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ANNUAL CYCLE ENERGY SYSTEM PERFORMANCE AND NATIONAL ECONOMIC COMPARISONS WITH COMPETITIVE RESIDENTIAL HVAC SYSTEMS

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INTRODUCTION

The Annual Cycle Energy System (ACES) project, sponsored by the U. S. Department of Energy, provides a technology that helps meet the demand for electrically based heating and cooling at substantially higher efficiencies than are obtainable with alternate systems.[1] The ACES provides space heating, air conditioning, and domestic water heating to residences and commercial buildings through the use of an electrically driven unidirectional heat pump that obtains its heat from water stored in an insulated underground tank. As the heat is extracted during the heating season, most of the water is frozen and the stored ice provides air conditioning in the summer, or at other times if needed. Thus, the water's heat of fusion is available as a heat source in winter and as a heat sink during periods of cooling needs. Because both the heating and cooling outputs of the heat pump are used, the resulting annual coefficient of performance (ACOP) is high. A simplified schematic diagram of a typical system is shown in Fig. 1.

The ACES achieves maximum energy conservation in applications where the annual ice production and cooling demands of a building result in a balance between heat extractions from the ice bin and heat deposits in it. In practice, an exact ice bin heat balance from building loads alone is unlikely because these loads vary with the annual weather and the building usage. Provision must be made in the ACES design to compensate for imbalances in the ice bin heat flows. This can be done through the use of an auxiliary solar convector panel, which provides for either the collection of heat for melting excess ice or for rejecting heat from the ice bin to the environment when the stored ice is depleted and supplemental cooling operation is required. Supplemental cooling requirements can be met by operating the ACES heat pump during off-peak nighttime and weekend hours, providing domestic hot water and rejecting excess heat to the environment. Cooling output is stored in the storage bin for use during peak cooling load hours. While this supplemental cooling technique does not conserve energy, it does enable the ACES to take advantage of off-peak electricity rates where they are available.

This paper summarizes the results of three years of experience with the ACES and conventional systems in two identical houses, the ACES house (using an ACES for heating, cooling, and water heating) and the control house (using conventional systems). The houses, located in a test complex on the University of Tennessee campus at Knoxville, Tennessee (Fig. 2), were unoccupied during the tests. Internal loads for a family of four

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were simulated by appliances and hot water used at the rate of 265 L/d (70 gal/d). The test period extended from November 1, 1977, through September 15, 1980.[2,3] Additionally the results of an analytical study comparing the ACES performance and economics with that of several conventional heating, ventilating, and air conditioning (HVAC) and water heating systems are summarized.[4] The ACOP, defined as follows, is used to compare ACES performance with that of alternative systems.

$$\text{ACOP} = \frac{\text{annual energy delivered for space heating, water heating, and space cooling}}{\text{annual energy consumption deliver these building services}}$$

DEFINITIONS

The ACES concept allows the designer flexibility and choice with respect to the method of ice formation and the bin energy storage capacity. It is useful to determine some terms that describe the various choices.

The ACES may employ either of two mechanical systems: (1) a brine chiller system which employs a submerged water-to-brine heat exchanger located in the ice storage bin to freeze the water surrounding the heat exchanger coils[2,3] or (2) a plate-type ice maker heat pump[5] which freezes water directly on the evaporator plates of the heat pump and periodically harvests the ice into the bin. Each offers certain advantages. With the brine chiller ACES, because of a higher ice-packing density in the storage bin, bin volumes of approximately half the size necessary for an ice maker ACES can be used. The brine chiller ACES also has simpler refrigeration circuitry, a lower refrigerant inventory and provides a nonfreezing energy transfer fluid, the brine, for transporting heat from supplemental sources such as a solar collector panel into the storage bin. The ice maker ACES, on the other hand, eliminates the ice bin heat exchanger with its attendant costs and need for fabrication in the field and makes modular design easier. Although both systems have compressor-only COPs of 3.0–3.1, the brine chiller appears to offer significantly better reliability and economics in most applications and is therefore used as the base case for ACES designs.[5,6]

The ACES transfers energy between the seasons of the year by storing the heat that is removed from the house during summer air-conditioning operation and then delivering the stored heat back to the house in the winter to provide space or water heating. To achieve maximum interseasonal energy transfer, the ice storage bin must have enough capacity to store all of the ice that can be produced during the heating season or all of the ice needed for summer air conditioning, whichever is smaller. A system with a bin of this size is defined as a *full* ACES, which is characterized by a late summer ice exhaustion date. In other words, cooling provided by stored ice makes the maximum possible contribution toward satisfying the house cooling needs. This is graphically represented in Fig. 3.

Systems with smaller bins than those corresponding to a full ACES can, of course, be fabricated. If this is done, the amount of ambient heat that must be collected from the environment to melt surplus ice, as well as the amount of supplemental cooling that must be provided, becomes greater. In order to provide an adequate reserve heating capacity, the smallest bin that should be considered for an ACES should have a thermal capacity sufficient to allow at least two weeks of heat pump operation during the coldest month of the year without having to collect any ambient heat. A system with a bin of this size is defined as a *minimum* ACES, and is a compromise to reduce capital costs of the system by reducing the energy conservation potential of the system. The ice inventory history of a typical minimum ACES is shown in Fig. 4. The stored ice contributes a much smaller fraction of the building's cooling needs.

ACES OPERATIONAL EXPERIENCE

The centerpiece of the residential ACES demonstration program is an experimental complex on the campus of the University of Tennessee, just outside of Knoxville (shown in Fig. 2). One of the buildings (the ACES demonstration house) is a 167-m^2 (1800-ft^2) residence with an insulated 69-m^2 (2440-ft^3) storage bin in the basement; this house utilizes a full ACES for heating, cooling, and domestic hot water production. Figure 5 shows a cutaway view of the ACES house. A second building (the control house) is an identically constructed residence with the same orientation, differing only in that its heating and cooling and water heating are provided by conventional systems. The two buildings are well-instrumented to permit direct comparison of the different systems.

