ORNL/CON-19

Conservation
—
Division of Buildings
and
Community Systems

Performance Report for the ACES Demonstration House August 1976 Through August 1977

Eugene C. Hise

OAK RIDGE NATIONAL LABORATORY
OPERATED BY UNION CARBIDE CORPORATION · FOR THE DEPARTMENT OF ENERGY

Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
Price: Printed Copy $5.25; Microfiche $3.00

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, contractors, subcontractors, or their employees, makes any warranty, express or implied, nor assumes any legal liability or responsibility for any third party's use or the results of such use of any information, apparatus, product or process disclosed in this report, nor represents that its use by such third party would not infringe privately owned rights.

ORNL/CON-19

Contract No. W-7405-eng-26

Energy Division

PERFORMANCE REPORT FOR THE ACES DEMONSTRATION HOUSE
AUGUST 1976 THROUGH AUGUST 1977

Eugene C. Hise

This work was supported by
the Department of Energy
Division of Buildings and Community Systems

Date Published: March 1978

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

CONTENTS

	Page
ABSTRACT	1
1. SUMMARY	1
2. MEASURED PERFORMANCES	3
2.1 Introduction	3
2.2 Description of Functions	3
2.2.1 Space heating with water heating	3
2.2.2 Water heating alone	4
2.2.3 Night heat rejection with water heating	4
2.2.4 Space cooling from stored ice	5
2.3 Performance Values	5
2.3.1 Energy inputs	5
2.3.2 Energy outputs	5
2.3.3 System operating temperatures	6
2.3.4 System performance	8
3. PROJECTED ANNUAL PERFORMANCE	13
4. LOAD MANAGEMENT	16
4.1 Heating Season Performance	16
4.2 Cooling Season Performance	17
5. SYSTEMS DESCRIPTION	21
5.1 The Building	21
5.1.1 Architectural	21
5.1.2 Thermal envelope	21
5.2 The Annual Cycle Energy System	21
5.3 The Ventilation Cycle	26
5.4 The Control System	27
5.4.1 Primary control functions	27
5.4.2 Secondary control functions	28
5.4.3 Control modes	28
5.5 The Data Acquisition System	29
5.5.1 Data sensing and on-site processing	29
5.5.2 DAS printout item description	30
5.6 The Loads Simulation Packages	33
5.6.1 Hot water	33
5.6.2 Electrical consumption and people	34
5.6.3 Infiltration due to door opening	34
REFERENCE	35
APPENDIX A — FIRST YEAR CHRONOLOGY	39
APPENDIX B — COMPONENT SPECIFICATIONS AND DESIGN DATA	43
APPENDIX C — SAMPLE CALCULATIONS AND CONDITIONS	49

PERFORMANCE REPORT FOR THE ACES DEMONSTRATION HOUSE
AUGUST 1976 THROUGH AUGUST 1977

Eugene C. Hise

ABSTRACT

The Annual Cycle Energy System demonstration house was constructed to demonstrate the energy conserving potential of an integrated space-heating and cooling and water-heating system consisting of a high-performance unidirectional heat pump, low-temperature thermal storage in the heat of fusion of ice, solar assistance, and passive heat rejection. The house was completed and preliminary operation began in July 1976. Continuous operation and complete data collection began in May 1977. Performance measurements and an analytically constructed integrated annual cycle predict an annual coefficient of performance (COP) of 3.16.

---

1. SUMMARY

The Annual Cycle Energy System (ACES) demonstration house is one of a complex of three houses built near Knoxville, Tennessee, to quantify various concepts of energy conservation in housing. The distinctive feature of the house is the Annual Cycle Energy System: an integrated space-heating and cooling and water-heating system composed of a high-efficiency unidirectional heat pump, low-temperature thermal storage, solar assistance, and passive heat rejection. The system is also integrated in time because it recycles heat on an annual basis between the thermal storage (ice bin) and the living space by pumping heat up from the bin to the living space in the heating season and letting the heat flow from the living space back down to the bin in the cooling season.

The second of the three houses, the control house, is identical in construction to the ACES house, but is heated by conventional heat pump or by resistance heat, is cooled by conventional heat pump and has a

conventional resistance-heated domestic hot-water tank. Its function is to permit a direct comparison of ACES with conventional systems, and its performance is discussed in this report only for purposes of comparison. The third house, the solar house, again has an identical thermal envelope but has a solar space- and water-heating system. The solar house is not discussed in this report.

The ACES house was completed and operation of the heat pump system was begun in July 1976. During the following year the auxiliary systems for data acquisition and loads simulation were installed and brought into full operation, performance testing in all modes and conditions was done, modifications and component replacements were made to improve performance, and component failures were repaired. The system did not operate throughout the first year in the complete and continuous fashion necessary to fulfill an annual cycle. However, the performance of the system has been analyzed in each operating condition, and an annual performance has been analytically constructed. The system exhibits a COP of 2.65 in space heating, 2.9 in water heating, and 10 to 13 in space cooling from chilled water or ice. Using the historical weather data for the annual cycle from October 1957 to October 1958 (a year selected as average among the 10 years for which data were available), an annual or integrated COP of 3.16 has been calculated.

The following are previously published reports relating to ACES:

E. C. Hise, J. C. Moyers, and H. C. Fischer, *Design Report for the ACES Demonstration House*, ORNL/CON-1 (October 1976).

H. C. Fischer et al., *The Annual Cycle Energy System: Initial Investigations*, ORNL/TM-5525 (October 1976).

H. C. Fischer, ed., *Summary of Annual Cycle Energy System Workshop I held October 29–30, 1975, at Oak Ridge, Tennessee*, ORNL/TM-5243 (July 1976).

2. MEASURED PERFORMANCES

2.1 Introduction

The ACES provides the functions of space heating, space cooling, and water heating; further, the space and water heating are frequently provided simultaneously, and much of the space cooling is provided from long-term storage. Thus, ACES is integrated in both function and time so that its true performance must be measured in terms of the electrical energy input required and the heating and cooling supplied by the system for the entire annual cycle. During this first year, the system did not operate in the continuous fashion necessary to fulfill an annual cycle; however, it has operated a large fraction of the time, and complete energy balances have been performed around the system operating in each of its functional modes. Using these measurements, a loads determination code, and a weather tape, an annual performance has been analytically constructed.

2.2 Description of Functions

The system performs three functions that require the operation of the compressor, namely, space heating with water heating, night heat rejection with water heating, and water heating only. It performs two functions that do not require the compressor, namely, space cooling with chilled water and space cooling by the ventilation cycle. The methods for calculating the COP in each of these functions (as opposed to the annual COP) are discussed in the following sections. A description of the physical system with flow schematics is given in Sect. 5.2.

2.2.1 Space heating with water heating

When the system operates to supply space heating, the domestic hot water is circulated through the desuperheater only, and brine is circulated in the space-heating condenser and fan coil circuit and in the evaporator and ice bin circuit. The COP is calculated as the useful heat delivered divided by the heat equivalent of all the electrical input. The useful heat is the sum of the measured heat delivery to the hot-water tank, the

measured heat delivery to the fan coil, the heat equivalent of the electrical input to the fan motor, and the heat input into the space from the surface of the hermetic compressor case and the several pump motors. The electrical input to the fan motor must be added in separately since it is, of course, all deposited in the circulating air stream but is not accounted for in the heat delivered to the fan coil by the circulating brine. Similarly, the heat of inefficiency from the compressor and pump motors is deposited directly into the air of the mechanical equipment room and circulated to the house. The electrical input is the sum of the inputs to the compressor, the fan, and the three circulating pumps.

2.2.2 Water heating alone

When the system operates to supply water heating only, the domestic hot water is circulated through the water-heating condenser and the desuperheater in series, and the brine is circulated through the evaporator and ice bin circuit. The COP is calculated as being the measured heat delivery to the hot-water tank divided by the heat equivalent of the electrical input to the compressor, the hot-water circulating pump, and the cooling circulating pump.

2.2.3 Night heat rejection with water heating

When the system operates to perform night heat rejection, the domestic hot water is circulated through the water-heating condenser and the desuperheater in series, the hot brine is circulated through the space heating condenser and solar panel-convector, and the chilled brine is circulated through the evaporator and the ice bin. Night heat rejection is performed to generate and store chilling capacity for a later air-conditioning demand, and therefore, the primary useful output is stored for later use. When that later air-conditioning function is performed, the system is credited with the measured cooling delivery and therefore cannot be credited with the stored cooling during night heat rejection. The heat delivery to the hot water is accounted for immediately (not upon use) and is therefore credited to the system during night heat rejection. The COP is calculated as being the measured heat delivery to the hot-water tank divided by the heat equivalent of the electrical input to the compressor and pumps.

2.2.4 Space cooling from stored ice

When the system operates to supply space cooling from the ice bin, brine is circulated through the ice bin and the fan coil in series. The COP is calculated as being the net cooling effect divided by the heat equivalent of the electrical input to the fan and the cooling circuit pump. The net cooling effect is the measured heat gain in the fan coil minus the heat equivalent of the electrical input to the fan and pump.

2.3 Performance Values

2.3.1 Energy inputs

The compressor, pumps, and fan are individually metered. Their energy draws vary somewhat as a function of temperature and control valve circuiting and are therefore tabulated as ranges (see Table 2.1).

Table 2.1. Energy inputs[a]

Unit	Energy input (Btu/h, full load)
Compressor	COM = 7700–9100
Fan	FAN = 1330–1380
Cooling circuit pump	PCL = 510–600
Heating circuit pump	PHT = 480–500
Domestic hot-water pump	PHW = 270–280

[a]Symbols are defined in Section 5.5.2.

2.3.2 Energy outputs

The system delivers services to the house in the form of space heating, space cooling, and water heating. In performing these services, it also transfers heat to and from the thermal storage and the solar panel-convector. Again, the magnitudes of these energy outputs vary with operating conditions and are presented as ranges (see Table 2.2). During the heating season of 1976–77, the ice accumulation remained well below the capacity of the thermal storage bin, so the solar panel-convector was not called upon to collect heat and no measurements of its performance in the heat-gathering mode were made.

Table 2.2. Energy outputs[a]

Function	Energy output (Btu/h)
Space heating	FCO = 19,000–26,000
Space cooling	FCI = 21,000–29,000
Water heating	HWO = up to 29,000
Heat rejection to solar convector	SPO = 18,500–24,000
Heat removal from thermal storage	BNI = 18,000–23,000

[a]Symbols are defined in Section 5.5.2.

2.3.3 System operating temperatures

The system temperatures are recorded as instantaneous values at the time of data logging, and therefore, a circulating-system temperature is valid if that system is circulating at the moment of data logging. The temperatures given in Table 2.3 are extracted from a number of applicable data sets and are representative of steady-state operation in the function shown. Temperatures logged during a startup transient or for a nonoperating function could be completely out of the range of temperatures shown.

