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Annual Cycle Energy System (ACES) Performance Report, November 1977 Through September 1978

A. S. Holman
L. A. Abbatiello

Conservation

Division of Buildings and Community Systems

OPERATED BY
UNION CARBIDE CORPORATION
FOR THE UNITED STATES
DEPARTMENT OF ENERGY

ORNL/CON-42

Contract No. W-7405-eng-26

Energy Division

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Date Published: May 1980

Prepared for the
U.S. Department of Energy
Division of Buildings and Community Systems

Prepared by the
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
operated by
UNION CARBIDE CORPORATION
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DEPARTMENT OF ENERGY

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ANNUAL CYCLE ENERGY SYSTEM (ACES) PERFORMANCE REPORT, NOVEMBER 1977 THROUGH SEPTEMBER 1978

A. S. Holman
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ABSTRACT

A single-family residence near Knoxville, Tennessee, is being used to demonstrate the energy-conserving features of the annual cycle energy system (ACES), an integrated heating and cooling system that utilizes a unidirectional heat pump and low-temperature thermal storage. A second house, the control house, is being used to compare the performance of the ACES with that of an electric-resistance heating and hot-water system combined with a central air conditioning system. The results of one year's operation, from November 1977 through mid-September 1978, showed that the ACES consumed 9012 kWhr of electricity and delivered 40.8×10^6 Btu (43.03×10^9 J) of heating, 19.8×10^6 Btu (20.89×10^9 J) of hot water, and 24.8×10^6 Btu (26.17×10^9 J) of cooling; the annual coefficient of performance (COP) was 2.78. The control house consumed 20,523 kWhr of electricity and delivered 41.3×10^6 Btu (43.57×10^9 J) of heating, 14.8×10^6 Btu (15.61×10^9 J) of hot water, and 23.2×10^6 Btu (24.41×10^9 J) of cooling; the annual COP was 1.13.

These loads were delivered in a test year in which the heating season was one of the most severe in the past 20 years and the cooling season was normal. In addition, the ACES reduced peak utility system demands significantly: a reduction from 11.7 to 3.1 kW was achieved in the winter and from 4.1 to 0.7 kW in the summer. The only problems encountered were a heat leak into the storage bin that was twice the calculated value and control logic errors that produced excessive hot water in the winter, requiring extensive use of the night heat-rejection mode in the summer. These problems are currently being corrected.

1. SUMMARY

A three-house complex in Knoxville, Tennessee, is being used to demonstrate various methods of energy conservation. The three houses are the annual cycle energy system (ACES) house, the solar house, and the control house. The solar house is not discussed extensively in this report.

The main feature of the ACES house is the annual cycle energy system (ACES). The ACES is an integrated heating and cooling system that employs a unidirectional heat pump, low-temperature thermal storage in the form of an ice-water storage bin, and a solar-convector panel that serves as either an external heat source or a heat sink. The ACES is also integrated in time, because thermal energy is transferred between the winter and summer months by the thermal-storage bin. During the winter, heat is removed from the bin and water is converted to ice as a by-product of the heating requirements. During the summer, the ice is used for cooling, thereby putting heat into the bin and completing the energy-transfer cycle.

The primary use of the control house is for comparison with the ACES and the solar houses. During the test year, the control house used an electric-resistance heating and hot-water system combined with a central air conditioner for cooling. The floor plans, thermal envelopes, and environmental exposures for the control and ACES houses are nearly identical. Neither house was occupied during the test year; however, internal loads for a family of 4 were simulated. Each day, 70 gal (0.26 m^3) of hot water was automatically consumed on a schedule, and 24 kWhr of electricity per day was used by internal equipment. Occupancy was not simulated, but doors were opened and closed by people working at the house. Total seasonal heating loads for the two houses were within 2% of each other. Based on this fact, the cooling loads were also assumed to be equal. Future plans include operating with an air-to-air heat pump in the control house to allow for direct comparison between ACES and conventional heat-pump systems and instrumenting the heat pump to determine whether the cooling loads are indeed equal. The solar house uses a solar heating and hot-water system.

This report covers the performance of the ACES during a one-year cycle from November 1977 to mid-September 1978. All components of the system operated admirably during this time; the system was down only two days. The winter weather for the year was very cold; summer weather was normal. The heating season degree days averaged 15% greater than normal, but two months (January and February) were the coldest in the last 20 years.

Data collected during the year showed an overall coefficient of performance (COP) of 2.78 for the delivery of 40.8×10^6 Btu (43.05×10^9 J) of heating, 24.8×10^6 Btu (26.17×10^9 J) of cooling, and 19.8×10^6 Btu (20.89×10^9 J) of hot water; 9012 kWhr of electricity was consumed. The COP for the system that combined an electric-resistance heating and hot-water system with central air conditioning was 1.13 for the delivery of 41.3×10^6 Btu (43.57×10^9 J) of heating, 23.2×10^6 Btu (24.48×10^9 J) of cooling, and 14.8×10^6 Btu (15.61×10^9 J) of hot water; 20,523 kWhr of electricity was consumed.

In addition to the savings in energy consumption, there was typically a 60% savings in the hourly integrated peak load on the utility in winter and summer operation; maximum peak-load reductions reached 83%.

The results of the year's operation revealed two aspects of the system that should be improved: (1) The compressor that was installed originally was sized to meet the house design day loads down to 17° F (−8.3° C), and the unit was capable of this. However, the number of hours below 17° F (−8.3° C) was considered sufficiently large to justify the installation of a compressor capable of carrying increased loads. (2) The other major item was the heat leak into the ice-water storage bin. Insulation for the bin was chosen on a predicted heat leak that was only about 50% of the actual heat leak. This higher heat leakage resulted in earlier than expected exhaustion of the ice supply and the low COP encountered during the night heat-rejection mode. To reduce the heat leakage, the bin was reinsulated.

In addition to the hardware modifications, two errors in the logic matrix were detected and corrected: (1) The strategy of producing hot water when space heating occurred resulted in excessive amounts of hot water which reduced the maximum amount of heating that could be delivered. This error was corrected by producing hot water only when it was required. (2) The second logic error was concerned with night heat rejection. The original logic called for this mode when the ice-water bin temperature exceeded 36° F (2° C) and the panel temperature was less than 80° F (27° C), but excessive amounts of cooling were wasted because of the bin's heat leak. By raising the initiating temperature to 44° F (7° C), the temperature difference between the bin and the earth was reduced and the heat leak and energy consumption were significantly reduced. The hardware items were corrected between mid-September and November 1978; we will report the operational results in future publications.

2. INTRODUCTION

The ACES program is an ongoing research, development, and demonstration (RD&D) effort sponsored by the U.S. Department of Energy (DOE) at the Oak Ridge National Laboratory (ORNL). The primary purpose of the ACES program is to study, develop, and evaluate the ACES engineering performance and its commercial viability.

The ACES concept was first brought to the attention of ORNL late in 1974 by Harry C. Fischer. Initial investigations of the system were carried out in a laboratory with small-scale equipment. These investigations[1] showed no major flaw in the concept; thus, a full-scale experiment was begun. For the experiment, a demonstration house[2] was constructed near Knoxville, Tennessee.

The purpose of the demonstration house is for studying the operational characteristics of the ACES in a residential application. This testing has provided a lot of information on the system and its components. The information has resulted in improvements in performance and reliability of the control system and equipment.

In addition to this performance report, reports have been issued on the shakedown and initial start-up period[3] and on the first winter's operation.[4]

3. ACES CONCEPT

The most fundamental principle of the ACES concept is that operation of a heat pump generates both heating and cooling. In most heat-pump applications, only one of these functions is utilized, but the ACES stores the cooling energy produced when heating is performed. Thus, both outputs of the heat pump are used.

It is, of course, possible to store either the heating or cooling energy. There are, however, two major advantages to storing the cooling energy. First and foremost is the fact that there exist readily available and very inexpensive storage media for saving the cooling energy—ice and water. The phase change of water occurs at a convenient temperature and the heat content per pound is high. Second, this country's population is primarily located in a heating zone, which means that generally it is possible to store all the annual cooling requirements from the heating demands available, but the reverse is not true.

Although the name ACES properly indicates annual cycle operations, the term has come to imply any heat-pump-based system that employs ice-water storage to accomplish interseasonal transfer of energy while maintaining constant-capacity heat-pump operation during the heating season. "ACES" is used, therefore, to designate systems ranging from a true annual cycle, such as the one discussed in this report, to small-bin storage systems capable of offering constant capacity heat-pump operation through two weeks of the coldest month without resorting to a supplemental energy source such as solar.

The major difference between annual cycle energy systems is the amount of interseasonal energy transfer accomplished. Interseasonal energy transfer is reduced and energy consumption increased if the size of the storage bin is decreased so that the cost is reduced. Additional expense is required to increase the size of the solar-convector panel (or any substituted heat source or heat sink) to ensure the heating and/or cooling capabilities. The basic features, components, and control systems are not affected by the amount of interseasonal energy transfer; thus, the choice of system size is based on economics. The primary economic influences are climate, component costs, electricity rates, and interest rates.

The following sections describe the major advantages of using the ACES concept, the equipment necessary to implement the concept, and the various control strategies that can be employed.

3.1 Features

The ACES has three major features that set it apart from the typical heat-pump installation: (1) a constant-capacity heat pump, (2) interseasonal energy transfer, and (3) off-peak energy use. The relative importance of these features varies according to the extent of interseasonal energy transfer being achieved; however, these features are common to all types of ACES.

3.1.1 Constant-capacity heat pump

Figure 3.1 shows the output for the ACES and the air-source heat pump vs outside temperature; also shown are the required structural heating loads. As the temperature decreases, the heating load for the structure increases and the air-source heat pump's output decreases. At some temperature, indicated by point A, the heating load and the air-source heat pump's output are equal. At temperatures below this point, the heat pump requires a supplemental heating source (shown by the shaded area) to continue to supply heating requirements. Because a supplemental heating source is not always required, it is usually supplied by low first cost electric-resistance heating elements. These heating elements, however, have a high operational cost. Because the output of ACES is independent of outside

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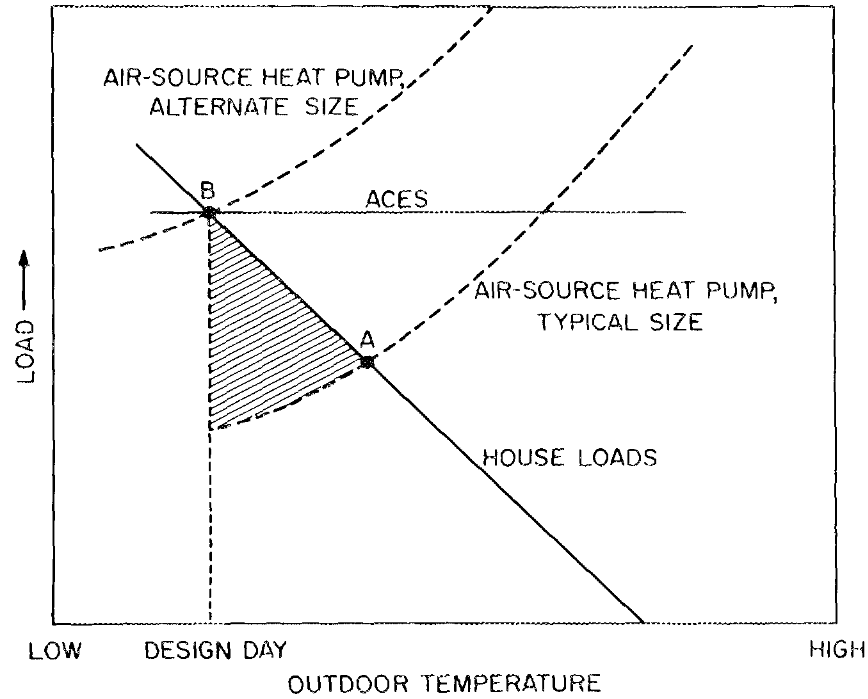

Fig. 3.1. Heat pump and ACES output vs temperature.

temperature, the system may be sized to meet the load at any given temperature and maintain the minimum amount of overcapacity at higher temperatures (point B, Fig. 3.1). If a typical heat pump were sized to meet the load at point B, its capacity would be higher for lesser loads and would, therefore, suffer larger cyclic losses than the ACES; it would also be improperly sized for cooling service. Sizing the ACES to carry the design day load removes the need for supplemental energy, thereby reducing the utility's peak load and the total energy consumption.

3.1.2 Interseasonal energy transfer

The second major feature of the ACES is that of interseasonal energy transfer. Because the cooling energy is saved when heating is performed, some portion of this stored energy can be later used to satisfy the structure's cooling requirements. Cooling generated in the winter can be transferred in time, by the storage bin, to the summer when it is needed. By meeting part or all of the cooling requirements with stored ice, the total annual consumption of energy is less than that of a system that must generate the cooling requirements instantaneously. Peak loads are also reduced because it is not necessary to operate a heat pump to provide cooling.

3.1.3 Off-peak energy use

Two ways in which ACES reduces peak loads have already been mentioned: through its constant capacity during heating and through the use of stored ice for cooling. It is also possible to reduce the peak load during the cooling season when the ice supply has been exhausted. The storage device can be chilled by operating the compressor during off-peak hours, and the cooling output can be saved for use during later peak-load periods. This technique does not reduce the compressor's peak load; however, it does shift the peak load to a time that is more desirable to the utility. The utility's peak load is thereby reduced during those periods when the utility is least able to supply the energy required.

3.2 Major Components

3.2.1 Ice-water storage bin

The primary difference between ACES and a typical air-source heat pump is the ice-water thermal storage bin. The bin serves as a low-temperature heat source for the heat pump when heating is required and as a storage device for the cooling output created during heating modes. This function is accomplished by utilizing the latent heat of fusion of water as it is converted to ice and then back to water.

3.2.2 Solar-convector panel

The solar-convector panel is the thermal-energy-balancing mechanism for the ACES. Any device capable of collecting and rejecting heat could be substituted for the panel; however, we assume that a panel is being used.

The purpose of the panel is twofold. Whenever the ice-water storage bin becomes full, the panel collects heat from the sun and from the ambient air, if possible, and uses this heat to melt some of the ice. This process ensures an adequate supply of water in the bin to meet future heating requirements. The alternate function of the panel is to reject heat from the heat pump when the ice supply has been exhausted and future cooling requirements are anticipated. The importance of the panel varies with the type of ACES installed; its function is determined by whether the local climate requires a heating or a cooling application.

3.2.3 Mechanical package

For the purposes of this discussion, the mechanical package is taken to be all the mechanical equipment associated with the ACES and the control system required to operate it. The major components of the mechanical package are the compressor, the various pumps, the fan, the refrigerant-to-brine heat exchangers, and the control system.

The function of the mechanical package is to transfer heat between the major external components: the indoor fan coil, the ice-water storage bin, the solar-convector panel, and, for this application, the domestic-hot-water tank. Transfer is done on command from the control system, which is responsible for determining which function(s) the ACES should be performing. Energy transfer between components is accomplished with a flowing heat-exchange medium, a methanol-water brine.

3.3 Control System

Operation of the ACES requires a control system capable of sensing various conditions and placing the ACES in the mode suited to the condition.

Currently, a programmable logic controller performs this function. This device was chosen because it provides the necessary flexibility to alter the logic. Once the logic is firmly established, a fixed controller can be designed to operate the ACES.

The input signals upon which the controller makes its decisions are:

1. house thermostat call for heating;
2. house thermostat call for cooling;

3. ice-bin ice inventory;
4. ice-bin water temperature;
5. hot-water thermostat call for water heating;
6. season indicator, status of last call by house thermostat (heating or cooling);
7. solar-panel temperature (no brine circulation).

The exact method for determining which mode to initiate is discussed in the following section.

3.4 Modes of Operation

In the following sections we discuss the most common modes of operation for the ACES and the control logic necessary to invoke them. Other ACES applications, especially large-building applications, may require modes not discussed in this report.

3.4.1 Primary modes

The ACES discussed in this report has three primary modes of operation: (1) space heating, (2) space cooling, and (3) domestic water heating. These modes are shown schematically in Figs. 3.2, 3.3, and 3.4, respectively.

In the space-heating and domestic-water-heating modes, heat is extracted from the storage bin, and that heat, along with the heat of compression, is used to heat the house and/or the domestic hot water. When heat is extracted from the storage bin, some of the water is converted to ice, which is stored for future cooling needs. Space heating is supplied when the house thermostat signals a need for heating. This action also resets the season indicator for the winter condition. Domestic hot water is produced when the compressor operates or when the hot-water thermostat calls for heat. Hot-water production has no effect on the season indicator.

The other primary mode of operation is space cooling. Cooling is performed by circulating brine between the fan coil and the storage bin, thereby melting some ice and using the heat absorption to cool the structure. Space cooling is provided when the house thermostat signals a need for cooling. This mode results in the season indicator being set for summer operation.

3.4.2 Balancing modes

In addition to the primary modes, there are two modes provided to keep the storage bin in thermal equilibrium—ice melting and night heat rejection (shown in Figs. 3.5 and 3.6, respectively). These two modes are required because it is unlikely that the bin capacity, ice production, and cooling requirements will be equal in any given year.

When the storage bin becomes full, its inventory must be reduced. To accomplish this, brine is circulated between the storage bin and the solar-convector panel; the heat collected by the panel is used to melt the ice. The ice-melting mode is invoked when three conditions simultaneously exist: (1) the ice bin is full, (2) the inactive solar panel temperature is 10° F (6° C) warmer than the bin, and (3) the season indicator shows "winter."

The night heat-rejection mode is used when the ice supply is exhausted before the end of the cooling season. In this mode, heat is extracted from the storage bin and, together with the heat of compression, is used to heat the domestic hot water. Any heat not required by the hot water is dissipated through the solar-convector panel. Heat is extracted from the bin, and cooling is produced and saved for later use. This mode is invoked if (1) the ice supply is exhausted, (2) the inactive panel temperature is less than 80° F (27° C), or (3) the season indicator signals "summer."

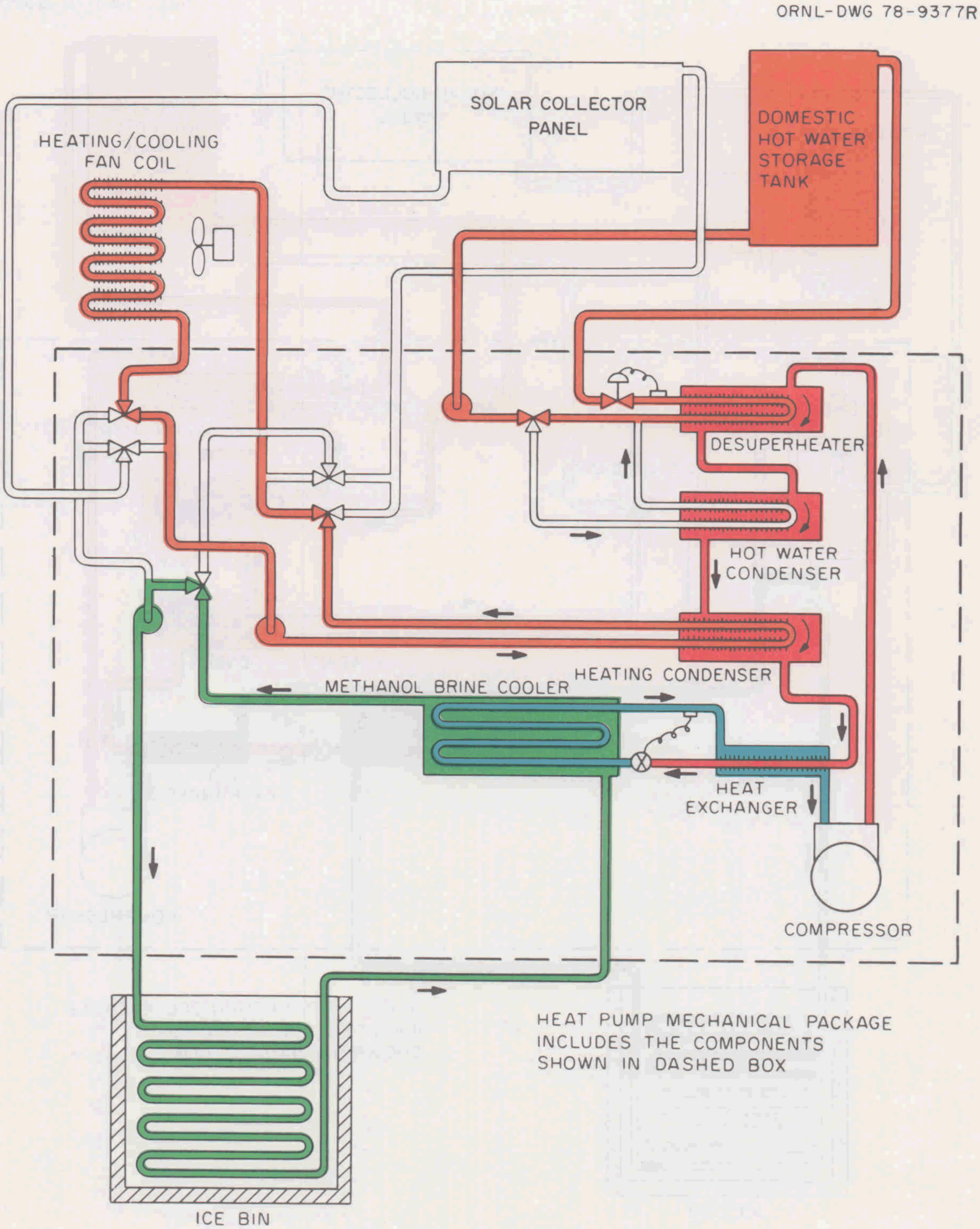

Fig. 3.2. Space-heating schematic.