The first full annual cycle test of the systems began on November 1, 1977, and continued until September 18, 1978, at which time the experiment was terminated to allow system upgrading and modification prior to the next heating season.[2] Because only small heating or cooling loads occur in late September and in October, the 10 months of actual operation closely approximated a full year's run. Operational performance was satisfactory during the first year, with minor control failures causing the system to be inoperative for about two days during the entire test year.

During late September and October 1978, additional insulation was added to the storage bin, increasing the level of insulation from about 0 to 6.7 $K{\cdot}m^2/W$ to 7.05 $K{\cdot}m^2/W$ and a new, more cost-effective ice bin coil system was added. Operation for the second test year began on December 1, 1978, and was completed on September 30, 1979. Third-year operation began on December 1, 1979, and continued through September 15, 1980. Again the system operated reliably and its performance agreed well with calculated predictions.[3]

During the first annual cycle test the control house was operated in the electrical resistance heating mode to establish base loads for the two houses. The heating load of the ACES house agreed with that of the control house to within 2%. During the next two years the control house used two commercially available heat pumps for space heating and space cooling. The heat pump used during the second year was a standard quality unit with an Air-conditioning and Refrigeration Institute (ARI) rated COP of 2.46 at 8.3°C (47°F). A more efficient heat pump having a rated COP of 3.11 at 8.3°C was used during the third year. Control house hot water was provided by an electric resistance (I^2R) heater during all three years.

1977/1978 Seasonal Performance

Because the ACES frequently provides both space and water heating simultaneously and because much of the space cooling is provided from stored ice, a by-product of heating, the true performance of the ACES must be measured in terms of the electrical energy input required and the heating, cooling, and water heating supplied by the system for an entire annual cycle. For the period November 1, 1977, to September 18, 1978, the ACES delivered a total of 43.0 GJ space heating, 20.9 GJ of water heating, and 26.2 GJ of space cooling. For this period, purchased energy was 9012 kWh, with a resulting demonstrated ACOP of 2.78. For the same period, the system in the control house delivered 43.5 GJ of space heating, 15.6 GJ of water heating, and 24.5 GJ of space cooling. The control house used a control electric resistance furnace for space heating, an electric water heater, and a central air conditioner for cooling. The purchased energy was 20,523 kWh, with a resultant ACOP of 1.13.

The design ice capacity of the bin, 51,700 kg, was attained on April 1, 1978, and it provided all of the cooling needs until July 27, 1978. Up to the time that the ice inventory was exhausted, the ACES ACOP for

heating, cooling, and hot water was 3.17. To maintain cooling capacity after the ice was exhausted the system was run in the supplemental cooling mode. Performance in this mode of operation was expected to be about equal in efficiency to an air-to-air heat pump but to have the potential advantage of operating during periods of off-peak electric rates. In practice, however, it soon became apparent that the ACES in this mode of operation was less efficient than the conventional system in the control house. This was caused by heat leakage into the bin and by the large internal load the mechanical package imposed upon the building cooling load. Not only was the ACES cooling the house, but it was also cooling a large amount of earth surrounding the bin. To minimize heat leakage, the controls of the system were reset so that the bin temperature stablized very near the apparent ground temperature of about 9°C. The mechanical room was vented directly to the outside to reduce the house internal load. This improved the ACES supplemental cooling mode efficiency, but the system continued to operate below expectations. By the end of the annual cycle, the ACOP had fallen to 2.78.

1977/1978 Weather Conditions

The performance of the ACES depends on the thermal loads that the system must deliver each year. These loads are a direct result of the interaction of the weather with the building's thermal envelope, and it is appropriate to characterize qualitatively the severity of the test-year weather and to compare it with an average weather year. Weather data used for the long-term comparisons is that compiled by the National Climatic Center at the Knoxville, Tenessee, McGhee-Tyson Airport station 13891. This station is located within 8 km (5 miles) of the test complex.

Heating season. November and December 1977 were relatively mild months, which produced only a slightly higher-than-normal heating demand. January and February 1978 were the coldest consecutive months on record in this area for the last 20 years. The remainder of the test year heating season moderated to nearly normal conditions.

The heating season had a total of 2104°C-days for the November 1977 to May 1978 period; this compares with the normal 1803°C-days anticipated for the same period, based on the long-term average. The severe winter required the ACES to deliver far more heating than was expected on the basis of average yearly requirements. For the test year, the system should have produced an ice inventory greatly exceeding the bin capacity and requiring panel operation to melt excess ice. However, bin capacity was not exceeded because of heat leakage through the uninsulated ice bin floor. As a result, the ice bin barely reached its maximum design capacity.

Cooling season. The 1977/1978 test-year cooling season was nearly normal, exhibiting a total of 890°C days as compared with 832°C-days for long-term average weather conditions. Thus, cooling requirements were nearly average for the year. The ice available should have lasted until late August, but because of the uninsulated bin floor, it was depleted in late July.

1978/1979 and 1979/1980 Performance

The experiences of the 1977/1978 season indicated some areas where system upgrading would pay significant performance dividends. The most important improvement needed was an increase in the level of insulation in the bin, both to reduce ice losses resulting from heat leakage to the ground and to make the supplemental cooling mode more efficient.

A shorter ice bin heat exchanger coil with 55% less surface area than the original coil, was installed between the first and second test years in order to lower system first costs. Although costs were reduced, the ACES heating season performance decreased by about 10% as a result.[3] Table 1 summarizes the ACES steady-state performance before and after the modifications were made.

The modified system was put into operation, and the second annual cycle started on December 1, 1978, and ran until September 30, 1979. For this period the ACES delivered 32.85 GJ of space heating, 19.1 GJ of space cooling, and 15.8 GJ of water heating while consuming 6719 kWh of electricity for an ACOP of 2.80. The control house system (with the ARI-2.46 heat pump) used 12,853 kWh to deliver the assumed same loads for space heating, 19.2 GJ of space cooling and 13.7 GJ of water heating for an ACOP of 1.42. Control house heat pump heating and cooling seasonal performance factors (SPFs) were 1.58 and 1.64.

