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ACES Tests at the Tech Site: 1981

Van D. Baxter

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ACES TESTS AT THE TECH SITE: 1981

Van D. Baxter

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OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
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RELATED REPORTS IN THIS SERIES

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

J. C. Moyers et al., *Design Report for the ACES Demonstration House,* ORNL/CON-1 (October 1976).

E. A. Nephew et al., *The Annual Cycle Energy System: Initial Investigations,* ORNL/TM-5525 (October 1976).

I. G. Cantor Company, *Investigation of Alternate Types of Water Storage Components to be Utilized in Conjunction with the ACES System,* ORNL/Sub-78/14233/1 (December 1977).

E. C. Hise, *Performance Report for the ACES Demonstration House August 1976 through August 1977,* ORNL/CON-19 (March 1978).

H. C. Fischer, *Development and Testing of a Single-Plate and a Two-Plate Ice-Maker Heat Pump,* ORNL/CON-21 (April 1978).

A. S. Holman and V. R. Brantley, *ACES Demonstration: Construction, Startup, and Performance Report,* ORNL/CON-26 (October 1978).

V. D. Baxter, *Intermediate Report on the Performance of Plate-Type Ice-Maker Heat Pumps,* ORNL/CON-23 (October 1978).

A. S. Holman et al., *Annual Cycle Energy System (ACES) Performance Report November 1977 through September 1978,* ORNL/CON-42 (May 1980).

M. L. Ballou et al., *MAD: A Computer Program for ACES Design Using Monthly Thermal Loads,* ORNL/CON-51 (March 1981).

R. E. Minturn et al., *ACES 1979: Capabilities and Potential,* ORNL/CON-48 (June 1980).

J. W. MacArthur, *Economic Analysis of Annual Cycle Energy Systems,* Energy Resources Center, Honeywell, Inc., ORNL/Sub-7470/1-3 (May 1980).

V. D. Baxter, *Ice-Maker Heat Pump Development: Final Report,* ORNL/CON-50 (September 1980).

L. A. Abbatiello et al., *Performance and Economics of the ACES and Alternative Residential Heating and Air Conditioning Systems in 115 U.S. Cities,* ORNL/CON-52 (March 1981).

L. A. Abbatiello, *The Annual Cycle Energy System Characteristics and Performance,* presented at the International Congress on Building Energy Management, Povoa de Varzim, Portugal (May 1980).

E. A. Nephew et al., *Theory and Design of an Annual Cycle Energy System (ACES) for Residences,* ORNL/CON-43 (May 1980).

J. L. Bledsoe et al., *Data Collection and Processing for the ACES,* ORNL/CON-59 (August 1981).

V. D. Baxter, *ACES: Final Performance Report December 1, 1978, Through September 15, 1980,* ORNL/CON-64 (April 1981).

E. A. Nephew and L. A. Abbatiello, *Performance and Economics of Eight Alternative Systems for Residential Heating, Cooling, and Water Heating in 115 U.S. Cities,* ORNL/CON-89 (November 1982).

M. H. Somerville and S. G. Penoncello, *The Design and Development of a Split Evaporator Heat Pump System,* University of N. Dakota Engineering Experiment Station, ORNL/Sub-7434/1 (December 1981).

G. H. Alexander, et al., *Development of Concepts for Low Cost Energy Storage Assemblies for Annual Cycle Energy System Applications,* Battelle Columbus Laboratories, ORNL/Sub-7800/1 (October 1981).

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ACES TESTS AT THE TECH SITE: 1981

Van D. Baxter

ABSTRACT

The Annual Cycle Energy System (ACES) in the ACES demonstration house at the Tennessee Energy Conservation in Housing complex near Knoxville, Tennessee, underwent short-term testing of the minimum ACES design concept and of several alternative energy collection/rejection modes using an outdoor fan coil unit.

Operation as a minimum ACES was about as expected. The heating coefficient of performance (COP) was about 4% less than that of a full ACES. The solar panel sizing methods used in an ACES performance modeling code were shown to be conservative for the panel type used at the demonstration house.

The system was tested in several modes with an outdoor fan coil in winter and summer. Use of the fan coil as the energy source for the space- and water-heating modes degraded system COP by about 10% over that when the ice bin was used. Ice melting can be done with a fan coil instead of a solar panel, but only if air temperatures exceed 4°C for a long enough time during the winter. Using the fan coil as the heat-rejecting mechanism instead of the panel for night heat rejection improved efficiency in that mode by 22% and improved supplemental cooling efficiency by about 5% over the previous year's performance.

1. SUMMARY

The Tennessee Energy Conservation in Housing (TECH) complex near Knoxville, Tennessee, was used for a series of short-term tests of the heating, ventilating, and air-conditioning (HVAC) system in the Annual Cycle Energy System (ACES) house to determine the validity of the minimum ACES design concept, the solar panel performance, and the effect of using an outdoor fan coil unit in lieu of a solar panel.

The ACES was modified by reducing the solar panel area to match predicted panel requirements for a minimum ACES (defined in Sect. 2) in Knoxville and by adding an outdoor fan coil unit. The system was tested in the minimum ACES mode and in the heating, cooling, and ice-melting modes using the outdoor fan coil unit in lieu of the storage bin or the solar panel.

1.1 Minimum ACES Tests

Performance in the minimum ACES mode matched predicted performance closely; the system heating coefficient of performance (COP) was 2.46 vs a predicted value of 2.42. The solar panel sizing methods of the monthly ACES design (MAD) computer program used to size the solar panel for these tests yielded a panel oversized by about 30% for the Knoxville climate. Modifying the sizing method to model more accurately the panel at the ACES house yielded a panel size about 20% larger than that needed in February 1981.

1.2 System Tests with Outdoor Fan Coil Unit

The following tests of the ACES were conducted using the fan coil unit:

1. space and water heating,
2. ice melting (both passive collection and active heat pumping),
3. night heat rejection and off-peak cooling, and
4. direct air conditioning.

1.2.1 Space and water heating

Space- and water-heating efficiency using outdoor air at air temperatures greater than 9°C as the energy source was 6 to 10% lower than that when using the ice/water storage bin. This performance reduction was caused by the additional energy consumed by the outdoor fan. Use of outdoor air in the heating season is not recommended, except possibly as a backup heating system in certain climatic areas.

1.2.2 Ice melting

The ice-melting modes tested were (1) use of the fan coil unit alone (passive) and (2) use of the coil as an energy source for the mechanical package while pumping heat to the bin (active). The passive mode test results showed that at air temperatures greater than about 4°C, the fan coil could be used in lieu of a solar panel, but at only one-half to one-tenth the efficiency because of the added fan energy consumption. The active mode was not useful for a brine-chiller mechanical package like the one used at the ACES house because the package is not properly designed for this mode of operation. In addition, the active mode was the least efficient mode tested for ice melting.

An outdoor fan coil unit is not a good substitute for the solar panel for ice melting in most northern areas. The passive mode will work, but only in regions having a sufficient number of hours with outdoor temperatures greater than 4°C.

1.2.3 Night heat rejection and off-peak cooling

When an outdoor fan coil unit is used, the steady-state efficiency in the night heat rejection mode is 22% better than when a solar panel is used, giving a COP of 1.98 vs 1.62. Three weeks of supplemental off-peak cooling operation in 1981 with the outdoor fan coil unit yielded an overall system COP of 2.34 for 4.94 GJ of space cooling and water heating energy deliveries. In 1980, the panel was used and a COP of 2.25 for 7.77 GJ resulted. An outdoor fan coil unit is therefore recommended over a solar panel for off-peak cooling with an ACES based on performance.

1.2.4 Direct air conditioning

Direct air-conditioning performance yielded a system COP of about 2.04 vs 2.34 for the off-peak cooling mode. The difference was caused by lower air temperatures during operation in the off-peak cooling mode.

2. INTRODUCTION

The Annual Cycle Energy System has been under development since 1975 by the Oak Ridge National Laboratory (ORNL); the work was sponsored by the U.S. Department of Energy (DOE). In brief, the ACES promotes energy conservation in buildings by providing space heating and cooling and domestic water heating using a unidirectional water-source heat pump; the heat pump extracts energy from an insulated storage tank, which, at the beginning of a heating season, contains only water (Fig. 1). As the heating season progresses, the water is converted to ice, which is stored to provide air conditioning when needed. The system has been extensively described, and its development has been documented in the reports listed at the front of this report.

During the cooling season, the compressor is operated only to provide hot water. The collector panel is installed and used as a supplemental energy source for the storage bin only if the demand for energy by the house over the heating season is likely to be greater than the energy that can be supplied by the latent heat of fusion of the stored water.

For greatest efficiency in an ACES installation, the interseasonal transfer of energy (i.e., storage of winter ice for summer use) must be maximized. This maximization is done by sizing the ice bin to hold either all the ice that can be made in winter or all the ice that can be used in summer, whichever is smaller. This system is called a full ACES and has been tested and evaluated for several years at the TECH site, in Knoxville, Tennessee (Fig. 2).[1,2] Results from these previous field tests have demonstrated that the full ACES performs efficiently and reliably and that computer programs written at ORNL to simulate its performance do so quite accurately.[3-5]

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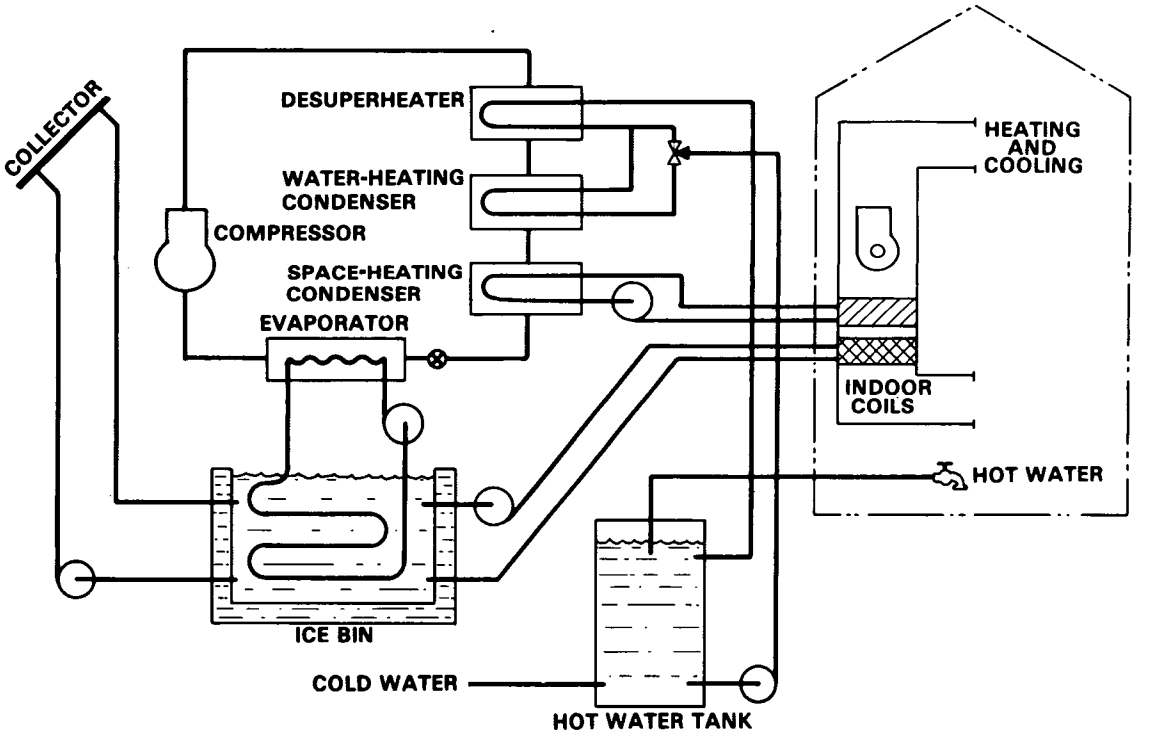

Fig. 1. Schematic of typical Annual Cycle Energy System.

