VAPOR COMPRESSION HEAT PUMP SYSTEM FIELD TESTS AT THE TECH COMPLEX

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ABSTRACT

The Tennessee Energy Conservation In Housing (TECH) complex has been utilized since 1977 as a field test site for several novel and conventional heat pump systems for space conditioning and water heating. Systems tested include the Annual Cycle Energy System (ACES), solar assisted heat pumps (SAHP) both parallel and series, two conventional air-to-air heat pumps, an air-to-air heat pump with desuperheater water heater, and horizontal coil and multiple shallow vertical coil ground-coupled heat pumps (GCHP). A direct comparison of the measured annual performance of the test systems was not possible. However, a cursory examination revealed that the ACES had the best performance, however, its high cost makes it unlikely that it will achieve wide-spread use. Costs for the SAHP systems are similar to those of the ACES but their performance is not as good. Integration of water heating and space conditioning functions with a desuperheater yielded significant efficiency improvement at modest cost. The GCHP systems performed much better for heating than for cooling and may well be the most efficient alternative for residences in cold climates.

INTRODUCTION

A heat pump can be generally defined as a machine used to transfer heat from a source at one temperature to a sink at a higher temperature. The purpose of this heat transfer can be either to cool the source (e.g., the interior of a house in summer, food in a refrigerator, etc.), to heat the sink (e.g., a house interior in winter, hot water, etc.), or both (e.g. heat water while storing ice for future cooling use.) Most heat pumps fit into two basic categories: vapor compression or absorption. Since the field tests discussed in this paper involved electrically driven, vapor compression systems, only the vapor compression cycle will be considered herein.

The field tests were conducted as part of the Tennessee Energy Conservation in Housing (TECH) program, initiated in 1975 as a joint venture of the Energy Research and Development Administration (forerunner of the Department of Energy (DOE)), the Oak Ridge National Laboratory (ORNL), the University of Tennessee (UT), and the Tennessee Valley Authority (TVA) with the objective of demonstrating methods of conserving energy in residences. Construction of the TECH complex was begun in 1976 when the first three houses and a garage/storage building were built. In 1979 the garage was converted into a fourth test structure and a fifth house and new storage building were added in 1981. Figure 1 is an aerial view of the site as it now exists on the UT Institute of Agriculture Experiment Station grounds near Knoxville, Tennessee.

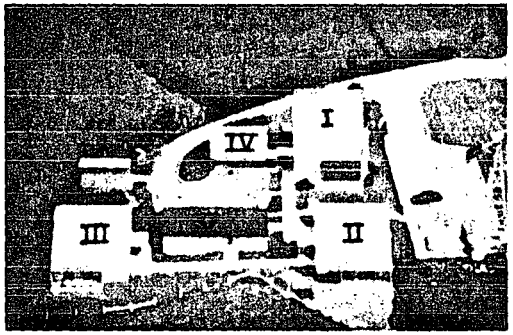

Figure 1. The Tennessee Energy Conservation in Housing Complex.

The three original houses are of identical floor plan (house III is a mirror image of houses I and II) with 170 m^2 of conditioned floor space and, insofar as possible, of identical construction. Several non-standard construction features are incorporated into the houses including exterior walls of 50.8 mm by 152.4 mm (2 in x 6 in) studs on 0.61 m (2 ft) centers to accommodate thicker wall insulation and air lock entries and magnetic weather stripping to minimize air infiltration. Complete construction details and floor plans may be found in Ref. [1] for house I and Refs. [2] and [3] for houses II and III.

THE VAPOR-COMPRESSION REFRIGERATION CYCLE

Mechanical compression systems (either electrically or engine-driven) are predominant among refrigeration cycles used by heat pumps. The coefficient of performance (COP) expresses the effectiveness of a refrigeration or heat pump system. It is a dimensionless ratio defined by the expression:

$$COP = \frac{\text{Useful heating or cooling effect}}{\text{Purchased energy required}}$$

A schematic of an ideal vapor-compression cycle along with temperature-entropy (T-s) and pressure-enthalpy (P-h) diagrams is shown in Fig. 2. Saturated, low-temperature, low-pressure vapor (state 3) is compressed to a higher temperature and pressure (state 4) by work input through the