For the final test period, December 1, 1979, through September 15, 1980, the ACES consumed 6447 kWh and delivered 34.4 GJ of space heat, 22.6 of space cooling, and 14.3 GJ of water heat for an ACOP of 3.08. Improved space cooling efficiency accounted for the ACES performance improvement in the third test year. Using the ARI-3.11 heat pump, the control house system delivered about 34.3 GJ of space heat, about 22.7 GJ of space cooling and 13.5 GJ of water heat while consuming 11,538 kWh for an ACOP of 1.73. Control house heat pump SPFs for 1979/1980 were 1.98 (heating) and 2.27 (cooling).

The improved system in the ACES house operated efficiently and reliably although some early problems with the control system caused about two weeks of outage in December 1979. Heat leakage into the bin at 0°C was reduced from 63 MJ/d experienced during the first year to about 37 MJ/d. Ice inventory for the second year exceeded bin design capacity, reaching 54,400 kg and it provided all of the house cooling until August 24, 1979. Maximum ice inventory reached 58,100 kg in 1980. Despite the fact that more ice was available in 1980 and no mechanical cooling was done in the house during May 1980, stored ice supplied the cooling needs only until August 14, 1980, due to an unusually hot summer.[3]

Second- and third-year weather conditions. The 1978/1979 and 1979/1980 heating seasons were each slightly colder than normal for the Knoxville area. From December 1978 through April 1979 there were 1664°C-days and for the same period in 1979/1980 there were 1636°C-days. Normal for the area is about 1572°C-days.

The 1979 cooling season was much milder than normal. There were 719°C-days from May through September compared to a normal 814°C-days for a 12% lower-than-normal cooling demand. In 1980, Knoxville experienced one of the hottest summers recorded in the area. From June through mid-September there were 882°C-days for a 25% greater-than- normal cooling demand.

Ice bin performance. The effectiveness of the ice bin in storing and delivering energy is fundamentally important to the efficiency of an ACES. Figure 6 shows ice inventory histories of the three test years, clearly indicating that significant amounts of cooling energy are stored during winter and used during summer.

During the first test year the bin had insulation levels of 6.7 $K \cdot m^2/W$ in the ceiling, 3.0 $K \cdot m^2/W$ in the walls, and a concrete slab floor with no insulation. The uninsulated floor allowed excessive amounts of ground heat to enter the bin, reducing the total amount of ice accumulated and causing a too rapid depletion of the ice in the summer.

The insulation added between the first and second test years raised the insulating value to about 7.1 $K \cdot m^2/W$ on all sides of the bin and reduced the bin heat leakage rate by about 40%. Ground temperatures around the bin were markedly affected by the insulation level. Figure 7 shows measured earth temperatures near the bin before and after the insulation level was increased. The improved insulation resulted in raising the earth temperature almost to the average undisturbed temperature level, as calculated by the Kusuda-Achenbach method.[7]

SYSTEM COMPARISONS

A summary of the ACES house and control house operation is presented in Tab. 2 along with predicted annual performance levels, calculated by the monthly ACES design computer program[8] for average weather condi-

tions. The ACES has delivered annual residential space heating and cooling and water heating loads while consuming only 57% as much electricity as the best conventional system tested. In addition to the savings in energy consumption, the ACES significantly reduced the peak hourly consumption compared to the three conventional systems. This reduction averaged about 40% over the two heat pump with I^2R water heating systems and 60% over the I^2R space heating system.

Heating season performance of the two air-source heat pumps is given in Tab. 3. The actual seasonal system performance of both heat pumps was much below the ARI-rated steady-state performance at both high- and low-temperature rating points. Cycling losses were felt to be the dominant losses and were so significant as to cause a decrease in system performance with increasing outside air temperatures above the heat pump's balance point.

SYSTEM ECONOMICS

A major question that must be asked of every new system is how does it compare with available alternatives? An attempt to answer the question was made by studying both the performance and economics of the ACES, a heat pump system, a solar system, and a conventional electric heating and cooling system in different locations in the United States.[4]

All systems evaluated were structured to provide the same services as an ACES, namely space and water heating and space cooling. The following assumptions were made in the analysis: (1) the period of evaluation (life cycle) is 20 years; (2) the real discount rate (time value of money after accounting for both inflation and taxes) for an individual is 2%; (3) the real escalation rate for the cost of electricity is 0%; and (4) all systems evaluated are readily available and mass produced. The results of the analyses for a number of alternative systems in a 167-m^2 (1800-ft^2) well-insulated residence in the Knoxville, Tennessee, area are summarized in Fig. 8. The ordinate is the 20-year life cycle cost of a given system, which is the sum of the initial installed cost and the present value of all maintenance and operating costs over the assumed 20-year life of the system. Operating costs are based on 4¢/kWh power and component replacement costs are included in the maintenance costs. The abscissa is the estimated annual energy consumption of the system in the Knoxville area. Figure 8 shows that, with the exception of the solar energy system, the life-cycle costs of the various systems are not greatly different, although there are appreciable differences in annual energy consumptions. Table 4 illustrates how the ACES compares both in performance and economics in several U.S. cities with an ARI 2.78 heat pump offered by a major retailer. This heat pump was used in the comparison because of the availability of accurate cost information. The table represents the results of a computer simulation of the system's performance in a house identical to the ACES house.[4] Figure 8 and Tab. 4 both show that the more energy-efficient systems have higher initial costs. Because the residential HVAC market reacts strongly to first-costs, it appears likely that federal or state incentives such as those now afforded solar energy will be necessary before the ACES will enter the market in substantial numbers.

In computing life cycle costs, the choice of the discount rate has a marked impact on the values derived. Two percent was used in the analyses leading to Fig. 8 and Tab. 4, but values can be found in the literature that range from 0% (Office of Technology Assessment) to 10% (Department of Energy) and higher.[9] Rates inferred by economists to represent consumer behavior may range as high as 100%.[10,11] Figure 9 indicates the effect that the assumed value of the real discount rate would have on the choice between three HVAC systems if power costs were 4¢/kWh and life cycle costs were the basis of selection.

