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REPORT

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PRELIMINARY DESIGN PACKAGE FOR PROTOTYPE SOLAR HEATING  
AND COOLING SYSTEMS

Prepared from documents furnished by

Honeywell, Inc.  
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2600 Ridgway Parkway N. E.  
Minneapolis, MN 44513

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## SECTION 1 INTRODUCTION

The information presented in this document summarizes the Heating Systems preliminary analysis and design activity. It is provided to NASA/MSFC to prepare for the Heating/Cooling Systems Preliminary Design Review to be held in Minneapolis, January 18 and 19, 1976. The scope of the information is as defined in Appendix B para. 4.1 of the contract statement of work. In addition, the Heating/Cooling Verification and Development plans are submitted with this document for NASA/MSFC approval.

The analysis presented in this document was made without site-specific data other than weather. Therefore, the results indicate performance to be expected under these special conditions.

The preliminary analysis for the 75-ton units for the commercial cooling systems was not completed. The market analysis has shown that multiples of 25-ton units represent a more marketable subsystem size. When site-specific data is available, the commercial cooling system sizes required can be achieved with multiples of the 25-ton units.

## SECTION 2

### MARKET ANALYSIS

A market analysis was completed by Lennox Industries to help establish criteria for cooling system designs. The following conclusions were drawn:

- The residential size cooling unit should be a nominal 3-ton unit.
- The commercial and multiple-family units should be 25-ton units.
- Development of a solar-assisted heat pump is required.

Table 2-1 shows sales distribution, by percentage, for Lennox residential cooling products sold in 1975 for each of their five Sales Divisions. Also indicated is the distribution when considering the entire lower 48 states and the marketing areas covered by each Sales Division. The 3-ton capacity is dominant in sales volume for each of the Sales Divisions except for the Midwest, where the 2.5-ton capacity has a slight edge.

With new standards for residential housing taking effect across the country, the sizes may tend to shift downward as the building requirements tend to reduce the maximum cooling and heating loads. However, for the near-term market, the 3-ton unit seems appropriate.

In the initial proposal, Honeywell sized a 75-ton unit for the commercial system based on the site load requirements of the RFP. An analysis of the market for 75-ton commercial units indicates that this size could be met with multiples of 25-ton units for the following reasons:

- Using a solar energy collection system, higher part load performance could be obtained with multiple 25-ton units rather than one 75-ton unit operating at low capacity.



Table 2-1. Lennox Cooling Units Sold in USA - 1975

Tons of Cooling	Percent Sales Distribution by Sales Division*					National Percentage
	Eastern	Southeast	Southwest	Western	Midwest	
1-1/2	6.9	4.6	4.0	4.0	5.5	5.3
2	21.7	9.5	11.4	13.9	26.1	18.6
2-1/2	23.7	17.9	14.3	16.5	26.4	21.0
3	26.1	31.2	32.6	29.0	24.3	27.9
3-1/2	7.9	11.2	11.2	11.0	6.0	8.8
4	6.9	13.4	14.2	13.1	5.8	9.5
5	6.8	12.2	12.3	12.5	5.9	8.9

## \* Marketing Areas for Each Sales Division:

- Eastern - Illinois (line just west of Chicago), Michigan, Indiana, Ohio, Kentucky, Tennessee, Pennsylvania, District of Columbia, North Carolina, New York, Maryland, Connecticut, Vermont, Maine, New Hampshire, Massachusetts, Virginia, West Virginia, Rhode Island, Delaware, New Jersey
- Southeast - South Carolina, Georgia, Florida, Alabama, Mississippi
- Southwest - Texas, Louisiana, Arkansas, Oklahoma, New Mexico
- Western - Arizona, Nevada, California, Utah, Oregon, Washington, Montana, Western half of Wyoming, Idaho
- Midwest - Wisconsin, Western half of Illinois, Missouri, Kansas, Nebraska, Iowa, Minnesota, North Dakota, South Dakota, Eastern half of Wyoming, Colorado

- Many more projects will be candidates for single 25-ton units than 75-ton units.
- There is much greater flexibility in matching to the load by using multiples of the 25-ton system rather than a single 75-ton system. Lennox has been marketing single-zone heating/cooling equipment in the commercial sector for many years by using equipment candidates in the 8- to 22-ton range.
- Lennox experience in marketing units is more applicable to the 25-ton capacity rather than the 75-ton size.

The use of heat pumps, specifically in the residential sector, is projected at an annual growth rate of 26 percent between 1972 and 1980. This, coupled with a declining availability of natural gas, indicates a shift toward electrical heating.

Heat pump systems, even with electrical strip heaters, will derive their energy from the least-depleting sources (hydro, coal, nuclear). In addition, higher-efficiency pumps, even served from oil and gas power systems, will have equal primary efficiency use.

The baseline heating/cooling systems delineated in the following sections have been sized using the preceding criteria as the basis.

## SECTION 3 BASELINE SYSTEMS

### 3.1 DESIGN APPROACHES

Honeywell's approach to the selection of solar heating and cooling systems is illustrated in Figure 3-1. Primarily, the approach consists of two parallel efforts: identification of all candidate solar heating and cooling subsystem components, and identification of subsystem constraints or evaluation criteria. The next step, preliminary subsystem selection, is designed to narrow the list of candidate subsystem components, using the defined constraints. The major components for system selection were collector type, storage medium, cooling subsystem, and auxiliary cooling method.

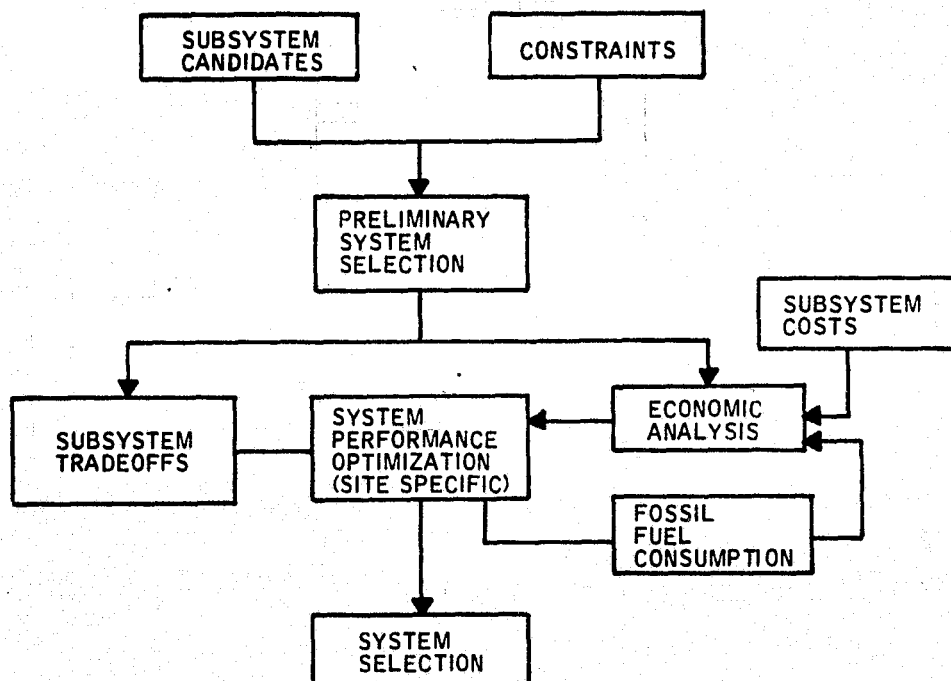


Figure 3-1. System Selection Flow Chart

The next step in system selection incorporates subsystem tradeoffs and economic tradeoffs to further reduce the number of workable and economically desirable systems. The subsystems or subsystem components upon which tradeoffs are examined are the following:

- Collector (same as heating)
- Storage
- Refrigeration section
- Rankine power section
- Auxiliary energy
- Working fluids
- Supplementary elements such as controls, piping, pumps, and heat exchangers

The economic analysis is made using methods which try to establish the most economical solar system.

The solar system performance is determined using Honeywell's Solar Systems Simulation Program including a Rankine Model. The economic analysis technique is used in conjunction with the simulation program to optimize the selected systems for the designated area and application.

### **3.1.1 Constraints and Design Criteria**

Constraints or evaluation criteria have been selected for purposes of performing preliminary and detailed tradeoffs of subsystem components. These include the following:



- Modularity -- Subsystem components that are of standard design and size that can be put together to achieve the desired capability. An example that meets this constraint is the flat-plate collector panel, a component of standard size that can be combined to create any desired collector area and the 25-ton Rankine engine, a standard subsystem size that can be combined for larger cooling loads.
- Scalability -- A subsystem component of standard design that can provide a progressive increase in capability by changing some of the components of that subsystem. An example of this type of subsystem component is the standard home furnace, the output capability of which can be increased by scaling burners and blower motor.
- Architectural Aspects -- Includes interface of solar heating systems on a building (especially collectors), impact on construction, and aesthetic qualities.
- Fuel Type Availability -- Assurance that local utilities will provide the type and amounts of fuel required.
- Economic Aspects -- Costs of procurement, installation, maintenance, and operation.
- Development Risks -- Availability of components within required time-frame, including subsystem design maturity.
- Maintainability -- Skill, knowledge, and training required to maintain system.

- Reliability -- Confidence in assuring continued system operation life cycle.
- Safety -- Safety of operation and use of system.
- Control Philosophy -- Control of solar heating system to use needed energy directly from collector to storage. Store excess energy and use auxiliary energy when required.

### 3.1.2 Preliminary Subsystem Component Selection

The preliminary subsystem selection is designed to narrow the list of subsystem candidates for final consideration. By using the constraints identified, advantages and disadvantages of each subsystem component are examined with respect to these constraints. Relative strengths and weaknesses are identified, and components can be ranked with respect to each other.

Subsystem candidates include all those solar heating and cooling system components that could be used to design heating systems. Baseline candidate subsystem components identified at the heating PDR are listed in Table 3-1. The table includes the cooling baseline subsystem components. Subsystem candidates are categorized by collectors, storage, auxiliary heating subsystems, working fluid, supplementary elements, solar power, air conditioning and auxiliary cooling subsystems.

It is evident that collectors, storage subsystems, and the cooling subsystem are the three most critical elements of heating system designs. Other subsystems, such as space heat, domestic hot water and controls, are easily defined after the selection of these primary subsystems.

Table 3-1. Subsystem Component Candidates

Subsystem	Component
Collector	<ul style="list-style-type: none"> <li>● Liquid flat plate</li> </ul>
Thermal storage media	<ul style="list-style-type: none"> <li>● Water</li> </ul>
Auxiliary heating subsystems	<ul style="list-style-type: none"> <li>● Fossil-fueled forced-air furnace</li> <li>● Fossil-fueled hydronic boiler</li> <li>● Heating-only heat pumps (optional)</li> </ul>
Working fluids	<ul style="list-style-type: none"> <li>● Water/ethylene glycol</li> <li>● Water</li> </ul>
Supplementary elements	<ul style="list-style-type: none"> <li>● Fans</li> <li>● Ducts</li> <li>● Controls</li> <li>● Piping</li> <li>● Pumps</li> </ul>
Solar power subsystem	<ul style="list-style-type: none"> <li>● Rankine power cycle</li> </ul>
Air conditioning subsystem	<ul style="list-style-type: none"> <li>● Direct expansion</li> <li>● Water chillers</li> </ul>
Auxiliary cooling subsystem	<ul style="list-style-type: none"> <li>● Electric motor</li> </ul>

The system providing the best match of subsystem components with the smallest development risk for space heating and cooling applications is a liquid, flat-plate collector with sensible heat water storage, a gas-fired, forced-air furnace as the auxiliary heating source, and a Rankine-powered, conventional air-conditioning subsystem with an electric motor as the auxiliary cooling source. In the event that gas is not available, heat pumps or electric furnaces may be considered in conjunction with the solar heating subsystem.

Numerous other workable systems exist for heating and cooling applications. Further, studies and tradeoffs would yield insights into the use of concentrators for high-temperature Rankine operation and the use of heat pumps for solar-assisted heating and cooling.

### **3.1.3 System Selection**

**3.1.3.1 Baseline Design Tradeoff Criteria** -- The heating/cooling (H/C) baseline design group generated several alternative baseline solar H/C systems for the single-family residence. These designs were then evaluated and a final selection was made according to the following basic criteria:

- Objectives of the program
- Satisfactory operation of each component in the system
- System simplicity
- System control
- Pumping energy in relation to solar energy contribution
- Tradeoff between yearly pumping energy cost and installed first cost
- System and component reliability

The information gained in developing a baseline design for the single-family residence can then be used to develop baseline designs for the multiple-family and commercial heating and cooling systems.

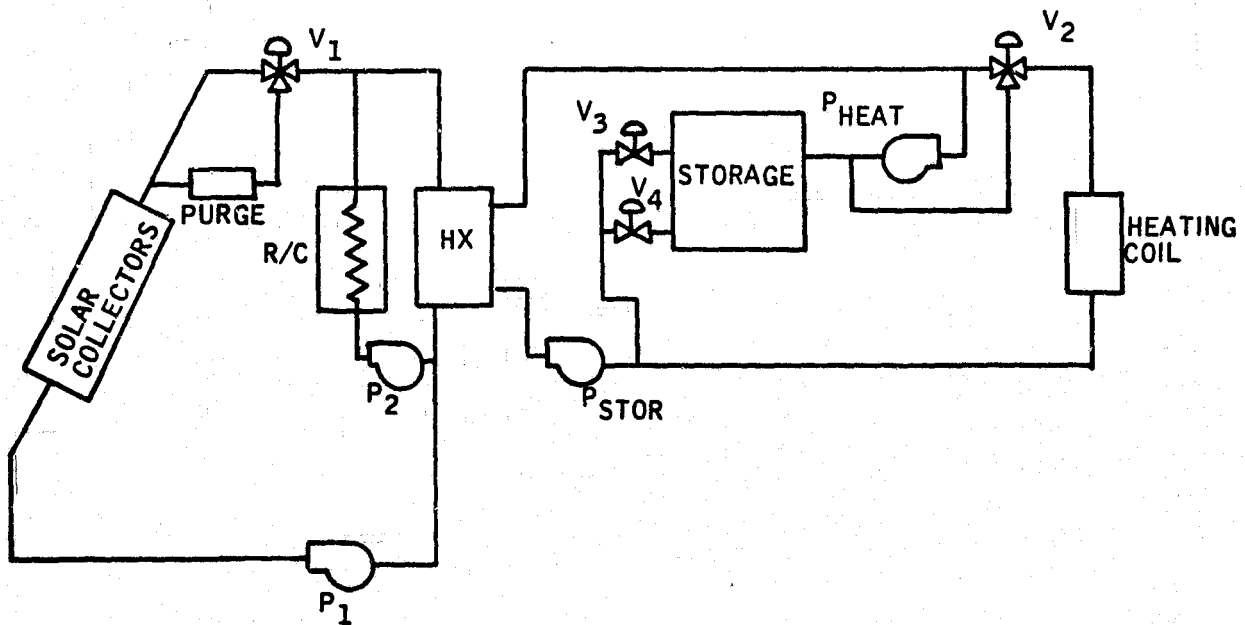
**3.1.3.2 Preliminary Alternate System Designs --** In arriving at a final single-family baseline design, many preliminary ideas were generated. This resulted in the formulation of the four alternative system designs shown in Figures 3-2 through 3-5 along with estimated pumping energy requirements.

The Preliminary Single-Family System No. 1 (Figure 3-2) is an H/C derivation of the solar heating system previously developed. This system provides for optimization of flows through the collectors and Rankine-cycle boiler on the glycol/water side of the system. This feature resulted in a serious problem with the efficiency of the R/C under certain conditions due to diluting of high-temperature fluid from the collectors with lower-temperature fluid from the R/C.

Preliminary Single-Family System No. 2 (Figure 3-3) is a very simple system which has only two pumps and has the heating coil in the glycol/water loop. It had been decided that the collectors and the R/C could operate at an equal flow, optimized for the R/C. The heat exchanger for the storage loop was placed ahead of the R/C for control purposes. This system had the advantage of being the lowest in first cost, but had the highest pumping-energy cost.

Preliminary Single-Family System No. 3 (Figure 3-4) is a modification of system 2 which adds a smaller pump on the glycol/water side for the heating and storage modes. This results in a lower pumping-energy cost but a higher first cost.

Preliminary Single-Family System No. 4 (Figure 3-5) is also a derivation of the solar heating system previously developed. Some of the features of systems 2 and 3 were also included.

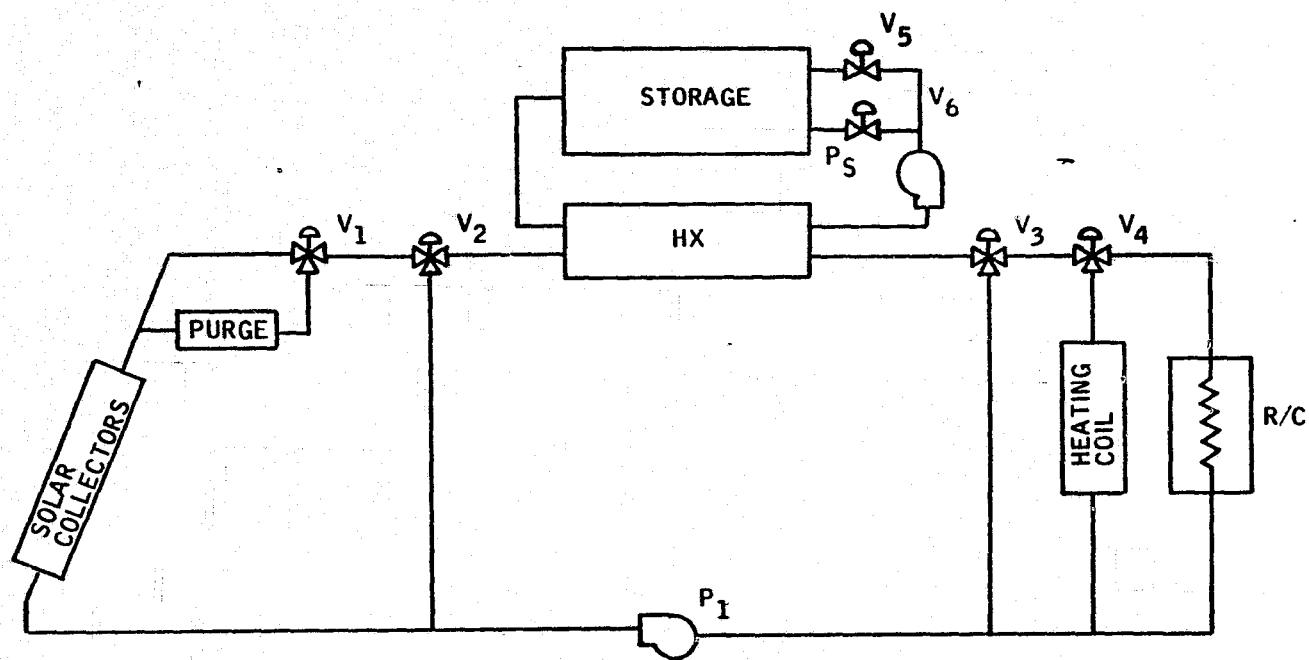


Single-Family Residence No. 1

System No. 1

Mode	$P_1$	$P_2$	$P_{stor}$	$P_{heat}$	Watts Total	Year (hr)	Year (kW-hr)
Direct Heating	380	---	380	---	760	517	393
Heat from Storage	---	---	---	380	380	124.9	47
Charge Storage	380	---	380	---	760	1180	897
Direct Cooling	380	564	---	---	944	514.7	486
Cooling from Storage	---	564	380	---	944	204.6	194
TOTAL -- Pumping kW-hr for year							2017

Figure 3-2. Preliminary Single-Family System No. 1



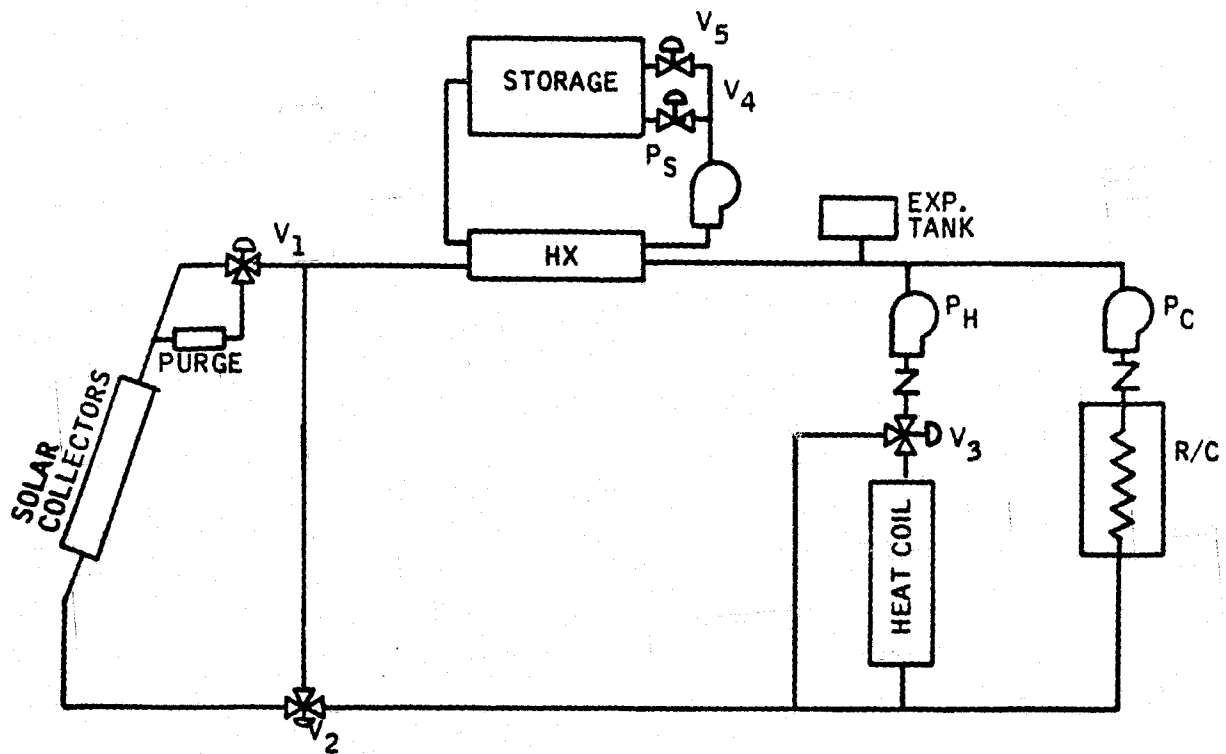
3-9

Single-Family Residence No. 2

System No. 2

Mode	$P_1$	$P_s$	Watts Total	Year (hr)	Year (kW-hr)
Direct Heating	702	---	702	517	363
Heat from Storage	702	380	1082	124.9	135
Charge Storage	702	380	1082	1180	1275
Direct Cooling	702	---	702	514.7	362
Cooling from Storage	702	380	1082	204.6	221
TOTAL -- Pumping kW-hr for year					2356

Figure 3-3. Preliminary Single-Family System No. 2



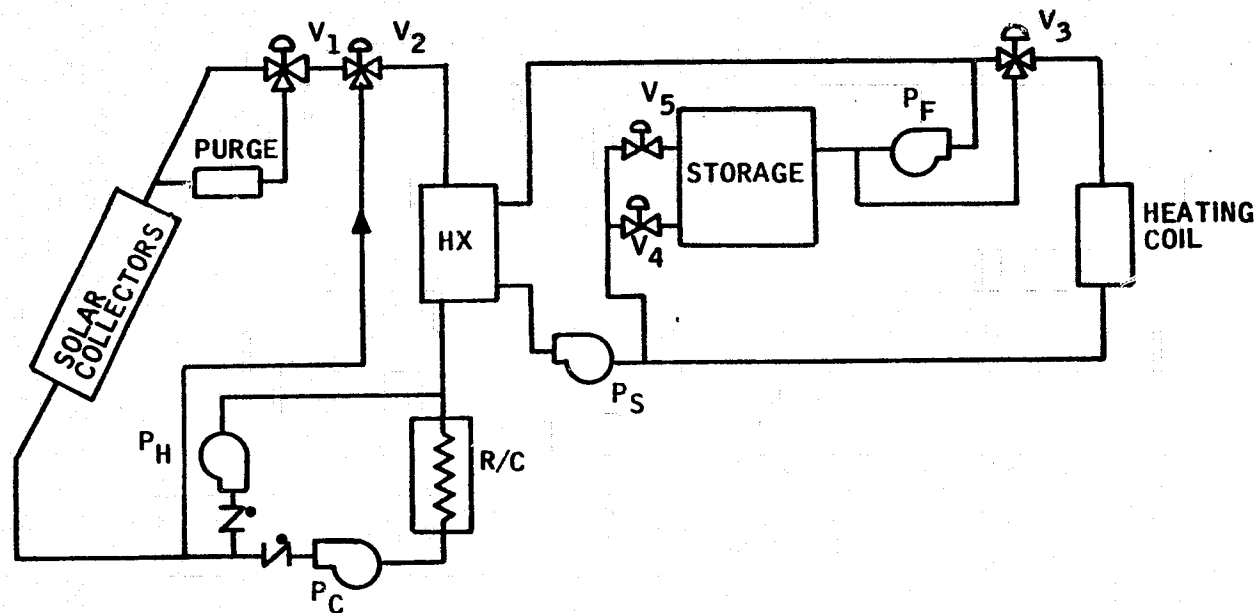
Single-Family Residence No. 3

## System No. 3

Mode	$P_H$	$P_C$	$P_S$	Watts Total	Year (hr)	Year (kW-hr)
Direct Heating	380	---	---	380	517	196
Heat from Storage	380	---	380	760	124.9	95
Charge Storage	380	---	380	760	1180	897
Direct Cooling	---	702	---	702	514.7	362
Cooling from Storage	---	702	380	1082	204.6	221
TOTAL -- Pumping kW-hr for year						1771

Figure 3-4. Preliminary Single-Family System No. 3





Single-Family Residence No. 4

System No. 4

Mode	$P_H$	$P_C$	$P_S$	$P_f$	Watts Total	Year (hr)	Year (kW-hr)
Direct Heating	380	---	380	---	760	517	393
Heat from Storage	---	---	---	380	380	124.9	47
Charge Storage	380	---	380	---	760	1180	897
Direct Cooling	---	702	---	---	702	514.7	362
Cooling from Storage	---	702	380	---	1082	204.6	221
TOTAL -- Pumping kW-hr for year							1920

Figure 3-5. Preliminary Single-Family System No. 4

**3. 1. 3. 3 Baseline System Selection** -- The four alternate designs were then evaluated according to the following analysis, with the results shown in Table 3-2.

Single-Family Systems 3 and 4 were considered to be very close in relative merit and were compared based on the data in Table 3-2 and previously mentioned criteria. As a result, single-family system 3 (Figure 3-4) has been selected as the baseline design.

The following are some of the design features of the single-family baseline design:

- The design is very flexible in terms of installation. The storage subsystem is the only water loop. Therefore, the major part of the entire system can be located conveniently inside or outside the residence regardless of environmental conditions.
- The solar collectors and the heating or cooling loads are in the same loop, thus providing very efficient use of solar energy in the direct heating and direct cooling modes.
- The pumping energy requirements are held to a minimum in all operating modes.
- The baseline design is operated by a simple control system, as discussed later in this section.

A complete description of the single-family baseline system follows later in this section.

**Table 3-2. Alternate Systems Evaluation Criteria**

Comparison Parameter	System No. 1*	System No. 2**	System No. 3	System No. 4
Year kW-hr and pumping energy cost at 4¢/kW-hr	2017 \$81	2356 \$94	1771 \$71	1920 \$77
Installed cost of pumps and control valves	\$1505	\$1400	\$1670	\$1805
Modes:				
• Direct heating	2 pumps	1 large pump; pump energy high	1 pump	2 pumps
• Heat from storage	***	2 pumps; pump energy high	2 pumps	***
• Charge storage	***	Pump energy high	***	***
• Direct cooling	2 pumps mix to R/C - poor	***	***	***
• Cooling from storage	***	***	***	***

\*Single-Family System No. 1 was discarded due to the diluting problem and its effect on R/C performance.

\*\*Single-Family System No. 2 was discarded due to the high pumping energy cost in relation to the two remaining alternatives.

\*\*\*No relative disadvantage.

The results of the single-family baseline design tradeoff study are applicable to the multiple-family and commercial heating and cooling systems. The multiple-family and commercial baseline systems have the same basic design features as previously described.

Complete descriptions of the multiple-family and commercial baseline systems follow later in this section.

**3.1.3.4 General Baseline System Description --** The proposed solar-assisted heating and cooling systems are single-loop solar collector systems interfacing with a conventional fossil-fuel-fired furnace and a Rankine-cycle air-conditioning subsystem. These proposed heating subsystem designs may be easily sized to fit a wide spectrum of applications. First, the collectors are modular and can be combined in arrays to satisfy site specific collector configurations. Second, the auxiliary subsystems are selected from a broad product line of fossil-fuel-fired warm-air furnaces, allowing many choices to fit site specific requirements. Third, the storage, hot-water, transport, and control subsystems are commercially produced items in a broad range of sizes. This subsystem modularity allows variations in system design to accommodate the variable performance requirements that are expected nationwide.

The Rankine-driven cooling subsystem will be designed and developed in two sizes: 2-ton and 25-ton. Lennox marketing analysis was used to select the most popular residential size on a nationwide basis. The 25-ton size represents the best size adaptable to larger multiple-family and commercial systems as individual or multiple systems.

The proposed solar systems are designed to maximize the amount of solar energy collected for use and storage. This is done by:

- Using high-performance flat-plate collectors
- A control system that minimizes collector inlet temperatures (maximizes energy into the building by increasing collector efficiency) and provides flexible multi-loop operation
- Independent heat exchangers using optimum transfer rate
- Direct collector to space heating, bypassing storage
- Direct collector to Rankine cooling, bypassing storage
- Storage which can be operated in parallel and independently from heating or cooling loop

System reliability and maintainability are assured through design features which indicate the following:

- A purge subsystem located to protect downstream components from overtemperatures
- A control system using simple logic
- A minimum of components in the system
- Manifolding external to the collector modules

Maintenance of space temperature, hence occupant comfort, is assured through the use of the following components and design techniques:

- A two-stage thermostat with a minimum differential for solar operation and auxiliary backup for heating and cooling
- Conventional furnace control of air temperature to the space.

The systems have been designed to minimize contamination of the potable water supply by the use of:

- A two-fluid-loop storage system that isolates the collector heat transfer fluid by two heat exchangers (two walls)
- A system in which domestic water pressure is higher than system pressures, thus assuring early detection of leaks.

### 3.2 SINGLE-FAMILY RESIDENTIAL HEATING AND COOLING SYSTEM DESCRIPTION

#### 3.2.1 System Description

The proposed system for a single-family residential heating and cooling system is a single-loop, solar-assisted, hydronic-to-warm air heating subsystem with solar-assisted domestic water heating and a Rankine-driven expansion air-conditioning subsystem. The system is composed of the following major components:

- Liquid cooled flat plate collectors
- A water storage tank
- A passive solar fired domestic water preheater
- A gas-fired hot-water heater
- A gas-fired warm-air furnace with hot-water coil unit

- A Rankine-driven direct-expansion air conditioner with auxiliary electric motor
- Water pump and cooling tower
- A tube-and-shell heat exchanger, four pumps, and associated pipes and valving
- A control system
- An air-cooled heat purge unit

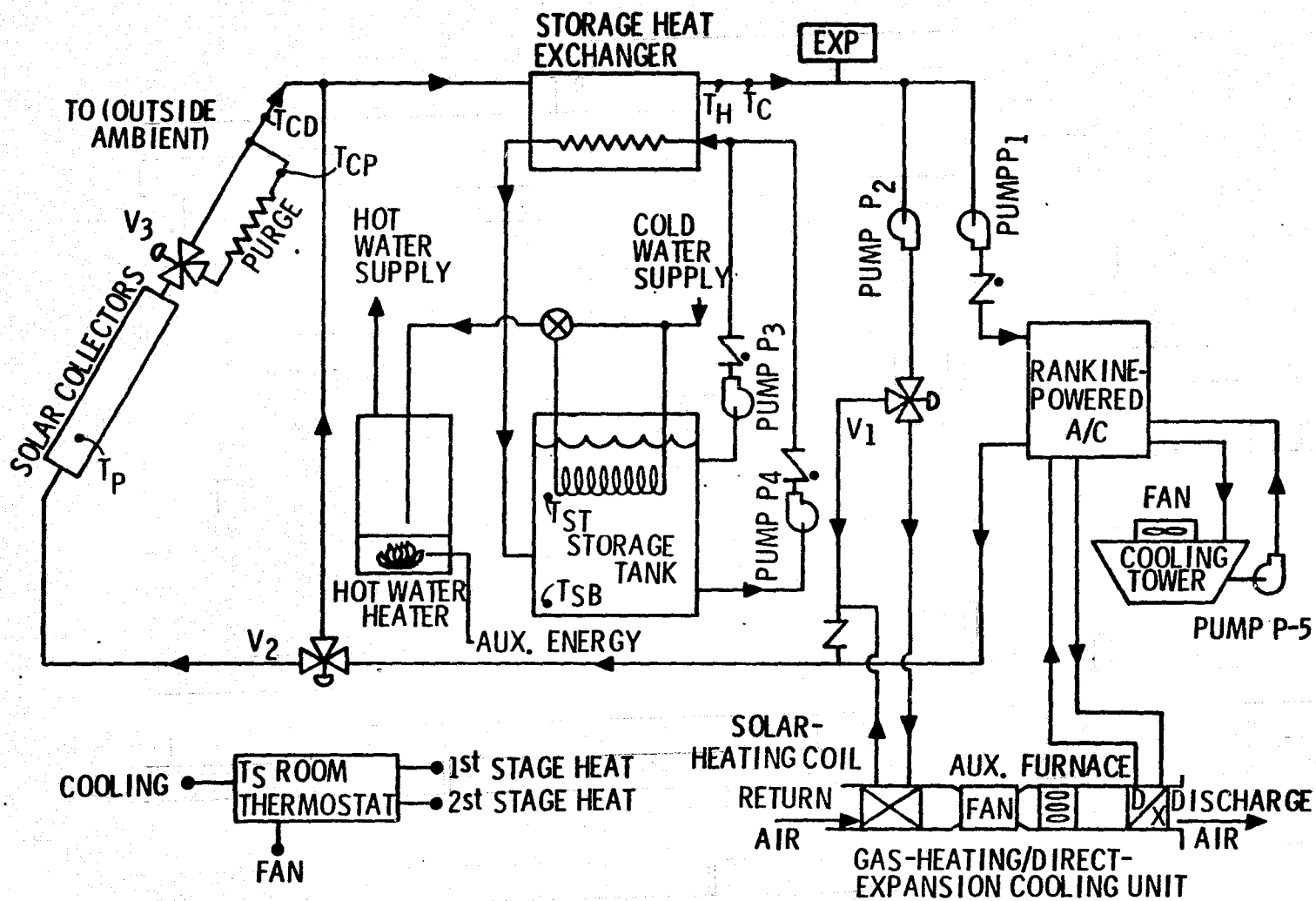
The arrangements of components within the system is as shown in Figure 3-6. The system consists of a glycol/water collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger. A domestic hot-water preheat coil is located in the storage tank.

The glycol/water collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump  $P_2$ , the Rankine boiler and pump  $P_1$ , the purge coil, and three control valves as required for the different modes of operation.

The water storage loop consists of the storage tank, storage pumps  $P_3$  and  $P_4$ , and the tube side of the storage heat exchanger.

The system provides 10 modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Rankine cooling from collectors
- Rankine cooling from storage



3-18

Figure 3-6. Single-Family Residential H/C System

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- Rankine cooling and storage simultaneously
- Electric motor auxiliary cooling
- Domestic hot water preheater
- Purge excess energy

**3.2.1.1 Heating Subsystem Operation** -- When solar energy is available and heating is required, the collectors supply heat directly to the furnace through the hot-water coil in the return air duct. Pump ( $P_2$ ) provides the heat-transfer fluid movement in this loop and the furnace blower moves the building air through the heat coil. When the heating demand is satisfied, valve  $V_1$  diverts the fluid around the hot-water coil and pump  $P_4$  operates, charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of stratification. During high solar radiation and low heating demand, both heating and storage loop operate simultaneously. If additional energy is still available; the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the furnace through the heat exchanger. Pump  $P_2$  drives the outside loop and pump  $P_3$  extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, the second-stage thermostat activates the auxiliary furnace until a comfortable temperature is maintained.

**3.2.1.2 Cooling Subsystem Operation** -- When solar energy is available and cooling is required, the collectors supply heat directly to the Rankine boiler. Pump  $P_1$  provides the heat-transfer fluid movement and, if necessary, can be sized differently than  $P_2$  to improve efficiency of the collector and Rankine operation. The Rankine drives a high Coefficient-of-Performance (COP)

compressor which, in turn, provides conventional direct-expansion cooling. When the cooling demand is satisfied, pump  $P_2$  is shut down and the system reverts to the storage mode explained in the heating subsystem.

During high solar radiation and low cooling demand, simultaneous cooling and storage is available, using pump  $P_1$  and pump  $P_4$ . If additional energy is still available, the purge coil operates by controlling the downstream temperatures to a preselected value. This is an infrequent mode and would occur if coils are oversized for a large heating demand.

When solar energy is not available or insufficient to operate the Rankine engine at the design horsepower (2.36 horsepower at 190°F collector outlet), an electric motor will operate the air conditioner independently or to make up the difference between the required horsepower and that supplied by the R/C. Storage is used to supply energy in the same manner as in the heating subsystem, except that pump  $P_1$  is the prime mover in the glycol/water loop. The baseline design uses a constant-speed compressor and therefore the electric motor is on-line at all times, supplying the balance of the required horsepower.

A variation of this concept is under study which uses a variable-speed R/C input to the compressor and thus variable-cooling output. The auxiliary motor operates only when the second-stage thermostat indicates the cooling load is not satisfied. In second stage, the system operates as a constant-speed system, delivering 3 tons of cooling.

**3.2.1.3 Specific Subsystem/Component Considerations** -- The heating/storage circuit pump,  $P_2$ , and the cooling circuit pump,  $P_1$ , are located ahead of the loads so as to provide a common location for the expansion tank. This provides for air elimination and system pressure control during all modes of operation.

When the system is heating or cooling from storage, control valve  $V_2$  diverts flow around the solar collectors.

When the system is in the purge mode, control valve  $V_3$  diverts flow through the purge coil which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps,  $P_3$  and  $P_4$ , are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump  $P_3$  draws hot water from the top of the storage tank for heating or cooling from storage. Pump  $P_4$  draws cooler water from the bottom of the storage tank for storage charging.

The domestic hot-water preheat coil, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The domestic hot-water coil in the storage tank transfers energy to the entering cold water. A thermostatic mixing valve keeps the temperature below  $140^{\circ}\text{F}$ . A conventional water heater is available downstream to add auxiliary energy as required to obtain  $140^{\circ}\text{F}$  water.

The location of each of the components within the systems is determined by the following criteria:

- Optimize performance of entire system.
- Provide reliable operation of each component.
- Provide control functions for each mode of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the loads so as to provide a means for inlet temperature control to the loads. This also provides for a convenient method for simultaneous storage charging.

The heating/storage circuit and the cooling circuit are placed in parallel, downstream of the storage heat exchanger. Each circuit is controlled by its pump and protected from backflow by a check valve.

### 3.2.2 Sequence of Operation -- Single-Family Residence

3.2.2.1 General -- Space heating and cooling is controlled by a two-stage heating ( $T_{SH}$ ) and a single-stage cooling ( $T_{SC}$ ) thermostat. First-stage heating ( $T_{SH1}$ ) is set to use solar energy, if available, and second-stage heating ( $T_{SH2}$ ) supplies the auxiliary heating if solar is not adequate. The cooling subsystem supplies a maximum of 3 tons output on demand. The horsepower required to drive the compressor is supplied by the Rankine power cycle and the electric motor. The amount of Rankine power supplied is dependent on the solar energy available.

The system control logic provides for: 1) collecting solar energy when available; 2) providing energy to the load on demand; 3) storing any excess energy; and 4) using solar energy before using auxiliary energy. A ladder diagram of the controls is shown in Figure 3-7. Figure 3-6 shows the location of the system control components.

3.2.2.2 First-Stage Heating From Collectors -- A demand for heat from the conditioned space activates the control system to first look for the energy in the collector, and through proper pump and valve selection, transfer the energy through the heating coil to the conditioned air. This mode involves valves  $V_1$  and  $V_2$ , pump  $P_2$ , and the furnace fan.

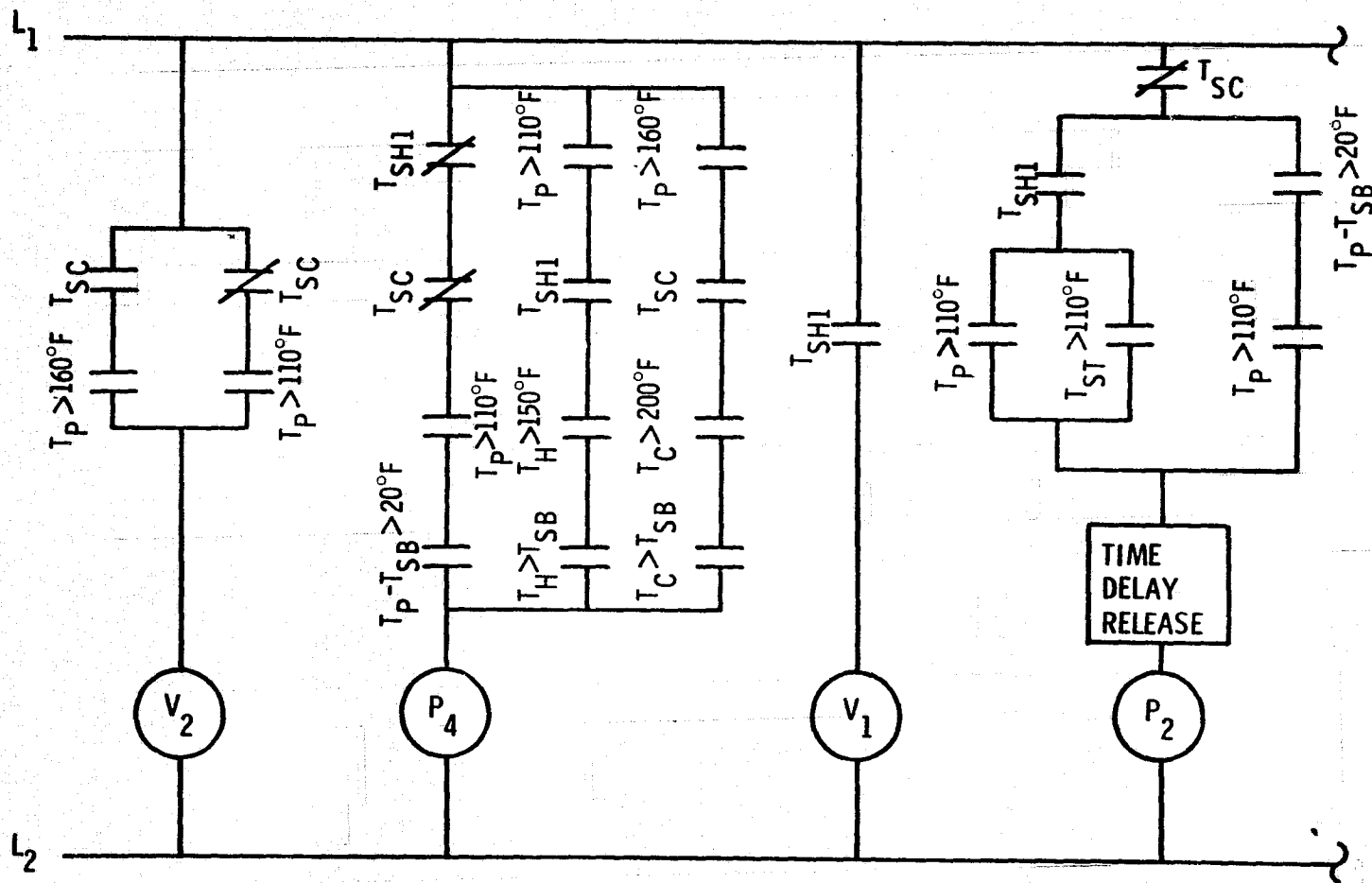


Figure 3-7. Single-Family Residence Heating/Cooling System Control Schematic

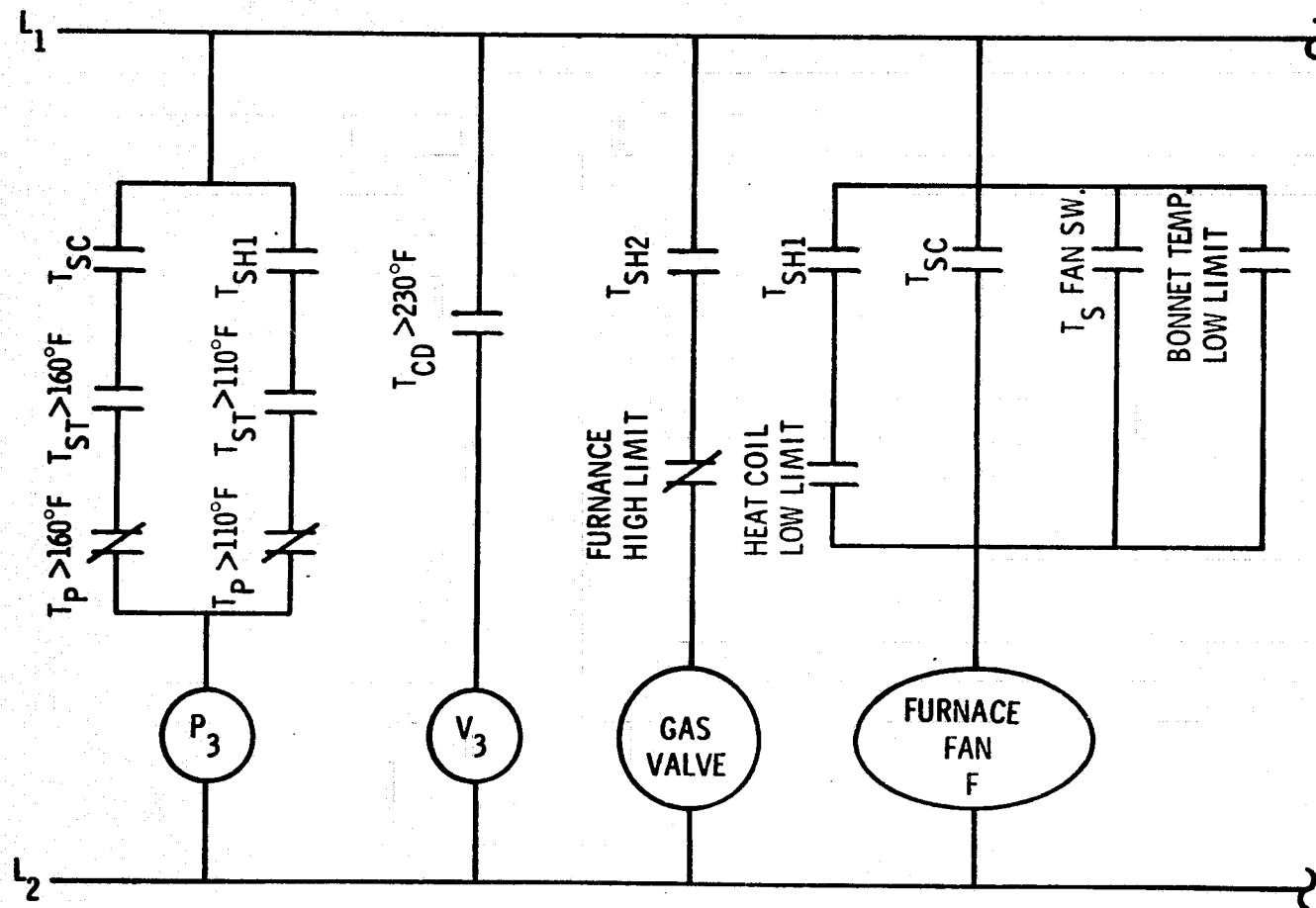


Figure 3-7. Single-Family Residence Heating/Cooling System Control Schematic (Continued)

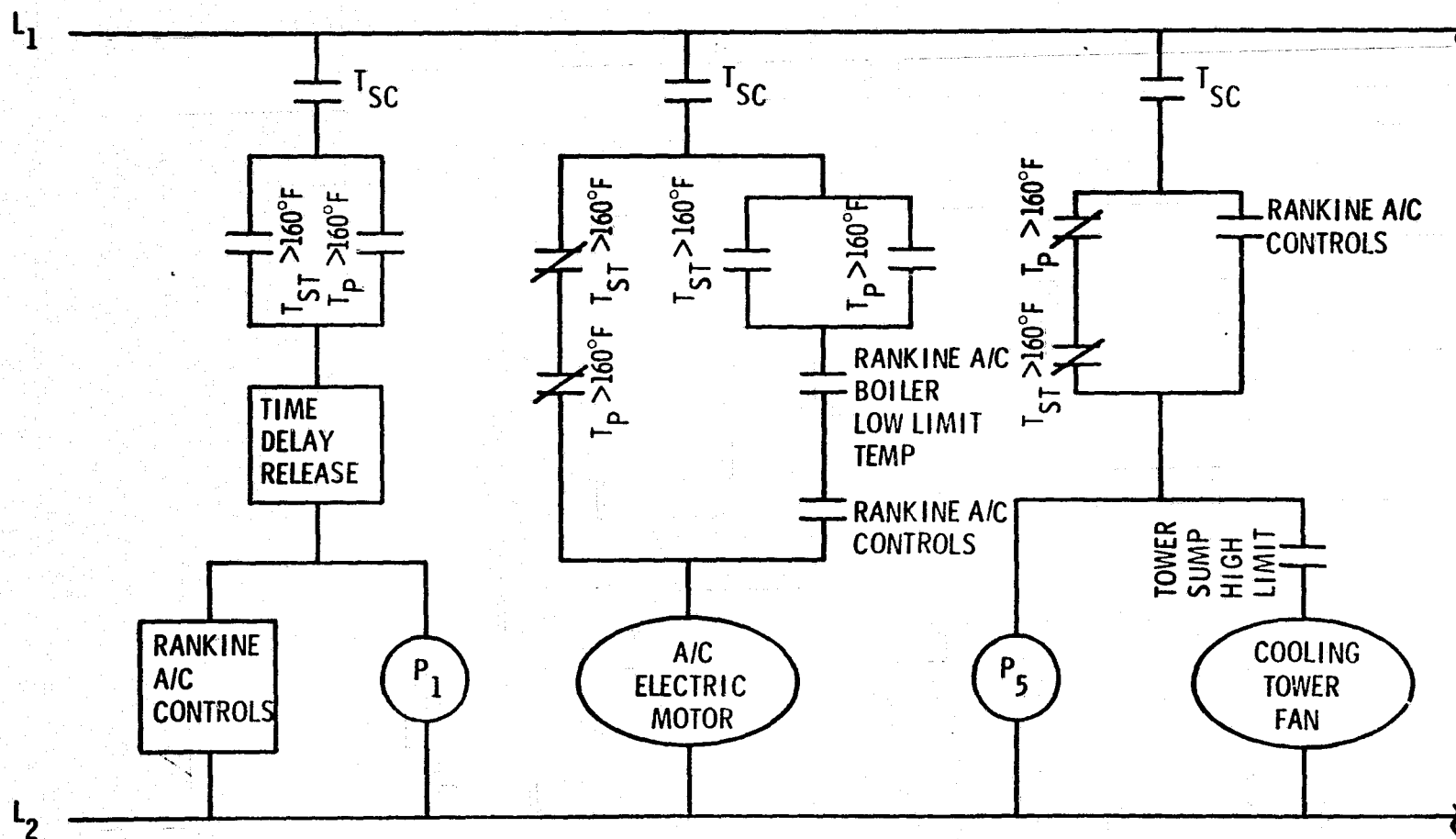


Figure 3-7. Single-Family Residence Heating/Cooling System Control Schematic (Concluded)

Whenever plate temperature,  $T_P$ , is greater than  $100^\circ\text{F}$  (adjustable) and there is a call for heating from the space thermostat ( $T_{SH1}$ ), pump  $P_2$  is activated. A shutdown time-delay relay prevents pump  $P_2$  from short-cycling. Valve  $V_2$  is positioned to direct flow to the collectors. Valve  $V_1$  is positioned to direct flow through the heating coil. The furnace fan motor is activated whenever there is a heating command from the thermostat and fluid temperature entering the heating coil is greater than  $100^\circ\text{F}$  (adjustable).

**3.2.2.3 Storage Charging While Heating** -- If the solar radiation is more than adequate to provide the heating load, the excess energy can be added to storage by activating the storage charging loop. This mode involves the same components as the heating from collectors plus pump  $P_4$ .

$P_4$  logic monitors excess energy ( $T_H > 150^\circ\text{F}$ ) and also determines that the bottom of the storage tank is at a lower temperature ( $T_H - T_{SB} > 20^\circ\text{F}$ , adjustable), activating the parallel storage mode.

**3.2.2.4 Charging Storage (Heating Season)** -- With solar energy available and no demand for heating, the control system transfers the collector energy to the storage tank. This mode involves  $V_1$ ,  $V_2$ ,  $P_2$ , and  $P_4$ .

Storage charging is done whenever there is no call for heating and  $T_P$  is greater than  $100^\circ\text{F}$  (adjustable) and  $T_P$  is greater than  $T_{SB}$  by  $20^\circ\text{F}$  (adjustable). Pumps  $P_2$  and  $P_4$  are activated and a shutdown time-delay relay prevents short-cycling of the pumps. Valve  $V_2$  is positioned to divert flow to the collectors as controlled by plate temperature  $T_P$ . Valve  $V_1$  is positioned to divert flow around the heating coil.

**3.2.2.5 Heating From Storage** -- As the availability of direct solar energy decreases and the heating demand persists, the central system first looks to storage to satisfy the demand. This mode involves  $V_1$ ,  $V_2$ ,  $P_2$ ,  $P_3$ , and the furnace fan.



Whenever  $T_P$  is less than  $100^\circ\text{F}$  (adjustable),  $T_{ST}$  is greater than  $100^\circ\text{F}$  (adjustable), and there is a call for space heat, valve  $V_2$  is positioned to divert flow around the collectors. Pump  $P_3$  is activated to discharge heat from the storage tank for space heating. Pump  $P_2$  is activated and valve  $V_1$  is positioned to direct flow to the heating coil.

**3.2.2.6 Second-Stage (Auxiliary) Heating** -- When the solar heating system can no longer satisfy the load, the temperature will continue to drop until the second-stage thermostat ( $T_{SH2}$ ) closes. If the furnace high-limit switch is in its normally closed status, the furnace is ignited, providing heat in a conventional way.

The solar heating operates independently of the conventional furnace heating subsystem and can simultaneously provide a part of the heating load. The fan will operate per the conventional bonnet temperature low limit.

**3.2.2.7 Direct Cooling From Collectors** -- A demand for cooling from the conditioned space activates the control system to first look for energy in the collector and, through proper pump and valve selection, transfer the energy through the Rankine boiler. The Rankine engine supplies shaft power to the electric motor shaft to the level that the available solar energy allows. This mode involves  $V_2$ ,  $P_1$ , furnace fan, cooling tower fan, and pump  $P_5$ .

Whenever plate temperature  $T_P$  is greater than  $160^\circ\text{F}$  (adjustable) and there is a call for cooling from space thermostat  $T_{SC}$ , pump  $P_1$  is activated and valve  $V_2$  positioned to direct flow to the solar collectors. Pump  $P_1$  discharges solar-heated fluid to the Rankine-cycle engine which drives the refrigeration compressor, providing cooling refrigerant to the furnace DX coil. The Rankine air conditioner controls prevent the auxiliary electric motor from operating until the Rankine boiler inlet has come up to a minimum operating temperature. Condenser water pump  $P_5$  is energized and the cooling tower fan is energized to run as controlled by sump water temperature. The cooling capacity of the refrigeration system is controlled by a conventional thermostatic expansion valve at the cooling coil.

**3.2.2.8 Storage Charging While Cooling** -- If the solar radiation is more than adequate to provide the cooling load, the excess energy can be added to storage by activating the storage charging loop. This mode involves the same components as the cooling from collectors plus pump  $P_4$ .

$P_4$  logic monitors excess energy ( $T_C > 200^\circ\text{F}$ ) and also determines that the bottom of the storage tank is at a lower temperature ( $T_C - T_{SB} > 20^\circ\text{F}$ , adjustable) and activating the parallel storage mode.

**3.2.2.9 Charging Storage (Cooling Season)** -- With solar energy available and no demand for cooling, the control system transfers the collector energy to the storage tank. This mode involves  $V_2$ ,  $P_2$ , and  $P_4$ .

**3.2.2.10 Cooling From Storage** -- As the availability of direct solar energy decreases and the cooling demand persists, the control first looks to storage to satisfy demand. This mode involves  $V_2$ ,  $P_1$ ,  $P_3$ , furnace fan, cooling tower fan, and pump  $P_5$ . Whenever  $T_P$  is less than  $160^\circ\text{F}$  (adjustable),  $T_{ST}$  is greater than  $160^\circ\text{F}$  (adjustable), and there is a call for space cooling, valve  $V_2$  is positioned to divert flow around the solar collectors. Pump  $P_3$  is activated to discharge heat from the storage tank to run the Rankine engine. All other functions and controls are similar to those for direct cooling.

**3.2.2.11 Auxiliary Cooling** -- Whenever there is a call for space cooling, the Rankine-cycle system is activated. A time delay will prevent the Rankine system electric motor from operating until the Rankine system boiler has reached a minimum operating temperature. The electric motor will supply the necessary energy for auxiliary cooling if solar-heated water is not available. It can also supply the additional shaft power if needed, during Rankine-cycle operation to assure a 3-ton output.

**3.2.2.12 Heat Rejector Control** -- Whenever the collector discharge temperature exceeds 230°F (adjustable) as sensed by  $T_{CD}$ , valve  $V_3$  is positioned to direct fluid through the purge coil. If a fan is required to extract additional energy (site specific), a fan cycling control will be incorporated to control purge outlet temperature.

**3.2.2.13 Domestic Hot-Water Heating** -- Whenever domestic hot water is drawn from the water heater, it is replaced by preheated water from a coil in the storage tank. A thermostatic mixing valve is used to regulate the hot-water supply temperature to 140°F maximum.

### **3.3 MULTIPLE-FAMILY RESIDENTIAL HEATING AND COOLING SYSTEM DESCRIPTION**

#### **3.3.1 System Description**

The proposed system for a multiple-family residential heating and cooling system is a single-loop, solar-powered, two-pipe hydronic heating and cooling system with a separate water storage and domestic hot-water heating loop. Being a two-pipe design, the central system is either in the heating mode or the cooling mode as determined by a Cooling Load Analyzer. The central heating is provided by either direct or stored solar energy and the central cooling is provided by a solar-powered Rankine engine/auxiliary electric motor-driven water chiller. The system is composed of the following major components:

- Liquid-cooled flat-plate collectors
- A water storage tank
- A solar-fired domestic water preheater with storage water tube bundle and circulating pump

- A gas-fired hot-water heater
- A water chiller driven by a Rankine engine and/or auxiliary electric motor
- A tube-and-shell heat exchanger, four central system pumps, and associated piping and valving
- A condenser water pump and cooling tower
- A control system
- An air-cooled heat purge unit with fan
- Twelve individual subsystems consisting of a circulating pump and a gas-fired warm-air furnace with a water heating/cooling coil

The arrangement of components within the system is shown in Figure 3-8. The system consists of a glycol/water collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger.

The glycol/water collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump  $P_2$ , the Rankine boiler and pump  $P_1$ , the purge coil, and three control valves as required for the different modes of operation.

The water storage loop consists of the storage tank, two storage pumps,  $P_3$  and  $P_4$ , and the tube side of the storage heat exchanger, and the solar DHW preheater pump,  $P_6$ .

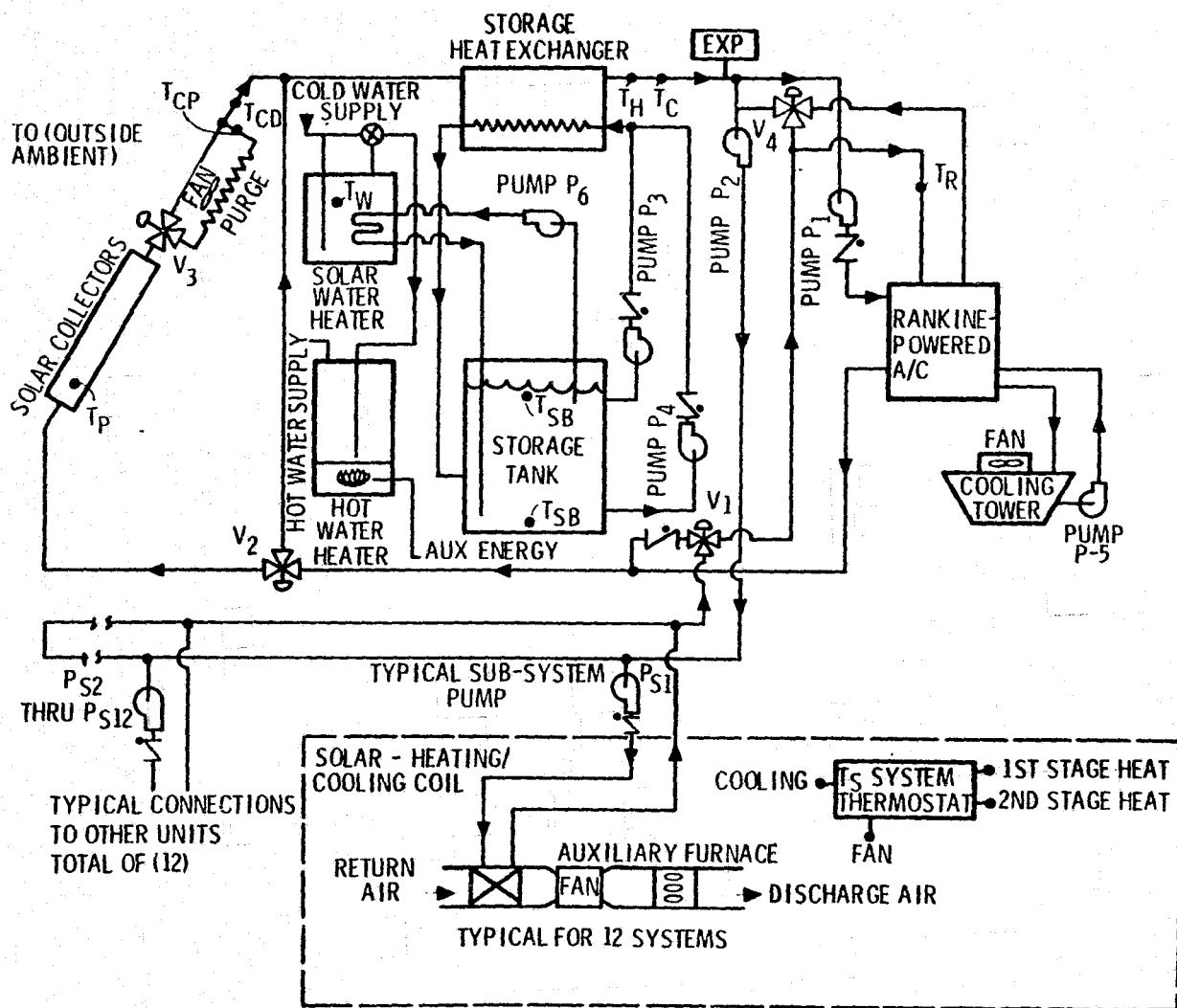


Figure 3-8. Multiple-Family Residential H/C System

The system provides 10 modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Rankine cooling from collectors
- Rankine cooling from storage
- Rankine cooling and storage simultaneously
- Electric motor auxiliary cooling
- Domestic hot-water preheater
- Purge excess energy

**3.3.1.1 Heating Subsystem Operation** -- When solar energy is available and heating is required, the collectors supply heat directly to the system. Pump  $P_1$  provides the heat transfer fluid movement in the primary loop, the sub-system pump circulates the secondary loop, and the furnace blower moves the space air over the heat coil.

During periods of high solar radiation and low heating demand, both heating and storage loop operate simultaneously, with the storage charge pump charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of thermal stratification. During periods of high solar radiation and low heating demand and with the storage tank fully charged, the system temperatures will increase and, as an overtemperature protective device, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the space through the heat exchanger. Pump  $P_2$  drives the primary loop and pump  $P_3$  extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, the heating load cannot be satisfied by the solar system and the second-stage thermostat activates the auxiliary furnace until a comfortable temperature is maintained.

**3.3.1.2 Cooling Subsystem Operation** -- When solar energy is available and cooling is required, the collectors supply heat directly to the Rankine boiler. Pump  $P_1$  provides the heat transfer fluid movement and, if necessary, can be sized differently than  $P_2$  to improve efficiency of the collector and Rankine engine operation. The Rankine engine drives a high COP water chiller, which, in turn, provides chilled water to the system. When the cooling demand is satisfied, pump  $P_2$  is shut down and the system reverts to the storage mode explained in the heating subsystem. During high solar radiation and low cooling demand, simultaneous cooling and storage is available. The purge coil unit will protect the system from excessive temperatures.

When solar energy is not available or insufficient to operate the Rankine engine at the design horsepower (20 horsepower at 190°F collector outlet), an electric motor is provided to operate the water chiller independently or to make up the difference for the required horsepower and that supplied by the Rankine engine. Storage is used to supply energy in the same manner as in the heating subsystem, except pump  $P_1$  is the prime mover in the glycol/water loop. The baseline design uses a constant-speed compressor input and therefore the electric motor is on-line at all times, supplying the balance of the required horsepower.

A variation of this concept is under study which uses a variable-speed R/C input to the compressor and thus variable cooling output. The auxiliary motor operates only when the water chiller controls indicate the cooling load is not satisfied. In this mode the water chiller operates as a constant-speed system delivering a maximum of 25 tons of cooling.

**3.3.1.3 Specific Subsystem/Component Design Criteria** -- The heating/storage circuit pump,  $P_2$ , and the cooling circuit pump,  $P_1$ , are located ahead of the loads so as to provide a common location for the expansion tank. This provides for air elimination and system pressure control during all modes of operation.

When the system is heating or cooling from storage, control valve  $V_2$  diverts flow around the solar collectors.

When the system is in the purge mode, control valve  $V_3$  diverts flow through the purge coil unit which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps  $P_3$  and  $P_4$  are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump  $P_3$  draws hotter water from the top of the storage tank for heating or cooling from storage modes of operation. Pump  $P_4$  draws cooler water from the bottom of the storage tank for storage charging of collected solar energy.

The solar domestic hot-water preheater, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The solar domestic hot-water preheater heats incoming cold water with solar-heated storage water through a tube bundle circulated by pump  $P_6$ . Storage water is drawn from the top of the storage tank and returned to the bottom to take advantage of thermal stratification.



A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

The location of each of the components within the system is determined by the following criteria:

- Optimize performance of entire system.
- Provide reliable operation of each component.
- Provide control functions for each mode of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the loads so as to provide a means for inlet temperature control to the loads. This also provides for a convenient method for simultaneous storage charging.

The heating/storage circuit and the cooling circuit are placed in parallel, downstream of the storage heat exchanger. Each circuit is controlled by its pump and protected from backflow by a check valve.

### 3.3.2 Sequence of Operation -- Multiple-Family Residence

3.3.2.1 General -- Space heating and cooling is controlled by individual space thermostats  $T_{S1}$  through  $T_{S12}$ , with each individual residence having independent heating control. First-stage heating is set to use solar energy, if available, while second-stage heating is set to use auxiliary energy if solar is not adequate. Each individual residence has independent cooling control, with the entire building system switching over from heating to cooling modes as determined by outside ambient temperature and a building cooling load analyzer. During the periods when the building system is in the cooling mode, each

individual subsystem has second-stage auxiliary heat available on demand. Cooling is set to use solar energy if available. If solar energy is not available or is not adequate, auxiliary energy will be added to satisfy the space building cooling load. The cooling subsystem supplies a maximum of 25 tons output on demand. The horsepower required to drive the compressor is supplied by the Rankine power cycle and the electric motor generator. The amount of Rankine power supplied is dependent on the solar energy available. The system control logic provides for: a) collecting solar energy when available, b) providing energy to the load on demand, c) storing any excess energy, and d) using solar energy before using auxiliary energy. A ladder diagram of the controls is shown in Figure 3-9.

**3.3.2.2 First-Stage Heating from Collectors** -- A demand for heat from an individual conditioned space activates the control system to first look for the energy in the collector and, through proper pump and valve selection, transfer the energy through the heating coil to the conditioned air. This mode involves valves  $V_1$ ,  $V_2$ , pump  $P_2$ , and the individual subsystem furnace fan and circulating pump.

Whenever plate temperature  $T_p$  is greater than  $100^\circ\text{F}$  (adjustable) and the outside ambient temperature is below  $60^\circ\text{F}$ , on a call for heating from any space thermostat through solar heating relay  $R_1$ , the main heating loop pump,  $P_2$ , is activated. A shutdown time delay relay prevents pump  $P_2$  from short-cycling. Valves  $V_2$  and  $V_1$  are positioned to direct flow to the collectors. The appropriate subsystem pump is activated and when the solar-heated fluid entering the heating coil reaches  $100^\circ\text{F}$  (adjustable), the subsystem furnace fan is activated. Each subsystem pump and piping circuit is provided with a check valve to prevent backflow.

**3.3.2.3 Storage Charging While Heating** -- If the solar radiation is more than adequate to provide the heating load, the excess energy can be added to storage by activating the storage charging loop. This mode involves the same components as the heating from collectors plus pump  $P_4$ .

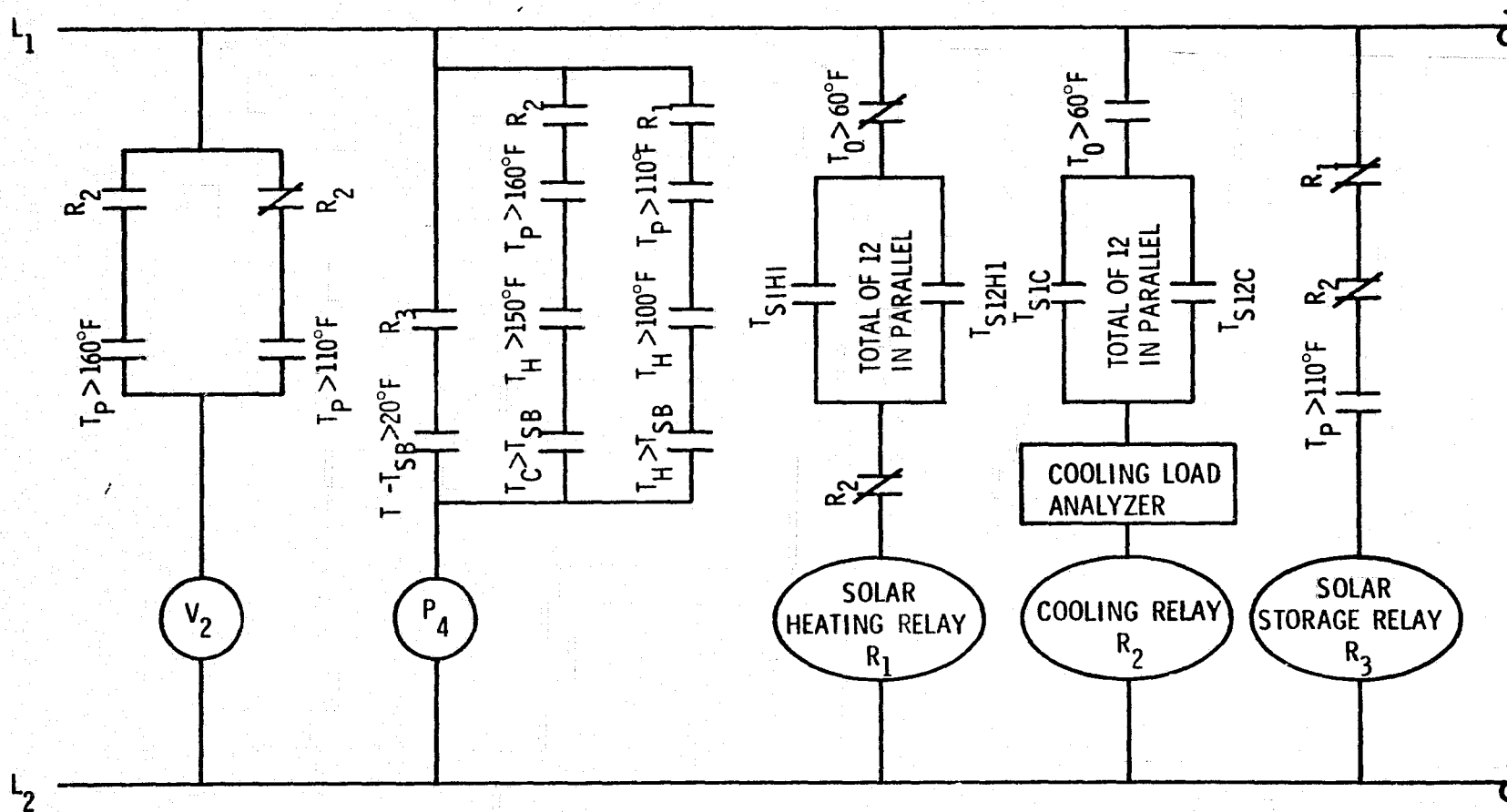


Figure 3-9. Multiple-Family Residence Heating/Cooling System Control Schematic

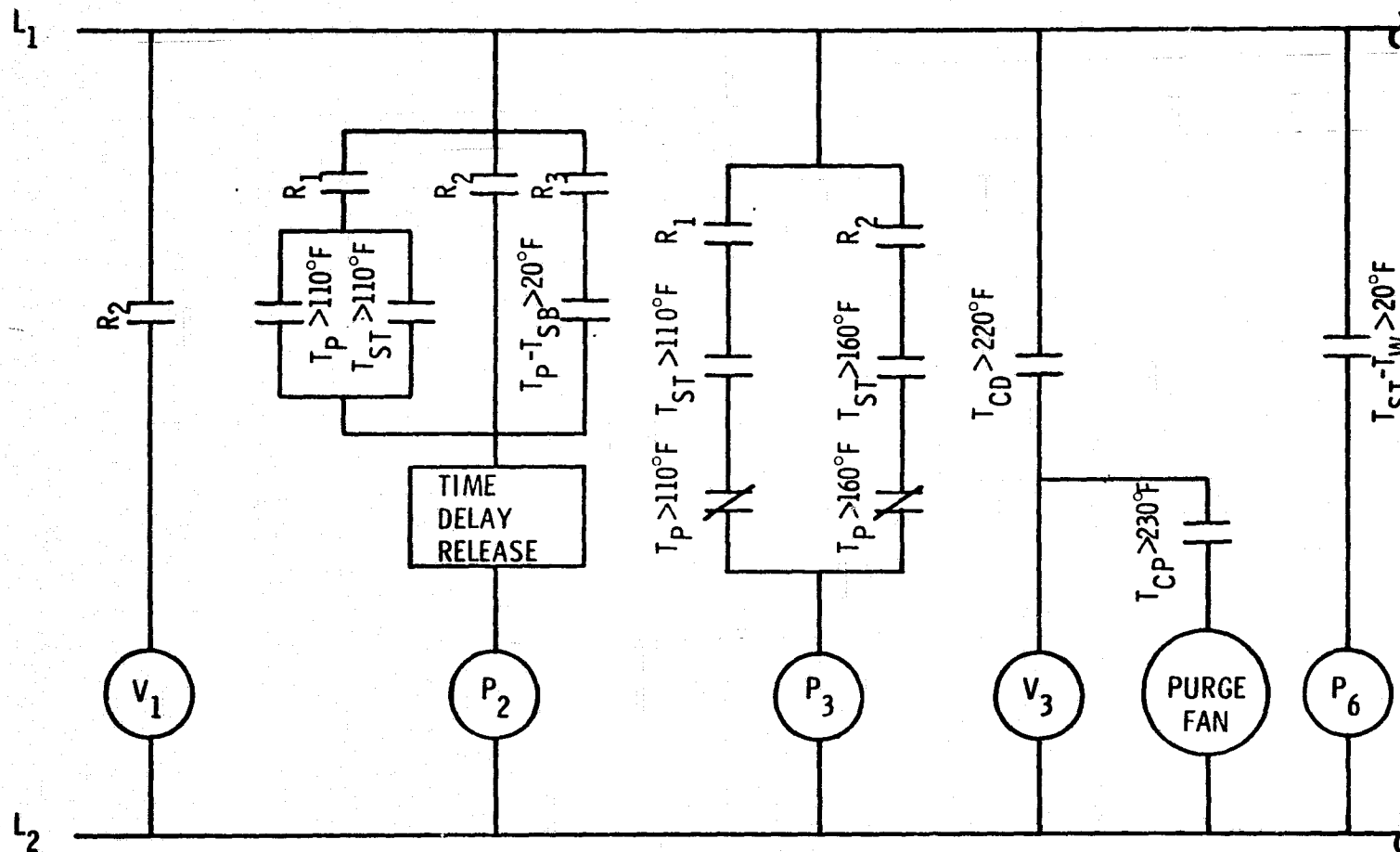
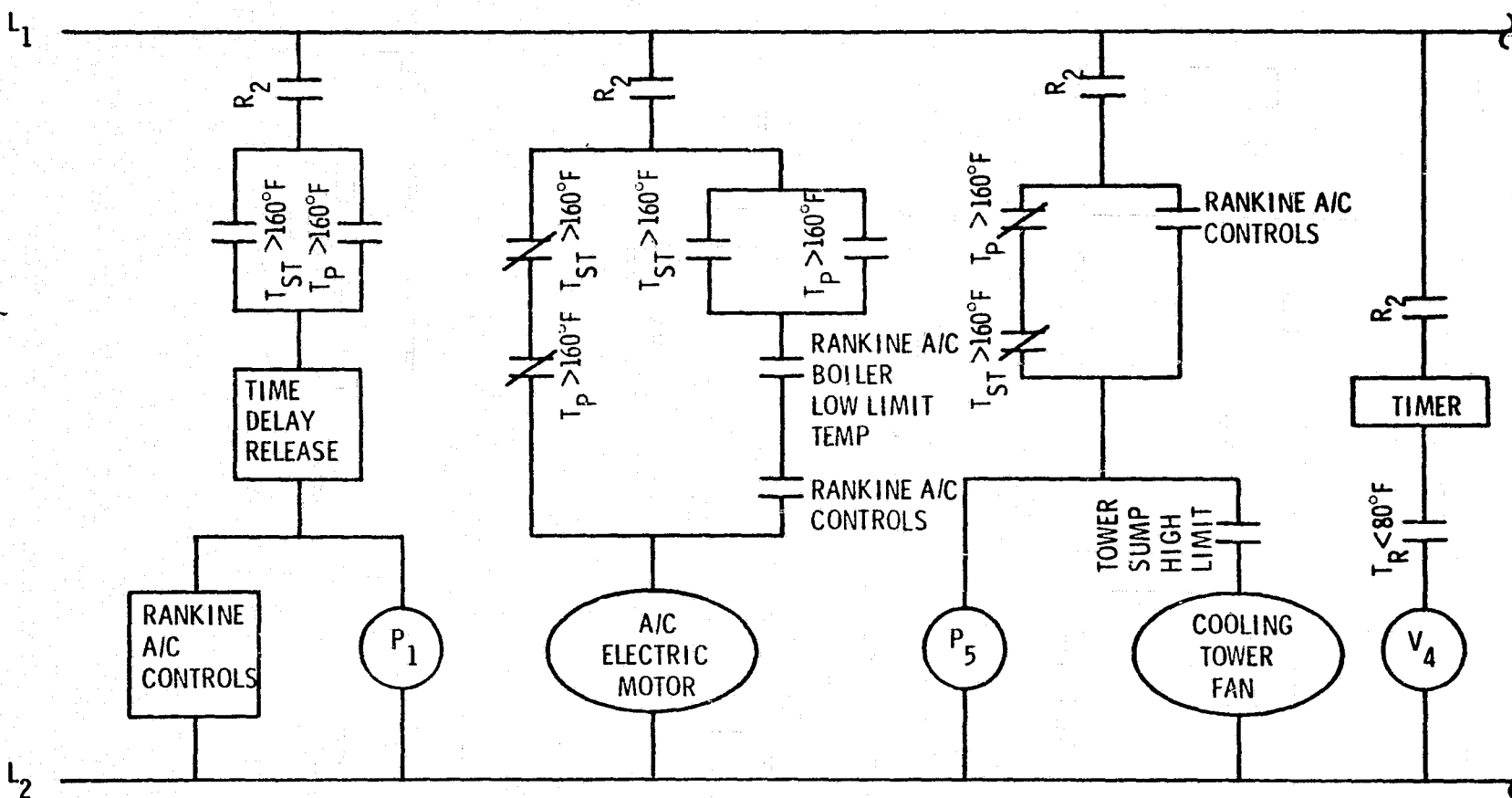
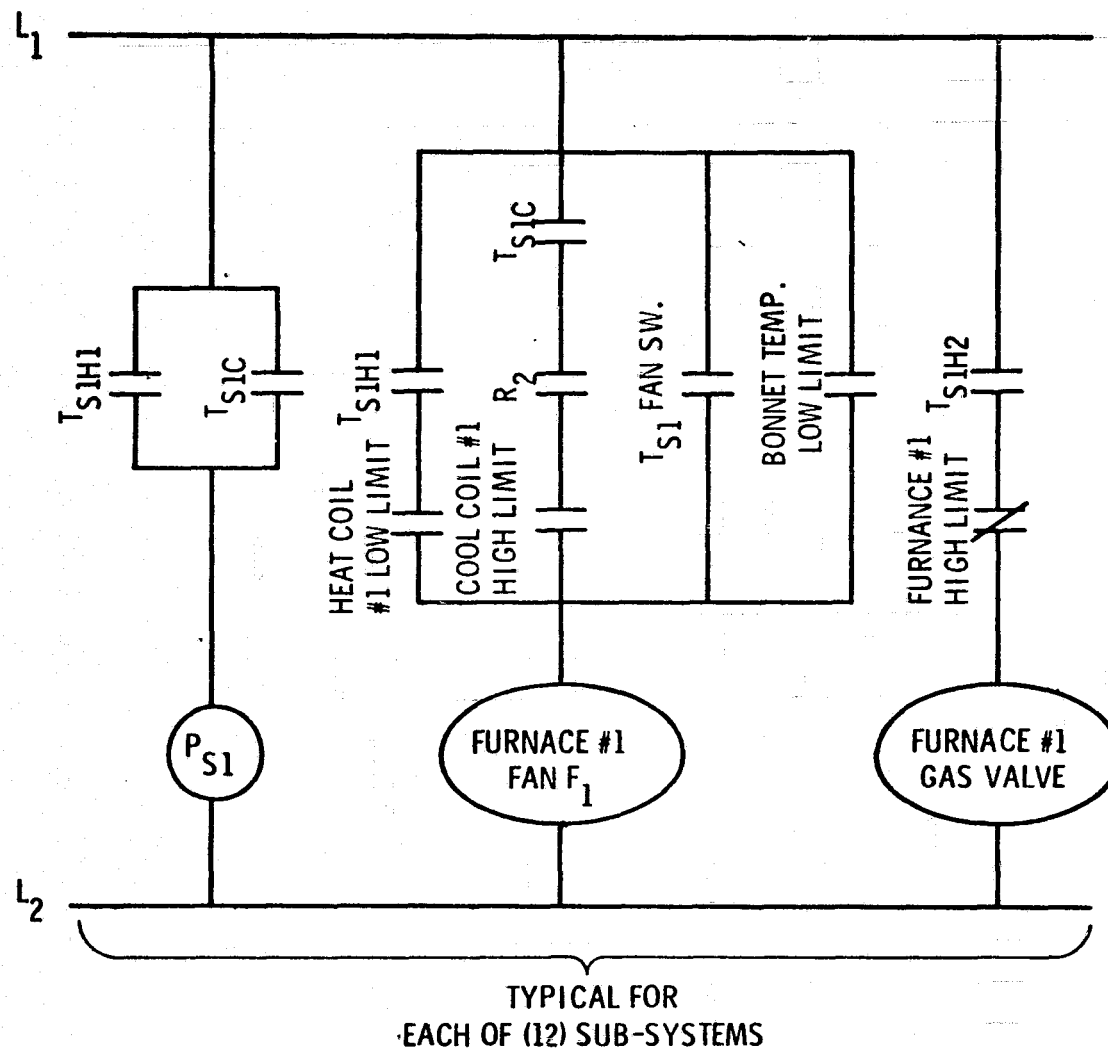


Figure 3-9. Multiple-Family Residence Heating/Cooling System Control Schematic (Continued)



3-39

Figure 3-9. Multiple-Family Residence Heating/Cooling System Control Schematic (Continued)



**Figure 3-9. Multiple-Family Residence Heating/Cooling System Control Schematic (Concluded)**

$P_4$  logic monitors excess energy ( $T_H > 150^\circ\text{F}$ ) and also determines that the bottom of the storage tank is at a lower temperature ( $T_H - T_{SB} > 20^\circ\text{F}$ , adjustable) before activating the parallel storage mode.