The design calculations indicated that at maximum ice inventory, the BNI* would be 23°F during steady-state heat extraction. The greatest ice inventory reached during this season was about 30% of maximum, at which time the minimum BNI recorded was 27°F and the compressor suction was 46 psig, corresponding to an evaporating temperature of 23°F. In steady-state operation, BNO approached bulk bin temperature within 0.5–1°F.

During steady-state space cooling, BNO equals FCI, and FCO equals BNI, with BNO achieving a 3–5°F approach to the bulk bin temperature. The higher temperature differential across the bin coil in space cooling rather than in heat extraction reflects the higher loading.

The primary mechanism for heat rejection from the radiant/convector panel is convection. The mean temperature difference between the brine stream and outdoor air temperature, therefore, varies with wind velocity and direction. The design convective coefficient was 2 Btu/ft^2·°F·h under essentially zero wind conditions.

*A list of symbols and their descriptions is given in Section 5.5.2.

Table 2.3. System temperatures

Symbol	Function	Temperature
BNI	Brine entering the bin coils	
	Heat extraction with ice on the coils	27–28°F
	Heat extraction from water above freezing	4°F below average bulk bin temperature
BNO	Brine leaving the bin coils	
	Heat extraction with ice on the coils	30–31°F
	Heat extraction from water above freezing	3–3.5°F above BNI
FCI	Brine entering the fan coil	
	Space cooling with ice	35–36°F
	Space cooling from chilled water	5°F above average bulk bin temperature
	Space heating with return air at 70°F	100°F
FCO	Brine leaving the fan coil	
	Space cooling	9–10°F above FCI
	Space heating	6–7°F below FCI
SPI	Brine entering the radiant/convector	
	Night heat rejection	25–40°F above outdoor dry bulb as a function of wind
	Solar collection	No data
SPO	Brine leaving the radiant/convector	
	Night heat rejection	6–8°F below SPI
HWI	Domestic hot water entering the condenser	55–140°F
HWO	Domestic hot water leaving the desuperheater	120–145°F

HWI has a very wide range of possible temperatures. The low temperature is that of the potable water entering the house in winter. The high temperature results from continued recirculation of the hot water during a lengthy night heat rejection operation. The minimum HWO is set by the thermostatic control valve, while the maximum again results from a lengthy night heat rejection.

2.3.4 System performance

2.3.4.1 Full-load, steady-state performance. The full-load, steady-state tests occurred, in some cases, during normal system operation, and for others it was necessary to override the automatic controls so as to operate the system in the desired functions only at steady state. The data for the test period were collected and analyzed by the data acquisition system (DAS). The COPs vary somewhat with temperatures and with the amount of ice on the submerged coils in the thermal storage bin. Each of the values tabulated is the average of a number of measurements made over the period of March through July 1977 (see Table 2.4).

Table 2.4. Full-load performance

Function	Full-load COP
Space heating with water heating	2.65
Water heating only	2.90
Space cooling with stored ice	12.70
Space cooling with the storage >32°F < 45°F	10.60
Night heat rejection with water heating[a]	0.50 (2.50)

[a]In the strict accounting used by the DAS, only the water heating is calculated as a useful output at the time of night heat rejection because credit is taken for the chilling when it is later used for space cooling. This procedure results in a COP of 0.5. If the chilling credit and the water-heating credit are taken at the time of operation, then a COP of 2.5 results.

Sample calculations of steady-state, full-load COP are shown in Appendix C.

2.3.4.2 Short-cycle performance. Of the several functions performed by the ACES, only those of space heating and water heating are subject to any significant alteration in performance as a result of part load or cyclic duty. There are few or no startup, shutdown, or equalization losses associated with space cooling with chilled water. The night heat rejection function could suffer cyclic losses; however, whenever that function is initiated it almost invariably runs for several hours. The several possible sources for alteration in performance at part load or cyclic duty result from thermal storage effects, temperature and pressure equalization effects, and the normal variations encountered in a residential application. The DAS properly accounts for each of these effects.

The heat source for the evaporator is the circulating brine stream, which, in turn, draws its heat from the bin. During steady-state operation, the brine leaving the bin will be 2–5°F below the bulk bin water temperature, but during shutdown the brine rises to bulk bin temperature. The volume of brine is such that for about 5 min after startup the evaporator is operating at the higher temperature. This effect tends to raise the performance of short, widely spaced cycles.

The domestic water-heating pump operates to circulate water from the storage tank bottom through the heat exchangers to the storage tank top, whenever the compressor operates, and establishes a steady-state temperature gradient across the heat exchangers. The refrigerant system uses a nonequalizing expansion valve so that on shutdown the inventory of refrigerant in the heat exchangers condenses and heats up the inventory of water. This stored energy is accounted for by the heat flow integrator on the next startup. Thus the measured performance for a short water-heating cycle reflects the previous operating history, especially compressor discharge enthalpy and duration of off cycle. As the length of the on cycle increases, the influence of the previous history diminishes.

During operation, the condenser is maintained at a high pressure and the evaporator at a low pressure. If, on compressor shutdown, the expansion valve permits the system to equalize, then the stored energy in the refrigerant can be dissipated with no useful product. On startup, the

energy required to recreate the differential is lost, and on short cycles this shutdown—startup loss can significantly affect the performance. However, this system uses a nonequalizing expansion valve, so the refrigerant system suffers little cyclic loss.

Similarly, during operation the brine side of the condenser is maintained at an elevated temperature and the evaporator at a lowered temperature. On shutdown, these systems ultimately converge by thermal conduction with little or no net useful heating or cooling. On startup the energy input to restore the differential delivers no useful product and thus lowers the COP.

The house and the ACES are designed to function as a normal residence with the normal variations in external and internal conditions and with no provisions for maintaining constant evaporating conditions (by controlling the water temperature or ice inventory in the storage bin), or for maintaining constant condensing conditions (by controlling the temperature of the water entering the water heating condenser and desuperheater). Thus, there is no way of conducting perfectly reproducible experiments.

Part-load space heating with water heating. Several sets of measurements were made for duty cycles of 10, 20, 30, and 40 min. The results were consistent, and the average performances are given in Table 2.5.

Table 2.5. Part-load performance for space heating with water heating

Duration of cycle (min)	Averaged COP
10	2.5
20	2.51
30	2.58
40	2.55
Average of full hour tests	2.65

Part-load water heating only. Several sets of part-load tests of water heating only for duty cycles of 10, 20, and 30 min were performed, and the averaged results are presented in Table 2.6. The competing effects of the several factors discussed in Sect. 2.3.4.2 show strongly in these part-load measurements.

Table 2.6. Part-load performance in water heating only

Duration of cycle (min)	Averaged COP
10	2.9
20	2.8
30	2.7
Average of full hour tests	2.9

2.3.4.3 Integrated performance. Since the different systems of the complex became fully operational at various times during the year, there are no long-term integrated performance data in the normal annual cycle mode. There is an 11-week period of nearly continuous operation and data from June 20 to September 2, 1977, during which time the ACES was operating in the night heat rejection function to store cooling for the required daily air conditioning. This is, of course, the least favorable operating mode for the ACES, since in the full annual cycle operating mode the space cooling for this period would have been supplied from the stored ice.

The analyses that can be performed over this period are the overall COP of the ACES and the inferred COP of the conventional heat pump in the control house. The pertinent measured data for these analyses are the space cooling delivered to the ACES house, the hot water delivered to the ACES house, the electrical consumption of the ACES mechanical package, and the electrical consumption of the control house heat pump. The assumed parameters are that the space cooling consumption of the control house is the same as for the ACES, and the electrical consumption of the control house water heater is the electrical equivalent of the measured heat supplied to the ACES hot-water tank. These assumptions are slightly favorable to the control house, since the occupancy level of the ACES house is higher due to workers and semiweekly tours, and since the various experiments performed for this report increased the operating time of the ACES.

Thus, while the data and analyses are not representative of normal operation, they provide a comparison of the conventional heat pump and water heater with a "worst condition ACES" (see Table 2.7).

Table 2.7. Integrated performance data for the period June 20 to September 2, 1977

Total space cooling supplied by ACES	17,582,848 Btu
Total water heating supplied by ACES	6,213,670 Btu
Total electrical input to ACES	3,271 kWh
Total electrical input to control house heat pump	2,629 kWh
Equivalent electrical input to control house for water heater (calculated as water heating supplied by ACES ÷ 3412 Btu/kWh)	1,820 kWh
Total services supplied by ACES (cooling + hot water)	23,796,518 Btu
Total input to ACES (3271 kWh × 3412 Btu/kWh)	11,160,652 Btu
COP	2.13
Total services supplied to control house (cooling + hot water)	23,796,518 Btu
Total electrical input to control house (4449 kWh × 3412 Btu/kWh)	15,183,818 Btu
COP	1.57

3. PROJECTED ANNUAL PERFORMANCE

A projected annual performance has been calculated based on the measured performances, presented in Sect. 2, and the heating, cooling, and water-heating loads from the original design calculations.[1] Table 3.1 displays the calculated total heating, cooling, and water heating required by the house for the year and the calculated electrical consumption for each of three systems: (1) resistance heat, electric water heater, and electric air conditioner; (2) air-to-air heat pump and electric water heater; and (3) ACES. The absolute and relative magnitudes of these values are, of course, dependent upon the weather cycle.

A cycle year starts with the beginning of the heating season, chosen here to be October 1 because no heating loads appeared in September, and extends to October 1 the following year, as opposed to following the calendar year of January 1 to January 1. Table 3.2 presents monthly values from October 1, 1957, to October 1, 1958 (chosen as a "typical" year). These values were derived in the following manner:

The space heating and cooling loads were calculated using a loads determination code and the hourly weather tape.

The water-heating loads were calculated for a 75-gal/day hot-water consumption heated from the appropriate ground water temperature for the month.

The net heat pumped from the bin is the heat removed from the bin to supply the indicated heating loads at the appropriate COP minus the heat deposited in the bin by the indicated cooling load.

The conductive gain to the bin is calculated from the insulating characteristics of the bin wall and the ground temperature profile for the month.

The total bin change is the net heat pumped from the bin minus the conductive gain.

The bin inventory at the end of the month is the total Btu of stored cooling capacity in the form of ice. There was no ice inventory at the beginning or end of the annual cycle.

Table 3.1. Projected annual performance for weather history of October 1957–October 1958

	Services supplied (10^6 Btu)	Energy consumed by system (kWh): Resistance heat, electric air conditioner, electric water heater	Air-to-air heat pump, electric water heater	ACES
Space heating	22.5	6,594	3,297[a]	4959
Water heating	16.1	4,718	4,718	
Space cooling	14.8	2,168[a]	2,168[a]	
Total	53.4	13,480	10,183	4959
Annual COP		1.16	1.54	3.16

[a]A seasonal heating COP of 2 is assumed for the heat pump. A seasonal EER of 6.8 (COP of 2 in cooling) is assumed for air conditioning.