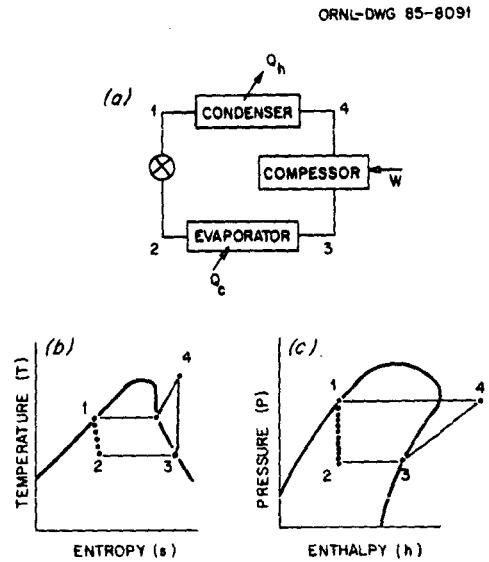

Figure 2. The ideal vapor-compression cycle.

compressor (W). Heat (Q_h) is released by the hot refrigerant through the condenser, heating its surroundings and leaving the refrigerant in a saturated liquid state (state 1). The liquid is throttled through the

expansion device to state 2. Heat (Q_c) is absorbed by the cold refrigerant through the evaporator, cooling its surroundings. The refrigerant leaves the evaporator as a saturated vapor at state 3 completing the cycle. Applying the steady-state steady-flow energy equation from the first law of thermodynamics on a unit mass flow basis we have:

$$W = h_4 - h_3,$$
$$Q_h = h_4 - h_1,$$
$$Q_c = h_2 - h_3, \text{ and}$$
$$h_1 = h_2.$$

COP's for heating (COP_h) and cooling (COP_c) are:

$$COP_h = \frac{Q_h}{W} = \frac{h_4 - h_1}{h_4 - h_3}$$
and
$$COP_c = \frac{Q_c}{W} = \frac{h_3 - h_2}{h_4 - h_3}$$

The COPs as defined above are ideal compressor-only COPs. In an actual system parasitic power for pumps and/or fans is required to transport the deliverable heating or cooling where it is needed. Thus, for a real system W would include power use for those parasitic devices as well as for the compressor.

The thermodynamic cycle of an actual system will depart significantly from the ideal cycle of Fig. 2. This departure occurs because of pressure drops within the system components (principally the compressor), heat losses from the compressor and connecting piping, compressor motor inefficiency, and temperature differences across system heat exchangers. These factors result in a significant departure from maximum potential performance.

The net effect of these departure or loss mechanisms is illustrated clearly in Fig. 3. An ideal vapor-compression cycle using R-22 as the refrigerant would have a COP of 21.45 when it is operating between the temperature limits of 21 and 8°C. Temperature differences across the heat exchangers, a necessary condition for operation of a real system, reduce the computed COP substantially. Heat exchanger refrigerant temperatures observed during steady-state tests of an air-to-air heat

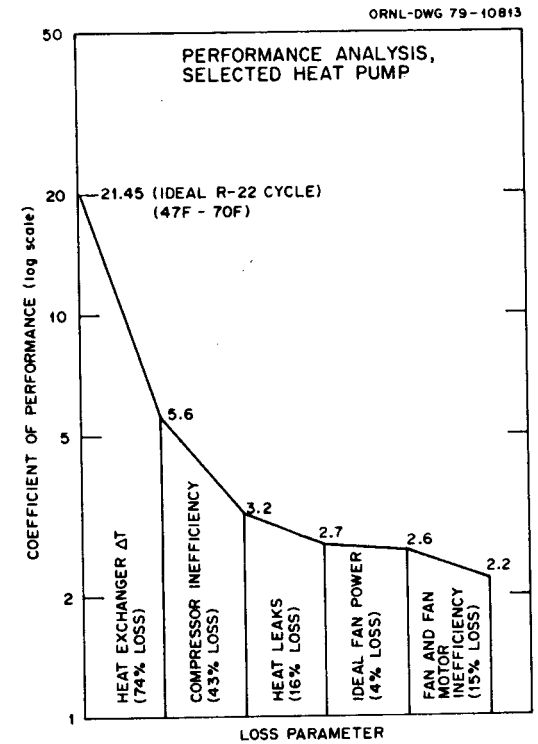

Figure 3. Effect of component inefficiencies on system efficiency.

pump, typical of those sold before 1978, at 8°C outdoor air temperature and 21°C indoor air temperature were 50°C condensing and -1.2°C evaporating. The computed COP using these temperature differences in an otherwise ideal system is 5.6, a 74% reduction from the maximum potential COP.

Departure from ideal operation within the compressor reduces the COP to 3.2, an additional 43% loss. Heat losses observed at the compressor shell reduce the COP an additional 16% to 2.7. Ideal fan power consumption (necessary to transport the heat throughout a house) would reduce the COP to 2.6; the observed fan and fan-motor inefficiencies

reduce the COP an additional 15% to the value of 2.2, which is about the observed value for the heat pump operating in steady state at these conditions [4].

