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SOLAR-ASSISTED HEAT PUMPS FOR THE HEATING AND COOLING OF BUILDINGS

**Six Month Technical Report
November 6, 1978 to May 5, 1979**

Contract No. EG-78-C-03-1719

AC03-78CS31719

**Prepared for
UNITED STATES DEPARTMENT OF ENERGY
WASHINGTON, D.C. 20545**

**Prepared by
General Electric Company
Corporate Research and Development
Schenectady, New York 12301**

May 1979

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GENERAL ELECTRIC

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FOREWORD

This second six-month technical report, "Solar-assisted Heat Pumps for the Heating and Cooling of Buildings," was prepared by the Thermal Branch of the Mechanical Systems and Technology Laboratory of the General Electric Company's Research and Development Center in Schenectady, New York. The report is being submitted to the United States Department of Energy in Washington, D.C., and describes work completed under Contract No. EG-78-C-03-1719 between November 6, 1978 and May 5, 1979.

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ABSTRACT

This report describes the General Electric Company's research and development efforts on solar-assisted heat pumps for the heating and cooling of buildings. Phase I of this contract requires the determination of the most cost-effective SAHP (Solar-assisted Heat Pump) system. The second six-months of the Phase I work, beginning November 6, 1978 and ending May 5, 1979, are presented in this report. The series (solar-assisted), parallel (solar-boasted), and standard heat pump systems with electric resistance heat backup are emphasized. Performance characteristics of all major SAHP components are determined and used in conjunction with the TRNSYS simulation program to obtain heating and cooling system performance for four different climates: Phoenix, Arizona; New York, New York; Fort Worth, Texas; and Madison, Wisconsin. Material and installation costs are obtained for the SAHP components and are used in a life-cycle cost analysis to determine the economic viability of the various systems. Ground coupling, evacuated tube collectors, and power demand distributions are studied. A marketing analysis is performed to assess the potential of SAHP systems in the residential marketplace.

The results of this study indicate that solar-boasted heat pumps (parallel) will be more cost effective than solar-assisted (series) heat pumps for the foreseeable future. Since all components for solar-boasted heat pumps are now commercially available, no further development work is needed to optimize the heat pumps for such a system. Furthermore, SAHP systems are not currently cost effective when compared to standard air-to-air heat pumps, nor is there evidence that they will be in the foreseeable future. Thus, the General Electric Company recommends that any further work under this contract be terminated.

EXECUTIVE SUMMARY

Solar-assisted heat pumps (SAHP) are potentially the most cost-effective solar heating and cooling systems. Under Phase I of Department of Energy Contract No. EG-78-C-03-1719, the General Electric Company has determined the most cost-effective SAHP systems. This report documents the second six months of the Phase I work.

SOLAR-ASSISTED HEAT PUMP COMPONENT MODELS

Two heat pumps are modeled in the cooling mode for use in the series (solar-assisted) and parallel (solar-boosted) systems. An air-to-water unit is used in the series system, while an air-to-air unit is used in the parallel system. A well insulated house is modeled and used to compute the heating and cooling loads from ambient dry bulb and wet bulb temperatures.

SYSTEMS STUDY

Parallel and series residential systems are simulated in cooling and heating in four different climatic regions; Fort Worth, Texas; New York, New York; Madison, Wisconsin; and Phoenix, Arizona. Commercial and multifamily structures are considered through an analysis of representative load profiles. The evaluation of the SAHP systems relative to peak power demand and different electricity pricing strategies is carried out. Ground coupling is explored as a means of improving SAHP system performance.

ECONOMIC ANALYSIS

A cost-estimating procedure for determining the installed consumer cost of a SAHP system is presented and used to determine candidate system costs. This permits a life-cycle cost comparison between SAHP systems and a reference system (air-to-air heat pump) to be performed.

MARKETING ANALYSIS

Existing government and solar industry marketing studies are presented and analyzed. These studies, in conjunction with the observed purchasing habits of consumers of central heat pump and air-conditioning systems, are used to predict the market share that SAHP systems can expect to enjoy.

CONCLUSIONS

The results of this study indicate that solar-boosted heat pumps (parallel) will be more cost effective than solar-assisted (series) heat pumps for the foreseeable future. Since all components for solar-boosted

heat pumps are now commercially available, no further development work is needed to optimize the heat pump for such a system. Furthermore, SAHP systems are not currently cost effective when compared to standard air-to-air heat pumps, nor is there evidence that they will be in the foreseeable future. Thus, the General Electric Company recommends that any further work under this contract be terminated.

Section 1

INTRODUCTION

Solar-assisted heat pumps (SAHP) are potentially the most cost-effective solar heating and cooling systems. The objective of Phase I of Department of Energy Contract No. EG-78-C-03-1719, the General Electric Company is to determine the most cost-effective SAHP system. This report documents the second six months of technical work which involved modeling and analyzing the candidate SAHP systems.

Conclusions from the first six months of technical work initiated and guided the direction of the simulations carried out during the reporting period. The reader is referred to the first six-month technical report [Ref. 1] in several instances in order to obtain back-up details of important technical decisions and assumptions. From the first six months of work, optimized equipment sizes and control strategies were developed. Preliminary component cost estimates and system life cycle costs were generated for the New York City climate.

Simulation efforts during the first six months of the contract concentrated on residential heating to quickly eliminate the least desirable candidate systems. It was concluded from this initial screening that series (solar-assisted) and parallel (solar-boosted) systems were the most promising candidates. Efforts during the reporting period, thus, concentrated on these systems. The scope of the simulation is expanded to include four climatic regions which are to be representative of major market centers. These climates are New York, New York; Phoenix, Arizona; Fort Worth, Texas; and Madison, Wisconsin. They are available as hourly data on Solmet weather tapes representative of weather conditions for an average year at these locations.

Cooling as well as heating performance is evaluated, necessitating the creation of a transient house model for cooling load generation and heat pump cooling performance maps. Component costs and system designs are reevaluated to obtain improved system costs. The application of SAHP systems to the heating and cooling of multifamily and commercial systems is also addressed, as is power demand distributions for candidate systems, including the cascade configuration (which was eliminated from detailed examination of life-cycle costs based upon the studies performed during the

Ref. 1. Heinz Jaster et al., Solar-assisted Heat Pumps for the Heating and Cooling of Buildings, First Six Months Technical Report prepared for the U.S. Department of Energy, Contract No. EG-78-C-03-1719, Corporate Research and Development, General Electric Company, Schenectady, New York, November 1978, Section 4.

initial six months of this study). Ways to improve the performance of SAHP systems, such as ground coupling or the use of evacuated tube collectors are considered. Finally a marketing and life-cycle costing analysis is performed to achieve the primary objective of Phase I -- the determination of the most cost-effective SAHP system and its potential to become a marketable product.

Section 2

SOLAR-ASSISTED HEAT PUMP COMPONENT MODELS

HEAT PUMP COOLING PERFORMANCE

Two types of heat pumps were modeled in the cooling mode for use in the series and parallel systems: an air-to-water unit for use in the series system and an air-to-air unit for use in the parallel system.

HEAT PUMP COMPONENT SPECIFICATIONS

The specifications of each of the heat pump components are listed in Table 2-1. These components are nearly identical with those used in the heat pumps operating in the heating mode. The differences include: 1) in cooling, the evaporator and condenser interchange roles; and 2) in cooling, the adiabatic and compressor motor efficiencies are increased due to the lower pressure ratio.

Table 2-1

HEAT PUMP COOLING SYSTEM COMPONENTS

<u>Compressor</u>	
Adiabatic efficiency	0.8
Motor efficiency	0.85
Clearance factor	0.0883
Geometric displacement rate	$1.36 \times 10^{-3} \text{ m}^3/\text{s}$ per nominal ton
<u>Refrigerant -- Air Evaporator (Nominal Five Ton) (Indoor Unit)</u>	
Finned-tube, cross-flow design	
Tube area	1.63 m^2
Four parallel refrigerant circuits,	each 1.11 cm i.d. and 11.5 m long
Air flow rate	$0.755 \text{ m}^3/\text{s}$
<u>Refrigerant -- Air Condenser (Nominal Four Ton) (Outdoor Unit)</u>	
Finned-tube, cross-flow design	
Tube area	1.82 m^2
Two parallel refrigerant circuits,	each 1.11 cm i.d. and 25.9 m long
Air flow rate	$1.510 \text{ m}^3/\text{s}$
<u>Water -- Condenser</u>	
Tube-in-tube counter-flow design	
Two parallel circuits	
Tube diameter	1.11 cm
Tube length	18.3 m/circuit
Water flow rate	$5.74 \times 10^{-4} \text{ m}^3/\text{s}$

SERIES SYSTEM

The "base case" series system heat pump utilizes a 5-ton nominal air evaporator, a water condenser, and a 2-speed compressor with geometric displacement rates corresponding to a 1.6- to 0.8-ton unit. This combination of component sizes is hereafter known as a 4-ton unit. The unit delivers 9.15 kW of cooling capacity at a COP (coefficient of performance) of 6 for water and air temperatures of 25 °C at full speed. Fan and/or pump power are not included in the COP value.

PARALLEL SYSTEM

The "base case" parallel system heat pump utilizes a 5-ton air evaporator, a 4-ton air condenser, and a 2-speed, 4-ton/2-ton compressor. This unit is also designated as a 4-ton unit. The unit delivers 14.5 kW at a COP of 2.9 for outdoor and indoor air temperatures of 35 °C and 25 °C, respectively, at full speed. Fan power is not included.

COOLING SIMULATIONS

In the cooling simulations, fractional sizes of the above 4-ton "base case" units were used; i. e. with a 2-ton series heat pump in cooling consisted of a 2.5-ton air evaporator with a 2-circuit 0.44-inch-diameter, 30-foot-long water condenser having a 4.5 gpm water flow rate, and a 0.8- to 0.4-ton, 2-speed compressor.

PERFORMANCE

Figures 2-1 through 2-5 show (for both series and parallel heat pumps) COP, heating capacity, and latent heat ratio vs. outdoor temperature or

Figure 2-1.

Air-to-air heat pump cooling performance -- COP and capacity as functions of outdoor temperature (half-speed operation)

Figure 2-2.

Air-to-air heat pump cooling performance -- COP and capacity as functions of outdoor temperature (full-speed operation)

Figure 2-3.

Air-to-air heat pump cooling performance -- latent heat ratio as a function of outdoor temperature

Figure 2-4.

Water-to-air heat pump cooling performance -- COP and capacity as functions of inlet water temperature

Figure 2-5.

Water-to-air heat pump cooling performance -- latent heat ratio as a function of inlet water temperature

water inlet temperature, and indoor relative humidity at full- and half-speed operation.

TRANSIENT HOUSE MODEL

A well insulated 167 m^2 house was modeled and used in the TRNSYS simulations to compute the heating load, the sensible cooling load, and the latent cooling load from inputs of ambient dry bulb and wet bulb temperatures. The house demand consists of conduction, infiltration, internally generated heat, and solar gains. The cooling loads are considered positive. The house internal temperature is assumed constant at 25.6°C if cooling is required and at 20.0°C if heating is required. The calculations are based on a house internal air mass of 499 kg , which reflects a house volume of 425 m^3 . Solar gain is based on windows placed on the north and south faces of the house. The variables and equations used for computing the house load are given in the following subsections where:

- M_a = house air mass, kg
- M_v = house vapor mass, kg
- C_p = specific heat of air = $1004 \text{ J/kg} \cdot ^\circ\text{C}$
- $\dot{M}_{a_{OD}}$ = air infiltration, kg/h
- $\dot{M}_{a_{ID}}$ = air exfiltration, kg/h
- $\dot{M}_i$ = internal water vapor generation, kg/h
- $\dot{M}_l$ = water vapor removed by heat pump, kg/h
- $\dot{M}_{v_{OD}}$ = vapor infiltration, kg/h
- $\dot{M}_{v_{ID}}$ = vapor exfiltration, kg/h
- ω_{ID} = indoor specific humidity
- ω_{OD} = outdoor specific humidity
- h_{fg} = latent heat of vaporization for water = $2.28 \times 10^6 \text{ J/kg}$
- K = overall house shell conductance, $\text{W}/^\circ\text{C}$
- T_{OD} = outdoor temperature, $^\circ\text{C}$
- T_{ID} = indoor temperature, $^\circ\text{C}$

Q_1 = internal sensible heat generation, W

Q_2 = internal latent heat generation, W

Q_3 = latent heat infiltration, W

Q_4 = net sensible heat infiltration, W

Q_5 = conduction gain, W

Q_6 = latent heat exfiltration, W

Q_7 = solar heat gain, W

Q_{sens} = sensible house load, W

$$= Q_1 + Q_4 + Q_5 + Q_7$$

$Q_{\text{sens}_{\text{HP}}}$ = sensible heat pump capacity, W

Q_{lat} = latent house load, W

$$Q_2 + Q_3 + Q_6$$

$Q_{\text{lat}_{\text{HP}}}$ = latent heat pump capacity, W

Q_{total} = total house load, W

$$= Q_{\text{sens}} + Q_{\text{latent}}$$

Q_{HP} = total heat pump capacity, W

$$= Q_{\text{sens}_{\text{HP}}} + Q_{\text{lat}_{\text{HP}}}$$

COMPONENTS OF HEAT GAIN

Exfiltration and Infiltration

The exfiltration and infiltration load depends on the amount of air entering and leaving the house for ventilation and from cracks in windows and doors. This air brings with it both moisture and heat which results in both a sensible and latent load. The volume of the house is 425 m^3 , with one air change every 2 hours.

Therefore

$$M_a = 499 \text{ kg}$$

$$\dot{M}_{a_{ID}} = \dot{M}_{a_{OD}} = 249.5 \text{ kg/h}$$

$$\dot{M}_{v_{OD}} = \dot{M}_a \omega_{OD}$$

$$\dot{M}_{v_{ID}} = \dot{M}_{a_{ID}} \omega_{ID}$$

where ω_{OD} is computed at every time step from the weather tape outdoor dry bulb and wet bulb temperatures, and ω_{ID} is computed at every time step from an indoor energy and mass balance.

Thus,

$$\begin{aligned} \text{Net sensible infiltration} &= \dot{M}_{a_{ID}} \cdot C_p (T_{OD} - T_{ID}) \\ &= 38.7 (T_{OD} - T_{ID}) \text{ W} \end{aligned}$$

$$\begin{aligned} \text{Net latent infiltration} &= \dot{M}_{a_{ID}} \cdot h_{fg} \cdot (\omega_{OD} - \omega_{ID}) \\ &= 1.7 \times 10^5 (\omega_{OD} - \omega_{ID}) \text{ W} \end{aligned}$$

Solar Heat Gain

Thermopane windows with partially drawn shades are assumed for modeling the solar heat gain. This type of window will typically allow 52 percent of the total radiation that reaches it to enter the house as a heat gain [Ref. 2]. Total radiation falling on a vertical plane facing north and south is computed by the solar processor in TRNSYS, using weather tape radiation data. The house window area (m^2) is assumed to be:

<u>Climatic Region</u>	<u>North</u>	<u>South</u>
Fort Worth, Texas	13.9	4.65
Madison, Wisconsin	4.65	13.9
New York, New York	4.65	13.9
Phoenix, Arizona	13.9	4.65

Ref. 2. Carrier Corporation, Carrier Handbook of Air-conditioning Systems Design, Mc Graw Hill Book Company, 1965, pp. 1-52.

This choice of window area allows the solar heat gain to assist in meeting the heating load in cold climates while not contributing much to the cooling load in warm climates.

Conduction Heat Gain

A time lag is associated with the conduction heat gain due to the thermal capacitance of the house structure, i.e., the thermal mass of the walls, ceiling, and floors. Therefore

$$Q_5 = K [T_{OD}(t-\tau) - T_{ID}]$$

where $T_{OD}(t-\tau)$ represents the outside dry bulb temperature at some time, τ hours, before the present time, t . The time lag, τ , was taken to be 4 hours for this study based on the thermal resistance of a typical wall structure. Variation of τ from 2 to 4 hours did not affect the results significantly. The conduction heat gain coefficient K is assumed to be $211 \text{ W/}^\circ\text{C}$.

Internal Load Generation

The internally generated load results from sensible and latent heat released within the house by the occupants, lights, and appliances. For this model, the following schedule representative of a typical family of four was used. It is based on average heat generation rates of people and appliances.

Sleeping hours 0 to 7 $Q_1 = 586 \text{ W}$
23 to 24

$Q_2 = 152 \text{ W}$

Working hours 7 to 23 $Q_1 = 1465 \text{ W}$

$Q_2 = 615 \text{ W}$

HOUSE CALCULATIONS

Water Vapor Mass Balance on House

$$\Delta M_v = \left(\dot{M}_{vOD} + \dot{M}_i - \dot{M}_l - \dot{M}_{vID} \right) \Delta t$$

$$M_a \left(\frac{\Delta \omega_{ID}}{\Delta t} \right) + \dot{M}_a \cdot \omega_{ID} = \dot{M}_a \cdot \omega_{OD} + \dot{M}_i - \dot{M}_l$$

where $\dot{M}_l$ is a function of the LHR (latent heat ratio) of the heat pump, which depends on ω_{ID} and T_{OD} for a given heat pump design. The latent heat ratio for different heat pumps as a function of T_{OD} and ω_{ID} and is presented in Figures 2-3 and 2-5.

Energy Balance on House

The sensible house load is assumed to be met by the heat pump under all conditions. Consequently, $Q_{sensHP} = Q_{sens} = Q_1 + Q_4 + Q_5 + Q_7$.

When cooling is required, the latent load of the house is given by $Q_{lat} = Q_2 + Q_3 + Q_6$. In general, this load will not be equal to the latent capacity of the heat pump given by $Q_{latHP} = LHR \cdot Q_{heat\ pump}$

where

$$LHR = f(T_{OD}, \omega_{ID}).$$

The difference between Q_{lat} and Q_{latHP} results in the humidity of the house increasing or decreasing according to the relation:

$$Q_{lat} - Q_{latHP} = \frac{\Delta \omega_{ID}}{\Delta t} \cdot M_a \cdot h_{fg}$$

This house model produces an overall yearly heating load in New York City equivalent to the simple house model used in the heating-only simulations performed during the first six months of the contract,

$$Q_{sens} (W) = 237 \frac{W}{^\circ C} (15.6 - T_{OD}) ^\circ C$$

Therefore, all of the heating simulation results are based on the above simple house model.