**3.3.2.4 Charging Storage (Heating Season)** -- With solar energy available and no demand for heating, the control system transfers the collector energy to the storage tank. This mode involves  $V_1$ ,  $V_2$ ,  $P_2$ , and  $P_4$ .

Storage charging is accomplished whenever there is a call for storage charging through solar storage relay  $R_3$  and  $T_P$  is greater than  $100^\circ\text{F}$  (adjustable) and  $T_P$  is greater than  $T_{SB}$  by  $20^\circ\text{F}$  (adjustable). Pumps  $P_2$  and  $P_4$  are activated and a shutdown time-delay relay prevents short-cycling of the pumps. Valve  $V_2$  is positioned to divert flow to the collectors as controlled by plate temperature  $T_P$ . Valve  $V_1$  is positioned to divert flow to the collectors.

**3.3.2.5 Heating From Storage** -- As the availability of direct solar energy decreases and the heating demand persists, the central system first looks to storage to satisfy the demand. This mode involves  $V_1$ ,  $V_2$ ,  $P_2$ ,  $P_3$ , and the individual furnace fan and circulating pump.

Whenever  $T_P$  is less than  $100^\circ\text{F}$  (adjustable),  $T_{ST}$  is greater than  $100^\circ\text{F}$  (adjustable) and there is a call for space heat, valve  $V_2$  is positioned to divert flow around the collectors. Pump  $P_3$  is activated to discharge heat from the storage tank for space heating. Pump  $P_2$  is activated and valve  $V_1$  is positioned to direct flow to the heating coil. All other controls and functions are similar to heating from collectors.

**3.3.2.6 Second-Stage (Auxiliary) Heating** -- Second-stage heating from any individual space thermostats  $T_{S1}$  through  $T_{S12}$  will command the auxiliary heat source to supply heat to that space. First-stage heating continues during second-stage heating. The blower motors for each individual space are controlled by their existing furnace controls.

**3.3.2.7 Direct Cooling From Collectors** -- A demand for cooling from the conditioned space activates the control system to first look for energy in the collector and, through proper pump and valve selection, transfer the energy through the Rankine boiler. The Rankine engine supplies shaft power to the electric motor's shaft to the level that the available solar energy allows. This mode involves  $V_2$ ,  $P_1$ , furnace fan, cooling tower fan, and pump  $P_5$ .

Whenever plate temperature,  $T_P$ , is greater than  $160^\circ\text{F}$  (adjustable) and there is a call for cooling from any one of several spaces through the Cooling Load Analyzer and the solar cooling relay  $R_2$ , then pump  $P_1$  is activated and valve  $V_2$  positioned to direct flow to the solar collectors. Pump  $P_1$  discharges solar-heating fluid to the Rankine-cycle engine which drives the Rankine air conditioning water chiller providing chilled water to main loop pump  $P_2$ . Valve  $V_1$  is positioned to direct main loop chilled water flow back to the Rankine air conditioner.

A time-delay relay prevents the pumps and the Rankine air conditioner from short-cycling. If the Rankine engine is not able to provide sufficient energy to satisfy the cooling load, auxiliary energy is added by an electric motor. A time-delay relay in the Rankine air conditioner controls prevents the auxiliary electric motor from operating until the Rankine engine has come up to a steady operating temperature. Condenser water pump  $P_5$  is energized and the cooling tower fan is energized to run as controlled by sump water temperature. The appropriate individual subsystem pump is activated and, when the chilled water entering the cooling coil drops to  $55^\circ\text{F}$  (adjustable), the subsystem furnace fan is activated. On start-up of the Rankine air conditioner, valve  $V_4$  modulates flow through the water chiller to maintain a maximum chilled water inlet temperature of  $80^\circ\text{F}$ . A timer controls valve  $V_4$  to provide full flow through the chiller after the Rankine engine is up to a steady operating temperature. The cooling capacity of the refrigeration system water chiller is controlled by conventional compressor capacity adjustment (i. e., on-off compressor cycling and/or cylinder unloading). If any



zone has a call for heat at a time when the Cooling Load Analyzer has the system in the cooling mode, second-stage auxiliary heating is available as described above.

**3.3.2.8 Storage Charging While Cooling** -- If the solar radiation is more than adequate to provide the cooling load, the excess energy can be added to storage by activating the storage charging loop. This mode involves the same components as the cooling from collectors plus pump  $P_4$ .

$P_4$  logic monitors excess energy ( $T_C > 200^\circ\text{F}$ ) and also determines that the bottom of the storage tank is at a lower temperature ( $T_C - T_{SB} > 20^\circ\text{F}$ , adjustable) before activating the parallel storage mode.

**3.3.2.9 Charging Storage (Cooling Season)** -- With solar energy available and no demand for cooling, the control system transfers the collector energy to the storage tank. This mode involves  $V_2$ ,  $P_2$ , and  $P_4$ .

**3.3.2.10 Cooling From Storage** -- As the availability of direct solar energy decreases and the cooling demand persists, the control first looks to storage to satisfy demand. This mode involves  $V_2$ ,  $P_1$ ,  $P_3$ , individual subsystem pumps, furnace fan, cooling tower fan, and pump  $P_5$ . Whenever  $T_P$  is less than  $160^\circ\text{F}$  (adjustable),  $T_{ST}$  is greater than  $160^\circ\text{F}$  (adjustable) and there is a call for space cooling, valve  $V_2$  is positioned to divert flow around the solar collectors. Pump  $P_3$  is activated to discharge heat from the storage tank to run the Rankine engine. All other functions and controls are similar to those for direct cooling.

**3.3.2.11 Auxiliary Cooling** -- Whenever there is a call for space cooling from the Cooling Load Analyzer, the Rankine-cycle system is activated. A time delay in the Rankine system controls will prevent the Rankine system electric motor from operating until the Rankine system boiler has reached a minimum operating temperature. The electric motor will supply the necessary

energy for auxiliary cooling if solar-heated water is not available. It can also supply the additional shaft power as needed during Rankine-cycle operation to assure a maximum 25-ton output.

3.3.2.12 Heat Rejector Control -- Whenever the collector discharge temperature exceeds 220°F (adjustable) as sensed by  $T_{CD}$ , valve  $V_3$  is positioned to direct fluid through the purge coil. If the temperature of the collector fluid leaving the purge coil,  $T_{CP}$ , rises above 230°F (adjustable), then the purge coil fan is activated. These controls prevent the collector fluid temperature from exceeding 230°F, thereby protecting the fluid itself and the various system components from excessive temperatures.

3.3.2.13 Domestic Hot-Water Heating -- Whenever domestic hot-water temperature,  $T_W$ , is less than storage tank temperature,  $T_{ST}$ , by 10°F (adjustable), pump  $P_6$  is activated to heat the domestic water from storage. A thermostatic mixing valve is used to regulate the hot-water supply temperature to 140°F (adjustable), leaving the solar water heater. The solar-heated domestic hot water then goes to the building hot-water heater where auxiliary heat is added as necessary to provide the required domestic hot-water temperature of 140°F (adjustable).

### 3.4 COMMERCIAL BUILDING HEATING AND COOLING SYSTEM DESCRIPTION

#### 3.4.1 System Description

The proposed system for a commercial heating and cooling system is a single-loop, solar-powered, two-pipe hydronic heating and cooling system with a separate water storage and domestic hot-water heating loop. Being a two-pipe design, the central system is either in the heating mode or the cooling mode as determined by a Cooling Load Analyzer. The central heating is provided by either direct or stored solar energy and the central

cooling is provided by three solar-powered Rankine engine/auxiliary electric motor-driven water chillers. The system is composed of the following major components:

- Liquid-cooled flat-plate collectors
- A water storage tank
- A solar-fired domestic water preheater with storage water tube bundle and circulating pump
- A gas-fired hot-water heater
- Water chillers driven by a Rankine engine and/or an auxiliary electric motor and controlled by a Cooling Load Sequencer
- A tube-and-shell heat exchanger, four central system pumps and associated piping and valving
- A condenser water pump and cooling tower
- A control system
- An air-cooled heat purge unit with fan
- Four individual subsystems consisting of a circulating pump and a gas-fired air-handling unit with a water heating/cooling coil

The arrangements within the system is as shown in Figure 3-10. It consists of a glycol collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger.

The glycol collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump  $P_2$ , the Rankine boilers and pump  $P_1$ , the purge coil, and three control valves as required for the different modes of operation.

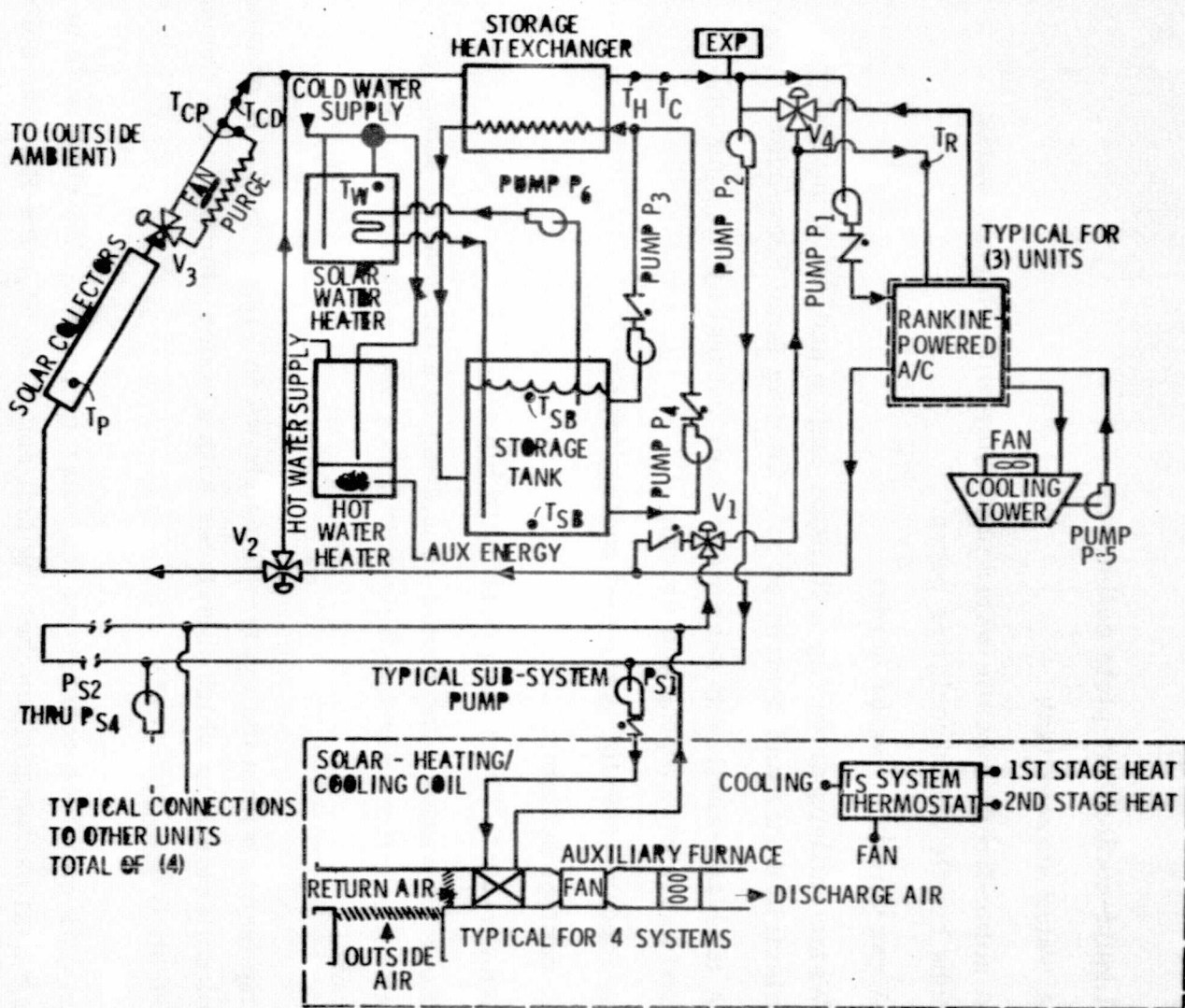


Figure 3-10. Commercial Building H/C System

The water storage loop consists of the storage tank, storage pumps  $P_3$  and  $P_4$ , and the tube side of the storage heat exchanger, and the solar DHW pre-heater pump,  $P_6$ .

The system provides 10 modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Rankine cooling from collectors
- Rankine cooling from storage
- Rankine cooling and storage simultaneously
- Electric motor auxiliary cooling
- Domestic hot-water preheater
- Purge excess energy

3.4.1.1 Heating Subsystem Operation -- When solar energy is available and heating is required, the collectors supply heat directly to the system. Pump  $P_1$  provides the heat-transfer fluid movement in the main loop, the subsystem pump circulates the subsystem loop, and the air-handling unit moves the building air over the heat coil.

During periods of high solar radiation and low heating demand, both heating and storage loop operate simultaneously, with the storage charge pump, charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of thermal stratification. During periods of high solar radiation and low heating demand with full storage charge, the system

temperatures will increase and as an overtemperature protective device, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the furnace through the heat exchanger. Pump  $P_2$  drives the main loop and pump  $P_3$  extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, the heating load cannot be satisfied by the solar system and the second-stage thermostat activates the auxiliary gas burner until a comfortable temperature is maintained.

**3.4.1.2 Cooling Subsystem Operation** -- When solar energy is available and cooling is required, the collectors supply heat directly to the Rankine boiler. Pump  $P_1$  provides the heat-transfer fluid movement and, if necessary, can be sized differently than  $P_2$  to improve efficiency of the collector and Rankine-engine operation. Each Rankine engine drives a high COP water chiller, which, in turn, provides chilled water to the system. When the cooling demand is satisfied, pump  $P_2$  is shut down and the system reverts to the storage mode explained in the heating subsystem. During high solar radiation and low cooling demand, simultaneous cooling and storage is available. The purge coil unit will protect the system from excessive temperatures.

When solar energy is not available or insufficient to operate the Rankine engine at the design horsepower (20 horsepower at 190°F collector outlet), an electric motor is provided to operate the water chiller independently or to make up the difference for the required horsepower and that supplied by the Rankine engine. Storage is used to supply energy in the same manner as in the heating subsystem, except pump  $P_1$  is the prime mover in the glycol loop. The baseline design uses a constant-speed compressor input and therefore the electric motor is on-line at all times, supplying the balance of the required horsepower.

A variation of this concept is under study which uses a variable-speed R/C input to the compressor and thus variable cooling output. The auxiliary motor operates only when the water chiller controls indicate the cooling load is not satisfied. In this mode each water chiller operates as a constant-speed system delivering a maximum of 25 tons of cooling.

**3.4.1.3 Specific Subsystem/Component Design Criteria** -- The heating/storage circuit pump,  $P_2$ , and the cooling circuit pump,  $P_1$ , are located ahead of the loads so as to provide a common location for the expansion tank. This provides for air elimination and system pressure control during all modes of operation.

When the system is heating or cooling from storage, control valve  $V_2$  diverts flow around the solar collectors.

When the system is in the purge mode, control valve  $V_3$  diverts flow through the purge coil unit which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps,  $P_3$  and  $P_4$ , are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump  $P_3$  draws hotter water from the top of the storage tank for heating or cooling from storage modes of operation. Pump  $P_4$  draws cooler water from the bottom of the storage tank for storage charging of collected solar energy.

The solar domestic hot-water preheater, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The solar domestic hot-water preheater heats incoming cold water with solar heated storage water through a tube bundle circulated by pump  $P_6$ . Storage water is drawn from the top of the storage tank and returned to the bottom to take advantage of thermal stratification.

A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

The location of each of the components within the system is determined by the following criteria:

- Optimize performance of entire system.
- Provide reliable operation of each component.
- Provide control functions for each mode of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the loads so as to provide a means for inlet temperature control to the loads. This also provides for a convenient method for simultaneous storage charging.

The heating/storage circuit and the cooling circuit are placed in parallel, downstream of the storage heat exchanger. Each circuit is controlled by its pump and protected from backflow by a check valve.

### 3.4.2 Sequence of Operation - Commercial Building

3.4.2.1 General -- Space heating and cooling is controlled by individual space thermostats  $T_{S1}$  through  $T_{S4}$ , with each individual zone having independent heating control. First-stage heating is set to use solar energy, if available,



while second-state heating is set to use auxiliary energy if solar is not adequate. Each individual zone has independent cooling control, with the entire building system switching over from heating to cooling modes as determined by outside ambient temperature and a building Cooling Load Analyzer. During the periods when the building system is in the cooling mode, each individual subsystem has second-stage auxiliary heat available on demand. Cooling is set to use solar energy if available. If Solar energy is not available or is not adequate, auxiliary energy will be added to satisfy the space building cooling load. The cooling subsystem consists of three Rankine engine / electric motor-driven 25-ton water chillers which provide a total cooling capacity of 75 tons. The amount of Rankine power supplied to the water chillers is dependent on the solar energy available. The system control logic provides for: a) collecting solar energy when available, b) providing energy to the load on demand, c) storing any excess energy, and d) using solar energy before using auxiliary energy. A ladder diagram of the controls is shown in Figure 3-11.

3.4.2.2 First-Stage Heating from Collectors -- A demand for heat from an individual conditioned zone activates the control system to first look for the energy in the collector and, through proper pump and valve selection, transfer the energy through the heating coil to the conditioned air. This mode involves valve  $V_1$ ,  $V_2$ , pump  $P_2$ , and the individual subsystem furnace fan and circulating pump.

Whenever plate temperature,  $T_p$ , is greater than 100°F (adjustable) and the outside ambient temperature is below 60°F, on a call for heating from any zone thermostat through solar heating relay  $R_1$ , the main heating loop pump,  $P_2$ , is activated. A shutdown time-delay relay prevents pump  $P_2$  from short-cycling. Valves  $V_2$  and  $V_1$  are positioned to direct flow to the collectors. The appropriate subsystem pump is activated and when the solar-heated fluid entering the heating coil reaches 100°F (adjustable), the subsystem furnace fan is activated. Each subsystem pump and piping circuit is provided with a check valve to prevent backflow.

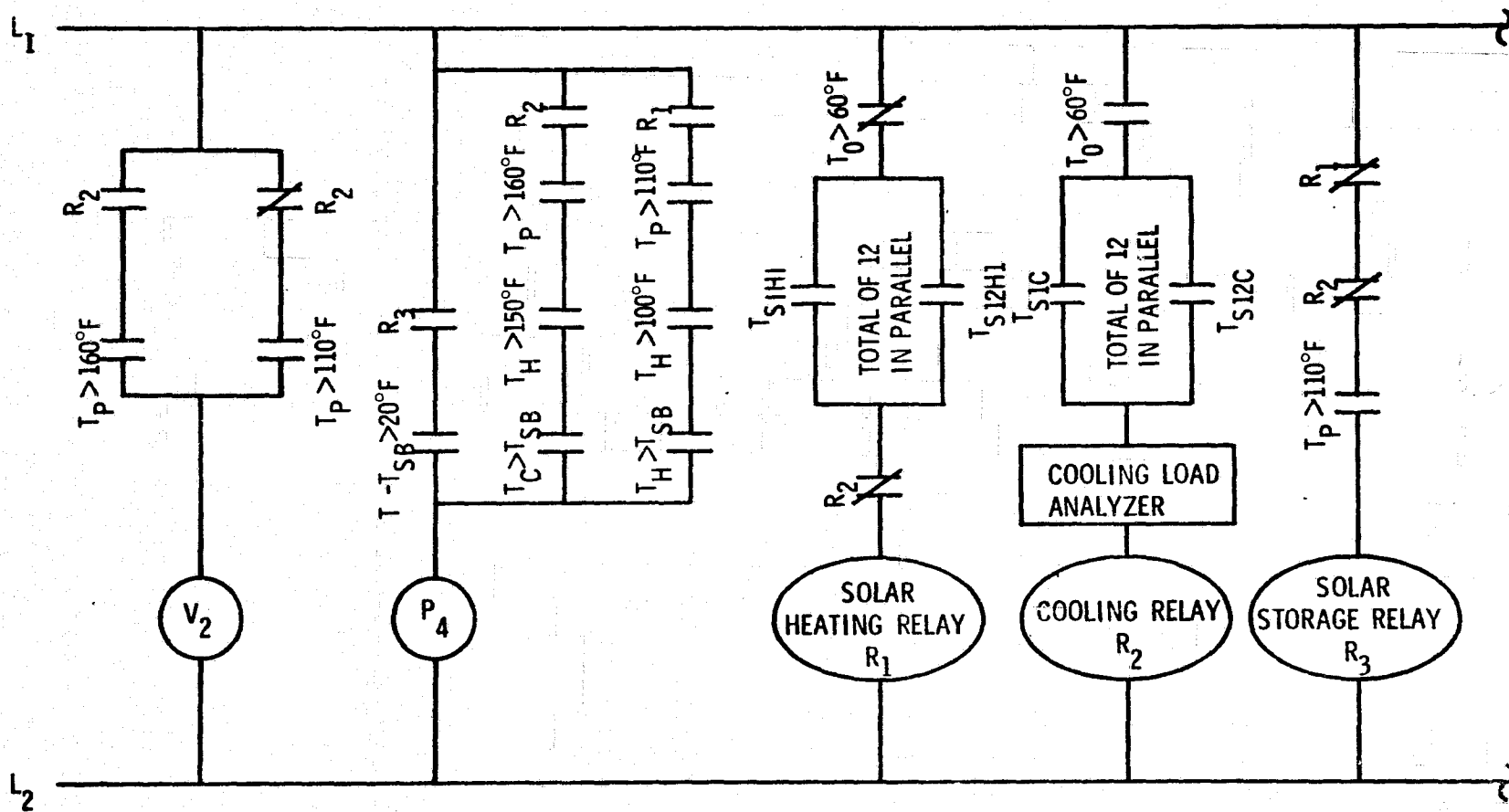


Figure 3-11. Commercial Building Heating/Cooling System Control Schematic

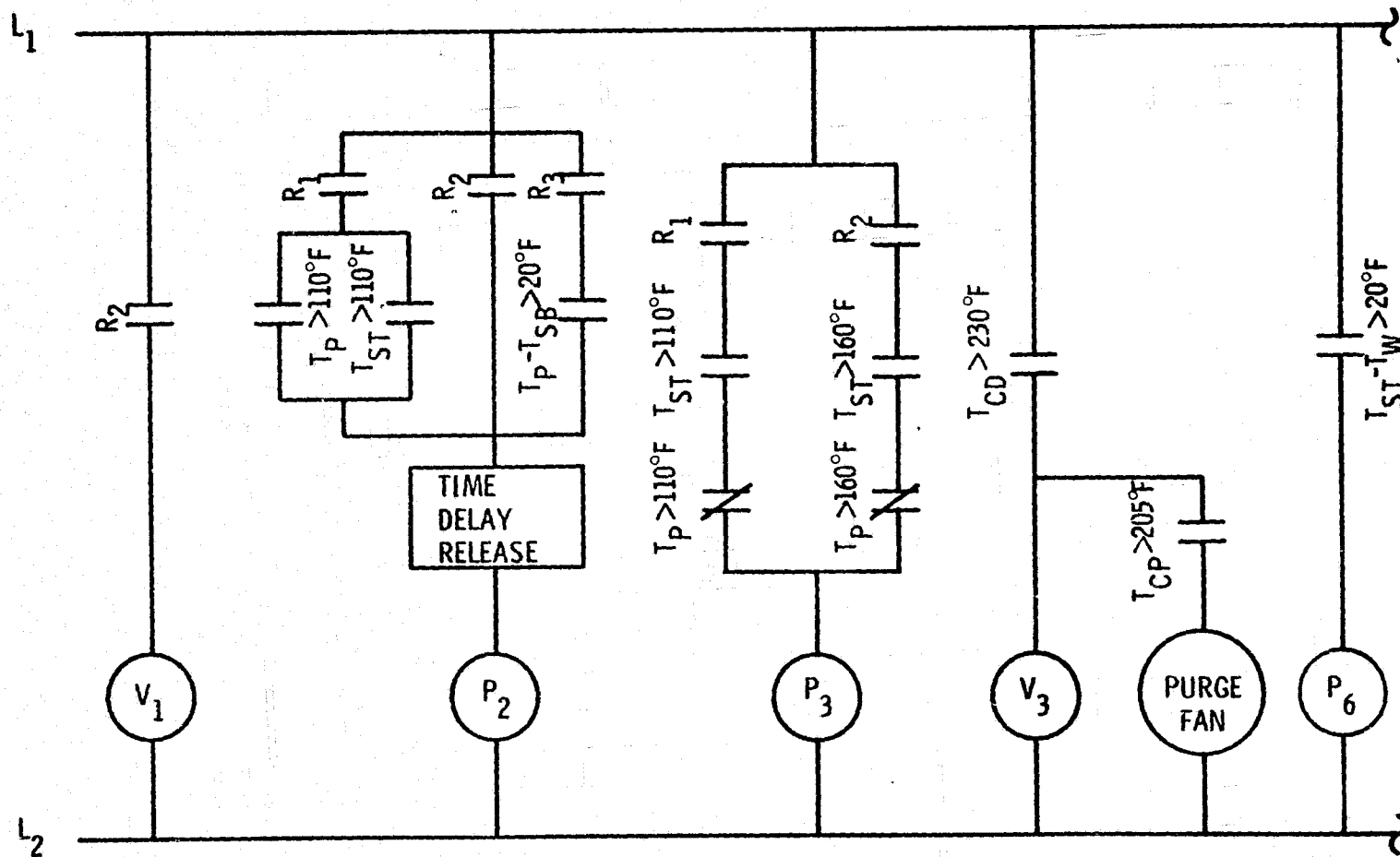


Figure 3-11. Commercial Building Heating/Cooling System Control Schematic (Continued)



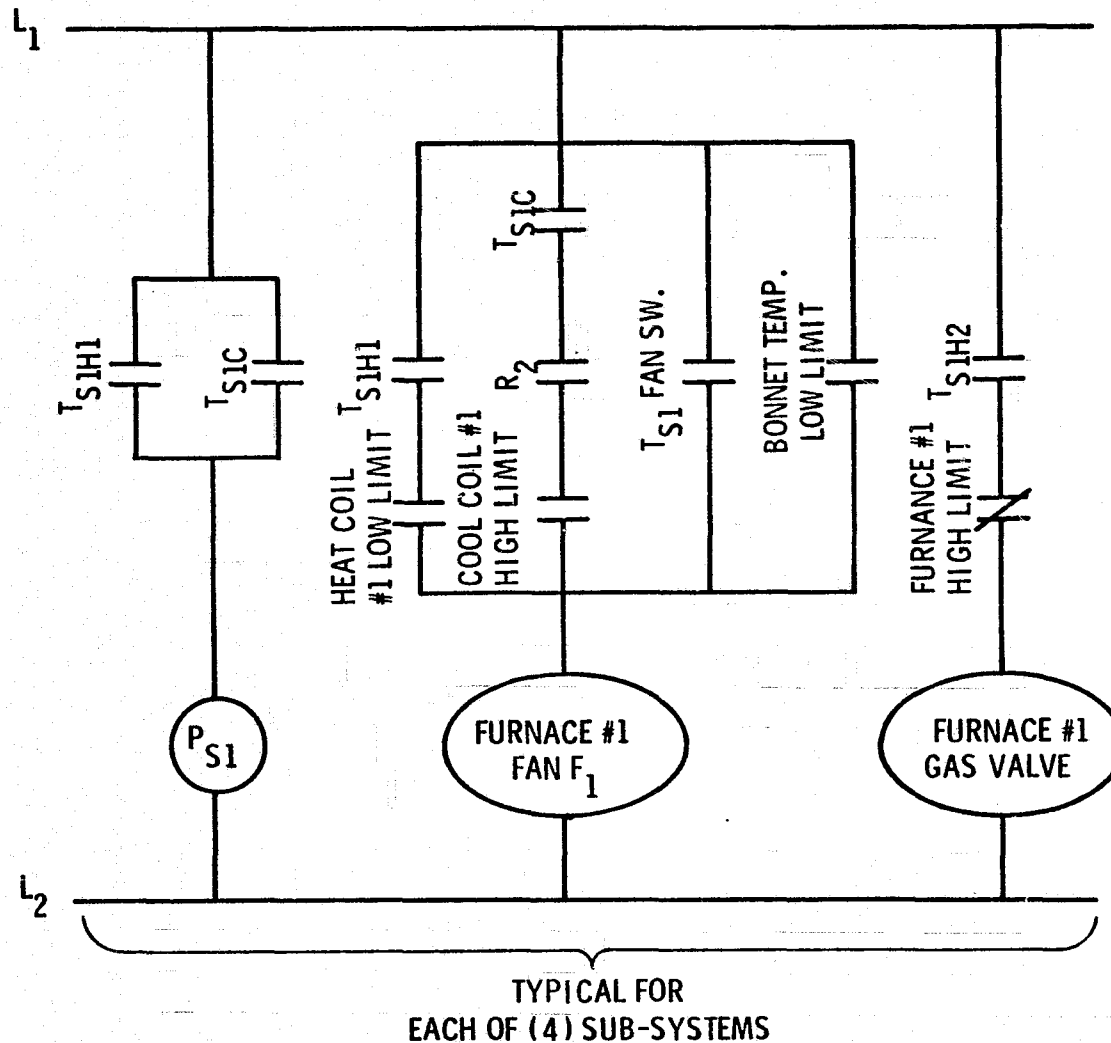


Figure 3-11. Commercial Building Heating/Cooling System Control Schematic (Concluded)

**3.4.2.3 Storage Charging While Heating** -- If the solar radiation is more than adequate to provide the heating load, the excess energy can be added to storage by activating the storage charging loop. This mode involves the same components as the heating from collectors plus pump  $P_4$ .

$P_4$  logic monitors excess energy ( $T_H > 150^\circ\text{F}$ ) and also determines that the bottom of the storage tank is at a lower temperature ( $T_H - T_{SB} > 20^\circ\text{F}$ , adjustable) before activating the parallel storage mode.

**3.4.2.4 Charging Storage (Heating Season)** -- With solar energy available and no demand for heating, the control system transfers the collector energy to the storage tank. This mode involves  $V_1$ ,  $V_2$ ,  $P_2$ , and  $P_4$ .

Storage charging is accomplished whenever there is a call for storage changing through solar storage relay  $R_3$  and  $T_P$  is greater than  $100^\circ\text{F}$  (adjustable) and  $T_P$  is greater than  $T_{SB}$  by  $20^\circ\text{F}$  (adjustable). Pumps  $P_2$  and  $P_4$  are activated and a shutdown time-delay relay prevents short-cycling of the pumps. Valve  $V_2$  is positioned to divert flow to the collectors as controlled by plate temperature  $T_P$ . Valve  $V_1$  is positioned to divert flow to the collectors.

**3.4.2.6 Heating from Storage** -- As the availability of direct solar energy decreases and the heating demand persists, the central system first looks to storage to satisfy the demand. This mode involves  $V_1$ ,  $V_2$ ,  $P_2$ ,  $P_3$ , and the individual furnace fan and circulating pump.

Whenever  $T_P$  is less than  $100^\circ\text{F}$  (adjustable),  $T_{ST}$  is greater than  $100^\circ\text{F}$  (adjustable) and there is a call for space heat, valve  $V_2$  is positioned to divert flow around the collectors. Pump  $P_3$  is activated to discharge heat from the storage tank for space heating. Pump  $P_2$  is activated and valve  $V_1$  is positioned to direct flow to the heating coil. Other controls are similar to those for heating from collectors.

3.4.2.7 Second-Stage (Auxiliary) Heating -- Second-stage heating from any individual space thermostat  $T_{S1}$  through  $T_{S4}$  will command the auxiliary heat source to supply heat to that space. First-stage heating continues during second-stage heating. The blower motors for each individual zone are controlled by their existing furnace controls.

3.4.2.8 Cooling with Outside Air -- Outside air will be used for cooling before mechanical cooling is activated. Where practical, an enthalpy controller sensing outside and return-air enthalpy will be used for controlling outside air cooling through the return-air and outside-air dampers. If there is a call for cooling and the outside air enthalpy is less than the return-air enthalpy, then outside air will be used up to a point of 100 percent outside air. On a further rise in space temperature ( $2^{\circ}\text{F}$ ), mechanical cooling will be activated.

3.4.2.9 Direct Cooling from Collectors -- A demand for cooling from the conditioned space activates the control system to first look for energy in the collector and, through proper pump and valve selection, transfer the energy through the Rankine boiler. The Rankine engine supplies shaft power to the electric motor's shaft to the level that the available solar energy allows. This mode involves  $V_2$ ,  $P_1$ , furnace fan, cooling tower fan, and pump  $P_5$ .

Whenever plate temperature  $T_P$  is greater than  $160^{\circ}\text{F}$  (adjustable) and there is a call for cooling from any one or several zones through the Cooling Load Analyzer and the solar cooling relay  $R_4$ , then pump  $P_1$  is activated and valve  $V_2$  is positioned to direct flow to the solar collectors. Pump  $P_1$  discharges solar heating fluid to the Rankine-cycle engines which drive the Rankine Air conditioner water chillers providing chilled water to primary loop pump  $P_2$ . Valve  $V_1$  is positioned to direct main loop chilled water flow back to the water chillers.

The Cooling Load Sequencer will bring on one or several of the Rankine-engine-driven water chillers in response to the system cooling load.

A time-delay relay prevents the pumps and the Rankine-driven water chillers from short-cycling. If the Rankine engines are not able to provide sufficient energy to satisfy the cooling load, auxiliary energy is added by electric motors on each water chiller. A time-delay relay in the Rankine air conditioner controls prevents the auxiliary electric motor from operating until the Rankine engine has come up to a steady operating temperature. Condenser water pump  $P_5$  is energized and the cooling tower fan is energized to run as controlled by sump water temperature. The appropriate individual subsystem pump is activated and, when the chilled water entering the cooling coil drops to  $55^{\circ}\text{F}$  (adjustable), the subsystem furnace fan is activated. On start-up of the Rankine air conditioner, valve  $V_4$  modulates flow through the water chillers to maintain a maximum chilled water inlet temperature of  $80^{\circ}\text{F}$ . A timer controls valve  $V_4$  to provide full flow through the chillers after the Rankine engines are up to a steady operating temperature. The cooling capacity of the refrigeration system water chillers is controlled by conventional compressor capacity adjustment (i.e., on-off compressor cycling and/or cylinder unloading). If any zone has a call for heat at a time when the cooling load analyzer has the system in the cooling mode, second-stage auxiliary heating is available as described above.

**3.4.2.10 Storage Charging While Cooling** -- If the solar radiation is more than adequate to provide the cooling load, the excess energy can be added to storage by activating the storage charging loop. This mode involves the same components as the cooling from collectors plus pump  $P_4$ .

$P_4$  logic monitors excess energy ( $T_C > 200^{\circ}\text{F}$ ) and also determines that the bottom of the storage tank is at a lower temperature ( $T_C - T_{SB} > 20^{\circ}\text{F}$ , adjustable), before activating the parallel storage mode.



**3.4.2.11 Charging Storage (Cooling Season)** -- With solar energy available and no demand for cooling, the control system transfers the collector energy to the storage tank. This mode involves  $V_2$ ,  $P_2$ , and  $P_4$ .

This logic is identical to charging storage during the heating season logic.

**3.4.2.12 Cooling From Storage** -- As the availability of direct solar energy decreases and the cooling demand persists, the control first looks to storage to satisfy demand. This mode involves  $V_2$ ,  $P_1$ ,  $P_3$ , individual subsystem pumps, furnace fan, cooling tower fan, and pump  $P_5$ . Whenever  $T_P$  is less than 160°F (adjustable)  $T_{ST}$  is greater than 160°F (adjustable), and there is a call for space cooling, valve  $V_2$  is positioned to divert flow around the solar collectors. Pump  $P_3$  is activated to discharge heat from the storage tank to run the Rankine engines. All other functions and controls are similar to those for direct cooling.

**3.4.2.13 Auxiliary Cooling** -- Whenever there is a call for cooling from the Cooling Load Analyzer, the Cooling Load Sequencer will bring on one or several of the water chillers in response to the system cooling load. If there is direct or stored solar energy available, the Rankine engines will be activated to drive the water chiller compressors. A time-delay relay in each Rankine-engine control system will prevent the auxiliary electric motor from operating until the Rankine system boiler has reached minimum operating temperature. The electric motor will supply the necessary energy for auxiliary cooling if solar-heated water is not available. Each electric motor can also supply the additional shaft power if needed during Rankine-cycle operation, to assure a maximum of 25 tons cooling output for each water chiller and a maximum system cooling output of 75 tons.

**3.4.2.14 Heat Rejector Control** -- Whenever the collector discharge temperature exceeds 200°F (adjustable) as sensed by  $T_{CD}$ , valve  $V_3$  is positioned to direct fluid through the purge coil. If the temperature of the collector fluid

leaving the purge coil,  $T_{CP}$ , rises above 230°F (adjustable), then the purge coil fan is activated. These controls prevent the collector fluid temperature from exceeding 230°F, thereby protecting the fluid itself and the various system components from excessive temperatures.

3.4.2.15 Domestic Hot-Water Heating -- Whenever domestic hot-water temperature  $T_W$  is less than storage tank temperature  $T_{ST}$  by 10°F (adjustable), pump  $P_6$  is activated to heat the domestic water from storage. A thermostatic mixing valve is used to regulate the hot-water supply temperature to 140°F (adjustable), leaving the solar water heater. The solar-heated domestic hot water then goes to the building hot-water heater where auxiliary heat is added as necessary to provide the required domestic hot-water temperature of 140°F (adjustable).

## SECTION 4 SYSTEM TRADE STUDIES

### 4.1 METHODOLOGY

A methodology for a comparison of solar-assisted heating and cooling system concepts has been established. The methodology or approach allows a complete examination of the solar system and subsystem candidates. Studies include variation in the key parameters that influence solar heating and cooling systems cost and performance. A systems simulation program was used to analyze the solar systems cost and/or performance as a result of variations in the key parameters.

The system tradeoff methodology is shown schematically in Figure 4-1.

### 4.2 CRITERIA

The criteria for selection of a solar heating and cooling system configuration will be the following:

- Minimum annual cost for conventional energy consumption.
- Reasonable mix of solar energy contribution to the building's cooling, heating, and hot water loads.
- Reasonable system payback
- Acceptable system safety
- System development risks
- Architectural considerations

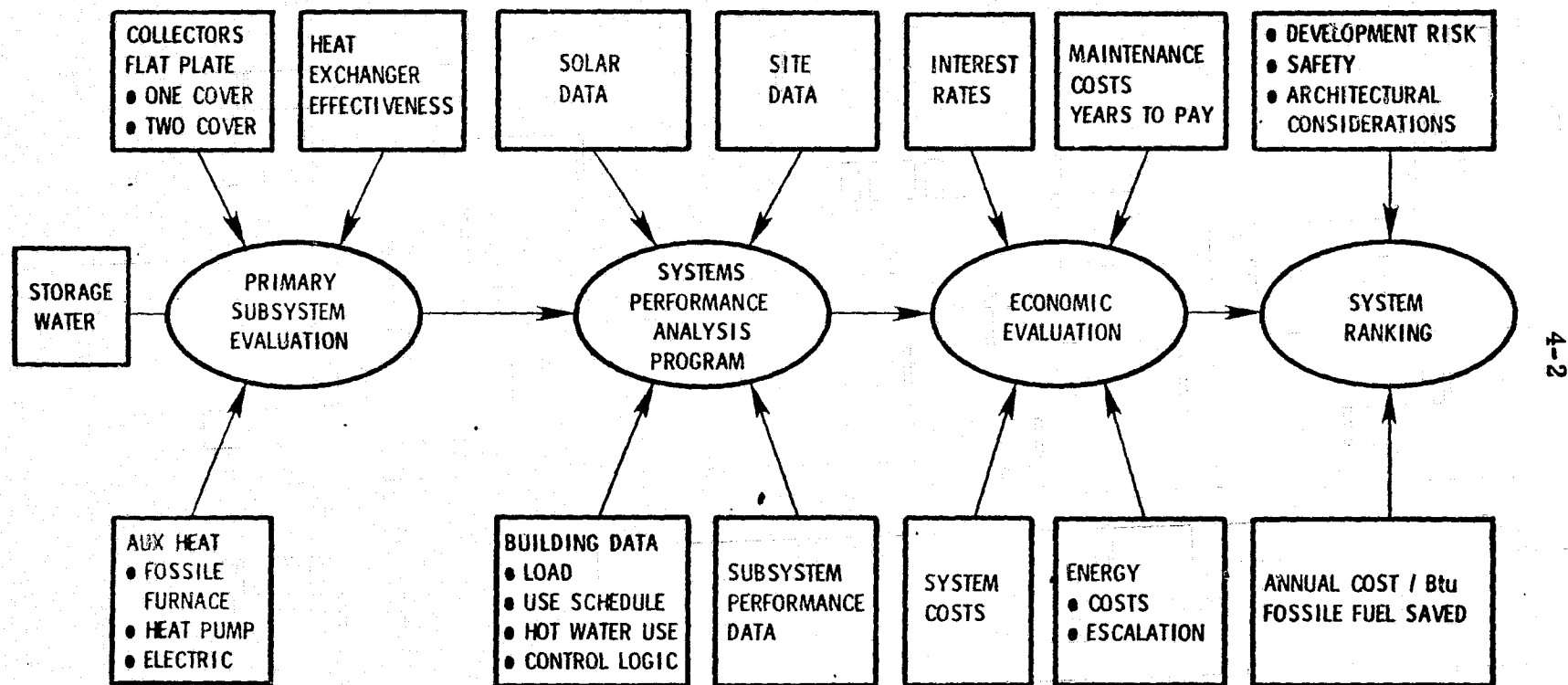


Figure 4-1. System Tradeoff Flow Chart

### 4.3 SITE AND BUILDING MODELS

The prediction of the performance and economics of a solar heating and cooling system is based on a simulation analysis using recorded weather data and calculated incident solar energy as modified by cloud cover data. The system is simulated by a digital computer code on an hour-by-hour basis. A preliminary list of selected cities for the solar heating and cooling systems has been identified.

Single-family heating  
and cooling

Pittsburgh, Pennsylvania

Single-family heating  
and cooling

Jacksonville, Florida

Multifamily heating  
and cooling

Dayton, Ohio

Multifamily heating  
and cooling

Columbus, Georgia

Commercial heating  
and cooling

Washington, D.C.

Commercial heating  
and cooling

Buffalo, New York

No specific sites in these cities have been identified at this writing. Of the above cities, Columbus, Georgia, was selected as a logical choice for formulating the baseline solar heating and cooling system. The location of this city in northern Georgia near Atlanta offers a reasonable balance of annual heating and cooling loads as well as acceptable annual solar radiation. This area also represents a significant marketing area for future solar heating and cooling systems.

A weather tape for Atlanta, Georgia, was employed to simulate the local environmental conditions for all three solar heating and cooling system designs. The mean daily total solar radiation for Atlanta is approximately 396 Langleys. The contoured charts of solar radiation are shown in Figures 4-2 through 4-4. These data were taken from the Climatic Atlas of the United States published in 1968 by National Climatic Center, Asheville, North Carolina. The national charts showing mean percentage of sunshine and total hours of sunshine are presented in Figures 4-4 and 4-3. An interpolation of these charts shows that Atlanta has approximately 2800 hours of sunshine annually, or approximately 60 percent of the possible sunshine.

The average annual heating degree days for Atlanta based on 65°F is 2961. (Reference ASHRAE Systems Handbook, Chapter 43.4). The monthly average heating degree days is shown in Table 4-1.

**Table 4-1. Average Monthly Heating Degree Days for Atlanta, Georgia**

January	636
February	518
March	428
April	147
May	25
June	0
July	0
August	0
September	18
October	124
November	417
December	<u>648</u>
Total	2961

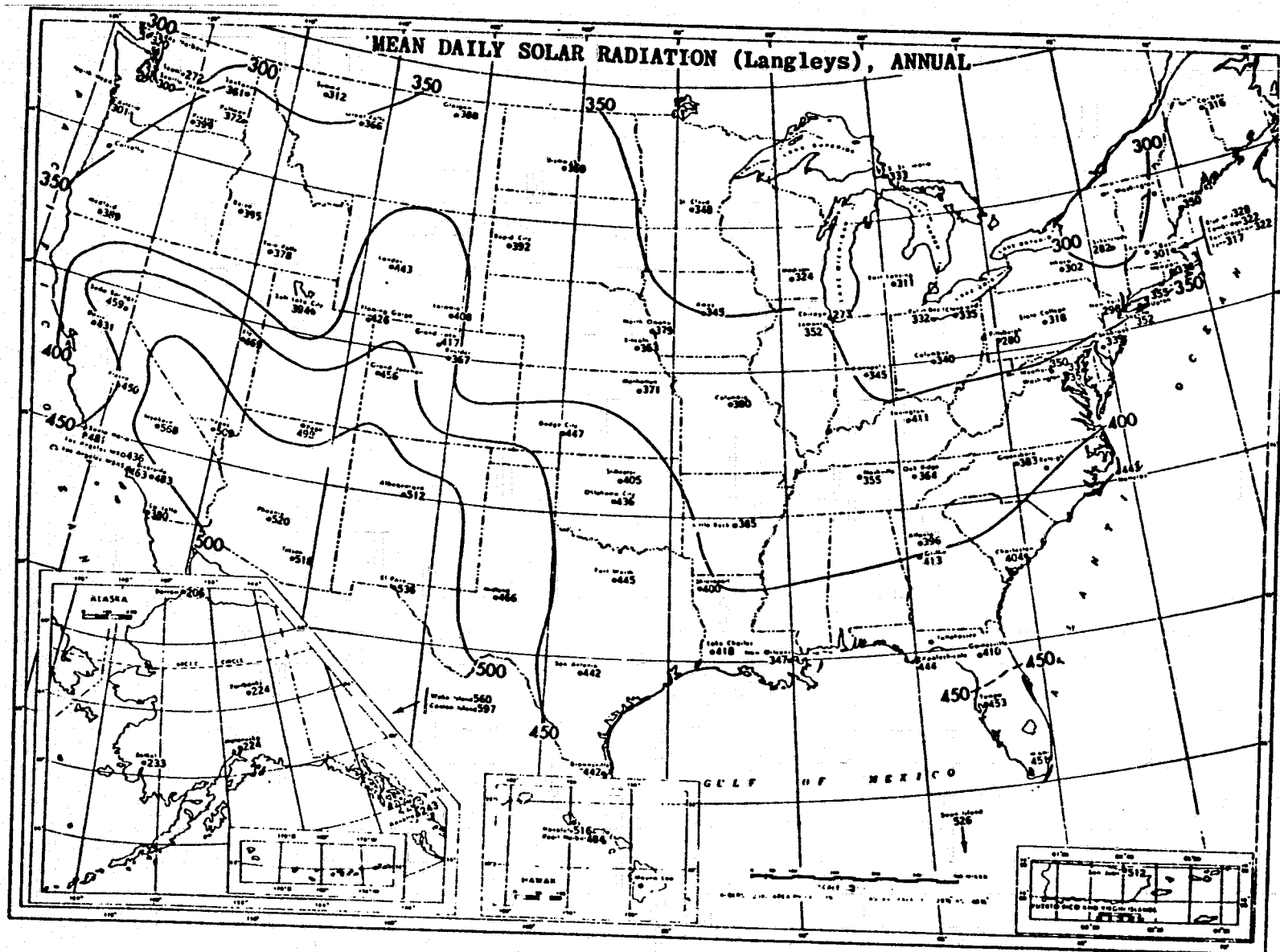


Figure 4-2. Mean Daily Total Radiation, Annual

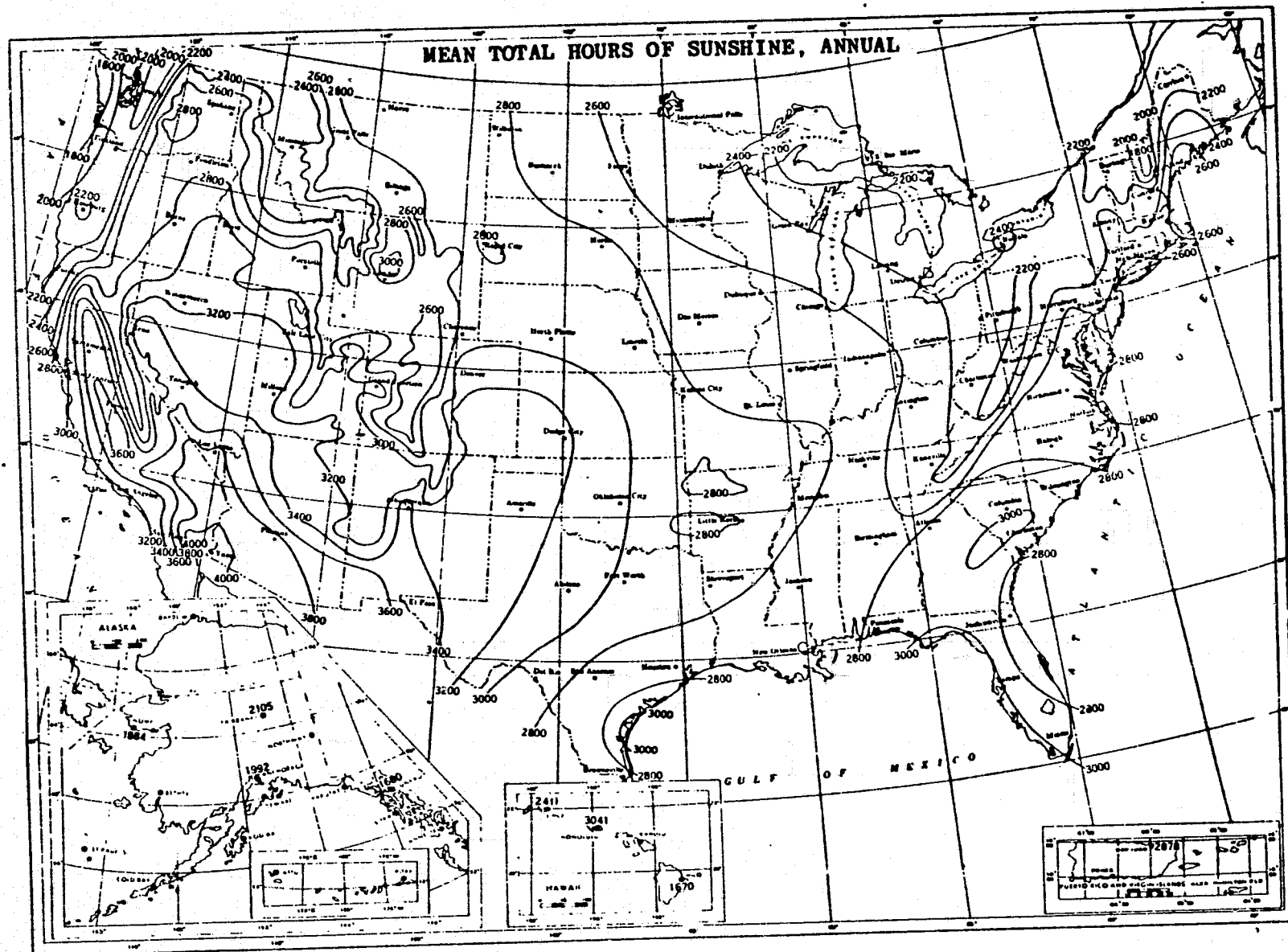


Figure 4-3. Mean Total Hours of Sunshine, Annual



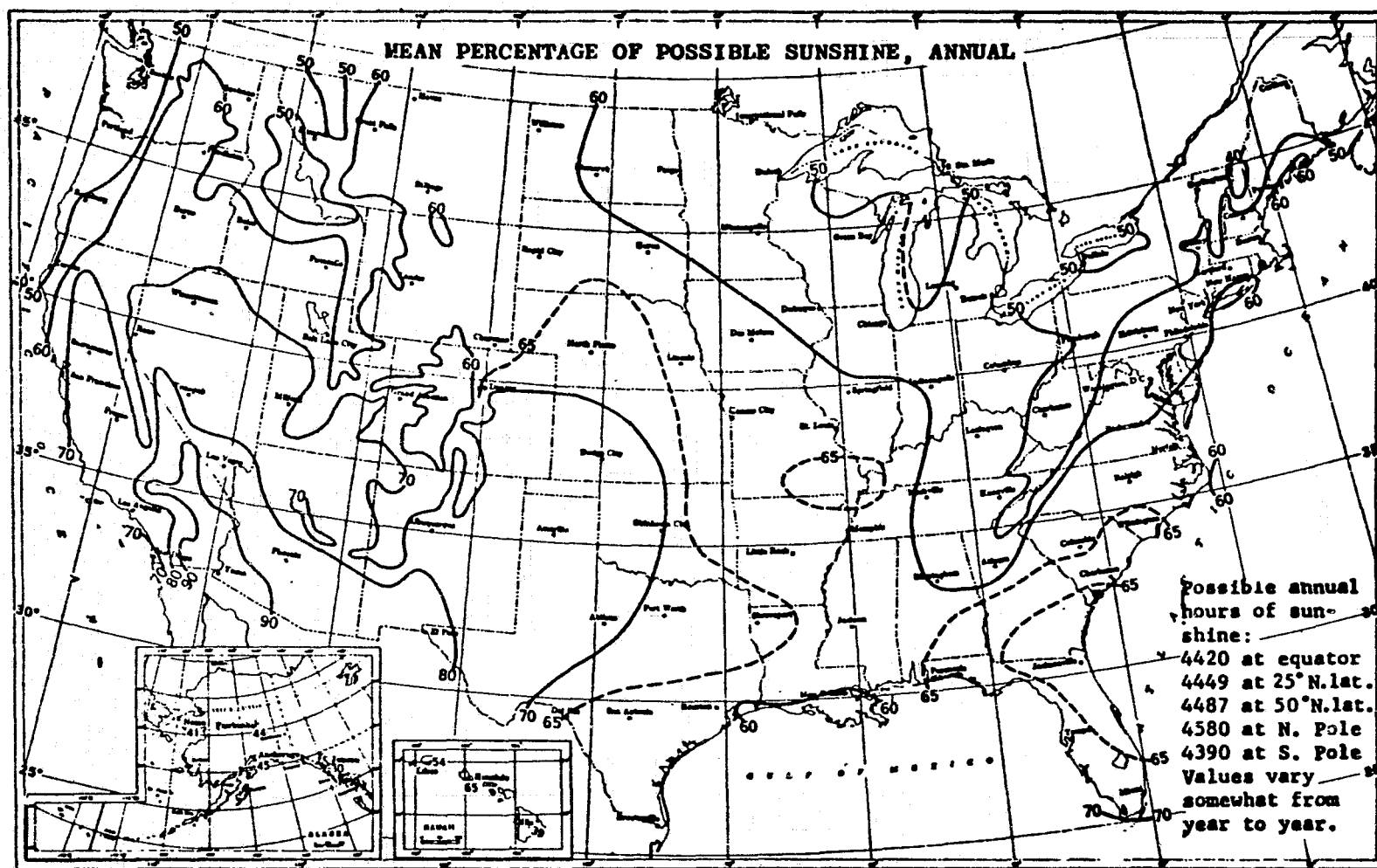


Figure 4-4. Mean Percentage of Sunshine, Annual

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System design conditions and building heating and cooling loads are:

- System Design Conditions -- The heating and cooling system design conditions are based on the requirements outlined in Interim Performance Criteria for Solar Heating and Combined Heating and Cooling Systems and Dwellings (IPC).
- Inside Design Conditions -- The heating plant capacity will be based on the maximum instantaneous block heat loss at outside winter design day conditions when calculated for an inside temperature of 70°F dry bulb, and the selected relative humidity for the project if applicable. The solar-assisted air conditioner capacity will be based on the maximum instantaneous block heat gain at outside summer design day conditions calculated for an inside environment of 76°F db and 50 percent relative humidity.
- Outside Design Conditions -- The design will be based on weather data given in the latest issue of the applicable ASHRAE Guide. The design day heating and cooling loads will be based on the 97 1/2 percent column for heating and the 2.5 percent column for cooling. These conditions for Atlanta are:

Winter:	23°FDB (97 1/2 percent column)
Summer:	92°FDB 77°FWB (2.5 percent column)
- Building Heating and Cooling Loads -- The building heating loads are based on ASHRAE standard 90-75 recommended requirements.. Since no specific buildings or sites have been identified, typical sizes have been used for calculating the energy demands for heating and cooling a single-family

and multifamily residence and a small commercial building, all located in Atlanta, Georgia.

#### 4.3.1 Single-Family Residence

The single-family residence was assumed to be a single-story, 2010 ft<sup>2</sup>, rectangular building (30 x 67), facing south. The composite "U" values for the walls and ceiling were taken from ASHRAE standard 90-75. For Atlanta with 2961 degree days, Figure 4-5 shows that for a type "A" building, the "U" value is 0.265 Btu/hr - ft<sup>2</sup>°F for walls. The "U" value for ceilings for a city having less than 8000 degree days is 0.05 Btu/hr - ft<sup>2</sup>°F. Figure 4-6 presents the "U" value for floors over unheated spaces such as an unfinished basement. The overall thermal transmission conductance is the combination of heat flow through the walls, floor and ceiling.

$$\begin{aligned}
 U_{\text{Floor}} &= 0.19 \text{ Btu/hr ft}^2\text{°F} & \text{Floor area} &= 30 \times 67 = 2010 \text{ ft}^2 \\
 U_{\text{Wall}} &= 0.265 \text{ Btu/hr ft}^2\text{°F} & \text{Wall area} &= 2 (30+67) \times 8 \text{ ft} = 1552 \text{ ft}^2 \\
 U_{\text{Ceiling}} &= 0.05 \text{ Btu/hr ft}^2\text{°F} \\
 UA_{\text{Tot}} &= 0.265 (194) (8) + 2000 (0.05 + 0.19) = 894 \text{ Btu/hr°F}
 \end{aligned}$$

This value includes the energy transfer through any opaque doors and windows included in the outside walls.

The above UA total is based on ASHRAE 90-75 design requirements which have been recommended for new buildings constructed with high thermal resistances and low air leakage so as to effectively utilize as well as conserve energy in the future. Presently constructed buildings and probably most buildings constructed in the immediate future may not meet these relatively high standards.

4-10

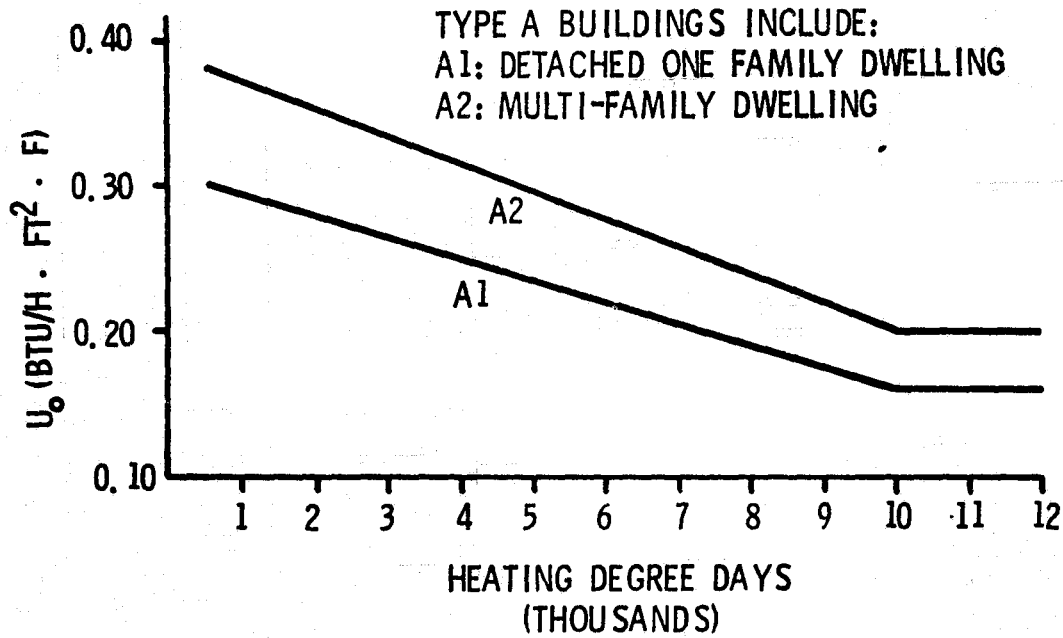


Figure 4-5. U<sub>o</sub> Values -- Type "A" Building

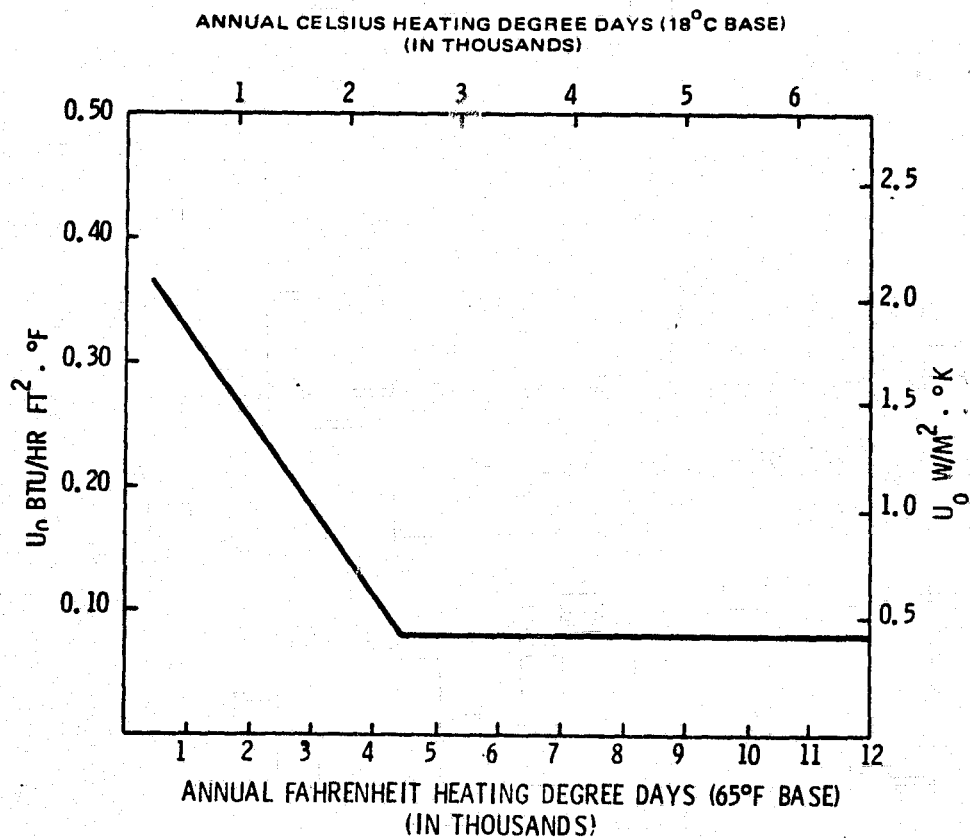


Figure 4-6. U<sub>o</sub> Values -- Floors Over Unheated Spaces

The infiltration through cracks and seams and doorways during daylight hours was assumed to be 1.0 air changes per hour. For a 2010 ft<sup>2</sup> house with an 9-ft ceiling this is 16,080 ft<sup>3</sup>/hr or 268 cfm of outside air which has to be heated and cooled.

The internal load schedule assumed to account for people, appliances, and lights is shown below.

8 a.m. - 8 p.m. -- 2550 Btu/hr

8 p.m. - 8 a.m. -- 1350 Btu/hr

The home is assumed to have 80 ft<sup>2</sup> of unshaded window area on the south-side. Solar loading on these windows has been determined from the following equation.

$$Q_{\text{sun}} = \text{Direct solar flux} \times (\text{window area}) \times (\text{window transmittance}) \times (\text{cosine of sun angle with window normal})$$

$$A_{\text{Window}} = 80 \text{ ft}^2$$

$$\tau_{\text{Window}} = 0.85$$

Figure 4-7 indicates schematically the composite factors defining the single-family residence load based on an Atlanta, Georgia, location and ASHRAE 90-75 construction requirements. The design day heating and cooling loads for this house are calculated as follows:

- Design Day Heating Load

External environment                      23°F DB (nighttime)

House environment                         70°F DB

$$Q_{\text{DD}} \text{ heating} = \text{Conduction} + \text{Infiltration} - \text{Internal load} - \text{Sun load}$$

$$894 (70-23) + 220 (1.08) \times (70-23) - 0 - 0$$

$$Q_{\text{Design day heating}} = 53.185 \text{ Btu/hr}$$

---


$$*1.08 = 60 \text{ min/hr} \times 0.24 \text{ Btu/lb}^\circ\text{F} \times 0.075 \text{ lb/ft}^3$$

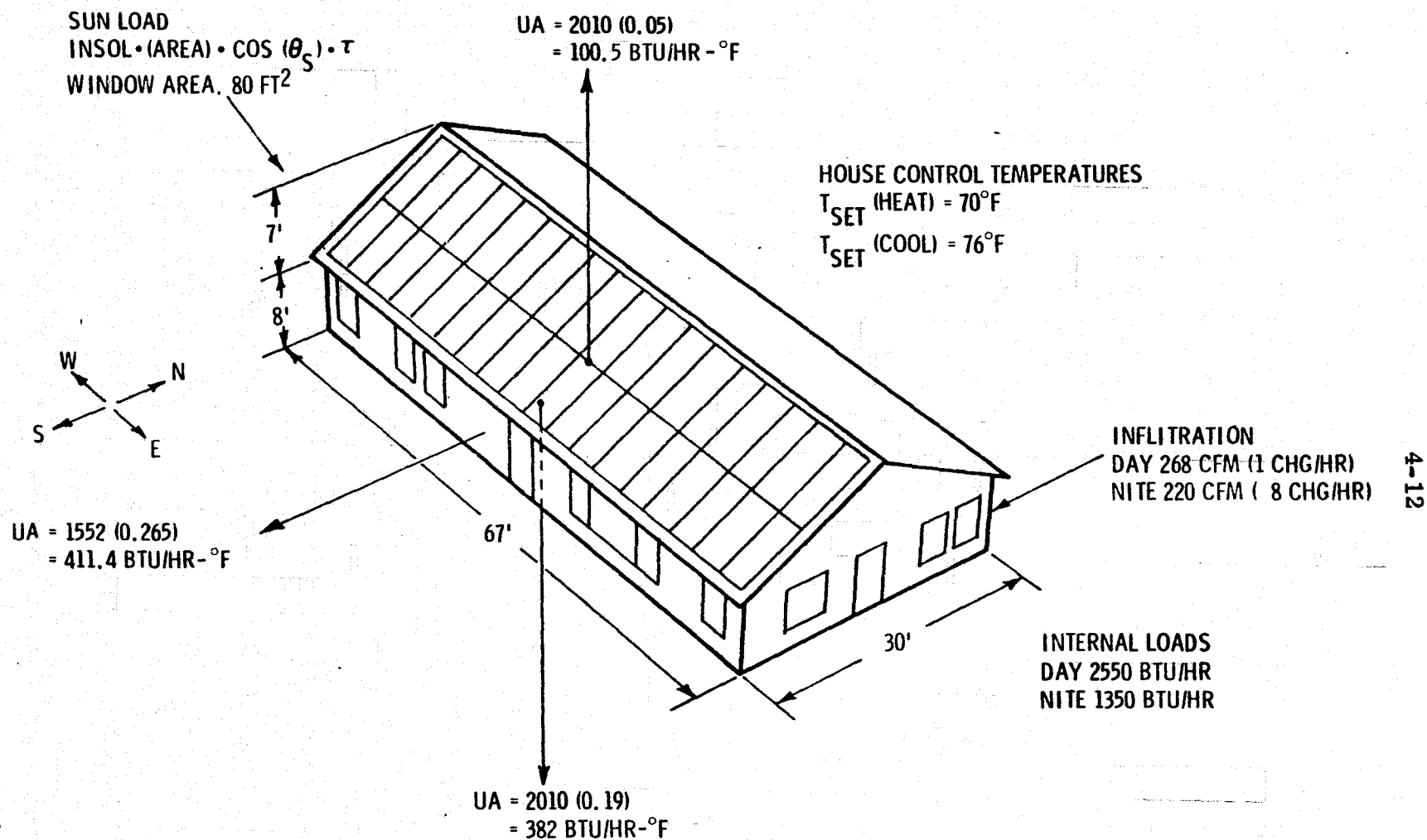


Figure 4-7. Single-Family Residence, Atlanta, Georgia

● Design Day Cooling Load

External environment

92°F DB (Noon)

77°F WB

House environment

76°F DB, 50% Relative Humidity

$$Q_{\text{Design day cooling}} = \text{Conduction} + \text{Infiltration} + \text{Internal Load} + \text{Sun load}$$

$$UA(\Delta T) + W(\Delta h) + \text{Inter. Load} + \text{IDN (A)} (\tau) \cos \theta$$

$$894 (92-76) + 268 (4.5) 11.7 + 2550 + 220 (80) (0.85) (\cos 78)$$

$$Q_{\text{Design day cooling}} = 33971 \text{ Btu/hr (2.83 tons)}$$

Therefore the capacity of the heating and cooling systems for the single-family residence in Atlanta were assumed to be:

Solar-assisted heating system

60,000 Btu/hr

Solar-assisted air conditioning system

36,000 Btu/hr (3 tons)

#### 4.3.2 Multifamily Residence

The multifamily residence was assumed to be a 12-unit, two story, rectangular building (80 x 120 ft), facing south. Figure 4-8 indicates schematically the multifamily residence model. The total volume is 153,600 ft<sup>3</sup> with a floor and roof area of 9600 ft<sup>2</sup> and a total peripheral wall area of 6400 ft<sup>2</sup>.

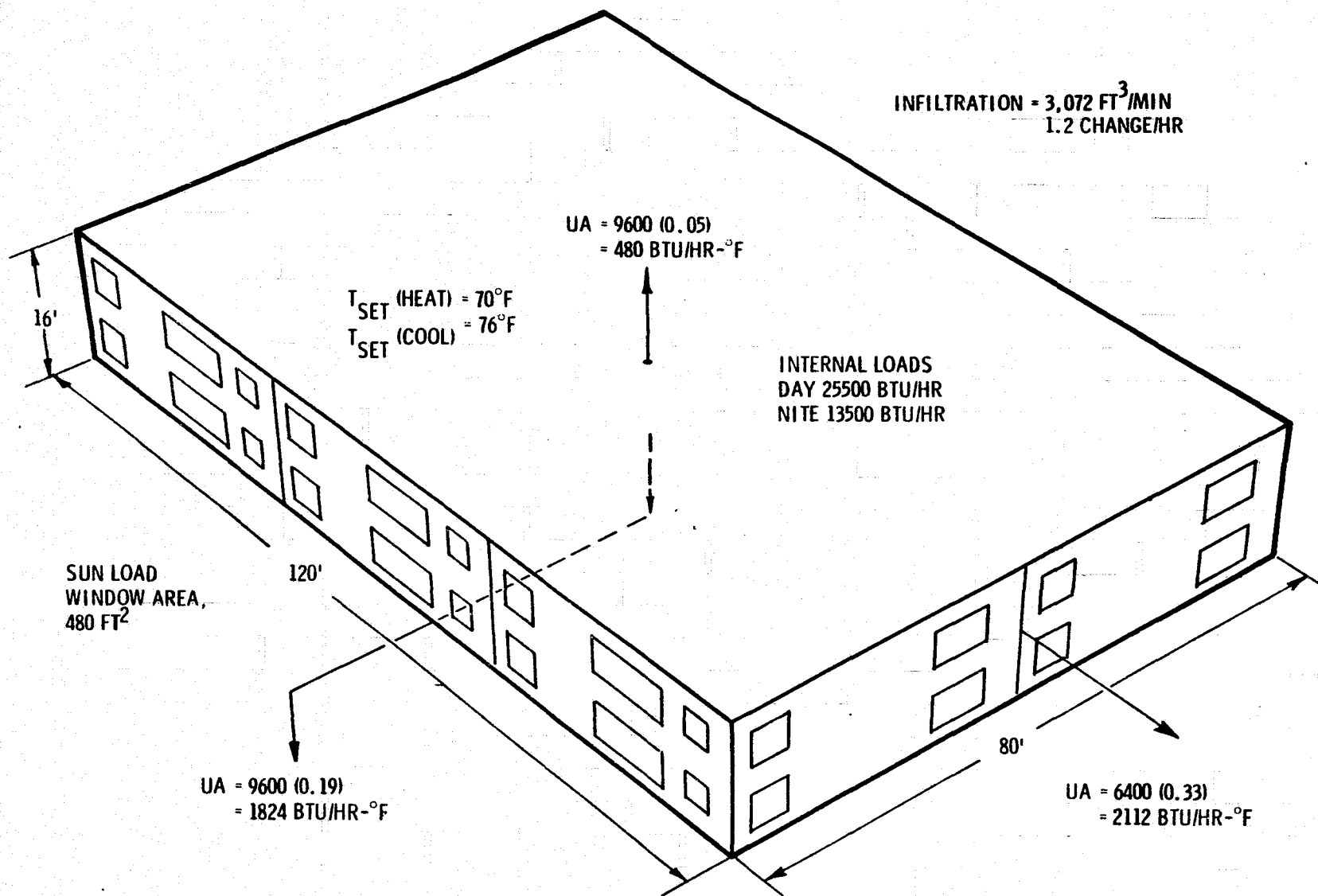
The following composite "U" values for the walls, roof and ceiling were taken from Figures 4-5 and 4-6 and paragraph 4.3.2.2. of ASHRAE 90-75.

Atlanta, 2961 degree days

$$U_{\text{wall}} (\text{Type A2 building}) = 0.333 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$$

$$U_{\text{floor}} (\text{unheated spaces}) = 0.19 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$$

$$U_{\text{Ceiling}} = 0.05 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$$



4-14

Figure 4-8. Multi-Family Residence, Atlanta, Georgia



Infiltration level was assumed to be 1.2 air changes/hr or 3072 CFM based on a large number of window seams and the high level of daily traffic in and out of the building entrances.

The following internal load schedule was assumed:

8 a.m. - 8 p. m.	25500 Btu/hr
8 p.m. - 8 a. m.	13500 Btu/hr

These internal loads account for people (200 Btu/hr/person) appliances, and lights.