The kWh consumed for heating are calculated from the heating load (space plus water) for the month and the appropriate COP. A COP of 2.6 is used as a seasonal average for space heating plus water heating. A COP of 3 is used for water heating only. During night heat rejection the electrical consumption is charged to chilling, and the water is heated from the reject heat.

The kWh consumed for space cooling are calculated from the space cooling load for the month and the appropriate COP. A COP of 12.7 is used for cooling from stored ice and a COP of 10.6 from chilled water.

The kWh consumed for night heat rejection are calculated for the space cooling required plus the conductive gain to the bin (total bin change) at a COP in cooling of 2.0.

No credit is taken for ventilation cycle cooling, since it will work equally well with any forced-air cooling system.

Table 3.2. Monthly load calculations

Month	Heating load	Cooling load	Water heating load	Heat pumped from bin	Conductive bin gain	Total bin change	Bin inventory E.O.M.	Heating	Cooling	Night heat rejection
	Btu × 10^6							Energy consumed (kWh)		
1957										
Oct.	1.0		1.3	1.45	1.27	0.18	0.18	259		
Nov.	1.9		1.4	2.10	1.02	1.18	1.36	372		
Dec.	3.4		1.6	3.15	0.81	2.34	3.70	564		
1958										
Jan.	6.1		1.7	4.91	0.68	4.23	7.93	879		
Feb.	5.9		1.5	4.66	0.54	4.12	12.05	834		
Mar.	3.4		1.6	3.15	0.64	2.51	14.56	564		
Apr.	0.6		1.4	1.26	0.76	0.50	15.06	225		
May	0.2	1.5	1.3	-0.55	1.00	-1.55	13.51	147	34.6	
June		3.5	1.1	-2.80	1.18	-3.98	9.53	107	80.8	
July		4.3	1.1	-3.56	1.36	-4.92	4.61	107	99.2	
Aug.		3.6	1.0	-2.94	1.47	-4.41	0.20	98	83.0	
Sept.[a]		1.9	1.1		1.37					
1-2		0.13	0.08	-0.05	0.10	-0.18	0.0	7	3.0	
3-30		1.77	1.02	3.05	1.27	0.0	0.0		49	447
Total	22.5	14.8	16.10		12.10			4163	349.6	447

Total services supplied = 53.4 × 10^6 Btu
Total kWh = 4959
Maximum ice inventory = 52.3 tons = 1867 ft^3

[a] Sept. 1-2: The cooling load is supplied from stored ice, and the compressor is operated to supply hot water.
Sept. 3-30: The compressor is operated in night heat rejection to supply the cooling load and supplies the hot water from rejected heat.

4. LOAD MANAGEMENT

The ACES requires less installed electrical generating capacity than other electric heating systems. During the heating season, its heating COP is high and independent of outdoor temperature. During the cooling season after the ice is exhausted, the electrical demand approximates that of a conventional properly sized air conditioner; however, the low-temperature storage permits the compressor to be operated at the most advantageous time rather than during the hottest hours of the day. During both the heating and cooling seasons the domestic hot water is heated at a COP of 2.6 to 3, thus reducing the demand for that service.

4.1 Heating Season Performance

The amount of heating energy that must be supplied by any heating system, of course, varies inversely with the ambient temperature; however, the electrical energy input required to supply that heat is a function of the efficiency of the system. Resistance heating has a COP of 1, and therefore its peak demand is the electrical equivalent of the peak heat load. The conventional air-to-air heat pump efficiency or COP varies with outdoor temperature. Its seasonal COP in the Knoxville climate is about 2, and thus its annual consumption will be about half of the resistance system. However, on the "coldest day" its COP approaches 1, and therefore its peak demand approaches that of the resistance system. The ACES heating COP remains at about 3 because it has a constant temperature heat source, and therefore its peak demand is about one-third that of the resistance or air-to-air heat pump systems.

The instantaneous peak demand for each of the heating systems could be higher than the hourly consumption, but it is assumed that for residential loads the hourly consumption approximates the diversified peak load. The consumption for ACES includes heating the domestic hot water. The other two systems probably should have some coincident water-heating peak added.

Table 4.1 shows the analytical prediction of the hourly peaks for several selected hours of past heating seasons. During the cold weather portion of the 1976–77 heating season, there were insufficient hours of coincident functioning of the ACES, data collection for ACES, and data collection for the control house to permit a comparison of actual consumptions comparable to those in Table 4.1. However, individual measurements of the total electrical draw of the ACES in full-load heating showed a typical peak hourly consumption of 3.3 kWh. Continuous data collected for the control house, using electric resistance heat from February 8 through February 22, 1977, had a daily peak hourly consumption of 6 kWh. During this period the minimum recorded outdoor temperature was 14°F, and the average temperature was 35°F. The electrical consumption for the control house does not include water heating.

Table 4.1. Predicted peak hourly electrical consumptions, heating season — demonstration house — three systems — Knoxville, Tennessee, weather data

Heating load (Btu/h)	Peak hourly consumption (kWh)			
	Resistance heat	Heat pump[a]	ACES[b]	Date
26,513	7.8	6.5–7.8	3.1	Feb. 1958
28,000	8.2	6.8–8.2	3.2	Jan. 1961
30,502	8.9	7.5–8.9	3.3	Dec. 1962

[a]Heat pump consumption assumes a COP of 1–1.2 during coldest hour.

[b]ACES consumption includes water heating; conventional systems do not.

4.2 Cooling Season Performance

During that part of the cooling season in which the ACES can supply the cooling load from stored ice, the electrical consumption is only that required for the circulating pump and blower and is much less than that for a conventional air conditioner. However, for a demand analysis it is necessary to consider also a situation in which the stored ice is consumed and cooling is supplied by compressor operation. In this

situation the ACES efficiency is approximately that of a good conventional air conditioner, and its consumption and demand will be approximately the same. However, the conventional system must supply the cooling load as it occurs, whereas the ACES compressor can operate during the coolest hours of the day or the week and store the cooling effect for use as needed. Thus, this load can be managed, if necessary, to be completely off the utility peak.

Figure 4.1 compares the analytical predictions of the typical daily electrical consumption for air conditioning by the ACES after stored ice is exhausted and for the conventional air conditioner in the control house. This analytical prediction was made to illustrate that, even when the ACES requires compressor operation to accomplish air conditioning, the bulk of the demand is moved to off-peak hours. Figure 4.2 is a portion of a computer plot comparing the actual consumption of the ACES and the control house for air conditioning for August 29—31, 1977, and illustrates that the load was moved to off-peak hours. Again, the electrical consumption for the ACES house includes water heating, while the control house does not. The ACES compressor was moved off-peak not on the basis of a timer, but rather on the basis of whether or not the temperature of a noncirculated tube of the radiant/convector was below 80°F.

ORNL-DWG 76-8808R

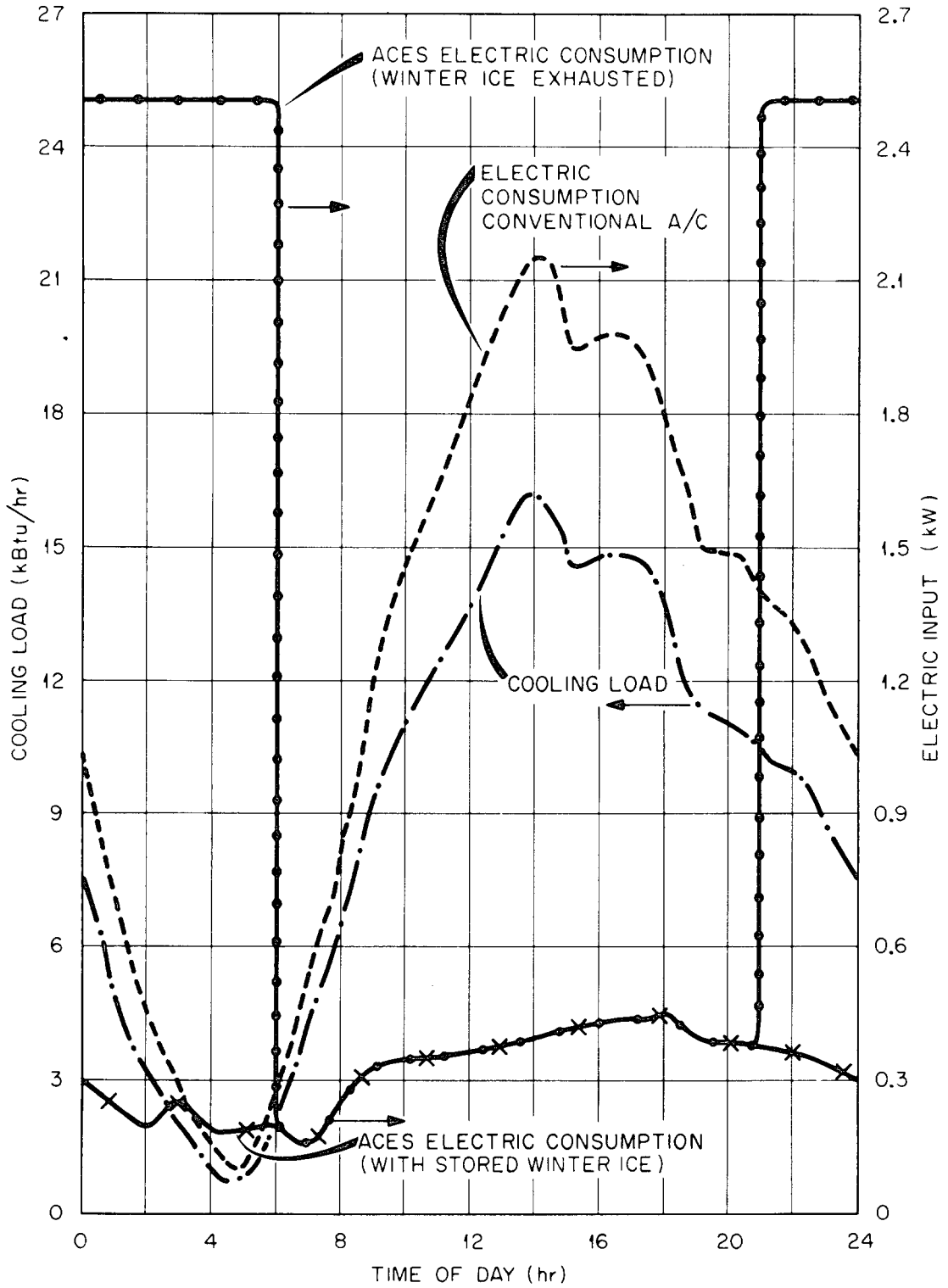

Fig. 4.1. Projected demand vs cooling load.

