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SOLAR AIR CONDITIONING STUDY

Richard Merriam

Arthur R. Little, Incorporated
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The state-of-the-art of solar cooling is evaluated to determine the near term performance potentials and life-cycle costs of the most promising approaches. The heat actuated absorption cycle, Rankine cycle, and desiccant dehumidification cycle are examined. The principles of operation are described, performance coefficients are reviewed, operating constraints are examined, and the commercial status of each approach is evaluated. An

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analysis of the major solar cooling demonstrations (as of 1976) is carried out. Savings-to-investment ratios are calculated for solar cooling systems in buildings in seven locations within the United States.

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ABSTRACT

The state of the art of solar cooling is evaluated to determine the near term performance potentials and life-cycle costs of the most promising approaches. The heat actuated absorption cycle, Rankine cycle, and desiccant dehumidification cycle are examined. The principles of operation are described, performance coefficients are reviewed, operating constraints are examined, and the commercial status of each approach is evaluated. An analysis of the major solar cooling demonstrations (as of 1976) is carried out. Savings-to-investment ratios are calculated for solar cooling systems in buildings in seven locations within the United States.

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Major contributions were made to the study by Dr. Peter Teagan of ADL, particularly in the areas of Rankine cooling and solar collector technologies.

Richard L. Merriam
Program Manager

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SUMMARY

A. BACKGROUND

Major efforts have been carried out in recent years in the development and commercialization of solar space conditioning equipment. The early markets have been identified as being concentrated in the small to intermediate size water and space heating systems for residential applications. Development and, particularly, commercialization of solar cooling systems have lagged because of the added technical complexities, higher first costs and lower energy cost savings achieved by solar cooling. From a national benefits viewpoint, however, solar cooling has a potential for significantly reducing our usage of precious fuel resources. In many larger buildings the cooling load is a significant if not the dominant space conditioning load.

There are a number of approaches for utilizing solar heat to achieve active space cooling. Most of these make use of an existing technology base for conventional space conditioning. However, development of cost-effective solar activated cooling machines using the existing technology is non-trivial. In addition, all approaches require a higher degree of sophistication in the solar collection and storage subsystems than is necessary for solar water or space heating.

The Civil Engineering Laboratory (CEL) of the Naval Construction Battalion Center (Port Hueneme), under sponsorship of the Naval Facilities Engineering Command (NAVFAC) is currently investigating the use of air conditioning systems driven in part or wholly by low intensity energy sources such as solar and wind energy. The present report has been prepared for CEL summarizing the present state of the art of solar cooling systems. The report examines the cost effectiveness of various solar cooling approaches, and provides a methodology for fairly simple calculations for solar space conditioning life-cycle costs.

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B. PROGRAM OBJECTIVES

The major objectives of this work were to:

- Review the state of the art of solar cooling;
- Determine the near term performance potential and commercial availability of the various approaches to solar cooling; and
- Make a preliminary assessment of the life-cycle costs of the most promising approaches.

C. PROGRAM APPROACH

Five major activities were carried out in the project: (1) various cooling system approaches were identified and characterized, (2) the status of each of the major approaches was reviewed, (3) regional climatic and energy cost data were collected, (4) a methodology for calculating life-cycle costs was developed, and (5) evaluations of systems life-cycle cost were made.

Analyses of cooling system performance were made for seven locations in the U.S.* The range of heating and cooling loads expected in these regions was bracketed by calculations carried out for typical construction single-family residence and office buildings. Parametric analyses were performed to determine the sensitivities of life-cycle costs to economic assumptions and technical performance characteristics of the various solar cooling approaches.

D. CONCLUSIONS

The study revealed the following findings:

1) The three most promising approaches to solar cooling are the heat actuated absorption machine, the Rankine Cycle heat engine and the desiccant dehumidification

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*Bremerton (Wa), Chicago (Ill), New London (Cn), Norfolk (Va), San Diego (Ca), Corpus Christi (Tx), Key West (Fl).

system. The absorption device uses the affinity of various liquids to achieve the equivalent of refrigerant compression. The Rankine Cycle is a specific form of a low temperature heat engine to create mechanical power to drive a vapor compression machine. Desiccant systems remove moisture from the air and, in some instances, can be used to provide cooling. A solar cell-heat pump system may also have promise in the longer term, but very little work has been done to date on this type of system.

2) The absorption approach has the broadest technical and commercial base. Only moderate alterations are required to existing equipment to achieve solar cooling. Heat activated absorption machines are presently produced in moderate volume and capital costs are close to the minimum achievable for this type of approach. The reliability of this type of system has already been established. Rankine Cooling systems are still in the development stage, but are similar to systems now being developed for waste heat bottoming cycle applications. Because of the relatively small commercial base for these systems, capital costs and reliability have not yet been established. However, the Rankine cycle approach offers significant performance potential in flexibility in operation over a wide range of input temperatures, and the option for a parallel electric drive on the vapor compression machine. Solar desiccant systems are also in the development stage, with an existing technology base in commercial and industrial drying applications. Large air handling equipment is normally required and lower COP's are generally achievable with desiccant systems (relative to the other approaches). Of the three major approaches to solar cooling, the desiccant approach has received the least amount of development effort, either under government or private sponsorship.

3) The major emphasis in solar driven absorption systems has been on the continuous, single-effect, lithium bromide unit. Presently, units are manufactured in a capacity range from 2 to 1700 tons of cooling. There is little industry effort in improving COP's or in the use of new fluids. COP's of commercial equipment are near the practical maximum achievable. However, some industry activity is presently underway to improve the design of smaller capacity equipment (25 tons and under) for solar applications. Other approaches, such as the use of water-ammonia, intermittant cycles, or refrigerant storage are likely to remain in the experimental stage in the near term.
4) The near term Rankine cooling systems are likely to operate in the low to moderate temperature regime (200-400°F) because of the technology requirements for solar collection at higher temperatures. The major development areas are: efficient expanders, efficient feed pumps, and controls. Few Rankine power systems specifically designed for solar application are currently available on a prototype basis, but more will be available when a significant solar cooling market is perceived by the industry.
5) Desiccant systems can operate at the same or slightly lower temperatures as absorption machines. The major development efforts for solar activated desiccant systems are in the solid desiccants--rotary wheel and moveable bed. Although the rotary wheel approach is applicable to all size ranges most of the development work has been for residential application. The wheel approach is the closest to commercialization with some prototype testing currently underway. However, there generally has been little activity on the part of desiccant system manufacturers towards developing systems for solar application.
6) Nearly all demonstrations of solar cooling equipment have employed single-effect lithium bromide absorption machines.

The demonstrations have shown that the present technology is adequate to achieve solar cooling. However, they have also shown the importance of proper attention to overall system design--specifically, the requirement for an efficient high temperature collector, a well designed control system, matching of component sizes and proper insulation of storage units.

7) Costs of solar equipment will continue to decline (in real terms) over the next 10-15 years as the solar industry develops. The major emphasis during this time period will likely be on heating equipment. High performance collectors will be introduced to the market for heating (as well as cooling) applications, but the collector costs will probably remain significantly higher than the costs of low temperature collectors. The costs of solar cooling machines will depend, to a large extent, on the market established for conventional applications of the technology since it is unlikely that the solar cooling market alone will be large enough to bring about the high volume productions required to achieve major cost reductions.

8) Solar cooling life-cycle costs are minimized by: low capital costs, steady use of the equipment throughout the year, displacement of high cost conventional energy, low maintenance costs, long lifetimes (greater than 10 years) and high insolation. Savings-to-investment ratios are highest for large systems where a heat activated cooling unit is conventionally used. Using capital cost data equivalent to a well developed industry and 1976 energy cost data, the SIR's are in the range of 0.1-0.15 for single family residences, and in the range 0.2-0.9 for office buildings. (For the same set of assumptions, the SIR values of a solar water heating system are 0.3 to 1.5). Using 1985 energy cost data, the SIR values for office buildings are in the range 0.4 to 2.0. A ten

year life and a 10%/year real discount rate represent conservative economic assumptions. A twenty year life nearly doubles the savings-to-investment ratio.

I. INTRODUCTION

A. BACKGROUND FOR SOLAR COOLING

The dramatic increases in energy prices in the past few years have acted as a major stimulus to a resurgence of commercial interest in solar energy. Solar can provide a means of conserving precious fuel resources and as a potential means of lowering the energy costs to the consumer. In addition, from a national benefits viewpoint, solar can lessen our dependence on the imports of oil or liquified natural gas. The potential contributions of solar space conditioning towards energy independence is a major motivating force behind the various governmental programs in support of development of a solar space conditioning industry.

The major efforts to date have been directed towards solar water and space heating systems because of their relative simplicity and cost effectiveness in the near term. Market studies have shown that the near term opportunity for solar space conditioning will be principally the residential market, where water and space heating are the dominant loads. Indeed, in a number of countries (notably Japan and Israel) use of solar energy for domestic water heating is an established practice. In the U.S., a number of manufacturing concerns are now marketing solar water heating packages or solar system components which can be used for space heating.

Solar cooling systems are inherently more complex than heating systems because the collected heat cannot be used directly to perform the desired function. The solar heat driven devices that are required to provide cooling to the conditioned space are additional pieces of equipment which increase the system capital cost and may reduce the overall system reliability. In addition, the required heat inputs to the cooling machines are generally higher than the cooling achieved (COP < 1) and the heat input stream temperatures necessary for operation are generally significantly higher than for solar water or space heating, thereby leading to lower efficiency collection or requiring the use of higher performance solar collectors.*

For many of the larger buildings (office building, apartments, government buildings and other commercial establishments) where occupancy levels are high, the cooling load is a major, or often times the dominant, load. Hence, energy conservation by solar space conditioning in these types of buildings requires consideration of solar cooling. As a rule, owners of these larger buildings tend to be less first cost sensitive, and are more likely to select between alternative HVAC** systems on the basis of life-cycle costing. Other things being equal, economics of scale tend to reduce the unit solar system costs (expressed in dollars per square foot of collector) of the systems required for the larger buildings. In addition, in some of the larger buildings, heat actuated cooling equipment (normally absorption type) may be used conventionally, thereby reducing the expenditures required for the solar cooling system.

B. CHARACTERISTICS OF SOLAR HEATING AND COOLING SYSTEMS

Figures I-1 and I-2 illustrate generalized residential and commercial sized HVAC systems with solar heating and cooling. The primary elements of the solar systems are: solar collector, thermal storage subsystem, heat actuated cooling machines, transport equipment and controls. All solar space conditioning functions are achieved by the use of heat energy, with a heat activated device required for the cooling functions. A backup heating system (called a "boiler") and cooling system are also shown in the figures. These auxiliary units are required to supplement the solar system when it is not capable of meeting the load by itself.

Generally, the backup cooling system in a residential sized system is a small (about 3 ton) air cooled vapor compression machine. In

*A brief review of the state of the art of solar collectors is given in Appendix A.