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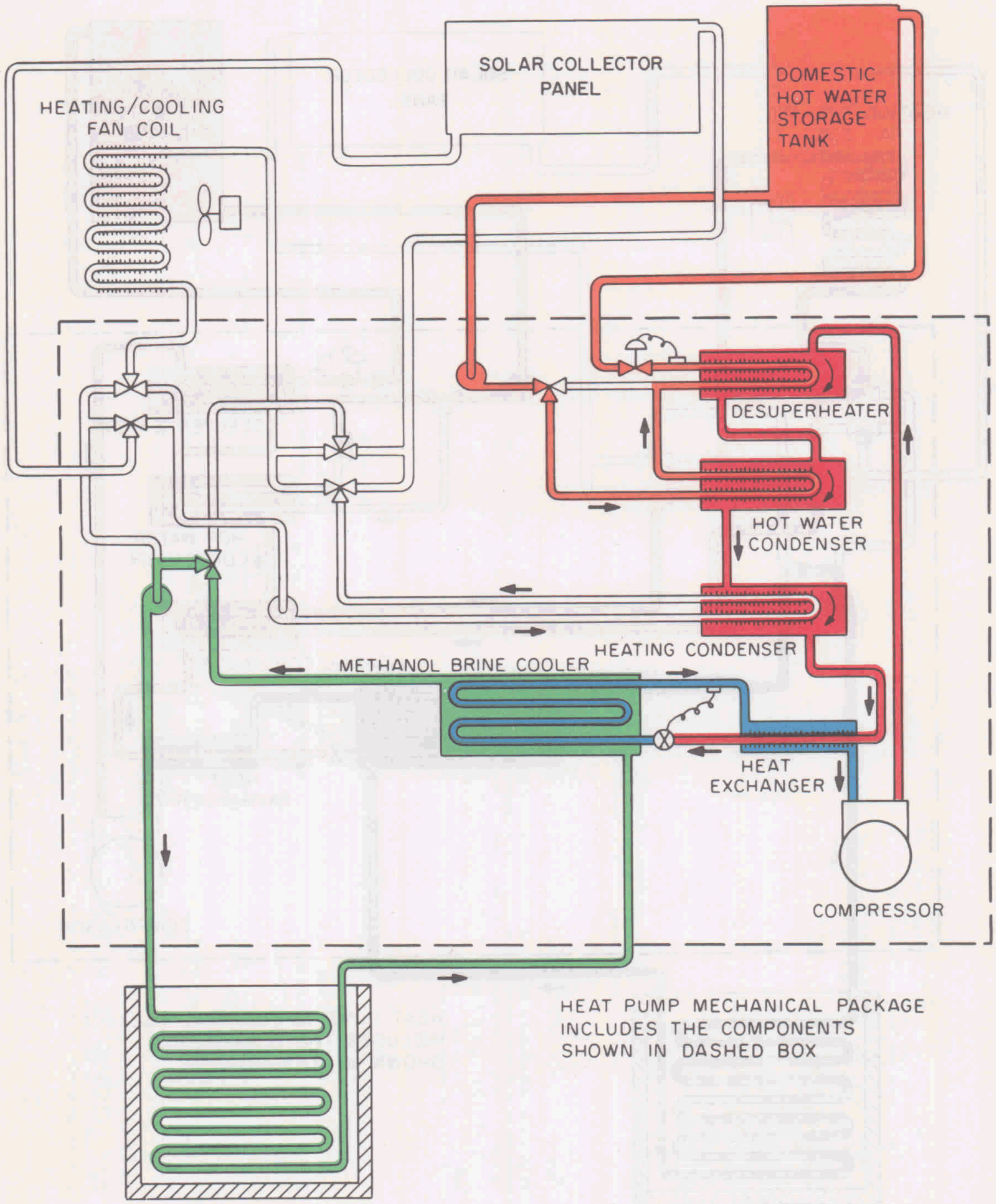

Fig. 3.3. Domestic hot-water schematic.

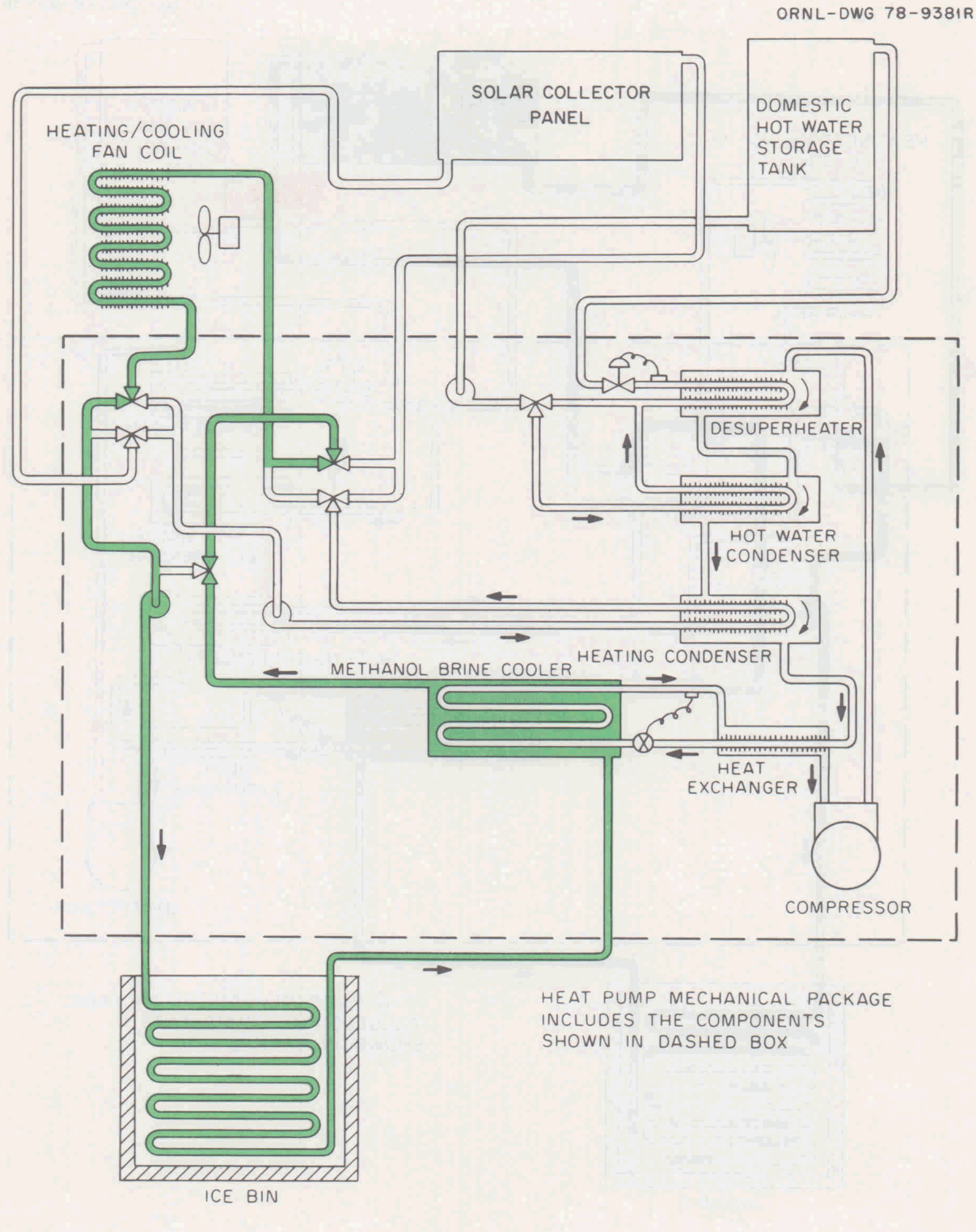

Fig. 3.4. Space-cooling schematic.

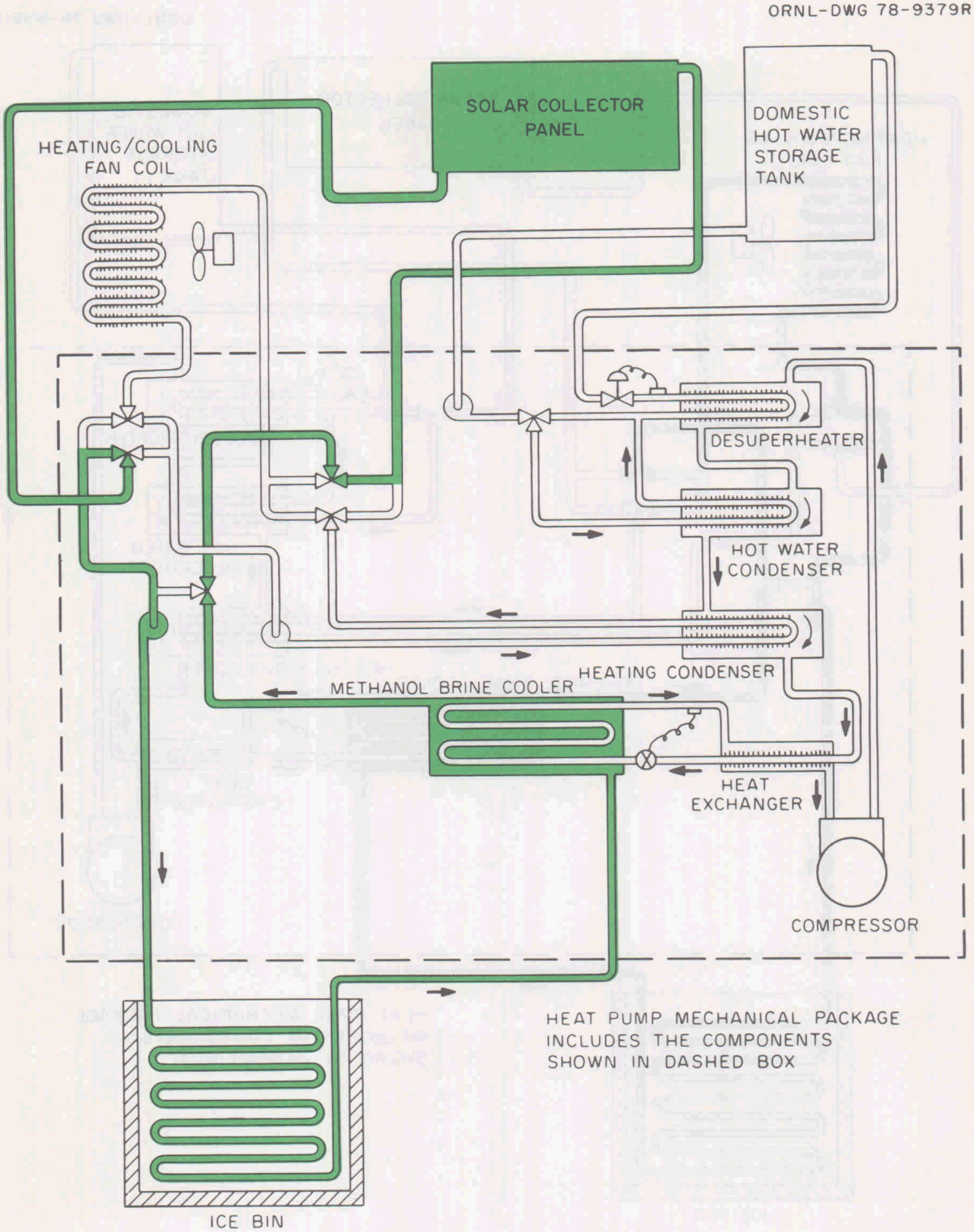

Fig. 3.5. Ice-melt schematic.

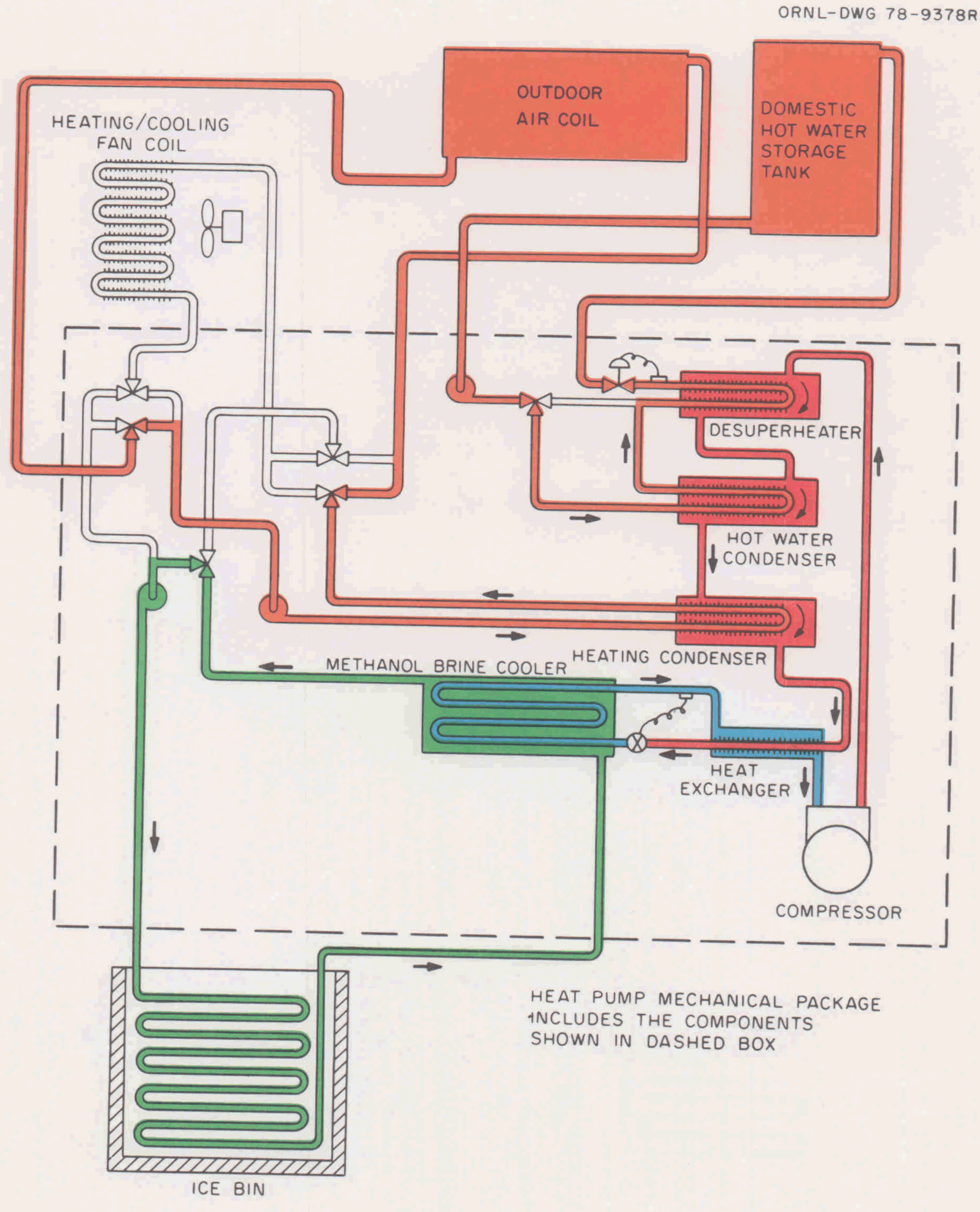

Fig. 3.6. Night heat-rejection schematic.

4. KNOXVILLE DEMONSTRATION

4.1 TECH Complex

To demonstrate the ACES concept, a test house was built near Knoxville, Tennessee. This house is part of a three-house complex called the Tennessee Energy Conservation in Housing (TECH) Project. The purpose of the project is to demonstrate energy conservation through improved thermal envelopes and through the use of innovative heating and cooling systems. The complex comprises the ACES house, in which the annual cycle energy system is used; the control house, in which a conventional heating and cooling system is used; and the solar house, in which a solar-heating and hot-water system is used.

The three-house complex, located on the agricultural farm of the University of Tennessee, south of Knoxville on Alcoa Highway, is a joint effort of ORNL, the University of Tennessee (UT), and the Tennessee Valley Authority (TVA), along with the cooperation of local industries. The complex has been operational since August 1976; experiments are being performed to evaluate the performance of the ACES and the solar-heating system, as well as the performance of the thermal envelopes.

4.2 Demonstration House

4.2.1 Construction

All three houses in the complex have similar floor plans; their exteriors are patterned after the requirements of the solar house. The ACES house is an 1800-ft^2 (167-m^3) single-family residence consisting of three bedrooms, two baths, a great room, a kitchen, a mechanical equipment room, and a partial basement. The thermal-storage bin is located under the northeast corner of the house.

The house was built with a better thermal envelope than is normally found in this area of the country, in an effort to reduce heating and cooling requirements. Double-pane insulated glass was used throughout; R-38 insulation was used in the ceiling where possible, and R-19 insulation was used in the crawl space, the sidewalls, below the floor, and the other ceiling areas. Table 4.1 gives a detailed description of the insulation of the demonstration house.

Table 4.1. Thermal insulation used in the demonstration houses

	Insulation thickness [in. (m)]	U-value [Btu hr^{-1} ft^{-2} °F^{-1} (W m^{-2} K^{-1})]
Sidewalls (R-19)	5-½ (0.14)	0.045 (0.256)
Ceiling, flat (R-38)	12 (0.305)	0.024 (0.136)
Ceiling, cathedral (R-19)	6 (0.152)	0.040 (0.227)
Floor (over crawl space) (R-19)	6 (0.152)	0.040 (0.227)
Floor (over ice bin) (R-38)	12 (0.305)	0.024 (0.136)
Windows	Double-glazed wood frame	0.58 (3.293)
Basement and crawl space		0.062 (0.352)

4.2.2 Simulated loads

Because of the experimental nature of the ACES, neither the ACES nor the control house are occupied. Internal loads and water use associated with occupancy have been simulated. Hot water consumption is 70 gpd (0.265 m^3/day) on the following schedule:

Time	Consumption [gal (m^3)]
6:00 AM	28 (0.106)
12:00 Noon	14 (0.053)
7:00 PM	14 (0.053)
11:00 PM	14 (0.053)

Internal loads from appliances are simulated by drawing 24 kWhr of electricity per day from resistance-heating elements and other electrical instruments that give off heat to the conditioned space. Both the ACES house and the control house have working refrigerators that are part of the internal electrical load. Door openings by people working at the complex simulate occupancy from the standpoint of ingress and egress. (Figure 4.1 shows an artist's conception of the ACES house.)

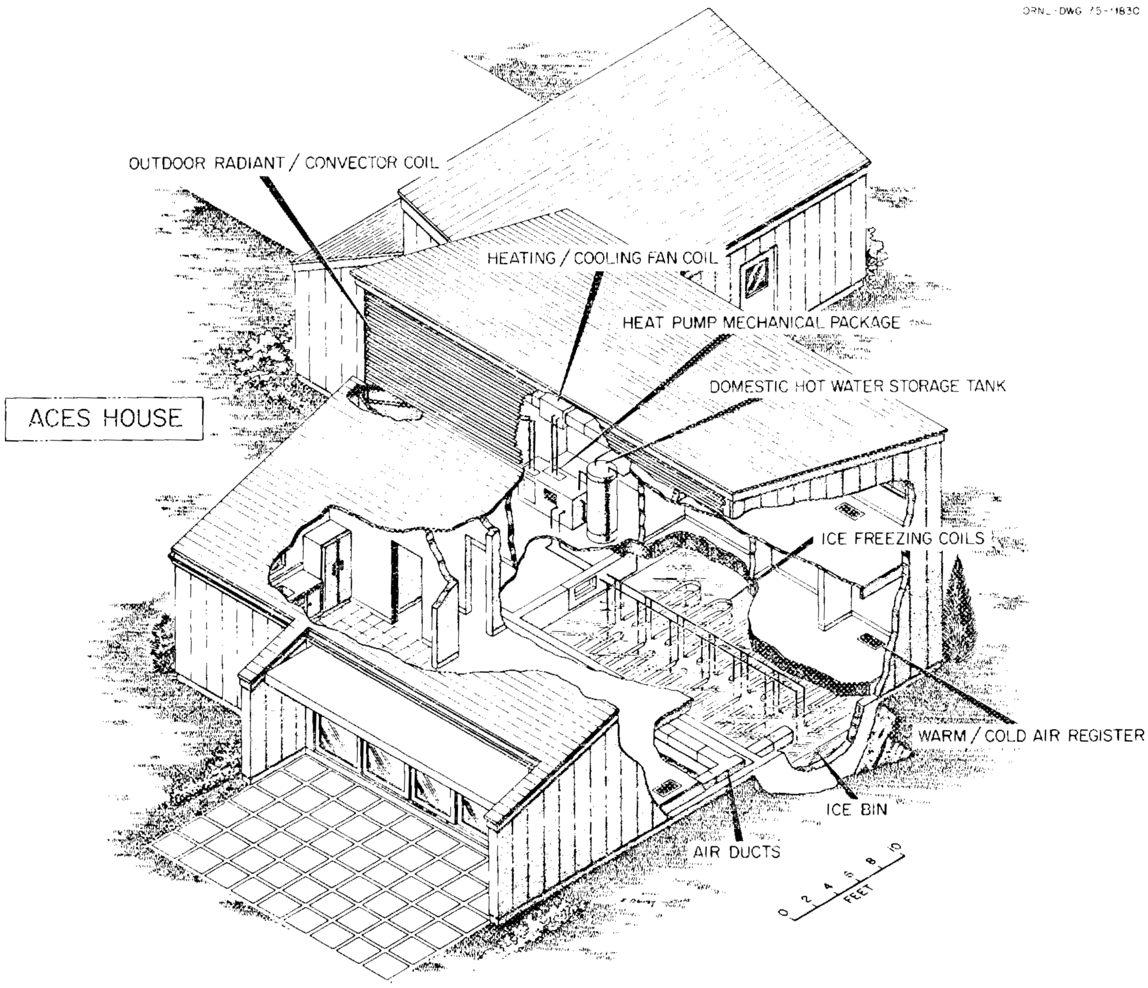

Fig. 4.1. Artist's conception of ACES house.