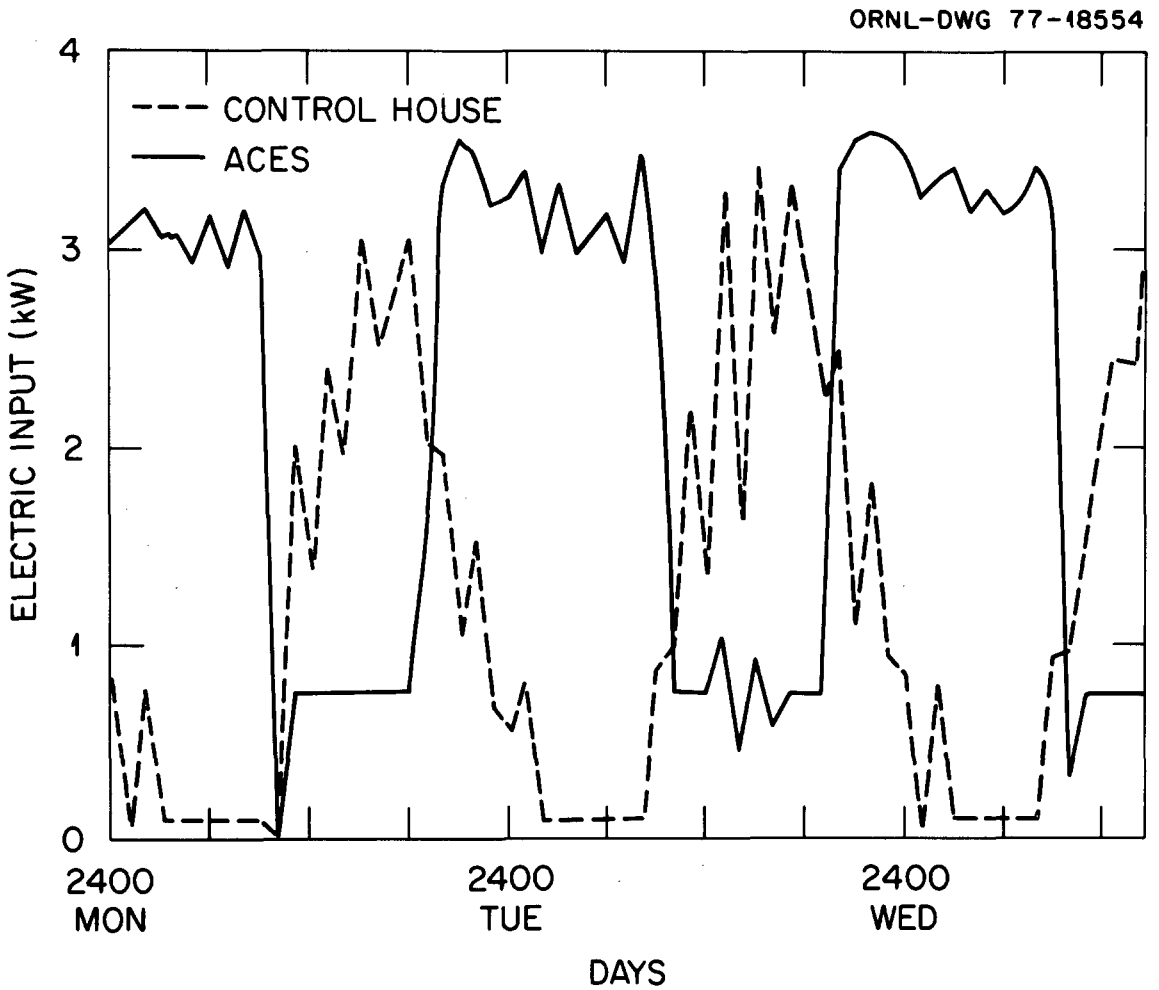

Fig. 4.2. Measured demand vs cooling load.

5. SYSTEMS DESCRIPTION

5.1 The Building

5.1.1 Architectural

The ACES demonstration house[1] is one of a complex of three houses built near Knoxville, Tennessee, to explore energy conservation in housing. The three are the ACES house, the solar house, and the control house. It is the design intent that the three houses be identical in floor plan, appearance, construction, and thermal characteristics, insofar as the differing requirements of the heating-cooling systems permit. The houses themselves (the thermal envelopes) are energy conserving and present the minimum of barriers to acceptance by the building industry and the public.

The house is a two-level, plus partial basement, frame structure having about 2000 ft^2 of living space distributed among three bedrooms, two baths, living-dining room, kitchen, utility room, and entry hall, shown pictorially in Fig. 5.1.

5.1.2 Thermal envelope

The thermal envelope of the house is better insulated than is conventional or than is required by *HUD Minimum Property Standards* for this climate zone. The insulation and the resulting U values for the demonstration house as designed and for *HUD Standards* are given in Table 5.1. Construction details are given in Table 5.2.

The basement walls and the crawl-space foundation are of reinforced concrete poured in Styrofoam forms. The resulting composite has an overall thermal resistance of $R = 16\ h \cdot ft^2 \cdot {}^\circ F \cdot Btu^{-1}$ ($U = 0.062\ Btu \cdot h^{-1} \cdot ft^{-2} \cdot {}^\circ F^{-1}$).

5.2 The Annual Cycle Energy System

The ACES is an integrated space-heating and cooling and domestic water-heating system. Its major elements are:

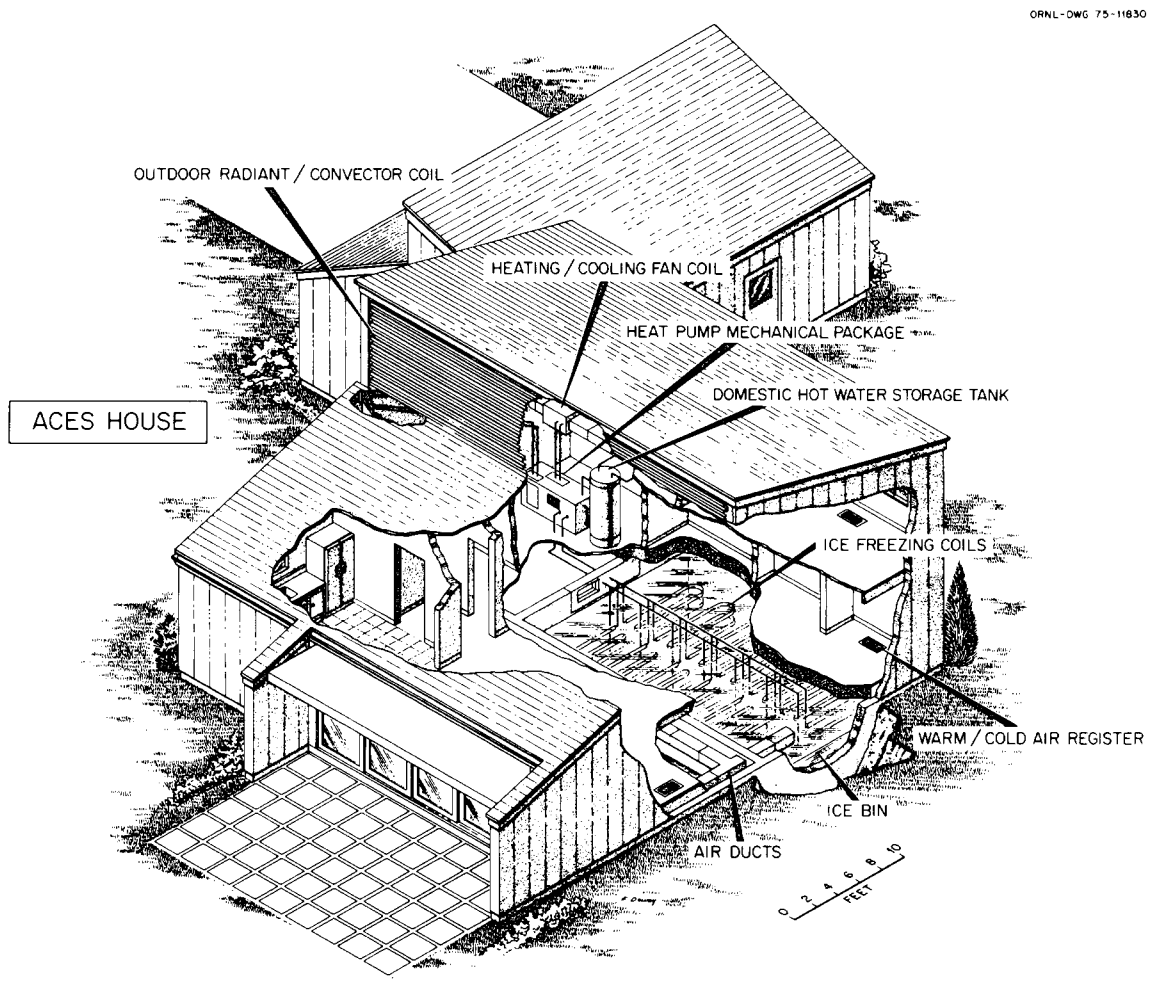

Fig. 5.1. Knoxville ACES demonstration house.

Table 5.1. Insulation details

	Demonstration house		HUD minimum property standards	
	Insulation	Btu/h·ft^2·°F	Insulation	Btu/h·ft^2·°F
Sidewalls	5-1/2 in.	0.045	Not specified	0.08
Ceiling, flat	12 in.	0.024		0.05
Ceiling, cathedral	6 in.	0.040		
Floor (wood floor over crawl)	6 in.	0.040		0.10
Windows	Double glazed	0.58	Single glazed	1.12
Basement and crawl space		0.062		

Table 5.2. Construction details

Exterior walls	1/2-in. gypsum board, 2 × 6 studs 16 in. O.C. with 5-1/2-in. glass fiber batt insulation, 25/32-in. fiber board sheathing, 5/8-in. exterior plywood
Ceiling	(760 ft^2 flat) 1/2-in. gypsum board, 2 × 4 trussed joists with 12-in. glass fiber batt insulation
	(367 ft^2, horizontal projection, cathedral) 1/2-in. gypsum board, 2 × 12 rafters with 6-in. glass fiber batt insulation, 2-1/2-in. vented air space, 5/8-in. plywood sheathing, building paper, asphalt shingles
Roof	(Over flat ceiling, 760 ft^2) 2 × 4 trussed rafters, 5/8-in. plywood sheathing, building paper, asphalt shingles
Floor	Carpet or sheet vinyl, 5/8-in. particle board underlayment, building paper, 5/8-in. plywood subfloor, 2 × 12 joists and 6-in. glass fiber insulation
Windows	Double glazed, wood frame
Door	Metal cased, 1-3/4-in. urethane foam core, with magnetic weather seal

1. A high-efficiency heat pump with refrigerant-to-brine heat exchangers on both the evaporating and condensing side.
2. Thermal storage on the low-temperature (evaporator) side.
3. An auxiliary heat source and sink.
4. A forced-air circulating system with a fan coil for space heating and cooling.
5. A refrigerant-to-water heat exchanger for heating and a tank for storing domestic hot water.