Section 3

SYSTEMS STUDY

RESIDENTIAL HEATING PERFORMANCE

The analysis of parallel and series systems in the heating mode for New York City weather was documented in detail in the first six months technical report [Ref. 1]. During the reporting period, these systems were simulated in the same manner for three other climatic regions: Fort Worth, Texas; Madison, Wisconsin; and Phoenix, Arizona. These areas were selected as being representative of the general climatic regions found in the United States and also were available in the SOLMET weather tape library.

The series systems employ a boosting and direct solar delivery control strategy, while the parallel systems use boosting and nondirect solar control. The selection of these control schemes is detailed in Section 3.3 of the first six months technical report. Inexpensive collectors are employed for both systems. Collector efficiency is given as $M = 0.75 - 8.5X$

where

$$X = (T_{\text{fluid}} - T_{\text{amb}})/I$$

$$T_{\text{fluid}} = \text{average fluid collection temperature, } ^\circ\text{C}$$

$$T_{\text{amb}} = \text{ambient temperature, } ^\circ\text{C}$$

$$I = \text{insolation on collector surface, } \omega/\text{m}^2$$

² Storage is via sensible liquid which is sized according to 0.08 m³ tank/m² collector area (2 gal/ft²). In sunnier climates, such as Arizona, a larger storage-size-to-collector ratio may be advantageous, as illustrated in Table 3-1. Heat pumps were sized to meet cooling loads in Arizona and Texas and to meet heating loads in New York and Madison. The simple heating house model mentioned in Section 2 was retained to keep all of the climatic comparisons on a common basis. The heating load for this house is given as

$$Q_{\text{load}} = 237 (15.6 - T_{\text{amb}})$$

where

$$Q_{\text{load}} = \text{house load, W}$$

$$T_{\text{amb}} = \text{ambient temperature, } ^\circ\text{C}$$

Table 3-1
HEATING PERFORMANCE
OF SOLAR-ASSISTED HEAT PUMP SYSTEMS

Climatic Region:		New York, New York	
Heating Load:		4.94×10^{10} J/Yr	
Standard Heat Pump SPF ⁽¹⁾ :		2.1	
Collector Area (m ²)	Parallel SPF (1.85-ton heat pump)	Series SPF (2.11-ton heat pump)	
25	2.78	--	
50	3.35	3.05	
75	3.85	4.12	
100	4.26	4.91	

Climatic Region:		Madison, Wisconsin	
Heating Load:		7.16×10^{10} J/Yr	
Standard Heat Pump SPF:		2.03	
Collector Area (m ²)	Parallel SPF (2.0-ton heat pump)	Series SPF (2.5-ton heat pump)	
25	2.21	--	
50	2.51	2.24	
75	2.84	2.88	
100	3.14	3.30	

Climatic Region:		Fort Worth, Texas	
Heating Load:		2.52×10^{10} J/Yr	
Standard Heat Pump SPF:		2.36	
Collector Area (m ²)	Parallel SPF (2.5-ton heat pump)	Series SPF (2.5-ton heat pump)	
25	3.45	--	
50	4.67	3.33	
75	5.47	4.18	
100	6.08	4.54	

Climatic Region:		Phoenix, Arizona	
Heating Load:		1.30×10^{10}	
Standard Heat Pump SPF:		2.72	
Collector Area (m ²)	Parallel SPF (3.0-ton heat pump)	Series SPF (3.0-ton heat pump)	
5	3.91	--	
10	5.92	5.87	
25	10.60	11.50	
50	10.50	--	
10 ⁽²⁾	--	6.33	

1) Seasonal Performance Factor

2) Reflects storage size of 0.16 m³ tank/m² collector

This load is representative of a well-insulated house with a living area of 167 m², as discussed in Appendix C of the first six months technical report. The results of the TRNSYS simulations of the parallel and series systems are summarized in Table 3-1. Also included for reference are standard single-speed air-to-air heat pump SPF's. The performance values shown in Table 3-1 form the basis for the economic comparisons of the two systems. They are presented graphically in Figures 3-1 through 3-4.

One of the basic motivations for developing SAHP systems is to take advantage of low-cost collectors by means of low fluid collection temperature. Evacuated tube collectors represent a more expensive collector design that yields high collection efficiency and relative insensitivity to collection fluid temperature. However, if evacuated tube collectors could be made inexpensively, their high efficiency would benefit SAHP system performance. In this spirit, simulations of series, parallel, and straight solar heating systems in New York City were implemented using a General Electric TC-100 collector model. This collector is a liquid-evacuated glass tube design with 10 tubes per panel. Efficiency of the TC-100 depends on both the magnitude and the nature of the incident radiation, i.e., whether it is direct or diffuse. During noontime, when the direct component of incident radiation is large, the efficiency of the TC-100 surpasses the flat-plate collector. In early morning and late afternoon, when incoming radiation is mostly diffuse, the efficiency of the TC-100 is low. Therefore, an hourly simulation program, such as TRNSYS, is necessary to accurately assess the TC-100's potential. The results are presented in Table 3-2.

Table 3-2

EVACUATED TUBE COLLECTOR PERFORMANCE
(TC-100 Collector -- New York, New York)

System	Collector Area (m ²)	SPF
Parallel	10	2.88
	25	3.97
	37.5	4.77
	50	5.52
Series	37.5	3.56
	50	4.35
	75	5.81
	100	6.73
Straight Solar	50	3.37
	100	6.67

Figure 3-1. Seasonal heating performance as a function of collector area -- New York City weather

Figure 3-2. Seasonal heating performance as a function of collector area -- Madison, Wisconsin weather

Figure 3-3.

Seasonal heating performance
as a function of collector area
-- Fort Worth, Texas weather

Figure 3-4.

Seasonal heating performance
as a function of collector area
-- Phoenix, Arizona weather

RESIDENTIAL COOLING PERFORMANCE

PARALLEL SYSTEM

The parallel system in the cooling mode is essentially a two-speed, air-to-air heat pump between the house and the outdoor air. The solar portion of the system plays no part in the cooling operation; hence, the cooling COP is independent of the collector area. To simulate the parallel system, the TRNSYS program is used to couple two-speed, air-to-air heat pumps of different sizes; the house model discussed in Section 2 and the weather tapes for different locations. The operating speed of the heat pump compressor is based on the sensible load and the humidity of the house. With the compressor operation at half speed, the cooling capacity and dehumidification ability are reduced as a result of increased evaporator temperature. However, half-speed cooling COP's are much higher, as evidenced in Figures 2-1 and 2-4. The control strategy for the determination of compressor speed is given in Figure 3-5.

Figure 3-5.

Cooling control strategy
for parallel system

The control strategy allows the heat pump to operate at half speed (high COP) as long as the house humidity is kept below 60 percent (with a 10 percent controller dead band) and the house sensible load is able to be satisfied. Humidity in the house is changed every time step during the computer simulation in response to the heat pump latent capacity, the outdoor humidity infiltration, and the internal latent load generation. All of these are computed by the house model. In New York weather, two-speed operation allows a 10 percent improvement in SPF (Seasonal Performance Factor) over a conventional single-speed heat pump. In Arizona, where the climate is drier and less dehumidification is needed, two-speed operation yields an 11 to 14 percent improvement in SPF, depending on heat pump size. Table 3-3 lists the results of the parallel SAHP system simulation. Included are the minimum heat pump size to satisfy the cooling loads, the sensible and latent house loads, and the cooling SPF.

Table 3-3

PARALLEL COOLING PERFORMANCE

Location	Sensible Load Latent Load (J/yr)	Heat Pump Size (full speed) (tons)	SPF
New York, New York	5.22×10^9 1.82×10^9	2.5	2.58
Fort Worth, Texas	2.54×10^{10} 1.04×10^{10}	2.5	2.52
Phoenix, Arizona	4.25×10^{10} 4.42×10^9	3.0	2.28

SERIES SYSTEM

Cooling is achieved with the series SAHP system through utilization of an air-to-water heat exchanger located in the air duct of an air-type solar collector. A schematic diagram of this system modification is shown in Figure 3-6. Dampers, located in the collector supply and return ducts, permit the circulation of outside air through the exchanger, thus allowing energy to be discharged from storage. In the solar collecting operational mode, these dampers are closed and the energy collected is transferred from the air to the storage water through the heat exchanger. Hence, in the collection mode there exists another drop in temperature (across the exchanger) which slightly degrades the heating system performance. Only the cooling performance of this system was modeled. Unlike the parallel system, the size of the collector for cooling by the series system has an effect on performance in that larger collectors require both larger heat exchangers and fans.

Figure 3-6. Schematic diagram showing series SAHP system in cooling mode

The collector heat exchanger, located in the supply duct, is assumed to have an effectiveness of 0.6, with an air-side pressure drop of 37 Pa (0.15 in.) at an air velocity of 2.54 m/s (500 fpm). The performance characteristics of this exchanger are similar to those of the exchanger in the indoor air handling system. The water-side pressure drop is 69 kPa (10 psi) at a flow rate of $6.3 \times 10^{-4} \text{ m}^3/\text{s}$ (10 gpm). The air-flow rate varies with collector size, while the water-flow rate is kept constant, since the air side heat-transfer resistance is the dominant one.

The collector water pump is sized to deliver $6.3 \times 10^{-4} \text{ m}^3/\text{s}$ (10 gpm) at a 138 kPa (20 psi) pressure drop (69 kPa each for both the heat exchanger and the piping). Using an overall pump efficiency of 35 percent (typical of pumps of this size), the required pump power is 250 W.

The flow rate required by the collector varies with the collector area. A flow rate of $0.01 \text{ m}^3/\text{s m}^2$ (2 cfm/ft²) is used, as suggested by the manufacturer of the Solaron air collector. For two collector panels in series, the pressure drop is 57 Pa (0.23 in. H₂O). For a $0.75 \text{ m}^3/\text{s}$ air-flow rate, sufficient for 75 m² of collector, the air pressure drops in the other parts of the duct system are:

Heat exchanger	37 Pa
Manifolding	15 Pa
Elbows (4)	25 Pa
External duct	<u>12 Pa</u>
	89 Pa

Plus

Collector panels	<u>57 Pa</u>
Total	146 Pa @ 0.45 m ³ /s

Assuming that systems having different collector areas (and correspondingly different air-flow rates) are designed with the same pressure drop, and using a fan impeller efficiency of 50 percent and a motor efficiency of 60 percent, the required collector fan power for various collector sizes is shown in Table 3-4.

Table 3-4

FAN SPECIFICATIONS FOR SERIES SYSTEM

Collector Area (m ²)	ΔP (Pa)	Air-flow Rate (m ³ /s)	Fan Power (W)
25	148	0.11	126
50	148	0.50	253
75	148	0.76	380
100	148	1.0	506

The storage tank is, as in the heating simulations, sized at 0.08 m³/m² (~2 gal/ft²) of collector area.

The control strategy for the cooling simulations is: the heat pump operates whenever a sensible load is detected (see parallel system house cooling model). The collector pump and the fan are switched on when the tank-ambient air temperature difference exceeds a preset value, T_{max} . They are switched off again when the temperature difference drops below a preset value, T_{min} . T_{min} is set at 4 °C, since at this point the total heat transferred in the exchanger is equal to the sum of the collector fan and pump power. Operation at temperature differences lower than 4 °C will not discharge the storage.

T_{max} was varied from 10 to 30 °C to obtain a minimum total power input to the system. It can be reasoned that, as T_{max} is raised, the fan and pump power decrease while the tank temperature increases, thereby lowering the COP and raising the heat pump power. It was found for New York

City that the optimum $T_{\max}$ was 20 to 30 °C, at which point the total seasonal fan and pump power comprised 15 to 35 percent of the total electrical power input to the system.

The heat pump used in this system is an air-to-water type operating in the cooling mode; the performance characteristics are presented in Section 2.

The results of the series SAHP system cooling simulation are listed in Table 3-5. Geographical comparisons of the parallel and series systems for New York City and Fort Worth, Texas weather are shown in Figures 3-7 and 3-8.

Table 3-5

SERIES SYSTEM COOLING PERFORMANCE

Climatic Region	Collector Size (m ²)	Sensible Load (J/yr)	Latent Load (J/yr)	Heat Pump Size (tons @ full speed)	Tank Size (m ³)	SPF
New York City	50	5.2×10^9	1.3×10^9	3	4	1.82
	75	5.2×10^9	1.3×10^9	3	6	1.94
	100	5.2×10^9	1.3×10^9	3	8	2.12
Fort Worth, Texas	60	2.7×10^{10}	7.9×10^9	3.5	4.8	1.82
	75	2.7×10^{10}	7.9×10^9	3.5	6	1.84
	100	2.7×10^{10}	7.9×10^9	3.5	8	1.94

COMMERCIAL AND MULTIFAMILY STRUCTURES

Commercial buildings are, in general, characterized by a large amount of internal heat generation during the daytime working hours and small load requirements during nonworking hours. In addition, office buildings are often zoned such that central core regions of the building may require cooling, while zones with exposed outside walls are heated. This leads to a building load profile which is markedly different from that of a simple family residence.

For purposes of this study, a small commercial building load tape computed by the National Bureau of Standards with the NBSLD program is used. Building loads are analyzed for New York, New York; Madison, Wisconsin; and Fort Worth, Texas. The building specifications are shown in Table 3-6.

Figure 3-7. Seasonal cooling performance as a function of collector area -- New York City weather

Figure 3-8. Seasonal cooling performance as a function of collector area -- Fort Worth, Texas weather

Table 3-6

COMMERCIAL BUILDING SPECIFICATIONS

Style: Low-rise office building

Orientation: North exposure faces due north

Size:	Length	30.5 m
	Width	30.5 m
	Height	7.3 m

Air Changes:	Minimum	0.25/h
	Maximum	1/h

Roof:	Area	930 m ²
	Thermal conductance	$U = 0.89 \text{ W/m}^2 \cdot ^\circ\text{C}$

Wall (All walls identical):		
Window area:	100 m ²	$U = 3.29 \text{ W/m}^2 \cdot ^\circ\text{C}$
Wall area:	234 m ²	$U = 0.857 \text{ W/m}^2 \cdot ^\circ\text{C}$

Floor on Grade:		
Area:	930 m ²	
	$U = 0.936 \text{ W/m}^2 \cdot ^\circ\text{C}$	

Zoning:

4 Identical perimeter zones
1 Core zone area:

455 m²

An internal heat generation schedule is used which reflects weekends, holidays, lights, equipment, and people. Hourly heating and cooling loads are provided for each building zone. The building load characteristics for different climates are summarized in Tables 3-7, 3-8, and 3-9. These characteristics result from a computer analysis of the NBSLD load tapes.

Based on Tables 3-7, 3-8, and 3-9, the following observations may be made:

1. Cooling loads play a major role in the building heating, ventilating and air conditioning requirements. In New York City, cooling is 61 percent of the total yearly load. Even in Madison, Wisconsin cooling is still 47 percent of the total load. This indicates that the cooling COP will be a dominant factor in deciding the optimum system. The SAHP system is primarily a heating system, and it becomes increasingly difficult to justify a large capital expenditure for solar collectors when the primary function of a system is cooling. Parallel systems have shown to be superior to series systems in cooling, as shown in the subsection "Residential Cooling Performance."

Table 3-7

BUILDING LOAD CHARACTERISTICS -- NEW YORK, NEW YORK

Month	Heating Load (kJ x 10 ⁻⁶)	Cooling Load (kJ x 10 ⁻⁶)
1	88.8	2.54
2	117.0	2.37
3	36.9	5.96
4	6.9	32.43
5	1.7	47.4
6	0.0	88.6
7	0.0	151.9
8	0.0	129.4
9	0.03	88.9
10	4.95	39.2
11	19.0	16.7
12	109.0	1.8

Zone	Yearly Heating Load (kJ x 10 ⁻⁶)	Yearly Cooling Load (kJ x 10 ⁻⁶)
South	52.85	99.76
West	69.03	90.51
North	77.47	69.04
East	68.99	92.38
Core	<u>116.0</u>	<u>255.4</u>
Total	384.3	607.1

Table 3-8

BUILDING LOAD CHARACTERISTICS -- MADISON, WISCONSIN

Month	Heating Load (kJ x 10 ⁻⁶)	Cooling Load (kJ x 10 ⁻⁶)
1	206.3	1.74
2	102.9	4.02
3	66.95	4.14
4	38.19	15.37
5	3.47	48.96
6	0.0	126.7
7	0.0	130.0
8	0.0	147.5
9	1.21	80.81
10	10.54	37.96
11	60.79	5.50
12	192.7	0.40
Zone	Yearly Heating Load (kJ x 10 ⁻⁶)	Yearly Cooling Load (kJ x 10 ⁻⁶)
South	98.42	104.6
West	121.2	97.31
North	131.6	65.55
East	117.7	98.91
Core	<u>214.1</u>	<u>236.8</u>
Total	683.0	603.1

Table 3-9

BUILDING LOAD CHARACTERISTICS -- FORT WORTH, TEXAS

Month	Heating Load (kJ x 10 ⁻⁶)	Cooling Load (kJ x 10 ⁻⁶)
1	33.08	23.87
2	31.86	17.19
3	23.26	42.90
4	0.13	98.55
5	0.0	133.5
6	0.0	222.9
7	0.0	230.8
8	0.0	248.5
9	0.0	179.8
10	0.36	109.2
11	3.11	56.98
12	25.47	11.59
Zone	Yearly Heating Load (kJ x 10 ⁻⁶)	Yearly Cooling Load (kJ x 10 ⁻⁶)
South	16.85	227.1
West	21.81	230.0
North	25.53	171.3
East	20.70	226.8
Core	<u>32.39</u>	<u>520.5</u>
Total	117.3	1376.0

2. Examination of a more detailed load analysis indicates that most of the heating load occurs during daytime working hours. This tends to improve the parallel system heating SPF, because the outdoor ambient temperature is, in general, higher during the daylight hours.
3. One possible advantage of a series system is its ability to use rejected heat from cooling one zone of the building to warm a different zone which requires heating. This simultaneous heating/cooling function of a series system could be accomplished by having individual heat pumps for each zone operating from one common storage tank, such as shown in Figure 3-9.

Figure 3-9. Series system with individual heat pump for each zone

The relative importance of storing rejected heat from cooling depends on the amount of overlap the heating and cooling loads have. Based on an optimistic view of the building load profile in New York City, 10 to 12 percent of the heating load (4 to 5 percent of the total load) could be provided by stored heat from cooling. In Madison, Wisconsin, 9 to 11 percent of the heating load (5 to 6 percent of the total load) and in Fort Worth, Texas, 5 to 7 percent of the heating load (0.4 to 0.6 percent of the total load) could be supplied by heat from cooling the building, especially from cooling the core zone.

The minor contribution that this mode of operation could make in satisfying the total building load indicates that the advantage of a series system which could use rejected building heat would be small.

