17 Research and New Concepts

Noam Lior

17.1 Integral and Direct Heating Systems

17.1.1 Space Heating Systems with Integrated Collectors

Included in this category are active solar heating systems in which solar collectors are an integral part of the building structure, that is, they replace an external facade or roof material. In this way the collectors serve a dual function, possibly leading to improved economy.

An early design was the roof-integrated Solaris trickle collector (Thomason 1960; Thomason and Thomason 1975). Since water condensation on the interior of the window panes constitutes an effective heat transfer process, the associated heat losses from such collectors were found to be higher than those from other flat-plate solar collectors—by almost a factor of two (Beard et al. 1976, 1978; Beard 1978)—and the layer of condensate on the glass also tended to reduce its transmittance, thus also reducing the solar energy absorbed by the collector. To minimize evaporation and avoid freezing and corrosion, Scientific Atlanta, Inc. (Beard 1978), manufactured a collector that used a film of silicone oil instead of water in a collector of similar design. Tests with this collector indicated that the overall efficiency was still quite low.

Higher efficiency is obtained when the water passes through tubes, as practiced in detached conventional solar collectors. For example, the Colorado State University Solar House I used a roof-integrated solar collector consisting of a roll-bond absorber laid on thermal insulation, which was mounted on the roof sheathing, with two glass plates above, and showed a collector efficiency slope of about 4.5 W/m^{2}C, twice better than that of the Solaris collector (Löf and Ward 1976; Karaki, Duff, and Löf 1978). The design, construction, and testing of a solar collector combined into a structural unit so that it would double as the building roof was successfully performed by the Los Alamos Scientific Laboratory (Moore, Balcomb, and Hedstrom 1974).

A significant effort to develop building-integrated air heating collectors was undertaken by Total Environmental Action, Inc. (Kohler et al. 1978; Moore, Temple and Adams, 1980). The collector design chosen is shown in figure 17.1. It was claimed that the installed costs are about half of those

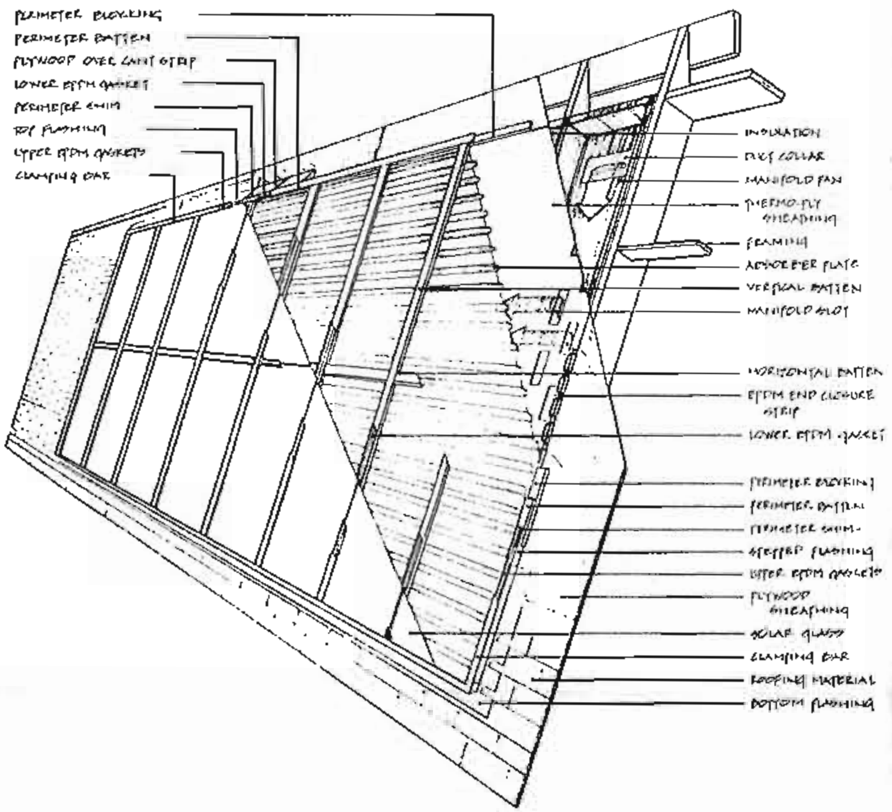


Figure 17.1
MODEL-TEA roof collector. Source: Temple and Adams (1980).

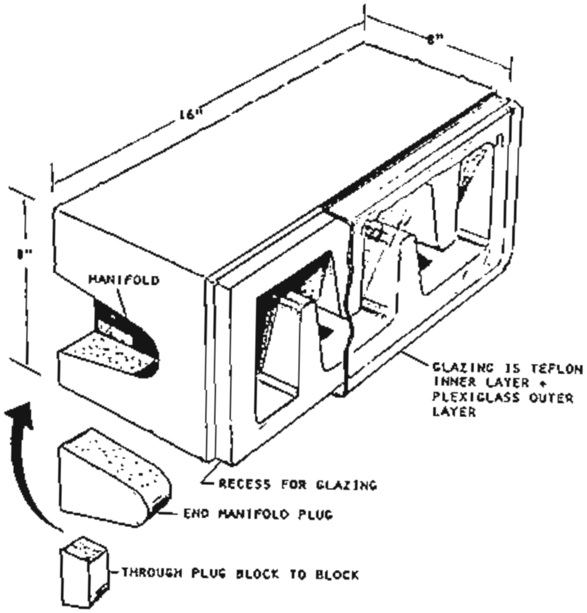
for commercially available air heating solar collectors and that the testing has shown that the performance is at least as good.

The study has confirmed the fact that eliminating leaks is perhaps the most critical aspect of site building an air collector. A smoke test was specified as an integral part of the construction process, and it has been reported that actual installations demonstrated that these collectors can be readily made air-tight. A low-cost air handling system and rock-bin thermal storage that could be easily incorporated into buildings were also developed. A comprehensive construction manual was prepared and made available (Temple and Adams 1980).

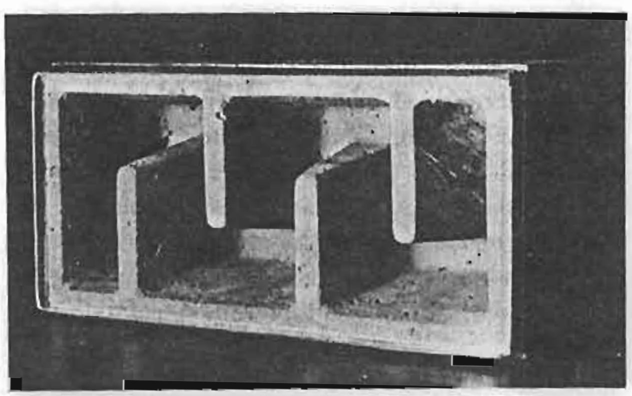
Another design of a collector integrated into the building roof was developed by Contemporary Systems, Inc. (Christopher 1979–1985). Manufactured collector modules, 2 feet (0.61 m) wide and 10 to 16 feet (3.05 to 4.88 m) long, are mounted between the rafters of a sloped roof or the studs of a vertical wall. The collectors form a maintenance-free durable outer shell for the building and thus replace any external sheathing or roofing materials. Rock-bin storage may be used with these collectors in one of the offered designs as well as in a hybrid active/passive system. In a performance monitoring and evaluation project of such a building by DOE in 1981–82, it was found that the system performed very well, delivered a solar fraction of 82%, and had a 10-year payback (DOE 1982). It should be noted that although a 15 to 20-year care-free lifetime was projected by the manufacturer for the plastic collector windows, there is no experimental evidence that such plastic covers would last or perform effectively for that long. Plastic windows of this and other types have failed after a relatively shorter time in many collector applications, and a useful life of about 5 years seems to be the state of the art. The short life will have negative impact on the economic outlook of such systems.

Forbes and co-workers from Mississippi State University have developed and tested solar air heaters integrated with preengineered metal buildings, primarily for agricultural applications, such as poultry broilers, farm shops, etc. (Forbes and McLendon 1977, 1979; McLendon, Forbes, and Hanks 1979). One of the walls of the building is built as a collector where the air flow channel is formed by horizontally corrugated steel sheeting on one side and vertically corrugated steel sheeting on the side exposed to the sun, with a 3-cm air gap between the sheets. Rock-bin storage was used. Efficiencies of up to 70% were observed but did not correlate with the $\Delta T/I$ ratio. The cost for the collector (in 1979) was stated to be less then $10/ft^2 (about $100/m^2). That design was adapted for commercial manufacturing and sale by Gulf State Manufacturing, Inc. (Starkville, MS).

Payne and Doyle (1978) calculated the energy needed for the production (including processing of the raw materials), transportation, installation, and maintenance of a typical solar collector with an aluminum or copper absorber and glass window, and found it to be about 10^6 Btu/ft^2 (11.34 GJ/m^2). Assuming that the energy collected over a heating season is about 100,000 Btu/ft^2 (1.13 GJ/m^2), they indicated that it would therefore take about 10 years of solar energy collection just to recoup this



a. The standard size (8" × 8" × 16") air-heating concrete block



b. A prototype of the standard size open-face air-heating concrete block with its outer glazing panel. This is one of several glazing options.

Figure 17.2
The concrete-block solar collector (Ketron, Inc.). (a) The standard size (8" × 8" × 16") air-heating concrete block. (b) A prototype of the standard size open-face air-heating concrete block with its outer glazing panel. This is one of several glazing options. Source: Payne (1978).

energy investment. Year-round use was predicted by them to recoup the energy in 3 to 5 years. A more recent estimate (considering higher quality commercial systems that collect 200,000 Btu/ft^2/year, or 2.26 GJ/m^2/year) indicates values of 1.6 to 1.9 years for solar space heat with fossil fuel backup (Cleveland et al. 1984). Payne and Doyle have consequently proposed that the use of collector materials which require substantially less energy should be considered, and have designed various collectors made of concrete and fired clay. The energy use for such collectors is almost three orders of magnitude lower, about 4,000 Btu/ft^2 (45.4 MJ/m^2). They have also built and tested a few units including concrete blocks that were designed to serve as solar collectors and, at the same time, replace structural blocks in south-facing walls. One of the designs considered is shown in figure 17.2. With a block-to-ambient temperature difference of about 5°C, efficiencies of about 70% were obtained. The support for this promising work stopped before the R&D was completed.

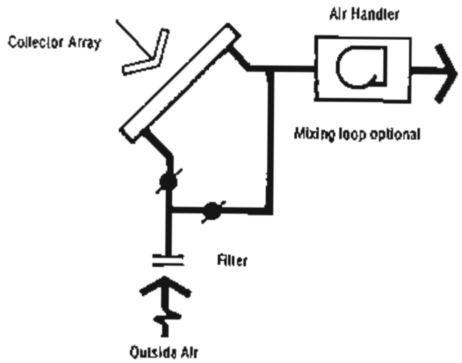
17.1.2 Direct Heating of Makeup Air, and Space Heating Assistance from Large Solar Water Heating Systems

Often the simplest and most cost-effective way to use solar energy for heating is by ducting the makeup air used in a heating system through air-heating solar collectors. In the simplest scheme, no storage or controls are needed, and the existing air handler is used to draw the air through the collectors, as shown in figure 17.3. Shown also is a collector bypass loop that can be controlled automatically to allow better temperature control of the makeup air (Solaron 1978).

When a large water heating load coexists with a space heating load, a solar heating system that supplies both loads in an effective combination would increase the annual solar contribution and most likely reduce costs. A combined system like this is shown in figure 17.4 (Solaron 1979), where the heat exchanger coil is used for water heating.

17.1.3 Direct Room Heating by Wall-Mounted Collectors

An integrated wall-mounted collector, such as those described in 17.1.1 above, or a factory-built collector mounted on the roof or south wall can be used in the simplest way by directing its heat to the space that it borders, without thermal storage. In that case, heat is supplied from the collector whenever needed and available, and an auxiliary heater can be



Heating outside air is often the simplest, most cost-effective use of the Solaron system. Outside air is drawn or blown through the collector array where the air is heated. BTU delivery is maximized in this system since the inlet air is at ambient temperature, thereby reducing losses from the collectors.

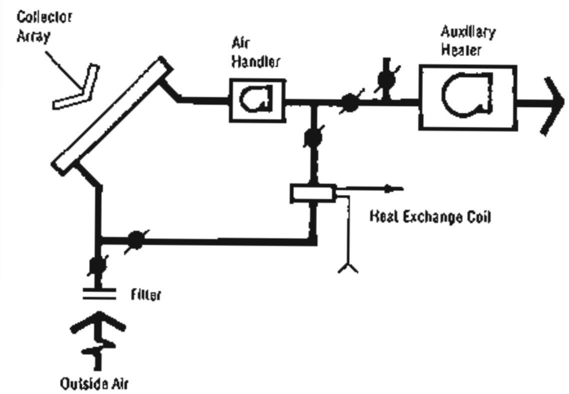
APPLICATIONS
- Make-up Air
- Process Hot Air for Industrial Drying
- Agricultural Air Drying

Figure 17.3
Makeup air and process hot air heating. Source: Solaron (1978).

used for periods when heat is needed but not available from the collector. A simple automatic controller can regulate this entire operation.

Reif (1981) describes the construction details of an air collector that is mounted on the exterior sheathing of a building wall. Air is driven in a horizontal direction by a thermostatically controlled fan through a channel formed between a corrugated sheet aluminum absorber (painted black) in front and an aluminum-foil-faced cardboard sheet in the back. The collector is insulated by a double-glazed window in front and by the wall insulation in the back. It was suggested that storage is typically not necessary in a regular-size house if the collector area does not exceed 200 ft^2 (18.6 m^2). The cost estimate (in 1981) for a 110 ft^2 (10.2 m^2) collector was about \$9/ft^2 in materials and about six days of work for two people.

Extensive residential use of factory-built daytime solar air heaters commenced in the mid-1980s, primarily in the western and south-central United States.



This system combines outside air and water heating. The combined system is often preferable in order to utilize the solar system throughout the year. In cases where hot air demands are intermittent, the system stores energy for hot water requirements.

