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ACES: Final Performance Report December 1, 1978, Through September 15, 1980

Van D. Baxter

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ACES: FINAL PERFORMANCE REPORT
DECEMBER 1, 1978, THROUGH SEPTEMBER 15, 1980

Van D. Baxter

DEPARTMENT OF ENERGY
Division of Buildings and Community Systems

Date Published: April 1981

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
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ACES: FINAL PERFORMANCE REPORT
DECEMBER 1, 1978, THROUGH SEPTEMBER 15, 1980

Van D. Baxter

ABSTRACT

A single-family residence near Knoxville, Tennessee, has been used for three years to demonstrate the energy conservation potential of the Annual Cycle Energy System (ACES). For the past two years the performance of the ACES has been compared with that of two different air-to-air heat pumps in an identical house, the control house. The two heat pumps had ARI rated COPs of 2.46 and 3.11 at 8.3°C, representing both standard and state-of-the-art-quality heat pumps now on the market. Control house hot water was supplied by an electric resistance heater each year, while hot water for the ACES house was supplied by the ACES mechanical package.

The ACES has achieved nearly all of its theoretical performance predictions and has verified its design criteria. This system has delivered residential heating and cooling services while consuming only 57% as much electricity as the best conventional alternative tested, and it is the best system for conserving electrical energy yet studied.

First-year performance has been summarized in an earlier report (ORNL/CON-42). Performance results for the second year (December 1978 through September 1979) showed that the ACES delivered 32.85 GJ of space heating, 19.17 GJ of space cooling, and 15.81 GJ of water heating while consuming 6719 kWh of electricity for an ACOP of 2.80. The control house (with the ARI 2.46 heat pump) consumed 12,853 kWh of electricity, delivering 32.85 GJ of space heating, 19.18 GJ of space cooling, and 13.74 GJ of water heating for an ACOP of 1.42. Third-year results (December 1979 through September 15, 1980) show that the ACES delivered 34.40 GJ of space heating, 22.65 GJ of space cooling, and 14.34 GJ of water heating while consuming 6447 kWh of electricity for an ACOP of 3.08. Using the ARI 3.11 heat pump, the control house consumed 11,358 kWh while delivering 34.29 GJ of space heating, 22.70 GJ of space cooling, and 13.57 GJ of water heating for an ACOP of 1.73.

In addition to reducing consumption, the ACES significantly reduced integrated peak utility demands. Winter peak hourly consumption was reduced from about 12.0 kWh to 3.3 kWh, and the summer air-conditioning peak consumption was reduced from about 4.1 kWh to 0.6 kWh. Instantaneous summer water heating peaks were reduced from 4.5 kW to 2.8 kW.

The 2.46 ARI heat pump demonstrated heating and cooling SPFs of 1.58 and 1.64, respectively, and the 3.11 ARI heat pump demonstrated SPFs of 1.99 and 2.27.

Reinsulation of the ice storage bin reduced heat leakage rates by about 40% and resulted in increasing ground temperatures by an average of 5.6°C over the first-year levels. The added insulation has been found to be necessary for an efficient ACES.

Measurement of the cooling loads in both houses during the summer of 1979 demonstrated equivalence of the thermal envelopes during the cooling season.

1. INTRODUCTION

The Annual Cycle Energy System (ACES) program is an ongoing RD&D effort of the Oak Ridge National Laboratory sponsored by the U.S. Department of Energy. Its primary purpose is to study, develop, demonstrate, and evaluate the ACES engineering performance and commercial viability.

The ACES promotes energy conservation in buildings by providing space heating and cooling and domestic hot water by means of a unidirectional heat pump with thermal storage and, where required, assistance from solar or other supplementary heat sources. The heat pump extracts energy from an insulated storage tank which, at the beginning of the heating season, contains only water. As the heating season progresses, the water is converted to ice, which is stored to provide summer air conditioning.

Experimental work in the program has been directed toward improving the efficiency and reliability of ACES components, such as the ice-maker heat pump and the solar/convector panels, and of the complete system. Field demonstrations have been designed to provide operational data for systems ranging from the full ACES, which provides the maximum possible interseasonal energy transfer, to smaller systems which may have tanks large enough to store only a few weeks' supply of energy. An ACES design handbook[1] has been developed for residential buildings so that the results of this program can be utilized by designers and builders across the country.

Analytical programs are being pursued to determine the economic competitiveness of the ACES with more conventional systems, to determine the marketability of the ACES in different areas of the country, and to determine the impact that widespread utilization of the ACES concept would have upon utility system loads. Computer design programs[2,3] have been developed and refined which accurately size the ACES components and determine the ACES configuration that is economically optimum for a given building in a given climate zone.

First brought to the attention of ORNL in 1974 by Harry C. Fischer, the ACES was investigated on a small scale in the laboratory. The results of those investigations[4] were favorable, and a decision was made to begin a full-scale test. This decision resulted in the construction of the ACES demonstration house[5] as part of the Tennessee Energy Conservation in Housing (TECH) complex near Knoxville, Tennessee.

Operation of the demonstration house and an identical control house has provided much information on the ACES and its performance relative to conventional heating, cooling, and water heating systems. This information has allowed improvement of the ACES controls and equipment and meaningful measures of its energy conservation potential as compared with conventional alternative systems.

Previous reports have been issued covering the shakedown and initial start-up period,[6] the construction and first winter's operation,[7] and the first full annual cycle test period.[8]

This report deals with the second- and third-year performance of the ACES and control houses and is the final report on the full ACES proof-of-concept test series. The experimental results have verified nearly all of the theoretical performance predictions making possible meaningful systems economic comparisons.[9,10]

The basic yardstick used in this report to compare the ACES with alternate systems is the annual coefficient of performance (ACOP), defined as follows:

$$\text{ACOP} = \frac{\text{annual heating, cooling, and water heating energy delivered}}{\text{annual electrical energy purchased}}$$

2. SUMMARY

The TECH experimental and demonstration complex near Knoxville, Tennessee, is being used to demonstrate various methods of energy conservation. Included in the complex are the ACES house, the solar house, and the control house. In addition, a garage structure is being used to test a solar-assisted heat pump system.

The main feature of the ACES house is its integrated space heating, water heating, and space cooling system employing a unidirectional heat pump, low-temperature thermal storage in an ice-water storage bin, and a solar/convector panel for use in balancing system loads. A major feature of the ACES is its interseasonal energy transfer through the thermal storage bin. During the winter, energy is removed from the bin to meet heating demands. The ice produced as a by-product of this operation is used to meet cooling demands the following summer, thus depositing energy into the bin and completing the annual cycle. The ACES has completed three full test years, the second and third of which form the basis of this report. Reference 8 contains a full report of the first test year.

Both the ACES and control house systems operated well throughout both years except for a three-week outage of the ACES in December 1978 and January 1979 caused by an incorrectly installed thermostat. This outage was accounted for in the data analysis.

Weather summaries show that the second year's winter had about 6% more degree-days than normal, and the summer had 12% fewer than normal. In the third year the winter had 4% more degree-days than normal, while the summer had 25% more than normal.

During the second test year the control house used a standard air source heat pump (ARI COP of 2.46 at 8.3°C) for heating and cooling. A high-efficiency (ARI COP of 3.11 at 8.3°C), state-of-the-art heat pump was used during the third year. Hot water was supplied by an electric resistance heater both years.

The floor plan, thermal envelope, and environmental exposure for the control and ACES houses are nearly identical. Though neither house was occupied, internal loads were simulated for a family of four. Hot

water was consumed automatically at the rate of 265 ℓ/day, and about 24 kWh/day of electricity was consumed to simulate appliance loads. Door openings by persons working at the complex and tour groups accounted for normal ingress and egress by occupants. The two houses have demonstrated measured heating loads within ±2% of one another during the first year's operation. House cooling loads were measured to be within 3% of each other during the second test year. The solar house and garage systems are not discussed in this report.

This report describes (1) modifications made to the ACES as a result of first year test findings and (2) ACES and control house performance levels for the test periods from December 1, 1978, to September 30, 1979, and December 1, 1979, to September 15, 1980.

Modifications made to the ACES prior to the second test cycle included installing a larger compressor in the mechanical package, reinsulating the ice storage bin, and replacing the bin heat exchanger with a shorter, more cost-optimized coil having only 45% as much surface area as the original. In addition, control logic changes were made to produce hot water only when required and to reduce the number of hours of night heat rejection operation.

Installation of the larger compressor and the smaller bin heat exchanger resulted in about a 10% reduction in steady-state and seasonal heating performance compared with first-year levels. Increased bin insulation levels were successful in reducing bin heat leakage by about 40%. This resulted in a projected improvement in ACOP during the second and third test years of about 10 to 15% over what would have been achieved with the original bin insulation. The bin interseasonal energy transfer was 84% of its thermal capacity in the second year and 89% in the third year, as compared with 68% in the first year. Ice production exceeded bin design capacity in both the second and third years, resulting in actual packing factors of 0.87 and 0.94 respectively. The excess ice was deliberately made in the second year to test the system performance under that condition.

In a test of solar panel heat collection performance, the gray-colored panel at the demonstration house collected an average of

61.2 kWh/day compared with a predicted collection of 61.5 kWh/day for a black-colored panel. This shows that the prediction method closely approximates actual performance. The panel performed adequately as a heat rejector. Its measured heat transfer coefficient in still air was about 7.5 $W/m^2 \cdot °K$, closely approximating the calculated value of 8.3 $W/m^2 \cdot °K$ for vertical-finned surfaces in still air.

The control house ARI 2.46 heat pump was modified in May 1979 by adding an intermediate brine heat transfer loop to the indoor unit between the airstream and the refrigerant evaporator. This heat transfer loop was instrumented to measure the control house cooling loads for comparison with those of the ACES house. As a result of this modification, however, the heat pump suffered a 24% COP degradation in the cooling mode. Therefore, the summer 1979 air-conditioning power consumption for the ARI 2.46 heat pump was calculated by using the cooling seasonal performance factor (SPF) of 1.64 measured in the summer of 1978.

Results of the second test year showed an overall ACOP of 2.80 for the ACES while delivering 32.85 GJ of heating, 19.17 GJ of cooling, and 15.81 GJ of hot water. The annual electricity consumption was 6719 kWh. This compares with an ACOP of 1.42 for the ARI 2.46 heat pump and electric resistance hot water system while delivering 32.85 GJ of heating, 19.18 GJ of cooling, and 13.74 GJ of hot water at an annual electricity consumption of 12,853 kWh. The predicted ACOP for the ARI 2.46 heat pump system is 1.38. Heating and cooling SPFs demonstrated by the ARI 2.46 heat pump were 1.58 and 1.64.

Third-year performance results showed that the ACES delivered 34.40 GJ of space heat, 22.65 GJ of space cooling, and 14.34 GJ of water heat while consuming 6447 kWh of electricity for an ACOP of 3.08 compared with a calculated value of 3.18. The control house with ARI 3.11 heat pump consumed 11,358 kWh while delivering 34.29 GJ of space heat, 22.70 GJ of cooling, and 13.57 GJ of water heat for an ACOP of 1.73. Calculations predict an ACOP of 1.67 for this system. The high-efficiency heat pump demonstrated heating and cooling SPFs of 1.99 and 2.27 respectively.

In addition to the savings in total energy consumption, there was a reduction in the peak hourly consumption during both years. This reduction averaged about 40% compared with the control house systems in both summer and winter, with maximum reductions reaching 85%.

The ACES has achieved all of its performance objectives, except that the water heating performance was degraded due to fouling of the heat exchangers. The system is capable of delivering residential space heating, space cooling, and water heating loads while consuming only about 57% as much electrical energy as the best conventional alternative yet tested.

3. THE KNOXVILLE DEMONSTRATION

3.1 The TECH Complex

In order to demonstrate the ACES concept, a test house and a control house were built near Knoxville, Tennessee. These houses are part of a three-house complex called the TECH project, whose purpose is to demonstrate energy conservation through improved thermal envelopes and the use of innovative heating and cooling systems.[8] The complex is composed of the ACES house; the control house, which employs conventional heating and cooling systems; and the solar house, which employs a solar heating and hot water system.

The complex, located on the agricultural farm of the University of Tennessee just south of Knoxville on Alcoa Highway, is a joint effort of ORNL, UT, and the Tennessee Valley Authority in cooperation with local industries. Operational since August 1976, the experiments being performed at the complex are used to evaluate the performance of the ACES and the solar heating system along with the performance of the thermal envelopes.

3.2 The Demonstration House

3.2.1 House construction

All three houses in the complex have similar floor plans and exteriors. The ACES house is a 167-m^2 single-family residence consisting of 3 bedrooms, 2 baths, a great room, a kitchen, a mechanical equipment room, and a partial basement. In addition, the thermal storage bin is located under the northeast section of the house.

All the houses were built with a better thermal envelope than was normally found in the area at the time of construction in an effort to reduce the heating and cooling requirements. Double-pane-insulated glass was used throughout, and 7.35 $m^2 °K/W$ insulation was used in the ceiling where possible, with 3.91 $m^2 °K/W$ insulation in the sidewalls, below the floor, and in other ceiling areas. A complete description of the insulation details of the demonstration houses can be found in Table 3.1.

Table 3.1. Thermal insulation used in the demonstration houses

	Insulation thickness (m)	U-value ($W/m^2 \cdot K$)
Sidewalls	0.14	0.256
Ceiling, flat	0.305	0.136
Ceiling, cathedral	0.152	0.227
Floor (over crawl space)	0.152	0.227
Floor (over ice bin)	0.305	0.136
Windows	Double glazed, wood frame	3.293
Basement and crawl space		0.352

3.2.2 Simulated loads

Because of the experimental nature of the ACES, neither the ACES house nor the control house is occupied. In order to account for internal loads and water use due to occupancy, these loads have been simulated. The hot-water consumption is 265 ℓ/day on the following schedule:

Time	Consumption (ℓ)
6:00 AM	106
12:00 Noon	53
7:00 PM	53
11:00 PM	53

Internal loads from appliances are simulated by drawing about 24 kWh/day, using resistance heating elements and electrical equipment. Both houses also have working refrigerators which are part of the internal electrical load. Door openings by persons working at the complex and visitor tours simulate occupancy from the standpoint of ingress and egress. Figure 3.1 shows an artist's conception of the ACES house.

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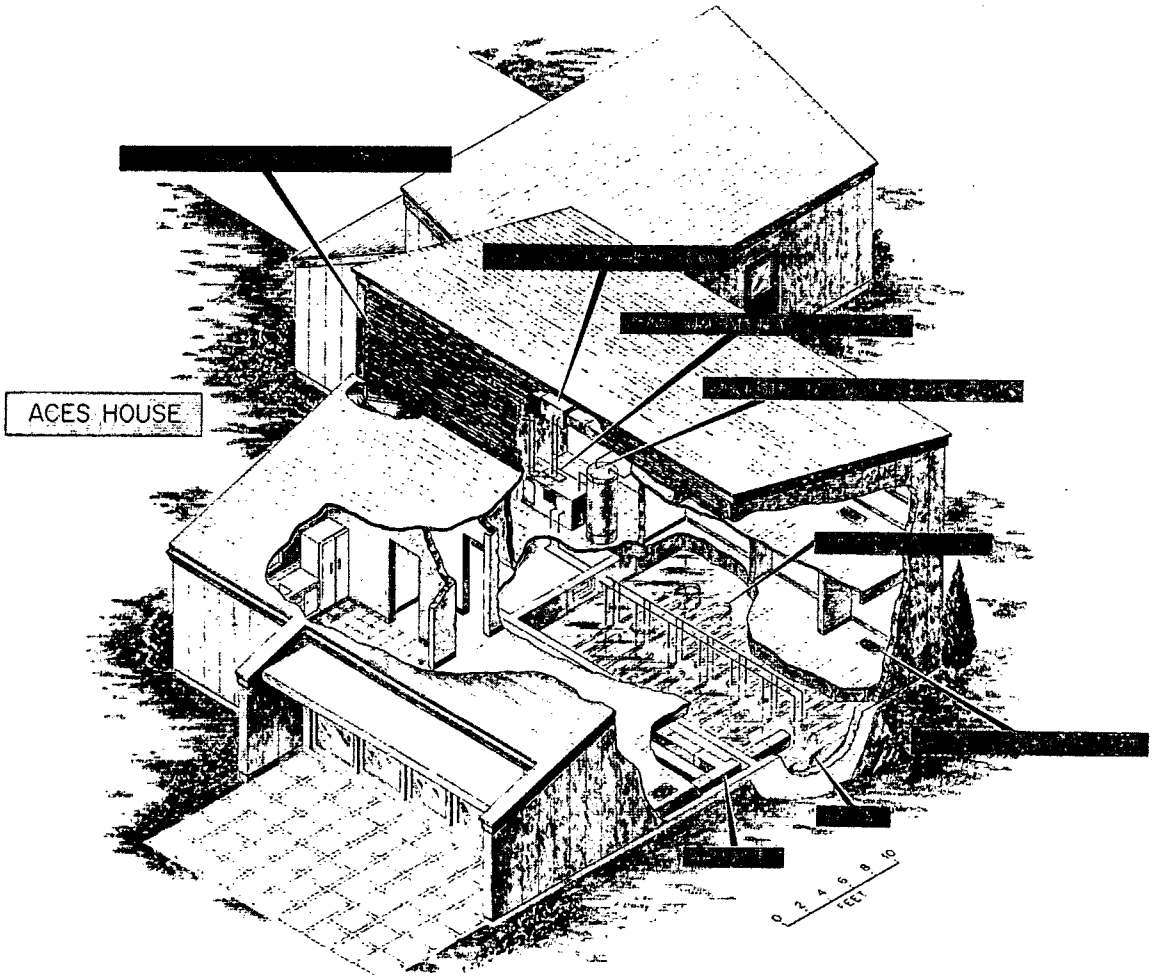

Fig. 3.1. Knoxville ACES demonstration house.

4. THE ACES CONCEPT

The ACES concept takes advantage of the fact that whenever a heat pump is operated, both heating and cooling are performed. In most heat pump applications, only one of these outputs is utilized while the other is wasted. The ACES, however, stores the cooling energy produced whenever heating is performed. Thus, both heat pump outputs are used.

Although the name ACES properly indicates annual cycle operations only, the term has come to imply a system which employs ice-water storage capable of maintaining constant-capacity heat pump operation during the heating season. Therefore, ACES is used to describe any system ranging from a true annual cycle, such as the one discussed in this report, through small bin storage systems.

The major difference between ACESs of various sizes is the amount of interseasonal energy transfer accomplished. Interseasonal energy transfer is reduced and energy consumption increased by decreasing the size of the storage bin. This generally requires an increase in the size of the solar/convector panel or other heat source/sink balancing system such that heating and/or cooling capabilities are ensured. The basic features, components, and control system are not affected by the extent of interseasonal energy transfer; rather, the decision is based on economics. Primary economic influences are component costs, electric rates, and interest rates. The economics are considered in detail in other reports.[3,9,10]

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

4.1 Features of the ACES

The ACES has three major features which set it apart from the typical heat pump installation. These features are: a constant-capacity heat pump, interseasonal energy transfer, and reduced peak energy use. The relative importance of these features varies depending on the extent

of interseasonal energy transfer being achieved; however, they are common to all types of ACESs.

4.1.1 The constant-capacity heat pump

Figure 4.1 shows the output for the ACES and a typical air source heat pump vs outside temperature, along with the heating load required by the structure. As the temperature decreases, the heating load for the structure increases while the air source heat pump heating output decreases. At some temperature (indicated by point A) the heating load and air source heat pump heating output are equal. At temperatures below this point the heat pump requires a supplemental heating source (shown by the shaded area) to enable it to continue to supply the heating requirements. The supplemental heating source for air source heat pumps is usually supplied by low-first-cost electric resistance heating elements, which, however, have a high operational cost. Because the output of the ACES is independent of outside temperature, the system may be sized to meet the load at any given design day temperature and maintain the minimum amount of overcapacity at higher temperatures (point B). If a typical heat pump were sized to meet the load at B, then its

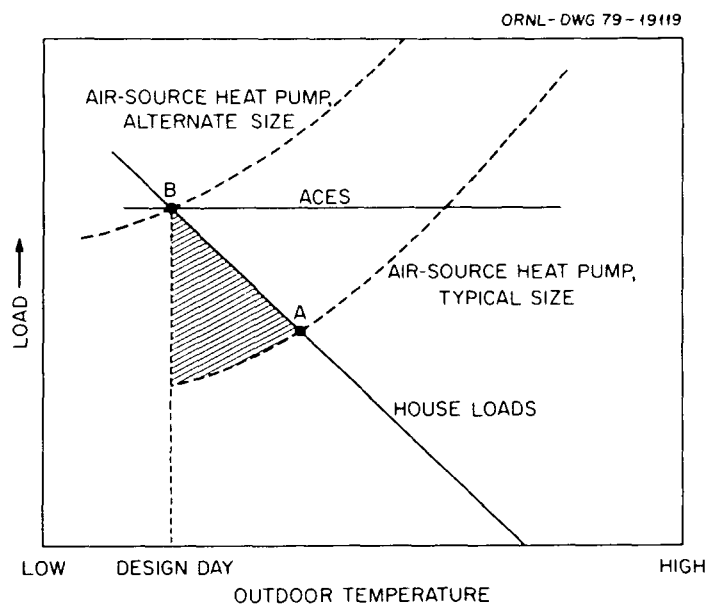

Fig. 4.1. Heat pump and ACES output vs temperature.

capacity at higher temperatures would be much greater than that of the ACES. It would, therefore, suffer larger cyclic losses than those of the ACES and would also be improperly sized for cooling services. Sizing the ACES to supply the design day load removes the need for supplemental energy, thereby reducing both the utilities' peak load and the total energy consumption.

4.1.2 Interseasonal energy transfer

The second major feature of the ACES is that of interseasonal energy transfer. Since the cooling output is saved whenever heating is performed, some portion of this stored energy can be used at a later time to satisfy the structure's cooling requirements. This means that the cooling generated in the winter can be transferred, 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 much less than that of a system which must generate the cooling requirements on demand. Peak loads are also reduced because it is not necessary to operate a heat pump to provide the cooling.

4.1.3 Peak energy use reduction

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

4.2 Major Components of the ACES

4.2.1 Ice-water storage bin

A primary difference between the ACES and a typical air source heat pump is the ice-water thermal storage bin, which 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 the heating modes. This function is accomplished by utilizing the latent heat of fusion of water as it is converted to ice and then, later, back to water.

4.2.2 Thermal energy balancing mechanism

In some regions of the United States a device is needed for collecting supplementary heat in winter and/or rejecting heat during supplemental cooling operation when the bin is out of ice.

Whenever the ice production capability is greater than the summer ice cooling requirements, heat is collected and used 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 balancing device is to reject heat from the heat pump and the storage bin when the ice supply is exhausted and additional cooling needs are anticipated.

At the ACES house no energy collection is normally required, but both functions can be accomplished by the 19.4-m^2 solar/convector panel shown in Fig. 4.2. This panel was sized for summer night heat rejection.

4.2.3 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 energy between the other major components: the indoor fan coil, the ice-water

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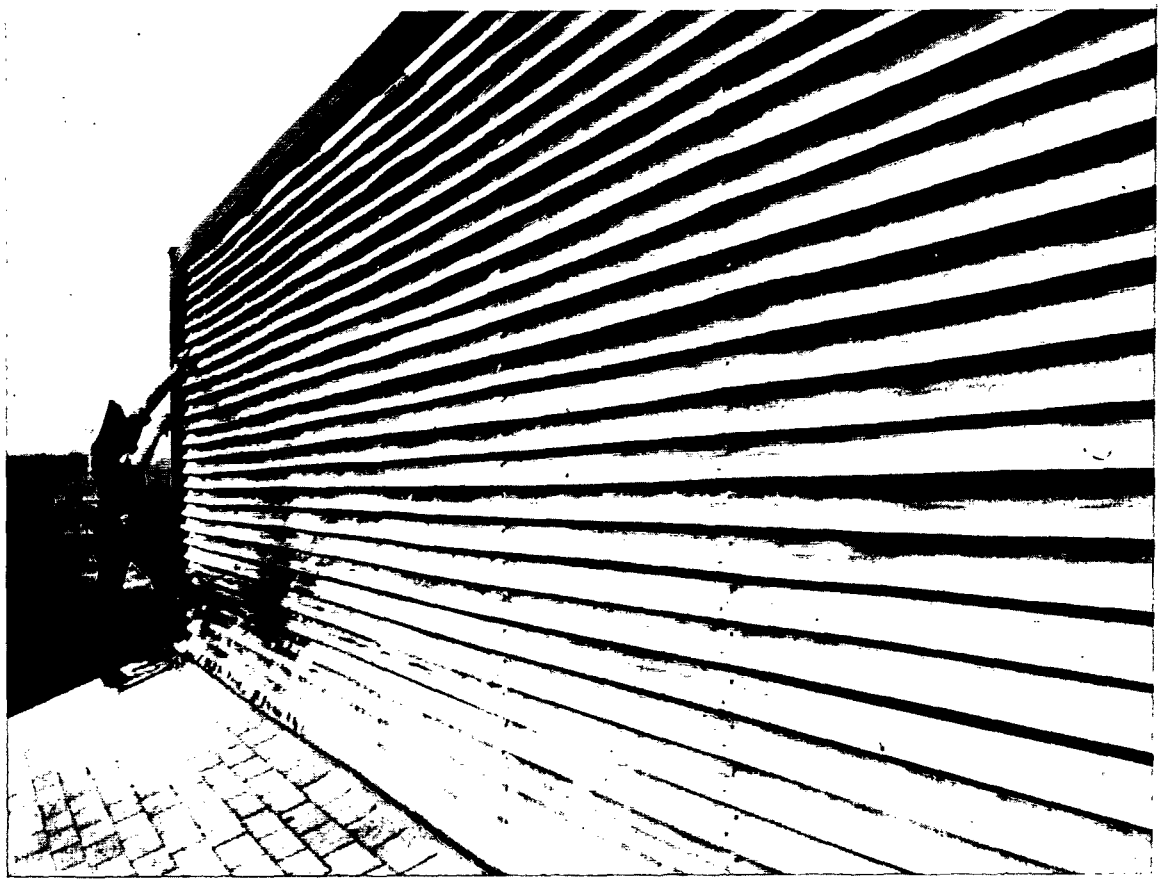

Fig. 4.2. Unglazed solar/convector panel at the ACES house.

storage bin, the balancing mechanism(s), and for this application the domestic hot-water tank. An intermediate fluid (brine) is used to facilitate energy transfer between the system components. All of this is done on command from the control system, which is responsible for determining which function(s) the ACES should be performing. Figure 4.3 shows the ACES house mechanical package.

4.3 The ACES Control System

It is necessary to have a control system capable of sensing various external conditions and placing the ACES in the correct mode of operation. At present, a programmable logic controller (Fig. 4.4) is used to perform this function. This device was chosen because it provides the

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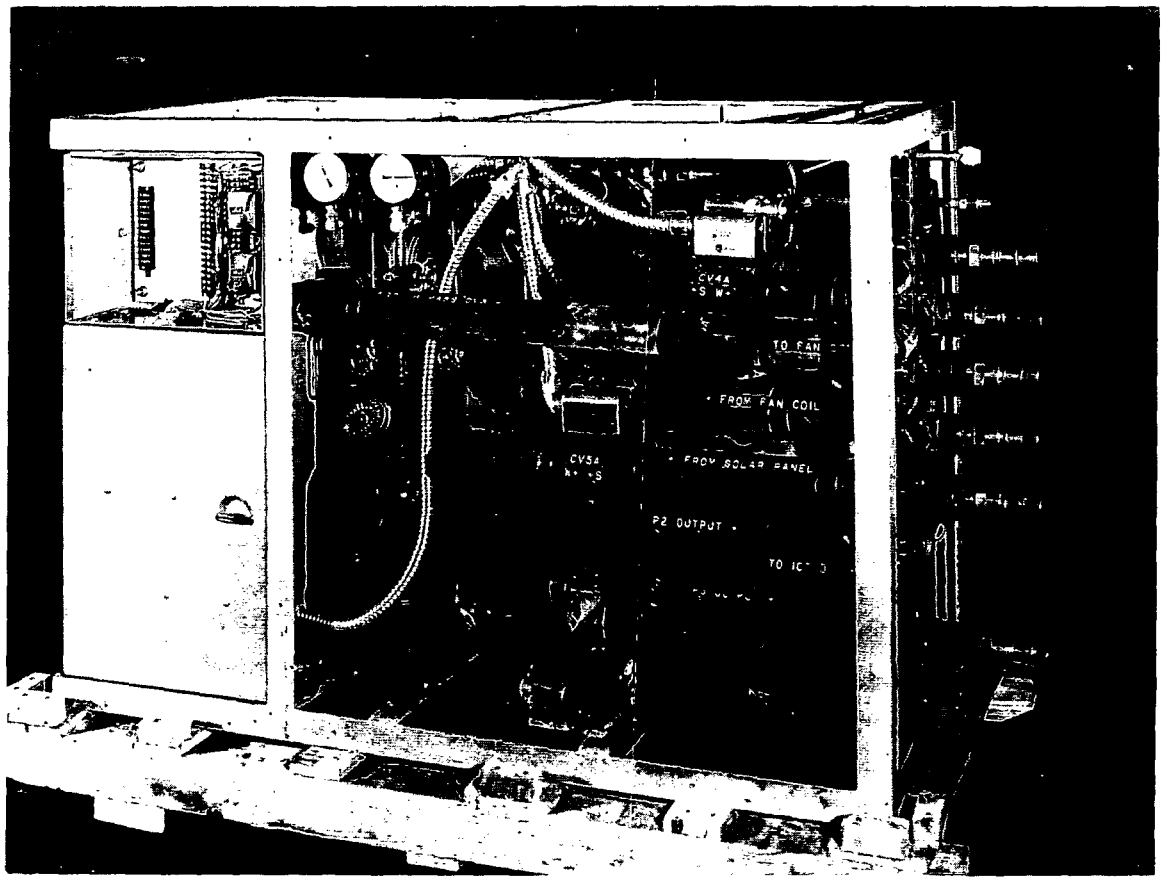

Fig. 4.3. ACES house mechanical package.

necessary flexibility to alter the logic, if needed. Once the logic has been 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 water level;
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);

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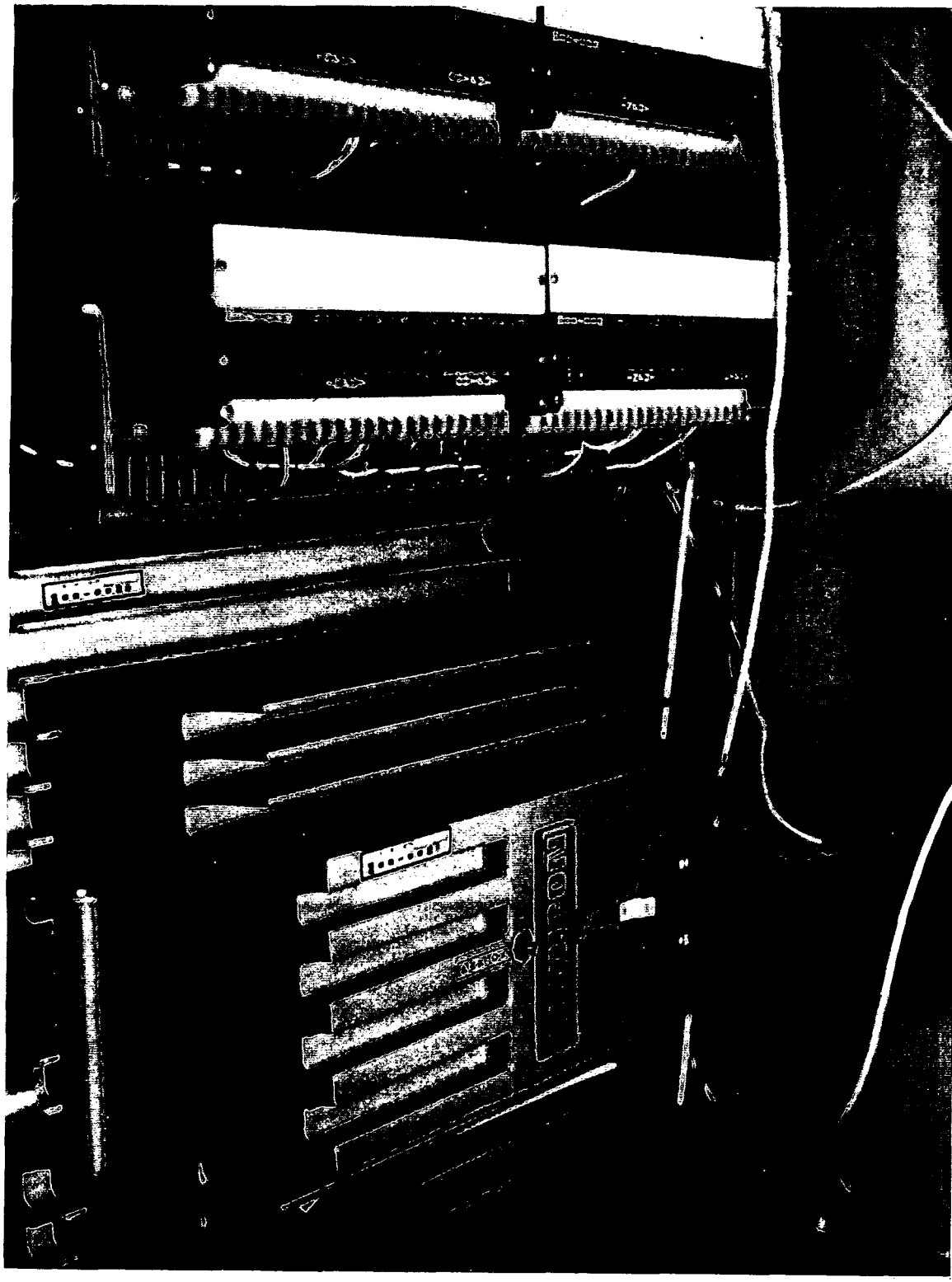

Fig. 4.4. ACES house programmable logic controller.