CONCLUSIONS

During a three-year period of field testing at Knoxville, Tennessee, the ACES has demonstrated an exceptionally high level of performance efficiency and has given reliable, essentially trouble-free operation. Algorithms

for simulating the performance of the system have been developed which give good agreement with the experimental results obtained at the test complex located on the University of Tennessee campus. The results obtained from the simulation of ACES operation in different climatic locations throughout the United States indicate that the system is broadly applicable in many regions of the country and can yield substantial energy savings relative to conventional HVAC systems.

Although the first costs of the ACES can exceed those of conventional systems by a factor of 2 or more, depending upon the climatic region, the annual operating costs promise to be much lower than those of alternative systems. In addition, the maintenance costs for the ACES are likely to be lower than those of comparable heat pump systems. As a result, even at today's power costs, the life cycle cost of the ACES is competitive with those of conventional systems in most parts of the country. Nevertheless, unless special subsidies or incentives are provided to offset the incremental first costs, it is unlikely that the ACES will achieve appreciable market penetration in the near future.

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TABLE 1. ACES steady-state COP

Mode	COP		
	1977/1978	1978/1979	1979/1980
Space heating and water heating (ice source)	2.70	2.59	2.59
Space heating only (ice source)	*a*	2.51	2.51
Water heating only (ice source)			
Summer	2.50	2.18	2.00
Winter	2.80	2.67	2.60
Space cooling			
Ice source	12.40	12.55	19.60
Chilled-water source	8.69	6.70	12.00

[a]This mode was not available during the 1977/1978 heating season.

TABLE 2. ACOP and power consumption, 1977–1980

Year	System	ACOP	Power consumption (kWh)
1977/1978	ACES	2.78	9,012
	I^2R space and water heating, central air-conditioner	1.13	20,523
1978/1979	ACES	2.80	6,719
	ARI-2.46 heat pump, I^2R water heating	1.42	12,853
1979/1980	ACES	3.08	6,447
	ARI-3.11 heat pump, I^2R water heating	1.73	11,358
Calculated[a]	ACES	3.18	6,437
	ARI-3.11 heat pump	1.67	11,283
	ARI-2.46 heat pump	1.38	14,820
	I^2R heat system	1.15	18,238

[a]Values for a full calendar year calculated by the MAD (Ref 8) program based on long-term weather averages.

TABLE 3. Heating season performance of the control house heat pumps

Month	Heating load (GJ)[a]	Average outdoor temperature (°C)	Supplemental electric heat required (kWh)	Total purchased power (kWh)	Monthly heating COP
			1978/1979 season[b]		
December	7.29[c]	3.3	120[c]	1211	1.67[b]
January	12.12	−0.2	336[c]	2042	1.65
February	8.41	2.6	221	1592	1.47
March	3.92	10.5	14	727	1.50
April	1.11	14.4	4	214	1.43
Total	32.85	6.2	695	5787	1.58
			1979/1980 season[d]		
December	7.70	3.9	118	1122	1.91
January	8.35	4.7	160	1090	2.13
February	9.95	0.8	266	1401	1.97
March	6.49	8.3	144	896	2.01
April	1.68	14.3	4	256	1.82
Total	34.18	6.4	692	4765	1.99

[a] 1 GJ = 0.948×10^6 Btu.

[b] Standard quality heat pump, ARI-rated COP of 2.46 at 8.3°C (47°F) and 1.7 at −8.3°C (17°F).

[c] Estimated.

[d] High-efficiency heat pump, ARI-rated COP of 3.11 at 8.3°C and 2.07 at −8.3°C.

TABLE 4. Comparison of performance and economics between the ACES and a heat pump system[a]

City	Annual energy consumption (kWh)	ACOP	First costs (1979 $)	Present worth of energy costs (1979 $)	Present worth of maintenance costs (1979 $)	20-year life-cycle cost (1979 $)
Albuquerque, NM						
ACES	6,404	3.39	8,596	5,987	1,929	16,512
Heat pump[b]	15,054	1.44	2,843	14,075	2,010	18,928
Lake Charles, LA						
ACES	9,706	2.16	5,958	5,493	1,929	13,380
Heat pump	14,357	1.46	3,067	8,125	2,010	13,202
Atlanta, GA						
ACES	6,733	3.03	8,038	4,296	1,929	14,264
Heat pump	14,313	1.43	2,885	9,131	2,010	14,026
Kansas City, MO						
ACES	7,409	3.42	9,949	4,830	1,929	16,708
Heat pump	17,839	1.42	2,957	11,630	2,010	16,597
Knoxville, TN						
ACES	6,437	3.18	8,316	3,208	1,929	13,453
Heat pump	14,295	1.43	2,885	7,124	2,010	12,019
Philadelphia, PA						
ACES	7,784	3.07	9,556	6,068	1,929	17,553
Heat pump	16,731	1.43	2,843	13,042	2,010	17,895
Richmond, VA						
ACES	6,670	3.14	8,298	5,644	1,929	15,872
Heat pump	14,624	1.43	2,885	12,374	2,010	17,269
San Diego, CA						
ACES	3,497	3.05	5,871	3,011	1,929	10,811
Heat pump	8,148	1.31	2,843	7,016	2,010	11,869
Seattle, WA						
ACES	7,331	2.73	7,339	1,360	1,929	10,628
Heat pump	13,650	1.47	2,843	2,527	2,010	7,380
Minneapolis, MN						
ACES	11,660	2.87	13,144	9,412	1,929	24,486
Heat pump	26,465	1.26	2,985	21,363	2,010	26,358

[a]Taken from Ref 4.

[b]ARI ratings. COP = 2.75 at 8.3°C; COP = 1.8 at −8.3°C; energy consumption figures include an electric resistance water heater.