5. DATA ACQUISITION PROCEDURES

Located in the ACES house is an automatic data acquisition system (DAS) that monitors the three houses in the TECH complex.

5.1 Data Acquisition System

Figure 5.1 is a schematic for the DAS. The data acquisition system consists of a computer, an interfacing section, and sensors. Currently, the DAS monitors 80 points on an hourly basis and records these data at the complex and also on tape for use at ORNL. For a complete list of the data points and other information about the data system, see Appendix A.

5.2 Monitoring Procedures

Two types of data are being recorded at the TECH complex: analog signals such as temperatures, and digital signals such as integrated heat flows and electrical consumption. The analog signals are selected and fed through a digital voltmeter; the signals are read by the computer and converted to the proper units. The digital signals are integrated external to the computer and the integration results are

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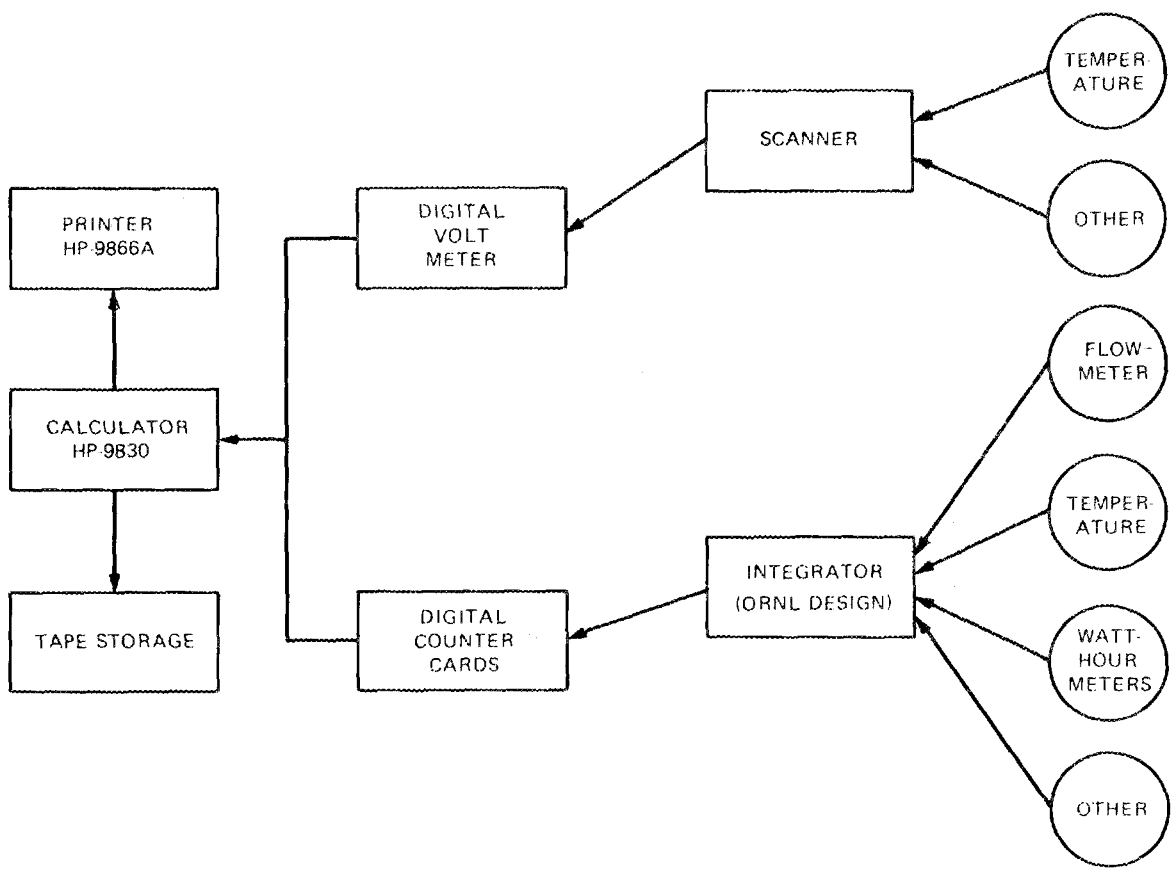

Fig. 5.1. Data acquisition system schematic.

stored on digital counter cards that are read by the computer and converted to the proper units. These data are read hourly. The analog data are the instantaneous values at the time of scan and the digital values are the sums for the hour. After the scan, the digital values are reset to zero, and the process begins again.

5.3 Data Reduction Techniques

Figure 5.2 is a block diagram for data processing at the ACES house. The data are recorded on magnetic cassette tapes at the house, and a printout is produced. The printout is used for performance testing at the TECH complex and for locating equipment or instrumentation malfunctions. The data on the cassette are transmitted to ORNL and become part of a permanent data base. These data are processed into weekly summaries of performance. Performance plots are used for determining system performance, for report preparation, and for studying thermal envelope performance. A sample of the various data outputs is provided in Appendix B.

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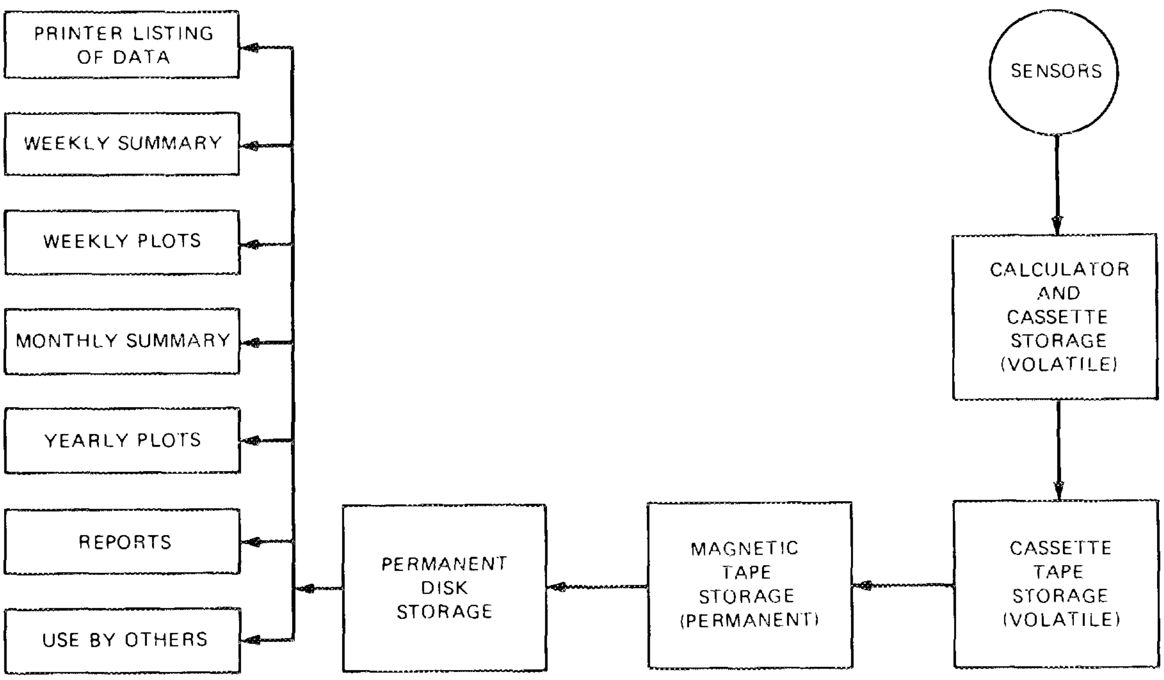

Fig. 5.2. Data-processing schematic.

6. ACES PERFORMANCE

6.1 Steady-State Performance

During the year, steady-state testing per se was not performed. However, it is possible to find periods of time in which the system operated in a steady-state manner. The following sections present the results for those periods.

6.1.1 Energy inputs and outputs

Table 6.1 gives the approximate power consumption of the compressor, pumps, and fan under full-load conditions. Each of these items was monitored individually to allow for determination of instances in which system performance could be improved. The actual power draws of the components vary slightly depending on the mode of operation, the ice-bin temperature, the amount of ice, and other factors; however, the values represent typical operating characteristics of the components.

Table 6.1. Component energy consumption

Device	Energy input (W)
Heating mode (ice source)	
Compressor	2260
Fan	380
Hot brine pump	143
Cold brine pump	166
Hot water pump	80
Night heat-rejection mode	
Compressor	2730
Hot brine pump	140
Cold brine pump	166
Hot water pump	80
Hot-water mode	
Compressor	1830
Cooling pump	166
Hot water pump	80
Cooling mode	
Cold brine pump	166
Fan	380
Ice-melting mode	
Cold brine pump	166

Table 6.2 presents the full-load capacities of the system in various operating modes. These values are also representative because the exact magnitudes vary slightly with several conditions.

6.1.2 System operating temperatures

Because the temperatures logged by the DAS are the instantaneous values at the times of the scans, the circulating temperatures are valid only if the system is operational and is not in a transient start-up

Table 6.2. Modal energy delivery rates

Mode	Energy delivery rate (W)
Space heating, fan coil	5910
Space cooling, fan coil (ice source)	7270
Space cooling, fan coil (chilled water source)	4744
Water heating only	5180
Night heat rejection, bin heat extraction	6091
Ice melting	0–8770 (variable)

period. Tables 6.3 and 6.4 depict normal operating temperatures selected from a number of representative surveys of both brine and refrigerant systems. These tables present the temperatures of the major components operating in the various modes and the typical flow rates. Heat flows are integrated during each hour of operation and recorded at the completion of the hour.

6.1.3 Coefficients of performance (COP)

The modal coefficient of performance was determined by selecting hours in which the ACES operated in a steady-state manner for a given mode. Table 6.5 gives the results of these calculations for the various modes. The bin heat-rejection COP during night heat rejection includes only evaporator heat flows and is a measure of equipment performance. The true night heat-rejection COP, or

Table 6.3. Typical brine-system temperatures and flow rates

	Space-heating mode	Space-cooling mode		Hot-water mode	Ice-melting mode	Night heat-rejection mode
		Ice	Chilled water			
Fan coil						
In, °F (°C)	105 (48)	37 (3)	50 (10)	*a*	*a*	*a*
gpm (ℓ/s)	5.5 (0.35)	5.5 (0.35)	5.5 (0.35)	*a*	*a*	
Out, °F (°C)	97 (36)	47 (8)	56 (13)	*a*	*a*	*a*
Ice bin						
In, °F (°C)	25 (−4)	47 (8)	56 (13)	25 (−4)	42 (6)	40 (4)
gpm (ℓ/s)	11.3 (0.71)	5.5 (0.35)	5.5 (0.35)	11.3 (0.71)	6.4 (0.40)	11.3 (0.71)
Out, °F (°C)	32 (0)	37 (3)	50 (10)	32 (0)	32 (0)	43 (6)
Solar panel						
In, °F (°C)	*a*	*a*	*a*	*a*	32 (0)	118 (48)
gpm (ℓ/s)	*a*	*a*	*a*	*a*	6.4 (0.40)	7.9 (0.50)
Out, °F (°C)	*a*	*a*	*a*	*a*	42 (6)	111 (44)
Hot water						
In, °F (°C)	*a*	*a*	*a*	90 (32)	*a*	135 (57)
gpm (ℓ/s)	*a*	*a*	*a*	0–3 (0.2)	*a*	0–3 (0.2)
Out, °F (°C)	*a*	*a*	*a*	120 (49)	*a*	145 (62)

[a]Not applicable.

Table 6.4. Typical refrigerant system[a] operating conditions

	English units	Metric units
Heating and water-heating mode (ice source)		
Superheated gas temperature	190° F	87.7° C
Condensing temperature	108° F	42.2° C
Condensing pressure	223 psig	1.537×10^6 Pa
Evaporation temperature	21° F	−6.1° C
Evaporation pressure	44 psig	3.0×10^5 Pa
Subcooled liquid temperature	44° F	6.6° C
Refrigerant gas temperature exiting the hot-water desuperheater	110° F	43.3° C
Water-heating mode (ice source)		
Superheated gas temperature	190° F	87.7° C
Condensing temperature	115° F	46.1° C
Condensing pressure	245 psig	1.75×10^6 Pa
Evaporation temperature	21° F	−6.1° C
Evaporation pressure	44 psig	3.0×10^5 Pa
Subcooled liquid temperature	45° F	7.2° C
Bin heat-rejection mode (45° F water source)		
Superheated gas temperature	192° F	88.8° C
Condensing temperature	125° F	51.6° C
Condensing pressure	280 psig	1.93×10^6 Pa
Evaporation temperature	38° F	3.3° C
Evaporation pressure	66 psig	4.5×10^5 Pa
Subcooled liquid temperature	63° F	17.2° C

[a]R22 refrigerant.

Table 6.5. Steady-state coefficients of performance (COP)

Mode	COP
Space heating with water heating	2.70
Water heating only	2.50
Space cooling with stored ice	12.4
Space cooling with chilled water	8.69
Bin heat rejection	1.99

supplementary cooling COP, must include both bin heat leakage and useful hot-water deliveries. The night heat-rejection COP is, therefore, greatly influenced by the actual systems application. Because the night heat-rejection COP is the result of a number of variables, it will not be included in this calculation. Appendix C contains the calculations used to derive Table 6.5.

6.2 Annual Performance

6.2.1 Energy consumption

Although the discussion on instantaneous performance is useful and informative, the true measure of ACES performance can only be obtained by observing the ACES for an entire year (because of the interseasonal energy transfer performed by the ACES). Until the transfer cycle has been completed, it is not possible to determine the exact ACES performance because of energy that is not accounted for. Figures 6.1 and 6.2 compare the performance of the ACES house and the control house. The control house had an electric-resistance heating and hot-water system and central air conditioning. This system was chosen because it allowed an accurate comparison of the performance of the two houses. Figure 6.1 shows the annual loads for the two houses along with the energy required to deliver those loads. The ACES house exhibited a COP of 2.78 for the year and the control house had a COP of 1.13. The

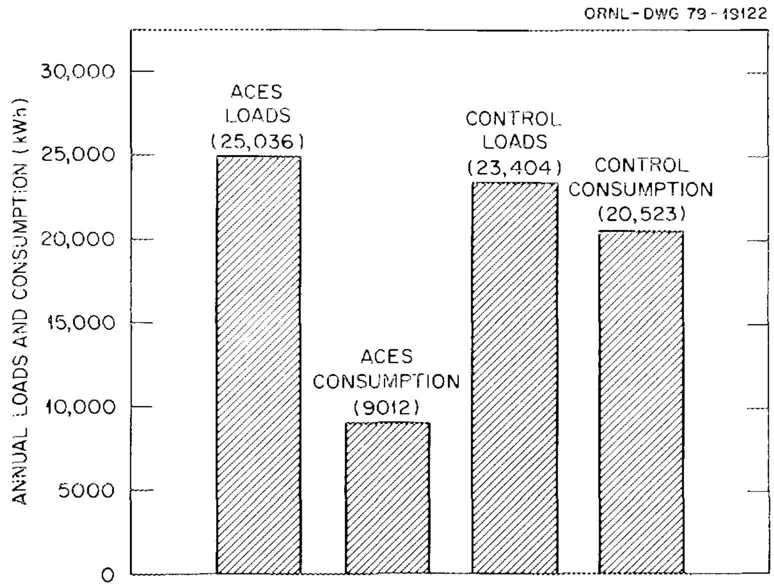

Fig. 6.1. Comparison of loads and energy consumption for the ACES and the control house.

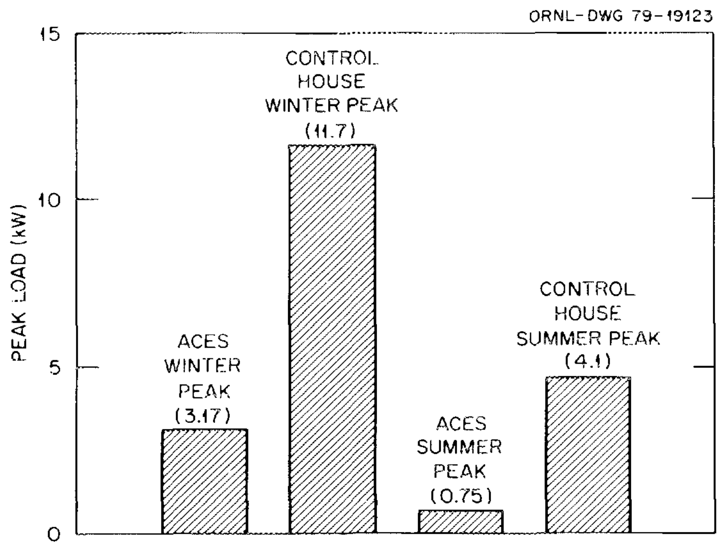

Fig. 6.2. Comparison of peak consumption for the ACES and the control house.

reduction in energy consumption for the ACES house was, therefore, 56% as compared with the control house.

Figure 6.2 shows the winter and summer utility peak loads for the two houses. The ACES house consistently exhibited peak loads markedly less than those of the control house. The winter peak reductions were caused by the lack of supplemental heat, by the method by which hot water was produced, and by the constant high efficiency performance of the ACES. The summer peak reductions were the result of providing all on-peak air conditioning from the storage bin rather than from operation of the compressor during the day, and by producing hot water with the compressor rather than using electric-resistance heat.

6.2.2 Weather conditions

The performance of the ACES depends on the thermal loads that the system must deliver each year. These loads are a direct result of the interaction of the weather with the building's thermal envelope, and it is appropriate to characterize qualitatively the severity of the test-year weather and to compare it with an average weather year.

Even though the TECH complex is fully instrumented to measure local climatological conditions, for long-term comparative purposes it is more convenient to use the Knoxville, Tennessee, McGhee-Tyson Airport, station 13891, weather data compiled by the National Climatic Center. The McGhee-Tyson Airport station is located within 5 miles of the TECH complex and has collected historical records for many years.

The degree-day method used by the National Climatic Center is an accepted method for quantifying the severity of a heating or cooling season. Actual weather data for the test year are listed in Table 6.6, together with 1941 to 1970 long-term average weather data from this same station. It is

Table 6.6. Test-year weather conditions

	Heating degree-days		Cooling degree-days	
	Normal[a] [°F (°C)]	Actual [°F (°C)]	Normal[a] [°F (°C)]	Actual [°F (°C)]
November	474 (263)	374 (208)		10 (6)
December	729 (405)	764 (424)		
January	756 (420)	1097 (609)		
February	630 (350)	846 (470)	8 (4)	
March	484 (269)	487 (270)	16 (9)	
April	173 (96)	148 (82)	32 (18)	37 (21)
May	47 (26)	74 (41)	152 (84)	130 (72)
June			315 (175)	319 (177)
July			409 (227)	432 (240)
August			381 (212)	384 (213)
September	10 (6)		208 (116)	302 (167)
October	175[b] (97)[b]		48[b] (27)[b]	
Test year total	3303 (1835)	3790 (2104)	1521 (845)	1614 (896)

[a]Long-term average, 1941–1970, measured in °F.

[b]Data not included in totals.

Source: National Climatic Center, *Local Climatological Data Annual Summary with Comparative Data, 1978, Knoxville, TN,* USCOMM-NOAA Asheville-1150, Asheville, N. C.

instructive to consider the test-year weather by season and by month of the year so that a qualitative impression of the influence of weather conditions on ACES performance can be gained.