More general information about vapor-compression and other heating, refrigeration, and air-conditioning systems may be obtained from references [5-7]. For an excellent overview of heat pumps and factors affecting their performance, see [8].

THE FIELD TESTS

Tech House I Tests

Two different heat pump systems have been tested in TECH house I. From 1979 through 1981 a series solar assisted heat pump (SSAHP) system was tested. In 1981 a horizontal coil ground coupled heat pump was installed and is still under test.

Series Solar Assisted Heat Pump. The SSAHP system is shown schematically in Fig. 4. Two different thermal storage tanks were available. The first, a 4.2 m^3 insulated concrete tank buried just below the surface of the ground in the crawlspace, was used in 1979-80 while the second, a 7.6 m^3 uninsulated steel tank buried 2.1 m deep just outside the house, was used in 1980-81.

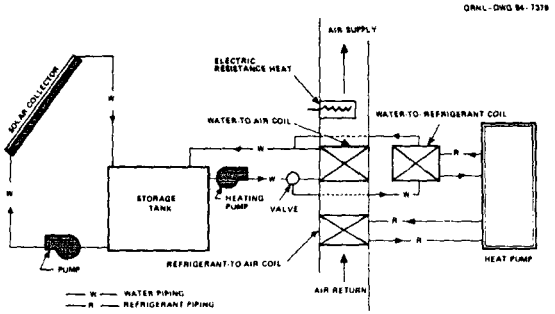

Figure 4. Series Solar-assisted heat pump (SSAHP) system schematic.

The system had a modified air-to-air heat pump added to a direct solar system to supply auxiliary heat in lieu of electric resistance heat. A refrigerant-to-water heat exchanger was added to the heat pump to serve

as the evaporator in the space heating mode. Water was circulated to this heat exchanger from the solar storage tank. An 8 kW electric resistance heater was used for third stage heating with the SSAHP system and the outdoor coil of the heat pump was utilized as the heat pump condenser in the cooling mode. Thus, the SSAHP could heat the space by extracting heat from the solar storage tank through the water-to-refrigerant coil and cool the space while rejecting heat through the outdoor coil. No domestic water heating was done by the SSAHP system.

Table I gives a summary of the SSAHP monthly and seasonal performance factors (SPF) for space heating-only. These SPFs are calculated by:

$$SPF = \frac{Q_t}{EC_t}$$

where

Q_t = total heating/cooling energy delivered (monthly or seasonal)

and

EC_t = total energy consumed to provide Q_t (monthly or seasonal)

Table I. Seasonal Performance Factors (SPF) for SSAHP system.

	Nov.	Dec.	Jan.	Feb.	Mar.	Season
1979-80 Season	4.55	4.43	1.70	1.80	3.27	2.42
1980-81 Season	–	2.74	2.67	3.06	5.43	3.16

Cooling load was not measured during the test period; therefore, cooling performance factors are not presented. When the system was switched from the insulated tank to the uninsulated tank a significant reduction in peak power consumption (11.2 kW to 4.3 kW) as well as an increase in SPF was realized because no electric resistance heat was required during the second test season. Heat transfer from the ground through the uninsulated steel tank and the increased storage volume appeared to contribute to this peak reduction. Other unquantified variables (weather severity, etc.) also played a role [12]. SSAHP heating SPFs for both years were higher than those of the air-source heat pumps tested in house III.

However, installation cost of an SSAHP is likely to be much higher than that of the conventional system. A more complete description of the SSAHP system and test results can be found in Refs. [12-14].

Horizontal Coil Ground-Coupled Heat Pump System. In 1981 and 1982 a horizontal coil ground-coupled heat pump (GCHP) system was installed. The system consisted of a unidirectional, water-to-water heat pump, an air handler with a water-to-air heat exchanger for distributing conditioned air to the space, and a ground-coupled heat exchanger. A 20% by weight methanol-water brine solution was used in the ground and fancoil loops to guard against freezing of the heat exchangers. Manually controlled three-way valves were used to direct the hot and cold brine streams to the proper heat exchangers for space cooling and heating. Fig. 5 is a schematic of the heat pump package.