7. equilibrium solar panel temperature;
8. time-of-day.

The output signals generated by the controller are:

1. compressor on/off,
2. indoor fan on/off,
3. cold-brine pump on/off,
4. hot-brine pump on/off,
5. hot-water pump on/off,
6. control valve position signal.

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

4.4 ACES Modes of Operation

The most common modes of operation for the demonstration house ACES and the control logic necessary to invoke them are presented below. Other ACES applications, especially large-building applications, may require modes not discussed in this report.

4.4.1 Primary modes

The ACES discussed in this report has three primary modes of operation:

1. space heating,
2. space cooling,
3. domestic water heating.

These modes are shown schematically in Figs. 4.5, 4.6, and 4.7 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 or the domestic hot water. When extracting heat from the storage bin, some of the water is converted into ice, which is stored for future cooling needs. Space heating is supplied whenever the house thermostat signals a need for heating. This

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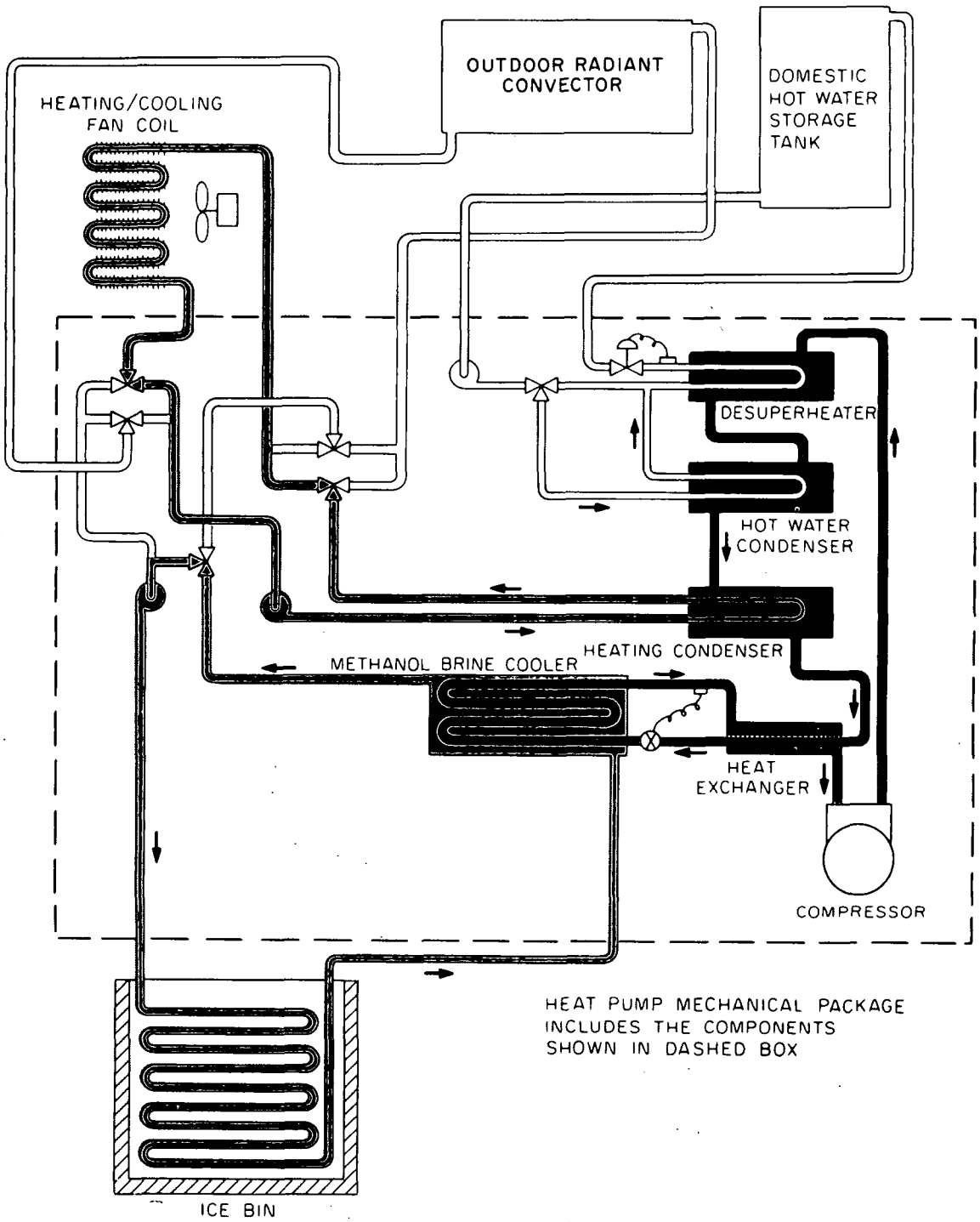

Fig. 4.5. Space heating schematic.

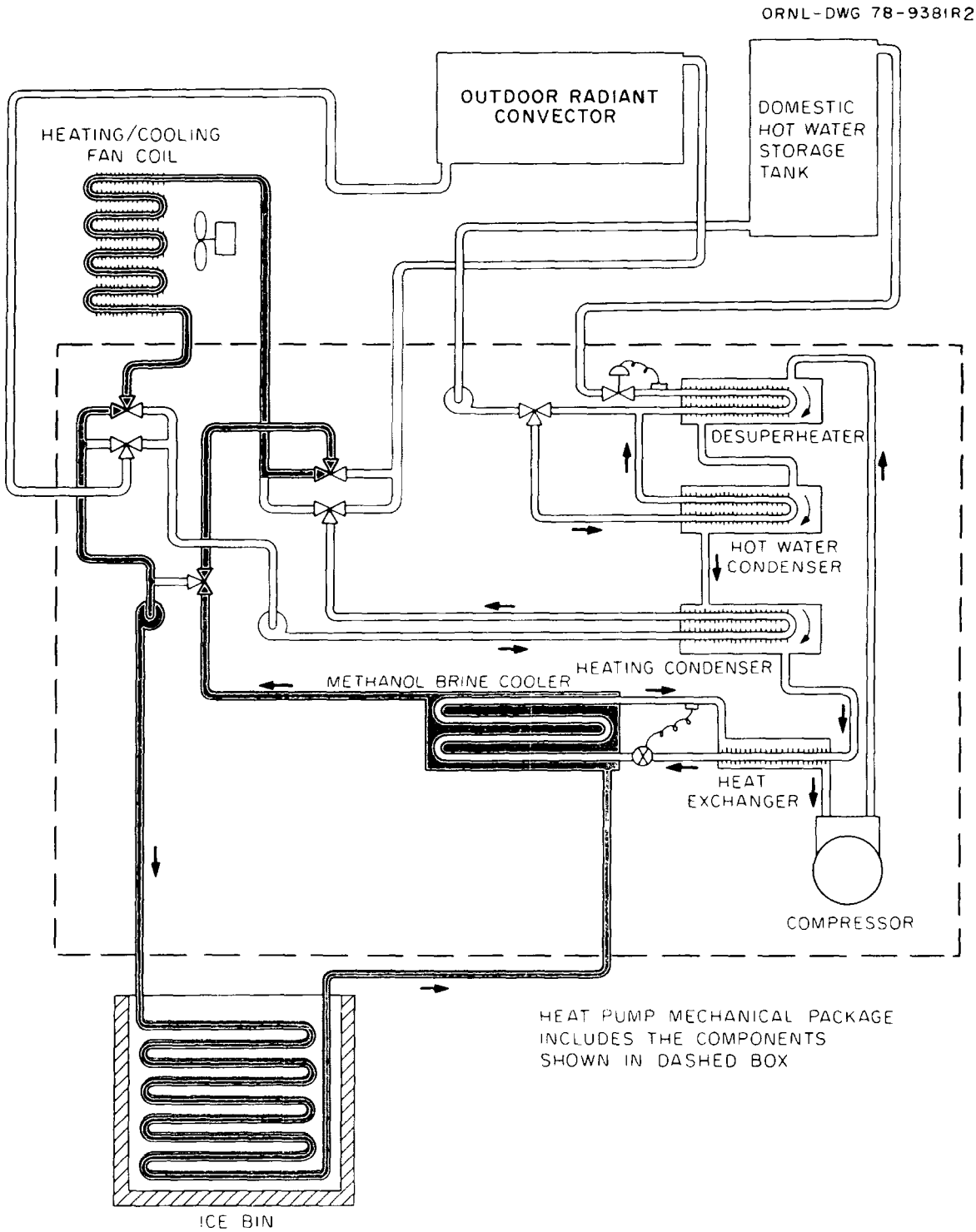

Fig. 4.6. Space cooling schematic.

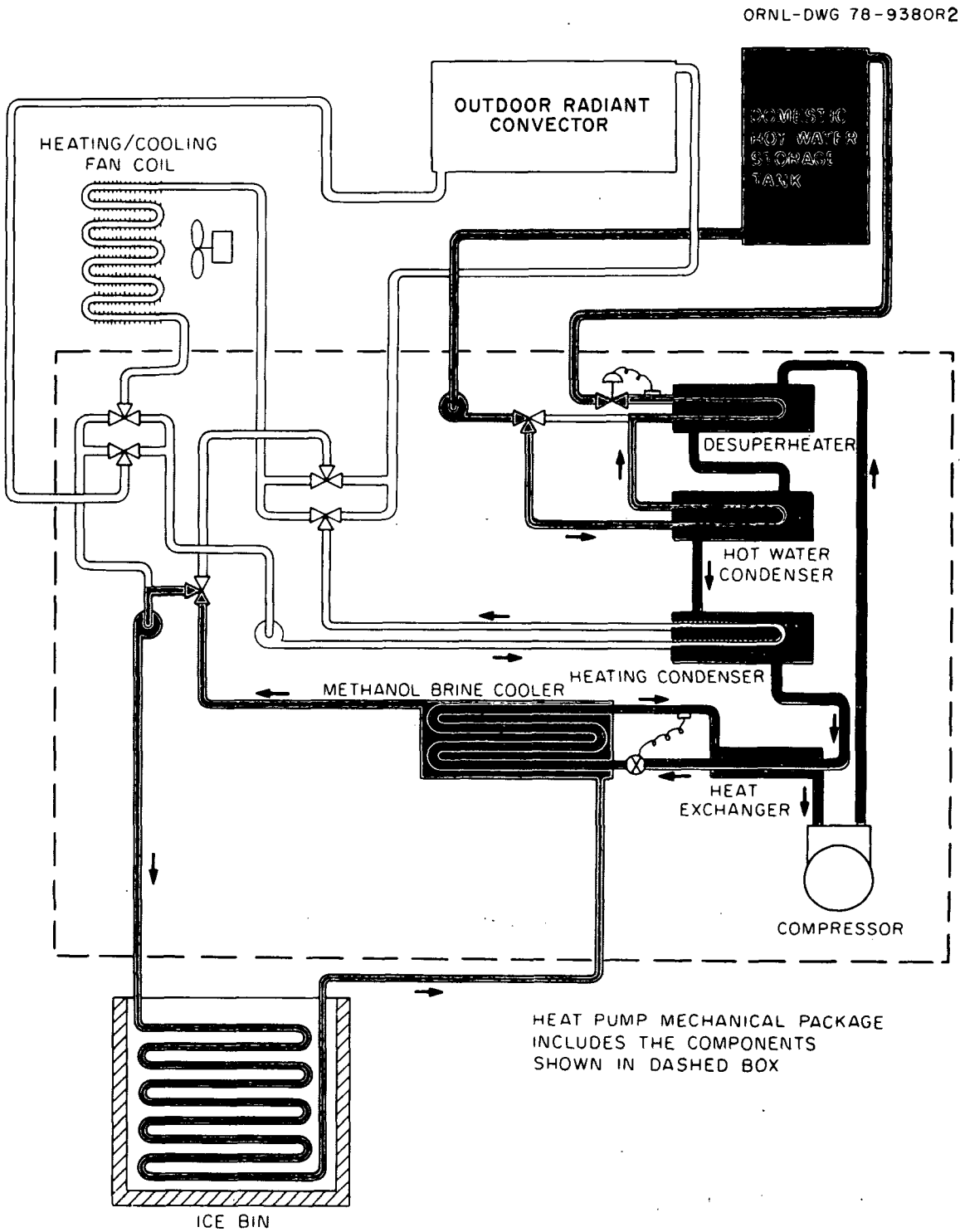

Fig. 4.7. Domestic hot water schematic.

action also resets the season indicator to "winter." Domestic hot water is produced whenever the hot-water thermostat calls for heat. Hot-water production can occur coincidently with the other primary modes and 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 this heat absorption to cool the structure. Space cooling is provided whenever the house thermostat asks for cooling. This mode results in the season indicator being set to "summer."

4.4.2 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. 4.8 and 4.9 respectively). These two modes are required because it is unlikely that the bin capacity, ice production, and cooling requirements will be exactly matched in any given year.

Ice melting is used to reduce the ice inventory should the storage bin become full and additional heating demands are expected. This is accomplished by circulating brine 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 initiated whenever three coincident conditions exist: (1) the ice bin is full, (2) the equilibrium solar panel temperature is 6°C warmer than the bin, and (3) the season indicator shows "winter." This mode is not required under normal winter conditions in Knoxville.

The night heat rejection mode is used whenever the ice supply has been 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. By extracting heat from the bin, cooling capacity is produced and saved for later use. This mode is invoked whenever (1) the bin

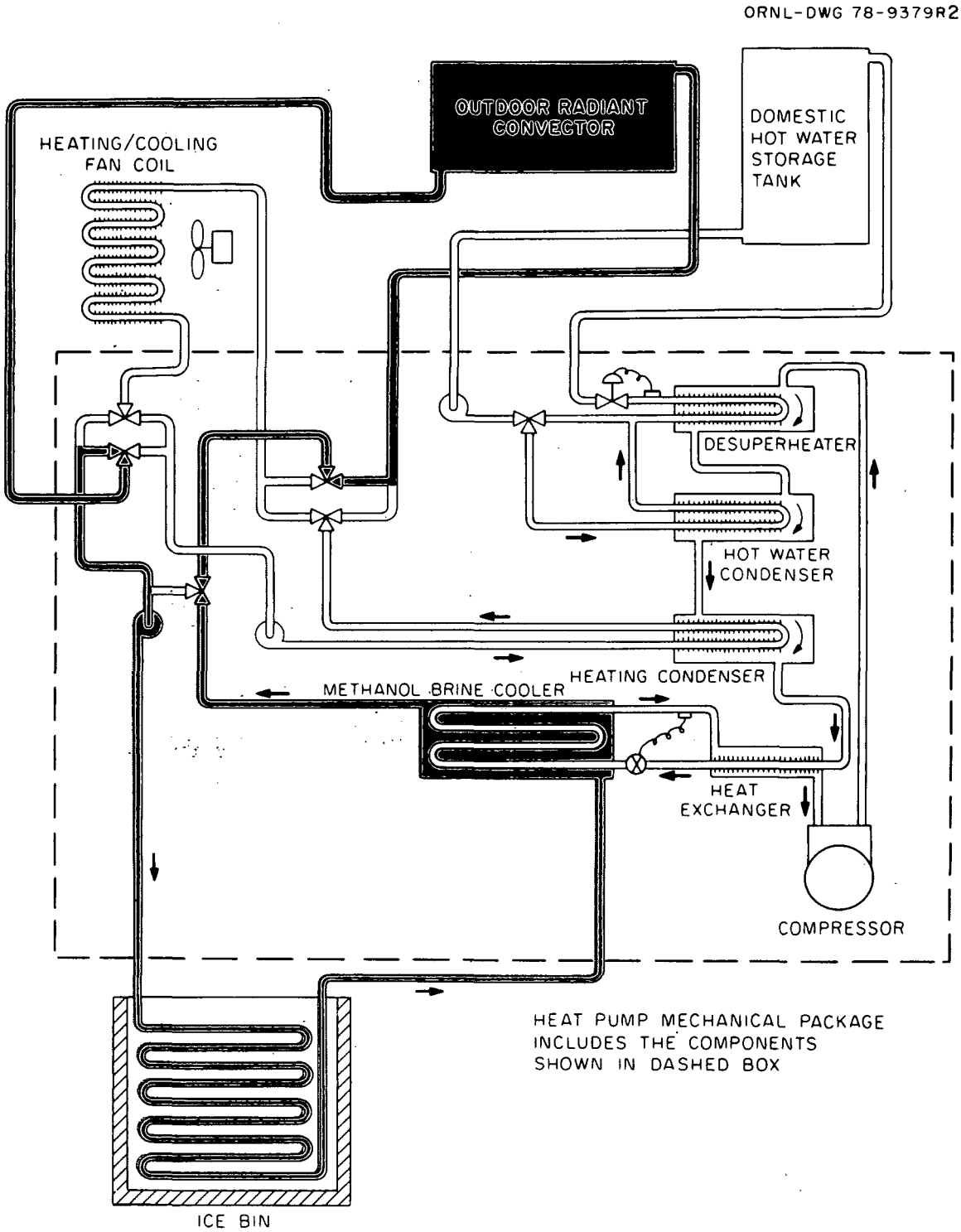

Fig. 4.8. Ice melt schematic.

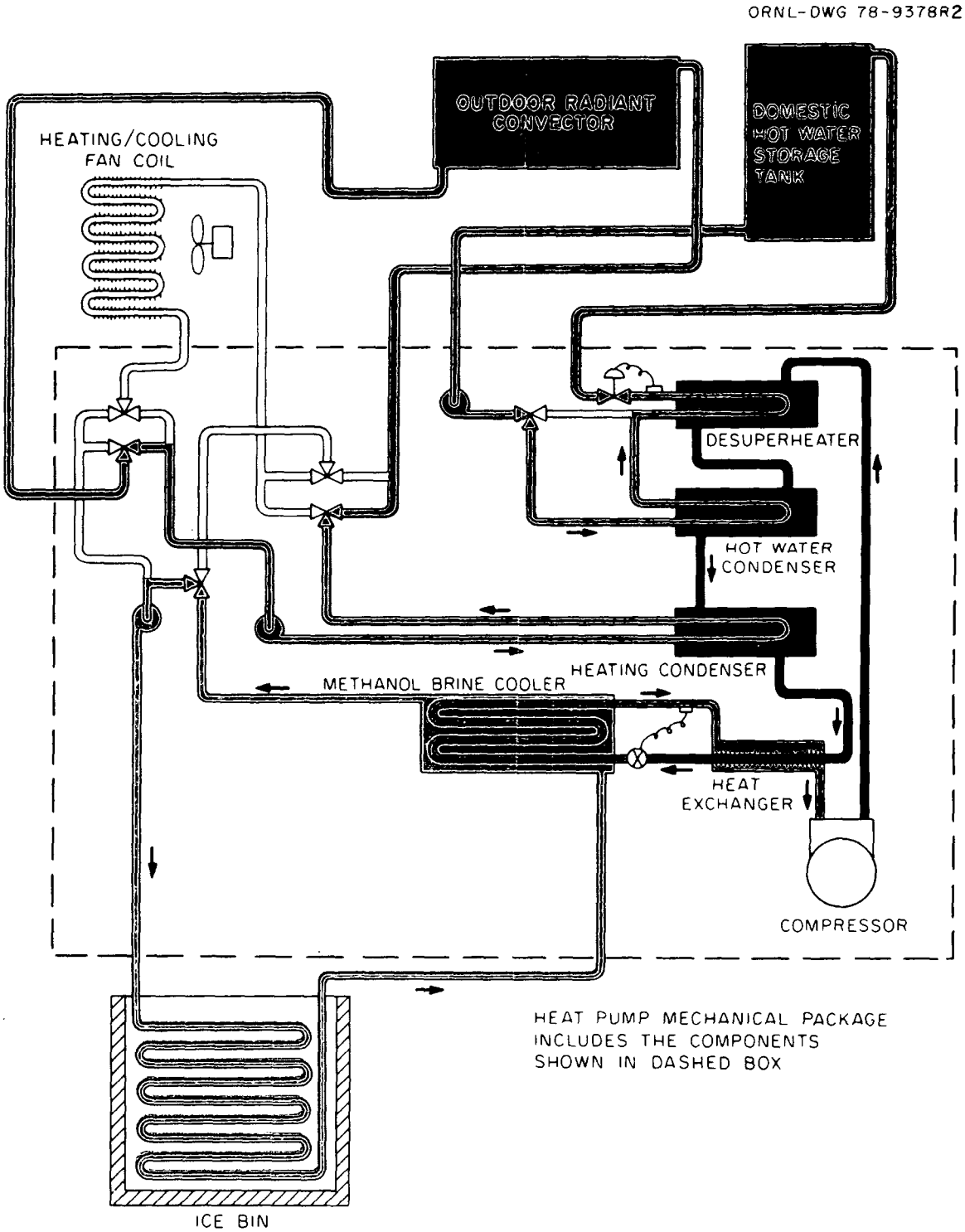

Fig. 4.9. Night heat rejection schematic.

temperature is over 7.2°C, (2) the equilibrium panel temperature is less than 27°C, (3) the season indicator shows "summer," and (4) the time-of-day signal indicates "nighttime."

5. DATA ACQUISITION PROCEDURES

Located in the ACES house is an automatic data acquisition system (DAS) which monitors the performance of the three houses and garage system at the TECH complex.

5.1 Data Acquisition System

A schematic of the DAS is shown in Fig. 5.1. It consists of a computer, an interfacing section, and sensors. Between mid-September 1978 and mid-November 1978, the DAS was modified to double its capacity to 160 data points and to improve its reliability. Upgrading modifications succeeded in improving the DAS uptime from 96% during the first

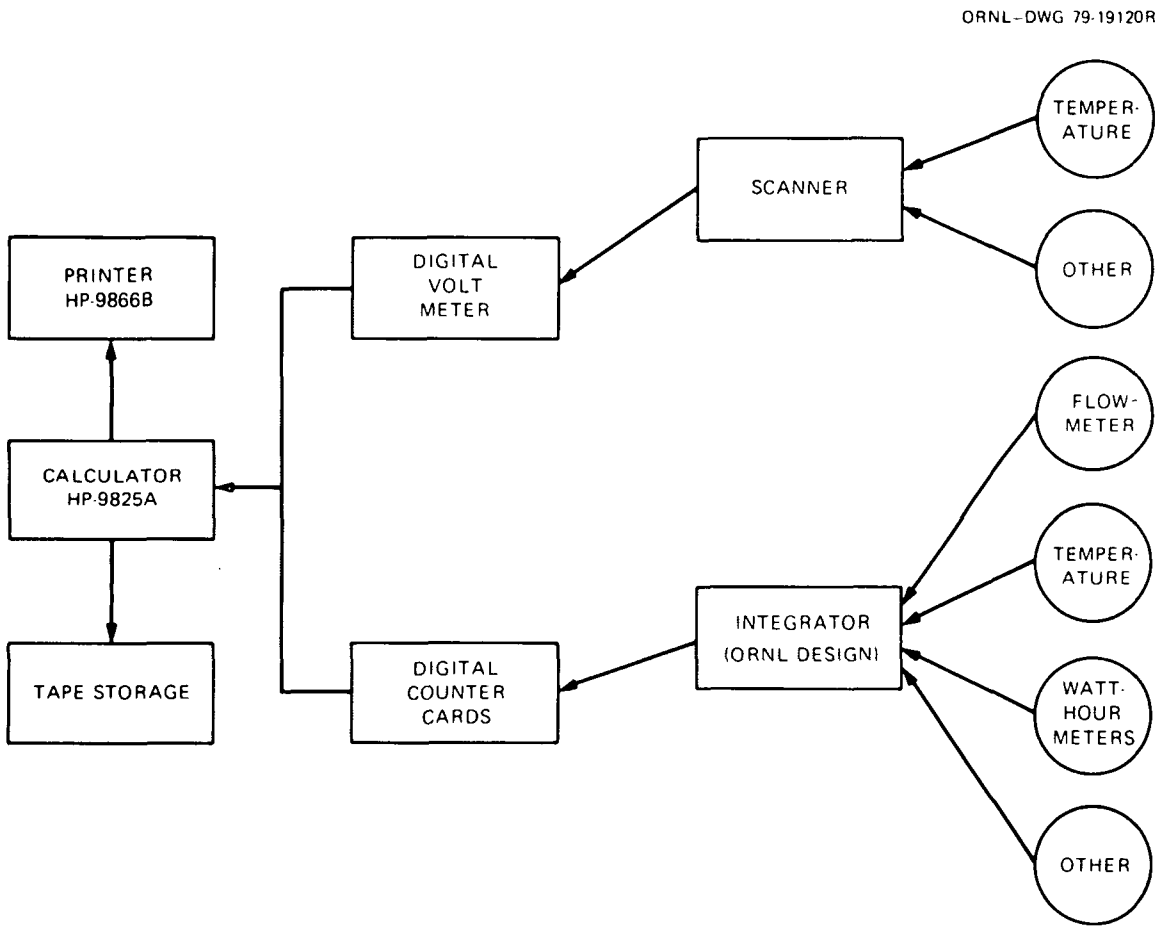

Fig. 5.1. Data acquisition system schematic.

year to over 99% during subsequent years. Over three-fourths of the second and third years' downtime could be traced to causes other than DAS breakdown. A photograph of the new DAS is shown in Fig. 5.2. The major modifications to the DAS include:

1. replacement of the HP-9830 calculator with a higher speed, higher capacity HP-9825A calculator;
2. replacement of the old digital volt meter with a newer, higher resolution digital volt meter;
3. doubling the number of analog and digital signals available;
4. upgrading the heat flow and watt-hour meter integrating circuitry.

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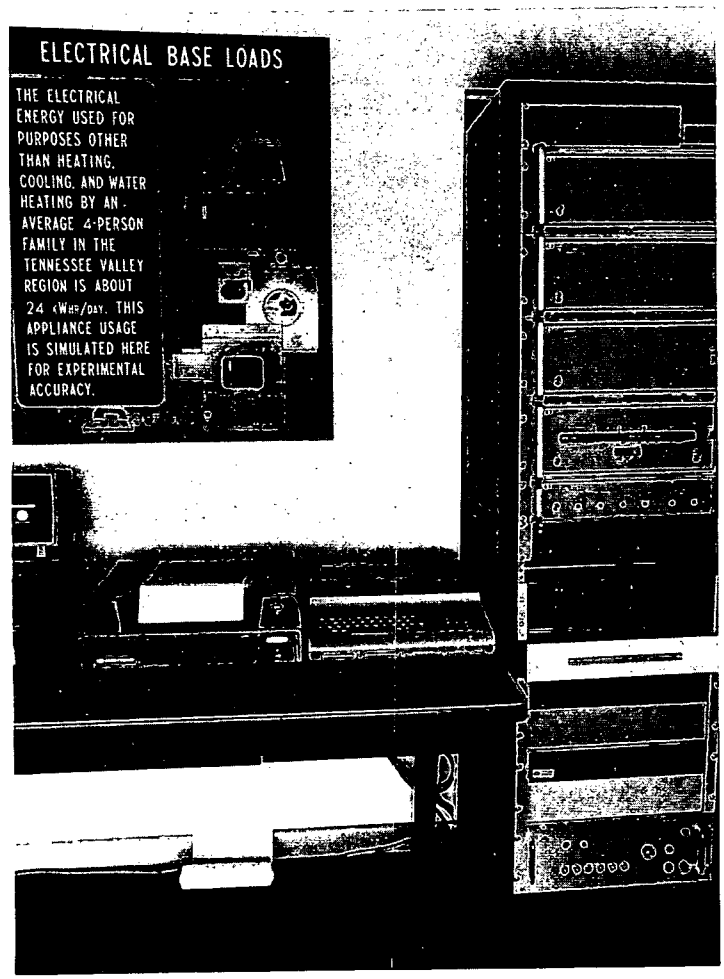

Fig. 5.2. ACES house data acquisition system.

Currently, the DAS monitors the 160 points on an hourly basis and records these data on paper at the complex and also on tape for use at ORNL. A complete list of the data points can be found in 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 volt meter which the computer reads and then converts to the proper units. Digital signals are integrated external to the computer, and the results are stored in digital counter cards which the computer reads and converts to the proper units. The computer scans the data hourly, with the analog data being the instantaneous value at the time of scan and the digital values being the sum for the hour. After the scan, the digital values are reset to zero and the process begins again. In addition, the DAS readings are backed up where possible with manual sensor readings; for example, watt-hour meters are read twice each week.

5.3 Data Reduction Techniques

Figure 5.3 shows 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. This 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, where they reside as a permanent data base. After being processed into weekly summaries, these data are used for determining system performance, for reports preparation, and for use by others studying thermal envelope performance. A sample of the various data outputs is provided in Appendix B.