**Heating ventilating and air conditioning.

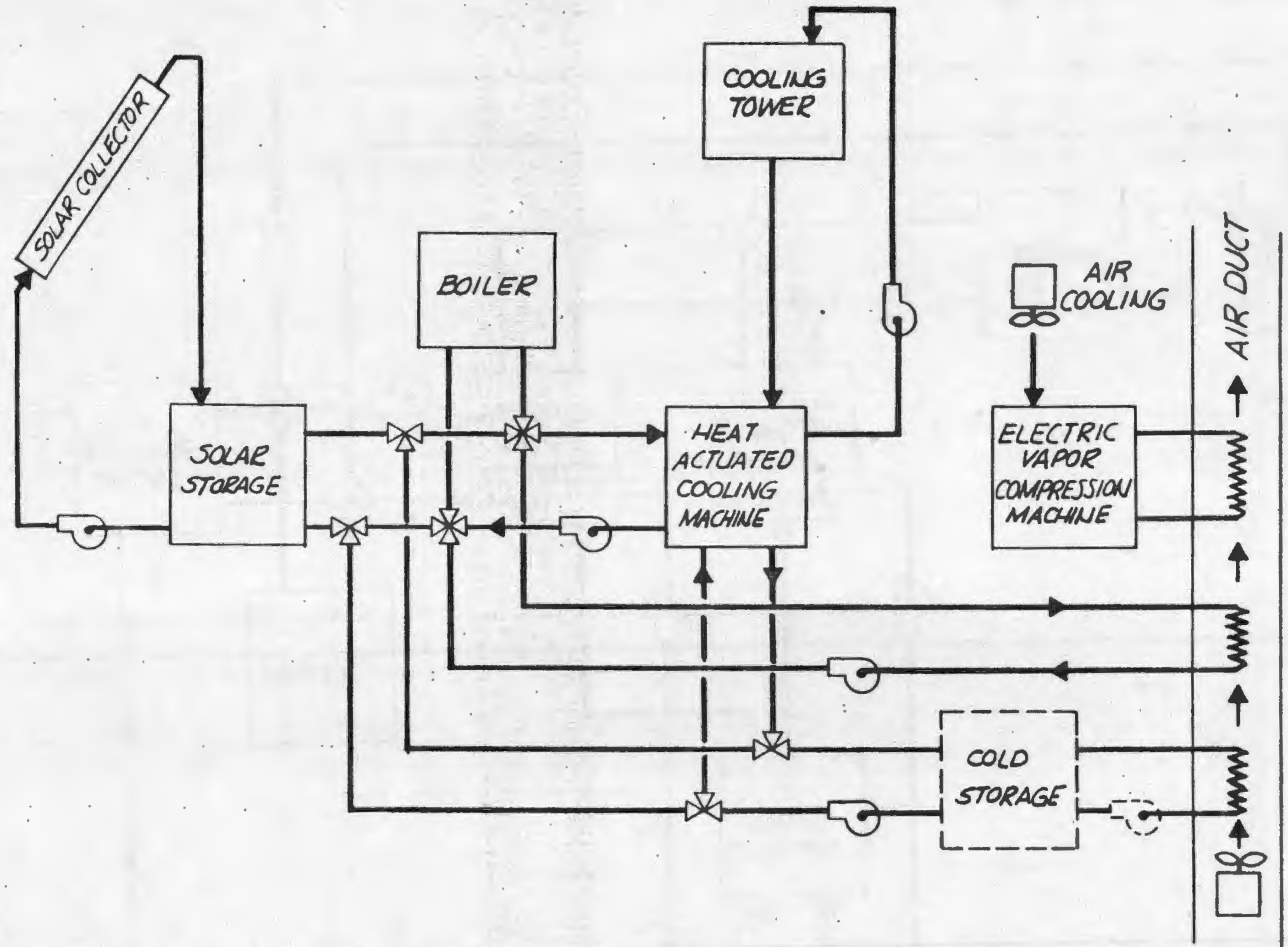

FIGURE I-1 GENERALIZED RESIDENTIAL SIZED HVAC SYSTEM WITH SOLAR HEATING AND COOLING

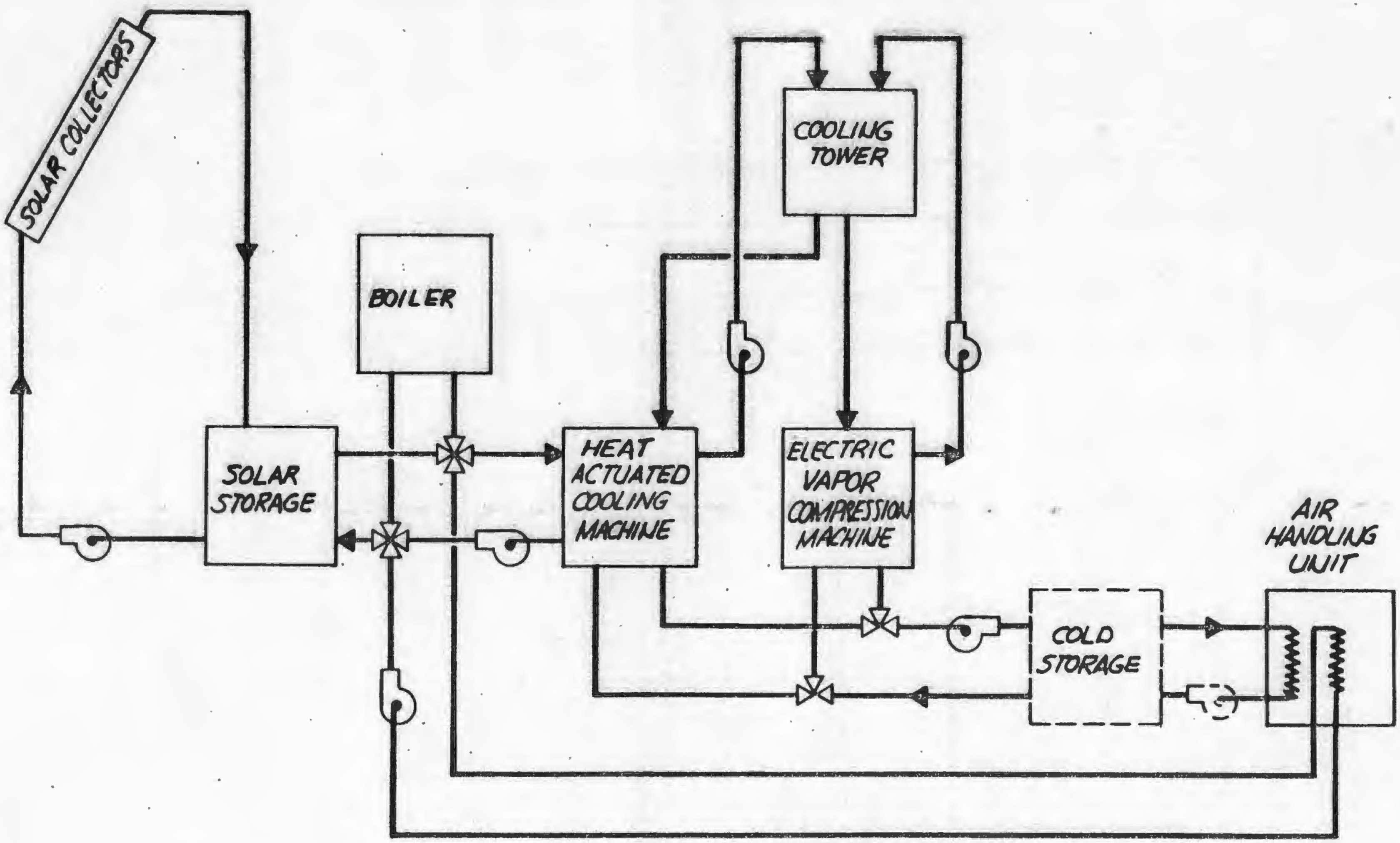

FIGURE I-2 GENERALIZED COMMERCIAL SIZED HVAC SYSTEM WITH SOLAR HEATING AND COOLING commercial sized systems, the backup cooling system is generally a water cooled electrically driven chiller or a lithium bromide absorption machine. Economizer cycles (circulation of outside air throughout the building when outside air temperatures are about 20°F lower than inside) are also often used in commercial sized HVAC systems.

Normally, the HVAC system delivers heat or cooling to the conditioned space upon demand. The solar thermal storage unit acts as a buffer between the space being conditioned and the solar collector subsystem. Hence, the storage unit provides an interface between time varying demands for heat or cool and time varying insolation. A cold storage unit can be introduced between the chiller and air handling unit or air duct. As will be seen later, incorporation of a cold storage unit can lead to significant improvements in the operating characteristics of the solar activated cooling equipment.

C. FACTORS FAVORING ECONOMICS OF SOLAR SPACE CONDITIONING

Energy conservation by use of solar heating and cooling equipment requires a major investment on the part of the building owner. The return on his investment is the cost saving achieved by partial displacement of conventional energy over the lifetime of the solar equipment.

From previous studies of solar system design and performance, a number of important general principles can be stated relating to the overall economics of solar space conditioning systems:

- For a given load, system annual efficiency is highest for small systems;
- System unit capital costs (dollars per unit solar collector area) are lowest for large systems;
- The most cost effective system meets only a fraction of the total load (a backup system will be required); and
- Unit costs of energy displaced by the solar system are generally more significant to overall economics than the insolation.

Hence, it is clear that the optimum size of a solar system is a function of a number of variables. We have found from previous studies (Reference 7) that the savings-to-investment ratio will be relatively insensitive to system size and near maximum for solar cooling systems providing between about 20-50% of the space conditioning load. When carrying out performance evaluations in the present work, systems were sized for this general range.

Overall system economics will also be improved by maintaining conditions leading to high annual system efficiency:

- Displacement of a steady load,
- Collection at "low-to-moderate" temperatures,
- Use of high COP devices and high performance collectors, and
- Low storage losses.

The final three of these conditions relate to the characteristics of the solar collection and space conditioning equipment. The first relates to the functions performed. Generally, utilization of the equipment for solar cooling alone will tend not be cost effective unless cooling is required year around. In the present work, savings-to-investment ratios have been calculated assuming that the solar equipment is utilized to provide heating as well as cooling. While the overall economics is now dependent on the ratio of the annual heating and cooling loads, to do otherwise would unrealistically penalize the solar system economics by allowing the capital intensive equipment to lie idle during significant portions of the year.

D. STUDY APPROACH

Figure I-3 illustrates the steps carried out in the analysis. The approach can be broadly classified into five major steps:

- Systems--various cooling system approaches were identified and characterized;
- Status--the status of each of the major approaches was reviewed;

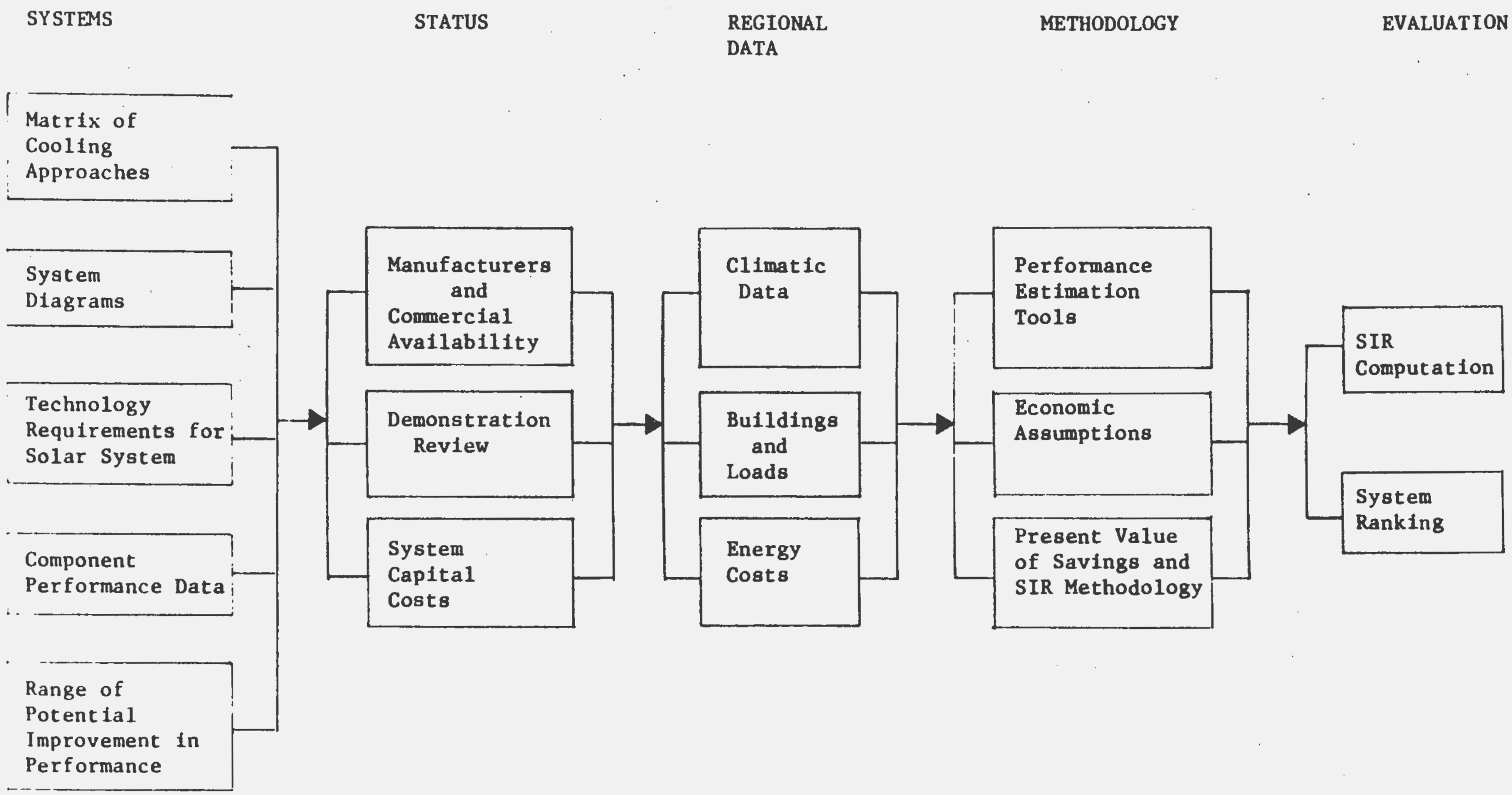

FIGURE I-3 STEPS IN ANALYSIS

- Regional data--climatic and energy cost data were collected and interpreted;
- Methodology--a methodology for calculating life-cycle costs was developed; and
- Evaluation--systems life-cycle costs were calculated and compared.

A matrix of systems and components were developed where the systems were characterized by application, commercial status and technology requirements of the supporting solar collection and storage subsystem. System diagrams were developed for the various approaches, and solar system component performance data were collected and summarized. Potential improvements in system performance were identified.

The commercial status of each of the candidate approaches was examined, and major manufacturers of equipment were identified. A review of pertinent demonstrations of solar cooling was made and conclusions developed relating to the systems demonstrated. Solar system capital costs were estimated for the candidate approaches.

Regional insolation, degree-day data and energy cost data were collected for seven geographic locations.* From the climatic data, heating and cooling loads were estimated for typical single family residences and typical office buildings in the seven locations.

A methodology was developed for determining the percentages of the annual heating and cooling loads met by the solar systems. Economic assumptions pertinent to life-cycle costing were reviewed, and a simplified approach to determining life-cycle costs and savings-to-investment ratios (SIR) was developed.

Using the data and methodology developed, computations were made for the SIR of the various cooling approaches. Parametric analyses were also carried out to determine the sensitivites of the life-

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*Bremerton, Chicago, New London, Norfolk, San Diego, Corpus Christi and Key West.

cycle costs to economic assumptions as well as to performance data. Based on these results, conclusions were drawn relating to the economic performance potentials of the various approaches and to the factors necessary to achieve cost effective solar cooling.