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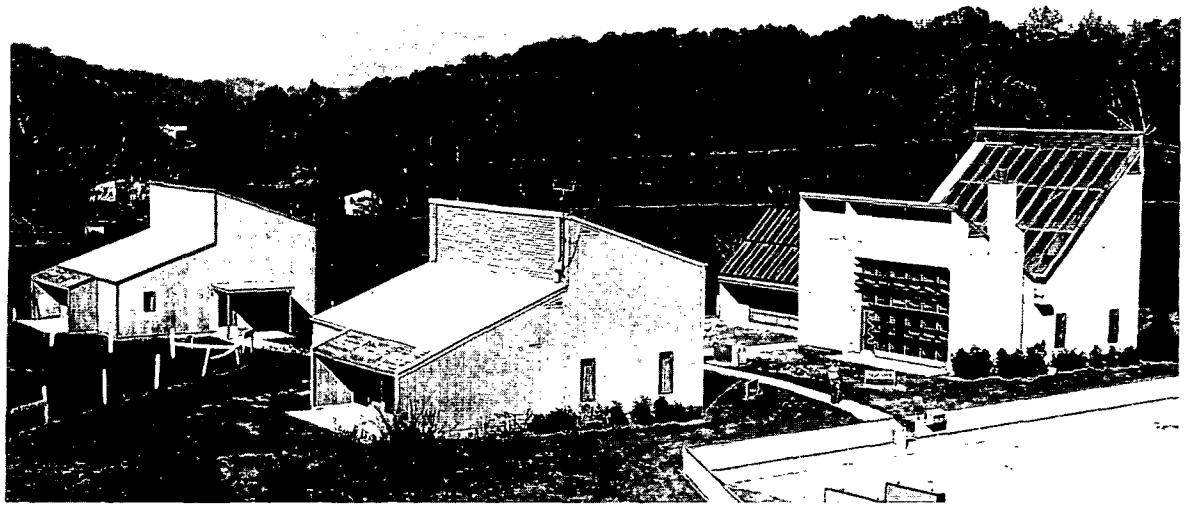

Fig. 2. Tennessee Energy Conservation in Housing complex, left to right: control house, Annual Cycle Energy System house, and solar house.

Knoxville is located in a climatic zone in which all the ice that can be generated during the heating season can be used during the cooling season. However, in many regions with colder climates, heating loads substantially exceed cooling loads and provisions must be made to remove excess ice so that it it does not exceed the capacity of the storage bin. In addition, it might be more economical, especially from a capital cost point of view, to construct a bin smaller than that needed for a full ACES and to replace the decreased energy capacity of the bin with energy from solar or other sources.[4,5] If solar collectors are to be used, a bin should be sized to supply, from latent heat alone, all the energy needed for at least two weeks of the coldest winter weather, assuming that the sun may not shine during this time. A system with a bin of this size is called a minimum ACES. Systems with bin sizes between those of the minimum and full ACES may, of course, be built.

In contrast to the full ACES, the use of low-temperature, unglazed solar panels to supplement bin energy has previously not been tested adequately. In Sect. 3 we describe some recent field tests on the use of solar panels to supply supplemental energy to the bin.

Although the ACES is very efficient, it is relatively expensive. This expense is a barrier to its widespread adoption in the residential heating and cooling market.[4,5] In many areas of the country, solar panels as a supplemental energy source are a significant cost factor. A number of sources other than solar panels might supply energy at significantly lower capital costs, and one of these, an outdoor fan coil unit, was of particular interest because it offered factory modular construction and simple field installation. Modifications to the system to accommodate such a unit and performance results from its use are described in Sect. 4.

3. MINIMUM ACES EXPERIMENT

The ice bin for the ACES test facility at the TECH site was designed to hold all the ice that could be produced in the heating season; consequently, there is no need for a supplemental source of energy. However, alternative means of cooling must be provided if all the ice made is not sufficient for all the cooling needed. In Knoxville, this was accomplished by

operating the compressor at night to chill water in the bin. The heat extracted from the bin in this operation, if not needed for heating domestic water, was dissipated by 19.4 m^2 of unglazed, extruded, finned aluminum tubes mounted on the upper south face of the ACES house. If needed, these tubes can also serve as a source of supplemental energy for the bin, collecting not only solar radiation but also picking up thermal energy from the surrounding air since the brine circulating through the tubes would be only slightly above 0°C.

There were two reasons for a series of tests using this installed radiator/convector panel. First, in the analytical work done at ORNL to calculate the expected ACES engineering performance in other geographic areas,[4-6] the parameters used to characterize the energy-gathering ability of the collectors were derived from an outdoor, but laboratory-scale, experiment done at ORNL in 1978.[6] Except for a 6-day run in December 1979,[2] these parameters have not been experimentally verified on a real house under actual weather conditions for an extended period of time. Second, the tests were run to verify under field conditions the calculational procedure used in the analytical work to estimate the weather/collector/bin interactions for the minimum ACES configuration.

3.1 System Modification and Data Acquisition System Upgrade

In the minimum ACES, as mentioned before, the bin is sized to hold enough latent heat to supply the needs of the house for two weeks of the coldest part of the winter. The radiator/convector panel is sized to prevent, during an average weather year, accumulated ice from exceeding the capacity of the bin. In operation, once the heating season begins, the panel is used to supply energy to the bin anytime the ambient temperature at the panels exceeds the bin temperature by a few degrees. Only toward the end of the heating season is ice allowed, by design, to accumulate in preparation for cooling from storage at a later date. The ORNL analytical work on ACES utilizes the computer program MAD.[3] MAD calculations for an average weather year in Knoxville indicate that the panel for a minimum ACES should have about 14.6 m^2 of surface exposed to solar radiation and ambient air, assuming that the entering fluid temperature is 0°C and that only one side of the panel is exposed to ambient air.[4,5] Assuming both sides of the panel are exposed to ambient air yields a panel size of 8.9 m^2. The panel at the ACES house does have both sides exposed to ambient air. To be conservative, however, since the panel is mounted very close to the building surface (within 50 mm), the larger size panel was chosen for these tests. The panel at the ACES house was reduced in size to 14.9 m^2 by removing 3 of the 13 panel circuits from the brine manifold (Fig. 3).

The ice bin in the ACES house, which has a volume of 69 m^3, is considerably larger than that needed for a minimum ACES in Knoxville, namely, 13.5 m^3. Rather than resort to the costly and time-consuming exercise of reducing the bin size, we opted to retain the larger bin and to see if we could prevent any net accumulation of ice during the cold months of January and February. Use of the existing bin produces an added complication in that heat leakage into the bin is a function of bin size, but, from our three years of experience with stored ice in the bin, we were able to make adequate computational corrections.

While the system was being modified to a minimum ACES design, and an outdoor fan coil unit (described in Sect. 4) was added, the data acquisition system (DAS) at the TECH site was modified both to increase the data collection and storage capacity and to streamline the transmittal of data from the site to the ORNL computer system. Collection capacity was increased from 160 to 235 points, and data storage capacity was increased from 250 kilobytes to over 6,000 kilobytes by the addition of a nine-track tape unit. Figure 4 is a photograph of the expanded DAS. Since the nine-track tapes are compatible with the ORNL computer system, the data can now be transferred directly from the tape to the IBM mass storage files. Previously the data were transferred to a cassette, then to a PDP-10 computer, and from there to the IBM system.[7] A schematic of the new data processing path is given in Fig. 5.

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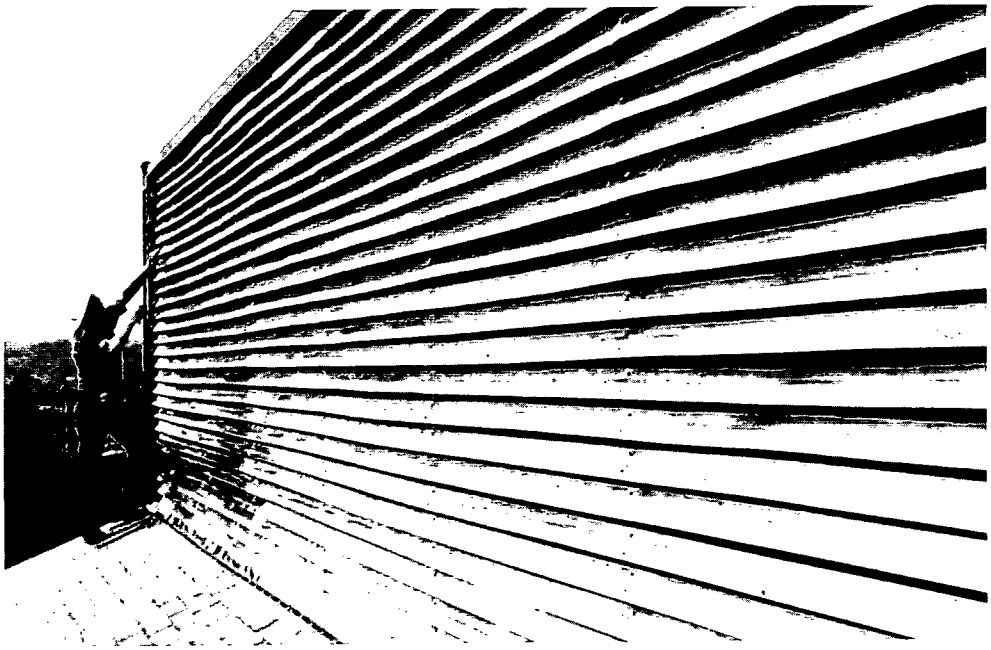

Fig. 3. Unglazed solar/convector panel at the Annual Cycle Energy System house.