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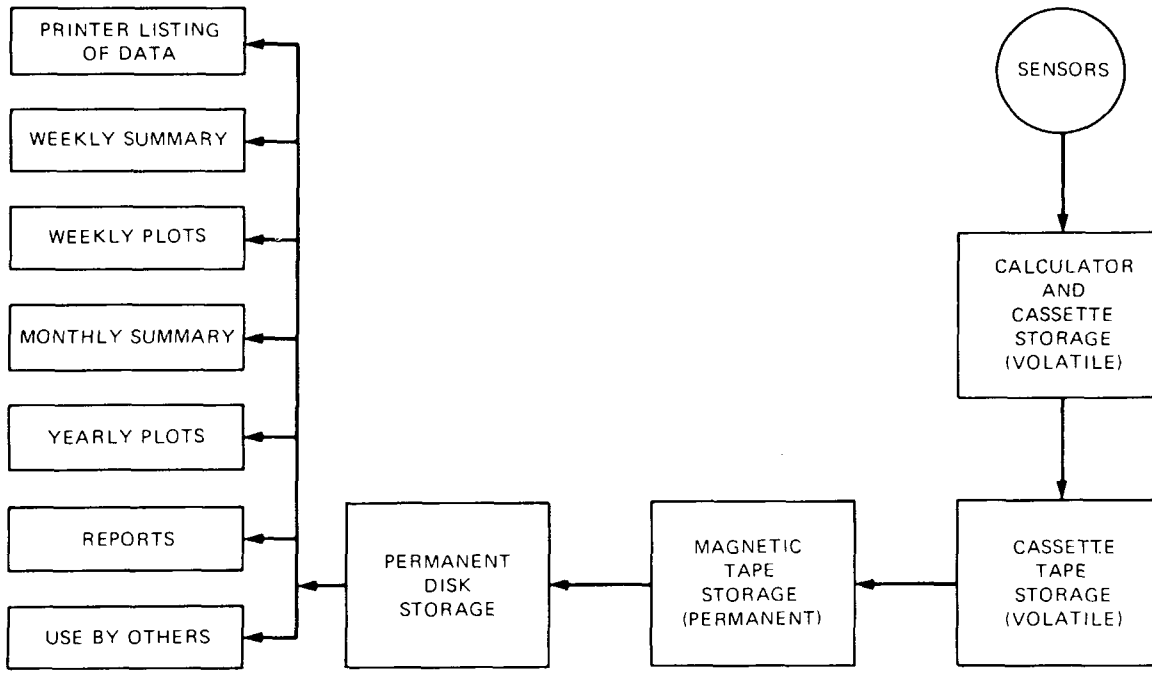

Fig. 5.3. Data processing schematic.

5.4 Data Accuracy

The resistance temperature devices (RTDs) essential to the heat flow measurements are calibrated to ±0.06°C. Watt-hour meters calibrated to about ±0.1% were obtained from a local utility. Two types of flowmeters are used: in-line turbine meters (±0.05%) for the ice bin, fan coil, and solar panel loops and a positive displacement meter (±1.5%) for the hot-water circuit.

According to ref. 11, the hourly heating and cooling loads are determined from the data as follows:

1. Heating load Q_H,

$$Q_H = BIN_i - BIN_o + SLP_i - SLP_o + Q_E - HWT_o \;;$$

2. Ice-phase cooling load (mechanical room unventilated),

$$Q_C = FAN_i - [BLWR + 0.5 \times COMP + 0.75 \times PUMPS] \;;$$

(Heat loss from the mechanical equipment is treated as an extra internal load, not as part of the house cooling load.)

3. Supplemental phase cooling load (mechanical room ventilated),

$$Q_C = FAN_i - BLWR \ ;$$

where

BIN_i, BIN_o = bin heat flows to and from the house respectively,
FAN_i = heat flow from the house air into the fan coil,
SLP_i, SLP_o = heat flows into and out of the house from the panel respectively,
HWT_o = heat flow to domestic hot water,
Q_E = ACES total equipment electrical power consumption,
BLWR = ACES fan power consumption,
COMP = ACES compressor power consumption,
PUMPS = ACES pump power consumption.

The accuracy limits of these measured loads are a function of the accuracies of the primary instruments (flowmeters and RTDs) and the accuracy of data conversion constants generated by in situ calibrations. Watt-hour meter accuracy must also be accounted for in the heating and cooling load determination.

Uncertainty analyses have shown that the accuracies of the hourly heating loads, water heating loads, and cooling loads are about ±3.4%, ±1.7%, and ±5%, respectively. When these hourly loads are summed up over an entire month, however, the overall accuracy improves to better than ±0.1% for all three loads. Occasional DAS outages and integrator malfunctions degrade these overall accuracy levels; however, overall accuracy of reported loads and power consumptions is felt to have been no worse than ±5% for any one month. Details of the uncertainty analysis are contained in Appendix D.

6. SYSTEM MODIFICATIONS

6.1 ACES Modifications

Between mid-September and December 1, 1978, the ACES house was shut down for the purpose of making several upgrading modifications to the system as dictated by first-year operational experience.[8] These included increasing heating capacity by installing a larger compressor, increasing the bin insulation and installing a shorter, more cost-optimized bin heat exchanger, and modifying control system logic to increase system operating efficiency.

6.1.1 Mechanical package modifications

On several occasions during the 1977-78 test year, the mechanical package was unable to maintain the house at set-point temperature.[8] This was caused in part by the fact that, as originally designed, part of the heat pump output always went into hot-water production and in part by the fact that the package was sized for a -8.3°C design day (97.5% point).

In an effort to correct this deficiency, two steps were taken. First, the hot-water control logic was modified so that water heating was done only when necessary. Second, the original compressor was replaced with a slightly larger model from the same manufacturer and product line in order to achieve a design day temperature of -13°C (99% point). Table 6.1 illustrates the rated cooling capacities and COPs of the two compressors at various evaporator temperatures.

Table 6.1. Comparison of original and new ACES mechanical package compressors[a]

Evaporator temperature (°C)	Cooling capacity (Wh)		COP	
	Original	New	Original	New
-6.7	5380	6155	2.52	2.46
0	6975	7810	2.85	2.89
4.4	8625	9555	3.18	3.33
8.3	9965	11225	3.68	3.62

[a]Condensing temperature = 43.3°C.

Another problem noted in the 1977-78 operation was the excessive power consumption of the ACES in the night heat rejection mode (supplemental cooling phase). The original control logic for night heat rejection initiated this mode whenever three conditions existed simultaneously: (1) bin temperature above 2.2°C, (2) equilibrium panel temperature below 27.0°C, and (3) season indicator at "summer" position.

The attempt to maintain the low bin temperature increased the bin heat leakage rate and therefore required the compressor to run longer. In addition, the location of the package inside the house caused excessive internal house loads requiring the compressor to operate for an extended period to reject those loads.

Immediate actions were taken to correct the situation during the first year.[8] These included raising the bin temperature set point to 7.2°C and isolating the mechanical equipment room from the rest of the house and ventilating it to the outside for the duration of the supplemental cooling phase. The control logic was further modified prior to the second year to include a 24-h timer in the control circuit. In order to further reduce power consumption, this timer permits night heat rejection mode operation only during specified hours.

6.1.2 Ice storage bin modifications

During the first test year, the most significant problem encountered was the magnitude of the heat leak into the ice bin. Actual heat leakage levels in 1977-78 were approximately 63.3 to 100.1 MJ/day.[8]

As a result of this discovery, it was decided to increase the insulation level of the bin floor and walls. This was done during October 1978. Figures 6.1 through 6.3 illustrate the installation of the bin insulation.

When the bin was drained and opened for the modifications, it was discovered that the original fiberglass batting in the bin ceiling had become soaked due to condensation. Therefore, it was replaced, as shown in Fig. 6.4, with Styrofoam* closed-cell insulation. In total, 17.8 cm of Styrofoam* was added to the ceiling, 15.2 cm to the floor, and 10.2 cm to the walls, for an insulation value of about 7.05 $m^2 \cdot K/W$ on all sides of the tank.

*"Styrofoam" is a registered trademark of the Dow Chemical Company.

ORNL PHOTO 7318-78

Fig. 6.1. Ice bin wall insulation.

ORNL PHOTO 7319-78

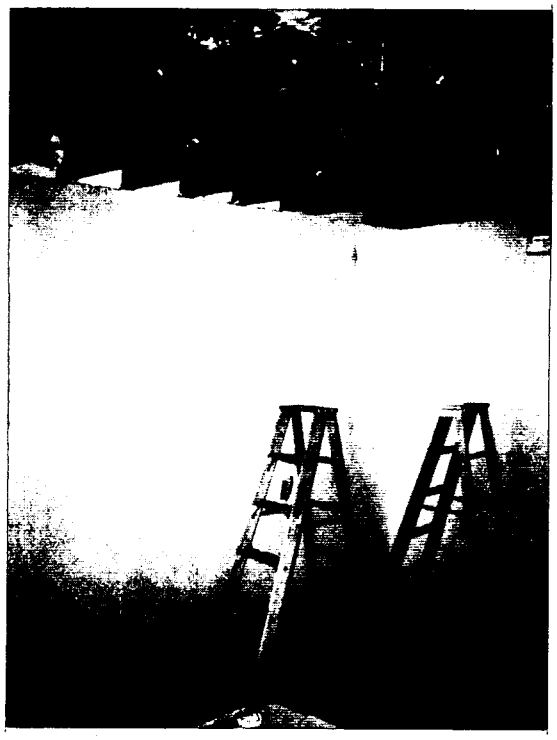

Fig. 6.2. Insulation installation.

ORNL PHOTO 7317-78

Fig. 6.3. Ceiling insulation and bin liner.

ORNL PHOTO 7316-78

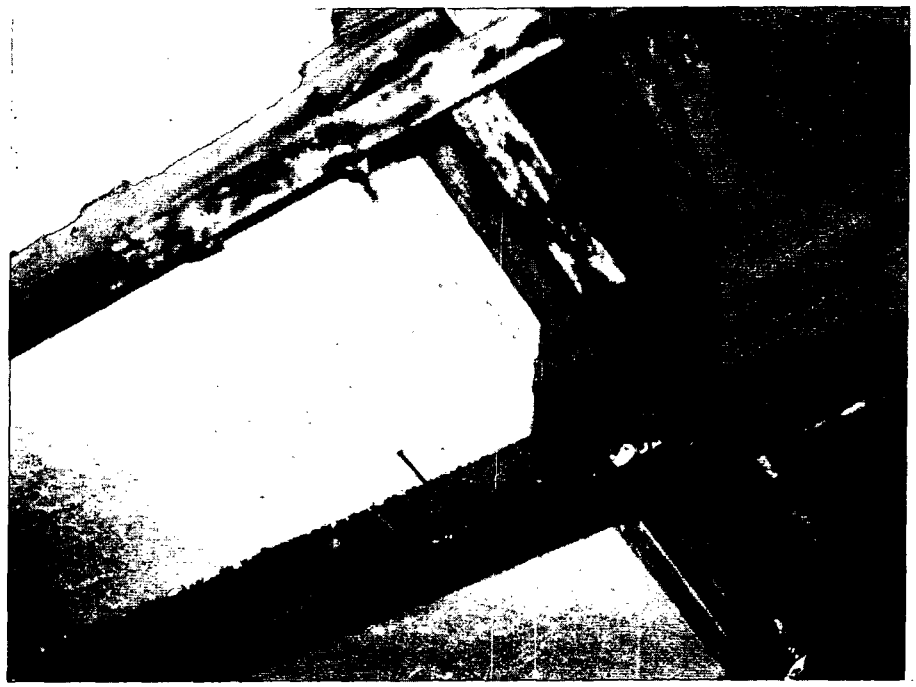

Fig. 6.4. Detail of ice bin ceiling insulation.

In addition to installing the insulation, the original bin heat exchanger, 549 m of 21.6-mm-ID plastic tubing, was replaced with 396 m of 15.9-mm-OD copper tubing. This was done in order to obtain a system with a more optimum balance between coil cost and power cost. According to ref. 1, the cost-optimized brine temperature for a power cost of 4¢/kWh (approximately the national average) is -9.4°C. For the demonstration house system, the design heat pump evaporator heat load is 5130 W, and the bin design volume is approximately 69 m^3 for a bin extraction rate density (ERD) of 74.4 W/m^3. From Fig. 6.5, the optimum tube spacing should be 0.71 m for 21.6-mm-ID tubing, whereas the original spacing was 0.33 m. A tube spacing equal to the bin ceiling joist spacing of 0.41 m was selected for the copper tubing because it provided a convenient means of coil support and resulted in a significantly shorter bin coil. This spacing is equivalent to a spacing of 0.55 m for 21.6-mm-ID plastic tubing illustrated as the actual design point in Fig. 6.5. Figures 6.6 and 6.7 illustrate the original and new bin heat

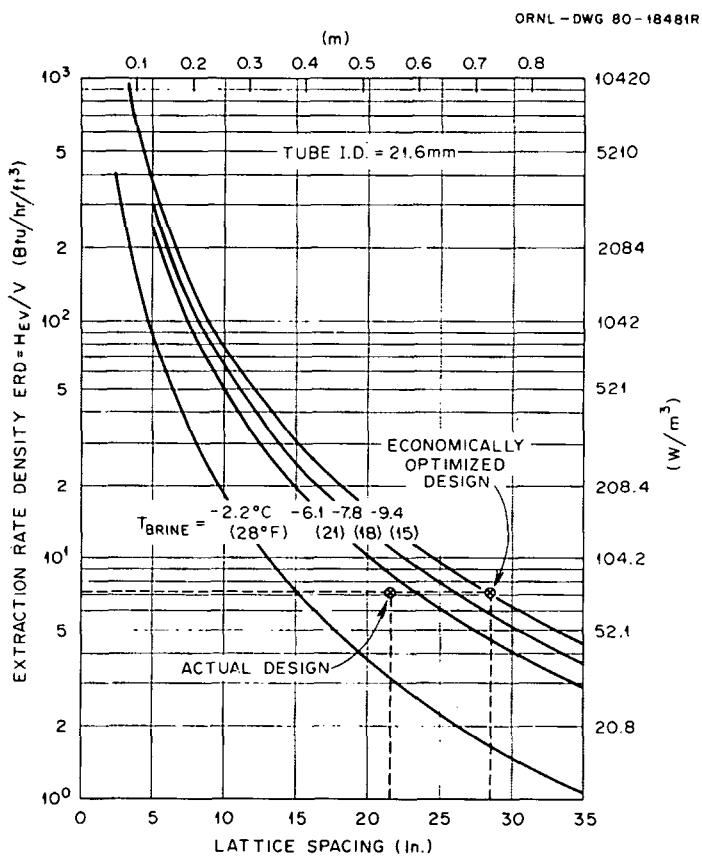

Fig. 6.5. Bin heat extraction rate density vs coil lattice spacing.

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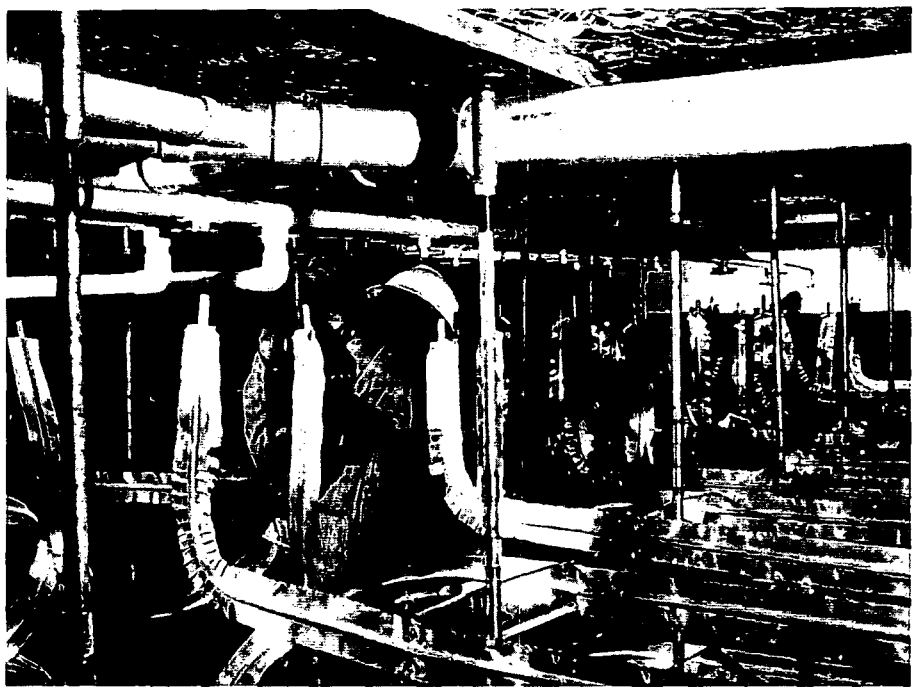

Fig. 6.6. Old ice bin heat exchanger coils and support structure.

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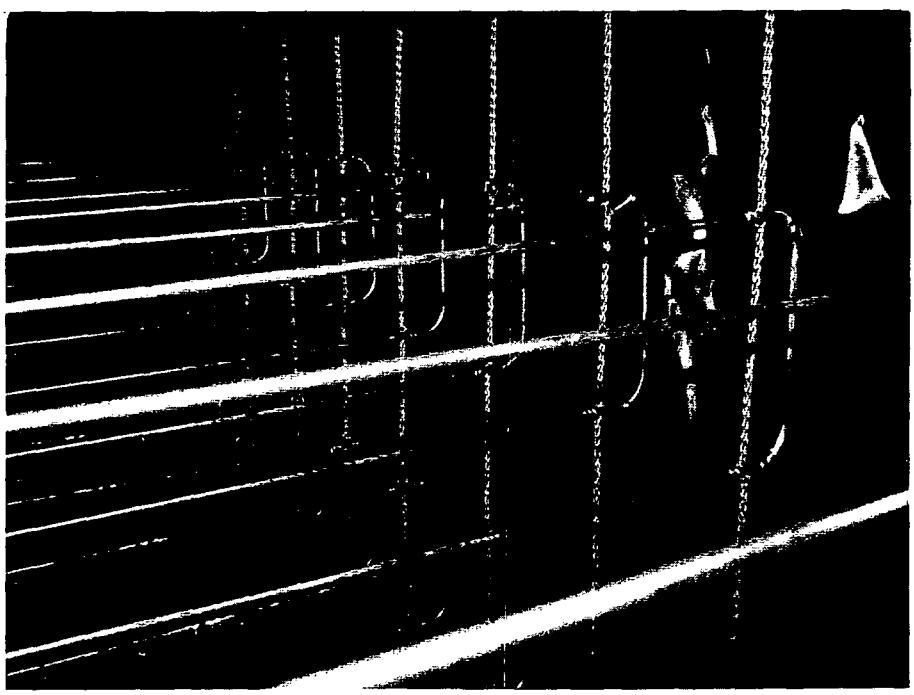

Fig. 6.7. New ice bin heat exchanger coils and support ropes.

exchangers respectively. The new coil was sized to operate with a -5.6°C average brine temperature.

The bin coil support system was another area where cost effectiveness could be improved through system simplification. The original system, shown in Fig. 6.6, used galvanized pipes with aluminum angle cross members attached firmly to the bin ceiling/house floor to support the coils and hold the ice mass under water. It was replaced by a system of polypropylene ropes as shown in Fig. 6.7. These ropes were hung from the bin ceiling joists and anchored, under tension, to ballast weights on the bin floor. The copper tubing was attached to the ropes, and the ballast was used to keep the ice mass submerged. This system was designed according to the principles outlined in ref. 1.

6.2 Control House Heat Pump Modifications

During the second test year, the control house used an average quality air-to-air heat pump (ARI COP of 2.46 at 8.3°C), shown in Figs. 6.8 and 6.9, for space heating and cooling. In May 1979, modifications were made to this heat pump for the purpose of instrumenting the unit to measure accurately the control house cooling load. This enabled accurate comparison of the thermal envelopes of the ACES and control houses during the cooling season. Figure 6.10 illustrates schematically the modifications made to the heat pump. Everything indicated within the dotted line was added to the system.

Basically, the modification consisted of disconnecting the indoor coil of the heat pump and replacing it with a refrigerant-to-brine coil. An air-to-brine coil was installed in the return air duct and connected to the refrigerant-to-brine coil and circulating pump. The instrumentation consisted of two RTDs in the inlet and outlet brine lines to the air coil and a flowmeter. Signals from these devices were processed by the DAS and recorded with the rest of the test data from the TECH complex.

The modified heat pump was replaced in November 1979 with a state-of-the-art heat pump having an ARI COP of 3.11 at 8.3°C. It was used to

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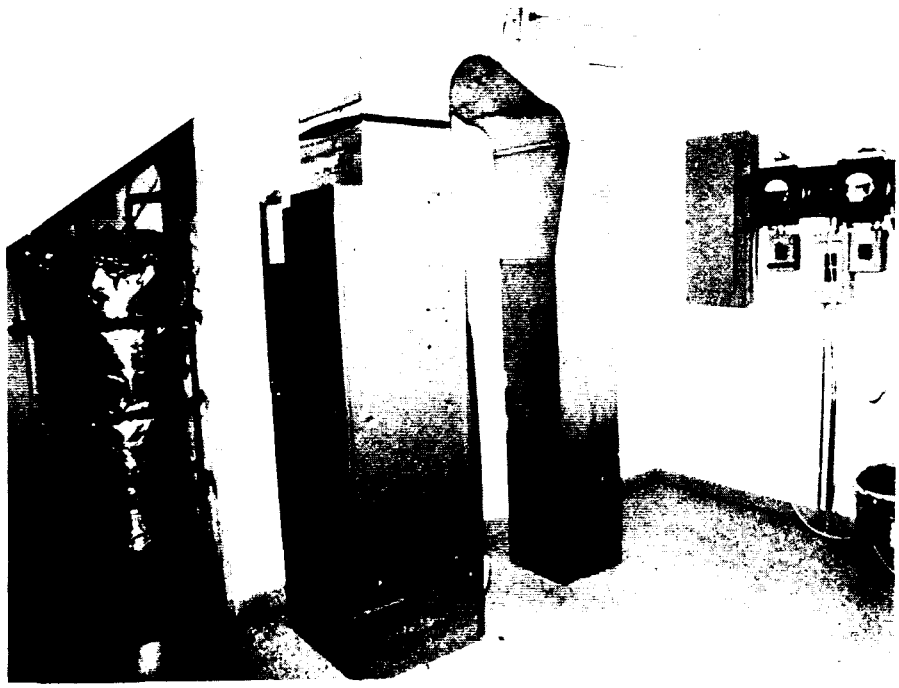

Fig. 6.8. Original control house heat pump indoor unit.

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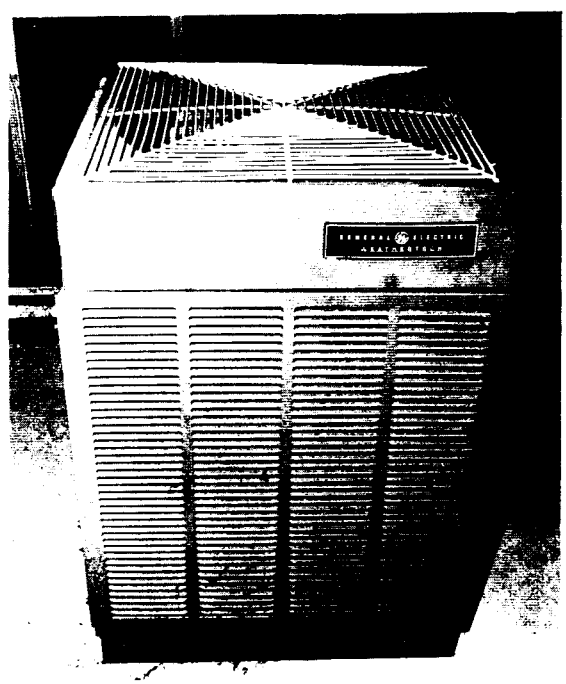

Fig. 6.9. Original control house heat pump outdoor unit.

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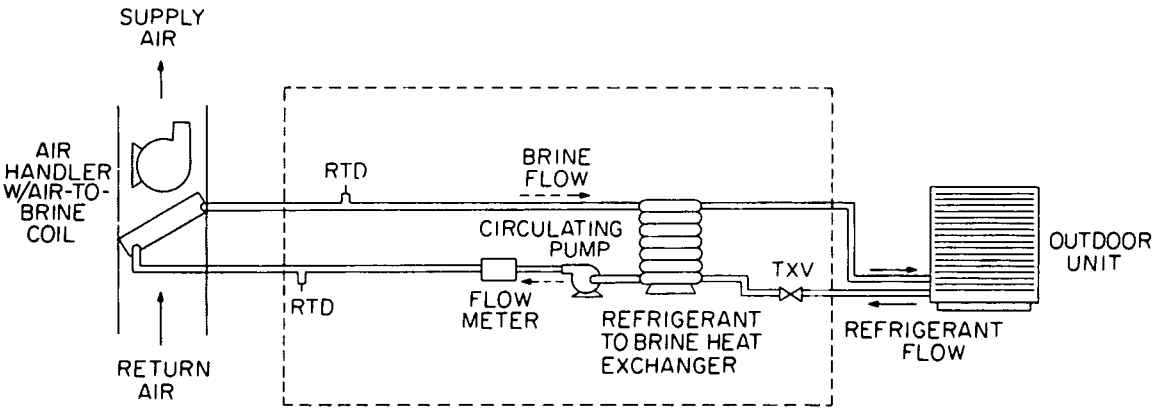

Fig. 6.10. Control house heat pump modifications for cooling load measurement.

experimentally determine the heating and cooling SPFs of a high-efficiency heat pump and to compare its performance with that of the ACES. Figures 6.11 and 6.12 show the indoor and outdoor sections of the new heat pump.

ORNL PHOTO 6903-80

Fig. 6.11. High-efficiency heat pump indoor unit.

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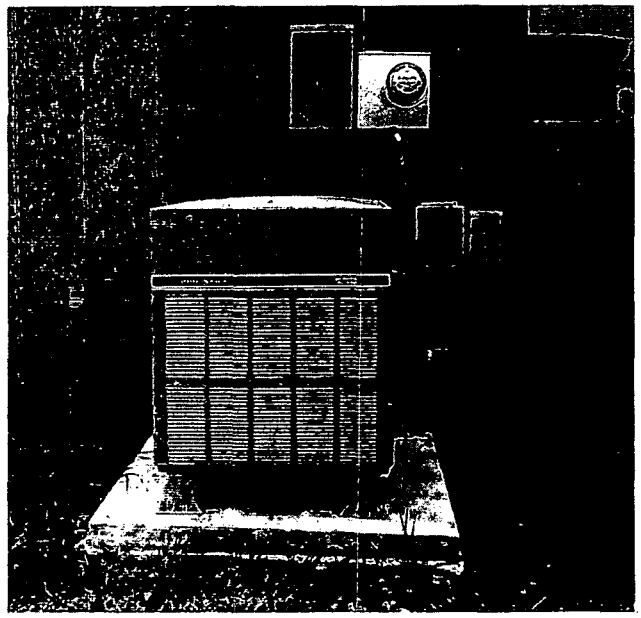

Fig. 6.12. High-efficiency heat pump outdoor unit.

7. STEADY-STATE PERFORMANCE

7.1 ACES

ACES steady-state performance was determined from periods of time in which steady-state operation occurred and from representative surveys of system temperatures and pressures. The following sections present the results obtained from these investigations.

7.1.1 Energy inputs, heat flows, and modal COPs

Table 7.1 gives power consumptions of each of the electrically driven devices under full load conditions in each operating mode. These

Table 7.1. Typical ACES component energy consumption by modes

Device	Energy input (W)
Heating only (ice source)	
Compressor	2490
Fan	480
Hot-brine pump	150
Cold-brine pump	165
Hot water only (ice source)	
Compressor	2600
Cold-brine pump	165
Hot-water pump	81
Heating and hot water (ice source)	
Compressor	2430
Fan	480
Hot-brine pump	150
Cold-brine pump	165
Hot-water pump	81
Night heat rejection (7.2°C water source)	
Compressor	2830
Hot-brine pump	150
Cold-brine pump	165
Hot-water pump	81
Cooling mode	
Fan (1979)	550
(1980)	215
Cold-brine pump	135
Ice melting	
Cold-brine pump	140

items are monitored individually in order to determine where performance improvements can be made and as a diagnostic aid in troubleshooting mechanical or data system malfunctions. The values presented represent typical operating characteristics.

Full-load brine- and water-side heat flows of the system in each mode are presented in Table 7.2. Here again, these are typical values.

Determination of modal-system steady-state COPs was made by selecting periods in which steady-state operation in a given mode was observed. Table 7.3 gives the results of these observations. The bin heat rejection COP is a measure of the bin heat removal performance during night heat rejection. Typical steady-state COP calculations are presented in Appendix E.

Table 7.2. Typical ACES heat flows by modes

Device			Heat flow (W)
Heating only (ice source)			
Fan coil output			6550
Bin extraction			4740
Hot water only (ice source)			
Hot water output			6200
Bin extraction			4740
Heating and hot water (ice source)			
Fan coil output			5415
Hot water output			1820
Bin extraction			5255
Night heat rejection (7.2°C water source)			
Hot water output			1840
Solar panel heat rejection			5800
Bin extraction			5130
Cooling			
Ice source:	Fan coil input	(1979)	9250
		(1980)	7230
	Bin deposit	(1979)	9540
		(1980)	7720
Chilled-water source:	Fan coil input	(1979)	5440
		(1980)	4380
	Bin deposit	(1979)	5840
		(1980)	4610
Ice melting			
Solar panel heat collection			0-12000
Bin deposit			0-12000

Table 7.3. ACES steady-state COP

Mode	COP 1978 - 1979	COP 1977 - 1978[a]
Space heating and water heating (ice source)	2.59	2.70
Space heating only (ice source)	2.51	
Water heating only (ice source):		
Summer	2.18	2.50
Winter	2.67	
Space cooling: Ice source	12.55	12.40
Chilled-water source	6.70	8.69
Bin heat rejection (7.2°C water source)	1.62	2.01

[a]From ref. 8.

Performance values for the unmodified mechanical package are included in Table 7.3 for comparison. As can be seen, the system's steady-state heating performance has deteriorated slightly. The major reason for this degradation is that a larger-capacity compressor was installed while the new bin heat exchanger had 55% less surface area than did the original heat exchanger. This resulted in lower brine temperatures exiting the evaporator as illustrated in Fig. 7.1. A secondary contributing factor is that the fan power draw increased following a failure in which the fan sheave slipped off the shaft. The replacement sheave was slightly smaller than the original, raising the air flow (from 0.38 m^3/s to 0.52 m^3/s) and the power consumption (from 380 W to 480-550 W). In March 1980 the fan belt was tightened somewhat, reducing the fan power draw to around 400 W while maintaining the air flow at about 0.52 m^3/s.

In May 1980 a larger sheave was installed on the fan, reducing the air flow to its design value of 0.38 m^3/s. With the fan belt tightened, the fan motor air-conditioning power consumption was lowered to 200-220 W vs 550 W in 1979. These changes reduced the ACES cooling capacity by 15 to 20%, but the ice source cooling COP was raised to 19.6 and the chilled-water source cooling COP to 12.0.