The preceding observations suggest that, from a performance viewpoint, SAHP systems are better suited to residential structures than to commercial buildings. The economics of commercial buildings, discussed in Section 4, "Economic Analysis," are also unfavorable. Therefore, the remaining analysis of the SAHP systems is focused on residential structures. Multifamily structures are assumed to have load profiles similar to single family houses and consequently are assigned equivalent SPF's. The economics of multifamily structures is also discussed in Section 4.

PERFORMANCE CONCLUSIONS

In climates with a substantial heating load (e.g., New York City and Madison, Wisconsin), one observes a crossover point in the performance of parallel and series systems (Figures 3-1 and 3-2). With sufficient collector area, series systems are able to outperform parallel systems. However, with a small collector area, the series heat pumps are "starved" for source heat which results in low heat pump COP and increased supplementary heating. This leads to the choice between a parallel system with lower performance at a lower price (small collector area) vs. a series system which has a higher SPF at a higher price (large collector area). Economics must inevitably dictate which is the most opportune system.

In milder climates (Fort Worth, Texas, and Phoenix, Arizona), a different situation exists (Figures 3-3 and 3-4). Here the heating load is small, and the equipment is sized for cooling, such that it is oversized relative to the heating load. In addition, the ambient temperatures are milder, dictating higher air-to-air heat pump COP's. In Fort Worth, this allows the parallel system to surpass the series system for all collector areas. In Phoenix, where there is more sunshine, the effect of the solar portion of the SAHP system is more apparent. Both the series and parallel systems deliver a large part of the heating load through the direct heat exchange of stored solar energy from the tank. This leads to a similar performance for both systems and a rapid increase in the SPF as a function of the collector area. Once the collector area exceeds about 20 m², virtually the entire heating load is supplied through the direct use of solar energy, and performance levels off. At this point, the series system enjoys a slight performance advantage due to its direct-heating control strategy.

The performance of parallel systems is independent of the collector-storage subsystem in the cooling mode. This allows the tank to remain at a high temperature (for preheating domestic hot water) without compromising the heat pump cooling COP. In the series system, cooling requires heat rejection from the tank to the ambient temperature. This places additional

ΔT 's in the heat rejection path and ultimately lowers the cooling COP's. It also requires that the tank temperature be kept low, otherwise the heat pump COP and reliability will suffer.

Based upon these results, the parallel systems are superior to series systems when operated in the cooling mode (Figures 3-7 through 3-8).

Summarizing the detailed findings of the subsection "Commercial and Multifamily Structures," one may conclude that the commercial building considered in this study does not offer an attractive application of the presently configured parallel and series systems, as compared to residential structures. This is primarily due to the increased emphasis on cooling performance and the inability of present systems to store rejected cooling heat long enough to have an appreciable impact on the building's yearly heating load. This is not to say, however, that all other commercial applications would exhibit the same load profile. Larger structures, or perhaps buildings which are not offices, may have a greater degree of heating/cooling overlap. However, this might lead to advantages for a heat-pump-only system with long-term storage as opposed to a SAHP system. Because of the variability of commercial buildings and their general emphasis on cooling, the remaining effort was focused on the residential SAHP systems. Multifamily residences were assumed to follow the same load profiles as single family residences, such that the same performance conclusions would apply.

LOAD DEMAND ANALYSIS

MARGINAL PRICING

The power demand on an electric utility varies with the time of day and with the season. The generation mix used to meet the demand has varying capital and fuel costs which are not generally reflected in the electric rates. Marginal pricing theory states that the price of the commodity should reflect the cost of producing an additional unit of the commodity (marginal cost). In particular, electricity delivered at different times should be considered as a variety of commodities with different marginal costs, and the pricing should reflect these changing costs of generation.

It is not true that marginal pricing schemes seek to improve the load factor and capacity factor of the utility in which they are instituted. Improving the load factor (i.e., raising it) only is relevant if electricity is not marginally priced; in which case the implicit assumption is that the cost of generating electricity can be made more constant so that flat electric rates are closer to the marginal rates. If consumer elasticity is zero, the load factor cannot be modified, and equitable pricing must be marginal to reflect the cost of generation. In much of the discussion that follows, time-of-day pricing and on-off peak pricing will be referred to not because

improved load factor is sought but because practical application of marginal costing will frequently be reflected in this type of rate.

The application of marginal pricing to utility rates is discussed at length in Reference 3. Some of the difficulties in applying such pricing are enumerated below.

DIFFICULTIES IN APPLYING MARGINAL PRICING

Strict application of marginal pricing would require a continuous monitoring of electricity use tied to the instantaneous fuel and capital costs. Reasonable application must include discrete changes in rate structure to make metering tractable and billing understandable by the consumer. The exact scheme to satisfy these requirements may take various forms:

- Seasonally adjusted flat rates
- Time-of-day rates
- Graduated-use rates
- Demand charges
- Seasonally adjusted time-of-day rates
- Temperature-coupled rates.

Of these rate types, only seasonal changing flat rates and graduated-use rates are easily implemented with current residential metering setups; however, these rates only grossly address marginal cost considerations. For instance, seasonal pricing might mitigate the higher cost of generation occurring in only a few summer days, and graduated rates (currently in existence in many utilities) account for fixed capital costs incurred by adding another user who may not be a substantial revenue source. Time-of-day rates and demand charge implementation will require new, more costly electric meters, representing a capital investment which some feel would increase the cost of all electricity more than the current marginal cost of peak generation.

A marginal pricing scheme based on load profiles which occur with flat electric rates will quickly become obsolete, since there will be some elasticity which will result in shifting of consumption. The value of long- and short-term elasticity for the residential sector is subject to considerable uncertainty, making difficult the task of assigning rates with any of the metering schemes mentioned previously.

Ref. 3. National Economic Research Associates, Inc., A Frame Work for Marginal Cost-Based Time-Differentiated Pricing in the United States: Topic 1.3, prepared for the Electric Utility Rate Design Study, February 21, 1977. Available through Electric Power Research Institute, Inc., Palo Alto, California.

In addition to the application problems associated with marginal pricing, regulatory commissions must consider subjective problems associated with apparent inequities that might occur when large commercial and industrial users have low rates because of their favorable load characteristics (i.e., deferrable or interruptable demand). The regulatory commissions will also consider consumer acceptance, environmental benefits from deferred generating plant construction, and state or federal mandates.

The various opinions of different regulatory commissions, the various load profiles and generation mix of the regulated utilities, plus the uncertainty of some factors governing the application of marginal pricing makes impossible specific statements about the form of future residential electric rates. It is possible to make generalizations about how a particular rate form will affect the cost of operating a particular residential heating, ventilating, and air conditioning system, and this is the approach followed below. The discussion concentrates on time-of-day pricing and seasonally adjusted flat rates. Power demand information is given as an indicator of worst-day utility loading and of consumer charges under a residential demand rate schedule.

CHARACTERISTICS OF SOLAR-ASSISTED HEAT PUMP/ELECTRIC DEMAND IN HEATING

In the evaluation of SAHP systems, two quantities have been used to define the characteristics of electric demand. These quantities are monthly averaged daily electric energy use and hourly peak power demand by month. The first of these quantities is an indicator of consumer billing with time-of-day rates and no demand charge. The second quantity can be used to evaluate demand charges and utility worst-day load.

All heating comparisons are based on New York City weather. The standard equipment sizings used are those found to be optimum for flat residential electric rates. Any other optimization would have to be based on a specific rate structure and would not be generally applicable.

Air-to-Air Heat Pump

Average daytime temperatures are higher than nighttime temperatures, so that thermal loads on the residence are reduced. In addition, solar gains during daytime periods further reduce the house load, and warmer ambient temperatures improve performance of the air-to-air heat pump. The combined effect on monthly averaged energy consumptions gives a curve which dips significantly during daylight hours. Such a curve is presented in Figure 3-10 for a 1.91-ton heat pump for the months of December, January, and February.

Figure 3-10. Energy use profiles in heating for New York City -- series, parallel, and ordinary heat pump

Figure 3-11. Peak demand profiles for New York City -- series, parallel, and ordinary heat pump

On the average, daytime loads are lower than nighttime loads, and heat pump performance is better. But, in a given month high loads and low temperatures may occur at any hour of the day, thus peak power profiles for the straight heat pump will generally be flatter than the energy use profiles. This means that an on-peak demand charge would seriously increase billing even though average use is favorable to utility loading. Curves of peak power demand (kWh/h) for the months of December, January, and February are presented in Figure 3-11. No storage is available with this system, so no possibility exists for modifying the load profiles.

Parallel Solar-assisted Heat Pump System

The limiting worst case performance of the parallel system is that of an air-to-air system. Consequently, it would be expected that average consumption would approach that of an air-to-air system in early morning hours when storage is depleted. Further, it is expected that daytime average use will drop significantly below that of the heat pump because of direct solar use. The behavior, shown in Figure 3-10 for a 1.85-ton heat pump with a 25-m² collector, is as expected. Peak power follows the trend of the air-to-air heat pump with a few exceptions during daylight hours due to direct solar use (Figure 3-11).

The two different modes of operation in this system allow some modification of the demand profile. However, as the results shown in Figures 3-10 and 3-11 indicate, the demand profiles with unsophisticated control are already favorable towards the daytime peak period for many utilities. If further profile modification is desired, the most likely form of sophisticated control would consist of changing the criteria for the operational mode with the time of day. This might allow, for instance, prolonged heat pump operation during off-peak hours in order to have greater solar contribution during peak hours. Sophisticated control would likely be coupled with changes in solar component sizing to effect demand and energy use profile changes. For instance, increased collector and tank sizes or an increased tank/collector size ratio would reduce the average and peak use. No sophisticated control was simulated, since it would have to be tailored for a specific rate structure.

Series Solar-assisted Heat Pump Systems

By virtue of the large collector sizes required by the series system, its daytime performance averages slightly better than the parallel system, and the stored solar energy keeps the average nighttime electricity use below the parallel system. The energy use profiles plotted for a 65-m² collector and a 2.18-ton heat pump are shown in Figure 3-10. The peak demand is quite different from the parallel SAHP or the air-to-air heat pump, alone, because it is determined by the resistance heat for the entire load when the

storage is depleted. Thus, the peak demand is significantly above the heat pump, alone, or the parallel SAHP system.

As in the parallel case, some alternative control schemes for load management are possible which vary the criteria for the operating mode. In particular, the range for heat pump operation could be expanded so that the stored energy is stretched into the night to reduce the peak power. Several simulations indicate that the overall energy efficiency suffers, but a negligible effect is seen on the peak power profile when the control temperature for a changeover to boosting is raised 5 °C. This indicates that the reduced system efficiency is not sufficient to stretch the storage into the cold nights which set the peak.

Increased storage sizes will eventually reduce the maximum peak, but as indicated in Figure 3-12 the size increase can be substantial in order to eliminate the monthly peak in the morning hours. Virtually all of the 20 percent improvement in seasonal efficiency resulting from the large component size (10 m³ storage/80 m² collector) increase is a result of the virtual elimination of the need for supplementing resistance heat. Furthermore, most of the displaced resistance heat is used at off-peak hours which would have low rates in a time-of-day pricing scheme, as shown in Figure 3-13.

SOLAR-ASSISTED HEAT PUMP LOAD MANAGEMENT IN HEATING

It has been shown that the average load profiles of all of the systems are all desirable in terms of utility loading, if the peak periods are approximately 9 am to 9 pm. High power demand does not appear to be a severe problem, except possibly with the series system. On this basis, it might seem that the economics of the SAHP systems would improve with marginal pricing. A complicating factor is that the overwhelming majority of U.S. utilities have their peaking problem in the summer, and this will be reflected in any sort of marginal rates. Typically, this would mean that with seasonally adjusted rates winter rates will be lower than the flat rate being replaced, and with seasonally adjusted time-of-day rates the on-peak rate might be lower than the previous flat rate level. This type of pricing effectively lowers the cost of electricity, making capital intensive heating systems even less economical.

Without altering the basic system configuration significantly, modifying the load profiles will require capital intensive additions in the form of increased collector and storage sizes. As noted in the case of the series systems, these types of modifications tend to reduce off-peak use faster than on-peak use, so the main body of capital investment goes towards decreasing the use of the least expensive energy. Increasing the collector and storage sizes will primarily affect off-peak usage in the parallel system as well. With such poor utilization of capital, it is doubtful that these expenditures could be justified.

Figure 3-12. Peak power profiles for various collector/tank sizes

Figure 3-13. Comparison of series performance with two collector/tank sizes

The load management techniques discussed thus far have only concerned themselves with methods which either do not significantly degrade overall energy efficiency or actually improve overall efficiency. These methods sought to meet the load during peak hours with the most efficient mode of operation but not at the expense of off-peak operation. If actual electric load deferral is desired, then resistance heat can be used with little additional capital cost but with a considerable energy penalty. Such modes of operation were not considered since electric rates low enough to warrant this inefficient operation imply that capital intensive heating systems of any kind could not be justified economically.

CHARACTERISTICS OF SOLAR-ASSISTED HEAT PUMP/ELECTRIC DEMAND IN COOLING

All of the cooling comparisons are based on Fort Worth, Texas, weather. In the cooling mode, the parallel system acts as a two-speed air-to-air heat pump.

Air-to-air Heat Pump (Two-speed)

Conventional air-to-air heat pumps (or equivalent air conditioners) are largely responsible for existing utility summer peaking problems because electric consumption increases with outdoor temperature. The average consumption is typified by the curves in Figure 3-14 and the peak consumption by the curves in Figure 3-15. Without storage nothing can be done to alleviate this problem, short of sacrificing comfort.

Series Heat Pump

The average energy use and peak demand profiles (Figures 3-14 and 3-15, respectively) are similar to those of the heat pump, alone, except they are slightly smoother due to the effects of thermal storage. The only option for reducing daytime peaks and energy use is to cool the storage vessel as low as possible at nighttime. A detailed study of this sort of process [Ref. 4] indicates that the best the series system could hope to do is a 25 percent peak reduction if an 11-m³ storage could be drawn down at night to 22 °C below the daytime maximum. This is extremely optimistic, since nighttime temperatures on hot days in Fort Worth, Texas, may not dip more than 10 °C below the daytime highs, and the change in the storage temperature can only approach this drop. Since other systems in the cited study offered peak reductions of 80 percent or better, no calculations were done to quantify peak reductions for warmer storage. Any significant effect on daytime peaks would require additional equipment for load deferral, such as discussed in the following subsection.

- Ref. 4. D.C. Golibersuch et al., Cool Storage Assessment Study, EPRI Report No. EM-468TPS 76-650, Electric Power Research Institute, Inc., Palo Alto, California, May 1977.

Figure 3-14. Energy use profiles in cooling for Fort Worth, Texas -- series and ordinary heat pumps

Figure 3-15. Peak power profiles in cooling -- Fort Worth, Texas

Alternate Hardware and Control Options

The principle difficulty with electric demand management in both the parallel and SAHP systems is that no reliable, efficient method exists for charging (or discharging in cooling) the storage during nighttime off-peak hours. This problem is remedied with the three-coil heat pump or with an additional air-to-water heat pump to charge or discharge the storage.

The additional initial cost of these devices must be offset with reduced electric bills. The economics will depend on the actual rate form, and it is not certain that there will be any improvement. Several examples illustrate possible problems.

- A summer rate which is coupled to the ambient temperature will not reduce the billing if nighttime off-peak temperatures do not drop below the temperature setpoint. This problem may be further aggravated by a net increase in total electricity used when the load is deferred.
- A rate structure which is already favorable to the SAHP systems in heating (i.e. time-of-day rates) can only result in marginally improved savings with the addition of capital intensive equipment for heating load deferral.
- A seasonally adjusted flat rate for summer peaking utilities will result in a lower winter rate than the annual flat rate it replaces. These lower winter rates will make capital intensive heating systems less attractive economically in climates with high heating loads and will eliminate any economic motivation for load deferral equipment.
- Severe differentials between on- and off-peak rates with a time-of-day structure may justify the additional equipment for SAHP system load deferral, but then it would be proper to include comparisons to inefficient hot-brick storage with resistance heat.

A decision on load deferral equipment obviously cannot be made without knowledge of the rate structure. Observations can be made about the load deferral capability of SAHP systems with an additional ambient air-to-storage heat pump. This configuration is called a cascade heat pump, since two stages are required to transfer heat from the ambient to the residence. In the series system, heat is transferred to the storage and then delivered by the inside heat pump to the residence. In the parallel system, heat is delivered to the storage and then transferred to the residence via the heat exchanger. Observations on the load deferral characteristics of such a SAHP system are discussed in the following subsection.

LOAD DEFERRAL CHARACTERISTICS OF CASCADE HEAT PUMP FOR SAHP SYSTEMS

Heating

Parallel System. The high storage temperatures in the parallel system prevent effective use of an additional air-to-water heat pump, thus in heating no significant additional load deferral is possible.

Series System. In the boosting or straight solar modes, the storage temperature is too high to accept charging from the outdoor heat pump. In the heat pump mode, the COP is high. The principal function of the outdoor heat pump should be to prevent the use of resistance heat by keeping the storage above the minimum value for the indoor heat pump. This control strategy is simulated for New York City weather by activating the outdoor heat pump when the tank temperature was within 0.5°C of the minimum tank temperature. The primary effect is to reduce the annual resistance heat use by a factor of 2.6 while not significantly affecting the performance of the series heat pump. The net effect, including the outdoor heat pump energy use, is a 5 percent improvement in seasonal heating efficiency.

The effect on the load profiles is pronounced only in the demand curve and in the early morning hours on the average energy use curves (Figures 3-16 and 3-17, respectively). In general, the benefit of the cascade heat pump is in the early morning hours which are not usually a problem for the utility.

Improving the load deferral beyond that offered under this simple scheme would require that the storage be heated during off-peak hours to temperatures suitable for direct heating. This is a very inefficient operation mode for the outdoor heat pump and would severely degrade its seasonal efficiency. Furthermore, this type of operation would eliminate the use of the series heat pump and eliminate any inherent efficiency advantage of the series system over the parallel system.

Cooling

This topic has been extensively studied in Reference 4, and the discussion in this subsection draws from this study. The key finding, which applies to both the series and parallel systems, is that the additional outdoor heat pump should be run in off-peak hours to cool the storage sufficiently so that it can be used for direct cooling. Partial cooling of the storage and use of the indoor heat pump with the favorable sink temperatures during on-peak periods is more energy inefficient and not as effective in deferring loads. Therefore, the parallel system with an additional heat pump will perform as well in cooling as the series system (equal storage size) with a cascade heat pump.