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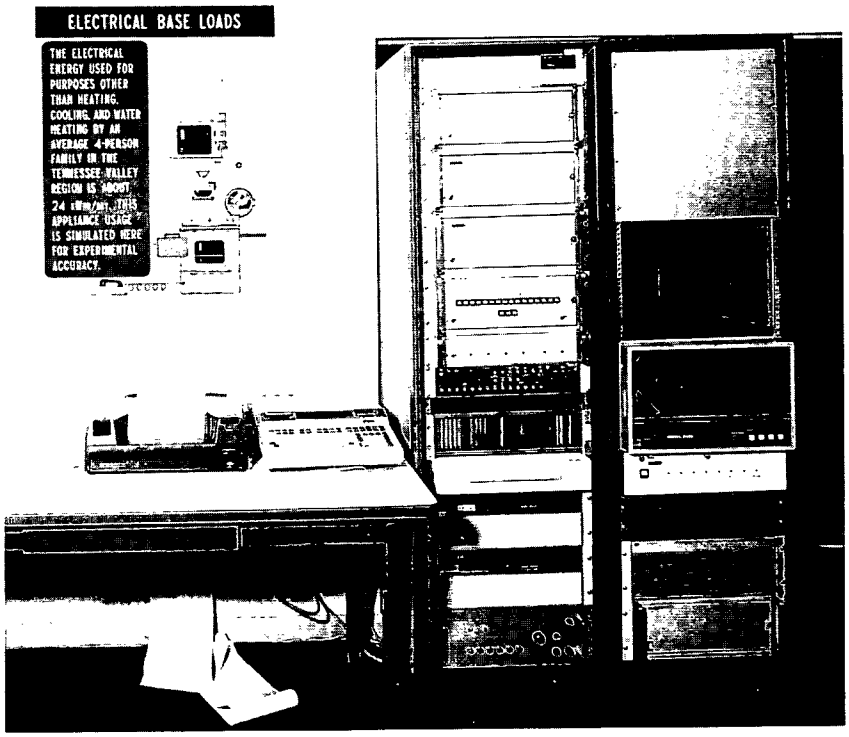

Fig. 4. Expanded data acquisition system.

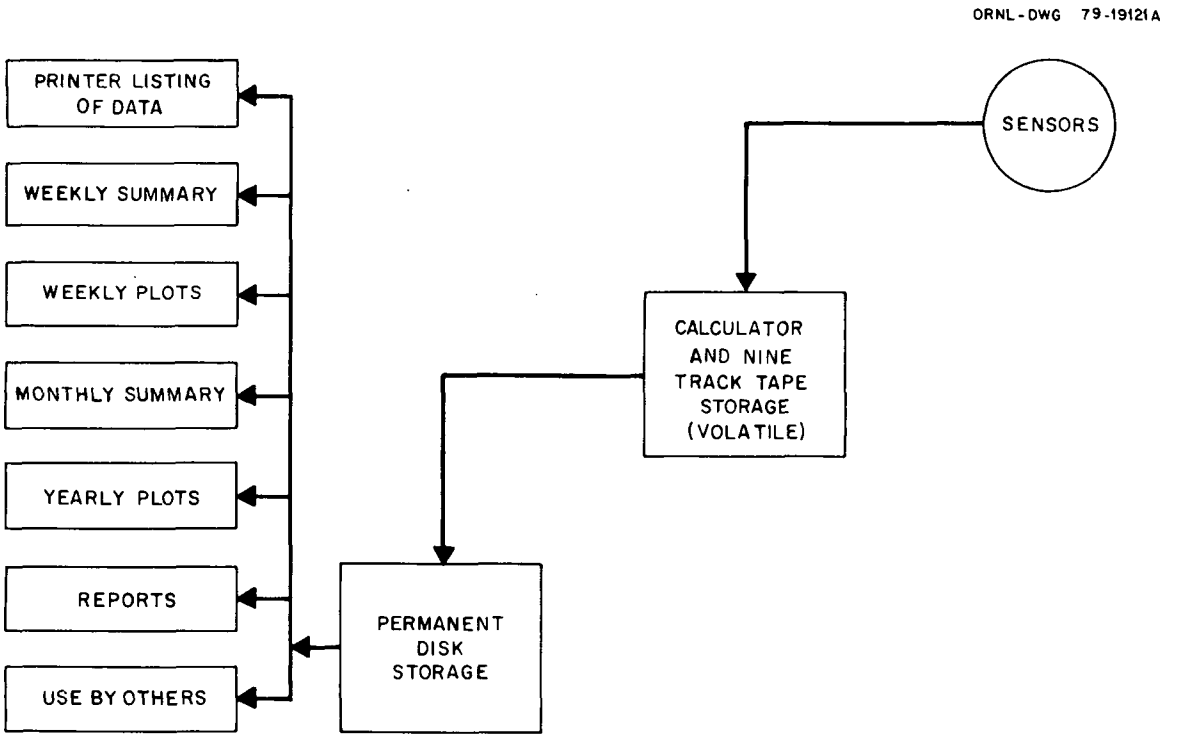

Fig. 5. New data processing path.

During this period, all flowmeters in the brine system were removed and returned to the manufacturer for refurbishing and recalibration. Delays in getting these items returned prevented start-up of the TECH site tests until late January 1981.

3.2 Experimental Results

From January 26 through March 2, 1981, the system was operated in the minimum ACES mode; the radiator/convector panel was used when ambient temperatures permitted. Table 1 shows performance results for the minimum ACES mode during this period compared with calculated results for an average-weather month of January and also shows a comparison of measured vs calculated performance results for a full ACES. January rather than February was chosen for these comparisons because the measured and calculated loads were a much better match. From Table 1 it is apparent that the extra energy consumed to operate a brine pump for collecting energy through the panel decreased the monthly COP by about 4%. Approximately 41 kWh extra electricity was consumed for this purpose during the test period compared to a predicted 35 kWh for average January weather conditions.

Figure 6 is a plot of the ACES ice inventory during the period when the panel was used. The plot shows that the ice inventory was held approximately constant over the first weeks of the test period and was decreased significantly during that latter part of the period. This indicates that the panel heat collection was more than adequate to maintain a constant ice inventory.

Table 1. Minimum ACES performance vs full ACES performance: measured and calculated

Mode	Minimum ACES		Full ACES	
	Measured January 26, 1981–March 2, 1981	Calculated average January[a]	Measured January 1980	Calculated average January[a]
Space heating,[b] GJ	8.44	9.07	8.73	9.07
Water heating, GJ	1.93	1.81	1.67	1.81
Electricity use, kWh				
For space and water heating	1131	1185	1122	1185
For solar collection	41	35	0	0
Total	1172	1220	1122	1185
COP				
Without solar collection	2.55	2.55	2.57	2.55
Overall	2.46	2.48	2.57	2.55

[a]L. A. Abbatiello et al., *Performance and Economics of the ACES and Alternative Residential Heating and Air Conditioning Systems in 115 U.S. Cities*, ORNL/CON-52 (March 1981).

[b]Discrepancy in calculated vs measured loads is accounted for by differences in average vs actual weather conditions.

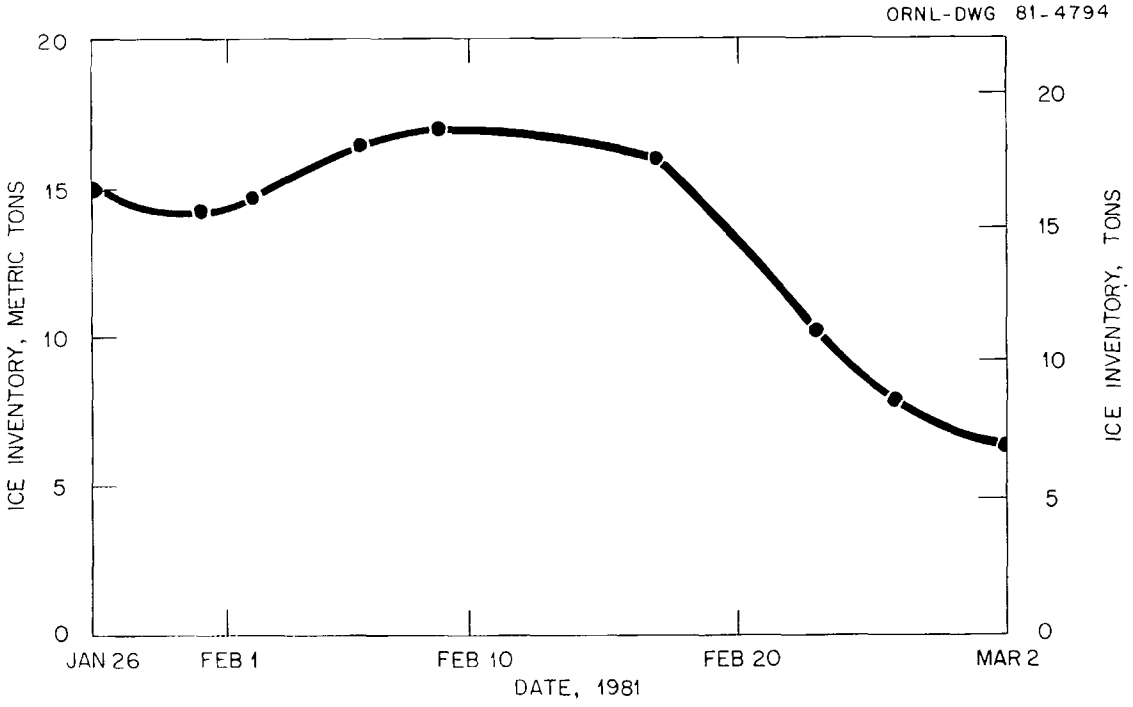

Fig. 6. Ice inventory: January 26, 1981, through March 2, 1981.

3.3 Verification of Panel Sizing Methodology

The panel sizing methods used by the MAD program were developed from a series of tests run on a test panel at Oak Ridge, Tennessee.[6] Data obtained were fitted to an expression, recommended by Bliss, of the form[8]

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

where

Q/A	=	panel heat flux (W/m^2),
F_r	=	panel efficiency factor,
α	=	absorptivity for solar radiation,
I	=	solar radiation intensity (W/m^2),
U	=	convection coefficient ($W \cdot m^{-2} \cdot K^{-1}$),
T_a	=	ambient air temperature (°C),
T_b	=	entering brine temperature (°C),
R	=	long wave reradiation heat loss [assumed = 47.3 W/m^2 (ref. 9)].

Analysis of the data obtained from the test panel showed that the convection coefficient and panel efficiency factor were well represented by the correlations

$$U = 1.443 \times S + 5.177 , \tag{1}$$

and

$$F_r = 0.941 - 0.005 \times U , \tag{2}$$

where S is the wind speed in kilometers per hour.

Panel performance data taken during the minimum ACES test, when analyzed using the Statistical Analysis System (SAS) program,[10] yielded the following correlations for U and F_r:

$$U = 1.127 \times S + 6.750, \tag{3}$$

and

$$F_r = 0.945 - 0.006 \times U . \tag{4}$$

The SAS analysis yielded correlations of the same form with constants of the same magnitude as those obtained from the original tests.

Using the actual February 1981 weather data and Eqs. (1) and (2), the MAD program was used to simulate the minimum ACES performance. Table 2 illustrates the measured panel performance for February 1981 vs average calculated February performance.

To have maintained the ice inventory at a constant level, it would have been necessary to supply 5.02 GJ from the panel and from bin heat leakage. The panel alone supplied 6.34 GJ, causing a calculated reduction in ice inventory of 7.7 t (8.49 tons). Measured ice loss, illustrated in Fig. 6, was 8.0 t (8.75 tons).