APPLICATIONS:
- This system combines the applications for make-up air/process hot air heating and process hot water heating listed above.

Figure 17.4
Combined makeup air/process hot air heating and process water heating. Source: Solaron (1979).

17.1.4 Space Heating by Window Concentrators

One way to increase the solar energy collection area without increasing the size of the absorber is to add flat mirrors that reflect solar energy to the absorber. Mirrors are much lighter and less expensive than absorbers, and thus cost reduction of the solar heat may be feasible. At the same time, their angle must be changed with time to obtain maximal capture of the insolation, and they must be kept relatively clean. A system like this was developed by Wormser Scientific Corporation (Pyramidal Optical Collector 1977), and several buildings were equipped with solar heating systems based on this type of concentrating collector.

Figure 17.5 shows this system installed in the attic, using a skylight as its aperture. The size of the aperture is changed with a movable reflective surface within the attic to follow the sun angle changes during the year. Stationary and movable reflective surfaces form a pyramid shape

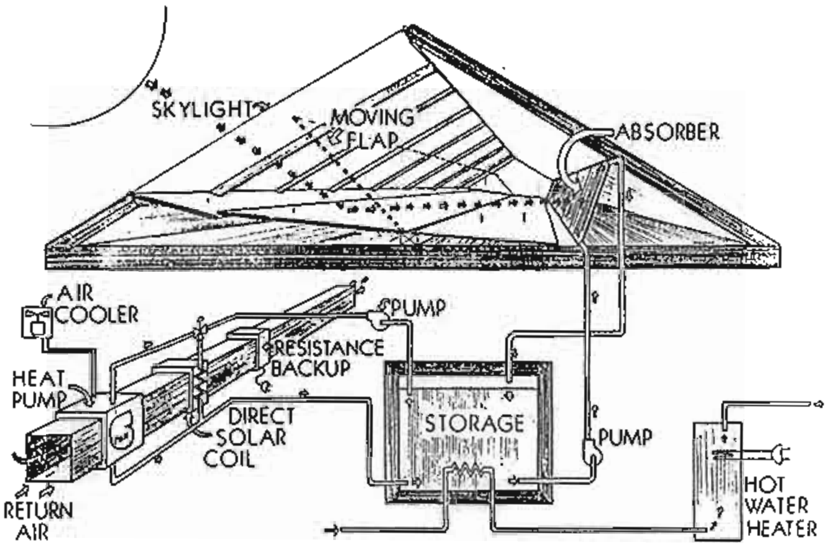


Figure 17.5
Pyramidal optics solar system. Source: Pyramidal Optical Collector (1977).

that directs the sunlight onto the absorber plate. The maximum theoretical concentration factor is 4. Although the system developers predicted a three-year payback, a number of reservations about this design, including the need to move the mirrors periodically, the occupation of potentially valuable attic space, and the actual operating experience, have stopped its further development.

17.2 Solar Space Heating Retrofit: Research and General Methodology

17.2.1 Introduction

Approximately 26% of the energy consumed in the United States is used for comfort heating and cooling, and service hot water heating in buildings. A significant satisfaction of this energy demand by the sun would be feasible over a reasonable time period only if a major fraction of the existing building stock (about 100 million in 1982) is retrofitted to solar energy use. Most of the effort in both active and passive solar heating and cooling of buildings has been so far oriented to components and new construction, and primarily to retrofit water heaters.

Having to be performed on an existing building that was not originally designed to include a solar system, retrofit is usually more costly and complex than the inclusion of a solar heating and cooling system into a new building that was designed to accommodate and integrate it with optimal installation procedures, performance, economics, and esthetics. Furthermore, a major barrier to the implementation of a countrywide solar retrofit program is the nonuniformity of existing buildings and, in many cases, the lack of adequate information about the buildings' structure and present condition, facts that tend to necessitate custom design and installation for each building.

The retrofit of existing buildings to solar heating and cooling has, nevertheless, an important national potential for the conservation of depletable energy resources and for the reduction of pollution, and in the longer term also for the reduction of comfort conditioning costs for the individual citizen. The review in this section is a very brief summary of work that has originated, in part, from the deliberations and conclusions of the Solar Retrofit Review Meeting held by the USDOE Solar Heating and Cooling R&D Branch in 1978 (unpublished; see acknowledgment). The review also includes the analysis of more recent information to present some of the main technical research aspects of active solar heating and hot water retrofit.

17.2.2 Potential for Saving Depletable Energy and Reducing Pollution

Out of the 26 quads of energy used annually by the residential and commercial sector, 18.5 are used for building service hot water and comfort heating and cooling (EIA 1981). The cost of this energy in 1981 was 89.7 billion dollars. DOE estimates that about 3.9 quads can be saved by energy conservation measures (DOE 1984). If the full energy conservation measures would indeed be implemented (a rather unlikely proposition), that would leave an annual consumption of 14.6 quads. Based on current economic constraints, solar systems are typically designed to supply around 50% of the total heating, service hot water, and air-conditioning load (although the latter is not a solar market yet). Consequently, if all buildings in the United States were retrofitted to solar heating and cooling (which is impossible in practice, but provides an upper limit), the potential annual saving in depletable fuels would be 7.3 quads, which represents an approximate cost of 35.4 billion dollars or 58% of the U.S. annual oil import. Furthermore, it would eliminate the generation of about 825 million tons (based on oil as fuel, or about 980 million tons for coal as fuel) of pollutants (including CO_2) annually.

Since only a fraction of the buildings can and will be retrofitted, the actual fuel replacement and savings would be lower, but even if, say, only one tenth of the current consumption would be supplied by solar energy by means of solar retrofit, the impact would be highly significant.

17.2.3 Technical Aspects

The general design principles specific to solar retrofit are: (1) minimal changes should be made to the building and existing heating/cooling system, (2) careful matching (interfacing) of the solar system with the existing heating/cooling system is necessary, and (3) adequate provisions for maintenance and repair should be ensured.

Even more than other solar energy applications, economical retrofit is favored by using lightweight, highly efficient solar collectors. Optimization of the spacing between collectors, which may allow some mutual shading to increase the overall amount of solar energy collected, is likely to be necessary (Lior, O'Leary, and Edelman 1977). Compact lightweight thermal storage is desirable.

Despite the fact that solar air heating systems are not subject to freezing problems and not prone to corrosion as compared to water heating types, the equipment is much more bulky and somewhat less efficient.

It is noteworthy that the existing heating system has most likely been designed for operating temperatures that are typically higher than those optimal in solar systems. Consequently, the interfacing of the solar and the existing system must be done carefully and with consideration of this difference (Dubin 1975).

The economics of solar energy applications in general has been discussed in many publications (cf. Kreith and West 1980; Ruegg and Sav 1980) and are a subject of continuous update. A survey by the U.S. Energy Information Administration conducted in 1981 indicated an *average* installed cost of $\$55/ft^2$ (single-family) and $\$27/ft^2$ (multifamily) for service hot water installations, and $\$43/ft^2$ (single-family) and $\$37/ft^2$ (multifamily) for combined water and space heating installations. At these prices and at present and near-future costs of fuel, the prospect for widespread use of solar energy is dim. Cost reductions will be attained by improved system design and manufacturing, and through transition to mass production and assembly. It should be noted that many manufacturers of solar systems and components find that the ultimate cost to the customers is overwhelmingly dominated by marketing costs (Löf, pers. com. 1987).

One way to improve the economics of solar retrofit is to design it in a way that would allow more than a single use, for example, generating additional sheltered space. Solar collectors may also alleviate maintenance costs: if a building needs a new roof or refacing with a new facade, the collectors can double up as a reliable new roof or facing material, and credit may be taken for the material that it displaces (cf. Balcomb et al. 1975).

In view of the need to reduce installation costs, and of many bad experiences with installed systems (cf. Jorgensen 1984; ESG 1984) that generated poor customer confidence in performance and reliability, the development and training of a ready work force of craftsmen skilled in solar retrofit is of great importance.

17.2.4 Some Key Research and Development Needs

CREATION OF A DATA BASE TO CLASSIFY GROUPS OF TYPICAL BUILDING/LOAD COMBINATIONS AMENABLE TO RETROFIT. A primary need for the implementation of a major solar retrofit program and the better definition of R&D requirements for that program is the conduct of a nationwide survey that would classify groups of typical building/load combinations amenable to retrofit. One of the amenability criteria should be the magnitude of energy conservation impact.

SYSTEM PRIORITIES:

1. Building service hot water
2. Space heating and low-to-medium temperature process heat
3. Heat for agricultural needs
4. Cooling

SOLAR COLLECTORS. The principal needs are the development of: (1) economical and light-weight methods and equipment for collector support and interface to the building; (2) durable, reliable, efficient, easy-to-install, relatively lightweight, and attractive collectors; and (3) dual-purpose collector and support-structure applications.

THERMAL STORAGE. The principle needs are: (1) the development of modular or collapsible tanks, bins, and associate materials, for easy access into existing buildings; (2) expansion of the existing R&D effort on low volume/mass storage systems (such as those using phase-change materials); and (3) study of the implementation of the thermal mass of the building for storage.

SPACE HEATING. Since solar heat is collected more efficiently at temperatures lower than those used in conventional space heating systems, it is necessary, for buildings that presently use hot water heating, to develop lower temperature convectors heated by liquid. R&D is also needed on system integration, interfacing with auxiliary backup, whole-system design (including control), and on economical and efficient, packaged solar space heating units. A space-heating retrofit handbook needs to be developed, which would include detailed design, assembly and maintenance instructions, and detailed case studies, past experience, and costs. The format and content of *Solar* for *Your Present Home* (Barnaby et al. 1978) and SERI's *Specification and Cost Manual for Energy Retrofits* (1984) may serve as a good beginning for this effort.

SYSTEM ASSEMBLY. The main R&D needs are: (1) development of design information about the spacing and piping of collector arrays, and its provision at the contractor level, and (2) development of the solar-related assembly and work methods, and on-site construction procedures for solar retrofit systems that represent minimal cost and disruption to the building's function. Related to the above, training of installation personnel at the vocational-school level is needed.

INSTRUMENTATION. Development of: (1) lower-cost, simple, and reliable instrumentation packages; (2) a clamp-on Btu meter that does not require the disassembly of existing pipes; and (3) human comfort-level measurement instruments, especially for the experimental evaluation of passive systems, is needed (cf. Boehm 1978, Lior 1979, Ferraro, Godoy, and Turrent 1982).

17.3 Solar Space Heating in Urban Environment: Research and General Methodology

17.3.1 Urban Characteristics Relative to Solar Heating

Despite the high potential and challenge, relatively little work has been done so far on the overall question of massive use of solar energy in cities.

Apart from futuristic architectural projects for the design of radically new cities, such as that by Soleri (1973), which are not in the scope of this chapter, the only studies found are the work at the University of Pennsylvania (Lior, Lepore, and Shore 1976; Lior, O'Leary, and Edelman 1977; Lior et al. 1978; Smith et al. 1976; Shore, Lepore, and Lior 1977; Lepore, Shore, and Lior 1978; Lior 1980), the ambitious Solar Cities and Towns program supported by the DOE and the National Endowment for the Arts, a brief summary of which is contained in IUD (1982), and a fair number of publications that describe specific solar energy projects in cities, such as the technical description of solar retrofits of individual buildings, partially referenced in section 17.2 above.

The Solar Cities and Towns program was initiated by the Department of Energy to provide models of energy conservation and solar design at urban scale. The Design Arts Program of the National Endowment for the Arts awarded related grants emphasizing the architectural and urban design implications. Some of the projects in this program are described below:

- In the effort to clarify the question of solar access and the legislation needed to ensure it in cities, the "solar envelope" approach to design and zoning was developed (Knowles 1982). A solar envelope is a volumetric set of limits in which development can occur without shadowing the natural or built surround at specified times of day and season.
- To develop a neighborhood energy strategy for the low-income neighborhood of Roxbury, Massachusetts (about 63,000 people), a study was made of building typologies and their amenability to solar energy use. Twelve housing types were identified, and recommendations for the best conservation and solar options were made (Schnee 1982).
- In combining the concepts of urban agriculture and solar energy, a schematic design, performance simulation, and cost-benefit analysis of an 1120 ft^2 (104 m^2) rooftop greenhouse were developed (Weinstein and Smith 1982). The greenhouse was designed for construction by local volunteer labor on top of a renovated six-story tenement building in the Bronx. The greenhouse is based on concepts of cooperative ownership and provides an opportunity for vegetable crop production at little cost to the tenants, as well as potential heating benefits for the apartment building. The estimated annual crop yield was $4,252 (in 1980 dollars), but the amount of

solar heat available for heating was too small. At the same time, the greenhouse provided better insulation for the roof.

17.3.2 Solar Philadelphia

Since the early 1970s, Philadelphia served as the cradle of some of the major ideas, studies, and projects for the introduction of solar energy into cities. The University of Pennsylvania (cf. Lior et al. 1976, 1977, 1978, Smith et al. 1976, Shore et al. 1977, Lepore et al. 1978, Lior 1980) initiated an effort in the early 1970s that was focused on the technical, economical, and social aspects of retrofit of row homes. This type of building constitutes more than three-quarters of the residential housing in that city and is also the dominant fraction of residential housing in many other major cities in the country. These houses are inherently well insulated, with rows of them built in a single structure often a street-block long. They usually have flat roofs, and the solar collectors may thus be placed on the roof and possibly shared by the entire block. All these attributes lead to the fact that row-homes are suitable for mass production and installation techniques of solar equipment, and thus have improved solar energy economics.

To demonstrate this concept and provide information for builders and homeowners, a row home near the campus ("SolaRow") was successfully retrofitted for space heating and domestic water, using about 500 ft^2 (47 m^2) of double-glazed flat-plate collectors on the roof and about 1,000 gallons (3.7 m^3) of hot water storage in the basement (cf. Lior et al. 1978). The system, shown in figure 17.6, was well instrumented, integrated with an automatic data acquisition system, and is still in operation. At about the same time, Drexel University (also in Philadelphia) has retrofitted one of its smaller dormitories to solar heating of domestic water, emphasizing the use of double-exposure flat-plate collectors installed on the roof, with flat mirrors used to reflect solar radiation to the back of the collector (Larson, Narayanan, and Savery 1976).