II. STATUS OF SOLAR COOLING TECHNOLOGY

A. GENERAL APPROACHES TO SOLAR COOLING

The major options for solar cooling are the absorption machine, heat driven (Rankine) refrigeration cycles, and desiccant systems. The basic approaches are illustrated in Figure II-1, where the location of solar heat input and cooling (heat extraction) are indicated for each approach.

The absorption approach utilizes the affinity of various combinations of liquids to achieve virtual compression of a refrigerant. Following generation of the refrigerant vapor by solar heat input, the remainder of the cycle is similar to the standard vapor compression cycle. The Rankine cooling cycle uses solar heat input to a heat engine to produce shaft power by expansion of the heated working fluid. The powered shaft is coupled with a conventional mechanical type vapor compression unit to achieve cooling. Desiccant systems use solid or liquid desiccant materials which can be regenerated by solar heat. The devices provide a means of dehumidification and can be combined with partial rehumidification to achieve cooling. A number of different system configurations are possible for the desiccant cooling approach.

A fourth approach, which has received very little attention to date, is the generation of electrical energy by solar cells which is then used to power a conventional vapor compression unit. Figure II-2 illustrates the concept of a solar cell powered heat pump system. In addition to the solar cell array and heat pump, the system contains: a power conditioning unit containing a battery pack, and other thermal storage and fluid transport components. The system would utilize available power from the electric utility during periods of low insolation.

A summary of the status of the major cooling approaches is given in Table II-1. Continuous absorption machines represent the best developed heat actuated air conditioning technology. Significant development efforts are required to bring the other approaches to commercial status. With the possible exception of the desiccant system, high performance* collectors are required. While some development efforts are underway on the solar heat activated cooling approaches, the solar cell-heat pump

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*Good collection efficiency at temperatures over 200°F.

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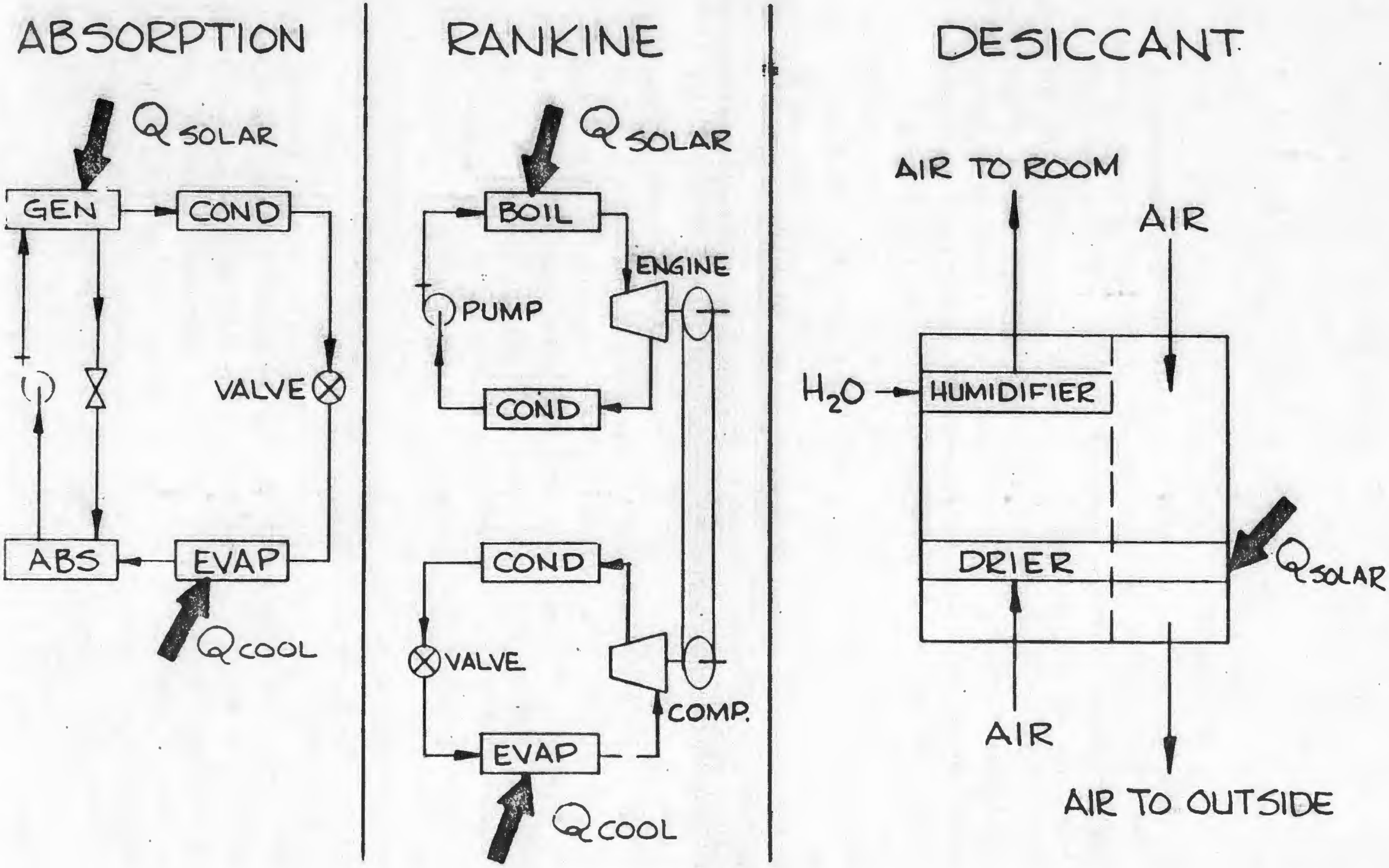

FIGURE II-1 SEVERAL APPROACHES TO SOLAR COOLING

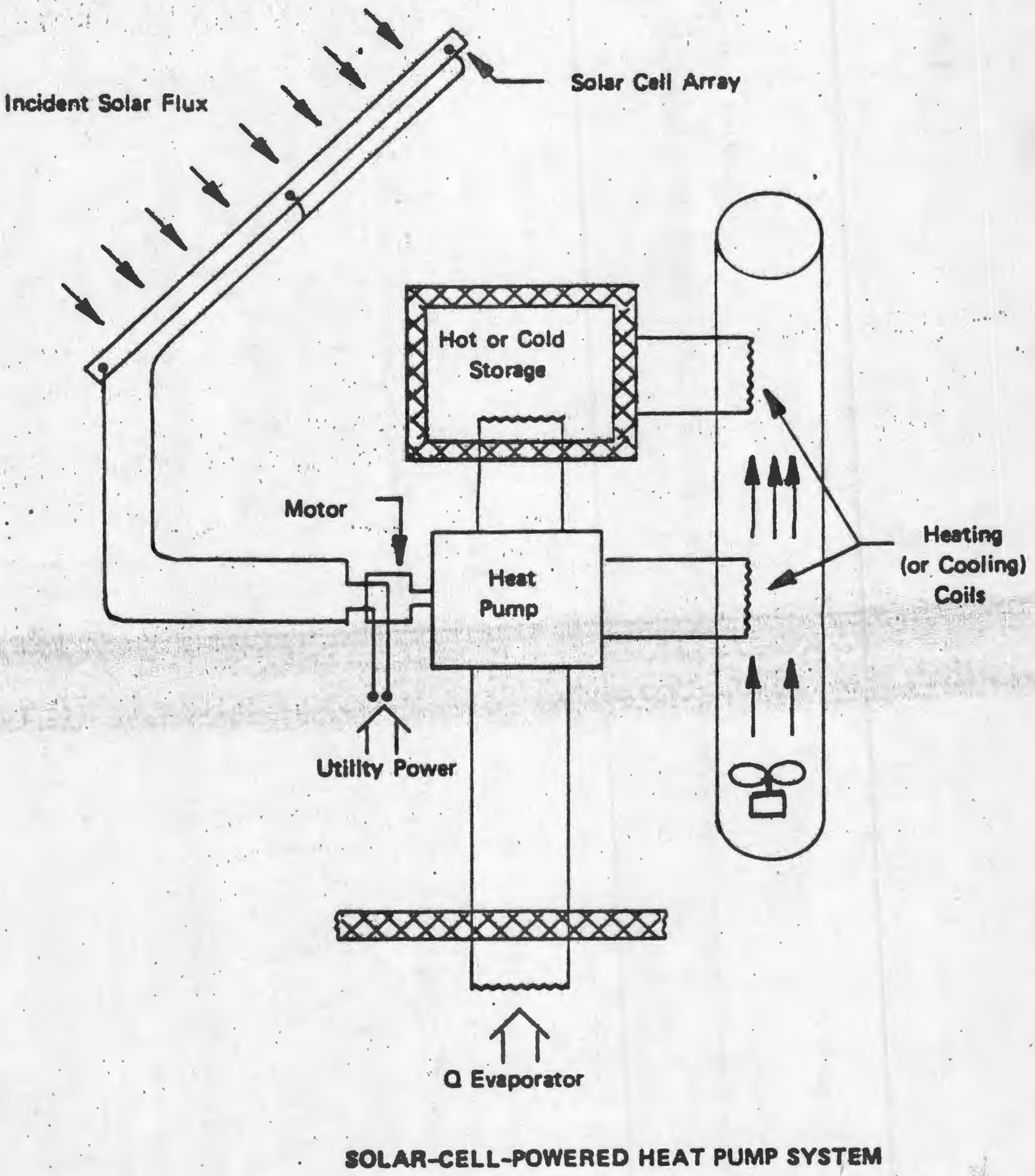

FIGURE II-2 SOLAR-CELL-POWERED HEAT PUMP SYSTEM

TABLE II-1 SOLAR COOLING APPROACHES

SYSTEM	COOLING FUNCTIONS	STATUS	TECHNOLOGY RESUIREMENTS		
			DEVELOPMENT	MAJOR COST REDUCTIONS	HIGH PERF. COLLECTOR
Absorption	Cooling				
Continuous		Available	Little		X
Intermittant		Lab Experiments	Major		X
Organic Rankine Cycle	Cooling	Prototype Demonstration	Major	X	X
Desiccant	Dehumidify				
Liquid	with	Early Demonstration	Moderate	X	
Solid	Evaporative Cooling	Lab Experiments	Moderate	X	
Solar Cell-Heat Pump	Cooling	Concept	Major	X	n/a

approach is in the conceptual state.

Detailed examinations of the state-of-the-art of the heat actuated cooling approaches are given in the following sections. An evaluation of the solar cell-heat pump is not undertaken in this report since no known development efforts are underway in this direction.

B. ABSORPTION CYCLE SYSTEMS

1. Theory Operation

a. Cycle Operation

The basic (single-effect) absorption cooling system is shown schematically in Figure II-3. The working fluid is a solution of refrigerant and absorbent which have a strong chemical affinity (absorption) for each other. This chemical reaction allows pumping of the refrigerant in the form of a refrigerant-absorbent solution rather than compressing it as a vapor, with a mechanical power input requirement considerably below that of a vapor compression cycle.

There are basically three temperature levels in the system. The high temperature level is in the generator heat exchanger where heat is input to energize the cooler. Intermediate temperature levels exist in the absorber and condenser heat exchangers through which cooling water flows for purposes of heat rejection. The low temperature level occurs in the evaporator where cooling is produced. Heat enters the absorption system through the generator and evaporator and is rejected from the absorber and condenser to the cooling water.

The system operates as follows. Heat is added to the solution in the generator from a high temperature heat source (e.g., solar thermal storage) where the refrigerant is vaporized. The vapor goes to the colder region (the condenser) in the high pressure side of the unit, where heat is removed and the vapor is liquified. A "concentrated" solution (weak in refrigerant) is left behind in the generator.
The liquid refrigerant then expands into the low pressure evaporator where it vaporizes and cooling is produced. The vaporized refrigerant flows to the low pressure absorber chamber where it combines with the concentrated absorbent from which the refrigerant was initially obtained.