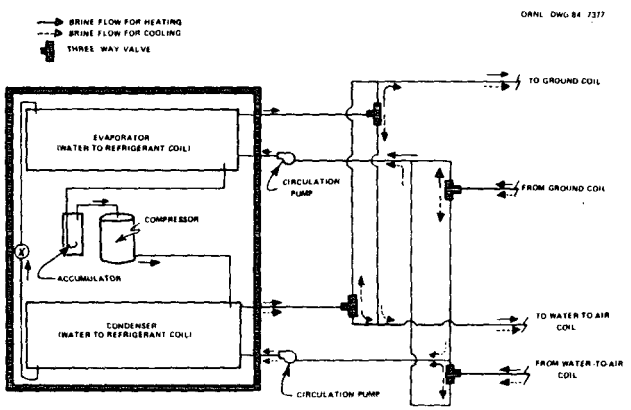

Figure 5. Ground-coupled heat pump (GCHP) mechanical package schematic.

The ground coil is 206 m long and consists of 32 mm nominal diameter polybutylene pipe buried approximately 1.2 m deep. A schematic of the coil layout is shown in Fig. 6.

Table II presents monthly and seasonal loads and performance factors for the 1983-1984 heating season and the 1984 cooling season. Since the heat pump mechanical package could be located either inside or outside the conditioned space, two seasonal performance factors (for space conditioning only) are presented [15].

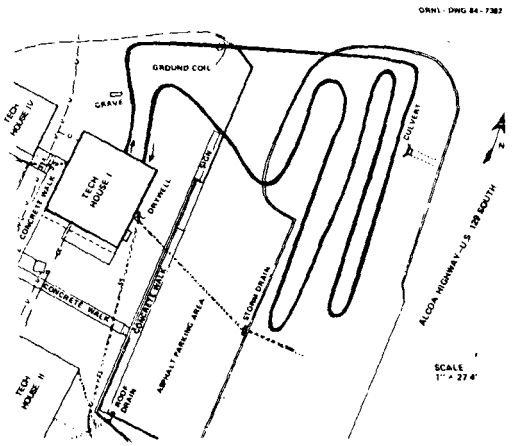

Figure 6. Layout of TECH house I ground coil.

SPFi is the performance factor for the case in which the heat pump package is located within the conditioned space as is the case for the test installation and is calculated by:

$$SPF_i = \frac{Q_{t_n}}{EC_t}$$

where Q_{t_n} = net heating/cooling delivered.

For the heating season

$$Q_{t_n} = Q_g + EC_t$$

where Q_g = heat extracted from the ground,

and for the cooling season

$$Q_{t_n} = Q_{l_c} - Q_i$$

where Q_{l_c} = cooling energy delivered through air handling unit

and

Q_i = heat losses to internal space from circulation pump(s) and compressor.

Table II. Horizontal coil GCHP heating and cooling performance factors.

Month	Load (kWh)	SPFi	SPFo
Nov. 83	1,609	2.80	2.30
Dec. 83	3,633	2.80	2.30
Jan. 84	3,401	2.50	2.00
Feb. 84	2,367	2.50	2.00
Mar. 84	2,308	2.40	2.20
Heating Season	13,318	2.59	2.16
June 84	1,242	1.34	1.72
July 84	1,296	1.43	1.70
Aug. 84	1,661	1.35	1.69
Sep. 83	416	1.20	1.79
Cooling Season	4,604	1.35	1.71
Annual Total	17,922	2.10	1.98

SPFo is the performance factor for the case where the heat pump package is located in an unconditioned space (i.e. a garage). It is calculated by

$$SPFo = \frac{Ql_{h/c}}{EC_t}$$

where

$Ql_{h/c}$ = heating/cooling delivered through air handling unit

As can be seen from Table II, locating the heat pump package within the conditioned space is clearly beneficial during the heating season but is a disadvantage for cooling operation. It is also clear that the GCHP's heating performance is very good but its cooling performance (even had the heat pump been outside the space) was poor.

This poor cooling performance was due to inadequate ground heat transfer. The primary causes were (1) the ground tended to dry out over the course of the summer resulting in reduced soil thermal conductivity,

and (2) the ground coil was undersized for cooling [16]. Cooling performance was also adversely affected by the fact that the heat pump used is not designed for space cooling applications.

Tech House II Tests

Systems tested in house II were the Annual Cycle Energy System (ACES) from 1976 through 1982 and a multiple vertical coil GCHP from 1983 to the present.