The ACES house ice bin is much larger than necessary for a minimum ACES, thus causing the disparity between measured and calculated heat leakage. Apart from heat leakage, however, the panel alone supplied about 30% more energy than necessary to maintain a constant ice inventory.

Table 2. Measured vs calculated solar panel performance for a minimum ACES in February 1981

	Measured (panel area, 14.9 m^2)	Calculated[a] (panel area, 14.4 m^2)
Loads, GJ		
Heating	6.71	5.50
Water heating	1.55	1.65
Supplemental heat needed for constant ice inventory, GJ	5.02	4.35
Actual heat input, GJ		
From panel	6.34	4.24
From leakage	1.27[b]	0.11
Ice inventory change, t	−7.72	0
Heating degree-days, °C·d		
February 1981	356[c]	
February average	350[c]	

[a]Calculated by MAD program using actual weather data for February 1981. (For average weather data, calculated panel area = 14.6 m^2, assuming one side of panel available for convection.)

[b]Heat leakage is much higher than calculated because the bin is about four times larger than it should be for a minimum ACES in Knoxville.

[c]Data from Knoxville weather station.

3.4 Discussion

The foregoing analyses and the sizing procedure used by MAD are based on a single panel design — unglazed, finned, aluminum tubing (Fig. 3). For this type panel, the sizing methodology used by MAD — assuming only one side freely exposed to ambient air — yields a panel area of 14.6 m^2, which is more than sufficient for the Knoxville area in 1981. As indicated earlier, however, assuming two sides available for convection yields a panel area of 8.9 m^2. A panel of this size would have collected only about 3.77 GJ instead of the 6.34 GJ collected by the experimental panel. This level of energy collection would not have been sufficient to keep the ice inventory constant in February 1981.

Both aforementioned designs are based on Eqs. (1) and (2) for heat transfer coefficient and panel efficiency and the assumptions that the fluid entering the panel was at 0°C and that the panel surface was black with an absorptivity (α) of 0.93. Data taken over the period from January 26 to March 2, 1981, suggest that Eqs. (3) and (4) better represent the heat transfer coefficient and panel efficiency for this application. The average fluid temperature entering the panel over the same period was about 1.4°C. In addition, since the ACES house panel is painted a light grayish brown, its α is estimated to be about 0.65.[11]

Using Eqs. (3) and (4), the demonstrated fluid entering temperature of 1.4°C, and the α of the actual panel in the design model, a design panel size of 14.1 m^2 was produced (as listed in Table 3). Compared to the test panel of 14.9 m^2, a panel of this size would have collected about 5.98 GJ of energy in February 1981, or about 19% more than was needed for constant ice inventory. Thus, given parameters and correlations that more closely model the actual Knoxville ACES solar panel operation, the MAD program yields panel sizes that are conservative by about 20% for Knoxville.

For panels of other than the finned tube type, experimental data should be obtained for radiative and convective heat transfer and panel efficiency and proper correlations generated for each type considered. Such panels should then be sized accordingly.

Table 3. Solar panel designs for a minimum ACES in Knoxville

	Area, m^2	Energy collection in February 1981, GJ
Actual panel used in Feb. 1981	14.9	6.34
MAD design—original assumptions[b]	14.6	6.21[a]
MAD design—two sides for convection	8.9	3.77[a]
MAD design—modified assumptions[c]	14.1	5.98[a]

[a]Estimated.

[b]One side for convection, 0°C entering fluid temperature, $\alpha = 0.93$, Eqs. (1) and (2).

[c]Two sides for convection, 1.4°C entering fluid temperature, $\alpha = 0.65$, Eqs. (3) and (4).

4. OUTDOOR FAN COIL UNIT TESTS

The following sections give the results of ACES tests using an outdoor fan coil unit as a substitute for the solar panel.

4.1 System Modifications

To allow system tests to be made utilizing ambient air as a heat source/sink, an outdoor fan coil unit (Fig. 7) was installed at the ACES house in March 1981. Table 4 contains the design specifications and laboratory performance test results for the coil. A schematic of the ACES with the fan coil unit and piping changes is shown in Fig. 8.

4.2 Heating Season Tests

4.2.1 Space and water heating

Table 5 gives the results of steady-state performance tests for space and water heating. Performance values using ice as an energy source are given for comparison. Fig. 9 is a schematic of this mode.

Two major conclusions may be drawn from Table 5. First, by using 9.4°C air as the heat source for the heat pump, the COP_r is slightly improved. Second, despite improved refrigerant system performance, overall system performance is reduced by 6% (COP = 2.44 vs 2.59) because of the extra energy consumed by the outdoor fan. If system efficiency using outdoor air at 9.4°C is to equal that using ice, the energy consumption of the outdoor fan has to be 80 W or less. A factor of 3.75 improvement in fan and motor efficiency would be required to achieve this consumption level. In comparison, the advanced fan-motor developed by Westinghouse[12] has an efficiency of 37% (about twice the efficiency of the fan-motor used).

Table 6 gives the results of two short-term tests in which the system primarily supplied water heating. Results indicate that system performance with the ice bin was 10% better than it was with outdoor air as the energy source, even though the average air temperature was 16°C. This was due to the increased energy consumption of the outdoor fan.

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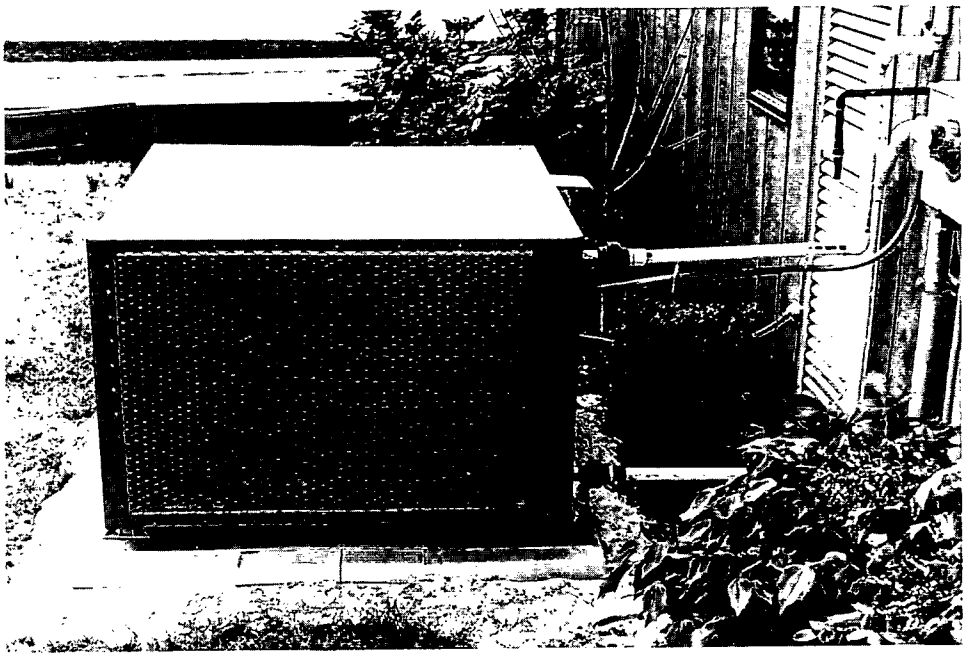

Fig. 7. Outdoor fan coil unit added to Annual Cycle Energy System.

Table 4. Air coil performance

	Winter	Summer
Design specifications		
Air flow, m^3/s	1.26	1.26
Capacity, W	9174	8734
Entering air, °C	4.44	32.22
Entering brine, °C	−5.56	40.56
Leaving brine, °C	−2.17	36.17
ΔP Brine side, kPa	10.10	4.13
ΔP Air side, mm H_2O	2.54	2.54
Brine flow, L/s	0.69	0.50
Actual performance		
Air flow, m^3/s	1.78	1.78
Capacity, W	4985[a]	8658
Entering air, °C	7.28	21.11
Entering brine, °C	−3.17	29.00
Leaving brine, °C	−0.22	25.11
Brine flow, L/s	0.44	0.50
ΔP Brine side, kPa	4.13	4.82
ΔP At design flow, kPa	10.20	4.82
ΔP Air side, mm H_2O	3.05	3.05
Fan motor efficiency, %	18	18
Capacity adjusted to design brine flow and ΔT, W	9027	9848

[a]Limited by ACES mechanical package capacity.

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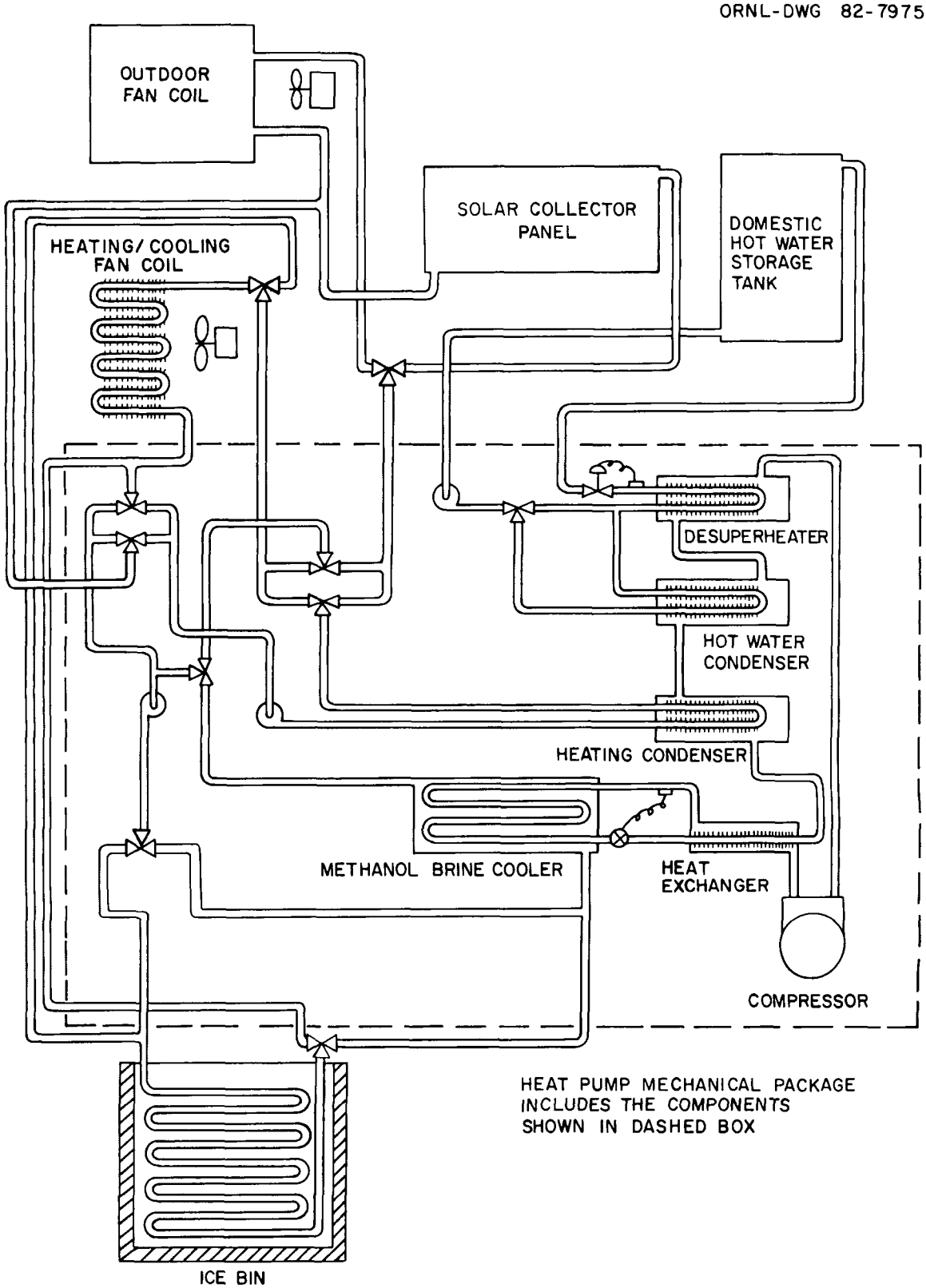

Fig. 8. Annual Cycle Energy System schematic with outdoor fan coil included.