The Philadelphia Solar Planning Project (cf. IUD 1982, Burnette & Assoc. 1980, Prowler and Legerton 1980a, 1980b, Coughlin and Ervolini 1981, Coughlin et al. 1980, Miller et al. 1980, Sims and Gilmore 1980, Wallenrod 1980, Flanagan et al. 1980, Burnette and Pauman 1981) represented the first comprehensive, citywide attempt to assess the potential for solar and energy conservation applications in a major Northeastern city and to implement them through policy changes within city departments. Some of the highlights are briefly described below.

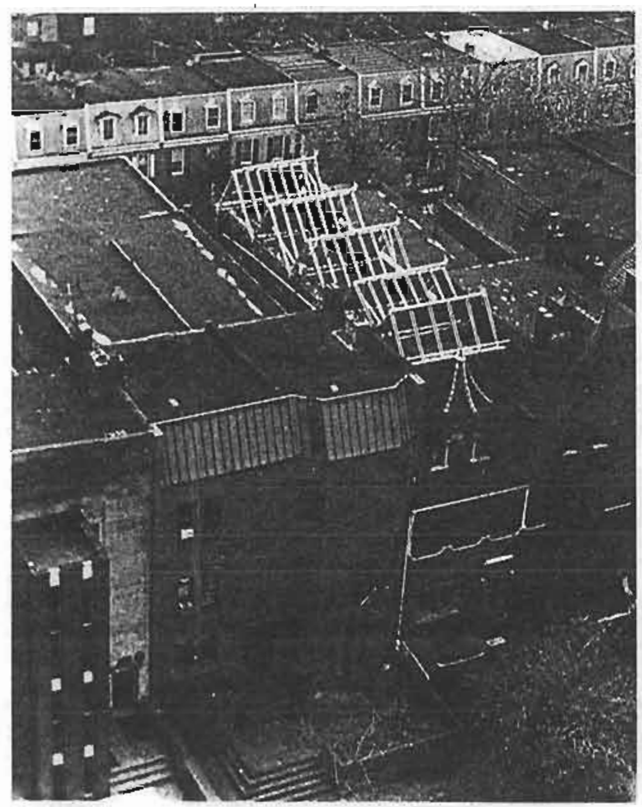


Figure 17.6
SolaRow, in a row-home neighborhood. Source: Lior (1980).

A residential housing survey indicated that four most numerous housing types accounted for 76% of the 484,000 residential buildings in the city, that over 80% are mid-row houses having two shared walls, and that over 90% have flat roofs with ample solar access (Prowler and Legerton 1980a, 1980b). An economic analysis of conservation and solar energy investments has indicated that all the conventional conservation measures considered (such as storm windows, insulation, etc.) have a payback period of less then 10 years, and all solar applications a payback of more then 10 years (at 1981 prices: \$4/1000 ft^3 gas, \$1/gallon oil, \$0.06/kWh electricity). The Trombe wall was determined to have the shortest payback among the solar installations (Coughlin and Ervolini 1981).

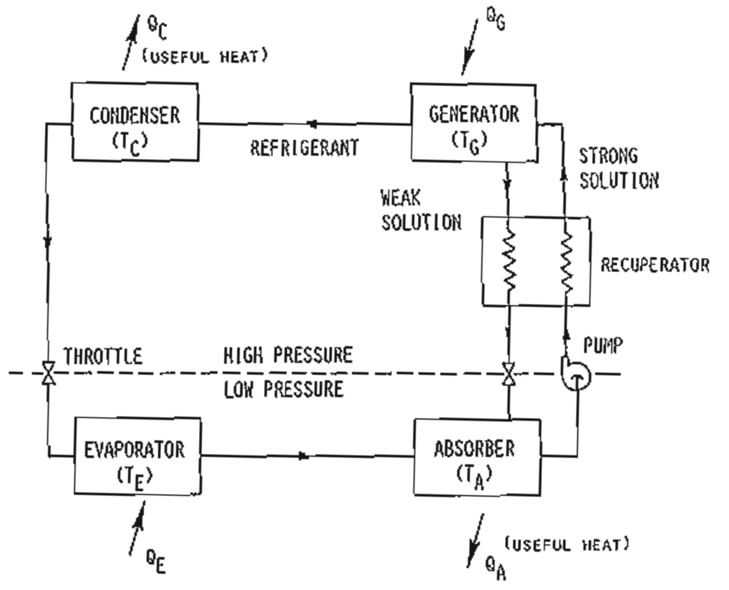


Figure 17.7
Absorption cycle heat pumps. (a) Standard absorption heat pump: $T_E < (T_A, T_C) < T_G$. Q_A and Q_C are useful heat outputs, Q_G is heat input at the highest temperature, and Q_E is "free" heat input. (b) "Reverse" absorption cycle heat pump: $T_C < (T_G, T_E) < T_A$. Q_A is the useful heat output, Q_E and Q_G are heat inputs from intermediate-temperature source, and Q_C is heat rejection to sink.

An economic input-output model for the Philadelphia Metropolitan Region was used to determine the impact of the implementation of conservation and solar measures on the economy, employment, and energy consumption in the region (Coughlin, Michel, and Cohen 1980). One of the findings was that the value of the sum of the direct and indirect benefits of these conservation and solar measures is approximately double the financial investment. Another was that the accompanying losses to the economy due to reduced demand for fossil energy are small relative to the economic gains, typically on the order of 10% or less.

17.4 Novel Solar-Driven Heat Pump Systems

17.4.1 Scope

The objective of this section is to briefly describe various novel solar-driven heat pump systems that include not only the heat pump, but also

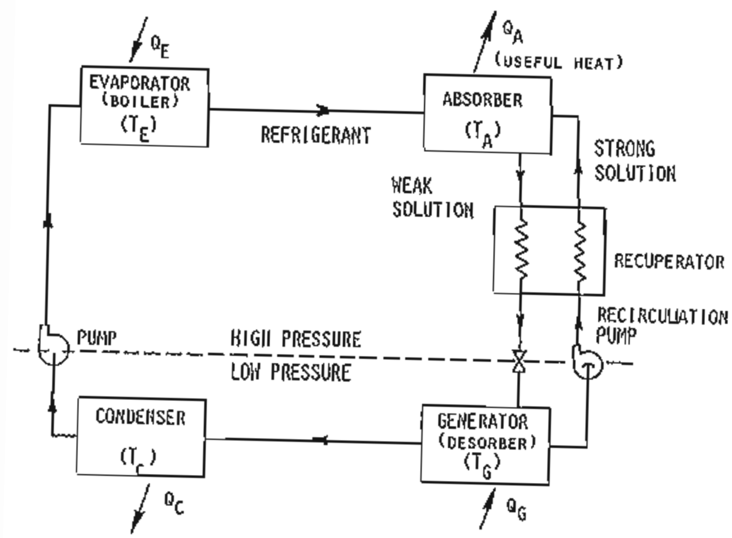


Figure 17.7 (continued)

the energy collection, storage, and demand elements. The heat pumps themselves are described in more detail in other chapters in this book. Electrically or engine-driven solar-assisted vapor-compression heat pumps are also not included in this chapter.

17.4.2 Absorption Heat Pump

Two different absorption cycles operate as heat pumps. One is the commonly used absorption cooling cycle, shown in figure 17.7a, in which there are (a) two heat inputs: one at a high temperature in the generator (Q_G at T_G) and one at a low temperature in the evaporator (Q_E at T_E), and (b) two useful heat outputs obtained from the absorber and condenser, at temperature levels (Q_A at T_A, and Q_C at T_C, respectively) between those of the two heat inputs. The temperature boosting is from the low-temperature source (applied to the evaporator) to the intermediate-temperature output (typically from the absorber and condenser), where that source could be the ambient, or low-temperature solar or waste heat. This type of system would be of benefit for heating only when the COP (based on the high-temperature heat quantity as the input) is sufficiently larger than 1, so that the trade-off of the high-temperature heat input for the lower temperature heat output is justified. From the systems standpoint, one advantage of this cycle is that it can be used year-round, for cooling in summer and for heating in winter, with minimal or no change. More detailed descriptions of this cycle are given in Niebergall (1959), Baughn and Jackman (1974), Schrenk and Lior (1975), Schwartz and Shitzer (1977), Harris and Shen (1977), Baughn and McDonald (1977), Knoche and Stehmeier (1979), and Lazzarin (1981).

The other absorption heat pump cycle, shown in figure 7b, has a heat input at an intermediate temperature, into the evaporator (which is sometimes called the "boiler" in this cycle; Q_E at T_E) and into the generator (here called the "desorber"; Q_G at T_G), it has a useful heat output from the absorber at a higher temperature (Q_A at T_A), and rejects heat at the lowest temperature from the condenser (Q_C at T_C). Compared to the previously described cycle, here the lower temperature heat input is boosted to the higher temperature without need for a high-temperature heat input. The same components and flow direction exist in both cycles, with one marked difference: by appropriate valving and pumps, the evaporator and absorber are on the high-pressure side, while the condenser and generator are on the lower-pressure side, the reverse of the pressure distribution maintained in the previous cycle. This cycle, sometimes called the "reversed absorption cycle", is described in more detail in Schwartz and Shitzer (1977), Lazzarin (1981), Isshiki (1977), Cohen, Salvat, and Rojey (1979), Perez-Blanco and Grossman (1982), and Kouremenos (1985).

The first system, *the absorption cooling cycle used as a heat pump*, has been successfully built and tested. LiBr absorption cooling units of 4.5 kW and 25 kW cooling capacity, manufactured by the Yazaki Corporation, have been tested in that mode of heating operation at the University of Padova (Lazzarin 1981). For a heat output at 35°C and generator temperatures of 80°C to 95°C, the COP of the smaller unit was between about 1.3 and 1.4 for evaporator temperatures above about 15°C, and has declined rapidly as the evaporator temperature fell below that level. For the larger unit, the COP was higher, reaching values of 1.6 to 1.7 for evaporator temperatures above 22°C. An ARKLA air-source gas-fired 7 kW ammonia-water cooling unit was built and tested as a heat pump to provide 14.65 kW heating (Kuhlenschmidt and Merrick 1982). For a delivered water temperature of 51°C to 52°C, the COP for an outdoor ambient temperature of 8.3°C was 1.26, and it declined to 1.12 for an outdoor temperature of −8.3°C. The electrical power requirement was small, 0.5

kW. For both types of heat pumps, an improvement in COP was recommended through further R&D.

Several conceptual solar-assisted heating systems based on this cycle have been proposed and analyzed (cf. Lauck et al. 1965, Schrenk and Lior 1975, Schwartz and Shitzer 1977, Harris and Shen 1977, Baugh and McDonald 1977, Cocchi et al. 1979, Lazzarin 1981, McLinden and Klein 1982). The most recommended system configuration is the one where the solar collectors supply the heat directly to the load when the temperature is high enough, and to the heat pump evaporator when it is not. With water as refrigerant (in a LiBr-water system, for example), the temperature of the evaporator must be kept, with possible assistance from solar heat, above 0°C. The fluid used for supplying heat to the load is preferably circulated in parallel through the absorber and condenser (Cocchi, Raffellini, and Stopponi 1979), since series circulation through the condenser and absorber provided a somewhat lower COP. The generator in this configuration is heated by a fossil-fuel source, since the efficiency of collectors for such high temperatures is relatively low and their cost relatively high.

Several studies have been made to examine the effects of using tanks of refrigerant and solution, integral to the absorption loop, as thermal storage (cf. Baugh and Jackman 1974, Semmens et al. 1974, Schrenu and Lior 1975, Harris and Shen 1977, McLinden and Klein 1982). The thermal storage is charged by supplying the energy required to separate the refrigerant from the absorbent solution (in the generator) and is discharged by releasing latent heat in the condenser and by the exothermic absorption of the refrigerant in the weak absorbent solution in the absorber. The condensed refrigerant (which is on the high-pressure side of the system) can also be stored and expanded into the evaporator whenever cooling is needed. The advantages of such thermochemical storage are the ability to store more energy per unit volume or weight than that of sensible heat storage systems for the same purpose, the almost isothermal nature of the storage system, and the typically ambient temperature of the storage system that minimizes heat losses and need for thermal insulation. The studies by McLinden and Klein (1982) and Semmens, Wilbur, and Duff (1974) propose only two storage tanks: one for the condensate and one for the absorbent solution. In intermittent and transient operation, characteristic for systems with thermal storage, this system configuration would, however, lead to changes in solution concentration that may either lead to crystallization or to higher-than-optimal generator temperatures. The ad-

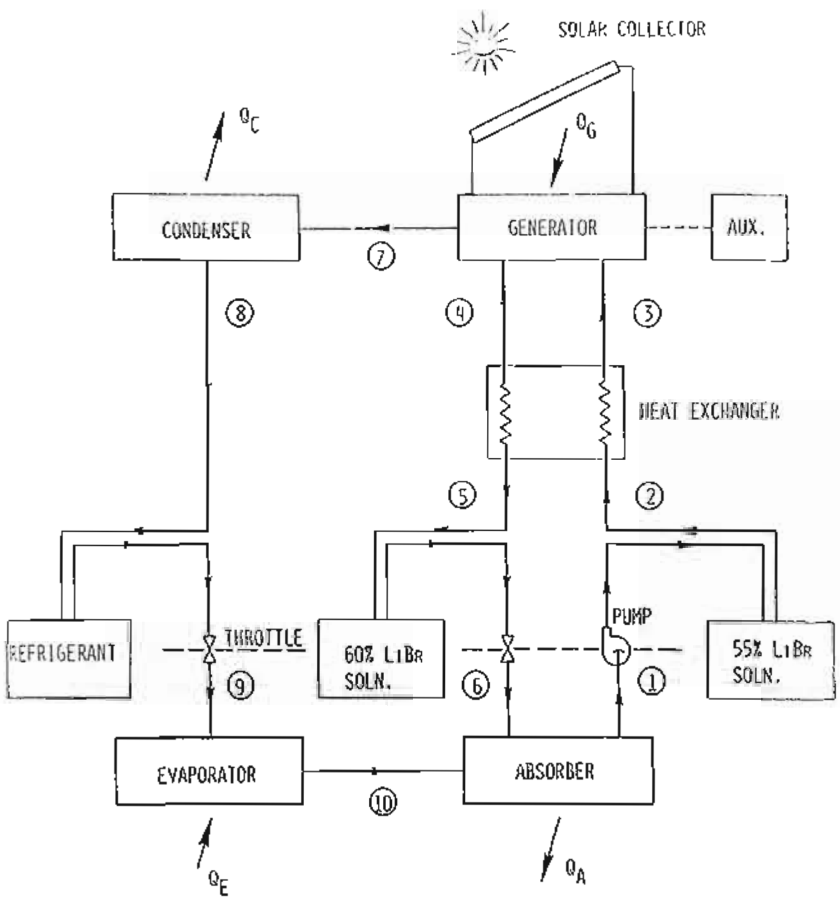


Figure 17.8
Schematic of solar-assisted absorption heat pump with integral refrigerant and solution storage. Source: Schrenk and Lior (1975).

dition of one more tank, as proposed by Baughn and Jackman (1974), Schrenk and Lior (1975), Harris and Shen (1977), and Baughn and McDonald (1977), and shown in figure 17.8, allows the maintenance of constant concentrations in the system throughout its operation, within the limits of stored mass.