Solar loading on the south windows is based on the following parameters:

Total window area on south face	480 ft <sup>2</sup>
Window transmission	0.85
Solar flux at noon on June 21	220 Btu/hr-ft <sup>2</sup>
Sun angle to window	78°F

The design day heating and cooling loads for the multifamily residence model described above in Atlanta, Georgia, is determined as follows:

● Design Day Heating Load

External environment 23°F (nighttime)

Building environment 70°F DB

$$\begin{aligned}
 Q_{DD \text{ Heating}} &= \text{Conduction} + \text{Infiltration} - \text{Internal load} - \text{Sun load} \\
 &= UA (\Delta T) + (W \Delta U) - \text{Loads} \\
 &= 4416(70-23) + 3072(70-23) 1.08 - 0 - 0
 \end{aligned}$$

$$Q_{DD \text{ heating}} = 363,487 \text{ Btu/hr}$$

● **Design Day Cooling Load**

External environment 92°F DB (noon)  
77°F WB

Building environment      76°F DB  
50 percent relative humidity

$$\begin{aligned} Q_{DD} \text{ Cooling} &= \text{Conduction} + \text{Infiltration} + \text{Internal load} + \text{Sun load} \\ &= 4416(92-76) + 3072(4.5)(11.7) + 25500 + 220(480)(0.85) \\ &= 276,558 \text{ Btu/hr (23 tons)} \end{aligned}$$

Therefore the capacity of the heating and cooling system for the multifamily residence assumed in the simulation analysis model were:

Solar-assisted heating system capacity	500,000 Btu/hr
Solar-assisted air conditioning system capacity	300,000 Btu/hr (25 tons)

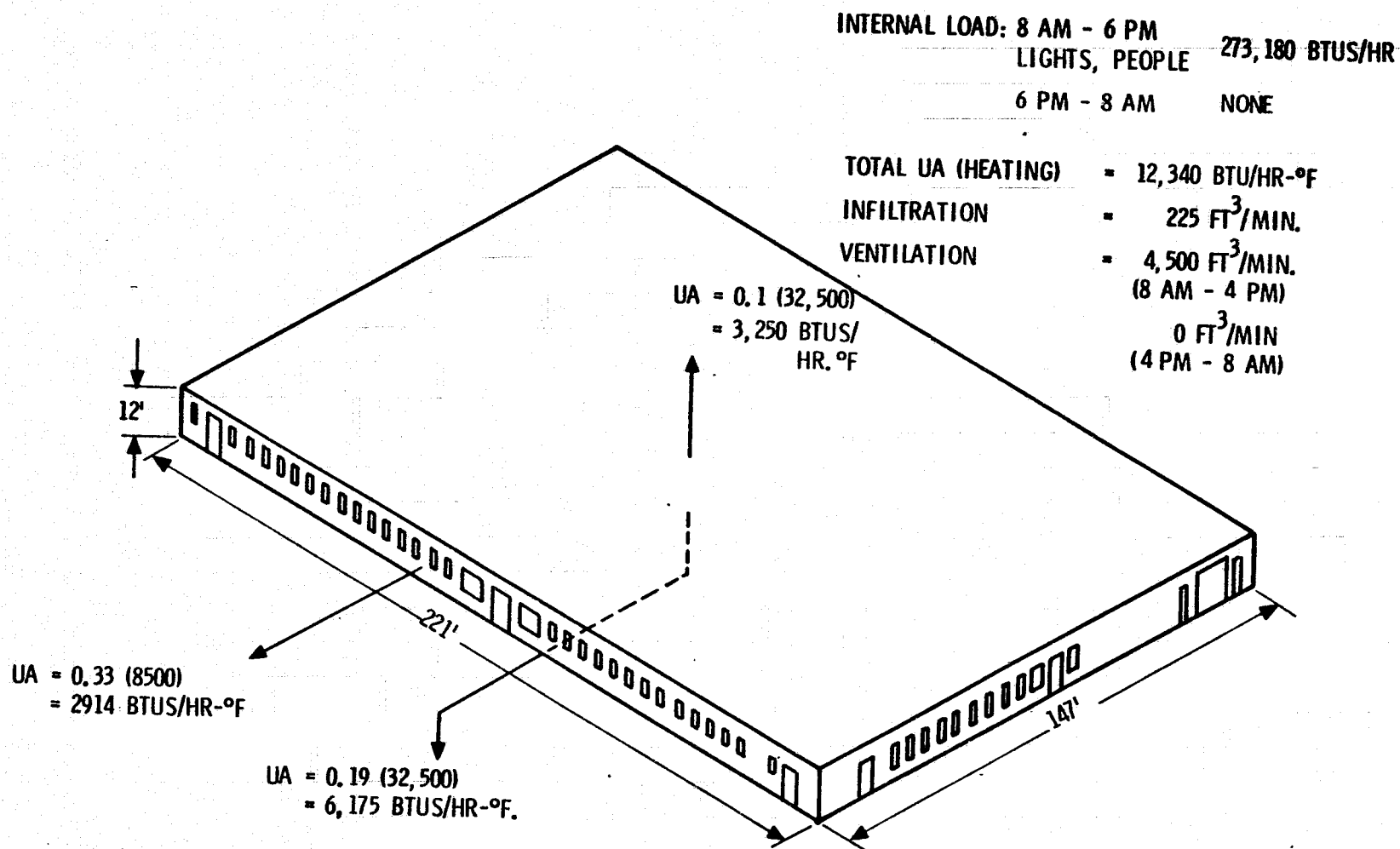
### 4.3.3 Commercial Building

The commercial building model is 32,500-ft<sup>2</sup> single-story, rectangular (147 x 221 ft) structure as shown in Figure 4-9. The building loads were calculated using ASHRAE 90-75 guidelines.

Heating - The "U" value for the commercial building walls was taken from Figure 4-10. For a building under three stories and 2961 degree days in Atlanta, the "U" value is 0.33 Btu/hr ft<sup>2</sup>F. The ceiling "U" value is 0.1 Btu/hr ft<sup>2</sup> and floor "U" is 0.19 Btu/hr ft<sup>2</sup>°F as shown in Figures 4-11 and 4-6, for 2961 degree days. The combined transmission conductance is calculated for a 12 foot-high wall as:

$$UA_{\text{conductance}} = 0.33 (736)(12) + 0.1 (32,500) + 0.19 (32,500)$$

$$UA_{total} = 12,340 \text{ Btu/hr F}$$



4-17

Figure 4-9. Commercial Building, Atlanta, Georgia

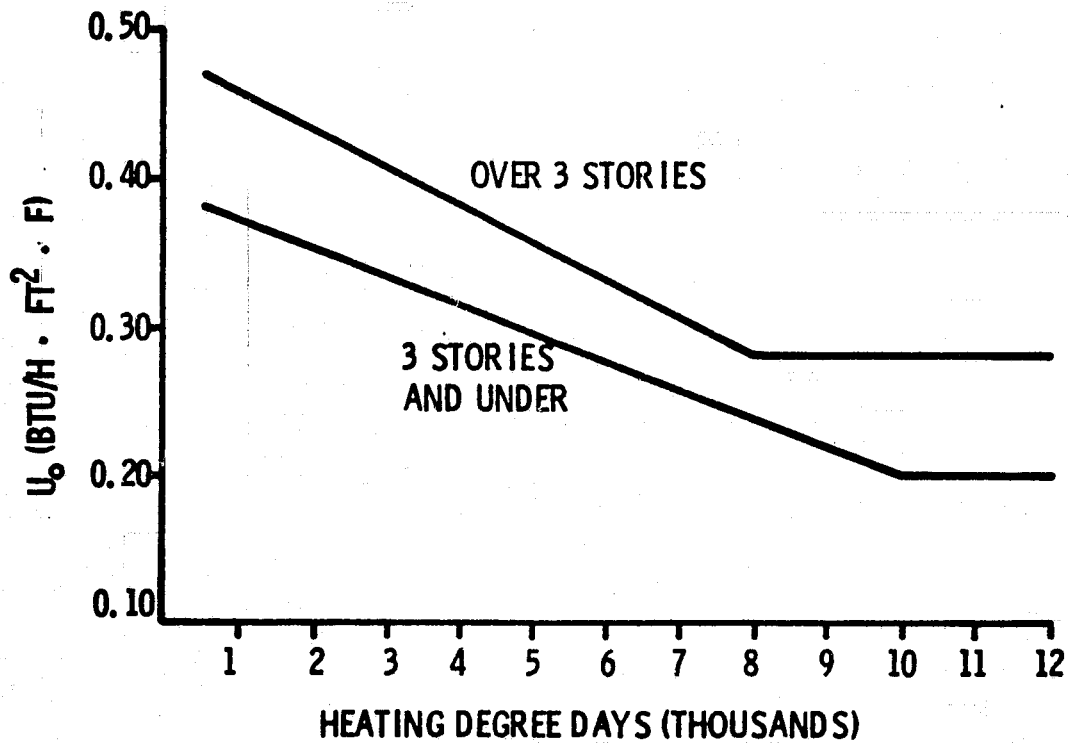


Figure 4-10.  $U_o$  Walls - Heating Type "B" Building

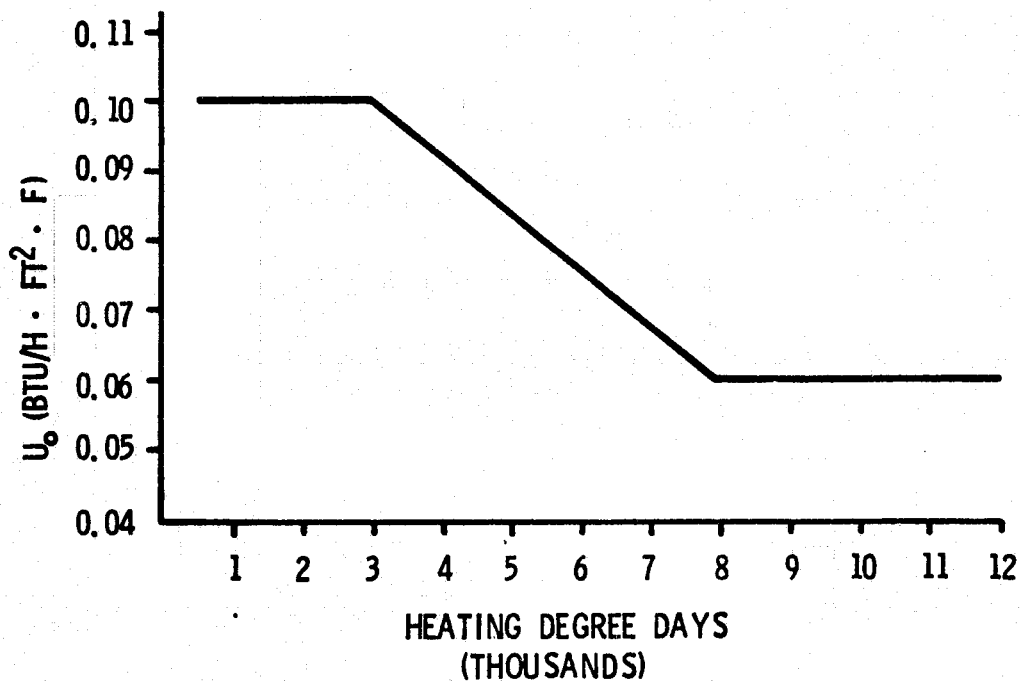


Figure 4-11.  $U_o$  -- Roofs and Ceilings Type "B" Buildings

The infiltration load was assumed to be the same as specified in RFP-404 as 225 cfm or  $UA_{\text{infiltration}} = 243 \text{ Btu/hr F}$ .

The following ventilation schedule was assumed:

4500 ft <sup>3</sup> /min (UA=4860)	8 a. m. - 7 p. m.
0	7 p. m. - 8 a. m.

The internal load schedule for people and lights was:

Lights 213,180 Btu/hr	8 a. m. - 7 p. m.
0 Btu/hr	7 p. m. - 8 a. m.
People 60,000 Btu/hr	8 a. m. - 7 p. m.
0 Btu/hr	7 p. m. - 8 a. m.

Solar loading was assumed negligible for the building.

The design day heating load for the Commercial building is calculated as follows:

$$\begin{aligned}
 Q_{\text{DD heating}} &= U A (\Delta T) + \text{cfm} (\Delta U) - \text{load} \\
 &= 12,340 (70-23) + 4725 (1.08)(70-23) - 0 \\
 &= 579,980 + 239,841 - 0
 \end{aligned}$$

$$Q_{\text{DD heat}} = 819,821 \text{ Btu/hr}$$

Cooling -- The cooling load is determined by a method similar to the heating load with the exception of the wall thermal transfer analysis. ASHRAE 90-75 defines an overall thermal transfer value (OTTV) as a function of degrees north latitude (Figure 4-12) for 35° North, the Atlanta OTTV is 32 Btu/hr ft<sup>2</sup>. This is equivalent to a "U" factor of 1.28 Btu/hr ft<sup>2</sup> F for a 25°F design day,  $\Delta T$ . Therefore, the commercial building conduction UA is computed as follows:

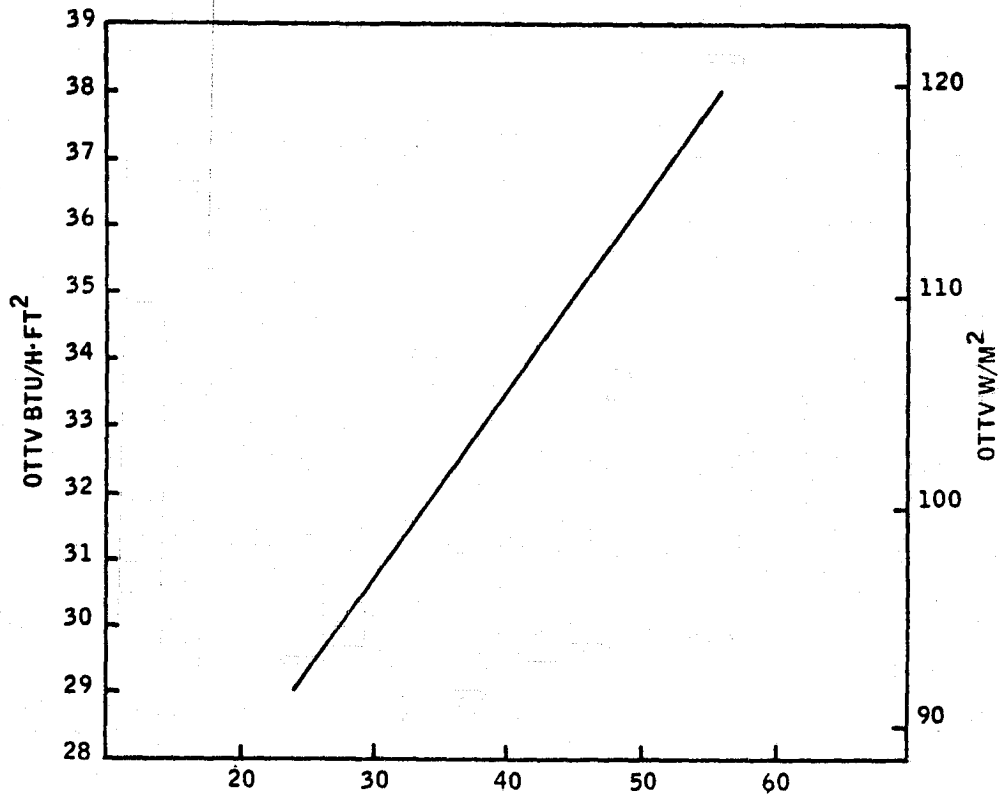


Figure 4-12. Overall Thermal Transfer Value -- Wall Cooling -- Type "B" Building -- Those Not Covered by Paragraph 4.3

$$\begin{aligned}
 UA_{\text{walls}} &= 1.28 \text{ Btu/hr ft}^2\text{°F} \times 736 \times 12 = 11,305 \text{ Btu/hr°F} \\
 UA_{\text{ceiling}} &= 0.1 \text{ Btu/hr ft}^2\text{°F} \times 32,500 \text{ ft}^2 = 3250 \text{ Btu/hr°F} \\
 UA_{\text{floor}} &= 0.19 \text{ Btu/hr ft}^2\text{°F} \times 32,500 \text{ ft}^2 = 6175 \text{ Btu/hr°F} \\
 UA_{\text{conduction}} &= 20,730 \text{ Btu/hr°F}
 \end{aligned}$$

The design day cooling becomes:

$$\begin{aligned}
 Q_{\text{DD cooling}} &= \text{conduction} + \text{infiltration and ventilation} + \text{internal load} \\
 &= 20,730 (92-76) + 4725 (4.5)(11.7) = 273,180 \\
 &= 853,631 \text{ Btu/hr (71.1 tons)}
 \end{aligned}$$

The capacity of the heating and cooling systems for the commercial building model in Atlanta, Georgia, were assumed to be:

- Heating system capacity - 800,000 Btu/hr
- Cooling system capacity - 75 tons

**Domestic Hot Water** -- The domestic hot water schedule for the three types of buildings is shown in Figures 4-13 through 4-15.

The energy requirements to heat the domestic hot water were calculated with the following assumptions (reference IPC).

Delivery temperature = 140 F

Supply temperature = well water temperature

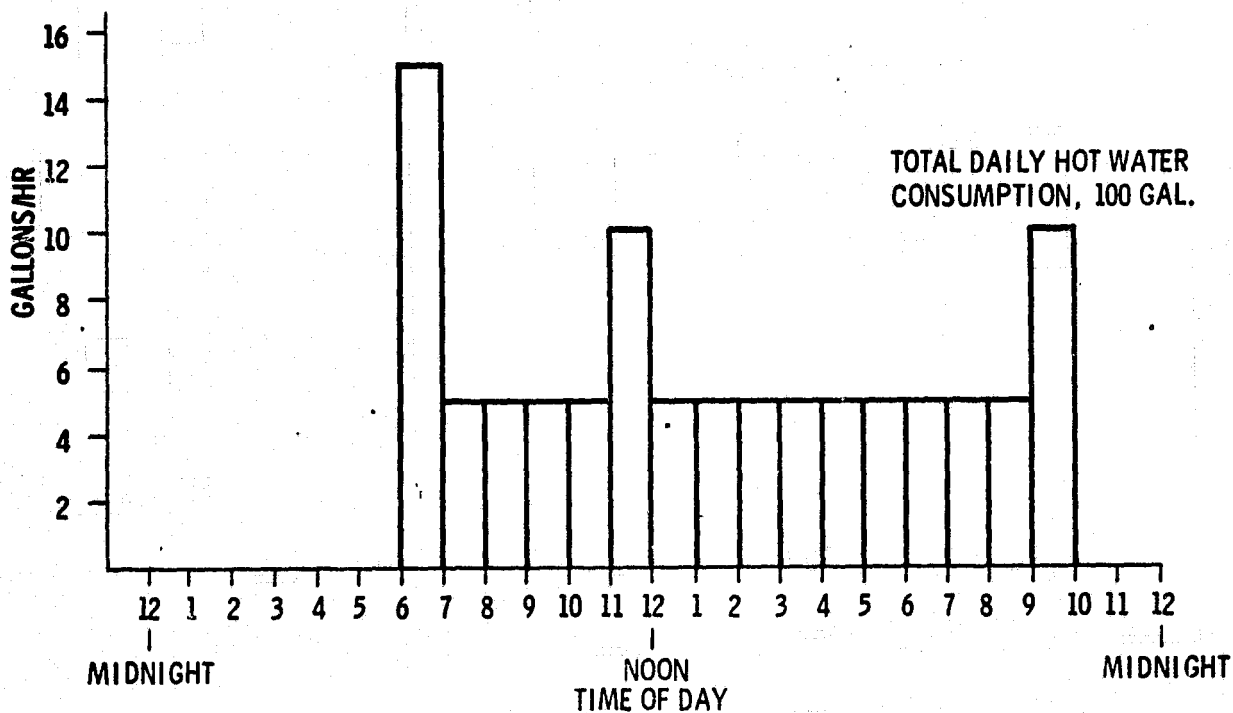


Figure 4-13. Schedule for Domestic Hot Water Consumption -- Single-Family Residence

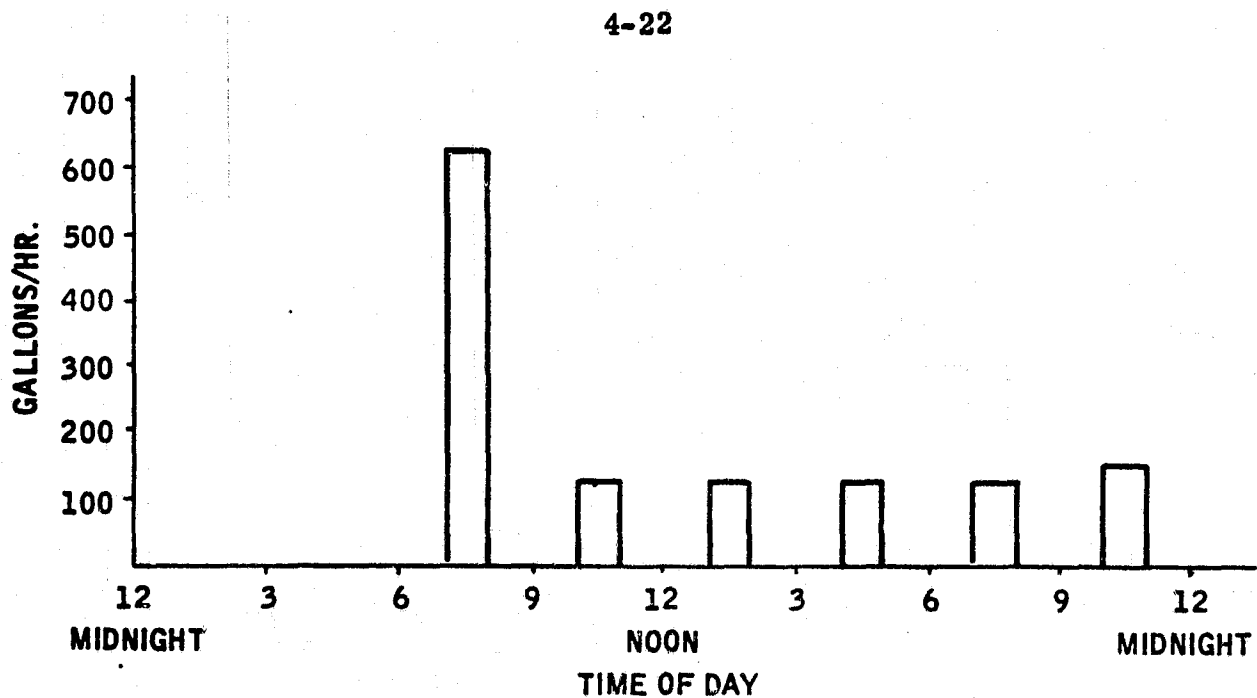


Figure 4-14. Assumed Use Profile for Domestic Hot Water -- Multifamily Residence

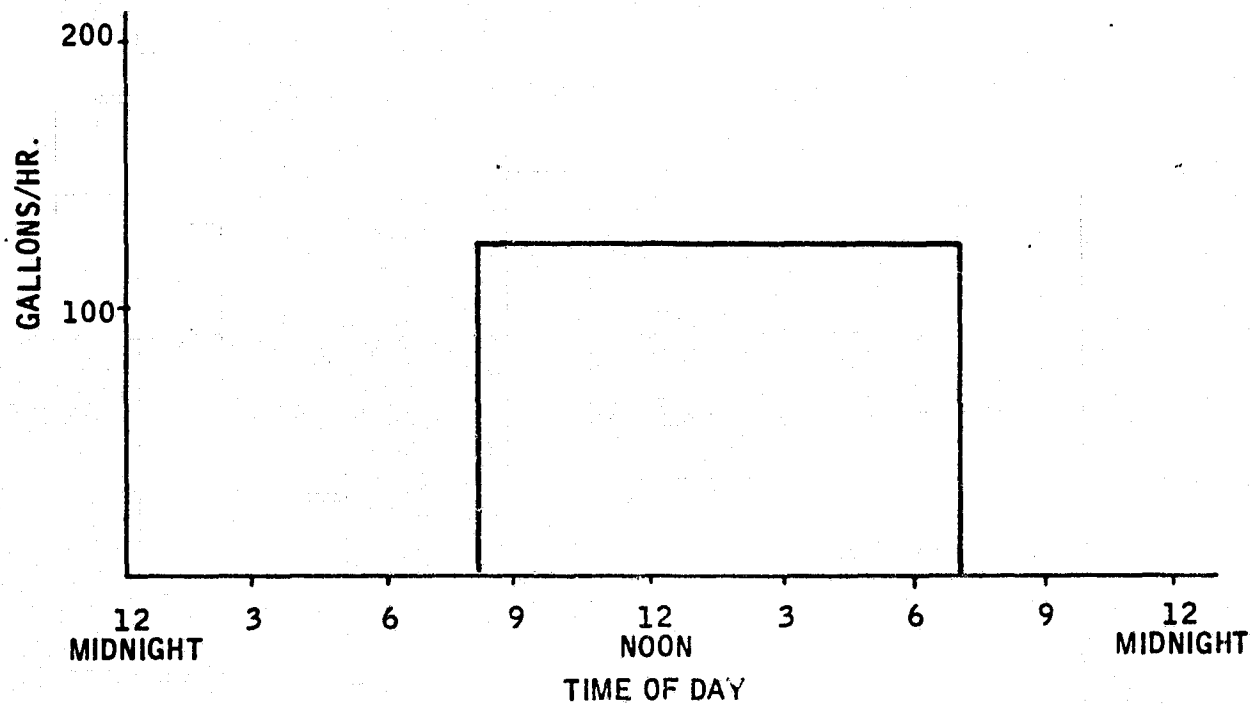


Figure 4-15. Assumed Use Profile for Domestic Hot Water -- Commercial Building



C-2

4-23

The well water temperature for Atlanta varies from 78°F to 52°F between summer and winter. (Reference NBSLD Computer Program for Heating and Cooling Loads in Buildings, NBS1R-74-574, T. Kusuda, November, 1974).

Weather Model -- The following hourly recorded data is taken from weather tapes provided by the National Climatic Center, Asheville, N.C. (Reference Airways TDF-14, Surface Observations Manual, Director National Climatic Center, Federal Building, Asheville, N.C., 28801).

- Dry bulb temperature
- Dew point temperature
- Wet bulb temperature
- Wind speed
- Barometric pressure
- Total cloud cover
- Type of cloud
- Occurrence of wet precipitation
- Occurrence of dry precipitation

Flat-Plate Collector Model -- The flat-plate model is based on a tilted flat plate facing south. Although the model allows both the tilt and direction to be specified, they are held constant for any one simulation run. The amount of energy collected is expressed by:

$$Q_{out} = K_1 Q_{inc} - K_2 (T_{in} - D_b)$$

where:

$Q_{inc}$  = Amount of solar radiation incident on the collector surface  
(calculated and modified by cloud data on an hourly basis)

$T_{in}$  = Temperature of fluid at collector inlet

$D_b$  = Ambient dry bulb temperature

$K_1, K_2$  = Constants for proposed Lennox collector

For two cover collector:

$$K_1 = 0.74$$

$$K_2 = 0.6$$

For one cover collector

$$K_1 = 0.8$$

$$K_2 = 0.68$$

The two-cover collector design was assumed throughout the analyses. The greater efficiency and resultant higher fluid exit temperatures associated with the two-cover design are essential for optimum Rankine cycle performance.

Storage -- Energy storage is accomplished through the sensible rise in temperature of a tank filled with water. For the purposes of our analysis, an insulated cylindrical steel tank with specified length and diameter was assumed. Energy supplied to the tank was assumed to be evenly distributed to all segments.

Economic Studies -- A computer subroutine was used to make economic comparisons between alternate systems and subsystems. The economic evaluations are based on annual cost requirement. The methodology for calculation of the annual cost requirement is based on formulas derived by Grant and Ireson<sup>1</sup> and Rosalie T. Ruegg<sup>2</sup>.

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<sup>1</sup>Eugene L. Grant and W. Grant Ireson, Principles of Engineering Economy, Ronald Press Company, New York 1964

<sup>2</sup>R. T. Ruegg, "Solar Heating and Cooling in Buildings: Methods of Economic Evaluation," NSB1R 75-712.

Figure 4-16 summarizes the procedure. The output is the annual revenue requirement and is the average dollar amount that would be required each year to cover the cost of operating and paying for the solar system.

The basic inputs needed for the procedure are as follows:

- System cost
- Annual hours of equipment operation (i. e. , pumps and fans)
- Energy demand for the building
- Maintenance and major replacement costs
- Local energy costs
- Energy escalation rates
- Local cost of money

Fuel costs are shown separately because it is expected that electricity will escalate at a different rate than natural gas or oil. Also, maintenance costs are separate since the timing of maintenance may not be demanded at a cost rate whereas the operating costs will be constant.

The system operating costs, including fuel and pumping/fan power, are brought to an equivalent basis for comparison purposes by computing the present value. The formula for this is:

$$PV = \frac{1}{(1+i)^n}$$

where  $i$  = interest rate or cost of money and  $n$  = number of interest periods or years. The interest rate selected for the study was 7 percent. The economic subroutine in the computer simulation allows for inserting various interest rates for study purposes.

The annual cost required by the homeowner to pay for the system as well as the operating and energy costs is predicted by summing the present value of fuel and operating costs over the life of the system, adding the system first

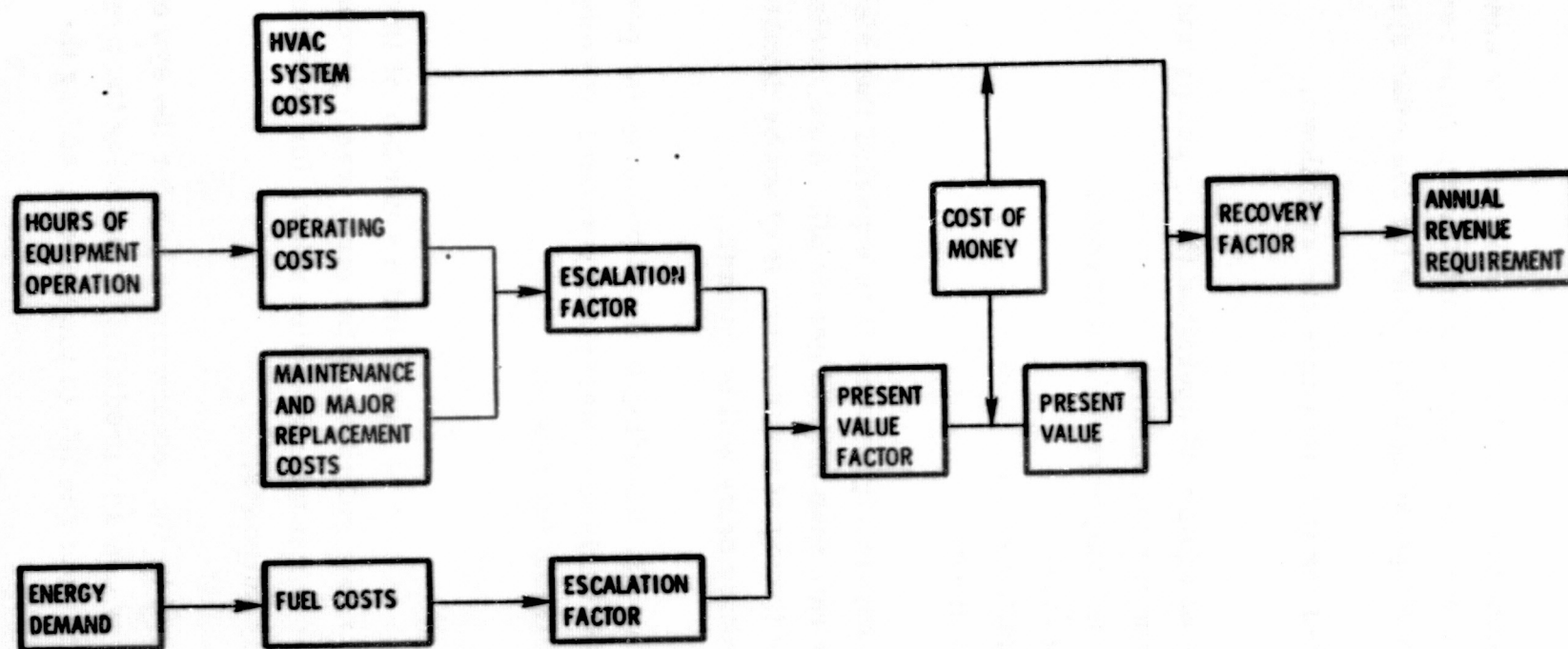


Figure 4-16. Level Annual Revenue Requirement



cost and then reducing this amount to a uniform payment by the capital recovery factor formula.

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1}$$

For a 20-year interest period and 7 percent interest, the capital recovery factor is 0.09439.

The maintenance costs are assumed small and thus have been neglected. Also not included are allowances for increases in property tax, increases in insurance, or other factors which might increase or decrease payments. Salvage value of equipment at the end of the amortization period was not considered.

Cost of Energy -- The cost of natural gas, oil and electricity for Atlanta for 1976 were based on current rate schedules recently obtained from Georgia Power Company, Atlanta Gas and Light, and Exxon Oil Company. These current monthly rates are:

• <u>Natural Gas<sup>+</sup> (1000 Btu/ft<sup>3</sup>)</u>			
<u>ft<sup>3</sup></u>	<u>\$ (SFR)</u>		<u>\$ (MFR)</u>
0 - 400	2.00 + 0.624/1000 ft <sup>3</sup>	first 5 days	1.924/1000 ft <sup>3</sup>
400 - 1500	1.854/1000 ft <sup>3</sup>	next 10 days	1.374/1000 ft <sup>3</sup>
1500 - 30,000	1.514/1000 ft <sup>3</sup>	over 15 days	1.274/1000 ft <sup>3</sup>
Over 30,000	1.284/1000 ft <sup>3</sup>		
• <u>Oil</u>			
No. 2	0.4/gal	Same as SFR	

---

<sup>+</sup>Includes purchase gas adjustment factor of \$0.624/1000 ft<sup>3</sup>

● Electricity\*

Minimum charge per month -	2.50
First 20 kwh	3.05
Next 80 kwh per kwh	0.04/kwh
Next 100 kwh per kwh	0.0359/kwh
Next 150 kwh per kwh	0.0251/kwh
Next 250 kwh per kwh	0.027/kwh
Over 600 kwh	0.0354/kwh

Projected Cost of Energy -- There appears to be many scenarios for the escalation of energy in the future. While future availability problems are speculative, the critical nature of energy use in today's society will obviously result in increased fuel costs. Energy costs, unless regulated nationally, will escalate at different rates for each geographical area.

Several escalation rates were assumed for the economic analysis so as to determine the sensitivity of solar economics to this important yet unpredictable factor. The escalation rates shown in column No. 1 of Table 4-2 were presented at the NASA contractors meeting of 3 August 1976. They are also the rates assumed in the Heating PDR analyses (reference A.D. Little, "Base Prices and Forecast", April, 1974).

For example, Column No. 1 predicts that electric power costs will escalate at 1.4%/year, gas at 4.2%/year and oil at 2.2%/year. In the year 2000, the cost of electricity will be about 40 percent more than today (i. e.,  $[1.014]^{24} = 1.396$ ) with gas up by 168 percent and oil up by 69 percent.

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\*Includes fuel adjustment charge of \$8.31/1000 kw-hr



Table 4-2. Fuel Escalation Rates Employed for the Heating/Cooling Economic Analyses

Energy Source	Power Cost Multiplier				
	Annual Escalation Rate				
Electrical Power			1.014	1.05	1.10
	Power Cost Multiplier	1st yr	1.014	1.05	1.1
		24th yr	1.396	3.22	9.85
Gas	Annual Escalation Rate		1.042	1.05	1.10
	Power Cost Multiplier	1st yr	1.042	1.05	1.10
		24th yr	2.684	3.22	9.85
Oil	Annual Escalation Rate		1.022	1.04	1.08
	Power Cost	1st yr	1.022	1.04	1.08
		24th yr	1.686	2.56	6.34

#### 4.4 SOLAR SIMULATION DESCRIPTION

##### 4.4.1 General Features

Honeywell has developed a general-purpose computer program (SUNSIM) for use in the design and evaluation of the solar heating and cooling demonstration systems.

Component models or relationships and loop constraints are used in the computer program to model multiple-loop solar systems as sets of non-linear differential equations. The differential equation may be integrated forward in time to determine fuel savings or linearized numerically for stability analysis using a fully automated modern control software package.

The closed-loop simulation structure illustrated in Figure 4-17 consists of three key functional blocks:

- A MAIN program which inputs data, controls the integration and linearization of the differential equations, samples the output and generates report quality output plots and tables.
- A first-order Adams-Bashforth integration STEP routine which updates state variables based on current and past values of the derivatives.
- A DERIVative routine which contains the differential equations used to model the system.

A derivative subroutine contains deterministic functions of time such as the diffuse and direct components of solar radiation on cloudy days and hourly weather data available for over 300 weather stations in the United States. Two different models may be used to compute the diffuse and direct components of solar radiation:

- An ASHRAE procedure based on weather data provided by the National Climatic Center and a cloud cover radiation model due to Kimura and Stephenson.
- Actual measurements of total radiation on a horizontal surface with an analytical estimate of diffuse and direct components based on a Liu-Jordan correlation. The radiation data is provided on tapes by the University of Wisconsin.



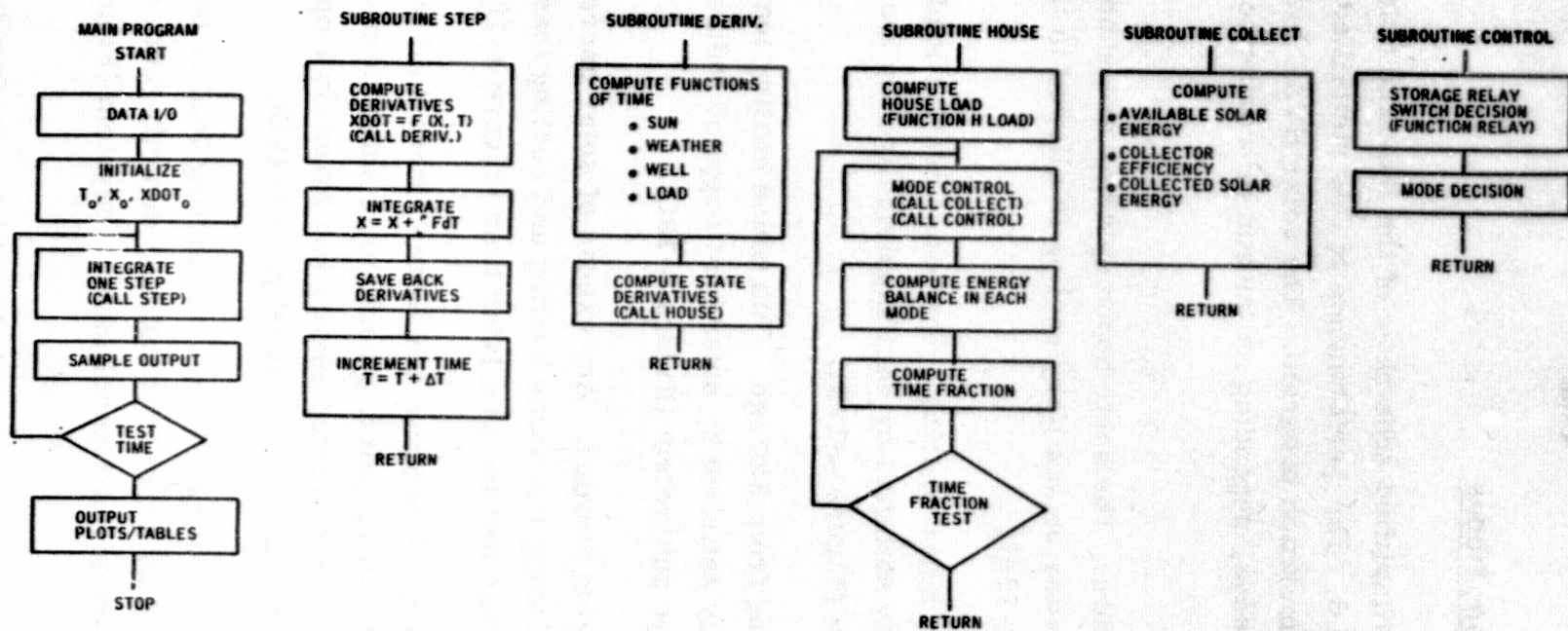


Figure 4-17. Structure of SUNSIM Software

#### 4.4.2 Solar HVAC System Modes

Figure 4-18 presents a simplified schematic of the Baseline Solar HVAC system discussed in Section 3. The performance of this simplified configuration was analyzed with the simulation program. The solar HVAC system may operate in any of nine modes, depending on the sun or weather conditions and on the house\* loads.

- Mode 1 - Shutdown. The solar house systems are shutdown; no auxiliary heating or auxiliary cooling is allowed in this mode (Figure 4-19A).
- Mode 2 - Direct Solar Heating. The house heating load is partially or fully satisfied by heat supplied directly from the solar collectors (Figure 4-19B).
- Mode 3 - Heating from Storage. The house heating load is partially or fully satisfied by solar heat supplied indirectly from the storage subsystem (Figure 4-19C).
- Mode 4 - Charging Storage. Any excess of solar energy subsequent to furnishing the house heating and cooling load is charged to storage for use at a later time (Figure 4-19D).
- Mode 5 - Purging. Whenever the solar system fluid temperature is near its boiling point, a heat purge subsystem is used to reject the excess collected energy and maintain loop temperature at acceptable levels (Figure 4-19E).
- Mode 6 - Direct Cooling. The house cooling load is partially or fully satisfied by solar heat supplied directly from collectors to the Rankine cycle - air conditioner subsystem (Figure 4-19F).

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\*The house shown in Figure 4-18 is symbolic of a single-family residence, a multifamily residence, or a commercial building.

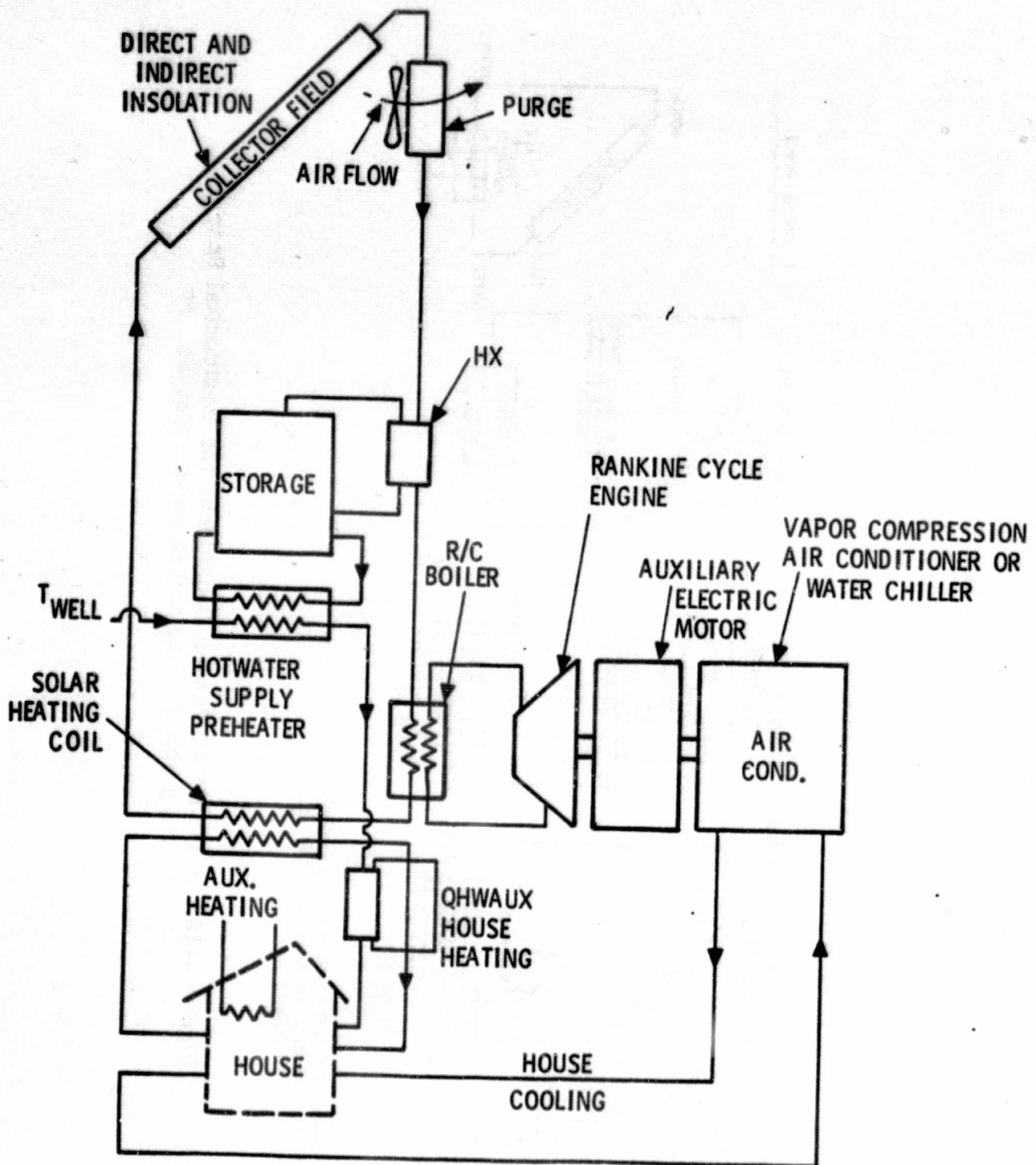


Figure 4-18. Schematic Diagram of a House Equipped with a Solar HVAC System







- Mode 7 - Cooling from Storage. The house cooling load is partially or fully satisfied by solar energy furnished from storage to the Rankine cycle - air conditioner subsystem (Figure 4-19G). An auxiliary heater or an auxiliary electric motor is turned on to satisfy the house heating and cooling loads anytime the solar energy supply is insufficient.
- Mode 8 - Sole Auxiliary Heating. Only auxiliary heating unit is operating in this mode (Figure 4-19H).
- Mode 9 - Sole Auxiliary Cooling. Only the auxiliary electric motor is driving the air conditioning unit in this mode (Figure 4-19I).

A general flow diagram for the house subroutine in the SUNSIM program is shown in Figure 4-20. The program control logic selects the modes in the following sequence of priorities:

- Solar energy directly from the collectors to meet the loads
- Solar energy directly from the storage tank to meet the loads
- Auxiliary energy sources to meet the load
- Storage charging when solar energy available and house loads have been satisfied.

#### 4.4.3 Solar HVAC Models

The solar heating and cooling systems for the SFR, MFR, and commercial building were analyzed with the above program. This code simulates the system's performance when subjected to actual hour-by-hour weather data. It calculates the heating and cooling load, hot-water load, and solar energy supplied to each load for a complete year of operation. The code determines

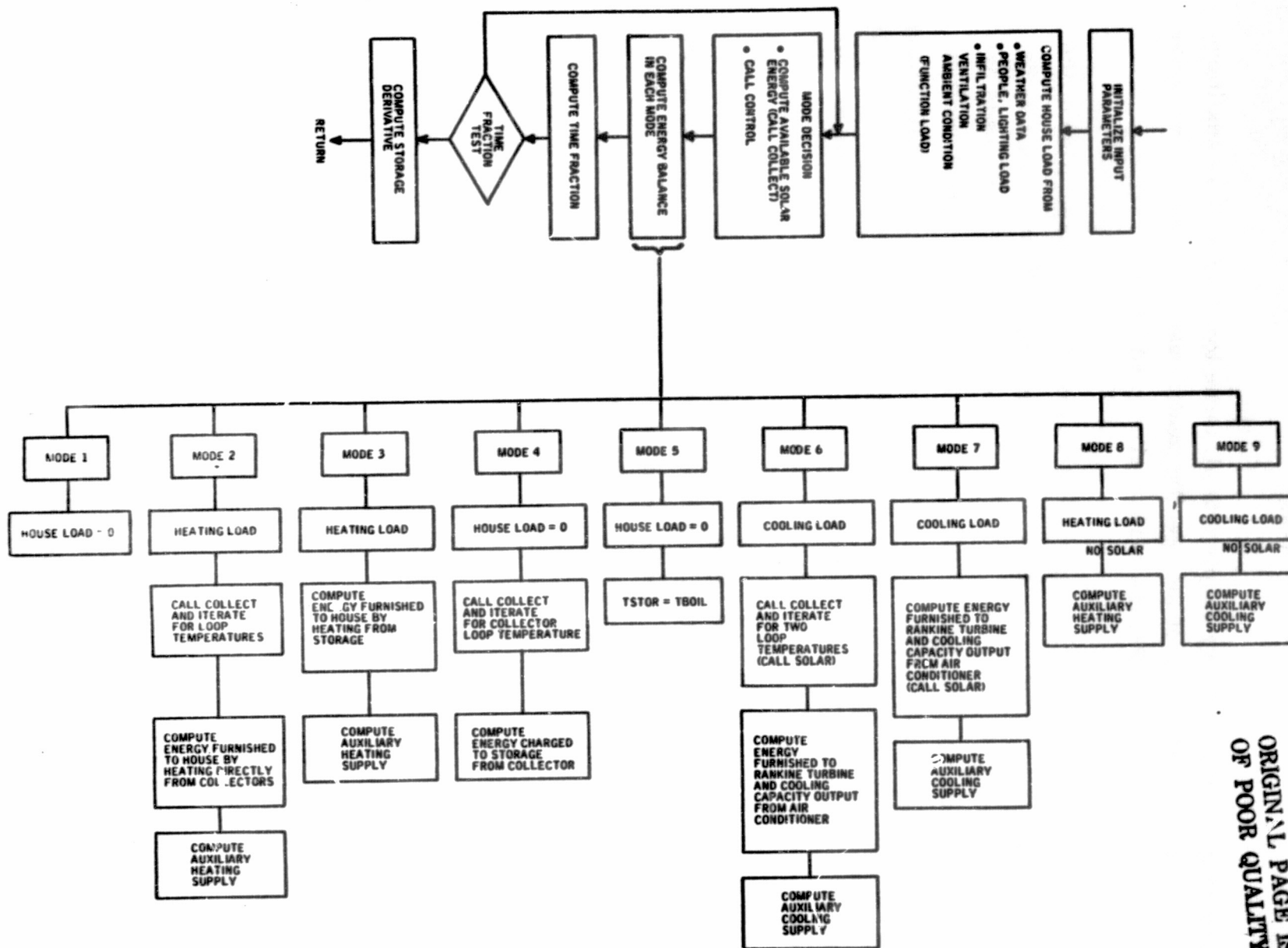


Figure 4-20. Flow Design for Subroutine House (SUNSIM)

the percentage of solar energy supplied to all three loads as well as auxiliary fuel required. A description of the specific models used to simulate the solar HVAC subsystems shown in Figure 4-18 is presented below.

Building Model -- The building load models as well as the computation of the design day loads for the SFR, MFR and commercial building in Atlanta were discussed previously in subsection 4.3. Hourly heating and cooling loads are computed based on weather data provided from a tape for the typical weather year specified for a given local. The loads are computed based on maintaining a constant and uniformly distributed air temperature of 70°F during the heating season and 76°F during the cooling season. It is assumed in all three models that heating and cooling loads do not exist simultaneously.

Collector -- The collector model used in the simulation code is based on actual test data. The flat-plate collector model expresses the energy gain per unit area of collector in terms of inlet fluid temperature, ambient temperature, incident solar radiation and solar incidence angle.

The incident solar flux is calculated by an ASHRAE subroutine called SUN and modified by hourly local cloud cover data read from a U.S. Weather Bureau tape. The outside ambient temperature is also read from the Weather Bureau tape, which is actual measured data for a specific city (Atlanta in this case). Fluid inlet temperature to the collector is calculated by performing an energy balance on the HVAC system loop several times per hour during the sunlight hours. This energy balance determines the resultant fluid temperature in the piping that returns to the collector array. The solar collector model is limited to steady-state unidirectional heat transfer.

Weather Data -- The weather data for Atlanta is read from U.S. Weather Bureau magnetic tapes purchased from the National Weather Bureau in Asheville, N.C. The information includes ambient wet and dry bulb temperatures, wind speed, dew point, cloud cover, day, and time of day data. The



Atlanta weather year 1952 has been defined by the U. S. Weather Bureau as representative for a typical year in this city.

Storage -- Energy storage is accomplished through the sensible rise in temperature of a tank filled with water. For the purposes of the analysis, an insulated cylindrical steel tank with a specified volume was assumed. The tank environment and the upper limit of temperature of operation will be the deciding factors in determining the thickness of insulation and the surface area of the tank which control the heat leakage. Energy supplied to the tank was assumed to be evenly distributed to all segments. Effects of storage tank temperature stratification were neglected for this analysis. However, this phenomenon is recognized as a potential system problem that will be studied during the detailed system design phase. The effect of the tank capacity on the solar system performance was considered in this analysis. An optimum and economical tank size was determined for each of the systems studied.

#### 4.4.4 Rankine Cycle Simulation

The simulation was used to evaluate the performance of a Rankine cycle heat engine in driving either a vapor compression cycle as a direct expansion unit or a water chiller to supply cooling to residences and buildings. Mathematical modeling of the Rankine engine was conducted to study the effects of various engine designs and sizes under variable operational conditions. A yearly performance simulation was undertaken to select a baseline design and determine, for a given set of design restrictions, an optimal operational range and control scheme.

The design and performance studies were carried out assuming the Rankine engine to be coupled to small-to-medium capacity air conditioning units capable of meeting the static load at design conditions. Capacity sizes representative of single and multifamily residential cooling requirements were

selected, namely 3- and 25-ton units. For off-design performance, a control strategy was required to augment the shaft power input to the compressor. An electric motor generator was used as a motor to supply auxiliary energy to the air conditioner, or as a generator to use the excess work of the turbine and thus match engine output to required load. Two types of control schemes were investigated: constant and variable speed. The following subsections illustrate the logic used by the computer program and the computational procedure followed for each of the two control modes.

Constant Speed -- Auxiliary energy required for system augmentation was one of the major characteristics used in the evaluation and selection of an optimum Rankine engine design. For the constant-speed approach, electrical energy was supplied to maintain the unit operating at the specified design capacity. But in order to evaluate the engine performance over a wide range of interface parameters, such as the temperature of the hot water available from the solar collectors and the wet and dry bulb temperatures of the ambient air, four different engine sizes were considered. For a given collector area and available solar energy, the various engine sizes have different heat addition requirements which could affect the operational temperature of the collectors and the storage tank and thus the overall thermal efficiency of the system. With the air conditioner design compressor input power for a residential unit at 2.4 hp, the 1 hp and 2 hp engines were undersizes while the 3 hp and 4 hp were oversizes at design conditions. Engine design temperatures were varied between 170°F and 210°F.

In the year-long simulation, the Rankine engine was modeled as a "black box" by using performance curves generated for all operating states within the desired temperature range. These curves were provided by Barber-Nichols Engineering and a sample is shown in Figures 4-21 and 4-22. The assumptions used in deriving these curves were assumed to hold throughout the simulation and no attempt was made to compute the condenser temperature as a function of outdoor wet bulb temperature. Refrigerant R-113 was assumed for the 3-ton system and both R-11 and R-113 were studied for the 25-ton system.

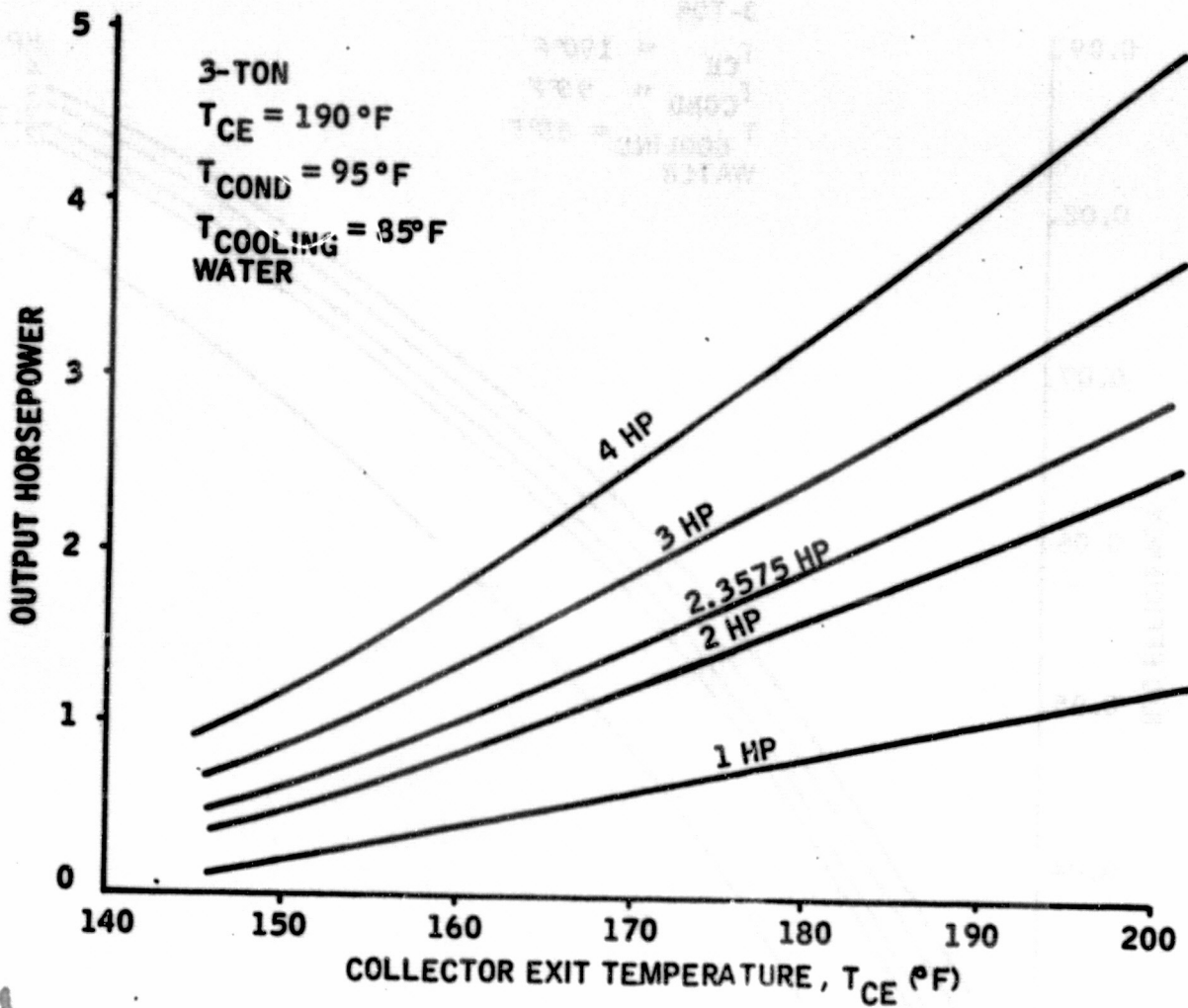


Figure 4-21. Output Horsepower versus  $T_{CE}$  at Constant Horsepower Off Design

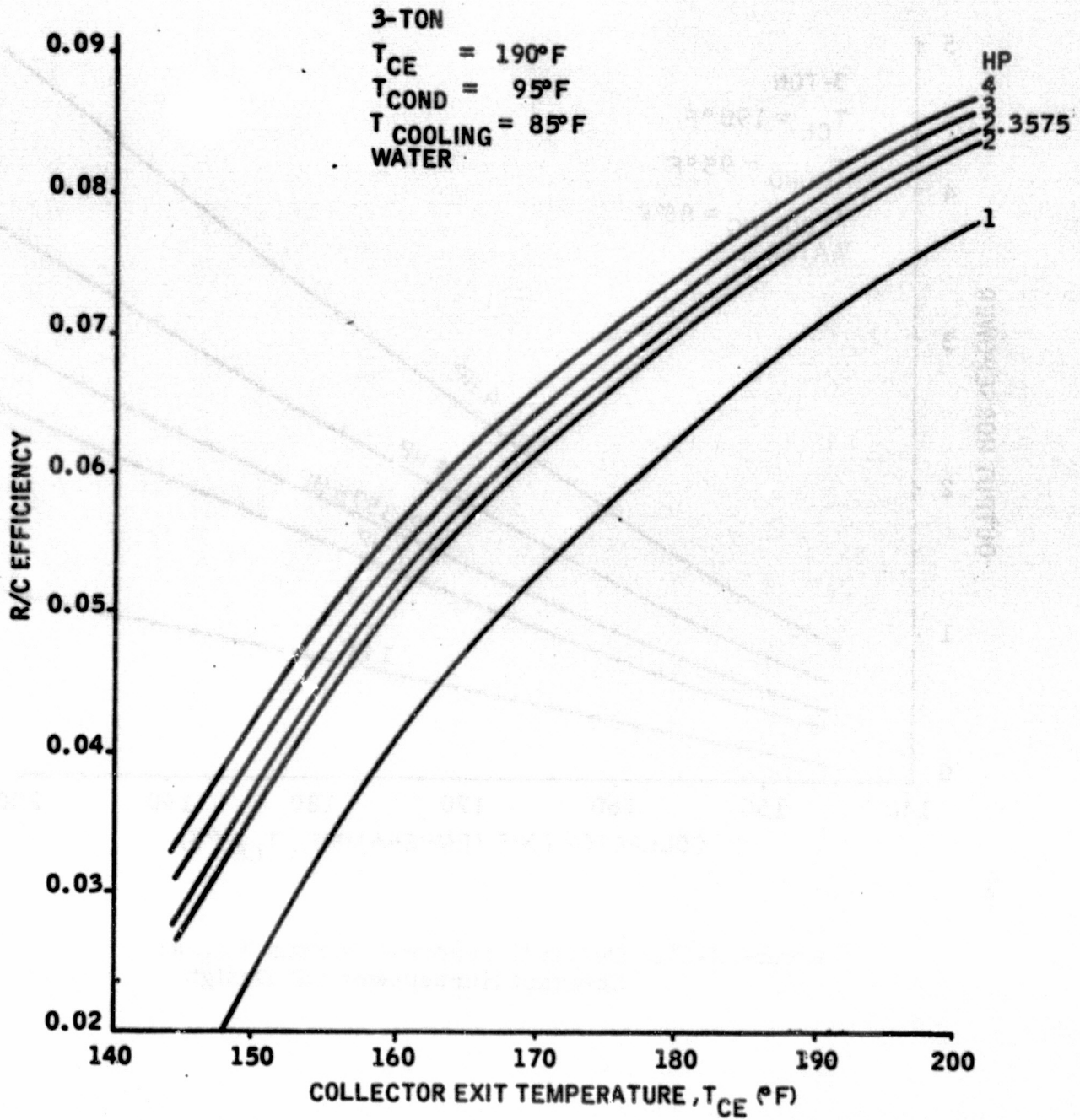


Figure 4-22. R/C Efficiency versus  $T_{CE}$  at Constant Horsepower Off Design



Basically, for a given engine size and design temperature, two cycle parameters were necessary to adequately describe its off design performance:

- 1) The turbine shaft power which, in the constant speed control mode, determines the amount of auxiliary power required to operate the unit at design capacity. Excess shaft power indicates that the motor is in a generating mode.
- 2) The Rankine cycle efficiency  $\eta_{RC}$  defined as

$$\eta_{RC} = \frac{\text{Turbine Shaft Power}}{\text{Heat Input}}$$

which is used to calculate the input energy to the vapor generator section of the Rankine engine. This amount of energy is completely provided by the solar system through the collectors or the storage tank.

Another parameter used in the analysis is the coefficient of performance (COP) which is defined as the ratio of cooling rate and the compressor power. For these tradeoff calculations, the COP was assumed independent of size and a constant value of 6.0 was adopted for the vapor compression and water chiller coolers.

Interpolation of these performance curves reduces the computational time substantially without affecting its accuracy. These cycle data were then interfaced with the corresponding components of the solar system to evaluate the performance of all system elements. Basically, in order to adequately model the solar energy input to the system and compute the system component characteristics, a computational interval of one-half hour was taken with the input variables assumed constant over the whole time interval. Then the calculations proceeded along the following steps:

- a) Fluid temperature at boiler inlet is determined through an iterative procedure to match the heat input requirements of the Rankine engine with the energy available directly from the collectors or the storage tank. A predetermined minimum inlet temperature is specified for initialization of the simultaneous iteration on each of the two loops.
- b) Using the boiler inlet temperature, the program computes the first cycle parameter, the turbine shaft power output, from the corresponding performance curve. If this power is high enough to offset the parasitic losses, the Rankine engine contribution is used to determine the auxiliary power required to maintain a constant power input to the compressor.
- c) The solar heat input is calculated from the second cycle parameter, the Rankine efficiency  $\eta_{RC}$ . An energy balance on the boiler section is performed to determine the fluid outlet temperature.
- d) System capacity defined as the product of shaft power and COP is compared to the space cooling load. If the capacity exceeds the load the fraction of the time period considered during which the air conditioner is "on" is calculated, energy expenditure from solar and auxiliary is computed for the time period. If the system is operating above design capacity, excess energy from the turbine shaft is converted to electrical power.
- e) Electrical energy input to satisfy the auxiliary power requirements was evaluated based on the motor efficiency at the corresponding part load operation. At full load, a design efficiency of 77 percent was assumed. Variation in efficiency for the motor generator is shown in Figure 4-23.

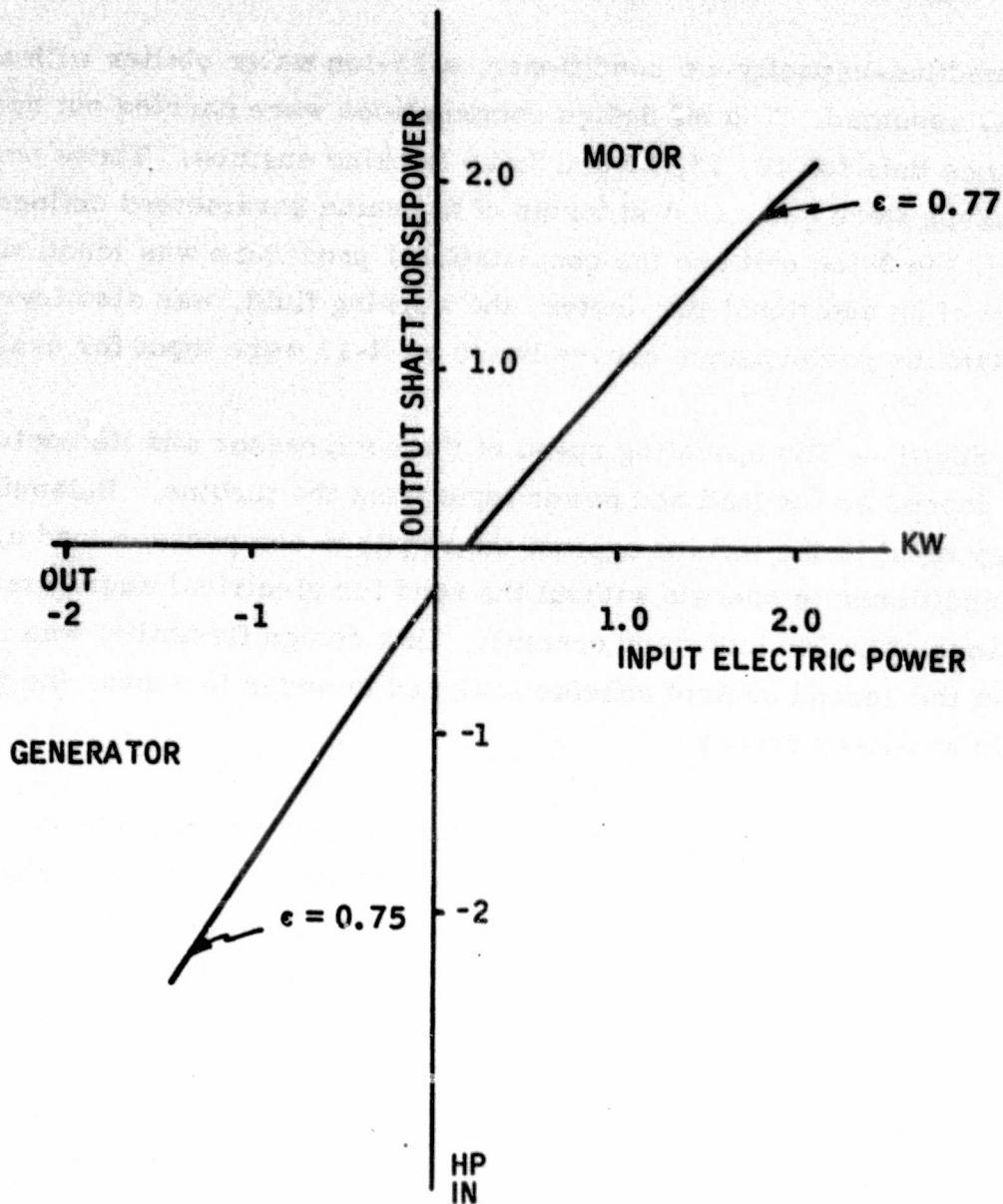


Figure 4-23. Motor Generator Variation in Efficiency



- f) Summations are made of the energy requirements and calculations are repeated for the next time interval.

For the medium-capacity air conditioner, a 25-ton water chiller with a COP of 6.0 was assumed. The off design computations were carried out based on performance data for 10, 15, 20 and 25 hp Rankine engines. These performance curves were generated in terms of the same parameters defined earlier for the 3-ton unit and the computational procedure was identical. The effect of an additional parameter, the working fluid, was also investigated. Rankine performance curves based on R-11 were input for evaluation.

Variable Speed -- The operating speed of the compressor and its cooling capacity depend on the load and power input from the turbine. Balancing the energy input to the turbine against the required compressor load allows the air conditioner to operate without the need for electrical augmentation over a range of boiler inlet temperature. This design flexibility was incorporated in the second control scheme analyzed in order to assess the potential savings in auxiliary energy.

Modeling at part load performance of the turbine and compressor was based on actual experimental data available from a previously built unit. Figure 4-24 shows the variation of system capacity and energy input requirement as a function of boiler inlet temperature. A design capacity of 3 tons at 215°F inlet was assumed. As this water temperature drops, the turbine power drops due to a rapid deterioration in the efficiency and the cooling capacity of the unit becomes a small fraction of its rated capacity. A minimum inlet temperature of 170°F was chosen as a cut-off limit corresponding to 30 percent of the total cooling capacity.

For comparison with constant-speed operation, the performance curves in Figure 4-24 were incorporated in the simulation along with a simplified control scheme. No modulation in the space temperature was allowed and the



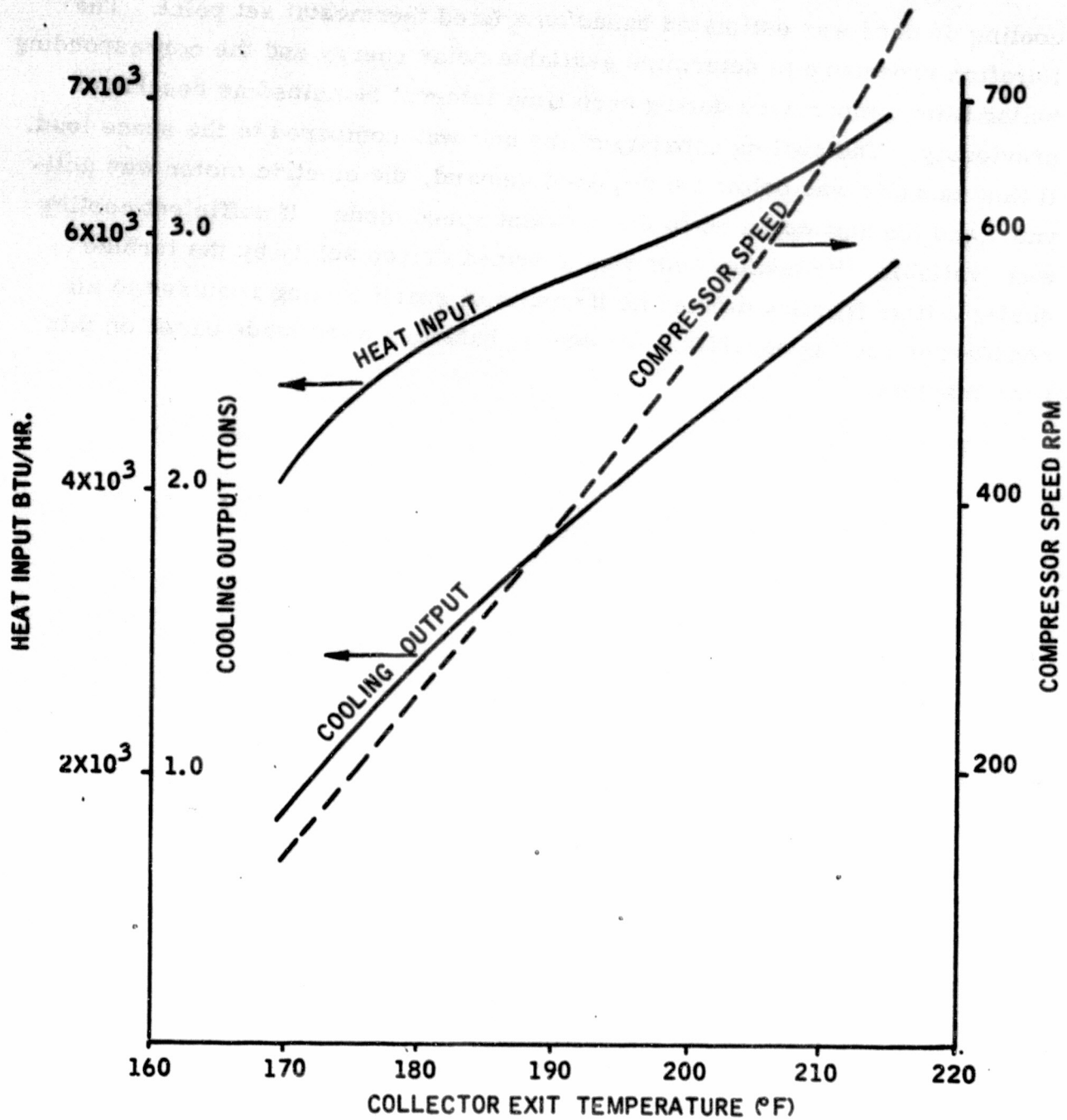


Figure 4-24. Variation of System Capacity and Energy Input Requirement as a Function of Boiler Inlet Temperatures

cooling demand was estimated based on a fixed thermostat set point. The iteration procedure to determine available solar energy and the corresponding boiler inlet temperature during each time interval remained as described previously. The cooling capacity of the unit was compared to the space load. If this capacity was below the required demand, the electric motor was activated and the unit operated in the constant speed mode. If sufficient cooling was available, the compressor was assumed driven solely by the turbine during a time fraction defined as the ratio of space cooling required to air conditioner cooling capacity. All energy balances were made based on this time fraction.

## SECTION 5

### BASELINE DESIGN OPTIMIZATION

#### 5.1 SOLAR HVAC SYSTEM DESIGN TRADEOFFS

Performance simulations and economic analyses of the selected solar-assisted heating and cooling systems were performed for systems for single-family residences, multifamily residences and commercial buildings. The tradeoff studies began with a "baseline" system, and various parameters that are significant to both cost and performance were varied separately to determine their individual effects on system performance and economics. The effects of these parameter variations have been studied for Atlanta, Georgia. The magnitude of these effects would be different in other localities.

##### 5.1.1 Single-Family Residence (SFR)

The baseline solar system modeled in the digital simulation program is shown schematically in Figure 3-6. It consists of a collector array with piping headers on both sides of the collectors for inlet and outlet of the collector fluid, a collector with two glass covers and a single storage tank filled with water for sensible heat storage. The system features direct flow of collector fluid to the Rankine cycle boiler or the house heating coil.

The most important system variable which greatly affects both performance and economics is the collector area. The collector area for the SFR was varied from 480 ft<sup>2</sup> to 1260 ft<sup>2</sup>. Figure 5-1 shows the percent of solar energy supplied to the Atlanta SFR heating, cooling and hot water loads. The solar contribution of these loads varies from about 20 percent, 42 percent and 77 percent for 480 ft<sup>2</sup> of collector area to about 72 percent, 75 percent and 90 percent for 1260 ft<sup>2</sup> system, respectively. The baseline 756 ft<sup>2</sup> collector

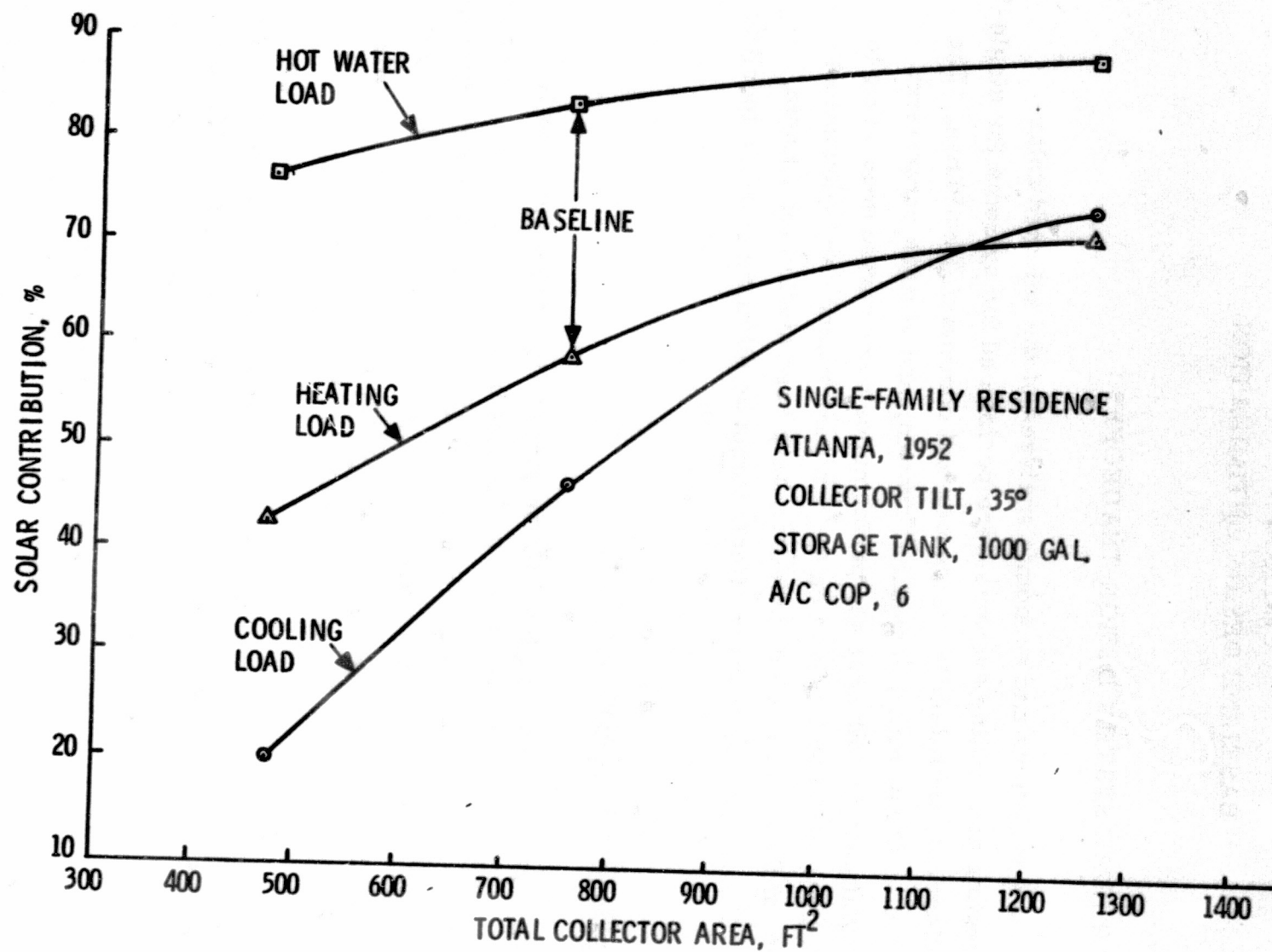


Figure 5-1. Solar Contribution versus Collector Area - SFR

area supplies 60 percent, 47 percent and 85 percent of the home's heating, cooling and hot water loads, respectively. These predictions were made for a system with a 1000-gallon storage tank.

It is obvious that the percent of energy supplied increases as the system collector area increases. However, at approximately  $1100 \text{ ft}^2$ , a break in the rate of relative contribution occurs. The solar contribution at this point to both heating and cooling is 70 percent. Figure 5-2 shows the additional auxiliary electric power required to support the solar cooling subsystem for the various collector areas.

Figure 5-2 also presents the impact of R/C design point thermodynamic efficiency on auxiliary cooling power. The baseline Rankine cycle design features a shaft output of 2.4 hp at an efficiency of 8 percent at  $190^\circ\text{F}$ . A decrease in this cycle efficiency increases the thermal energy input requirements to the R/C boiler. Figure 5-3 presents the impact of R/C efficiency on a solar contribution to the SFR cooling load. The auxiliary electric cooling power logically increases as  $\eta_{\text{RC}}$  decreases; however, an equally significant result is that the solar contribution decreases rapidly.

Both Figures 5-2 and 5-3 may be used to determine whether it's more cost effective to increase overall cooling system performance through improvements in R/C parameters (heat exchanger effectiveness, regenerators, flow rate, etc.) or by merely adding collector area. For example, Figure 6-57 in Section 6 indicates that  $\eta_{\text{RC}}$  may be improved from 6.5 percent to near 8 percent through the addition of a regenerator and an increase in heat exchanger effectiveness. However, this would cost about \$600.00. To achieve the same cooling system performance (i. e., auxiliary power consumption) for an  $\eta_{\text{RC}} = 6.5$  percent. Figure 5-2 indicates that an additional  $170 \text{ ft}^2$  of collectors would be required. However, these installed collectors would cost approximately  $170 \text{ ft}^2 \times \$17.85/\text{ft}^2 = \$3,034.00$  and thus it may be concluded that improvements in R/C parameters for cooling performance is more cost effective than adding collection area.