Figure 3-16. Energy use profiles for New York, New York -- series, cascade

Figure 3-17. Peak demand profiles for New York, New York -- series, cascade

In the case of a 12-hour peak period, peak power reductions of 80 to 90 percent were found to be possible with only a 10 percent increase in total energy use. It would be necessary to increase the size of the storage in both the series and parallel cases in order to achieve this performance. The parallel system would require a smaller storage increase than would the series system, because off-peak cooling requirements can be met with the air-to-air heat pump instead of through the storage. The hottest 24-hour period determines the size of the storage; typically 25 percent of the load occurs off peak so that the parallel storage could be 25 percent smaller than the series storage for the same performance and size of the outdoor heat pump.

Whether or not any load deferral equipment can be justified economically will depend on the specific rate structure. However, in no case will the series system gain a cost performance advantage over the parallel system.

LOAD DEMAND SUMMARY

It is highly unlikely that the load management capabilities of the SAHP systems, with or without cascade heat pumps, will affect the basic conclusions about the relative economics of the parallel and series SAHP systems because:

- Both systems are equally poor in cooling without an additional outdoor air-to-water heat pump.
- Both systems have equally good efficiency in cooling if an additional outdoor heat pump is included, but the parallel system will cost slightly less and will not have seasonal switchover problems with the storage.
- Parallel, series, and heat pumps, alone, all have average heating load profiles which are favorable to utility loading without any special control or equipment.
- It is difficult to further improve on peak performance of the SAHP systems because of their already high efficiency. Additional storage and collector sizes tend to improve off-peak performance the most.

There are some additional reasons to believe that the series system economics may be adversely affected by marginal pricing:

- Winter rates will average lower than ordinary flat rates for the majority of the country's utilities. The average lower cost of electricity will be detrimental to capital intensive systems.
- High differential in time-of-day rates may justify load deferral with the use of resistance heat in off-peak hours. Such a situation would exclude the use of series systems, since storage would be kept at temperatures suitable for direct heating.

GROUND COUPLING

Recently, a great deal of interest has been shown in the use of ground as a thermal storage medium for heat collected during warm periods and to supply heat when it is needed during cold periods. The availability of a medium is especially desirable in applications for solar and SAHP systems. The concept is that the collector, which is underutilized in a regular solar system during the warm season, can be used to collect the solar energy for storage in the ground. This stored energy can then be utilized in the cold months when the house heating demand is high. Such a coupling of the ground storage with the solar system will reduce the collector area needed in regular-type systems. Since solar collector size is the dominant factor in the cost of solar and SAHP systems [Ref. 1], it has been felt that this storage technique makes these systems more economically competitive to alternative systems.

A quantitative study of a ground-coupled SAHP system is not available from previous works, therefore, a limited study was carried out under this contract. While this study does not give a conclusive answer, it provides some interesting information that will be useful to those working in this field. This section documents development of the ground storage model and the results obtained from a TRNSYS simulation using this ground storage model for a ground-coupled SAHP system. A general discussion of the difficulties which would be encountered in a thorough study of ground-coupled SAHP systems is presented together with some recommendations and reservations.

ONE-DIMENSIONAL GROUND STORAGE MODEL

In a ground-coupled SAHP system, the temperature of the fluid flowing through the heat exchanger pipes buried in the ground (referred to as "ground exchanger" in the subsequent discussion) varies in accordance with the energy collected from the collector, the energy supplied to the house, the energy loss from the tank, etc. Cycling of the circulating pump for the ground exchanger also changes the heat-transfer condition in the ground storage. Therefore, an acceptable model for the ground storage has to accommodate transient effects. Several assumptions for simplification of the transient problem are made in the model used for this program. They are:

- The pipes are spaced so close (or a sufficiently high conductivity material is placed around the pipes) that they can be treated as a planar exchanger.
- The planar exchanger is so large that edge effects can be neglected, and the horizontal temperature distribution is uniform compared to vertical distribution.
- The ground thermal properties (e.g., conductivity, density, capacity, etc.) are spatially invariant.

These assumptions imply that the ground temperature distribution can be calculated in a one-dimensional manner.

A schematic sketch of the model is shown in Figure 3-18. Mathematically, the model can be formulated as follows.

$$\frac{\partial T}{\partial t} = K \frac{\partial^2 T}{\partial x^2} \quad (3-1)$$

where T is the ground temperature, and K is the thermal diffusivity of the soil.

Figure 3-18. Ground storage model

The boundary conditions of thermal conduction in the ground storage depend on the condition of the circulating pump. When the pump is on and water is flowing from the tank through the ground exchanger, the adjacent ground temperature will be about the same as the water temperature, i. e.

$$T(0, t) = T_w(t) \quad (3-2)$$

where $T_w(t)$ is the average instantaneous water temperature throughout the pipes. This is reasonable because the conductance on the water side is much larger than the conductance on the ground side. In fact, the Biot number, $B_i = hd/k$, is very large (≥ 6000); where h is the heat-transfer coefficient of the water, k is the conductivity of the ground, and d is the diameter of the pipe. At the top and bottom of the storage, the ground temperature depends on the heat loss condition, i. e.,

$$\frac{\partial T}{\partial x} (\pm D, t) = \epsilon \quad (3-3)$$

where D is the distance from the exchanger pipe to the top and bottom of the storage, and ϵ is defined as heat loss characteristic.

When the pump is off, the heat transfer across the pipes is terminated. Therefore, the thermal condition at the planar exchanger can be treated as perfectly insulated, i.e.,

$$\frac{\partial T}{\partial x}(0, t) = 0 \quad (3-4)$$

In the model, it was assumed that the initial temperature of the soil throughout the storage area is constant, i.e.,

$$T(x, 0) = T_s \quad (3-5)$$

The one-dimensional transient ground storage model described by Equations 3-1 through 3-5 has been solved using the method of eigenfunction expansion [Ref. 5]. The procedure is straightforward, but tedious, and is omitted here. The solution of the equations is given in the following:

Pump on

$$T(x, t) = Tw(t) + \epsilon x + \sum_n \phi_n \cdot e^{-\lambda_n K(t-t_{on})} \left\{ -\frac{2}{D} \left[\frac{Tw(t_{on})}{\sqrt{\lambda_n}} - \frac{b_o}{\sqrt{\lambda_n}} + \epsilon \left(\frac{(-1)^n}{\lambda_n} \right) \right] + \sum_m \frac{2b_m}{D} \cdot e^{-(m\pi)^2 K(t_{on}-t_{off})} \alpha_{nm} \right\} - \sum_n \frac{2}{\sqrt{\lambda_n} D} \beta_n \cdot \phi_n$$

where

$$\beta_n = \sum_m \cdot e^{-\lambda_n K(t-t_m)} \Delta Tw_m$$

$$\alpha_{nm} = \frac{2D * (2n+1)}{[(2n+1)^2 - 4m^2]} \pi$$

Ref. 5. P.W. Berg and J.L. McGregor, Elementary Partial Differential Equations, Holden-Day Publication, 1966, Ch. 4.

$$\begin{aligned}
 b_o &= Tw(t_{off}) + \frac{\epsilon D}{2} + \sum_m \frac{1}{D\sqrt{\lambda_m}} e^{-\lambda_m K(t_{off} - t_{on})} \\
 &\quad \left\{ -\frac{2}{D} \left[\frac{Tw(t_{on})}{\sqrt{\lambda_m}} - \frac{b_o}{\sqrt{\lambda_m}} + \epsilon \left(\frac{(-1)^m}{\lambda_m} \right) \right] \right. \\
 &\quad \left. + \sum_n \cdot \frac{2b_n}{D} \cdot e^{-(n\pi)^2 K(t_{on} - t_{off})} \alpha_{mn} \right\} - \sum_m \frac{2}{\sqrt{\lambda_m} D} \beta_m \\
 b_n &= \frac{2\epsilon}{D} \cdot \frac{1}{\left(\frac{n\pi}{D} \right)^2} \cdot ((-1)^n - 1) + \sum_m \cdot \frac{2\alpha_{mn}}{D} \cdot e^{-\lambda_m K(t_{off} - t_{on})} \\
 &\quad \left\{ -\frac{2}{D} \left[\frac{Tw(t_{on})}{\sqrt{\lambda_m}} - \frac{b_o}{\sqrt{\lambda_m}} + \epsilon \left(\frac{(-1)^m}{\lambda_m} \right) \right] \right. \\
 &\quad \left. + \sum_n \cdot \frac{2b_n}{D} \cdot e^{-(n\pi)^2 K(t_{on} - t_{off})} \alpha_{mn} \right\} - \sum_m \frac{2}{\sqrt{\lambda_m} D} \cdot \beta_m \cdot \alpha_{mn}
 \end{aligned}$$

where

- λ_n - eigenvalue
- ΔTw_m - water T change at time t_m
- t_{off} - instant of time when pump is turned off
- t_{on} - instant of time when pump is turned on
- ϕ_n - eigenfunctions

TRNSYS SIMULATION

Coupling the ground storage model with the TRNSYS program, a simulation of a ground-coupled SAHP system is made for a $UA = 237 \text{ W/}^\circ\text{C}$ house in the New York City area. A schematic configuration of the ground-coupled SAHP system is shown in Figure 3-19. The control strategy for the ground-coupled SAHP systems is the same as that in General Electric's previous

study for SAHP systems without ground coupling [Ref. 6]. In this simulation, the control for the circulating pump of the ground exchanger is based on the difference between the tank temperature, T_{tank} , and the ground temperature, T_g , 3 cm away from the exchanger pipes; i.e., the pump is on for charging the ground with heat when

$$T_{\text{tank}} - T_g > \Delta T \text{ and } T_{\text{tank}} > 280^\circ\text{K}$$

and for discharging the ground when

$$T_g - T_{\text{tank}} > \Delta T \text{ and } T_g > 275^\circ\text{K}$$

Otherwise, the pump is off.

In the case considered, ΔT is chosen to be 2°C with a 1°C dead band.

The present simulation is made for a house with a 25 m^2 collector area and a 85 m^2 (area) by 4 m (depth) ground storage. The pipes are laid at the mid-height of the storage. The ground storage is assumed to be filled with dry sand and to be perfectly insulated at the boundaries.

The accumulated energy results of calculations for June through November are listed in Table 3-10. The system collects about one third of the total solar energy available. Approximately three-quarters of this collected energy is stored in the ground by the end of November. Since there is assumed to be no heat loss at the storage boundaries, the total amount of energy in the ground storage is an optimistic value.

The water tank temperature variation over the six-month simulation period is shown in Figure 3-20. The tank temperature reaches a peak at the end of September and starts to drop sharply because of the house heating demand in October and November. In order to show the ground storage behavior, the history of the temperature distribution in the ground is given in Figure 3-21. From the slope of the curves, it can be seen that the energy charging rate is higher during the earlier months, as one might expect. Figure 3-21 also shows that in November the tank is starting to take energy from the ground to supply the house heating demand.

The ground storage has been modeled with a number of simplifications. Nevertheless, coupling this one-dimensional transient model into the TRNSYS simulation raises some computational difficulties. Therefore, one may wish to use a simple energy balance method to model the ground exchanger. In other words, one may use a bulk temperature for the ground

Ref. 6. Heinz Jaster, Sec. 3.

Figure 3-19. Schematic diagram of series SAHP system with ground storage

Table 3-10

GROUND-COUPLED SOLAR-ASSISTED
HEAT PUMP PERFORMANCE

($J \times 10^{-9}$ in New York, New York Climate
June 1 to November 30)

Q_{sun}	Q_{coll}	Q_{tank}	Q_{elec}	Q_{pump}	$Q_{\text{h.p.}}$	Q_{grd}	Q_{load}
60.3	20.7	6.0	0.5	0.5	1.4	15.7	6.5
Q_{sun} : Total solar insolation on the inclined 25-m ² flat collector							
Q_{coll} : Total solar energy collected							
Q_{tank} : Total heating supplied from the tank							
Q_{elec} : Total electricity needed except for the circulating pump for the ground exchanger							
Q_{pump} : Total electricity needed for the circulating pump							
$Q_{\text{h.p.}}$: Total heat delivered through the heat pump							
Q_{grd} : Total energy saved in the ground storage							
Q_{load} : Total house heating demand

Figure 3-20. Ground-coupled tank temperature as a function of time

Figure 3-21. Ground temperature as a function of distance from storage source pipe

and characterize the heat transfer between the buried pipes and the ground by a constant effectiveness (or thermal conductance at the pipes). In order to evaluate this model, the thermal conductance across the planar exchanger is calculated using the one-dimensional model. The calculated results are shown in Figure 3-22. The large deviation of this value with respect to time clearly indicates that a constant conductance model is inappropriate.

Figure 3-22. Variation of thermal conductance with time of day

At present, the TRNSYS simulation for a ground-coupled SAHP system is only successful during the warm months (June through November), because a very small time step is required for a stable and convergent computation in the cold months (December through February). This causes a serious problem with respect to computational time and cost. A small computation time step is needed because: 1) the numerical error caused by the discrete variation of the temperature of the water flowing through the pipes is severe, 2) the ground temperature profile around the planar exchanger is strongly nonlinear, and the calculation using an insufficiently small length scale for the heat flux across the exchanger is erroneous.

DIFFICULTIES

In reality, the ground storage contains nonuniform soil whose thermal properties vary greatly. Such nonuniformities are further compounded by the uneven distribution of the moisture in the soil. Since the ground temperature varies, there is a diffusion of moisture from the warmer to cooler regions in the soil, due to vapor pressure differences. This moisture migration changes the specific heat as well as the conductivity. For instance,

the surroundings of the heat-absorbing pipes will have a high thermal conductivity due to the increase in moisture content. However, incorporation of these inherent effects into a real system cannot be properly understood without conducting an experimental investigation. In fact, there are many other effects (such as, moisture freezing, soil types, local precipitation, and snow cover) which will also support the need for experimental study.

Our experience based on the preliminary modeling effort on the ground storage indicates that several features should be incorporated in a mathematical model to optimize the ground storage design. This model should be three-dimensional in order to determine the best configuration of the pipe layout. The transient model must be able to handle frequent changes in the boundary conditions. If it is planned to use the TRNSYS program in a simulation of the performance, the integration scheme in the present version of the TRNSYS program has to be modified. This is due to the convergence problems inherent to the scheme [Refs. 7 and 8]. Experience has proven that it is unfeasible to optimize a ground storage design through only a numerical investigation. Experimental work must be concurrently carried out.

RECOMMENDATIONS AND RESERVATIONS

An experimental investigation of the ground storage should focus on (but not be limited to) the following:

- Pipe-field configurations
- Effects of storage size variation
- Boundary insulations
- Soil types
- Effects of control strategies

A recent study [Ref. 9] on a ground-coupled heat pump system (without solar assistance) shows an SPF = 3.25 in contrast to an SPF = 2.1 for an air-to-air heat pump. Thus, on a cost basis, ground storage may be more effective for a system without solar assistance. Marketing of ground storage may be difficult because: 1) space requirements, 2) unavailability of a thorough design guide, and 3) possible damage to the vegetation above it.

Ref. 7. A. Orbach, R. Fischl, and P.R. Herczfeld, "An Improved Integration Method for Solar System Control Modeling Using the TRNSYS Program," Proceedings of the First Workshop on the Control of Solar Energy Systems for Heating and Cooling, Hyannis, Mass., 1978.

- Ref. 8. R. Bruno and R. Kersten, "Models and Methods for the Analysis and Optimization of Solar Energy Systems," Proceedings of the Energy Use Management Conference, Tuscon, Ariz., Vol. III, 1977, p. 643.
- Ref. 9. V.J. Ginschel, "Betriebsergebnisse einer Erdreich-Wasser - Wärmepumpe," Elektrowärme in Technischen Ausbau elektrowärme, International Edition A, Vol. A6, 1977, p. 35.

Section 4

ECONOMIC ANALYSIS

INTRODUCTION

STATUS OF ECONOMIC STUDIES AS OF NOVEMBER 1978

In the first six months technical report [Ref. 1], the basic procedure for comparing the costs of providing space heat with conventional and solar heating systems was described. This included a life-cycle costing methodology and a component cost data base to calculate the system capital cost inputs needed for the life-cycle cost calculations. The life-cycle costing methodology contained all of the essential elements which go into figuring the actual costs of owning and operating a heating system, including tax effects and preferences for the time distribution of payments. Factors for computing the present values of various types of payments and tax deductions were tabulated. With the exception of rental property, which was not addressed, this discussion is consistent with a recently published account of methods used by the Department of Energy's Solar Heating and Cooling Branch Working Group on System Simulation and Economic Analysis [Ref. 10] and other related studies [Refs. 11 through 14].

The development of a data base for computing the installed cost to the consumer of heating systems represents a major endeavor, which was only partly completed during the first six months reporting period. Essentially, an attempt was made to create a solar system estimating guide, which could

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- Ref. 10. A. M. Perino, A Methodology for Determining the Economic Feasibility of Residential or Commercial Solar Energy Systems, (Supplement giving updated economic parameters) Report No. SAND 78-0931, Sandia Corporation, Albuquerque, N.M., 1979.
- Ref. 11. R. T. Ruegg, Solar Heating and Cooling in Buildings: Methods of Economic Evaluation, Report No. NBS1R 74-712, National Bureau of Standards, Washington, D.C., 1975.
- Ref. 12. U.S. Congress Office of Technology Assessment, Application of Solar Technology to Today's Energy Needs, U.S. Government Printing Office, Washington, D.C., 1977.
- Ref. 13. A. E. McGarity, Solar Heating and Cooling: An Economic Assessment, U.S. Government Printing Office, Washington, D.C., 1977.
- Ref. 14. Joint Economic Committee, The Economics of Solar Home Heating, U.S. Government Printing Office, Washington, D.C., March 13, 1977.

be used very much like such guides as Mean's [Ref. 15], the Building Cost File [Ref. 15], or the Dodge Manual [Ref. 16] are used to obtain the costs of complete systems by considering the material, the installation labor, and the indirect costs associated with each subsystem and component. Although existing guides are useful in developing a solar system estimating guide, the treatments are not detailed enough to allow valid comparisons between differently configured systems. Also, much of the information in the existing guides applies most closely to commercial rather than residential construction practices. In order to obtain costs for solar systems, the information in the various estimating guides was supplemented with quotes from manufacturers and local distributors and informal estimates from local installers.

The costing procedures developed in this early effort were utilized to get installed system costs for conventional (nonsolar) and for series, parallel, and three-coil SAHP systems for one-family residences. These costs, while developed in some detail, were not refined to the extent that they were in the subsequent reporting period. However, they did allow a screening of the different types of systems according to life-cycle cost viability in one weather pattern (New York City).