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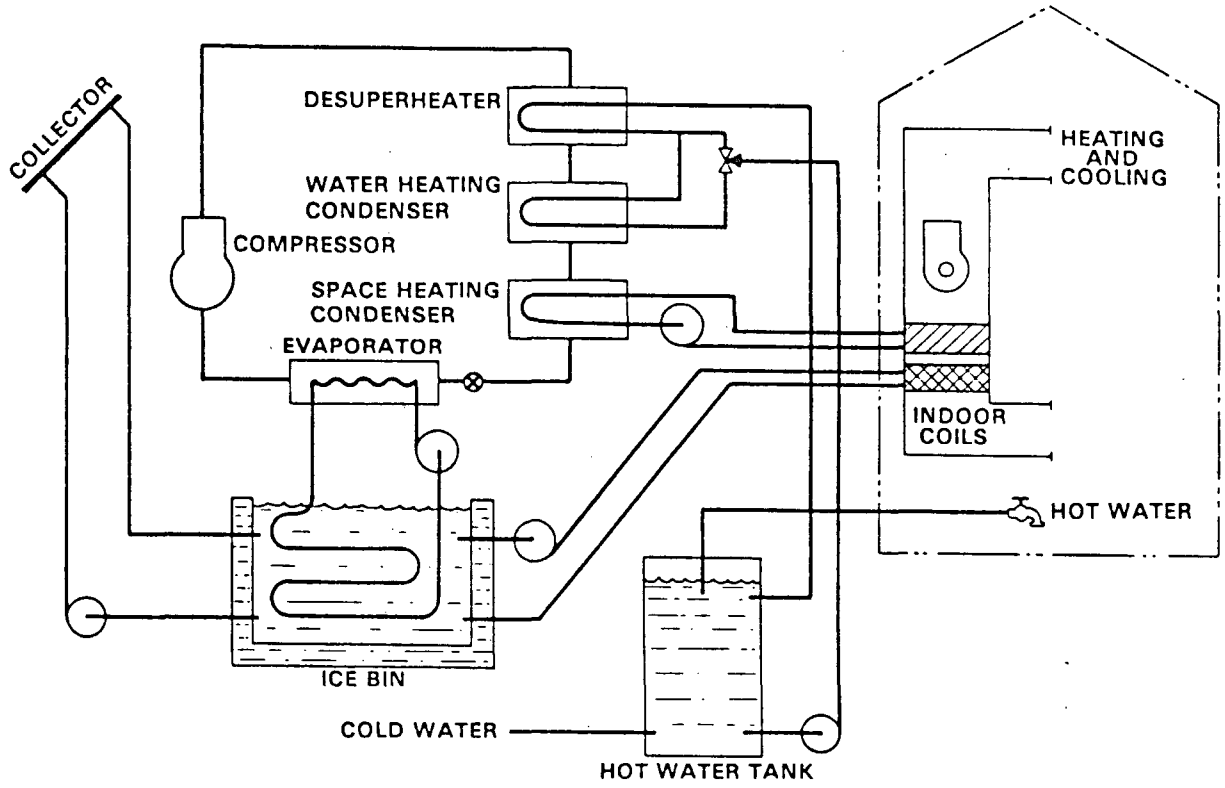

Fig. 1 Typical ACES schematic

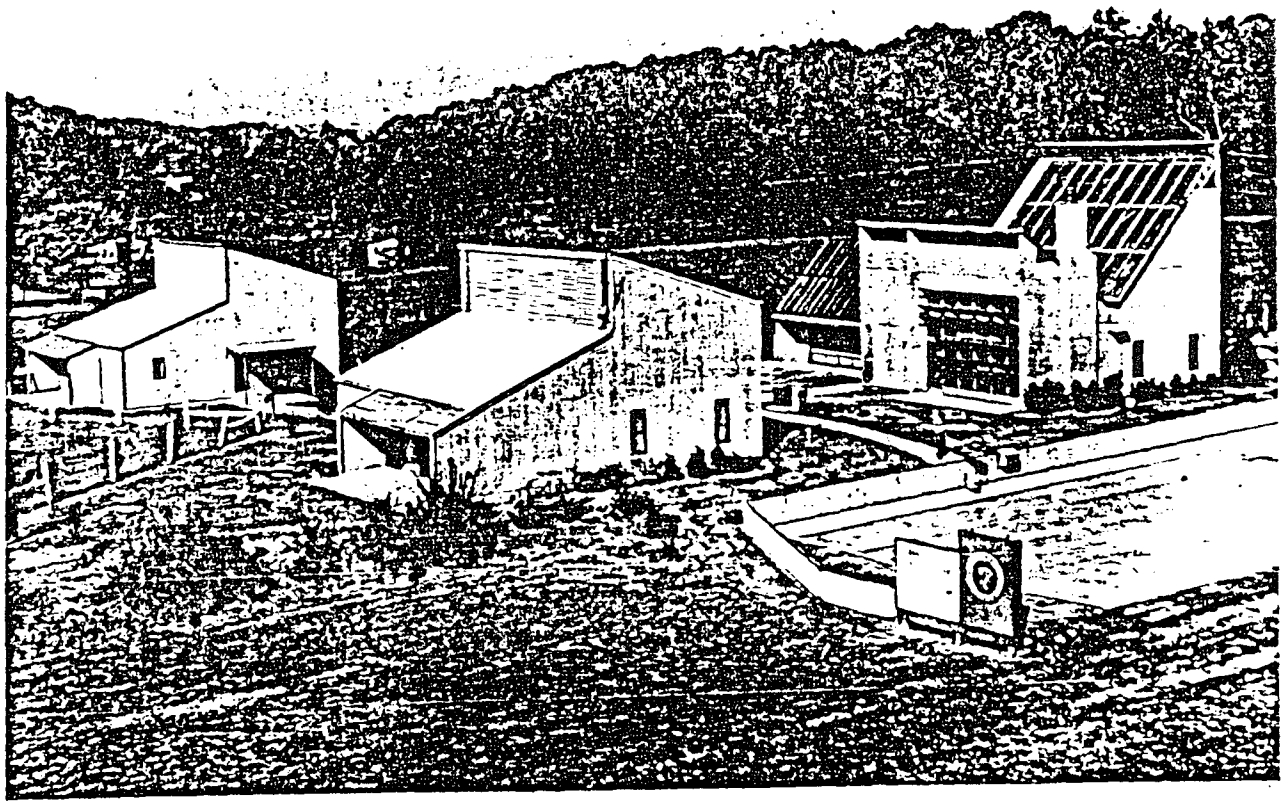

Fig. 2 Photo of TECH complex
l. to r.; control house, ACES house
Solar house

ORNL-DWG 79-20376R

- INTERSEASONAL ENERGY TRANSFER
- UTILIZES BOTH OUTPUTS
- CONSTANT-CAPACITY HEAT PUMP OPERATION
- DIURNAL LOAD MANAGEMENT