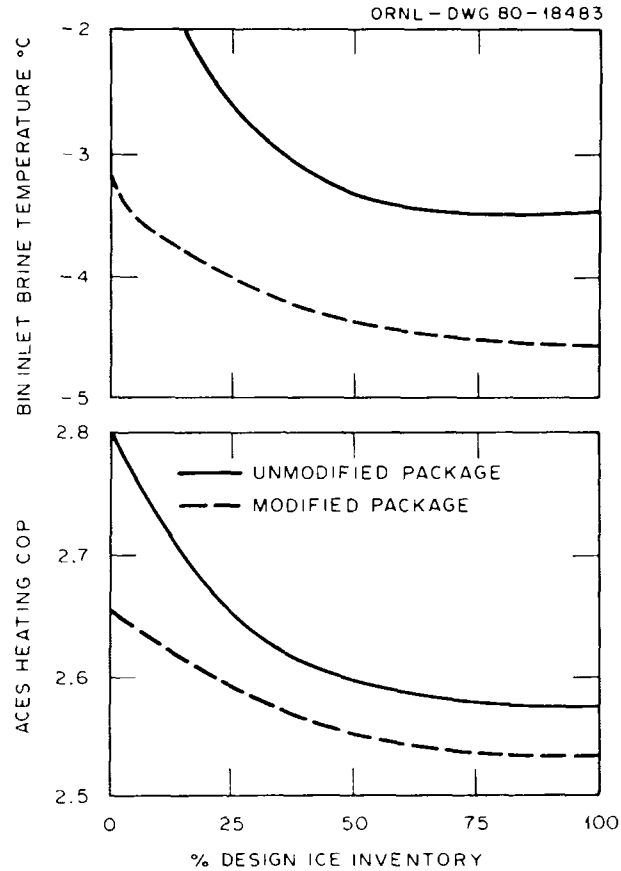

Fig. 7.1. Brine temperature and COP vs ice inventory.

The system water heating COP fell from 2.50 in 1977-78 to 2.18 in 1978-79. This drop is attributed primarily to the larger compressor and smaller bin heat exchanger combination. The water heating COP deteriorated to 2.0 in the third year due to fouling of the refrigerant-to-water heat exchangers in the mechanical package. Periodic chemical cleaning of these heat exchangers should alleviate this problem and is recommended for all ACES installations.

7.1.2 System operating temperatures

Table 7.4 presents typical brine temperatures selected from a number of data scans taken when the system was operating. Flow rates were taken from the in-line turbine flowmeters. Typical refrigerant temperatures and pressures, presented in Table 7.5, were obtained from several representative surveys of the refrigerant system.

Table 7.4. Typical ACES brine system temperatures and flow rates

Device	Mode						
				Space cooling			
	Space heating and water heating	Space heating only	Water heating only	Ice source	Chilled-water source (9.0°C)	Night heat rejection (7.2°C water source)	Ice melting
Fan coil							
In, °C	35.1	38.7		2.2	11.4		
Out, °C	31.9	34.8		8.4	15.3		
Flow, ℓ/s	0.44	0.44		0.36	0.36		
Ice bin							
In, °C	-4.71	-4.3	-4.5	8.4	15.4	0.8	1.1-11.4
Out, °C	-2.71	-2.5	-2.9	2.1	11.4	3.0	0.5- 3.6
Flow, ℓ/s	0.71	0.71	0.71	0.36	0.36	0.71	0.41
Solar panel							
In, °C						43.8	0.6- 3.7
Out, °C						40.5	1.1-11.3
Flow, ℓ/s						0.51	0.41
Hot water							
In, °C	28.0		30.8			58.5	
Out, °C	46.9		50.2			62.3	
Flow, ℓ/s	0-0.19		0-0.19			0.19	

Table 7.5. Typical ACES refrigerant system operating temperatures and pressures

	Mode			
	Space heating and water heating	Space heating	Water heating	Night heat rejection
Compressor discharge temperature, °C	101.2	101.4	104.4	110.1
Temperature leaving desuperheater, °C	57.1	83.3	54.7	70.3
Temperature leaving hot-water condenser, °C	50.1	76.3	45.6	60.5
Temperature leaving heating condenser, °C	37.9	39.4	42.9	46.0
Temperature entering expansion valve, °C	36.4	36.4	41.0	44.6
Evaporating temperature, °C	-6.7	-6.7	-6.1	0
Temperature leaving evaporator, °C	-3.0	-3.0	-2.0	0.7
Compressor suction temperature, °C	19.4	21.2	21.4	27.3
Discharge pressure, kPa	1542	1576	1896	1931
Suction pressure, kPa	303	308	315	393

7.2 Control House Heat Pumps

Steady-state performance of the original control house heat pump was measured periodically during the heating season. Results of these tests are presented in Table 7.6 along with the ARI rating.

Table 7.6. Control house ARI 2.46 heat pump steady-state performance; as-built configuration

	On-site test	ARI rating point
Outdoor temperature, °C	11.1	8.3
Heat output, W	10,856	9085
Outdoor unit power, W	3,680	
Indoor unit power, W	530	
Total power, W	4,210	3700
Space heating COP	2.58	2.46

After the unit was modified to enable cooling-load measurement, its steady-state performance was again measured. Comparison of this performance with that of the unmodified unit is illustrated in Table 7.7 for the cooling mode. As can be seen, there was a 24% reduction in the steady-state cooling COP while the cooling capacity was reduced by 40%.

Table 7.7. Heat flow instrumented heat pump performance vs rated performance of original unit (cooling mode)

	On-site test	ARI rating point
Outdoor temperature, °C	31.1	35
Cooling capacity, W	5300	8792
Outdoor unit, W	2830	
Indoor unit, W	510	
Total power, W	3340	4300
COP	1.59	2.04

The modified heat pump was replaced with a higher-efficiency heat pump in November 1979. ARI ratings along with the results of two on-site performance tests are presented in Table 7.8.

Table 7.8. ARI ratings and experimental performance test results for the control house state-of-the-art heat pump (ARI 3.11)

	On-site test 1 actual performance	On-site test 2 actual performance	High-temperature ARI rating	Low-temperature ARI rating
	Heating Mode			
Outdoor temperature, °C	9.7	8.0	8.3	-8.3
Heating capacity, W	10,892	10,328	9965	5276
Outdoor unit, W	3,147	3,086		
Indoor unit, W	395	389		
Total power, W	3,542	3,475	3200	2550
COP	3.08	2.97	3.11	2.07
	Cooling Mode			
Outdoor temperature, °C	35.2	26.5	35	
Cooling capacity, W	9,379	10,424	9818	
Outdoor unit, W	3,411	3,037		
Indoor unit, W	405	404		
Total power, W	3,816	3,441	3900	
COP	2.46	3.03	2.52	

8. ANNUAL CYCLE PERFORMANCE

8.1 Annual Energy Consumption and Loads

Due to the interseasonal energy storage feature of the ACES, the true measure of its performance can only be determined from data for an entire year of operation. Figures 8.1 and 8.2 compare the ACES with the heat pump and electric resistance (I^2R) hot-water systems of the control house. Figure 8.1 illustrates the annual loads and energy consumption required to meet those loads. The ACES exhibited an ACOP of 2.80 in the second year and 3.08 in the third year, while the control house demonstrated ACOPs of 1.42 and 1.73 respectively. Calculations predict ACOPs of 3.18 for ACES, 1.67 for the ARI 3.11 heat pump/I^2R water heat system, and 1.38 for the ARI 2.46 heat pump/I^2R water heat system.[2]

Purchased energy consumption reductions of 48% for 1978-79 and 43% for 1979-80 were demonstrated by the ACES over the two heat pump/I^2R water heating systems.

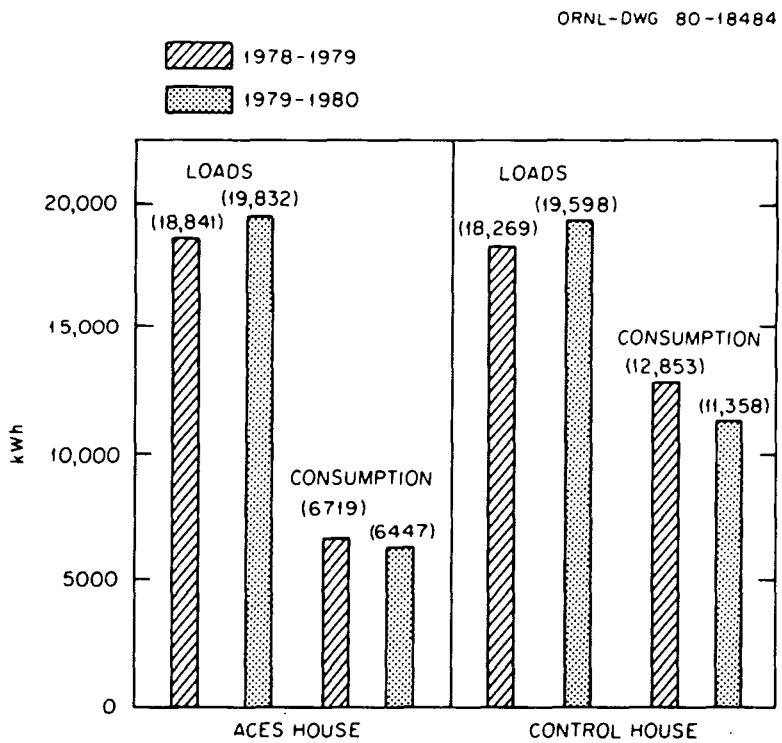

Fig. 8.1. Comparison of ACES and control house loads and power consumptions.

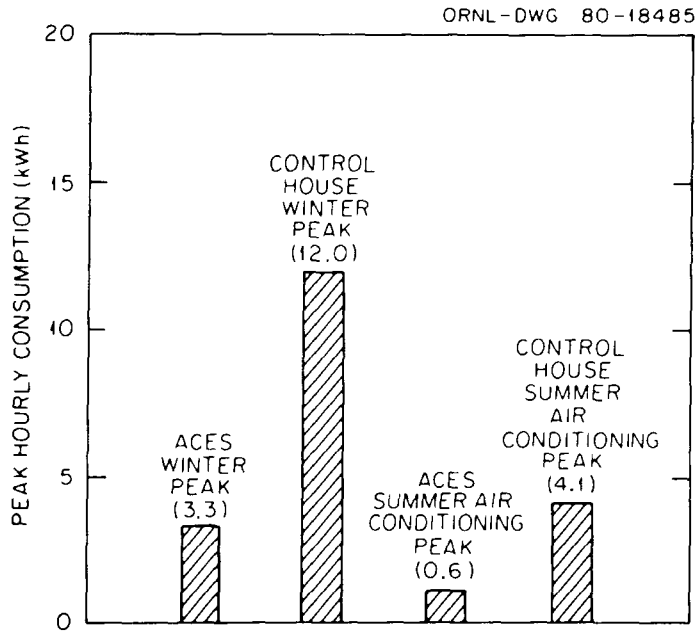

Fig. 8.2. Comparison of ACES and control house peak hourly consumption.

Peak hourly consumption for heating and summer air conditioning is shown in Fig. 8.2. As illustrated, the ACES consistently exhibited much lower peak loads than did the heat pump/I^2R water heating systems. Winter peak reductions were achieved through the absence of a need for supplemental heat, using the heap pump for water heating, and through the ACES constant performance. Summer air-conditioning reductions resulted from providing all on-peak air conditioning from the storage bin rather than operating the compressor. Instantaneous summer water heating peaks were reduced from about 4.1 kW to about 2.8 kW.

Table 8.1 presents monthly loads and power consumptions for both houses along with ACES ice bin performance information. The control house heating load was not measured directly. However, based on results obtained during the first year's operation,[8] its load was assumed to be equivalent to the ACES house load. Due to the ACES system outages in late December and early January of the second year, heating loads for that period were estimated. The details of this estimation and other data corrections are contained in Appendix C.

Table 8.1. Monthly performance summary for the ACES and control houses

Month	Control house				ACES house				Ice bin			Internal loads	
	Loads (GJ)			Electrical consumption (kWh)	Loads (GJ)			Electrical consumption (kWh)	Ice inventory (GJ)	Estimated heat leakage (GJ)	Avg. tank temperature (°C)	Control (GJ)	ACES (GJ)
	Heating	Cooling	Hot water		Heating	Cooling	Hot water						
1978-1979 Test Year													
December 1978	7.29[a]	0	1.65	1,670	7.29[a]	0	1.75	997[a]	1.43	1.53	1.7	3.04	3.50
January 1979	12.12	0	1.80	2,542	12.12	0	1.84	1,539	7.69	1.21	0	3.23	2.99
February 1979	8.41	0	1.60	2,037	8.41	0	1.70	1,121	13.50	1.09	0	2.82	2.46
March 1979	3.92	0	1.56	1,160	3.92	0	1.71	610	15.17	1.21	0	2.63	2.52
April 1979	1.11	0	1.42	611	1.11	0	1.53	284	15.65	1.17	0	2.51	2.41
May 1979	0	0.58	1.36	476	0	0.58	1.57	215 (622)[c]	16.92[b]	1.21	0	2.66	2.74
June 1979	0	4.15	1.16	1,023	0	4.24	1.35	278	11.71	1.90	0	2.54	2.56
July 1979	0	4.33	1.01	1,015	0	4.47	1.11	249	5.30	2.14	1.1	2.79	2.36
August 1979	0	6.85	1.09	1,462	0	6.02	1.41 (1.18)[d]	534	0	1.28	4.4	2.66	2.77
September 1979	0	3.28	1.09	857	0	3.86	1.79 (1.09)[d]	892	0	0.50	7.2	2.24	2.82
October 1979[e]	0	0.03	1.33	382	0.25	0.13	1.22	240	0	0.88	7.2		
November 1979[e]	3.26	0	1.40	845	3.26	0	1.55	513	0	1.89	1.1		
Total	32.85	19.18	13.74	12,853	32.85	19.17	15.81 (14.88)[d]	6,719		12.52		27.12	27.13
1979-1980 Test Year													
December 1979	7.70	0	1.62	1,573	7.70	0	1.66	994	3.42	1.53	0	2.83	3.08
January 1980	8.35	0	1.64	1,546	8.35	0	1.63	1,078	7.64	1.88	0	2.97	2.97
Febuary 1980	9.95	0	1.60	1,845	9.95	0	1.68	1,265	14.24	0.38	0	2.63	2.43
March 1980	6.49	0	1.71	1,370	6.49	0	1.72	893	18.42	0.63	0	2.57	2.39
April 1980	1.68	0	1.56	691	1.79	0	1.52	382	19.42	0.92	0	2.36	2.44
May 1980	0.10	0	1.56	484	0.10	0.03	1.57	219	19.42	1.06	0	2.27	2.43
June 1980	0	5.26	1.24	990	0	5.26	1.15	252	12.55	1.54	0	2.32	2.47
July 1980	0	8.35	1.10	1,392	0	8.35	1.10	289	3.20	1.89	2.0	2.65	2.59
August 1980	0	6.70	1.02	1,055	0	6.62	1.49 (1.02)[d]	621	0	0.19	8.0	2.54	2.51
September 1980	0	2.39	0.51	412	0	2.39	0.82 (0.51)[d]	455	0	0.13	12.8	1.22	1.33
Total	34.29	22.70	13.57	11,358	34.40	22.65	14.34 (13.56)[d]	6,447		10.15		24.36	24.64

[a]Estimated.

[b]Includes 2.17 GJ added during bin excess capacity experiment in May 1979.

[c]Includes 407 kWh consumed during bin excess capacity experiment.

[d]Numbers in parentheses are actual hot-water requirements. Delivered loads were higher due to overheating of water during night heat rejection operation.

[e]Not included in totals. Control house heat pump replaced.

Analysis of the internal house loads indicates that the control house had 0.35 GJ greater internal loads than did the ACES house in the 1978-79 heating season. The ACES internal loads exceeded those of the control house in the 1979 cooling season by 0.36 GJ. In both cases these differences are less than 2.5% of the measured loads. This is within the estimated accuracy level of the load measurement for the second test year; therefore, the differences are not considered significant.

In the 1979-80 heating season, the control house had internal loads greater than those of the ACES house by 0.05 GJ. Adjustment of the control house heating load by this amount does not significantly affect either the heat pump SPF or the ACOP.

8.2 Seasonal Performance

The following sections present a breakdown of the annual performance of both ACES and control houses by season, pointing out the primary areas of the ACES energy conservation potential and peak load leveling ability.

For the purposes of this report, the heating season is defined to run from December 1 to April 30, and the cooling season from May 1 to September 30. November is normally considered part of the heating season in the Knoxville area; however, the modification work prior to the second year's operation delayed the start of the test year until December.

8.2.1 Heating season

The ACES displayed significant reductions in energy consumption during the observed heating seasons. Table 8.2 shows the space and water heating loads incurred by the ACES and control houses and the energy necessary to meet those loads. Heating season electricity consumption by the ACES is 43% less than that of the heat pump/I^2R hot-water system in the second year and 34% less than that of the state-of-the-art heat pump/I^2R hot-water system in the third year. In addition, ice was stored for summer cooling during the heating process.

Table 8.2. Winter loads and energy consumptions.

Function	Control house Load (GJ)	Control house Consumption (kWh)	ACES house Load (GJ)	ACES house Consumption (kWh)
	December 1, 1978, to April 30, 1979			
Heating	32.85	5786	32.85	
		1.58[a]		4551
Water heating	8.04	2234	8.54	
Total	40.89	8020	41.39	4551
		1.41[b]		2.53[b]
	December 1, 1979, to April 30, 1980			
Heating	34.18	4766	34.30	
		1.99[a]		4612
Water heating	8.13	2259	8.21	
Total	42.31	7025	42.51	4612
		1.67[b]		2.56[b]

[a]Seasonal performance factor for space heating.

[b]Seasonal performance factor for space heating and water heating.

The ACES heating season energy conservation is a result of its constant capacity and performance characteristic. As shown in Fig. 7.1, the ACES COP was virtually independent of the bin ice inventory, clearly illustrating its fixed capacity. This feature eliminates the need for supplemental heating, since an ACES can be sized to deliver any design day load. Delivery of domestic hot water at a COP of 2.22 in 1978-79 and 2.0 in 1979-80 as opposed to 1 for standard electric water heaters also contributes to the ACES' energy conservation.

The reduction in water heating COP from the second test year to the third test year is a result of fouling of the mechanical package refrigerant-to-water heat exchangers. Regular chemical cleaning (about once every four years) could prevent excessive fouling.

As in the first year,[8] the ACES mechanical package exhibited only minor performance degradation due to cycling. The ACES heating season COPs were 2.53 in the second year and 2.56 in the third year. Steady-state COPs for space and water heating ranged from 2.51 to 2.67.

The original control house heat pump, ARI rated 2.46 at 8.3°C and 1.7 at -8.3°C, showed a heating season SPF of 1.58. Its performance degradation was due to a combination of cycling losses, frosting-defrosting losses, and the requirement of supplemental electric resistance heat. Table 8.3 shows the monthly performance of this heat pump during the heating season. The COP varied from 1.65 in January (average temperature -0.2°C) to 1.43 in April (average temperature 14.4°C). This clearly shows the severe performance penalty, presumably from cycling, imposed on the heat pump during periods of low heating need.

Table 8.3. Heating season performance of the original control house heat pump[a]

Month	Heating load (GJ)	Average outdoor temperature (°C)	Supplemental electric heat required (kWh)	Total purchased power (kWh)	Monthly heating COP
December	7.29[b]	3.3	120[b]	1211	1.67[b]
January	12.12	-0.2	336[b]	2042	1.65
February	8.41	2.6	221	1592	1.47
March	3.92	10.5	14	727	1.50
April	1.11	14.4	4	214	1.43
Total	32.85	6.2	695	5787	1.58

[a]General Electric Weathertron heat pump, ARI rated COP of 2.46 at 8.3°C and 1.7 at -8.3°C.

[b]Estimated.

Monthly performance of the high-efficiency heat pump is given in Table 8.4. This heat pump has demonstrated an SPF of 1.99 compared with ARI ratings of 3.11 at 8.3°C and 2.07 at -8.3°C. This illustrates the combined effects of frosting-defrosting, cycling, and supplemental heat. Total supplemental heat requirements were about 692 kWh. Neglecting the supplemental resistance heating, the heat pump pumped heat seasonal COP is 2.16. For the average ambient temperature of 6.4°C, this indicates cycling and frosting-defrosting performance degradations of 28% from an expected pumped heat seasonal COP of 2.99 based on rated steady-state performance.

Table 8.4. High-efficiency heat pump[a] heating season performance

Month	Load (GJ)	Average outdoor temperature (°C)	Supplemental electric heat required (kWh)	Total consumption (kWh)	COP
December	7.70	3.9	118	1122	1.91
January	8.35	4.7	160	1090	2.13
February	9.95	0.8	266	1401	1.97
March	6.49	8.3	144	896	2.01
April	1.68	14.3	4	256	1.82
Total	34.18	6.4	692	4765	1.99

[a]General Electric Weathertron II ARI rated COP of 3.11 at 8.3°C and 2.07 at -8.3°C.

Load management is of great interest to electric utilities because of their problem of maintaining adequate on-line generating capacity during high peak periods. The ACES constant-capacity feature enabled it to reflect a low peak load to the utility throughout the heating season. Figures 8.3 and 8.4 show typical weekly energy requirements of the ACES vs those of the control house with the original and the state-of-the-art heat pumps respectively. Both consumption curves have the same shape, but the peak hourly load imposed on the utility by the ACES was only 31% as high as that of the control house. Thus the ACES can reduce winter peak loads by as much as 69% compared with heat pump/I^2R hot-water systems.

8.2.2 Cooling season

Stored-ice phase. The distinguishing feature of an ACES is the interseasonal energy transfer which occurs when ice produced during the heating season is stored and used to meet space cooling demands the following summer. With stored ice the ACES cooling COP is 12.2 to 19.6 compared with steady-state COPs of 2.0 to 3.0 for conventional air conditioners.[12] Thus, cooling season energy consumption is significantly reduced by the ACES.

Table 8.5 shows the measured cooling loads and power consumptions for both ACES and control houses during the stored-ice phase. Due to the performance degradation caused by the addition of the heat flow

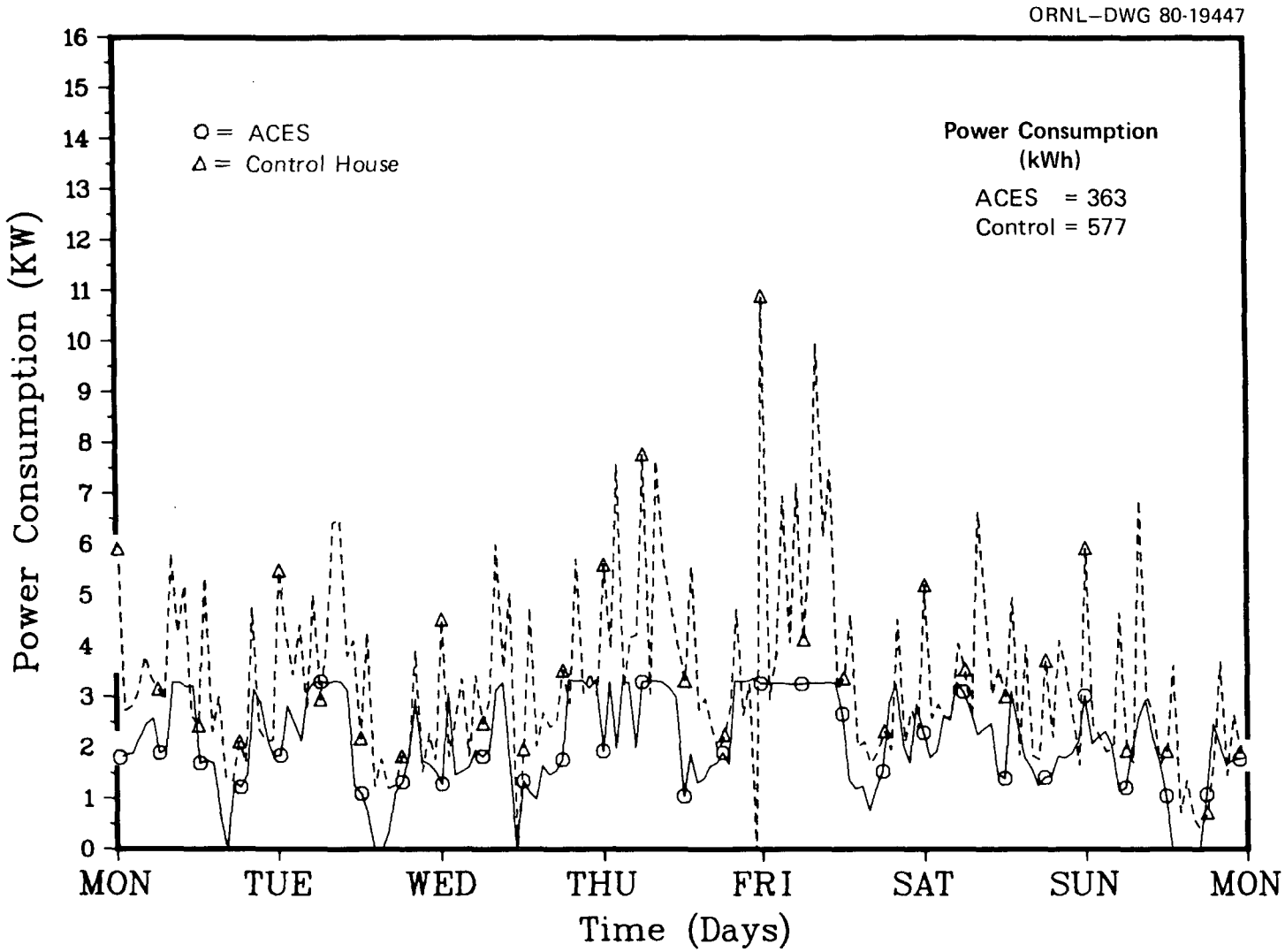

Fig. 8.3. ACES and control house power consumption for week beginning Jan. 29, 1979.

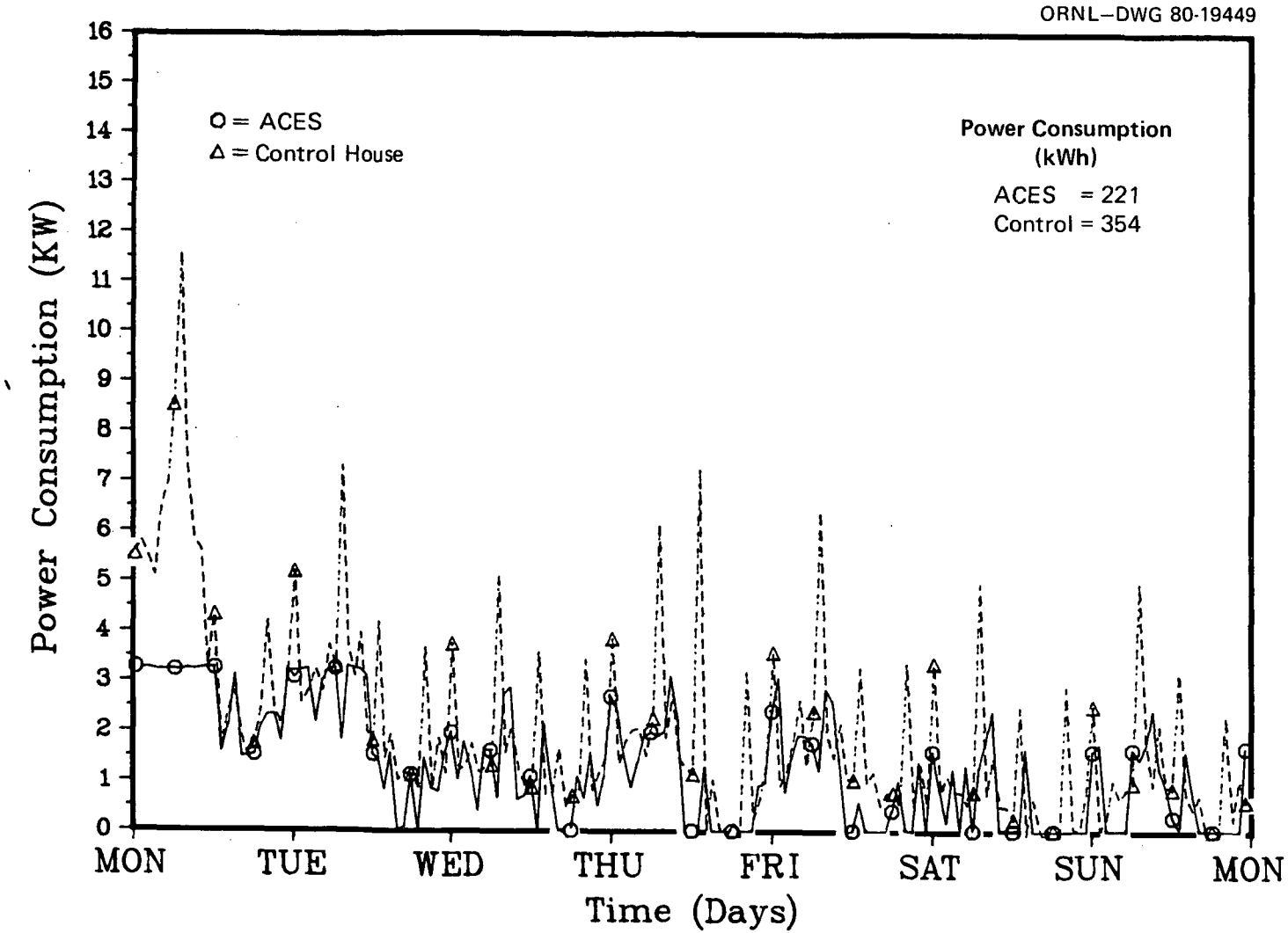

Fig. 8.4. ACES and control house power consumption for week beginning Mar. 3, 1980.

Table 8.5. Summer energy consumptions and loads; stored-ice phase.

Function	Control house Load (GJ)	Control house Consumption (kWh)	ACES house Load (GJ)	ACES house Consumption (kWh)
	May 1, 1979, to Aug. 24, 1979			
Cooling	14.51	2457[a] (2839)	13.92	424
Water heating	4.38	1218	5.01	617
Total	18.89	3676	18.93	1041
	June 1, 1980, to Aug. 14, 1980			
Cooling	16.73	2124	16.65	255
Water heating	2.78	773	2.76	401
Total	19.51	2897	19.41	656

[a]Calculated from cooling load and 1978 SPF of 1.64. Uncorrected consumption in parentheses.

measurement loop, during the second year, the control house heat pump power consumption was calculated by using the 1979 measured cooling load and a cooling SPF of 1.64 (measured in 1978[8]) as follows:

$$\text{heat pump power consumption} = \frac{\text{cooling load}}{1.64} .$$

The ACES reduced power consumption by 72% for this period during the 1979 season and by 77% in 1980. The cooling SPF for the control house during the 1980 season was 2.27.

Peak load reduction during the summer was quite significant, as illustrated in Fig. 8.5. Air-conditioning peak consumption for the ACES was about 0.68 kWh vs 4.1 kWh for the control house. Water heating produced peak loads of about 2.9 kWh for the ACES compared with 4.5 kWh for a standard electric water heater.