The second system, *the reversed absorption cycle*, was tested at the Oak Ridge National Laboratory with the primary objective of making use of industrial waste heat in the 60°C to 90°C range (Huntley 1982). The test unit was constructed by ARKLA Industries by using several modified

components of their standard 88 kW (25 ton) LiBr-water chiller, to result in a heating capacity of 42 kW. For hot water input temperatures of 60°C to 80°C, and condenser cooling water inlet of 14.7°C to 34.6°C, the COP was 0.38 to 0.5, and the temperature boost 15.1°C to 32.1°C. The parasitic power was relatively small, with electric COPs of 50 to 70. One may conclude that the relatively small temperature boost and COP at these conditions do not make this heat pump commercially attractive. A two-stage heat pump of this type was proposed and analyzed, indicating potentially larger temperature boosts, at the expense of a more complex system using somewhat more parasitic power (Perez-Blanco and Grossman 1982).

17.4.3 Chemical Heat Pump

The absorption heat pumps discussed above are part of the broader family of chemical heat pumps. All of the chemical heat pumps described here share the same principle: heat is stored through two reversible thermochemical reactions, one that releases the more volatile component and the other that absorbs it (sometimes the second reaction is simply a phase-change process). Due to the difference in concentration in the two reactions, they occur at two different temperatures. Going in one direction, the process transfers heat from the high temperature region to the low. Reversal of the reaction allows the transfer of heat from the lower temperature to the higher, which thus produces the heat pump effect, i.e., the boosting of a low temperature energy source to a higher level. Another attractive feature is the high energy storage density involved with such reactions, because it includes here not only sensible heat but primarily heat of reaction and/or phase change. The reactants may be either a liquid and a gas (the gas being the volatile component), or a solid and a gas. More details about the fundamental processes involved and their thermodynamic characteristics can be found in Offenhartz (1976) and Raldow and Wentworth (1979). Some of the key system-related aspects are described below.

Apart from the related work on absorption cooling, very little work has been done on chemical heat pumps until the 1970s. The energy crisis has prompted the study of a number of such concepts. The DOE Office of Energy Systems Research supported five studies for systems based on calcium chloride/methanol, magnesium chloride/water, sulfuric acid/water, ammoniated salts, and paired metal hydrides. Other systems were under development privately, including a zeolite/water cooling system, and a sodium hydroxide/water chiller to operate at temperatures of about 160 F

(71°C); a Na_2S/H_2O system is under development in Sweden; and an ispopropanol/hydrogen/acetone system in France (Mezzina 1982).

The system based on the *reaction of $CaCl_2$* and *CH_3OH* vapor was investigated theoretically and experimentally by the EIC Corporation (Offenhartz 1978; Offenhartz and Brown 1979). The reactions are:

$$CaCl_2 \cdot 2CH_3OH(\text{solid}) \rightarrow CaCl_2(\text{solid}) + 2CH_3OH(\text{gas}) \quad (\text{at } T_2),$$

$$CH_3OH(\text{gas}) \rightarrow CH_3OH(\text{liquid}) \quad (\text{at } T_1).$$

If a temperature T_1 of 40°C is wanted for heating or for heat rejection, T_2 must be about 130°C for regeneration in the charging mode. These conditions allow a pressure drop from the solid $CaCl_2 \cdot CH_3OH$ to the condensing methanol, and a respective transport of the methanol vapor in that direction.

Experiments (Offenhartz and Brown 1979) have proven the concept in principle and also identified some of the problems: the difficulty in producing good rates of mass and heat transfer with this solid inorganic salt without appreciable pressure drop in the methanol vapor, the need to keep the system sealed against air in-leakage, the doubling of the volume of $CaCl_2$ during the transformation to $CaCl_2 \cdot CH_3OH$, and corrosivity. It was also found that a temperature driving force of 14°C to 20°C was needed to drive the reaction at the required rates.

The *sulfuric acid/water* system was studied and tested by Rocket Research Company (Clark and Hiller 1978; Hiller and Clark 1979a, 1979b; Clark and Carlson 1980; RRC 1982). In the charging mode, a mixture of H_2SO_4 and H_2O is heated, and the water is removed by boiling at a pressure maintained by a cooled condenser. The boiling point keeps rising with the concentration of the acid in this process, going from about 60°C at a concentration of 70%, to 170°C at 95%. Thus the first reaction is that of separation of water from the acid, and the second is condensation. When discharging, the concentrated acid solution is cooled to a point where its vapor pressure becomes lower than that of the water in the condenser, and the process reverses: heat supplied at the condenser temperature causes the water to evaporate and flow to the acid solution, with which it recombines. This reaction is exothermic and raises the temperature of the solution. Heat can thus be withdrawn from the solution for some useful purpose, at a temperature higher than that of the condenser, resulting in the heat pump or temperature boosting action. This process is

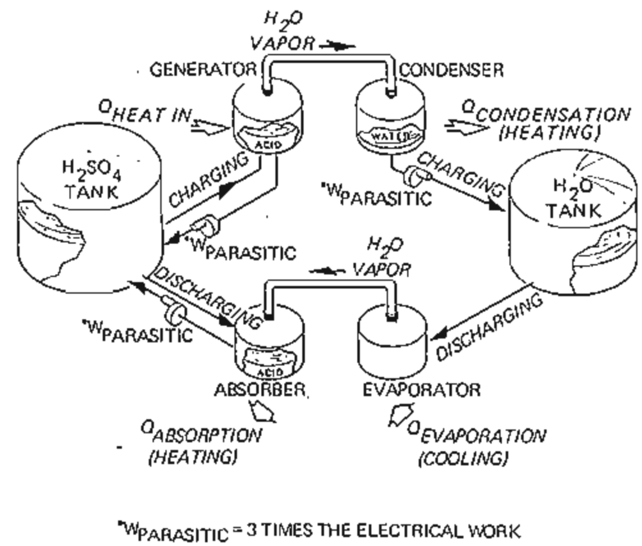


Figure 17.9
Chemical heat pump/chemical energy storage system schematic. Source: Clark and Carlson (1980).

rather similar in principle to that in a conventional absorption heat pump with internal energy storage. The heating COP was found in the laboratory and in theoretical predictions to be about 1.6 for heating and about 0.6 for cooling.

A system flow diagram for both heating and cooling is shown in figure 17.9. The test facility operated well with charging temperatures between about 100°C and 200°C. A major effort was made to develop an industrial heat pump to boost the temperature of waste heat sources at up to about 250°F (121°C) to temperatures of up to about 380°F (193.3°C). Temperature boosts of up to about 113°F (62.8°C) were obtained at a thermal COP of 0.2–0.4.

Metal hydrides are typically metal alloys of AB_5 composition, which were found to reversibly absorb and desorb large amounts of hydrogen at a relatively constant pressure with excellent kinetics. For example, stoichiometric $LaNi_5$ can absorb over six hydrogen atoms while undergoing a 25% lattice expansion, with over 95% of the hydrogen equilibrium pressure attained in a few minutes at room temperature. The absorption reaction is exothermic and the desorption endothermic. These properties, combined with the fact that different hydrides have different temperatures for the same hydrogen pressure plateau, indicate that a heat pump can be constructed by using the hydriding reaction with two metals, one at the higher temperature (T_H) and one at the lower (T_L) (cf. Gruen, Mendelsohn, and Sheft 1978; Argabright 1982a, 1982b).

The process is in principle similar to that of the previously described chemical heat pumps and is depicted in figures 17.10a (heat amplification) and 17.10b (temperature boosting). Four tanks are used: 1 and 2 for the warm-side alloy, and 3 and 4 for the cold-side alloy. One can see from figure 17.10a that approximately two units of heat are produced at T_M for each unit of heat supplied at T_H. The temperature-boosting cycle shown in figure 17.10b uses an intermediate temperature heat source (at T_M) and a low temperature (T_L) source (say, the ambient) to produce heat at the highest of the three temperatures, T_H. Here the COP is lower than 1 (about 0.5), but temperature-boosting is achieved.

As shown in figure 17.10, the cycles must operate intermittently, requiring charge cycles between cycles of useful output. Because of the high cost of the alloys (about \$20/1b for $LaNi_5$ in 1982), which constitutes the major cost item in the proposed heat pump, it is necessary to minimize the quantity of the alloy per unit energy output. This can be accomplished by reducing the cycle time in this intermittent cycle and by minimizing the thermal losses due to the intermittent heating and cooling of the heat pump structure itself. Thus it is desirable to provide high rates of heat transfer with a low thermal mass heat exchanger. The DOE-supported effort by the team of Southern California Gas Company and Solar Turbines, Inc. has indeed been focused in this direction. Using $LaNi_5$ and $MmNi_{4.15}Fe_{0.45}$ (Mm is Mischmetal) as the high and low temperature alloys, respectively, they developed heat exchangers consisting of finned copper tubes in a staggered tube bundle arrangement in which the hydride powder is stored in the annular spaces between the fins. Water is the heat transfer medium that flows through the tubes. Tests were performed (for refrigeration only) with the approximate temperatures being: $T_H = 200°F$ (93.3°C), $T_M = 85°F$ (29.4°C), and $T_L = 45°F$ (7.2°C). Due to the general problem of inadequate cycling rate and change in the federal research program directions, the funding of this project was discontinued before the necessary R&D for the solution of these problems was completed, and thus before a complete operating heat pump system could be constructed.

Possible environmental impact and hazards due to the use of the hydrides and hydrogen were not discussed or explored.

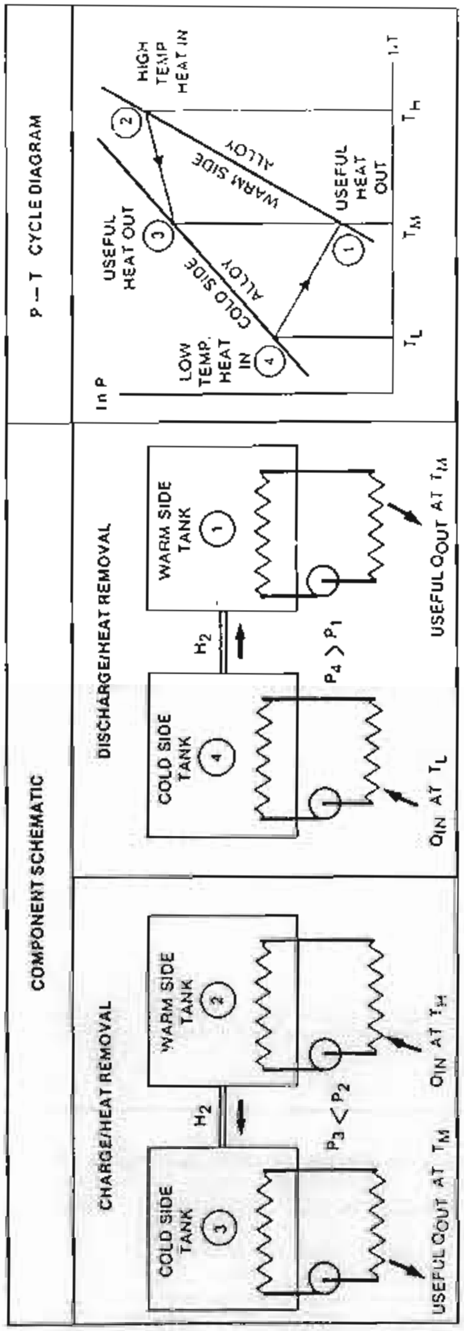
An *open cycle desiccant* heat pump was developed by Robison and co-workers (Robison 1982; Robison and Griffiths 1984), analysis on such concepts was also made by others (cf. Schlepp and Collier 1981), based on the reaction between the humidity in the air and a desiccant. The desiccants tried by Robison were triethylene glycol, calcium chloride, and mixtures of calcium chloride and lithium chloride. A heat pump based on this principle was installed in a 2800 ft^2 (260 m^2), house in South Carolina and provided all the heating and cooling for a few years. The authors estimate a cost of around \$2,000 (in 1984) for a 3-ton heat pump of this kind, but more detailed work and economic analysis need yet to be performed to draw reliable conclusions on the commercial viability of this concept.

A comparative study of the energy performance and economics of the sulfuric acid/water and methanolated salt chemical heat pumps discussed above and of conventional and emerging-technology HVAC systems was conducted by TRW, Inc. (1981). It was concluded that solar-driven chemical heat pumps consume less resource energy than either baseline or emerging systems, that chemical heat pumps can be cost-competitive in specific residential applications and locations (for example, in Albuquerque but not in Boston), and that their application in commercial buildings is not attractive either from the energy-conservation or from the cost-saving standpoints.