The severity of the heating season determines the heating loads that the system must deliver, the energy that is consumed during heating, and the production of ice that results from delivery of the heating requirements. The maximum amount of energy available for interseasonal energy transfer is limited by either the heating demands or, if ice production exceeds bin capacity, by the bin capacity.

Heating season. During this test year, ice production almost exactly equaled bin capacity. The qualitative and quantitative characteristics of the weather leading to these results are described in the following paragraphs.

November 1977 was a relatively mild month, which produced only a slightly higher than normal heating demand. January and February 1978 were both severely cold months, each within the top two highest heating-demand months of the last 20 years. Collectively, these two months were the coldest two consecutive months on record for the last 20 years. January and February 1978 were recorded as requiring 1943 degree-days (1079° C-days) of heating, but the record for a previous two-month period was 1775 degree-days (986° C-days) for January and February 1970. The remainder of the test year heating season moderated to nearly normal conditions.

The total seasonal heating degree-days was 3790 degree-days (2104° C-days) for the November 1977 to May 1978 heating season; this compares with the normal 3303 degree-days (1835° C-days) anticipated for the November to May period, based on the National Climatic Center long-term average. November 1977 to May 1978 was exceptionally cold; it was the fourth coldest winter of the last 20 years.

The severe winter required the ACES to deliver far more heating than was expected on the basis of average yearly requirements. For the test year, the system should have produced an ice inventory greatly exceeding the bin capacity and requiring panel operation to melt excess ice. Bin capacity was not exceeded; the bin was filled only to its maximum design capacity.

Cooling season. The test-year cooling season can be characterized as nearly normal as judged by the National Climatic Center's tabulation of cooling degree-days (in Table 6.6). Each month of the summer exhibited only slight deviations from the long-term cooling degree-day average. The only exception was September 1978, which remained hotter somewhat longer than usual. Because the experiment was terminated on September 18, average cooling can be assumed even for this month.

The cooling season exhibited a total of 1614 degree-days (896° C-days) as compared with 1521 degree-days (845° C-days) for long-term average weather conditions. Cooling should be considered as having been average for the year.

6.2.3 Ice-bin effectiveness

The ice bin is the distinguishing feature that sets the ACES apart from other heating and cooling concepts. To operate successfully, the phase-change material (ice) must be well insulated from the environment. The insulation is accomplished by two methods. First, the bin is located underground in an environment in which temperatures fluctuate less than the normal ambient air temperatures. Second, the bin is insulated from those areas that will experience temperatures deviating greatly from 32° F (0° C).

The effectiveness of the ice bin in storing and delivering energy is of fundamental importance to the success of the ACES.

Figure 6.3 shows the ice inventory during the test year; the figure illustrates what is meant by interseasonal energy transfer. The curve is proportional to the amount of energy transferred because each English ton of ice corresponds to 288,000 Btu (3.04×10^8 J) of stored cooling energy. The figure

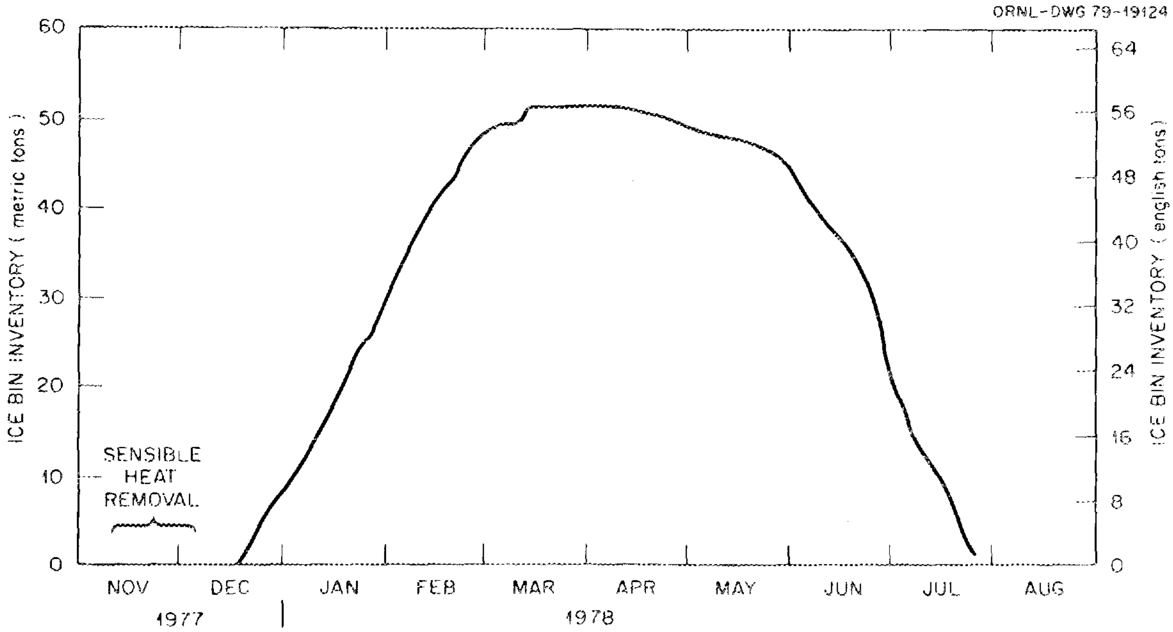

Fig. 6.3. ACES ice-bin inventory, 1977–1978.

clearly shows that significant amounts of cooling energy were produced during the winter and used during the summer, thereby reducing energy consumption for summer cooling.

The ice bin in this installation was originally constructed as an integral component of the structure. The walls of the ice bin form a portion of the foundation of the house, and the first floor of the house forms the top of the bin. The top of the bin was insulated with an R-38-equivalent fibrous mineral wool batting contained in asphalted kraft-paper units. The sides were constructed of Foam Form blocks with integral concrete and structural steel elements. As constructed, the walls exhibited an insulation rating equivalent to R-17. The bin floor was a simple concrete pad cast over a bed of crushed rock, with no insulation provided.

When the bin was originally constructed, provisions were made to measure the earth temperature at a distance of 1 ft (0.31 m) from the tank sidewall and at 8 points directly below the bin floor [2 rows of thermocouples at 1, 2, 3, and 4 ft (0.3, 0.6, 0.9, and 1.23 m) below the bin floor]. These 20 thermocouples have been monitored both before starting the test and periodically throughout the test year. The earth-temperature thermocouple locations and a typical representation of actual ground temperature isotherms is shown in Fig. 6.4.

Figure 6.5 presents the results of the measurements throughout the year, as well as average undisturbed earth temperatures. The undisturbed earth temperatures are calculated by the Kusuda-Achenbach[5] method and are presented for comparison. The thermocouple grid matrix detected the thermal influence of the ice bin on the surrounding earth about four months after ice was first stored in the bin. Temperature stratification was observable by comparing the thermocouple measurements taken close to the bin floor with those taken further from the floor. The thermocouples located furthest below ground level [13 ft (4 m)] did not appear to be significantly influenced by the presence of the ice-bin heat leakage. The actual heat leakage into the ice bin was about two to three times that originally calculated. Table 6.7 shows the original heat-leakage calculations.

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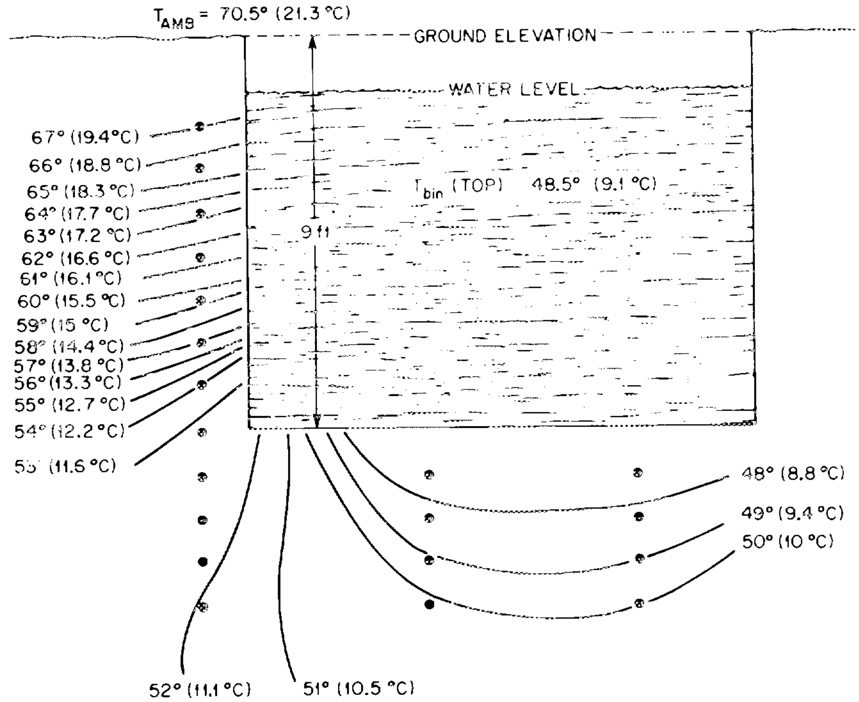

Fig. 6.4. Late summer ice-bin earth-temperature isotherms, 8/3/77.

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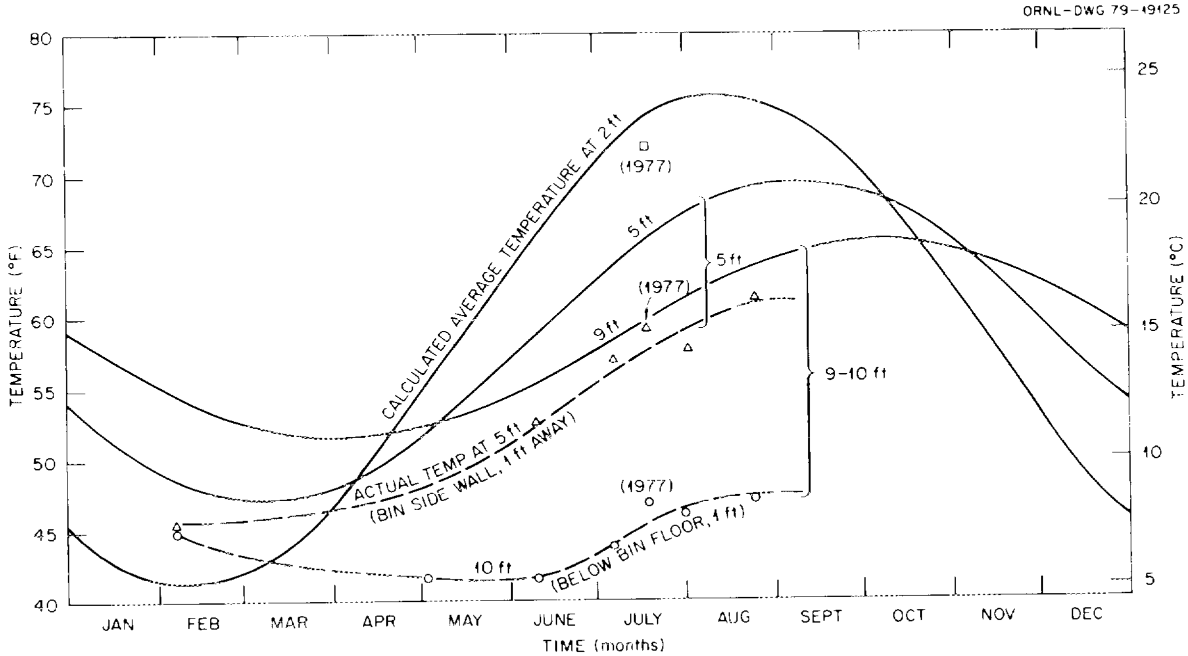

Fig. 6.5. Calculated earth temperatures and actual temperatures surrounding the ice bin.

Table 6.7. Original heat-leakage calculations for the ice bin[a]

	Monthly heat leakage [Btu $\times 10^6$ (J $\times 10^9$)]
January	0.641 (0.6762)
February	0.508 (0.5359)
March	0.599 (0.6319)
April	0.708 (0.7469)
May	0.933 (0.9843)
June	1.104 (1.1647)
July	1.271 (1.3409)
August	1.379 (1.4549)
September	1.286 (1.3568)
October	1.212 (1.2787)
November	0.978 (1.0318)
December	0.804 (0.8482)

[a]The tank temperature was assumed to be 32°F (0°C) all year.

The water contained within the bin always exhibited some temperature stratification. A series of studies was conducted to measure this stratification and typical late summer results are shown in Fig. 6.6.

Ice formation phase. During the winter, house space-heating and hot-water loads caused the water to be converted to ice. The ratio of ice-bin heat extraction to the amount of heat delivered to the house for space heating and water heating is given by the expression:

$$\alpha(H) = \frac{COP(H) - 1}{COP(H)} .$$

Here, COP(H) is the combined space- and water-heating steady-state COP; for the demonstration house system $\alpha(H) = 0.630$.

During the period of December 20, 1977 (the date of first ice formation), to April 1, 1978 (the date of maximum ice inventory), the system delivered a total of 35.75×10^6 Btu (37.7×10^9 J) space- and water-heating loads; the system had the potential of producing 22.52×10^6 Btu (23.76×10^9 J) equivalent of ice (71.09 metric tons). The estimated actual bin heat leakage would have naturally destroyed 5.56×10^6 Btu (5.86×10^9 J) (17.55 metric tons), leaving 53.54 metric tons. The remaining amount is very close to the actual ice inventory achieved on the first of April; however, the estimate does not take into account the period required to utilize the water tank's sensible energy. The sensible energy stored in the bin must also be accounted for if an accurate prediction of ice inventory is to be made. The time from November 1, 1977, to December 20, 1977, was needed to extract the bin's sensible energy.

In addition to evaluating the performance actually demonstrated during the extremely cold test-year winter, it is instructive to consider what would have happened had the house experienced an average weather year. The average weather year heating requirements of the house are estimated to be 30.58×10^6 Btu (32.26×10^9 J), and the date of initial ice formation is estimated to be January 1. For an average weather year, the ice-production potential would be limited to about 9.35×10^6 Btu (9.86×10^9 J) (29.5 metric tons). This amount of ice would have been in storage on April 1 during an average weather year with the as-built ice-bin construction.

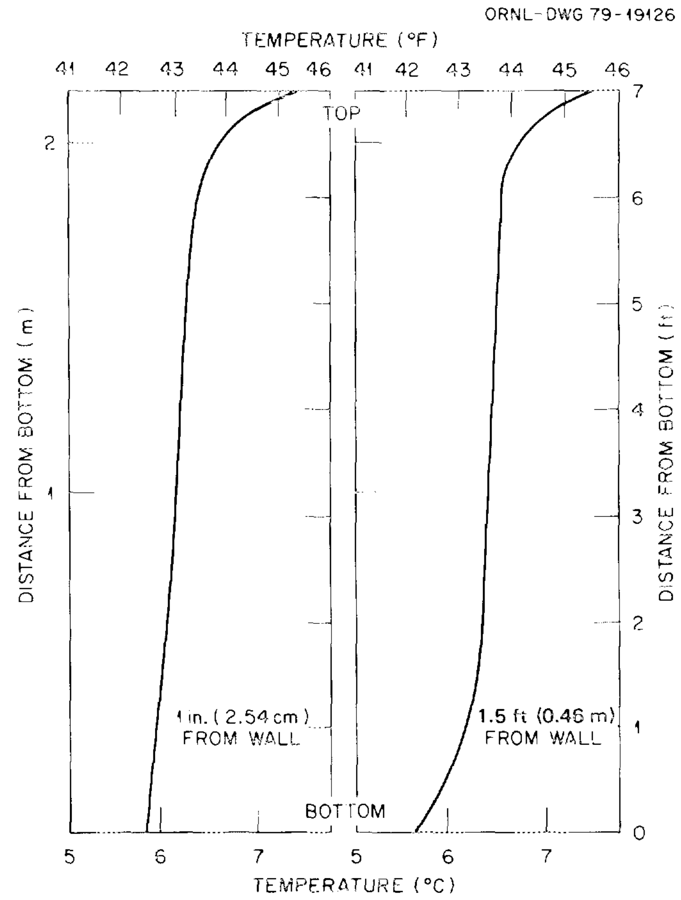

Fig. 6.6. Summer ice-bin temperature profiles, 6/30/77.

Spring season. During the spring, space-heating and space-cooling loads are small and bin energy extractions result primarily from water-heating loads. Water-heating loads did not impose a sufficient bin energy extraction rate to counteract bin heat leakages during these months. The stored cooling capacity of the bin declined and at the beginning of May, the system had 15.55×10^6 Btu (49 metric tons), or 94% of its design storage capacity.

Cooling season. In East Tennessee, significant cooling requirements usually begin in May. The test year was quite normal because substantial cooling loads were required by the middle of May. All cooling requirements were satisfied by using stored ice, until the ice and all usable bin sensible energy were exhausted on July 27, 1978. A total of 12.64×10^6 Btu (13.33×10^9 J) of useful cooling loads had been successfully transferred from the winter season and delivered to the house. Effective interseasonal energy transfer was achieved.

The control house required the purchase of 2274 kWhr of electricity to deliver approximately the same cooling loads during this period, but the ACES required the purchase of only 298 kWhr to provide space cooling.

The ice bin effectively delivered 68% of its total thermal capacity in useful cooling loads. Bin heat leakage and delivery energies reduced the useful energy to this level from the 94% of capacity that had existed at the beginning of the cooling season.

It is informative to evaluate what would have happened had the system experienced an average winter and entered the month of May with all the ice that would have been produced in an average year.

After an average winter, the ice inventory remaining on May 1 would have been 8.54×10^6 Btu [(9.01×10^9 J) (26.96 metric tons)]. This quantity of ice could have provided stored-ice cooling only until late June, at which time the system would have entered the night heat-rejection mode.

Night heat rejection bin performance. When the system enters the night heat-rejection mode of operation, energy leakage into the bin must be rejected by the heat pump, along with the cooling loads of the house and the compressor input energy. High bin heat leakage rates therefore increase the purchased energy requirements. After the system entered the night heat-rejection mode, an effort was made to allow the bin water temperature to rise and thus minimize heat leakage into the bin by reducing the temperature difference with respect to the surrounding earth. The effort was successful, reducing purchased energy requirements to about 13% more than normal air conditioning requirements. However, compromises in the desired building dehumidification level had to be made and indoor relative humidities reached 70%. The system cooling capacity was also substantially reduced.

Figure 6.3 shows the ice inventory as a function of time. At the completion of stored-ice cooling, the bin water level had risen as a result of about 330 lb (150 kg) of additional water that accumulated in the ice bin over the seven-month period. The increase in water level is attributed to condensation of atmospheric water vapor. During the first part of the year (up to April 10, 1978), a very poorly fitting window for viewing the process allowed air infiltration into the bin. A well-sealed triple-pane window was installed to correct this problem.

6.3 Seasonal Performance

Although it is true that the actual performance of the ACES can only be determined over an entire year, interesting aspects of the ACES show up when the performance is studied on a seasonal basis.

6.3.1 Heating season

To a homeowner the primary feature of an ACES is energy conservation. Energy conservation shows up dramatically when stored ice is used for space cooling, but energy conservation during the heating season is also significant. Table 6.8 shows the heating loads incurred by the ACES and control houses and the energy required to meet these loads. The heating season is defined here as November 1, 1977, through April 30, 1978. The ACES shows a savings of 63% over an all-electric-resistance-heating

Table 6.8. Winter energy consumption

(November 1, 1977, through April 30, 1978)

Function	Control house Load [Btu × 10^{-6} (J × 10^{-9})]	Control house Consumption (kWhr)	ACES Load [Btu × 10^{-6} (J × 10^{-9})]	ACES Consumption (kWhr)
Heating	41.02 (43.28)	12,019	39.98 (42.18)	5,468
Hot water	9.67 (10.20)	2,834	11.42 (12.05)	
Total	50.69 (53.48)	14,853	51.40 (54.23)	5,468

system. This savings is accomplished while storing ice for later use to further reduce energy consumption.

Energy conservation during the heating season is a direct result of the fixed-capacity, high-efficiency, and no-cycling operational features of the ACES. Figure 6.7 shows the temperature of the brine inlet and the COP as a function of ice inventory. The flatness of the COP curve clearly indicates the uniformly high efficiency of the ACES: the COP varies from only 2.8 in the condition with no ice to 2.6 when 50 metric tons of ice are present. The fixed-capacity feature also removes the need for supplemental heating, usually supplied by resistance heating elements at a COP of 1; the ACES has a COP of 2.6. (It is interesting to note that the ACES system experienced little performance degradation because of cycling.) The remainder of the energy conservation arises from delivering the domestic hot water at a COP of 2.5, rather than the COP of 1 characteristic of resistance heating elements.

Another advantage that ACES offers is peak-load leveling. This feature is extremely important to electric utilities because one of their major problems is on-line generating capacity. The ACES provides peak-load leveling by maintaining a fixed COP regardless of external conditions. This means that the ACES has a peak-load curve with the same shape as that of electric-resistance systems; however, the magnitude of the peak load is reduced by (COP − 1)/COP. Figure 6.8 shows the purchased energy requirements for the ACES and those for a resistance heating system during a typical cold winter week. The ACES has a consumption curve with the same shape as that of the electric-resistance system; however, the magnitude of consumption is only about 1/COP as much. This means that the ACES reduces the peak load by 63% when compared with an electric-resistance heating system.