Supplemental phase. Upon exhaustion of the ice supply, it was necessary to operate the ACES in the supplemental cooling phase. In this phase the system operated off-peak in the night heat rejection mode to chill the water in the storage bin to meet daytime air-conditioning needs. As noted in ref. 8, several problems were evident during the

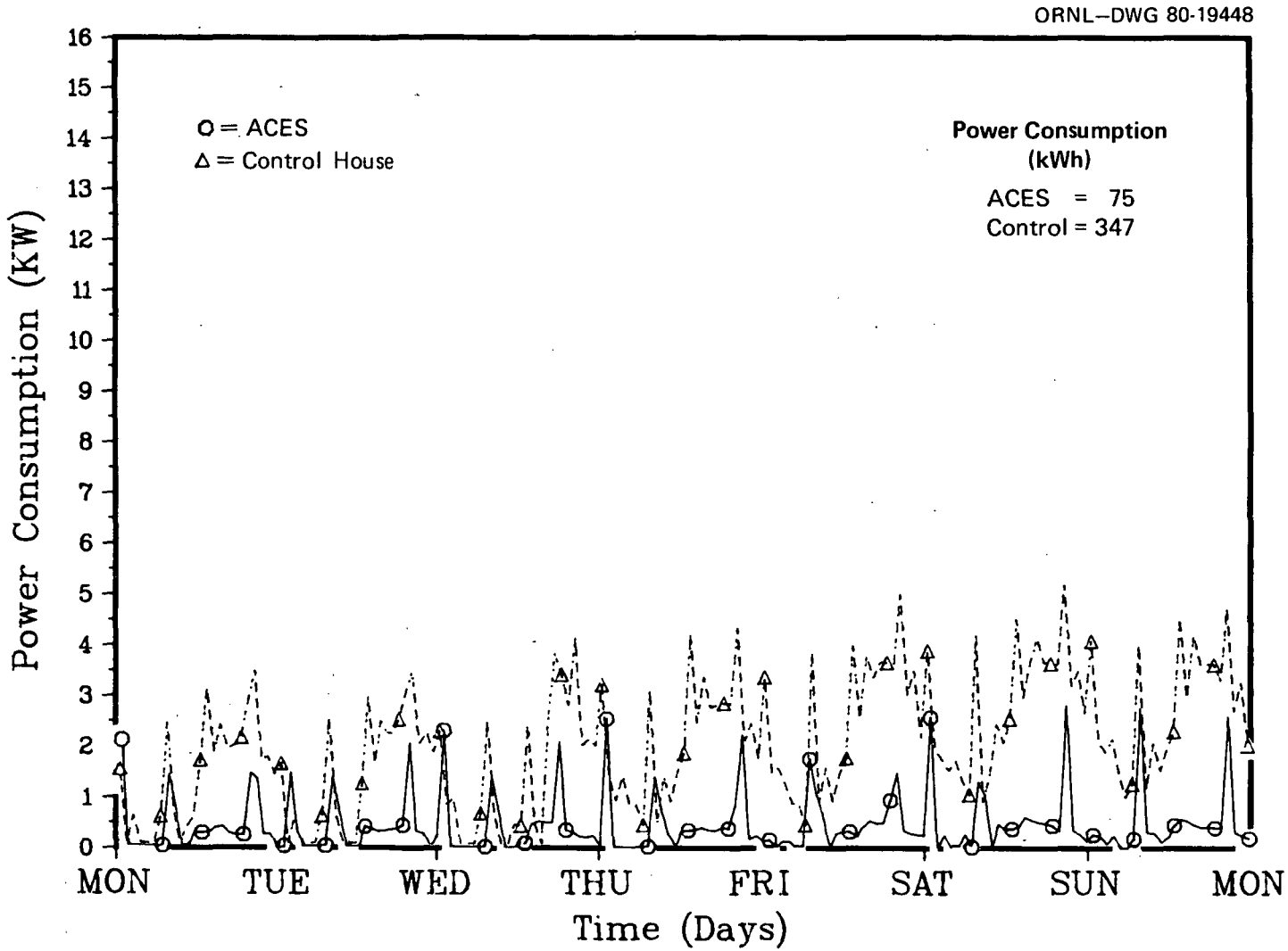

Fig. 8.5. ACES and control house power consumption for the summer week beginning June 4, 1979 (ice cooling).

first year's night heat rejection operation, including (1) excess compressor operation due to improper control settings and (2) excess cooling loads due to having the mechanical equipment in the conditioned space. Before night heat rejection operation began in the second year, appropriate control logic changes were made as noted earlier, and the mechanical room was isolated and vented to the outdoors.

Table 8.6 shows loads and power consumptions of both houses for the supplemental phase. The success of the efforts to reduce power consumption during the supplemental cooling phase is reflected in the fact that in the second year the ACES consumed about 3% less electricity than did the control house. The ACES consumed 12.7% more electricity than did the control house for this phase during the first year.[8] In the third year, ACES consumption was 0.9% more than that of the control house.

Table 8.6. Loads and energy consumptions during supplemental cooling phase

Function	Control house		ACES house	
	Load (GJ)	Consumption (kWh)	Load (GJ)	Consumption (kWh)
	Aug. 24, 1979, to Oct. 1, 1979			
Cooling	4.67	791[a]	5.25	
		(915)		1127
Water heating	1.32	367	2.26[b]	
			(1.32)	
Total	5.99	1158	7.51[b]	1127
			(6.57)	
	Aug. 14, 1980, to Sept. 15, 1980			
Cooling	5.96	650	5.96	
				961
Water heating	1.09	302	1.81[b]	
			(1.09)	
Total	7.05	952	7.76[b]	961
			(7.05)	

[a]Estimated from cooling load and 1977-78 cooling SPF of 1.64. Actual consumption in parentheses.

[b]ACES water heating load excessive due to overheating of water in night heat rejection phase. Actual requirements are approximately those of the control house.

Reducing fan energy consumption and carefully controlling the hours of compressor operation to match the house cooling requirements improved the ACES supplemental cooling COP; during 1979 the supplemental cooling-phase COP was 1.85, while in 1980 the value was 2.25.

Total cooling and water heating energy requirements for the 1979 and 1980 cooling seasons are given in Table 8.7. The ACES reduced cooling season energy consumption by 55% over the ARI 2.46 heat pump/I^2R hot-water system and by 58% over the ARI 3.11 heat pump/I^2R hot-water system.

Peak load reduction in the supplemental cooling phase is accomplished by running the mechanical package in the night heat rejection mode

Table 8.7. Cooling season loads and energy consumptions

Function	Control house		ACES house	
	Load (GJ)	Consumption (kWh)	Load (GJ)	Consumption (kWh)
	May 1, 1979, to Oct. 1, 1979			
Cooling	19.18	3248[a]	19.17	
		(3763)		
		1.64[b]		2168
Water heating	5.70	1585	7.27[c]	
			(6.33)	
Total	24.88	4833	26.44[c]	2168
		1.43[d]	(25.50)	3.39[d]
	May 1, 1980, to Sept. 15, 1980			
Cooling	22.70	2824	22.65	
		2.27[b]		1836
Water heating	5.43	1509	6.14[c]	
			(5.41)	
Total	28.13	4333	28.79[c]	1836
		1.83[d]	(28.06)	4.36[d]

[a]Calculated from load and 1978 SPF of 1.64. (Actual consumption in parentheses.)

[b]Seasonal performance factor for cooling.

[c]ACES water heating loads excessive due to overheating of water in night heat rejection mode. Actual requirements are approximately those of the control house.

[d]Seasonal performance factor for space cooling and water heating.

during off-peak, night time periods to store cooling energy for use in the peak daytime hours. Figure 8.6 shows the energy requirements of the ACES and control houses during a typical week of supplemental cooling operation.

Heat pump cooling season performance. This section gives a brief summary of the air-conditioning performance of both the standard quality heat pump (instrumented for heat flow measurement) and the state-of-the-art heat pump. The standard quality heat pump was originally rated by ARI as having a 2.04 cooling COP at 35°C; however, the modifications made to enable cooling load measurement degraded this performance to 1.59. The state-of-the-art heat pump has an ARI rated cooling COP of 2.52. Table 8.8 shows the performance of the heat flow instrumented unit during the 1979 cooling season together with the expected performance of the unmodified version. The actual cooling SPF was found to be 1.41 compared with the steady-state COP of 1.59 at 31.1°C, indicating cycling losses of at least 11%. It should be pointed out, however, that the capacity of the modified unit was only 60% of that of the original, as noted in Table 7.7. Therefore, the unmodified heat pump would be expected to incur greater cycling losses.

Table 8.9 shows the monthly cooling season performance of the high-efficiency heat pump. This unit yielded a cooling SPF of 2.27 compared with its rating of 2.52 at 35°C. Based on the ARI rating, air-conditioning cycling losses of this heat pump were 12%.

8.3 Weather Conditions

Actual weather data for the test years along with long-term average weather data are given in Table 8.10. These data were taken from recordings by the National Climatic Center at the Knoxville, Tennessee, McGhee-Tyson Airport, Station 13891, located about 11 km from the TECH site. Degree-days are referenced to 18.3°C for both heating and cooling.

For the 1978-79 heating season, December and March produced lower than normal heating loads. January and February, however, were much colder than normal, equivalent to the January-February 1970 period with 984°C-days. April was about normal, with 93°C-days.

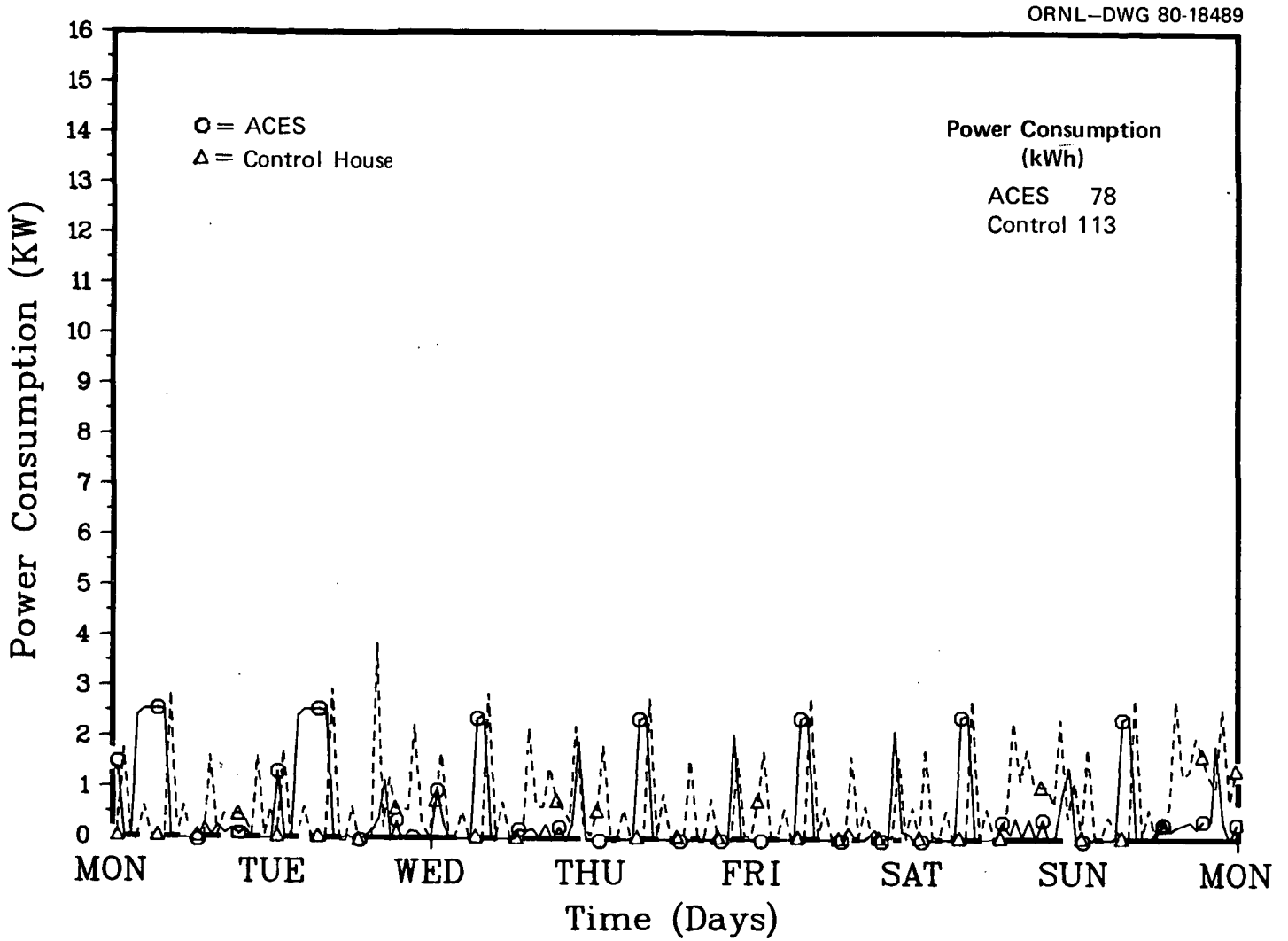

Fig. 8.6. ACES and control house power consumption for week beginning Sept. 24, 1979.

Table 8.8. Heat flow instrumented heat pump air-conditioning performance (1979 season)

Month	Cooling load (GJ)	Power consumption (kWh)		Average outdoor temperature (°C)	Actual monthly COP
		Actual[a]	Calculated[b]		
May	0.58	114	98	18.5	1.41
June	4.15	803	702	21.7	1.43
July	4.33	850	733	22.9	1.41
August	6.85	1361	1159	23.7	1.40
September	3.28	643	556	20.6	1.42
Total	19.18	3771	3248	21.5	1.41

[a]The control house heat pump was modified to accurately measure the cooling load in 1979 at the expense of performance.

[b]Calculated using measured 1978 cooling SPF of 1.64.

Table 8.9. State-of-the-art heat pump air-conditioning performance (1980 season)[a]

Month	Cooling load (GJ)	Power consumption (kWh)	Average outdoor temperature (°C)	Monthly COP
June	5.26	646	23.5	2.26
July	8.35	1086	26.7	2.13
August	6.70	772	25.8	2.41
September	2.39	270	24.5	2.46
Total	22.70	2775	25.2	2.27

[a]Data collected from June 1 to Sept. 15, 1980; natural ventilation was the only cooling method used during May 1980.

The 1979-80 heating season weather was similar to that of the 1978-79 season overall. December and January were warmer than normal, while February, March, and April were colder than normal.

The total heating degree-days for the December 1978 to May 1979 period were 1664°C-days compared with the normal 1572°C-days. This winter produced a 6% higher than normal heating demand. For the December 1979 to May 1980 period, there were 1636°C-days or a 4% higher than normal heating demand.

Table 8.10. Weather conditions

	Degree-days,[a] heating			Degree-days,[a] cooling		
	Normal (°C)	Actual (°C) 1978-79	Actual (°C) 1979-80	Normal (°C)	Actual (°C) 1978-79	Actual (°C) 1979-80
December	405	378	389			
January	420	547	397			
February	350	437	453	4		
March	269	209	294	9	8	
April	96	93	103	18	11	9
May	26	26	18	84	62	76
June				175	134	175
July				227	176	299
August				212	222	292
September	6			116	124	116[b]
Total	1572	1689	1654	845	738	967[b]

[a]Referenced to 18.3°C.
[b]Through September 15.

The second test year cooling season weather was much milder than normal. May, June, and July were significantly cooler than normal, while August and September were about normal. The cooling season exhibited 719°C-days from May through September compared with the normal 814°C-days, or a 12% lower than normal cooling demand.

The 1980 cooling season was one of the hottest on record for the Knoxville area. There was a total of 766°C-days from June through August for a 25% greater than normal cooling load.

8.4 Ice Bin Performance

The ice bin is the component which stores for summer use the ice produced during the winter and provides a constant-source temperature for the heat pump. Figure 8.7 shows the ice inventory history of the past three years. In order to accomplish its storage function successfully, the bin must be well insulated from its environment. As originally constructed, the bin was integrated into the house structure. Two of

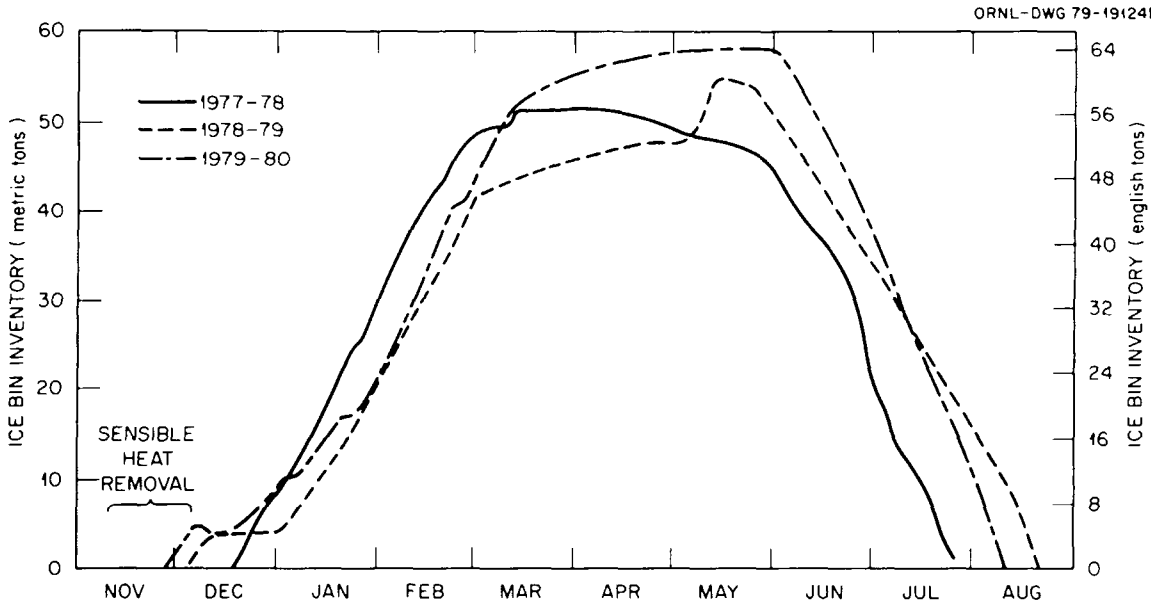

Fig. 8.7. Ice inventory history, 1977-80.

the walls were part of the foundation, and the bin ceiling was part of the first floor.

First-year test results showed that the original insulation levels (6.7 $m^2 \cdot K/W$ ceiling, 3.0 $m^2 \cdot K/W$ walls, and 0.7 $m^2 \cdot K/W$ floor) were inadequate to maintain sufficient ice storage. In an effort to reduce the heat leakage, the tank was reinsulated to about 7.05 $m^2 \cdot K/W$ on all six sides. Leakage rates were reduced by an average of about 40% from the rates experienced during the first year; however, they were still more than twice as great as theoretical estimates. Table 8.11 shows estimated actual bin leakages for all three years along with a theoretical estimate, calculated by the MAD program.[2]

The location of the ACES house bin is at least partially responsible for the disparity between the calculated and measured leakages. For the calculations the bin is assumed to be surrounded on all sides by earth, whereas the ACES house bin has two exposed surfaces. The bin roof is the floor of the heated space, and one wall, containing an access hatch and a viewing window, is half exposed to the basement. Uncertainties in predicting ground temperature also have an effect.

Table 8.11. Theoretical and estimated actual ice bin heat leakage

Month	Theoretical Calculation[a]		Actual estimates					
			1977-78[b]		1978-79		1979-80	
	Heat leak (GJ)	Tank temperature (°C)	Heat leak (GJ)	Tank temperature (°C)	Heat leak (GJ)	Tank temperature (°C)	Heat leak (GJ)	Tank temperature (°C)
November	0.54	3.1	0.79	7.2			1.89	2.2
December	0.55	0	1.59	4.4	1.53	1.7	1.53	1.7
January	0.44	0	2.05	0	1.21	0	1.88	0
February	0.37	0	1.57	0	1.09	0	0.38	0
March	0.44	0	1.72	0	1.21	0	0.63	0
April	0.53	0	1.86	0	1.17	0	0.92	0
May	0.70	0	2.31	0	1.21	0	1.06	0
June	0.81	0	2.74	0	1.90	0	1.54	0
July	0.95	0	3.60	2.7	2.14	1.1	1.89	2.0
August	0.93	1.2	1.84	7.2	1.28	4.4	0.19	8.0
September	0.59	7.2	1.91	8.8	0.50	7.2	0.13	12.8
October	0.51	7.0			0.88	7.2		

[a]Calculated by Manual ACES Design (MAD) program (see ref. 2).

[b]Ice bin floor uninsulated.

Ground temperatures were measured periodically throughout the test year. Results of these measurements are illustrated in Fig. 8.8 along with those made during the previous year. Theoretical earth temperatures were calculated by the Kusuda-Achenbach method.[13] As can be seen, the added insulation substantially reduced the effect of the bin on the surrounding earth.

The effectiveness of the extra insulation in improving the interseasonal energy transfer performance of the bin is discussed in the following sections.

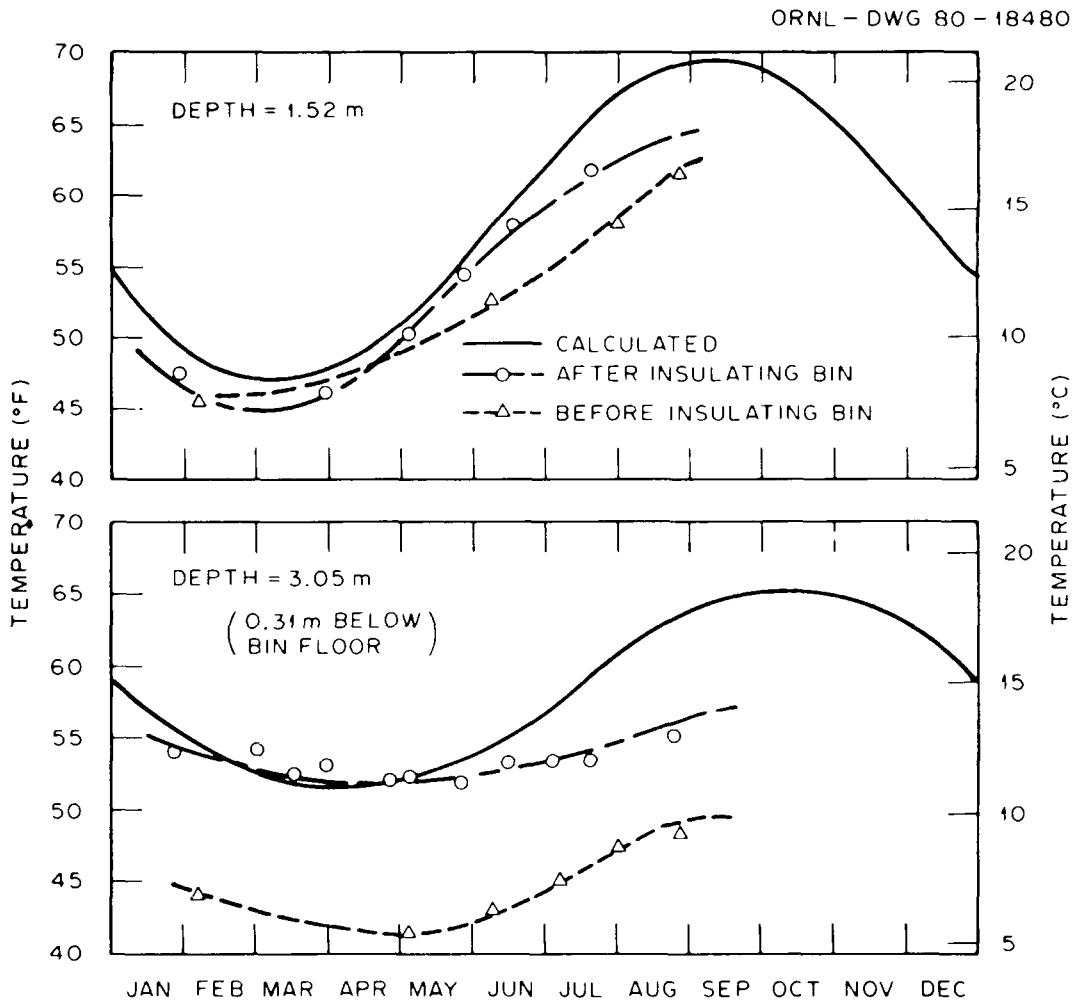

Fig. 8.8. Calculated and measured earth temperatures around ice bin.

8.4.1 Heating season

During the 1978-79 season, ice first appeared on December 11, 1978. However, due to previously discussed mechanical problems, ice production did not begin until January 8, 1979. From January 8 until April 30, the ACES delivered 28.99 GJ of space heating and water heating at a COP of 2.53. The fraction of these loads delivered from the ice bin was

$$\alpha(H) = \frac{COP - 1}{COP} = 0.605 \ .$$

This yields a total heat withdrawal from the bin of 17.54 GJ, equivalent to 52.5 metric tons (t) of ice. Subtracting the ice destroyed by actual heat leakage, 13.17 t, this yields a net ice production estimate of 39.33 t. Actual ice production from January 9 to April 27, 1979, was 37.40 t. Had the system operated for the full winter, net ice production for 1978-79 would have been approximately 55.62 t.

In the third year, ice production began on November 26, 1979. From that date until April 30, 1980, 43.81 GJ of space heating and water heating was delivered at a COP of 2.56. For this COP, α_H = 0.609, and the estimated heat withdrawal from the bin is 26.68 GJ (79.9 t of ice). About five days of solar panel operation melted 1.58 GJ (4.7 t), and the estimated heat leakage melted 5.66 GJ (16.9 t). This yields an estimated net ice production of 58.3 t (19.44 GJ) compared with the actual production of 58.1 t (19.42 GJ).

Ice production with the original bin insulation level and the second-year heating load would have totaled only 45.15 t (15.05 GJ), and only 48.5 t (16.18 GJ) with the third-year heating load.

Every effort was made in the third year to produce and store as much ice as possible. Design studies using MAD have shown that the ACES house bin should have a 10% larger ice capacity: 57.1 vs 50.9 t. It was attempted to overcome this shortfall by exceeding the design ice capacity of the existing bin. When the ice capacity reached the point where further production would have resulted in spillage, water was removed gradually as more ice was made. The total ice production of

58.1 t resulted in a packing factor of 0.94 and exceeded the estimated ice inventory requirement by 1.0 t.

8.4.2 Spring season

In late April and May of both years, the ACES essentially delivered only water heating loads. During this time, heat withdrawals for water heating almost exactly balanced the heat leakage, with the result that no ice was lost. In the first year, leakage reduced the inventory by some 2.75 t during this period.[8]

In early May 1979, an experiment was run to fill the bin to approximately 105% of design ice capacity. This was done to test heat pump performance and the new coil support structure under excess ice conditions and to replace the ice inventory lost due to the December outage. About 2.17 GJ (7.62 t) of ice was added to the bin inventory.

The coil support structure performed as expected. There were no noticeable effects other than slackness in some of the ropes. This slackness could have been avoided with proper pretensioning during installation.

Mechanical system performance with excess ice is summarized in Table 8.12. As can be seen, the overall performance differs only slightly from the typical performance values given earlier. The average brine temperature was -4.7°C compared with a design value of -5.6°C. This indicates that an adequate factor of safety exists in the bin coil design procedure.

Operation with 10 to 12% excess ice in April 1980 resulted in a 5% performance degradation compared with March 1980. In March the ACES demonstrated a COP of 2.55; this was reduced to 2.43 in April. The drop in performance is illustrated in Fig. 8.9.

8.4.3 Ice-phase cooling season

In the second year the house began experiencing significant cooling loads around May 14, 1979. Until August 24, 1979, all cooling requirements were satisfied using stored ice. Of the 54.39 t (18.17 GJ) of ice in storage on May 14, 14.03 t (4.69 GJ) was melted by heat leakage. However,

Table 8.12. ACES heat pump performance under excess ice conditions[a]

Component power consumption	
Compressor	2213 Wh
Cold-brine pump	144 Wh
Hot-water pump	71 Wh
Energy flows	
Hot water	5078 Wh
Bin heat removal	3707 Wh
COP	
Summer hot water	2.09
Winter hot water[b]	2.53
Refrigerant conditions	
Discharge pressure	1722-1929 kPa
Condensing temperature	46.1-51.7°C
Suction pressure	283-310 kPa
Evaporating temperature	-6.1°C
Brine temperatures	
Bin inlet	-5.50°C
Bin outlet	-3.96°C
Hot-water inlet	27.8°C
Hot-water outlet	48.7°C

[a]Hot-water test 1222-1310 h, May 8, 1979, 53.2 t of ice in bin.
[b]Takes credit for waste heat flow to heated space.

the water heating load of 5.01 GJ deposited an additional 3.01 GJ of ice equivalent into the bin, leaving a net of 16.49 GJ available to meet a cooling load of 13.92 GJ. Delivery energies and mechanical equipment loads consumed the remaining 2.57 GJ. The bin therefore delivered 71% of its total May 14 ice capacity in useful cooling loads. An additional sensible cooling capacity of 1.46 GJ was counteracted by heat leakage, causing night heat rejection to begin immediately upon ice exhaustion.

With the original bin insulation level, natural ice production would have yielded only 45.15 t (15.05 GJ). If all of this ice were available at the start of the cooling season, it could have met the second-year house cooling needs only until early August.

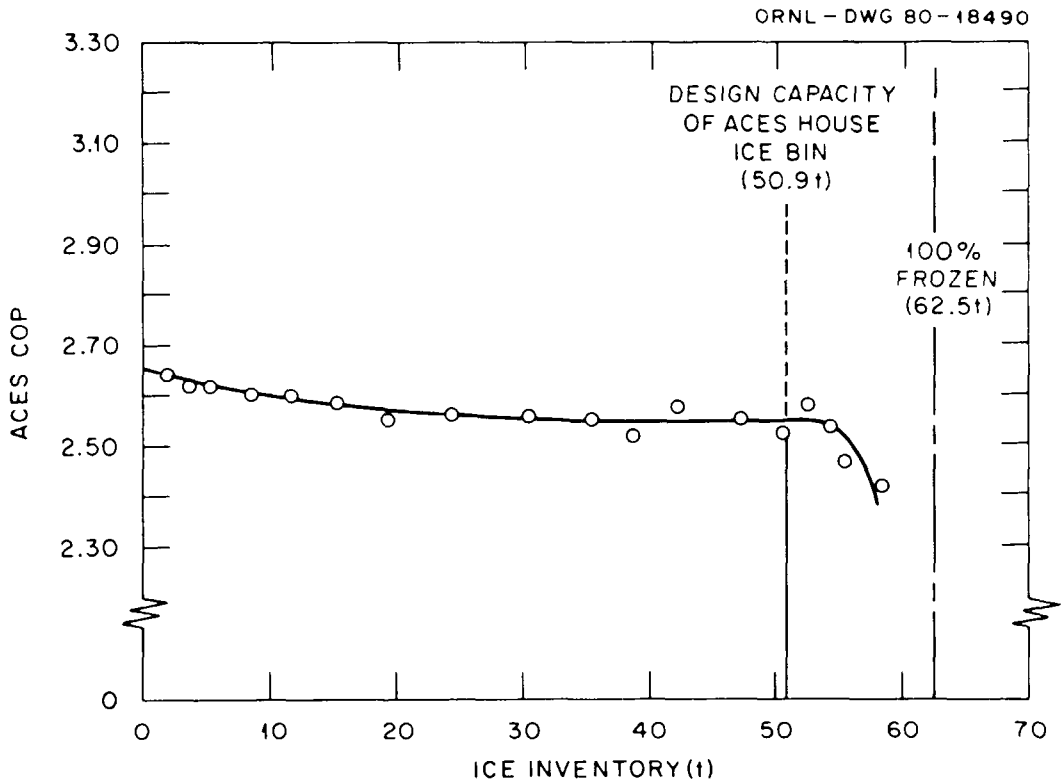

Fig. 8.9. ACES COP vs ice inventory, 1979-80.