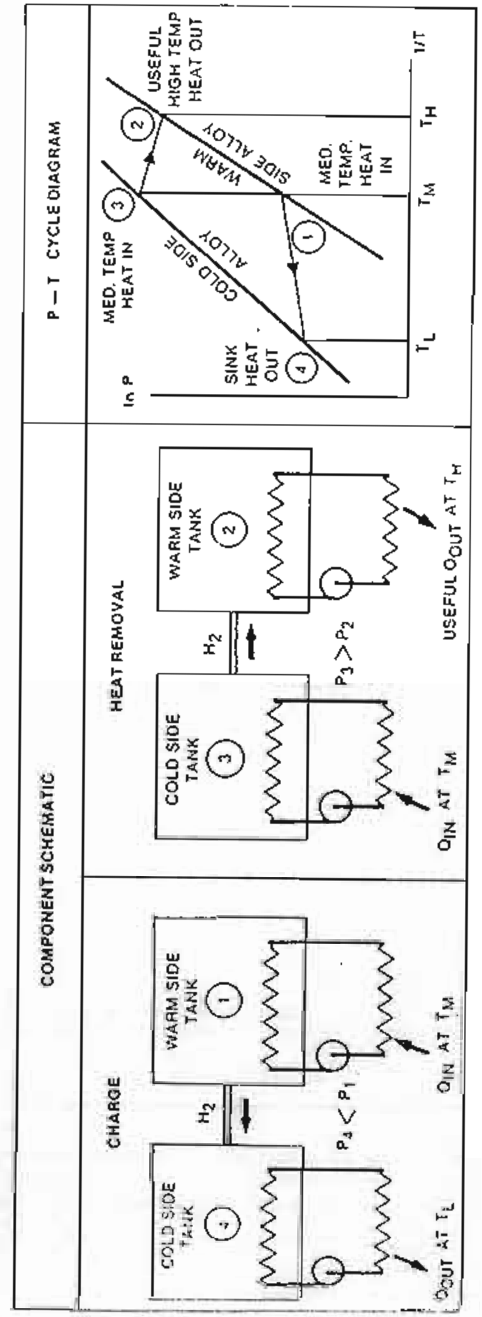
17.4.4 Rankine Cycle Heat Pump

This chapter focuses on overall *heat pump system* innovations; more information on mechanical cooling systems is given in chapter 19 of this volume. Solar-assisted heat pump systems that are not driven by solar engines are not treated here; information can be found in chapter 12 of this volume.

The major characteristics of five of the main Rankine cycle heat pumps studied are described in table 17.1 (cf. Koai, Lior, and Yeh 1984). Four of the systems use organic fluids in the power cycle, and so do most of the Rankine cooling systems. The advantages of using organic fluids are their high molecular weight (typically 100 or more), which provides improved cycle efficiency in a less costly single stage expander, and a positive slope $\partial T/\partial S$ vapor saturation limit curve, which produces a dry (superheated)



(a) Heat amplification



(b) Temperature boosting

Figure 17.10
Metal hydride heat pump cycles. Source: Argabright (1982a). (a) Heat amplification. (b) Temperature boosting.

Table 17.1
Rankine-cycle solar heat pump projects

Investigator	Power cycle fluid	Engine	RPM	Compressor	Cooling cycle fluid	Solar supply temp., °C	Cooling capacity, tons	Comments	
United Technologies Research Center	R-11	Turbine	45,000	Centrifugal	R-11	143	18	same-shaft turbo compressor	a
Garrett-Air Research	R-11	Turbine	24,800–82,600	Centrifugal	R-11	93	3, 25, 75	same-shaft turbo compressor	b
Battelle Laboratories	R-12	Pivoting-tip rotary-vane	3,600	Pivoting-tip rotary-vane	R-12	100	3		c
General Electric Corp.	FC88 (fluorinated hydrocarbon)	Rotary vane	1,200	Reciprocating	R-22	100–140	3, 10		d
University of Pennsylvania	steam	Turbine	15,300	1750 RPM Commercial (Trane) open, reciprocating; or other	R-22	100	25	Hybrid cycle: 20% of the energy supplied by fuel to super-heat steam to 600°C	e

References: a. Biancardi, Sitler, and Melikian (1982), Melikian et al. (1982).
b. Rousseau and Noe (1976), McDonald (1978).
c. Fischer (1978).
d. Eckard (1976), Graf (1980).
e. Lior (1980), Lior, Yeh, and Zinnes (1980), Lior and Koai (1984a, 1984b), Sherburne and Lior (1986), Koai, Lior, and Yeh (1984).

vapor upon expansion from the saturated state, thus avoiding problems associated with droplet formation in the turbine. In general, organic fluids also have disadvantages, such as possible toxicity, flammability, corrosivity, instability at high temperatures, susceptibility to oxygen and contaminants, and mutual degradation when in contact with lubricants used in the engine/compressor. Several have performed well, however, for the duration of the tests (up to several thousand hours with several fluids, and several million hours with tri-chloro-benzene at temperatures up to 200°C); for further references see chapter 19 of this volume. Thermal stability limitations also do not permit superheating by the addition of heat, an important detriment to the potential improvement of cycle efficiency.

Steam is used as the working fluid in one of the systems described below (Koai, Lior, and Yeh 1984; Lior 1977; Lior, Yeh, and Zinnes 1980; Lior and Koai 1984a, 1984b, Sherburne and Lior 1986; Curran and Miller 1975), and although it has the disadvantage of a negative $\partial T/\partial S$ curve and a molecular weight at least 6 times lower than most organic fluids used in Rankine cycles, thereby requiring a more complex and multi-stage expander, it is stable, nontoxic, nonflammable, and relatively noncorrosive. It has a much lower back-power ratio (requires less pumping power), its properties are well known, and it can be easily superheated by the addition of heat to increase cycle efficiency. Its cost is relatively negligible, a fact that also allows economical integration of water-based thermal storage with the steam and power generation components (cf. Lior and Koai 1984a).

The solar Rankine heat pump development program, which was probably the most successful in meeting design and performance goals, was conducted by the United Technologies Research Center (UTRC) with its subsidiary, Hamilton Standard Division (Biancardi, Sitler, and Melikian 1982; Melikian et al. 1982). Primarily supported by DOE, the most advanced model developed and tested was the "MOD-2", with a capacity of 18 tons (63.3 kW) cooling and 110–170 kW heating (figure 17.11). The turbocompressor used was developed specifically for this application, consisting of a single-stage fixed-geometry turbine and centrifugal compressor mounted on a common shaft and operating at a design speed of about 42,000 RPM.

Cooling tests in the laboratory, using a water-cooled condenser (but with the water discarding heat to ambient air in an outdoor heat exchanger) at the ASHRAE rated conditions of 95°F (35°C) ambient air and

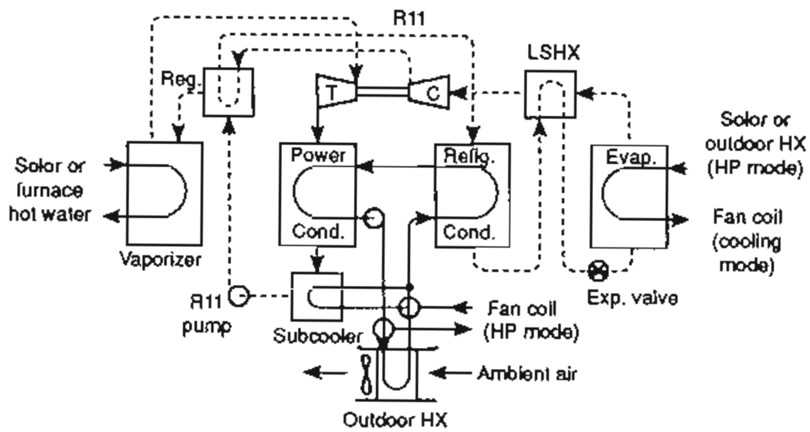


Figure 17.11
UTRC "MOD-2" heat pump module schematic. Source: Melikian et al. (1982).

45°F (7.2°C) chilled water output, with a vapor generator inlet water temperature (equivalent to about 10°C less than the solar collector outlet temperature) of 295°F (146°C), have indicated a cooling COP of about 0.64, turbine efficiency of 76.5%, and compressor efficiency of 78%. Extrapolating from data obtained through about 120 hours of testing of the MOD-2 and from the seasonal COP of 0.39 for a UTRC earlier model heat pump adapted to cooling only, which operated for two years at a site in Phoenix, Arizona, a seasonal cooling COP for the MOD-2 may conservatively be estimated to be about 0.44. At present and near-future costs of conventional energy and solar system components, neither this nor the other solar Rankine heat pumps described below were found to have significant market penetration potential.

UTRC also presented a conceptual design of a more advanced solar Rankine heat pump, the MOD-3, emphasizing improvements of the parallel solar-assisted heat pump concept as opposed to the MOD-2 series solar assisted concept. As shown in figure 17.12, vapor compression loop improvements considered include two-stage compressors with inter-stage flash economizer, condenser subcooler, liquid suction heat exchanger, and compressor exhaust recuperator. Power loop improvements include two-stage expansion with regenerative feed heating, and turbine and exhaust recuperation. Additionally, better matching of the turbine and compressor designs is proposed by use of multi-staging or intermediate gearboxes, to

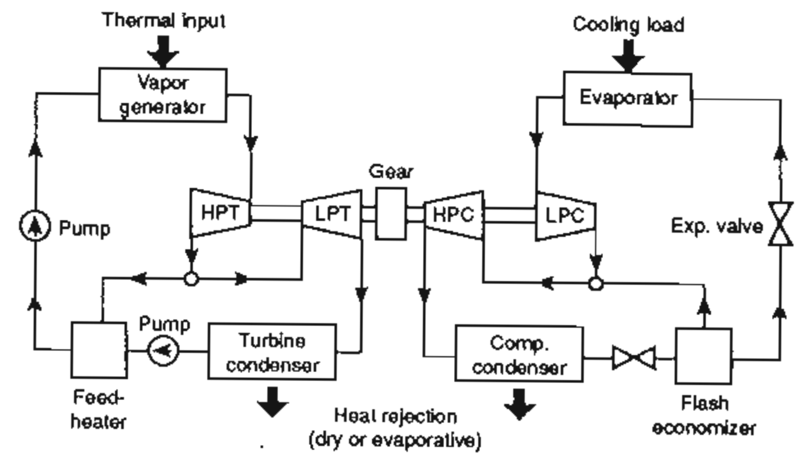


Figure 17.12
UTRC-proposed advanced-design Rankine cycle heat pump. Source: Melikian et al. (1982).

ensure that both turbine and compressor will be running near their best efficiency points over a wide range of operating conditions. It was predicted that such a unit would double the COP of the MOD-2 at a relatively-low increase in cost and may thus achieve commercial viability.

The only solar Rankine heat pump system using steam was proposed and developed by the University of Pennsylvania (Lior 1977; Lior, Yeh, and Zinnes 1980; Lior and Koai 1984a, 1984b; Sherburne and Lior 1985, 1986; Curran and Miller 1975). Depicted in figures 17.13 and 17.14, and named SSPRE ("solar steam powered Rankine engine"), this is a hybrid solar-powered/fuel-assisted power cycle that is used to drive a conventional vapor compression heat pump. The underlying principle of this cycle is the use of energy from two different temperature levels to arrive at (1) a better thermodynamic matching with the energy sinks in the power cycle, and (2) improved system economics. The design conditions for the system that was developed under DOE contracts specified the supply of solar energy at about 100°C to convert the water into steam, and the supply of heat from a fuel-fired, or from point-focusing solar concentrators—Sherburne and Lior (1985), superheater to superheat the steam to 600°C, the top temperature compatible with engineering materials used in conventional power plant technology. Exhaustive analyses both by the University of Pennsylvania researchers and others (Koai, Lior, and Yeh 1984; Lior

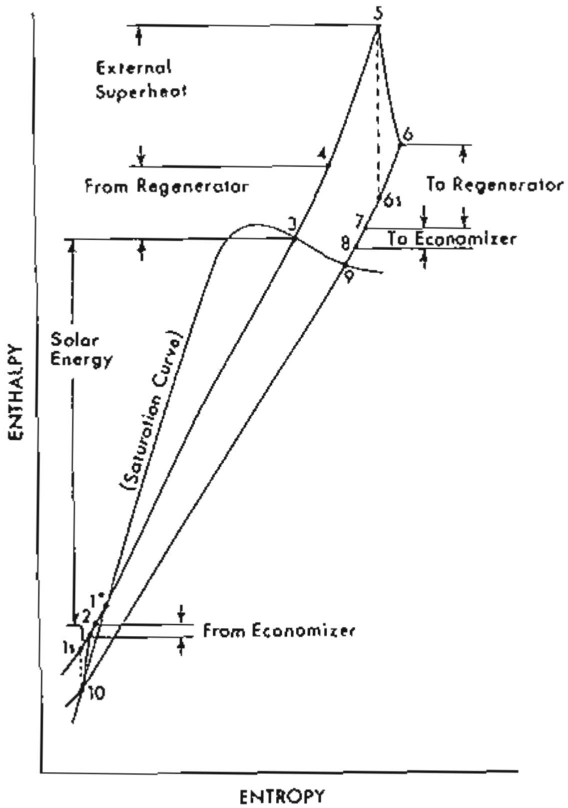


Figure 17.13
The solar-powered/fuel-assisted hybrid Rankine cycle (SSPRE) Mollier diagram. Source: Lior (1977).

