from the system simulation results are presented below. These recommendations include both system analysis approach methodological and system thermal/economic matters for this and future studies.

9.1 CONCLUSIONS

The combined solar heat pump systems included in this study are at best marginally competitive, on a life-cycle cost basis, with conventional oil and electric furnace systems. However, the payback periods are in the 15-to-25 year range. The combined solar heat pump systems are <u>not</u> economically competitive with conventional gas furnace or stand-alone heat pump systems for residential space heating, cooling, and water heating. The liquid series system comes the closest of any of the systems but still is not life-cycle cost competitive. These conclusions are in general agreement with other studies (1,4).

The combined solar heat pump systems do offer the potential for significant energy savings as compared to conventional furnace systems and the stand-alone heat pump. The cost of that savings, however, is beyond that which the average consumer can be expected to pay. Presently, the first cost of the combined solar systems ($12,000 to $20,000) is the major concern of the consumer, especially when the conventional alternatives are so much less ($2000 to $2500). Furthermore, it is possible that the same energy savings could be obtained for less cost using a combination of conventional technologies, passive techniques, and conservative measures.

Future Improvements

It appears that, in the next five-year timeframe, it is unlikely that any of the combined solar heat pump systems studied in this report will be installed for purely economic reasons. It remains to be determined what, if anything, can be done to these systems to make them competitive.

Costs for each system are itemized in Appendix E. Barring manufacturing process or materials breakthroughs, the parallel system prices are firm. Those marketing parallel systems do not foresee price decreases. The prices listed for series systems already include low-cost, site-built collectors and an optimistic estimate of the liquid-to-air heat pump cost. Prices on other series system components are firm. The collector cost sensitivity analysis (subsection 8.4) did not offer any encouraging directions toward significant system cost reductions.

One possibility for cost improvement lies in factory-built housing. Factory-built housing is a specialized market where solar space heating has economic promise. Here, the site-built construction process can occur in a controlled environment. An alternative possibility is the distribution of appropriate materials through building supply houses for on-site fabrication. However, with on-site labor included, it is not clear that site-built collectors can be installed for prices below those quoted in Appendix E for the series system.

Another possibility is the application of photovoltaic (PV) systems to the residential market. Hybrid PV/thermal collection systems have been proposed. The critical question is how cheaply PV systems can be installed since, currently, they are more expensive than thermal systems.

The competitive position of thermal systems would be greatly enhanced if a practical means of long-term energy storage existed so that backup systems could be dispensed with. However, none of the storage concepts investigated to date show significant promise: chemical methods require high temperatures to drive the reaction; phase change methods are expensive and not particularly compact; and sensible heat storage methods are impractical due to size and stand-by losses. Perhaps the chemical method is the most promising if the proper constituents and high temperature collectors can be identified.

9.2 RECOMMENDATIONS

Recommendations fall into two categories: those concerning the comparative framework/methodology developed for this study and those concerning the results of the heat pump simulation study. Recommendations with regard to the methodology are listed below, followed by the recommendations concerning the heat pump study.

9.2.1 Comparative Methodology Recommendations

In order to improve the comparative methodology, consensus must be reached on all of the problem points identified in subsection 8.2.2.

- The current analysis tools are adequate for impressing thermal comfort standards on active forced air systems, but must be extended if passive or radiant heating methods are to be included in the comparison. In either case, a comfort zone must be agreed upon and advocates of each system must recommend the control set points and deadbands appropriate for that system.

- The temperature level control heat pump model developed for this study should receive critical review and improvement if necessary.

- The residential load model developed for this study should receive critical review and improvement if necessary. In addition, an extension is necessary if passive and radiant heating systems are to be compared on a uniform thermal comfort basis.

- Agreement is needed on how to size components of solar systems when they are not yet competitive and life-cycle costing cannot be used. The sizing method used in this study should be critiqued.

- Advocates of each system should suggest the controls which they believe are most advantageous for their system.