ACES System Tests. From November, 1977, through September, 1982, TECH house II was used for testing of a residential ACES [9,10,11]. ACES is an integrated space conditioning and water heating system consisting of a unidirectional heat pump with low-side thermal storage and, if needed, an auxiliary solar collector or outdoor fancoil. Fig. 7 is a schematic showing the basic ACES components. During winter the heat pump extracts energy from an insulated storage tank filled with water

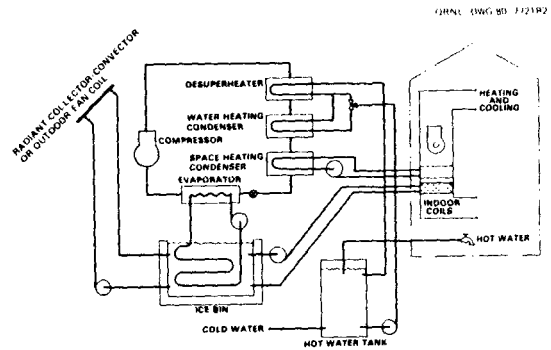

Figure 7. Typical Annual Cycle Energy System (ACES) schematic.

and boosts it to provide space and water heating. As the heating season progresses, the water is converted into ice to be used for space cooling the following summer. The solar collector or fancoil would be used to supply extra heat to the storage tank (in areas where more ice could be made than was needed) or to provide a heat sink for late-summer, off-peak air-conditioning operation (for areas where the winter's ice production is insufficient to meet the building cooling needs). Use of the thermal storage gives the ACES its two primary energy conserving features: (1) a

constant-capacity heat pump (eliminating the need for electric resistance backup heating) and (2) interseasonal transfer of cooling energy from winter to summer (greatly reducing energy consumption for cooling). A review of the theory and design requirements of residential ACES is given in [17].

Table III gives the results of four annual test cycles performed at house II. The annual performance factors (APFs) given in Table III are for space conditioning and water heating and are calculated by

$$APF = \frac{\text{Space heating} + \text{space cooling} + \text{water heating}}{\text{annual purchased energy consumption}}$$

During the 1977-78 test it became apparent that heat transfer from the earth to the thermal storage bin was much higher than anticipated causing early summer ice exhaustion [9]. Insulation levels on the bin walls, floor, and ceiling were increased in October, 1978, to reduce this heat transfer level. Stored ice was maintained until late August of the 1978-79 and 1979-80 test cycles as a result [10]. Indoor air flow and fan energy use for cooling were reduced in 1980 with a resultant increase in annual performance factor (APF). Component tests in 1981 indicated that use of an outdoor fancoil rather than a solar collector/convector would significantly increase late-summer off-peak cooling operation [11]. This feature was incorporated prior to the 1981-82 test year.

The ACES is the most energy conserving system for providing annual space conditioning and water heating needs for a Knoxville area residence that we have tested. Annual energy use for providing these services was

Table III. Annual Cycle Energy System test results.

Test Year	1977-1978	1978-1979	1979-1980	1981-1982
Test Period	11/1/77-9/18/78	12/1/78-9/30/79	12/1/79-9/1/80	9/28/81-9/28/82
Space Heating (GJ)	43.07	32.85	34.40	37.31
Space Cooling (GJ)	26.21	19.17	22.65	24.40
Water Heating (GJ)	20.84	15.81	14.34	14.16
Annual Energy Use (kWh)	9012	6719	6447	7360
APF	2.78	2.80	3.08	3.05

about 50% less than that required by a high efficiency air-source heat pump/electric water heater system in TECH house III [10]. A survey of field test results of numerous solar systems indicated that ACES has a better APF than solar systems as well [18]. Economic analyses have shown that energy savings for ACES will take about 25 years to pay back its incremental first cost (about $7,000-$9,000 for the Knoxville system in a mature industry) relative to air-source heat pump systems under present economic conditions [19-21]. The estimated performance of ACES across the United States is shown in Fig. 8. This map shows that ACES is most efficient in the central U.S. where there is a good balance between heating and cooling needs.

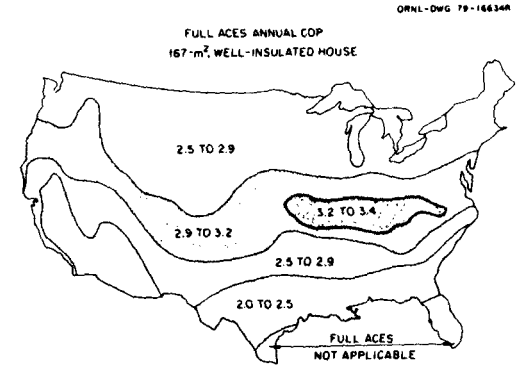

Figure 8. Full ACES APFs for the United States for a 167-m^2, well-insulated house.

Multiple Shallow Vertical Coil Ground-Coupled Heat Pump. Between October, 1982, and June, 1983, the GCHP system was installed in TECH house II. As illustrated schematically in Fig. 9, this system consisted of a water-to-water heat pump with a water-to-air fancoil, identical to those used for the TECH house I GCHP, and a ground coil composed of six vertical heat exchangers. These heat exchangers ranged in depth from 18.3 to 30.5 m and were composed of 38.1 mm diameter high density polyethylene tubing arranged in a U-tube configuration. As for the house I system, a 20% by weight methanol-water brine was used in the ground and fancoil loops. Testing has just been completed and reports will be available in the future.