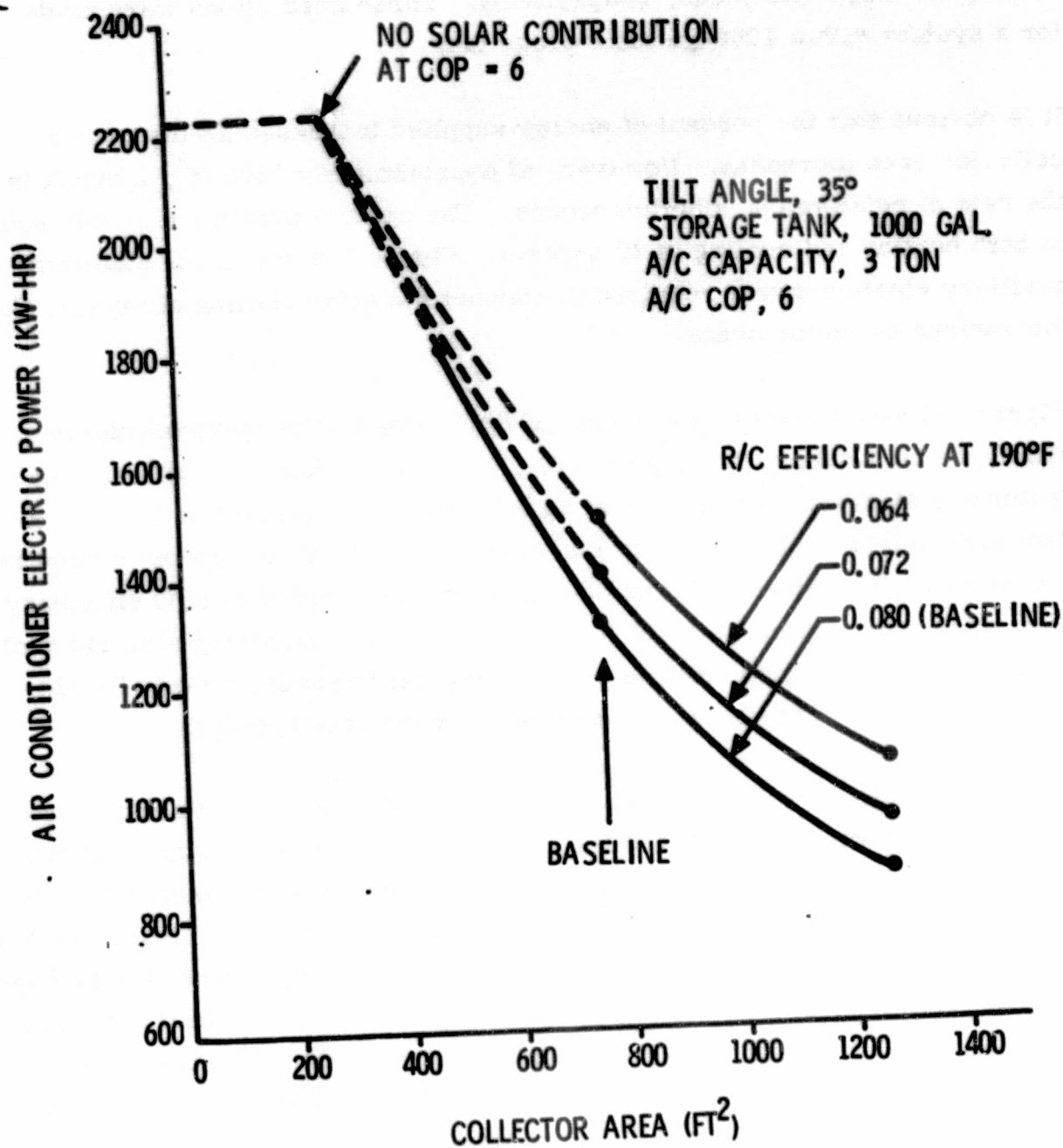


Figure 5-2. Cooling Auxiliary Power versus Collector Area - SFR

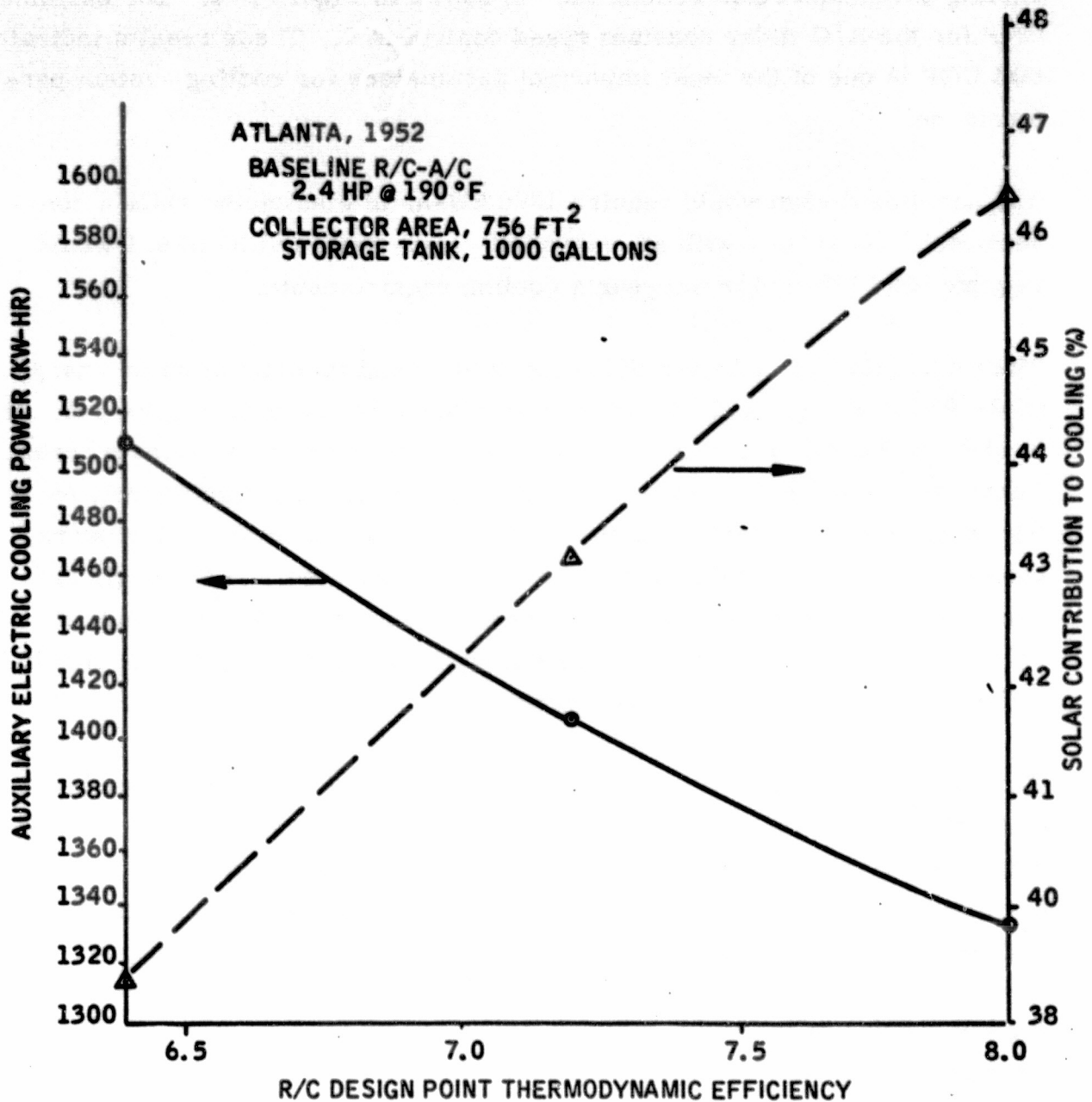


Figure 5-3. Cooling Subsystem Performance versus R/C Efficiency - SFR

The impact of the vapor compression air conditioner COP on auxiliary cooling power and solar contribution is shown in Figure 5-4. The baseline COP for the A/C under constant speed control is 6. These results indicate that COP is one of the most important parameters for cooling system performance.

The baseline design would require 1300 kW-hr of electricity while a conventional A/C system with an energy efficiency ratio (EER) of 6.1 would require 5926 kW-hr for the year's cooling requirements.

The importance of collector tilt angle on the contributions of solar energy to the SFR heating, cooling and hot water loads is shown in Figure 5-5. It is obvious that tilt angle is not as sensitive a parameter as collector area. Contributions to heating and hot water loads do not vary significantly for a tilt angle variation from 20 to 50 degrees. Cooling contribution appears near maximum at 20 degrees tilt. It must be repeated that tilt angle and relative solar contributions are site specific and that a different mix of relative solar contributions and sensitivity to tilt angle can be expected in different locations and different latitudes.

The baseline tilt angle for this study is 35 degrees (the approximate latitude of Atlanta). The collector tilt angle of 20 degrees is approximately the same angle as typical roof construction (4' in 12' or 18.4 deg) and thus from an architectural as well as optimized cooling contribution, a tilt angle of 20 degrees may be a logical choice.

Figure 5-6 shows the number of solar Btus supplied directly to the SFR heating, cooling and hot water loads and the total Btus collected for the year. At 35 degrees tilt, both the number of Btus collected and supplied directly to the loads is maximum. The energy difference between the Btus collected and those supplied directly to the loads is composed of the following:



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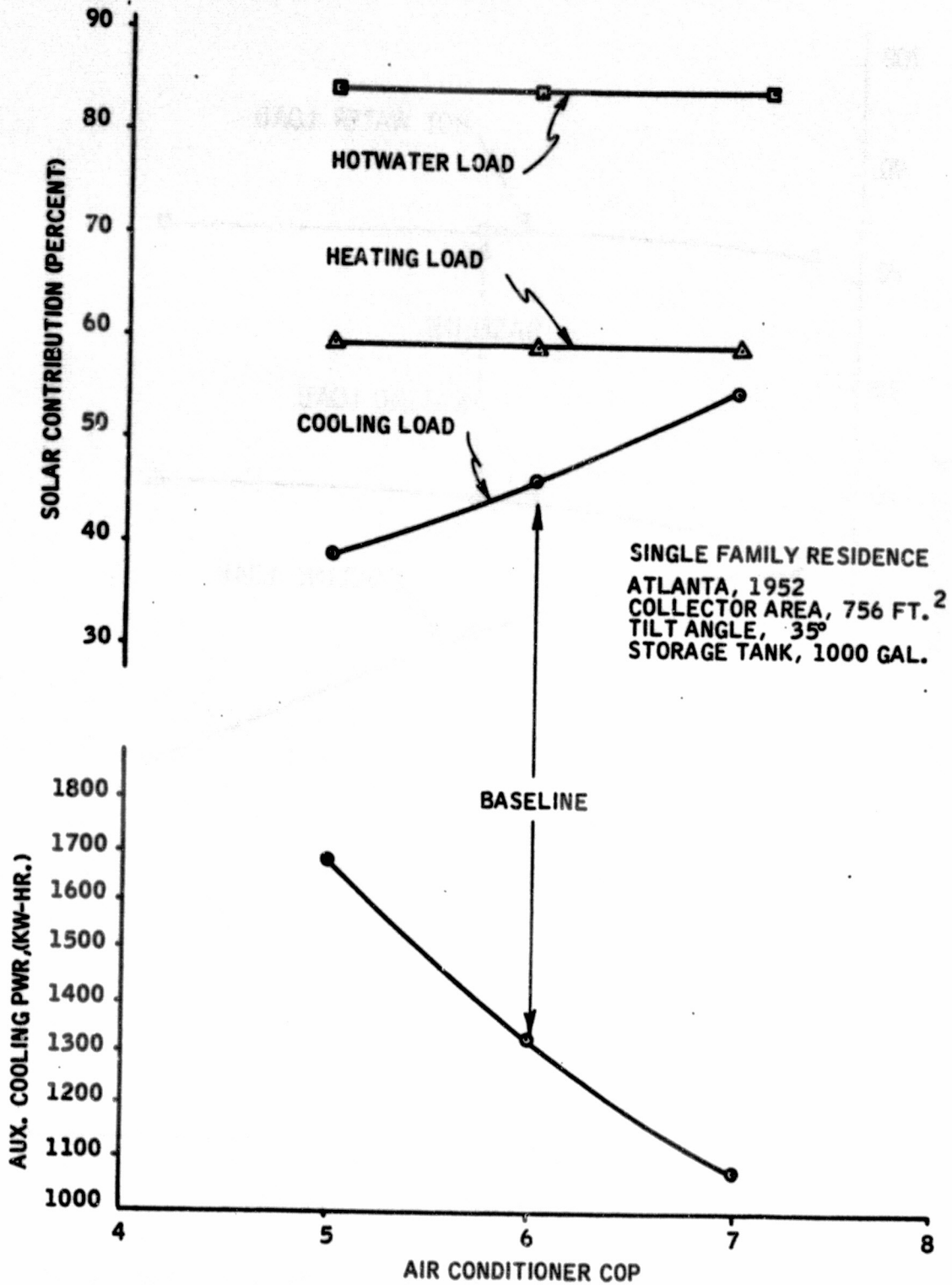


Figure 5-4. Cooling Subsystem Performance versus Air Conditioner COP - SFR

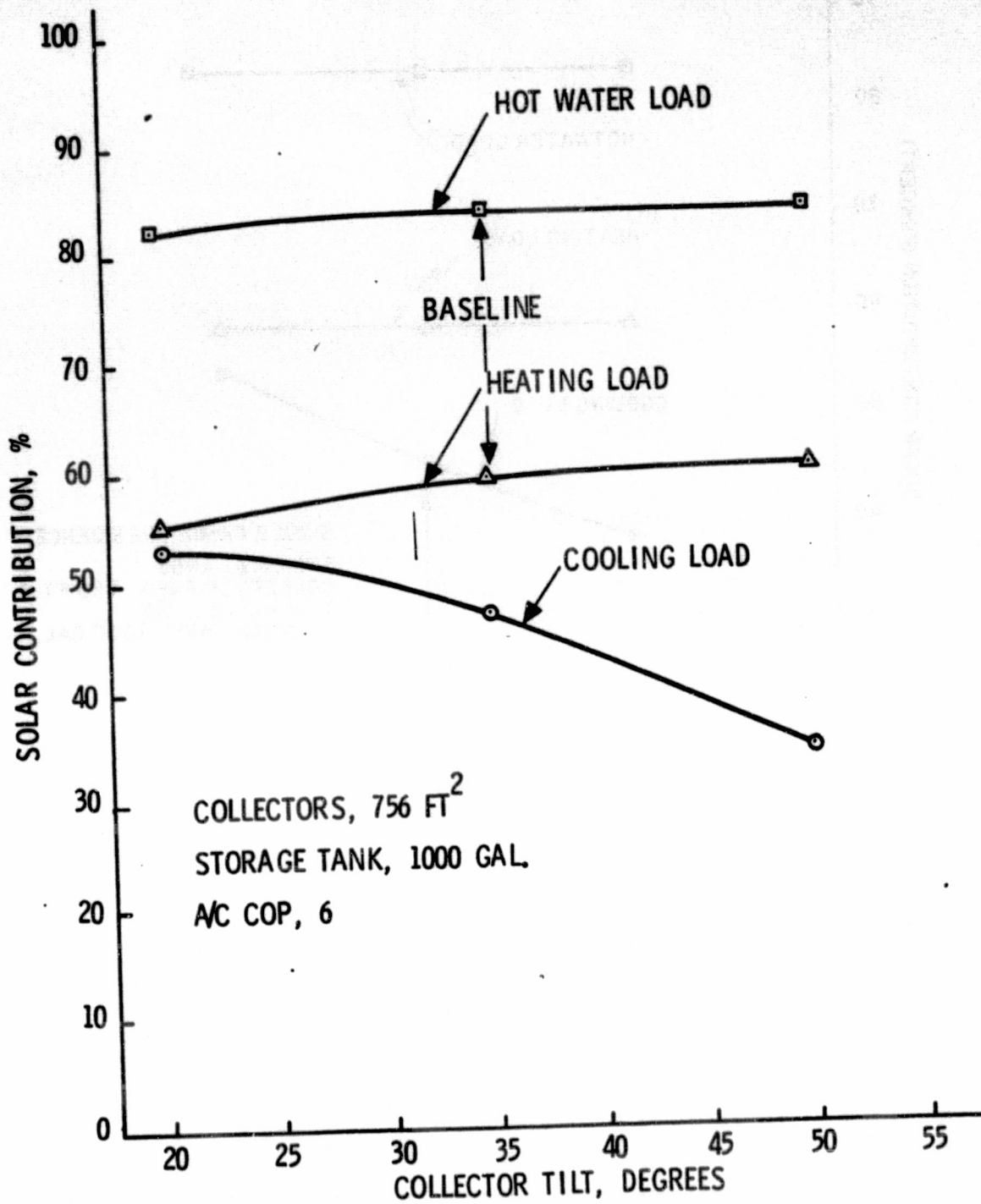


Figure 5-5. Solar Contribution versus Collector Tilt - SFR

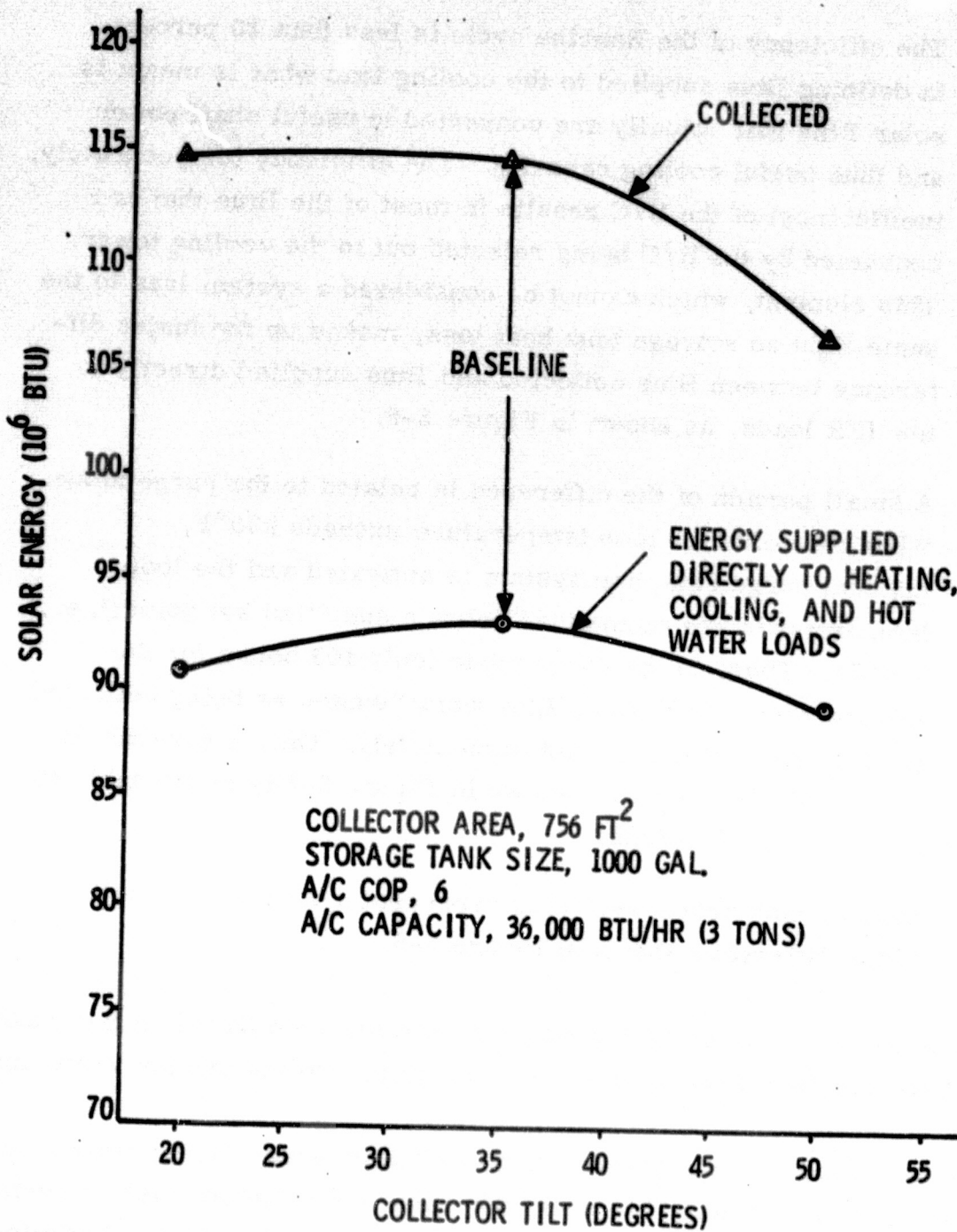


Figure 5-6. Solar Energy Supplied versus Collector Tilt - SFR

- a) The efficiency of the Rankine cycle is less than 10 percent. In defining Btus supplied to the cooling load what is meant is solar Btus that actually are converted to useful shaft power and thus useful cooling capacity. The efficiency (or accurately, inefficiency) of the R/C results in most of the Btus that are consumed by the R/C being rejected out to the cooling tower. This element, which cannot be considered a system loss in the same light as storage tank heat loss, makes up the major difference between Btus collected and Btus supplied directly to the SFR loads, as shown in Figure 5-6.
- b) A small portion of the difference is related to the purge mode. When the collector loop temperature exceeds 230°F, the heat purge rejection system is activated and the loop temperatures are maintained below a specified set point (i. e., 220°F). Thus, in the purge mode (only 106 hours for the Atlanta year, 1952) solar Btus were counted as being collected but they were also rejected immediately. Only a very small portion of the difference shown in Figure 5-6 is attributable to these purge conditions.
- c) Storage tank heat losses also represent a very small portion of the difference shown in Figure 5-6.

The baseline tilt angle of 35 degrees has been selected based on the maximum solar Btus supplied directly to the SFR heating, cooling and hot water loads.

The performance of the system with the collector array facing away from south was not simulated. It is well known that the optimum system performance is achieved with collectors facing due south. Previous calculations have shown that variations of 30 deg east or west reduce performance about 3 percent.



The performance of a 756 ft<sup>2</sup> solar HVAC system with different size storage tanks is shown in Figure 5-7. The percent of solar energy supplied to the heating load varies from about 52 percent for a 500-gallon tank to approximately 62 percent for a 1500-gallon tank. A 1000-gallon storage tank was selected as the baseline size based on the reasonable mix of solar contributions shown in Figure 5-7 and on the near minimum in auxiliary electric power required for the SFR cooling load.

Heat losses from the solar HVAC storage tank were neglected for these system tradeoff analyses. However, storage tank heat losses were included in the baseline system performance analysis discussed subsequently in subsection 5.2. Analysis performed previously showed that a tank UA of 9.44 (4-inch thick of fiberglass insulation), the solar heating system would provide about 2 percent less energy over the year than if heat losses were completely eliminated. If the storage tank is located inside the home, the heat loss would decrease the home's heat load and the net effect would be the same as a perfectly insulated storage tank. Of course, storage tank heat loss would increase the home's cooling load if the tank were located inside the home. Thus, tank losses were neglected for these tradeoff analyses so as to not influence the tradeoff conclusions.

The baseline solar HVAC system for the SFR has a heat exchanger (Hx) between the solar collector glycol loop and the hot water storage tank subsystem. The effectiveness of this heat exchanger on system performance is shown in Figure 5-8. The baseline Hx effectiveness was assumed to be 0.55, a reasonable level of performance for a glycol/water exchange condition for the flow rates defined for the baseline solar system. Figure 5-8 indicates that solar contributions to the house heating, cooling and hot water loads are not strongly influenced by this heat exchanger's effectiveness. Note that this result may offer an area for system cost reduction since annual electric power for cooling is only affected by about 3 percent.

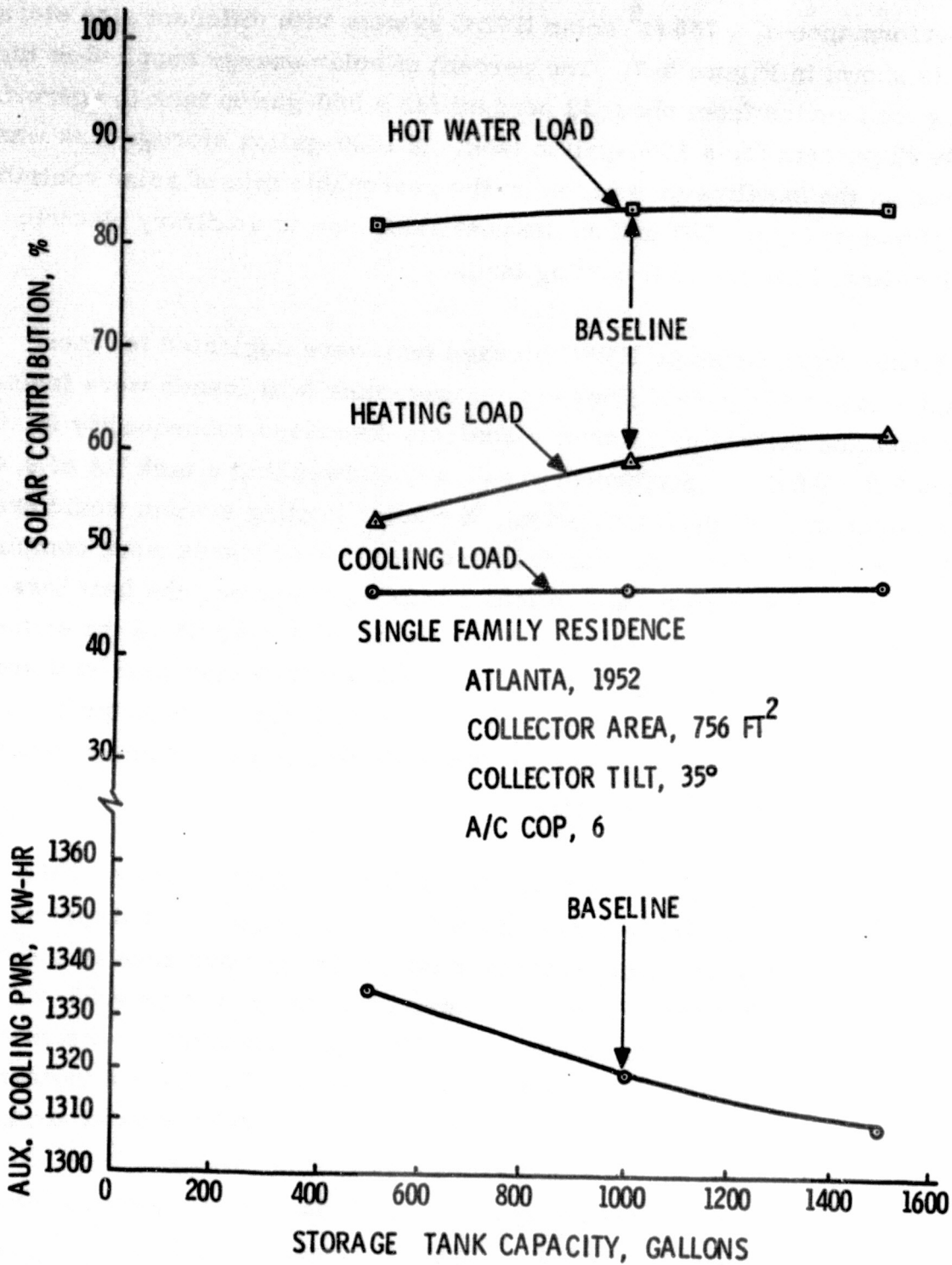


Figure 5-7. Subsystem Performance versus Storage Tank Capacity - SFR

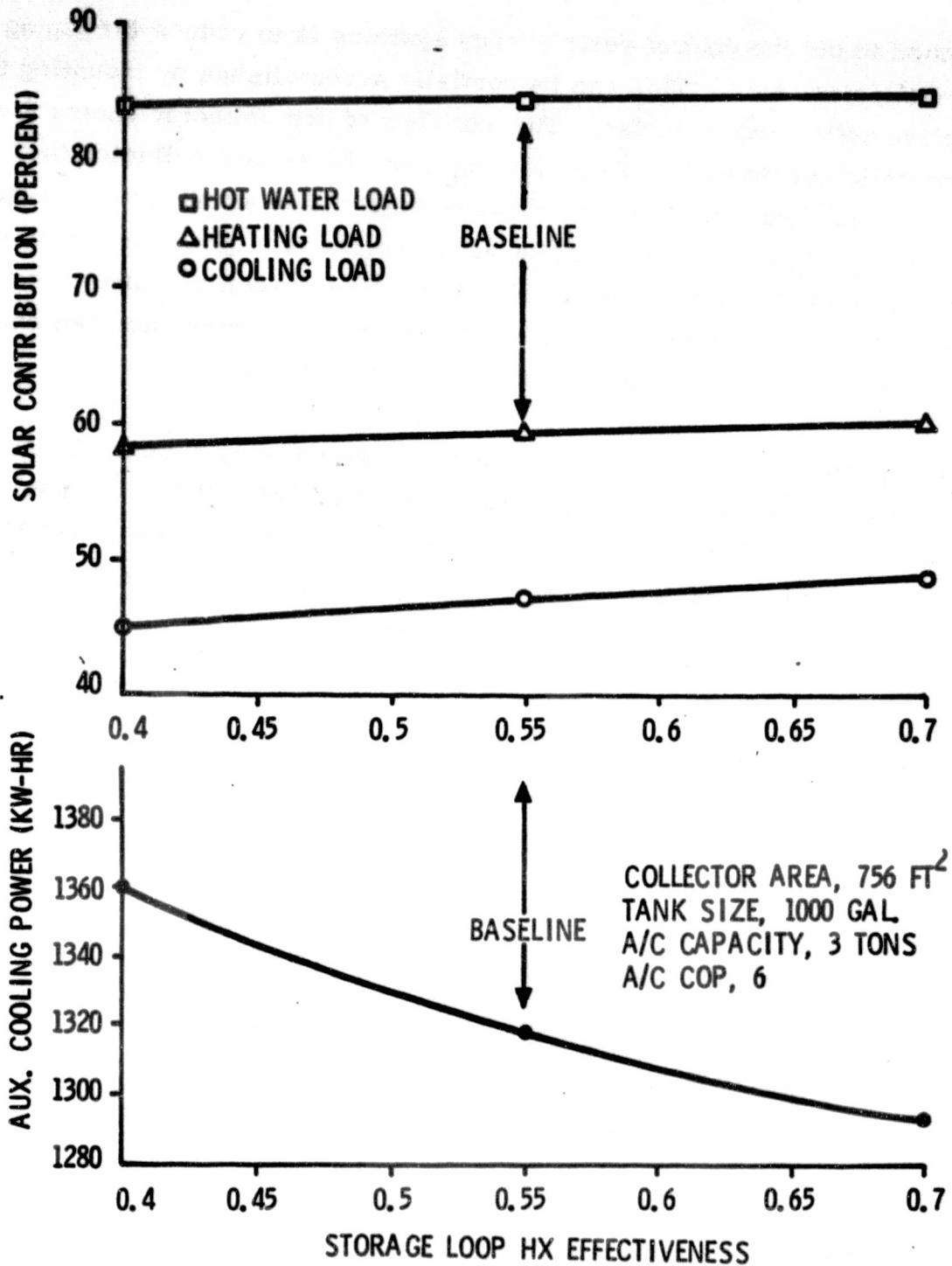


Figure 5-8. Subsystem Performance versus Heat Exchanger Effectiveness - SFR

A method to cut the cost of solar energy systems is to reduce the piping and solar collector costs. This can be partially accomplished by mounting two flat-plate collectors in series. The exit flow of one collector enters the second collector directly. Fluid exiting from the second collector then goes into the piping header. The only disadvantage of this arrangement is that the performance of the second collector in series is slightly degraded because the fluid inlet temperature is higher. The system performance of a solar HVAC system with two collectors arranged in series has been shown in the past to be only degraded by a fraction of a percent.

In conclusion, a system analyses has been performed to determine the effect of variations in subsystem parameters on overall solar HVAC system performance. A baseline system for a single-family residence was established from these tradeoff studies and the thermal performance and economic analysis of this system are discussed subsequently in subsection 5.2.1. This baseline system features the following parameters for the Atlanta, Georgia SFR.

• Collector tilt angle	35 deg
• Number of collectors	42
• Total collector area	756 ft <sup>2</sup>
• Storage tank capacity	1000 gallons
• Storage tank heat exchanger effectiveness	0.55
• Air conditioner capacity	3 tons
• Air conditioner COP	6
• Heating system capacity	60,000 Btu/hr

#### 5.1.2 Multifamily Residence (MFR)

The baseline solar system simulated by the computer program was discussed previously in Section 3 and is shown in Figure 3-8. The system has the same



important features as the Atlanta SFR System. The results of the SFR system tradeoff analyses indicated that a collector tilt angle of 35 deg was an optimum selection. Thus, the two major system variables that were analyzed for the MFR were collector area and storage tank size.

Figure 5-9 presents the effect of collector area variation on solar HVAC performance. At the baseline area of 6300 ft<sup>2</sup>, the solar HVAC supplies 42 percent, 65 percent, and 84 percent of the building's cooling, heating and hot water loads, respectively. Increasing the area to 10,800 ft<sup>2</sup>, can raise the solar contribution to the same respective loads to 67.5 percent, 77 percent and 89.5 percent. The auxiliary electric power required for the water chiller subsystem is shown in Figure 5-10. Increasing the collector area from the baseline of 6300 ft<sup>2</sup> to 10,800 ft<sup>2</sup> will result in a reduction of the annual cooling electric power from 12,700 kW-hr to 8600 kW-hr.

The effect of storage tank capacity on system performance is shown in Figure 5-11. Again, as in the SFR system, storage tank size does not strongly influence overall system performance parameters.

The baseline MFR annual thermal performance and economic analysis is discussed subsequently in subsection 5.2.2. This baseline features the following system parameters for the Atlanta MFR.

• Collector tilt angle	35 deg
• Number of collectors	350
• Total collector area	6300 ft <sup>2</sup>
• Storage tank capacity	8333 gallons
• Storage tank heat exchanger effectiveness	0.55
• Water chiller capacity	25 tons
• Chiller COP	6
• Heating system capacity	600,000 Btu/hr

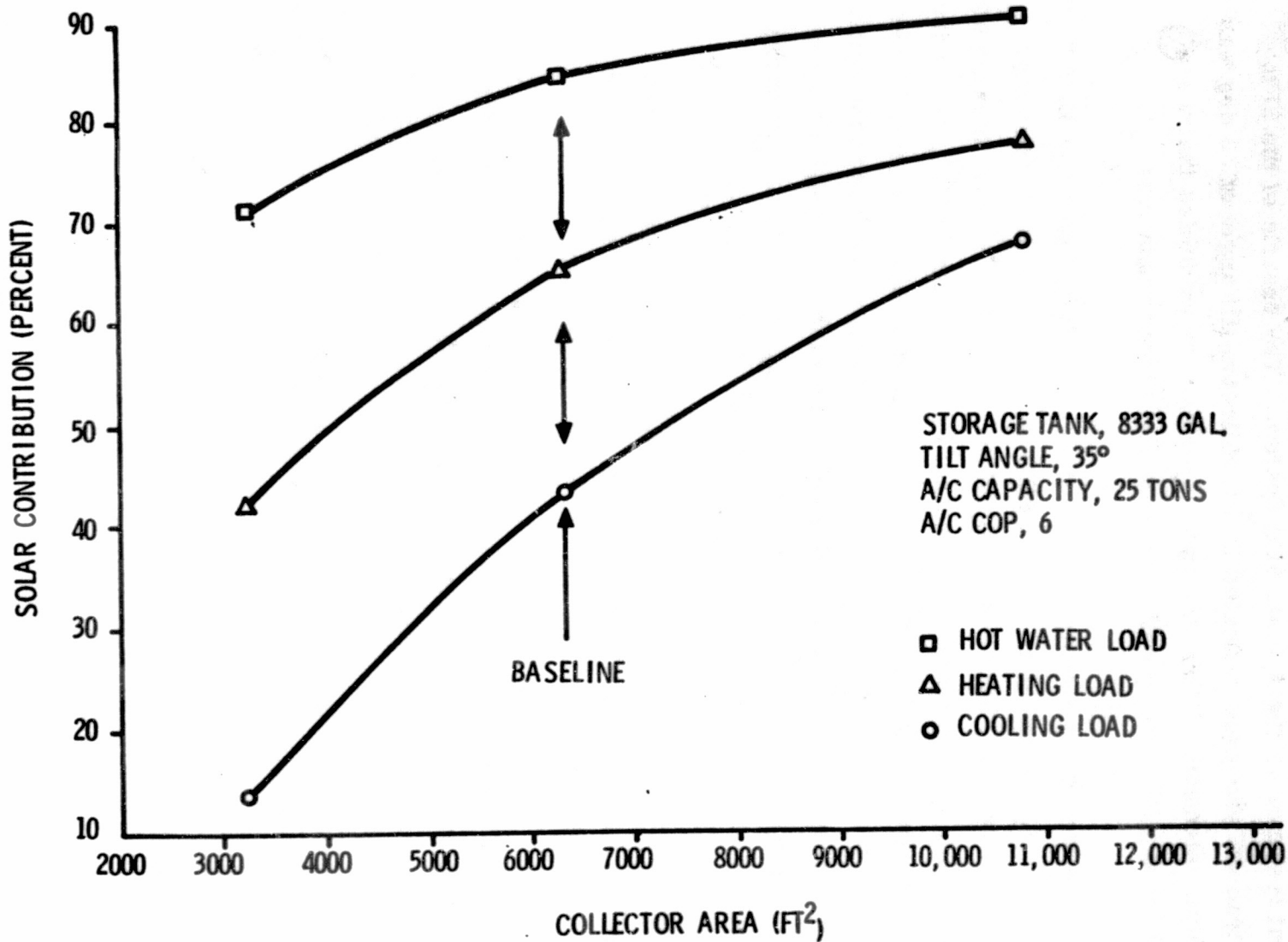


Figure 5-9. Solar Contribution versus Collector Area - MFR

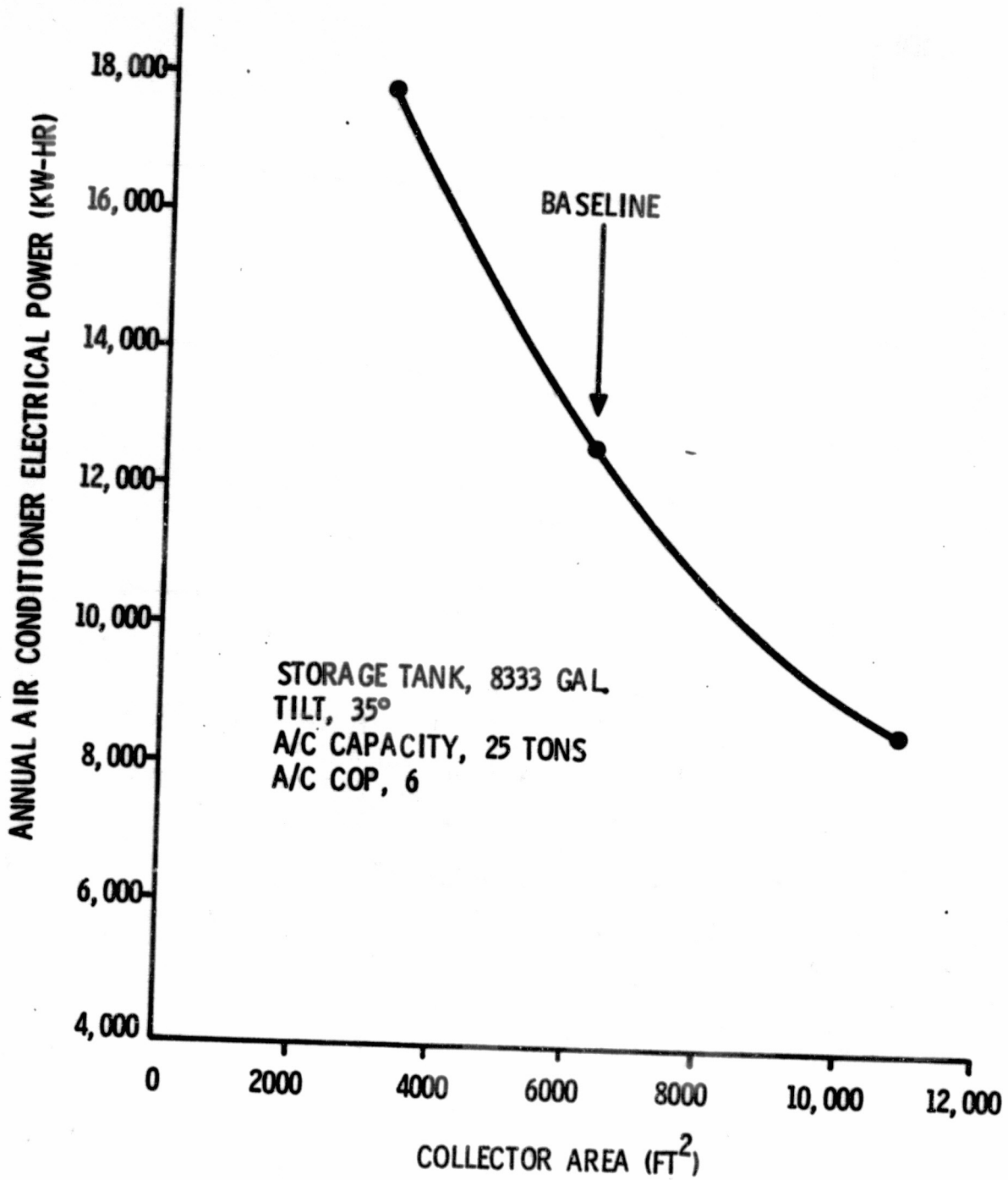


Figure 5-10. Cooling Auxiliary Power versus Collector Area - MFR

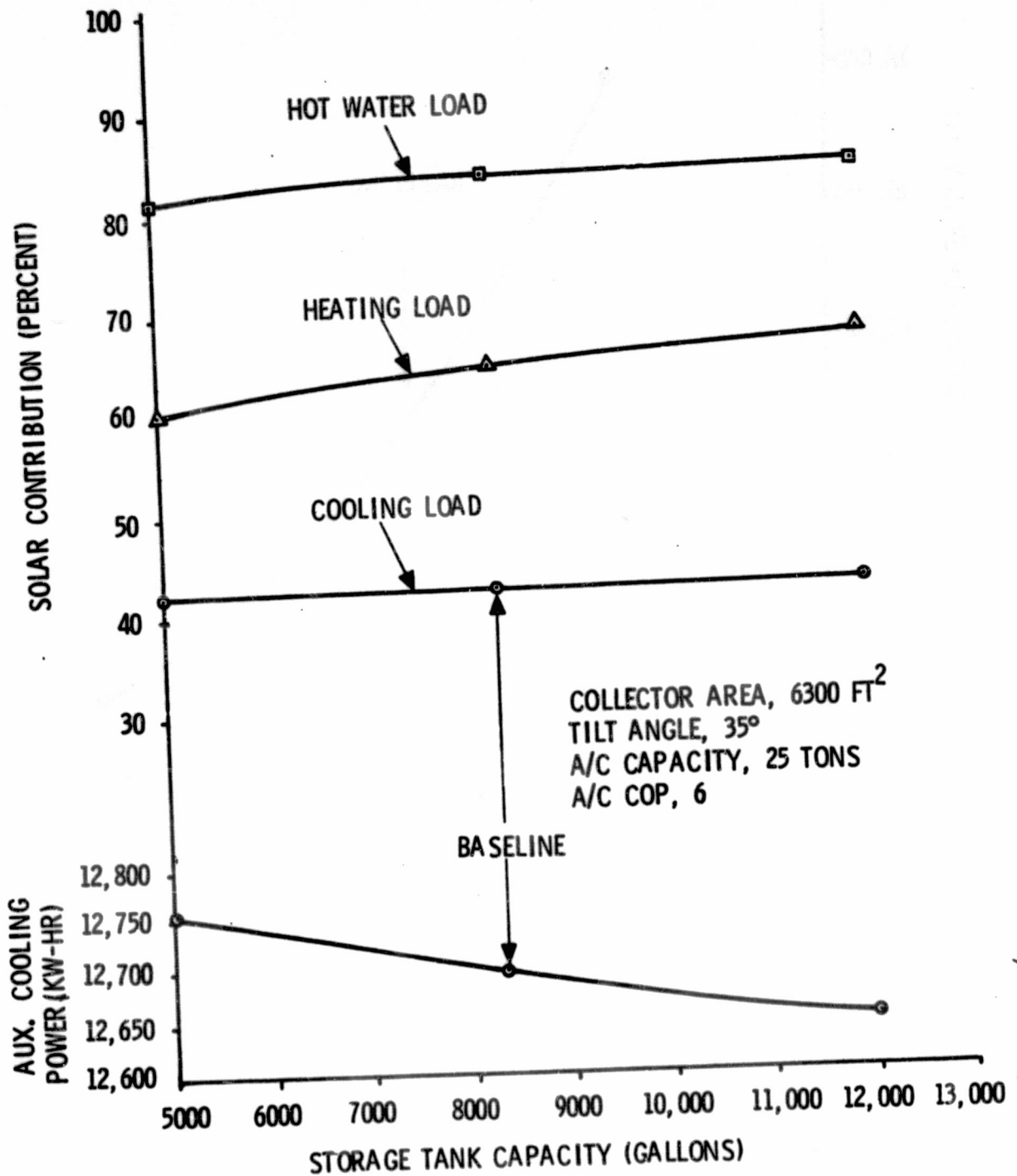


Figure 5-11. Subsystem Performance versus Storage Tank Capacity - MFR

### 5.1.3 Commercial Building

The commercial building model was discussed previously in Section 4. A baseline solar HVAC system was formulated for the commercial building model in Atlanta, Georgia, based on three 25-ton R/C-A/C subsystems performing the cooling task. The thermal performance of this large solar HVAC system is discussed in subsection 5.2.3. This system, which is merely an expansion of the MFR solar system, has the following system features:

• Collector tilt angle	35 deg
• Number of collectors	670
• Total collector area	12,060 ft <sup>2</sup>
• Storage tank capacity	15,600 gallons
• Water chiller capacity (three 25-ton R/C-A/C units)	75 tons
• Water chiller COP	6
• Heating system capacity	800,000 Btu/hr

## 5.2 BASELINE SYSTEM PERFORMANCE AND ECONOMIC ANALYSIS

### 5.2.1 Single-Family Residence System

The recommended solar assisted heating, cooling and hot water system for a single-family residence is a hydronic-to-warm air heating system, a Rankine cycle-vapor compression air conditioner operated at constant speed, and a domestic hot water preheat system. The baseline system consists of the major components shown in Figure 3-6 and listed in Table 5-1.

The performance of this solar HVAC system for an Atlanta, Georgia, house (2010 ft<sup>2</sup>) is shown in Figures 5-12 through 5-14. The space heating load

Table 5-1. Single-Family Residence (2010 ft<sup>2</sup>) Atlanta, Georgia

## Baseline Solar HVAC System Design:

## a) Cooling Subsystem

3-ton air conditioner  
 COP = 6  
 Constant speed control  
 Rankine cycle design point  
 2.4 hp at 190°F  
 $\eta_{R/C}$  at 190°F = 8 percent  
 Control range, 150°F - 200°F

## b) Heating Subsystem

Capacity, 60,000 Btu/hr  
 Hx coil effectiveness = 0.6

## c) Storage Subsystem

Tank capacity, 1000 gallons  
 Storage Hx effectiveness, 0.55

## d) Solar Collector Subsystem

Area 756 ft<sup>2</sup> (total) 625 ft<sup>2</sup> (effective)  
 Number of collectors, 42  
 Flow rate, 12 gpm

## e) House Loads

UA = 894 Btu/hr°F  
 Infiltration, 268 cfm (day) 220 cfm (night)  
 Internal load, 2550 Btu/hr (day) 1350 Btu/hr (night)  
 Sun load on 80 ft<sup>2</sup> windows  
 Design day heating load, 51,835 Btu/hr based on  
 $T_{AM} = 23^{\circ}\text{F}$ ,  $T_{House} = 70^{\circ}\text{F}$   
 Design day cooling load, 33,971 Btu/hr (2.83 tons) based on  
 $T_{AM} = 92^{\circ}\text{F}$ ,  $T_{House} = 76^{\circ}\text{F}$

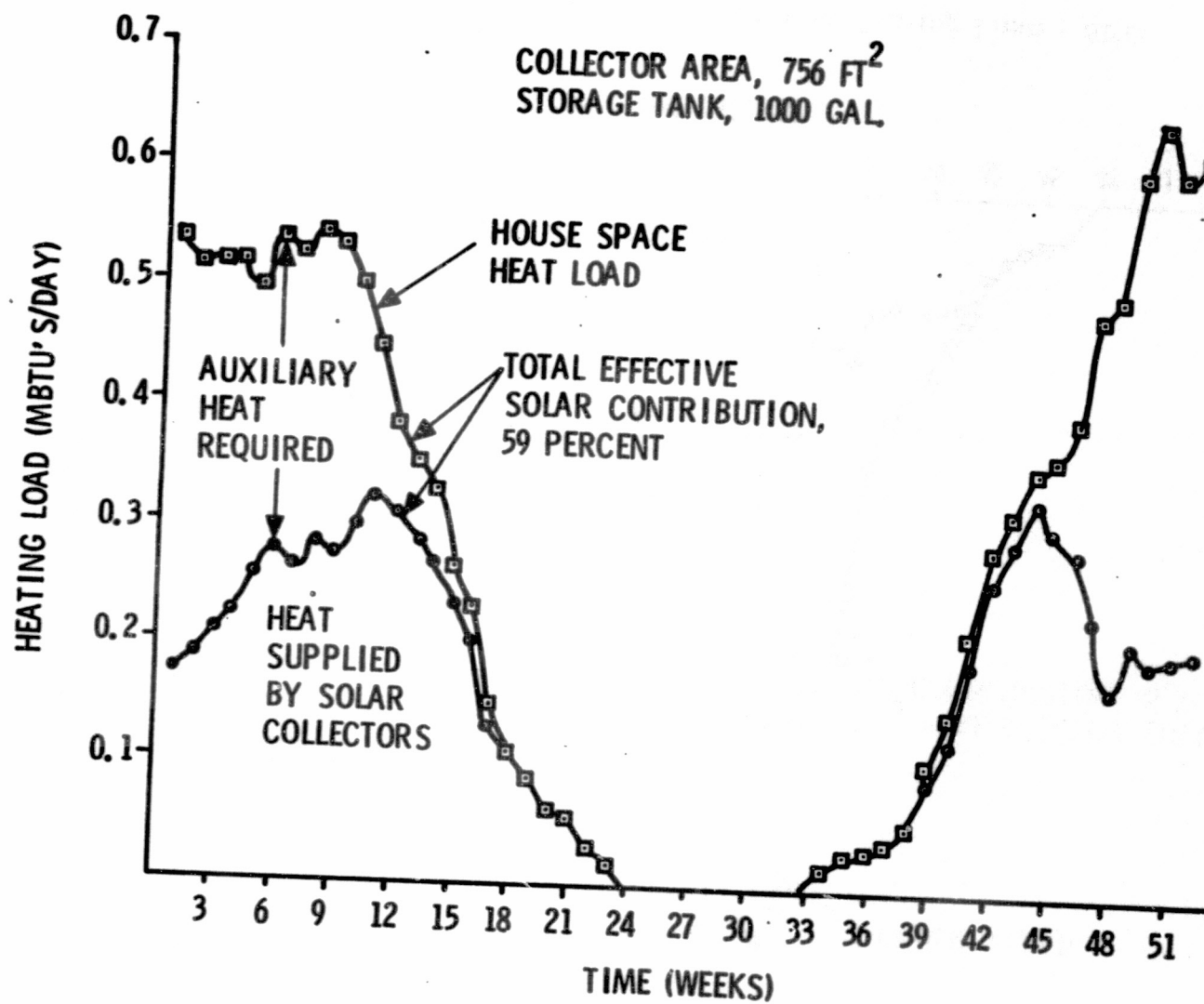


Figure 5-12. Weekly Solar Contribution to Heating Load - SFR

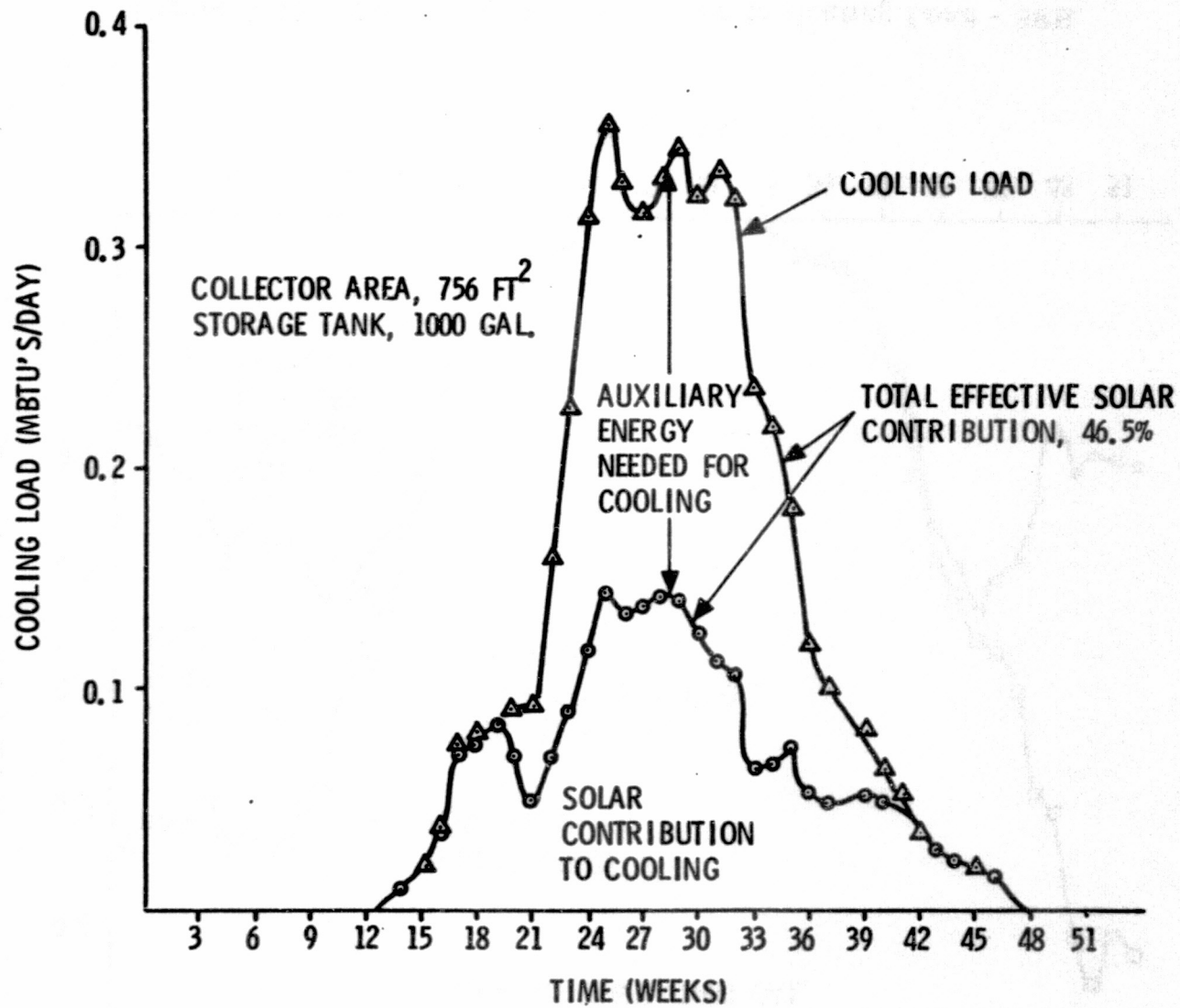


Figure 5-13. Weekly Solar Contribution to Cooling Load - SFR



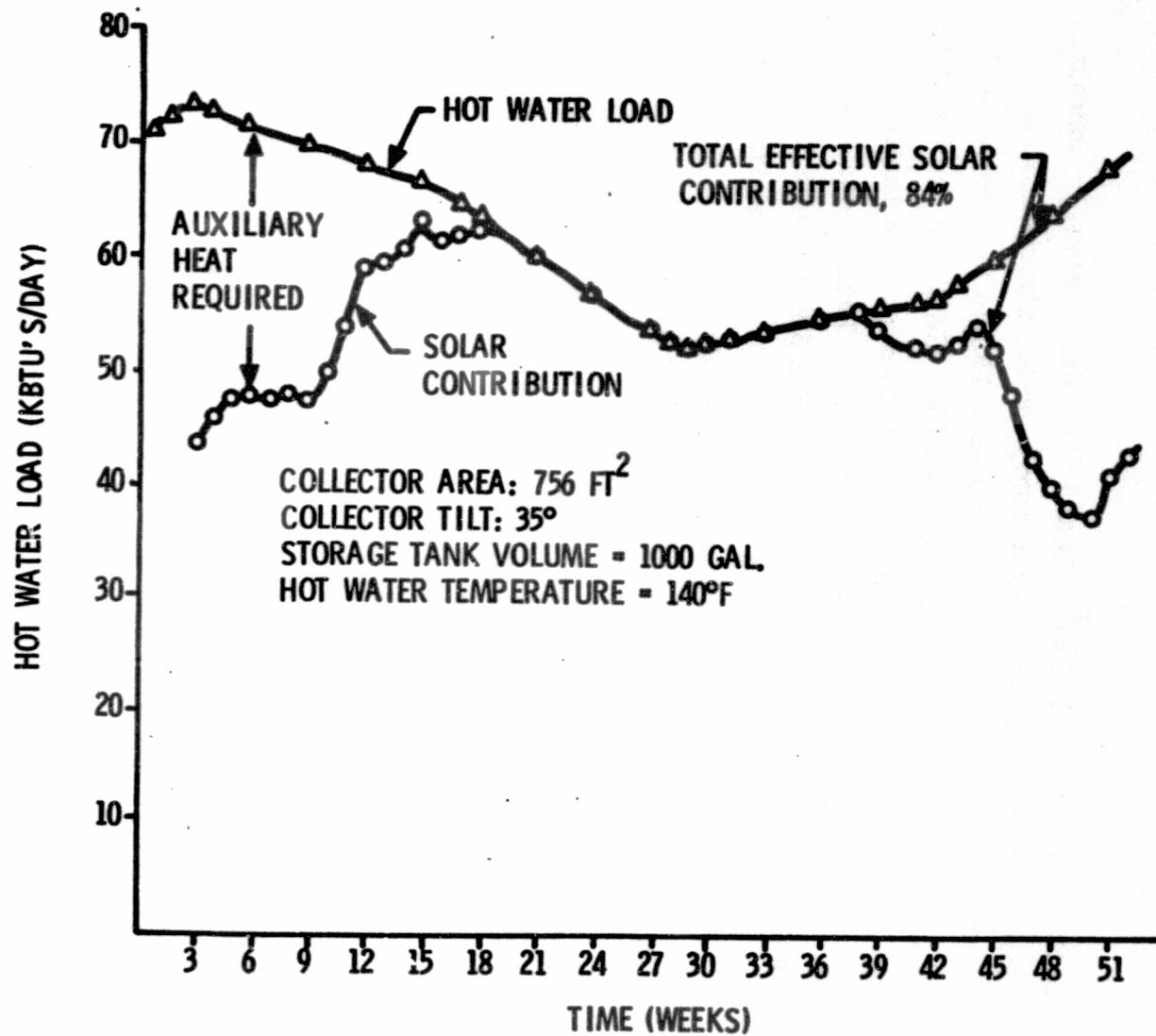


Figure 5-14. Weekly Solar Contribution to Hot Water Load - SFR

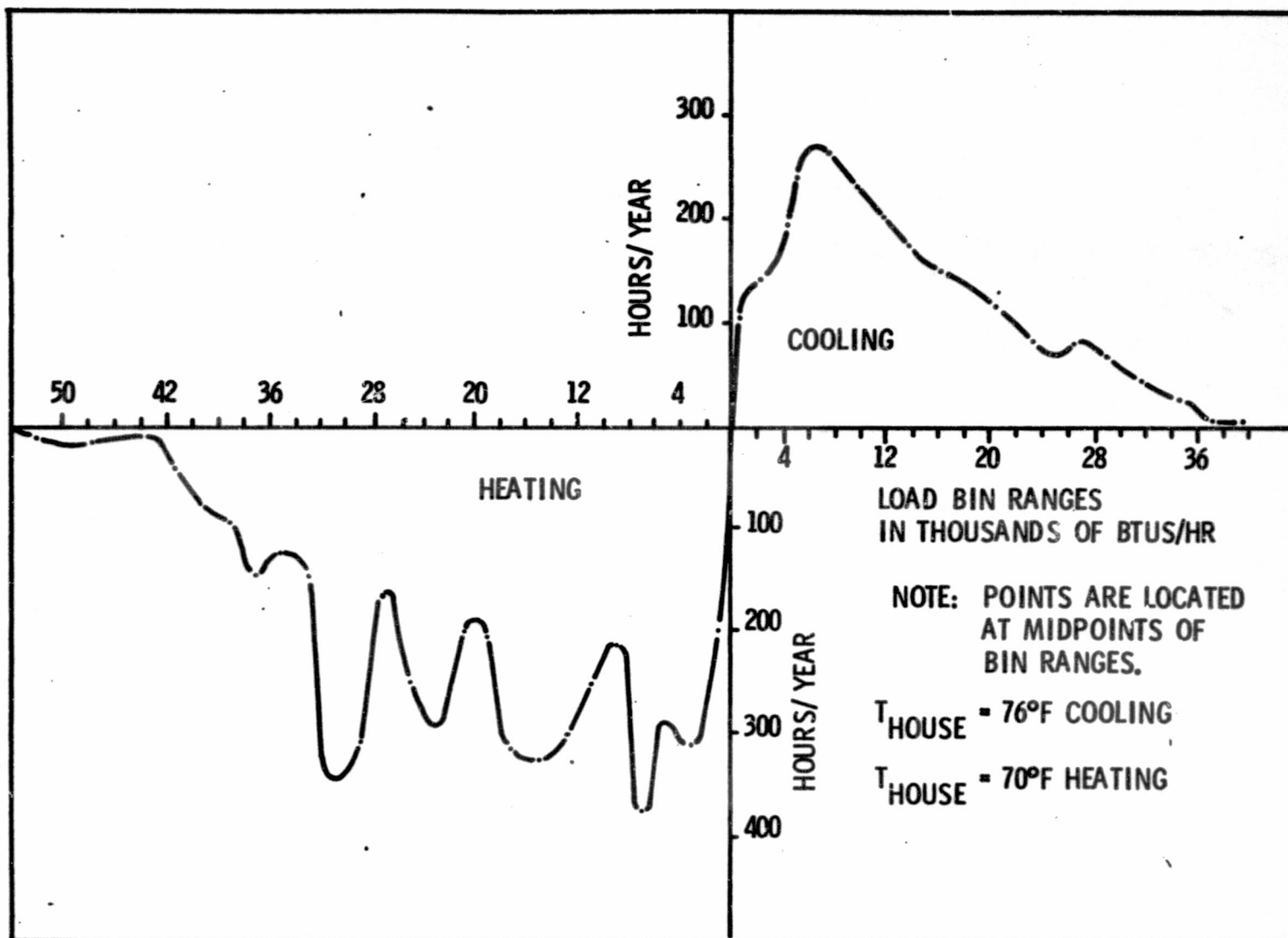
for the typical year, 1952, is  $96.43 \times 10^6$  Btu, of which  $56.95 \times 10^6$  Btu, or 59 percent, is supplied by solar energy. The yearly service hot water load is  $22.5 \times 10^6$  Btu of which  $18.89 \times 10^6$  or 84 percent is supplied by solar.

The annual cooling load is  $36.15 \times 10^6$  Btu and the solar system in combination with the Rankine cycle driving a vapor compression air conditioner supply  $16.8 \times 10^6$  Btu, or 46.5 percent. Figures 5-15 and 5-16 present the annual cooling and heating loads for the Atlanta SFR. Note that the Rankine cycle air conditioner operates most of the time (800 hours) at a house-cooling load of 4000 to 8000 Btu/hr, or about 0.5 ton. Total annual time that the air conditioner is on is 937 hours. Less than 5 hours occur at a house cooling load in excess of 3 tons (36,000 Btu/hr). An unusually high cooling load occurred in June, 1952. The heating loads shown in Figure 5-15 indicate less than 14 hours at a load in excess of 52,000 Btu/hr.

The operation of the pumps and fans for the solar HVAC system consumes 3805 kW-hr annually. This represents a cost of \$135.46 per year based on an electrical charge of \$0.356/kW-hr.

The following table presents the fuel costs in support of the solar HVAC system.

Heating System (required, $39 \times 10^6$ Btu)	
Electrical (\$0.0356/kW-hr)	\$412.13
Oil (\$0.40/gallon)	140.99
Gas (\$0.0017/ft <sup>3</sup> )	83.89
Cooling System (required, $19.35 \times 10^6$ Btu)	
Electrical (\$0.0356/kW-hr)	47.53
Hot Water System (required, $3.5 \times 10^6$ Btu)	
Electrical (\$0.0356/kW-hr)	37.75
Oil (\$0.40/gallon)	12.91
Gas (\$0.0017 ft <sup>3</sup> )	7.68



5-25

Figure 5-15. Cooling and Heating Load Histogram - SFR

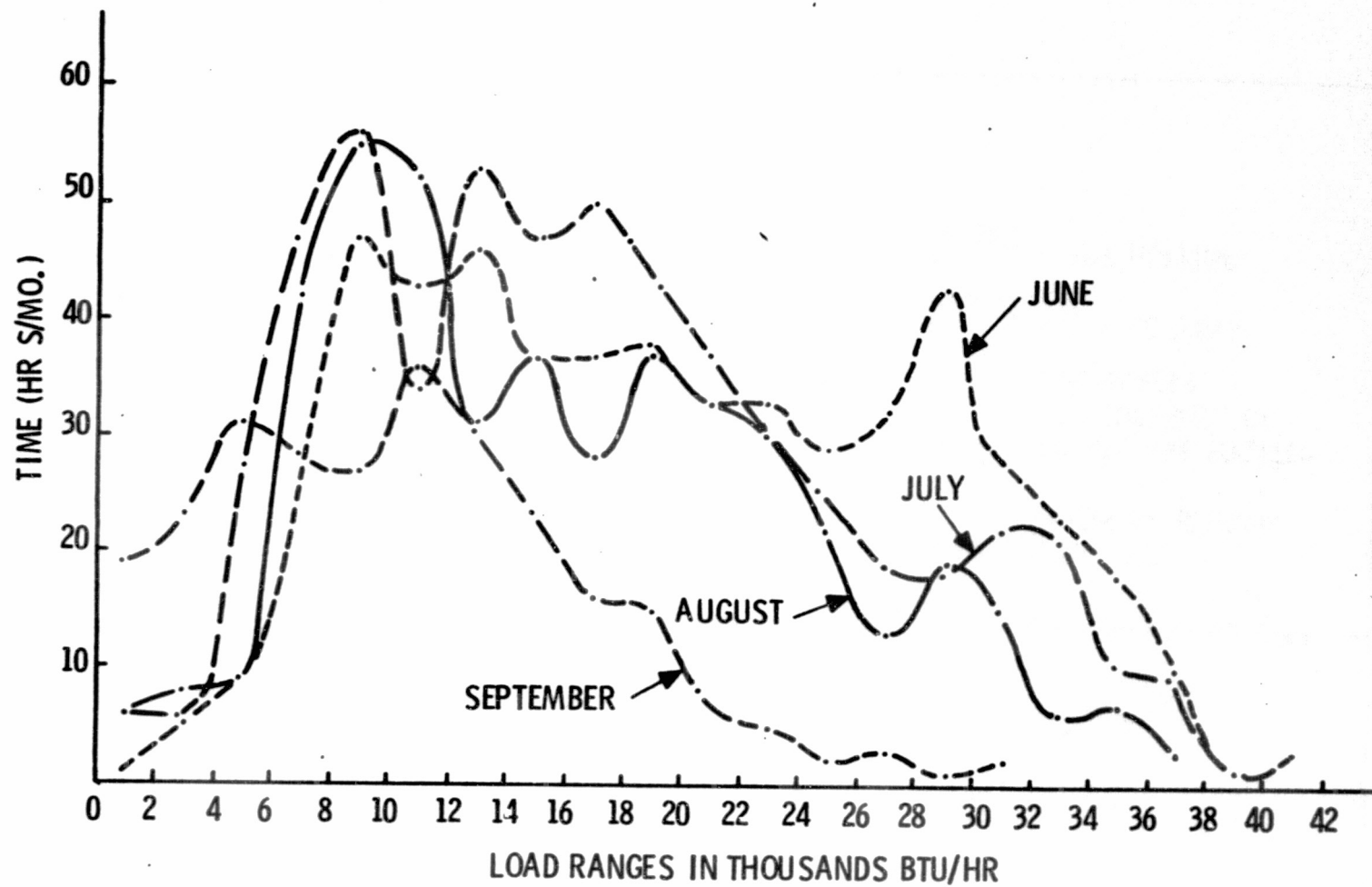


Figure 5-16. Cooling Load for SFR

The aforementioned fuel costs, plus the pumping and fan costs, represent the annual cost to the homeowner to heat, cool, and provide hot water to the home. An economic analysis has been made to determine the relative economic merits of the solar HVAC system in conserving conventional fuels and to determine the potential payback period for the capital investment associated with the solar system.

The estimated cost for just the baseline solar HVAC system is determined from the following equation.

$$\text{Cost (including installation)} = \$11,661 + \$17.85 (\text{collector area})$$

For the baseline collector area of  $756 \text{ ft}^2$ , the total system cost for the Atlanta SFR is \$25,155.60. This represents an annual cost of \$2,374.44 based on a 20-year loan at 7 percent interest.

Assuming that all the auxiliary heating, cooling and hot water systems are based on electric power\*, the total annual electric power requirement (including pumps and fans) in support of the solar HVAC system is \$632.87. If one adds this to the mortgage P&I and assumes that electrical power rates escalate at 10 percent/year over the next 20 years, the solid curve shown in Figure 5-17 can be constructed. This curve represents the future annual cost for the homeowner for the payoff of the solar HVAC system capital investment as well as the auxiliary electrical power costs to heat, cool and provide hot water.

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\* This assumption is not considered unrealistic due to the following factors:

- a) uncertainties in future gas reserves as well as gas rates make any analysis of this fuel source questionable
- b) oil rates are subject to similar uncertainties as well as international politics

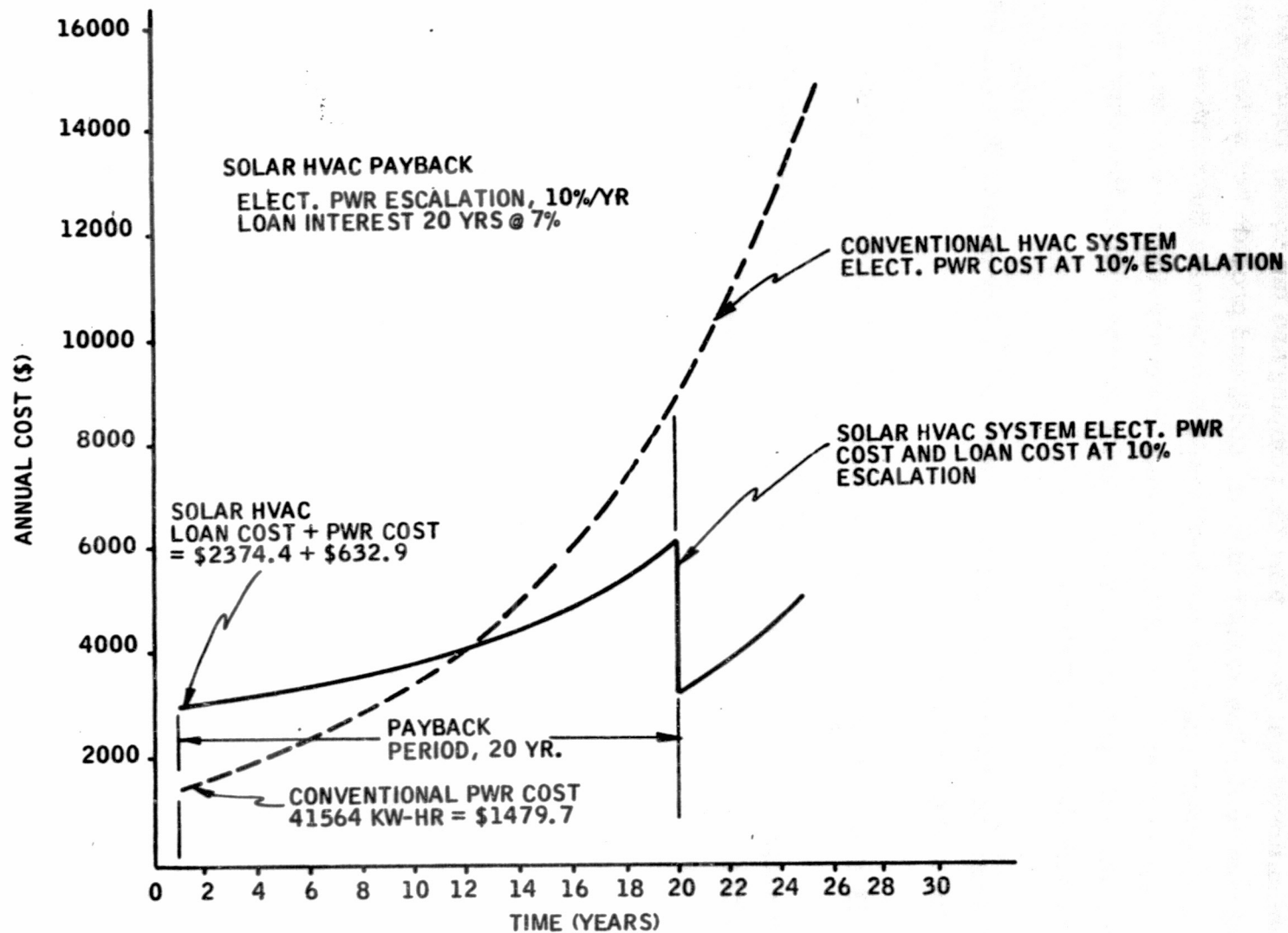


Figure 5-17. Solar HVAC Payback - SFR

The annual cost to meet the same total house loads with a conventional all-electric HVAC system is estimated below:

$$\text{Heating} = \frac{96.43 \times 10^6 \text{ Btu}}{3413 \text{ Btu/kW-hr}} = 28,253 \text{ kW-hr}$$

$$\text{Cooling} = \frac{36.15 \times 10^6 \text{ Btu}}{\text{EER}^* = 6.1} = 5.926 \text{ kW-hr}$$

$$\text{Hot Water} = \frac{22.51 \times 10^6 \text{ Btu}}{3413 \text{ Btu/kW-hr}} = 6,595 \text{ kW-hr}$$

\*EER (energy efficiency ratio) is estimated in ASHRAE 90-75 to be 6.1.

Total annual power requirement for the conventional HVAC system is 41,564 kW-hr or \$1,479.68. Now if one escalates this annual cost by 10 percent, the dashed curve in Figure 5-17 is obtained. The area under these curves over 20 years represents a total dollar outlay. It is obvious that solar system payback is just over 20 years based on a comparison of areas.

Now one could discount these results by not accepting the 10 percent escalation figure. However, let us assume that this analysis is merely an example to demonstrate the sensitivities of solar HVAC system economic analysis.

The curves indicate two very basic facts:

- a) Solar HVAC system economic payback within a 20-year period will require significant fuel escalation rates in the future.
- b) Reducing the initial system cost will also strongly reduce payback period.

As an example of the latter, consider that the cost of the baseline solar HVAC system for the SFR can be reduced as a result of mass production and cost reduction techniques.

Low-cost solar HVAC system for SFR --

R/C - A/C	\$3,000 each (assumed \$8,000 above)
Collector	\$6.00/ft <sup>2</sup> (assumed \$13.50/ft <sup>2</sup> above)
Collector Installation	\$4.35/ft <sup>2</sup>
Remaining System	\$3,661.00 (heat exchangers, storage, etc.)

The "low-cost" solar HVAC system would be \$14,485.60 which is \$1,367.30 per year on a 20-year, 7 percent loan. Assuming that this system obtains the same thermal performance merits as the baseline system, the yearly cost figures for this system as compared to a conventional HVAC system are shown in Figure 5-18. At an electric power escalation factor of 10 percent the payback period for the "low-cost" solar HVAC is about 11 years. Of equal significance is a total power cost savings of over \$19,000.00 in the 20-year period. At an electric power escalation rate of 5 percent, the payback period increases to about 19 years.

In conclusion, solar HVAC systems for single-family residences can be shown to be economically competitive with reasonable payback time periods only if the following trends occur in the future:

- Fuel costs must escalate from 5 to 10 percent per year.
- A solar HVAC system that exhibits acceptable thermal performance must be designed, built and installed at low cost.



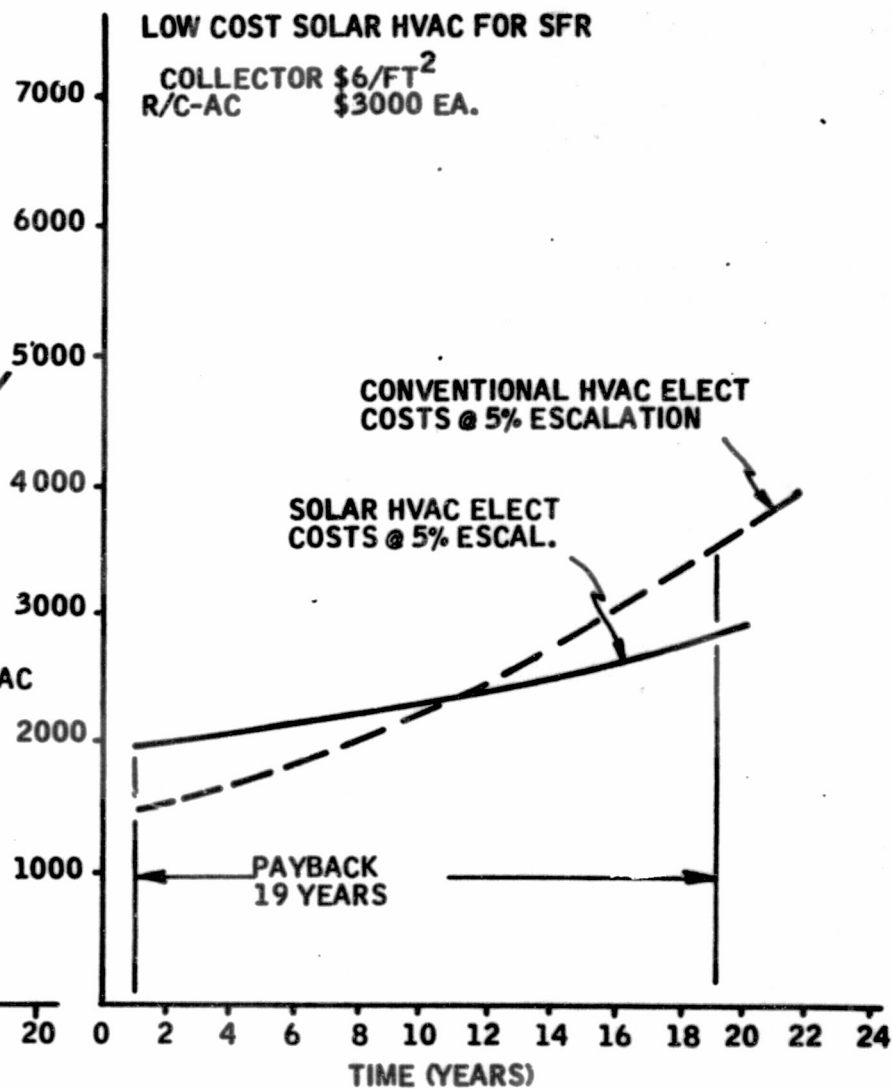
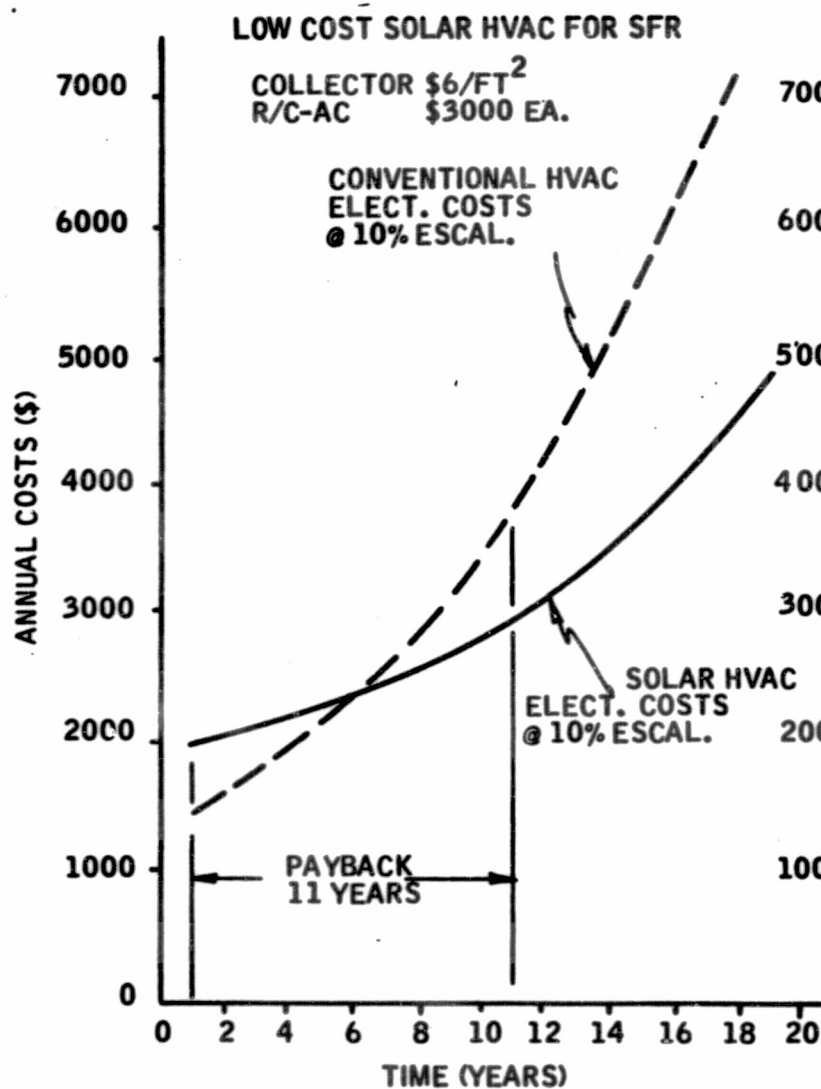


Figure 5-18. Low Cost Solar HVAC Payback - SFR

### 5.2.2 Multifamily Residence (MFR)

The recommended solar-assisted heating, cooling and hot water system for the multifamily residence is a hydronic-to-warm air heating system, a Rankine cycle driving a 25-ton water chiller which operates at constant speed, and a domestic hot water preheat system. The baseline solar HVAC system consists of the major components shown in Figure 3-8 and listed in Table 5-2.