It is suggested that all proposed technologies for new residential space heating, cooling, and water heating be compared using the framework/methodology developed here. These include:

- Advanced solar system concepts

- Passive measures

- Conservation measures

- Control concepts.

The use of the methodology and system elements developed in this study would allow uniform and meaningful cost/benefit analyses among various studies.

It is suggested that similar comparative frameworks be established for other potential solar market areas such as:

- Residential retrofit
- Commercial new construction
- Commercial retrofit
- Agricultural applications
- Industrial applications.

By building the comparative framework around a specific energy-consuming market, technology development can be mission-oriented on a systems level.

9.2.2 Heat Pump Study Recommendations

Recommendations arising from the results and conclusions of the heat pump simulation study are as follows:

- Further development of combined solar heat pump systems should proceed carefully as justified by policy-level or other noneconomic factors, unless the thermal and/or cost improvements expected substantially change their competitive basis as explained herein.
- An attempt should be made to identify other applications more economically suitable for combined solar heat pump systems than in residential heating and cooling.
- The regions with no natural gas available and high electric costs should be identified. These regions are the only near-term possibilities for "competitive" combined solar heat pump systems.
- Advanced storage concepts and hybrid PV/thermal configurations should be analyzed for solar heat pump system applications. These concepts were not included in the present study, and their cost/benefit is unknown.

REFERENCES

1. J. M. Davis, "The Development of Cost-Effective Solar-Assisted Heat Pumps for Heating and Cooling Applications," Proceedings AS/ISES Conference, Denver, August 1978.

2. J. W. Andrews, et al., "A Solar-Assisted Heat Pump System for Cost-Effective Space Heating and Cooling," Report BNL 50819, Brookhaven National Laboratory, March 1978.

3. J. W. Andrews, "Development of a Cost-Effective Solar Assisted Heat Pump System," Proceedings AS/ISES Conference, Denver, August 1978.

4. W. Kahan, et al., "The Economic Constraints on Solar Energy, Heat Pump and Solar-Assisted Heat Pump Systems," Proceedings AS/ISES Conference, Denver, August 1978.

5. J. W. Mitchell, et al., "Combined Solar-Heat Pump Systems," Solar Age, July 1978, pp. 24-28.

6. T. L. Freeman, et al., "Performance of Combined Solar-Heat Pump Systems," Proceedings AS/ISES Conference, Denver, August 1978.

7. T. L. Freeman, et al., "Performance of Combined Solar-Heat Pump Systems," Solar Energy, Vol. 22, No. 2, 1979, pp. 125-135.

8. R. L. Oonk, et al., "Modelling of Combined Air Based Solar/Heat Pump Heating and Cooling Systems," Proceedings AS/ISES Conference, Denver, August 1978.

9. "Cost-Effective Control Systems for Solar Heating and Cooling Applications," Final Report DOE Contract EG-77-C-03-1592, Honeywell, September 1978.

10. J. G. Asbury, et al., "Solar Energy and Electric Utilities: Can They Be Interfaced?" Science, 195, 1977, pp. 445-448.

11. J. G. Asbury, et al., "Assessment of Energy Storage Technologies and Systems Phase 1: Electric Storage Heating, Storage Air Conditioning and Storage Hot Water Heaters," Report ANL/ES-54, Argonne National Laboratory, August 1976.

12. Climatic Atlas of the U.S., U.S. Department of Commerce, NOAA, 1968.

13. "Phase 1/Base Data for the Development of Energy Performance Standards for New Buildings," HUD Contract H-2689, AIA Research Corporation, January 1978.

14. W. Kahan, Singer Corporation, private communication, November, 1978.

15. ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating and Conditioning Engineers, New York, 1977.

16. SOLMET Volume 1 - Users Manual, National Climatic Center, Ashville, NC, 1977.

17. "Procedure for Determining Heating and Cooling Loads for Computerizing Energy Calculations: Algorithms for Building Heat Transfer Subroutines," Energy Calculations1, HS-1000-3/77, ASHRAE, 1975.