PROGRESS FOR THE PERIOD NOVEMBER 1978 TO MARCH 1979

The major effort in this reporting period was to refine and generalize our methods of economic analysis. Data on component costs, labor costs, and overheads were obtained by the Applications Engineering Group of the General Electric Company's Central Air Conditioning Business Division and combined with the earlier results. Estimates of heat pump installed costs, including start-up and warrantee costs, have been used. Both conventional, single-speed, air-to-air heat pumps and non-conventional, two-speed and air-water units were analyzed. Assumptions for labor rates, overheads, and markups believed to be more appropriate for residential contractors were used. All of the data are presented in terms of costs to the general contractor.

In this reporting period, our analysis of cost-performance trade-offs has been extended to include some aspects of systems installed in rental

Ref. 15. Building Construction Cost Data 1976, Edited by R. S. Godfrey et al., R. S. Means Inc., Duxbury, Mass., 1975.

Ref. 16. McKee-Berger-Mansueto Inc., Building Cost File - 1975 Unit Prices Eastern Edition, Construction Publishing Company, Inc., New York, 1974.

Ref. 17. Dodge Building Cost Services, 1975 Dodge Manual for Building Construction Price and Scheduling, McGraw-Hill, New York, 1975.

property. For the rental-dwelling analysis, a four-apartment complex with a single, large-scale SAHP system was considered. The differences between these systems and those for the single-family, homeowner-occupied residences can be classified as follows: performance differences, installed cost differences, and financial differences. Performance differences arise from the large scale of the storage, the different physical layout of the system, and the different load characteristics for a multiapartment structure. The resources were not available to study the detailed performance differences. Therefore, the seasonal performance factors were taken from corresponding residential systems. Loads for each apartment were taken as 75 percent of the loads for a single-family system. Although there are no doubt some subtle performance differences between multi- and single-family systems, it is believed that neglecting them does not lead to significant changes in the conclusions.

Most of the efforts on systems for rental property were devoted to determining the differences in installed costs and to the financial considerations affecting the trade-offs of capital costs and operating costs for homeowner and rental investments. It was assumed that the rental complex is directly owned by a syndicate of fairly high income investors. With this assumption, it is found that for comparable assumptions for lifetime and operation and maintenance costs, differences in discount rates, taxes, and other parameters tend to cancel. In the end, there are no significant differences between the capital and fuel cost trade-off for homeowner or investor-owned residential property.

Some differences exist in the installed capital cost associated with multifamily systems. Economics of scale show up primarily in controls, collector pumps and piping, storage tanks, and startup costs. However, the difference in capital cost per unit of solar energy delivered for multifamily and single-family systems is not large for the examples treated.

One example of the capital vs. operating cost trade-off for a system installed in a commercial property owned by a corporation was also considered. The financial and tax treatment of this situation leads to a significantly different result than that for residential property, with the solar system turning out to be much more difficult to justify for the commercial case.

Costing results are presented for six systems, the conventional heat pump (nonsolar), the parallel SAHP, and the series SAHP — each for single family and multifamily dwellings. Based on work during the previous reporting period, the multisource SAHP systems were eliminated. It was discovered that some of the flexibility built into the previous systems, namely the provision for direct heating from the solar collectors without going through the storage, is not worth the extra systems complexity. Although the resources were not available to go back and modify the previous work on

residential systems, the multifamily systems were designed with simpler equipment layouts. Performance is scaled from the corresponding more complex residential systems. Any differences involved due to this procedure have negligible effects on the conclusions.

COST ESTIMATING PROCEDURE

This section contains tables and plots that can be used to obtain costs for SAHP systems installed in new residential construction. The costs are those passed on to the general contractor. The average expected selling price to the ultimate owner is given by the general contractor's costs plus a fair profit. Actual selling prices for particular systems will depend on variations in costs in different parts of the country, variations in costs among individual contractors, and fluctuations in competitive market conditions at the time the property is sold. Despite price variations, the calculation of the average selling price based on costs is well established in price estimation. When used with an appropriate account of marketability factors, this is the best way of understanding the relative costs to the consumer of differently equipped buildings.

The costs presented here are based on a SAHP system being installed by a mechanical contractor in new single-story residential construction. It is assumed that these costs also apply to small multiapartment buildings. The systems are assumed to be installed from separate components (non-modular) at the rate of 100 per year. All of the components are purchased as stock items from plumbing-heating distributors. The component costs are installed costs as would be quoted to general contractors. The installed costs include the mechanical contractors' overhead and profit. The system used for overhead and profit is a dual-rate system with separate overhead rates for both material and labor. Material and labor costs are taken to represent a national average. Cost amounts apply to billings and wages in the last quarter of 1978. (The consumer price index for this period, with 1967 = 100, is 200 [Ref. 18].)

The procedure for obtaining the installed cost is summarized by the following formulas:

Material cost to general contractor equals

Material x overhead x profit = material cost to general contractor

Material = cost of material to mechanical contractor
from distributor

Overhead = 30 percent

Profit = 10 percent profit on material cost to general
contractor

Ref. 18. U.S. Department of Commerce, Business Conditions Digest,
Vol. 19, No. 2, 1979, p. 84.

Labor cost to general contractor equals

Man-hours x labor rate x overhead x profit = labor cost to general contractor

Man-hours	=	time in hours to install component
Labor	=	mechanic rate based on a mechanic and a helper working together
Mechanic rate	=	\$8.50/h
Helper rate	=	\$5.00/h
Combined role	=	\$6.75/h
Overhead	=	60 percent
Profit	=	10 percent profit on labor cost to general contractor

The material cost plus the labor cost equals the installed component cost to the general contractor. The final cost to the customer has an additional 12 percent markup by the general contractor. The costing procedure is schematically illustrated in Figure 4-1.

Figure 4-1. Schematic representation of costing procedure

The installed cost of heat pumps is determined for air-to-air and water-to-air heat pumps. Due to the different methods of costing heat pump components, a separate discussion is provided in Appendix A, "Heat Pump Costs."

The remainder of this section consists of plots and tables that make possible computation of the total installed costs to the general contractor for each of the components which might be used in SAHP systems. The plots give the installed component costs plotted against a parameter measuring size or capacity. Tables 4-1 through 4-10 give data for the materials and labor portions of the various components. Finally, Table 4-11 gives the prices for miscellaneous components not described in the earlier plots and tables. A list of the components is given below:

- Plastic and copper tubing
- Armaflex insulation
- Plastic tubing with 0.5-in. insulation
- Copper tubing with 0.5-in. insulation
- Gate valves
- Globe valves
- Strainers and swing check valves
- 24-V control valves
- Pumps
- Rectangular duct work -- 1-in. fiberglass ductboard
- Flexible insulated round duct
- Heat pumps
- Heat exchangers
- Storage tanks
- Miscellaneous components

The data shown in the tables were obtained from the following sources:

National Association of Plumbing, Heating, and Cooling Contractors
Labor Estimating Guide

Raub Supply Company Price Manual

C & M Company - Duct Work Estimating Procedures

Grundfos Pump Company

TACO Pump Company

Milton Roy Company

ALCO Valve Company

American Supply Company

Figure 4-2. Pipe installed cost to general contractor:
 "A" - CPCV, 100 psi, 180 °F plastic tubing;
 "B" - Type "L" copper tubing;
 "C" - CPCV, 199 psi, 180 °F plastic tubing with 0.5-in. Armaflex insulation;
 and "D" - Type "L" copper tubing with 0.5-in. Armaflex insulation

Table 4-1

ARMAFLEX INSULATION

Tubing Size (inches)	Armaflex 0.5-in. Wall Mechanical Contractor Costs per Foot ⁽¹⁾	Material Cost to General Contractor Including Overhead and Profit	Labor Man-hour Per foot Including Fittings ^(2,3)	Labor Cost to General Contractor Including Overhead and Profit	Installed Cost per Foot to General Contractor
0.5	\$0.43	\$0.57	0.04	\$0.45	\$1.02
0.75	0.47	0.63	0.04	0.45	1.08
1.0	0.53	0.71	0.04	0.45	1.16
1.25	0.59	0.78	0.04	0.45	1.23
1.5	0.67	0.89	0.04	0.45	1.34
2.0	0.87	1.16	0.04	0.45	1.61

1) Material costs to mechanical contractor are based on actual quotes for June through August 1978.

2) Labor based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.

3) Two inches and below = 4.0 h for 100 ft = 0.04 h/ft.

Table 4-2

CPVC PLASTIC TUBING WITH 0.5-IN. INSULATION

Pipe Size (inches)	CPVC Plastic Pipe Mechanical Contractor Cost per Foot ⁽¹⁾	Fittings (10%)	Total Pipe Mechanical Contractor Cost to General Contractor	Pipe Cost to General Contractor, Including Overhead and Profit	Labor Man-Hour, Per Foot Including Fittings ^(2,3)	Labor Cost to General Contractor Including Overhead and Profit	Installed Cost per Foot of Pipe to General Contractor	Installed Cost per Foot of Insulation to General Contractor	Installed Cost per Foot of Insulated Plastic Pipe to General Contractor
0.5	\$0.19	0.02	\$0.21	\$0.28	\$0.12	\$1.35	\$1.63	\$1.02	\$2.65
0.75	0.35	0.04	0.39	0.52	0.125	1.41	1.93	1.08	3.01
1.0	0.50	0.05	0.55	0.73	0.150	1.69	2.42	1.16	3.58
1.25	0.80	0.08	0.88	1.17	0.158	1.78	2.95	1.23	4.18
1.5	1.05	0.10	1.15	1.53	0.180	2.03	3.56	1.34	4.90
2.0	1.80	0.18	1.98	2.63	0.210	2.37	5.00	1.61	6.61

- 1) Material costs to mechanical contractor are based on actual quotes for June through August 1978.
- 2) Labor based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.
- 3) Example: 10 ft of pipe = 0.04 man-hour/ft x 10 ft = 0.40 man-hour/10 ft
2 ells of fittings/10 ft = 0.40 man-hour/fitting x 2 = 0.80 man-hour for 2 fittings
1.2 man-hour/10 ft = 0.12 man-hour/ft

Table 4-3

COPPER TUBING WITH 0.5-IN. INSULATION

Tubing Size	Type "L" Tubing Mechanical Contractor Cost per Foot ⁽¹⁾	Fitting 10% ⁽¹⁾	Total Tubing Mechanical Contractor Cost per Foot	Tubing Cost to General Contractor Including Overhead and Profit	Labor Man-hour Per Foot Including Fittings ^(2,3)	Labor Cost to General Contractor Including Overhead and Profit	Installed Cost per Foot of Tubing to General Contractor	Installed Cost per Foot of Insulation to General Contractor	Installed Cost per Foot of Insulated Tubing to General Contractor
0.5	\$0.29	0.03	\$0.31	\$0.41	0.12	\$1.35	\$1.76	\$1.02	\$2.78
0.75	0.47	0.05	0.52	0.69	0.125	1.41	2.10	1.08	3.18
1.0	0.67	0.07	0.74	0.98	0.150	1.69	2.67	1.16	3.83
1.25	0.95	0.10	1.05	1.40	0.158	1.78	3.18	1.23	4.41
1.5	1.17	0.12	1.29	1.72	0.180	2.03	3.75	1.34	5.09
2.0	1.82	0.18	2.00	2.66	0.210	2.37	5.03	1.61	6.64

- 1) Material costs for mechanical contractor are based on actual quotes from June through August 1978.
- 2) Labor based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.
- 3) Example: 10 ft of pipe = 0.04 man-hour/ft x 10 ft = 0.40 man-hour/10 ft
2 ells or fittings/10 ft = 0.40 man-hour/fitting x 2 = 0.80 man-hour for 2 fittings
1.20 man-hour/10 ft = 12 man-hour/ft

Figure 4-3. Gate valves installed cost to general contractor

Table 4-4
GATE VALVES

Nominal Pipe Size (inches)	Manufacturer and Model No.	Cost to Mechanical Contractor ⁽¹⁾	Material Cost to General Contractor Including Overhead and Profit	Labor Man-hour per Valve ⁽²⁾	Labor Cost to General Contractor Including Overhead and Profit	Installed Cost to General Contractor
0.5	Hammond Bronze Gate Valve 125 lb.	\$ 4.23	\$ 5.63	0.40	\$4.51	\$10.14
0.75	Hammond Bronze Gate Valve 125 lb.	5.03	6.70	0.42	4.74	11.44
1.0	Hammond Bronze Gate Valve 125 lb.	7.15	9.51	0.50	5.64	15.15
1.25	Hammond Bronze Gate Valve 125 lb.	10.00	13.30	0.54	6.09	19.39
1.5	Hammond Bronze Gate Valve 125 lb.	12.85	17.09	0.60	6.77	23.86
2.0	Hammond Bronze Gate Valve 125 lb.	18.23	24.25	0.70	7.90	32.15
0.5	NIBCO Boiler Drain	1.70	2.26	0.40	4.51	6.77

1) Material costs to mechanical contractor are based on actual quotes for June through August 1978.

2) Labor man-hours based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.

Figure 4-4. Globe valves installed cost to general contractor

Table 4-5

GLOBE VALVES

Nominal Pipe Size (inches)	Manufacturer and Model No.	Cost to Mechanical Contractor ⁽¹⁾	Material Cost to General Contractor Including Overhead and Profit	Labor Man-hour per Valve ⁽²⁾	Labor Cost to General Contractor Including Overhead and Profit	Installed Cost to General Contractor
0.5	Hammond Bronze Globe Valve 125 lb.	\$ 4.92	\$ 6.54	0.40	\$4.51	\$11.05
0.75	Hammond Bronze Globe Valve 125 lb.	5.71	7.60	0.42	4.74	12.34
1.0	Hammond Bronze Globe Valve 125 lb.	8.33	11.07	0.50	5.64	16.71
1.25	Hammond Bronze Globe Valve 125 lb.	11.78	15.67	0.54	6.09	22.76
1.5	Hammond Bronze Globe Valve 125 lb.	15.72	20.91	0.60	6.77	27.68
2.0	Hammond Bronze Globe Valve 125 lb.	23.35	31.05	0.70	7.90	38.95

1) Material costs to mechanical contractor based on actual quotes for June through August 1978.

2) Labor man-hours based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.

Figure 4-5. Strainer and swing check valves costs to general contractor: "A" - Swing check valves; and "B" - Strainer

Table 4-6
STRAINERS AND SWING CHECK VALVES

STRAINER						
Nominal Pipe Size (inches)	Manufacturer and Model No.	Cost to Mechanical Contractor ⁽¹⁾	Material Cost to General Contractor Including Overhead and Profit	Labor Man-hour per Strainer ⁽²⁾	Labor Cost to General Contractor Including Overhead and Profit	Installed Cost to General Contractor
0.5	Trane TS-0.5 in.	\$ 7.58	\$10.09	0.40	\$4.51	\$14.60
0.75	Trane TS-0.75 in.	7.84	10.42	0.42	4.74	15.16
1.0	Trane TS-1.0 in.	10.98	14.58	0.50	5.64	20.20
1.25	Trane TS-1.25 in.	14.50	19.28	0.54	6.09	25.37
1.5	Trane TS-1.5 in.	21.56	28.68	0.60	6.77	35.45
2.0	Trane TS-2.0 in.	31.16	41.45	0.70	7.90	49.35

SWING CHECK VALVE						
0.5	Hammond Bronze 125 lb. Check Valve Copper	5.37	7.15	0.40	4.51	11.66
0.75	Hammond Bronze 125 lb. Check Valve Copper	6.82	9.07	0.42	4.74	13.81
1.0	Hammond Bronze 125 lb. Check Valve Copper	8.95	11.90	0.50	5.64	17.54
1.25	Hammond Bronze 125 lb. Check Valve Copper	12.60	16.76	0.54	6.09	22.85
1.5	Hammond Bronze 125 lb. Check Valve Copper	15.87	21.11	0.60	6.77	27.88
2.0	Hammond Bronze 125 lb. Check Valve Copper	17.05	22.67	0.70	7.90	30.57

1) Material costs to mechanical contractor based on actual quotes for June through August 1978.

2) Labor man-hours based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.

Figure 4-6. Control valves installed cost to general contractor: "A" - Curve used for three-way valves; "B" - Curve used for two-way valves. There are considerable variations in valve costs. See Table 4-7 for valve characteristics corresponding to plotted points.

Table 4-7

TWENTY-FOUR VOLT CONTROL VALVES

Valve Size (Inches)	GPM and Pressure Drop	Manufacturer and Model No.	Cost to Mechanical Contractor ⁽¹⁾	Material Cost to General Contractor Including Overhead and Profit	Labor Man-hours ⁽²⁾			Labor Cost to General Contractor Including Overhead and Profit Rate at 6.75	Installed Cost to General Contractor
					Piping	Wiring	Total		
0.5	8 gpm at 5 psi	ALCO 231WB08P04	\$40.77	\$ 54.22	0.40	0.30	0.70	\$ 7.90	\$ 62.12
0.75	16 gpm at 5 psi	ALCO 231WB12P06	49.79	66.22	0.42	0.30	0.72	8.11	74.33
1.0	30 gpm at 5 psi	ALCO 231WB16P08	75.53	100.45	0.50	0.30	0.80	9.00	109.45
0.75	6 gpm at 1 psi	TACO No. 571 Heating Zone Valve	26.33	35.01	0.42	0.30	0.72	8.11	43.12
1.0	10 gpm at 2 psi	TACO No. 572 Heating Zone Valve	27.53	36.61	0.50	0.30	0.80	9.00	45.61
1.25	16 gpm at 3 psi	TACO No. 573 Heating Zone Valve	32.23	42.86	0.54	0.30	0.84	9.47	52.33
0.5	9 gpm at 5 psi	Honeywell V8044A1010 Diverting	25.50	33.92	1.00	0.30	1.30	14.65	48.57
0.75	16 gpm at 5 psi	Honeywell V8044A1044 Diverting	26.75	35.58	1.00	0.30	1.30	14.65	50.23

1) Material cost to mechanical contractor based on actual quotes for June through August 1978.

2) Labor man-hours based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.

Figure 4-7. Pumps installed cost to general contractor

Table 4-8

PUMPS

Manufacturer and Model No.	Pump (hp)	Feet of Head at 10 GPM	Cost to Mechanical Contractor ⁽¹⁾	Pump Cost to General Contractor Including Overhead and Profit	Labor Man-hours ⁽²⁾			Labor Cost to General Contractor Including Overhead and Profit	Installed Cost to General Contractor
					Piping	Wiring	Total		
Milton Roy GPPS-45H-1	1/7	17	\$88.50	\$117.70	0.5	1.0	1.5	\$16.90	\$134.60
Grundfos UP26-64	1/12	14	80.00	106.87	0.5	1.0	1.5	16.90	123.57
TACO 008-FZ	1/25	8.5	92.56	123.10	0.5	1.0	1.5	16.90	140.00

1) Material costs to mechanical contractor are based on actual quotes for June through August 1978.