1977; Lior and Koai 1984a, 1984b; Curran and Miller 1975) have shown that when about 20% to 26% of the total energy input is supplied by fuel (in the superheater), the power cycle efficiency is essentially doubled above that of organic fluid Rankine cycles, which operate at similar solar collector temperatures, to values of about 15% (seasonal) to 18% (at design, with condensation at 46°C). This doubling of the efficiency halves the required area of solar collectors, which is the primary cost-component of all solar power cycles. Furthermore, solar energy at 100°C can be obtained from flat-plate or evacuated collectors, a major advantage in cost and reliability when compared to concentrating collectors with tracking. It is

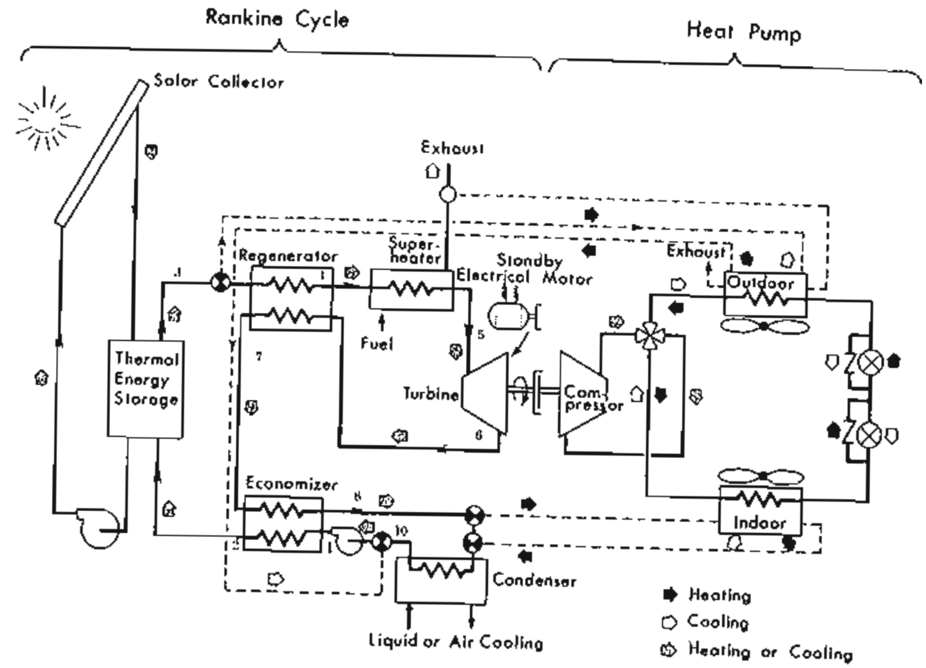


Figure 17.14
The University of Pennsylvania solar-powered/fuel-assisted hybrid Rankine cycle driven heat pump: system diagram. Source: Lior (1977).

also noteworthy that since steam generation occurs at 100°C, the highest pressure in the system is atmospheric.

Computer simulations (Koai, Lior, and Yeh 1984; Lior 1977; Lior and Koai 1984a, 1984b) have indicated that this system should have a cooling COP of about 0.62 under ASHRAE standard test conditions while operating with solar collector water output of 98°C. This is about the same COP as obtained in tests of the UTRC system described above. However, the UTRC system was operating at the much higher collector water temperature of about 156°C, which necessitates tracking solar concentrators and more expensive thermal storage and insulation. Using water-cooling, instead of air-cooling and making two other relatively simple changes, it was predicted that a cooling COP of about 1.35 could be attained by the SSPRE system.

Operating in the heating mode, the SSPRE heat pump system was compared (through computer simulation, Lior [1977]) with ten other heating

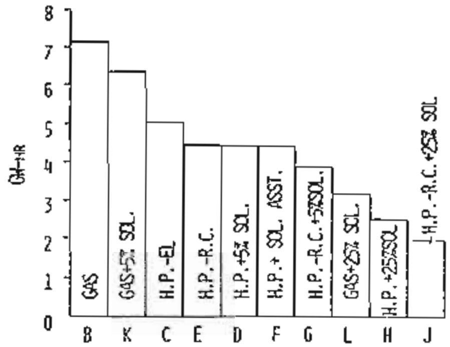


Figure 17.15
Annual resource-energy consumption for heating with use of SSPRE heat pump—30-story office building, New York. Source: Lior (1977).

systems, nine of which are shown in figure 17.15. Applied to a New York skyscraper with a design heat load of 3MW, figure 17.15 shows that the SSPRE system has the lowest resource energy consumption of all systems compared; for example, a 3.5-fold decrease in annual resource energy consumption for heating occurs when the SSPRE system with collector area of about 27,000 ft^2 (about 2,900 m^2) is used instead of a gas-fired heating system.

Since low-horsepower commercial steam turbines that operate under the conditions of the SSPRE cycle have an efficiency typically below 50%, a novel 30-HP radial-flow 10-stage turbine with 25-cm-diameter counter-rotating rotors, which uses reaction blading, was designed at the University of Pennsylvania and built for operation in the system. The design predicts an efficiency of 75% and excellent off-design performance. Preliminary tests, which had to be interrupted due to a leakage problem, have indicated efficiencies up to 79.5%. The design speed is 15,300 RPM, significantly lower than that of the organic fluid Rankine cycle turbines used, and thus with a potentially higher reliability. The speed-reduction gear box incorporates an over-running clutch system to allow the automatic engagement of a backup electric motor.

Another novel feature, made possible by the use of steam as the working fluid, is the incorporation of a combined thermal-storage/steam-generation system. The heat from the collectors is stored in a hot water tank that releases steam to the turbine by flashing when it is exposed to the lower pressures governed by the power-cycle condenser. Consequently, the same water is used both as the thermal storage medium and the power-cycle working fluid, with resulting economy in equipment.

The entire 25-ton system was constructed, and both component and system testing was started; the work was interrupted, however, by major reductions in DOE solar energy budgets.

The Rankine-driven heat pump work may be summarized by saying that the organic fluid cycles have been demonstrated, improved, and made more reliable primarily by using good engineering. The SSPRE hybrid steam cycle was the least conventional and had the highest predicted performance and economic potential but has not been advanced yet through the stages of shakedown and full testing. The discontinuation of these programs under DOE support are a consequence primarily of poor economic prospects at present fuel prices (for more details on the economic status and commercialization prospects, see also chapters 19 and 22 in this volume).

17.4.5 Note on Performance Criteria

The definition of indices of performance of energy systems that have more than one type of useful output (say, heating *and* cooling) and/or input (say, fuel *and* electricity) poses a problem, because the unit value and cost of these energy quantities may be different (cf. Bonne [1978]). Simply added up in energy units, the efficiency or COP may be misleading: for example, a higher efficiency or COP may actually result in a higher cost-per-unit desired energy output because of a shift to using a larger amount of the more expensive energy type.

The definition of energy-performance indices for solar heating and cooling systems poses even greater difficulties, primarily because a larger number of variables must be specified to describe the conditions under which the performance criterion is to be determined.

Taking these considerations into account, Lior, Yeh and Zinnes (1980) proposed a new performance criterion, the "economic COP" (with the symbol COP$):

$$\text{COP\$} = \frac{H_0}{AH_s + BH_f + CE},$$

where

H_0	is total useful energy for heating and cooling;
H_s	is solar energy input;

H_f	is fuel energy input;
E	is electric energy input;
A, B, and C	individual energy source costs of solar heat, fossil fuel energy, and electricity inputs, respectively, in \$/energy unit.

A, B, and C may be the present-value life-cycle costs, or calculated by any other method that the customer wishes to use in the evaluation of heating/cooling options. This method complicates somewhat the evaluation of the coefficient of performance (relative to a ratio based on energy), but it still is a relatively small penalty in view of this straightforward resolution of a complex problem. It is worth noting again that C is also a function of the overall system configuration.

This COP\$ must be determined for the same cooled/heated space and ambient conditions, possibly as recommended by ASHRAE and ARI standards, but it should also be determined for a "typical" cooling and heating season for several loads and geographic locations.

To those interested in the overall supply, demand, consumption, and conservation of fuel resources, a COP based on resource energy, COP_r, should be calculated as follows:

$$\mathrm{COP}_r = \frac{H_0}{H_f + E/\eta},$$

where η is the efficiency of conversion of heat to electricity, usually taken at 0.25 to 0.3, which would allow comparing the solar-powered unit to those that are entirely electric-powered:

$$\mathrm{COP}_{r,e} = \frac{H_0}{E/\eta},$$

or fossil-fuel fired:

$$\mathrm{COP}_{r,f} = \frac{H_0}{H_f}.$$

17.5 Large-Scale Concepts

17.5.1 Space Heating with Annual Energy Storage

It is an appealing idea to collect solar heat during the summer, when it is more abundant and less needed, store it, and then use it in winter (alongside with the heat collected in winter), when it is more needed and less available. Several studies were conducted to determine the effect of storage size on the solar fraction gained, on required collector area for a desired solar fraction (Speyer 1959; Hooper and Cook 1980; Drew and Selvage 1980; Braun, Klein, and Mitchell 1981; Sillman 1981), and on the heat losses incurred from the storage device (Hooper and Attwater 1977). The common conclusion was that for a desired solar fraction, the use of very large storage reduces the required solar collector area significantly. For the climatic conditions of the northern states, it was found that for buildings of more than 10,000 ft^2 (955 m^2) floor area, annual storage systems providing 100% of the space heating requirement from the solar source were more cost effective than shorter term storage systems supplying their optimum proportion of the load (usually about 60%). A consensus conclusion from these studies is that small district heating systems in which a single storage is shared by the community appear to offer the most cost-effective means of solar heating for smaller buildings in the same region. It is generally felt, however, that clearer evidence of the cost advantages of systems with long-term energy storage vs. those with short-term energy storage still needs to be generated.

Quantitative guidelines for the effect, optimization, and design of solar heating systems with seasonal storage were obtained from a computer simulation for a district of 50 family houses in Boston, Medford, Oregon, Bismarck, Nevada, and Albuquerque, New Mexico, (Sillman 1981). Several types of building loads and energy conservation measures were considered. The results are summarized in figure 17.16 and 17.17.

It is noteworthy that the solar fraction in any region could actually decrease with increased storage volume if the collector area is below a certain size, because of the accompanying drop in temperature below useful levels. At the other limit, if the solar fraction needs to be raised in systems with diurnal-type storage, the larger collector area required for that purpose produces temperatures above allowed limits for a sizable fraction of the time (especially in summer), and thus cost-optimization does not favor high solar fractions, typically showing a minimal cost at a solar fraction of about 40–70%. A number of easy-to-use design tools have been developed for the sizing of such systems (Baylin and Sillman 1980; Braun, Klein, and Mitchell 1981; Drew and Selvage 1980).

In the three northern cities, the optimal design required about 1 to 2.5 m^3 of water storage per m^2 collector. Due to the much higher insolation,

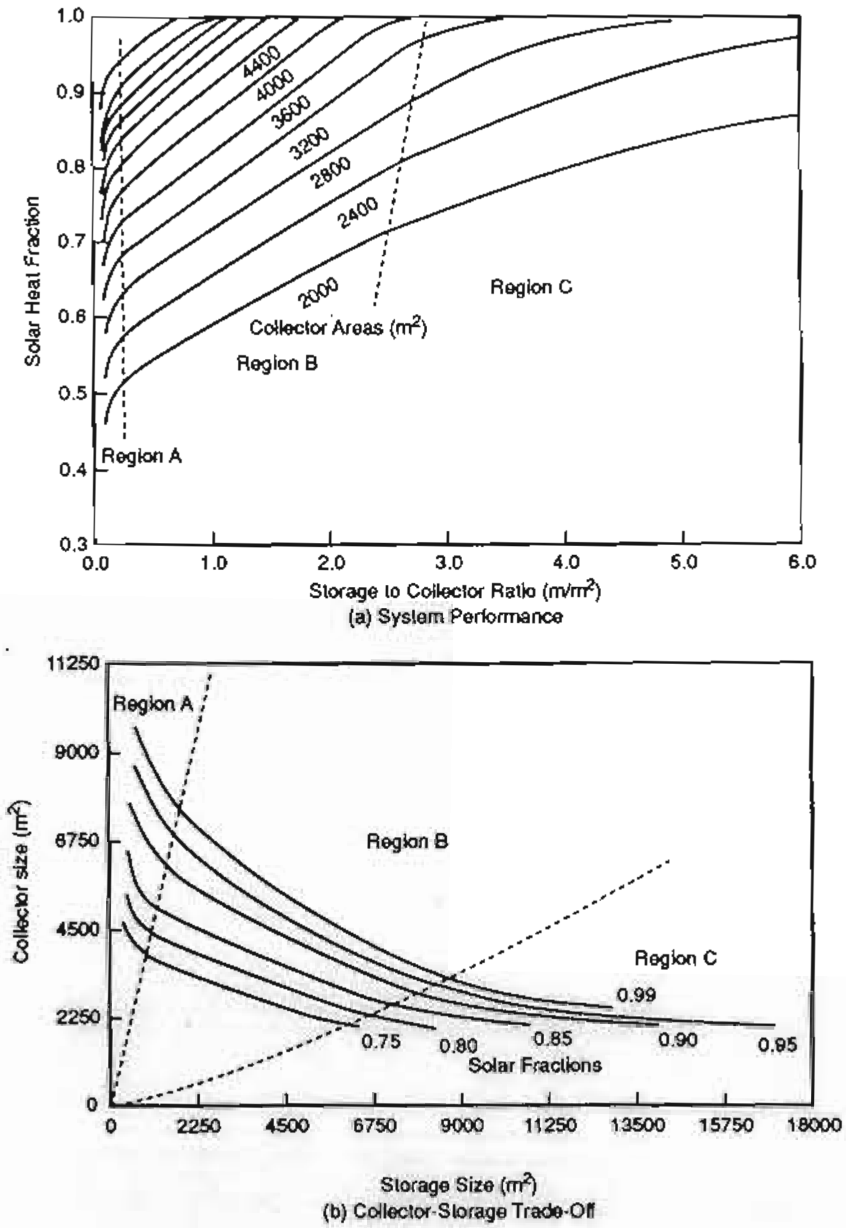


Figure 17.16
System performance and collector storage tradeoff curve for a typical annual storage system. Source: Sillman (1981).