Table 5. ACES space- and water-heating performance: outdoor fan coil unit vs ice/water bin heat exchanger

	Evaporator energy source	
	Outdoor air[a]	Ice/water bin[b]
Heat flows, Wh/h		
From energy source	5660	5255
Indoor fan coil	5610	5415
To hot water	1175	1820
Evaporator		
Entering brine, °C	2.4	−2.7
Leaving brine, °C	−1.0	−4.7
Brine flow, L/s	0.44	0.71
Indoor fan coil		
Entering brine, °C	37.2	35.1
Leaving brine, °C	33.6	31.9
Brine flow, L/s	0.40	0.44
Component energy consumption, Wh/h		
Compressor	2590	2430
Indoor fan	475	480
Cold brine pump	142	165
Hot brine pump	139	150
Hot water pump	81	81
Outdoor fan	300	
COP[c]	2.44	2.59
COP_r[d]	2.18	2.16
Refrigerant conditions		
Compressor discharge pressure, kPa	1572	1542
Condensing temperature, °C	43.8	42.9
Condenser exit temperature, °C	39.0	37.9
Evaporating temperature, °C	−4.2	−6.7
Compressor suction pressure, kPa	324	303
Compressor suction temperature, °C	8.5	18.4

[a]Temperature = 9.4°C.
[b]Temperature = 0°C.
[c]Coefficient of performance (energy source heat flow plus compressor, i.d. fan, and pump watts per system watts).
[d]Compressor only cooling COP, energy source heat flow/compressor watts.

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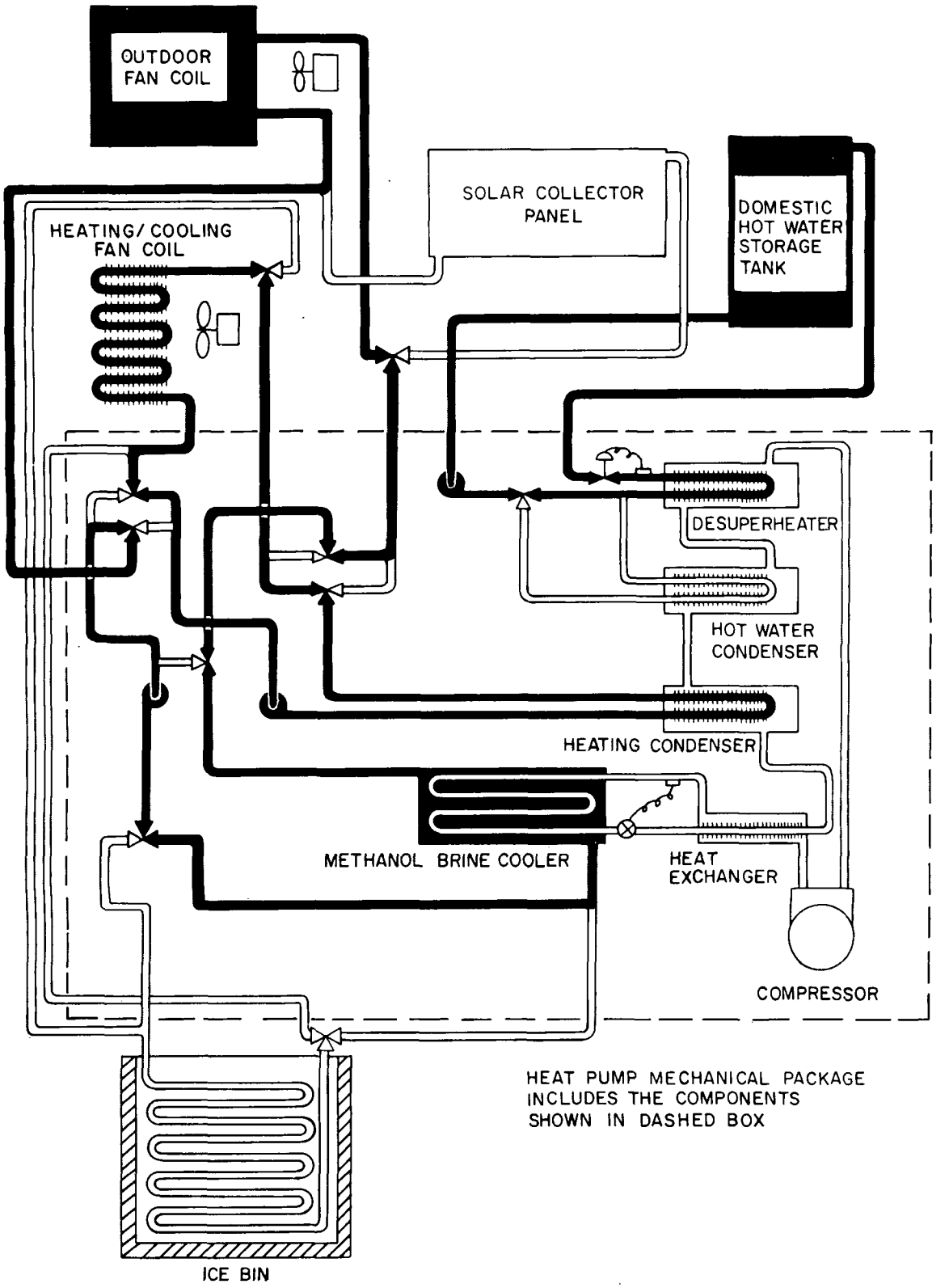

Fig. 9. System schematic of space heating and water heating using the outdoor fan coil (compressor on).

Table 6. ACES performance: ice source vs air source

	Ice source[a]	Air source[b]
Test duration, days	5	4
Space heating, kWh	16.3	10.3
Water heating, kWh	81.8	60.9
Energy consumption, kWh		
Compressor	37.1	27.3
Indoor fan and pumps	3.4	2.6
Outdoor fan		2.7
COP[c]		
System	2.4	2.2
Compressor only	2.6	2.5

[a]Temperature = 0°C.
[b]Temperature = 16°C.
[c]Coefficient of performance.

These tests results indicate that the efficiency of the fan and/or its motor will have to be substantially improved if a brine-chiller ACES is to perform as efficiently in the heating mode with outdoor air as with the ice bin. Thus, an outdoor fan coil unit is not justified if it is to be used only in heating mode operations.

4.2.2 Ice melting using outdoor fan coil unit

The ice-melting modes tested using the outdoor fan coil unit as an alternative to the solar panel were (1) a passive mode requiring only a circulating pump and the outdoor fan (Fig. 10) and (2) an active mode requiring compressor operation (Fig. 11).

4.2.2.1 Passive mode

Results of passive tests are summarized in Table 7. The daytime performance of the panel is far superior to that of the fan coil unit, as expected. However, the coil can collect more energy from the surroundings when solar radiation is not available than can the panel.

These results indicate that in areas having a sufficient number of hours of outdoor temperature greater than about 4°C, an outdoor fan coil unit could perhaps be used instead of a solar panel to provide ice melting. Energy consumption for this purpose would be increased, however, by a factor from 2 to 10 if this were done.

Table 7. Passive ice-melting performance: solar panel vs outdoor fan coil

	Solar panel				Fan coil unit	
	Clear day		Night			
Air temperature, °C	11.5	3.5	10.4	4.0	11.0	3.3
Heat collection, Wh/h	8170	7420	3765	1300	7850	1020
Energy consumption,[a] Wh/h	135	135	135	135	435	435
Melting COP[b]	60.5	55.0	27.9	9.6	12.1	2.3

[a]Circulating pump and outdoor fan.
[b]Melting coefficient of performance equals the amount of heat collected divided by the amount of energy consumed.

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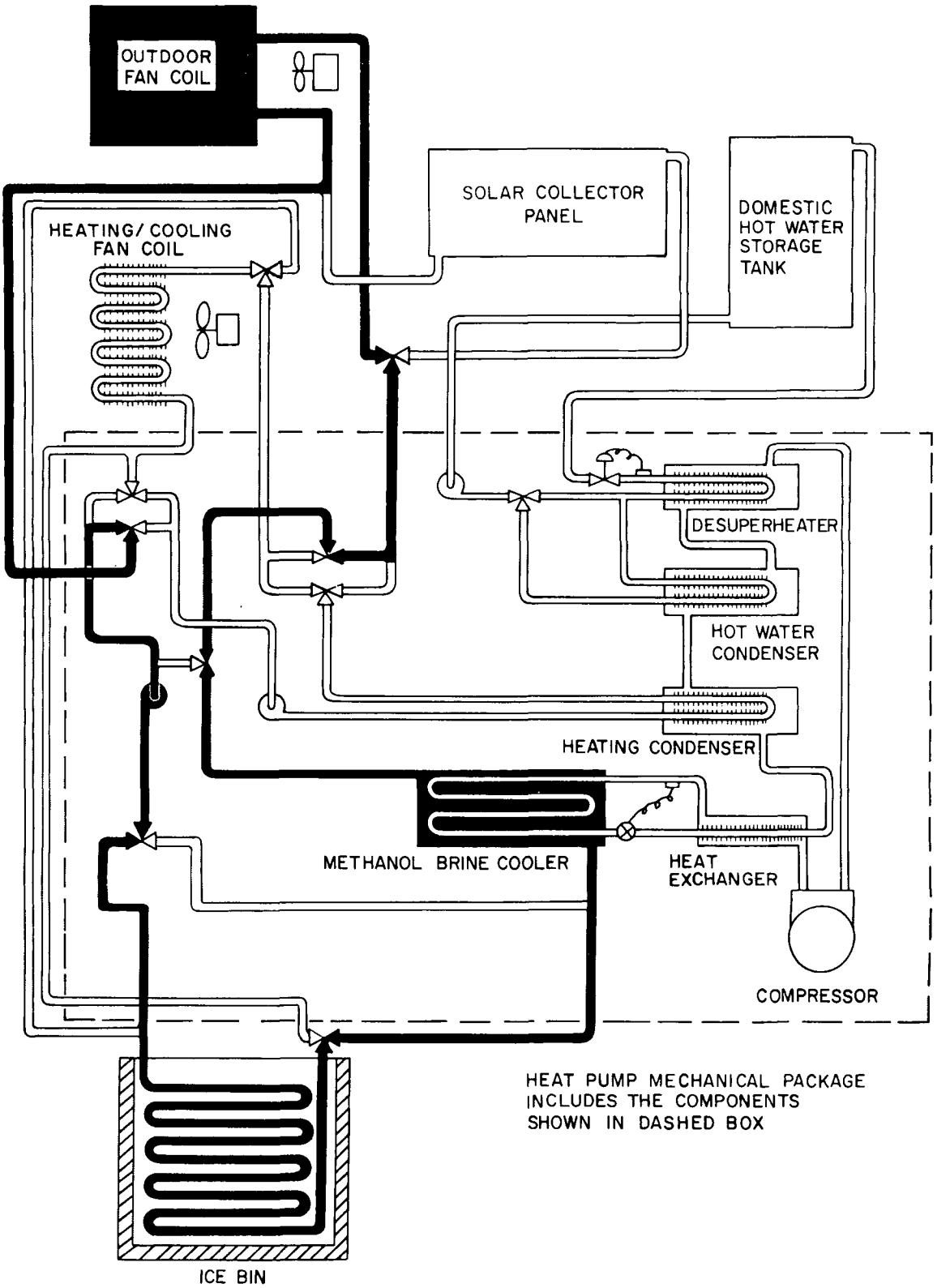