6.3.2 Cooling season

Stored-ice cooling. The cooling season is defined here as May 1, 1978, through mid-September 1978.

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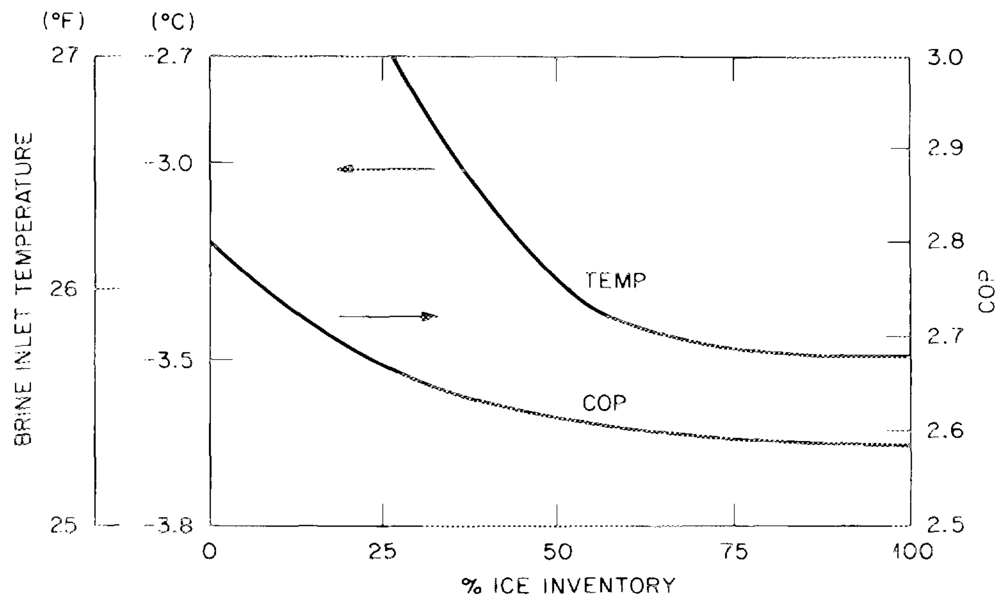

Fig. 6.7. Temperature of the brine inlet as a function of ice inventory (space-heating mode).

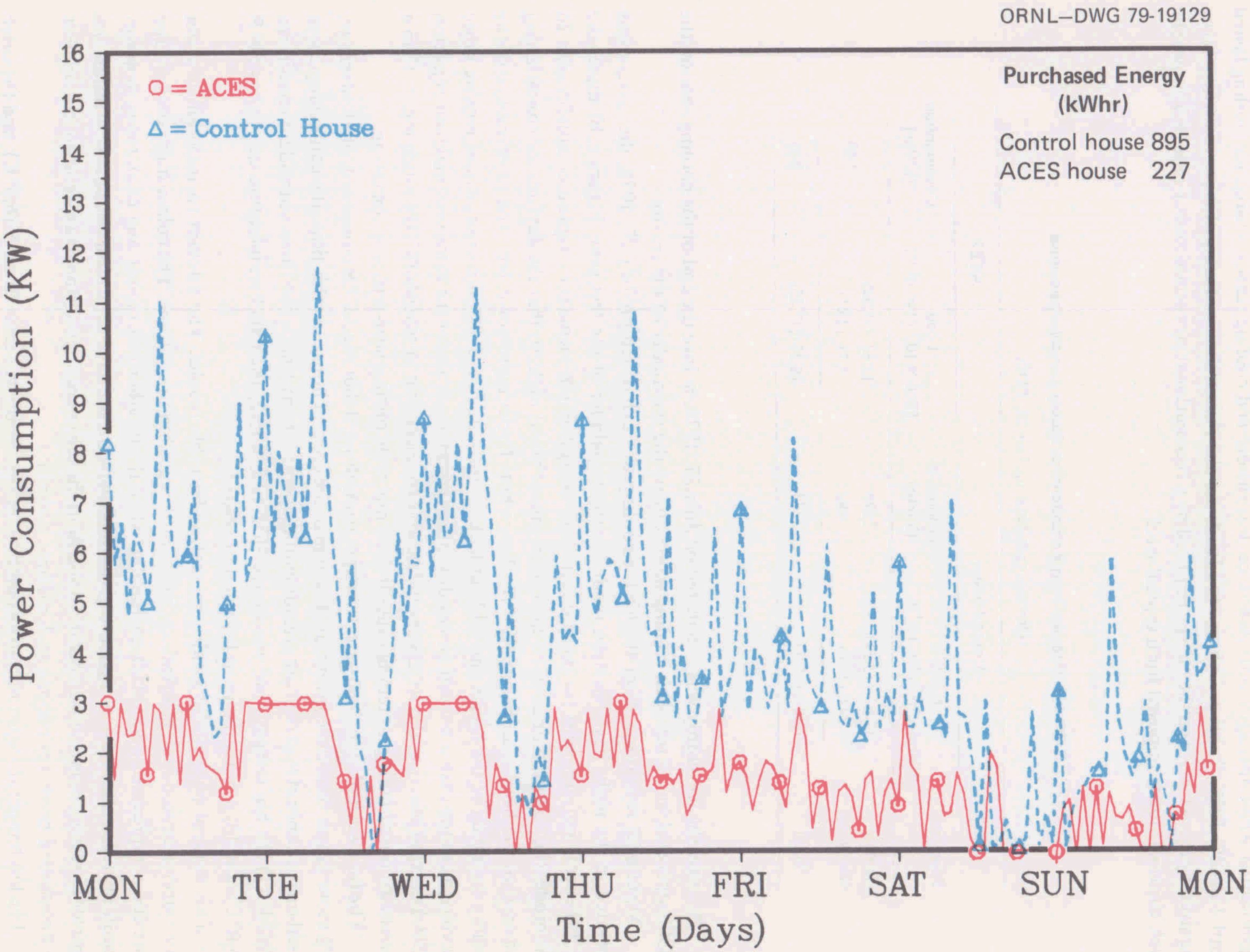

Fig. 6.8. Comparison of ACES and control houses, week beginning 1/2/78 (heating-season week).

When stored ice is available the cooling COP for the ACES is 12.4; conventional air conditioners have steady-state COPs of 2.05 to 2.93. Thus, it is apparent that the ACES reduces energy consumption significantly when operating with stored ice. For the defined cooling season, stored-ice cooling lasted until July 27, 1978. Table 6.9 shows the loads on the ACES house during this period, the energy required by the ACES, and the energy required by a conventional air conditioner. Figure 6.9 shows the peak electrical loads imposed for a typical week.

Table 6.9. Summer energy consumption during stored-ice operation

(Ice was exhausted on July 27, 1978)

Function	Control house		ACES	
	Load [Btu × 10^{-6} (J × 10^{-9})]	Consumption (kWhr)	Load [Btu × 10^{-6} (J × 10^{-9})]	Consumption (kWhr)
Cooling	12.64 (13.34)	2274	12.64 (13.34)	840
Hot water	3.41 (3.60)	999	3.71 (3.91)	
Total	16.05 (16.94)	3273	16.35 (17.25)	840

Because the ice supply was exhausted on July 27, 1978, before the end of the cooling season, the ACES operated in the off-peak cooling mode during the remainder of the season.

Night heat rejection. After the ice for cooling was exhausted on July 27, 1978, the system was operated in the night heat-rejection mode for the remainder of the test year. Figure 6.10 shows the ACES energy requirements for this mode and the energy requirements for a typical air conditioner. In this mode, the compressor extracts heat from the ice bin at night so that the daytime air conditioning demands of the house can be supplied by the ice bin. The compressor is required to extract both the cooling demands of the house and all heat leakage into the bin. This, of course, would require more purchased energy than a normal air conditioning system operating at the same seasonal heat-rejection COP. In actual practice, the ACES should be able to reject heat at seasonal COPs much higher than a normal air conditioning system because it does not cycle during heat-rejection operation.

The first week of night heat-rejection operation showed that the ACES consumed, on the average, 33% more power than the conventional system in the control house. At this time, the control logic was based on the concept that the compressor would operate at any time when three coincident conditions existed: (1) the bin temperature was above 36° F (2.2° C), (2) the panel temperature was below 80° F (26.6° C), and (3) a potential need for cooling existed.

This control philosophy yielded several undesirable results. The attempt to maintain low bin temperatures caused high heat leakage rates into the ice bin to continue. The colder it got outside, the more the compressor operated. Occasionally, when the weather was cloudy and cool, the system ran continuously for almost 24 hr. Because of this unsatisfactory situation, an effort was made to adjust the controls to restrict compressor operation and limit the bin energy extraction to a rate sufficient to meet the anticipated needs for cooling and bin heat leakage.

The bin temperature for initiating compressor operation was reset to 44° F (6.6° C), and the solar panel cut-off temperature was changed so that compressor running time was decreased. The combined effect of these changes was to decrease the average daily duration of compressor operation. Decreasing

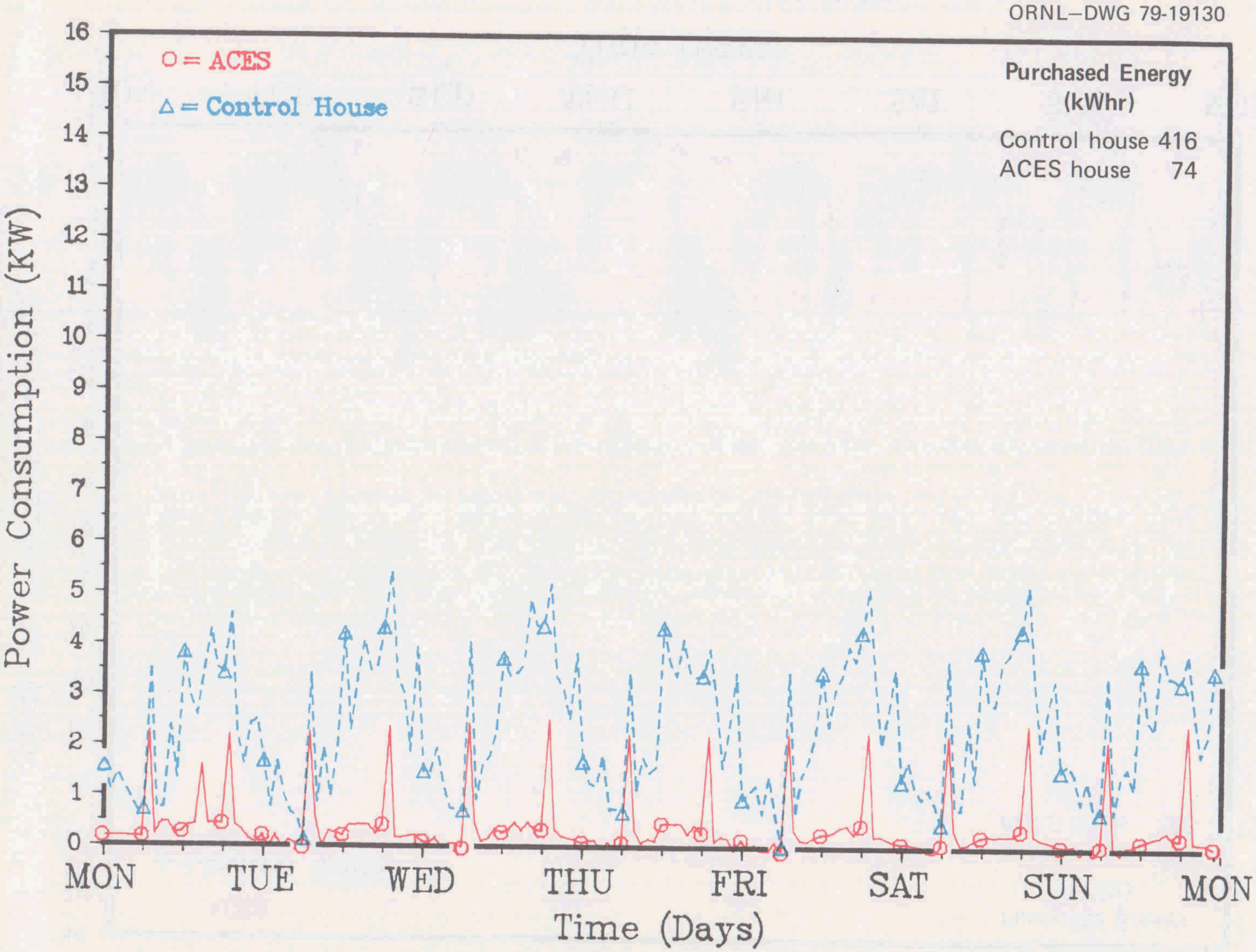

Fig. 6.9. Comparison of ACES and control houses, week beginning 6/26/78 (stored-ice cooling).

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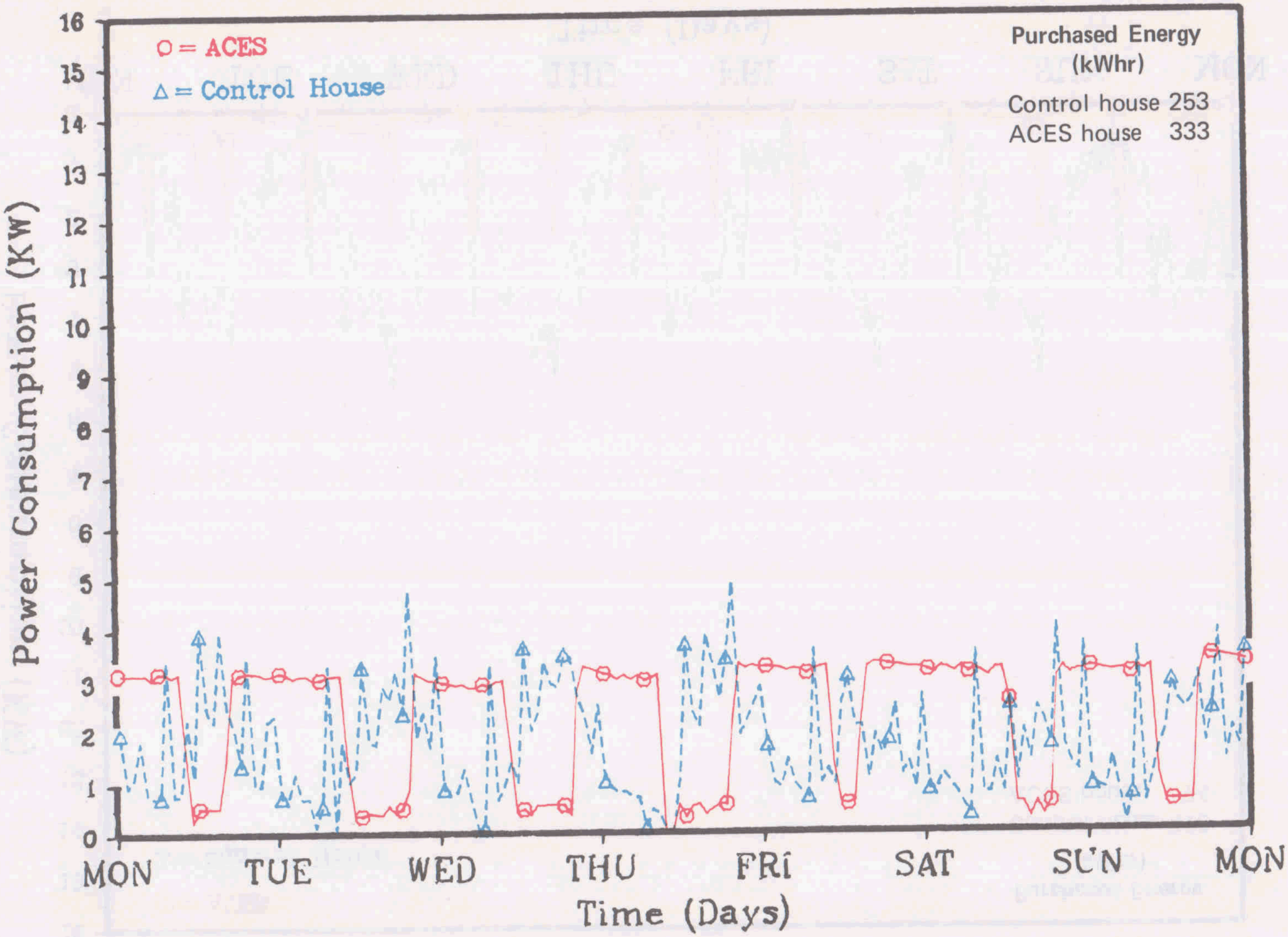

Fig. 6.10. Comparison of ACES and control houses, week beginning 7/31/78 (night heat rejection).

compressor operation time allowed the average bin temperature to rise and reduced the heat leakage into the bin. Frequent adjustments were made to maintain desired operating time.

The compressor operated continuously during periods of low outdoor temperature. The extended duration of compressor operation was related to the rate of heat extraction from the ice bin. The rate of heat extraction was not sufficiently rapid to reduce the bin temperature below the compressor cut-off point because of cooling demands and bin heat leakage.

The compressor and pumps are located inside the ACES house to facilitate use of waste heat during the winter months. When the compressor must operate in the summer, waste heat is detrimental because it too must be rejected along with normal house demands and bin heat leakage. Because the compressor must operate to reject its own waste heat, the cooling COP of the ACES is significantly reduced and purchased power requirements are increased. It is estimated that the supplemental cooling COP of the ACES was reduced by heat from the equipment to about 80% of the cooling COP of the control-house system which has the compressor outdoors. To eliminate the increased system demands, the mechanical package room was isolated from the remainder of the house and a window in the room was opened. Even with these corrective actions, the ACES exhibited a lower cooling COP than the control-house system for the remainder of the test year.

Table 6.10 gives the energy requirements for the ACES and the control house during the period in which the ACES operated in the off-peak cooling mode. In this mode the ACES required 12.7% more energy to meet its cooling and hot-water requirements than did the control house because: (1) some of the cooling energy from the bin was lost to the surroundings before it could be used; (2) additional cooling was required because the compressor was located inside; and (3) energy was required both to store the cooling energy and later to retrieve it. The bulk of this energy is used during off-peak hours, and compressor heat can be easily vented outside to reduce the amount of extra energy required. Table 6.11 gives energy requirements for the entire summer, including the domestic-hot-water requirements.

Control-house air conditioning. The system that was compared with the ACES during the cooling season was a commercial Weathertron heat pump operating in the air conditioning mode. This unit has a 2.04 steady-state cooling COP at the 95°F rating point, as rated by the Air Conditioning and Refrigeration Institute. The unit was periodically checked and found to be operating at the specified performance level.

Table 6.12 shows the performance of this unit during the test-year cooling season. The average seasonal cooling COP was 1.64, which indicated that cycling losses in the air conditioning mode are in

Table 6.10. Summer energy consumption during off-peak cooling operation

(July 27, 1978, through September 18, 1978)

Function	Control house		ACES	
	Load [Btu × 10^{-6} (J × 10^{-9})]	Consumption (kWhr)	Load [Btu × 10^{-6} (J × 10^{-9})]	Consumption (kWhr)
Cooling	10.02 (10.57)	1891	12.15 (12.82)	2704
Hot water	1.73 (1.83)	506	1.73 (1.83) 4.66 (4.92)[a]	
Total	11.75 (12.40)	2397	13.88 (14.65)[a] 16.81 (17.74)[a]	2704

[a]Hot water requirements differ because of the method of production. Actual requirements are those for the control house.

Table 6.11. Total summer energy consumption

(May 1, 1978, through September 18, 1978)

Function	Control house Load [Btu × 10^{-6} (J × 10^{-9})]	Control house Consumption (kWhr)	ACES Load [Btu × 10^{-6} (J × 10^{-9})]	ACES Consumption (kWhr)
Cooling	22.66 (23.91)	4165	24.79 (26.15)	3542
Hot water	5.14 (5.42)	1505	5.14 (5.42)	
			8.37 (8.83)[a]	
Total	27.80 (29.33)	5670	29.93 (31.57)	3542
			33.16 (34.98)[a]	

[a]Hot-water requirements differ because of the method of production. Actual requirements are those for the control house.

Table 6.12. Control house air conditioning performance

(Standard *Weathertron* heat pump rated at a COP of 2.04 at 95°F outside air temperature)

	Cooling demand [Btu × 10^{-6} (J × 10^{-9})]	Purchased power [Btu × 10^{-6} (kWhr)]	Monthly cooling COP
May	1.50 (1.58)	1.00 (293)	1.50
June	4.98 (5.25)	3.00 (879)	1.66
July	6.54 (6.90)	3.982 (1167)	1.64
August	6.30 (6.65)	3.86 (1131)	1.63
September	3.34 (3.52)	1.982 (581)	1.69
Seasonal total	22.66 (23.90)	13.82 (4051)	1.64

excess of 20%. If cycling losses are neglected, the system performance should have been better than the rating point because outside air temperatures during the test-year cooling season seldom reached 95° F. For this evaluation, the cooling requirements are assumed to be identical.