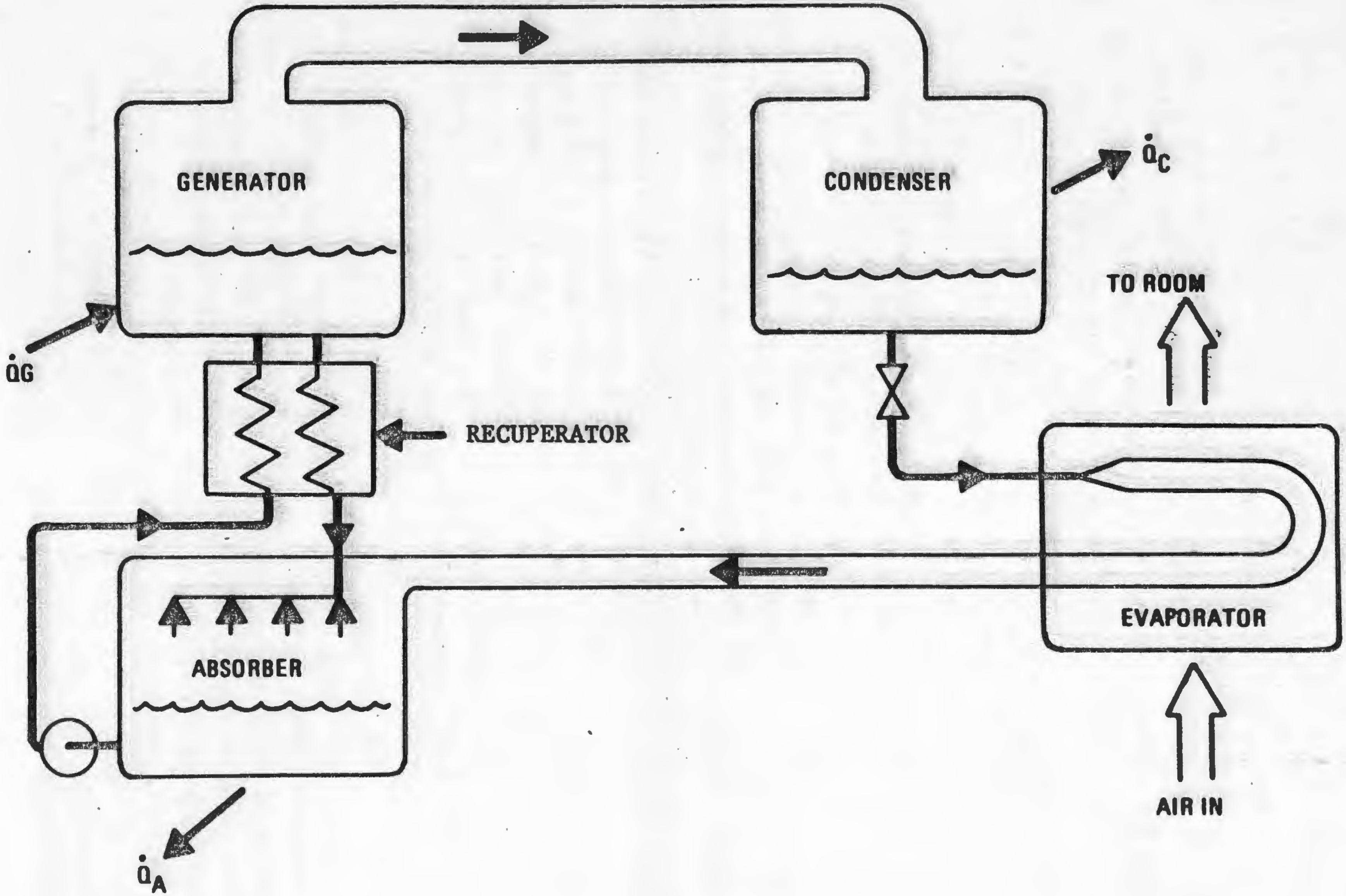

FIGURE II-3 ABSORPTION SYSTEM SCHEMATIC (REFERENCE 9)

Since this recombination reaction is exothermic, heat must be removed from the absorbant to maintain its temperature at a sufficiently low value to ensure a high affinity for the refrigerant. The solution is then pumped back to the generator so the cycle can continue. A recuperator is used to exchange heat from the high temperature concentrated solution, circulated from the generator to the absorber, to the low temperature solution pumped back to the generator. The recuperator reduces the net heat input requirement to the system by about 25%.

The affinity of the solution for the refrigerant is determined by the solution concentration and temperature in the absorber chamber. The pressure in the absorber (established by the absorber temperature and refrigerant vapor pressure) equals evaporator pressure and, therefore, determines the temperature at which refrigerant vaporization occurs. Similarly, the temperature of the condenser determines the pressure in the condenser and generator. Corresponding to the pressure and to the solution concentration in the generator (prior to heat input) a minimum temperature must be maintained to ensure continued vaporization of the refrigerant. Hence, for each heat rejection temperature and evaporator temperature, a minimum temperature is required to produce cooling. Physically, this corresponds to the saturation temperature of the solution. (Below this temperature, the cycle will not operate.) The minimum generator temperature is also dependent on the flow rate of the solution between the absorber and generator, with a higher generator temperature required for lower flow rates.

Figure II-4 defines this minimum theoretical generator temperature required to maintain an evaporator temperature of 45°F as a function of the heat rejection temperature,* with an infinite circulation rate of the absorbent. The figure is valid for the two most common refrigerant-absorbent combinations: water-lithium bromide and ammonia-water. The temperatures are defined internal to the machine. For example, if cooling water at 85°F were supplied to the machine with a 15°F temperature drop across the absorber and condenser heat exchangers,

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*Assuming condenser and absorber heat rejection occur at the same temperature.

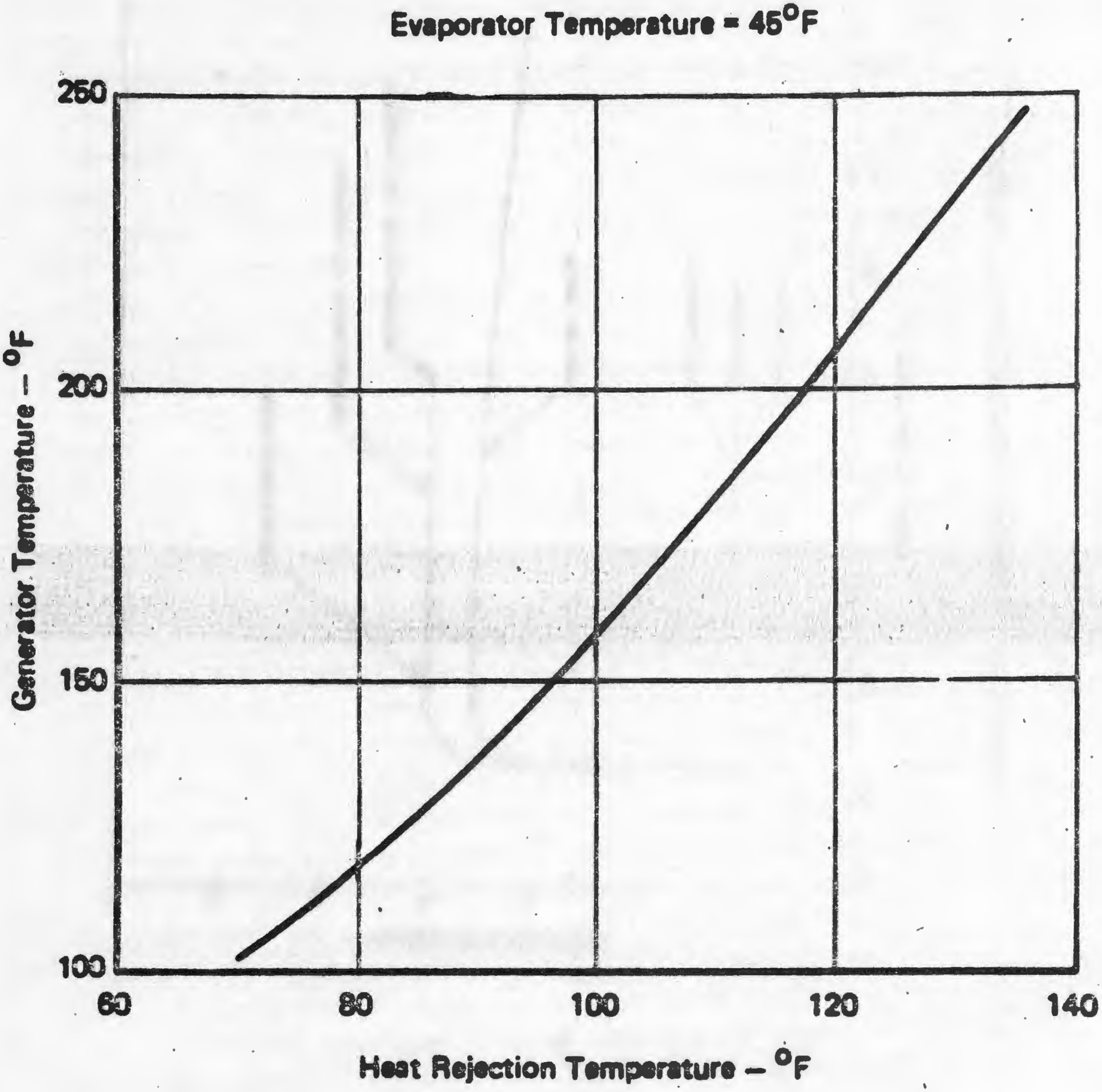


FIGURE II-4 MINIMUM GENERATOR TEMPERATURES FOR LITHIUM BROMIDE AND AMMONIA ABSORPTION CYCLES the minimum generator working fluid temperature would be 160°F--corresponding to an inlet temperature of about 175°F with a 15°F temperature drop across the generator heat exchanger. Although not shown in the figure, the minimum required generator temperature is reduced at higher evaporator temperatures. As a rough approximation, a one degree rise in evaporator temperature yields a one degree drop in required generator temperature.

The plot shows that the minimum generator temperature is quite sensitive to the heat rejection temperature, explaining why most solar-powered absorption machines are water cooled. For air cooling, the required generator temperature will be higher.** A heat-rejection temperature of 130°F, for example, implies a minimum generator temperature to about 225°F. In practice, air cooled water-ammonia units are designed to operate with generator temperatures of about 350°F--a consequence of the reasonably low circulation rates dictated by high cycle pressures and significant pump work.

b. Performance Characteristics

The coefficient of performance of an absorption air conditioner, defined as the ratio of the heat transfer rate into the evaporator to the heat transfer rate into the generator, can be calculated as a function of the various temperature levels. Figure II-5 presents the theoretical coefficient of performance of ammonia-water and lithium-bromide air conditioners. The results, which are taken from Reference 9 were obtained by assuming a condenser temperature 10°F higher than the absorber temperature and assuming a recuperator effectiveness of 90%.

The theoretical coefficient of performance is seen to be relatively insensitive to variations in generator, absorber and condenser temperature, once the minimum generator temperature level is reached to sustain operation. This arises because the heat of vaporization of the refrigerant is nearly independent of generator temperature, and is the major term in the generator heat load. The figure also shows a higher theoretical COP for the water-lithium bromide system than for the ammonia-

**Other problems can arise at high generator temperatures as will be discussed later.

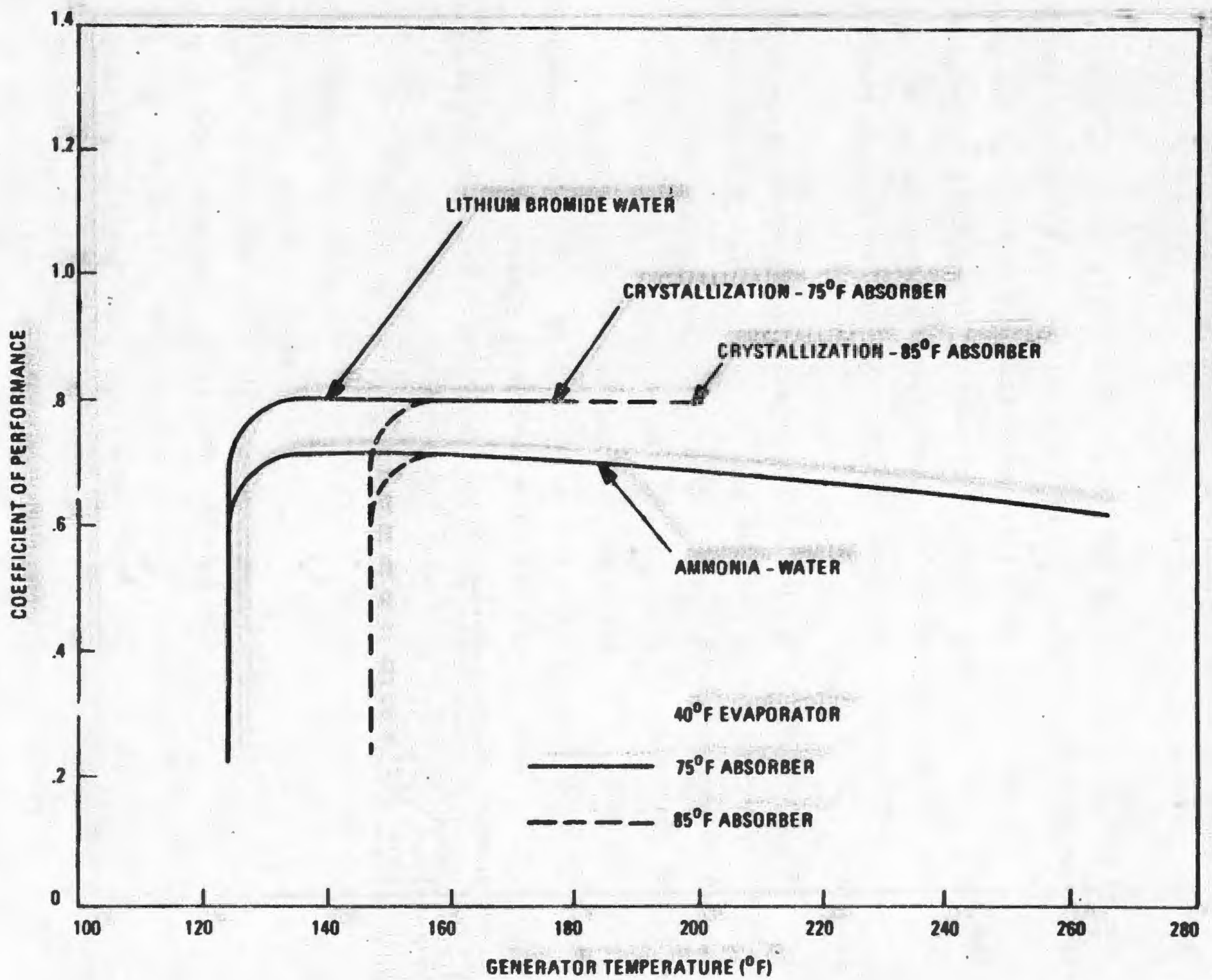


FIGURE II-5   ABSORPTION SYSTEM PERFORMANCE (REFERENCE 9) water system, but also indicates a maximum generator (working fluid) temperature corresponding to the onset of crystallization. (The phenomenon of crystallization and its attendant limitations on performance of a lithium bromide unit will be discussed later.)