In the third year, the house began using stored ice to meet cooling demands on June 2. Sensible and latent storage met all cooling needs until August 14, 1980. Of the 19.42 GJ (58.1 t) of ice in storage, leakage consumed 3.67 GJ. Sensible storage of 1.46 GJ and storage from hot-water production of 1.36 GJ left a net 18.57 GJ in storage to meet cooling needs of 16.64 GJ. Delivery energies and equipment loads consumed 1.93 GJ. Had the bin had only the original insulation, stored cooling would have been exhausted in mid-July.

The effect of the added bin insulation on annual performance is shown in Table 8.13. With the extra insulation, the bin interseasonally transferred 84% of its total cooling capacity in the second year and 89% in the third year, in contrast to projected transfers of 69 and 62% respectively, with the original construction.

The ice-phase cooling season could be extended by effective use of an economy cooling cycle (outside air ventilation). Original calculations

Table 8.13. Bin interseasonal energy transfer summary for new and original insulation levels

	Second-year cooling loads and ice production		Design bin capacities	Third-year cooling loads and ice production	
	Original insulation[a]	Modified bin		Original insulation[a]	Modified bin[b]
Latent cooling storage (GJ)	15.05	18.17	17.40	16.18	19.42
Sensible cooling storage (GJ)	1.46	1.46	2.11	1.46	1.46
Total storage (GJ)	16.51	19.63	19.51	17.64	20.88
Cooling storage from hot-water production (GJ)	2.33	3.01		2.08	1.36
Heat leakage (GJ)	7.44	6.15		7.07	3.67
Net energy available for cooling (GJ)	11.40	16.49		10.97	18.57
% design capacity	58	85		56	95
% total storage	69	84		62	89
Mechanical equipment loads (GJ)	1.56	2.02		0.98	1.59
Delivery losses (GJ)	0.43	0.55		0.18	0.34
Useful cooling delivered (GJ)	9.41	13.92		9.81	16.64
% design capacity	48	71		50	86
% total storage	57	71		56	80
Ice runout date	8/1	8/24		7/16	8/14
Total ACES annual power consumption (kWh)	7402	6719		7306	6447
ACOP	2.55	2.80		2.73	3.08

[a]Estimates.

[b]Does not include May 1980 data (no mechanical cooling was done in May).

estimated that an economy cycle could reduce the annual cooling load on the storage bin by as much as 7.39 GJ, or approximately 25% of the total load.[5] Such a reduction would have delayed the ice runout date by several weeks for both test years and would possibly have eliminated the supplemental phase entirely, with a significant increase in ACOP.

An attempt was made to simulate the effect of an economy cycle during the 1980 cooling season. In May, the windows were left open in both houses, shedding an estimated 1.70 GJ of mechanical cooling requirements. Also, the house thermostats were set up from 25.6°C to 27.2°C in August and September, reducing cooling requirements by an estimated 1.80 GJ. These measures effectively extended the ice runout date by about two weeks and raised the ACES ACOP. The fact that the 1980 summer was much hotter than normal prevented the ice from lasting through late August, as predicted for normal weather conditions.

In the summer of 1978, an automatic economy cycle delivered only about 1% of the load.[8] The control system used, however, was not anticipatory and therefore did not take proper advantage of the cooling potential of the outdoor air. With a better control scheme, such an economy cycle could deliver a significant portion of the house cooling needs. It is recommended that either an automatic or manual economy cycle be used with the ACES.

8.4.4 Supplemental cooling season

When in the night heat rejection mode, heat leakage into the bin must be rejected along with the house cooling loads. The first year's high leakage rates forced increased energy consumption. Maintaining higher bin temperatures reduced leakage but also reduced cooling capacity and humidity control. Adding insulation to the bin reduced heat leakage rates during this period by an average of 34% from the first-year rates. Control logic changes made to limit the hours of night heat rejection operation combined with the lower leakage rate reduced energy consumption to 3% less than that required for the control house heat pump/I^2R water heat system during the second year. In addition, bin temperatures were maintained 1.1 to 2.8°C lower than in the first year, giving slightly better humidity control and cooling capacity.

8.5 Solar/Convector Panel Performance

8.5.1 Winter heat collection

Although ice melt mode operation has not been required during any of the three test years, the panel was operated for a six-day period in December 1979 to determine its performance in this mode. Experiments conducted on a test panel at ORNL[1] indicated that daily heat collection could be theoretically described by the equation

$$Q/A = F_f[\alpha I + 2U(T_a - T_b) - R] ,$$

where

Q/A = heat collection per unit panel area (W/m^2, $Btu/h \cdot ft^2$),
F_f = fin efficiency factor,
α = panel surface absorptivity for solar radiation,
I = daytime average solar radiation intensity (W/m^2, $Btu/h \cdot ft^2$),
T_a = outdoor air temperature (°C, °F),
T_b = entering brine temperature (°C, °F),
R = long-wave radiative heat loss (W/m^2, ($Btu/h \cdot ft^2$),
U = convection coefficient, ($W/m^2 \cdot °C$, $Btu/h \cdot ft^2 \cdot °F$) .

Analysis of the data collected on the test panel shows that the fin efficiency factor and the convection coefficient are well represented by the correlations

$$F_f = 0.941 - 0.028U ,$$
$$U = 0.912 + 0.409S ,$$

where U has the units of $Btu/h \cdot ft^2 \cdot °F$, and S is the wind speed in miles per hour.[1] The long-wave radiative heat loss, R, was estimated to be 47.3 W/m^2 (15 $Btu/h \cdot ft^2 \cdot °F$).

Table 8.14 shows the comparison between the actual and calculated panel heat collection tabulated in order of increasing solar energy availability. Since the panel at the demonstration house was painted a

Table 8.14. ACES radiant/convector panel heat collection performance, December 1979

Total daily insolation[a] (kWh)	Average wind speed (km/h)	Ambient air temperature (°C)	Entering brine temperature (°C)	Fin efficiency factor	Daily heat collection	
					Calculated (kWh)	Observed (kWh)
31.3	14.93	9.0	1.6	0.809	58.4	39.4
53.8	4.19	11.0	3.7	0.886	50.6	46.9
69.6	8.93	9.7	2.6	0.852	46.0	43.6
88.9	3.41	8.9	2.0	0.891	77.1	84.3
94.2	6.00	7.9	2.0	0.873	73.0	74.8
97.4	8.53	7.0	2.6	0.855	73.5	78.5
Total					378.6	367.5
Daily average					63.1	61.2

[a]Total solar energy available on surface inclined 30° from vertical.

grayish brown to match the house exterior, its absorptivity was estimated to be 0.73.[14] Calculated daily heat collection levels were within ±8.5% of the observed values, except for the day with high wind speed and low insolation. The calculated performance was 48% higher than the observed performance for that day. Reference 1 estimates a daily average collection of 61.5 kWh for December, using a panel having a surface absorptivity of 0.93. This indicates that the prediction method is slightly conservative.

The test results show that the panel operated effectively, collecting 367.5 kWh in six days of operation compared with total December 1979 space and water heating loads of 2601 kWh. Further, they indicate that the theoretical prediction method approximates actual operation very well; however, to fully verify the theory, the panel should be operated for an entire winter month.

8.5.2 Heat rejection

Night heat rejection performance has been adequate for each of the test years. The panel has continued to demonstrate a heat transfer coefficient of 7.1 to 8.0 $W/m^2 \cdot K$ in near calm conditions compared with a handbook guess of 11.4 $W/m^2 \cdot K$. For vertical-finned surfaces in still air, the calculated value is 8.28 $W/m^2 \cdot K$. Table 8.15 shows the average night heat rejection performance of the panel for 1979.

Table 8.15. Panel night heat rejection performance

Wind speed (km/h)	Temperature (°C)		Heat rejection rate (kW)	Heat transfer coefficient ($W/m^2 \cdot K$)
	Outdoor air	Average brine		
2.6	20.0	42.0	6.1	7.1
2.6	14.0	39.0	7.6	8.0
5.1	15.0	39.0	7.6	8.4
9.9	15.0	37.0	7.5	8.9
13.0	20.0	34.0	7.5	13.5
24.4	20.0	31.0	7.3	17.8

9. EFFECTS OF MODIFICATIONS

As a result of the first year's operation, several modifications, described in Chap. 6, were made to the ACES prior to the start of the second test year. These modifications were made to correct the excessive bin heat leakage and the deficiencies in the mechanical package and control system. In addition to the ACES modifications, the control house heat pump was modified in May 1979 to enable accurate measurement of the control house cooling load. The effects of the ACES modifications on its performance and the effectiveness of the heat pump modifications on the measurement of the control house cooling loads are discussed in the following sections.

9.1 Mechanical Package

Modifications were made to the mechanical package in order to correct three major deficiencies:

1. insufficient heating capacity,
2. excessive water heating during periods of high heating loads,
3. excessive power consumption during night heat rejection.

To correct these deficiencies, a larger-capacity compressor was added, and control logic changes were made to produce hot water only when needed and to reduce the hours of night heat rejection operation.

Both the larger compressor and the hot-water control logic alteration had desirable effects on the heating capacity, but the control change was the most effective. Table 9.1 shows full-load capacities for the package before and after the modifications. As can be seen, the effect of the larger compressor in increasing capacity was minimal due primarily to the reduced heat transfer area in the storage bin. Total capacity in the space heating and water heating combined mode was increased by only 0.3 kW and heating capacity by only 0.1 kW. Altering the hot-water control logic and allowing the package to devote its full output to space heating had a much more significant impact, raising the heating capacity by 1.54 kW or 23%.

Table 9.1. ACES system capacity before and after modifications

1977-78 season	Heating and hot-water mode (ice source): Total capacity = 8.3 kW Heating capacity = 6.7 kW Hot-water capacity = 1.6 kW COP = 2.70
1978-79 season	Heating and hot-water mode (ice source): Total capacity = 8.6 kW Heating capacity = 6.8 kW Hot-water capacity = 1.8 kW COP = 2.59 Heat only mode (ice source): Heating capacity = 8.24 kW COP = 2.50

Table 9.2 illustrates actual delivered hot-water loads in the ACES and control houses for the past two test years. Control logic changes were successful in preventing excessive water heating energy deliveries in high-heating-load months.

Table 9.2. Hot-water energy delivered to the ACES and control houses, 1977-80

Month	Design loads[a] (GJ)	Delivered water heating loads (GJ)					
		1977-78		1978-79		1979-80	
		Control	ACES	Control	ACES	Control	ACES
November	1.50	1.57	1.55			1.21	1.27
December	1.72	1.76	2.82	1.65	1.75	1.62	1.66
January	1.80	1.94	2.46	1.80	1.89	1.64	1.63
February	1.63	1.70	1.92	1.60	1.70	1.59	1.67
March	1.69	1.75	1.79	1.56	1.71	1.69	1.71
April	1.48	1.47	1.52	1.42	1.53	1.56	1.52
May	1.33	1.42	1.61	1.36	1.57	1.56	1.57
June	1.13	1.16	1.29	1.16	1.35	1.24	1.15
July	1.07	1.16	1.15	1.01	1.11	1.10	1.10
August	1.08	1.07	3.01	1.09	1.41	1.02	1.49
September	1.12	0.61	1.76	1.09	1.79	0.51	0.82
October	1.36			1.33	1.22		
Total	16.96	15.61	20.88	15.07	17.03	14.74	15.59

Night heat rejection control logic changes made in the first year included raising the bin temperature set point.[8] A 24-h timer was installed in the control circuit in August 1979, making it possible to physically limit the hours of operation in this mode. This capability enabled the ACES power consumption in the supplemental cooling season to be reduced to about the same as that required by a central air conditioner and electric water heater.

9.2 Ice Storage Bin

The only major deficiency in the ice bin performance during the first year was the magnitude of the heat leak from the ground into the bin. Originally the ice bin floor was uninsulated. Polystyrene foam insulation was added to the bin floor and walls to correct the situation. In addition, 196.9 mm of the polystyrene foam was added to the bin ceiling to replace 304.8 mm of fiberglass batting that had become water-soaked during the first year.[8]

The added insulation reduced bin heat leakage rates by an average of about 40% from first-year levels. As shown in Table 8.13, by imposing the original heat leakage rates on the second-year loads, it is estimated that the ACES would have yielded an ACOP of only 2.55 instead of the 2.80 demonstrated. It is therefore imperative that the bin be heavily insulated for maximum energy conservation.

Another modification made to the ice storage bin involved the in-bin heat exchanger. The original coil was replaced by a shorter coil having about 45% as much heat transfer surface area. In addition, a simpler coil support system was installed. These modifications are illustrated in Figs. 6.6 and 6.7.

The shorter coil, together with the larger compressor, contributed to a reduction in steady-state and seasonal heating efficiency. However, it did demonstrate the feasibility of operation with shorter, more cost-optimized coils.

The new coil support system, illustrated schematically in Fig. 9.1, worked well throughout the ice production phase and the ice cooling phase

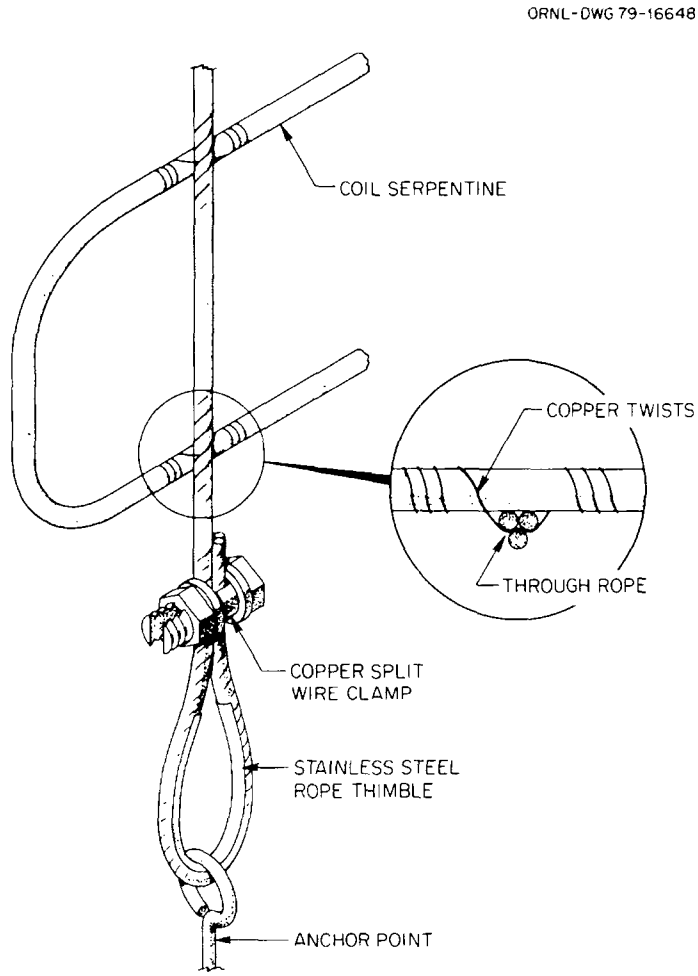

Fig. 9.1. Detail of coil support structure for ice bin heat exchanger. Taken from Ref. 1.

until about the last three weeks of the second-year ice cooling phase. At that time the ice melted away from the coils and support ropes and floated to the surface. As shown in Fig. 9.2, the ice mass was restrained only by the brine supply and return headers. There was some concern that pressure from the ice might rupture one or both headers.

To prevent a reoccurrence of this phenomenon, about 43 m of 25.4-mm-OD, sched-80 PVC pipe was attached to the coil support structure, as illustrated in Fig. 9.3, to provide support for the ice when it melts free of the coils. This additional support prevented the ice mass from floating up during the third-year cooling season.

ORNL PHOTO 4362-80

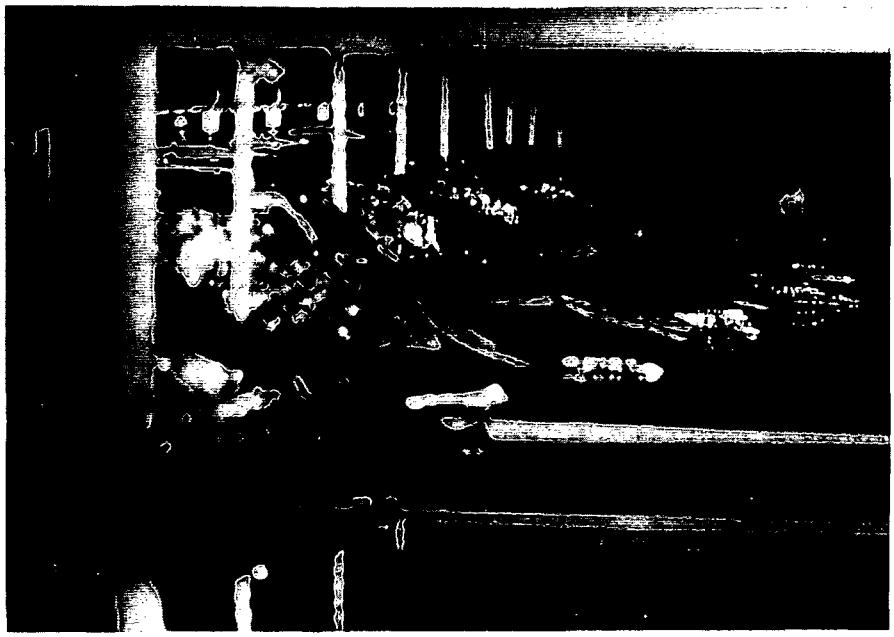

Fig. 9.2. Ice mass in ACES bin after melting free of coils.

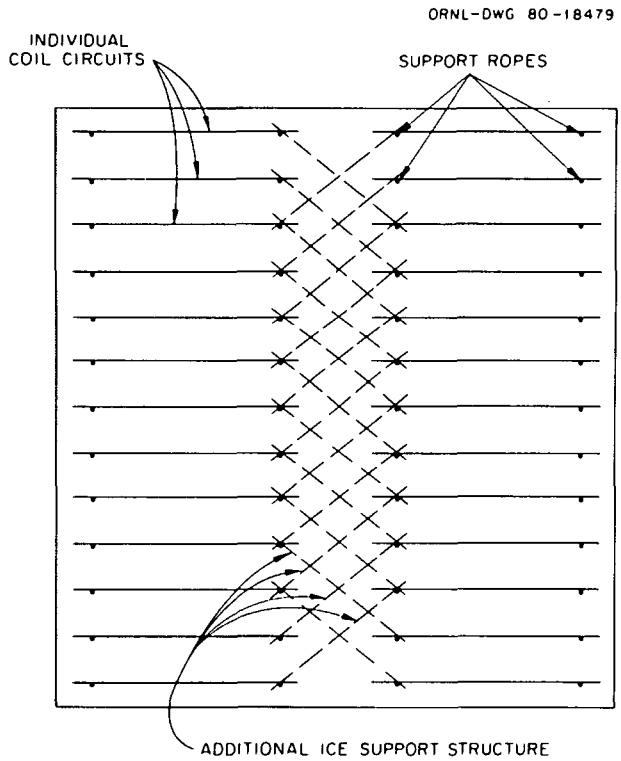

Fig. 9.3. Plan view of ACES house ice bin, showing added supports attached to coils to prevent ice float-up.

9.3 Control House Cooling Load Measurement

In May 1979 the control house heat pump was modified and instrumented for cooling-load measurement. This enabled the comparison of the control house and ACES house cooling season thermal envelope performance.

Table 9.3 gives the measured cooling loads for the 1979 cooling season in both houses. During the months of June and July, the measured cooling loads on the two houses were within ±2% of each other, well within the experimental error expected. Agreement was not as good in August and September, however. During August the control house load was 15% greater than that of the ACES house. Use of the control house during that period was increased due to the presence of construction crews on the TECH site. This increased use, primarily as a rest stop, caused increased occupancy and infiltration loads.

Table 9.3. ACES and control house 1979 cooling loads

	Cooling load (GJ)	
Month	ACES	Control
May	0.58	0.58
June	4.24	4.15
July	4.47	4.33
August	6.02	6.85
September	3.86	3.28
Total	19.17	19.18

In September the ACES load was about 15% greater than the control house load. This was due primarily to an imbalance in the internal loads during the month. The fact that the ACES operated in the night heat rejection mode all month is felt to have had little effect because the mechanical room was isolated and vented to the outside.

The total seasonal cooling loads were nearly equal for the two houses. Although there were wide differences in the loads in two months, these discrepancies can be traced to causes other than differences in thermal envelope performance. Thus the assumed equivalence of the ACES

and control thermal envelopes in the cooling season has now been experimentally verified.

The ratio of the latent to total cooling loads was nearly equivalent for both houses as well. Table 9.4 shows the monthly condensate collections and the ratios of latent load to total load for both houses.

Table 9.4. ACES and control house latent cooling performance, 1979

	Condensate collection (kg)		Latent-to-total cooling ratio	
Month	ACES	Control	ACES	Control
June[a]	210.9	231.2	0.20	0.24
July	512.9	411.9	0.28	0.23
August[b]	496.6	455.1	0.26	0.21
Total	1220.4	1098.2	0.25	0.22

[a]Condensate collected from June 12, 1979.

[b]Measured to Aug. 24, 1979.

10. CONCLUSIONS AND RECOMMENDATIONS

Several major conclusions can be drawn from the three years of ACES demonstration house operation. They are listed here.

1. The ACES has significant energy conservation potential. Power consumption was reduced by 56% over that for an identical house with an electric heat, central air-conditioner, electric (I^2R) hot-water system; by 48% over an ARI 2.46 at 8.3°C air-to-air heat pump/I^2R hot-water system; and by 43% over an ARI 3.11 at 8.3°C air-to-air heat pump/I^2R hot-water system. Table 10.1 lists the ACOPs and power consumptions of the ACES and comparative systems for the three test years, along with the predicted values.

Table 10.1. ACOP and power consumption, 1977-80

	Year and system	ACOP	Power consumption (kWh)
1977-78:	ACES	2.78	9,012
	I^2R Heat and water heat, central air-conditioner	1.13	20,523
1978-79:	ACES	2.80	6,719
	ARI 2.46 Heat pump, I^2R Water heat	1.42	12,853
1979-80:	ACES	3.08	6,447
	ARI 3.11 Heat pump, I^2R Water heat	1.73	11,358
Calculated:[a]	ACES	3.18	6,437
	ARI 3.11 Heat pump	1.67	11,283
	ARI 2.46 Heat pump	1.38	14,820
	I^2R Heat system	1.15	18,238

[a]Values for a full calendar year calculated by the MAD (ref. 2) program based on long-term weather averages.

2. Peak hourly consumption reductions of up to 73% were demonstrated in the winter over all three comparative systems. In addition, summer air-conditioning peak consumption was reduced by up to 85%.

3. Adding insulation to the bin walls, floor, and ceiling reduced heat leakage and improved the interseasonal energy transfer capability by an estimated 47%. Insulation of bins to at least 7.1 $m^2 \cdot K/W$ is recommended in all installations.

4. The ACES demonstrated an ability to perform efficiently with a bin containing up to 95% ice. This indicates that adequate factors of safety exist in designs employing 80% packing factors.

5. Measurement of the control house cooling loads during 1979 demonstrated that the ACES and control house cooling loads were within ±3% of each other. Heating loads for the two houses were demonstrated to be within ±2% of one another during the 1977-78 season.

6. Changes in the ACES hot-water control logic eliminated excessive water heating in times of high heating loads and boosted the system space heating capacity.

7. Reducing heat leakage and changing night heat rejection control logic to limit the hours of operation in that mode combined to reduce supplemental cooling season power consumption to about the same level as that of a central air-conditioner/I^2R water heating system.

The ACES has proven itself capable of accommodating varying weather conditions with no effect on its efficiency. In addition, it has worked reliably in all three test years. The only significant equipment outage occurred in December 1978, when an improperly wired thermostat admitted AC feedback into the DC control system, causing repeated pump failures. This problem was corrected, and no further control-system-related outages

occurred. In the latter part of the 1979-80 test year, the compressor was shut off by the high-pressure cutoff switch, when trying to heat water. This indicated fouling of the heat exchangers caused by scale buildup. Regular chemical cleaning (using a food-grade ice machine cleaner for instance) of the water heating system will eliminate this problem. Such action is recommended for the ACES mechanical package before the next test period.

The ACES control system is simple to operate. All a homeowner need worry about is where he sets the heating and cooling thermostats and possibly the water heating thermostat. Other control devices (bin thermostat, outdoor thermostat, ice-level control, night heat rejection timer, etc.) could be preset when installed. Periodic inspection, say during an annual routine service call, would be all the attention normally required by technical personnel.

The TECH site is a very good comparative test bed for evaluation of residential systems. It is strongly recommended that the facilities at the TECH site continue to be used for evaluation of the annual performance of advanced heating and cooling systems for residences.

Operation of the demonstration ACES has verified that the ACES design criteria, detailed in the design manual, work adequately and that it has met the theoretical performance predictions outlined in ref. 2 except that the water heating COP was somewhat lower than expected due to fouling of the heat exchangers. This verification has enabled meaningful economic comparisons to be made between the ACES and competitive heating, water heating, and air-conditioning systems.[9,10] Findings of the study reported in ref. 10 indicate that the ACES is break-even in life-cycle costs over a 20-year study period in most locations in the United States. This is a particularly significant conclusion in light of the fact that the ACES is the best residential heating and cooling system for conserving electrical energy yet studied.

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Appendix A

DATA POINTS

Table A.1 is a list of the data points measured by the expanded data acquisition system (DAS), along with those points calculated from the measured values. Since the expansion and upgrading modifications of October and November 1978, the DAS has demonstrated a 99.5% uptime during the test period, running 24 h a day, 7 days a week.

Table A.1. Data system input listing

Data number	Type	Description	TECH site location
1	Integrated	BINo (Heat pump-to-bin heat flow)	ACES house
2	Integrated	BINi (Bin-to-heat pump heat flow)	ACES house
3	Integrated	SLPo (Solar panel heat rejection)	ACES house
4	Integrated	SLPi (Solar panel heat collection)	ACES house
5	Integrated	Pclg WHM Kh=3.6(Cooling pump)	ACES house
6	Integrated	Phtg WHM Kh=3.6(Heating pump)	ACES house
7	Integrated	Phwt WHM Kh=3.6(Hot-water pump)	ACES house
8	Integrated	COMP WHM Kh=3.6(Compressor)	ACES house
9	Integrated	BLWR WHM Kh=3.6(Blower)	ACES house
10	Integrated	ACES WHM Kh=7.2(ACES total)	ACES house
11	Integrated	FANo (Heat flow out of fan coil)	ACES house
12	Integrated	FANi (Heat flow into fan coil)	ACES house
13	Integrated	HWTo (Heat flow into hot water)	ACES house
14	Integrated	ECSd (Econ. cycle operating time, DH)	ACES house
15		Digital output	
16	Integrated	ECSh (Econ. cycle operating time, H_{27})	ACES house
17		Spare	
18		Spare	
19	Integrated	SAMP WHM Kh=3.6(Air handler)	Garage
20	Integrated	HPID WHM Kh=3.6(Heat pump indoor)	Garage
21	Integrated	HPOD WHM Kh=3.6(Heat pump outdoor)	Garage
22	Integrated	OFFP WHM Kh=12(Off-peak heater)	Garage
23	Integrated	HPAX WHM Kh=12(Heat pump aux. ht.)	Garage
24	Integrated	SCSP (Solar coll. to space heat flow)	Garage
25	Integrated	HPSP (Heat pump to space heat flow)	Garage
26	Integrated	SCPB (Solar collector to bed heat flow)	Garage
27	Integrated	PBSP (Bed to space heat flow)	Garage
28	Integrated	TPWR WHM Kh=12(Total power)	Garage
29		Spare	
30		Digital input	
31	Integrated	SOLC (Solar collector heat flow)	Solar house
32	Integrated	WTRC (Water-to-refrigerant coil heat flow)	Solar house
33	Integrated	WTAC (Water-to-air coil heat flow)	Solar house
34	Integrated	HWTR (Hot-water heat flow)	Solar house
35	Integrated	SOLR WHM Kh=7.2(Solar total)	Solar house
36	Integrated	BLWR WHM Kh=3(Blower)	Solar house
37	Integrated	SHWT WHM Kh=3(Hot-water heater)	Solar house
38	Integrated	Pscl WHM Kh=3(Collector pump)	Solar house
39	Integrated	DHTR WHM Kh=12(Duct heater)	Solar house
40	Integrated	Phtg WHM Kh=3(Heating pump)	Solar house
41	Integrated	HTPU WHM Kh=3(Heat pump outdoor unit)	Solar house

Table A.1 (continued)

Data number	Type	Description	TECH site location
42	Integrated	RTAC (Refrigerant-to-air coil heat flow)	Solar house
43		Spare	
44		Spare	
45		Not used	
46	Integrated	Qrad (Solar radiation, 45°)	Weather station
47	Integrated	Wspd (Average wind speed)	Weather station
48	Integrated	CHTP WHM Kh=7.2(Control-house total)	Control house
49	Integrated	CHHW WHM Kh=3.6(Hot-water heater)	Control house
50	Integrated	CHIU WHM Kh=12(Indoor unit)	Control house
51	Integrated	CHOU WHM Kh=3.0(Outdoor unit)	Control house
52	Integrated	FANi (Heat flow into fan coil)	Control house
53	Integrated	FANo (Heat flow out of fan coil)	Control house
54		Spare	
55		Spare	
56		Spare	
57		Spare	
58		Spare	
59		Spare	
60		Not used	
61	Analog	Ice level	ACES house
62	Analog	BINl (Bin low temp.)	ACES house
63	Analog	BINh (Bin high temp.)	ACES house
64	Analog	BINi (Bin inlet temp.)	ACES house
65	Analog	BINo (Bin outlet temp.)	ACES house
66	Analog	FANi (Fan coil inlet temp.)	ACES house
67	Analog	FANo (Fan coil outlet temp.)	ACES house
68	Analog	SLPi (S.P. inlet temp.)	ACES house
69	Analog	SLPo (S.P. outlet temp.)	ACES house
70	Analog	HWTi (HW inlet temp.)	ACES house
71	Analog	HWTo (HW outlet temp.)	ACES house
72	Analog	Hdry (House dry bulb temp.)	ACES house
73	Analog	Hwet (House wet bulb temp.)	ACES house
74	Analog	SLPu (Solar panel equilibrium temp.)	ACES house
75		Spare	
76		Spare	
77		Spare	
78		Spare	
79		Spare	
80	Analog	Pebble bed #1 temperature (top)	Garage
81	Analog	Pebble bed #2 temperature	Garage
82	Analog	Pebble bed #3 temperature	Garage
83	Analog	Pebble bed #4 temperature	Garage
84	Analog	Pebble bed #5 temperature	Garage

Table A.1 (continued)

Data number	Type	Description	TECH site location
85	Analog	Pebble bed #6 temperature	Garage
86	Analog	Pebble bed #7 temperature (bottom)	Garage
87	Analog	Space dry bulb	Garage
88	Analog	Space wet bulb	Garage
89		Spare	
90		Spare	
91		Spare	
92		Spare	
93		Spare	
94		Spare	
95		Spare	
96	Analog	Rain (Rainfall)	Weather station
97	Analog	Wdir (Wind direction, 0° ≡ N)	Weather station
98	Analog	Pbar (Barometric pressure)	Weather station
99	Analog	Tdew (Outdoor dew point)	Weather station
100	Analog	Tdry (Outdoor dry bulb)	Weather station
101		Spare	
102	Analog	SLPi (Collector inlet temp.)	Solar house
103	Analog	SLPo (Collector outlet temp.)	Solar house
104		Spare	
105	Analog	SLPu (Solar panel equilibrium temp.)	Solar house
106	Analog	SLPp (Solar panel pumped temp.)	Solar house
107		Spare	
108	Analog	T#2h (Tank #2 high temp.)	Solar house
109	Analog	T#2L (Tank #2 low temp.)	Solar house
110	Analog	T#1h (Tank #1 high temp.)	Solar house
111	Analog	T#1L (Tank #1 low temp.)	Solar house
112	Analog	FANi (Fan coil inlet temp.)	Solar house
113	Analog	FANo (Fan coil outlet temp.)	Solar house
114	Analog	Hdry (House dry bulb temp.)	Solar house
115	Analog	Hwet (House wet bulb temp.)	Solar house
116		Spare	
117		Spare	
118		Spare	
119		Spare	
120		Spare	
121		Spare	
122		Spare	
123		Spare	
124		Spare	
125		Spare	
126		Spare	
127		Spare	
128		Spare	
129		Spare	

Table A.1 (continued)

Data number	Type	Description	TECH site location
130		Spare	
131		Spare	
132		Spare	
133		Spare	
134		Spare	
135	Analog	AIRo (Air temp. leaving indoor unit)	Control house
136	Analog	FANi (Fan coil inlet temp.)	Control house
137	Analog	FANo (Fan coil outlet temp.)	Control house
138	Analog	Hdry (House dry bulb temp.)	Control house
139	Analog	Hwet (House wet bulb temp.)	Control house
140	Analog	CRLS (Crawl space temp.)	Control house
141	Calculated	ACES equipment power consumption	
142	Calculated	Solar radiation, 30° surface	
143	Calculated	Solar radiation, horizontal surface	
144	Calculated	Outdoor air specific volume	
145	Calculated	Economy cycle cooling load	
146		Spare	
147		Spare	
148		Spare	
149		Spare	
150		Spare	
151		Spare	
152		Spare	
153		Spare	
154		Spare	
155		Spare	
156		Spare	
157		Spare	
158	Calculated	Stime (Integration time, min)	
159	Calculated	Month/day	
160	Calculated	Hour/min.	