6.4 Monthly Performance

Table 6.13 presents monthly information for the major components of the ACES. Since the ice-melting by solar panel operation only occurred for a few hours during March, its contribution to the monthly performance has been omitted. The two most important items in Table 6.13 are the heat leak into the storage bin and the amount of cooling that had to be stored to meet the cooling loads after the ice was exhausted. For August, 9.29×10^6 Btu (9.80×10^9 J) of cooling was stored to meet the 6.3×10^6 Btu (6.65×10^9 J) of cooling required. This difference results from the heat leak into the bin, which was a direct loss in the off-peak air conditioning mode and the increased cooling loads resulting from operation of the compressor.

Table 6.13. Delivered loads and power consumption in the ACES house

	Loads [Btu × 10^{-6} (J × 10^{-9})]			Ice bin			Electrical consumption (kWhr)
	Heating	Cooling	Hot water	Ice inventory [Btu × 10^{-6} (J × 10^{-9})]	Estimated actual heat leakage [Btu × 10^{-6} (J × 10^{-9})]	Average tank temperature [°F (°C)]	
1977							
November	3.64 (3.84)		1.47 (1.55)	0	~0.75 (0.79)	45 (7.2)	505
December	8.06 (8.50)		2.67 (2.82)	0	1.51 (1.59)	40 (4.4)	984
1978							
January	12.63 (13.33)		2.33 (2.46)	2.59 (2.74)	1.94 (2.05)	32 (0)	1707
February	9.66 (10.19)		1.82 (1.92)	9.36 (9.88)	1.49 (1.57)	32 (0)	1246
March[a]	4.72 (4.98)		1.70 (1.79)	15.26 (16.10)	1.63 (1.72)	32 (0)	712
April	1.26 (1.33)	0.26 (0.27)	1.44 (1.52)	16.42 (17.32)	1.76 (1.86)	32 (0)	314
May	0.61 (0.64)	1.50 (1.58)	1.53 (1.61)	15.55 (16.41)	2.19 (2.31)	32 (0)	304
June	0.13 (0.14)[b]	4.98 (5.25)	1.22 (1.29)	14.11 (14.89)	2.60 (2.74)	32 (0)	280
July	0.11 (0.12)[b]	6.54 (6.50)	1.09 (1.15)	6.71 (7.08)	2.90 (3.60)	37 (2.7)	546
August		7.55 (7.97) 6.30 (6.65)[c]	2.85 (3.01)	0	1.74 (1.84)	45 (7.2)	1546
September		4.01 (4.23) 3.34 (3.52)[c]	1.67 (1.76)	0	~0.86 (0.91)	48 (8.8)	868
Total	40.82 (43.07)	24.84 (26.21)	19.79 (20.88)		19.37 (20.98)		9012

[a]0.263 × 10^6 Btu of ice melting by panel operation was done this month.
[b]Summer heating loads were the result of systems testing.
[c]Denotes actual loads on control house; these are also the approximate useful loads in the ACES house, the difference from the preceeding number being increased by internal loads from compressor operation.

6.5 Economy Cycle Operation

The economy cycle uses outside air to provide some or all of the cooling requirements of the structure. Use of outside air is normally accomplished by the opening of windows when it is cool outside. So that this concept could be evaluated, an automatic system was built into the ACES heating, ventilating, and air conditioning system.

Basically, the economy cycle feature is used under two conditions. First, when the enthalpy of the outside air was less than that of the house air and the house required cooling, dampers were positioned so that the return air was exhausted to the outside; the outside air was pulled through the cooling coil and delivered to the house. Because enthalpy of the outside air was less than that of the house air, the cooling load was reduced. Operation in the second condition occurred when the enthalpy of the outside air was below 27 Btu/lb (12.9×10^3 J/kg); under this condition, outside air could provide all the cooling. Therefore, the circulating brine system was turned off, and outside air was pulled in through the coil, and delivered to the house. House air was exhausted to the outside.

The economy cycle system operated from the end of May until early July 1978, at which time it was disabled. During this operational period, the system achieved some economy cycle cooling [about 100,000 Btu (1.05×10^8 J), or about 2% of load], but it was far less than the anticipated 25% of load. Investigation revealed two causes for this poor performance. First, the daily house demands for air conditioning did not coincide with the availability of low-enthalpy outside air. Second, the system had to wait until it was too late to enter this mode; it was not anticipatory.

Because economy cycle system malfunctions occurred occasionally, it was decided to disable the system to preserve the accuracy of heat flow information for ACES performance evaluation. This experience with the economy cycle concept is not conclusive; other control philosophies coupled with more reliable sensors might make the concept more effective.

7. OPERATIONAL EXPERIENCES

Several important characteristics of the ACES were learned from the data obtained during the test year. These items will be discussed as they relate to the major components of the ACES.

7.1 Mechanical Package

The mechanical package includes the major mechanical equipment (heat exchangers, pumps, valves, compressor, piping, etc.), the control equipment, and the control logic necessary to operate the ACES. With the exception of downtime resulting from improper testing of the system (improper testing of the high- and low-pressure cutout switches, for example), the system operated for the entire year without major difficulties.

The most significant problem with the mechanical equipment was insufficient heating capacity. During the design phase, a 17°F (−8.3°C) winter design-day temperature was chosen; this figure corresponds to the 97.5% value given in the *ASHRAE Handbook of Fundamentals*.[6] The system was capable of supplying all the heating requirements down to this temperature; however, there was a significant number of hours below the design-day temperature when the lack of capacity was noticeable by a drop in the house temperature [7°F (4°C)]. Some problems in comparing the performance of the ACES and control houses resulted. The control house had sufficient resistance-heating capacity to maintain house temperature under all conditions during the year; the ACES house did not. Plans have been made to correct this deficiency by installing a larger compressor.

The remaining problems involved logic errors built into the ACES controller. The original logic called for hot water to be produced when space heating occurred; this was accomplished by using the desuperheater to heat water continuously. Two problems for the ACES resulted. First, syphoning some of the system output into the hot water reduced the output of the system for space heating, thereby lessening capacity even more. Second, this logic caused excessively high hot-water temperatures, which resulted in unnecessary power consumption. Table 7.1 shows the energies delivered to each hot-water system.

Table 7.1. Actual hot-water delivered energies[a] of the ACES and control houses

	ACES [Btu $\times 10^6$ (J $\times 10^9$)]	Control house[b] [Btu $\times 10^6$ (J $\times 10^9$)]
1977		
November	1.47 (1.55)	1.49 (1.57)
December	2.67 (2.82)	1.67 (1.76)
1978		
January	2.33 (2.46)	1.84 (1.94)
February	1.82 (1.92)	1.61 (1.70)
March	1.70 (1.79)	1.66 (1.75)
April	1.44 (1.52)	1.39 (1.47)
May	1.53 (1.61)	1.35 (1.42)
June	1.22 (1.29)	1.10 (1.16)
July	1.09 (1.15)	1.10 (1.16)
August	2.85 (3.01)	1.01 (1.07)
September	1.67 (1.76)	0.576 (0.608)
Total	19.79 (20.88)	14.80 (15.61)

[a]70 gpd water draw (265 liters/day).

[b]Electric water heater maintained at 120°F (48.9°C) delivery temperature.

To alleviate this problem, the logic was changed so that water heating was only performed when the hot-water thermostat indicated a need for heat. This change allowed the entire output to be used for space heating most of the time, thereby increasing the output capacity of the system. Additionally, less energy was consumed because unnecessarily high hot-water temperatures were not produced.

Another logic error occurred during the off-peak air conditioning mode. The original logic called for the night heat-rejection mode to begin when the solar-panel temperature was below 80° F and the storage-bin temperature was above 36°F (2.2°C). Maintaining a low bin temperature resulted in significant amounts of energy being wasted because of increased heat leakage into the bin. Because the heat leakage into the bin is directly proportional to the temperature difference between the bin and the ground, the minimum bin temperature was raised to 44° F (26.7°C) which resulted in decreased heat leak and improved performance in the night heat-rejection mode of operation.

7.2 Solar-Convector Panel

The solar-convector panel can perform dual functions in an ACES. The panel can provide ice-melting capabilities, or it can operate as the heat-rejection component of the compressor after the ice has been exhausted.

The ACES demonstration house, as constructed with the large bin, does not require any ice-melting capability for design-year weather conditions. The solar-convector panel for the demonstration house was sized solely to meet heat-rejection requirements.

7.2.1 Panel heat collection

Although the loads delivered by the ACES during the test year greatly exceeded the original design loads, the bin heat capacity was never exhausted. For this reason, the panel was not used to collect large amounts of energy for melting ice.

The capability of the panel to collect energy was evaluated during a three-day test period (March 5, 6, and 7) when the system was forced into the ice-melt mode. Daytime, outdoor, ambient-air temperatures ranged from 19 to 40° F (−1.2 to +4.4° C). The panel was capable of collecting significant quantities of energy at all temperatures above 28° F (−2.2° C) while the sun was shining. On a day averaging 28° F (−2.2° C), the panel demonstrated a capability of collecting 40 Btu h^{-1} ft^{-2} (45.4×10^4 J h^{-1} m^{-2}). The following day, when the temperature averaged 38° F (3.3° C), the average collection rate was 114 Btu h^{-1} ft^{-2} (129.5×10^4 J h^{-1} m^{-2}). Total heat collection for the test period amounted to 0.26×10^6 Btu (0.274×10^9 J). Because of the brief period of operation in the collection mode, no conclusions can be made about the long-term effectiveness of this panel. The panel appears, however, to operate as theory predicts.

7.2.2 Night heat rejection

After July 27, the system was operated in the night heat-rejection mode. This mode of operation subjected the panel to a variety of conditions. The most severe operating condition for the panel occurs when no wind is present. The original panel design assumed a heat-transfer coefficient of 2.0 Btu h^{-1} ft^{-2} °F^{-1} (11.36 W m^{-2} K^{-1}). Further laboratory experimental work has shown that a heat-transfer coefficient of 1.46 Btu h^{-1} ft^{-2} °F^{-1} (8.29 W m^{-2} K^{-1}) is more realistic for near-calm air conditions.

Performance during the latter portion of the test year indicates that a heat-transfer coefficient of 1.3 Btu h^{-1} ft^{-2} °F^{-1} (7.38 W m^{-2} K^{-1}) was achieved in near-calm conditions and that the heat-transfer rate increased severalfold when a wind was blowing.

7.3 Ice-Water Storage Bin

Probably the most significant thing learned about ACES during the year's operation was the magnitude of the heat leak into the bin. A great deal of time and effort was spent during the design phase to predict the heat leakage. The decision not to install heavy insulation around the bin proved to be a mistake.

The actual heat leakage into the bin was about 60,000 to 95,000 Btu/day (63.3 to 100×10^6 J/d), which is roughly equivalent to melting 0.21 to 0.33 tons of ice per day. This leakage was more than twice the amount predicted.

The early calculations were based solely on the conduction of heat through the ground. A substantial amount of apparent heat transfer through the ground can be caused by moisture migration under favorable conditions. It is not possible to isolate the amount of increased heat leakage induced by moisture migration during the test year.

During the test year, a noticeable deflection in the heat-exchanger support structure was noticed. The structure deflection was carefully monitored for the remainder of the year to determine if there was a problem. The deflection measurements correlated linearly indicating that failure was not eminent; and, because the coils and support structure were replaced, it will not be possible to determine whether after this test year, this would have been a long-term problem.

When the system was shut down at completion of the test year, the ice-bin batting insulation was removed and found to be extremely wet from condensation. This problem was corrected by replacing the batting with Styrofoam closed-cell insulation, which does not absorb moisture.

7.4 Water Conditioning

The ACES bin and mechanical package contain various metals in direct contact with either the methanol-water brine solution or the bin water. Water conditioning to prevent bacterial growth and corrosion was found to be necessary very early in this experiment. Corrosion caused failure of the original aluminum bin coil during the first year of operation. Since that coil was replaced, effectiveness of the water treatments used to correct deficiencies has been monitored. An assessment of these treatments follows.

7.4.1 Brine system

The brine system contains several different materials in contact with both hot and cold brine solutions. A treatment for the brine system developed in late 1976 was designed to inhibit corrosion in three metal systems: aluminum, copper, and iron. The following table shows the desired concentrations of the inhibitors and the concentrations used.

Material	Concentration (ppm)
Nitrite-borate	1000
Silicate	500
Toluolytriazole	5
Chemtrol 19	30
Boric acid	(Adjust pH to 7.5–8.0)

In January 1977, the methanol-water brine system was treated with a formulation of nitrite-borate, silicate, toluolytriazole, boric acid, and microbiocide materials. Aluminum, copper, and iron corrosion coupons inserted in the brine system were removed after 1000 h of operation, and the corrosion rates were found to be negligible. Additional corrosion coupons, which were installed in the system on December 13, 1977, and removed on April 17, 1978, and September 18, 1978, were analyzed for corrosion. The following table presents the results of the corrosion rate evaluations.

Coupon material	Yearly corrosion rate	
	inches	millimeters
Aluminum	0.0049	0.123
Copper	0.0008	0.020
Iron	0.0044	0.110

These corrosion rates are considered negligible; however, small amounts of corrosion on the aluminum coupon occurred at the edges where the metal was somewhat stressed due to "punching out" the coupon.

7.4.2 Ice-bin water

After the ice bin was filled, the water was treated with 1 to 2 gal of 5% sodium hypochlorite (Clorox)(to achieve 1 ppm of free chlorine) to prevent bacterial growth. This treatment was apparently successful because the water has remained clear, which is an indication of the absence of bacteria. The water's pH was adjusted to 7.5 by using hydrochloric acid.

Table 7.2 lists the results of detailed water chemistry analyses, both at the initial filling and near the time of maximum ice inventory. The ice-bin water contained traces of iron, zinc, and cadmium, probably from corrosion of the cadmium-coated steel hose clamp and the galvanized pipe hanger. Water treatment of the bin and the brine circuit apparently was successful.

Table 7.2. ACES ice-bin water analyses

Trace material	Original water charge	Ice-bin water @ 55 tons ice[a] (mg/liter)
Aluminum	<0.05	<0.05
Cadmium	<0.002	<0.004
Copper	0.004	0.01
Iron	<0.06	0.12
Lead	<0.01	0.02
Zinc	0.04	2.5
Total dissolved solids[b]	132	384
Total hardness[c] ($CaCO_3$)	66	157
P-Alk.	<1	<1
M-Alk. ($CaCO_3$)		93
Chloride	15	67
Sulfate		57

[a]With 55 tons of ice in the bin, the remaining water should have been concentrated approximately 2.94 times.

[b]Total dissolved solids (initial concentration) $132 \times 2.94 = 388$ (actually analyzed 384) mg/liter.

[c]Total hardness (initial concentration) $66 \times 2.94 = 194$ (actually analyzed 157) mg/liter.

8. SUBSEQUENT MODIFICATIONS

The ACES was shut down on September 18, 1978, so that modifications could be made before next year's operation. The modifications decided on were:

1. increasing the heating capacity of the ACES,
2. reducing the amount of heat-transfer surface in the bin,
3. adding more insulation to the bin to reduce heat leakage,
4. modifying control system logic to correct deficiencies.

Because of the design of the mechanical package, the capacity of the ACES may be increased by the installation of a larger compressor. A larger compressor was installed, and it is believed that the system will now be able to meet heating requirements down to ~8° F (−13° C) outside temperature [instead of the 17° F (−8.3° C) originally designed]. The larger compressor will facilitate comparisons between the ACES and the control house because the ACES house will now be able to maintain its inside temperature during even the coldest days.

To reduce the heat leakage of the storage bin, 4 in. (0.1 m) of Styrofoam insulation was added to the sidewalls, 7-3/4 in. (0.2 m) was added to the top, and 6-3/4 in. (0.15 m) was placed over the floor. The overall R-value was thus raised to at least 37 on all surfaces; the added insulation is expected to reduce heat losses drastically.

The original bin design contained about 1800 lin ft (533 m) of tubing. Later economic analyses revealed that this amount was excessive from the standpoint of a cost/benefit ratio and that the design should have been based on 600 lin ft (184 m). Although the excessive amount of tubing was not detrimental from an operational point of view, a decision was made to reduce the tubing to 1300 lin ft (400 m) as a concession to cost-benefit considerations. Thirteen hundred lin ft (400 m) of 5/8 in. OD copper tubing was installed in the bin. As a final measure, the tank liner was replaced with another heavier material.

The data acquisition system (DAS) was also upgraded. The new DAS contains an improved calculator (computer) and has twice the data-point capacity of the old system. These improvements will allow for additional experiments to be carried out at the TECH complex. Control logic changes were also initiated to correct the shortcomings discovered during the year.

9. COMPARATIVE ASSESSMENT OF ACES PERFORMANCE

This report has compared the performance of the ACES house with that of the control house, which employs a resistance-heating and hot-water system with central air conditioning. The comparison has proved extremely useful in determining the equality of the loads on the two houses and for comparing the performance of ACES with that of a resistance-heating system. Current plans call for the ACES house to be experimentally compared with a conventional air/air heat-pump system for the next year's operation to allow a direct evaluation of the ACES in relation to a heat pump. We have estimated, however, what the relative performance of the ACES and conventional heat-pump system would have been during the test year of this report. For comparisons a fossil-fueled system with central air-conditioning was also considered.

Table 9.1 gives the seasonal performance factors for the two systems in the heating season, cooling season, and for water heating. The systems are:

1. electric resistance—electric-resistance space and water heating with central air conditioning,
2. fossil fueled—fossil-fueled space and water heating with central air conditioning.

Table 9.2 gives the loads and energy consumption of the ACES for the test year, along with the assumed consumption for the other systems. Energy consumptions are presented in kilowatt-hours, although the two different energy sources are typically represented in different units. The primary importance of this table is to show the relative energy consumptions of the three systems depicted.

Table 9.1. Seasonal performance factors of two heating and cooling systems

	Resistance heating	Fossil fueled
Cooling	1.64	1.64
Heating	1	0.6
Hot water	1	0.6

Table 9.2. Energy consumption for alternative systems

	Resistance heating (kWhr)	Fossil fueled (kWhr)	ACES (kWhr)
Heating	11,954	19,523	
Cooling	4,427	4,427	9,012
Hot water	5,798	9,663	
Total	22,179	33,613	9,012

Table 9.3 presents the amounts of natural resources that were consumed to meet the given loads. A 28% network efficiency was assumed for electricity consumption and a 10% transportation loss was attributed to the fossil fuel consumed. This table gives an estimate of the quantities of nonrenewable resources consumed by the various systems to meet the same load.

Table 9.3. Resource consumption of alternative systems

	Loads [Btu × 10^{-6} (J × 10^{-9})]	Resource consumption [Btu × 10^{-6} (J × 10^{-9})]		
		Resistance	Fossil	ACES
Heating	40.80 (43.05)	145.7 (153.72)	75.55 (79.71)	
Cooling	24.78 (26.15)	53.96 (56.96)	53.96 (56.96)	109.85 (115.90)
Hot water	19.79 (20.88)	70.68 (74.57)	36.65 (38.67)	
Total	85.37 (90.08)	270.34 (285.25)	166.16 (175.34)	109.85 (115.90)

Additionally, we can assess the relative performance of a house equipped with a conventional heat pump and electric water heating by assuming only seasonal coefficients of performance (COP) for the heating season. For comparison, the heating-season COP for a conventional heat pump should be about 1.50 in Knoxville. The cooling-season cooling COP was 1.64 and, for water heating, 1.

These COPs would have resulted in a comparative test house consumption of 7809 kWhr for space heating, 4427 kWhr for space cooling, and 5798 kWhr for water heating. Total annual consumption would have been 18,034 kWhr to deliver loads identical to those met by the ACES, resulting in a resource consumption of 219.82 × 10^6 Btu (231.92 × 10^9 J), and in a resource efficiency of 39%. Resource efficiency is the percentage of a natural resource which is utilized at the end use.

Table 9.4 gives the resource efficiency for each system as a whole. Resource efficiency is defined as delivered load divided by natural resources consumed. As can be seen, the ACES had a significantly higher resource efficiency than did the other systems. In addition, the ACES is not consuming a critical fossil fuel. ACES electricity can be produced from a number of natural resources, many of which are not as limited or as valuable as are the fossil fuels, gas, and oil.

Table 9.4 Resource efficiency (in %) of alternative systems

Resistance heating	Heat[a] pump	Fossil fired	ACES
32	39	51	74

[a]Assumed performance.

The primary purposes of this section were (1) to demonstrate that the ACES is indeed an energy-conserving system in that it reduces the total amount of energy that must be consumed, and (2) to point out that the sources of energy for the ACES can be derived from resources less valuable than those used for fossil systems.

10. CONCLUSIONS

The basic conclusion regarding the year's operation was that the test was successful. Both the ACES and the DAS proved to be most reliable and generally, their performance met or exceeded expectations, with the exception of bin heat leakage and night heat rejection.

Areas in which problems occurred can be summarized briefly as follows.