The theoretical COP is also dependent on the evaporator temperature, and tends to rise with increasing temperature. Hence, it is frequently advantageous to operate with moderately high evaporator temperatures (50-55°F) to minimize the net heat input requirements. However, the maximum theoretical COP is less than unity since the evaporator heat transfer is equal to the heat of vaporization of the refrigerant (plus any superheat), while the heat input to the generator is equal to the heat of vaporization of the refrigerant, plus an effective heat of solution, plus unrecovered sensible heat in the recuperator, plus heat to superheat the refrigerant vapor. If the solution heat exchanger (recouperator) has an effectiveness less than unity, the unrecovered sensible heat must be supplied by the generator and discharged by the absorber. A heat exchanger can be placed to pre-cool the liquid refrigerant leaving the condenser by the vapor leaving the evaporator as an additional means of increasing the coefficient of performance. However, the gain in COP is usually small (about 2%) and is usually not considered sufficient to justify the investment.

The capacity of the unit is also related to the operating temperatures. Above the minimum required temperature, the capacity increases almost linearly with generator temperature, as more refrigerant is vaporized from the solution. (The variation in capacity with generator temperature is typically 1.5 - 2.0%/°F, expressed as a percentage of nominal capacity.) However, above a certain generator temperature, some carry over of solution to the refrigerant circuit can occur, resulting in a loss in coefficient of performance due to inhibiting the performance at the evaporator and excessive heating of solution in the generator with some heat removed directly in the condenser. An increase in evaporator temperature will increase the operating capacity at a constant generator temperature, or allow a decrease in required generator temperature at constant capacity. The pressure (and saturation temperature) in the evaporator is controlled by the absorbent concentration in the absorber, and the absorbent concentration is determined by the heat input to the generator; hence, a higher pressure in the absorber requires less refrigerant vaporization in the generator.

With a reduction in cooling water temperature, the required generator temperature to maintain a given capacity will also drop. Typically, a 2°F drop in cooling water temperature yields a 3°F decrease in the required generator temperature. However, the maximum capacity of the machine will drop as the cooling water temperature decreases. If the generator temperature is not decreased, solution carry over may occur as discussed earlier. The reduction in the maximum capacity as the cooling water is decreased is a natural function of: the increased velocity and specific volume of the refrigerant through the eliminator separating the generator and condenser, the reduction of pressure difference between the generator and absorber to move the absorbent through the recouperator and piping, and the reduction in pressure between the condenser and evaporator to move the refrigerant condensate to the evaporator. Although the maximum capacity decreases with lower cooling water temperatures, the COP generally tends to increase. Hence, there are definite performance advantages to operating at lower cooling water temperatures.

c. Control Considerations

Because of the sensitivity of capacity and COP to the various temperature levels (generator, absorber, condenser, evaporator), fairly sophisticated controls are required to obtain near optimum performance of an absorption unit. Other areas of concern are: (1) limiting the generator temperature on the high side to avoid crystallization problems inherent with lithium-bromide units, (2) maintaining refrigerant free of contamination and increasing operating COP by reducing solution circulation rate at low loads, and (3) reducing losses in cooling performance caused by frequent cycling of the machine.

In most early installations of absorption cooling systems, controls and bypass arrangements were used to supply constant cooling water temperatures to the machine. Presently, there are efforts on the part of designers to allow machine operation with variable cooling water temperatures to take advantage of increased COP and reduced required generator temperatures with lower cooling water temperatures. Generally, with reduced ambient temperatures--and reduced cooling load--lower cooling water temperatures will be available to the machine. As noted above, the machine's maximum capacity will decrease as the cooling water temperature drops and a decrease in generator temperature may be required to prevent solution carry over to the refrigeration circuit.

As the load drops and the required generator temperature drops, the COP may be reduced unless the solution flow rate is decreased. Such a decrease in flow rate (brought about by controls) will increase the concentration and thus the COP. However, care must be exercised to avoid crystallization problems in the lithium bromide unit at too high an absorbent concentration.

Part load conditions may also cause the machine to cycle on and off at moderately frequent intervals. During the "off" period, the machine may cool down, requiring the expenditures of additional heat energies to bring the unit back to operating temperature at the beginning of an "on" period. An experimental study of the warm-up transient losses in a small tonnage lithium bromide unit has shown required warm-up times of upwards of ten minutes (Reference 11). This type of loss in operating performance can be reduced in a solar system by the method of integration into the rest of the system. (An example is discussed later where the results of demonstrations of solar cooling are analyzed.)

2. Continuous Absorption Machines

a. Single-Effect Type

Lithium Bromide--Water

The lithium-bromide single-effect water-cooled cycle is the most common commercial absorption machine. Several manufacturers (Carrier, York, Trane and Arkla) market units of 100 tons or greater capacity. Arkla has stock equipment down to 15 tons, the 100-ton and 25-ton units being available in water-fired versions. Arkla formerly marketed 3-ton direct fired heating and cooling machines, and has modified for water firing and supplied a number of these machines for solar cooling demonstration projects. A modified 25-ton machine, optimized for solar operation with full rated capacity at 195°F water, is being made available by Arkla. Yazaki Company in Japan also markets lithium-bromide machines of various capacities, including a 2-ton residential sized unit.

The single-effect lithium bromide cycle has some very attractive aspects. The low (sub-atmospheric) cycle pressures minimize pumping power. In the smaller units, a thermally driven vapor life pump (bubble pump) is used to circulate the solution from the absorber to the generator.* In addition, because of the low vapor pressure of lithium-bromide, it can be contained in the high pressure side of the system without great difficulty.

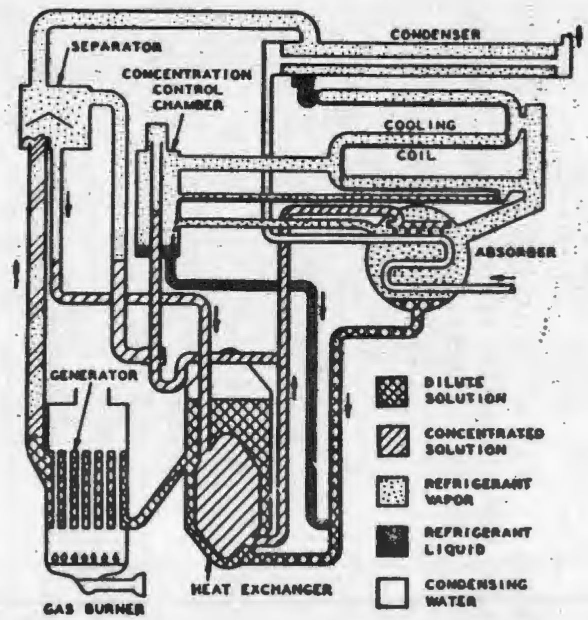
Figure II-6 illustrates cycle arrangements for two single-effect lithium bromide-water machines. The direct fired unit (Figure II-6A) is similar to the Arkla 3-ton unit. The unit differs from the simplified cycle diagram shown in Figure II-3 by the substitution of a vapor lift pump for a mechanical solution pump and the addition of a concentration control device. The figure shows a "gas burner" for heat input; in the version modified for solar firing, the burner is replaced by a hot water heat exchanger.

In this smaller unit, the evaporator coil is constructed of finned tubes over which passes the air to be cooled. The absorber is a cylindrical shell which contains a coil through which cooling water is circulated. The solution flowing into the top of the absorber is distributed over the entire outside surface of the coil to expose the maximum area of absorbent to the refrigerant vapor. The concentration control chamber provides a liquid seal so refrigerant vapor cannot flow directly from the generator into the absorber.

Figure II-6B shows a large tonnage steam-fired water chiller similar in design to the one shell units manufactured by Trane. In this unit, as in most large tonnage machines, a mechanical solution pump is used. Pumps are also employed for circulating absorber and refrigerant fluids.

Figure II-7 shows idealized operating cycles for lithium-bromide coolers of several manufacturers. The figure shows vapor pressure and

*Pumping action relies on a density difference between fluids entering and leaving the high pressure side of the system. The bubble pump is limited to low differential heads and low flow rates.



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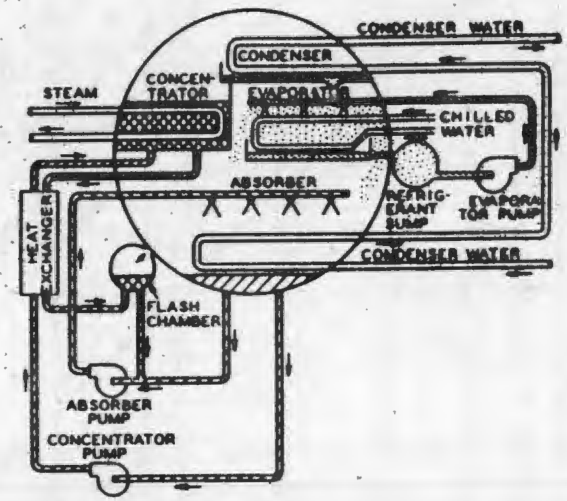
A. Direct Fired Heating and Cooling Unit



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B. Steam Fired One-Shell Water Chiller

FIGURE II-6 SINGLE-EFFECT LITHIUM BROMIDE CYCLE

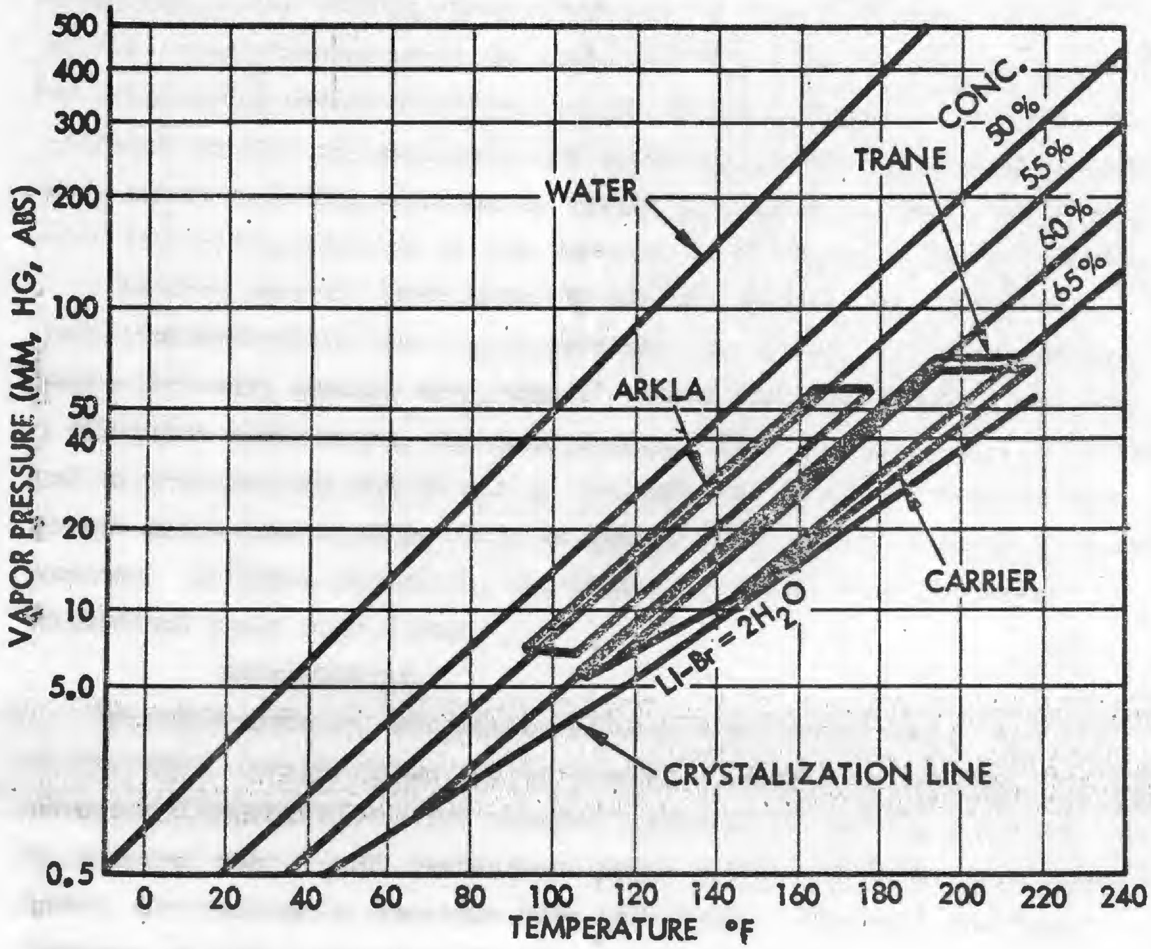

FIGURE II-7 OPERATING CONDITIONS OF VARIOUS VAPOR ABSORPTION REFRIGERATORS (REFERENCE 4) temperature states corresponding to constant solution concentrations. In addition, the conditions where crystallization can occur are also shown as a function of temperature. The operating cycles shown in the figure represent the pressure-temperature states of the absorbent-refrigerant solution between the absorber and the generator. The high pressure side corresponds to conditions in the generator; the low pressure part of the cycle corresponds to conditions in the absorber. As the temperature to the generator increases, the solution concentration increases. After vaporization of the refrigerant in the generator, the solution is circulated towards the absorber for heat removal. If the solution concentration is too high upon leaving the generator, crystallization of the solution can occur at the outlet end of the recuperator (prior to entering the absorber). Hence, input temperatures at the generator are limited below a maximum temperature to avoid crystallization problems.