The arrangement of these components in the house is shown pictorially in Fig. 5.1, and a flow schematic is shown in Fig. 5.2. The heat pump (compressor) always extracts heat from the evaporator (20% methanol—brine cooler), which in turn extracts heat from the ice storage bank via the circulating brine system. The heat pump always discharges its heat to two of the three high-temperature heat exchangers: the desuperheater, the hot-water condenser, and the space-heating condenser. Thus the refrigerant cycle is completely contained within the mechanical package and is a nonreversing cycle.

Space heating is accomplished by circulating brine from the heating condenser through the fan coil and transferring the heat to the living space by the circulating air system. Space cooling is accomplished by circulating brine from the ice storage bank through the fan coil and transferring heat to it from the circulating air system.

The selector valves choose the brine circuit for heating or cooling as required. The domestic water is heated by recirculation through the desuperheater during the space-heating season and through the desuperheater and the hot-water condenser during water heating and night heat rejection.

The radiant/convector panel is circuited to be able to accept or reject heat as needed. During the heating season, the panel can accept solar and convective heat and deposit it in the ice bank via the circulating brine system. This mode of operation can be automatically selected when needed (for instance, if the ice inventory exceeds 80% of the bank capacity). Ice formation is restricted to 80% of bank capacity

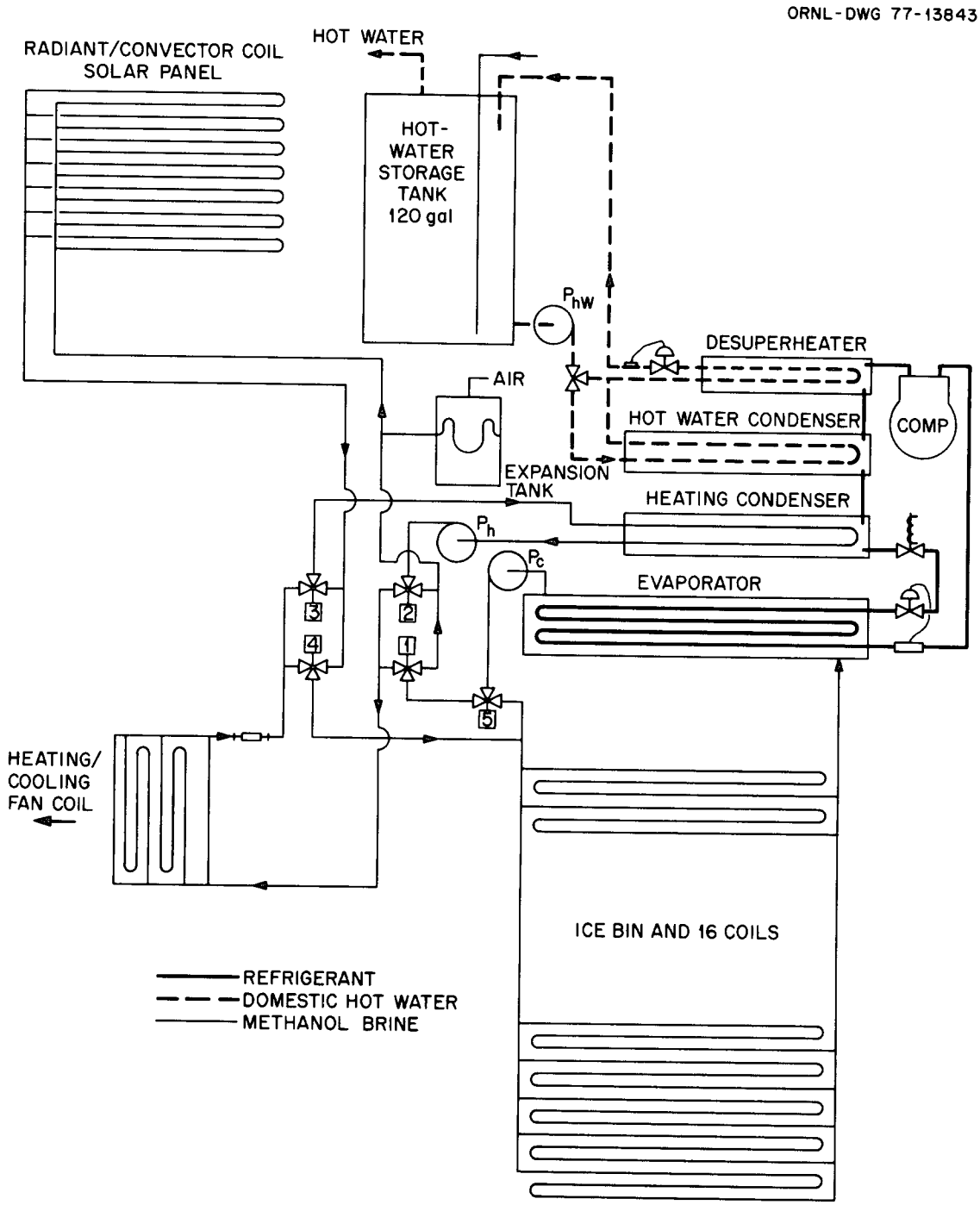

Fig. 5.2. As-built piping schematic for ACES Knoxville demonstration house.

to avoid expansion damage to the container. During the cooling season, the panel can reject heat convectively from the space-heating condenser via the circulating brine system. This mode of operation can be automatically selected when needed (for instance, if the ice inventory is exhausted, the bank temperature exceeds 36°F, and outdoor ambient temperature is below 80°F). This mode permits compressor operation under favorable conditions for heat rejection and off the usual utility peak load. The design specifications for the system and components are given in Appendix B.

5.3 The Ventilation Cycle

In any building the internal loads — that is, the heat from lighting, cooking, appliances, etc.; the heat from the occupants; and the solar energy entering through the windows and structure — will hold the indoor temperature above the outdoor temperature. The outdoor temperature at which the internal loads will maintain the desired indoor temperature is called the break-even temperature. This break-even temperature is a function of the size of the building, the degree of insulation, and the internal loads, and thus can vary widely with different buildings. However, for any building there is that span of outdoor temperatures below the desired indoor temperature and above the break-even temperature for which the building will require cooling. The conventional air-conditioning system must supply this cooling by refrigerating the indoor air. However, if the system is equipped with a ventilation cycle, the cooling will be supplied by ventilating with outdoor air, thus expending only blower energy and no refrigeration energy.

In the demonstration house the heating-cooling duct system is equipped with two motor-operated dampers so arranged that when in the normal position, the blower circulates indoor air through the coil; and in the ventilation cycle position, the blower draws outside air into the house, discharges indoor air outside, and bypasses the coil. The control system places the dampers in the ventilation cycle position whenever the

room thermostat requires cooling and the outdoor thermostat recognizes that the cooling can be accomplished by ventilation.

5.4 The Control System

The external control for the system is like any other space heating-cooling system: a room thermostat having a heating and a cooling temperature setting. The internal control system is different, but of about the same complexity as that of a conventional air-to-air heat pump.

The required sensing elements are the room thermostat, the hot-water storage tank thermostat, an outdoor temperature thermostat, a radiant/convector thermostat, and an ice inventory sensor.

5.4.1 Primary control functions

The primary control functions are as follows:

1. Space-heating demand — signaled by the room thermostat; turns on the compressor, the brine circulating pumps, and the circulating air blower.
2. Water-heating demand — signaled by the hot-water storage tank thermostat; turns on the compressor, the brine circulating pumps, and the water circulating pump. During the heating season this mode would seldom occur because the compressor operation for space heating will supply adequate hot water.
3. Space-cooling demand, chilled brine — signaled by the room thermostat; turns on the chilled-brine circulating pump and the circulating air blower.
4. Space-cooling demand, ventilation cycle — coincident signal from the room thermostat, indicating space-cooling demand, and from the outdoor thermostat indicating that the demand can be satisfied with outdoor air; operates the motor-operated dampers and turns on the circulating air blower. Outdoor air is drawn into the house (through a screened and filtered louver) and house air discharges outside to cool the house without using the chilled brine.

5.4.2 Secondary control functions

The secondary control functions, which may or may not occur during a season, are as follows:

1. Full ice bin — an ice inventory in excess of 80% during the heating season signaled by the ice inventory sensor indicates the need for supplementary heat to the ice bin. A coincident signal of available supplementary heat to the radiant/convector thermostat will cause brine to circulate between the ice bin and radiant/convector to melt ice. The 80% setting leaves an adequate safety factor for bad weather.
2. Exhausted ice bin — an exhausted ice inventory during the cooling season, signaled by an ice bin temperature above 32°F, indicates the need for supplementary compressor operation. A coincident signal that the radiant/convector thermostat is below 80°F turns on the compressor and the brine circulating pumps, permitting the system to refrigerate the bin and reject heat to the radiant/convector panel.

5.4.3 Control modes

The ACES can be operated in different control modes to accommodate different requirements of the climate, the structure, the occupants, and the choice of installed equipment. The system in the demonstration house was designed to provide full annual cycle heating and cooling for this house in this climate; however, certain control and equipment flexibility is provided to permit simulating operation in other climates and with different system arrangements.

The system is being operated in the design mode. During the heating season, all of the heat required for space heating and water heating is withdrawn from the ice storage bank with no contribution from the radiant/convector unless, through unusual weather or system operation, the ice inventory exceeds 80% of the capacity of the ice storage bank.

In that event, solar energy will be collected until the ice inventory is reduced below 80%. During the cooling season, the space cooling is accomplished by circulating the chilled brine from the ice storage bank coils through the fan coil. If the ice is exhausted before the end of the cooling season, as is probable in a year having a mild winter and a hot summer, the compressor is permitted to operate at night when the air temperature falls below a predetermined temperature of about 80°F to chill the water and perhaps make ice. The thermal storage capacity of the water between 35 and 45°F is 1.5×10^6 Btu, which is sufficient to supply about a week of cooling.

5.5 The Data Acquisition System

A DAS has been installed to monitor and record the weather data, the temperatures in the houses, and the energy consumption and performance of the space conditioning systems. The DAS takes information from temperature, pressure, flow, watt-hour, etc., sensors; processes some of it to obtain hourly integrations of energy flows; prints a real-time, on-site summary, hourly or on demand; and records the hourly summary on magnetic tape. The real-time printouts are used for on-site system analyses, trouble shooting, and system and component performance determinations. The magnetic tapes are transmitted each week into a major computer system for permanent storage and for processing by several data reduction programs to produce graphic and tabular analyses of the performance of the several systems.