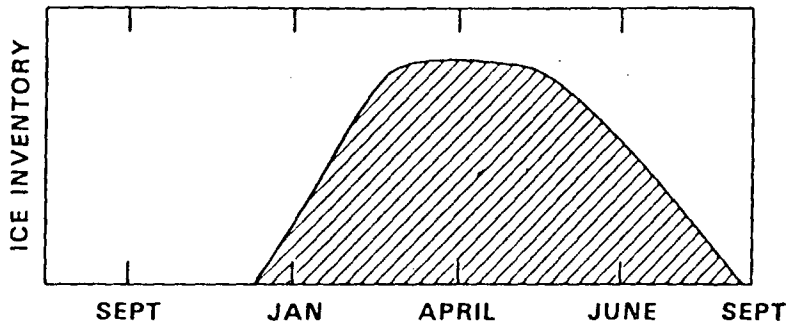

Fig 3. Full ACES ICE INVENTORY

ORNL-DWG 79-20375R

- UTILIZES BOTH OUTPUTS (SUMMER)
- CONSTANT-CAPACITY HEAT PUMP OPERATION
- DIURNAL LOAD MANAGEMENT

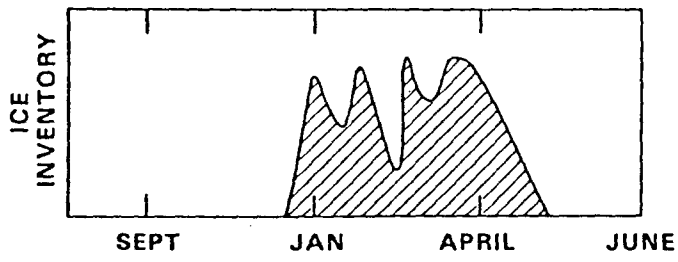