Table 5-2. Multifamily Residence (9,600 ft<sup>2</sup>) Atlanta, Georgia

#### Baseline Solar HVAC System Design:

##### a) Cooling Subsystem

25-ton air conditioner  
COP = 6  
Constant speed control  
Rankine cycle design point  
20 hp at 190°F  
 $\eta_{RC}$  at 190°F = 8.6 percent  
Control range, 150°F - 200°F

##### b) Heating Subsystem

Capacity, 600,000 Btu/hr  
Hx coil effectiveness, 0.6

##### c) Storage Subsystem

Tank capacity, 8,333 gallons  
Storage Hx effectiveness, 0.55

##### d) Solar Collector Subsystem

Area 6300 ft<sup>2</sup> (total) 5227 ft<sup>2</sup> (effective)  
Number of collectors, 350  
Flow rate, 100 gpm

##### e) House Loads

UA = 4,416 Btu/hr°F  
Infiltration, 3072 cfm (day and night)  
Internal load, 25,500 Btu/hr (day) 13,500 Btu/hr (night)  
Sun load on 480 ft<sup>2</sup> windows  
Design day heating load, 349,987 Btu/hr based on  
 $T_{AM} = 23^{\circ}F$ ,  $T_{House} = 70^{\circ}F$   
Design day cooling load, 276,558 Btu/hr (23 tons) based on  
 $T_{AM} = 92^{\circ}F$ ,  $T_{House} = 76^{\circ}F$

The thermal performance of this system for an Atlanta, Georgia, MFR is shown in Figures 5-19 through 5-21. Solar contributions to the three loads are summarized below.

	<u>Annual Btu</u>	<u>Solar Supplied Btu</u>	<u>Solar Contribution</u>
Heating Load	$624.5 \times 10^6$	$407.0 \times 10^6$	65%
Cooling Load	$325.0 \times 10^6$	$139.9 \times 10^6$	43%
Hot Water Load	$283.2 \times 10^6$	$238.3 \times 10^6$	84%

The annual heating and cooling load distributions for the Atlanta MFR are shown in Figure 5-22. The 25-ton Rankine cycle water chiller operates most of the time at 130,000 to 150,000 Btu/hr, or about 12 tons. The total annual time that the air conditioner operates is 1009 hours, with only 3 hours at a load over 300,000 Btu/hr, the 25-ton capacity of the cooling system.

The annual power consumption of the pumps and fans for this solar HVAC system is 42,678 kW-hr. This represents an annual cost of \$1,519.34 based on \$0.0356/kW-hr. The auxiliary fuel costs in support of the solar HVAC system are listed below:

Heating System ( $217.5 \times 10^6$ Btu)	
Electrical (\$0.0356/kW-hr)	\$2,271.11
Oil (\$0.40/gallon)	776.93
Gas (\$0.0017/ft <sup>3</sup> )	462.28
Cooling System ( $185.1 \times 10^6$ Btu)	
Electrical (\$0.0356/kW-hr)	452.00
Hot Water ( $44.9 \times 10^6$ Btu)	
Electrical (\$0.0356/kW-hr)	468.33
Oil (\$0.40/gallon)	160.62
Gas (\$0.0017/ft <sup>3</sup> )	95.33

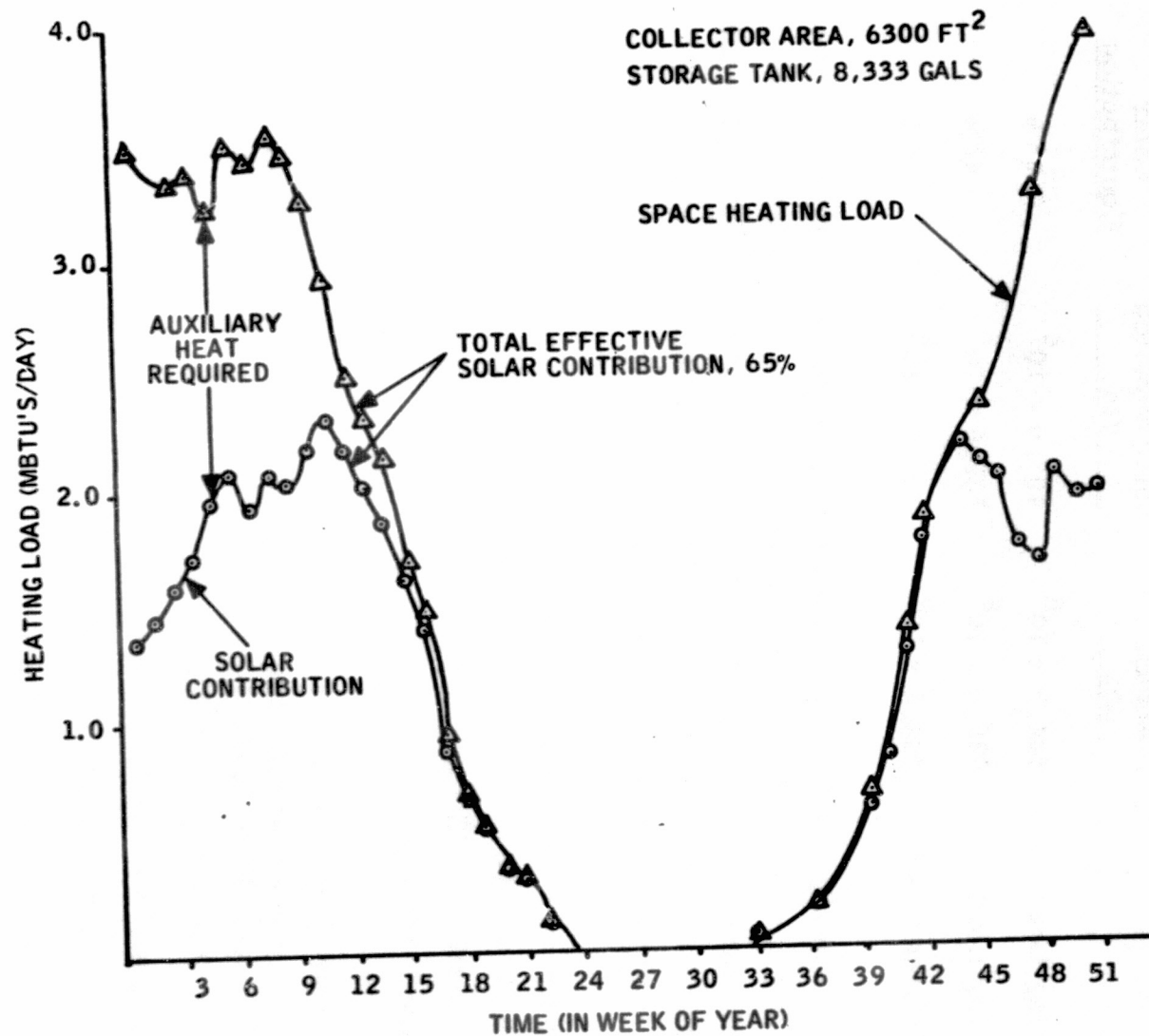


Figure 5-19 Weekly Solar Contribution to Heating Load - MFR

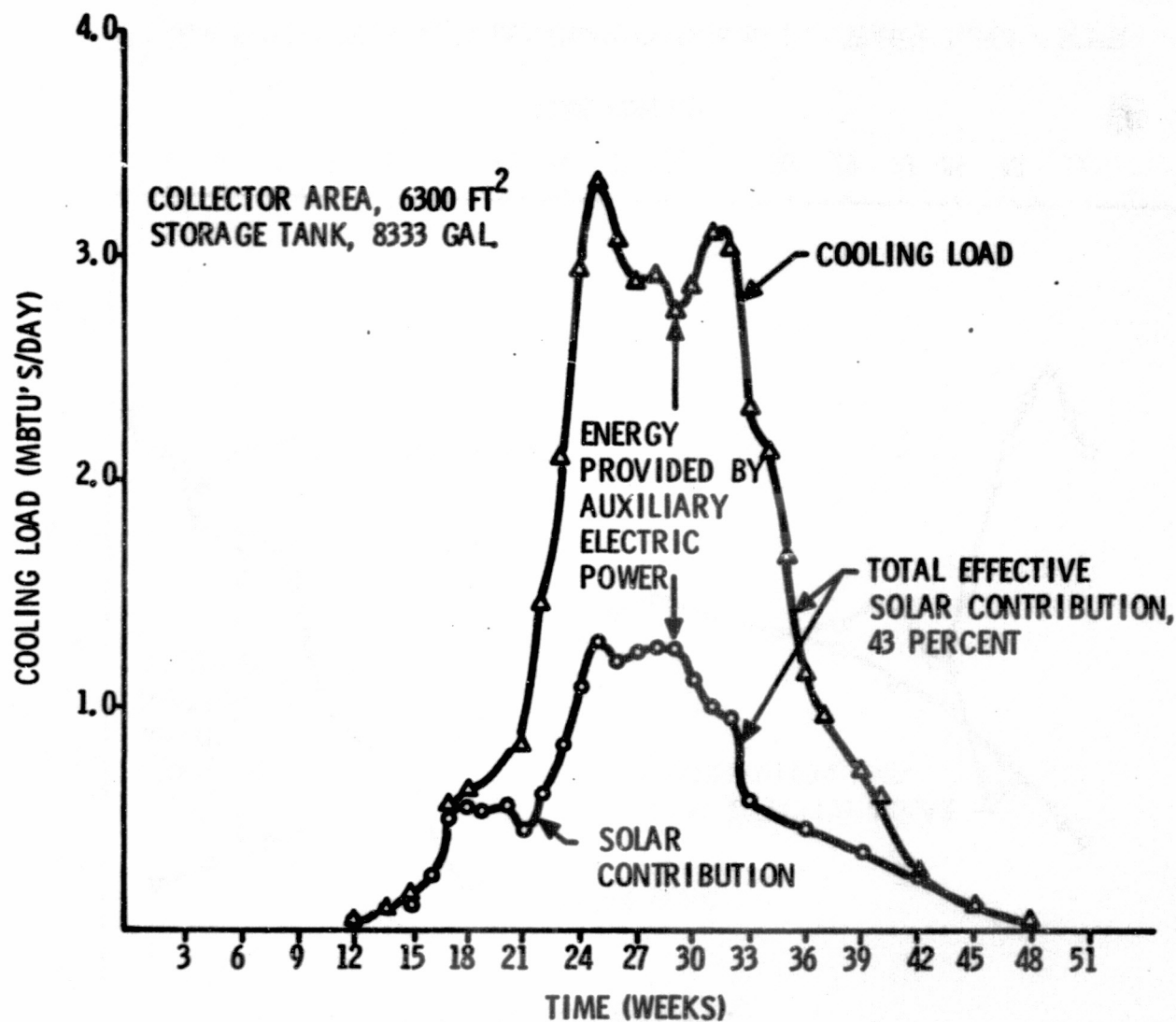


Figure 5-20. Weekly Solar Contribution to Cooling Load - MFR

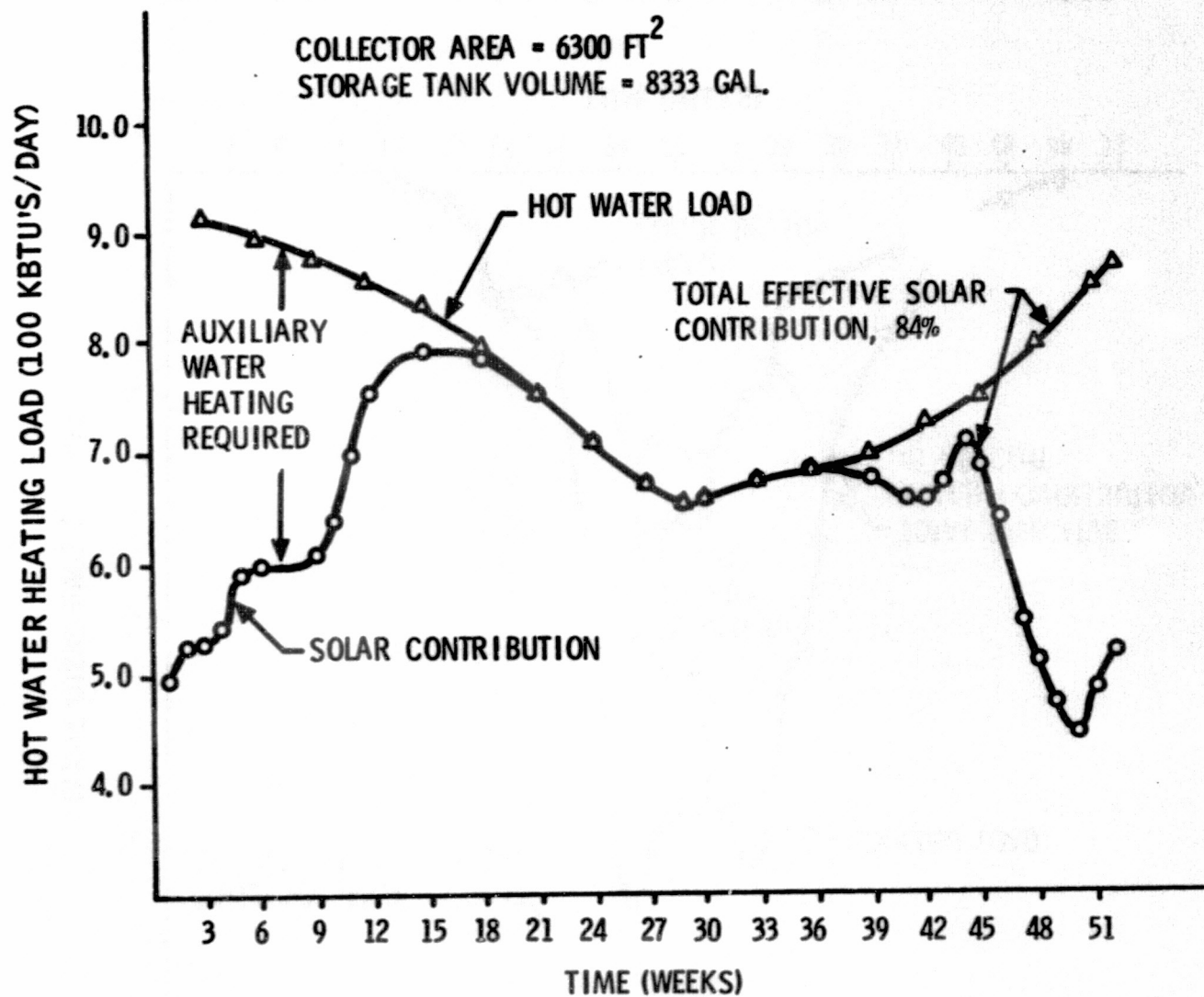


Figure 5-21. Weekly Solar Contribution to Hot Water Load - MFR

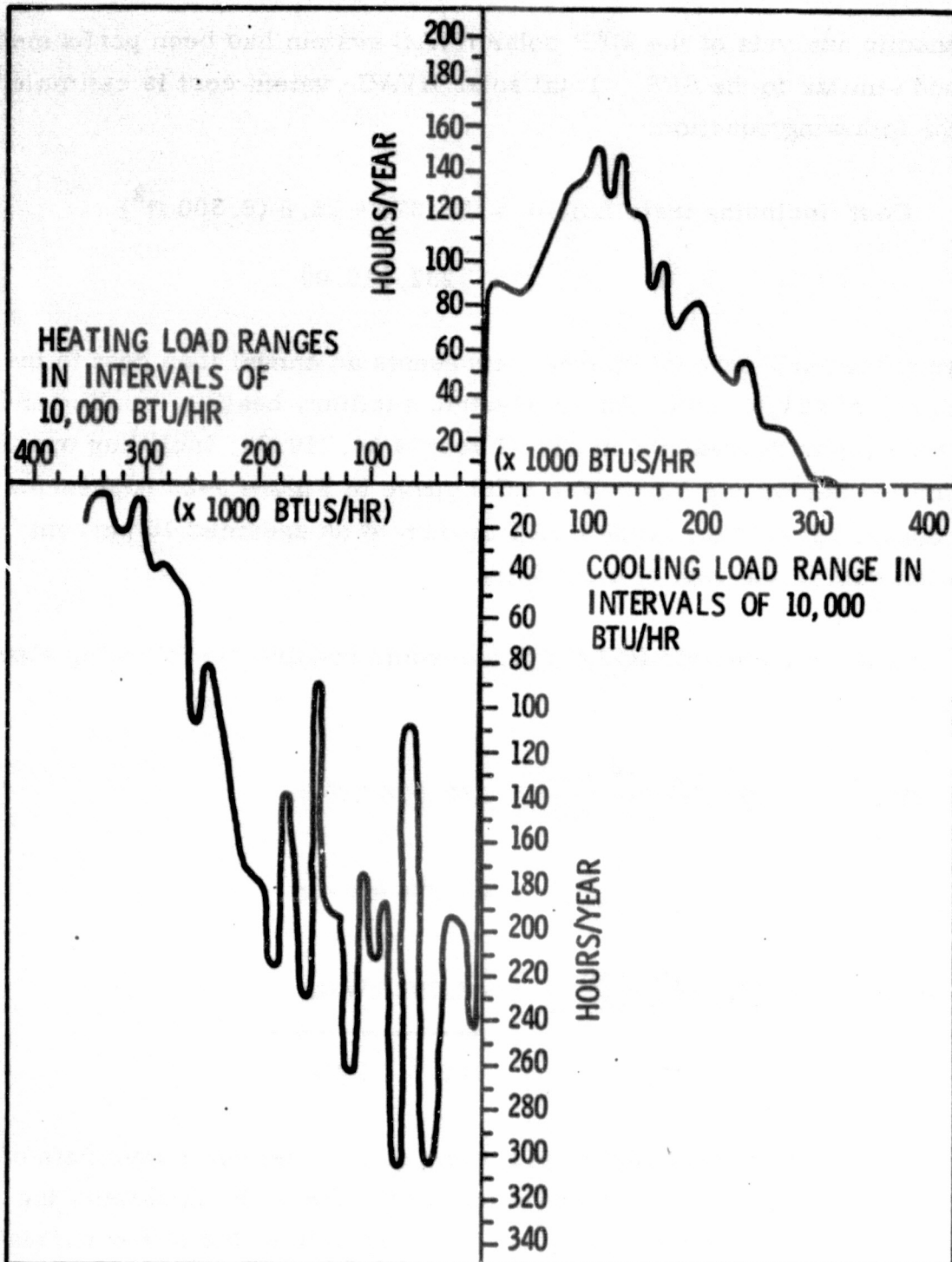


Figure 5-22. Cooling and Heating Load Histogram - MFR



An economic analysis of the MFR solar HVAC system has been performed in a method similar to the SFR. Total solar HVAC system cost is estimated from the following equation:

$$\begin{aligned}\text{Cost (including installation)} &= 70,376 + 25.8 (6,300 \text{ ft}^2) \\ &= \$232,916.00\end{aligned}$$

A 20-year loan at 7 percent interest represents an annual loan cost to the MFR owner of \$21,984.94. An all-electric auxiliary heating, cooling and hot water system represents an annual cost of \$4,710.78, including operating costs for the pumps and fans. The solid curve in Figure 5-23 represents the future annual costs of the solar HVAC system at an assumed 10 percent escalation rate in the cost of electricity.

A conventional all-electric HVAC system would require the following electric power.

Heating	=	$\frac{624.5 \times 10^6 \text{ Btu}}{3413 \text{ Btu/kW-hr}}$	=	182,976 kW-hr
Hot Water	=	$\frac{283.2 \times 10^6 \text{ Btu}}{3413 \text{ Btu/kW-hr}}$	=	82,976 kW-hr
Cooling	=	$\frac{325 \times 10^6 \text{ Btu}}{6.8^* \text{ Btu/kW-hr}}$	=	47,794 kW-hr
Total				<hr/> 313,747 kW-hr

This represents an annual cost of \$11,169.42 at an electric power rate of \$0.0356/kW-hr. The dashed curve shown in Figure 5-23 represents the annual cost to operate the conventional HVAC system at the power escalation rate of 10 percent for the next 20 years. The system payback period appears to be 21 years. This time is in excess of the assumed 20-year operating life

\*EER (energy efficiency ratio) is estimated in ASHRAE 90-75 to be 6.8.



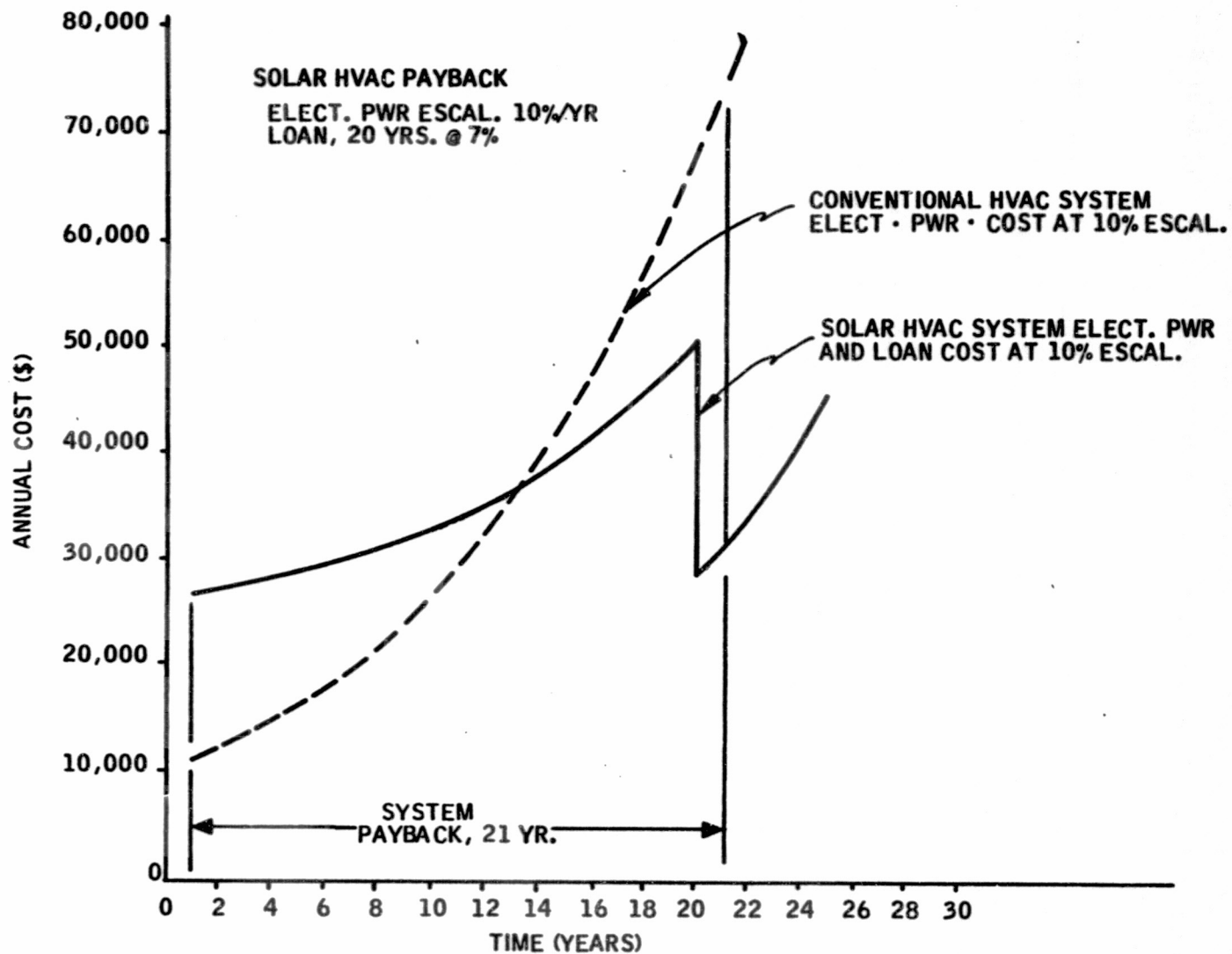


Figure 5-23. Solar HVAC Payback - MFR

for the solar HVAC system. However, this figure does illustrate that if solar HVAC systems demonstrate operational life times in excess of 20 years, a significant cost advantage will exist in the years following satisfaction of the loan.

No attempt has been made at this time to estimate a "low cost" solar HVAC system for the MFR; however, it is expected that conclusions similar to those reached for the SFR system would result:

"Solar HVAC systems may be economically competitive with conventional systems provided: 1) their present fabrication and installation costs can be reduced; and 2) the cost of conventional power escalates from 5 percent to 10 percent per year."

### 5.2.3 Commercial Building

The recommended solar-assisted heating, cooling and hot water system for the commercial building is an expansion of the MFR system. The major system parameters for the commercial building solar HVAC system are listed in Table 5-3. The cooling subsystem consists of three 25-ton Rankine cycle-driven water chillers instead of a large individual 75-ton R/C - A/C cooling subsystem.

The collector area of 12,060 ft<sup>2</sup> was selected to provide at least a 40 percent solar contribution to the cooling load. The thermal performance of the base-line system in meeting the building's heating, cooling and hot water loads is shown in Figures 5-24 through 5-26 and summarized as follows.

Table 5-3. Commercial Building (32,500 ft<sup>2</sup>) Atlanta, Georgia**Baseline Solar HVAC System Design:****a) Cooling Subsystem**

Three 25-ton air conditioners, total 75 tons  
COP = 6

Constant speed control

Rankine design point

20 hp at 190°F

$\eta_{RC}$  at 190°F = 8.6 percent

Control range, 150°F - 200°F

**b) Heating Subsystem**

Capacity, 800,000 Btu/hr

Hx coil effectiveness, 0.6

**c) Storage Subsystem**

Tank capacity, 15,600 gallons

Storage Hx effectiveness, 0.55

**d) Solar Collector Subsystem**

Area 12,060 ft<sup>2</sup> (total)      10,010 ft<sup>2</sup> (effective)

Number of collectors, 670

Flow rate, 300 gpm

**e) Building Loads**

UA = 12,340 Btu/hr°F

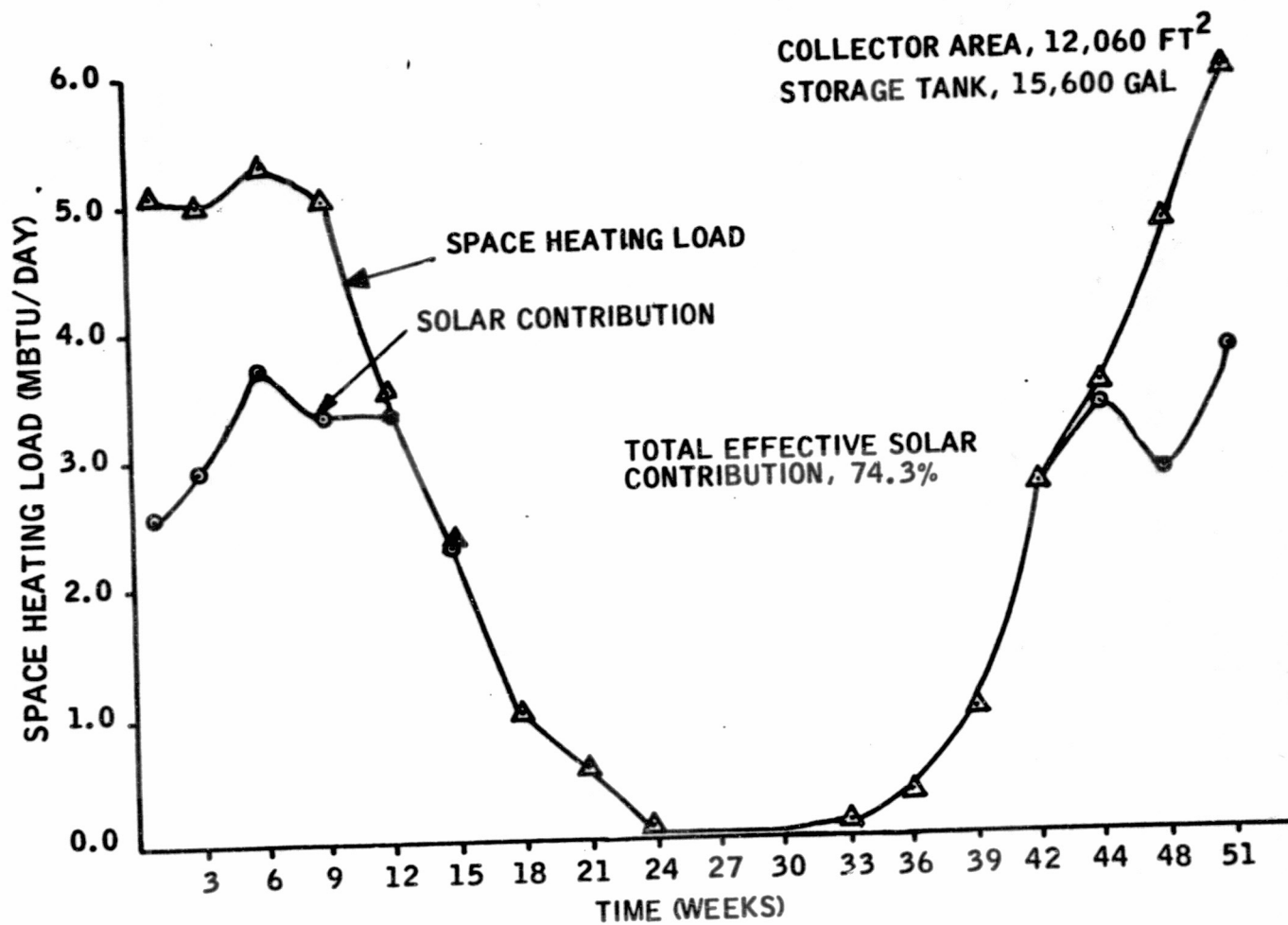
Infiltration, 225 cfm

Ventilation, 4,500 cfm (day)      0 cfm (night)

Internal load, 273,180 (day)      0 (night)

Design day heating load = 819,921 Btu/hr

Design day cooling load = 853,631 Btu/hr



5-42

Figure 5-24. Weekly Solar Contribution to Heating Load - COM.

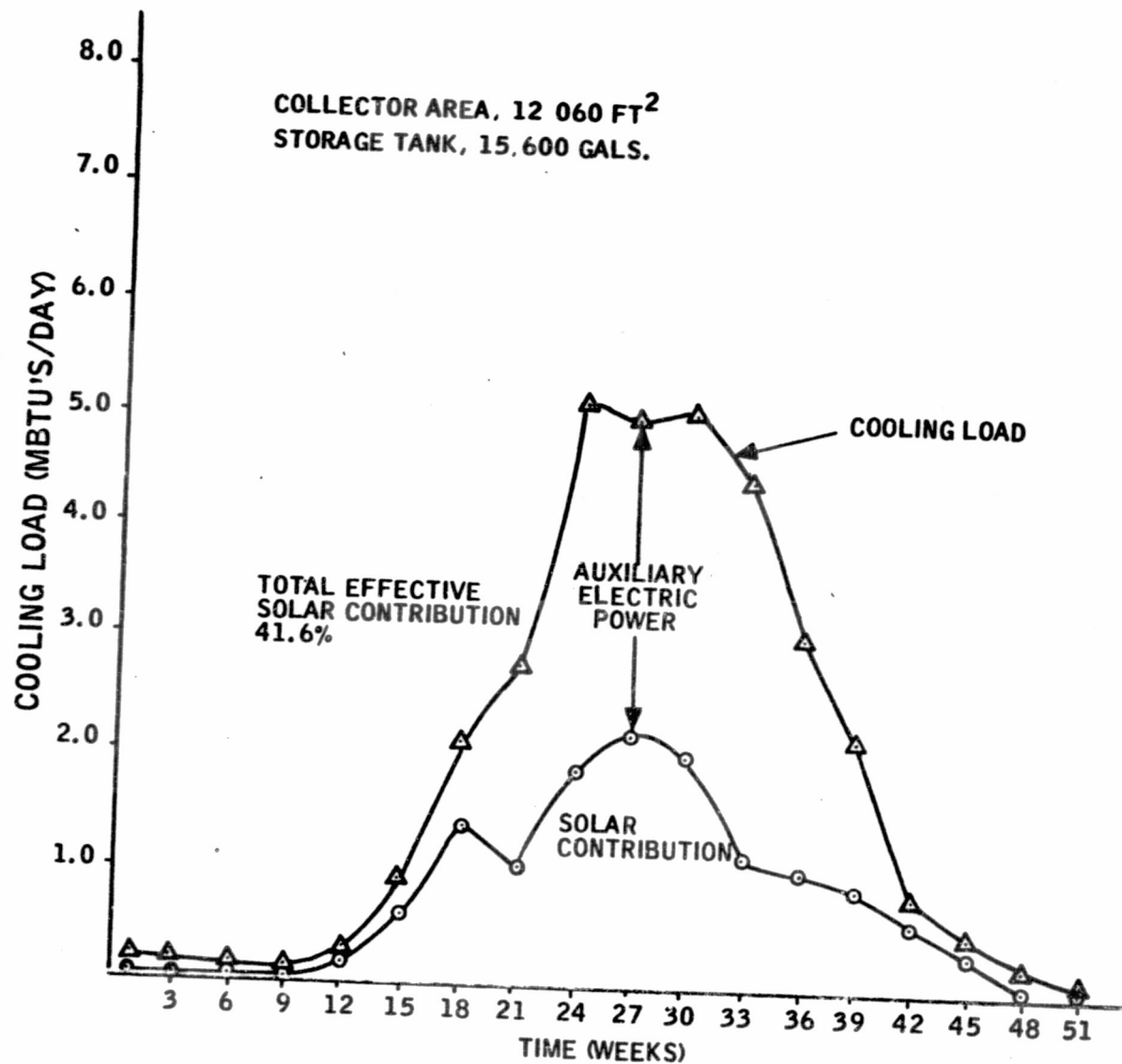
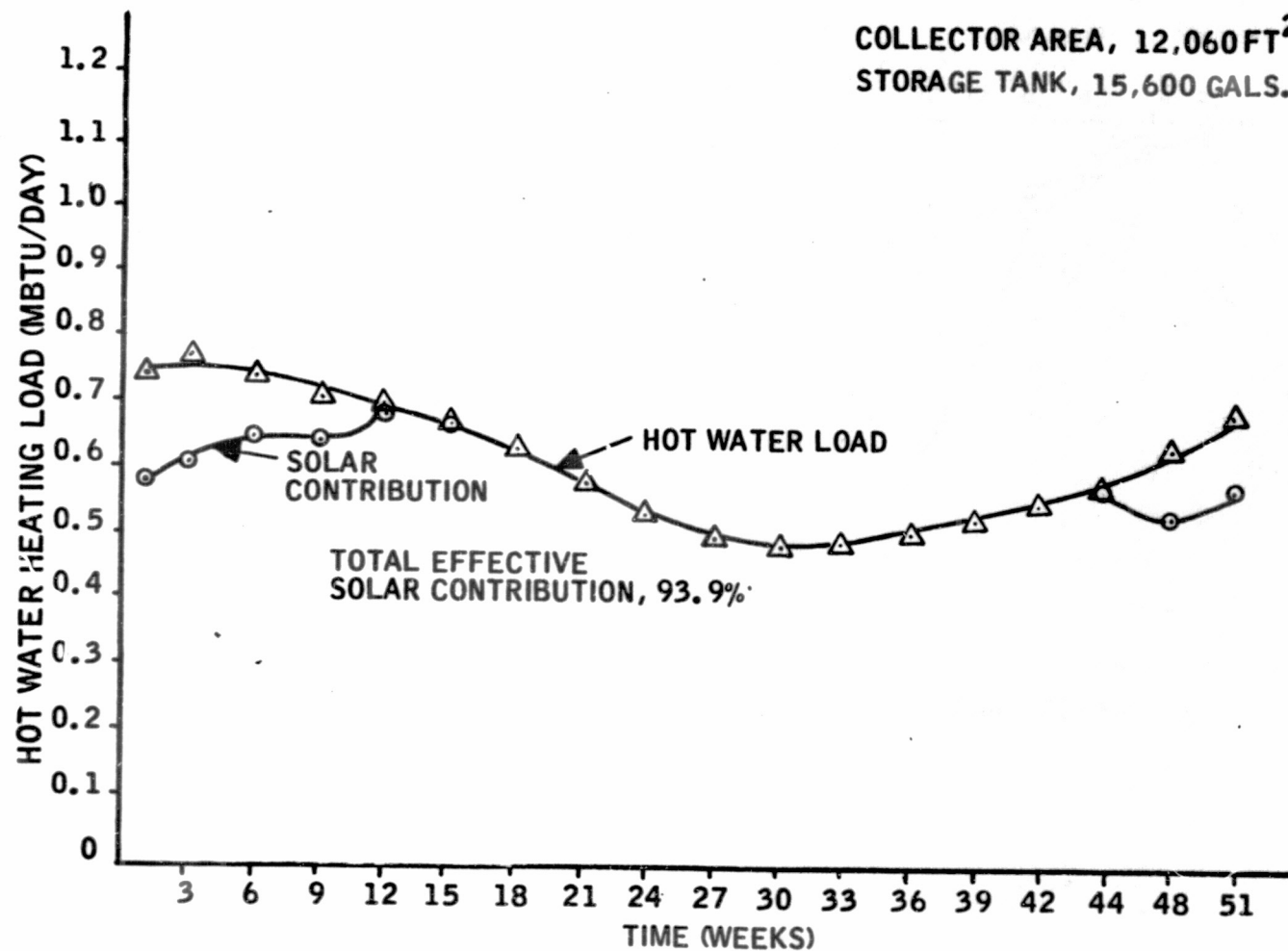


Figure 5-25. Weekly Solar Contribution to Cooling Load - COM.



5-44

Figure 5-26. Weekly Solar Contribution to Hot Water Load - COM.

	<u>Annual Load</u> Btu	<u>Solar Supplied</u> Btu	<u>Solar</u> <u>Contribution</u>
Heating Load	$925.3 \times 10^6$	$687.5 \times 10^6$	74.3%
Cooling Load	$702.7 \times 10^6$	$292.4 \times 10^6$	41.6%
Hot Water Load	$226.4 \times 10^6$	$212.5 \times 10^6$	93.9%

The annual costs for conventional fuels to support the commercial building solar HVAC system are listed below.

Heating ( $237.8 \times 10^6$  Btu)

Electrical (\$0.0356/kW-hr)	\$ 2482
Oil (\$0.4/gallon)	844
Gas (\$0.0017/ft <sup>3</sup> )	505

Cooling ( $410.3 \times 10^6$  Btu)

Electrical (\$0.0356/kW-hr)	884
-----------------------------	-----

Hot Water ( $13.9 \times 10^6$  Btu)

Electrical (\$0.0356/kW-hr)	146
Oil (\$0.4/gallon)	50
Gas (\$0.0017/ft <sup>3</sup> )	30

An analysis of the pumping and fan power required to operate this large solar HVAC system as well as an economic analysis has not been made as of this writing. The commercial building's internal loads and zone heating and cooling requirements are extremely site specific. Therefore, the economic analysis of this system will be performed at a time when more specific site information is available.

### 5.3 CONCLUSIONS

This section has summarized the thermal performance of baseline solar HVAC systems for a single-family residence, a multifamily residence and a commercial building in Atlanta, Georgia. These performance figures will vary with location. However, it is believed that these results for Atlanta reflect the realistic impact of solar HVAC systems in a future major marketing area. Economic analyses have been performed to illustrate what must happen to result in solar heating and cooling systems being cost competitive. The results indicate that solar system fabrication and installation costs must be reduced and present-day fuel rates must escalate significantly in order to defend solar energy on an economic basis and this certainly must be done if commercial sales are to be successful.



## SECTION 6

### SUBSYSTEM DESIGN AND DEVELOPMENT APPROACH

#### 6.1 GENERAL

Each of the baseline solar HVAC systems previously described is made up of the following subsystems:

- Collectors
- Storage
- Space heating
- Auxiliary energy
- Hot water
- Transport
- Controls
- Site data acquisition
- Electrical
- Cooling

Many of these subsystems are common to all of the systems or are made up from different models of the same basic equipment. The subsystems are formed from off-the-shelf hardware whenever possible to minimize development.

Several of the subsystems contain components that require sizing to optimize performance while maintaining minimal component costs. It is recognized that sizing some equipment, such as pumps, requires more detail than is presently available for proper matching of system load characteristics. Those components which require critical sizing are:

- Water-to-air heat exchangers
- Water-to-water heat exchangers
- Collector area

- Pumps
- Piping size
- Hot-water heaters
- Auxiliary energy
- Total cooling capacity

Past experience indicates that water-to-air heat exchangers should be sized at 75 percent of peak demand for heating. As an illustration, the peak heating demand for the residential dwelling in the RFP was 80,000 Btu/hr. The heat exchanger was sized to supply 60,000 Btu/hr with 1000 cfm of return air at 70°F and a solar water supply of 140°F at 15 gpm. Optimization of the heat exchanger will involve efforts to supply the load at lower collector outlet temperatures once site specific information is available.

Practical limits exist as to air flow and return-air temperature which affect the heat-exchanger sizing as well as obvious physical constraints.

The collector areas will be sized on the basis of loads and the amount of collector area which could reasonably be installed on the allowable building space. Analysis has shown that these may not necessarily be optimum but are reasonably close without the benefit of building particulars. Further work on collector array optimization will be done.

The pumps will be sized on the basis of pressure drops through the collectors, heat exchangers, estimate of the length of pipe and number of fittings, and a piping schematic.

The hot-water heaters will be sized on the basis of domestic water supply recommended in ASHRAE and heating the water from 40°F to 140°F without the benefit of solar heating. The solar water heaters will be sized on the same draw basis but with solar-supplied heat at an average temperature of 140°F.

The remaining components in the subsystems are sized either by maximum load or by their relationship with other components in the subsystem, such as cooling towers, condenser and chilled water pumps, and electrical ratings of the electrical components.

## 6.2 COLLECTORS

The collector subsystem is a product which is presently being manufactured by Lennox Industries and is considered developed to the prototype level. In addition, it was recognized that the collectors are modular, suitable for retrofit or new construction, and can be combined to provide any subsystem size. It was also recognized that a study of possible flow configurations for various collector arrays should be made. Of concern is the possibility of nonuniform flow in the collector arrays which can lead to serious degradation of performance. A simulation program was developed from which guidelines will be established to determine the number, size, and arrangement of the collector modules and supply and return headers in the collector arrays. The results of this study will be used to design the collector arrays when specific site locations have been determined.

### 6.2.1 Parameters

A study of applicable Interim Performance Criteria (IPC) was made to determine what parameters of the solar collector to investigate. In the case of the collector, the development activity fell into two main areas: function and mechanical configuration.

During collector development, much effort was expended to design a highly efficient collector. Selective coatings and glass transmittance were investigated thoroughly. It was determined that the efficiency of collection was a

fixed design characteristic at the beginning of qualification and that efficiency of a newly produced collector was not a relevant qualification criterion. At the same time, however, it was apparent that the effect of environmental exposure on efficiency had not been sufficiently investigated, so that efficiency degradation was determined to be a qualification criterion.

The mechanical configuration of the collector was investigated during development and yielded a design that permitted easy integration with the structure and a long service life. One element of the mechanical configuration which was not verified during development was exposure to mechanical loads inherent in a structural application.

As a result of reviewing the work done during development, it was determined that the following tests needed to be conducted to complete qualification of the collector:

- Degradation due to:
  - Solar exposure
  - Pollutants
  - Thermal exposure
  - Outgassing
- Mechanical loads due to:
  - Internal pressure
  - Roof loads
  - Hail

After determining the parameters to be tested, an investigation was made to find or develop suitable test techniques. Wherever feasible, the approach was to make use of existing standard test techniques. Toward this end, the IPC was used as a baseline to standardize the tests. Referenced test standards were used whenever practical. This approach resulted in the following series of tests.

### 6.2.2 Collector Subsystem Qualification Status

The collector subsystem will be qualified in part by a formal Qualification test in accordance with Document F3437-T-101. This test was initiated on 18 October 1976. It is progressing on schedule and completion is expected on or about 30 May 1977. Qualification status is currently as follows:

- Test Number 3.1, Pressure:

Objective: Ascertain that the system does not leak under a hydrostatic pressure of 150 psig.

Status: Not started.

Results: None.

- Test Number 3.2, Service Loads:

Objective: Determine the ability of the collector to withstand a distributed load of 50 psf. This is to simulate typical roof loads of snow and wind.

Status: Complete.

Results: The collector withstands a uniform load of 50 psf with no damage or plastic deformation. A load of 78 psf applied for engineering information also resulted in no damage. The load consisted of uniformly distributed sandbags.

- Test Number 3.3, Hail:

Objective: To determine the ability of the collector to withstand impact of 1.25-inch hail at 82 fps without fluid leakage in the system.

Status: Complete

Results: No damage resulted to any component of the collector due to impact of 1.25-inch hailstone at 82 fps. This test level was increased for engineering information with the following results:

- 2-inch hail, 107 fps - no damage
- 2.5-inch hail, 116 fps - glass breakage (outside cover)
- 3.0-inch hail, 131 fps - glass breakage

This test was conducted by firing an ice sphere from a compressed-gas launcher.

● Test Number 3.4, Solar Degradation:

Objective: To determine if degradation results from up to 6-month exposure near Phoenix, Arizona.

Status: Pre-exposure efficiency tests conducted on two collectors. Both collectors are now undergoing exposure at Desert Sunshine Exposure Tests, Phoenix, Arizona.

Results: None to date.

● Test Number 3.5, Pollutants:

Objective: To determine if coupon samples of collector components are degraded by exposure to:

- Ozone
- Sodium chloride
- Sulfurous acid
- Nitric acid
- Hydrochloric acid



Status: Exposure of coupons is in progress for all pollutants with the exception of ozone. The ozone test will begin on or about 15 January 1977.

Results: None to date.

- Test Number 3.6, Thermal Degradation:

Objective: To determine if coupon samples of the collector are degraded by exposure to maximum service temperature.

Status: Exposure of coupons is in progress.

Results: None to date.

- Test Number 3.7, Insulation Outgassing:

Objective: To determine if outgassing of organic materials results from day-night cycling of the collector.

Status: Three cycles on the solar simulator are complete. One-hundred day-night cycles in Phoenix, Arizona, will begin on or about 15 January 1977.

Results: Requirements met for three solar simulation cycles.

### 6.3 ENERGY STORAGE SUBSYSTEM

The collected solar energy which cannot be used for space heating or domestic hot-water heating at the time of its availability will be stored in the solar storage water tank. The stored energy would be available for future use (e.g., at night or on cloudy days). A system tank size of 1000 gallons represents reasonable size for residential use, while 8500 gallons seems feasible for multiple-family and commercial use. The flat heat tank is sized to store excess solar energy throughout the year without water boiling in the tank.

The actual water temperature in the storage tank will vary depending upon the energy requirements for space heating/cooling and domestic hot-water heating. The storage system will use stratification as much as possible for maximum utilization of the collectors and stored energy. Valving will permit storage charging and recharging with the activation of the storage pump and proper control signals. A layer of insulation will reduce heat losses. Actual location and size of the storage tank will be determined after specific sites have been selected.

#### 6.4 SPACE HEATING SUBSYSTEM

Space heating for residential and multiple-family use is provided using a forced-air recirculating system. The fan is integral to the commercially available auxiliary energy furnace. Heating of the air from the solar energy source occurs through heat exchange in a hot-water coil which is installed in the return-air cabinet. Coil capacities of 45,000 to 100,000 Btu/hr are available.

The commercial units for space heating/cooling are provided using a forced-air recirculating system with fresh-air makeup. The fan(s) is integral to the rooftop-mounted auxiliary energy furnace. Heating of the air from the solar energy source occurs through heat exchange with a hot-water/chilled-water coil which is installed in each rooftop unit. The cooling capacity of each coil is 300,000 Btu/hr.

#### 6.5 AUXILIARY ENERGY SUBSYSTEM

The residential and multiple-family gas-fired furnace (Figure 6-1) is equally applicable to residential and small business or commercial installations. The low height, quietness of operation, and modern cabinet design permit



installation in a recreation or family room, basement, utility room, or closet. A low-boy-type installation is made possible with the addition of a return-air cabinet to the high-boy furnace. The low height of the furnace, due to the design features of the heat exchanger, will allow ample space for basement installations. The return-air cabinet can be installed on either side of the furnace. Direct expansion evaporator units and matching condenser units, electronic air cleaners, and power humidifiers can be added for a complete, all-season total comfort installation. Quiet operating blowers have sufficient capacity to handle all air volume requirements.

The commercial unit (Figure 6-2) will be a modification of an existing commercial rooftop unit of 500,000 Btu/hr heating capacity. The number of rooftop units required will be determined after site specific information is available.

The combination of gas-fired heating and cooling units with bottom handling of conditioned air are designed primarily for rooftop installation with optional fresh-air intake. A separate roof frame mates to the bottom of the unit and when flashed into the roof permits weatherproof duct connection and entry into the conditioned area. No additional roof curbing or flashing is required. The single-package unit can also be installed on a slab at grade level with end-handling of conditioned air. The insulated single cabinet houses gas-fired heaters, belt drive blowers, air filters, and the optional air intake dampers which are shipped complete with all controls wired.

The development status of the auxiliary energy and space heating subsystems is as follows:

- Residential-Multiple-family Units -- Since the space heating subsystems will be the same for single-family and multiple-family applications, and the auxiliary energy subsystem will also be the same other than the size, they will be treated the same as residential units.

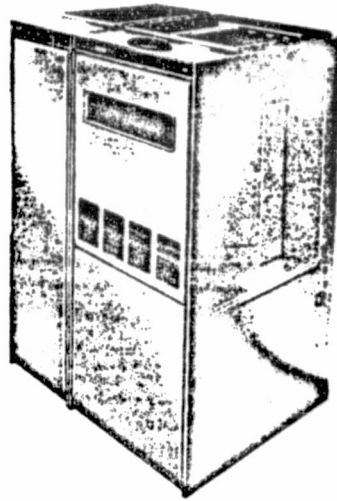


Figure 6-1. Lennox Gas-Fired Furnace

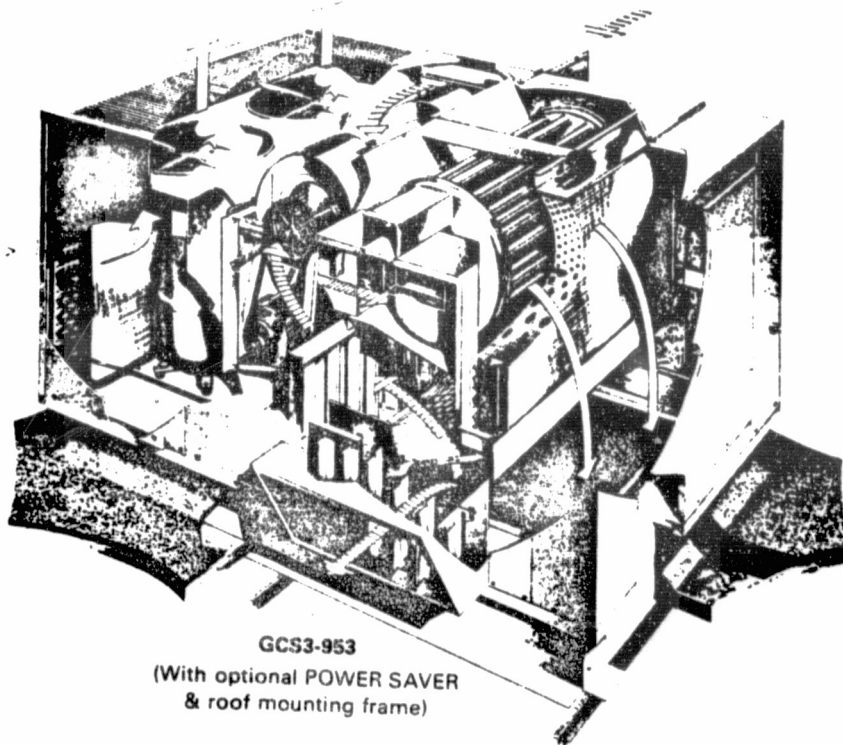


Figure 6-2. Lennox Rooftop Heating/Cooling Unit

Lab tests have concluded that based on Seely tests IPC No. 2.1 , the existing furnace motor will withstand a return-air temperature of 140°F. It appears that a special high-temperature motor will not be necessary. The operational characteristics of the furnace have also been test-verified as unaffected when integrated with the space-heating subsystem.

The only deviation required on the furnace will be a wiring change which will turn on the furnace blower when the thermostat calls for first-stage solar heating. This eliminates the need for a sure-start fan control.

The space-heating subsystem has been defined to be cabinet-designed to contain a filter, coil, and a temperature-limit control which will control the discharge air temperature from the coil. Some components have been acquired; some will be fabricated for assembly and verification tests after final detailing.

- Commercial Units -- A rooftop unit will be used to adapt a water coil located upstream from the blower and gas-fired heat section. The same coil will be used for both hot-water heating and chilled-water cooling. Efforts are being made to size the coil and lay out the coil location in relationship to a GCS3-1853-500 unit. Plans also call for investigating the possibility of using a new line of Lennox rooftop DSS1 units for the commercial application if the time required for development of this new line is feasible.

Based on the market analysis, it is clearly indicated that there is no need for heating-only commercial applications; this would make it advantageous to use the DSS1 units in the commercial application because of the possible time savings involved.

The coil will be sized for the cooling cycle and the same coil would likely turn out to be oversized for the heating cycle. However, in this situation, it is better to have only one size coil.

Again, the motors will be test-verified to ascertain that they will withstand 140°F air flow-through. Further, the air-handling capacity will be determined. Since the coil is upstream from the gas-fired heat exchanger in chilled-water cooling operation, condensation could occur in the heat exchanger. A provision for draining of condensate will be made.

## 6.6 HOT-WATER SUBSYSTEM

The hot-water subsystem consists of two hot-water heaters in series. The first heats the incoming domestic water with solar energy, while the second boosts the water to the final deliverable temperature, if added heat is necessary.

The solar energy which cannot be used for space heating will be used by the domestic hot-water preheater by using heat from the solar water storage tank or water preheating as shown in Figures 6-3 and 6-4. The water temperature in the preheater will not be controlled and thus will vary as domestic water use and the storage tank temperature vary. The inlet to the preheater will be city water. The outlet from this preheater will go to the input side of the domestic hot-water heater through a three-way control valve. The components of this subsystem are:

- Solar water-to-water preheater

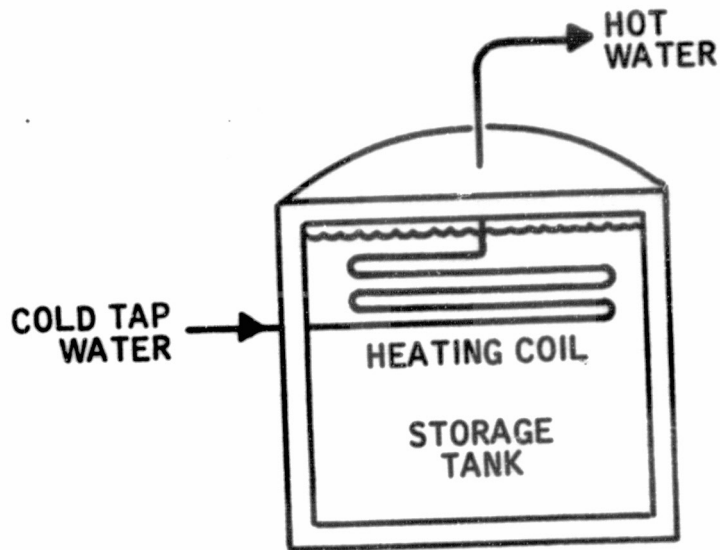


Figure 6-3. Solar Preheat DHW Subsystem for Single-Family Residence Applications

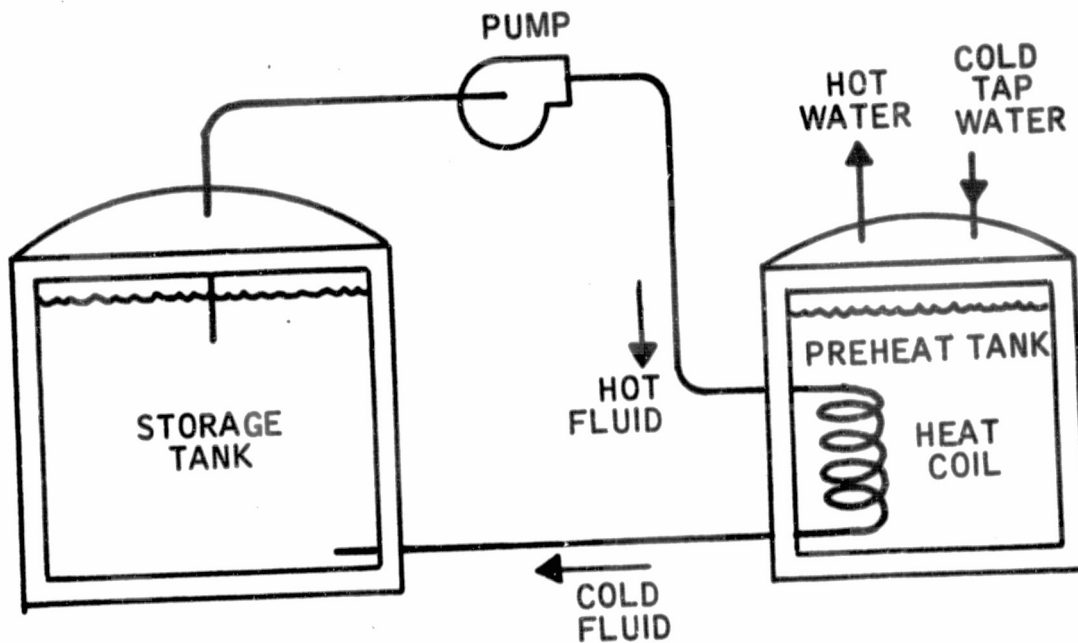


Figure 6-4. Solar Preheat DHW Subsystem for Multiple-Family Residence and Commercial Applications

- Gas-fired hot-water heater
- Self-contained three-way control valve

The domestic hot-water heater is sized using the requirements of the specific site.

The system can be modeled by using processes operating batchwise and involving heat transfer in determining the average temperature differences to describe the phenomena. The solar water preheat subsystem falls into this category (simple tank with heating coil). Figures 6-3 and 6-4 show schematics of the solar preheat DHW subsystems for single-family residence, multiple-family residence, and commercial applications.

Figure 6-5 shows the temperature rise of domestic water through a coil as a function of UA values for various tank temperatures. Figure 6-6 shows the same data plotted as coil outlet temperature as a function of coil length for various coil sizes for one tank temperature. Assuming a 50-foot coil of 0.75-inch copper tubing in a 140-gallon tank, a 55°F temperature rise would result. This corresponds to a UA of about 1100. Figure 6-7 shows pressure drop for various coil sizes. For the single-family unit, the pressure drop is insignificant. Figure 6-8 shows additional data on the 0.75-inch preheat coil. A storage tank temperature of 170°F would provide all of the 140°F domestic hot-water required for a flow of 2 gpm.

For the multiple-family and commercial applications, the system essentially "reverses its function." The coil is in a domestic water preheat tank, and the concern is how rapidly this tank can recover for various fluid temperatures, flow rates and coil UA values. Figure 6-9 shows the relationship between the recovery rate of the 326-gallon tank with heating water at 160°F as a function of the UA for various heating water flow rates. Figure 6-10 expresses the same data as Figure 6-9 for a 0.75-inch coil of various lengths,

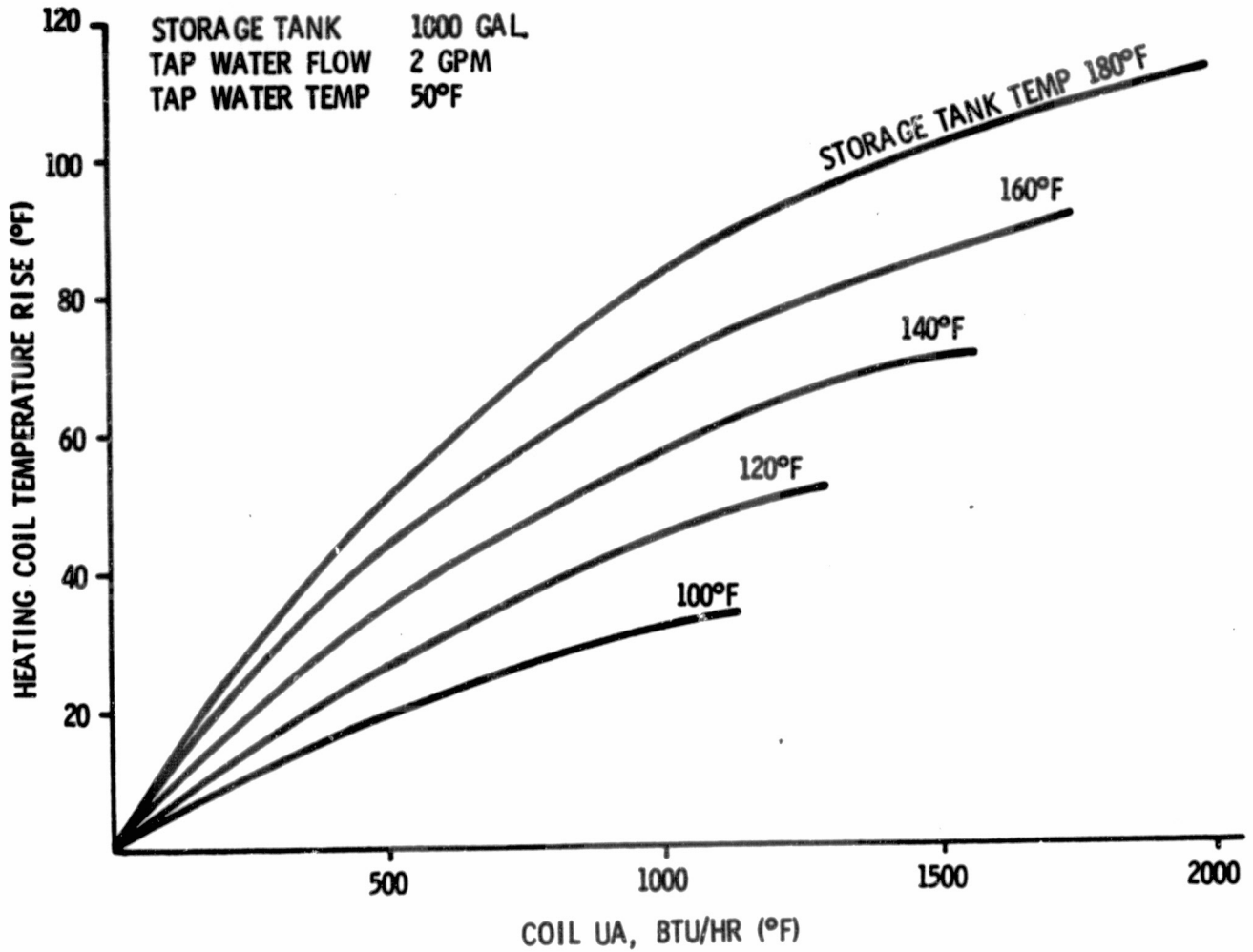


Figure 6-5. DHW Temperature Rise for Single-Family Residence Preheat Subsystem

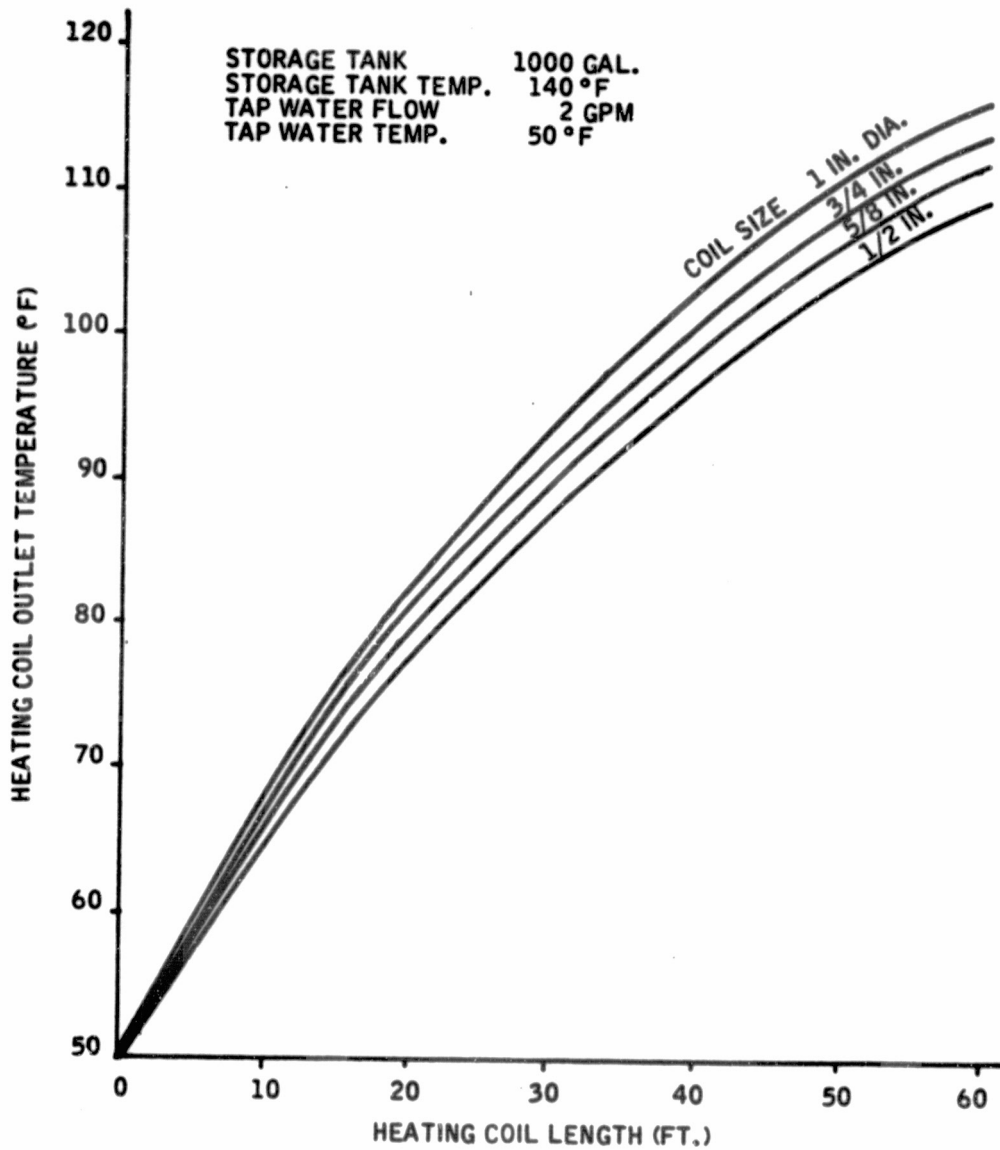


Figure 6-6. DHW Temperature Rise for Single-Family Residence Preheat Coil



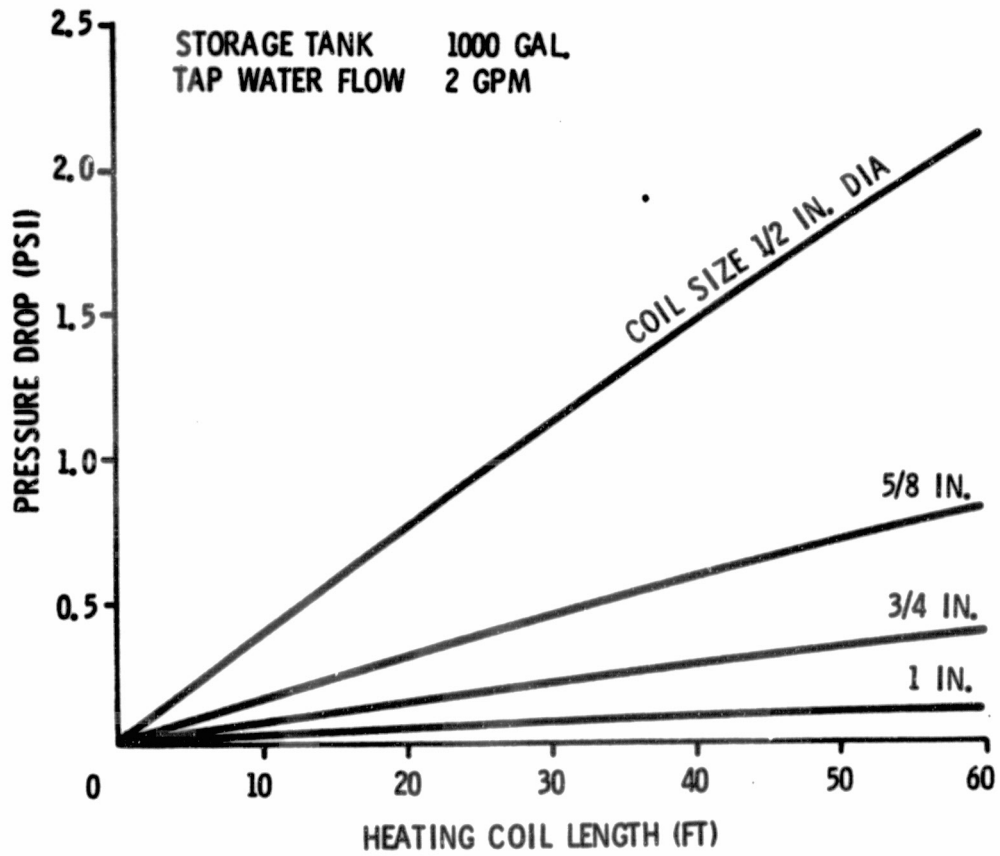


Figure 6-7. Pressure Drop for Single-Family Residence Preheat Coil

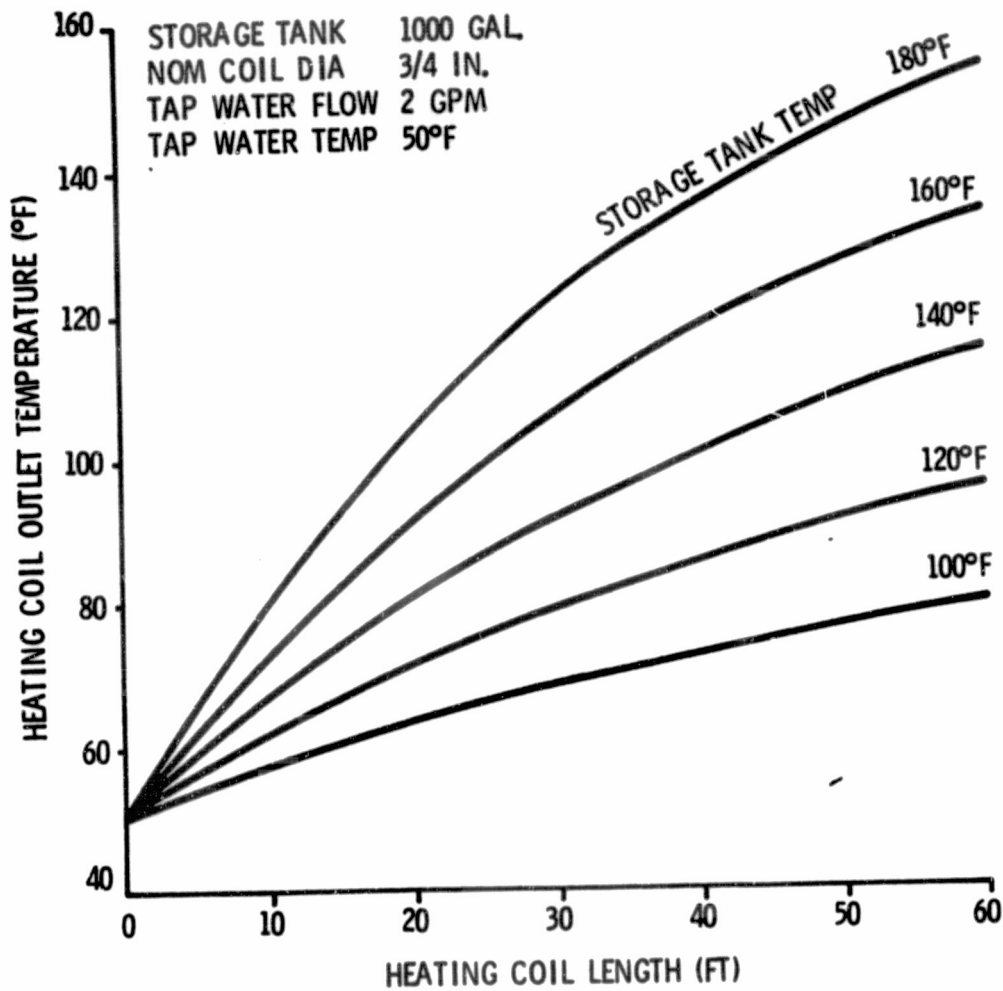


Figure 6-8. DHW Temperature Rise for Single-Family Residence 0.75-inch Solar Preheat Coil

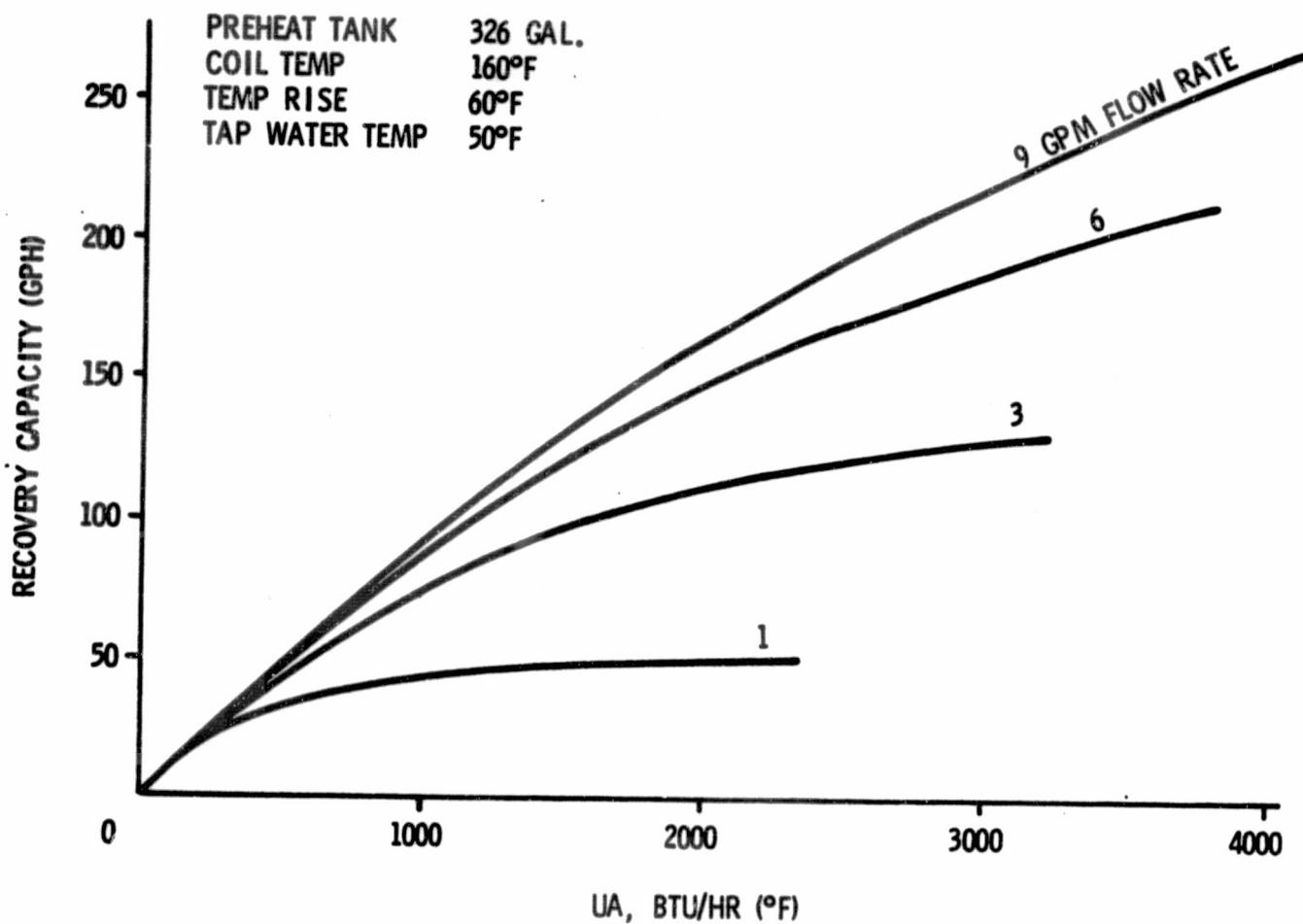


Figure 6-9. Recovery Capacity of DHW in Multiple-Family Residence Preheat Subsystem as Function of Heating Coil UA

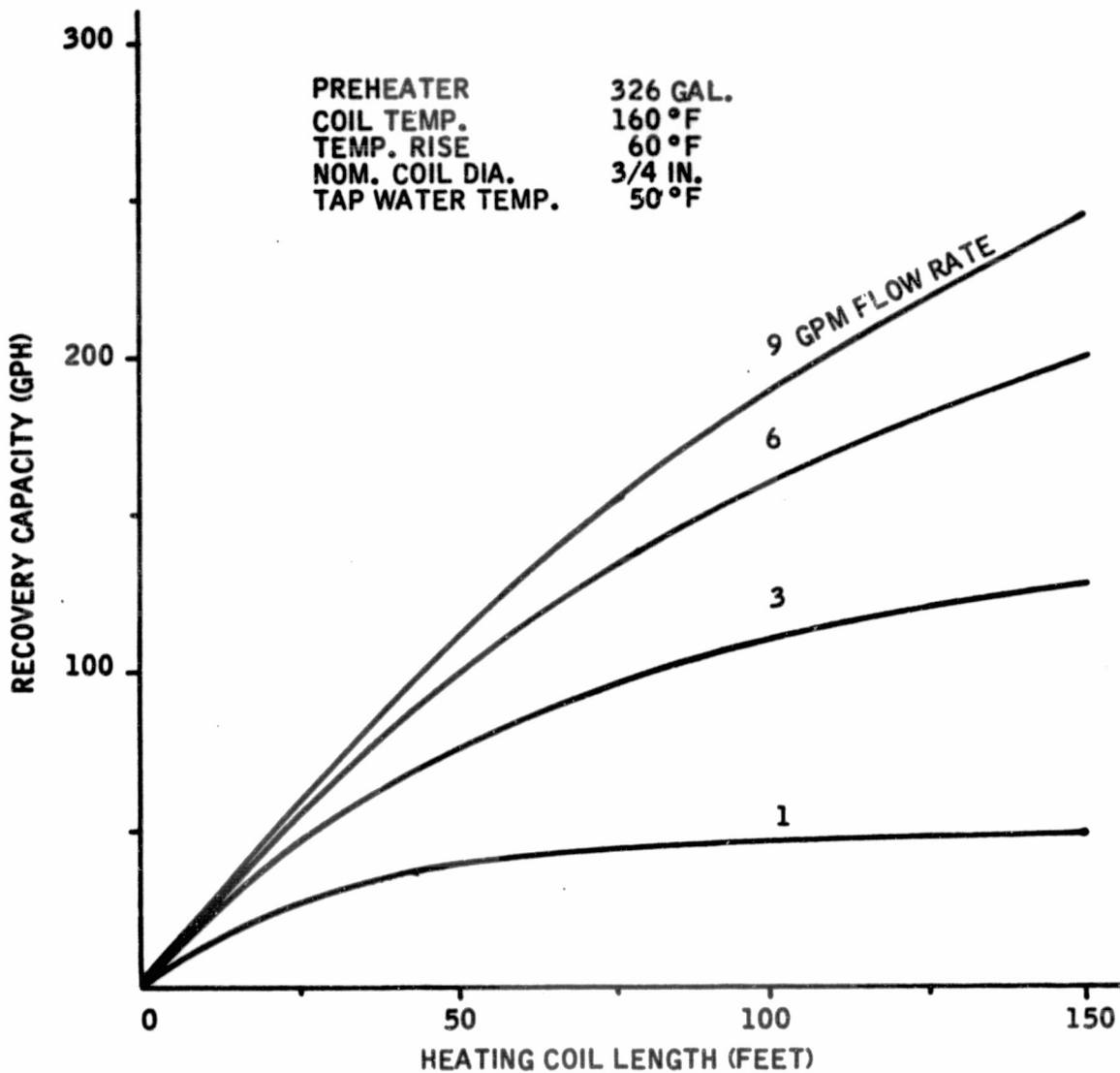


Figure 6-10. Recovery Capacity of Heating Coil in Multiple-Family Residence Application as Function of Coil Length

while Figure 6-11 gives the recovery capacity as a function of flow rate. Figure 6-12 can be used for predicting performance of a 0.75-inch heating coil of 150 feet effective length. Commercially available units with recovery rate of 300 gallons/hour are available.