5.5.1 Data sensing and on-site processing

A description of the general nature of the data sensing and on-site processing is given to aid in understanding the DAS printouts.

1. All circulating system temperatures are read at the instant of printout and are therefore meaningful only if that portion of the system is in operation at the time of reading. These temperatures are used primarily for real-time system analyses, and the instantaneous temperatures are therefore the ones desired.

2. The indoor and outdoor temperature and wind direction measurements similarly are instantaneous. The wet bulb and dew point are directly sensed (not calculated) with appropriate instruments.
3. Rain is in units of inches to date and is reset at the beginning of each month.
4. All integrated measurements described below are printed in terms of quantity per unit of time since the counters were zeroed. The time unit is normally 1 hr; however, when the DAS is commanded to print at shorter time intervals, then the integration is for the elapsed time from the preceding zero. The printout notes the event and the time of each zeroing.
5. SUN is directly measured by a pyranometer facing 10° west of south and inclined 45° above horizontal. This orientation was chosen to reproduce the orientation of the collectors on the adjacent solar house. The printout is the integration for the preceding time in units of Btu/ft^2.
6. All circulating system heat flows are calculated from the temperature differential, volumetric flow, and fluid properties, and integrated for the time period. The flows are printed in Btu.
7. The electrical consumptions are integrated from the watt-hour meter disk rotation and are then converted to and printed out as Btu.
8. The heat balance entries are calculated from the appropriate heat flows and electrical inputs to compressor, pumps, and fan.

5.5.2 DAS printout item description

A DAS printout is reproduced in Fig. 5.3. The meaning of the symbols is described below.

ACES Performance Data

COM	Compressor	Electrical consumption converted to Btu
FAN	Fan	
PCL	Cooling-circuit pump	

```
                    ACES/SOLAR HOUSES - HOURLY LOG
  7/  7/77   90002 HRS  FILE 42        COUNTERS ZEROED  7/  7/77   90012 HRS
****ACES PERFORMANCE DATA****
BNI   2720  COM   4330  FAN   1380  PCL    510  PHT    210  PHW    130
FCO      0  HWO   2470  FCI  16450  BNO   9110  SPI      0  SPO  11060
QIN  23925      QOUT  22640     QIN/QOUT   1.06
HTG COP   0.65  CLG COP  15.83  ECYCMIN   0.00
CH PWR   4520
****ANALOG DATA****
BNI  60.31  BNO  55.55  FCI  55.76  FCO  60.21
SPI 104.44  SPO  85.65  HWI 136.32  HWO 145.90
HDB  77.67  HWB  72.52  BNL  46.24  BNH  54.23  SPL  79.70
TONS ICE   0.00
SHDB  77.20  SHWB  72.18  CHDB  78.41
****WEATHER DATA****
TODB  77.3  TODEW 74.4  SUN  41.70  WDR 281  RAIN 0.00
PRESS 29.13 AV WSPD  1.39
```

Fig. 5.3. On-site DAS hourly log for 0800–0900, July 7, 1977. The system accomplished space cooling (FCI = 16,450), hot water heating (HWO = 2470), and night heat rejection (SPO = 11,060) each for a portion of the hour.

PHT	Heating-circuit pump	Electrical consumption converted to Btu
PHW	Domestic hot-water-circuit pump	
HWO	Heat delivered to domestic hot water	
FCI	Heat into the circulating system from the fan coil (space cooling)	
FCO	Heat out to the fan coil from the circulating system (space heating)	
BNI	Heat into the system from the thermal storage bin	
BNO	Heat out to the bin from the system	
SPI	Heat into the system from the solar panel	
SPO	Heat out to the solar panel from the system	
QIN	Summation of heat and energy put into the brine system	
QOUT	Summation of heat taken out of the brine system	
QIN/QOUT	The heat balance	
HTG COP	The useful heat delivered divided by the electrical input	
CLG COP	The useful cooling delivered divided by the electrical input	
ECYCMIN	The number of minutes of operation of the ventilation cycle	
CHPWR	Electrical consumption of the heat pump in the control house	

Analog Data

BNI	Temperature of circulating stream entering the bin
BNO	Temperature of circulating stream leaving the bin
FCI	Temperature of circulating stream entering the fan coil
FCO	Temperature of circulating stream leaving the fan coil
SPI	Temperature of circulating stream entering the solar panel
SPO	Temperature of circulating stream leaving the solar panel
HWI	Temperature of the potable water entering the water-heating condenser
HWO	Temperature of the potable water leaving the water-heating desuperheater
HDB	Dry-bulb temperature in the ACES house
HWB	Wet-bulb temperature in the ACES house
BNL	Water temperature at the bottom of the thermal storage bin

BNH	Water temperature at the middle of the thermal storage bin
SPL	Temperature of a dummy (noncirculating) solar panel element
TONS ICE	Quantity of ice in the thermal storage bin
SHDB	Dry-bulb temperature in the solar (University of Tennessee) house
SHWB	Wet-bulb temperature in the solar (University of Tennessee) house
CHDB	Dry-bulb temperature in the control house

Weather Data

TODB	Outdoor dry-bulb temperature
TODEW	Outdoor dew point
SUN	Solar insolation
WDR	Wind direction, degrees from north
RAIN	Monthly accumulation in inches
PRESS	Atmospheric pressure, inches of mercury
AV WSPD	Hourly average of wind speed, miles per hour

5.6 The Loads Simulation Packages

The experimental and demonstration functions of the ACES and control houses preclude anything approaching normal occupancy, and they are therefore not occupied. The normal loads imposed on a space-conditioning and water-heating system by occupancy are simulated or accounted for by three loads devices.

5.6.1 Hot water

A total of 75 gal/day of hot water is drawn on the following schedule:

0700 h	25 gal
1200 h	15 gal
1900 h	20 gal
2300 h	15 gal

5.6.2 Electrical consumption and people

The normal use of lighting and appliances releases heat into the residence which reduces the heating load and increases the cooling load. This normal internal load is simulated with electrical resistance heaters which consume an average of 1 kW used on the following schedule:

0000–0700 h	500 W
0700–0900 h	1500 W
0900–1700 h	1000 W
1700–2200 h	1500 W
2200–2400 h	1000 W

Occupancy by a family of four is assumed with an occupancy schedule of:

0800–1700 h	2 people
1700–0800 h	4 people

The heat released by this occupancy is simulated by additional resistance heat.

Both houses contain operating refrigerators, and the ACES house contains the DAS. Appropriate deductions are made in the simulators to adjust for these loads.

5.6.3 Infiltration due to door opening

It is assumed that the frequency and duration of door openings by a family of four will be approximated by visitors during the regularly scheduled tours and numerous special visitations. The doors are instrumented to record the number of openings and the duration of open time so that the assumption can be checked and corrected if necessary.

REFERENCE

1. E. C. Hise, J. C. Moyers, and H. C. Fischer, *Design Report for the ACES Demonstration House*, ORNL/CON-1 (October 1976).

APPENDICES

Appendix A. FIRST YEAR CHRONOLOGY

A brief review of the major events during the first year follows. A more complete chronology will be made in the Construction and Startup Report to be published at a later date.

1976

July	The house construction was completed and the mechanical system installed. The mechanical system was placed in operation, accomplishing night heat rejection and making ice, and air conditioning as needed from the stored ice. The house was dedicated and officially opened July 30.
August	Mechanical system continued in normal operation. Energy and material balances were made around the system and components to measure system and component performance. The two brine circulating pumps were shown to have a "wire to water" efficiency of 8 and 10%. "Wire to water" efficiency is the product of the pump efficiency and the motor efficiency.
September	Mechanical system continued in normal operation. Identified and ordered replacement circulating pumps with "wire to water" efficiency of 25 to 30%. Identified and ordered a new design, limited production, high-efficiency compressor.
October	Mechanical system continued in normal operation. Installed the new compressor and pumps and repeated all performance measurements.
November	The data acquisition terminal was placed in the house and the sensor cable connections begun. The weather

(November)	station was received and installed. The finned aluminum heat exchange surface in the bin developed several leaks. Analyses indicated that the localized corrosion cells had been initiated or contributed to by a previously unsuspected biological growth. The aluminum tubing was replaced with 1-in. polyethylene tubing. Chlorination and periodic culture sampling of the bin water were initiated.
December	The vinyl ice bin liner failed due to improperly fitted corners. The corners were reinforced and repaired. Installation of the DAS neared completion.

1977

January	The DAS began on-line operation. The refrigerant evaporator (brine chiller) developed a leak, and a replacement unit was installed.
February	The desuperheater and condenser for domestic water heating were recircuited as shown in Fig. 5.2 for more efficient operation. Complete energy balances and performance measurements were made using the DAS on-line.
March	The loads packages were installed to simulate the internal loading and hot-water usage of normal occupancy. The DAS developed an intermittent failure that interrupted data collection.
April	DAS computer failure was analyzed and repaired. All systems in normal operation. Performance testing and system tuning continued.
May	Normal operation.

June	Detected a slight mixing of heated and cooled brine streams resulting from several three-way valves failing to completely seal. Installed later-design trim and valves supplied by manufacturer.
July	Normal operation.

Appendix B. COMPONENT SPECIFICATIONS AND DESIGN DATA

1. GENERAL

The following information is as presented in the *Design Report for the ACES Demonstration House*,[1] except where later field changes were made as noted.

The mechanical installation isometric, Fig. B.1, indicates the location and relative size of the components and the routing of the piping.