Fig 4. MINIMUM ACES ICE INVENTORY

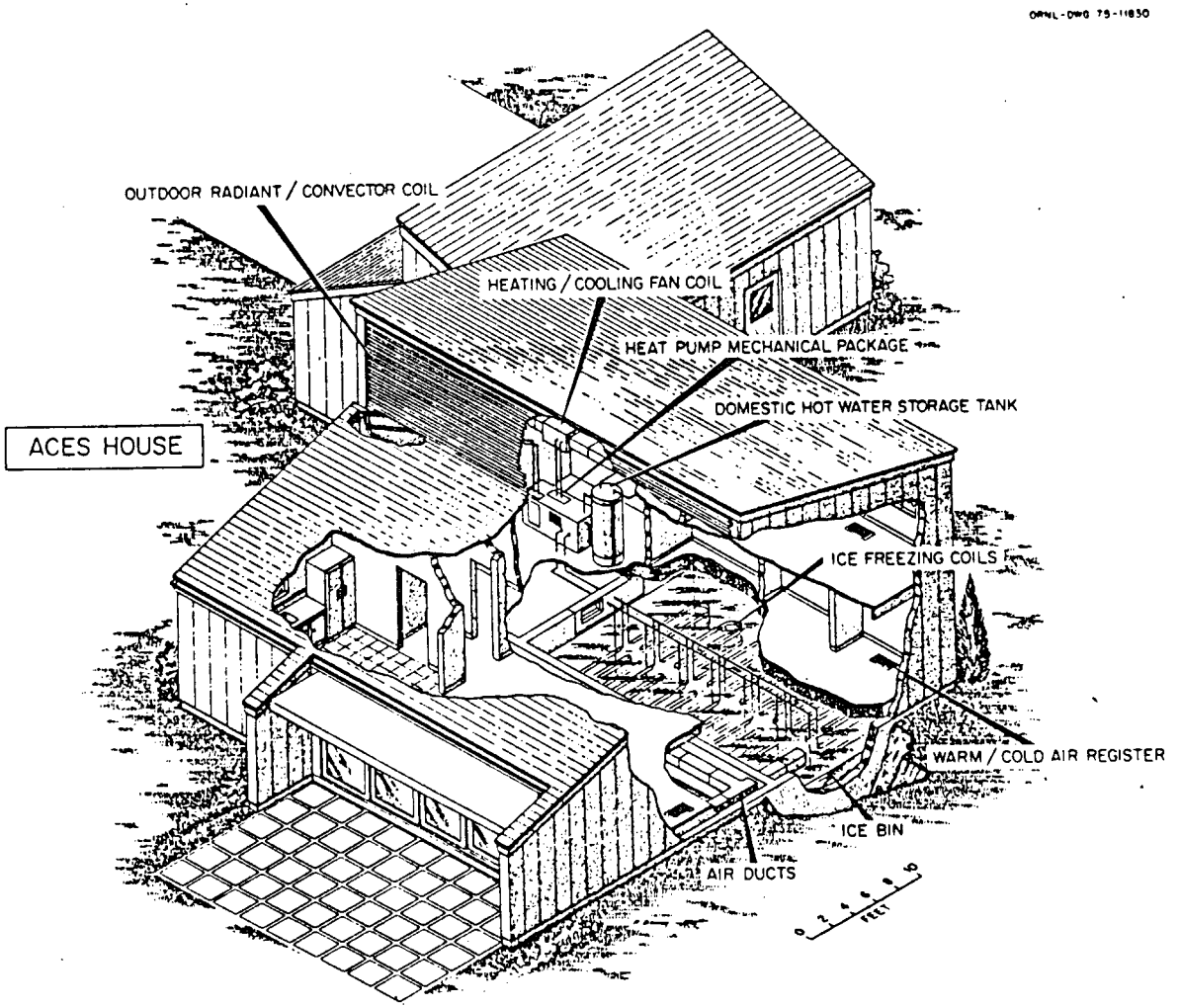

Fig. 5 Cutaway view of ACES house

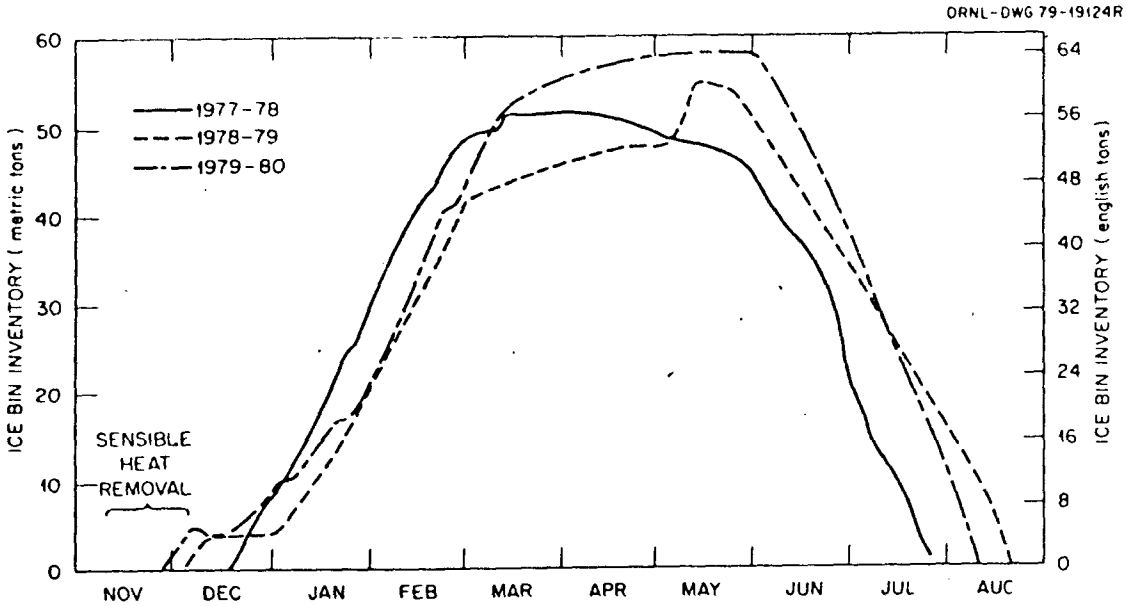

Fig. 6 Ice inventory, 1977-1980