2) Labor based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.

Figure 4-8. Duct installed cost to general contractor

Table 4-9

RECTANGULAR DUCTBOARD 1-IN. FIBERGLASS

Size (inches)	Material Cost per ft. ² to Mechanical Contractor ⁽¹⁾	Material Cost per ft. ² to General Contractor Including Overhead and Profit	Material (ft. ² per foot)	Material Cost per Linear Foot to General Contractor	Man-hours per ft. ² to Construct Duct ⁽²⁾	Labor Cost per Linear Foot Including Overhead and Profit to Install Duct	Fabricated Cost per Linear Foot to General Contractor	Man-hours per ft. ² to Install Duct ⁽²⁾	Labor Cost per Linear Foot Including Overhead and Profit to Install Duct	Installed Cost per Linear Foot to General Contractor
6 x 6	\$0.51	\$0.69	2.7	\$1.86	0.05	\$1.52	\$3.38	0.05	\$1.52	\$ 4.90
8 x 6	0.51	0.69	3.0	2.07	0.05	1.69	3.76	0.05	1.69	5.28
10 x 6	0.51	0.69	3.4	2.35	0.05	1.91	4.26	0.05	1.91	6.17
12 x 6	0.51	0.69	3.7	2.55	0.05	2.08	4.63	0.05	2.08	6.71
14 x 6	0.51	0.69	4.0	2.76	0.05	2.25	5.01	0.05	2.25	7.06
16 x 6	0.51	0.69	4.4	3.03	0.05	2.47	5.50	0.05	2.47	7.97
18 x 6	0.51	0.69	4.7	3.24	0.05	2.64	5.88	0.05	2.64	8.52
20 x 6	0.51	0.69	5.0	3.45	0.05	2.81	6.26	0.05	2.81	9.07
22 x 6	0.51	0.69	5.4	3.72	0.05	3.04	6.76	0.05	3.04	9.80
24 x 6	0.51	0.69	5.7	3.93	0.05	3.21	7.14	0.05	3.21	10.35
26 x 6	0.51	0.69	6.0	4.14	0.05	3.38	7.52	0.05	3.38	10.90
28 x 6	0.51	0.69	6.4	4.41	0.05	3.60	8.01	0.05	3.60	11.61
30 x 6	0.51	0.69	6.7	4.62	0.05	3.77	8.39	0.05	3.77	12.16

1) Material cost to mechanical contractor based on actual quotes for June through August 1978.

2) Labor man-hours based on C & M Company, Mechanical Contractor in Pennsylvania.

Figure 4-9. Flexible insulated round duct installed cost to general contractor

Table 4-10

FLEXIBLE INSULATED ROUND DUCT

Size Diameter (inches)	Cost to Mechanical Contractor ⁽¹⁾	Material Cost to General Contractor Including Overhead and Profit	Labor Man-hours per Foot ⁽²⁾	Labor Cost to General Contractor Including Overhead and Profit	Installed Cost per Foot to General Contractor
5	\$1.75	\$2.32	0.10	\$1.13	\$3.45
6	1.84	2.44	0.10	1.13	3.57
7	2.07	2.75	0.10	1.13	3.88
8	2.29	3.04	0.11	1.23	4.27
10	2.68	3.56	0.12	1.35	4.91
12	3.41	4.53	0.15	1.69	6.22
14	4.12	5.47	0.17	1.92	7.39

1) Material cost to mechanical contractor based on actual quotes for June through August 1978.

2) Labor man-hours based on National Association of Plumbing, Heating, and Cooling Contractors Labor Estimating Guide.

Figure 4-10. Heat pump installed costs to general contractor. Installed cost includes outdoor unit, air handler, refrigerant lines, resistance heat, warranty costs, and installation labor based on a non-modular system.

Figure 4-11. Heat exchanger installed cost to general contractor

Table 4-11

MISCELLANEOUS COMPONENTS

Item	Size	CFM	Material Cost to General Contractor	Labor Man-hours	Labor Cost to General Contractor	Installed Cost to General Contractor
Floor Diffuser	2 1/4 in. x 12 in.	100	\$ 3.61	0.5	\$ 5.63	\$ 9.24
Floor Diffuser	4 in. x 12 in.	120	4.00	0.5	5.63	9.63
Ceiling Diffuser	6 in. Ø	75	5.85	0.5	5.63	11.48
Ceiling Diffuser	8 in. Ø	125	6.60	0.5	5.63	12.23
Ceiling Diffuser	10 in. Ø	175	7.55	0.5	5.63	13.18
Return Grills	14 in. x 8 in.	300	1.65	0.5	5.63	7.28
Return Grills	24 in. x 8 in.	500	2.70	0.5	5.63	8.33
Return Grills	30 in. x 8 in.	600	3.10	0.5	5.63	8.73
Floor Diffuser Boots	2 1/4 in. x 12 in.	100	2.64	0.5	5.63	8.27
Floor Diffuser Boots	4 in. x 12 in.	130	3.00	0.5	5.63	8.63
Air Handler	2 tons	800	273.00	8.0	90.00	363.00
Air Handler	2 1/2 tons	1000	305.00	8.0	90.00	395.00
Air Handler	3 tons	1200	362.00	8.0	90.00	452.00
Air Handler	3 1/2 tons	1400	399.00	8.0	90.00	489.00
Air Handler	4 tons	1600	444.00	8.0	90.00	534.00
Electric Heater	15 kW	All	94.00	1.0	11.30	105.30
Heat Pump - 2 Speed	2 tons	--	897.00	6.0	68.00	965.00
Heat Pump - 2 Speed	2 1/2 tons	--	1065.00	6.0	68.00	1133.00
Heat Pump - 2 Speed	3 tons	--	1224.00	6.0	68.00	1296.00
Heat Pump - 2 Speed	3 1/2 tons	--	1410.00	6.0	68.00	1478.00
Heat Pump - 2 Speed	4 tons	--	1591.00	6.0	68.00	1659.00
Refrigerator Line Set	25 feet	--	80.00	1.5	17.00	97.00
Heat Exchanger	--	--	--	3.0	34.00	--
Control Center - Standard Heat Pump	--	--	35.25	2.0	23.00	58.25
Control Center - Solar (nonmodular)	--	--	499.00	8.0	90.00	589.00
Control Center - Solar (modular)	--	--	499.00	4.0	45.00	544.00
Control Center Wiring - (standard heat pump)	--	--	7.00	2.0	23.00	30.00
Control Center Wiring - (solar)	--	--	16.00	4.0	45.00	61.00
Relay Solar (nonmodular)	--	--	17.30	0.5	5.63	22.93
Module (installation)	--	--	--	8.0	90.00	--
Startup - Standard Heat Pump	--	--	--	8.5	95.00	--
Startup - Solar (modular/nonmodular)	--	--	--	12.00	135.00	--
Water Storage Tank	82 gal.	--	100.00	3.0	34.00	134.00
Electric Water Heater	30 gal.	--	--	--	--	185.00
Electric Water Heater	52 gal.	--	--	--	--	256.00
Expansion Tank	8 gal.	--	35.00	1.0	11.30	46.30
Thermal Insulation for Storage Tank	--	--	--	--	--	1/ft ²

- COATED STEEL TANK, INSTALLED UNDERGROUND
20-YEAR WARRANTY GALVANIC PROTECTION
- ◆ COATED STEEL TANK, INSTALLED UNDERGROUND
- STEEL TANK, INSTALLED
- ▽ STEEL FIXED ROOF OIL TANK, INSTALLED
- FIBERGLASS TANK, U.L. LISTED, INSTALLED UNDERGROUND
- PRESTRESSED CONCRETE TANK, INSTALLED
- △ PRECAST CONCRETE SEPTIC TANK, INSTALLED UNDERGROUND

Figure 4-12. Buried tanks installed cost to general contractor. Sources: 1) Means building construction cost data, 1976, pp. 18, 20, 22, 34, 184, 205-206 and 258; and 2) Office of Technology Assessment Report: Solar Energy Applied to Today's Energy Needs, Washington, D.C. Corrected by a factor of 1.3 for escalation to 1978.

LIFE-CYCLE COSTING METHODOLOGY

EXTENSION TO RENTAL PROPERTY

The last six months technical report (Ref. 1) described the life-cycle cost calculation for a homeowner investing in his own residence. The calculation for a nonresident investor considering a SAHP system for rental property is different. The difference is due to the different return expected for rental property investments compared with investments in one's own home, and also because of the different tax treatment of fuel expenses and depreciation for rental property.

Just as life-cycle costing is not the most common procedure used for judging homeowner investments, it is also not the most common procedure

for rental property. For example, the concept of "cash-on-cash" return is commonly applied to improvements which are anticipated to appreciate in value as the property itself appreciates. However, the cash-on-cash return concept, like the simple payback concept, is only an approximate measure of the key factors affecting an investment. Life-cycle cost, with its emphasis on return on investment, should be the preferred procedure for judging the relative attractiveness of different options.

The notation used in the previous report is followed here, namely:

PV	= present value referred to beginning of month one
d	= monthly compounded rate
i	= interest rate on loan
ϵ	= escalation rate (different for energy, maintenance, and replacement parts)
t	= incremental tax rate
T	= life-cycle period
e_0	= monthly cost at beginning of costing period
MP (ϵ, d, T)	= factor which relates monthly escalating payments to present value
TP (i, d, T)	= factor which gives present value of interest portion of level mortgage payments
DP (d, N)	= factor which gives present value of depreciation
IC	= initial capital cost

The only new factor to be considered is depreciation. Depreciation enters into the economic calculation for rental property because it can be used to offset taxable income. This is a valuable consideration for a high income tax bracket investor. Generally, it is advantageous to charge off depreciation as rapidly as possible within the rules allowed by the Internal Revenue Service. Commonly used depreciation methods include "straight line," "double declining balance," and "sum of years digits." The lifetimes used are governed either by the actual expected life of the asset, or by a set of guideline ranges (asset depreciation ranges or ADR's). For a solar-assisted heat pump system, it appears that a tax life of eight years would be allowable [Ref. 19]. It will be assumed that the depreciation method is "sum of years digits" for which the factor DP is given by

$$DP(d, T_t) = \frac{2}{d} \left[\frac{T_t - MP(0, d, T_t)}{T_t (T_t + 1)} \right]. \quad (4-1)$$

Ref. 19. J.K. Lasser, Your Income Tax, Simon and Schuster, New York, 1978, Sec. 5.41.

The meaning of the discount factor to be used for rental property life-cycle costing is slightly different than for the homeowner case. Here, the monthly compounded rate, d , should be taken as the after-tax rate of return available to the investor from investments of the same character of risk offered by the SAHP system. This rate is generally higher than the consumer discount rate which applies to the typical homeowner.

When tax effects are considered, the elements of the present value for the life-cycle costs can be organized as follows:

$$\begin{aligned}
 PV = & \text{initial down payment} \\
 & + \text{present value of loan payments} \\
 & + (1-t) \times \text{present value of O\&M (operation and maintenance)} \\
 & \quad \text{net of income tax effects} \\
 & + (1-t) \times \text{present value of property taxes and insurance} \quad (4-2) \\
 & \quad \text{net of income tax effects} \\
 & - t \times \text{present value of interest payment} \\
 & - t \times \text{present value of depreciation} \\
 & + (1-t) \times \text{present value of energy costs} \\
 & - \text{tax credits}
 \end{aligned}$$

For the calculation of the rental property life-cycle cost, the economic parameters listed in Table 4-12 are used.

Table 4-12
ECONOMIC PARAMETERS FOR LIFE-CYCLE COSTING
FOR RENTAL PROPERTY

General Inflation Rate	6% ⁽¹⁾
Discount Rate (rate of return)	11% ⁽¹⁾
Life-cycle Period	15 years ⁽²⁾
Combined Federal-State Income Tax Rate	50%
Tax Life for Depreciation	8 years ⁽³⁾
Tax Credit Rate	10% ⁽⁴⁾
Loan Interest Rate	9% ⁽¹⁾
Down Payment	20%
Real Estate Taxes and Insurance	1% ^(5, 6)
Price Inflation for Energy	
Electricity	8.5%
Oil	9.5%
Gas	10.5%
Prices of Energy for Late 1978	Regionally determined
National Average Energy Prices	
Electricity	\$9.72/GJ (3.5¢/kWh)
Oil	\$3.83/GJ (55¢/gal)
Gas	\$3.32/GJ (35¢/therm)
Operation and Maintenance Cost Rate	5% ⁽⁶⁾

- 1) Per annum compounded monthly
- 2) True useful life
- 3) Applicable for heating equipment
- 4) Solar tax credit for business investment. Heating systems do not qualify for standard investment tax credit
- 5) Assumes 0.5 percent real estate tax rate for solar systems
- 6) Increases with general inflation

For these parameters, the values of the individual terms in Equation 4-2 are as given in the following. The initial capital expenditure is denoted IC, and the initial monthly expenses for electricity, oil, and gas are denoted by e(elect), e(oil), and e(gas).

$$\begin{aligned}
 PV = & 0.2 \text{ IC (down payment)} + 0.714 \text{ IC (loan payments)} \\
 & + 0.264 \text{ IC (O\&M)} + 0.053 \text{ IC (property taxes and insurance)} \\
 & - 0.242 \text{ IC (tax benefit for interest)} - 0.379 \text{ IC (tax benefit} \\
 & \quad \text{for depreciation)} - 0.090 \text{ IC (solar tax credit-} \\
 & \quad \text{might apply to only a portion of IC)} + 75.1 \text{ e(elect)} \\
 & + 80.6 \text{ e(oil)} + 86.7 \text{ e(gas)} \qquad (4-3)
 \end{aligned}$$

The terms proportional to IC can be summed to give the final result for the present value of the cash flows for an investment in rental property, assuming that the solar tax credit applies to any increments in IC, namely:

$$PV = 0.520 \text{ IC} + 75.1 \text{ e(elect)} + 80.6 \text{ e(oil)} + 86.7 \text{ e(gas)} \qquad (4-4)$$

Expressed in simple payback terms, the investment will be justified if the payback, based on initial savings, is about 13 years for oil. A slightly longer payback is justified for gas (13.4 years) and a slightly shorter one for electrical energy savings (12 years).

Because tax benefits substantially affect the capital cost for down payments and loan payments, the result of Equation 4-4 is very sensitive to assumptions made for noncapital costs which are taken as proportional to IC. In particular, the operation and maintenance costs, which were taken as 5 percent per year following the suggestion of Reference 10, are larger than all of the other capital-dependent costs. If the operation and maintenance costs were negligible, the allowed payback periods would be more than doubled!

It is instructive to compare the present value calculation for rental property with that for the resident homeowner case. The latter case has been extensively discussed in the first six months technical report (Ref. 1), where calculations have been made with parameters which are characteristic of a homeowner's point of view. In certain respects, there are differences between the homeowner parameters and the rental property parameters which are not directly related to differences in the true economics of the two cases. This is particularly true of the lifetime assumptions (15 years vs. 20 years) and the operation and maintenance costs (5 percent/year vs. negligible). To provide a better understanding of the relation of the rental and homeowner cases, let us consider two sets of homeowner parameters: the base case established in the first six months technical report and a comparison case more nearly consistent with the assumption for rental property. These parameters are given in Table 4-13, with the various factors needed for the present value calculation listed in Table 4-14.

Table 4-13

ECONOMIC PARAMETERS FOR LIFE-CYCLE COSTING FOR A RESIDENT HOMEOWNER

Parameter	Base Case	Comparison Case
General Inflation Rate	6% ⁽¹⁾	6% ⁽¹⁾
Consumer Discount Rate	6% ⁽¹⁾	7% ⁽¹⁾
Life-cycle Period	20 years ⁽²⁾	15 years ⁽²⁾
Combined Federal-State Income Tax Rate	37.5%	37.5%
Tax Credit	30% on first \$2000 20% on next \$8000 10% on installation costs	20% average
Loan Interest Rate	10%	9.5%
Down Payment	20%	20%
Real Estate Tax	exempt	0.5% ⁽³⁾
Insurance	neglected	0.5% ⁽³⁾
Price Inflation for Energy		
Electricity	8%	8.5%
Oil	8%	9.5%
Gas	8%	10.5%
Prices of Energy for Late 1978	regionally determined	
National Average Energy Prices		
Electricity	\$9.72/GJ (3.5¢/kWh)	
Oil	\$3.83/GJ (55¢/gal)	
Gas	\$3.32/GJ (35¢/therm)	
Operation and Maintenance	neglected ⁽³⁾	5% ⁽³⁾

- 1) Per annum compounded monthly
2) True useful life
3) Increases with general inflation

Table 4-14

FACTORS USED IN PRESENT VALUE CALCULATIONS FOR RENTAL PROPERTY AND FOR A RESIDENT HOMEOWNER

<u>Rental Case</u>	
MP (0, 9%, 180 mo)	98.59
MP (0, 11%, 180 mo)	87.98
MP (6%, 11%, 180 mo)	126.66
MP (8.5%, 11%, 180 mo)	150.16
MP (9.5%, 11%, 180 mo)	161.23
MP (10.5%, 11%, 180 mo)	173.44
TP (9%, 11%, 180 mo)	0.4843
DP (11%, 96 mo)	0.7577
<u>Homeowner Base Case</u>	
MP (0, 10%, 240 mo)	103.62
MP (0, 6%, 240 mo)	139.58
MP (6%, 6%, 240 mo)	240
MP (8%, 6%, 240 mo)	294.97
TP (10%, 16%, 240 mo)	0.8675
<u>Homeowner Comparison Case</u>	
MP (0, 9.5%, 180 mo)	95.76
MP (0, 7%, 180 mo)	111.26
MP (6%, 7%, 180 mo)	217.55
MP (8.5%, 7%, 180 mo)	279.78
MP (9.5%, 7%, 180 mo)	311.13
MP (10.5%, 7%, 180 mo)	347.09
TP (9.5%, 7%, 180 mo)	0.6147

The present value formula for the base case is:

$$\begin{aligned} \text{PV (base case)} = & 0.2 \text{ IC (down payment)} + 1.078 \text{ IC (loan payment)} \\ & + 0(\text{O\&M}) - 0.260 \text{ IC (tax benefit for interest)} \\ & - \text{tax credit} + 295.0 \text{ e(electricity)} + 295.0 \text{ e(oil)} \\ & + 295.0 \text{ e(gas)} \end{aligned} \quad (4-5)$$

Collecting terms proportional to IC gives:

$$\begin{aligned} \text{PV (base case)} = & 1.018 \text{ IC} - \text{tax credit} + 295.0 \text{ e(electricity)} \\ & + 295.0 \text{ e(oil)} + 295.0 \text{ e(gas)} \end{aligned} \quad (4-6)$$

The results for the comparison case of a homeowner are:

$$\begin{aligned} \text{PV (comparison)} = & 0.2 \text{ IC (down payment)} + 1.162 \text{ IC (loan payment)} \\ & + 0.906 \text{ IC (O\&M)} + 0.091 \text{ IC (insurance)} \\ & + 0.057 \text{ IC (real estate tax)} \\ & - 0.184 \text{ IC (tax benefit for interest)} \\ & - 0.187 \text{ IC (tax credit)} + 279.8 \text{ e(electricity)} \\ & + 311.13 \text{ e(oil)} + 347.09 \text{ e(gas)} \end{aligned} \quad (4-7)$$

Collecting terms proportionally to IC gives:

$$\begin{aligned} \text{PV (comparison)} = & 2.045 \text{ IC} + 279.8 \text{ e(elect)} + 311.1 \text{ e(oil)} \\ & + 347.1 \text{ e(gas)} \end{aligned} \quad (4-8)$$

Comparing Equations 4-6 and 4-8, one notes that the changes in assumptions, particularly regarding operation and maintenance costs, have changed the present value associated with initial capital cost by about a factor of two. Thus, while the base case homeowner calculation indicates that a solar investment of \$10 000. can be justified for an initial monthly savings in electricity costs of only \$31.80, the comparison case requires an initial monthly savings of \$73.09. For rental property (Equation 4-4), the required initial monthly savings is \$69.24. When the result of the comparison case is compared with that for rental property, it is seen that the solar investment is actually slightly easier to justify for rental property.