Appendix B

SAMPLE DATA OUTPUT

Table B.1. Equipment and weather data summary for week beginning Jan. 29, 1979[a]

Average outside dry bulb temperature (°C)	-1.08
Maximum outside dry bulb temperature (°C)	8.75
Minimum outside dry bulb temperature (°C)	-11.90
Degree-days heating referenced to 18 (°C)	132.35
Degree-days heating referenced to 16 (°C)	118.35
Degree-days heating referenced to 14 (°C)	104.35
Degree-hours cooling referenced to 21 (°C)	0.00
Degree-hours cooling referenced to 23 (°C)	0.00
Degree-hours cooling referenced to 25 (°C)	0.00
Total heating load delivered (kWh)	794.47
Total cooling load delivered (kWh)	0.00
Total economy load delivered (kWh)	0.00
Total hot-water load delivered (kWh)	120.31
Maximum heating load delivered (W)	8493.00
Maximum cooling load delivered (W)	0.00
Maximum economy load delivered (W)	0.00
Maximum hot-water load delivered (W)	4815.40
Total ACES power consumption (kWh)	362.63
Total control-house equipment power consumption (kWh)	576.92

[a]Control house equipped with an ARI 2.46 air-to-air heat pump and electric resistance water heating during the 1978/1979 test year.

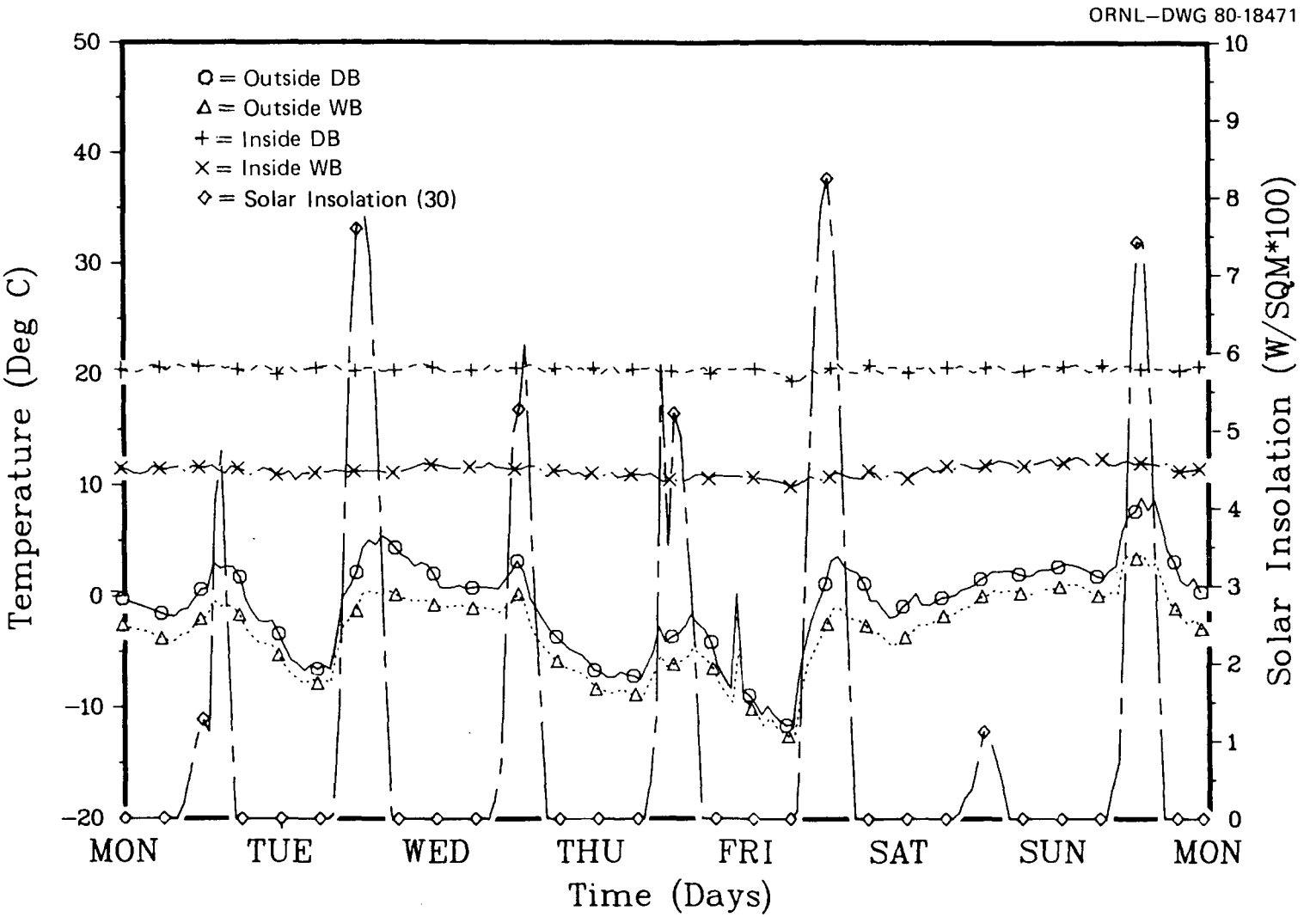

Fig. B.1. ACES house temperature and insolation plot, week beginning Jan. 29, 1979.

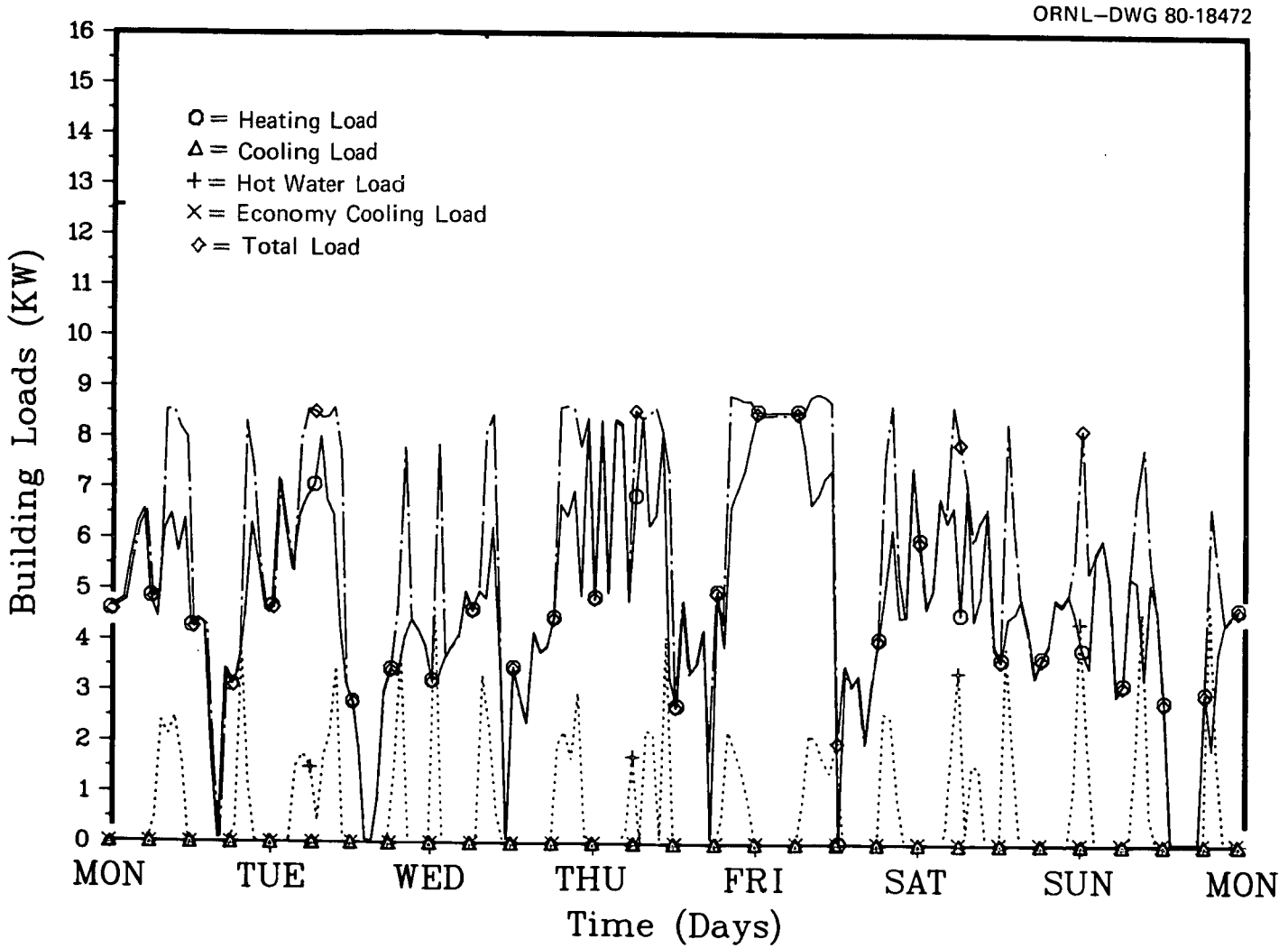

Fig. B.2. ACES building thermal load plot, week beginning Jan. 29, 1979.

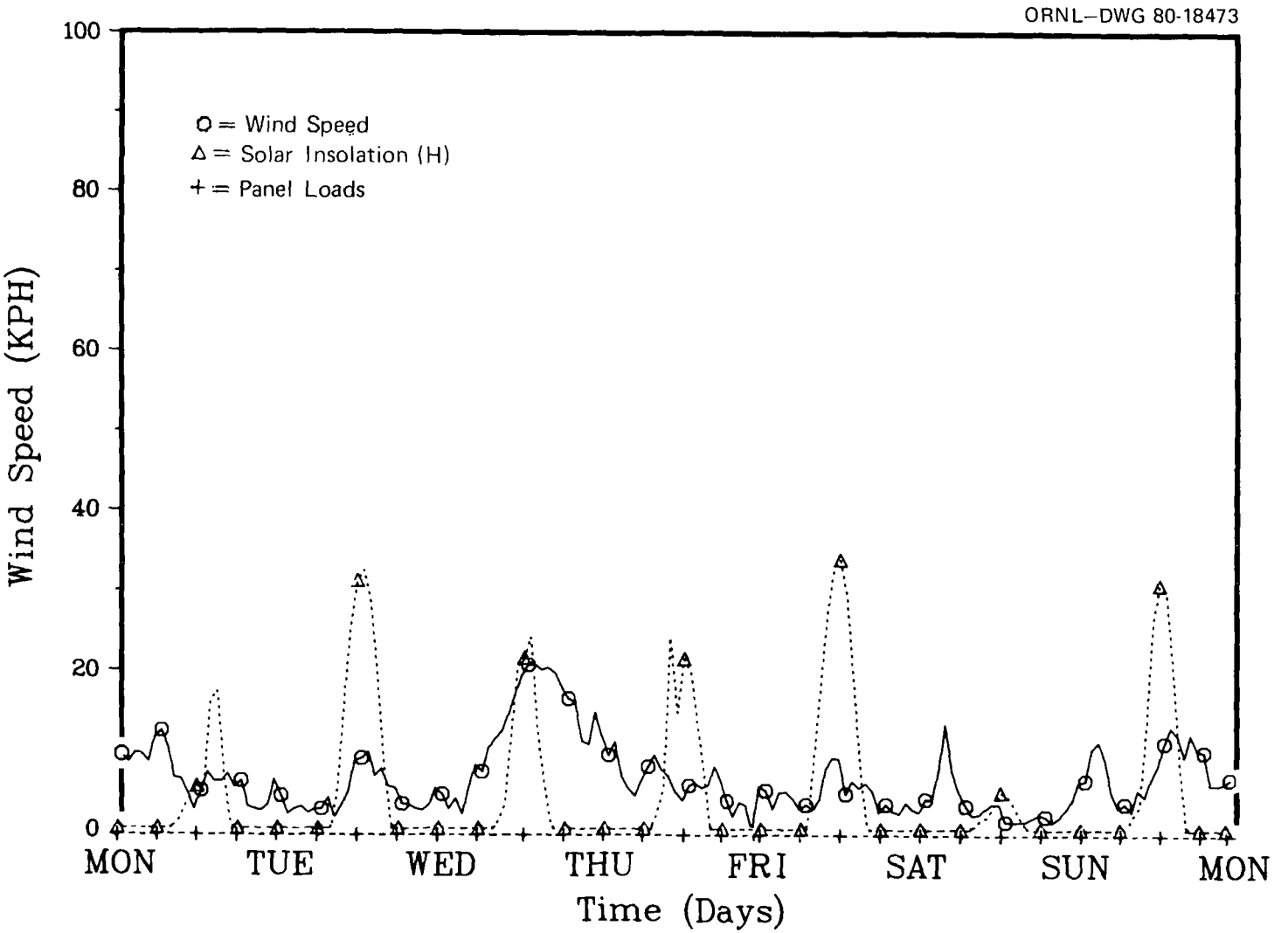

Fig. B.3. Wind speed and solar insolation plot, week beginning Jan. 29, 1979.

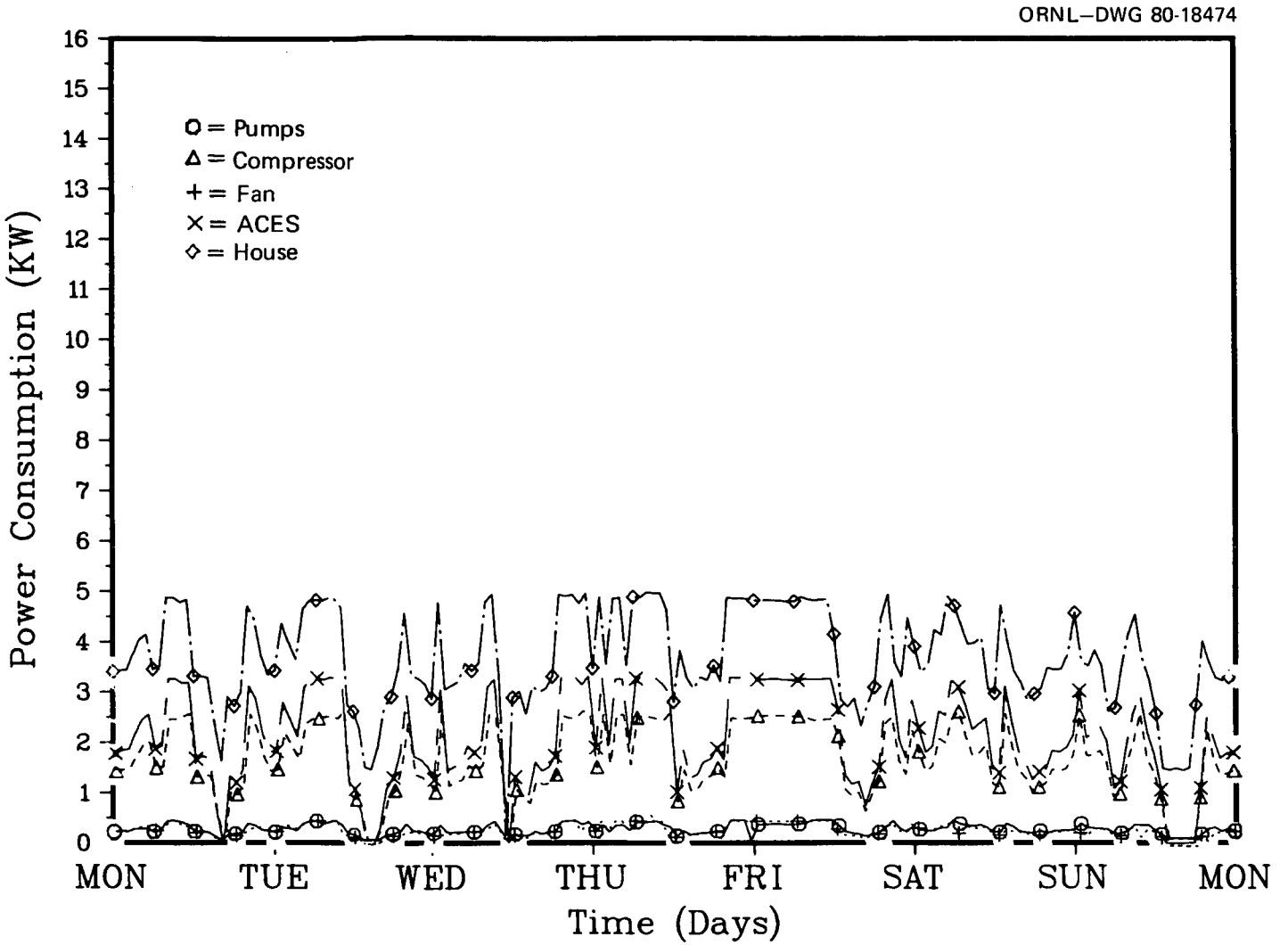

Fig. B.4. Electrical consumption plot for ACES components, week beginning Jan. 29, 1979.

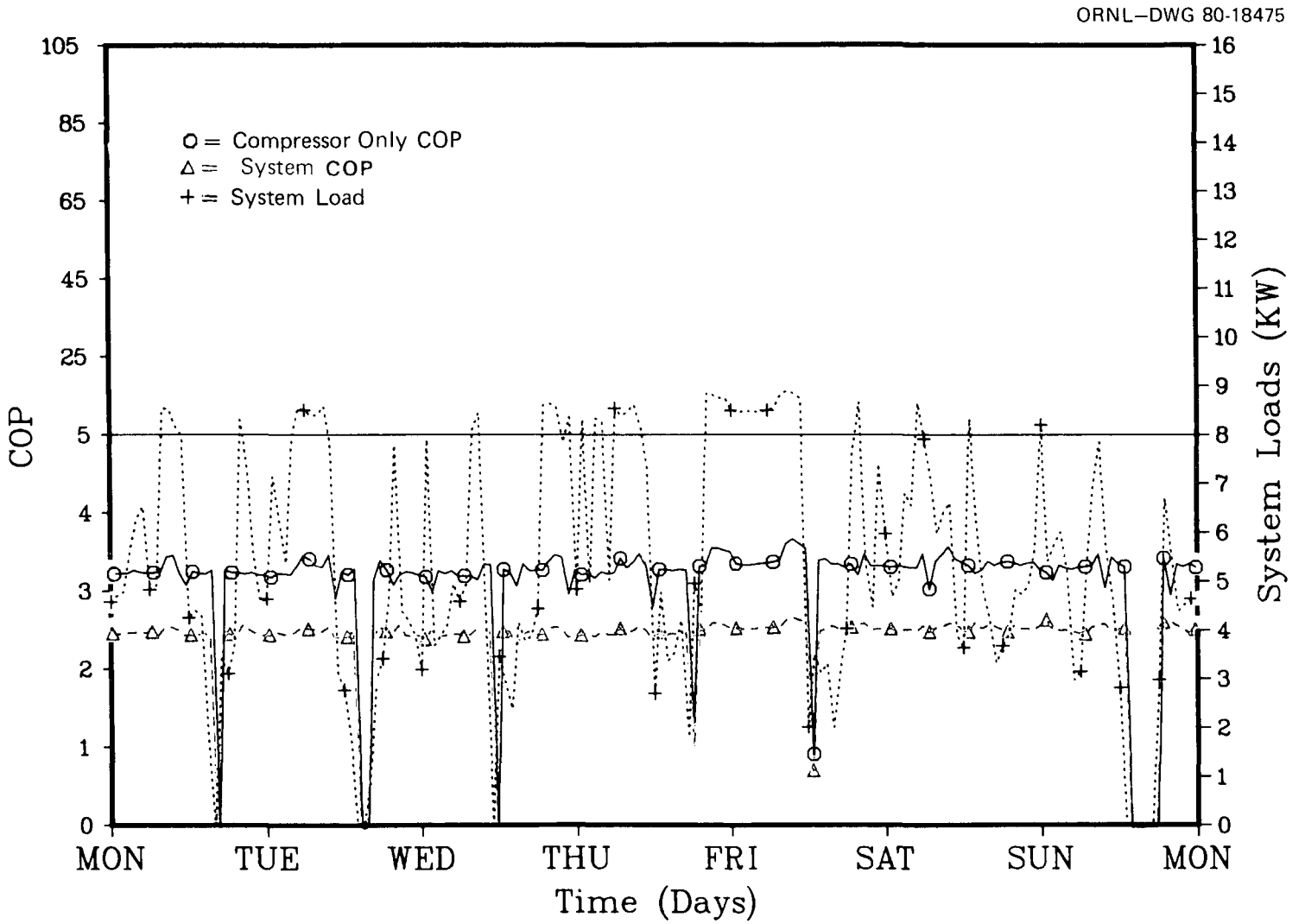

Fig. B.5. ACES coefficients of performance and load plot, week beginning Jan. 29, 1979.

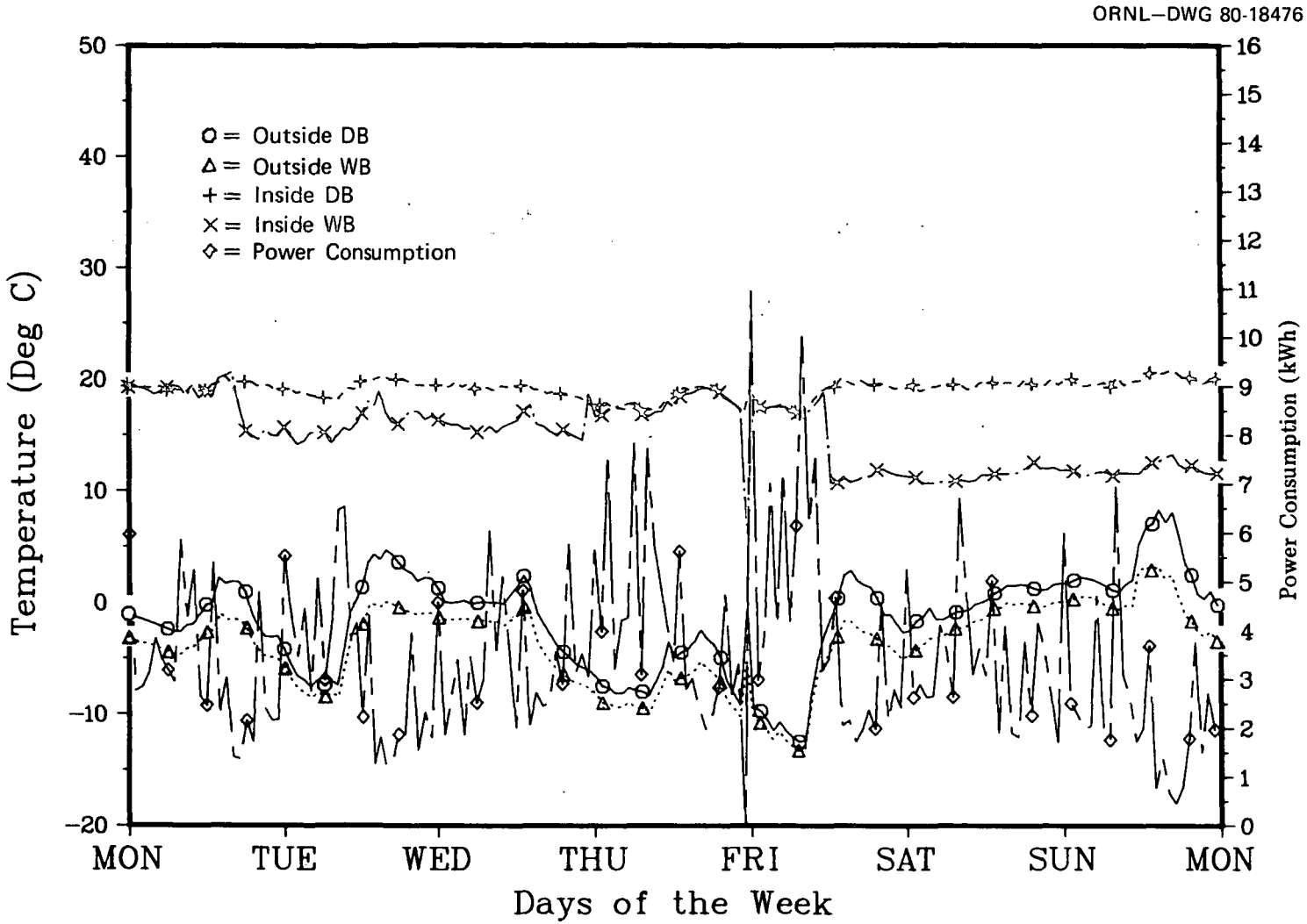

Fig. B.6. Control house weather and power consumption plot, week beginning Jan. 29, 1979. (ARI 2.46 air-to-air heat pump.)

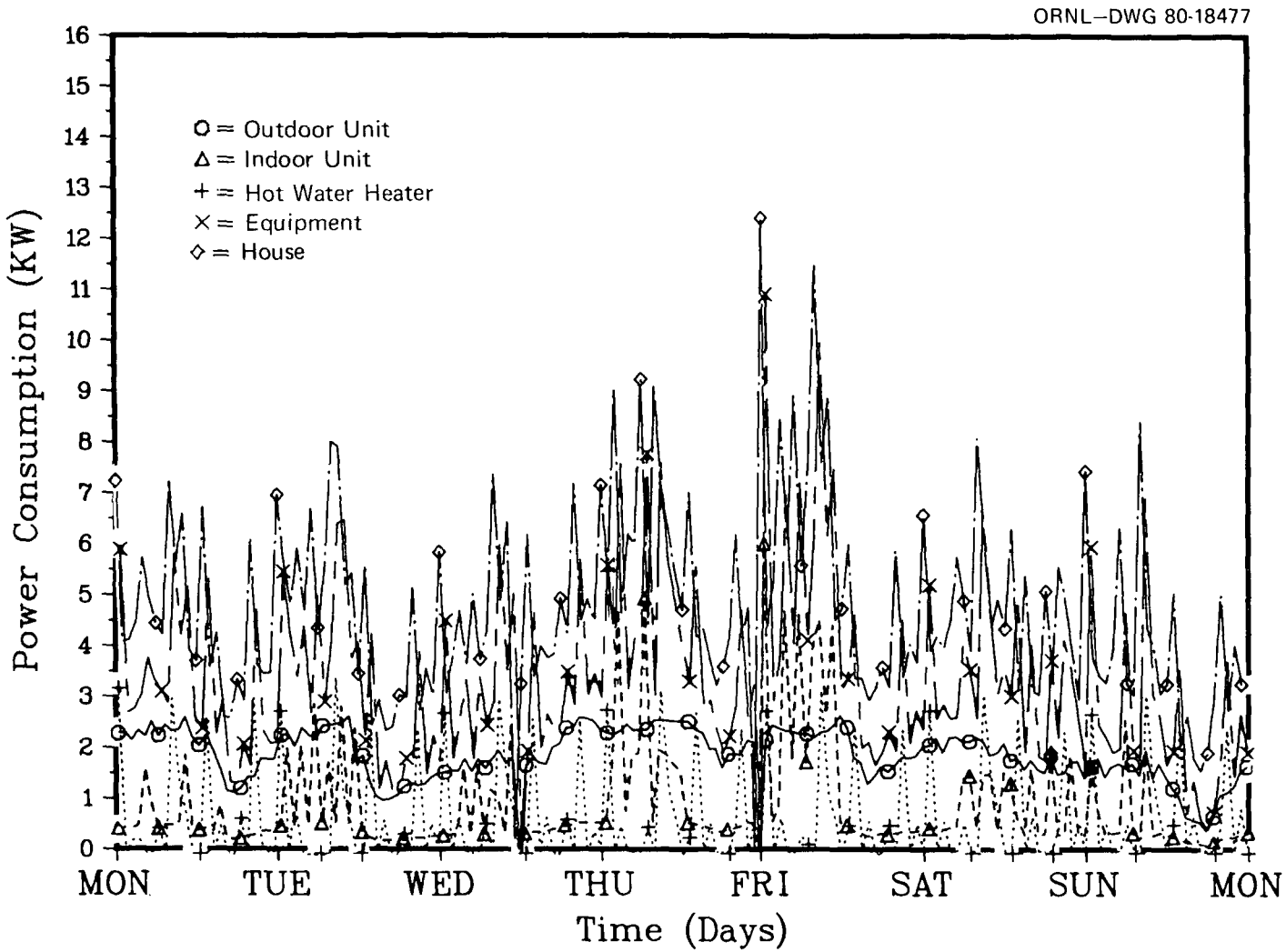

Fig. B.7. Control house component power consumption plot, week beginning Jan. 29, 1979. (ARI 2.46 air-to-air heat pump.)