18. E. A. Carter and A. M. Patel, "Evaluation of Methods of Calculating Solar Radiation on an Inclined Surface," Proceedings, Systems Simulation and Economic Analysis for Solar Heating and Cooling Conference, San Diego, June 1978.

19. F. Benford and J. E. Brock, "A Time Analysis of Sunshine," Trans. Am. Illum. Eng. Soc., 34, 200, 1939.

20. G. Clark, "An Assessment of Passive Cooling Processes and Systems in the United States," September 1977 - February 1978 Progress Report, DOE Contract EG-77-C-03-1600, Trinity University, 1978.

21. J. A. Duffie and W. A. Beckman, Solar Energy Thermal Processes, Wiley-Interscience, 1974, p. 76.

22. T. L. Freeman, "Evaluation of the Typical Meteorological Years for Solar Heating and Cooling System Studies," Final Report, Altas Corporation (SAI subcontractor on DOE Contract No. EM-78-C-04-4261), March 1979.

23. I. J. Hall, et al., "Generation of a Typical Meteorological Year," Volume 2.2, Proceedings AS/ISES Conference, Denver, 1978.

24. Local Climatological Data, National Climatic Center, Ashville, NC, 1977.

25. Characteristics of New Housing: 1977, Construction Reports C-25-77-13, U.S. Department of Commerce, 1978.

26. Thermal Performance Guidelines for One and Two Family Dwellings, National Association of Home Builders, 1977.

27. J. J. Mutch, "Residential Water Heating, Fuel Consumption, Economics and Public Policy," Report R1498, Rand Corporation, 1974.

28. Technical Bulletin No. 78-3, Texas Engineering Experiment Station, July 1978.

29. Handbook of Air Conditioning Design, McGraw-Hill, 1965.

30. Personal communication with City of Madison, WI, January 1979.

31. "TRNSYS Manual, A Transient Simulation Program," Report #38, University of Wisconsin Engineering Experiment Station, Version 9.2, October 1977.

32. S. A. Klein, et al., "A Method of Simulation of Solar Processes and Its Application," Solar Energy, Vol. 17, No. 1, 1975, pp. 29-37.

33. S. A. Klein, W. A. Beckman, and J. A. Duffie, "TRNSYS - A Transient Simulation Program," ASHRAE Transactions, Vol. 82, Part I, 1976.

34. M. J. Pawelski, MS, Thesis Mechanical Engineering, University of Wisconsin, Madison, WI, 1976.

35. P. L. Versteegen, et al., "Survey of Currently Used Simulation Methods," Final Report to DOE under Contract EM-78-C-04-4261, Science Applications, Inc., November 1978.

36. National Air Conditioning Contractors of America, Manual J, 1975.

37. J. W. Mitchell, Ph.D., Professor, Mechanical Engineering, University of Wisconsin, personal correspondence, 1979.

38. R. L. Oonk, MS, Thesis Mechanical Engineering, University of Wisconsin, Madison, WI, 19__.

39. R. L. Oonk, et al., "Comparison of Simulated and Measured Performance of the CSU Solar House I Heating System," Proceedings, ISES Meeting, Los Angeles, 1975.

40. M. J. Pawelski, et al., "Comparisons of Measured and Simulated Performance for CSU House I," submitted to ASHRAE Trans., 1979.

41. I. J. Hall, et al., "Generation of Typical Meteorological Years for 26 SOLMET Stations," SANDIA Laboratories Report, to be published.

42. Bulletin 38BQ-6P, Carrier Corporation, 19__.

43. "Energy Efficiency Program for Room Air Conditioners, Central Air Conditioners, Dehumidifiers and Heat Pumps," SAI-77-858-LJ, Final Report to DOE under Contract CR-04-60724-00, Science Applications, Inc., March 1978.

44. W. H. Parken, et al., "Factors Affecting the Performance of a Residential Air-to-Air Heat Pump, "ASHRAE Trans., 83, Part 1, 1977, pp. 839-849.