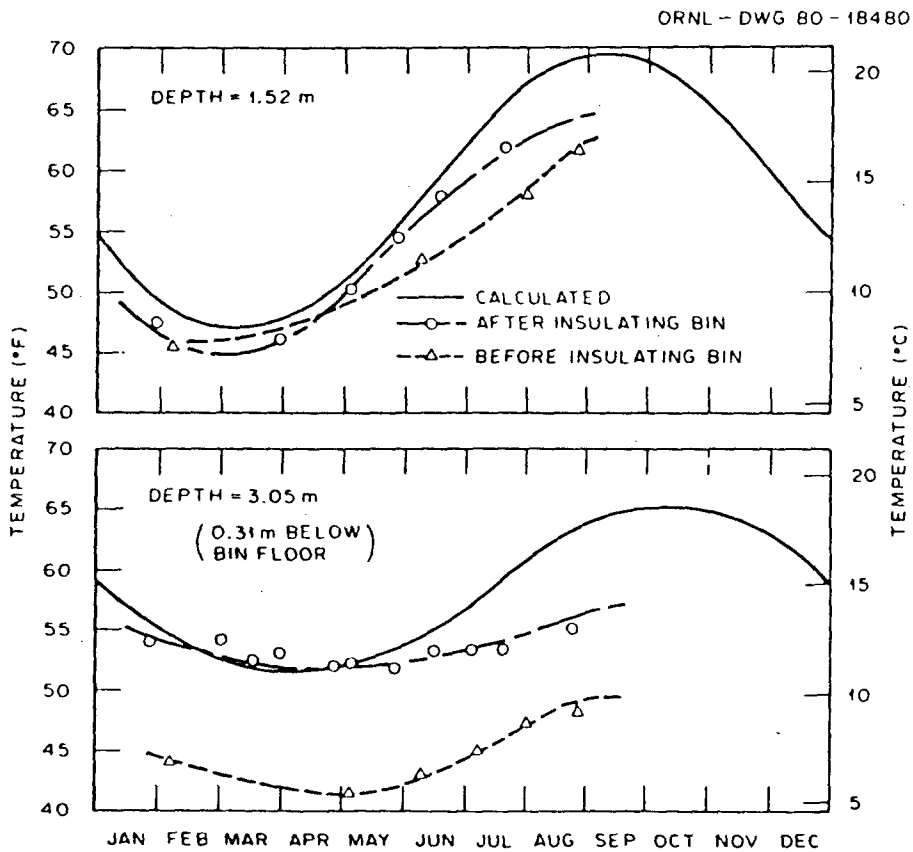

Fig 7. Earth temperatures near ice bin

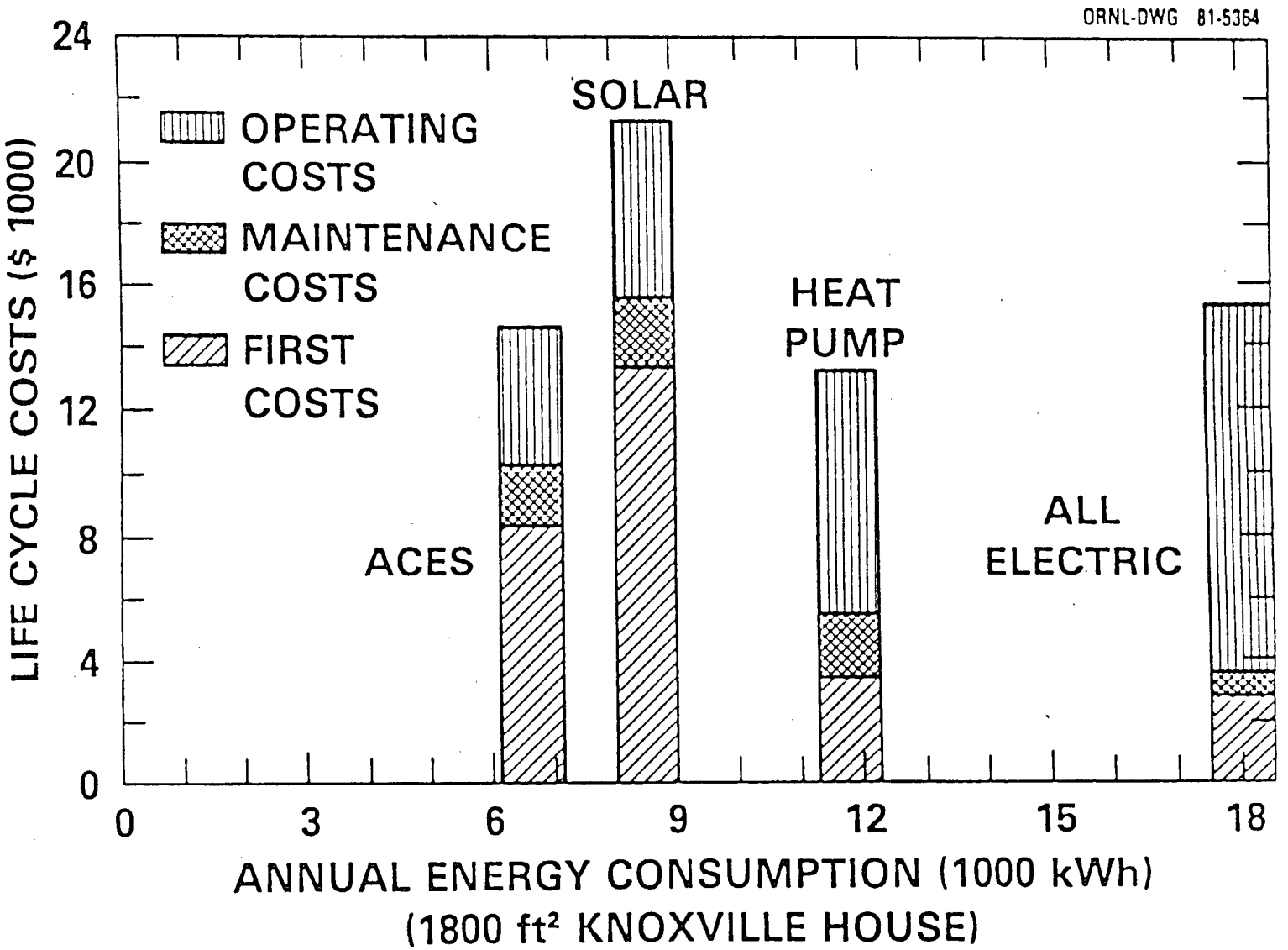

Fig. 8 System life cycle costs

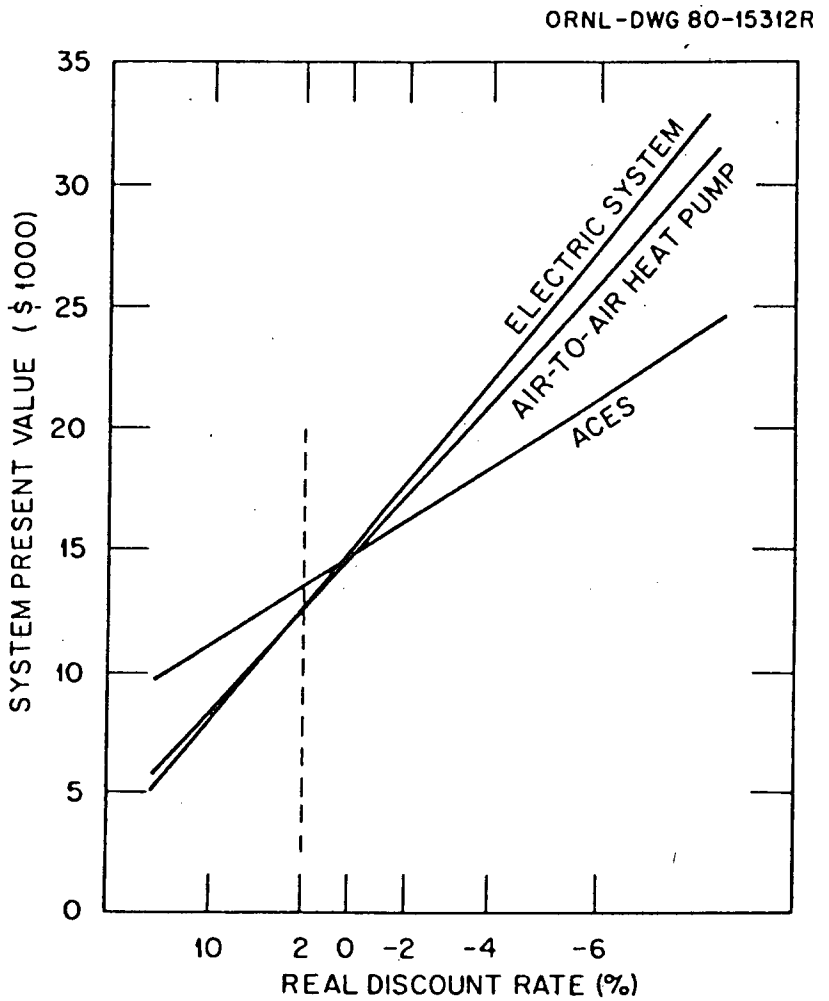

Fig. 9 Life cycle cost vs real discount rate