1. There was an unexpectedly large heat leak into the storage bin (a well-insulated bin is required to accomplish interseasonal energy transfer).
2. There were control logic errors that caused excessive amounts of energy to be used in the night heat-rejection mode.
3. Incorrect logic caused excessive amounts of hot water to be produced during the coldest portion of the winter.
4. There were increased cooling loads caused by waste heat from the mechanical package in the night heat-rejection mode.
5. There was insufficient heating capacity to maintain house temperature during the coldest periods.
6. There was condensation on the insulation in the storage bin.

Other conclusions about the performance of the ACES have already been mentioned in this report but are reiterated here.

1. Winter energy consumption was reduced by 63%.
2. Winter peak loads were reduced by 74%.
3. Summer energy consumption was reduced by 37%.
4. Summer peak utility loads were reduced by between 33% and 82%.
5. Annual energy consumption was reduced by 56%.
6. The ACES exhibited no cyclic losses during the heating season.
7. Stored ice provided 2274 kWhr of equivalent cooling purchased energy.

All these comparisons are in relation to the control house, which used electric-resistance heating and hot water, and a central air conditioning system. Future operation will allow for these same comparisons to be made against a conventional air/air heat-pump installation.

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ACKNOWLEDGMENTS

The authors want to recognize other members of the project team without whose efforts this work would not have been possible: J. C. Moyers served as group leader during this period and provided administrative support; E. C. Hise served as task leader and provided technical support; G. G. Slaughter designed, installed, and calibrated the data acquisition system used for this project; V. R. Brantley was the onsite technician who maintained the house and carefully monitored the equipment; Jean Bledsoe assisted in data reduction; and H. C. Fischer, who first brought the annual cycle energy system (ACES) concept to the attention of the Oak Ridge National Laboratory (ORNL).

APPENDICES

Appendix A

DAS DATA POINTS

This appendix contains information on the data aquisition system (DAS) used at the Tech Complex. Included is a list of all the data points measured and those points calculated from the measured values. The DAS has been operational since August 1976. Since that time, several major modifications were made in an effort to improve the overall reliability of the system. During the test year, the DAS as a whole demonstrated a 96% uptime by running 24 hr per day, 7 days per week. The overall accuracy of the DAS was about 5% during this period. This estimate considers the general operational availability, the time when significant failures occurred, and the sensor-to-storage accuracy measured by testing.

Table A.1 is a list of the data points along with the units and types of signals measured.

Table A.1. Data points

Signal	Unit	Type
1. Ice-bin level	Tons	Analog
2. Bin temperature low	°F	Analog
3. ACES house dry bulb	°F	Analog
4. Control house crawl-space temperature	°F	Analog
5. Bin coil inlet	°F	Analog
6. Bin coil outlet	°F	Analog
7. Fan coil inlet	°F	Analog
8. Fan coil outlet	°F	Analog
9. Solar panel inlet	°F	Analog
10. Solar panel outlet	°F	Analog
11. Domestic-hot-water inlet	°F	Analog
12. Domestic-hot-water outlet	°F	Analog
13. Bin temperature high	°F	Analog
14. Control house dry bulb	°F	Analog
15. Uncirculated panel dry bulb	°F	Analog
16. Control house wet bulb	°F	Analog
17. Not used		
18. Solar pump 1 inlet	°F	Analog
19. Solar collector inlet	°F	Analog
20. Solar collector outlet	°F	Analog
21. Solar house hot water tank high	°F	Analog
22. Solar house hot water tank middle 1	°F	Analog
23. Solar house hot water tank middle 2	°F	Analog
24. Solar house hot water tank low	°F	Analog
25. Solar house collector tank 1 high	°F	Analog
26. Solar house collector tank 1 low	°F	Analog
27. Solar house fan coil inlet	°F	Analog
28. Solar house fan coil outlet	°F	Analog
29. ACES house wet bulb	°F	Analog
30. Collector tank 2 high	°F	Analog
31. Collector tank 2 low	°F	Analog
32. Solar house dry bulb	°F	Analog
33. Solar house wet bulb	°F	Analog
34. Outside dry bulb	°F	Analog
35. Outside dew point	°F	Analog
36. Barometric pressure	in. Hg.	Analog
37. Not used		
38. Wind direction	deg	Analog
39. Not used		

Table A.1 (continued)

Signal	Unit	Type
40. Rainfall	in.	Analog
41. Bin heat output	Btu	Integrated
42. Bin heat input	Btu	Integrated
43. ACES panel input	Btu	Integrated
44. ACES panel output	Btu	Integrated
45. Cooling pump energy input	Wh	Integrated
46. Heating pump energy input	Wh	Integrated
47. Hot water pump energy input	Wh	Integrated
48. Compressor energy input	Wh	Integrated
49. Fan energy input	Wh	Integrated
50. ACES house energy input	Wh	Integrated
51. ACES fan coil output	Btu	Integrated
52. ACES fan coil input	Btu	Integrated
53. ACES hot water output	Btu	Integrated
54. Control house heat output	Wh	Integrated
55. Digital data	N/A[a]	N/A
56. Solar collector heat input	Btu	Integrated
57. Solar fan coil output	Btu	Integrated
58. Solar hot water output	Btu	Integrated
59. Solar house energy input	W	Integrated
60. Solar house fan energy input	W	Integrated
61. Solar hot water pump energy	W	Integrated
62. Solar collector pump energy	W	Integrated
63. Solar duct heater energy	W	Integrated
64. Solar fan coil pump energy	W	Integrated
65. Delta enthalpy	Btu/lb	Integrated
66. Solar radiation incident	Btu/ft^2-h	Integrated
67. Average wind speed	mph	Integrated
68. Control house energy input	Wh	Integrated
69. Economy cycle elapsed time	sec	Integrated
70. Digital data	N/A	N/A
71. Economy cooling (calculated)	Btu	Integrated
72. Integration time for scan	sec	Integrated
73. Date, month/day	N/A	N/A
74. Date, year	N/A	N/A

[a]Not applicable.

Appendix B

DATA PLOTS PRODUCED BY THE DAS

Table B.1. Equipment and weather data summary, week beginning 7/24/78

Average outside dry-bulb temperature, °C	23.96
Maximum outside dry-bulb temperature, °C	32.34
Minimum outside dry-bulb temperature, °C	18.16
Degree days heating referenced to 18°C	0.00
Degree days heating referenced to 16°C	0.00
Degree days heating referenced to 14°C	0.00
Degree hours cooling referenced to 21°C	552.66
Degree hours cooling referenced to 23°C	368.31
Degree hours cooling referenced to 25°C	223.75
Total heating load delivered, kWh	0.00
Total cooling load delivered, kWh	307.35
Total economy load delivered, kWh	0.00
Total hot water load delivered, kWh	81.41
Maximum heating load delivered, W	0.00
Maximum cooling load delivered, W	6453.69
Maximum economy load delivered, W	0.00
Maximum hot water load delivered, W	2004.69
Total ACES power consumption, kWh	248.29
Total control house equipment power consumption, kWh	304.68

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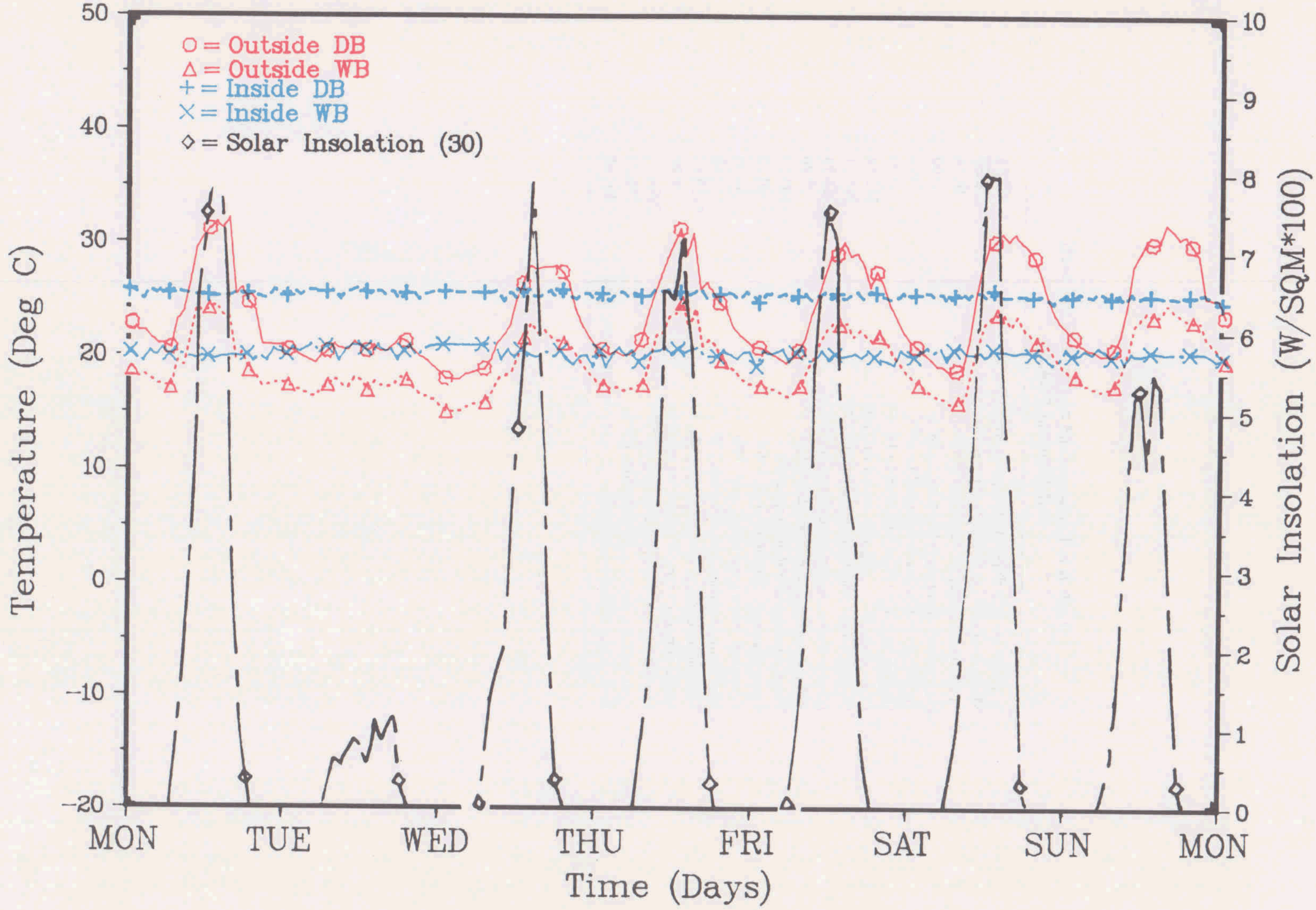

Fig. B.1. Temperature and insolation plot, week beginning 7/24/78.

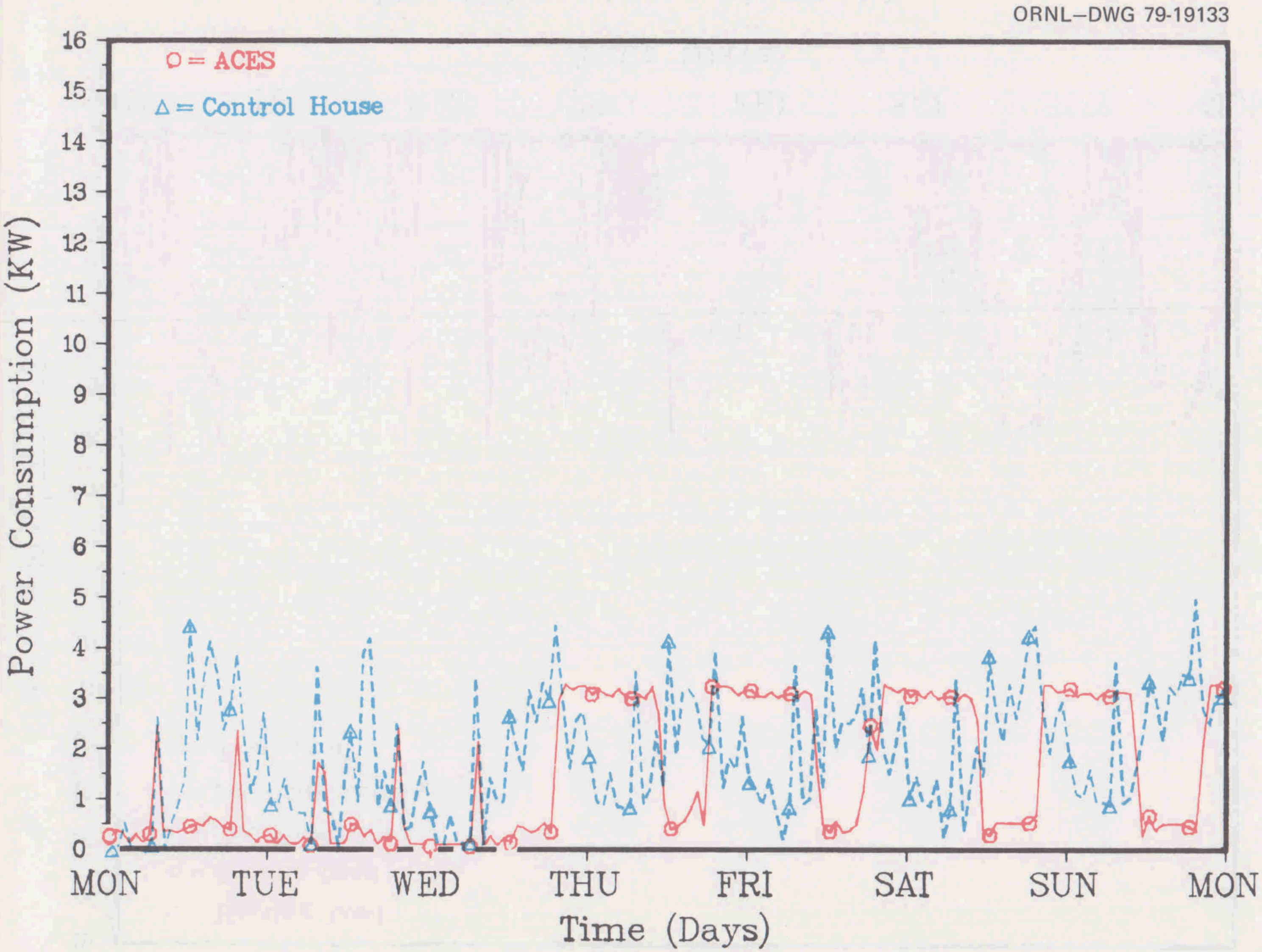

Fig. B.2. Power-consumption plot for the ACES and control houses, week beginning 7/24/78.

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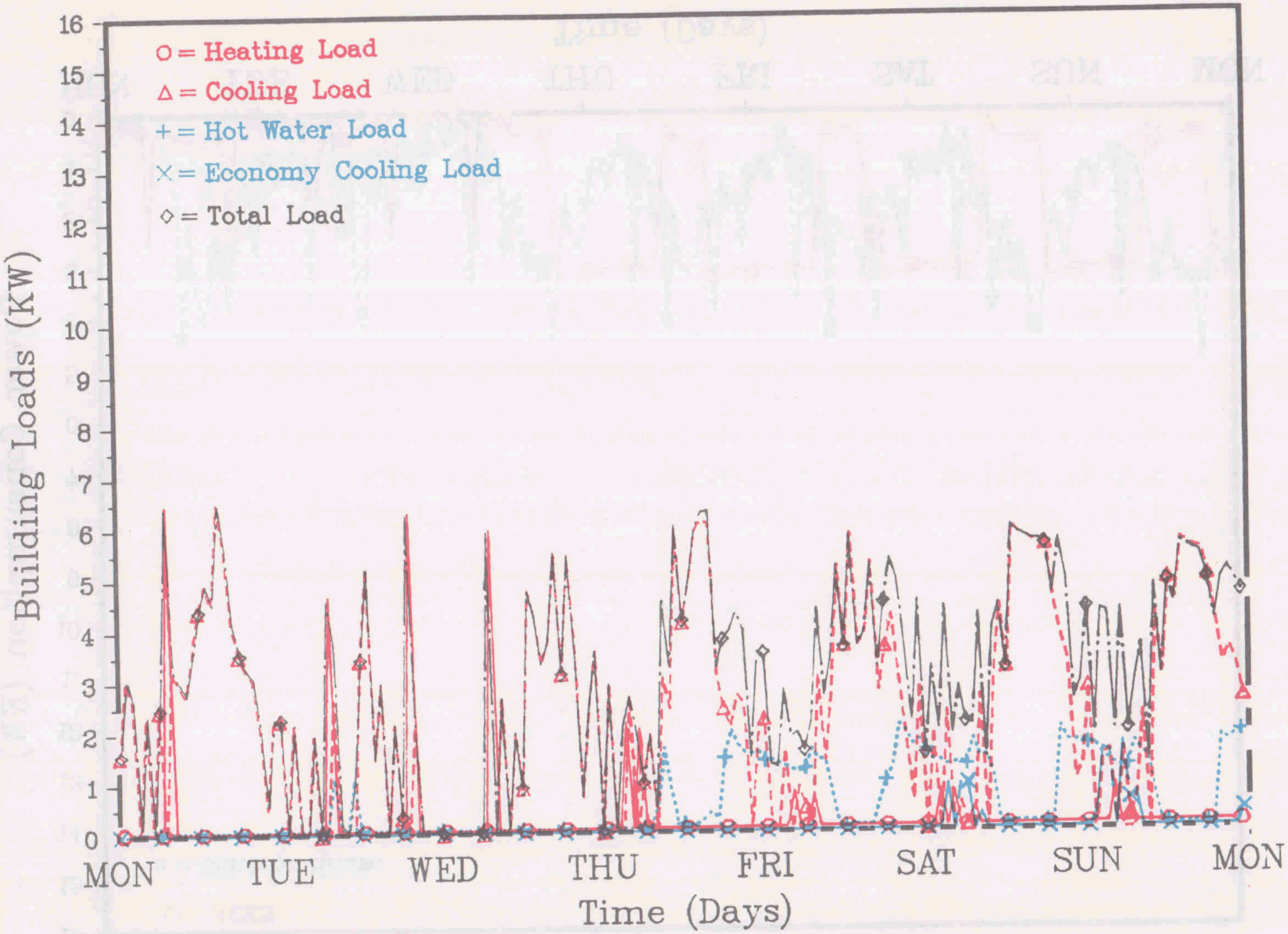

Fig. B.3. ACES building thermal-load plot, week beginning 7/24/78.

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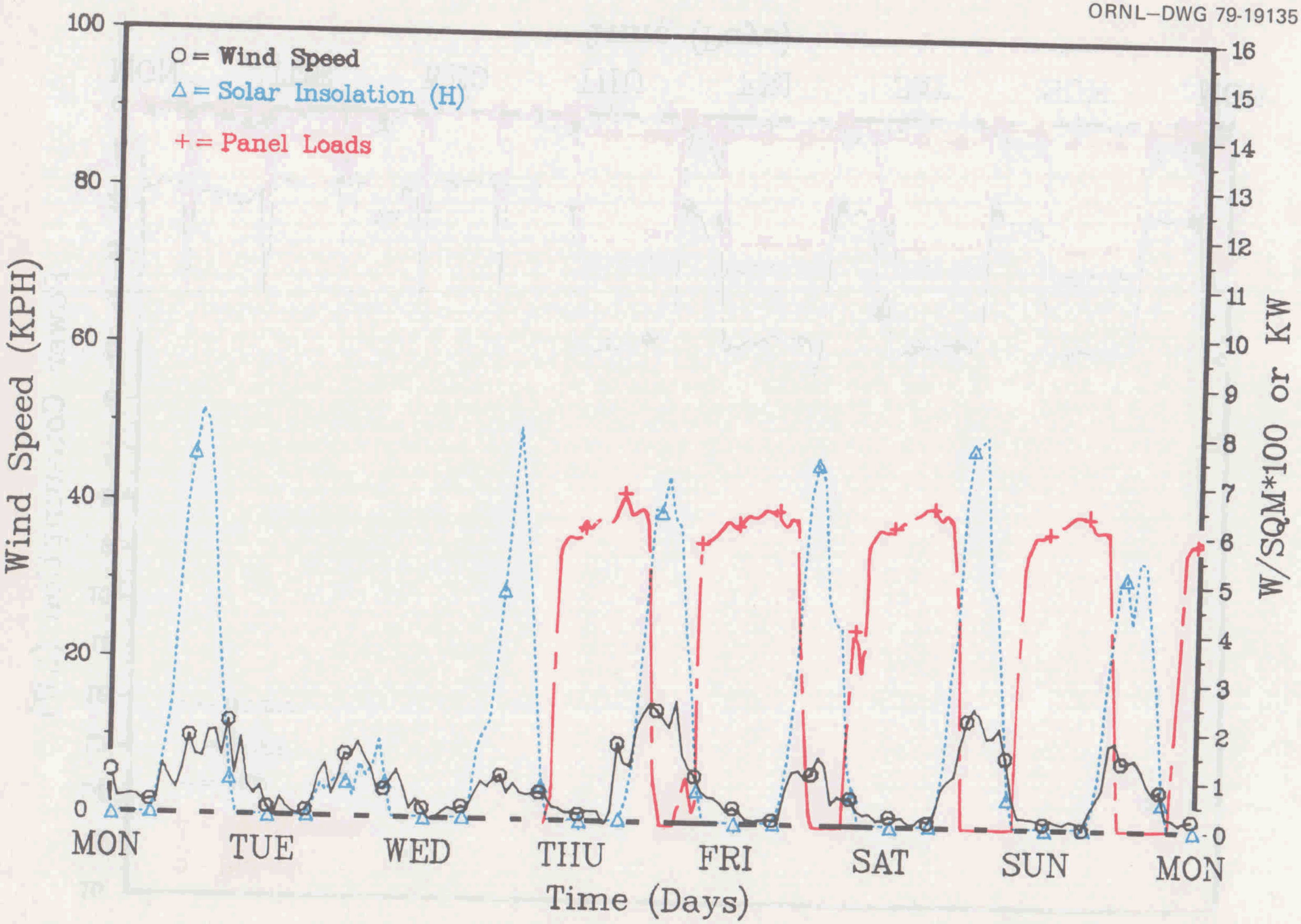

Fig. B.4. Wind speed and solar-insolation plot, week beginning 7/24/78.