2. THE MECHANICAL PACKAGE

The mechanical package was custom assembled by Bohn Aluminum and Brass Company. It contains the heat pump (the refrigerant compressor), the desuperheater, the water-heating condenser, the space-heating condenser, the brine cooler (evaporator), the circulating pumps, and the control valves. The heat exchangers are steel and aluminum construction, and there are no copper-bearing materials in any brine circuits:

Cooling capacity	25,000 Btu/h at 25°F brine outlet
Heating capacity	30,000 Btu/h at 25°F brine outlet, 105°F condensing
Dimensions	5 ft long × 3-1/2 ft high × 3 ft deep
Compressor design load	3500 W
Refrigerant	R-22
Total pump load	455 W

During the course of the year as more efficient components were identified, the compressor and two of the circulating pumps were replaced. The changes resulted in the following operating characteristics:

Cooling capacity	21,000 Btu/h
Heating capacity	28,000 Btu/h
Compressor design load	2000 W
Total pump load	390 W

ORNL-DWG 76-8807

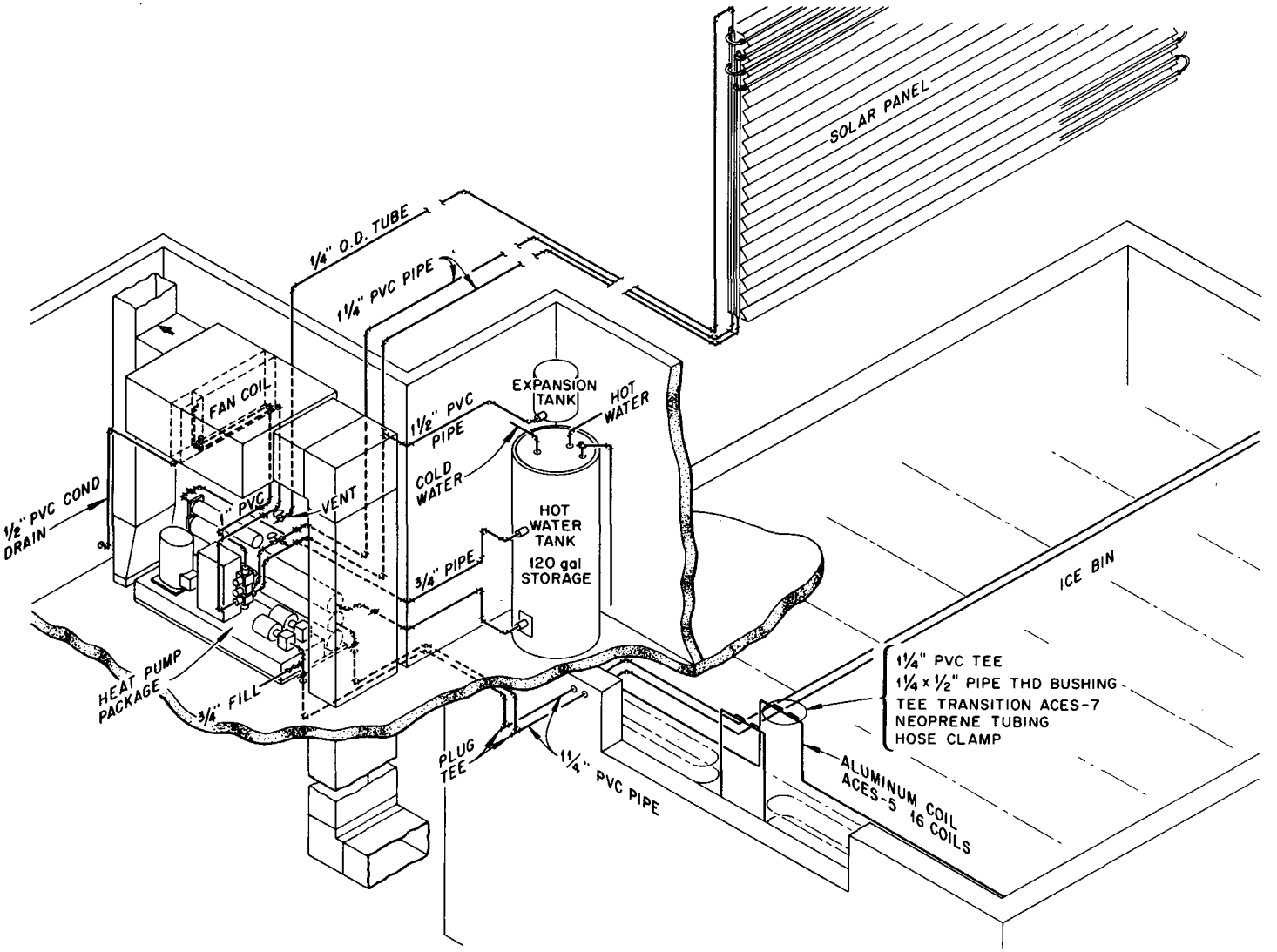

Fig. B.1. System isometric.

3. ICE STORAGE TANK

The ice storage tank is constructed as an integral part of the basement-foundation of the house. The floor slab and footings are a continuous pour (no expansion joint). Poured into the footings are vertical reinforcing bar stubs, a 2 × 4 in. key to tie the vertical walls, and a 1 × 4 in. rubber water stop. The walls are a reinforced concrete monolith poured in Foam Form. The floor and walls are finish coated with an asphalt-urethane preparation (Lion Oil NOKORODE 705) primarily to prevent chemical interaction between the concrete and water and as an added waterproofing. The top or cover is the floor above, with 12-in. fiberglass batts between the floor joists:

Dimensions	19 × 17.5 × 7.5 ft fill level (2500 ft^3)
Maximum permissible ice inventory	2000 ft^3
Total heat of fusion available	16.5×10^6 Btu
Sensible heat between 45 and 32°F	2.1×10^6 Btu
Average daily cooling load range	100,000—250,000 Btu/day

4. ICE BUILDING COILS

The coils are constructed of an extruded aluminum tube (Fig. B.2) having an outside diameter of 1/2 in. and an integral axial fin with an overall width of 3 in. The tubing is mounted on support structures that extend from the tank floor to the floor joists overhead to support the weight or resist the bouyancy as the case may be:

Tubing quantity	1950 ft
Number of parallel circuits	16
Center distance	13 in.
Maximum permitted ice diameter	13 in.

ORNL-DWG 76-8806

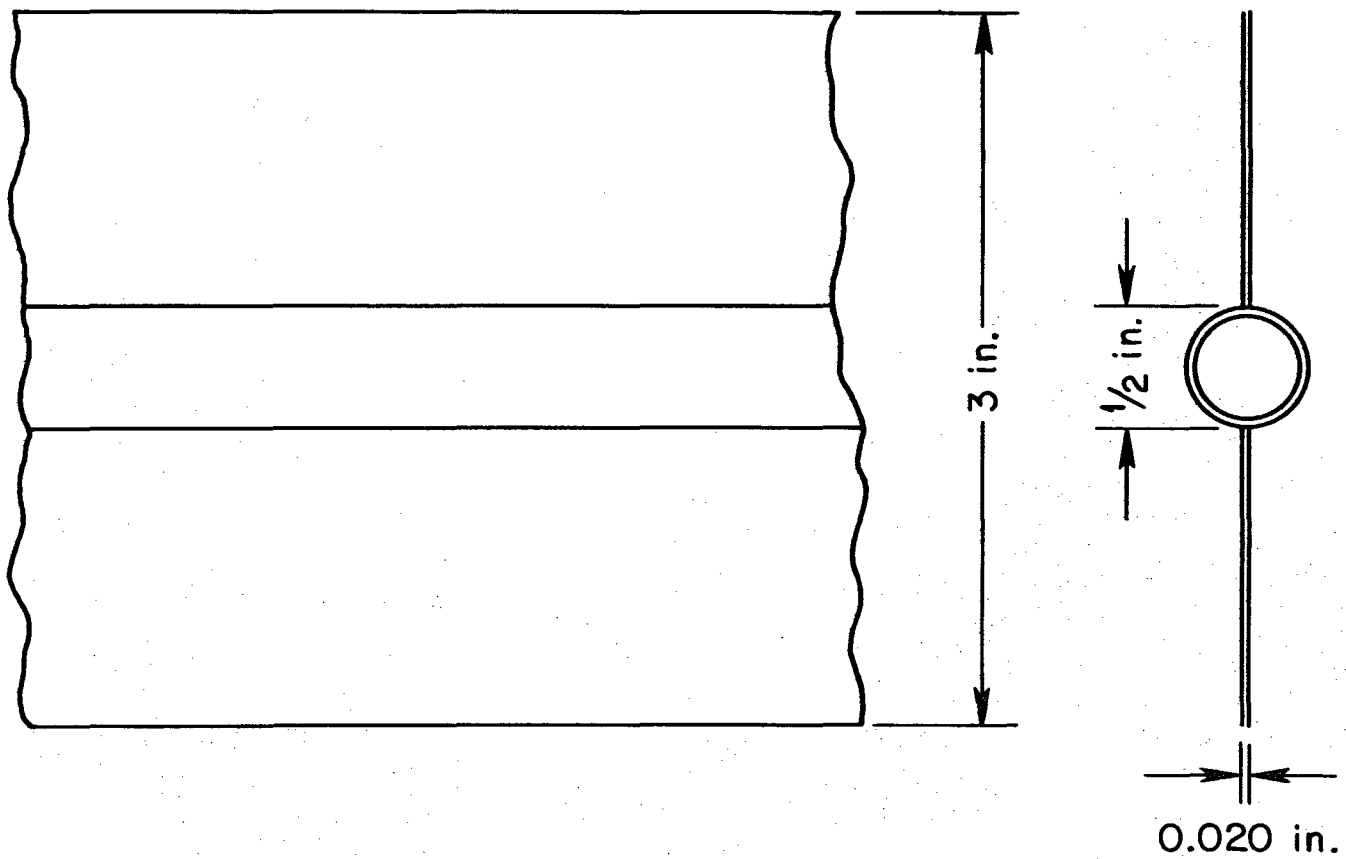

Fig. B.2. Radiant/convector and ice coil tubing.

Thermal loading	12 Btu/ft·h
Brine-to-water ΔT at maximum ice diameter	9°F

The aluminum tubing suffered a number of perforation failures in November 1976, induced by microbiological action, and was replaced by 1-in.-OD × 0.093-in.-wall low-density polyethylene tubing. The design calculations for this tubing predicted about a 1°F greater brine-to-water ΔT. No change in performance has been observable.