The proposed system for single-family units will be as shown in Figure 6-3. During detailed design, tradeoffs between cost, performance, and size will be updated and finalized, while the commercial and multiple-family units will use commercially available units such as Patterson-Kelley or Bell and Gosset water-fired preheaters.

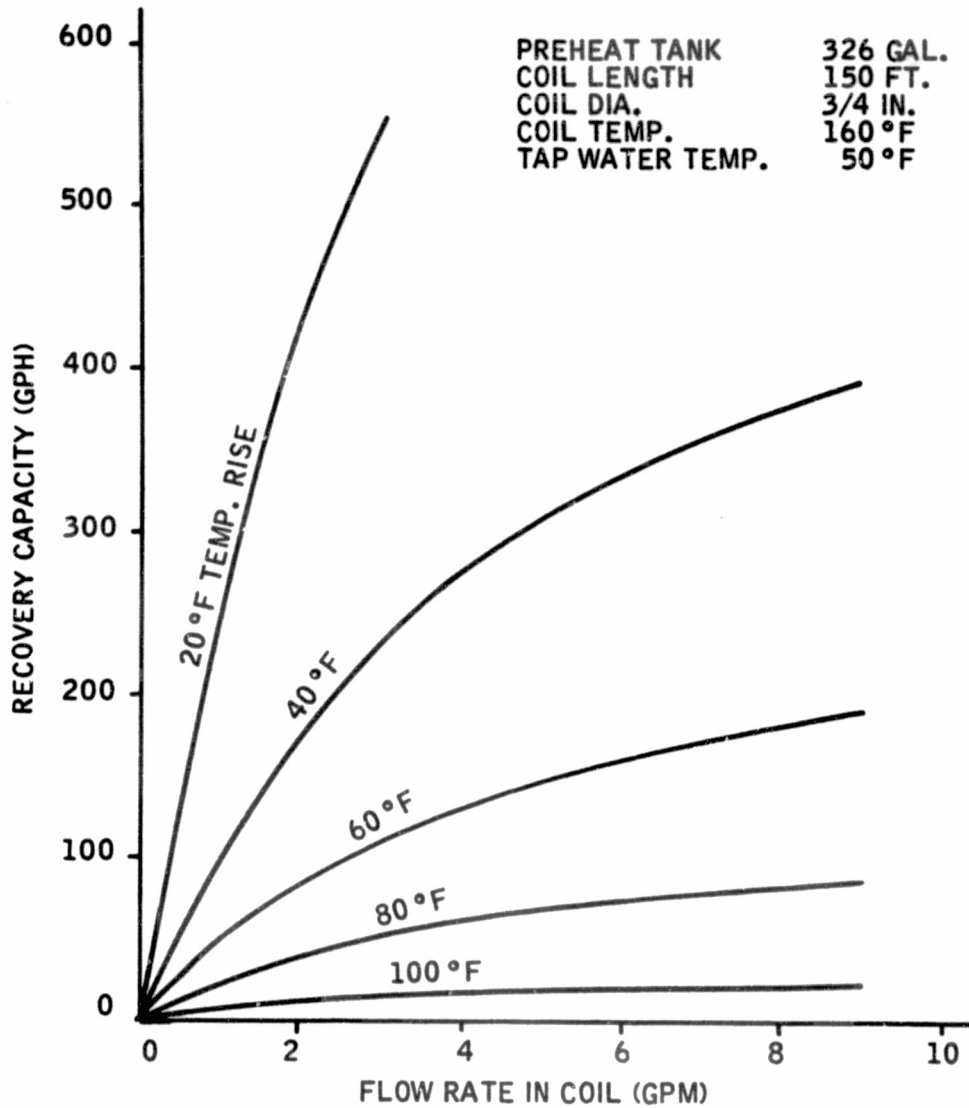


Figure 6-11. Recovery Capacity of Heating Coil in Multiple-Family Residence Application as Function of Coil Flow Rate

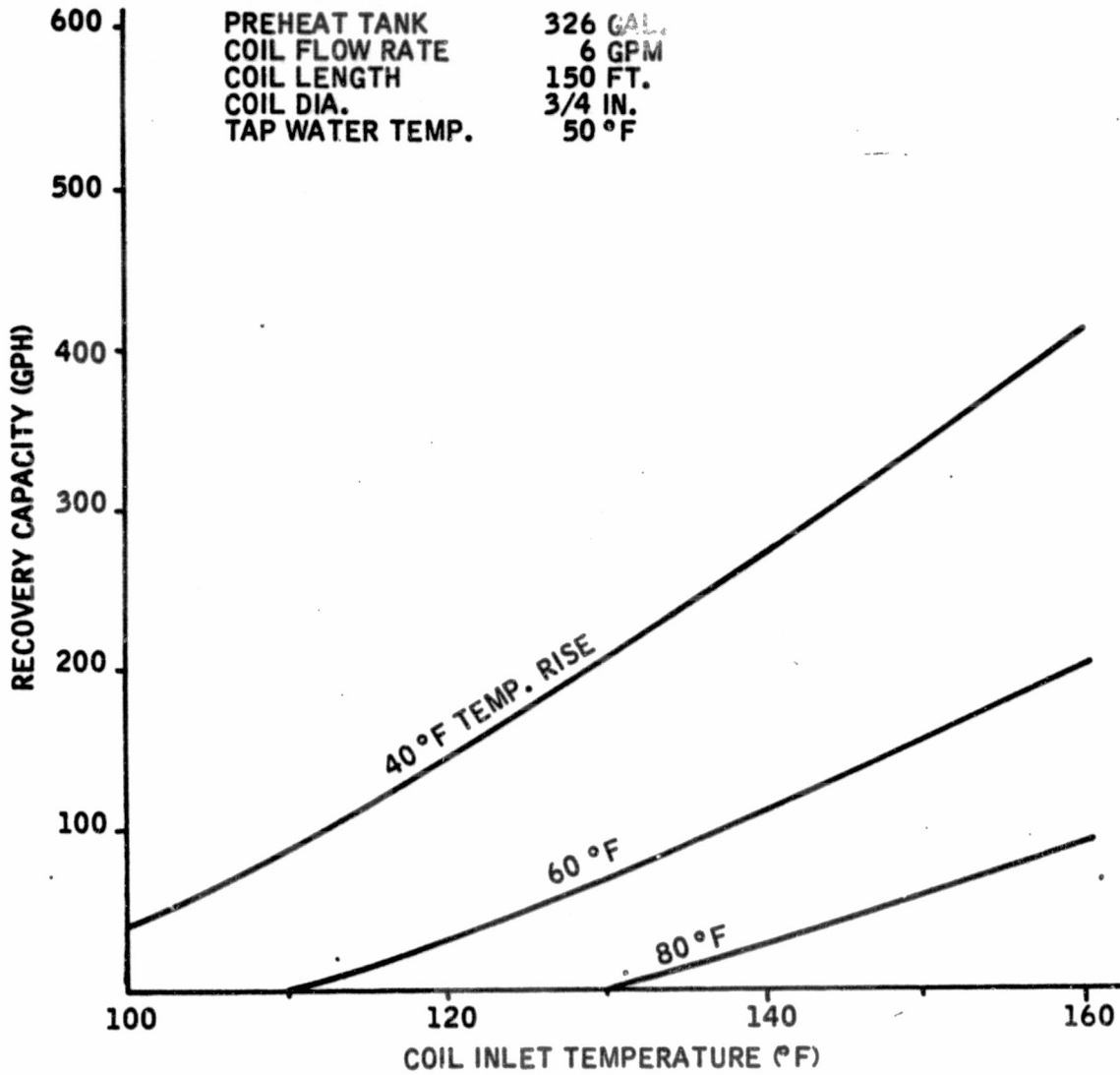


Figure 6-12. Recovery Capacity of Heating Coil in Multiple-Family Residence Application as Function of Coil Temperature

## 6.7 ENERGY TRANSPORT SUBSYSTEM

The energy subsystem consists of components necessary to ensure that subsystem components interface with one another and that necessary components are supplied for safe and efficient operation of the system. The following components are included in the transport subsystem:

- Piping sized for the system and site
- Expansion tank with fittings, ASME-rated
- Air separator and pressure-relief valves, ASME-rated
- Flow control devices with flowmeters
- Circulating pumps, centrifugal with machinable impellers
- Tube-and-shell heat exchanger
- Purge coil for dissipating excessive solar energy

System components are sized upon available data from the sites, or calculated, or obtained from by analysis of system performance. Further refinements in system sizing of pumps and piping can be expected when specific buildings are determined and piping layouts are completed. This could cause considerable variation in the projected size of pumps, pipe and valving.

All valving except control valves are excluded from the transport system. These valve sizes and locations can best be determined after building layouts and will be included in the installation specifications and pricing.

The most critical components on sizing are the heat exchangers. An analysis was made to determine the effects of using various heat exchangers with different "effectiveness" values and mass flowrates. Such an analysis is clearcut for all modes of operation except heating and cooling from storage. In these cases, the transport subsystem consists of two direct-type heat



exchangers coupled by the circulation of a heat-transfer medium. The thermal performance of the subsystem when in these modes of operation was investigated. The storage media and liquid-coupling working fluid were both water. The overall "effectiveness" is related to the subsystem parameters for  $C_L > C_s > C_c$  by the following equation:

$$\epsilon = \frac{1}{\frac{1}{\epsilon_c} + \frac{C_c}{C_L} \left( \frac{1}{\epsilon_s} \frac{C_L}{C_s} - 1 \right)}$$

For the condition  $C_s > C_L > C_c$ , the coupling loop-to-storage loop heat-flow capacity ratio,  $C_L/C_s$ , drops out. The individual terms of the equation are defined in Figures 6-13 through 6-21.

The results of the thermal performance of the transport subsystem for cooling and heating from storage modes of operation are presented in Figures 6-13 through 6-20. The conclusions of the analysis for the cooling mode in single-family residence parameters' design range are:

- A 12-gpm liquid-coupling loop flowrate has been selected. An increase in flowrate should be accompanied by an increase in storage-loop flow to prevent any decrease in subsystem performance. However, the decrease in subsystem performance is minimal.
- The storage-loop flowrate should be 7 to 9 gpm. Storage flow variation has only a small effect on the overall thermal performance of the subsystem; effectiveness increases 3.5 percent/gpm increase in flow rate.
- The storage-side heat-exchanger effectiveness should be 0.5 to 0.6. A 1 percent overall effectiveness increase can be expected per 5 percent increase in the component heat-exchanger effectiveness.

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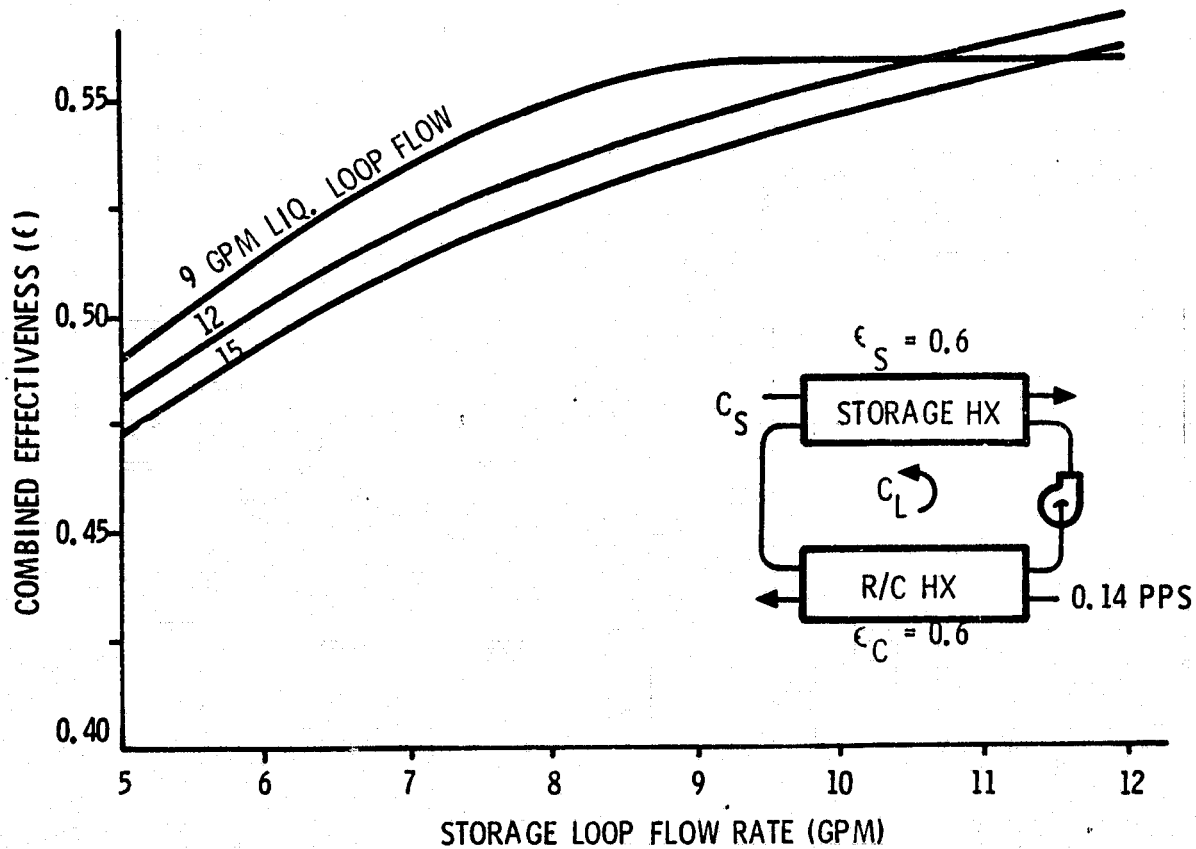


Figure 6-13. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

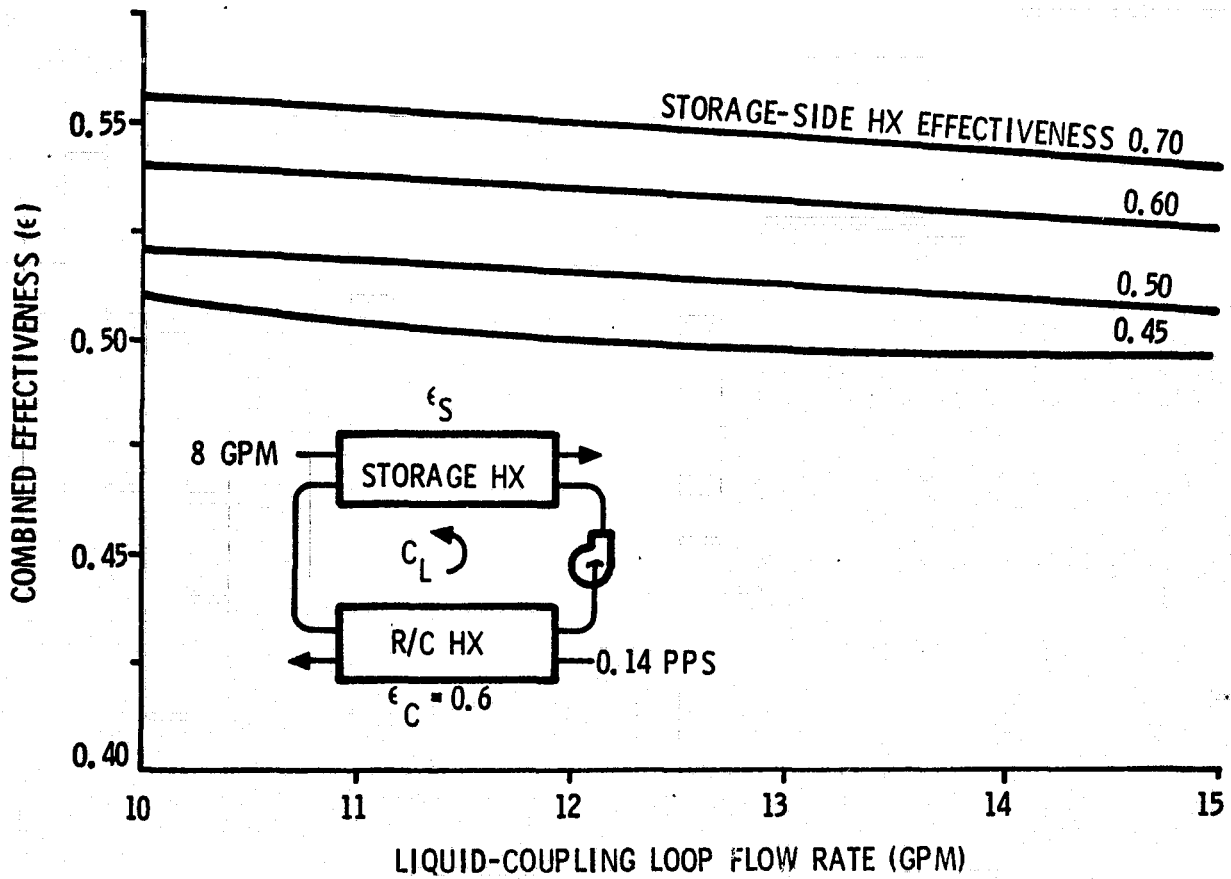


Figure 6-14. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

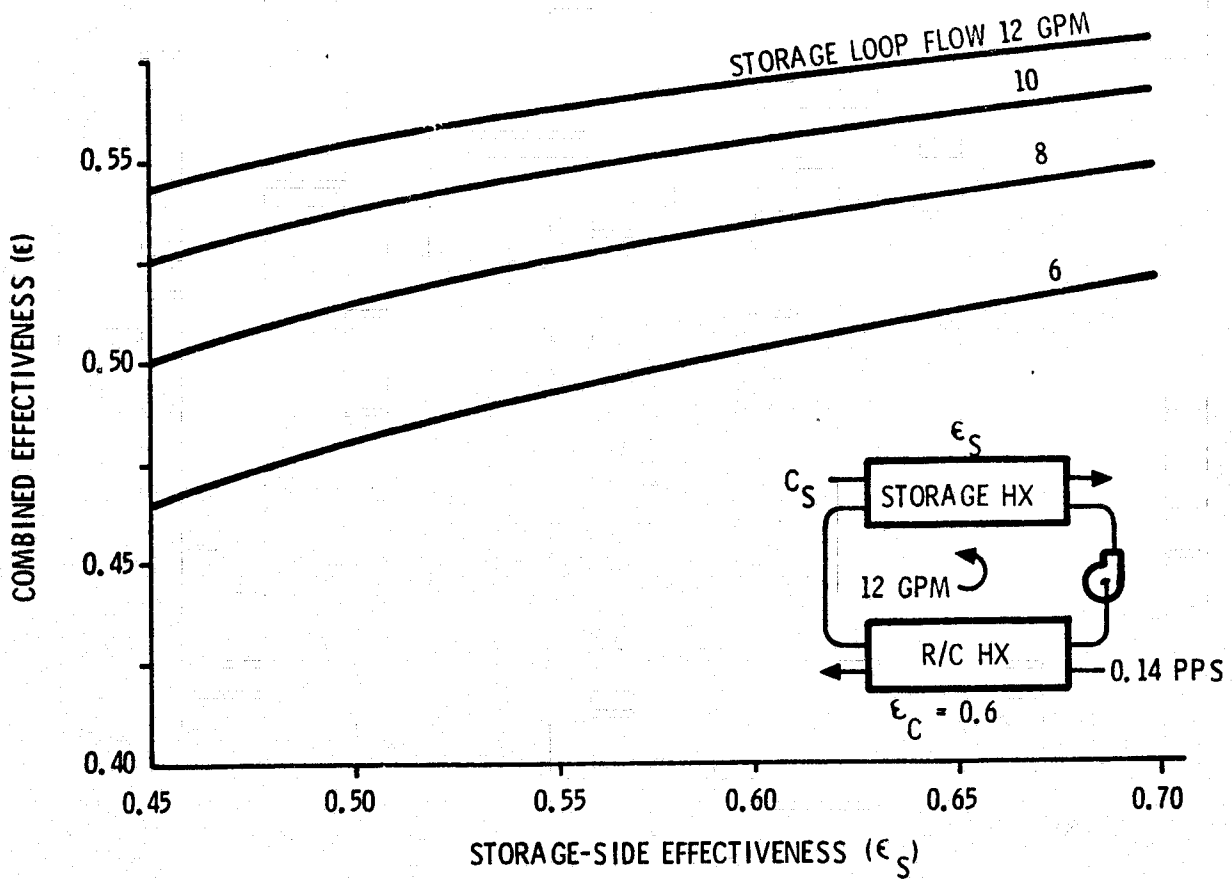


Figure 6-15. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

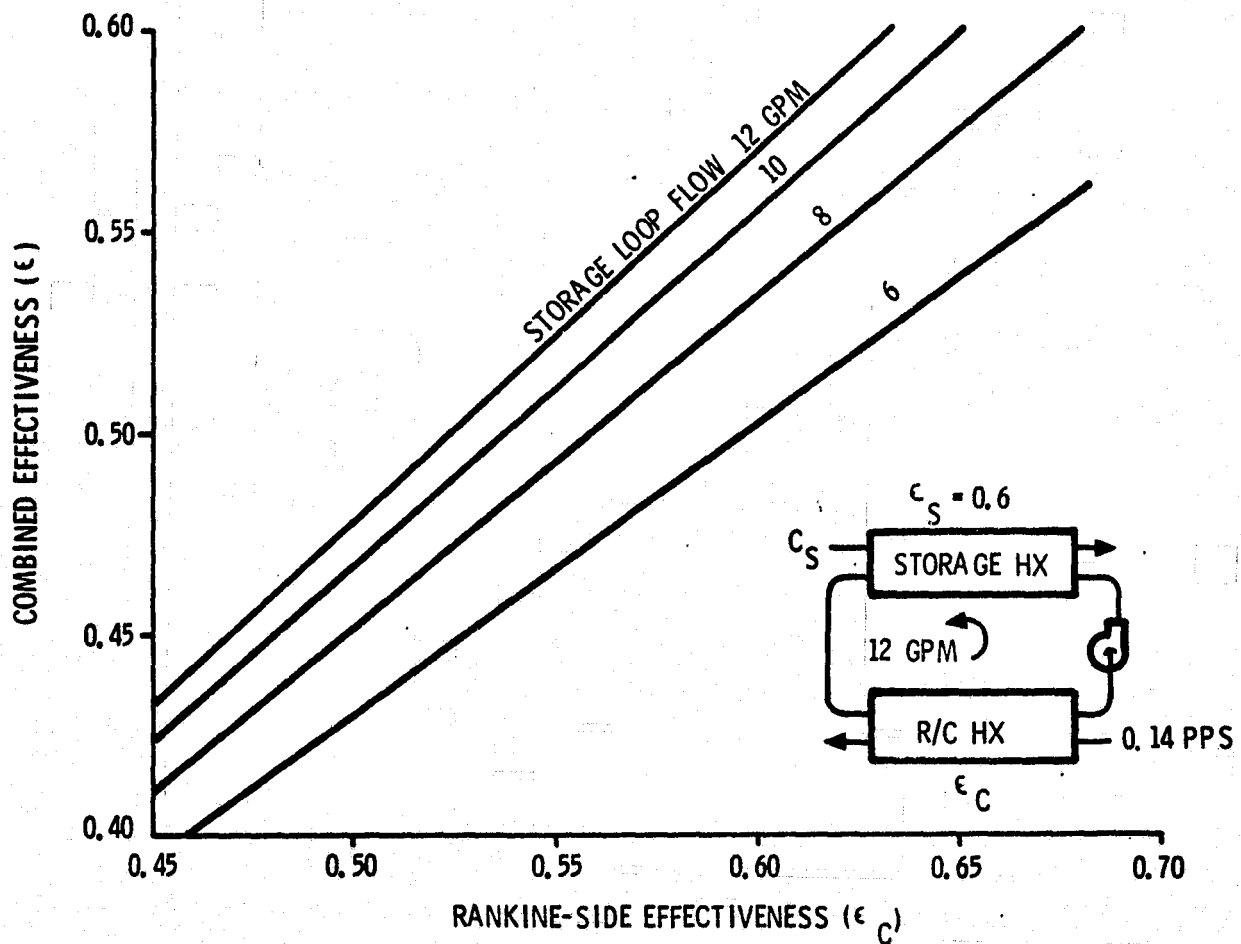


Figure 6-16. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

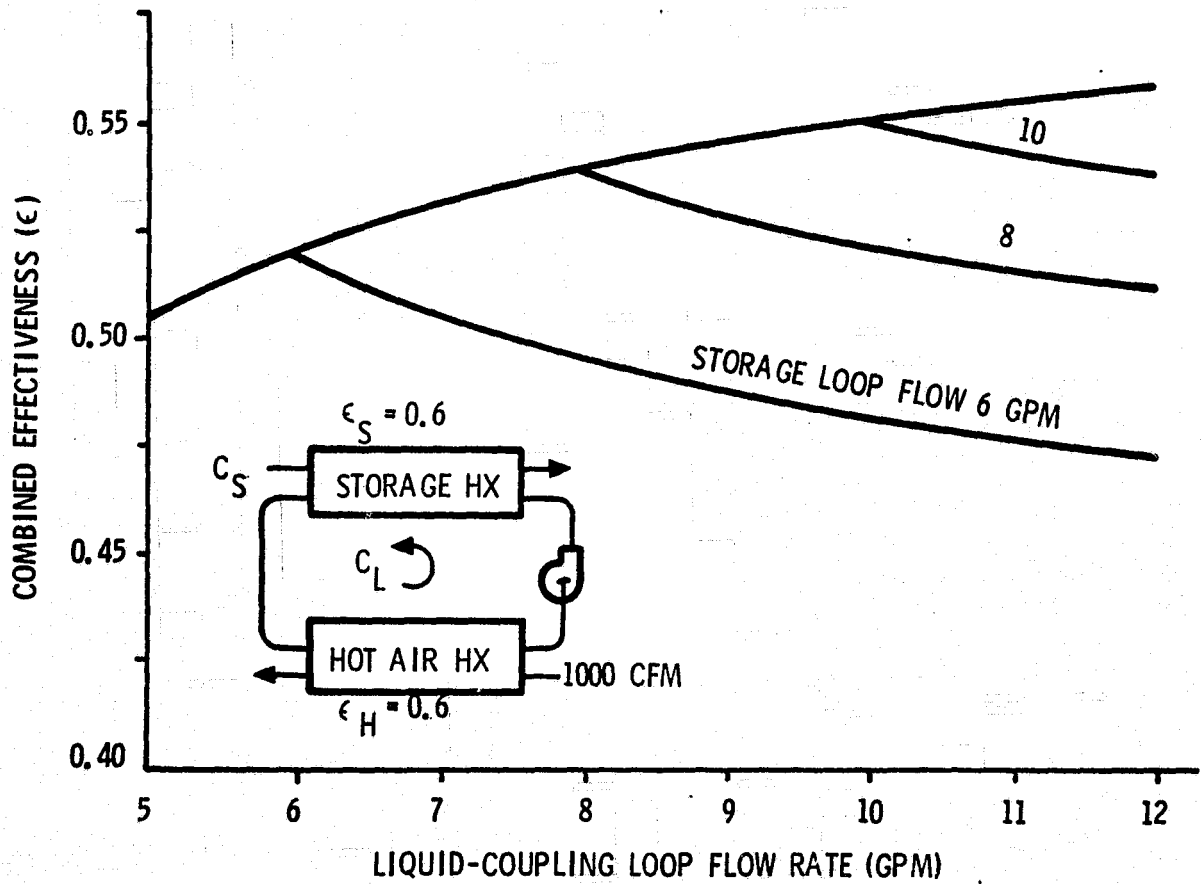


Figure 6-17. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

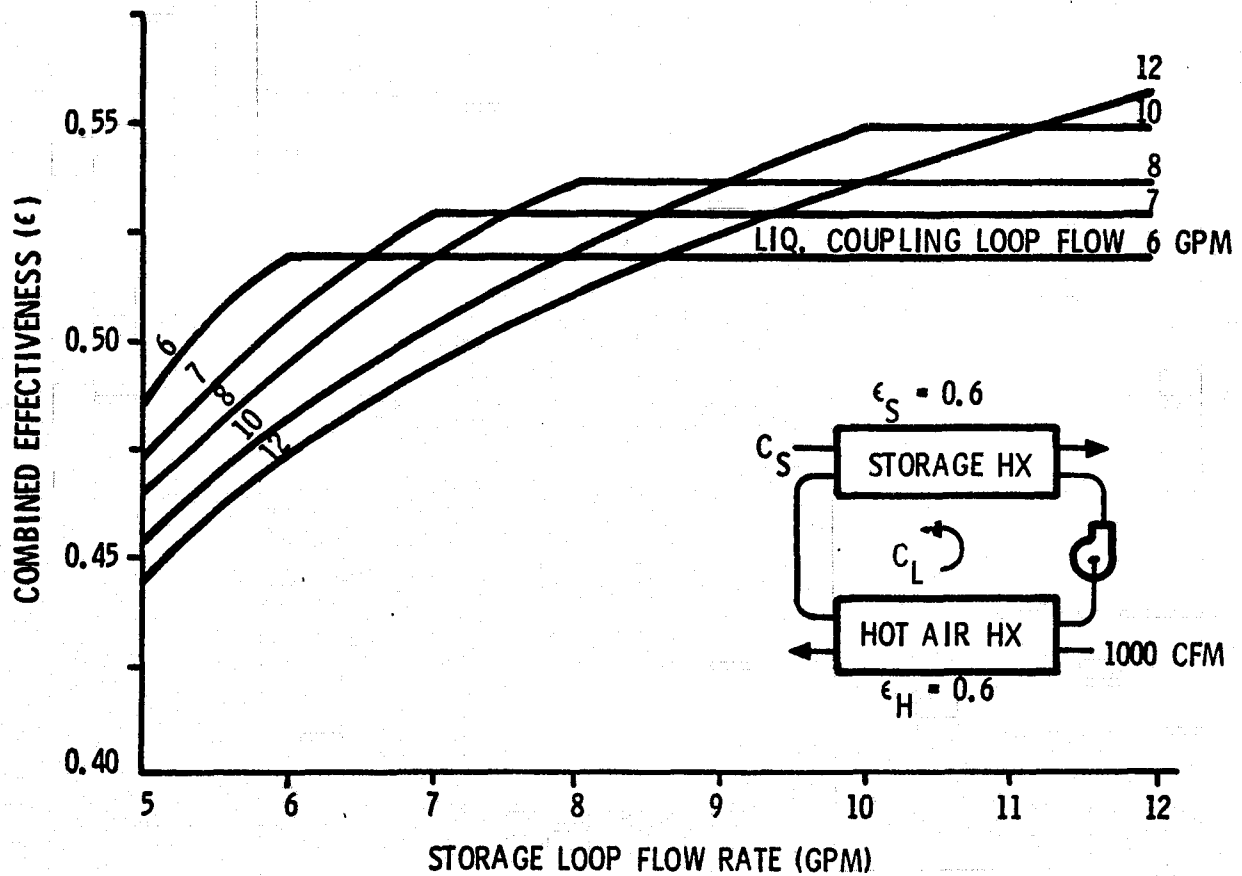


Figure 6-18. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

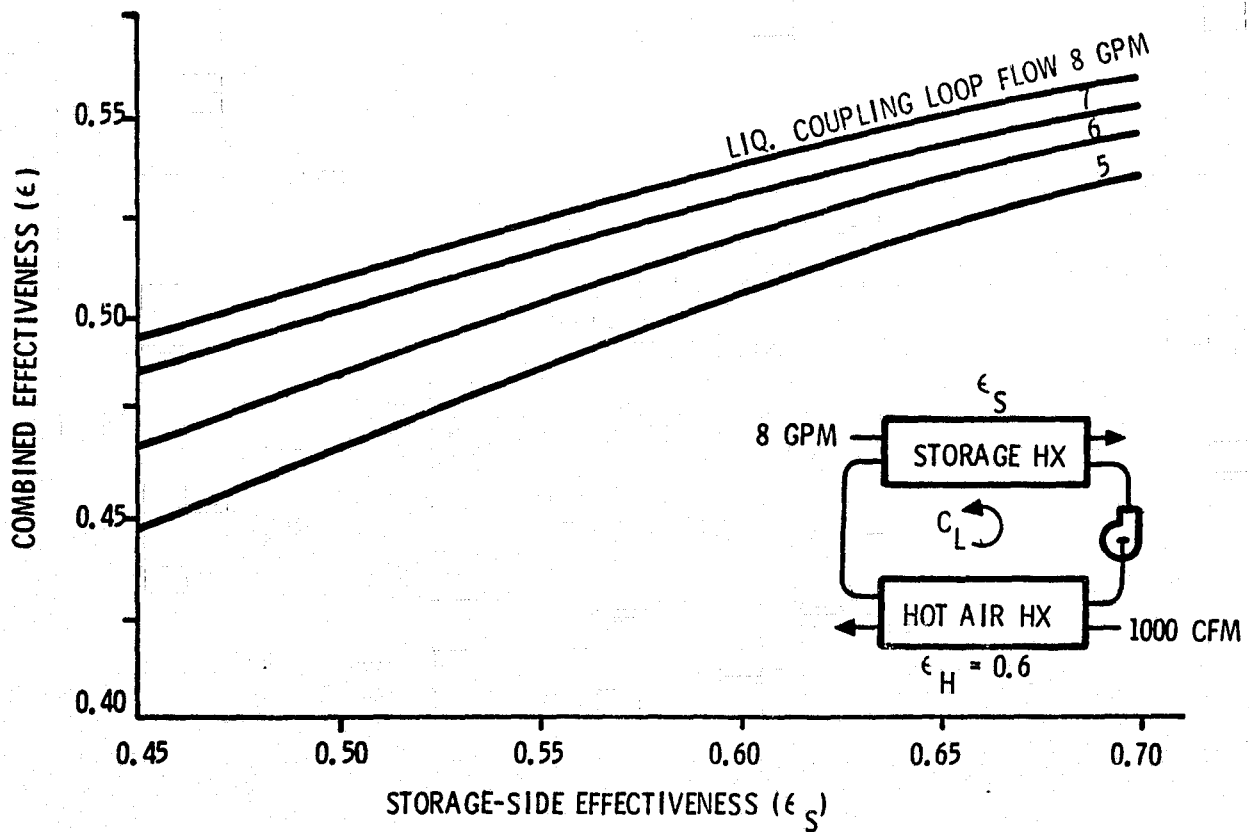


Figure 6-19. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem



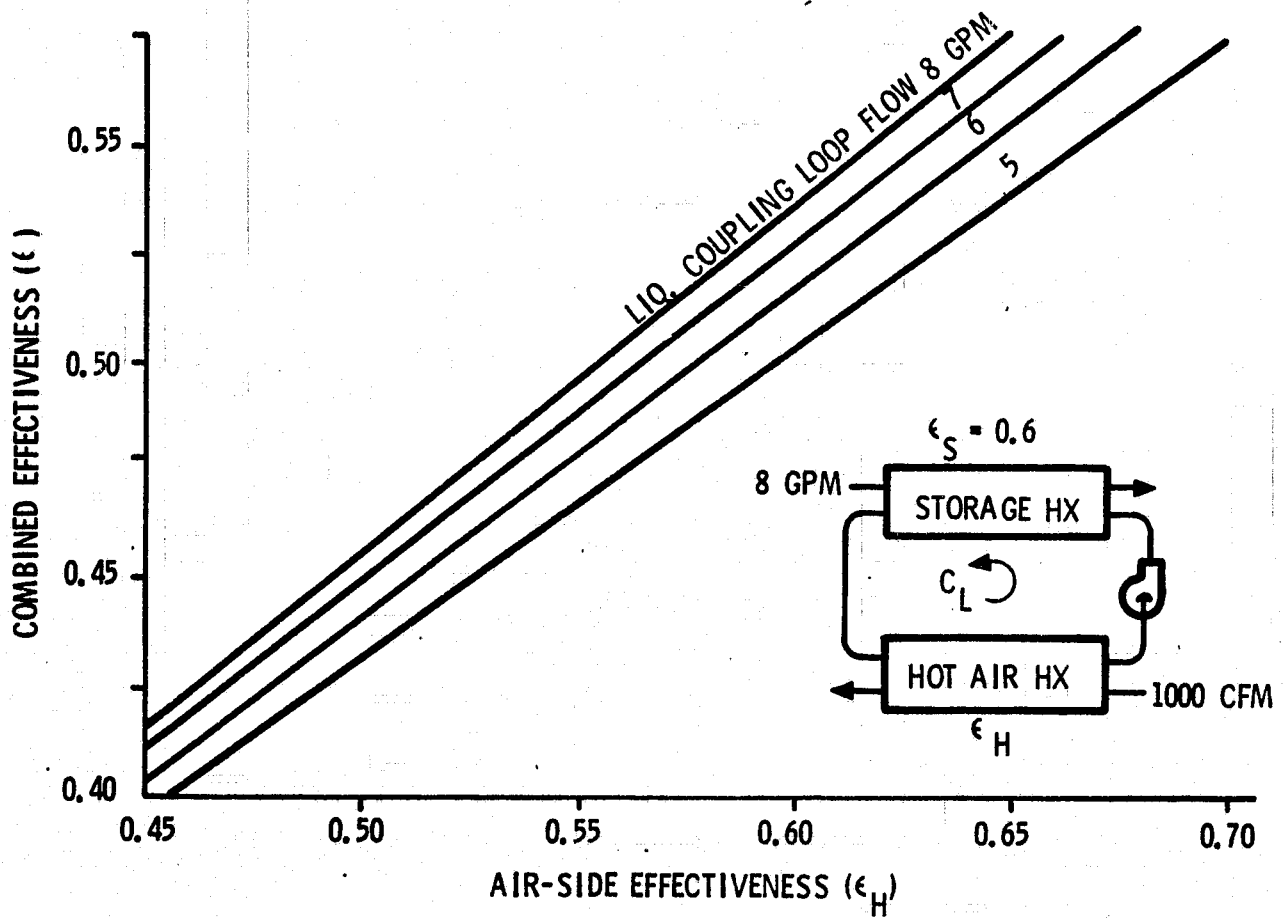


Figure 6-20. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

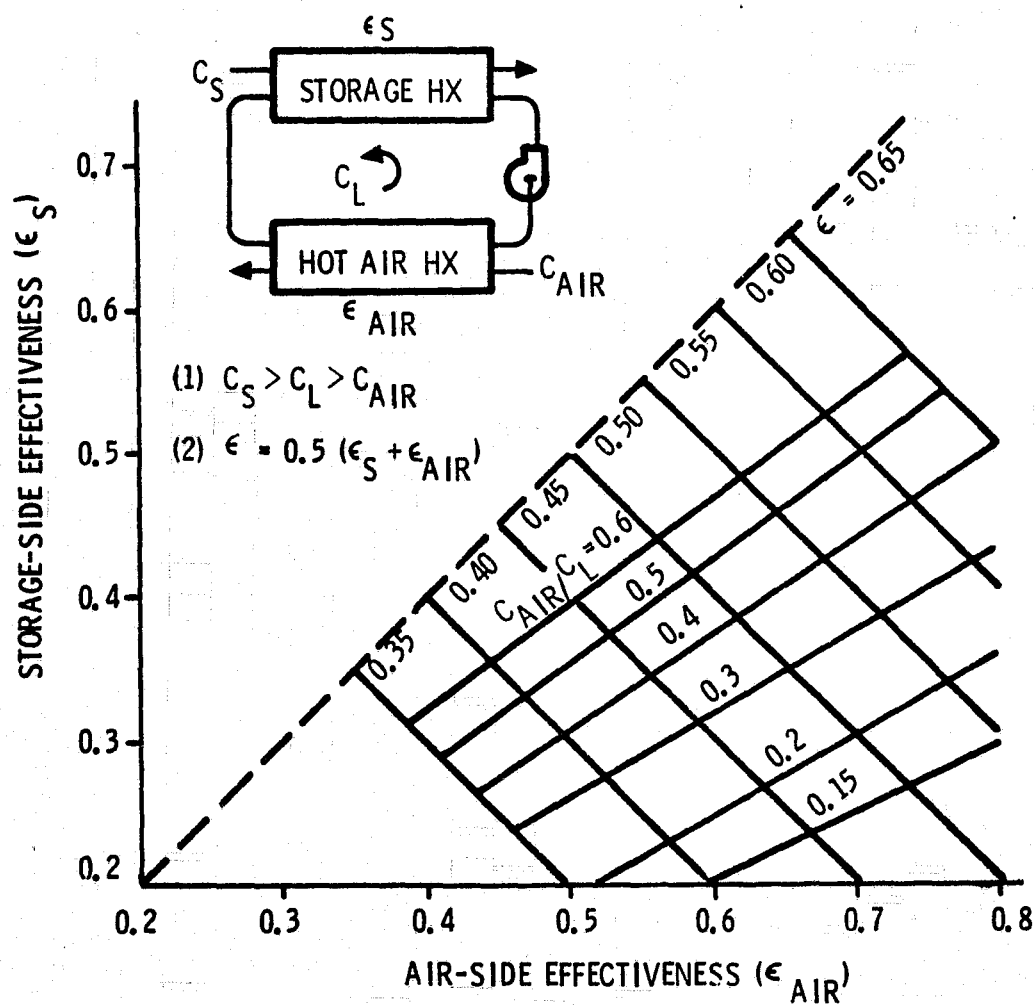


Figure 6-21. Direct-Type Heat-Exchanger Combinations for Maximum Liquid-Coupled Loop Effectiveness

- The cooling-side heat-exchanger performance is the dominant influence on the overall effectiveness of the transport system. An 8 percent overall gain is obtainable per 10 percent improvement in the cooling-side heat-exchanger performance. This is due to the small cooling media flow capacity rate relative to that of the liquid-coupling loop.

The conclusions of the analysis for the heating mode in the parameters' design range are:

- The optimum overall effectiveness occurs when the liquid coupling-to-storage loop heat-flow capacity ratio is unity. However, operation at less than unity ratio values results in an insignificant drop in overall effectiveness.
- The 7- to 9-gpm storage-loop flowrate for cooling also holds for heating purposes. Storage flow variation has a minor effect on the overall thermal performance; effectiveness increases 5 percent/gpm increase in flowrate.
- The liquid-coupling loop flowrate should be 6 to 8 gpm, and variations of flowrate in this range have a negligible effect on the overall subsystem performance. Unlike the cooling mode, increases in flowrate need not be accompanied by a storage-loop flow increase, provided that storage-loop flow is the larger of the two.
- The same conditions exist for the storage-side heat-exchanger effectiveness in the heating mode as exist in the cooling mode.
- The same conditions exist for the air-side heat-exchanger effectiveness in the heating mode as exist for the cooling-side heat exchanger in the cooling mode.

Figures 6-21 and 6-22 describe the results for component heat-exchanger optimization for the heating-from-storage mode of operation. Figure 6-21 gives component heat-exchanger combinations which optimize the subsystem overall effectiveness. The constriction which must be satisfied is that only heat-exchanger combinations with equal average effectiveness magnitudes can be compared [i.e., equal values of  $\epsilon = 0.5 (\epsilon_s + \epsilon_{air})$ ]. Figure 6-22 gives the overall subsystem effectiveness for the conditions of Figure 6-21. These results provide a basis for thermal performance and heat-exchanger combination cost analysis.

Three working-fluid candidates for the baseline heating/cooling solar systems were compared. The fluids were 50 percent aqueous solutions of Dowtherm-J (alkylated aromatic liquid), Dowfrost (propylene glycol), and Dowtherm SR-1 (ethylene glycol). General properties and cost of each fluid, and their performance relative to water were evaluated. The performance constriction was that all fluids produce an equal heat-transfer effect. Table 6-1 gives a cost comparison for the three fluids. Figure 6-23 and Table 6-2 shows fluid performance over the temperature range 100° to 215°F. Based on these results alone, Dowtherm-J was dismissed from further consideration. Dowtherm SR-1 shows somewhat better performance than Dowfrost; however, Dowfrost is somewhat less expensive.

Table 6-1. Cost Comparison of Dowtherm-J, Dowfrost, and Dowtherm SR-1 in Dollars/Gallon

Quantity	Dowtherm-J	Dowfrost	Dowtherm SR-1
4000 gal	\$3.25	\$2.70	\$2.90
70 drums	3.38	3.14	3.37
10 drums	3.87	3.31	3.56

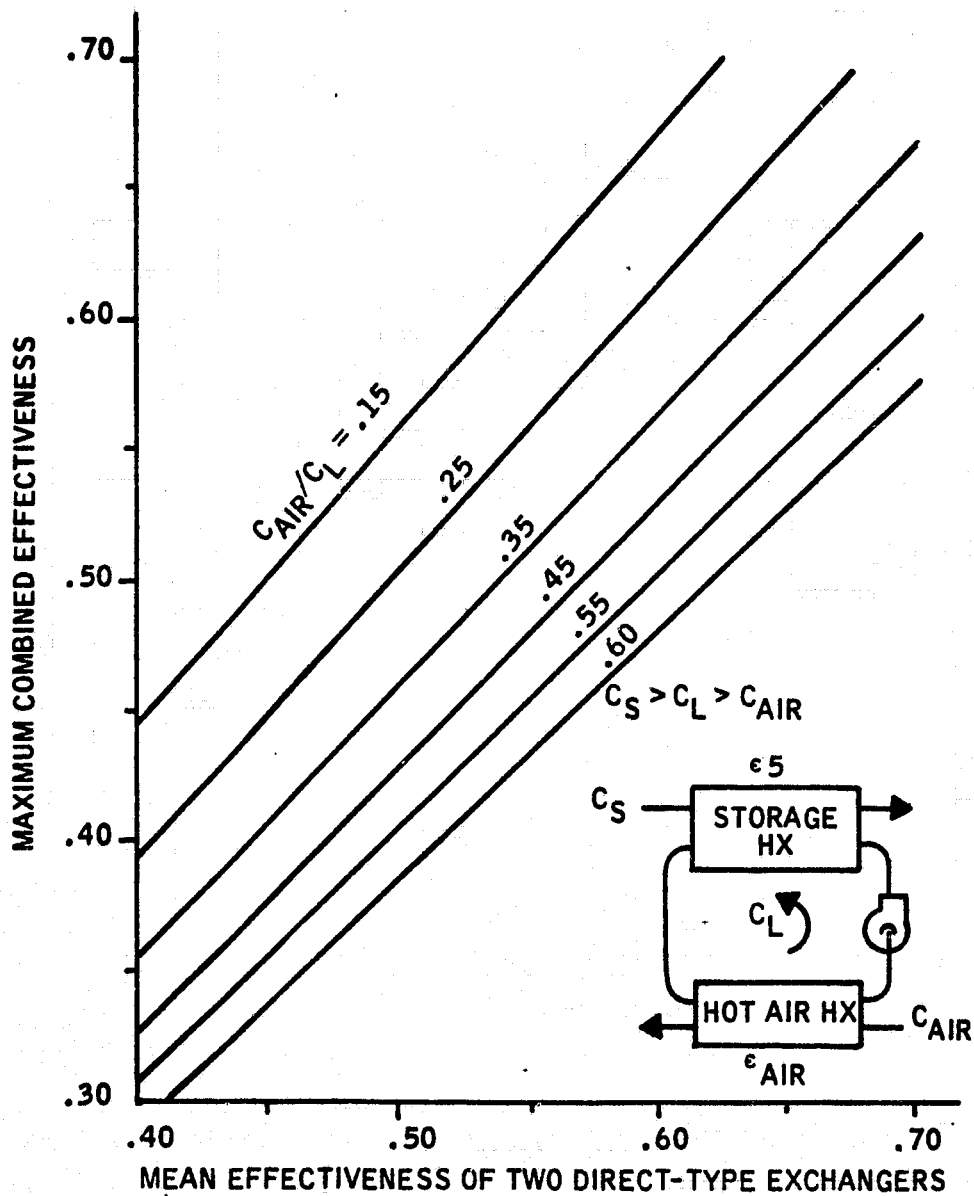


Figure 6-22. Maximum Effectiveness of Liquid-Coupled, Dual-Exchanger Heat Transport Loop

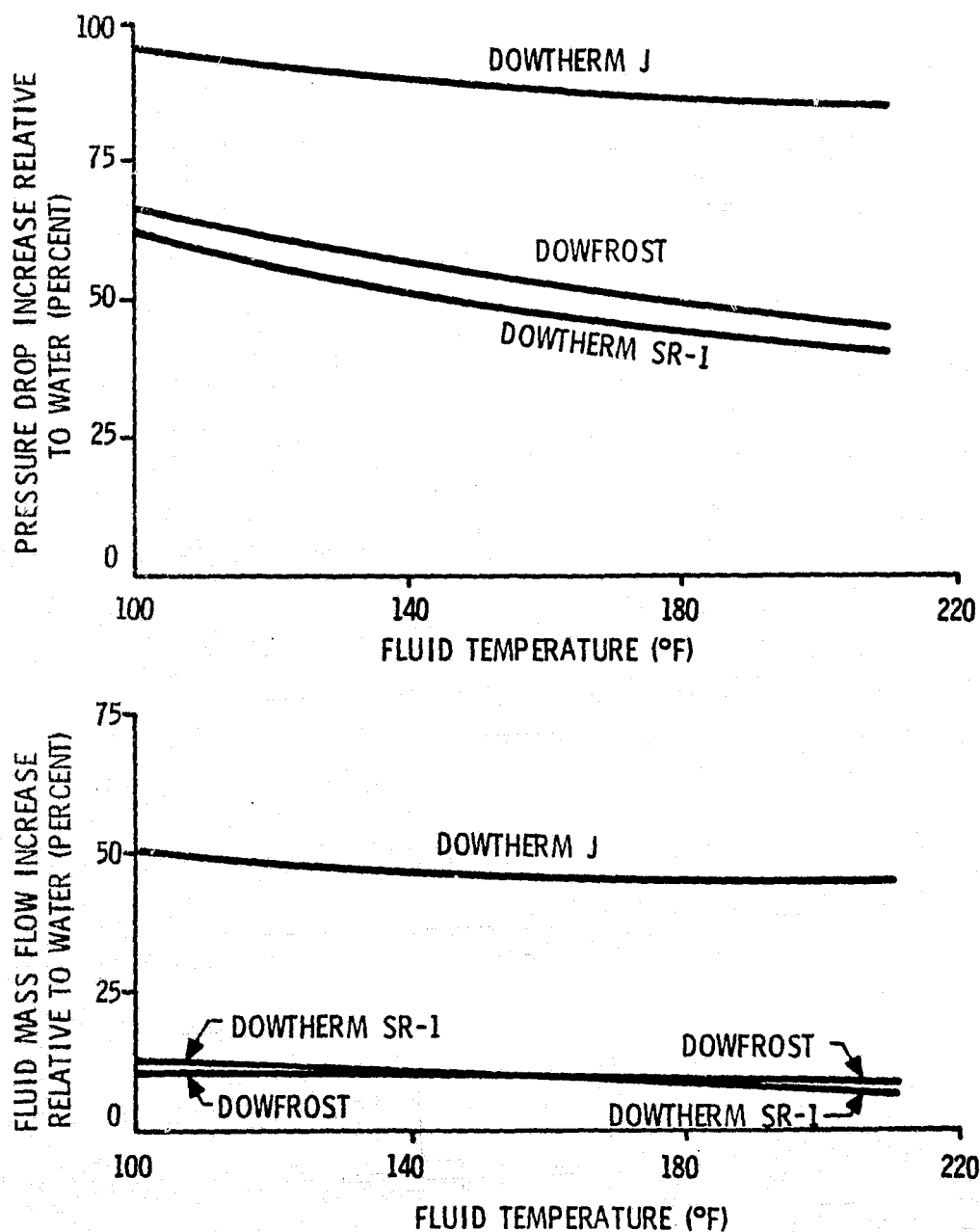


Figure 6-23. Performance of 50-Percent Aqueous Solution Relative to Water for Condition of Equal Heat Transport

**Table 6-2. Working Fluid Performance with Equal Heat Transport Criteria**

Parameter	Water			Dowtherm-J			Dowfrost (propylene glycol)			Dowtherm SR-1 (ethylene glycol base)		
Dilution (% by volume)				50			50			50		
Pressure (psia)	14.7			14.7			14.7			14.7		
Freeze point (°F)	32			-100°F for pure sol.			-28			-36		
Boil point (°F)	212			575°F for pure sol.			214			230		
Temperature (°F)	100	140	180	100	140	180	100	140	180	100	140	180
Density (lbm/ft <sup>3</sup> )	62.0	61.4	60.6	57.6	56.7	55.8	64.1	63.0	62.0	66.2	65.3	64.4
Specific heat (B/lbm-°F)	0.997	0.998	1.002	0.724	0.733	0.744	0.870	0.885	0.900	0.830	0.850	0.870
Kinematic viscosity (ft <sup>2</sup> /hr)	0.0266	0.0186	0.0139	0.0284	0.0200	0.0152	0.121	0.0625	0.0423	0.100	0.0625	0.0423
Heat capacity Rate ratio	1.0	1.0	1.0	0.665	0.678	0.685	0.903	0.910	0.919	0.889	0.907	0.923
Flow capacity ratio for same heat transfer	1.0	1.0	1.0	1.505	1.476	1.461	1.108	1.099	1.088	1.125	1.103	1.083
Reynolds number ratio relative to water	1.0	1.0	1.0	1.409	1.373	1.335	0.244	0.327	0.358	0.299	0.328	0.356
Friction factor ratio relative to water	1.0	1.0	1.0	0.942	0.946	0.951	1.280	1.216	1.197	1.235	1.215	1.198
Pressure drop ratio relative to water	1.0	1.0	1.0	1.952	1.902	1.870	1.625	1.507	1.450	1.669	1.573	1.495

In general, in addition to low-temperature freeze protection and high boiling point, Dowfrost and Dowtherm SR-1 exhibit numerous similar characteristics. Some of these are:

- Stability over wide temperature range
- Low coefficient of thermal expansion. If the aqueous-glycol solution undergoes complete freeze-up, upon thawing it will not crack the container.
- Noncorrosivity with inhibitor additives
- Completely nonflammable when above 20 percent aqueous solution
- Extremely low toxicity
- Annual check of the solution inhibitor level is normally required.

It should be noted that Dowfrost and its inhibitor meet the requirements of the Food Additives Regulation which lists those materials "generally recognized as safe" for use in foods.

Aqueous solutions of both Dowfrost and Dowtherm SR-1 appear to be capable working fluids for the baseline heating/cooling solar system. However, before a final decision is made, several other prospects, such as silicone oil, shall be investigated.



## 6.8 CONTROLS SUBSYSTEM

The control subsystem design is based on the overall system definition that includes related subsystems such as collectors, solar storage, energy transport, and auxiliary heating/cooling. Control logic is determined so that solar energy is collected and used to minimize the consumption of conventional energy while maintaining conditions in the space consistent with the type and duration of occupancy. Through engineering analysis, operating modes that could be hazardous or lead to discomfort are eliminated. The heating or cooling system demand for energy is satisfied by using solar energy before adding conventional energy required to satisfy the load. Interrelated subsystems that affect the operating efficiency of other subsystems are also analyzed. This analysis determines a control logic that leads to a higher overall system efficiency. For example, if stored energy is higher than collected energy, the control subsystem prevents dissipation of the stored energy into the collector subsystem.

The control logic is used to design a control schematic of the subsystem and to determine the operating temperatures, pressure, flow, and differential-temperature requirements of the system. This information is then used to determine component requirements. The component requirements are then used to establish availability of off-the-shelf deliverables and items requiring complete component development. Off-the-shelf deliverable controls would include standard thermostats, control valves, motor operators and linkages, switching and time-delay relays, time clocks, etc. These are all standard components as used in the HVAC industry.

The differential-temperature controller is also a standard production item. This controller, when used with the proper thermistor sensors, is designed as a module capable of providing a variety of automatic control functions in the switching of circulating pumps, valves, dampers, motors, and other accessories used in solar control systems. It has a solid-state differential

amplifier with a three-pole switching relay. Control functions can be modified by changing the connections of the differential resistors, setpoint resistors, and thermistor sensors. This module can be used as a:

- Differential-temperature control (relay makes on temperature differential rise)
- Setpoint temperature control (relay makes on temperature rise)
- Setpoint temperature control (relay makes on temperature drop)

The differential-temperature controller has the following design features:

- Modular construction (one basic module provides a variety of solar control functions)
- Solid-state differential amplifier
- Three-pole switching relay (two N.O. and one N.C. isolated contacts)
- Integral transformer for powering the low-voltage control circuit
- Color-coded leadwires for line-voltage connections
- Exposed terminal strip with screw terminals for low-voltage connections
- Plug-in differential resistors
- Mounts in any position on a standard 4 x 4-inch junction box
- Interchangeable thermistor sensors

Control equipment requiring complete component development would include the control panel and the motor control panel.

The logic control panel will provide the control logic needed to efficiently and safely operate the solar-energy collection, transport, and auxiliary-energy subsystems. This is more complex than conventional systems of similar size and application. For this reason, the control logic is contained on a printed-circuit card using multipole relays or solid-state devices whose control and interconnection requirements are pre-engineered and prewired. This allows high reliability and contributes to simplification of the installation and maintenance of the control subsystem. To accomplish this, a logic control panel (panels) will be developed and tested. It is anticipated that two unique logic panels will be required, one for the heating-only subsystem and one for the heating and cooling subsystem. The additional logic specific to single-family, multiple-family, or commercial applications will be addressed in the master control panel discussed below.

The master control panel will also require complete component development. As stated above, the control system requirements are more complex for combined solar-energy and conventional-energy systems than are those for conventional systems alone. In order to ease the installation and maintenance problems, a master control panel for each unique control system will be developed. It is anticipated that all components of the control subsystem not required to be remotely located to sense temperature or control flow will be enclosed in the master control panel. This includes the differential-temperature controller, logic control panel, time clock, time-delay relays, and power relays. Electrical connections to remote sensors and actuators will be easily identified via terminal connections. The use of the master control panel also allows a control subsystem to be fabricated, prewired, pretested, and completely checked out during acceptance and before delivery to installation sites. By using standard components in addition to the logic control panel controller, six unique master control panels will be developed. They are: 1) heating-only single-family; 2) heating-only multiple-family; 3) heating-only commercial; 4) heating/cooling single-family; 5) heating/cooling multiple-family; and 6) heating/cooling commercial. As site specific data become available, these control panels will be developed.

### 6.8.1 Development and Qualification Tests

During development, the components listed above will be tested at the breadboard level using complete simulation of each operational mode for the system. In addition, failure modes, such as power restoration after power loss, will be tested to ensure that no damage or unsafe condition that can be reasonably prevented will occur due to control subsystem malfunction. In addition, tests will be run to determine the best human-engineering factor for the control system, with the goal being to have a system requiring no more "operator expertise" than presently required for a conventional system of similar size and application. The IPC and other applicable criteria will be used to determine development and performance requirements.

At the completion of breadboard testing, printed-circuit board testing of the logic controller will be done to assure design and development accuracy. No tests are anticipated for the control subsystems during the qualification phase, as components used during development will be standard manufacturer's catalog items tested during development, or tested in the acceptance phase as discussed under acceptance testing.

### 6.8.2 Acceptance Testing

Each of the 12 master control panels will be tested using all system operation modes as described for the heating and heating/cooling subsystems. This includes tests of the logic controller and differential-temperature controller. Remote thermostat sensors and control valves are mass-produced standard catalog items used in large quantities in similar applications and service and will not be tested prior to installation.

## 6.9 SITE DATA COLLECTION SUBSYSTEM

The preparation of the detailed flow schematic used to identify the instrumentation desired will be completed shortly after the specific sites are identified. Typically, this effort will be done in parallel with the preparation of the installation drawings. The evaluation factors listed in Table 3-1 of referenced SHC-1006 will be used when preparing this instrumentation-oriented schematic. The, specific sensors will be selected from the Acceptable Sensor List, reference Table 3-2 of SHC-1006, and a Prepared Instrumentation Plan prepared. This plan will be site-specific (i.e., six plans for heating/cooling systems) dependent. When approved (i.e., AIP), it will be forwarded for the site owner/architect. Following that, the site owner/architect will prepare the detailed construction drawings, and, using these, the HVAC contractor will provide the installation costs. A modified list of action "steps" shown in Table 1-1 of SHC-1006 is included herein as Table 6-3.

Table 6-3. Instrumenting Steps\*

Step	Action by	Action
1	NASA	Supplies "instrumentation installation guidelines" to Honeywell.
1A	NASA	NASA/MSFC supplies sites and site data.
2	Honeywell	Prepares complete solar system schematics showing instrumentation locations for each site.
3	Honeywell	Submits Proposed Instrumentation Plan (PIP) to NASA for review and approval (without costs).
4	NASA	Reviews and modifies PIP as necessary. Supplies Honeywell with an Approved Instrumentation Plan (AIP)
4A	Honeywell	Provides site owner/architect with AIP.
4B	Architect	Provides Honeywell with detailed construction drawing.
4C	Honeywell	Obtains costs to install from HVAC contractor.
5	NASA	Negotiates a contract modification with Honeywell to implement the AIP.
6	NASA	Provides and ships all necessary instrumentation as Government-Furnished Equipment (GFE) to the demonstration site for HVAC contractor installation and checkout.
7	NASA	Arranges for the installation of the necessary telephone transmission equipment to accommodate the Site Data Acquisition Subsystem (SDAS).
8	Honeywell	Informs ERDA that the AIP has been implemented and that all instruments have satisfactorily undergone checkout.
9	NASA/ ERDA	Delivers and installs an SDAS and confirms the satisfactory transmission of the data to the Central Data Processing Facility.
10	ERDA/ NASA	Receives and evaluates the incoming performance data, and prepares monthly and annual performance reports using standardized data reduction procedures.
11	NASA	Mails performance reports for each site to Honeywell within 30 days after the end of each performance period.
12	ERDA/ NASA	Removes SDAS from the site at the end of the test period.
13	Honeywell	Upon NASA's direction, removes all instrumentation which can be cost-effectively removed and returns to NASA at the end of the test period.

\*Reference SHC-1006, Table 1-1: Procedure for Instrumenting Solar Heating and Cooling Commercial Demonstration Projects.

## 6.10 ELECTRICAL SUBSYSTEM

The electrical subsystem consists of those components necessary to deliver power to the subsystems. The wiring, conduit, and installation hardware are best determined after specific buildings have been identified. At that time, complete drawings and specifications can be completed. The components of this subsystem which can be identified and can be designed for electrical requirements are:

- Cooling tower fan motor starter
- Purge coil fan motor starter
- Magnetic motor starters for pumps
- Fused disconnect switches
- Circuit breakers
- Air handler

Equipment sizing is based on National Electrical Manufacturer's Association (NEMA) standards, and all items purchased will be in accordance with Underwriter's Laboratory requirements and will bear the UL label of acceptance.

## 6.11 COOLING SUBSYSTEM

### 6.11.1 Cooling Subsystem Schedule

The cooling subsystem design and hardware activity schedule is shown in Figure 6-24. The coordinated design runs in parallel with performance interface data exchanged continuously throughout the detail design activity. Of significance is the major milestone of parts ordering. This occurs approximately 1 February for the Rankine-cycle 3-ton units and during March for the 25-ton units.

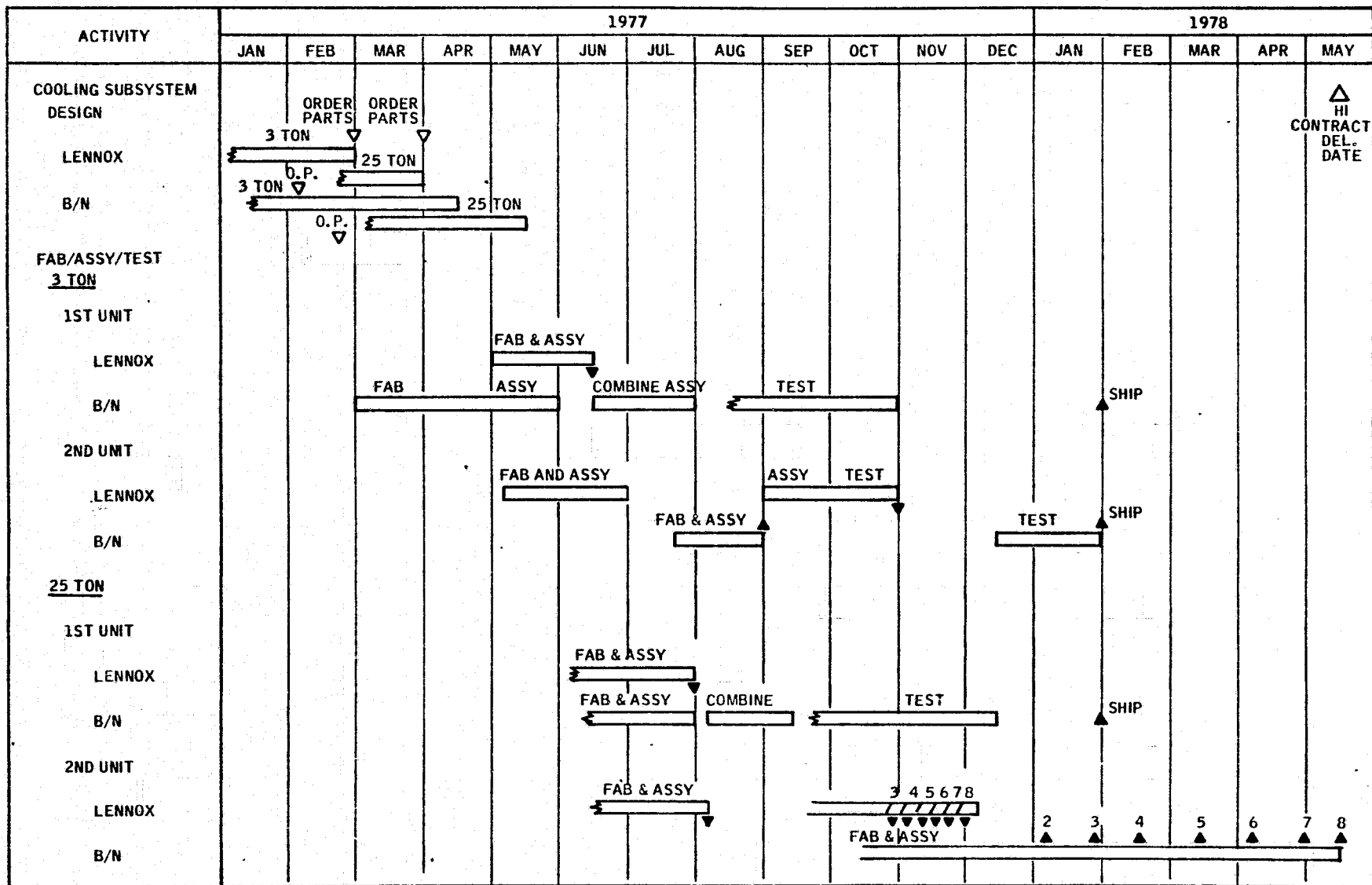


Figure 6-24. Cooling Subsystem Schedule



The fabrication of the 3-ton units at Barber-Nichols needs to start early in February in order to get the development work done and still leave enough time to produce the units for delivery. The constraining activities are the fabrication and test of the 25-ton units. Both 3-ton units will be completed by February 1978.

The shipping arrows on the chart show physical movement of the hardware from one facility to another. On the first 3-ton unit, the respective elements are fabricated at Lennox and Barber-Nichols; then Lennox ships their portion to Barber-Nichols for integration and testing. On 3-ton unit No. 2, Barber-Nichols ships the Rankine-cycle equipment to Lennox for final assembly so packaging and sound tests can be conducted using their facilities. Then, the unit is shipped to Barber-Nichols for final testing.

On the 25-ton units, all assembly integration is done at Barber-Nichols, as well as full unit performance testing.

### 6.11.2 Cooling Subsystem Description

**6.11.2.1 General** -- Systems developed for this project will be new in design, much different from any product that Lennox currently produces. The equipment includes two sizes, one to meet the needs of the residential market and one to meet the needs of the multiple-family/commercial market.

The model codes for the residential and the multiple-family/commercial units are RSH1 (Rankine-powered high side) and RCWS1 (Rankine-powered chilled-water system). The code letters and series designation will be followed by the size, phase, dash number, and voltage designation. The complete model numbers are as follows:

- Residential unit - RHS1-411V-1P
- Multiple-family/Commercial unit - RCWS1-3053V-1Y

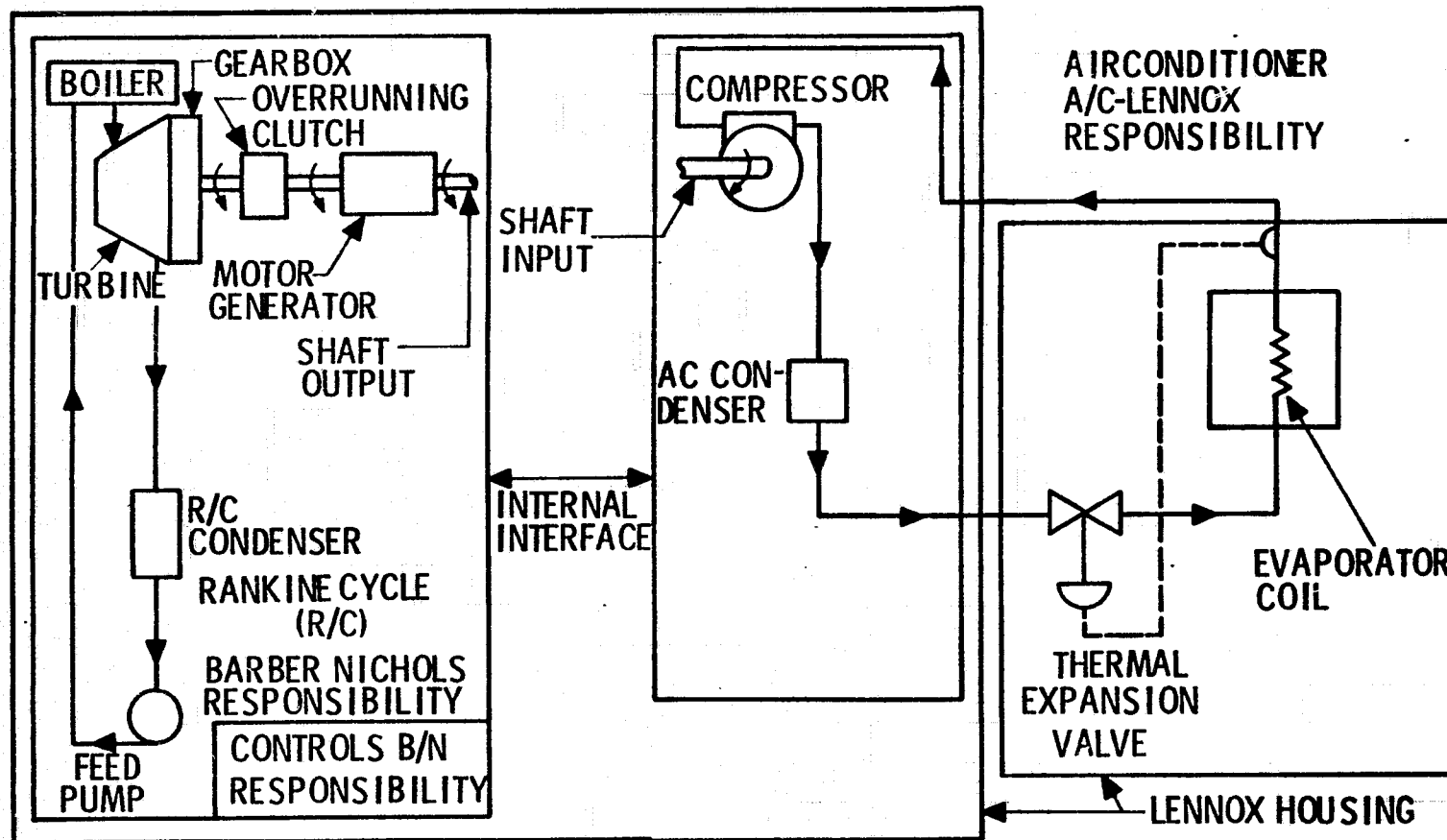
**6.11.2.2 Interface --** Under present agreement, Lennox is to supply the refrigeration subsystems, sheet-metal cabinetry, and structural framework containing both the refrigeration subsystem and the Rankine-cycle engine. Barber-Nichols Engineering will develop and supply the Rankine-cycle engines.

Both residential (Figure 6-25) and multiple-family/commercial (Figure 6-26) refrigeration subsystems use open-type compressors and water-cooled condensers. The residential unit is a "split system" by category and is linked to a direct-expansion evaporator coil. The multiple-family/commercial unit is "packaged" by category and incorporates a liquid chiller as the evaporator to supply chilled water to the conditioned space.

**6.11.2.3 Operation --** The cooling subsystem consists of two distinct thermodynamic portions, the Rankine-cycle (R/C) power loop and the air-conditioning (A/C) loop. The combined RC/AC subsystem is shown schematically in Figure 6-27.

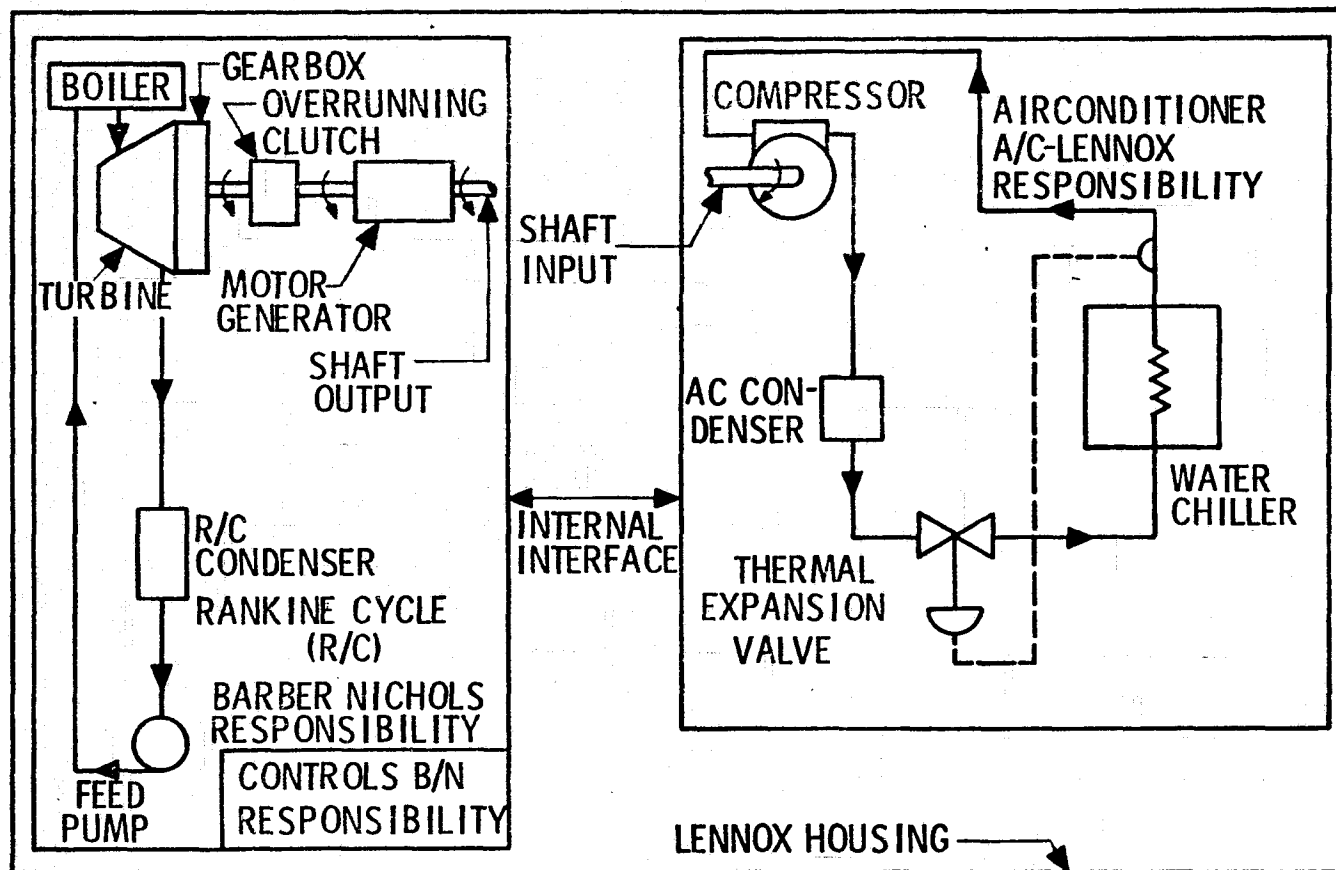
In the Rankine cycle, working fluid is pumped from a water-cooled condenser through a regenerator to the boiler, which extracts heat from solar collector water. The regenerator is a liquid-to-vapor performance improvement heat exchanger operating within the R/C loop. Vapor leaving the boiler is admitted to nozzles which feed a turbine rotor. Turbine exhaust vapor passes through the vapor side of the regenerator and returns to the condenser, completing the Rankine cycle.

Turbine rotational speed is reduced by a gearbox whose low-speed shaft is connected by an overrunning clutch to a motor-generator and air-conditioning compressor. This configuration permits total input power to the A/C loop from the solar-powered R/C. If the R/C system cannot keep up with the cooling demand (i.e., if it is a partially cloudy day), rated cooling can be maintained with the help of the motor.



6-51

Figure 6-25. Residential Cooling Subsystem Interface



6-52

Figure 6-26. Multiple-Family/Commercial Cooling Subsystem Interface

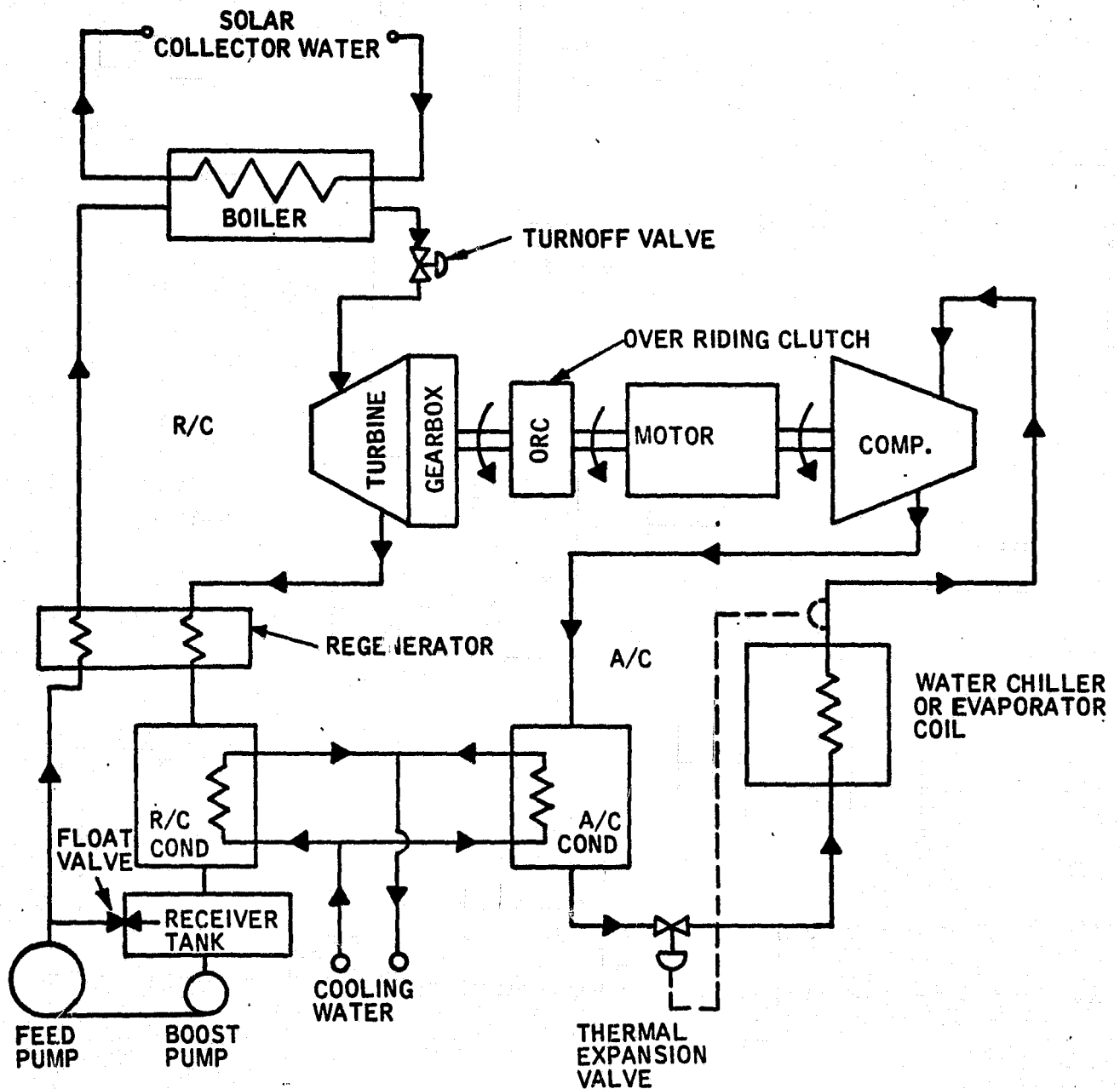


Figure 6-27. General System Schematic for Rankine-Cycle Air-Conditioning System

In the air-conditioning cycle, a compressor receives low-pressure refrigerant vapor from the evaporator (or chiller) and pressurizes it. The high-pressure vapor then enters a water-cooled condenser where the latent heat of vaporization is removed, leaving high-pressure liquid refrigerant. This liquid is allowed to pass through a thermal expansion valve to the low-pressure portion of the loop (i.e., evaporator). The expansion of this high-pressure liquid produces a mixture of refrigerant liquid droplets and vapor at a low temperature (about 45°F). This low temperature provides sufficient temperature difference to extract heat from air or water, as desired, and thus supply cooling. The energy taken from the air or water in the cooling process supplies heat of vaporization to the droplets of refrigerant liquid, producing refrigerant vapor. This low-temperature, low-pressure vapor then flows to the compressor, completing the A/C cycle.