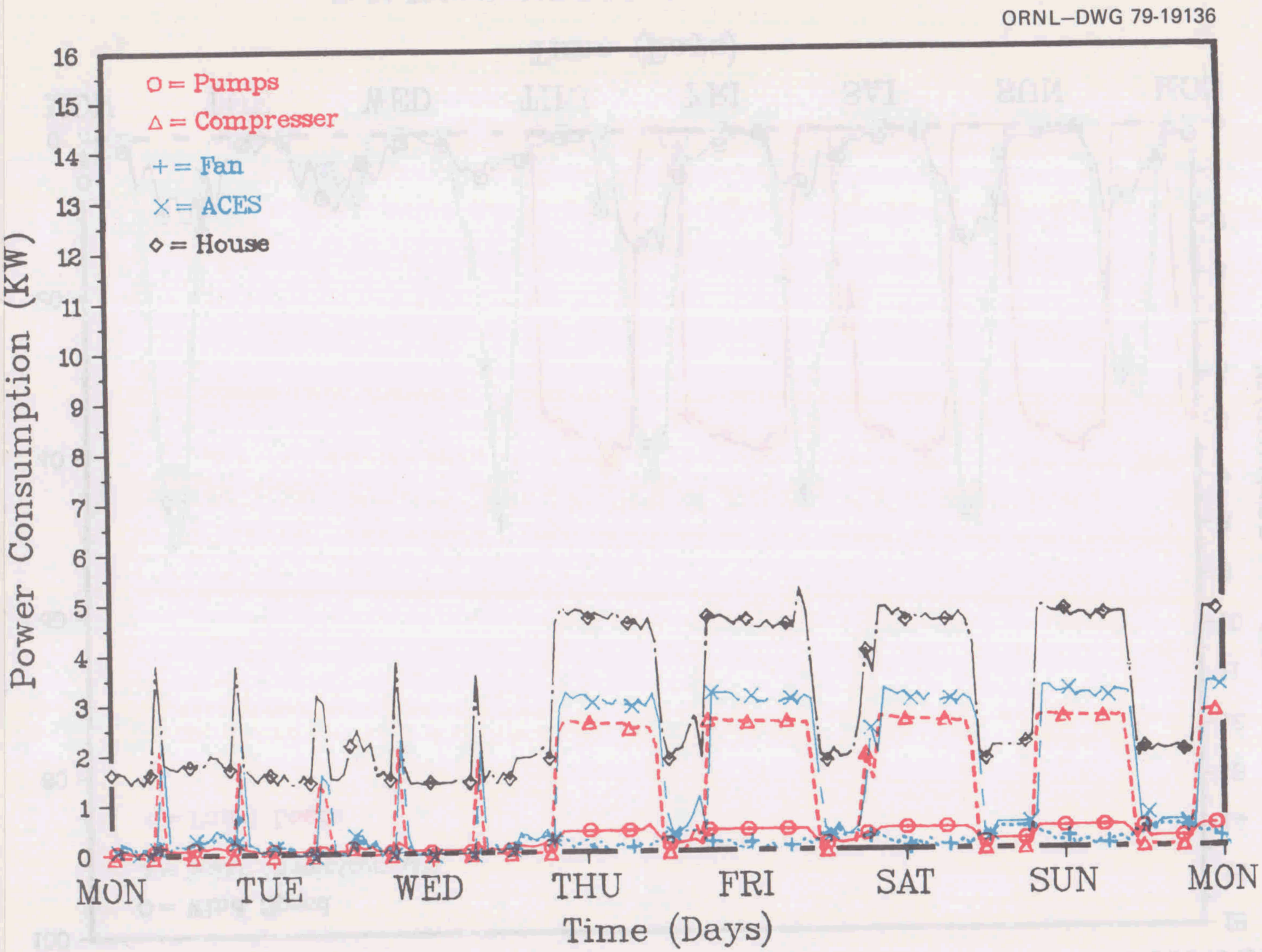

Fig. B.5. Electrical consumption plot for ACES components, week beginning 7/24/78.

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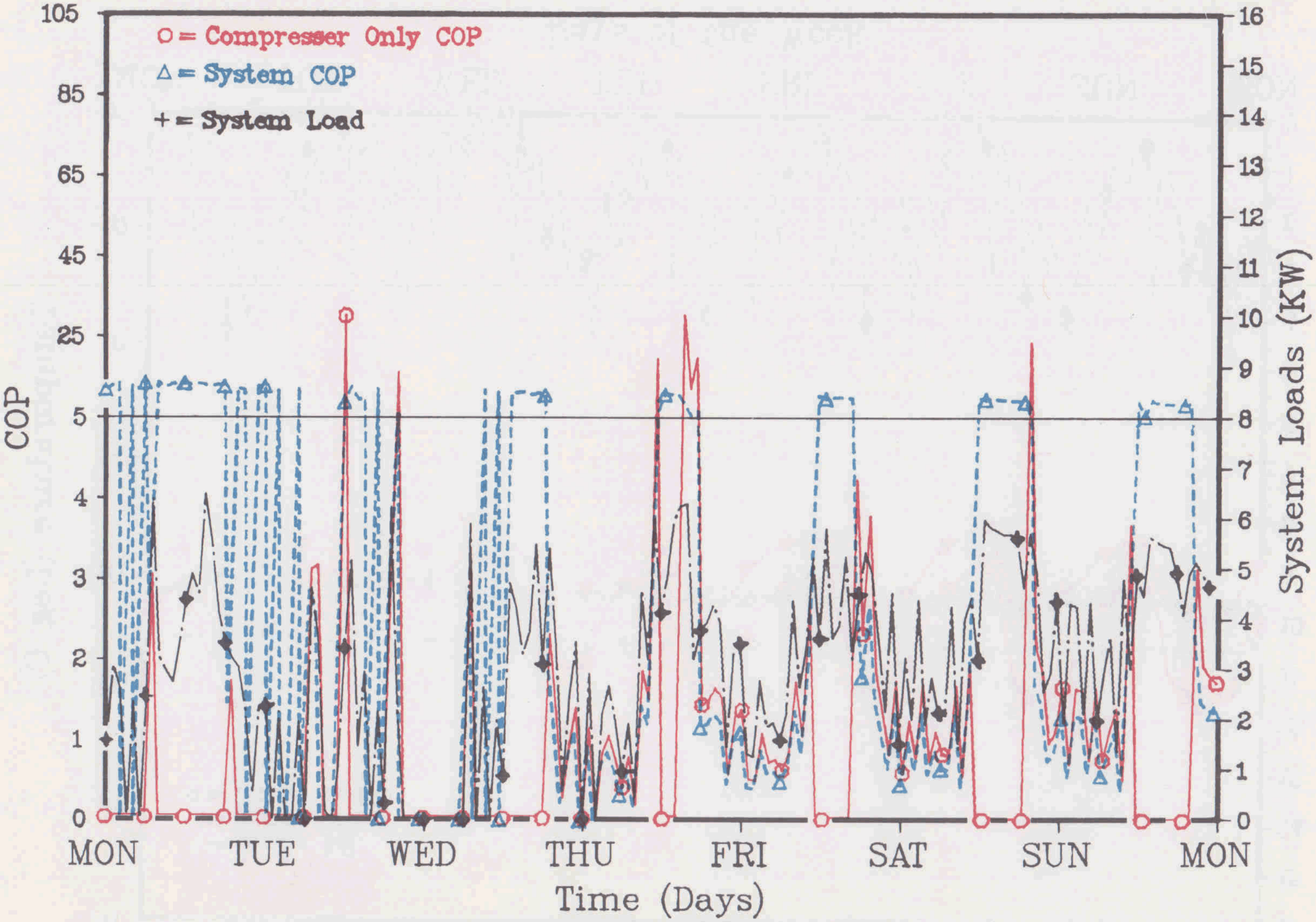

Fig. B.6. ACES coefficients of performance and loads plot, week beginning 7/24/78.

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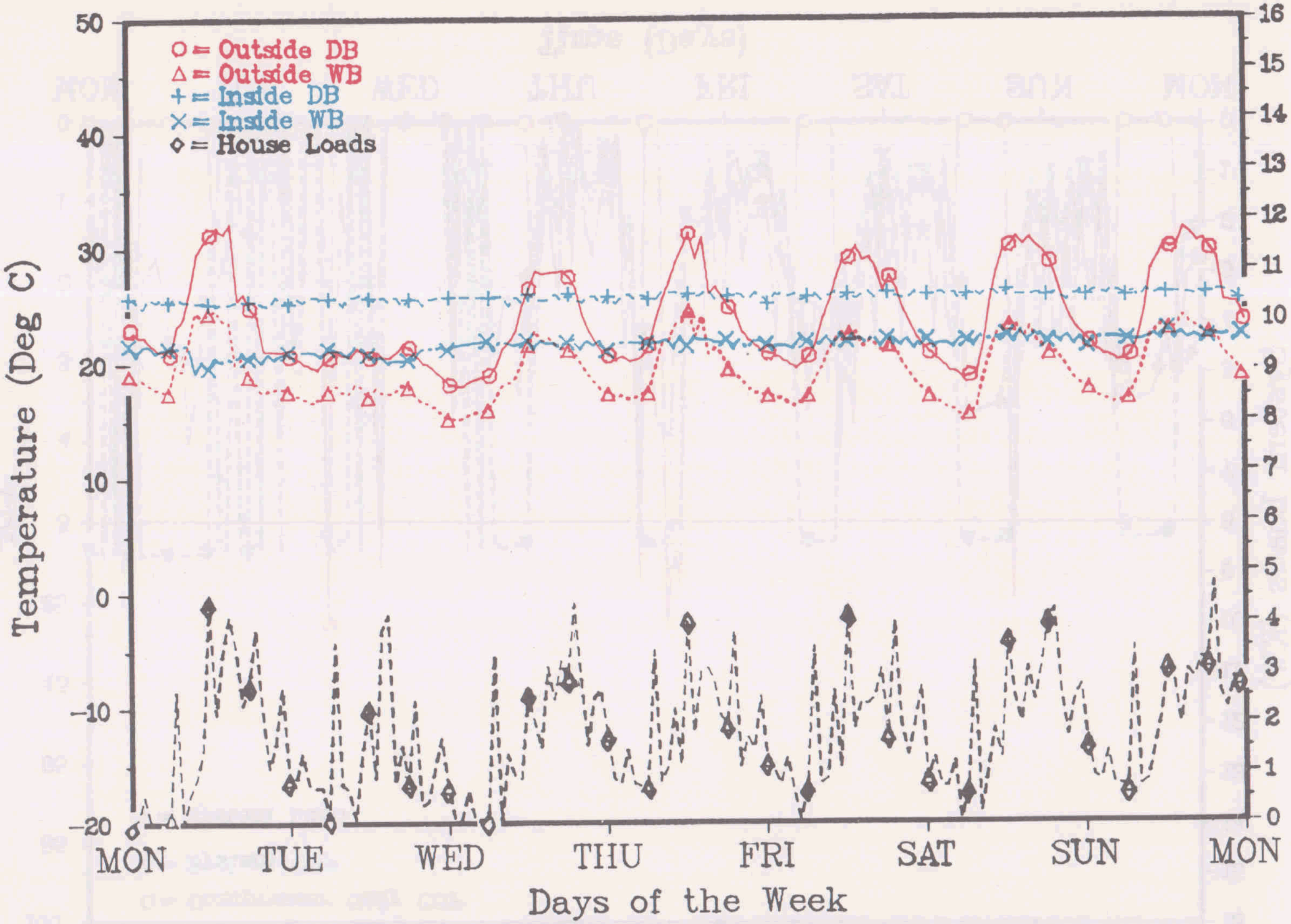

Fig. B.7. Control-house weather and power-consumption plot, week beginning 7/24/78.

Appendix C

SAMPLE CALCULATIONS

C.1 Space Heating with Hot Water COP

Because the mechanical package is located inside the house, all the electrical input goes toward heating the house. For this reason the heat output equals the heat removed from the bin plus the electrical input. The COP is the heat output divided by the electrical input.

	Electrical inputs (Whr)		Energy outputs (Whr)
Compressor	2260	Heat from bin	5160
Fan	380	Electrical consumption	3029
Pumps	389		
Total	3029	Total	8179

COP = Output/Input = 8179/3029 = 2.70

C.2 Space Cooling (Ice Melting)

Useful cooling is defined as the energy extracted by the fan coil minus the energy input to the fan; the COP is the useful energy divided by the energy input.

	Useful cooling (Whr)		Energy input (Whr)
Heat extracted by the fan coil	7270	Fan	380
Less fan energy + 3/4 pump	504	Pump	166
Total	6766	Total	546

COP = Useful Cooling/Energy Input = 6766/546 = 12.4

C.3 Water Heating*

The COP for water heating is the heat into the hot water divided by the electrical input. In this mode, no credit is taken for the heat given off by the compressor and pumps.

	Heat output (Whr)		Electrical input (Whr)
		Compressor	1830
		Pumps	246
Total	5180	Total	2076

COP = Output/Electrical Input = 5180/2076 = 2.50

*In actual practice hot water needs are generally satisfied by simultaneous production with space heating during most of the winter.

If, during the winter, a credit is taken for the heat given off by the electrical components, the COP is determined by:

	Heat output (Whr)
Heat removed from the bin	3340
Electrical input	2076
Total	5416

COP = Output/Input = 5416/2076 = 2.61

C.4 Night Heat Rejection

The COP for night heat rejection is best quantified by calculating the bin heat rejection COP(BR), but it can be defined in several ways. First, if only the hot water produced was considered, the COP would be the hot water produced divided by the electrical input:

		Electrical Input (Whr)
Hot water = 2100 Wh	Compressor	2730
	Pumps	386
	Total	3116

COP = Hot Water/Electrical Input = 2100/3116 = 0.67

If the energy stored in the bin was also considered useful, the COP would equal the stored cooling plus the hot water, divided by the electrical input. This definition excludes the energy lost by heat leakage into the bin because this is a function of how long the energy is stored. Calculation of the actual night heat-rejection COP(SC) must consider the loads delivered and the bin heat leakage. For an ideal hour without bin heat leakage and maximal hot water production, the COP would be:

	Useful Energy (Whr)
Hot water	2100
Stored cooling	6091
Total	8191

COP = Useful Energy/Electrical Input = 8191/3116 = 2.63

The bin heat rejection COP(BR) is

$$\mathrm{COP(BR)} = \frac{6091 - 0.25(166)}{2730 + 166 + 140} = 1.99 \, .$$

Appendix D

MECHANICAL PACKAGE DESIGN SPECIFICATIONS

The following list contains the design specifications for the mechanical package.

1. Compressor

a. The refrigerant compressor shall be Tecumseh model AH8532E, R-22, with a capacity of 250 to 260 lb/hr at 20°F (saturated) evaporating temperature and 110°F (saturated) condensing temperature.
b. The compressor electric circuit shall be provided with a starting capacitor sized to permit the compressor to start against a nonequalized, no-pump-down system at a differential of 226 psig to 41 psig (110 to 20°F).

2. Domestic-Hot-Water Desuperheater

a. The capacity of the desuperheater shall be 6000 Btu/hr when heating domestic hot water entering at 55°F, at a flow of 4 gpm, and with the desuperheating compressor discharge entering at 226 psig (230°F) and leaving as a desuperheated vapor at not less than 111°F.
b. The domestic-water-side pressure drop through the desuperheater shall not exceed 1 psi at a flow rate of 4 gpm.
c. The desuperheater should be arranged for countercurrent flow. (Valve CV1 will control the flow of domestic hot water through the desuperheater so that the discharge temperature is 120°F.)

3. Domestic-Hot-Water Condenser

a. The capacity of the condenser shall be 18,500 Btu/hr when heating domestic hot water entering at 55°F, at a flow rate of 4 gpm, and with the condensing compressor discharge refrigerant (entering from desuperheater) from 226 psig to 227 psig (111 to 100°F).
b. The domestic-water-side pressure drop through the domestic-hot-water condenser shall not exceed 1 psi at a flow rate of 4 gpm.
c. The domestic-hot-water condenser should be arranged for countercurrent flow. (Valve CV2 will bypass the domestic water flow around the domestic-hot-water condenser when the space-heating is operational. However, it will divert this flow through the domestic-hot-water condenser when the space-heating condenser is not operational and when the compressor is required for domestic-hot-water heating only.)

4. Space-Heating Condenser

a. Duty
   (1) When the domestic-hot-water desuperheater is at a capacity of 6000 Btu/hr. 18,500 Btu/hr when heating 8 gpm methanol and water (25 wt % methanol) from 97.23 to 102.36°F when condensing R-22 refrigerant entering at 226 psig (111°F).
   (2) When the domestic-hot-water desuperheater is not in operation. 24,500 Btu/hr when heating 8 gpm methanol and water (25 wt % methanol) from 97.23 to 104°F when condensing R-22 refrigerant entering at 226 psig (230°F).
b. The methanol-solution-side pressure drop shall not exceed 2 psi at a flow rate of 8 gpm. The refrigerant-methanol solution flow should be arranged for countercurrent flow.

5. Evaporator (Methanol-Brine Cooler)

a. The evaporator's capacity shall be 17,500 Btu/hr when cooling 12 gpm of methanol and water (25 wt % methanol) from 26.45 to 24.55°F when the R-22 evaporator temperature is 41 psig (20°F).
b. The methanol-solution-side pressure drop shall not exceed 4 psi at a flow rate of 20 gpm.

6. Shell and Tube Arrangement General

a. All shell and tube vessels—desuperheater, both condensers, and evaporator (brine cooler)—shall be ASME construction: 300 psig refrigerant side and 125 psig on water or methanol side. The tube bundles shall be copper of not less than 0.035-in. wall thickness.
b. Preferably, the shells shall be arranged in a vertical tier, stacked top to bottom in sequence of refrigerant flow.
c. The tiered shell stack should be located at the rear of the mechanical package assembly in supports that permit easy removal of a single shell from the right end of the mechanical package (where removal space is available).

7. Accumulator

The vendor shall submit alternative proposals for the accumulator—with and without internal heat exchanger for liquid subcooling.

a. Proposal A without subcooling
Under this proposal, the accumulator shall be a shell of sufficient size to accept the entire refrigerant charge to protect the system from liquid overflow onto the compressor. The internal suction pipe shall have an oil lifter.
b. Proposal B with subcooling
Under this proposal, the accumulator shall be a shell-and-coil design, with the coil designed to subcool 226-psig high-pressure liquid from an entering temperature of 110°F to a leaving temperature of 60°F at an evaporator temperature in the shell of 20°F, with a shell of sufficient size to accept the entire refrigerant charge to protect the system from liquid overflow onto the compressor. The internal suction pipe shall have an oil lifter.
c. Under Proposal A, the vendor shall include manual shutoff valves and a bypass to allow for test runs of the system, both with and without liquid subcooling, so that ORNL can ascertain the relative cycle efficiency with subcooling.

8. Fan-Coil Air Unit

a. The unit furnished will be one Carrier model 42BH-1 horizontal draw-through air conditioner complete with 16-gauge casing, 2 side access panels, 4-row cooling coil with 1/2-in.-OD copper tube/aluminum plate fins tested for 350 psig with manual air vent, insulated drain pan, and discharge and return duct collars. The permanent flat filter will be removable from the side.
b. The unit is to be furnished with Vee-belt drive for field mounting of motor. The Vee-belt drive will be sized so that the unit can deliver 800 cfm at 1 in. total pressure.
c. The vendor shall furnish a separate, additional variable-pitch motor sheave (for field adjustment).

9. Fan-Coil Motor

The unit furnished will be one 1/3-hp, 1757-rpm, 115-V, single-phase, 60-cycle, high-efficiency, energy-saver fan motor as manufactured by Gould, Inc., Century Electric Motor Division, St. Louis, Missouri.

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