Fig. 10. Schematic of the passive ice-melting mode (compressor off).

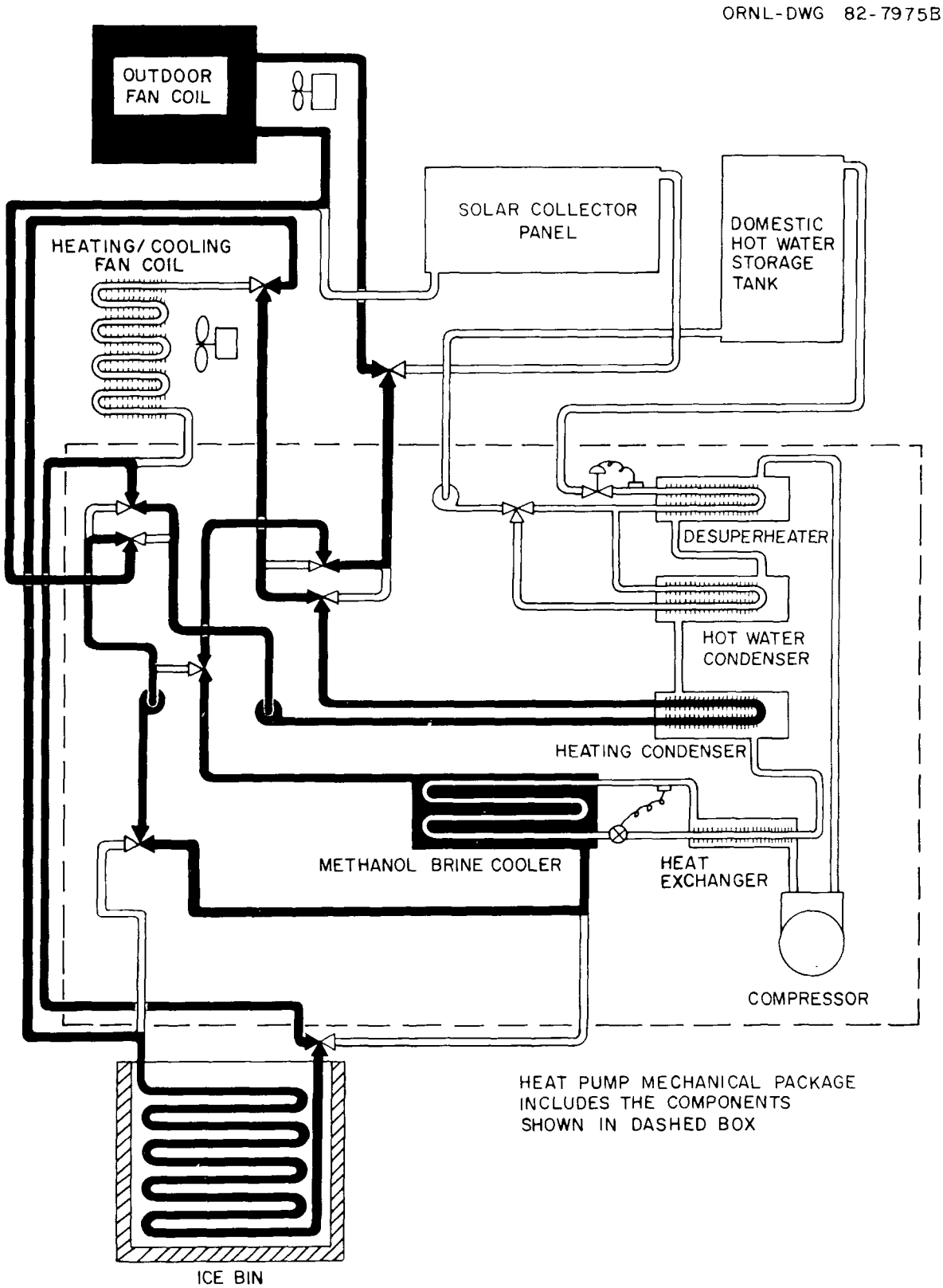

Fig. 11. Schematic of the active ice-melting mode (compressor on).

4.2.2.2 Active mode

Use of the fan coil unit in an active mode in which the compressor is used to pump heat from the outdoor air to the bin was also investigated. Results of system tests are not encouraging (Table 8). The melting COP of 2.25 is not as good as that obtained in the passive mode at an air temperature of 3.3°C. Brine flow in the bin loop had to be throttled to less than 25% of its normal value to prevent the low-pressure safety control from shutting off the compressor.

The active mode has the potential to achieve melting COPs of about 3 or better (see Appendix A for a sample calculation) for outdoor air temperatures of around −5°C, and it could be of use in areas where air temperatures are not high enough for use of the passive mode. To achieve such COPs, however, would require redesigning the mechanical package to enable it to operate efficiently at the low heat source/sink temperature differences encountered in the active melting mode while not degrading its heating efficiency. Even so, these efficiencies are not as high as those attainable by the passive mode.

Table 8. Active ice-melting mode test results

Air temperature, °C	5.0
Outdoor coil	
Flow, L/s	0.45
T_{in}, °C	−1.6
T_{out}, °C	1.5
Heat flow, Wh/h	5300
Bin	
Flow, L/s	0.09
T_{in}, °C	20.6
T_{out}, °C	0.2
Heat flow, Wh/h	6534
Energy consumption	
Compressor, Wh/h	2320
Fan pumps, Wh/h	580
Melting COP[a]	2.25

[a]Melting coefficient of performance equals the bin heat flow divided by the energy consumption.

4.3 Cooling Season Tests

The system with the outdoor fan coil unit was tested in two different cooling modes: (1) off-peak cooling storage or night heat rejection (Fig. 12) and (2) direct air conditioning (Fig. 13).

4.3.1 Off-peak cooling

Test results of steady-state performance in the night heat rejection mode — fan coil unit vs solar panel — are shown in Table 9. Performance was clearly superior when the outdoor fan coil unit was used as a nighttime heat rejector. Use of the fan coil unit enabled the compressor to operate with a much lower discharge pressure. The resultant decrease in compressor energy consumption more than outweighed the added energy consumption of the outdoor fan, thus yielding a 22% performance improvement.

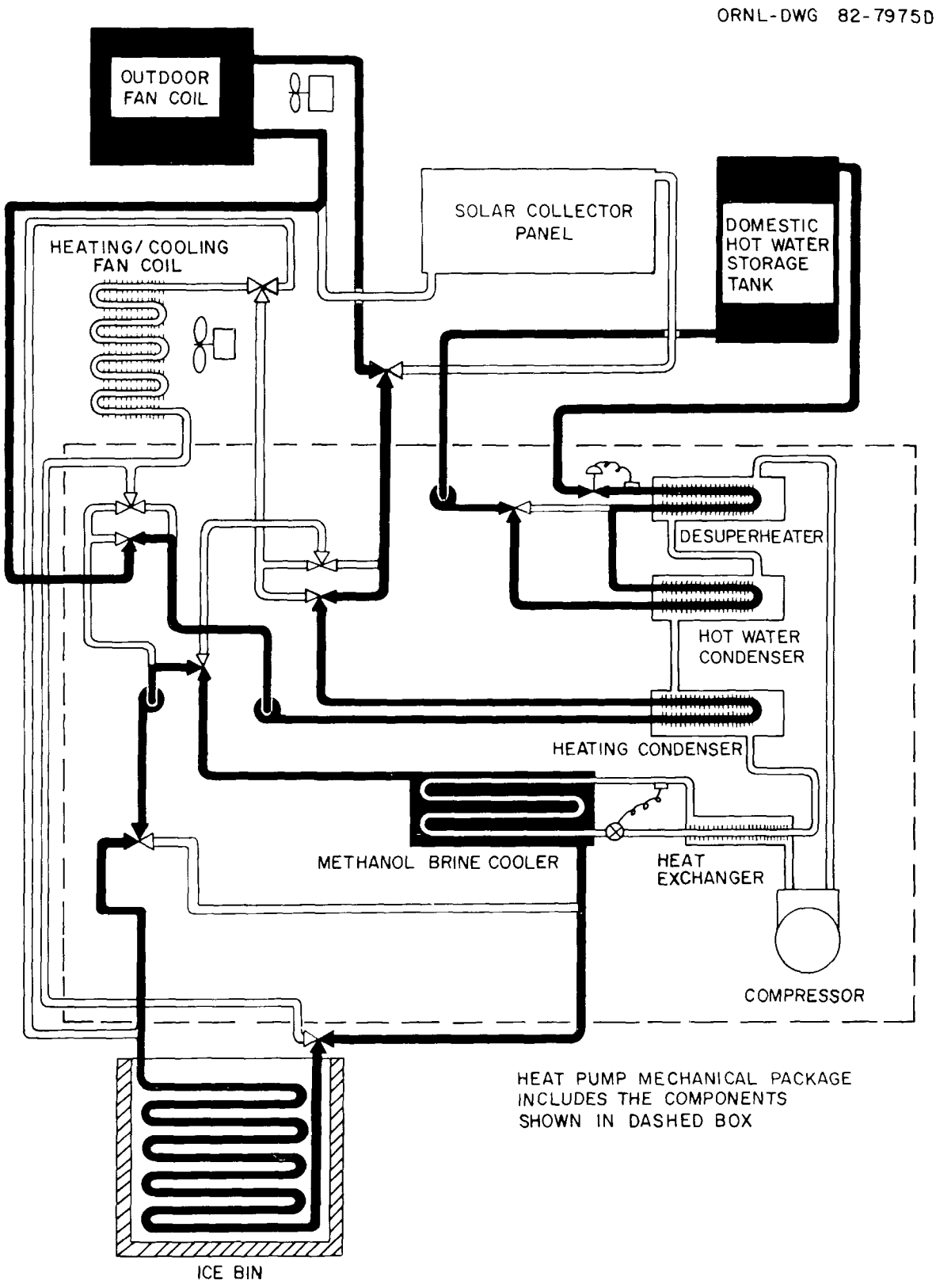

Fig. 12. Schematic of the night heat rejection mode (compressor on).