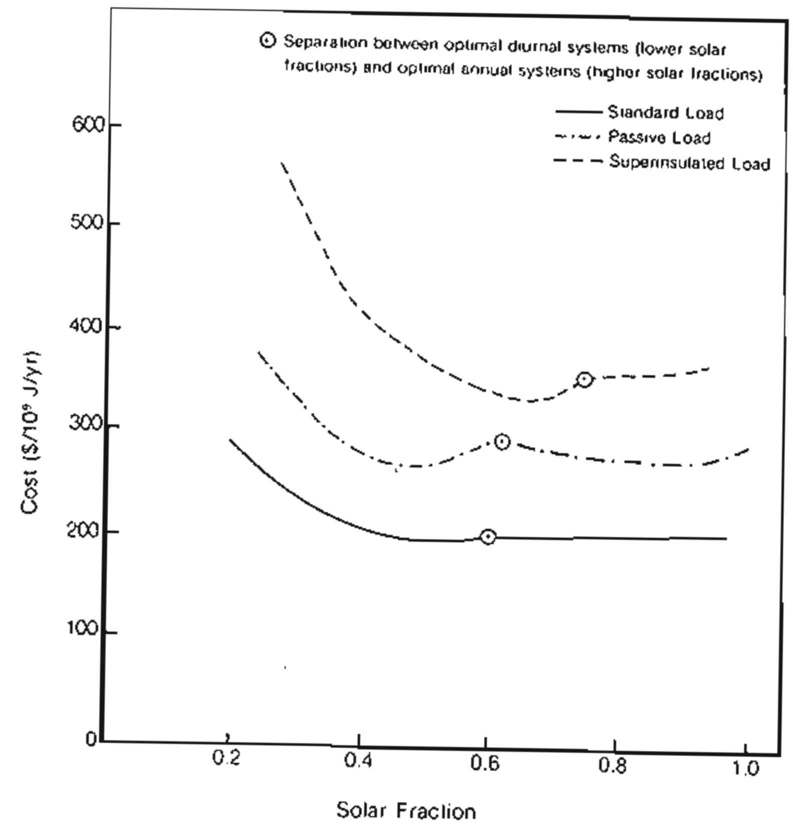


Figure 17.17
System cost per unit heat delivered vs. solar fraction for combined space heat and hot water systems. Source: Sillman (1981).

5 m^3 of storage were needed per m^2 of collector in Albuquerque, New Mexico. Depending on the particular system and location chosen, using the economic parameters listed under figure 17.17 and a capital discount rate of 10%, the seasonal storage system was reported to break even against other heat sources costing 4–9¢/kWh.

A more formal optimization conducted by Lund (1984) for a 60° latitude location (Helsinki), with a system that includes a heat pump using the lower-temperature stored heat as the source, indicated, contrary to what was predicted in the previous analyses, that the present-value life-cycle cost of a system with 50% solar fraction would be lower than half that of one with a 90% solar fraction. For a partially passive heated system, a 2% annual fuel-escalation rate and a real 4% discount rate, Lund predicted a life-cycle energy cost of 8.9¢/kWh for a typical system. It should be noted that the studies from Sweden and Finland use electric power costs that are significantly lower than those in the United States and thus their final cost conclusions are not applicable here.

In view of the traditional use of district heating in Europe, significant interest exists there to develop central solar heating plants with seasonal storage (Bankston 1982; Lund et al. 1982). Six such systems were built in Sweden and one in Finland, described briefly in table 17.2. These systems have in general operated well. It would be of value to the program to try and validate the theoretical design and prediction methods (quoted above) with the performance results from these operating plants. Much of the activity is indeed integrated through the International Energy Agency Solar Heating and Cooling Program, Task VII on Central Solar Heating Plants with Seasonal Storage, and the Energy Conservation through Energy Storage Program (a recent review is given in Bankston [1986]).

A 27,400-gallon (103.7 m^3), 12-ft (3.66 m) deep experimental system for annual collection and storage of solar energy was constructed in 1976 at the University of Virginia (Beard et al. 1979). It was operated and evaluated for one collection-heating cycle from February 1977 through January 1978, and it was found that the heat losses were excessive. At the end, the collector suffered wind and snow damage and the experiment was terminated.

An active project in Hatfield, Massachusetts, heated a 5,000-m^2 school building with a system that includes a 2,000-m^3 insulated earth-storage unit charged by 170 m^2 of solar collectors and discharged through 27 water source heat pumps to different building zones. The system has been monitored by the University of Massachusetts since 1982 (Krupezak et al. 1985).

17.5.2 Space Heating by Solar Ponds

17.5.2.1 Shallow Ponds

The shallow solar pond (SSP) is a large-area horizontal solar collector that consists of a layer of water, a few centimeters deep, resting on a thermally-insulating base material, with one or two sheets of glazing placed over the top. A transparent (typically plastic) evaporation-suppres-

Table 17.2
Solar central heating projects with seasonal storage in Sweden and Finland

	Collector			Storage			
Location	Type	Area (m^2)	Heat Pump	Type	Volume (m^3)	Planned Solar Contribution (%)	Application
Studsvik, Sweden	CPC[a] on rotating lid	120	No	Water in earth pit	640	100	Office building
Ingelstad, Sweden	Line-focus PTC[b]	1,250	No	Water in concrete tank	5,000	50	52 single-family houses
Lambohov, Sweden	Single-glazed, selective FPC[c]	2,900	Yes (storage to consumer)	Water in rock	10,000	85	55 single-family houses
Sigtuna (Sunstore project), Sweden	Black FPC with one glass; black FPC uncovered	36 126	No	Rock, 40 bore holes	10,000	80	One large single family house
Kingsbacka (Sunclay project), Sweden	Black FPC, uncovered	1,600	Yes (storage to consumer)	Clay, submerged U-tube	85,000	60	School for 800 students
Lyckebo, Sweden	Selective FPC, covered by one glass and two Teflon sheets	4,320	Not yet decided	Rock cavern	1000,000	15	500 dwellings
Kerava, Finland	Single-glazed FPC	1,100	Yes	Water in rock + rock with 54 bore holes	1,500 + 300 equivalent in rock	45	44 apartments

Source: Bannston 1982, Lund et al. 1982.
a. CPC: compound parabolic concentrator
b. PTC: parabolic-trough concentrator
c. FPC: Flat-plate collector

sion layer is in contact with the top water surface, and the water rests on a black radiation-absorbing bottom surface. Several designs have been developed and tested since the early work by Willsie and Boyle was published in 1909, and a detailed description of the state of the art, especially of the more recent work at the Lawrence Livermore Laboratory, was given by Clark and Dickinson (1980), and co-workers (Casamajor and Parsons 1979). The purpose of the ponds is to heat water to about 40–70°C (100–160°F), at which the ponds operate at a typical average daily efficiency of about 30–50%.

Intended ultimately for providing process heat for the Sohio uranium mining and milling complex (near Grants, New Mexico), an SSP prototype test facility, consisting of three 3.5m by 60m modules and hot and cold storage reservoirs, was constructed by researchers at Lawrence Livermore Laboratory. Temperatures of up to the 60°C range were obtained in summer, up to the 40°C range in fall, and up to 19.5°C in winter. The all-inclusive installed costs were \$60.20/m^2 (\$5.50/ft^2) in 1975 dollars. At the economic conditions of that time, it was estimated that the price should come down to about \$40/m^2 to be competitive with oil at \$15/barrel.

An SSP system was designed and constructed by the same group to provide heating for army barracks at Fort Benning, Georgia (LLNL 1985). The overall pond area was 26,500 m^2 (composed of 80 pond modules and covering 11 acres of land) to supply 2,000 m^3 of hot water per day for the barracks and laundry. The project was constructed by the Army Corps of Engineers for an approximate cost of \$4.5 million and planned to save more than 11,000 barrels of oil per year.

17.5.2.2 Salt-Gradient Ponds

This section will describe only the application of salt-gradient ponds to the heating of buildings; other details about solar pond principles, design, construction, and operation can be found in a number of references, such as chapter 11 of *Economic Analysis of Solar Thermal Energy Systems*, volume 3 of this series, Tabor (1981), Tabor and Weinberger (1981), Nielsen (1980, 1986), and a brief manual on this topic was prepared by Fynn and Short (1982). Table 17.3, taken from that manual, shows the estimated solar pond areas needed to meet an annual average load at a given latitude, average annual insolation, and temperature difference.

Produced typically at temperatures of 40°C to 90°C, the heat from salt-gradient solar ponds is very well suited for heating service water and

Table 17.3
Initial estimate for salt-gradient solar pond areas (in m^2) need to meet an annual average load at a given latitude, average annual insolation, and temperature difference

Tropical and subtropical climate

Latitude 0–29°N
Insolation 500 Langleys
77 Btu/hr sq ft
242 Watts/sq m

Temperature difference °C	Load (kilowatts) 30	60	120	300
33	692	1359	2683	6632
44	889	1730	3396	8347
55	1229	2364	4600	11224

Mediterranean to northern U.S. climate

Latitude 30–43°N
Insolation 400 Langleys
61 Btu/hr sq ft
193 Watts/sq m

Temperature difference °C	Load (kilowatts) 30	60	120	300
33	1056	2065	4066	10025
44	1572	3036	5922	14482
55	2952	5572	10695	25781

Intermediate climate

Latitude 44 29°N
Insolation 300 Langleys
46 Btu/hr sq ft
145 Watts/sq m

Temperature difference °C	Load (kilowatts) 30	60	120	300
33	2115	4102	8027	19682
44	5619	10549	20173	48463

Northern European climate

Latitude 50–53°N
Insolation 200 Langleys
31 Btu/hr sq ft
97 Watts/sq m

Temperature difference °C	Load (kilowatts) 30	60	120	300
33	38872	69410	128052	297918

Source: Fynn and Short (1982).

buildings. Methods of heat extraction from the ponds are discussed in Wittenberg and Etter (1982). An early study of the applicability of solar ponds to space heating (Rabl and Nielsen 1975) proposed the possibility of using a heat pump during periods in which the pond temperature falls below values useful for direct heating of space and supplying then the pond heat to the heat pump evaporator. This concept was successfully tried by Shah, Short, and Fynn (1981a). Rabl and Nielsen's calculations (for a system without a heat pump) for heating a home that has a heat load of 25,000 Btu/degree(°F)/day (47.475 MJ/degree(°C)/day) indicated that about 130–140 m^2 of pond area would be needed in Boston, Seattle, and Columbus, Ohio, 60 m^2 in Albuquerque, New Mexico, and 240 m^2 in Fairbanks, Alaska. The pond depths were 3 to 5 m, and the average temperatures about 60–70°C. The produced heat cost (including the costs of salt) was reported to be 1.1 ¢/kWh ($3.22 per million Btu; $3.05/GJ). Because of the lower relative costs of pond construction, and lower edge heat losses, this cost was reduced to 0.41 ¢/kWh ($1.20 per million Btu; $1.14/GJ) when a larger pond to supply heat to twenty houses was considered. Lebeouf (1980, 1981) studied the application of such ponds to district heating and cooling for the climates of Fort Worth, Texas, and Washington, D.C.

A number of salt-gradient solar ponds have been built in this and other countries (principally in Israel for power production). For space heating, a 155 m^2 (1668 ft^2), 3 m deep pond was built in 1975 by the Ohio Agricultural Research and Development Center at Wooster, Ohio, to supply heat to a greenhouse (Badger et al. 1977; Shah, Short, and Fynn 1981b, 1982). The maximum efficiency achieved was 12%. As indicated above, the system was converted in 1979 to a solar pond heat pump system (Leboeuf 1981). A 2,000 m^2 (about half acre) pond, about 3 m deep, was built in Miamisburg, Ohio, for heating a community recreational building and outdoor swimming pool (cf. Wittenberg and Harris [1979, 1980, 1981]). The estimated cost of the heat delivered by the pond at that time was $8.95/GJ ($9.45/million Btu), competing favorably with the cost of fuel oil.

The largest salt-gradient solar pond in the United States was constructed in Chattanooga, Tennessee, by the Tennessee Valley Authority (Chinery and Siegel 1982) and has an area of 4,000 m^2 (one acre). The largest salt-gradient solar pond in the world was constructed in Israel for producing 5 MW peak electric power; it has an area of 250,000 m^2 (68.6 acres).

A comprehensive study of the potential of salt-gradient solar ponds in the United States (Lin 1982) has concluded that conventional salt-gradient

solar ponds can provide heat at sufficiently high temperatures for space heating and service water heating in all of the regions except Alaska. The minimal economical size is a half-acre pond (about 2,000 m^2), to supply heat to a number of houses or a large commercial or apartment building. The availability of low-cost land in the proximity of the heated buildings is a limiting factor, since vacant land in most developed areas is scarce and costly. The total U.S. pond potential for heating buildings and service water was estimated to be 3.27 quads/year. The capital costs were estimated (in 1981 dollars) to range from $31/m^2 to $87/m^2 ($2.90/ft^2 to $8.10/ft^2). Assuming a discount rate variation from 11% to 20%, the cost of delivered heat ranged from $6 to $54.7 per million Btu ($5.69 to $51.84 per MJ).

In comparison with flat-plate solar collectors ($400/m^2 to $800/m^2 installed, 30–35% efficiency) and concentrating solar collectors ($750/m^2 to $1,200/m^2 installed, 50–60% efficiency), salt-gradient solar ponds ($30/m^2 to $90/m^2, 15–20% efficiency) appear to be the least expensive solar collector/thermal storage systems available. It should, however, be noted that the economic analyses of solar ponds have not taken marketing costs into account; that the mere potential for construction and its subsequent inception drive land costs up, resulting possibly in land costs that are much higher than estimated by Lin (1982) and other economic studies; and that the experience with solar ponds and with their economic analysis is not yet sufficient to decisively substantiate the present claims.

U.S. government funding for salt-gradient pond research and development ended in fiscal year 1984.