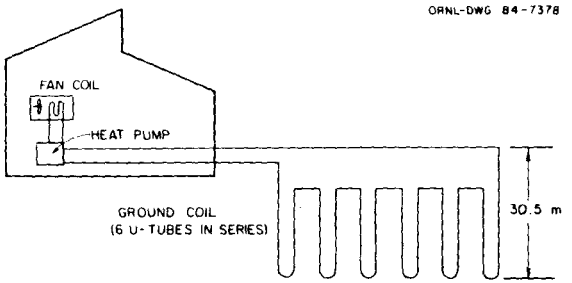

Figure 9. TECH house II ground coupled heat pump (GCHP) schematic.

Table IV gives the results of seasonal performance factor tests (for space conditioning only) from July, 1983, through April, 1984. The SPFi's include the effect of the intermediate brine loop between the heat pump and the air handler and the crankcase heater energy consumption. Actual installations would probably use water-to-air heat pumps so SPFi would be somewhat better than that of this experimental system. The same, of course, holds true for the horizontal coil GCHP. Both the heating and cooling SPFi's were better than those of the TECH house I system primarily because most of the house II ground coil was in saturated soil or rock and, thus, was not subject to wide temperature swings or summertime soil dryout associated with relatively shallow pipe burial depths. Heating performance was better than that of any of the air-source heat pumps tested in house III as well, but cooling performance was not as good as that of the high efficiency air-source heat pump. Unlike that of the house I system, however, the cooling performance of the vertical coil GCHP was reasonable. In addition, the overall APF (for space conditioning only) of the vertical coil GCHP was superior to that of the air-source systems.

Tech House III Experiments

Air-Source Heat Pump System Tests. During the 1978-79 and 1979-80 test years, two air-source heat pump/electric water heater systems were tested in house III [22]. A standard efficiency heat pump (COP = 2.46 at 8.3°C) was used for the 1978-79 season (heat pump I) while a high efficiency (COP = 3.11 at 8.3°C) unit from the same manufacturer (heat pump II) was used during subsequent years. Table V summarizes the results of the annual performance of each system.

Table IV. Multiple shallow vertical coil GCHP heating and cooling performance.

Period	Load (kWh)	Energy Consumption (kWh)	SPFi
6/27-7/29/83	2,094	1,174	1.78
7/29-8/24/83	2,200	1,284	1.71
8/24-9/30/83	1,233	701	1.76
9/30-10/10/83	122	76	1.73
Cooling Season	5,649	3,235	1.75
10/31-11/28/83	1,206	390	3.09
11/28/83-1/3/84 *	2,383	784	3.04
1/3-1/30/84	3,475	1,243	2.80
1/30-3/2/84	3,214	1,157	2.78
3/2-4/2/84	2,133	771	2.77
4/2-4/26/84	383	150	2.55
Heating Season	12,994	4,495	2.85
Annual Totals	18,443	7,730	2.39

*Does not include 12/22-12/30.

Comparing Tables III and V shows that the APFs (including water heating) of the air-source systems are lower than those of the ACES.

In addition to the system comparisons with ACES, it was desired to measure the actual seasonal performance of air-source heat pumps and compare it to their rated performance. Fig. 10 shows measured monthly heating COPs and rated steady-state COPs plotted against outdoor temperature for both heat pumps tested. It is interesting to note that, while the steady-state COP values increased with increasing temperature the field measured monthly values tended to decrease. Cycling losses at high ambient temperatures degraded performance by as much as 45% compared to rated steady-state performance.

After the 1980 test year, instrumentation was added to heat pump II in an attempt to measure directly the magnitude of the different dynamic losses that affect a heat pump's field performance (frosting, defrosting, and cycling) [23]. Table VI illustrates the fraction of measured annual energy consumption due to the various losses.

Table V. Air-source heat pump/electric water heater system test results.

	1978-1979	1979-1980
Test Period	12/1/78-9/30/79	12/1/79-9/15/80
Space Heating (GJ)	32.85	34.29
Space Cooling (GJ)	19.18	22.70
Water Heating (GJ)	13.74	13.57
Energy Cons. (kWh)	12853	11358
System APF	1.42	1.73
Heat Pump Only*		
SPF, heating	1.58	1.99
SPF, cooling	1.64	2.27
APF	1.60	2.10

* not including water heating.