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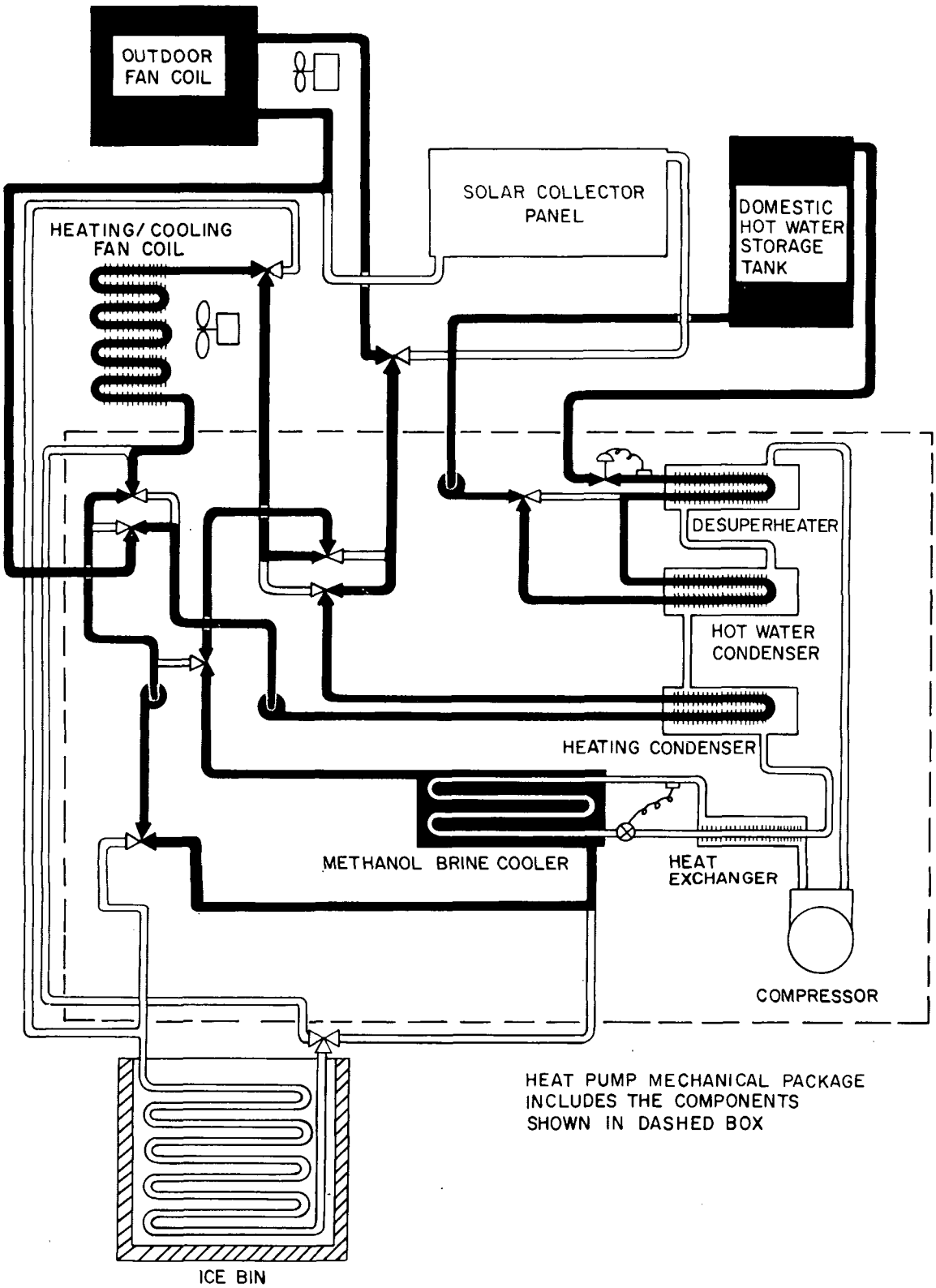

Fig. 13. Schematic of the direct air-conditioning mode (compressor on).

Table 9. Night heat rejection performance: outdoor fan coil unit vs solar panel

	Heat rejection device	
	Outdoor fan coil	Solar panel
Air temperature, °C	21.0	20.0
Heat flows, Wh/h		
To air	6600	5880
To hot water	1300	1840
From ice bin	5700	5130
Ice bin		
Bin temperature, °C	12.80	5.5
Entering brine, °C	10.00	0.8
Leaving brine, °C	12.40	3.0
Brine flow, L/s	0.6	0.7
Rejector		
Entering brine, °C	29.6	43.8
Leaving brine, °C	25.4	40.5
Brine flow, L/s	0.5	0.5
Hot water		
Entering water, °C	54.4	58.5
Leaving water, °C	57.5	62.3
Flow, L/s	0.19	0.19
Energy consumption, Wh/h		
Compressor	2280	2830
Cold brine pump	146	165
Hot brine pump	140	150
Hot water pump	80	81
Outdoor fan	290	
Bin heat removal COP, $(COP_{BR})^a$	1.94	1.59
Refrigerant		
Discharge pressure, kPa	1343	1931
Condensing temperature, °C	37.6	51.9
Evaporating temperature, °C	−8.2	0.0
Suction pressure, kPa	276	393
Suction temperature, °C	18.0	27.3

[a]COP = coefficient of performance.

$$COP_{BR} = \frac{\text{bin heat removal}}{\text{energy consumption}} .$$

Table 10 gives a comparison of the ACES supplemental cooling performance in 1980 vs that in 1981. Use of the outdoor fan coil unit resulted in a system COP of 2.34 vs 2.25 using the panel.

An outdoor fan coil unit is recommended for the ACES wherever supplemental cooling is required.

4.3.2 Direct air conditioning

Steady-state test results for the direct air-conditioning mode utilizing the fan coil unit as a heat rejector are given in Table 11. The COP for this mode was found to vary from about 2.1 to about 2.2 as outdoor temperature decreased. Longer-term test results show that internal equipment loads (compressor, pumps, and indoor fan) from an unventilated mechanical room degrade performance significantly as illustrated in Table 11.

Table 10. ACES supplemental cooling performance: solar panel vs outdoor fan coil

	Solar panel: 8/14/80 - 9/15/80	Outdoor fan coil: 8/24/81 - 9/14/81
Cooling load, GJ	5.96	3.76
Water heating load, GJ	1.81	1.18
Energy consumption, kWh	961	586
COP_{SC}[a]	2.25	2.34

[a]Supplemental cooling coefficient of performance equals the amount of cooling plus water heating energy delivery divided by the amount of energy consumed.

Table 11. ACES direct air-conditioning steady-state test results

Air temperature, °C	23.0	31.0
Cooling load delivered, Wh/h	5125	5493
Water heating delivered, Wh/h	1600	1500
Energy consumption, Wh/h		
Compressor	2260	2540
Indoor fan	206	206
Cold brine pump	128	128
Hot brine pump	139	139
Hot water pump	80	80
Outdoor fan	280	280
COP[a]	2.18	2.07

[a]Coefficient of performance equals the amount of cooling plus water heating energy delivered divided by the amount of energy consumed.

By ventilating the mechanical room properly, the long-term integrated direct air-conditioning COP was raised from 1.8 to 2.0 (Table 12). Even so, it was much below the performance of a standard heat pump with desuperheater water heater. This was due to the additional heat exchanger temperature differences imposed by the intermediate brine heat transfer fluid loops as reflected in higher compressor energy consumption and to the additional parasitic energy consumption of the circulation pumps.

The ACES is more efficient when operating in the off-peak supplemental cooling mode than in the direct air-conditioning mode, as can be seen by comparing the results in Tables 10 and 12.

Table 12. ACES direct air-conditioning performance vs control house heat pump desuperheater water heater system

	7/13/81 - 7/27/81		9/14/81 - 9/17/81	
	ACES	Control	ACES	Control
Cooling load delivered at fan coil, kWh	1245.2	1142.4	117.0	115.5
Internal equipment loads, kWh	343.4[a]	6.9	4.5	0.7
Cooling load delivered to house, kWh	901.8	1135.5[b]	112.5	114.8
Water heating delivered, kWh	452.6	243.9	24.9	24.0
Compressor energy consumption, kWh	566.4	391.9	49.2	38.0
Parasitic energy consumpton, kWh	192.4	100.3	18.0	9.6
COP[c]	1.8	2.8	2.0	2.9
Average house temperature, °C	26.4	25.3	25.8	25.3
Average outdoor operating temperature, °C	27.1	28.6	24.0	24.6

[a]Mechanical room not ventilated to outside.

[b]The higher control house cooling load was due to thermostat control problems that resulted in very long on cycles and a lower average indoor temperature.

[c]Coefficient of performance.

4.4 Discussion

4.4.1 Heating season

As mentioned previously, three different heating season operational modes were tested: direct space heating and water heating, passive ice melting, and active (mechanical package assisted) ice melting.

In general, direct space heating using outdoor air as the energy source was not as efficient as using ice as the energy source due to the extra energy consumption of the outdoor fan. However, this mode may still be useful as a backup heating system if the bin becomes too full of ice to be used efficiently.

In regions near the full ACES zero panel area line for a house similar to the Knoxville ACES house (Fig. 14) where supplemental heat collection is not normally required, use of the outdoor air as a backup heat source may eliminate the need for supplemental heat collection during harsher than normal winters. That is, the system would use the ice bin until full capacity was reached which, because of the large bin volumes required in the region, would occur in the latter part of the winter. The system would then use the outdoor fan coil backup until the end of the winter or until heat leakage melted enough ice to allow use of the storage bin again. Alternatively, the coil could be used in the passive melting mode, in lieu of a solar panel, to melt the excess ice.

For regions north of the zero panel area line where some supplemental heat collection will be required almost every year, the passive melting mode will suffice only if there are enough hours in which the outdoor temperature is 4°C or higher. If the number of such hours is insufficient, another means of energy collection must be used.