The smaller lithium bromide units tend to operate at lower solution concentrations to prevent crystallization. Hence, a higher percentage of rated capacity can often be obtained with the smaller units when firing by solar heat input using flat plate collectors. The larger cooling units generally operate at higher concentration--and, therefore, higher generator temperatures--to allow operation at higher absorber temperatures, thereby reducing the cost expenditures on the heat rejection equipment.

In the smaller units which employ a bubble pump for circulation of the absorbent solution, crystallization may also be a problem at low generator temperatures because of low circulation rates resulting from a decrease in the density difference between the generator and absorber. Substitution of a mechanical pump between the absorber and generator eliminates this problem.

As noted earlier, lower generator temperatures are required at higher evaporator temperatures, and/or at lower water cooling temperatures. Hence, from a standpoint of minimizing the potential for crystallization as well as obtaining high coefficients of performance, it is desirable to use high evaporator temperatures where possible, and to take advantage of low cooling water temperatures when available. In some of the larger tonnage machines automatic decrystallization is accomplished by circulating heated solution through the recuperator. In addition, optional devices are available for detecting the onset of crystallization with dilution of the solution by liquid refrigerant. These devices will dilute the solution in the instance of an electric power failure.

Industry sources (References 48 and 49) report that commercial type lithium-bromide absorption machines can operate reliably for many years with only general maintenance. Other than operation at too high a generator temperature, the main problems that can arise are corrosion and crystallization due to air or hydrogen in the unit. Some of the larger units have a purge cycle to remove noncondensable gases from the machine. To limit corrosion, the inhibitor should be analyzed and maintained about once a year.

Water-Ammonia

The water-ammonia absorption cycle is also commercially availabe in the small tonnage range (up to 10 tons). This cycle has no solubility limitation on the absorber temperature and the generator can be operated over a wide temperature range without problems of crystallization. Hence, air cooling is feasible with this cycle. However, the cycle operates at high pressures (70-300 psi) requiring mechanical circulation of the solution, with relatively high circulating power, thereby discouraging high circulation flows. Because the vapor pressure of the absorbent, water, is not low at operating pressures a rectifying unit is required to prevent circulation of the water to the refrigerant loop.

Design generator temperatures of the commercially available air cooled machines are quite high, about 350°F. Operation near 50% capacity can be obtained at 250°F input temperature. Nominal COP values are near 0.5-0.6.

Table II-2 provides a comparison between the water-lithium bromide and ammonia-water absorption cooling systems. The required generator input temperatures are normally higher for the ammonia-water systems. However, because ammonia is the refrigerant, the machines may be used

TABLE II-2 COMPARISON BETWEEN H_2O-LiBr AND NH_3-H_2O SYSTEMS

AREAS OF COMPARISON	H_2O-LiBr	NH_3-H_2O
Capacity (commercially available)	30-700 tons	2-10 tons
Generator Input Temperatures	180-260°F	250-300°F
Cooling Coil Temperature	45°F	5°F
Water Cooling Required	Yes	No
Pressure Range	0.2-2.2 psia	70-350 psia
US Building Code Restriction on Indoor Application	No	Yes
Salt Precipitation	Yes	No
Rectifier Needed	No	Yes

for refrigeration, providing cooling temperatures down to 5°F. Because of the high pressures of operation and the toxicity of ammonia, there are restrictions on applications of the ammonia-water cycle machines. The machines cannot be used to directly cool the building air, but must be used as a chiller.

Less emphasis has been placed on the use of ammonia-water cooling systems for solar cooling because of the higher operating temperatures required. Although the ammonia-water units can be air cooled, collection efficiencies at the required operating temperatures are normally too low for flat plate collector systems.

There is very little activity in the industry aimed at evolving new or improved ammonia-water cooling systems. The design and research activity is currently centered on the university level. Experimental units have been operated at the University of Florida (Reference 17) and at the University of California (Reference 15). The objectives of the development work being carried out at the two universities are to obtain moderately high COP at generator operating temperatures near 200°F, evaporator temperatures in the region of 45°F-55°F, with air cooling of the condenser and absorber units. The possibility of air cooling appears to be the primary advantage offered by the ammonia-water absorption unit. However, accomplishing these goals will require new designs. Because of the lack of industry interest in this area, it is unlikely that an economically viable chiller (one with an acceptable first cost and high COP) will be available in the near term.

b. Double-Effect Type

The double-effect lithium bromide machine utilizes two stages of concentration to increase the amount of refrigerant generated per unit heat input and thereby improve coefficient of performance. Figure II-8 is an illustration of a two stage absorption machine manufactured by the Trane Company. The super heated refrigerant from the generator drives additional refrigerant from the absorber solution in the generator/condenser while the refrigerant from the first generator condenses. The total refrigerant vapor generated per unit heat input in the first and second stages is almost twice

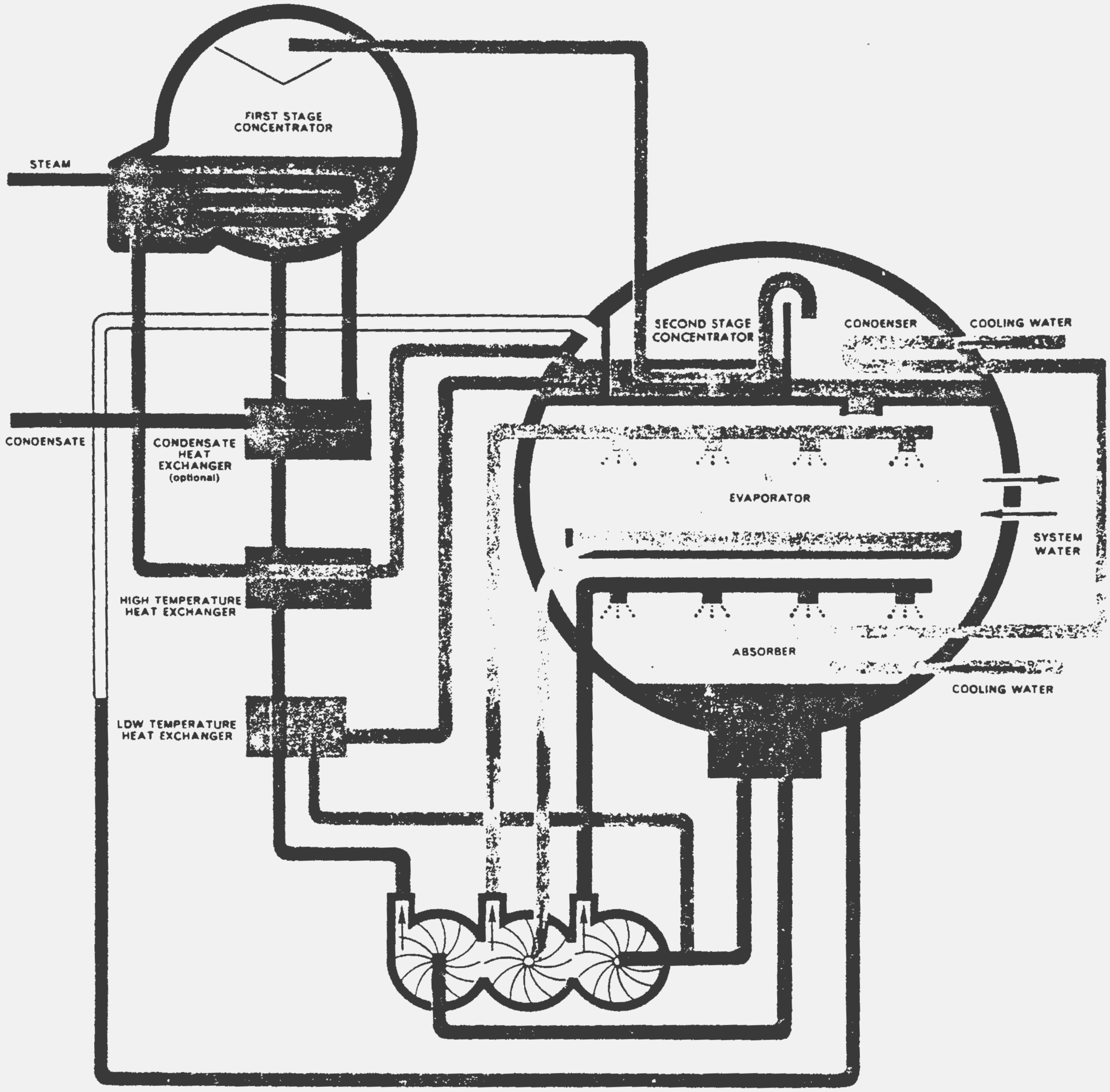

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FIGURE II-8 **DOUBLE-EFFECT LITHIUM BROMIDE ABSORPTION MACHINE** that generated in the single effect machine. For fixed absorber, evaporator, and condenser temperatures, the fraction of the refrigerant from the first stage generator that is condensed is determined by the generator temperature. Figure II-9 illustrates the variation in coefficient of performance as a function of the generator temperature for two working fluid absorber temperatures. These theoretical COP's were calculated (Reference 9) assuming a 40°F evaporator temperature and a condenser temperature 10°F above the absorber temperature. The recuperators were assumed to be 90% efficient. The double-effect lithium bromide machine normally achieves a COP of about 1.1 for generator temperatures above 220°F compared to COP's of 0.65-0.75 for the single-effect machine.

Double-effect lithium bromide absorption machines are marketed by the Trane Company in the U.S., in capacities greater than 380 tons, and by several Japanese manufacturers. Operational reliability with the double-effect machines is reported to be similar to that obtained with the single-effect machine. Since double-effect machines are normally used in applications where cooling is required throughout the year, normal maintenance requires that the inhibitors be analyzed on a six month basis. The double-effect machine is more expensive than the single-effect unit, but can have favorable life cycle costs. Because of the high generator temperatures required for operation, cost effective solar cooling with the double-effect machine would require the use of highly efficient collectors or concentrating collectors. To date, there are no demonstrations of solar cooling reported with double-effect machines.

c. Alternative Fluids

Over the years other working fluids have been investigated for use in the absorption cycle. One of the more promising of these liquids is sodium thiocyanate as an absorbent while ammonia is the refrigerant. This solution is non-toxic, inexpensive, non-explosive, non-corrosive (with iron) and offers the advantage of having a low vapor pressure, thereby eliminating the need for a rectifying unit between the generator and condenser. The elimination of the rectifier has the

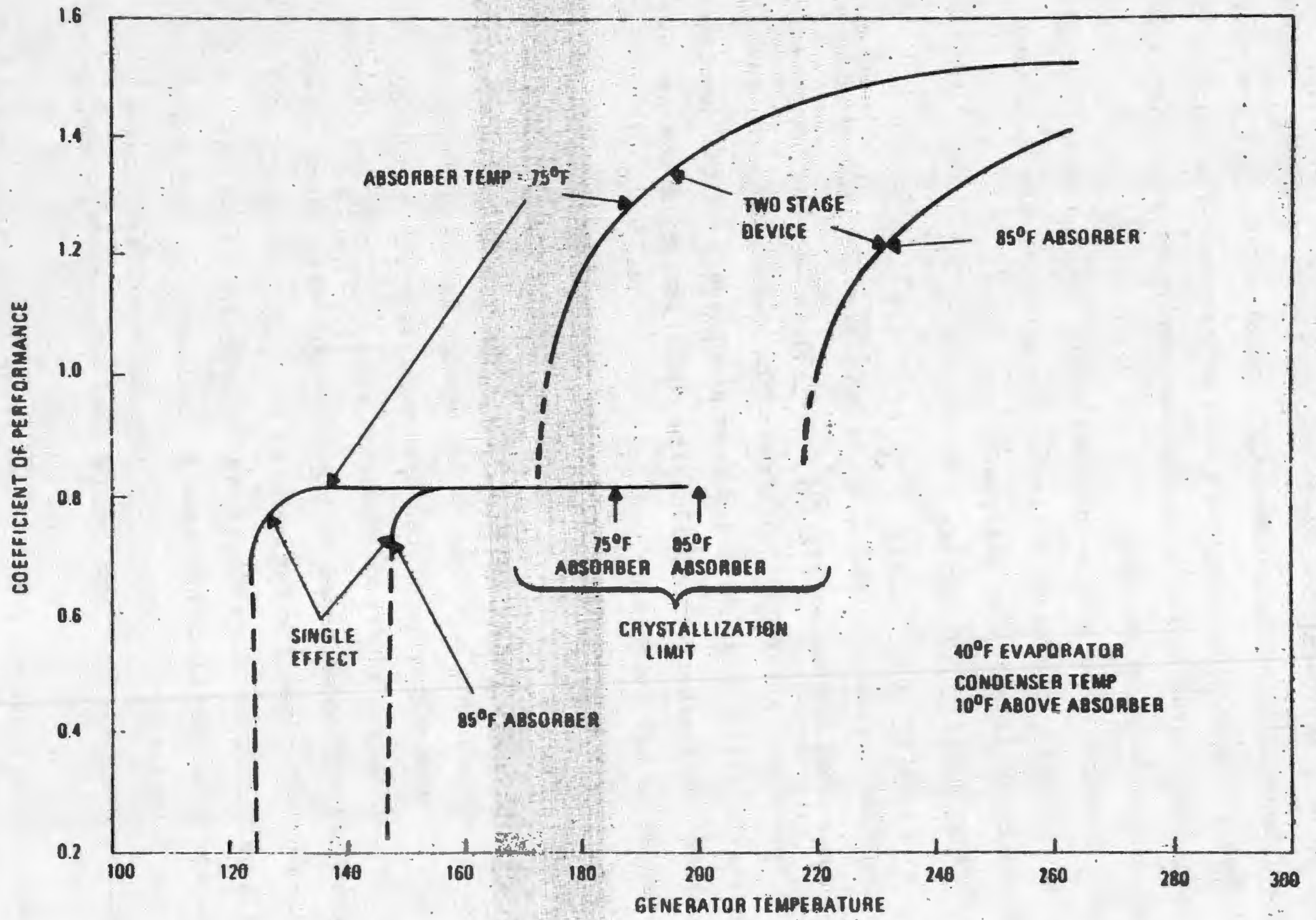


FIGURE II-9 TWO-STAGE ABSORPTION SYSTEM PERFORMANCE (REFERENCE 9) promise of both improving COP, and reducing capital expense. Investigations (theoretical and experimental) carried out for the performance of an absorption cooling unit using this refrigerant-absorbant combination have indicated operating COP's in the same range or lower than obtainable with the ammonia-water solution.