### 6.11.3 Refrigeration Design Approach

The Lennox approach to the NASA-404 Program is to develop a product which will economically bind together energy efficient hardware, consumer safety, producibility, and marketability in a common package. An economical design will be achieved by striving to select those components which are in common use in today's industry. The method by which these components are applied will differ from current designs, however.

The skin for the cabinetry of both the residential unit and the multiple-family commercial unit will be an adaptation of cabinetry currently used to contain Lennox CHA9 and DSS1 product lines. The present cabinets will be dimensionally modified to meet the package requirements of the 50 inches wide by 40 inches high by 56 inches long residential unit, and the 84 inches wide by 65 inches high by 120 inches long multiple-family commercial unit. Figure 6-28 is an isometric view of the residential unit.

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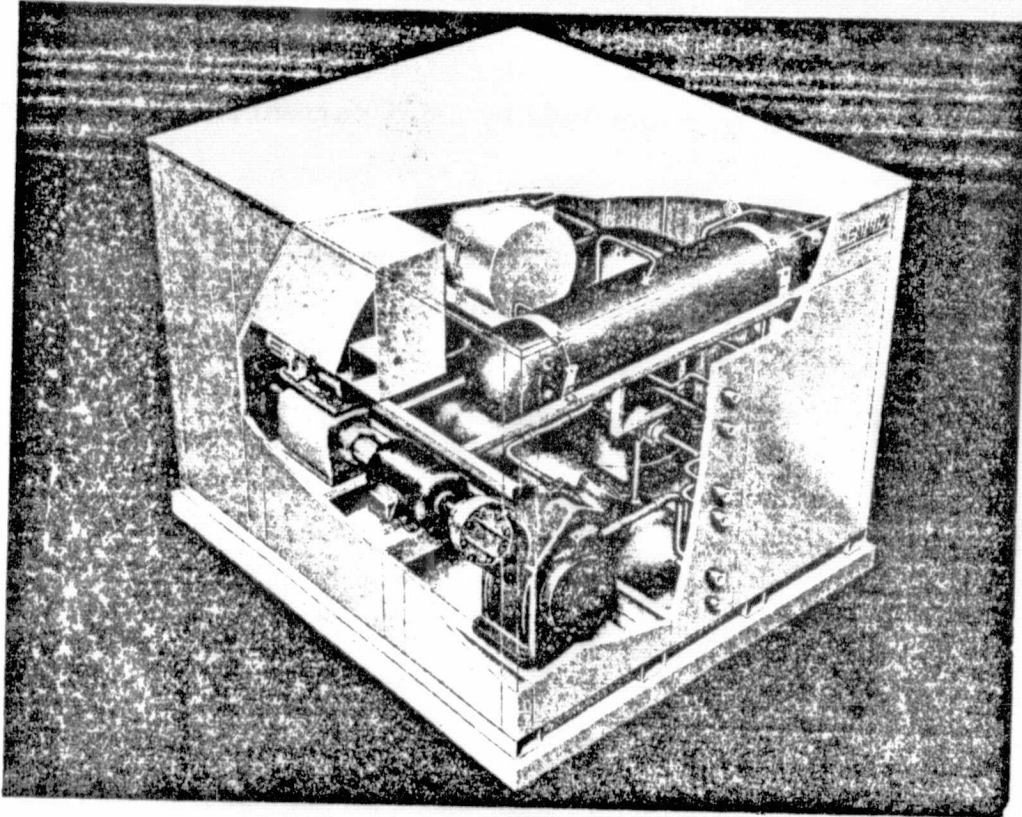


Figure 6-28. Isometric View of Residential Refrigeration Unit

The units will be painted with Lennox industrial-grade acrylic enamel finish. This finish meets the Lennox specifications, which have been developed in accordance with American Society for Testing Material (ASTM) Standard D-1654 (Evaluation of painted or coated specimens subject to corrosive environments) and Standard D-714 (Evaluating degree of blistering of paints). Testing consists of a 500-hour salt spray test, which is conducted in accordance with ASTM Standard B-117.

Marketable aesthetics will be maintained by using current Lennox silhouette and color schemes.

Manufacturability will be achieved by using the concept of modularity. This concept will use advanced subassembly of several modules that will bolt and braze together to form the final package. This will be a vital format for economical manufacturability. The Lennox refrigeration module and the Barber-Nichols heat-exchange module and power module occupy separate sections of the cabinet as shown in Figure 6-29.

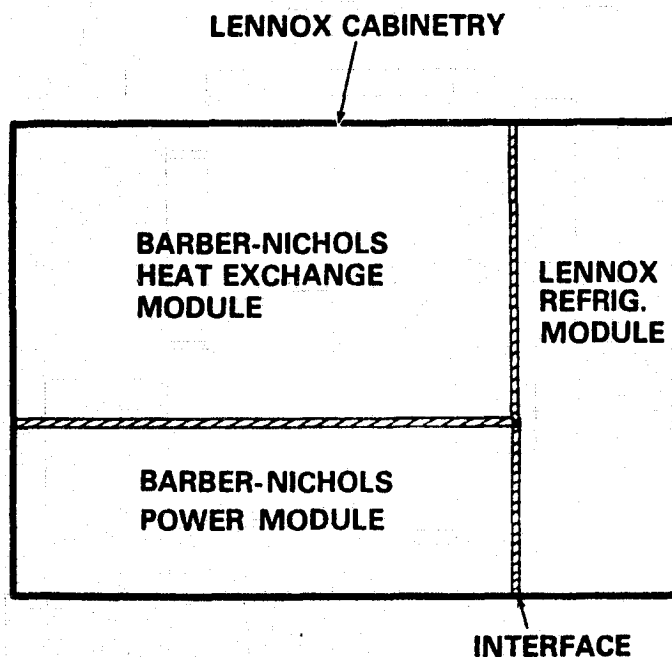


Figure 6-29. Plan View of Residential Unit Modularity



This independence allows Lennox to assemble the refrigeration module into the cabinet for shipment to Barber-Nichols for final assembly and test. Independent module tests can be performed during assembly to determine performance prior to final assembly.

Both the residential unit and the multiple-family commercial unit will be designed for ease of installation and serviceability. Lennox feels that if the installer and servicemen's job is made easier, he will then do a better job during the initial installation and a more thorough job maintaining the system.

Field connections will be centered in one location of the unit. All plumbing connections will be made in this location, external to the cabinetry. This central location will include large access panels to facilitate ease of entry to moving mechanical components and electrical controls as shown in Figure 6-30.

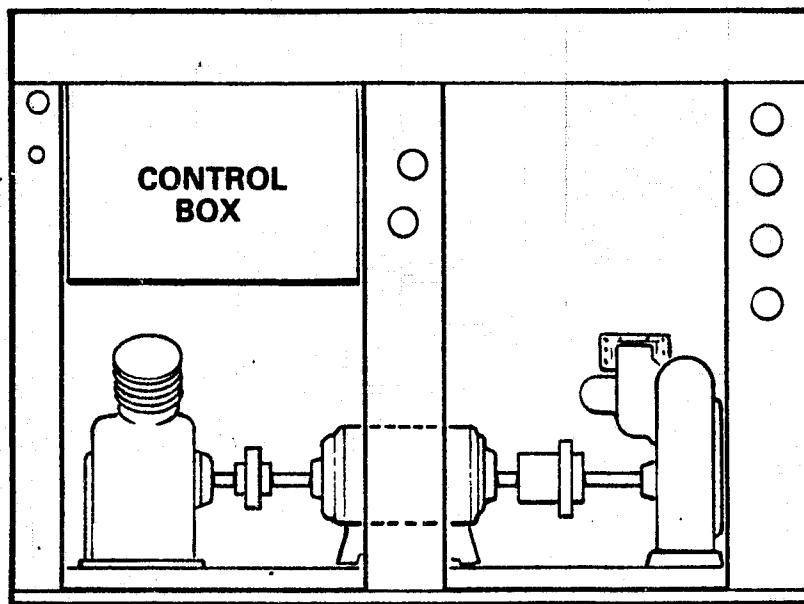


Figure 6-30. Multiple-Family/Commercial Unit Serviceability  
(Access Panels Removed)

6.11.3.1 Component Hardware -- Lennox will make every attempt to make this program as practical as possible. Component selection will incorporate the off-the-shelf concept in its design whenever possible. By using this concept, Lennox feels that an extremely practical cost-effective system will result. An energy-efficient system will result by applying standard components in an unconventional manner.

Lennox is researching possible compressor options available in the marketplace. The present options being examined are (Figure 6-31):

- Automotive (reciprocating and swash plate)
- Reciprocating
- Centrifugal
- Rotary

Elements of concern for compressor selection include:

- Capacity
- Brake horsepower input
- Coefficient of performance
- Input speed
- Lubrication
- Reliability
- Physical size and weight
- Refrigerant type
- Cost

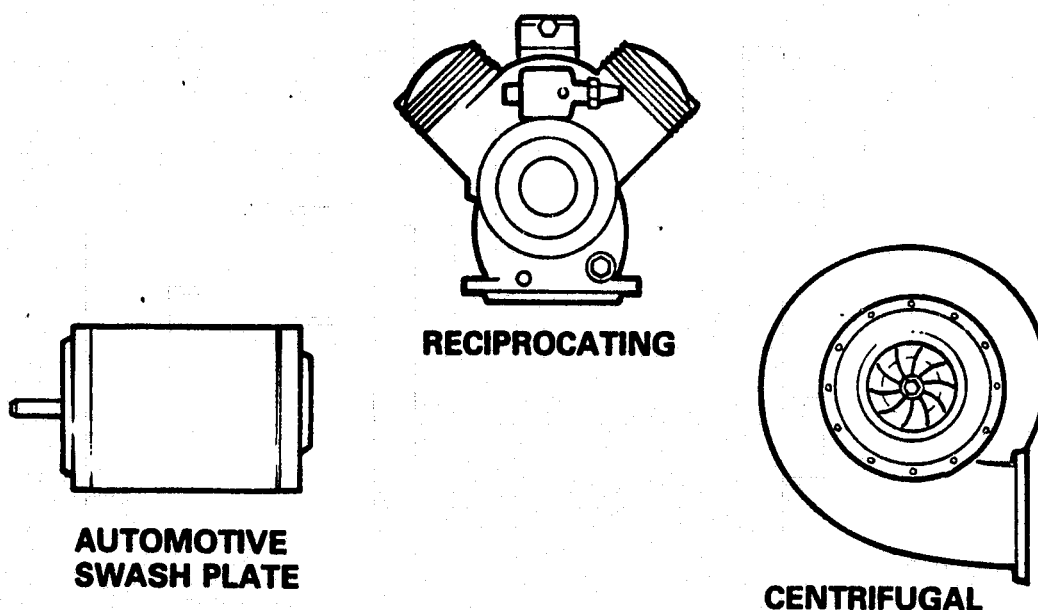


Figure 6-31. Compressor Options

Each of these items is under consideration in a time study scheduled for completion 15 January 1977. Results of the study will be presented at the Preliminary Design Review (PDR). These results determine the final interface between the compressor, motor, gearbox, and turbine. Horsepower has been selected. The remaining critical parameter is rpm, which depends on the type of compressor. It is desirable that this rpm be in a range compatible with a motor synchronous speed (1700 or 3600 rpm) so that the direct drive baseline can be maintained.

Shaft alignment and coupling interface will be carefully coordinated between Lennox and Barber-Nichols (Figure 6-31). Present plans call for rigid mounting of the Rankine-power module and refrigeration compressor on a set of common rails. The rails will be fixed to the unit base by spring mounts. This will allow the rigid assembly to be free floating and hence eliminate any extreme amounts of vibration from being transferred the main body of the unit.

In addition to the spring mounts, flexible connections will be included in all plumbing lines that are interfaced between the Rankine-power module, the refrigeration compressor, and the remaining unit modules as shown in Figure 6-32.

Detailed analysis of evaporator and coil considerations are in process and the status of these trades will also be presented at the PDR.

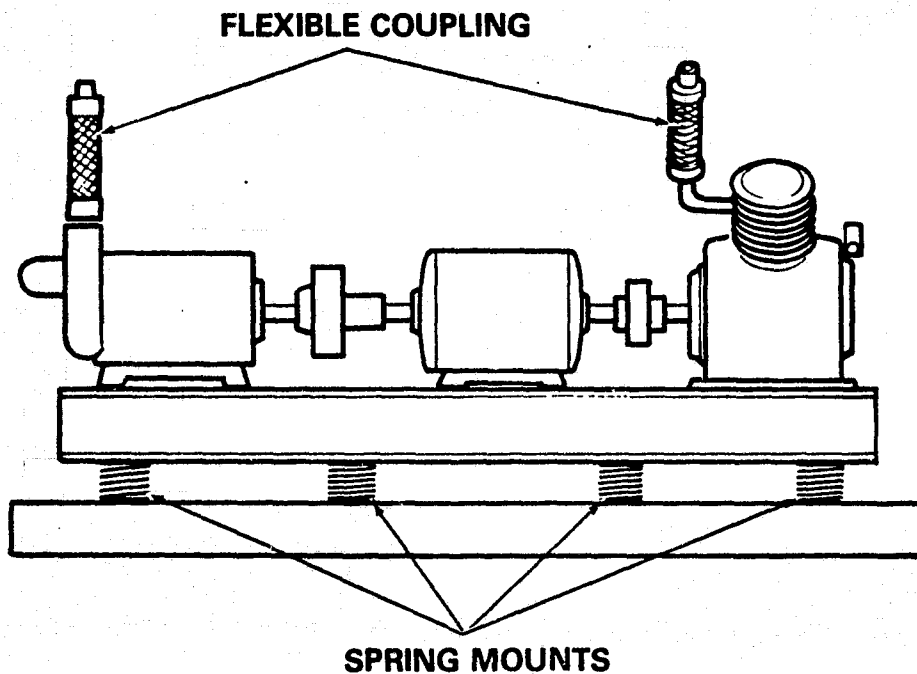


Figure 6-32. Shaft Alignment and Vibration Isolation

### 6.11.3.2 Controls --

- Residential Refrigeration Module (Figure 6-33) -- Standard controls to be included in the residential refrigeration module are:
  - Electrical: High-pressure switch, low-pressure switch, oil safety switch, crankcase heater and relay (outdoor installations)
- Multiple-family/Commercial Refrigeration Module (Figure 6-34) -- Standard controls to be included in the multifamily-commercial refrigeration module are:
  - Mechanical: Thermostatic expansion valve, compressor capacity control
  - Electrical: High-pressure switch, low-pressure switch, oil safety switch, crankcase heater and relay (outdoor installations) Freeze stat.

Lennox will, whenever possible, use control components currently applied in Lennox equipment to assure component reliability.

### 6.11.3.3 Refrigeration Module Performance -- Present design goals call for the following performance characteristics for the refrigeration modules:

- Residential Subsystem:
  - Capacity - 36,000 Btuh
  - Bhp Input - 2.4 Bhp
  - COP - 6.0\*

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\* Per ARI Standard 210-75

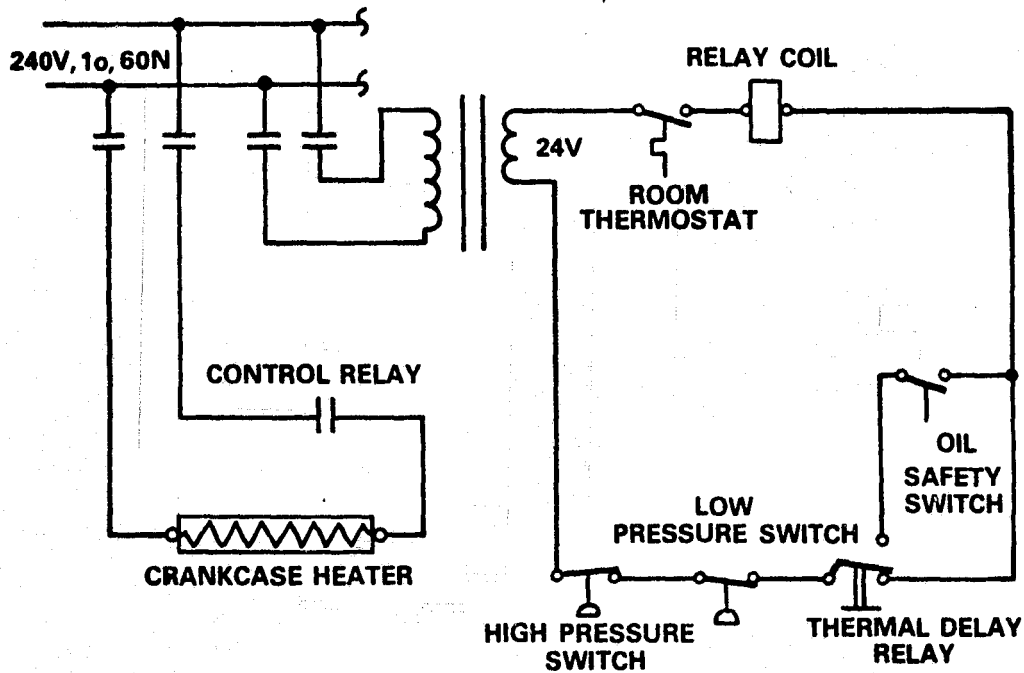


Figure 6-33. Residential Refrigeration Module Control Circuitry

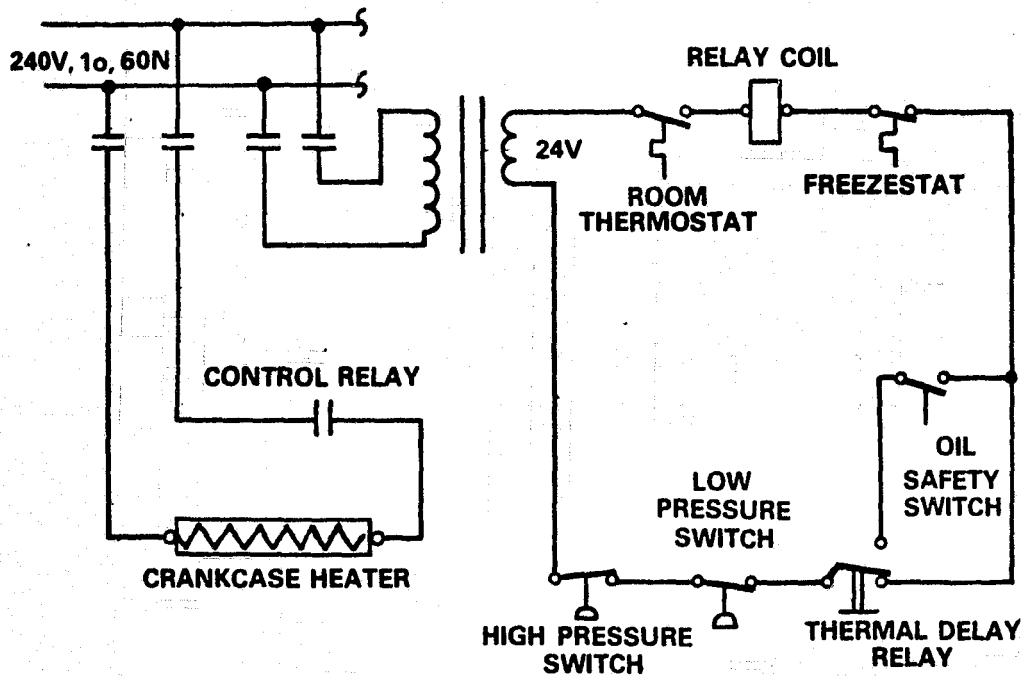


Figure 6-34. Multiple-Family/Commercial Refrigeration Module Control Circuitry

- **Multiple-family/Commercial Subsystem:**

- Capacity - 300,000 Btuh
- Bhp Input - 19.6 Bhp
- COP - 6.0\*\*

Sound rating numbers will not be a set objective under this project. There will be no sound numbers established by testing; nevertheless, Lennox will remain sound conscious during all phases of equipment design. To give example of sound conscious design, three key factors will be incorporated into the units: 1) The units will be completely encased by cabinetry; 2) Rotating mechanical components will be isolated from the main body of the unit by vibration dampeners; and 3) Fiberglass insulation will line the entire cabinet.

#### 6.11.4 Rankine-Cycle Subsystem

The Rankine-cycle subsystem power loop was described in Section 6.11.1 and is again shown in Figure 6-35. This section discusses the working fluid selection study and the preliminary hardware investigation that has been conducted.

6.11.4.1 Working Fluid Considerations -- Two fluids were considered to be viable candidates for use in the Rankine cycle: Refrigerants 113 and 11 (R-113 and R-11). Both are relatively high-molecular-weight fluorinated and chlorinated carbon compounds. Many criteria were used to narrow the large fluid candidate field to these two. The most significant considerations were cycle efficiency, thermodynamic and transport properties, turbomachinery considerations, toxicity, and flammability.

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\*\* Per ARI Standard 590-76

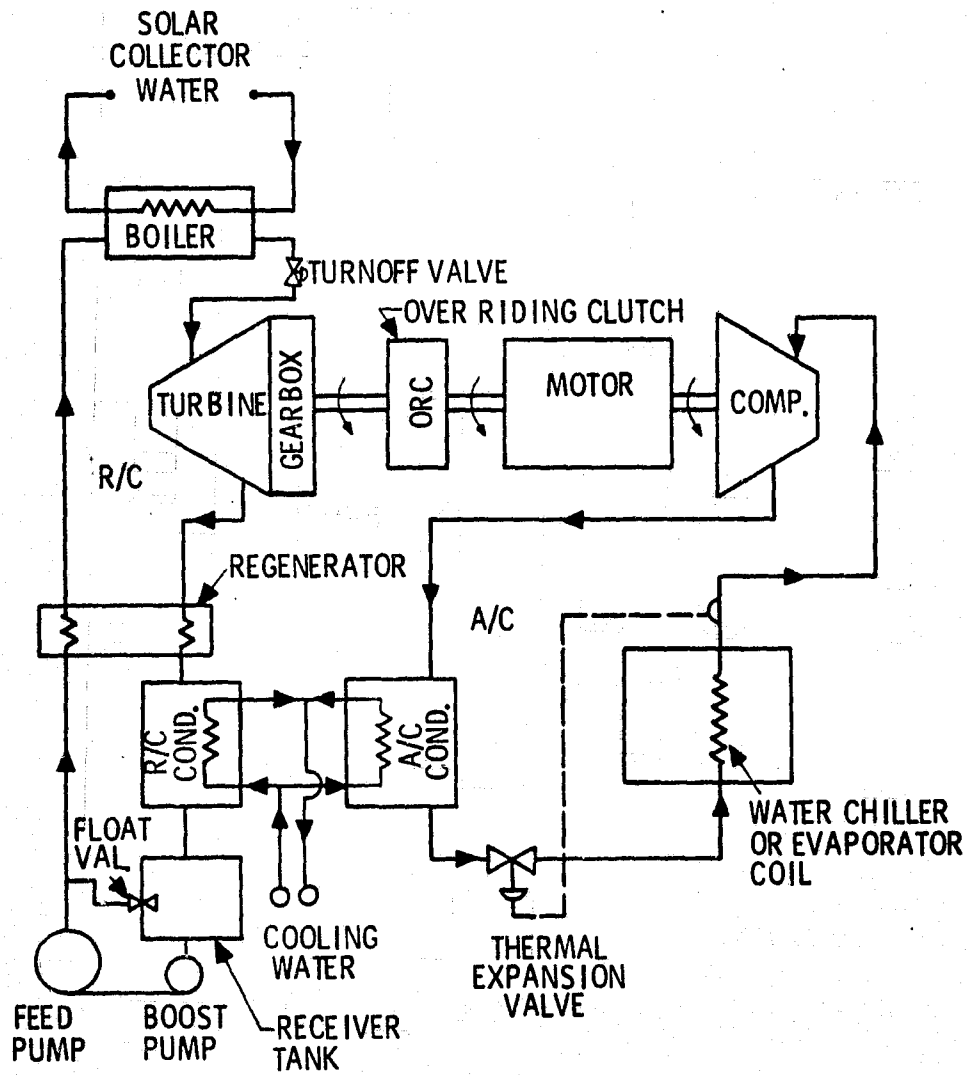


Figure 6-35. General System Schematic for Rankine-Cycle Air-Conditioning System



Comparing the two fluids in the temperature range shown in Table 6-4, a 5 percent advantage in cycle efficiency is obtained with R-113. To attain this performance, however, the R-113 requires a regenerator to recover some of the superheat remaining in the turbine exhaust vapor. The R-11 cycle, due to the fluid's thermodynamic properties, contains little or no superheat in the turbine exhaust and so does not require a regenerator.

For the same operating temperatures, R-11 operates at a slightly higher pressure than R-113, as shown in Table 6-4.

Table 6-4. Fluid Operating Temperature Comparison

Heat Exchanger	Fluid	
	R-113	R-11
Boiler pressure at 190°F	47 psia	90 psia
Condenser pressure at 95°F	10 psia	22 psia

This fact favors R-11 because low-pressure condenser systems generally are less attractive from a leakage standpoint.

The same cycle operating conditions also produce different turbomachinery for R-113 and R-11 systems. For the 190°F boiling and 95°F condensing cycle, an R-113 turbine demonstrating 72 percent efficiency and producing 3 horsepower rotates at 39,000 rpm and is 2.4 inches in pitch diameter, while an R-11 turbine with the same power and efficiency is a 60,000-rpm, 1.6-inch unit. Since gearbox losses increase rapidly with speed, a slower R-113 turbine with the same efficiency is much more attractive. For the 3-ton system, the best R/C fluid choice is unquestionably R-113. Its higher cycle efficiency and turbomachinery advantages clearly offset the disadvantages of the regenerator and the below atmosphere condenser.

In the 25-horsepower range, a 77 percent efficient R-113 turbine has a 22,000-rpm, 5.1-inch rotor, while an equally efficient R-11 turbine turns at 33,400 rpm and is 3.6 inches in diameter. However, a R-11 turbine rotating at the same speed as the 77 percent R-113 machine (22,000 rpm) would sacrifice only two points in efficiency with the speed reduction, and the R-11 system would not require a regenerator. For these reasons, the best fluid system choice for the multiple-family dwelling (25-ton air conditioner) is unclear. It is concluded that both R-113 and R-11 would offer reasonable approaches. R-113 was selected from the simulation study for the 25-ton development system, however, due to its slightly higher overall performance.

**6.11.4.2 Three-Ton Hardware** -- The following sections describe in some detail the status of the selection process for various pieces of hardware for the 3-ton unit.

**6.11.4.2.1 Turbine** -- A usual approach to achieve optimum system efficiencies in a single-fluid hermetic system would be to use an expander of the same type as the compressor (i.e., a piston expander with a piston compressor and a turbine expander with a centrifugal compressor). For the present 3-ton system, however, a turbine prime mover was selected because of the lower cost and higher prototype reliability when compared with a positive-displacement expander. Barber-Nichols has had extensive experience in the design and fabrication of turbomachinery for use in systems similar to the present application. The measured performance of a 2-inch turbine was reported by Robert Barber.\* A design point efficiency of 72 percent was achieved by this small radial inflow turbine.

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\* Barber, Robert E., "Solar Air-Conditioning Systems Using Rankine-Power Cycles - Design and Test Results of Prototype 3-Ton Unit," Barber-Nichols Engr. Co., Arvada, Colorado.

A preliminary turbine performance analysis was completed using the non-dimensional parameters of specific speed ( $N_s$ ) and specific diameter ( $D_s$ ). These parameters relate system cycle conditions (flowrate and available energy) and design variables (rotational speed and turbine diameter) to achievable performance for various turbine concepts. To perform this preliminary analysis, a turbine speed of 35,000 rpm (based on a tradeoff study relating turbine efficiency, size, and gearbox loss to turbine rotational speed) was selected for the 3-ton unit. The above similarity analysis and tradeoff study resulted in a 2.5-inch turbine.

The turbine type offering the best potential performance for selected design conditions is a radial inflow turbine. This type of turbine with the selected design conditions was found to match well with previous turbines designed and developed by Barber-Nichols. The turbine rotor is similar to that found in automobile turbochargers. In this application, existing turbine rotor and housing hardware can be modified to result in an extremely cost-effective turbine prime mover capable of efficiencies greater than 70 percent.

A preliminary estimate of off-design turbine performance was then made for a range of Rankine-cycle off-design conditions. The primary measure of turbine efficiency is the ratio of turbine wheel tip speed to the working fluid isentropic spouting velocity ( $U/C_o$ ), where:

$$U = N\pi D/720$$

$$N = \text{turbine rotational speed, rpm}$$

$$D = \text{wheel tip diameter, in.}$$

$$C_o = \sqrt{2gJ\Delta H'}$$

and

$$\Delta H' = \text{overall isentropic head, Btu/lb}$$

As may be noted, the ratio  $U/C_o$  accounts for changes in both turbine rotational speed and cycle off-design parameters affecting the overall isentropic head. Figure 6-36 presents the variation in turbine efficiency with off-design  $U/C_o$ . The figure was constructed using data obtained previously with a radial inflow turbine similar to that proposed for the present application. This procedure is intended to account for first-order influences on turbine efficiency. Secondary changes due to off-design pressure ratio, Reynolds number, and Mach number will be evaluated during the final turbine performance design analysis.

Final turbine design parameters, including nozzle and rotor definition, will be completed during the final design. Barber-Nichols will rely heavily on tested performance of these components in selecting an optimum turbine configuration. Design and off-design turbine performance will also be predicted during this phase using a digital computer program written for that purpose.

6.11.4.2.2 Gearbox -- The following specifications have been outlined as preliminary requirements for the 3-ton-unit gearbox. The input speed to the gearbox will be approximately 35,000 rpm, the output shaft speed 1750 rpm or lower depending on compressor selection, and the maximum gearbox power loss of 0.2 horsepower. The gearbox will have a hermetic seal either at the high-speed shaft or the low-speed shaft to contain the working fluid. The life expectancy should be in excess of 3000 hours. The gearbox should be designed to allow minimum production costs. Its power throughput will be approximately 3 horsepower.

Several approaches have been considered for meeting the above specification requirements. These approaches consider the development time available and the probable risk in reaching the above design goals. Previous designs have shed some light as to possible paths that can be taken.

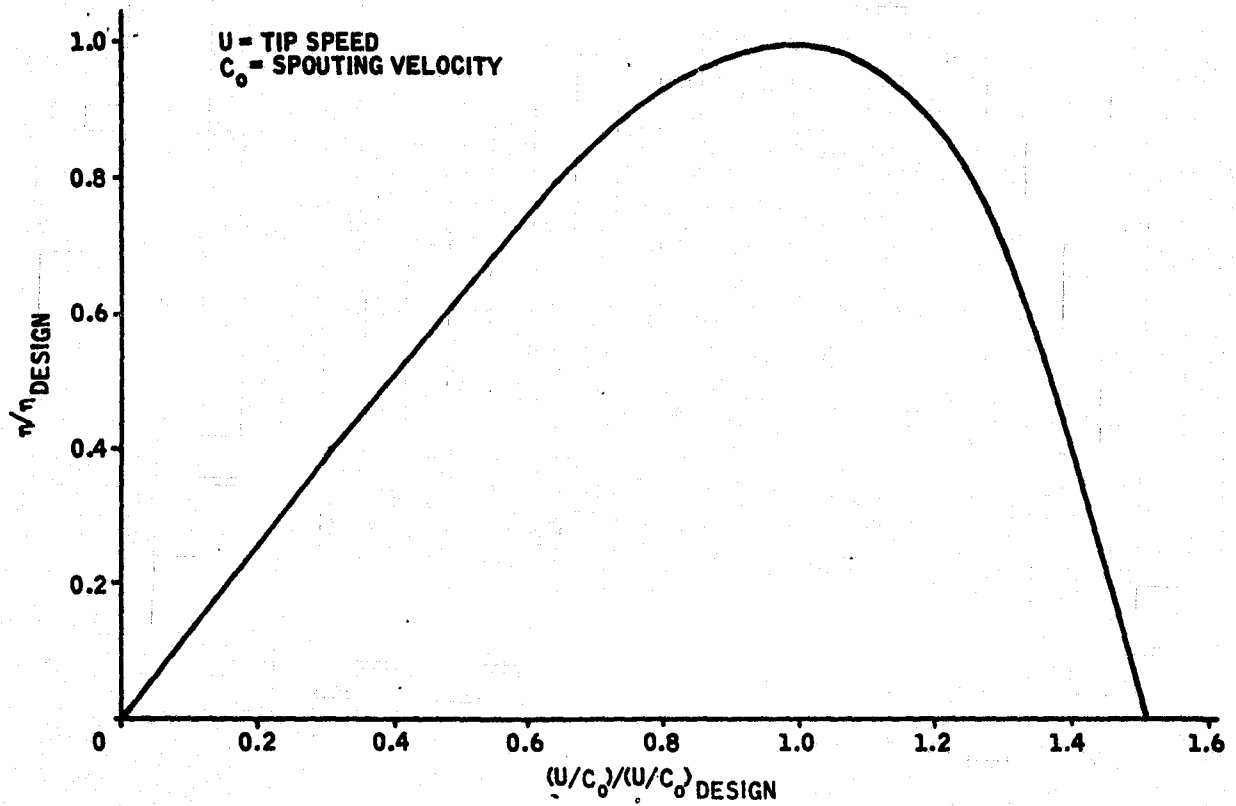


Figure 6-36. Variation in Turbine Efficiency with Off-Design  $U/C_0$

Two previous 3-ton air-conditioning units have been in operation for various periods of time. Each of these units used a commercially available gearbox modified for high-speed service. The first unit developed had a double-face seal on the output shaft, thus providing the hermetic seal at that point. It was also canned inside a liquid-filled chamber that acted as a heat sink for cooling. The high-speed turbine shaft had a single-face seal to prevent gross working fluid migration through the turbine end of the gearbox. The gearbox was vented to the low-pressure condenser to prevent liquid buildup in the gears. In the design of the second unit the hermetic seal was moved to the high-speed shaft assembly to allow the gearbox to be vented to ambient air and, thus, it only required a low-pressure lip seal on the output shaft. Both of these units used a splash lube system developed to minimize power loss. The second unit did not have a cooling chamber around it but used a cold-water plate bolted to the bottom of the gearbox.

The primary advantage in using the commercially available gearboxes was in the availability of existing precision gearing that could operate at high speeds with minimum modification. These units were rugged and oversized, resulting in minimum development time. The advantage of the first unit over the second lies in the area of cooling. There was more than sufficient cooling available to handle the power losses of that gearbox. The reason this unit was not used in the second system was because of the power loss due to the high-density working fluid vapor in the gearbox chamber. Three face seals in the assembly also contributed to this higher loss. Initially there was a chance that oil migration from the gearbox would never return to the gearbox but would remain in the rest of the system. It was found during the operation of the second unit that it was much noisier than the first unit. This occurred because of considerable noise damping available in the liquid chamber surrounding the gearbox in the first unit. Also, the cold-plate-mounted gearbox was just marginally cooled, and any attempts at adding sound damping around the unit resulted in overtemperature.

Another approach is under consideration in an effort to solve the above problems. It was noted that there are many high-reduction gearboxes on the market in the hand and power tool industry that are presently in production and from which production costs could be extracted on a high-volume basis. One such unit that appears to be very well adapted to this application is a unit made by Milwaukee Tool Company that has a high-speed motor driving a diamond drill unit for cutting holes in concrete. This unit is designed as a commercial product and its rated capacity in that particular product is 2 horsepower with 1000-rpm output. Its input speed is 20,000 rpm, and it is a triple-reduction unit. This unit has smaller gears than the other units and because of this has a lower power loss. Projected power losses for this unit are: 0.2 horsepower, including seals, thus meeting specifications. Gear tangential velocities are very low in this unit, thus presenting no windage loss problems if a low-speed hermetic seal is used. It has the build-in capability of changing from one speed to another which may be modified to allow an internal clutching mechanism that would disengage the turbine during motor operation. The unit is presently in production in quantities of 500 or more per run, and existing component costs are comparable with previously designed units. The advantage of using this unit in this particular system lies primarily in the low power loss. Additionally, production costs can be very closely estimated. The primary disadvantage of using this unit in the proposed system lies in the fact that the unit has not been previously used in a similar application.

There are several possible techniques for lubricating and cooling the gearbox. One is a conventional approach of using a shaft-driven gear pump to pump fluid from a sump through a cooler-filter and on through various lube jets directed at the bearings and gears. The other system developed uses the gearbox with a splash system in which the oil is collected in the sump of the gearbox and splashed over various components in the gearbox for lubrication and cooling. This will require a method of cooling the oil. This latter system is the one used in the previous 3-ton air-conditioner units.

Another approach that will be investigated is a mist lubrication/cooling system. This would greatly reduce the losses in the bearings. The primary problems in such systems are generating the mist and providing some kind of throughflow to provide the cooling. The system under study for this program uses a gas-driven mist generator similar to those used in paint spray systems. The driving gas (hot Freon) is bled off at the boiler. This gas is metered through a control valve and is then expanded through a standard paint-spraying nozzle. This spraying nozzle provides a siphoning effect, sucking up oil and generating a mist which is sprayed into the gearbox. The gearbox sump, which is external to the gears themselves, is vented to the condenser, allowing the gas introduced with the spray to migrate to the condenser. The gearbox is cooled by direct contact with liquid Freon from the boost pump being injected directly into the oil.

The system does require two primary control valves. One valve controls the proper gas flow through the mist-generating nozzle by maintaining a certain pressure drop across the mist-generating nozzle. The cooling is controlled by the second valve which is a thermal valve that cycles on and off to maintain a certain temperature in the oil reservoir. A primary advantage in such a system lies in the fact that all components involved in mist-generating and cooling of the gearbox are static systems that are not subject to reliability problems and premature failure.

One development item in this particular system is the control of the oil migration back to the gearbox. Air-conditioning systems currently operate under similar conditions, and it is proposed that a conventional approach be followed.

**6.11.4.2.3 Boiler Feed Pump** -- There are several possible approaches for the boiler feed pump. The primary consideration is whether to use a direct-driven or auxiliary-motor-driven unit. A second choice would be positive-displacement or centrifugal. The means of driving affects the start-up characteristics of the cycle (requirement for start-up pump) and the



possible efficiency (speed) of the pump. The second choice is related to control methods and matching characteristics of the components for off-design operation. Barber-Nichols has experience with all types of boiler feed pumps.

A boiler feed pump candidate that was recently acquired is a diaphragm pump. These have been used with Freon but not R-113 specifically. They are very-low-speed units that must be driven by a belt reduction system or a gear reducer. In discussions with the distributor's engineers, the shortest life components are the Viton diaphragms which are presently rated at 1500 hours at the pump's maximum rated speed and pressure. Since lower speeds and lower pressures can greatly increase life, it was recommended that the pump be tested for our application operating at a lower speed. Since the pump will be operating at a lower speed and at lower pressure, the life values should improve dramatically. This pump will be tested to verify performance and NPSH requirements. Then, possibly, some simple modifications to the pump will be apparent that will improve characteristics and possibly also add additional life.

A boost pump is applied in the system to provide a sufficient net positive suction head (NPSH) to meet the requirements of the main boiler feed pump. The pump is located at the discharge of the condenser in the system hot well. A bypass line is provided around the pumps to assure a quantity of liquid to the pump and provide stability in system operation.

Two off-the-shelf candidate boost pumps are currently under test at the Barber-Nichols facility. Manufacturers of these pumps are Sundstrand and Grundfos. Both units are hermetic, and both use a wet motor. The bearings are lubricated by the working fluid. These inexpensive pumps were designed for use in hot-water systems, and no mechanical modifications are anticipated to use either pump in R-113.

The pump impellers are designed with backward-curved vanes. A typical characteristic curve, with the delivered head increasing at low pump flow-rates, was found for both pumps during testing. This characteristic will provide quite stable pump operation over the range of flowrate.

6.11.4.2.4 Motor -- The 3-ton air-conditioning system will use a nominal 3-horsepower, single-phase motor. The typical efficiency for a production motor in this category is around 73 percent. These motors will retain this efficiency until loads below 50 percent are reached, at which time the efficiency will decrease steadily until the "no load" condition is attained. Typical "no load" power consumption will be around 0.3 kW. These motor efficiencies do contribute to the overall system efficiency and any improvement also improves the overall solar energy conversion efficiency. Single-phase motors of this power rating with efficiencies as high as 90 percent can be designed and fabricated. A high-efficiency motor such as this would be expensive, and at this time, Barber-Nichols is not aware that any of the manufacturers of production-quantity motors are considering the design or marketing of a high-efficiency single-phase motor.

6.11.4.2.5 Controls -- The basic Rankine-cycle power system requires only one active component for complete control. These power systems tend to be self-balancing and can be adjusted to run without any active control components. To handle a wide range of off-design operating conditions efficiently, a float-activated bypass valve is used to match pump delivery to system flow requirements. The pump characteristics at off-design conditions are the only factors that require adjustment to keep the system balanced or operating at its best performance. As shown on the system schematic, Figure 6-35, a reservoir collects the condensed fluid and supplies it to the pumps for the system. A float bypass valve is placed in this tank and tries to hold the reservoir level fixed by increasing or decreasing the bypass flow from the pump back to the reservoir. This action tends to balance the total system fluid inventory and thus aids the self-regulation of the system. This bypass

valve performs a second function during high-power-level operation and keeps the pump from depleting the reservoir and cavitating the pump. The bypass also allows the pump to be somewhat oversized and able to provide the increased flows when required. This reservoir-level-control approach, coupled with the proper fluid charge, enables a system to operate from a basically zero power level up to a maximum limited by the capacity of the pump to meet required head/flow requirements.

In addition to the basic Rankine-cycle loop control, a system requires a control logic center to integrate the Rankine-cycle system and the air conditioner with the rest of the solar-energy collection system and the building temperature control. This logic center will cycle the air conditioner on and off with or without the Rankine-cycle power system to meet the commands received, while monitoring various performance parameters to protect the combined cooling system from damage in the event of equipment malfunctions. The logic control for these Rankine-cycle-powered cooling systems will be composed of simple relay logic using switch-type sensors for the required inputs and for sending electric power to the various control components required.

Each cooling system will be controlled primarily by an on or off command from a central solar controller. A voltage will be supplied from this controller whenever the building requires cooling. Upon receipt of this command, the logic will activate the air conditioner by turning on either the electric motor or the Rankine-cycle power system or both. The decision to use the Rankine-cycle power will be a function of the availability of solar heating water above a minimum-temperature level. The performance studies being made at the present time will establish the requirement or basis for establishing the various combinations of power source use desired. If the studies indicate that supplementary electrical power is needed over a significant portion of the cooling day, then there will probable be no benefit in providing a control mode for using the Rankine-cycle power system with the electric-motor shutoff. If the study shows that there is a significant performance

advantage to running with the Rankine-cycle power system only, then logic will be required to select this mode of operation. One of the simplest approaches to provide logic for the decision choosing Rankine-cycle power only or both power sources is to use a two-level thermostat for the building. With a two-level thermostat, two separate switch closures are provided with a finite temperature difference between them. The first switch closure will be used to command the Rankine-cycle power system to power the air conditioner. If the Rankine-cycle power is sufficient to support the cooling load, then the thermostat will work around this setpoint. If the cooling load is too large for the Rankine-cycle power system, building temperature will climb until it activates the second switch which, in turn, will energize the electric motor and it will pick up its share of the air-conditioning load. Building temperature then will cycle about the second setpoint. The decision to use the Rankine-cycle power upon a command for cooling is dependent only on the availability of solar water hot enough to provide useful power for the system. The energizing of the Rankine-cycle system for cooling would thus be overridden by a temperature switch that would allow the Rankine-cycle system to come on if the water temperature is high enough and shut it off if too low. This logic panel, at a minimum, would control the motor starter and the solenoid valve between the Rankine-cycle boiler and the turbine. If, in the development of the final system configuration, a clutch is used to isolate the electrical motor when the Rankine-cycle system is on, the control center will control this clutch in parallel with the Rankine-cycle solenoid valve.

Both the Rankine-cycle power system and the air-conditioning system have physical parameters that require monitoring for overall system protection. In the air-conditioning system, the typical parameters monitored to ensure proper performance are compressor discharge pressure, suction pressure, and compressor oil pressure. The Rankine-cycle system parameters monitored will be lube oil pressure and lube oil temperature. Sensors will be provided to monitor these parameters and will be used as interlocks in the logic chain to prevent system start-up or provide system shutdown in the event their setpoints are exceeded.

**6.11.4.2.6 Parasitic Losses** -- For the Rankine-cycle system to operate, support is required from various electrically-powered accessories. The 3-ton demonstration unit built for the Honeywell van uses an electric clutch, solenoid valve, boost pump, and a logic control box with relays and motor starter. On the demonstration model, these accessories use a total of 170 watts of power. The boost pump requires 70 watts, the clutch 40 watts, and the relay logic control box 50 watts. Fifty watts is not a true reflection of the control logic requirements because at least 35 of these watts are consumed in a DC power supply required to power the clutch. The control approach, and thus the components required, has not as yet been established for this system. The electrical clutch used in the demonstration system was included so the Rankine-cycle power system could drive the induction motor in a generating mode. This generating mode is not being considered for these 3-ton systems. The elimination of the clutch can save 75 watts of this parasitic power. There is a possibility, however, that an electrically-operated clutch will provide overall performance improvement if used to decouple the turbo gearbox when there is insufficient power from the Rankine cycle power system to drive the air conditioner compressor. This clutch would not have to be electrically-powered but could be mechanical with solenoid operation. The control approach used in the overall system configuration will be based on minimizing these parasitic power requirements while achieving overall system performance objectives.

There are other power system parasitic losses besides the electric ones that have been mentioned. The gearboxes required to reduce or match the turbine to the electric-motor speed will consume approximately 0.2 horsepower with the 3-ton unit. If, in the design of the cooling system compressors, the selection requires a nonstandard motor speed, a belt-drive reduction system will be required. Power losses in these belt drives are typically 2 percent or less of the power level transmitted. A physical packaging configuration that requires jack shafts or idlers to control the belts will add even more loss to the mechanical power transmission system. In the A/C operating

mode, where the Rankine-cycle power system is not producing power and all of the power is supplied by the electric motor, there is a need to decouple the Rankine-cycle drive from the motor and compressor. There is a power loss associated with driving the turbine backwards through the gearbox. One technique is to use an overrunning clutch that prevents power feedback into the gearbox. These overrunning clutches, however, also have parasitic losses when used in the override mode. All of these loss factors tend to reduce the overall efficiency of this Rankine-cycle-powered air-conditioning system. Each one of these loss factors will be evaluated carefully and design approaches selected that minimize their effect on the performance of the cooling subsystem.

**6.11.4.2.7 Rankine-Cycle Heat Exchangers** -- Barber-Nichols' time-proven approach for selecting low-temperature Rankine-cycle heat exchangers is to use commercially available units wherever possible, thereby reducing cost, lead time, and development. The preheater, boiler, and condenser will be purchased from nationally recognized vendors. We are not aware of any commercially available heat exchangers suitable for the Rankine-cycle regenerator. This unit therefore will be designed and assembled by Barber-Nichols.

Prices for these types of heat exchangers were used to estimate heat-exchanger costs during the R/C optimization discussed in Subsection 6.11.5. Additional features of the heat exchangers are discussed below:

- **Preheater** -- The preheater is a commercially available industrial-type heat exchanger commonly used as an oil cooler, with water as the heat sink. For this application, solar collector water is placed in the tubes and the R/C working fluid flows with low-pressure drop through the shell. A one-pass, counter-flow configuration is used to maximize heat transfer with minimum cost.

The tube-side heat transfer surface may be mechanically cleaned without disturbing the shell-side piping. The fixed tube sheets are welded or brazed to the shell (depending on manufacturer) which eliminates any joints on the R/C side of the unit. These units are designed by their manufacturer for temperatures and pressures in excess of our requirements and their large production quantities result in low cost. Barber-Nichols' design approach has been verified by testing to improve confidence in our predicted performance.

- Boiler -- The boiler transfers heat from water to a boiling organic fluid. This heat exchanger is functionally very similar to water chillers commonly found in air-conditioning systems. In fact, Barber-Nichols' recommended approach is to use a commercially available water chiller for the R/C boiler. This recommendation is based on our experience with low-temperature Rankine cycles using this type boiler. Forced-convection designs are preferred because the relatively small water-to-organic fluid temperature differences anticipated for this application are too small to produce the natural convection required in pool-type boilers. We have had excellent experience with the forced-convection evaporators.

These units are designed for use with R-12 or R-22 and, therefore, have a pressure capability well in excess of that required with R-113. This will enhance system reliability and safety. The large production quantities of these readily available heat exchangers result in low cost and short lead times. A variety of types and configurations are available to suit our design requirements. These units are designed to have low-pressure drops and are compatible with the various Freons and with water and water-glycol solutions.

- Condenser -- The R/C condensers have requirements very similar to condensers for air conditioning and refrigeration units. They both desuperheat and condense a fluorocarbon fluid using water as the heat sink. Therefore, commercially available air conditioning and refrigeration condensers are proposed for the Rankine-cycle condensers. These units are designed for 300 psig or more and their high pressure capability will greatly enhance system reliability and safety. Their welded tube sheets eliminate gasketed joints on the Freon side, reducing leaks and simplifying maintenance. Several designs feature all copper water channels and an epoxy-coated tube sheet and water plate to prevent pitting caused by galvanic action. Units feature mechanically cleanable tubes and low pressure drops. Units are designed for use with cooling towers.
- Regenerator -- The regenerator is a vapor-to-liquid heat exchanger with a high effectiveness. Because of the required low-pressure drops and large required heat-transfer area, we are unaware of suitable commercially available units. Therefore, Barber-Nichols' approach is to design and build a heat exchanger specifically for this application.

This heat exchanger uses a heavy-duty industrial oil cooler core that is specially manufactured to Barber-Nichols' specifications. The design incorporates copper tubes for high heat transfer and economical assembly, and aluminum fins because of their light weight, low cost, and excellent heat-transfer characteristics. The external fins incorporate serrations for strength and durability, and incorporate a sunburst collar to impart turbulence and increase heat transfer.



Barber-Nichols has built several of these units and has test data to back up our extensive analysis to ensure the best heat-exchanger design possible. Test data and analysis show that this concept results in high heat transfer and extremely low pressure drops.

The heat-exchanger shell is designed in accordance with Section VII, Division I, of the ASME Pressure Vessel Code. The units will withstand full vacuum on both the tube and shell sides. This capability enhances reliability by simplifying system controls and valve requirements. Each unit is fully pressure-tested before it is connected to the system. The regenerators incorporate fittings designed for the particular application to minimize joints and fittings.

Similar units have been built and used in numerous systems and their superior performance and reliability have been verified.

**6.11.4.3 Twenty-five Ton Hardware** -- The following describes in some detail the status of the selection process for various pieces of hardware for the 25-ton unit. In most cases this is basically the same as for the 3-ton and will not be repeated here. The areas that are different will be discussed.

**6.11.4.3.1 Turbine** -- The turbine for the 25-ton unit will be similar to that for the 3-ton unit, differing in speed and diameter. There will be a slight improvement in efficiency to about 77 percent.

**6.11.4.3.2 Gearbox** -- The gearbox under consideration for the 25-ton Rankine-cycle air-conditioning system is an existing-design gearbox used in other Rankine-cycle turbine-driven units under previous development by Barber-Nichols. The unit is basically a ground-gear, double-reduction

gearbox using a wet sump with a positive-pressure oil system. This unit is oversized for the 25-ton unit and, as such, has proportionally large losses (2 horsepower). These losses can be reduced through further development. One area that could prove fruitful is to use a mist lube system as mentioned in the previous 3-ton air-conditioning section. Another area would be to reduce seal loss by using a low-speed seal rather than a high-speed seal.

Depending on the results of development tests with the 3-ton gearbox, a gearbox unit is available from Milwaukee Tool Company that may be adaptable to this power range. This unit could be modified as previously stated for the 3-ton unit, thus providing a low-loss unit.

6.11.4.3.3 Pumps -- The main boiler feed pump for the 25-ton unit will probably be driven by the gearbox. An efficiency of 60 percent is expected for this pump. The boost pump and start-up pump, if required, will be similar to those for the 3-ton unit.

6.11.4.3.4 Motor -- The 25-ton air-conditioning system requires a 20-horsepower, three-phase motor. Typical efficiencies for this category of motor are around 86 percent at full load, with this efficiency being maintained to power levels of 50 percent. The Century Electric Division of Gould has produced and is presently marketing a new, high-efficiency, three-phase motor. A 20-horsepower motor from the Gould "E plus" line has a guaranteed efficiency of 90.5 percent at full load. These motors retain their efficiency down to the 50 percent load also. A secondary benefit achieved with this motor line is an improved power factor. Power factors of around 0.9 are achieved as compared with 0.8 for a standard production unit today. This improvement in performance results in a cost factor of 1.25 for these motors over the standard production model.

6.11.4.3.5 Controls -- Same as for 3-ton system.

6.11.4.3.6 Parasitic Losses -- With a 25-ton air-conditioning system, the same accessories as for the 3-ton system are required. The boost pump naturally is bigger and will require up to 500 watts of power. Controls power should be identical to that required in the 3-ton system. A clutch, if required for this larger system, will have to be of a mechanical configuration and solenoid-actuated at these higher power levels. With the proper clutch configuration, the solenoid-actuator will require power only when engaging and disengaging the clutch. The gearbox loss for the 25-ton unit will be under 2 horsepower.

The other losses described in the section on parasitic losses in the 3-ton unit also pertain to the 25-ton unit and will not be repeated here.

6.11.4.3.7 Heat Exchangers -- Same as for the 3-ton system.

#### 6.11.5 Rankine-Cycle Optimization Study

The optimization study by Honeywell with inputs from Lennox and Barber-Nichols involved carrying out cycle computations to determine power output and efficiency at various operating conditions. System characteristics provided the basic data necessary for the evaluation of an optimum design. A number of parameters were considered in the selection and development of the baseline schematic including:

- Cycle temperatures that are compatible with present day flat-plate technology. This requirement limited the inlet temperature to the engine boiler to a maximum of about 220°F. Various design temperatures below this upper limit were investigated.
- Engine design and size with the minimum auxiliary power requirement over the range of operating conditions. The engines were coupled to small-to-medium-capacity air conditioners. Design capacities of 3 tons and 25 tons were selected for the single-family and multiple-family residences, respectively.
- Rankine-engine working fluid based on the temperature limitations of the cycle and its thermodynamic states. The refrigerant contribution to the requirement of minimal maintenance on the unit over a long operating life was also considered. Both R-113 and R-11 were studied for the medium-size engines.
- System control to optimize the use of collected solar energy and minimize auxiliary power requirement. Control reliability and simplicity in interfacing with the overall system were also factors in the evaluation criteria. In the present study, two control schemes were investigated: constant-speed and

variable-speed. Basically, in constant-speed operation, auxiliary power from the electric motor is supplied to maintain the air conditioner at its design capacity. In the variable-speed mode, the cooling capacity of the unit varies with the fluid inlet temperature and is electrically enhanced only when it fails to meet the space cooling requirements.

The effects of these parameters and various design values were studied through component and system models incorporated in a year-long simulation. Additional cycle computations were then made based on the operating condition selected from the simulation. These last cycle conditions then established some cost figures for the cycle. The subsections that follow describe these computations in some detail and present a summary of the major results of the simulation.

6.11.5.1 Off-Design Study -- The cycle off-design operation was analyzed to determine the performance during the time that insufficient insolation was available to operate at the design conditions. These data were the input to the simulation program which determined the auxiliary power required to operate the air conditioner at design capacity. This off-design study was completed for a range of design collector-exit temperatures from 160°F to 210°F. At each collector-exit design temperature the off-design operation performance was determined for a collector-exit temperature of 10°F above the design value and as much as 40°F below the design value.

The off-design study involved a complete cycle calculation at the design condition (collector-exit temperature), and then the cycle was recomputed using the following described assumptions as to the off-design operation of each component. The cycle calculation for the design study will be described first, then the assumption for the off-design computation will be discussed.

The basic cycle was assumed to operate between the temperature limits of the collector-exit design temperature and the temperature of the cooling tower water at 85°F. The design flowrates of the hot source and cold sink were 4 gpm/ton and 3 gpm/15,000 Btu/hr, respectively. A 10°F approach temperature was assumed in both the boiler and the condenser. The pressure loss on the working side was assumed to be 5 percent of the absolute pressure on both the high-pressure side and the low-pressure side of the cycle. The working fluid was R-113 for the 3-ton unit. This choice of working fluid implied the use of a regenerator which had an assumed effectiveness of 0.80. The efficiency of the turbine and pumps were assumed to be 0.72 and 0.50, respectively. This is based on past experience and the analysis described in the turbine and pump sections of this report. The gearbox loss (0.2 horsepower) was included in the analysis, so that the design power is the power out of the gearbox. The design power was 1, 2, 3 and 4 hp for the 3-ton unit. The results of this design-cycle computation provided the baseline conditions for the off-design performance calculation.

The typical off-design operating conditions result from the lack of sufficient solar input which the cycle experiences at a lower than design collector-exit temperature. A similar off-design condition is possible during periods of high solar insolation but with low cooling requirement. The collector-exit temperature would then be above the design condition. In either case, the change in the high temperature of the cycle, with the assumption of hardware fixed by the design condition, will change the performance of the cycle.

Basing the computation on the temperature of the working fluid as it leaves the boiler (rather than the collector-exit temperature), the working fluid temperature was allowed to increase or decrease by some amount, such as 10°F, to represent the off-design condition. This then defined the boiler pressure (saturated) and the pressure input to the turbine. The flow through the turbine was determined by the choked condition in the nozzles. Therefore, the change of the flow due to the change in the cycle conditions could be

determined. The boiler-and-condenser-approach temperature difference is assumed to vary directly with the flowrate. This, in turn, defined the collector temperature and condenser coolant temperature. The frictional pressure drop in the heat exchangers varied with the square of the flowrate. The turbine efficiency varied with the isentropic spouting velocity as described in the turbine section. The pump efficiency was assumed to remain constant but the pump work varied with the head rise. The actual flow in the pump was the design flow, since the speed was constant. Any excess was bled back around the pump.

The off-design computations were carried out for the same conditions as described for the design condition. These are presented in Table 6-5. The plot for the power output and the Rankine-cycle efficiency is shown in Figures 6-37 and 6-38 for a design collector-exit temperature of 190°F. This represents the typical plot, and the curves for all the other conditions are similar in shape. These curves were used as the input data for the Honeywell simulation program to determine the optimum design temperature and power level. Added to these graphs are curves for a power level of 2.3575 horsepower which was the power selected by the simulation study.

Table 6-5. Off-Design Study Power Levels  
(Values in Horsepower)

Cycle	Design-Point Collector Exit Temperature, °F					
	170	180	190	200	210	220
3-Ton/R-113	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4
25-Ton/R-113	15, 25, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25
25-Ton/R-11	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25

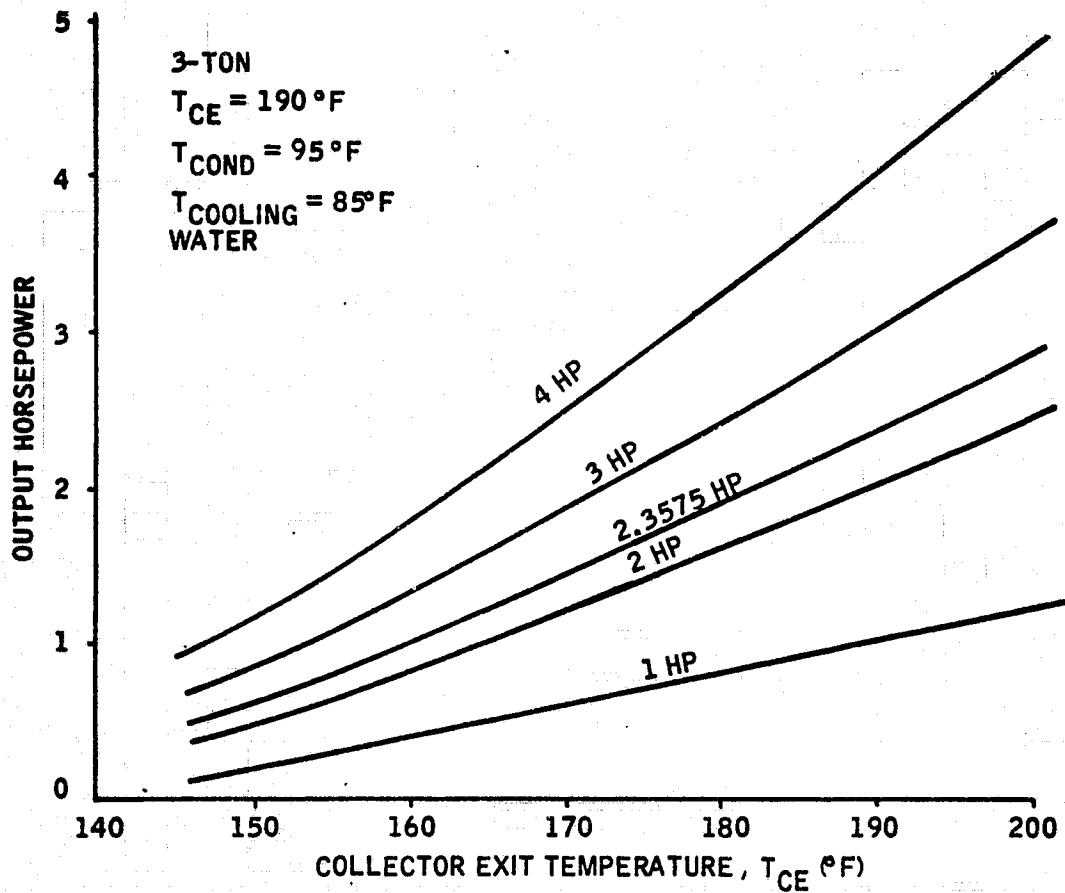


Figure 6-37. Output Horsepower versus  $T_{CE}$  at Constant-Horsepower Off-Design



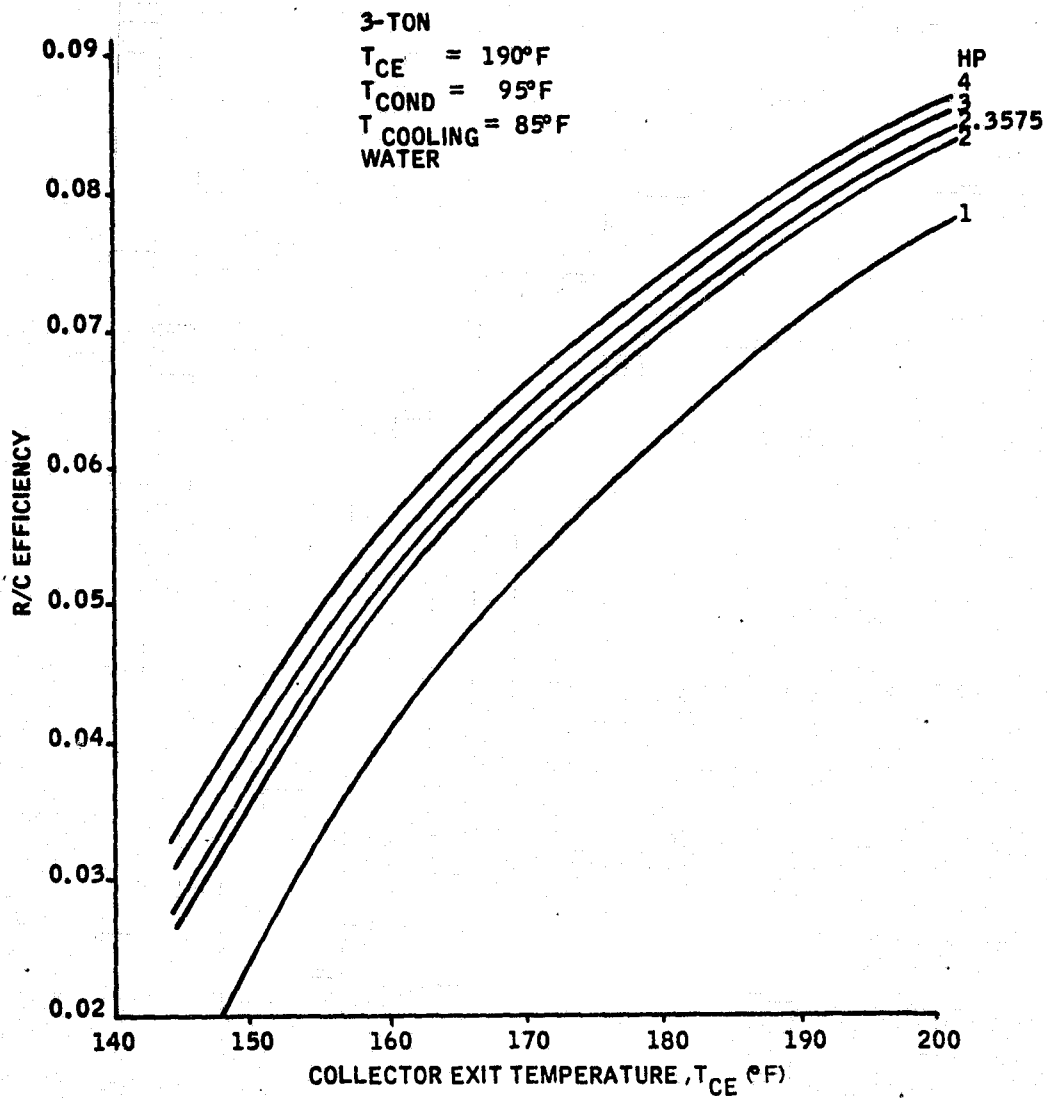


Figure 6-38. Rankine-Cycle Efficiency versus  $T_{CE}$  at Constant-Horsepower Off-Design

6. 11. 5. 2 Simulation Study and Results -- The results of the off-design study were input into the Honeywell simulation program to determine an optimum collector temperature and optimum power level. This simulation study is fully described in Section 4 of this report. Along with the power output and efficiency, the feed pump capacity and the control methodology are two other aspects of Rankine-cycle operation that can affect the results. These were also input into the program.

One significant limiting facet of the Rankine-cycle operation is the capacity of the boiler feed pump. Usually, this limit is 10 percent over design but it can be made any desired value. It is a question of trading off design performance with maximum capacity. The effect this has on the simulation is that the Rankine-cycle operation (power output and efficiency) levels fall off at some point above the design point rather than just continuing on up the curves presented. This was input at  $10^{\circ}\text{F}$  above the design point.

The other major factor input to the program was the method of controlling the cycle. One such method is to use the motor as a generator to limit the speed when an excess of solar energy is available. Along with this there are a number of alternatives, such as variable-speed operation when the off-design condition allows the Rankine-cycle air conditioner to move to off-design speed. Then, at some selected point, the motor/generator would cut in to bring the speed back to the design level.

Figure 6-39 shows the effects of various engine sizes and design temperatures on the yearly consumption of auxiliary power needed to meet the cooling requirements of a single-family residence in Atlanta. With a compressor COP of 7.0 the 2-horsepower engine provides the design power requirement of a 3-ton air conditioner. The results of the simulation for a fixed collector area of  $756\text{ ft}^2$  show that the heat addition requirements of the 3-horsepower and 4-horsepower Rankine engines were larger than the amount available from the solar system. The largest Rankine engine forced the collectors to operate at the lowest possible temperature allowed by the controller, and the

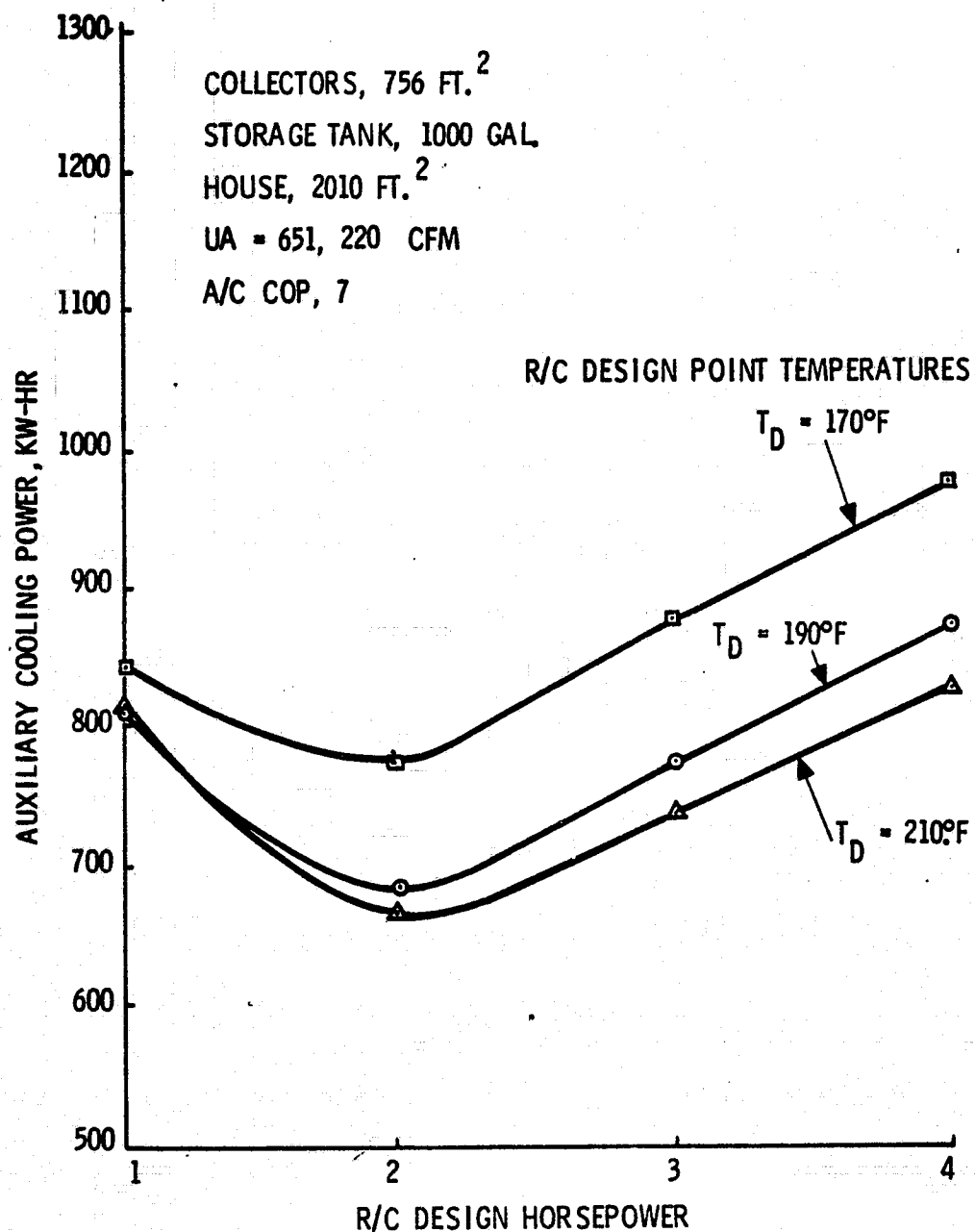


Figure 6-39. Auxiliary Cooling Power versus R/C Design Horsepower at Varying Design Temperatures - 3-Ton Unit

improvement in the solar system efficiency did not offset the decrease in the engine thermal efficiency. Thus, the overall operating efficiency of the large engine system decreased and its auxiliary power requirements increased. The smallest engine considered, the 1-horsepower engine, proved to be undersized for the cooling loads considered. From the results of these simulations, it is apparent that the 2-horsepower engine output was more evenly matched to the power requirements of the air conditioner a greater percentage of the time, with a minimum electrical power expended in providing the added system capacity. Another parameter which was investigated was the effect of the design temperature on the cooling performance of the Rankine engine. As expected, the engine performance improved with increasing temperatures, but this improvement was insufficient to overcome the degradation in collector efficiency. Figure 6-40 illustrates the effect of temperature on various system efficiencies. As the operational temperature drops below 200°F, the combined efficiency ( $\eta_{\text{coll}} \times \eta_{\text{RC}}$ ) for the 190°F and 210°F designs shows a minimal variation. Consequently, the difference in performance becomes so small that the 190°F design point was selected as the preferred choice. These combined results also indicate that for the location and weather data analyzed, the 210°F design is operating in the lower temperature range a great percentage of the time. The selection of 2 horsepower as a reasonable optimum was based on a Compressor COP of 7.0. But data from presently available components indicated a more realistic value of 6.0, which increased the baseline Rankine-engine size to 2.36 horsepower for the 3-ton cooling system.

The results from the variable-speed simulation showed a net 2.6 percent yearly savings in auxiliary energy consumption over a similar constant-speed design in Atlanta. The selection of a different site with different cooling requirements and solar input can have a large effect on the choice of the best control option and may provide a greater savings for the variable-speed control.

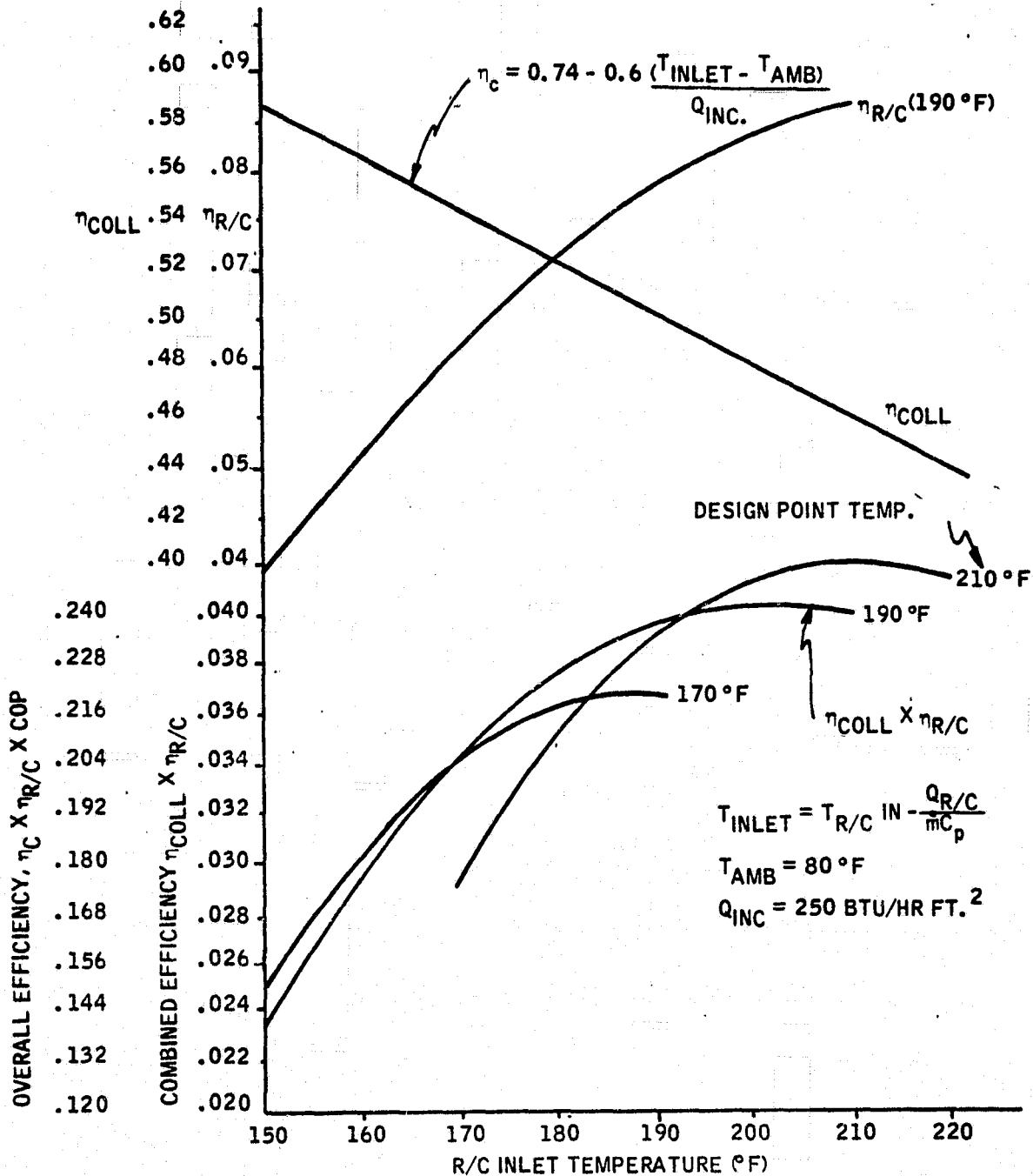


Figure 6-40. Effect of Inlet Temperature versus System Efficiency

The simulation study for the 25-ton unit was completed for both R-113 and R-11 as working fluids. Figure 6-41 illustrates the results for the various engine sizes considered. Since R-113 shows improved efficiency with an approximately 8 percent decrease in auxiliary energy consumption, this fluid was chosen to complete the optimization study.

The selection of the optimum design based on the results illustrated in Figure 6-42 was similar to the 3-ton unit. Rankine-engine design powers between 15 and 20 horsepower showed no major effect on the auxiliary energy consumption for the 190°F and 210°F designs. Further, the total auxiliary energy consumed was low and, in particular, the differences between the various power levels were very small. Thus, the selected design power level was the value required to drive the air conditioner at its design load. For a 25-ton air conditioner with a Compressor COP of 6.0, a 19.6-horsepower engine was chosen.