45. G. C. Groff, et al., "A Computer Simulation Model for Air-Source Heat Pump System Seasonal Performance Studies," presented at the Oklahoma State University Heat Pump Technology Conference, October 18-19, 1976.

46. Engineering Applications Manual, Solaron Corporation, 19__.

47. S. A. Klein, et al., "Design Procedure for Solar Air Heating Systems," Solar Energy, 1977, p. 509.

48. J. Kohler, "Rules of Thumb for Sizing Storage," Solar Age Magazine, April 1978, p. 23.

49. S. A. Klein, et al., "A Design Procedure for Solar Heating Systems," Solar Energy, 8, 1976, pp. 113-127.

50. FCHART Users Manual, EES Report 49-3, University of Wisconsin, June 1978.

51. SOLCOST Space Heating Handbook, DOE/CS-0042/3, Supt. of Doc., GPO, Stock No. 061-000-00148-4, July 1978.

52. A. M. Percino, "A Methodology for Detemining the Economic Feasibility of Residential or Commercial Solar Energy Systems," SAND 78-0931, SANDIA Laboratories, January 1979.

53. M. Davis, Program Manager, DOE/CS, SHAC RAND D, Personal Communication, 1978.

54. National Solar Heating and Cooling Demonstration Program Project Expeience Handbook, DOE, September 1978.

55. T. A. King, J. S. Moore, and H. T. Hale, "An Analysis of the Construction Costs of Ten Commercial Solar Energy Systems," The Second Solar Heating and Cooling Commercial Demonstration Program Contractor's Review, December 1978.

56. Proceedings of First New England Site Built Solar Collector Conference, May 1978.

57. Means Building Construction Cost Data, 1979.

58. Dodge Manual for Building Construction Pricing and Scheduling, 1978.

59. Application of Solar Technology for Today's Energy Needs, Volume 1, Office of Technology Assessment, June 1978.

60. "An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate," EPRI EM-319, Vol. 1, Westinghouse Electric Corporation, January 1977.

61. "SHACOB: Requirements Definition and Impact Analysis," EPRI ER-808, Volume 4, Aerospace Corporation, June 1978.

62. Cost Optimization and Estimating for Four Solar Collector Designs, Dubin-Bloom Associates, March 1978.

63. P. O. Fanger, Thermal Comfort: Analysis and Applications in Environmental Engineering, McGraw-Hill, New York, 1970.

64. ASHRAE Systems Handbook, American Society of Heating, Refrigeration and Air Conditioning Engineers, New York, 1973.

65. National Environmental Systems Contractors Association, Manual G.

66. National Environmental Systems Contractors Association, Manual K.

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16. Abstract (Limit: 200 words)
This study performed an analysis of series and parallel configured solar heat pump systems for residences. The year-round thermal performance for all the heating, cooling, and hot water system configurations were determined by simulation and compared against conventional heating and cooling systems in several geographic locations. The series and parallel combined solar heat pump systems investigated are at best marginally competitive, on a 20-year life-cycle cost basis, with conventional oil and electric furnace systems. The combined solar heat pump systems are not economically competitive with conventional gas furnace or stand-alone heat pump systems for residential space heating, cooling and water heating. The combined solar heat pump systems do offer the potential for significant energy savings as compared to conventional furnace systems and the stand-alone heat pump. The cost of that savings, however, is beyond that which the average consumer can be expected to pay. It appears unlikely that during the next five years any of the combined solar heat pump systems studied here will be installed for purely economic reasons. Further development of parallel and series combined solar heat pump systems should no longer be pursued, unless justified by policy level or other noneconomic factors.

17. Document Analysis

a. Descriptors: Heat Pumps; Solar-Assisted Heat Pumps; Residential Sector; Performance; Sizing; Load; Life-Cycle Cost; Cost; Comparative Evaluations; Solar Heating; Solar Cooling; Domestic Hot Water Systems; Statistical Models

b. Identifiers/Open-Ended Terms

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c. UC Categories

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