Table VI. Heat pump II energy use due to dynamic losses 1982-83 heating and cooling seasons

Component	% Total Energy Consumed
Steady-state Energy Use to Meet Load	75.8
Frosting Losses	2.3
Defrosting Losses	5.9
Start-up Transient Losses	9.8
Off-cycle Parastics	6.2
Total	100.0

Air-Source Heat Pump/Desuperheater Water Heater System. During the 1981-82 test year, in alternate months with the dynamic loss measurement test, heat pump II was operated with a desuperheater water heater to determine its effect on system performance compared to that of the same heat pump with an electric resistance water heater [24]. The desuperheater was a small heat exchanger designed to recover excess or wasted heat from the refrigeration cycle and use it to heat domestic hot water. A schematic of the system is shown in Fig. 11, with test results given in Table VII.

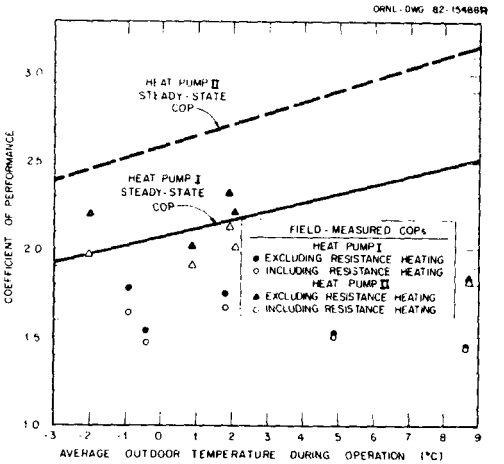

Figure 10. Air-source heat pump steady-state and field-measured COP vs. temperature.

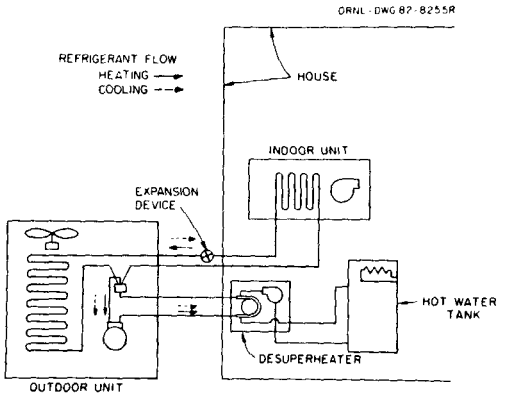

Figure 11. Schematic of air-source heat pump/desuperheater water heater system.

Test results showed a 14% improvement in APF for the desuperheater system over the electric water heater system due almost entirely to summertime heat recovery water heating by the desuperheater. Heating seasonal performance of the systems proved to be nearly identical, a result

Table VII. Performance of heat pump II with desuperheater vs. electric water heater, 1981-82

	Desuperheater	Electric
Space Heating (GJ)	18.58	19.65
Space Cooling (GJ)	13.85	11.96
Water Heating (GJ)	8.26	8.69
Energy Use (kWh)	5782	6525
SPF, Heating	1.68	1.63
SPF, Cooling	2.50	1.90
APF	1.96	1.72

not unexpected since in winter the desuperheater reduces the space heating capacity of the heat pump forcing increased reliance on electric backup space heating. Energy savings of the desuperheater system could be expected to recoup the incremental cost of the desuperheater unit in 7 to 9 years based on 1981 electricity costs.

Tech House IV Experiments

Parallel Solar Assisted Heat Pump System With and Without Off-peak Electric Heat Storage. The parallel solar assisted heat pump (PSAHP) system, illustrated schematically in Fig. 12, has three main components. A 29.4 m^2 air heating solar collector array collected solar energy when available. The individual collectors had a double glazing of fiberglass and teflon film and were inclined at 45° and oriented 10° west of south. Thermal storage was provided by a 7.4 m^3 pebble bed (located in the conditioned space) containing 25.4 mm nominal diameter crushed limestone. A standard 8.8 kW air-source heat pump provided backup heat for the system [25]. No water heating was done.

During the 1979-80 heating season an attempt was made to minimize on-peak energy consumption by using a 15 kW electric heater to heat the pebble bed during utility off-peak hours. In on-peak periods, the system operated like a conventional PSAHP. That is, solar heat or heat from storage, if available, was used to heat the space. If neither was available the heat pump provided space heating. When no space heating was required and solar energy was available, it was used to charge the pebble

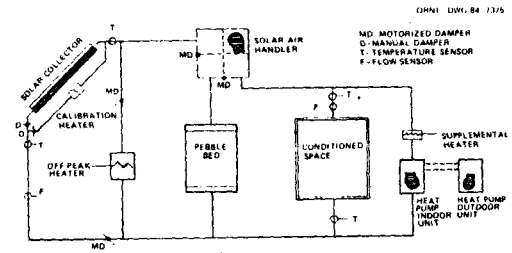

Figure 12. Parallel solar-assisted heat pump (PSAHP) system schematic.

bed storage. During the 1980-81 and 1981-82 heating seasons, the system was operated as a straight PSAHP without off-peak storage heating.