In summary, the simulation results were used to select the proper design temperature and power levels for each of the system capacities considered. With present plans to provide multiple 25-ton units to supply the cooling requirements of the commercial building, the following designs were selected:

- 3-Ton - Constant Speed:
  - $T_D = 190^\circ\text{F}$  control range (150°F to 200°F)
  - Horsepower at 190°F = 2.36
  - R-113 working fluid
- 25-Ton - Constant Speed:
  - $T_D = 190^\circ\text{F}$  control range (150°F to 200°F)
  - Horsepower at 190°F = 19.6
  - R-113 working fluid

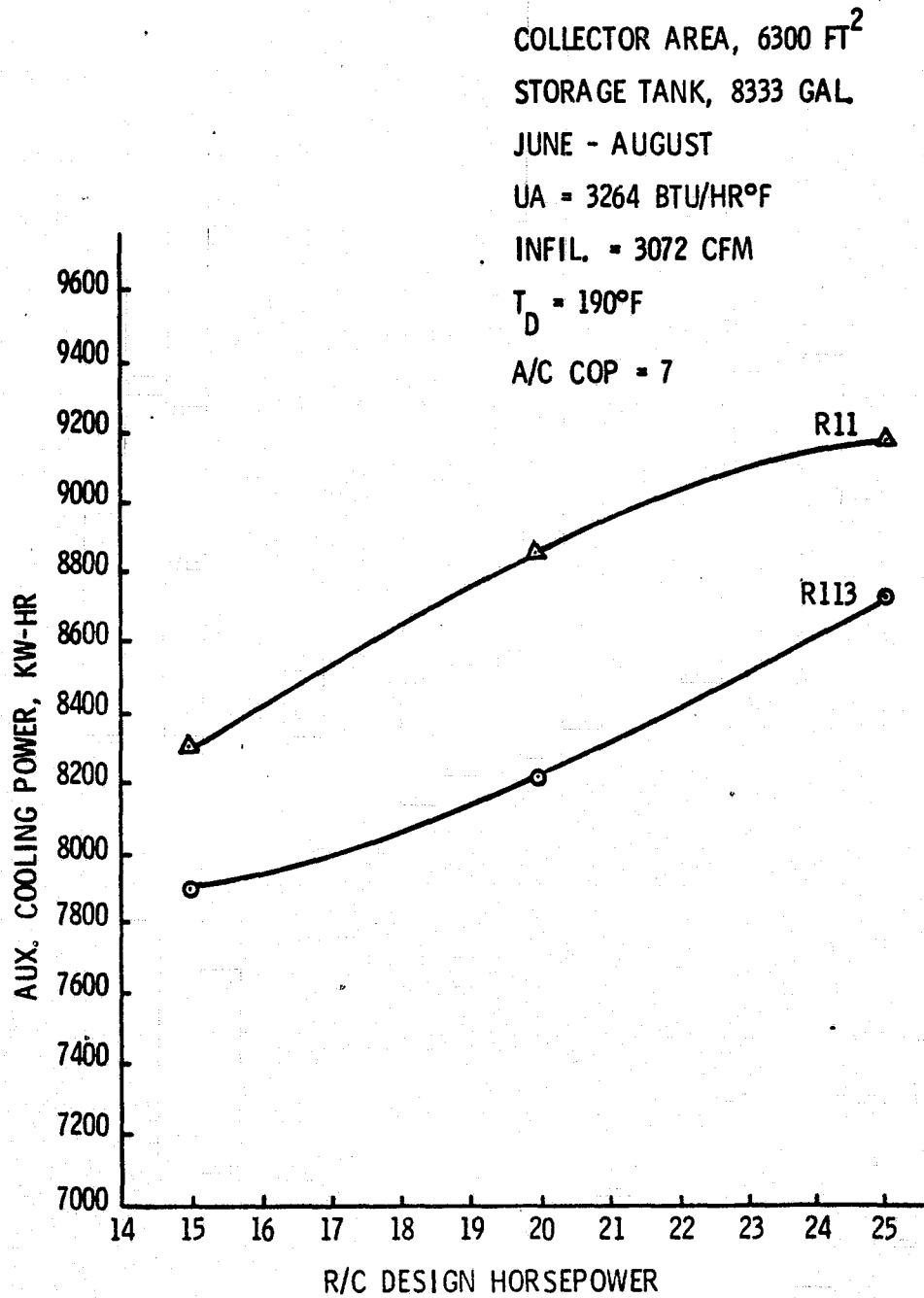


Figure 6-41. Refrigerant Effect on Rankine-Cycle Design Horsepower versus Auxiliary Cooling Power for Atlanta Multifamily Residence

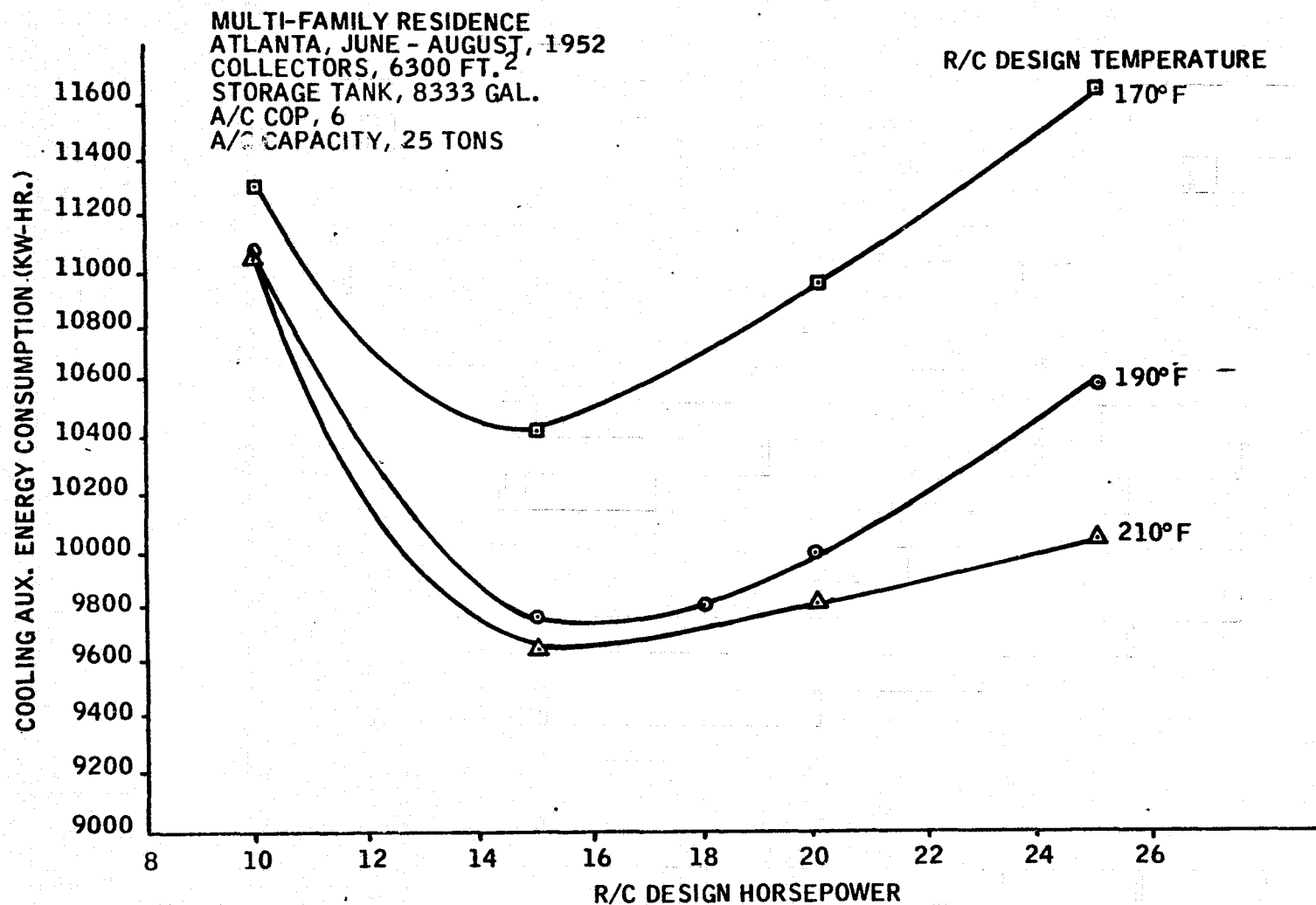


Figure 6-42. Auxiliary Cooling Power versus R/C Design Horsepower at Varying Design Temperatures - 25-Ton Unit



**6.11.5.3 Design Point** -- The temperature limits of the system ( $190^{\circ}\text{F}$  and  $85^{\circ}\text{F}$ ) have been selected along with the power level from the simulation study. At this point, it is then necessary to select the cycle conditions that will operate within these limits with components (particularly the heat exchangers) that are reasonably economical. This selection process is described in the subsections that follow. Both the 3-ton and 25-ton units are discussed together, as the computations are the same and the conclusions are similar. As part of this study, the site selection can have a large effect on the cycle and component selection (see Section 5.1.1.).

**6.11.5.4 Cycle Cost and Efficiency** -- There are many variables that affect the cost and efficiency of the Rankine cycle. A study was conducted to determine the simplest and most economical cycle to use as a baseline for comparison. Improvements were then made to this cycle which improved the efficiency while increasing the cost. The results of this analysis is given in Table 6-6 for the 3-ton unit and Table 6-7 for the 25-ton unit. These data are then used in the selection process to define the design cycle, as described in Section 6.11.5.7.

The Rankine-cycle costs for large quantities are 93.3 percent hardware, and of these hardware costs, 24 percent are heat exchanger costs (22 percent of overall costs). The cost of the heat exchangers is very sensitive to cycle conditions, particularly to the temperatures of boiling and condensing. The conditions used in the study are as follows. The collector-exit temperature was held constant at  $190^{\circ}\text{F}$ , COP was fixed at 6.0 (2.36 horsepower for 3-ton unit and 19.6 horsepower for 25-ton unit), and condenser water temperature and flowrate were fixed at  $85^{\circ}\text{F}$  and 3 gpm/15,000 Btu/hr. The regenerator effectiveness was varied from zero (no generator) to 1. Two condensing temperatures were selected for water-cooled condensers ( $95^{\circ}\text{F}$  and  $100^{\circ}\text{F}$ ) and for air-cooled condensers ( $120^{\circ}\text{F}$  and  $140^{\circ}\text{F}$ ). The turbine efficiency and gearbox losses were set at 72 percent and 0.2 horsepower for the 3-ton system and 77 percent and 2.0 horsepower for the 25-ton system, as previously described. The pump efficiency was 50 percent and 60 percent for the 3-ton and 25-ton systems, respectively.

Table 6-6. Design Cost/Efficiency Matrix for 3-Ton Rankine-Cycle Unit

Parameter						T <sub>ce</sub> = 190°F COP = 6.0 (2.3575 hp)
Boiler Temperature, T <sub>o</sub> (°F)	Boiler Water Flow Rate (gpm/ton)	Condensed Temperature (°F)	Regenerator Effectiveness (ε)	R/C Efficiency	Cost Ratio (Cost/baseline)	Remarks
174.97	4.0	100	0.0	0.0657	1.00	BASELINE
174.97	4.0	100	0.8	0.0702	1.09	
177.2	4.0	100	0.8	0.0720	1.17	
177.2	4.0	100	0.8	0.0672	1.05	
180	6.0	100	1.0	0.0755	∞	Variable regenerator effectiveness.  T <sub>cond</sub> = 100°F
180	6.0	100	0.8	0.0741	1.15	
180	6.0	100	0.6	0.0728	1.08	
180	6.0	100	0.4	0.0715	1.06	
180	6.0	100	0.2	0.0703	1.04	
180	6.0	100	0.0	0.0691	1.03	
176	4.0	100	0.8	0.0711	1.56	
176	6.0	100	0.8	0.0711	1.12	
180	6.0	95	1.0	0.0802	∞	Variable regenerator effectiveness.  T <sub>cond</sub> = 95°F
180	6.0	95	0.8	0.0787	1.22	
180	6.0	95	0.6	0.0772	1.14	
180	6.0	95	0.4	0.0758	1.11	
180	6.0	95	0.2	0.0744	1.09	
180	6.0	95	0.0	0.0731	1.07	
190	7.88	95	0.8	0.0861	∞	Variable T <sub>o</sub>
185	7.88	95	0.8	0.0824	2.15	
180	6.0	95	0.8	0.0787	1.22	
175	6.0	95	0.8	0.0748	1.18	
170	6.0	95	0.8	0.0707	1.17	
174.97	4.0	95	0.8	0.0747	1.18	
174.97	4.0	95	0.0	0.0697	1.04	
178.25	4.0	95	0.8	0.0773	1.24	
180	6.0	85	0.8	0.0880	∞	Variable T <sub>cond</sub> .  T <sub>o</sub> = 180°F
180	6.0	90	0.8	0.0835	∞	
180	6.0	95	0.8	0.0787	1.22	
180	6.0	100	0.8	0.0741	1.15	
180	6.0	105	0.8	0.0694	1.17	
180	12.0	140	0.8	0.0379	1.69	Air-cooled condenser*
180	8.0	120	0.8	0.0563	1.58	

\*COP is less than 6.0 if A/C condenser is air-cooled.

Cost ratio determined for the system that delivers 2.3575 horsepower.

Table 6-7. Design Cost/Efficiency Matrix for 25-Ton Rankine-Cycle Unit

Parameter						$T_{ce} = 190^{\circ}\text{F}$ $\text{COP} = 6.0 \text{ (19.546 hp)}$
Boiler Temperature, $T_o$ ( $^{\circ}\text{F}$ )	Boiler Water Flow Rate (gpm/ton)	Condensed Temperature ( $^{\circ}\text{F}$ )	Regenerator Effectiveness ( $\epsilon$ )	R/C Efficiency	Cost Ratio (Cost/baseline)	Remarks
176.58	4.0	100	0.0	0.0707	1.0	BASELINE
180	6.0	100	1.0	0.0794	$\infty$	Variable regenerator effectiveness. $T_{\text{cond}} = 100^{\circ}\text{F}$
180	6.0	100	0.8	0.0780	1.27	
180	6.0	100	0.6	0.0767	1.13	
180	6.0	100	0.4	0.0754	1.07	
180	6.0	100	0.2	0.0742	1.03	
180	6.0	100	0.0	0.0730	1.01	
176.58	4.0	95	0.8	0.0800	1.39	$T_{\text{ppboiler}} = 3.0^{\circ}\text{F}$
176.58	4.0	95	0.0	0.0749	1.06	
178.97	4.0	95	0.8	0.0819	1.50	
180	6.0	95	1.0	0.0843	$\infty$	Variable regenerator effectiveness. $T_{\text{cond}} = 95^{\circ}\text{F}$
180	6.0	95	0.8	0.0828	1.39	
180	6.0	95	0.6	0.0813	1.21	
180	6.0	95	0.4	0.0799	1.15	
180	6.0	95	0.2	0.0786	1.11	
180	6.0	95	0.0	0.0773	1.07	
190	7.42	95	0.8	0.0905	$\infty$	Variable $T_o$
185	7.42	95	0.8	0.0869	4.76	
180	6.0	95	0.8	0.0828	1.39	
175	6.0	95	0.8	0.0787	1.33	
170	6.0	95	0.8	0.0744	1.32	
180	6.0	85	0.8	0.0925	$\infty$	Variable $T_{\text{cond}}$
180	6.0	90	0.8	0.0878	$\infty$	
180	6.0	95	0.8	0.0828	1.39	
180	6.0	100	0.8	0.0780	1.27	
180	6.0	105	0.8	0.0731	1.32	

Cycle efficiency, and heat transfer in Btu/hr in the preheater, boiler, re-generator, and condenser were then calculated for a given set of cycle conditions. This heat transfer and, thus, the heat-exchanger size, is characterized in the boiler and condenser by the "pinch point" temperature difference. This is shown in Figure 6-43 for the boiler. Given the cycle conditions and power output, the amount of heat input is some defined quantity. Thus, the hot fluid from the collectors (flowrate defined at 4 gpm/ton) will have a prescribed temperature drop ( $\Delta T_2$ ). For a given approach temperature difference ( $\Delta T_1$ ), there will be some pinch point temperature ( $\Delta T_{pp}$ ). A similar diagram can be constructed for the condenser, and there will be a condenser pinch point temperature difference.

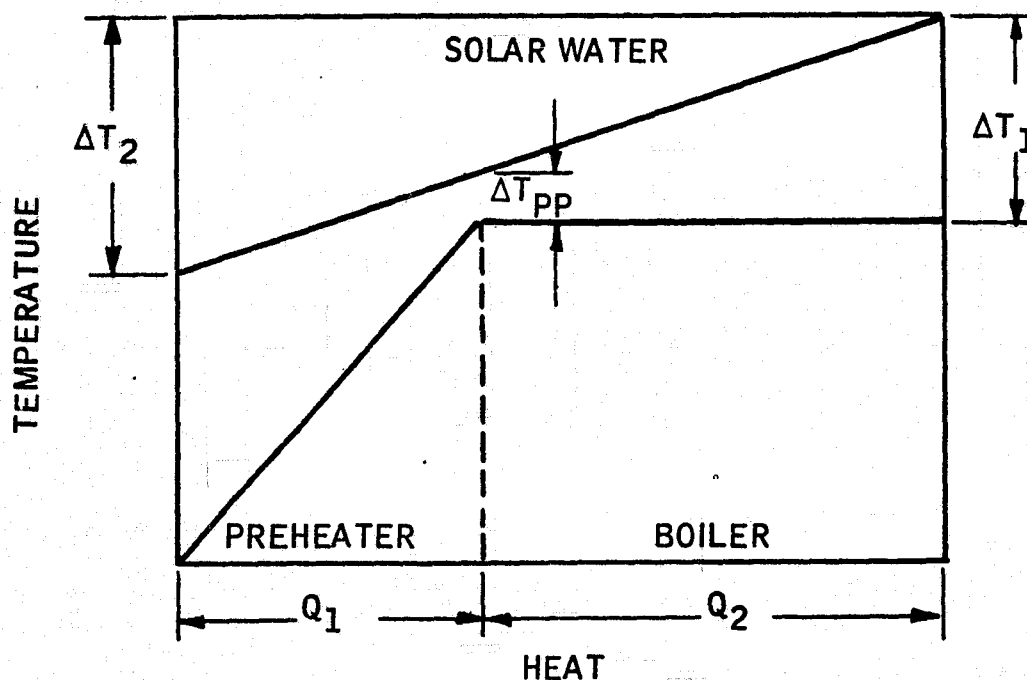


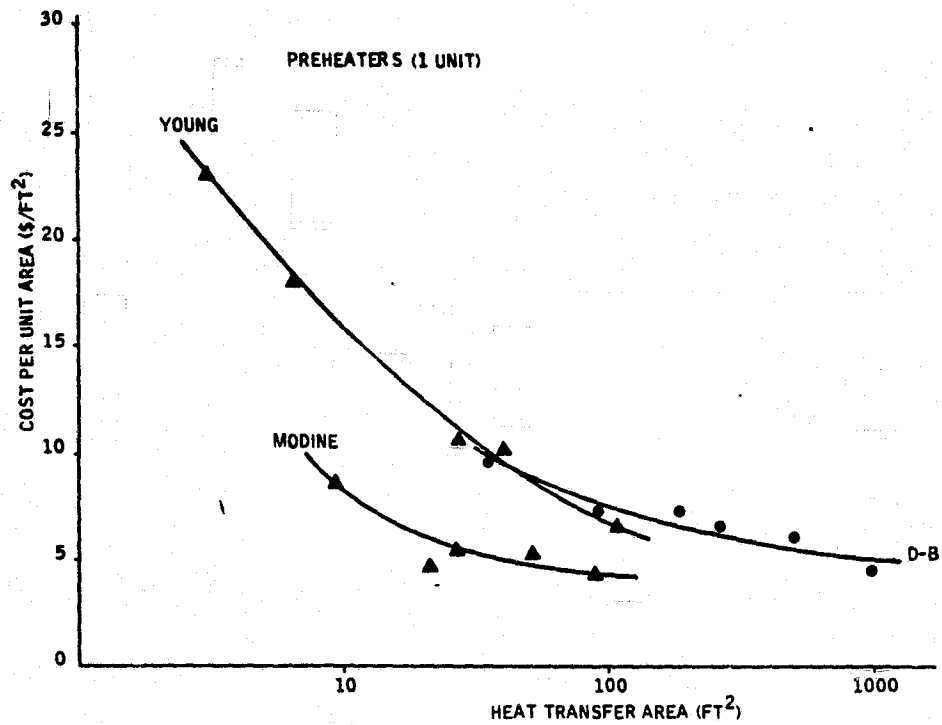
Figure 6-43. Fluid Temperature Changes in Preheater and Boiler

Values of UA were calculated for the preheater, boiler, and regenerator using the log mean temperature difference (LMTD) method. The cost of each heat exchanger was then determined from the calculated UA value and curves for cost per UA versus UA. These curves were generated from heat exchanger manufacturers' catalogs by the following procedure.

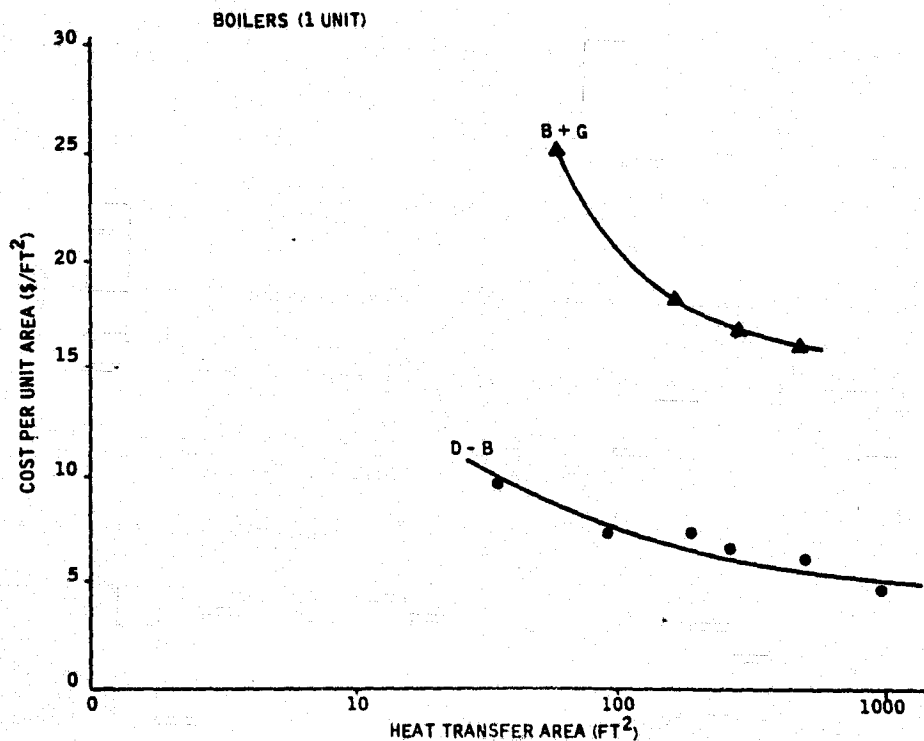
To determine heat-exchanger cost versus heat-transfer area and UA for preheaters, boilers, and condensers, values of area and UA were obtained from heat-exchanger manufacturers' catalogs, technical representatives, or were calculated from the tables and data published in the catalogs. Manufacturers' representatives were contacted to obtain current resale costs and heat-transfer areas for the heat exchangers selected at random from those listed in the catalogs. The costs and areas were then plotted as cost per unit area ( $\$/\text{ft}^2$ ) versus total heat-transfer area ( $\text{ft}^2$ ) and a smooth curve was drawn through the points. In each case, a minimum of four samples was obtained. (Figures 6-44, 6-45 and 6-46.)

For cost versus UA, the current resale costs obtained from the manufacturers' representatives were again used. However, values of UA were calculated from the catalogs using the tabulated catalog data for the particular heat exchanger (preheater, evaporator, or chiller). The values of hot and cold fluid inlet temperature, heat capacity, and flowrate used to generate the tables were used to calculate log mean temperature differences (LMTD) and values of UA so that correction factors would not have to be calculated. The costs and UA were then plotted as cost per UA versus UA and a smooth curve was drawn through the points. (Figures 6-47, 6-48 and 6-49.)

In plotting cost versus UA, two phenomena were noted. For the condensers, the cost versus total heat-transfer area plotted as a separate curve for each manufacturer (Figure 6-46). However, when plotted as cost versus UA, all data for the four manufacturers plotted as a single curve (Figure 6-49). This indicates that the condenser price is based on UA and it doesn't make



**Figure 6-44. Preheaters - Cost per Unit Area versus Heat Transfer Area**



**Figure 6-45. Boilers - Cost per Unit Area versus Heat Transfer Area**

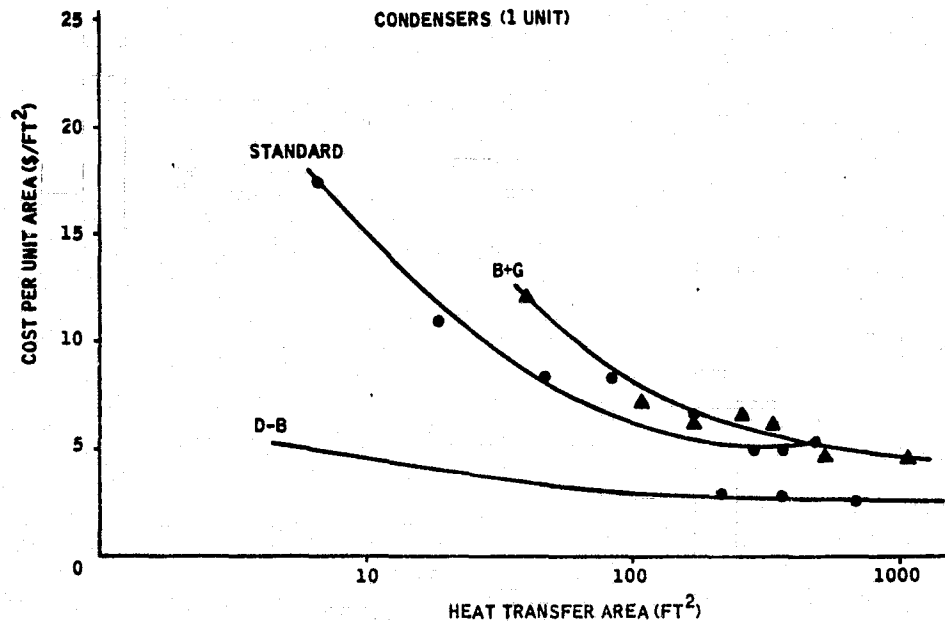


Figure 6-46. Condensers - Cost per Unit Area versus Heat Transfer Area

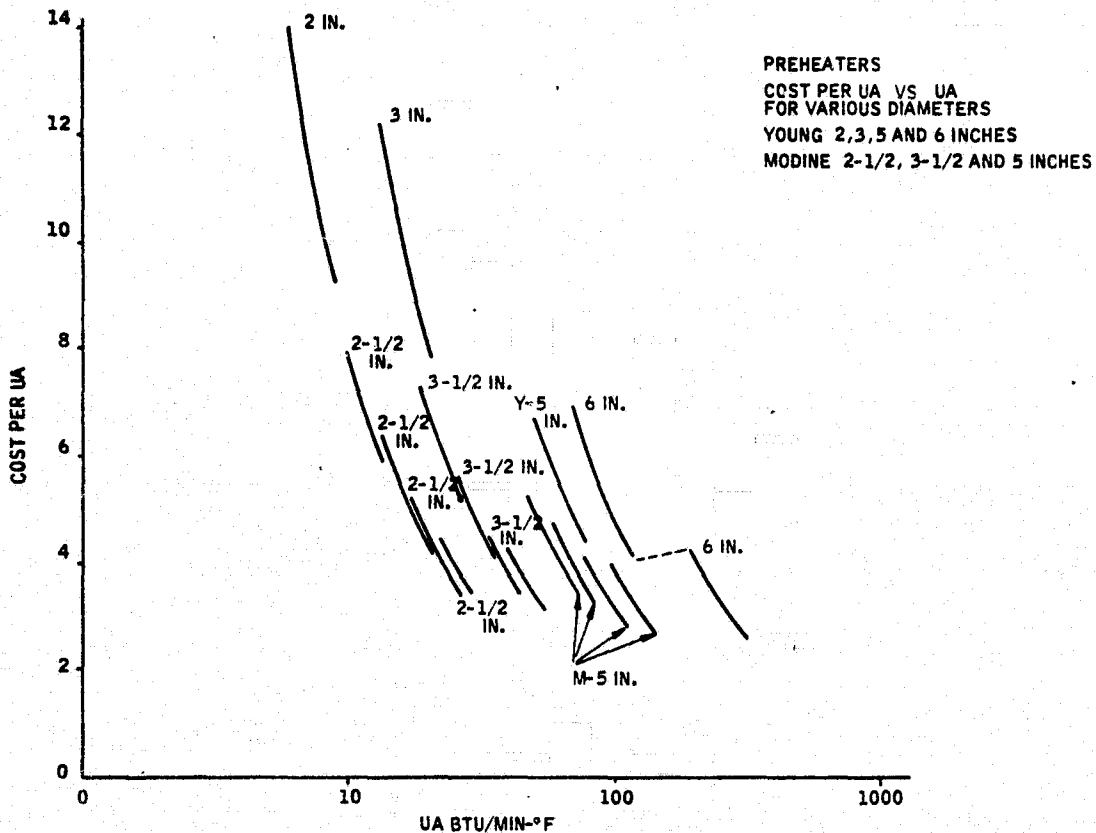


Figure 6-47. Preheaters - Cost per UA versus UA

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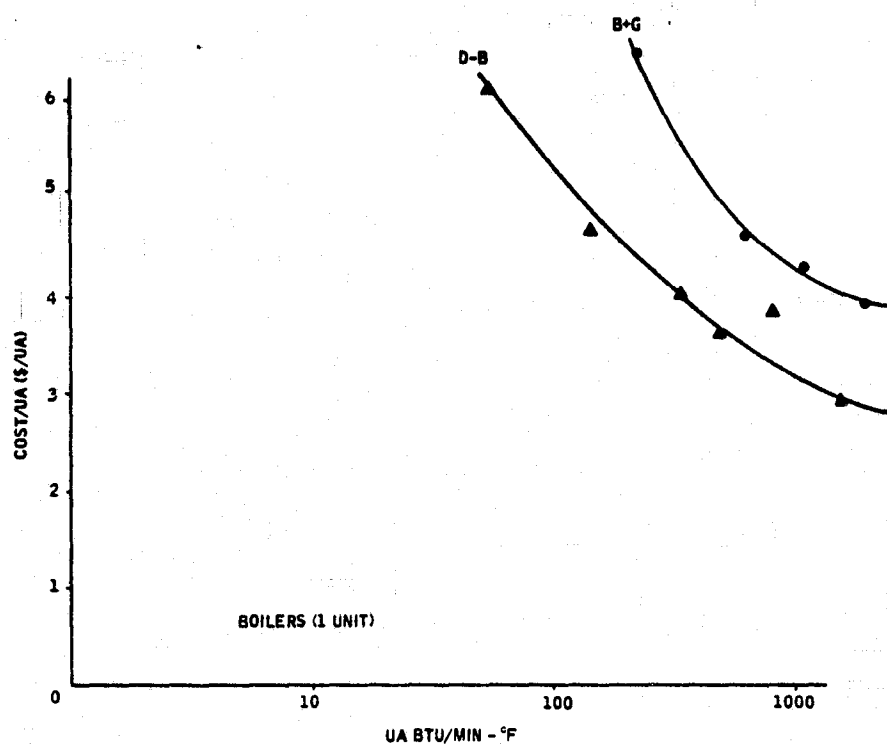


Figure 6-48. Boilers - Cost per UA versus UA

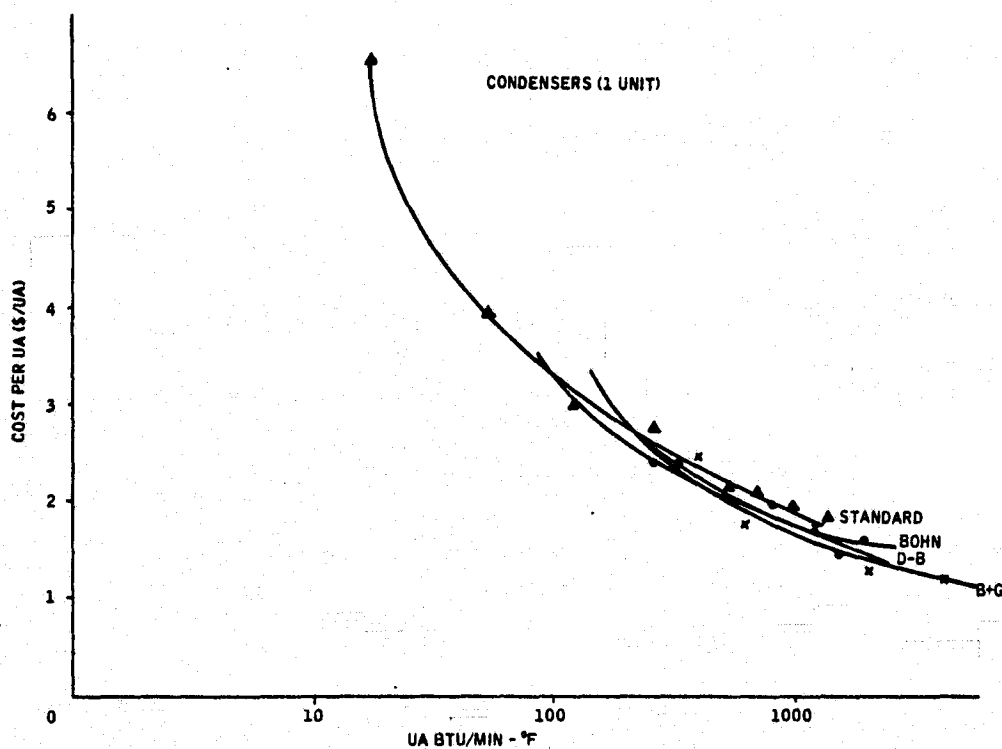


Figure 6-49. Condensers - Cost per UA versus UA



any difference whose condenser is purchased, the price will be approximately the same for a given value of UA. Therefore, a particular manufacturer's model can be selected based on total UA required and physical geometry required to fit the given transfer area plotted as one curve for each manufacturer's condensers.

When preheater cost versus UA was plotted (Figure 6-47), a series of curves for each manufacturer as a function of shell diameter and tube length was obtained. Therefore, for a given value of UA, each manufacturer had at least two preheaters which could be chosen, one being more expensive per UA than the other. To choose the proper heat exchanger and determine its total cost knowing the total UA required, it was also necessary to determine acceptable shell-and-tube pressure drops so the curve for a preheater with the right tube length and shell diameter could be used to determine cost. Since some preheaters were tube-and-shell without fins and others had finned tubes, the cost versus UA method of determining heat-exchanger cost is more accurate and simpler to use because it is not necessary to determine fin efficiencies and value of U.

When boiler cost was plotted against UA (Figure 6-48) instead of total heat-transfer area, the cost differential between manufacturers dropped markedly. For example, plotting cost per unit area versus total heat-transfer area (Figure 6-45), the cost ratio between two selected manufacturers per unit area was about 3. However, when plotted as cost per UA versus UA, the cost ratio per UA for the same two manufacturers was about 1.6. Thus, unless the values of U and fin efficiency are known, it is difficult to compare boilers knowing only the required heat-transfer area. Assuming the same value of U and fin efficiency for each manufacturer's heat exchanger, the cost difference obtained using the cost versus heat-transfer area plot would not be the same as that obtained using the cost versus UA plot.

The cost of regenerators was established from previous experience at Barber-Nichols, since there are no commercial units which are suitable for Barber-Nichols' application (Figure 6-50). Since the data are limited to several units, the

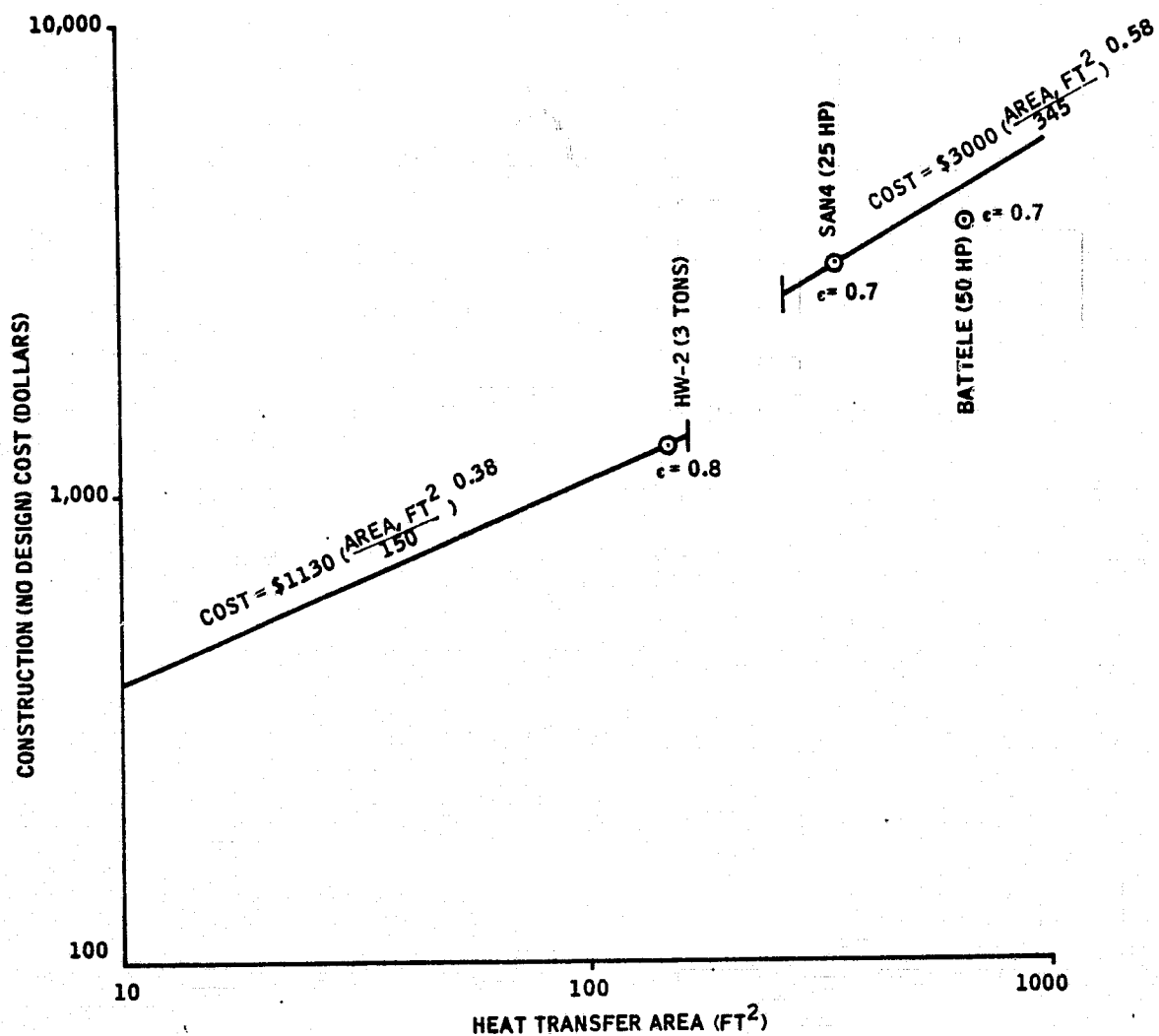


Figure 6-50. Regenerator - Construction Costs versus Heat-Transfer Area

projections are somewhat lower in confidence level than for the preheaters, boilers, and condensers. Also, they are very dependent on the design and thus the values could change somewhat with a new design. The data were correlated using the heat-transfer surface area, since the  $U$  is relatively constant for the limited number of data points.

Since the condenser pinch point was very small (about  $0.2^{\circ}\text{F}$ ) for a  $95^{\circ}\text{F}$  condensing temperature, both the LMTD and NTU method of determining UA were used, since both methods are very sensitive to small pinch point temperatures. Again, the condenser cost was determined by using the plot of cost per UA versus UA.

When the heat exchangers are purchased in quantities greater than one, the cost savings varies from 2 percent to about 20 percent depending on the quantity purchased. The final heat-exchanger costs were extrapolated to a large quantity figure. They were then added into the cost of the Rankine system to obtain a total cost. This was then divided by the baseline cost figure to obtain a cost ratio.

The results of this cost-efficiency study are for use in the selection of a design cycle as described in Section 6.11.5.7, "Selection of Design Cycle."

**6.11.5.5 Effect of Collector Flowrate on Efficiency** -- The Rankine-cycle efficiency is very sensitive to collector flowrate. Therefore, an analysis was made to determine this sensitivity. The effect of collector flowrate on the cycle takes the form of changing the pinch point temperature differential, which, in turn, results in a different boiling temperature.

The analysis required the computation of a correlation between boiling temperature and heat input to the cycle. This was done through a least-squares method of linear regression. This then allowed a boiler temperature to be computed based on a certain value of pinch point at a specified collector

flowrate. This then lead to a cycle calculation which resulted in a Rankine-cycle efficiency.

The results of this endeavor are shown in Figures 6-51, 6-52, and 6-53 for the 3-ton unit, and in Figures 6-54 through 6-56 for the 25-ton unit. It is apparent from these figures that dramatic improvements can be made in cycle efficiency by increasing the collector flowrate. However, this also will increase the primary loop pumping power and is not considered in the baseline design.

**6.11.5.6 Air Cooling** -- An investigation was conducted to examine the feasibility of air cooling the low-temperature end of the Rankine cycle. This would eliminate the cooling tower with its associated cost and maintenance problems.

The basic change in the cycle is the replacement of the water-cooled condenser with an air-cooled condenser. The temperature at the low end of the Rankine cycle then becomes the design ambient plus whatever temperature change is required across the heat exchanger for the air and Freon films. Air (gases in general) is inherently a poor heat-transfer medium and requires a temperature difference approximately an order of magnitude greater than that required by water to transfer the same amount of heat. This means that with air-cooled condensers the temperature differences will be 20° to 40° as compared with 5° to 10° with water cooling, all other conditions being equal.

To determine the actual effect on the cycle, two cases were computed. The first case used a fairly standard size air-cooled condenser which had an approach temperature difference (condensing temperature less entering air temperature) of 40°F. The second case used a much larger unit but with a temperature difference of 20°F. Both cases used an ambient air temperature of 100°F, which is site dependent but is probably a typical value.

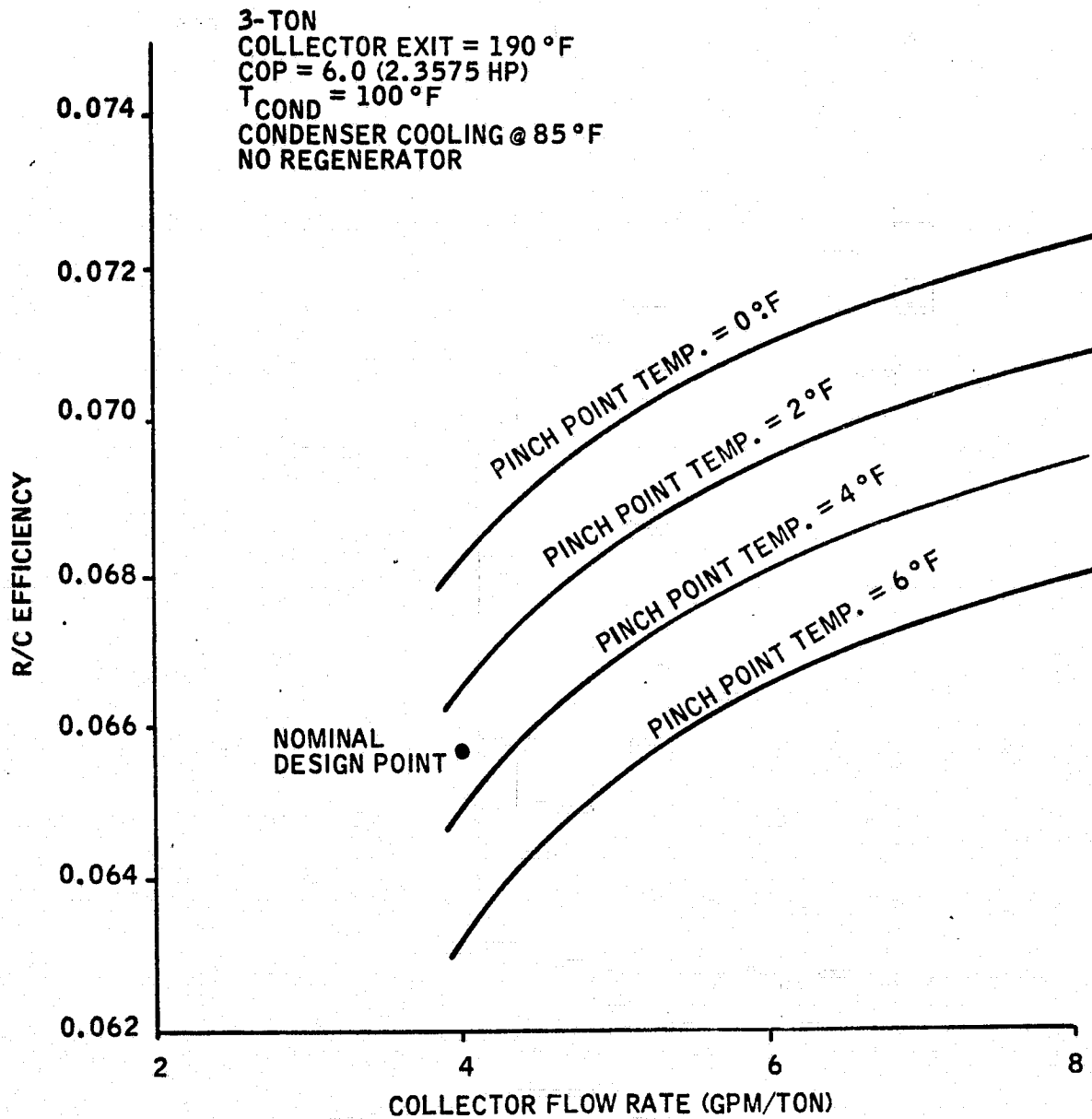


Figure 6-51. Effect of Collector Flow Rate on Rankine-Cycle Efficiency for 3-Ton Unit with No Regenerator and  $T_{COND} = 100^{\circ}\text{F}$

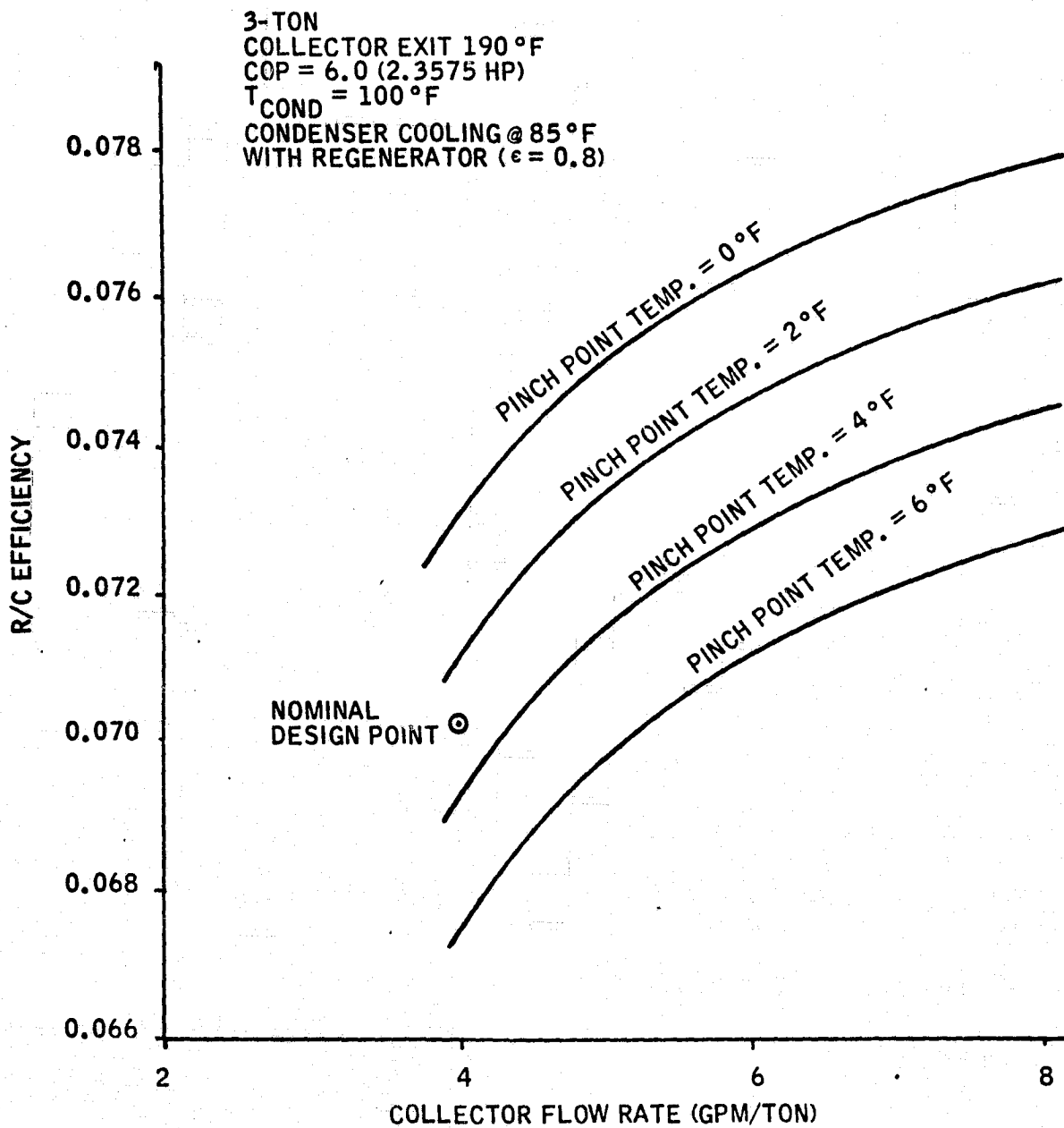


Figure 6-52. Effect of Collector Flow Rate on Rankine-Cycle Efficiency for 3-Ton Unit with Regenerator and  $T_{\text{COND}} = 100^\circ\text{F}$

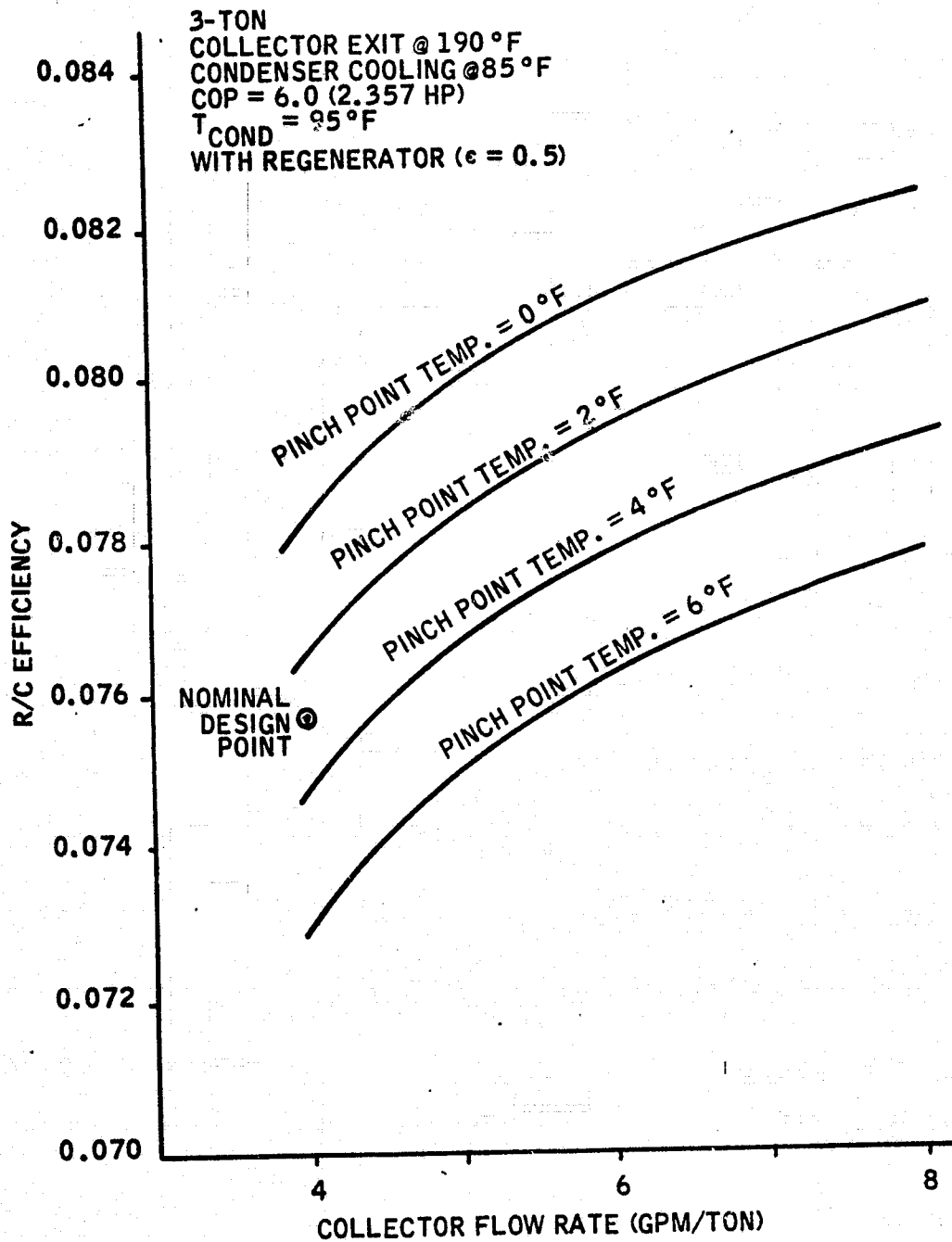


Figure 6-53. Effect of Collector Flow Rate on Rankine-Cycle Efficiency for 3-Ton Unit with Regenerator  
 $T_{COND} = 95^{\circ}\text{F}$

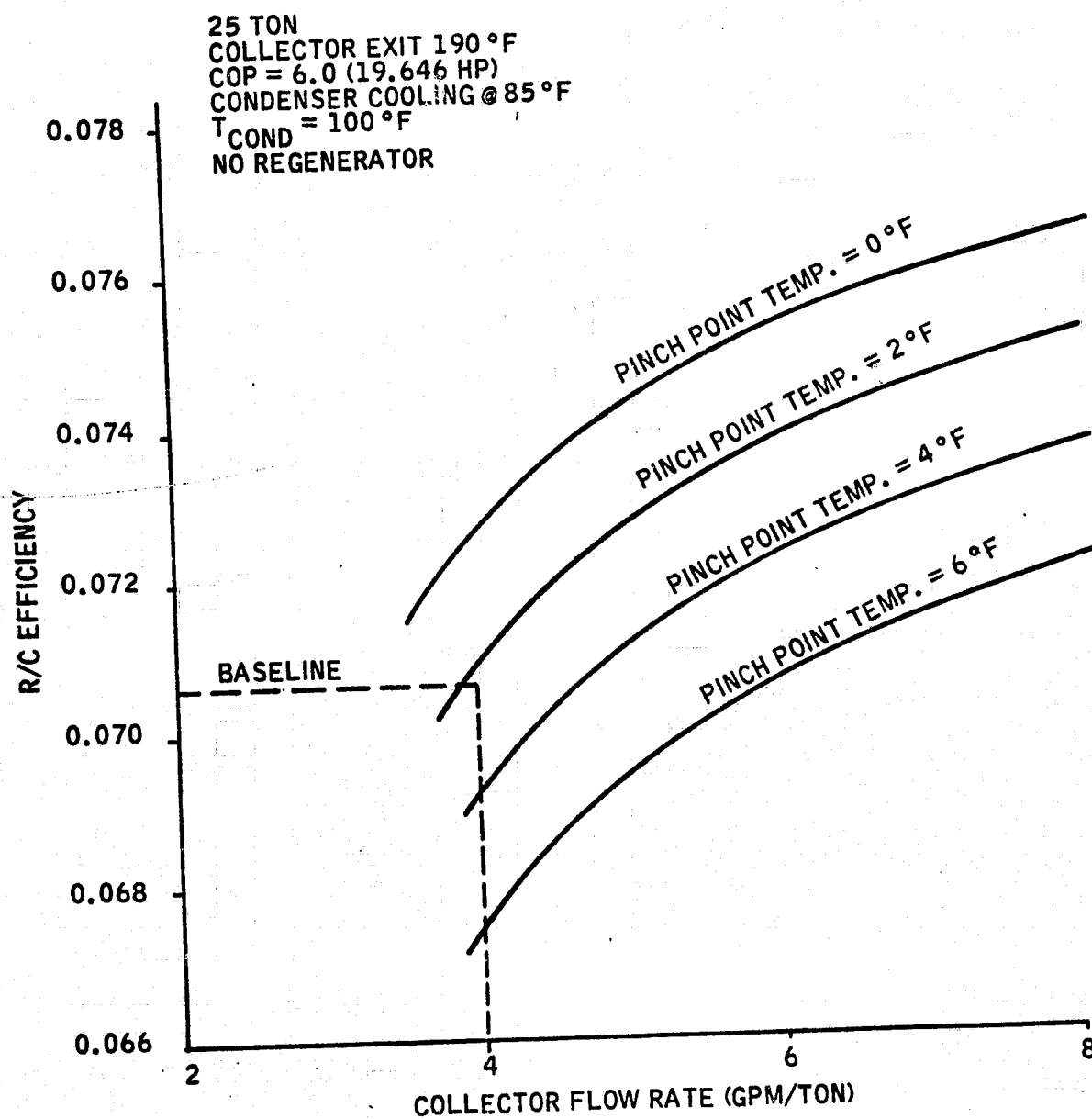


Figure 6-54. Effect of Collector Flow Rate on Rankine-Cycle Efficiency for 25-Ton Unit with No Regenerator and  $T_{COND} = 100^\circ\text{F}$



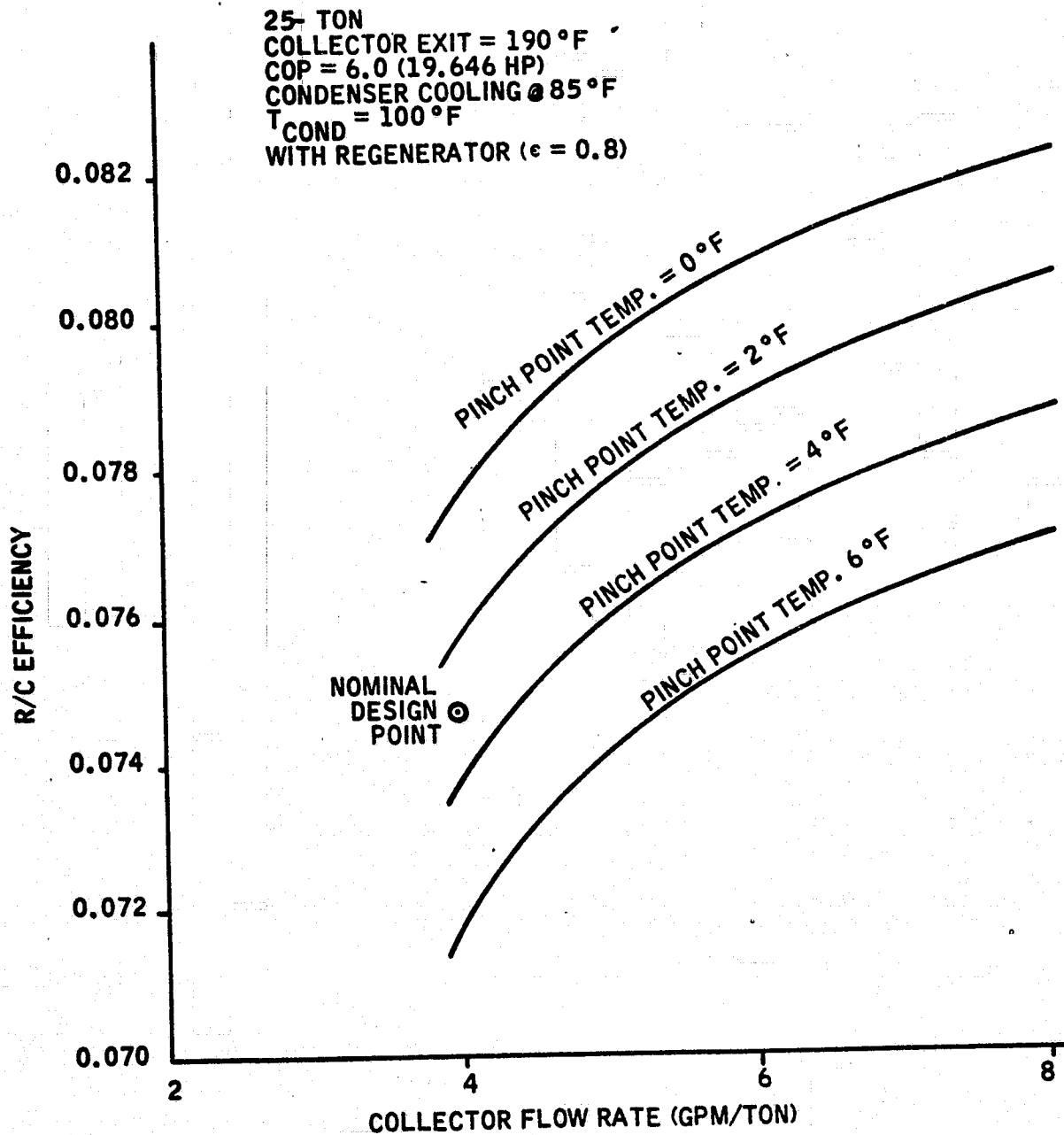


Figure 6-55. Effect of Collector Flow Rate on Rankine-Cycle Efficiency for 25-Ton Unit with Regenerator and  $T_{COND} = 100°F$

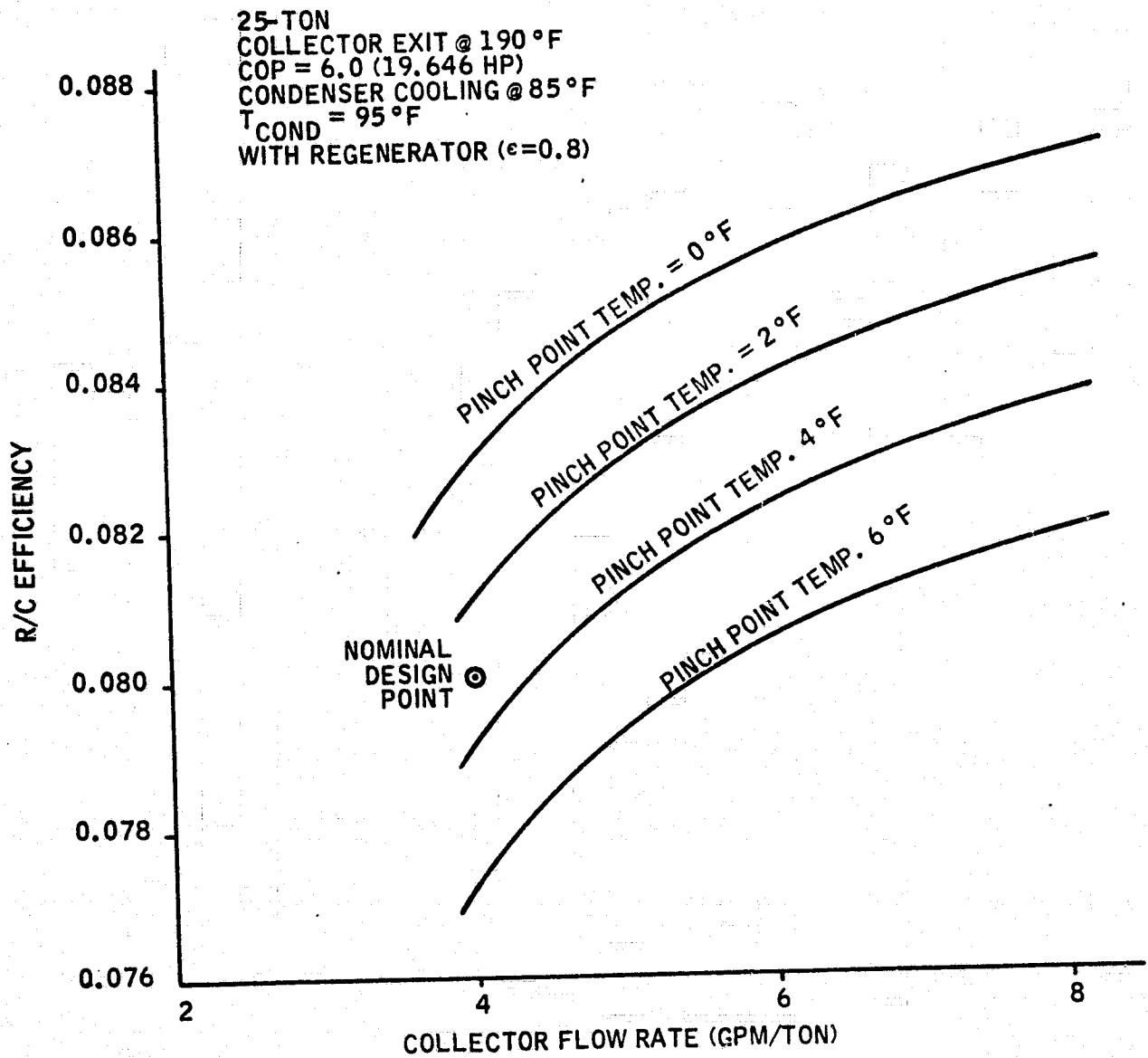


Figure 6-56. Effect of Collector Flow Rate on Rankine-Cycle Efficiency for 25-Ton Unit with Regenerator and  $T_{COND} = 95°F$

The results of this analysis are presented in Table 6-8. As can be seen, the efficiency of the Rankine cycle suffers greatly in going to air cooling while the cost of the cycle actually increases. Another factor, not shown in the table or analysis, is that the air-conditioner cycle will also suffer a large penalty in efficiency (COP), in fact, even to a greater extent than that suffered by the Rankine cycle. This is because the change in condenser temperature represents a larger percentage change in the temperature limits of the air-conditioner cycle than it did for the Rankine cycle.

Table 6-8. Air-Cooled Rankine Cycle

Cycle	Condensing Temp (°F)	Condenser Approach ( $\Delta T$ ) (°F)	Rankine Cycle, $\eta$ (%)	Cost Factor* ( $\frac{\text{cost air-cooled}}{\text{cost water-cooled}}$ )
Water Cooled	95	10	7.9	1.00
Air	120	20	5.6	1.58
Air	140	40	3.8	1.69

\* Cost of cooling tower not included.

The result of this analysis is that air cooling is not a reasonable alternative. In fact, it readily demonstrates that a vast improvement can be made in the performance of the air conditioners presently on the market by using water cooling. It might not be cost-effective with the cost of the cooling tower included, but it would be extremely energy-effective.

**6.11.5.7 Selection of Design Cycle** -- In the cost-efficiency analysis it became apparent that a baseline cycle could be achieved by careful engineering but would not take extensive development nor inordinately large and expensive heat exchangers. Once the baseline cycle was defined, an analysis was made to determine what possible improvements in the cycle could be made and

what their costs would be. This then led to a plan of action that dictates each step in improvement and the cost of that step. Thus, the cycle can be improved to whatever extent is desired from a cost standpoint.

The baseline cycle for the 3-ton unit is based on a collector temperature of  $190^{\circ}\text{F}$  (at 4 gpm/ton) and a condenser coolant flow of 3 gpm/15,000 Btu/hr at  $85^{\circ}\text{F}$ . The power out of the gearbox is 2.36 horsepower, which is the power required to drive a 3-ton air conditioner at a COP of 6.0. The working fluid is R-113 (a fluorocarbon refrigerant). This fluid superheats as it passes through the turbine; thus, it has some energy available for regeneration (however, a regenerator is not included in the baseline unit).

The heat-exchanger size (and cost) is largely controlled by its pinch point temperature difference. Obviously, the pinch point cannot be negative (heat flow in wrong direction). Starting with a large pinch point (implies small heat exchanger), the size (surface area) of the heat exchanger must increase as the temperature difference decreases (small pinch point) in order to transfer the same amount of heat. For the Rankine cycle, it is more involved than this, since the efficiency and amount of heat input change as the pinch point changes. The heat exchanger theoretically becomes infinite in size as the temperature difference goes to zero. In reality, there is a minimum value (around  $2^{\circ}\text{F}$  for R-113) required to get the boiling mechanism to take place. For the baseline cycle, the boiler preheater heat exchangers are sized based on a pinch point of  $3^{\circ}\text{F}$ , and the condenser is sized on a pinch point of  $5^{\circ}\text{F}$ .

The other heat exchanger that has a major effect on cycle efficiency and cost is the regenerator. This device is a gas (poor heat transfer)-to-liquid heat exchanger with the requirement of low-pressure drop on both sides. Available commercial units are not suitable for Barber-Nichols applications, therefore a regenerator must be designed and built specifically for this purpose, resulting in high cost. Therefore, it was not included in the baseline cycle, causing a loss in efficiency.

The baseline turbine is assumed to be 72 percent efficient. This is possible with careful engineering but without extensive development. The pump is assumed to be 50 percent efficient with the same criteria as the turbine.

The baseline cycle has an efficiency of 0.0657. The most reasonable improvement to make in this cycle (strictly from consideration of the cycle and not the system) is to increase the collector flow rate to 6 gpm/ton. This allows a higher boiling temperature at the same pinch point. The improvement in the cycle is shown in Figure 6-57. Since this change demands a larger pump and more power, it will adversely affect the cost of the system operation and probably would not be feasible. Therefore, this increased flowrate was not included in the adjustments to the cycle.

The next most logical change to make is a decrease in condenser temperature (pinch point) to nearly 95°F. It is possible to closely approach this value because of the method of determining the condenser coolant flow (based on heat rejected) and due to some desuperheating at the condenser inlet. The improvement and cost increase are also shown in Figure 6-57.

The addition of a regenerator is the next improvement. It was assumed that its performance would be such that 80 percent of the superheat from the turbine exhaust would be transferred to the boiler feed liquid. This can be achieved with careful design and some development. Less effective (and smaller) regeneration can be used but the cost-effectiveness will not be as good. The result is shown in Figure 6-57. It was assumed that the decrease in condenser temperature would be done before the regenerator was added, so the costs and improvements were added to this point. This also increases the pinch point to 4°F since the amount of heat added is decreased.

The final change involves decreasing the boiler pinch point to 1°. The effect of this change on cost and performance was determined; however, the change would require extensive development and probably isn't possible to achieve.

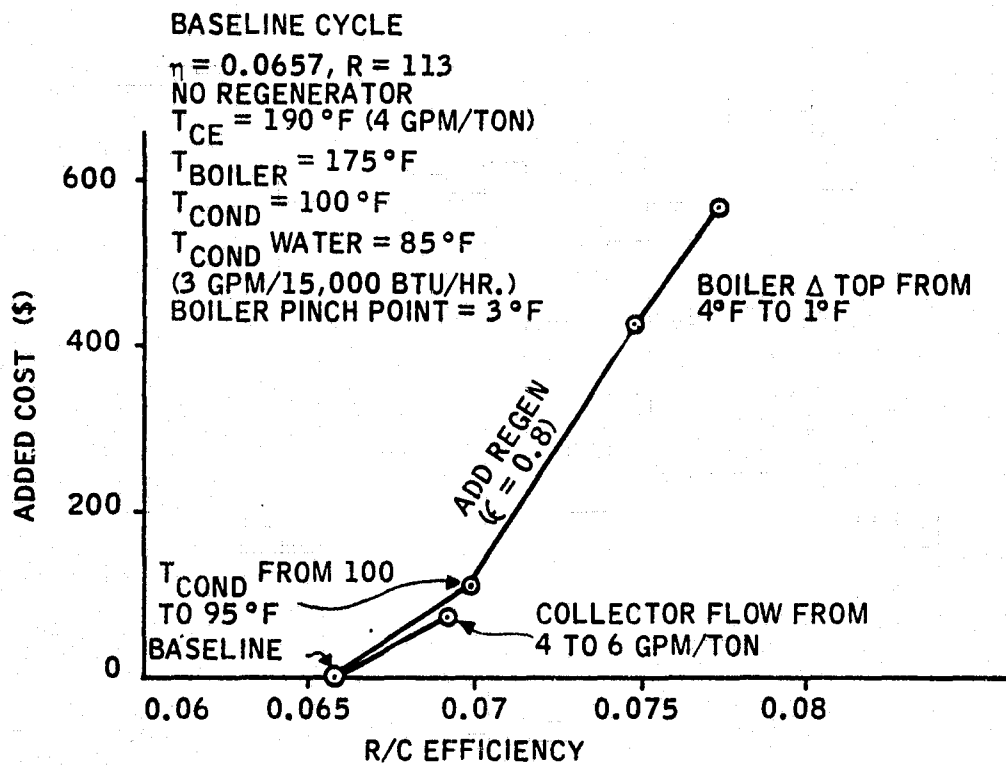


Figure 6-57. 3-Ton Cycle Improvement Costs

Similar results are shown in Figure 6-58 for the 25-ton unit. The baseline cycle again has a collector temperature of  $190^{\circ}\text{F}$  (at 4 gpm/ton) and a condenser coolant flow rate of 3 gpm/15,000 Btu/hr at  $85^{\circ}\text{F}$ . The power output is 19.6 horsepower, which is the power required to run the 25-ton air conditioner if COP is 6.0. The working fluid is R-113 but a regenerator was not included in the baseline unit. The boiler preheater pinch point is  $2.3^{\circ}\text{F}$  and the condenser pinch point is  $5^{\circ}\text{F}$ . For this case, the pinch point in the boiler with the regenerator added becomes  $3^{\circ}\text{F}$ .

An efficiency of 0.07 in Figure 6-57 shows an added cycle cost of \$130. In Figure 5-2, the same auxiliary contribution is obtained using 0.07 efficiency and  $60\text{ ft}^2$  less collector area (\$600 at  $\$10/\text{ft}^2$ ) for a net savings of \$470. Of course, the higher the cost per square foot, the more pronounced the savings.

#### 6.11.6 Cooling Subsystem Cost Effectiveness

When evaluating the cost effectiveness of the solar cooling system, one must compare the system cost savings with conventional cooling approach. The site selection also has a strong effect on the final system design and economic effectiveness. This subject will be addressed in detail at the PDR using the latest design and cost figures. A handout will be presented representing this section.

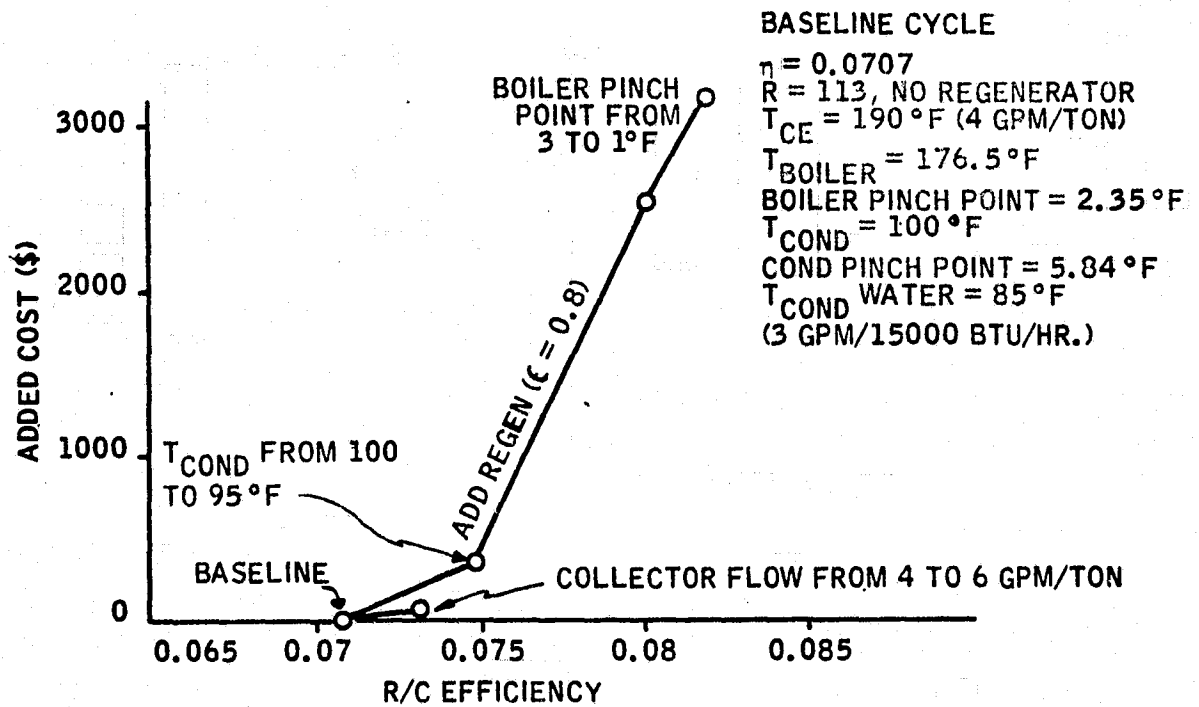


Figure 6-58. 25-Ton Cycle Improvement Costs



## SECTION 7 SPECIAL DATA

The following subsections delineate the schedule and data information requested by Appendix B, paragraph 4.1 in the contract S. O. W.

### 7.1 DRAWING LIST

The attached chart (Figure 7-1) outlines the drawings required to define each of heating and heating/cooling systems. To date, only top-level schematics and a reference to the appropriate appendix of the System Specification SK 140021 are shown as being completed. As the site-specific design is accomplished, an installation drawing will be created and identified for each system and assembly drawings prepared for each subsystem.

### 7.2 TYPE I, II AND III DOCUMENTS

The following Type I documents are being submitted with this document and are expected to be approved at the Preliminary Design Review. We are also submitting Item 3 below with this document.

1. Heating and Cooling Systems Development Plan--F3437-P-201
2. Heating and Cooling Systems Verification Plan--F3437-P-202
3. Preliminary Hazard Analysis--F3437-P-203

The following Type II document has been revised and resubmitted for approval.



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1. Quality Assurance Plan-F3437-Q-101--Revised 12/10/76.

The following Type I or Type II documents are approved:

1. Qualification Tests for Collector Subsystem--F3437-T-101
2. Safety and Health Plan
3. New Technology Reporting Plan--F3437-P-104
4. Work Breakdown Structure and Dictionary--F3437-A-101

The System Performance Specification is not being submitted at this time as Honeywell lacks the site data to prepare the revision.

### 7.3 HEATING/COOLING SYSTEMS SCHEDULE INFORMATION

#### 7.3.1 Site Data Acquisition Subsystem Schedule

The sensors furnished by MSFC as part of the Site Data Acquisition Subsystem will be required for the Heating and Cooling Systems as listed below. These dates are based on the key assumption that the site-ready date and the contractual delivery date of 5/15/78 are approximately coincident. Any significant change in this relationship will alter these dates. Honeywell assumes the balance of the data acquisition subsystem to be available (on site) at site-ready date.

<u>Subsystem</u>	<u>Sensor Need Date</u>
Single Family Residential	1 September 1977
Multifamily Residential	1 September 1977
Commercial Applications	1 September 1977

### 7.3.2 Installation Drawing Schedule

The installation drawings for the Heating and Cooling Systems will be completed approximately 15 May 1977 providing the H/C site data is received by 15 February 1977.

### 7.3.3 Prototype Design Review

The current plan shows the Heating and Cooling System Prototype Design Review as occurring approximately 1 August 1977. This is directly dependent on receipt of H/C site data by 15 February 1977.

## 7.4 PROTOTYPE DESIGN REVIEW DATA

The data Honeywell recommends to be used to accomplish the Prototype Design Review is as follows:

- System Specifications for SF, MF, and Commercial
- Installation drawings for each site\*
- System top assembly drawing or bill of material with site specific callouts\*
- Subsystem assembly drawing or bill of material with site specific callouts\*
- Procurement specifications\*
- Verification data and status summary

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\* This element is shown in Figure 7-2.

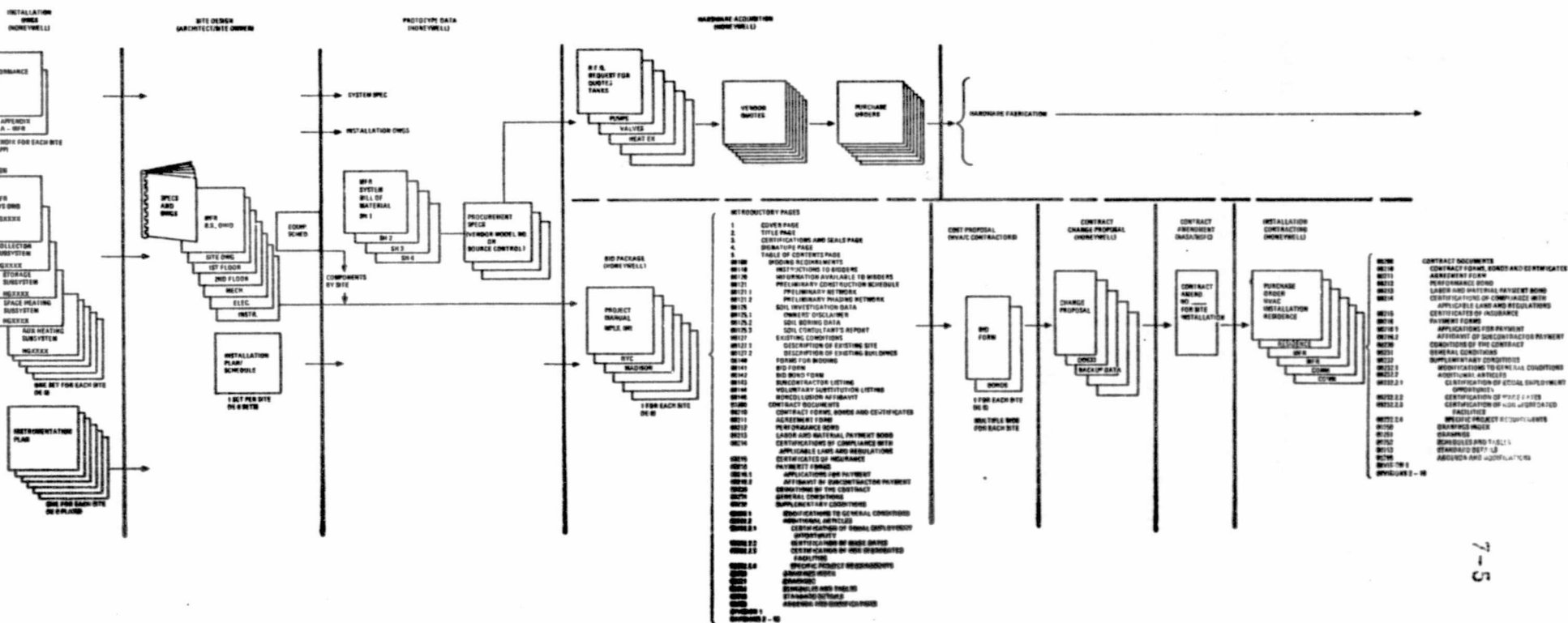


Figure 7-2. Engineering Data Flow to Prototype Design Review

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- Final hazard analysis
- List of special handling, installation and maintenance tools
- Review of support philosophy and spare parts requirements
- Proposed format and outlines for installation, operation and maintenance manuals
- Other Type I, II and III documentation
- Results of analysis and tradeoff studies