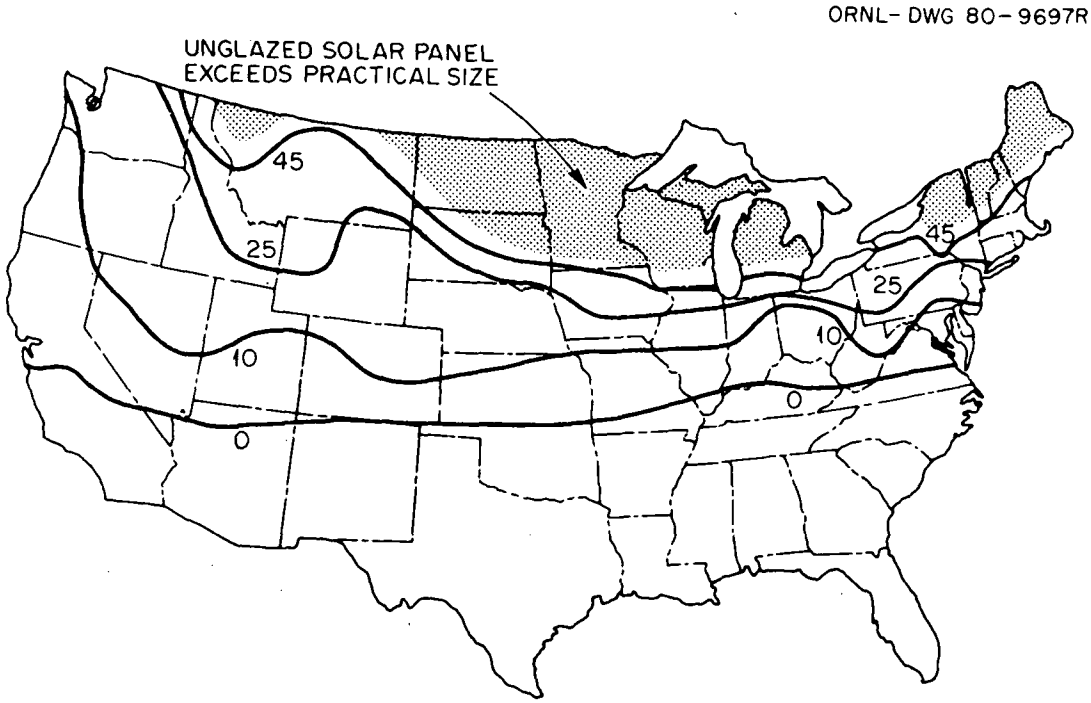

Fig. 14. Panel area in square meters for a full Annual Cycle Energy System (ACES) in a house similar to the ACES demonstration house.

Use of the heat pump to collect energy from the air and pump it to the bin has been shown to be inefficient with the brine-chiller system. Although this method shows some promise, it is not likely to work well without significant design changes to the brine chiller's refrigerant circuit. In any event, it would be the most energy-consuming means of collecting supplemental energy of all alternatives studied.

4.4.2 Cooling season

Of the two cooling season modes tested, off-peak supplemental cooling is the most efficient, probably because in the supplemental mode the unit operates at lower average air temperatures thus lowering the head pressure and compressor energy consumption. For this reason, the off-peak cooling mode is recommended over direct air conditioning for brine-chiller ACES when the ice supply is exhausted.

5. CONCLUSIONS AND RECOMMENDATIONS

The following conclusions are drawn from the tests done in 1981 at the TECH site.

1. Unglazed solar/convector panels are an effective means of supplying environmental energy to an ACES storage bin, except in extreme northern regions of the United States.

2. The method used in the MAD computer program to size solar panels (which assumes only one side of the panel is available for convective heat collection) is conservative for panels composed of finned, aluminum tubes. For Knoxville, the minimum ACES predicted panel area was about 30% larger than necessary for maintaining a constant ice inventory.

3. Modification of the panel design correlations and of the entering fluid temperature and absorptivity to more accurately model the test panel resulted in a panel area about 20% larger than necessary.
4. Brine-chiller ACESs deliver heating energy more efficiently when using an ice/water medium as the energy source than when using outdoor air.
5. At air temperatures above about 4°C, an outdoor fan coil unit alone can be used in a passive mode to collect environmental energy for ice melting in lieu of a solar panel.
6. Using the compressor to pump heat to the bin from the outdoor air is not a viable mode for a brine-chiller ACES of the design used at the ACES demonstration house.
7. Use of an outdoor fan coil unit is more efficient than use of a solar panel for night heat rejection.
8. Night heat rejection operation to store cooling energy for later use is more efficient than direct air-conditioning operation for brine-chiller ACESs.

Use of an outdoor fan coil unit to replace the solar panel for wintertime supplemental energy collection is not recommended unless supplemental heat requirements are unusually small and a sufficient number of hours in which the air temperature is greater than 4°C are expected to be available. An outdoor fan coil is recommended for all ACESs in areas where some supplemental cooling must be provided.

Care is advised in using the MAD program methodology for sizing solar panels for the ACES. For panels of unglazed, finned aluminum tubing (Fig. 3), the sizing method is conservative. For different panel types, it is recommended that experimental data for both convective and solar heat gain be obtained before attempting to size the panel using the MAD program.

APPENDIX A

SAMPLE CALCULATIONS

Active melting mode potential performance using outdoor air-to-brine fan coil unit

Objective: pump heat from air at −5°C to ice/water at 0°C

Evaporator temperature:

assume $\Delta T_{air-brine} = 10$ C°,
assume $\Delta T_{brine-R\text{-}22} = 3$ C°,
then $\Delta T_{air-R\text{-}22} = 13$ C°,
evaporator temperature = −18 C°.

Condenser temperature:

assume $\Delta T_{ice-brine} = 4$ C°,
assume $\Delta T_{brine-R\text{-}22} = 3$ C°,
then $\Delta T_{ice-R\text{-}22} = 7$ C°,
condenser temperature = 7 C°.

Note: assumed temperature differences are based on observations of ACES mechanical package performance.

Further assumptions:

60 C° discharge superheat,
20 C° suction superheat,
40% compressor shell heat loss,
25% parasitics (fan and pump energy equals 25% of compressor energy).

These values are typical of ACES mechanical package operation.[4,5]

Figure A.1 illustrates the refrigeration cycle for the melting mode on a pressure-enthalpy (p-h) diagram.

From R-22 property tables:

At point 1,

T = −18 + 20 = 2°C,
P = 165 kPa (24 psig), and
h = 255.75 kJ/kg (110.0 Btu/lb).

At point 2,

T = 7 + 60 = 67°C,
P = 524 kPa (76 psig), and
h = 296.67 kJ/kg (127.6 Btu/lb).

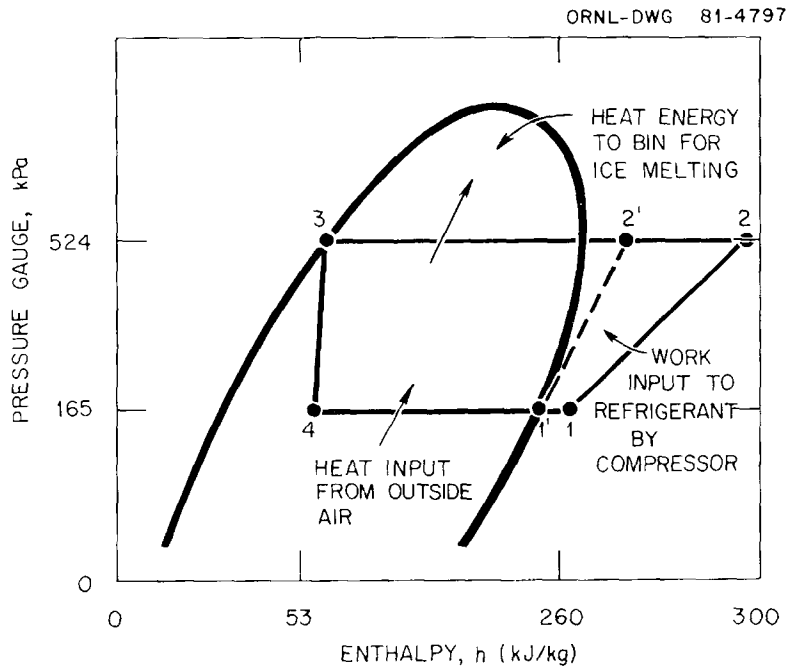

Fig. A.1. Pressure-enthalpy diagram showing refrigerant cycle for active melting mode.

At point 3,

T = 7°C,
P = 524 kPa, and
h = 53.01 kJ/kg (22.8 Btu/lb).

At point 1′,

T = −18°C,
P = 165 kPa,
h = 242.96 kJ/kg (104.5 Btu/lb).

At point 2′,

T = 22.2°C
P = 524 kPa,
h = 262.73 kJ/kg (113.0 Btu/lb).

The melting COP is calculated as follows:

$$COP_{melt} = \frac{h_2 - h_3}{1.25[(h_2 - h_1)/0.6]} ,$$

$$COP_{melt} = \frac{296.67 - 53.01}{2.08(296.67 - 255.75)} , \text{ and}$$

$$COP_{melt} = 2.9 .$$

For cycle 1′ – 2′ – 3 – 4 – 1′ (18°C evaporator temperature and 7°C condenser temperature) with no compressor shell heat loss, but including parasitics,

$$COP_{melt} = \frac{h_{2'} - h_3}{1.25(h_{2'} - h_{1'})} = 8.5 \ .$$

REFERENCES

1. A. S. Holman et al., *Annual Cycle Energy System (ACES) Performance Report November 1977 through September 1978*, ORNL/CON-42 (May 1980).
2. V. D. Baxter, *ACES: Final Performance Report December 1, 1978, through September 15, 1980*, ORNL/CON-64 (April 1981).
3. M. L. Ballou et al., *MAD: A Computer Program for ACES Design Using Monthly Thermal Loads*, ORNL/CON-51 (March 1981).
4. L. A. Abbatiello et al., *Performance and Economics of the ACES and Alternative Residential Heating and Air-Conditioning Systems in 115 U. S. Cities*, ORNL/CON-52 (March 1981).
5. E. A. Nephew and L. A. Abbatiello, *Performance and Economics of Eight Alternative Systems for Residential Heating, Cooling, and Water Heating in 115 U. S. Cities*, ORNL/CON-89 (November 1982).
6. E. A. Nephew, L. A. Abbatiello, and M. L. Ballou, *Theory and Design of an Annual Cycle Energy System (ACES) for Residences*, ORNL/CON-43 (May 1980).
7. J. L. Bledsoe and D. R. Miller, *Data Collection and Processing for the ACES*, ORNL/CON-59 (November 1981).
8. R. W. Bliss, "The Derivation of Several Plate Efficiency Factors Useful in the Design of Flat-Plate Solar Heat Collectors," *J. Solar Energy* 3(4), 55 (1959).
9. R. W. Bliss, "Atmospheric Radiation Near the Surface of the Ground: A Summary for Engineers," *J. Solar Energy* 5(3), 103 (1961).
10. *Statistical Analysis System (SAS) User's Guide 1979 Edition*, ed. by J. T. Helwig and K. A. Council, SAS Institute, Inc., Raleigh, N.C.
11. *ASHRAE Handbook: 1981 Fundamentals*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1981, p. 2.8.
12. Advanced Electric Heat Pump I: Final Report, ORNL/Sub/79-24712/3 (in preparation).

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