17.6 Photovoltaic/Thermal (PV/T) Hybrids

Much synergism exists between photovoltaic and thermal collectors of solar energy. They can share common components, such as the transparent cover, frame, absorber, and supports. Since the efficiency of the photovoltaic cells increases as the temperature decreases, it is desirable to cool them and use the discarded heat. The total energy conversion efficiency of a combined system increases and the total collector area needed for a given energy load decreases. Concentrating PV/T collectors in particular are inherently hybrid photovoltaic-thermal collectors because of the requirement to cool the photovoltaic cells. Since excess electric energy could conceivably be sold back to the utility, while excess heat is of no use at present, the combined unit is bound to have a better efficiency than any single unit. If the incremental cost of a hybrid system, above that of a single-purpose system, is lower than the value of the additional energy obtained, such hybrid systems could also have an economic advantage.

Early studies (Wolf 1976) showed through computer analysis that a liquid heating solar collector that contained photovoltaic (PV) cells can supply the diversified electric, domestic hot water, and space heating loads of a home in the northeastern United States. The state of the art was advanced through a number of theoretical and experimental studies (Boër, Higgins, and O'Connor 1975; Kern 1979; Russell 1979; Loferski 1982; Raghuraman 1979, 1980; Hendrie 1982). It became clear that in cases where the heat load was high and when the solar thermal collector contribution was significant, the combined photovoltaic/thermal collector system was more cost effective than systems containing PV modules only or side-by-side PV modules and solar thermal collectors.

Another conclusion derived from these studies was that the PV/T collector needs to be designed so that it can attain the maximal combined efficiency. For example, the first PV/T panels that were constructed under DOE sponsorship by ARCO-Solar and Spectrolab, and tested by the MIT Lincoln Laboratories showed disappointing results (Hendrie 1982). The designs included both air-cooled and liquid-cooled collectors, and were simply extensions of existing, commercially available solar thermal collectors, with only minor modifications for conversion to PV/T units. Neither has attained the required performance of at least 6.5% maximum-point electrical efficiency at 15V and thermal efficiency of 40% and a maximum cell-to-average fluid temperature difference of 15°C at the test conditions of 35°C difference between outlet fluid and ambient temperatures, 45°C cell temperature, and 1 kw/m^2 insolation level. The test results were also well below typical solar thermal collector efficiency values at the same conditions. Analysis of the results indicated several avenues of improvement, including the improvement of the thermal conductance between the absorber and the heat transfer fluid (by better conducting bonds and by improved heat convection through jet impingement or increased surface area), improvement of the thermal insulation of the window, improved cell absorptance by texturing on both sides (because the cells are partially transparent to solar radiation), addition of a second absorber to capture radiation that is transferred through the cell-mounting absorber, and maximal cell packing. Tests have shown that the electrical efficiency at the

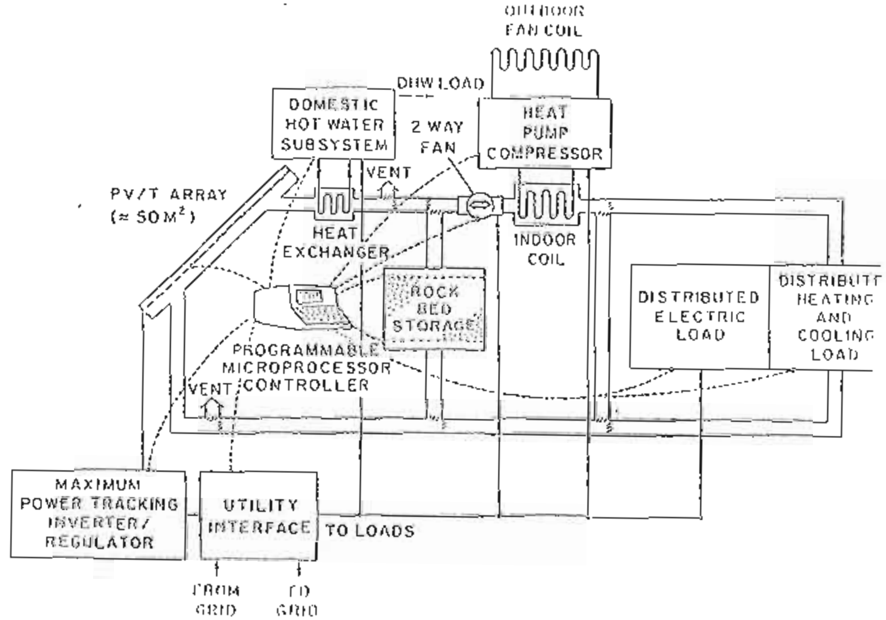
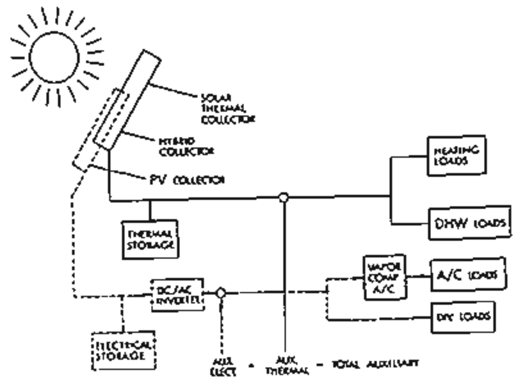


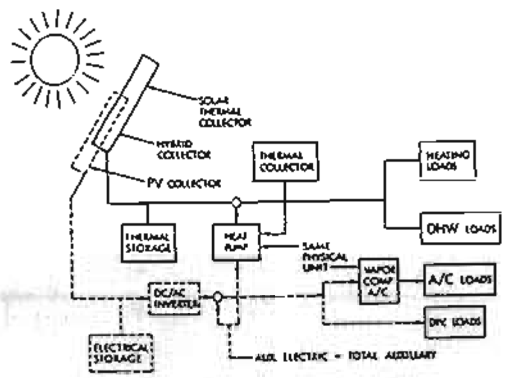
Figure 17.18
PV/T system schematic. Source: Loferski (1982).

above conditions reached 9.8%, and the thermal efficiency 42%, a marked improvement over the first-generation units. Further improvements were recommended but not carried through due to termination of funding.

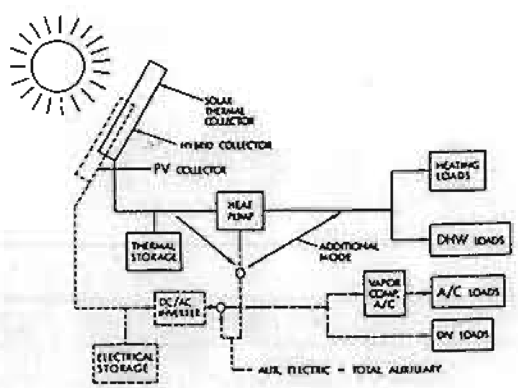
The work at Brown University (Loferski 1982) resulted in the development of an advanced PV/T air-heating collector that incorporated a number of the above-recommended improvements. A 360 ft^2 (33.5 m^2) collector system of this type was installed as a part of the roof of a prefabricated building having a 1,200 ft^2 (111.6 m^2) floor area. All electric loads in the house were connected to the AC electric service, which, in turn, was connected to a synchronous inverter. Excess electric power was sold to the utility. As shown in figure 17.18, the solar heat is first used to heat domestic water and then, according to demand, is sent either directly to heat the house or to the rock thermal storage. The parallel air-to-air heat pump is used when the heat supplied from the collectors is insufficient. The system capacity is 2.86 peak kW, and it can displace about 6000 kWh per annum from the 11,000 kWh total needed for the building. About half of the



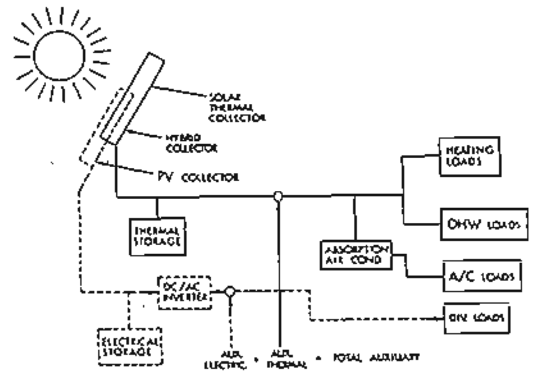
(a) Baseline Hybrid Solar
Power System



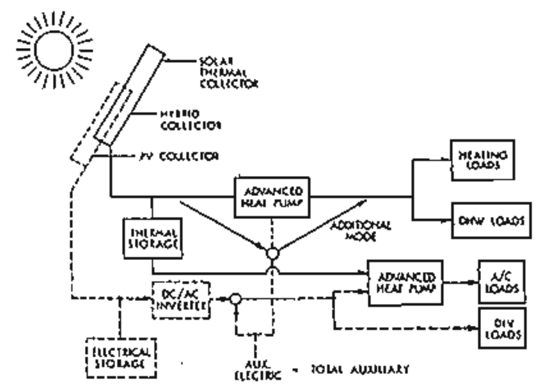
(b) Parallel Heat Pump Hybrid
Solar Power System



(c) Series Heat Pump Hybrid
Solar Power System



(d) Absorption Cooling Hybrid
Solar Power System



(e) Series Advanced Heat Pump
Hybrid Solar Power System

Figure 17.19
Five hybride PV/T systems. Source: Kern and Russell (1978).

energy supplied is thermal. The add-on cost of the PV/T was \$54,570, resulting in a payback period of 91 years (the PV part alone has a payback of 95 years). Assuming that DOE cost goals for PV cells of \$1/peak Watt and for other components were met, the PV/T add-on costs would be reduced to \$13,700, resulting in a payback of 23 years. Although the system was built, it was not run due to termination of funding.

Computer simulation studies were conducted to compare five different hybrid PV/T systems (Kern and Russell 1978), shown in figure 17.19, with a conventional system that provides heating from an oil or gas-burning furnace and cooling from a central vapor-compression electric-powered air conditioner. In addition to thermal storage, batteries were used in the simulation for electric energy storage. All of the systems were applied to a 10,000 ft^2 (930 m^2) office building and a 1200 ft^2 (111.6 m^2) single-family residence at four climatic regions: Phoenix, Arizona, Miami, Florida, Boston, and Ft. Worth, Texas.

The calculations have shown that the series advanced heat pump provides the largest energy savings at all locations. At the same time, the direct solar heat system with vapor compression air-conditioning is the most economical due to its lowest first cost. The relative need of thermal vs. electric energy depends on the nature of the load: the fraction of PV to thermal energy should increase as the fraction of cooling-load to heating-load increases.

A prototype PV/T concentrating collector, an E-Systems cylinder-shaped line focusing Fresnel lens system with a concentration ratio of 25 : 1, aperture of 0.914 × 2.44 m, and 46 silicon cells 2.3 × 2.3 cm each, mounted on the focal line, was tested by the University of Arizona (Wood et al. 1982). A cost analysis, using the actual cost of this PV/T system (\$300/m^2) indicated that it is not economically attractive and that the dominant factor is the cost of the PV cells.

In addition to the analyses and design methods developed by Kern (1979), Russell (1979), and Kern and Russell (1978) discussed above, several other contributions have been made to the design methodology of hybrid PV/T systems (Wood et al. 1982; Florschuetz 1979; Venkateswaran and Anand 1980; Herman and Schwinkendorf 1982; Schwinkendorf 1982, 1984). A weathertight and simple method for mounting solar thermal or photovoltaic collectors as an integral part of a structure is described by Rost, Amerduri, and Groves (1981).

An analysis for the optimization of a spherical-reflector/tracking-absorber (SRTA) concentrator for combined hot water and PV electricity generation resulted in recommendations for optimization (Bar-Lev, Waks, and Grossman 1982). A summary of the technical viability of flat-plate PV/T solar collectors is given by Andrews (1981) and of experimental results with such collectors by Kern and Pope (1982).

Acknowledgment

Section 17.2 (Solar Space Heating Retrofit) draws, in part, on some of the conclusions of the Solar Retrofit Review Meeting organized by the USDOE Solar Heating and Cooling R&D Branch, held in March 1978 at Columbia, Maryland. The author, who served as the chairman of the session on R&D and systems development, is grateful to the general chairman, Mr. Herbert Yim, and to the meeting participants for their contributions, particularly to Mr. J. M. Davis and Dr. F. H. Morse from the USDOE, who initiated and sponsored the meeting.

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18 Solar Cooling—Introduction and Summary

Michael Wahlig

18.1 Introduction

Three main approaches to solar cooling have been pursued actively over the past ten years: mechanical, absorption, and desiccant cooling. Each of these is the subject of a succeeding chapter in this section (schematic diagrams of the three approaches are given in figures 18.1, 18.2, and 18.3).

In both the mechanical and absorption approaches, the cooling effect is produced by the evaporation of a pure refrigerant fluid in a closed container (an "evaporator"), a cooling process identical to that occurring in a conventional electric-driven vapor compression air conditioner. In fact, a solar-driven mechanical cooling unit contains a vapor compression chiller as a subcomponent. The cooling effect in a desiccant cooling system is generated by the evaporation of water into an air stream that has been dried by the cycle, using essentially the same cooling mechanism as that produced by a conventional evaporative cooler in a dry climate.

The details of the thermodynamic cycles for the three approaches are given in their respective chapters. This introductory chapter will present general concepts applicable to all these cooling approaches, relative advantages among them, and a brief picture of how their developments have fared over the past ten years. The latter part of this chapter will summarize the status and outlook for each approach.

18.1.1 General Perspective for Solar Cooling

All of the solar cooling technologies considered here are solar thermal ones; i.e., the solar energy is converted to thermal energy (i.e., heat) via a solar collector, and then the heat is used to drive a cooling cycle.

In this respect, there is a basic difference between solar heating and solar cooling. For heating, the collected solar heat may be applied directly to a building heating load. For cooling, the solar-collected heat must be used to drive an energy conversion device, using a thermodynamic cycle to produce cooling. The output cooling effect produced (i.e., the amount of heat removed from the building) divided by the input heat used to drive the cycle is the efficiency of the cooling cycle. Traditionally, this efficiency is called the thermodynamic coefficient of performance, or COP, of the cooling cycle. Typically, the COP value and the required driving temperature vary with the load and ambient temperatures.

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