There appears to be little interest within the industry in the use of new fluids. The major interest appears to be the reduction of capital costs in existing designs.

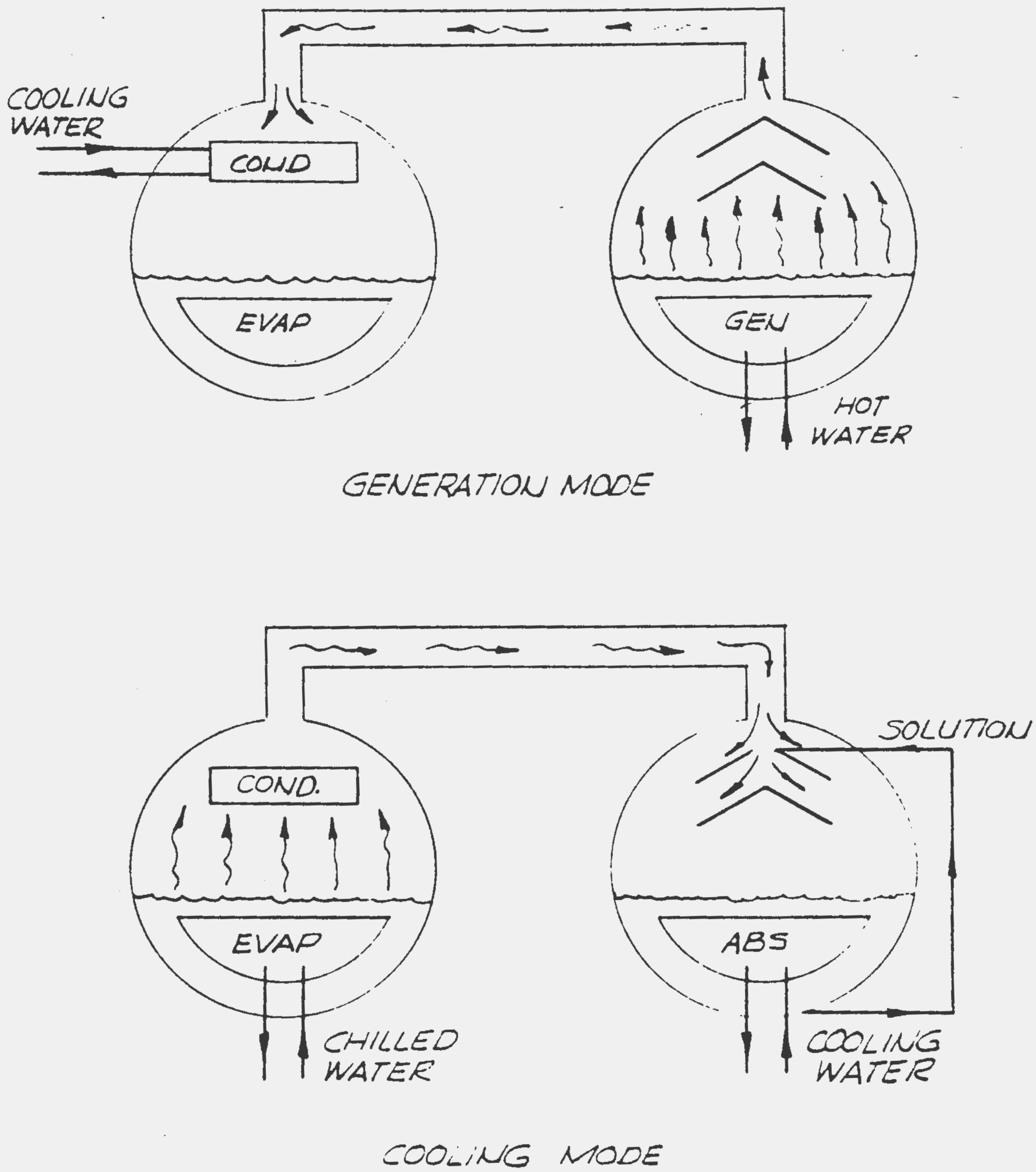
3. Experimental Approaches

a. Intermittent Cycles

Intermittent cycle absorption units have been devised in which a coupled two-vessel system serves the function of the four components found in continuous absorption machines. One vessel contains the absorbent-refrigerant mixture and serves alternately as the generator and absorber. The other vessel contains the refrigerant and serves as the condenser and evaporator.

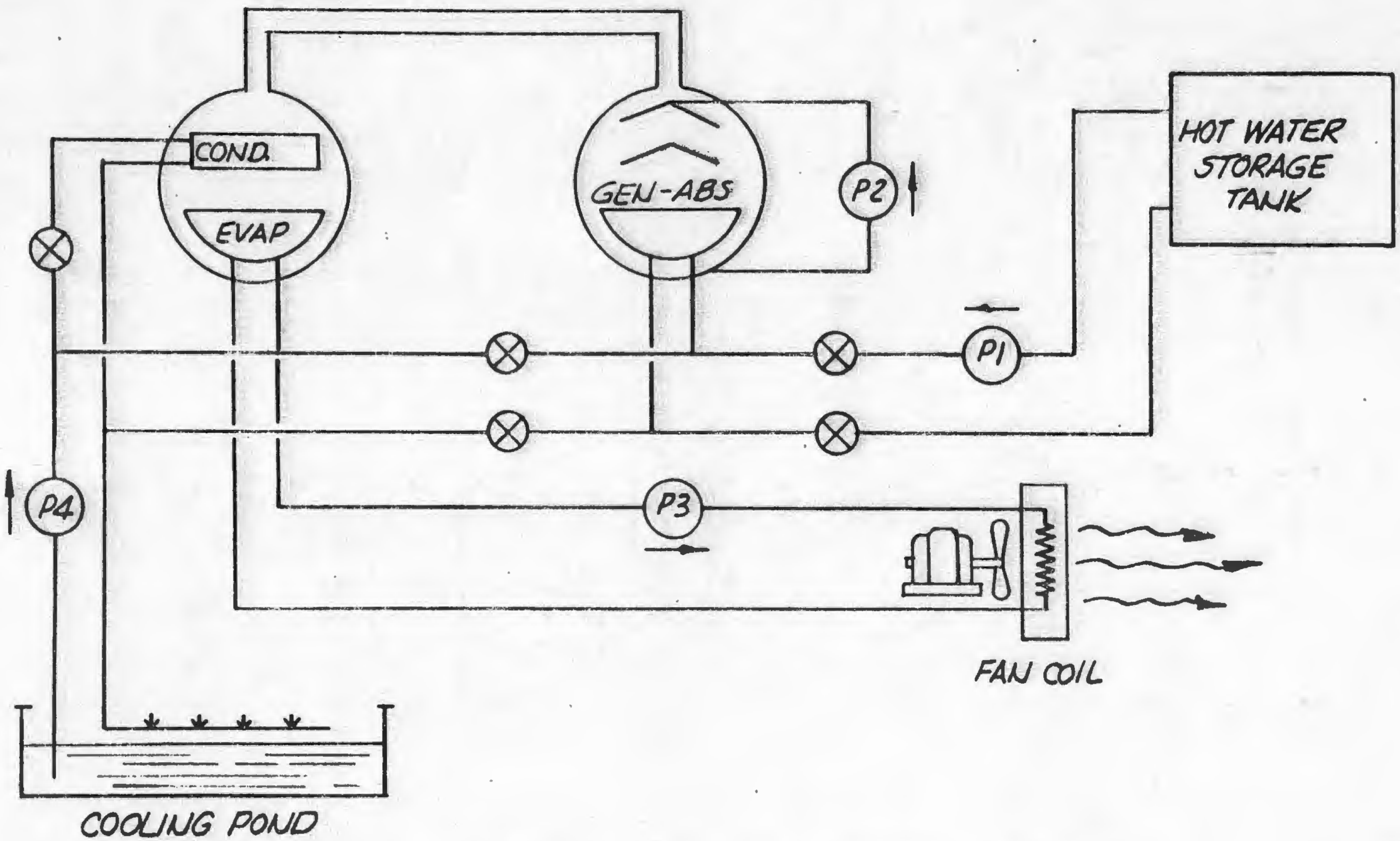
Figure II-10 shows a schematic diagram of the operation of an intermittent (ammonia-water) absorption air conditioning system at the University of Florida (Reference 18). Installation of the intermittent system into a working solar system is illustrated in Figure II-11. The operation of the system is as follows. First, heat is applied to the generator/absorber vessel causing refrigerant to be generated and recondensed in the condenser/evaporator vessel. Air cooling or water cooling can be used to cause condensation of the refrigerant. After the refrigerant is vaporized from the solution, the generator/absorber vessel is allowed to cool and it assumes the function of the absorber. The condenser/evaporator vessel acts as an evaporator and can be used to accomplish useful cooling.

The advantages offered by the intermittent system are simplicity of design and construction, and the elimination of a refrigerant pump. The disadvantages are: the machine cannot provide continuous cooling without adequate cold storage, component sizes tend to be large in order to provide for a refrigerant storage, and high pressures are



SOURCE: UNIVERSITY OF FLORIDA RESEARCH DOCUMENT

FIGURE II-10 SCHEMATIC DIAGRAM OF THE OPERATION OF AN INTERMITTENT ABSORPTION AIR CONDITIONING SYSTEM



SOURCE: UNIVERSITY OF FLORIDA RESEARCH DOCUMENT

FIGURE II-11 SCHEMATIC DIAGRAM OF THE INSTALLATION OF THE SOLAR POWERED INTERMITTENT AMMONIA/WATER ABSORPTION AIR CONDITIONING SYSTEM

obtained in the condenser (especially when air cooled). The coefficient of performance of an intermittent cycle is generally lower than that obtainable with a continuously operating system because of the heat losses associated with alternately heating and cooling the unit. (In a continuous cycle machine a recuperator between the generator and absorber reduces these heat losses.)

The intermittent cycle has generally been proposed in the context of providing refrigeration. Intermittent cycle units are not presently marketed commercially for cooling.

b. Refrigerant Storage

Studies have been carried out to evaluate the performance of absorption systems utilizing refrigerant storage. The refrigerant would be accumulated during hours of high insolation. and expanded to provide cooling at other times to meet the load. Storage is also needed in the absorber to accommodate sufficient absorbent to keep the concentration within allowable limits.

The advantages of refrigerant storage include:

- The volume required to store the cooling capacity is small because of the high heat of vaporization of the refrigerant.
- The refrigerant could be stored at near ambient temperatures where heat losses would be minimized.
- Storage takes place at low or moderate pressures.

At present only analytical studies have been carried out. Further analytical work and testing would be required to prove the feasibility of this approach. An alternative to refrigerant storage would be storage of chilled water (in an external storage unit) provided by a solar fired absorption chiller.

c. Hybrid Systems

A hybrid solar air conditioning system has been proposed which employs the use of vapor compressors between the generator and condenser and between the evaporator and absorber of an absorption unit. The potential advantages stated for such an approach (Reference 21) are a lower intital cost than for the absorption system and a lower operating cost than with the vapor compression system. The proposed system is illustrated in Figure II-12. The vapor compressors allow higher compression of the refrigerant and greater cooling per flow of refrigerant than is achievable with the absorption unit. Only a limited amount of analysis has been performed on this system to date. Unless clear advantages in terms of cost effective cooling were offered by such an approach it is unlikely that sufficient incentives would be offered to manufacturers to develop and bring such a device to the market place.

4. R & D Areas

The primary areas of current research and development on solar activated absorption units are: 1) means of eliminating or reducing crystallization of the absorber, and 2) methods of increasing operating COP. In the smaller units where bubble pumps are presently used to circulate the solution, mechanical pumps can be employed to eliminate the problems of crystallization associated with failure of the bubble pump to operate at low generation temperatures. Under sponsorship by the NSF, Arkla has included this modification to its three ton absorption unit (Reference 10). Arkla is also developing a freeze resistant cooling tower. The conventional cooling tower would be eliminated as a separate package and the costs associated with the labor and material for installing the cooling water lines would also be eliminated.

Cooling system operational COP's can be increased by the use of more efficient or larger heat exchangers. By the use of enlarged heat exchangers and proper control of the fluids to assure operation at relatively constant concentrations, coefficients of performance in the neighborhood of 0.8 to 0.85 could be obtained with 85°F cooling water. However, increased capital costs are associated with the larger heat exchangers. Currently there is little interest in the industry in improving the COP of the conventional fired units. The major interest appears to be in the area of reduced costs.