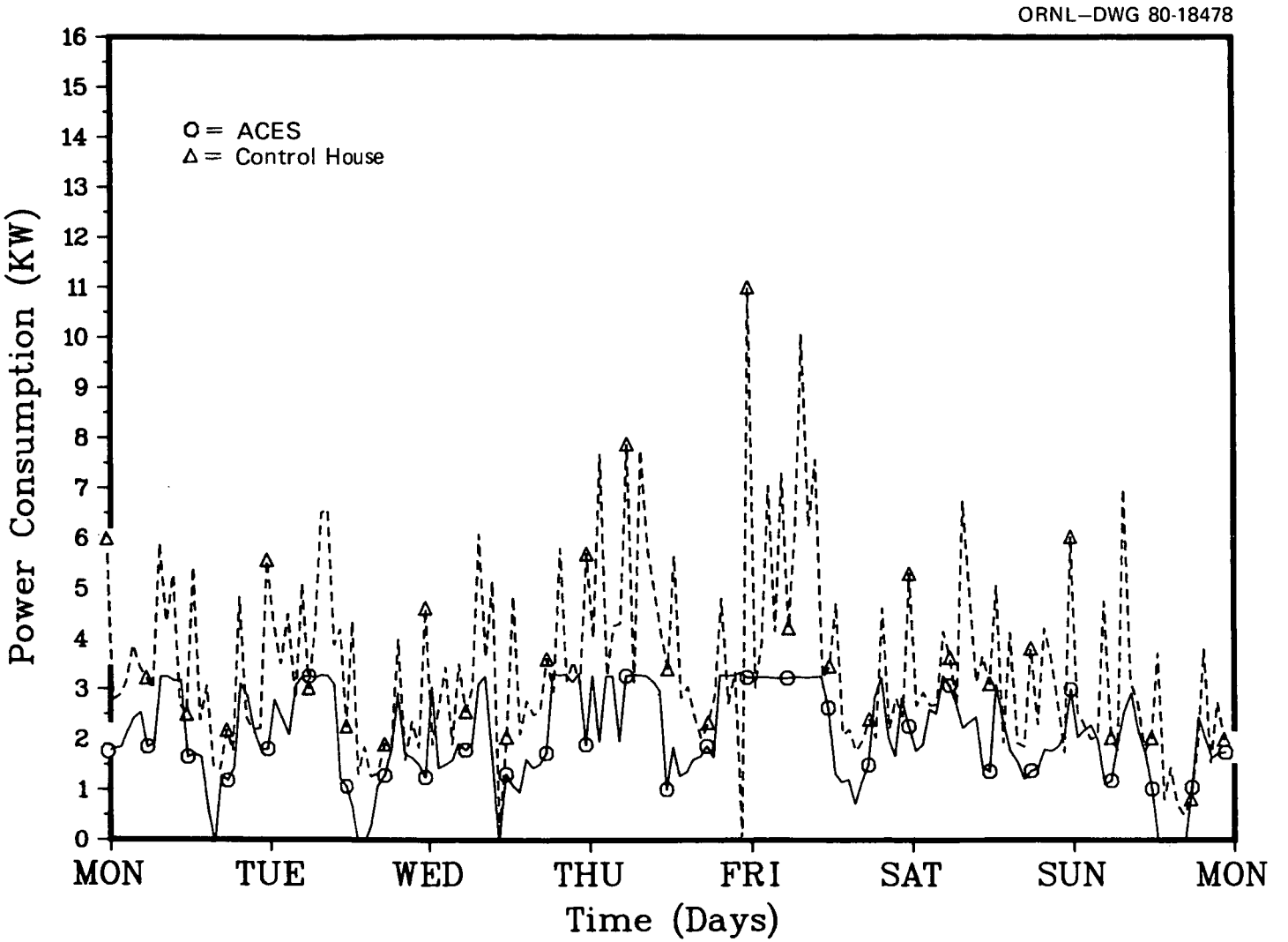

Fig. B.8. Power consumption plot for the ACES and control houses, week beginning Jan. 29, 1979. (Control house ARI 2.46 heat pump.)

Appendix C

DATA ADJUSTMENTS

Occasional disturbances in the data collection due to either complete or partial DAS outages, excess hot-water consumption, leaving breakers off, etc., necessitated adjustment of the collected data to reflect true loads and power consumptions. Almost all of these disturbances occurred in the second test year. The major adjustments are outlined in this appendix. Critical measurements were backed up with alternate data sources as far as was practical. All electrical power consumptions were read manually on a twice-weekly basis.

C.1 December 1978 Heating Load

Owing to control problems with the ACES, the actual heating loads for December and the first week in January were not completely measured. Figure C.1 shows measured monthly house heating load in GJ plotted as a function of monthly degree-days. For the month of December 1978, with 378°C-days, this graph estimates the heating load to be about 7.29 GJ.

For the first week in January, the heating load was estimated by using the actual control house heat pump power consumption and the heating COP it demonstrated during subsequent weeks of similar weather conditions. This adjustment gave a total January heat load estimate of 12.12 GJ, which compares well with the demonstrated house thermal load performance illustrated in Fig. C.1.

C.2 Control House Indoor Fan on Continuously

Air infiltration testing is done at the control house, using sulfur hexaflouride gas as a tracer. It is necessary to have the indoor fan on to distribute the tracer gas evenly throughout the structure. Several times the fan was left running continuously, necessitating an adjustment of the heat pump power consumption to reflect the actual loads. Table C.1 gives a summary of the periods of continuous fan operation and the power consumption adjustment applied to the measured consumption.

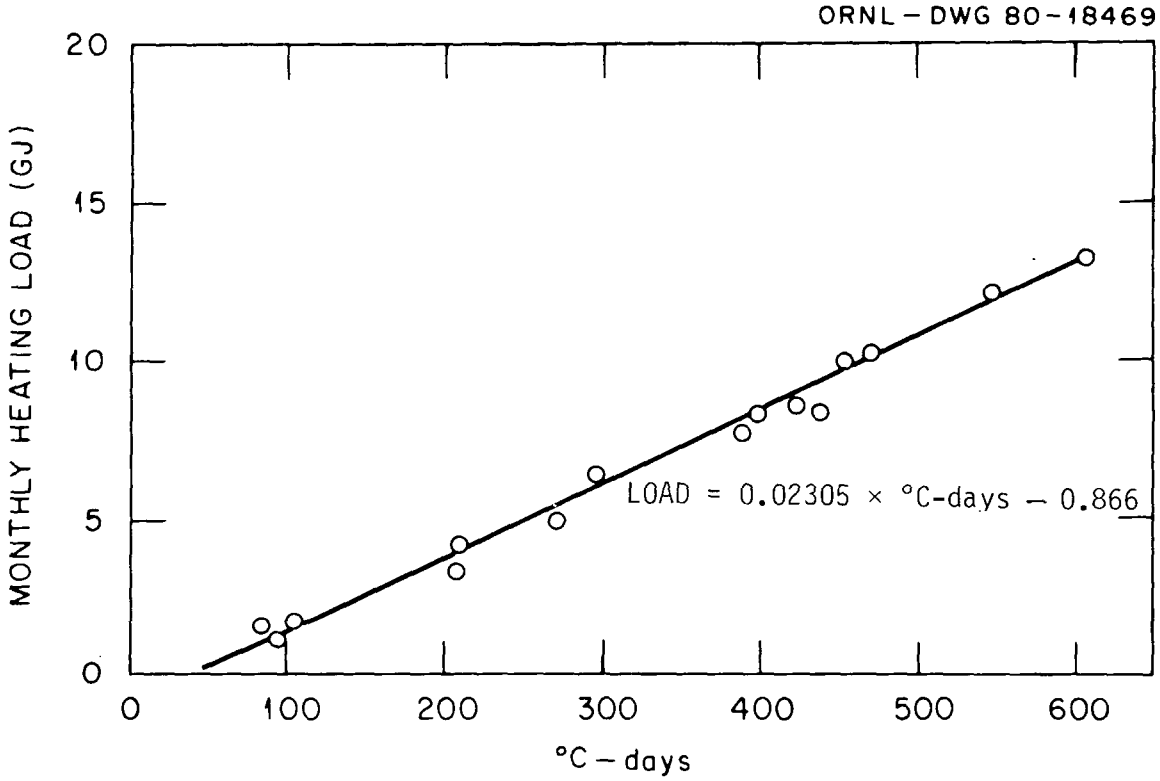

Fig. C.1. Monthly house heating load vs degree-days.

Table C.1. Corrections to control house power consumption for continuous fan operation

Month	Days of continuous operation	Adjustment to power consumption (kWh)
December 1978	11	19.4
January 1979	2	3.5
February 1979	3	4.9

As an example, for the week beginning February 5, 1979, there was a three-day period during which the fan ran continuously, consuming 38.7 kWh. The data indicate that the average ratio of heat pump indoor unit to outdoor unit power was

$$\frac{CHIU}{CHOU} = 0.19 \ .$$

By this ratio the indoor unit should have used only 23.9 kWh during that period. This is a difference of 14.8 kWh. Subtracting this amount from the measured week's power consumption yields

$$598.9 - 14.8 = 584.1 \text{ kWh} \ .$$

However, to correct for the 14.8 kWh of heat delivered by the fan, an amount must be added to the adjusted power consumption to reflect heat pump operation to deliver this load. This is calculated by

$$14.8/COP_H \ ,$$

where COP_H = average heating only COP for February
= 1.5.

Thus the total correction, P', is

$$P' = 14.8 - 14.8/1.5$$
$$= 4.9 \text{ kWh} \ ,$$

and the adjusted power consumption is

$$598.9 - 4.9 = 594.0 \text{ kWh} \ .$$

C.3 Excessive Hot Water Use in Control House

On March 19, 1979, the solenoid valve controlling the measured use of hot water in the control house became stuck in the open position.

This resulted in 161 kWh of excessive water heating by the resistance elements. A similar occurrence on January 1, 1980, caused an excess of 77 kWh in consumption. Power consumption was adjusted by subtracting the excess power consumed from the measured consumption for those weeks.

C.4 Control House Breaker Off

The control house heat pump and water heater were inadvertantly left off for one day from February 26 to February 27, 1979. Approximately 92 kWh was added to the power consumption to reflect the total heating and hot-water loads for that week.

C.5 Calibration Error

From May through mid-July 1979, the ACES brine system heat flows were 2.3% high due to a calibration error. All brine system heat flows for that period were adjusted downward by 2.3%.

C.6 Fan Coil Heat Flow Outage

In late August of both 1979 and 1980, lightning storms resulted in the loss of the FAN_i heat flow (and thus the direct measure of cooling load) at the ACES house. The BIN_0 heat flow and the fan (BLWR) and cooling pump (Pclg) power consumption measurements were unaffected. It was possible, therefore, to determine fairly accurately the FAN_i heat flow for the outage period by

$$FAN_i = \frac{FAN_i}{BIN_0} \times \frac{BIN_0}{BLWR} \times BLWR \ .$$

The ratios FAN_i/BIN_0 and $BIN_0/BLWR$ were determined from observations of the data during periods of no outages.

C.7 UPS Failure and Programming Error

During the week of February 25, 1980, two failures occurred which resulted in complete DAS outages lasting a total of 39 h. First, the uninterruptible power supply (UPS) failed, disrupting power to the DAS for 15 h. The second failure occurred because the DAS was not properly programmed to account for leap years. As a result, data for February 29 were not recorded on the data storage cassette.

The following additions were made to the heating and water heating loads to reflect the total power consumptions for that week:

ACES and control house heating load	0.67 GJ
ACES water heating load	0.10 GJ
Control house water heating load	0.09 GJ

Appendix D

EXPERIMENTAL ERROR ANALYSIS

D.1 Heat Flow Measurement

Figure D.1 shows a typical schematic of the interface circuitry used in the DAS integrator to determine heat flows. Heat flows are experimentally determined by the following expression:

$$HF = K \times [f_{flow} \times f_{\Delta T}] , \qquad (1)$$

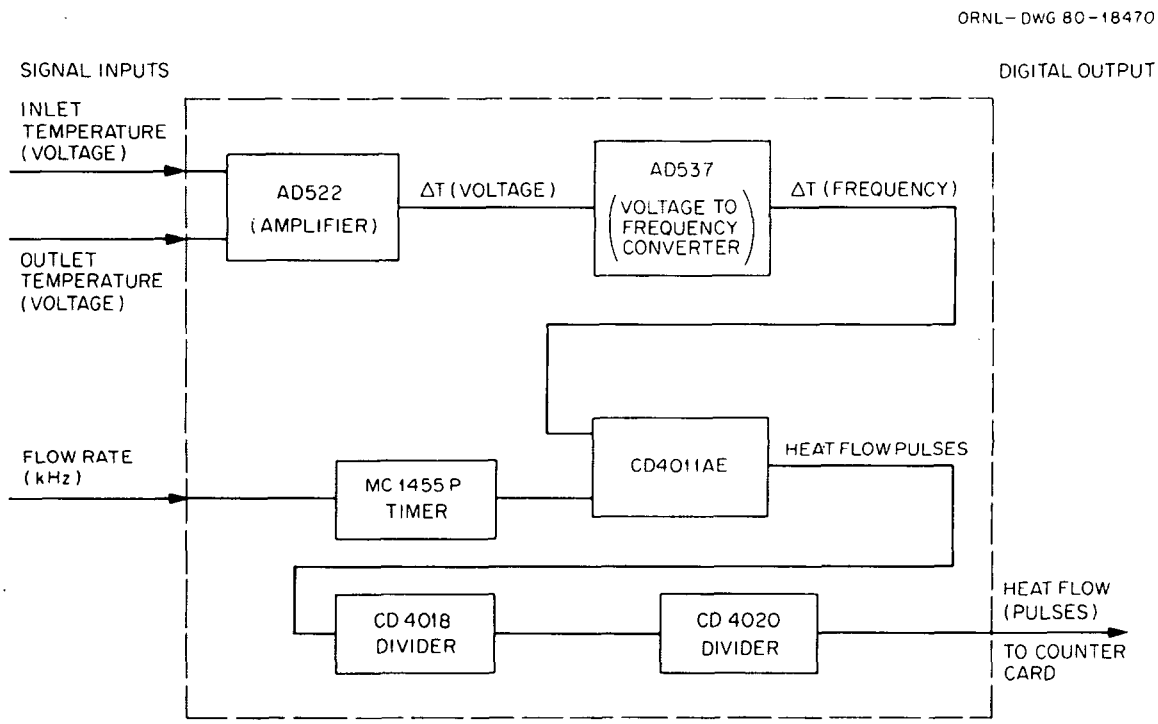

Fig. D.1. Typical heat flow integration circuit.

where

HF = measured heat flow,

K = data constant to convert the digital heat flow signal to the proper units,

f_{flow} = flowmeter signal in kHz,

$F_{\Delta T}$ = frequency signal in kHz proportional to the temperature difference.

Barring malfunctions, there are two major sources of error in the heat flow measurements: (1) uncertainties in the flow and ΔT measurement and (2) uncertainty in the data constant determination. Errors due to drift in the electronic circuitry can be neglected because (1) the equipment is located in a nearly constant temperature environment and (2) frequent calibrations adjust for any long-term drift. The repeatability of the electronics for the BIN_i measurement is shown in Table D.1.

Table D.1. ACES heat flow and power consumption under steady-state, full-load conditions, Jan. 25, 1979

Hour	BIN_i (Wh)	Q_E (Wh)	BIN ΔT (°C)	Q_H (Wh)
0000-0100	4954	3296	1.81	8250
0100-0200	4954	3286	1.79	8240
0200-0300	4954	3288	1.82	8242
0300-0400	4954	3286	1.78	8240
0400-0500	4954	3283	1.91	8237
0500-0600	4954	3276	1.82	8230

Flow measurements are made with turbine flowmeters (in the bin, fan coil, and solar panel loops) of ±0.05% accuracy or positive-displacement flowmeters (in the hot-water loop) of ±1.5% accuracy. Temperature measurements are made using RTDs calibrated to within ±0.06°C. Figures D.2 and D.3 show a typical flowmeter and RTD pair used for heat flow measurement.

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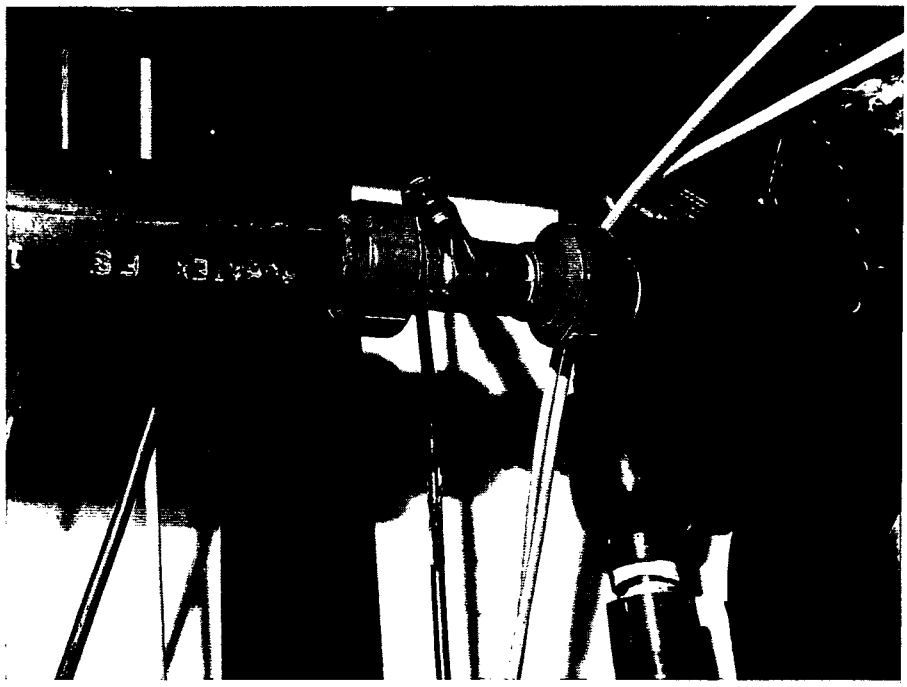

Fig. D.2. Flowmeter for fan coil heat flow measurement.

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Fig. D.3. RTD pair for fan coil heat flow measurement.

Data constants are determined by performing in situ calibrations. The system is run in steady-state mode and the flow rate and temperature difference measured. A heat flow is calculated and ratioed to the DAS calculated heat flow to determine the constant as follows:

$$K = K' \times \left(\frac{\text{DAS heat flow}}{\text{calculated heat flow}}\right) ,$$

where K' is the old data constant.

The uncertainty in the heat flows can be determined from Eq. (1) as follows, using the method described by Holman:[1]

$$U_{HF} = \left[\left(\frac{\partial HF}{\partial K} \times U_K\right)^2 + \left(\frac{\partial HF}{\partial f_{flow}} \times U_{flow}\right)^2 + \left(\frac{\partial HF}{\partial f_{\Delta T}} \times U_{\Delta T}\right)^2\right]^{1/2} ,$$

where U_K, etc., are the uncertainties of each component. Carrying out the differentiation,

$$U_{HF} = \left[\left(f_{flow} \times f_{\Delta T} \times U_K\right)^2 + \left(K \times f_{\Delta T} \times U_{flow}\right)^2 + \left(K \times f_{flow} \times U_{\Delta T}\right)^2\right]^{1/2} .$$

Dividing by $K \times f_{flow} \times f_{\Delta T}$,

$$\frac{U_{HF}}{HF} = \left[\left(\frac{U_K}{K}\right)^2 + \left(\frac{U_{flow}}{f_{flow}}\right)^2 + \left(\frac{U_{\Delta T}}{f_{\Delta T}}\right)^2\right]^{1/2} . \qquad (2)$$

For the heat flow, BIN_i, we have the following demonstrated averages:

$U_K/K = \pm 2.1\%$,

$\Delta T = 1.9°C$,

$U_{\Delta T}/f_{\Delta T} = \pm 4.2\%$,

$U_{flow}/f_{flow} = \pm 0.05\%$.

For these values,

$$U_{BIN_i}/BIN_i = \pm 4.7\% \ .$$

Table D.2 contains average temperature differences and uncertainties for the major heat flows.

Table D.2. Heat flow uncertainties

Heat flow	Nominal ΔT (°C)	$U_{\Delta T}/\Delta T$ (%)	U_K/K (%)	U_{flow}/f_{flow} (%)	Heat flow uncertainty (%)
BIN_i	1.9	4.2	2.1	0.05	4.7
BIN_o					
Ice cooling	6.2	1.3	3.9	0.05	4.1
Supplemental cooling	3.4	2.0	3.9	0.05	4.4
FAN_i					
Ice cooling	6.2	1.3	4.2	0.05	4.4
Supplemental cooling	3.4	2.0	4.2	0.05	4.6
FAN_o	4.0	1.9	2.2	0.05	2.9
HWT_o	20.2	0.4	0.8	1.50	1.7
SLP_o	4.0	1.9	2.0	0.05	2.8
SLP_i	4.1	2.0	2.0	0.05	2.8

D.2 Heating and Cooling Load Uncertainties

D.2.1 Heating loads

Hourly heating loads are determined using the following equation:

$$Q_H = [BIN_i - BIN_o + SLP_i - SLP_o + Q_E - HWT_o] \ (Wh) \ .$$

Since the solar panel has operated for only brief test periods, the effect of the SLP_i and SLP_o heat flows on the heating load can be neglected.

The electrical consumption Q_E is the sum of the ACES equipment power consumption,

$$Q_E = COMP + BLWR + Pclg + Phtg + Phwt \ ,$$

and its uncertainty averages about ±0.08%. Expressing the heating load as

$$Q_H = BIN_i - BIN_o + Q_E - HWT_o ,$$

the uncertainty for each hourly calculation can be expressed as

$$U_{Q_H} = \left[\left(\frac{\partial Q_H}{\partial BIN_i} \times U_{BIN_i}\right)^2 + \left(\frac{\partial Q_H}{\partial BIN_o} \times U_{BIN_o}\right)^2 + \left(\frac{\partial Q_H}{\partial Q_E} \times U_{Q_E}\right)^2 + \left(\frac{\partial Q_H}{\partial HWT_o} \times U_{HWT_o}\right)^2\right]^{1/2} ,$$

using the method described by Holman.[1] For the space heat only mode, nominal values are

BIN_i = 4954 Wh,	U_{BIN_o} = 232.8 Wh,
Q_E = 3286 Wh,	U_{Q_E} = 2.6 Wh,
HWT_o = 0,	$U_{BIN_o} \leq$ 0.6 Wh,
$BIN_o \leq$ 10 Wh,	
Q_H = 8230 Wh,	

U_{Q_H} = 232.8 Wh or U_{Q_H}/Q_H = ±2.8%.

For the space heat and water heat mode,

BIN_i = 5255 Wh,	U_{BIN_o} = 247.0 Wh,
$BIN_o \leq$ 10 Wh,	$U_{BIN_o} \leq$ 0.6 Wh,
HWT_o = 1820 Wh,	U_{HWT_o} = 30.9 Wh,
Q_E = 3306 Wh,	U_{Q_E} = 2.6 Wh,
Q_H = 6691 Wh,	U_{Q_H} = 249 Wh,

or U_{QH}/Q_H = ±3.7%.

For the water heat only mode

BIN_i = 4740 Wh,	U_{BIN_i} = 222.8 Wh,
$BIN_o \leq$ 10 Wh,	$U_{BIN_o} \leq$ 0.6 Wh,
HWT_o = 6200 Wh,	U_{HWT_o} = 105.4 Wh,
Q_E = 2840 Wh,	U_{QE} = 2.3 Wh,
Q_H = 1336 Wh,	U_{QH} = 246.5 Wh,

or $U_{QH}/Q_H = \pm 18.4\%$.

Typically, about 88% of the heating load is supplied by the heating-only mode, 9% by the space and water heat mode, and 3% by the water-heat-only mode. Therefore the weighted average of the hourly heating load uncertainty is

$$U_{Q_H}/Q_H = \pm 3.4\%.$$

This can be considered a maximum error. In actual practice, measured heat flows and power consumption seldom deviated by more than ±2% from a mean value when in full load, steady-state operation, as illustrated in Table D.1.

When the hourly heating loads are summed over an entire month, the maximum error is reduced to about ±0.1%. For example, the February 1979 heating load was 8.41 GJ or 2336 kWh. The average hourly load was 3.5 ±0.12 kWh.

The monthly load is determined by

$$Q_{H_m} = Q_{H_1} + Q_{H_2} + \ldots + Q_{H_i} \quad ,$$

where i = number of hours in the month, and the uncertainty of this load can be determined by

$$U_{Q_{H_m}} = [U_{Q_{H_1}}{}^2 + U_{Q_{H_2}}{}^2 + \ldots + U_{Q_{H_i}}{}^2]^{1/2} \ .$$

For February 1979, with the load distributed evenly over 672 h, the uncertainty would be

$$U_{Q_{H_m}} = 3.0 \text{ kWh},$$

or

$$\frac{U_{Q_{H_m}}}{Q_{H_m}} = \frac{3.0}{2336} = 0.001.$$

If the package operated at full-load capacity, it would require about 290 h to deliver the load. The uncertainty under this condition would be

$$U_{Q_H}/Q_{H_m} = 0.002.$$

D.2.2 Cooling loads

With the mechanical room unventilated (stored-ice cooling phase), hourly cooling loads are calculated by

$$Q_C = FAN_i - [BLWR + 0.5 \times COMP + 0.75 \times PUMPS].$$

For full-load operation with water heating, nominal values are

$FAN_i = 9250$ Wh,	$U_{FAN_i} = 407$ Wh,
$BLWR = 550$ Wh,	$U_{BLWR} = 0.6$ Wh,
$COMP = 2600$ Wh,	$U_{COMP} = 2.6$ Wh,
$PUMPS = 240$ Wh,	$U_{PUMPS} = 2.6$ Wh,
$Q_C = 7220$ Wh,	$U_{Q_C} = 407$ Wh,

or $U_{Q_C}/Q_C = 0.056.$

Without water heating,

FAN_i = 9250 Wh, U_{FAN_i} = 407 Wh,

BLWR = 550 Wh, U_{BLWR} = 0.6 Wh,

PUMPS = 160 Wh, U_{PUMPS} = 0.2 Wh,

Q_C = 8580 Wh, U_{Q_C} = 407 Wh,

or $U_{Q_C}/Q_C = 0.047$.

With the mechanical room ventilated (supplemental cooling phase), the cooling load is calculated by

$$Q_C = FAN_i - BLWR.$$

For full-load operation,

FAN_i = 5440 Wh, U_{FAN_i} = 250 Wh,

BLWR = 550 Wh, U_{BLWR} = 0.5 Wh,

Q_C = 4890 Wh, U_{Q_C} = 250 Wh,

or $U_{QC}/Q_C = 0.051$.

Here again, summing up the hourly cooling loads over each month has the effect of lowering the monthly uncertainty to around ±0.1%.

D.3 Factors Affecting DAS Accuracy

The foregoing analyses have estimated hourly and seasonal accuracies for the DAS, assuming normal operation. Occasional complete-system outages and individual circuit or component malfunctions which require that the data be adjusted reduce the overall DAS accuracy level. Estimated overall monthly accuracy levels for reported loads and power consumptions from December 1978 through September 1980 are listed in

Table D.3. These levels are based on the theoretical uncertainties and conservative estimates of the effects of perturbations caused by malfunctions, incorrect calibrations, outages, and other external error sources.

Table D.3. Accuracy levels for reported power consumptions and loads

Month	Loads (%)		Power consumption (%)	
	ACES	Control	ACES	Control
Dec. 1978				2.0
Jan. 1979	5.0	5.0	2.0	2.0
Feb. 1979	2.0	2.0	2.0	2.0
Mar. 1979	2.0	2.0	2.0	2.0
April 1979	2.0	2.0	2.0	2.0
May 1979	5.0	5.0	2.0	2.0
June 1979	0,-3	2.0	2.0	2.0
July 1979	0,-2	2.0	2.0	2.0
Aug. 1979	5.0	5.0	2.0	2.0
Sept. 1979	2.0	2.0	2.0	2.0
Dec. 1979	0.5	0.5	0.5	0.5
Jan. 1980	0.5	0.5	0.5	0.5
Feb. 1980	2.0	2.0	2.0	2.0
Mar. 1980	0.5	0.5	0.5	0.5
April 1980	0.5	0.5	0.5	0.5
May 1980	0.5	0.5	0.5	0.5
June 1980	1.0	0.5	1.0	0.5
July 1980	0.5	0.5	0.5	0.5
Aug. 1980	2.0	2.0	0.5	0.5
Sept. 1980	0.5	0.5	0.5	0.5

Reference for Appendix D

1. J. P. Holman, *Experimental Methods for Engineers*, pp. 37-44, McGraw-Hill, New York, 1971.

Appendix E

SAMPLE CALCULATIONS

E.1 Space Heating with Hot-Water COP

Because the mechanical package is located inside the thermal envelope of 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 input (Wh)		Energy output (Wh)
Compressor	2430	Heat from bin	5255
Fan	480	Electrical consumption	3306
Pumps	396		
Total	3306		8561

COP = output/input = 8561/3306 = 2.59

E.2 Space Heat Only

	Electrical input (Wh)		Energy output (Wh)
Compressor	2490	Heat from bin	4955
Fan	480	Electrical consumption	3285
Pumps	318		
Total	3285		8240

COP = output/input = 8240/3285 = 2.51

E.3 Space Cooling (Ice Phase)

Useful cooling is defined as the energy extracted by the fan coil minus the energy input to the fan and the pump motor heat loss; the COP is the useful cooling divided by the energy input.

	Useful cooling (Wh)		Energy input (Wh)
Heat extracted by the fan coil	9250	Fan	550
Less fan energy + 3/4 pump	651	Pump	135
Total	8599		685

COP = useful cooling/energy input = 8599/685 = 12.55

E.4 Water Heating

The COP for water heating is the heat into the hot water divided by the electrical input. During the summer, when no credit is taken for the heat given off by the compressor and pumps, the COP is determined by:

	Heat output (Wh)		Electrical input (Wh)
Water heat output	6200	Compressor	2600
		Pumps	246
Total	6200		2846

COP = output/electrical input = 6200/2846 = 2.18

During the winter, a credit is taken for the heat given off by the electrical components; thus, the COP is determined by:

	Heat output (Wh)
Heat removed from the bin	4740
Electrical input	2846
Total	7586

COP = output/input= 7586/2846 = 2.67

E.5 Night Heat Rejection

The COP for night heat rejection is best quantified by calculating the bin heat rejection COP [COP(BR)]. 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 supplemental cooling phase COP must consider the loads delivered and the bin heat leakage.

	Useful energy (Wh)		Electrical input (Wh)
Stored cooling	5130	Compressor	2830
		Cold brine	165
		Hot-brine pump	151
Total	5130		3146

$$COP(BR) = \frac{5130 - 0.25(165)}{3146} = 1.62$$

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