5. RADIANT/CONVECTOR PANEL

The radiant/convector panel occupies a south-facing vertical break in the roof about 8 ft high × 34 ft long (see Fig. 5.1). The tubing is that shown in Fig. B.2. The support members are 2 × 6 in. wood studs cut to a sawtooth form to orient each tube fin 30° from the vertical (normal to the noon sun at winter solstice) and separate them vertically to prevent shadowing. The tubes are nailed to the studs through the fins. The supply and return headers are at one end and return bends at the other to form 13 parallel circuits. No glazing or insulation is installed, and free convective airflow is permitted through the panel. The tubes are painted a flat brown:

Number of tubes	26
Length of tubes	32 ft
Radiant surface area	208 ft^2
Convective surface area	415 ft^2
Design convective coefficient	2 $Btu/ft^2 \cdot h \cdot °F$
Average incident solar energy Dec. 15–Feb. 15	9000 $Btu/ft^2 \cdot week$
Average convective energy Dec. 15–Feb. 15	240 $Btu/ft^2 \cdot week$
Average total available energy Dec. 15–Feb. 15	9480 $Btu/ft^2 \cdot week$, or 2×10^6 Btu/week for the whole panel
Average house requirement	1.2×10^6 Btu/week

6. FAN COIL

The fan coil unit located in the main air duct contains the finned tube brine coil, the circulating air blower, and the motorized damper for the ventilation cycle:

Face area	2.5 ft^2
Number of rows	6
Liquid-side pressure drop	6 ft H_2O
Liquid flow	6–8 gpm
Airflow	1200 ft^3
Air-side pressure drop	0.25 in. H_2O

7. BRINE

The circulating brine is 20% methanol and 80% water:

Freezing point	4.5°F
Boiling point	182°F
Density	60.4 lb/ft^3
Specific heat	0.97 Btu/lb·°F

Appendix C. SAMPLE CALCULATIONS AND CONDITIONS

1. SAMPLE CALCULATION OF FULL-LOAD SPACE HEATING PLUS WATER HEATING

May 27, 1977

0800 to 0900

Electrical energy inputs (Btu)		
Compressor	COM =	8030
Fan	FAN =	1340
Pumps	PCL =	570
	PHT =	460
	PHW =	270
Heat input (Btu)		
Into the system from the bin	BNI =	17,750
Heat outputs (Btu)		
Into hot water from the system	HWO =	3,900
Into the fan coil from the system	FCO =	19,900
Direct space heating from motor losses		
Fan motor	FAN =	1340
Pumps	0.75(PCL + PHT + PHW) =	975
Compressor	0.25(COM) =	2007

It is assumed that all of the electric energy to the fan heats the circulating air, 75% of the energy to the pumps heats the air (25% is deposited in the circulating brine), and 25% of the energy to the compressor leaves the shell to heat the air (75% is deposited in the refrigerant).

Energy balance around the refrigerant and brine system:

Q_{in} = BNI + 0.75(COM) + 0.25(PCL + PHT + PHW) = 24,097 Btu

Q_{out} = FCO + HWO = 23,800 Btu

Q_{in}/Q_{out} = 1.01

Coefficient of performance:

Energy input = COM + FAN + PCL + PHT + PHW = 10,670 Btu

Useful heat delivered = FCO + HWO + FAN + 0.25(COM) + 0.75(PCL + PHT + PHW) = 28,122 Btu

COP = useful heat/energy in = 2.64

2. SAMPLE CALCULATION OF SPACE COOLING WITH STORED ICE

May 31, 1977

1200 to 1300

Electrical energy inputs (Btu)		
Fan	FAN =	1410
Pump	PCL =	480
Heat inputs to the system (Btu)		
From the fan coil	FCI =	28,910
From the pump	0.25(PCL) =	120
Heat outputs from the system (Btu)		
Into the bin	BNO =	30,980
Into the conditioned space	FAN =	1,410
	0.75(PCL) =	360

Energy balance around the brine system:

Q_{in} = FCI + 0.25(PCL) = 29,030 Btu

Q_{out} = BNO = 30,980 Btu

Q_{in}/Q_{out} = 1.07

Coefficient of performance:

Energy input = FAN + PCL = 1,890 Btu

Useful cooling = FCI - FAN - 0.75(PCL) = 27,140 Btu

COP = useful cooling/energy input = 14.4

3. SAMPLE CALCULATION OF FULL-LOAD WATER HEATING

July 7, 1977

1100 to 1200

Electrical energy inputs (Btu)

Compressor	COM =	8780
Pumps	PCL =	590
	PHW =	270

Heat inputs (Btu)

Into the system from the bin	BNI =	23,170
From the pumps	0.25(PCL + PHW) =	215
From the compressor	0.75(COM) =	6,585

Heat output (Btu)

Into the hot water	HWO =	29,650

Energy balance around the refrigerant system:

Q_{in} = BNI + 0.75(COM) + 0.25(PCL + PHW) = 29,970 Btu

Q_{out} = HWO = 29,650 Btu

Q_{in}/Q_{out} = 1.01

Coefficient of performance:

Energy input = COM + PCL + PHW = 9,640 Btu

Useful hot water output = HWO = 29,650 Btu

COP = useful output/energy input = 3.07

INTERNAL DISTRIBUTION

1. L. A. Abbatiello
2. S. I. Auerbach
3. M. Baker
4. V. D. Baxter
5. S. E. Beall
6. D. J. Bjornstad
7. J. R. Buchanan
8. R. S. Carlsmith
9. W. S. Chern
10. T. J. Clifford
11. F. A. Creswick
12. R. M. Davis
13. J. G. Delene
14. J. R. DeMonbrun
15. C. R. Eichelberger
16. H. C. Fischer
17. W. Fulkerson
18. V. O. Haynes
19. N. E. Hinkle
20. E. A. Hirst
21. E. C. Hise
22. A. S. Holman
23. L. Howard
24. J. R. Jackson
25. P. L. Johnson
26. S. I. Kaplan
27. D. Kaserman
28. R. S. Livingston
29. H. M. Long
30–49. P. M. Love
50. J. W. Michel
51. R. E. Minturn
52. W. R. Mixon
53. J. C. Moyers
54. E. A. Nephew
55. D. L. O'Neal
56. H. Postma
57. M. W. Rosenthal
58. T. H. Row
59. J. M. Schreyer
60. G. G. Slaughter
61. R. L. Spore
62. E. G. Struxness
63. D. B. Trauger
64. G. U. Ulrikson
65. D. J. Walukas
66. T. J. Wilbanks
67. H. E. Zittel
68. A. Zucker
69. Biology Division Library
70–71. Central Research Library
72. Document Reference Section
73. Laboratory Records – RC
74–76. Laboratory Records Department
77. ORNL Patent Office

EXTERNAL DISTRIBUTION

78. Mr. Ben Adams, President, Crouch and Adams, Inc., 106 Administration Road, Oak Ridge, Tennessee 37830
79. Mr. Arthur G. Baitz, Assistant Vice President and Director of Planning, Robertshaw Controls Company, 1701 Byrd Avenue, P. O. Box 26544, Richmond, Virginia 23261
80. Mr. Hamp Barnett, Power Research Staff, Tennessee Valley Authority, 1310 Commerce Union Bank Building, Chattanooga, Tennessee 37401
81. Mr. J. Temple Bayliss, Virginia Energy Office, 823 East Main Street, Richmond, Virginia 23219
82. Mr. Albert Bedinger, Sr., 1128 Keowee Avenue, SW, Knoxville, Tennessee 37919
83. Mr. Albert Bedinger, Jr., Environment Center, University of Tennessee, 346 South Stadium Building, Knoxville, Tennessee 37916

84. Mr. Ed Bottum, President, Refrigeration Research, Inc., 525 North Fifth Street, Brighton, Michigan 48116
85. Mr. Irwin G. Cantor, President, Irwin G. Cantor/Consulting Engineers, 919 Third Avenue, New York, New York 10022
86. Dr. Louis-Marie Chounet, Batiment 200, Lab. de L'Accelerateur Lineaire, 91405 Orsay, France
87. Mr. Gordon D. Duffy, Executive Editor, Air Conditioning News, 700 East Maple, Birmingham, Michigan 48011
88. Mr. Nathan R. Feldman, Director, Mechanical Engineering, Veterans Administration, Office of Construction, Lafayette Building, 811 Vermont Avenue NW, Washington, D.C. 20420
89. Mr. Leo Foster, Elliott Lewis Corporation, 2301 Cherry Street, Philadelphia, Pennsylvania 19103
90. Mr. W. F. Hagen, President, Turbo Refrigerating Company, Shady Oaks Drive, Expressway Industrial Park, Post Office Box 396, Denton, Texas 76201
91. Mr. Richard P. Hankins, Jr., Torrence, Dreelin, Farthing & Buford, Inc., Suite 600, Seaboard Building, Richmond, Virginia 23230
92. Rear Admiral John Harllee, U.S.N. (Ret.), Director of Field Services, Americans for Energy Independence, Suite 1405, 1500 Wilson Boulevard, Arlington, Virginia 22209
93. Mr. William A. Knox, 111 Edgewood Avenue, Columbia, Missouri 65201
94. Mr. Jim Leggoe, Philadelphia Electric Company, 2301 Market Street, Philadelphia, Pennsylvania 19101
95. Mr. Herbert Lindahl, Manager, Advanced Engineering, Bohn Aluminum and Brass Corporation, Heat Transfer Division, 1625 East Voorhees Street, Danville, Illinois 61832
96. Mr. John F. Lowrey, Manager, Applications Department, Philadelphia Electric Company, 2301 Market Street, Philadelphia, Pennsylvania 19101
97. Mr. Calvin D. MacCracken, President, Calmac Manufacturing Corporation, 150 South Van Brunt Street, Englewood, New Jersey 07631
98. Mr. Jim McCallum, Executive Editor, Air Conditioning and Refrigeration Business, 614 Superior Avenue West, Cleveland, Ohio 44113
99. Mr. Patrick C. McKelvey, Instructor/Coordinator, Electrical Power Technology, Cincinnati Technical College, 3520 Central Parkway, Cincinnati, Ohio 45223
100. Mr. Ben Miller, Manager of Engineering, REMCOR Products Company, 321 East Grand Avenue, Chicago, Illinois 60611
101. Mr. Kenneth J. Nemeth, Executive Director, Southern Interstate Nuclear Board, 7 Dunwoody Park, Suite 104, Atlanta, Georgia 30341
102. Mr. Neno Nenov, Senior Engineer, Process and Product Development Department, Union Carbide Corporation, Linde Division, Post Office Box 44, Tonawanda, New York 14150
103. Mr. Aharon Nir, Isotope Department, Geoscience Group, Weizmann Institute, Rehovot, Israel

104. Mr. J. A. O'Brien, Director, Bedford Administrative Operations, The Mitre Corporation, Bedford, Massachusetts 01730
105. Mr. A. H. Rosenfeld, Lawrence Berkeley Laboratory, Building 50B, Room 5246, Berkeley, California 94720
106. Dr. David P. Ross, Senior Scientist, Energy Resources and Conservation Division, Science Applications, Inc., 2028 Powers Ferry Road, Suite 260, Atlanta, Georgia 30339
107. Dr. Mason H. Somerville, Manager, Engineering Experiment Station, University of North Dakota, Grand Forks, North Dakota 58202
108. Mr. Richard F. Todd, Director of Education and Training, National Environmental Systems Contractors Association, 1228 Seventeenth Street NW, Washington, D.C. 20036
109. Mr. Robert G. Werden, Werden & Associates, Inc., Post Office Box 414, Jenkintown, Pennsylvania 19046
110. Institute of Energy Analysis, ORAU Library
111. Research and Technical Support Division, DOE-ORO
112–138. Technical Information Center, DOE, Post Office Box 62, Oak Ridge, Tennessee 37830
139–600. External Energy Conservation Distribution Mailing List and Energy Conservation Office (9102-1, Room 2)

☆ U.S. GOVERNMENT PRINTING OFFICE: 1978-748-189/414