Now, let us consider briefly the economic analysis of a solar heating installation in a building owned by a corporation. Here, the analysis is done differently than for the case of ownership by an individual because of the extra level of taxation before corporate earnings are translated into spendable consumer income.

The present value of the "revenue requirement" which the corporation needs in order to get its expected return on investment is computed. Two

cases are considered: one quite favorable to solar investment and another which is less favorable. The parameters are summarized in Table 4-15. Only systems using electrical energy will be considered. The result for the present value in the favorable case with a 20-year life and a generous tax treatment and a low maintenance cost is:

$$PV = 1.384 IC + 160 e(\text{elect}). \quad (4-9)$$

Table 4-15

PARAMETERS FOR ECONOMIC CALCULATION
FOR CORPORATION-OWNED SOLAR SYSTEMS

Debt Fraction		40%
Debt Cost		9.5%
Common Stock Fraction		60%
Return for Common Stock		18.67%
Weighted Return on Capital		15%
Income Tax Rate (State and Federal Effective Combined Rate)		50%
"After Tax" Discount Rate		13.10%
Lifetime*	10 years	20 years
Capital Recovery Factor	18.50%	14.32%
Investment Tax Credit	10%	20% (on solar equipment)
Sum of Years Digits Tax Depreciation	7-year tax life	15-year tax life
Operation and Maintenance	5%	1%
Ad Valorem Taxes and Insurance	1.9%	1.6%
Present Value of Capital Related Costs	1.608 IC	1.384 IC

For the less favorable case, with a 10-year life, the present tax credits (for a building solar space heating system) and a 5 percent per year maintenance cost is:

$$PV = 1.608 IC + 96.2 e(\text{elect}). \quad (4-10)$$

In payback terms, the most favorable case requires a simple payback of 9.6 years. The payback for the least favorable case is 5 years.

The conclusions of the above discussion are: First, the results are very dependent on whether high (5 percent of IC per year) or low (negligible) amounts are assumed for operation and maintenance. For comparable assumptions for operation and maintenance costs and lifetime and reasonable assumptions for other parameters, the very different tax effects and discount

*Possible resale value of capital equipment is neglected.

rate assumptions for homeowner and rental property installations cancel out to a large extent. A simple payback (based on initial energy savings) of about 12 years applies in either case when the high operation and maintenance costs assumption is used. Finally, a system owned by a corporation was considered. Here, the finances are considerably weighted against the solar system. A simple payback in the range of 5 to 10 years would be required to justify a solar investment.

INSTALLED SYSTEM COSTS

The methodology presented in the section "Cost Estimating Procedure" (page 4-4) is used to calculate the initial capital costs for the candidate SAHP systems and the reference nonsolar heat pump systems. Six systems are considered, three each for single family and multifamily dwellings. In each case, a schematic system layout (Figures 4-13 through 4-18) has been constructed. From these layouts, major pieces of equipment and approximate lengths of pipes and ducting have been identified. These in turn have been categorized and costs estimated and tabulated (Tables 4-16 through 4-21). Final system costs are given for two different assumptions for collector costs for a typical collector area.

ECONOMIC CONCLUSIONS

The economic factors governing the competitive position of conventional nonsolar heat pump systems and SAHP systems for homeowner, rental property, and corporate ownership cases have been examined. The differences in ownership are associated with differences in the scale of the systems and, in principle, with different functions for the building which would affect the load characteristics. Ownership also has an important effect on the finances for capital-intensive relative to fuel-intensive investments. These aspects, the extent to which they have been dealt with, and the principal conclusions are summarized below.

Engineering Aspects

- Component performance as a function of system size -- not treated; not expected to be significant.
- System performance due to load differences -- not treated, significant difference expected for nonresidential structures, but single family and multifamily essentially proportional.
- Economies of scale for components and installation -- considered for the case of a multifamily complex. The effect is about a 10 to 15 percent savings in installed capital cost for the solar portion of the system, based on the same solar energy delivery.

Financial Aspects (treated in some detail)

- Homeowner case -- results are very sensitive to the assumption for operation and maintenance costs. The required simple payback is from 12 to 20 years.
- Rental property -- despite great differences in input assumptions, return, tax treatment, etc., the required payback is the same as comparable homeowner cases.
- Corporation ownership -- capital intensive systems are more difficult to justify. The required simple payback is 5 to 10 years.

SYSTEM COST COMPARISONS**EXPLANATION OF COSTING GRAPH**

Figure 4-19 shows the graph used to compare the life-cycle cost of any candidate SAHP system with a base system used in constructing the graph. In this case, a single-speed, air-to-air heat pump is the base system. On the ordinate is plotted the ratio of the base system SPF to the candidate system SPF, while on the abscissa is plotted the ratio of the candidate system initial cost to the base system initial cost, minus one. The negative-sloping lines represent the constant life-cycle cost, i.e., all of the systems that lie on a particular line possess the same life-cycle cost. The fact that the lines have a negative slope agrees with the fact that as the SPF of a system increases one can afford to pay a higher initial cost for the system and still obtain an equal life-cycle cost. An initial electricity cost is indicated on each constant life-cycle cost line; this electricity cost is assumed to escalate at 8 percent.

SYSTEM COST COMPARISONS

Figure 4-20 shows the life-cycle cost results for parallel and series systems in New York, New York. Included are the SPF and the initial cost of the base case, single-speed heat pump. Solar collector costs are treated parametrically, ranging from \$5 to \$10/ft² of collector area. Each curve on the graph represents (for either a series or parallel system with a collector cost of \$5 or \$10/ft²) a varying collector area. Increasing areas are obtained by moving down and to the right along the curve. Each point represents a certain collector area as defined below.

Series: 50, 75, and 100 m²

Parallel: 25, 50, 75, and 100 m²

To determine the minimum life cycle cost, the optimum collector area, and the initial electricity cost for any candidate system, one simply draws a line tangent to the system curve which intersects the ordinate at unity.

The optimum systems for New York City are derived from Figure 4-20 and summarized in Table 4-22.

It can be seen that the parallel system becomes cost effective (with respect to the single-speed heat pump) at an initial electricity rate of 5.8¢ and 8.5¢/kWh as opposed to 7.3¢ and 12.8¢/kWh for the series system. The optimum series systems have larger collector areas than the parallel systems and, as a result, higher SPF's. However, the high cost of the series system collector array forces them to become cost effective at higher initial energy costs.

Figure 4-21 and Table 4-22 show the system cost results for Fort Worth, Texas. Here again, the parallel system becomes cost effective at 9.3¢ and 14.0¢/kWh at lower initial electrical rates than for the series system. Here the difference in threshold energy costs (between the parallel and series systems) is larger than that for the systems in the New York City climate. This is the result of two factors:

- The Fort Worth annual heating load is smaller than that of New York City. Hence, the electricity rate must be larger to make the total energy cost correspond to that in the New York City life-cycle cost.
- Parallel systems perform well in warmer climates (where the ambient air (source) temperature is high) when compared to series systems, for equal collector areas.

Due to the reduced heating load and the increased solar insolation, smaller collector areas for both series and parallel systems are required, when compared to the optimum systems in New York.

An examination of the results for Madison, Wisconsin (Figure 4-22 and Table 4-22) shows an opposite trend from those of New York City and Fort Worth, Texas. In this colder climate (with a correspondingly higher heating load), the series system performs well compared to the parallel system which utilizes a very cold source. The threshold energy costs for the series and parallel systems are very close. Also, the optimum collector areas are somewhat larger for both systems.

The cost comparisons for Phoenix, Arizona are shown in Figure 4-23, where the same overall trends as for Fort Worth, Texas, can be seen.

The cost comparisons for the New York City systems utilizing the GE TC-100 collector are shown in Figure 4-24. This collector is a high-efficiency, evacuated-tube type, the cost of which is likely to be in the \$20 to \$30/ft² range. It can be seen from Figure 4-24 that the threshold energy costs at which the parallel system becomes cost effective when compared to the base system varies from 4.5¢/kWh to 11¢/kWh for collector costs ranging from \$10/ft² to \$30/ft². The threshold energy costs for the series system vary from 7.5¢/kWh to 14¢/kWh for collector costs ranging from \$10/ft² to \$20/ft².

Figure 4-13. Reference nonsolar, single-family, heat pump system

Table 4-16

**COSTS FOR REFERENCE NONSOLAR, SINGLE-FAMILY,
HEAT PUMP SYSTEM**

Pipe	--
Valves	--
Pumps	--
Heat exchanger	--
Controls: control unit	\$ 58.25
wiring	30.00
Ducting: ducts	963.10
grilles	100.57
Tank	--
Collectors	--
Heat pump (2.5-ton, 1-speed)	1470.00
Start-up	95.00
Miscellaneous charges*	205.00
Total Cost to General Contractor	\$2922.00
Cost to Purchaser	\$3275.00

*Allowance for higher charge associated with concrete pad, contingency allowance, return checkout, etc. (Such charges are not included for solar systems.)

Figure 4-14. Parallel, single-family, SAHP system

Table 4-17

COSTS FOR PARALLEL, SINGLE-FAMILY, SAHP SYSTEM
(Based on 50-m² Collector)

Pipe:	12 m (Length)	1.90 cm (Diameter)	120
	30 m	2.54 cm	360
Valves:	1 2-way		50
	1 3-way		65
	2 Strainers		16
Pumps:	Collector, 100 W		145
	Storage, 60 W		120
Heat Exchanger (air-to-water)	0.5 m ²		125
Controls:	Control center		544
	Wiring		61
	Relays (2 for pumps)		183
Ducting:	Ducts		963
	Grills		101
Tank (Insulated), 4-m ³			750
Collector		Case 1 ⁽¹⁾	Case 2 ⁽²⁾
		5380	2690
Solar Heat Pump (2.5-ton, 2-speed)			1700
Start-up			135
Cost to General Contractor		10 752	8062
Cost to Purchaser		12 042	9030

1) \$10/ft² installed cost.

2) \$5/ft² installed cost.

Figure 4-15. Series, single-family, SAHP system

Table 4-18

COSTS FOR SERIES, SINGLE-FAMILY, SAHP SYSTEM
(Based on 50-m² Collector)

Pipe:	18 m (Length) 1.90 cm (Diameter)	\$ 180
	30 m 2.54 cm	360
Valves:	1 2-way	50
	2 3-way	130
	3 Strainers	50
Pumps:	Collector, 100 W	145
	Storage, 60 W	120
	Source, 60 W	120
Heat Exchanger (air-to-water)	0.5 m ²	125
Controls:	Control Center	544
	Wiring	61
	Relays (3 for pumps)	175
Ducting:	Ducts	963
	Grilles	101
Tank (insulated), 4-m ²		750
Collector	Case 1 ⁽¹⁾	\$5380
	Case 2 ⁽²⁾	2690
Solar Heat Pump (2.5-ton, 2-speed, water source)		1400
Start-up		135
Cost to General Contractor	10 789	8099
Cost to Purchaser	\$12 084	\$9071

Figure 4-16. Reference nonsolar, multifamily, heat pump system

Table 4-19

**COSTS FOR REFERENCE NONSOLAR,
MULTIFAMILY, HEAT PUMP SYSTEM**

Pipe	--
Valves	--
Pumps	--
Heat exchangers	--
Controls: Control Unit	\$ 233
Wiring	90
Ducting: Ducts	2890
Grilles	300
Tank	--
Collectors	--
Heat Pumps (2-ton, 2-speed, 4 units)	6400
Start-up	90
Total Cost to General Contractor	\$10 003
Cost to Purchaser	\$11 203

Figure 4-17. Parallel, multifamily, SAHP system

Table 4-20
COSTS FOR PARALLEL, MULTIFAMILY,
SAHP SYSTEM

Pipe:	40 m (Length) 1.90 cm (Diameter)	\$ 400
	30 m 2.54 cm	360
	30m 3.18 cm	390
Valves:	5 Strainers	83
Pumps:	1 Collector, 500 W	400
	2 Storage, 60 W	240
	2 Storage, 100 W	290
Heat Exchangers (4 units, air-to-water),	0.5-m ²	500
Controls:	Control Center	544
	Wiring	183
	Relays 5 for Pumps	234
Ducting:	Ducts	2890
	Grills	303
Tank (Insulated), 12-m ³		1770
Collector	Case 1 ⁽¹⁾	Case 2 ⁽²⁾
	\$16 140	8070
Solar Heat Pump (2-ton, 2-speed)		6400
Start-up		270
Cost to General Contractor	\$31 094	\$23 024
Cost to Purchaser	\$34 825	\$25 787

1) \$10/ft² installed cost.

2) \$5/ft² installed cost.

Figure 4-18. Series, multifamily, SAHP system

Table 4-21
COSTS FOR SERIES, MULTIFAMILY, SAHP SYSTEM

Pipe:	60 m (Length)	1.90 cm (Diameter)	\$ 600
	30 m	2.54 cm	360
	30 m	3.18 cm	390
Valves:	4 3-way		260
	5 Strainers		83
Pumps:	1 Collector, 500 W		400
	2 Storage, 100 W		290
	2 Storage, 150 W		352
Heat Exchangers (4 units, air-to-water),	0.5-m ²		500
Controls:	Control Center		544
	Wiring		183
	Relays (5 for Pumps)		234
Ducting:	Ducts		2890
	Grills		303
Tank (Insulated), 12-m ³			1770
			Case 1 ⁽¹⁾ Case 2 ⁽²⁾
Collector			\$16 140 8070
Solar Heat Pump (2-ton, 2-speed air-to-water)			5400
Start-up			270
Cost to General Contractor			\$30 969 \$22 899
Cost to Purchaser			\$34 685 \$25 647

1) \$10/ft² installed cost.

2) \$5/ft² installed cost.

Figure 4-19. Basic graph for comparing life-cycle costs

Figure 4-20. Life-cycle cost results for parallel and series systems in New York, New York

Table 4-22

SYSTEM COST COMPARISONS FOR HEATING SEASON

Climatic Region	System	¢/kWh Competitive	Collector Area (m ²)	Seasonal Performance Factor	Initial Cost with Tax Credit (\$ in thousands)
New York, New York	Series \$ 5/ft ²	7.3	70	3.82	8.69
	\$10/ft ²	12.8	65	3.62	12.20
	Parallel \$ 5/ft ²	5.8	30	3.00	5.64
	\$10/ft ²	8.5	25	2.72	6.59
Fort Worth, Texas	Series \$ 5/ft ²	17.5	60	3.69	7.99
	\$10/ft ²	>20	--	--	--
	Parallel \$ 5/ft ²	9.3	25	3.47	5.51
	\$10/ft ²	14.0	20	3.28	6.18
Madison, Wisconsin	Series \$ 5/ft ²	8.3	90	3.12	10.57
	\$10/ft ²	14.4	85	3.08	15.31
	Parallel \$ 5/ft ²	8.0	60	2.67	7.74
	\$10/ft ²	14.0	55	2.60	10.66

Figure 4-21. Life-cycle cost results for parallel and series systems in Fort Worth, Texas

Figure 4-22. Life-cycle cost results for parallel and series systems in Madison, Wisconsin

Figure 4-23. Life-cycle cost results for parallel and series systems in Phoenix, Arizona

Figure 4-24. Life-cycle cost results for parallel and series systems for New York, New York; system utilizing GE TC-100 collector

Section 5

MARKETING ANALYSIS

This section addresses the market potential for SAHP systems from two different approaches. In the first approach, existing government and solar industry marketing studies are presented and analyzed. The second approach utilizes an inductive method in which the observed purchasing habits of consumers of central heat pump and air-conditioning systems are used to predict the market share that SAHP systems can expect to enjoy.

SURVEY OF EXISTING STUDIES

Figure 5-1 shows various independent estimates of the cumulative number of solar homes from 1977 to 1987. It can be seen that these estimates

Figure 5-1.
Estimates of cumulative
number of solar homes

vary by as much as 500 percent in the later years. The most optimistic estimate is that of the SEIA (Solar Energy Industry Association). However, since that estimate is consistently more than double the estimates of the DOE (Department of Energy) and other government agencies, it is felt that the SEIA estimate is likely to be too optimistic and hence was discounted. The average annual growth rate predicted for each study is also shown in Figure 5-1. It can be seen that the variation in the annual growth rates between the government studies, DOE, RUPI/GESD (Regional Urban Planning Institute/General Electric Space Division), CRS (Congressional Research Service), and OTA (Office of Technology Assessment) is only about 6 percentage points. The average of these annual growth rates is 31 percent, as compared with the 56 percent estimate by SEIA.

Each of the curves in Figure 5-1 shows the total number of solar installations, including those for domestic hot water, solar cooling, and space heating. The RUPI/GESD study includes a breakdown of solar systems into these three categories, which is presented in Table 5-1. Solar space heating and heating/cooling systems are expected to grow at an average rate of 54.4 percent per year, from 2700 systems in 1979 to 84 000 systems in 1987. It can also be seen that the percentage of these systems when compared to all solar systems (in any given year) rises from a mere 6 percent in 1979 to 28 percent in 1987 (Figures 5-1 and 5-2 and Table 5-2).