Table VIII is a summary of the system performance factors for all three heating seasons. The 1979-80 overall SPF of 1.61 was lower than that of subsequent years due to the off-peak electric heat storage feature. Eliminating that option resulted in much improved overall efficiency. From these results it is obvious that an overall SPF penalty must be paid in order to incorporate off-peak electric heating into the system. An on-peak to off-peak electric rate differential of greater than 7 to 1 would be required in order to offset the added energy consumption imposed by using electric heat for off-peak storage charging [26].

Table VIII. PSAHP system seasonal performance factors (space heating only).

Month	1979 - 1980 Season	1980 - 1981 Season	1981 - 1982 Season
November	2.59	2.65	2.95
December	1.77	2.52	2.16
January	1.32	2.01	1.39
February	1.46	2.39	2.55
March	1.68	3.59	3.87
Season	1.61	2.56	2.11

SYSTEM COMPARISON SUMMARY

A cursory comparison of the performance of the field tested systems is included in this section. It must be clearly understood, however, that this comparison is based on test results from a single location. They should not be construed to be generally applicable to all locations.

A direct annual performance comparison of the various systems tested at the TECH site is not possible since not all delivered the same loads under the same weather conditions. However, a rough comparison, made by assuming certain load deliveries, is offered in Table IX. For this comparison, all systems that delivered no water heating were assumed to have an electric water heater delivering 14.2 GJ/yr (same as ACES during 1981-82). Similarly, for the solar systems, a cooling load of 24.4 GJ/yr was assumed at a cooling SPF of 2.27 (same as heat pump II tested in house III). In addition, the two solar systems were assumed to deliver a 37.3 GJ/yr heating load at their demonstrated heating SPFs. Measured heating and cooling SPFs (sans water heating) and estimated system costs are also given in Table IX for comparison.

Several observations can be drawn from the comparison in Table IX and the test results presented in this paper. One is that integration of the water heating and space conditioning functions (via use of a desuperheater) yields significant performance benefits for a modest investment. Use of a desuperheater could have improved the performance of the other heat pump systems by at least as much as, if not more than, it did that of the air-source heat pump.

The GCHP systems performed only marginally better than the air-source heat pump in Knoxville. Had this comparison been done at a more northern location, however, the GCHPs would have compared more favorably due to their much better heating performance. Indeed, field tests of GCHP systems in the Syracuse, NY, area yielded heating and cooling SPFs in excess of 3.0.[27] Similarly, test results for the solar assisted systems show that they perform better during warmer months. Thus, their heating performance should be better in warmer locations and in areas that receive more sunshine. The SAHP systems tested offer no advantages over air source heat pumps for cooling.

The ACES is by far the most energy-efficient system tested at the TECH site. This may not be the case in the extreme southern and

Table IX. System performance and cost comparison for Knoxville, TN.

System	APF	Heating SPF[e]	Cooling SPF[e]	Estimated Costs (1985 $)
SSAHP	2.07[b,c,d]	3.16	-	15000+
PSAHP with no off peak electric heating	1.92[b,c,d]	2.56	-	15000+
Air Source Heat Pump[e]	1.73	1.99	2.27	4000
Air Source Heat Pump with desuperheater	1.96	1.99	2.27	4500
Ground coupled heat pumps[f]				5000-6000
Horizontal coil	1.75[d]	2.59	1.35	
Vertical coil	1.92[d]	2.85	1.75	
ACES	3.05	-	-	15000+

[a] assumed 37.3 GJ heating load delivered at demonstrated system SPFH.

[b] assumed 14.2 GJ hot water load delivered at COP of 1.0.

[c] high efficiency heat pump system with electric water heater.

[d] Water-to-water heat pump systems (energy consumption of two circulation pumps included), located inside conditioned space.

[e] Space conditioning service only (no water heating).

[f] assumed 24.4 GJ cooling load delivered at SPFC of 2.27

northern areas of the U.S. It is unlikely that either ACES or the solar assisted heat pumps will achieve wide-spread use due to their much greater complexity and cost of installation [14]. It is probable that the air source heat pump/desuperheater combination will be difficult to match economically for climates similar to and warmer than Knoxville. For colder climates, GCHP systems (with desuperheaters) may well offer the best alternative of the systems discussed in this paper.

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