Table 5-1

RUPI/GESD MARKET FORECAST --
RESIDENTIAL SOLAR ONLY

	1979	1980	1981	1982	1983	1984	1985	1986	1987
Cumulative Solar Installations, x 10 ³	100	153	216	293	386	511	680	905	1205
Annual Solar Installations, x 10 ³	45	53	63	77	93	125	169	225	300
Solar Applications Installations, x 10 ³									
Domestic Hot Water	423 (94)*	49 (92)	57 (90)	67 (87)	79 (85)	100 (80)	128 (76)	167 (74)	216 (72)
Space Heating	2.25 (5)	3.55 (6.7)	4.4 (7)	7.5 (9.8)	10.2 (11)	19 (15.2)	30 (18)	45 (20)	63 (21)
Heating/Cooling	0.45 (1)	0.95 (1.8)	1.9 (3)	2.5 (3.2)	3.7 (4)	6 (4.8)	10 (6)	14 (6)	21 (7)
Heating and Heating/Cooling	2.7 (6)	4.5 (8.5)	6.3 (10)	10 (13)	13.9 (15)	25 (20)	40 (24)	59 (26)	84 (28)

*Numbers in parentheses indicate percent of total number of systems.

Table 5-2

RUPI/GESD SOLAR ENERGY MARKETING STUDY

Year	Cumulative Installations	Space Heating and Heating/Cooling Installations	Percent Increase Annually Cumulative	Percent Increase Annually Heating and Heating/Cooling
1979	100 000	2700		
1980	153 000	4500	53	67
1981	216 000	6300	41	40
1982	293 000	10 000	36	59
1983	386 000	13 900	32	39
1984	511 000	25 000	32	80
1985	680 000	40 000	33	60
1986	905 000	59 000	33	48
1987	1 205 000	84 000	33	42
		Average	36.6	54.4

Figure 5-2.

Estimated cumulative solar systems market. Source: RUPI/GESD study

An estimate of the SAHP market split among manufacturers is shown below:

<u>Manufacturer</u>	<u>Percent of Market</u>
A	25
B	10
C	10
D	8
E	8
All others	39

The letters A-E represent future SAHP system manufacturers whose market shares are typical of those of the manufacturers of HVAC equipment. Figure 5-3 and Table 5-3 show, using the data from the RUPI/GESD study, the expected cumulative SAHP systems sales for the five largest manufacturers, assuming that all of the residential solar heating and heating/cooling systems are SAHP systems.

Due to the wide variations in these predictions, and the fact that the purchasing habits of consumers were not considered in these studies, the following inductive approach was considered.

INDUCTIVE APPROACH

An inductive approach to determine the marketability of SAHP systems is a method where the purchasing habits of consumers of heat pumps are generalized to apply in the purchasing of SAHP systems. It is expected that

Figure 5-3. Cumulative SAHP systems market by manufacturer

Table 5-3

**CUMULATIVE SAHP SYSTEMS MARKET
BY MANUFACTURER**

Year	Total Systems	Systems By Manufacturer			
		A	B and C	D and E	Other
1979	2700	675	270	216	1053
1980	4500	1125	450	360	1755
1981	5300	1575	630	504	2457
1982	10 000	2500	1000	800	3900
1983	13 900	3475	1390	1112	5421
1984	25 000	6250	2500	2000	9750
1985	40 000	10 000	4000	3200	15 600
1986	59 000	14 750	5900	4720	23 010
1987	84 000	21 000	8400	6720	32 760

the same factors that influence the decision to purchase a particular heat pump system will be important factors in the purchase of a SAHP system; hence, the heat pump consumer habits are used as a base.

Some factors that influence the purchasers of heat pumps are:

- Initial cost
- Years to payback
- Life-cycle cost
- Energy savings

The relationship between some or all of these factors may influence the consumers' decision as to whether or not to purchase a more expensive, more efficient model over a less expensive model of standard efficiency. Once the important factors and their relationships are determined and the total annual heating market is known, it will be possible to calculate the annual market for any SAHP system (series or parallel), if the cost and energy efficiency of these systems is known.

It has been the General Electric Company's experience with the sale of heat pumps and other HVAC equipment that the most important factor which influences the prospective buyer is the payback period. It should be noted that energy savings affect the payback period directly. A system that is energy efficient but has a high initial cost and a long payback period will not be attractive to consumers.

Life-cycle cost, while being absolute in its ability to determine the least expensive system and being effective as a means to market conventional heat pumps and vent dampers for gas-fired warm-air furnaces, is not likely to influence a homeowner's decision to purchase a SAHP system. Some of the reasons for this are:

1. The most cost-effective SAHP system has a payback of more than four times that of a regular air-to-air heat pump when both are compared to an electric-resistance heating system.
2. The longer the payback period, the greater the reluctance of the consumer to invest in a capital venture. This is true because he will have displaced what previously may have been discretionary income into a very nonliquid form.
3. The current federal tax credit to homeowners offers the best return on investment to a "solar system" with an installed cost of \$2000. This will probably offer the best opportunity to those who market solar-assisted domestic hot water systems. Since this represents as much as 90 percent of the solar-panel market today, it can only help to augment this market while offering very little incentive for SAHP.
4. Until the manufacturers have developed an approach of packaging components for SAHP systems into easily installed standard models, the installation cost cannot approach an optimum. The modular concept would save labor on the job site and greatly reduce the "fear factor" of the installer, who currently offsets this factor by adding contingency dollars to his price. In order for a manufacturer to take the modular approach, some viable market or demand which he can serve must exist.

Figure 5-4 is an estimate based on marketing experience of the relationship between the fraction of consumers who prefer a more expensive heat pump of higher efficiency (over a standard heat pump) and the payback period of that higher quality unit. Since this is a subjective relationship, a range is shown rather than a line. As can be seen, the fraction of converted consumers drops rapidly as the payback period increases; after 5 years, less than 20 percent of the prospective consumers will purchase the more expensive heat pump. As expected, as the payback period goes to zero, all of the consumers will choose the better unit. The curve possesses this shape because:

- The potential for technological obsolescence increases drastically with time.

Figure 5-4.

Payback vs. percent of market

- As the payback period increases, the time to recover what was formerly discretionary income increases; thus, the consumer will have to "do without" for a longer period.
- Since most studies indicate that the average family relocates every 5 years, systems which have payback periods in excess of this will be very difficult to sell.

The following exercise is used to predict the residential solar-assisted heat pump system market in 1985. Given that:

IC_B = Initial base system cost, \$

IC = Initial cost of candidate system, \$

e = Annual energy savings of candidate system over base system, kWh/year

s_n = Electricity cost during n^{th} year, \$/kWh

m = Years payback

Then

$$IC - IC_B = S_1 e + S_2 e + \dots S_m e$$

where

$$S_n = S_1 (1+i)^{n-1}$$

and

S_1 = First year electricity cost

i = Electricity cost annual escalation rate

Using the above equations

$$\frac{IC - IC_B}{e} = S_1 (1+i)^0 + S_1 (1+i)^1 + \dots + S_1 (1+i)^{m-1}$$

or

$$\frac{\Delta IC}{S_1 e} = K = 1 + (1+i) + \dots + (1+i)^{m-1}$$

Figure 5-5 shows the years of payback m versus the quantity K for electricity cost escalation rates of 0 and 8 percent.

Figure 5-5.

Payback vs. quantity for electricity cost escalation

If the information presented in Figure 5-4 is combined with the information shown in Figure 5-5 for various incremental initial costs and annual energy savings, the plots shown in Figure 5-6 can be produced. These plots were constructed assuming an energy cost escalation rate of 8 percent. Hence, for any candidate SAHP system (when compared to a base system, e.g., a two-speed heat pump), if the first year energy savings and the incremental initial cost is known, the fraction of consumers who will choose the SAHP system over the two-speed heat pump system can be determined.

Figure 5-6. Incremental initial costs and annual energy savings

The national single-family new construction rate for the next 5 years as well as the fraction of new residences that will utilize heat pumps have been estimated in the past. Using these data as a base, the total number of series and parallel SAHP systems that could be sold can be calculated. For this calculation, the base system (a single-speed heat pump) is assumed to have an initial cost of \$4735 (in 1985) and an SPF of 2.1. The annual house heating load is assumed to be 13 720 kWh, as obtained from the New York City simulation. A comparison of both the series and parallel SAHP systems, using the results of the SAHP simulations for New York City and the aforementioned base system, is shown in Table 5-4.

Each of the systems represented in Table 5-4 possesses a payback period of 22 to 25 years, hence only about 10 percent of the single-speed heat pump system consumers would opt to purchase a SAHP system. Using the estimated single new-family construction rate, it is found that this translates into a sale of approximately 4800 SAHP systems in 1985. Thus, the market for SAHP systems is very small when compared with the rest of the residential new construction market.

Table 5-4

**MARKETING COMPARISON OF SERIES
AND PARALLEL SAHP SYSTEMS
(Based on New York, New York Simulation)**

Base System (Single-speed, air-to-air heat pump)						
Seasonal Performance Factor					2.1	
Initial Cost (no tax credit)					\$4735 in 1985	
1985 Electricity Cost					6¢/kWh (3.5¢ at 8 percent escalation)	
Annual House Load					13 720 kWh	
Base System First Year Electricity Cost					\$392	
Energy Escalation Rate					8 percent	
General Inflation Rate					6 percent	
Collector Cost \$5/ft ² in 1978 escalated at 6%					\$7.50/ft ² in 1985	
System	Collector Area (m ²)	Initial Cost 1985	Seasonal Performance Factor	Energy Cost 1985	First Year Energy Savings 1985	Payback Years at 8 Percent Escalation
Series	50	\$13 580	3.05	\$270	\$122	25.0
	75	16 950	4.12	200	192	23.3
	100	19 760	4.91	168	224	24.3
Parallel	25	10 500	2.78	296	96	23.0
	50	13 330	3.35	246	146	22.5
	75	16 540	3.85	214	178	24.2
	100	19 360	4.26	193	199	25.0

Other factors which will inhibit or restrict the SAHP market are:

- Uncertainty of future fuel costs
- Structural limitations for collector location are related to size and orientation
- Tax incentives are not proportionate to incremental costs
- Lack of solar industry standards hurts credibility of product with the consumer
- Lack of solar regulation regarding protection of solar access is not widely in place

Section 6

SUMMARY AND CONCLUSIONS

A study that is to specify the "best" solar-assisted heat pump for energy conservation must necessarily ask and answer the question: What national energy savings result from the market introduction of a candidate system? The work performed to date, and documented in this report, has dealt exclusively with this question.

A candidate system's national energy savings potential depends not only on its performance but also on the number of installations and, therefore, on its cost and on the performance and costs of already, or soon to be, commercially available alternative and competing heating systems. To address these questions, the following work was carried out under Phase I:

- Designed candidate SAHP systems
- Established component performance models and estimated component performance
- Estimated component costs
- Optimized component sizes and types for a balance of performance and cost
- Optimized system control strategies
- Estimated system seasonal energy use for installations in four climate zones
- Estimated system installed consumer costs
- Estimated costing parameters such as loan interest rates, future costs of energy, consumer discount rates, tax effects
- Computed SAHP system life-cycle costs
- Computed seasonal energy use and first installed costs of alternative heating systems: air-to-air heat pumps, straight solar systems, solar-boosted heat pumps
- Computed life-cycle costs of alternative heating systems
- Compared all of the systems on the life-cycle cost basis while varying those parameters which are least known and which have the greatest effect on life-cycle cost: e.g., future cost of electrical energy, solar collector installed cost
- Performed a market study to estimate the number of solar-assisted heat pumps which are likely to be sold

The results of this work led to the following observations:

- For optimum performance, heat pumps for SAHP systems should have capacity modulation.
- Currently used refrigerants (R-22, R-12) for air-to-air heat pumps and air conditioners are suitable for use in solar-assisted heat pumps.
- Heat pumps which are to serve SAHP systems will operate at higher evaporator temperatures than air-to-air heat pumps; condenser temperatures are similar.
- No basic technological developments are required for the fabrication of heat pumps which are to serve SAHP (series) systems.
- Inexpensive, low performance solar collectors are more cost effective than high performance collectors for solar-assisted (series) and solar-boosted (parallel) heat pump systems.
- Control strategy affects system efficiency dramatically; simultaneous operation of a water-to-air heat exchanger and a heat pump condenser at some storage temperatures will improve system seasonal efficiency by approximately 15 percent.
- Residential and multifamily dwellings are economically more attractive applications for solar-assisted heat pumps than commercial buildings.
- Solar-assisted heat pump systems require larger collectors and are more efficient than solar-boosted heat pump systems at their respective economic optimum sizes.
- Solar-assisted heat pump systems are less efficient in their air conditioning mode of operation than air-to-air heat pumps.
- Solar-assisted and solar-boosted heat pumps are more cost effective than straight solar systems.
- Series solar-assisted heat pumps are more cost effective than multi-coil or cascade systems.
- Solar-boosted heat pumps are more cost effective than solar-assisted heat pumps.
- Direct solar or combined solar/heat pump systems are not cost competitive with commercially available air-to-air heat pumps, nor is there evidence that they will be in the foreseeable future.

Appendix A

HEAT PUMP COSTS

The final price to the ultimate consumer for a heating system depends on many factors, such as the cost of equipment, the amount and skill of the installing labor, the required sales effort, the remoteness and concentration of the market, the amount of customer training, and the service and repair to name a few of the important ones. In addition, there are time-related trends that influence the price, e.g., design trends, social customs, inflation, and political regulation. A brief statement on some of the more tangible factors is made here to provide a background for heat pump costs.

RELATIONSHIP OF PRICE TO COST

In a free, competitive market the customer's price bears an established empirical relationship to the manufacturing cost. The multiplying factor is strongly influenced by the sales density. Typical factors run from 2 for over the counter goods to over 3 for installed equipment.

COST VS. VOLUME

Only minor cost improvements can be expected in the air-conditioning industry due to increases in volume. The industry is both mature and competitive. In 1978, unitary shipments for residential sizes was 2 600 000 units. This is sufficient volume for the half-dozen major manufacturers to have mass production techniques and automation as extensively as is likely to occur in the near future. This means that very little reduction in manufacturing costs can be expected from just increased production only.

COST VS. INFLATION

Although prices may exhibit short-term variations, the cost of manufactured products are tied directly to factors that are controlled by inflation, namely labor, materials, and overhead. Therefore, in the long run costs will reflect exactly the rate of inflation.

COST VS. PERFORMANCE TRENDS

The cost of a heat pump depends largely on the level of performance desired, which in turn depends on the economics of heating and cooling vs. the value of the comfort received. Although no one can say "Utopia" will not be discovered tomorrow, it can be observed that no major technological break throughs have been made in the heat pump business in the last 45 years. This is largely because the basic laws of nature governing thermodynamics in this field have been pretty well understood for 100 years. Looking at the

trend in heat pump performance over the past 45 years shows that the efficiency of the product follows the customer's concern over the cost of power.

Figure A-1 shows the trend with time of the SPF of General Electric heat pumps together with an average price to the customer of electricity expressed in 1977 dollars. It shows that where there is concern over power costs, consumers will pay more for higher efficiency and vice versa.

Figure A-1. a) Heat pump efficiency; and b) Price of electricity in 1977 dollars

EQUIPMENT PRICING BY COMPONENTS

In an established business with a mature product, the price is established based on the product cost and other considerations. The product cost in turn is related to the manufacturing cost which is basically labor, materials, and overhead. When starting with a new design, the final price is arrived at in inverse order. That is, all of the material, the labor, the variable overhead for all of the components, the unscheduled materials

(paint, a squirt of oil here and there, temporary ties, etc.), warehousing, shipping, warranty, etc. are all added together to form the manufacturing cost. Then the product cost is determined by adding margins to the manufacturing cost for development, risks, and profit, after which a further markup in the distribution chain establishes the consumers price.

In this study, which requires the prices of several sizes and arrangements of new equipment, the approach selected was to start with the price of an established piece of equipment similar to the new design required and then modify the established price by the price difference of the components that had to be changed. In this way many small items and costs associated with a product would not be inadvertently lost as sometimes happens when starting the costing procedure from scratch.

To accomplish this, the equipment was divided into its major components which would be either replaced or varied in size relative to other components. The price of each of the major components was then established and plotted as a function of the major controlling variable of its size.

COMPRESSOR PRICE TRENDS

The price of a motor-compressor is a strong function of its horsepower; with the volumetric displacement of the compressor being of secondary importance. This is because the motor accounts for over half the cost, and its price varies directly with the power.

In any series of mechanical designs for a compressor, changes in the displacement are obtained by varying the stroke. This is only a minor cost differential. For a given application such as an air-to-air heat pump, the compressor displacement and the horsepower are related so that either can be used as a pricing parameter. However, when a change is made to a water-to-air heat pump, especially where the water temperatures are beyond the normal limits, the displacement and the horsepower have a different relation. The displacement will be smaller due to the higher density of the suction gas, while the power required will remain about the same. Therefore, the price of the new series of compressors will be the same function of horsepower but not of displacement. Although a few changes in the mechanical design of the compressor will be required to handle the wider range of water temperatures, this should not materially effect the price.

WARRANTY PRICE TREND

The cost of the warranty is included in the price of the product and depends not only on the reliability of the product but on the cost of the warranty repairs, as well as how liberal the company is in including

near-warranty problems. For heat pumps, the warranty costs increase with the size of the equipment. Since the compressor is by far the biggest cost factor, the total system warranty costs correlate well with the compressor horsepower.

INDOOR AIR-HANDLER PRICE TREND

The function and application of the indoor air-handler does not change much between the different systems considered. The price of this component can be well correlated to its capacity rating.

OUTDOOR AIR-TO-AIR UNIT COMPONENT PRICE **(less compressor and warranty)**

The price of the basic outdoor section of a split heat pump less the compressor, the compressor electrical components, and the warranty will vary with the system capacity.

WATER-TO-AIR UNIT COMPONENT PRICE **(less compressor and warranty)**

Although the prices for standard water-source heat pumps are available, they are not suitable for this study because the standard water-source heat pump is not usable over the range of water temperatures postulated for the solar-assisted series system. It was necessary to divide the system into known cost components, substitute where required to get a workable system, and then reconstitute the price.

Figure A-2 represents the results of applying the heat pump component pricing trends discussed previously. The heat pump installed costs to the general contractor are given as a function of the type and size of the heat pump. A contractor markup of 12 percent would be applied to determine the installed consumer cost. Figure A-2 does not include controls, ducting, or start-up charges.

Figure A-2. Heat pump component pricing trends