Another area of importance in obtaining high operating COP's is the control of the temperatures and fluid flow rates at the generator

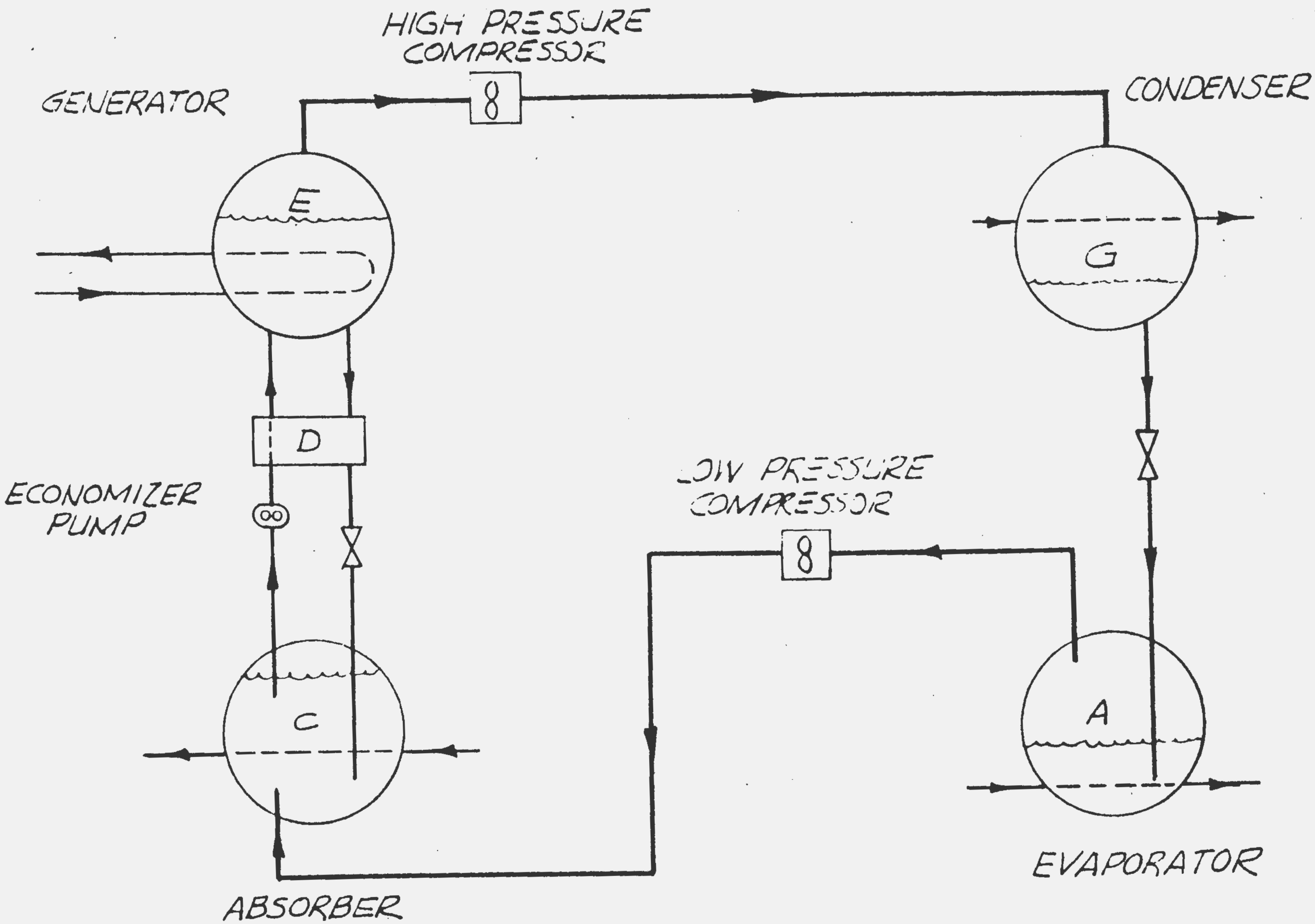


FIGURE II-12   HYBRID SOLAR AIR CONDITIONING SYSTEM and heat rejection stations of the absorption unit. As mentioned earlier, improved COP can be obtained by taking advantage of available low cooling water temperatures with associated reduced generation temperatures. At cooling loads considerably below the rated capacity of the unit, solution flow control could increase the COP.

5. Summary of Technology Status

Table II-3 summarizes the status of the various approaches to solar fired absorption cooling. Only the continuous single-effect and double-effect machines are commercially available. Of these, water-lithium bromide units are available in the widest capacity range, and are more compatible with current technology solar collectors. The main advantage of ammonia-water systems is its compatibility with air cooling. However, in larger installations, water cooling is commonly used in conventional air conditioning systems. Of the remaining approaches, the refrigerant storage concept appears to hold the greatest promise for solar applications, but very little work has been done on this approach to date.

As will be discussed in a later section of this report, cost effectiveness requires that the cooling machines be low first cost, and, hence, commercially available (at some point) or require only slight modifications of available equipment. Because the solar cooling market is not perceived to be large in the near term, the industry has shown only moderate interest in the development of new cooling equipment specifically adapted for solar applications.

C. ORGANIC RANKINE CYCLE ENGINE COOLING SYSTEMS

1. Introduction

One approach for providing solar air conditioning is to use solar heat to drive a heat engine which in turn drives a conventional vapor compressor cooling loop. In principle, any external heat input engine could be used within this basic system concept including:

- Rankine cycles using water (steam plants)
- Stirling engines

TABLE II-3 ABSORPTION SYSTEM APPROACHES

TYPE	STATUS	HEAT REJECTION	TYPICAL OPERATING TEMPS (°F)	TYPICAL COP'S	ADVANTAGES	DISADVANTAGES
Continuous Single-Effect						
• H_2O-LiBr	Commercial	Cooling Tower	170 - 240	0.6 - 0.8	Low Pressure	Crystalization
• NH_3-H_2O	Commercial in Small Tonnage	Air/Cooling Tower	120 - 240	0.1 - 0.5	Capable of Refrigeration	High Pressure Need for Rectifier
• NH_3-N_aSCN	Conceptual	Air/Cooling Tower	120 - 240	0.1 - 0.5	Rectifier Not Needed	High Pressure
Continuous Double-Effect (LiBr)	Commercial	Cooling Tower	300 - 400	1.0 - 1.2	Low Pressure	
Refrigerant Storage						
• Continuous	Conceptual	Cooling Tower	170 - 240	0.5 - 0.7	Match Cooling to Sun's Availability	
• Intermittent	Experimental	Cooling Tower	120 - 240	0.1 - 0.5	Potential Cost Savings	
Hybrid	Conceptual	Air/Cooling Tower	170 - 240		Higher Potential Cost Savings Higher COP	

- Closed Brayton Cycle Systems
- Hybrids of the above

The "Stirling Engines" and Closed Brayton Cycles are best suited for high temperature applications. The use of water in Rankine cycle systems is definitely appropriate at temperatures above 500°F and for higher power applications (probably over 1000 Hp) where freezing is not a problem.

For low to moderate temperature applications (200-400°F) consistent with flat plate or low level focusing collectors, the organic Rankine engine offers the best combination of high thermal efficiency combined with simple, low cost, mechanical hardware.

2. System Description

A schematic of the overall system is shown in Figure II-13. The system consists of:

- A solar collector array which is used to heat the working fluid of a Rankine cycle engine loop via a counterflow heat exchanger.
- An organic Rankine cycle engine power loop.
- A vapor compression cooling loop.

The portion of the system unique to this application is the organic Rankine cycle power loop used to drive the cooling system. The essential elements of the power system are:

- expander
- feedpump
- boiler
- condenser

With some working fluids, the vapor leaving the expander is highly super heated. To increase the cycle efficiency a regenerator may be placed at the outlet of the expander to use the super heat energy for preheating the fluid from the feedpump before it flows into the "boiler."

One of the important advantages of the solar engine driven cooling system indicated by Figure II-13 is that there are two options available:

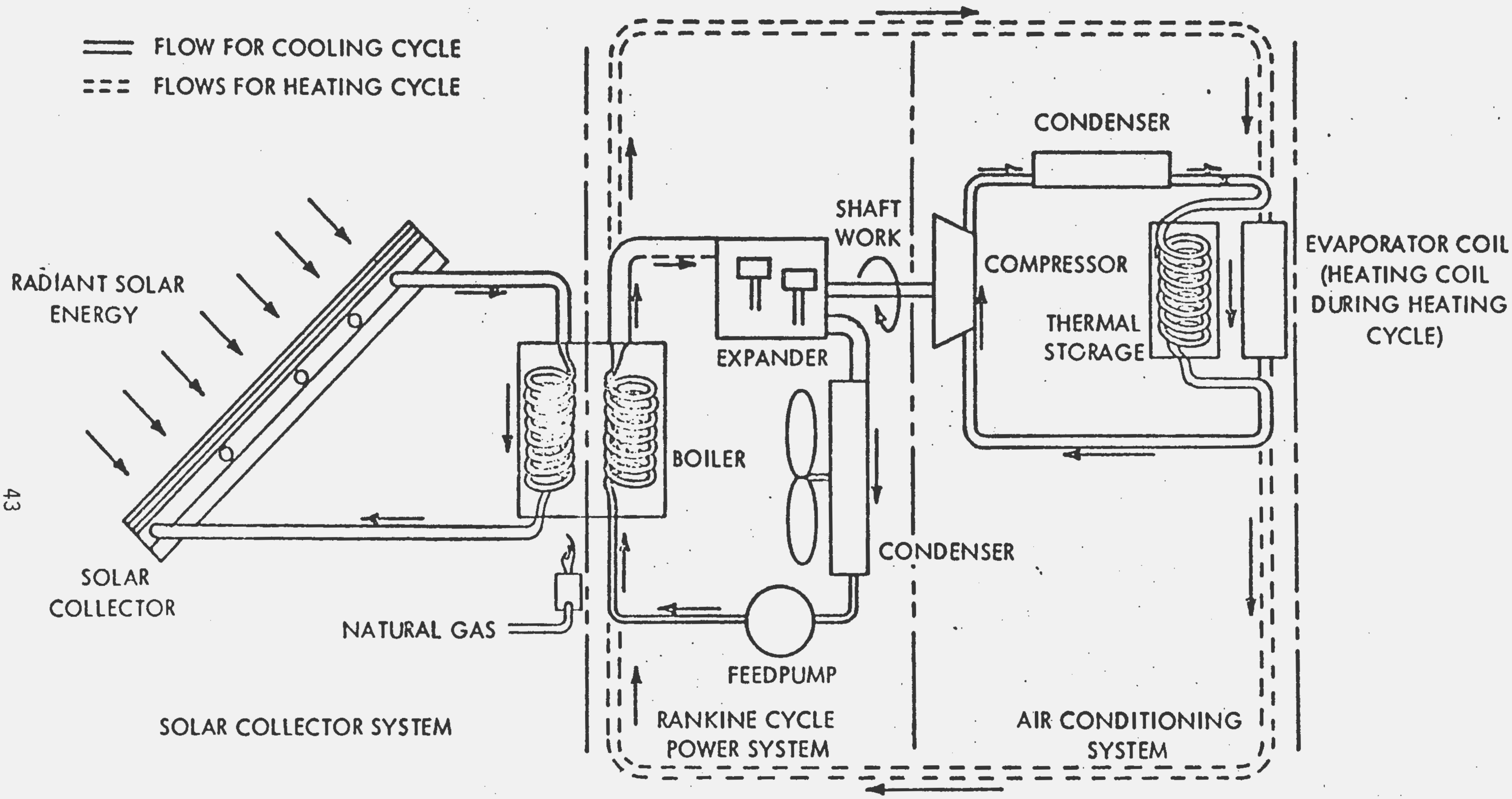


FIGURE II-13 SCHEMATIC OF A RANKINE-CYCLE SOLAR COOLING SYSTEM

- The compressor can be operated by an electric drive operating in parallel with the engine output.
- Additional heat can be provided to the engine boiler by a flame fired system.

In general, it is preferable to drive the compressor directly with electric power. This consumes considerably less fuel in the power plant (utility power, diesel engine/generator, etc.) than by directly flame firing the unit. There may be situations, however, when providing supplemental heat is preferable such as when steam is available as a result of other processes or there is insufficient electrical generating capacity.

3. Engine Cycle Description

Many fluids can be considered for use in the engines. Figure II-14 shows the pressure-enthalpy diagram of a Rankine power cycle using one of the candidate fluids (R-114) operating in a temperature range of interest for solar applications. The cycle indicated would utilize a regenerator. This cycle is described below:

- In a counterflow heat exchanger, heat is transferred from the hot fluid (possibly pressurized water) in the solar collector loop to the boiling working fluid. The working fluid leaves the boiler at an elevated temperature and pressure (212°F, 250 psi).
- The vapor then passes through the expander to produce work. The vapor leaving the expander is slightly superheated, but at a pressure only a few psi above the condensing pressure (45-60 psi).
- The exhaust vapor then passes through a regenerator heat exchanger, where it gives up its superheat energy to preheat the liquid coming into the boiler.
- The working fluid leaves the regenerator at very near saturated conditions and then goes to the condenser, where it is condensed at a temperature of between 80 to 120°F, depending on the type of condenser configuration used (i.e., evaporative cooled or spray-cooled).

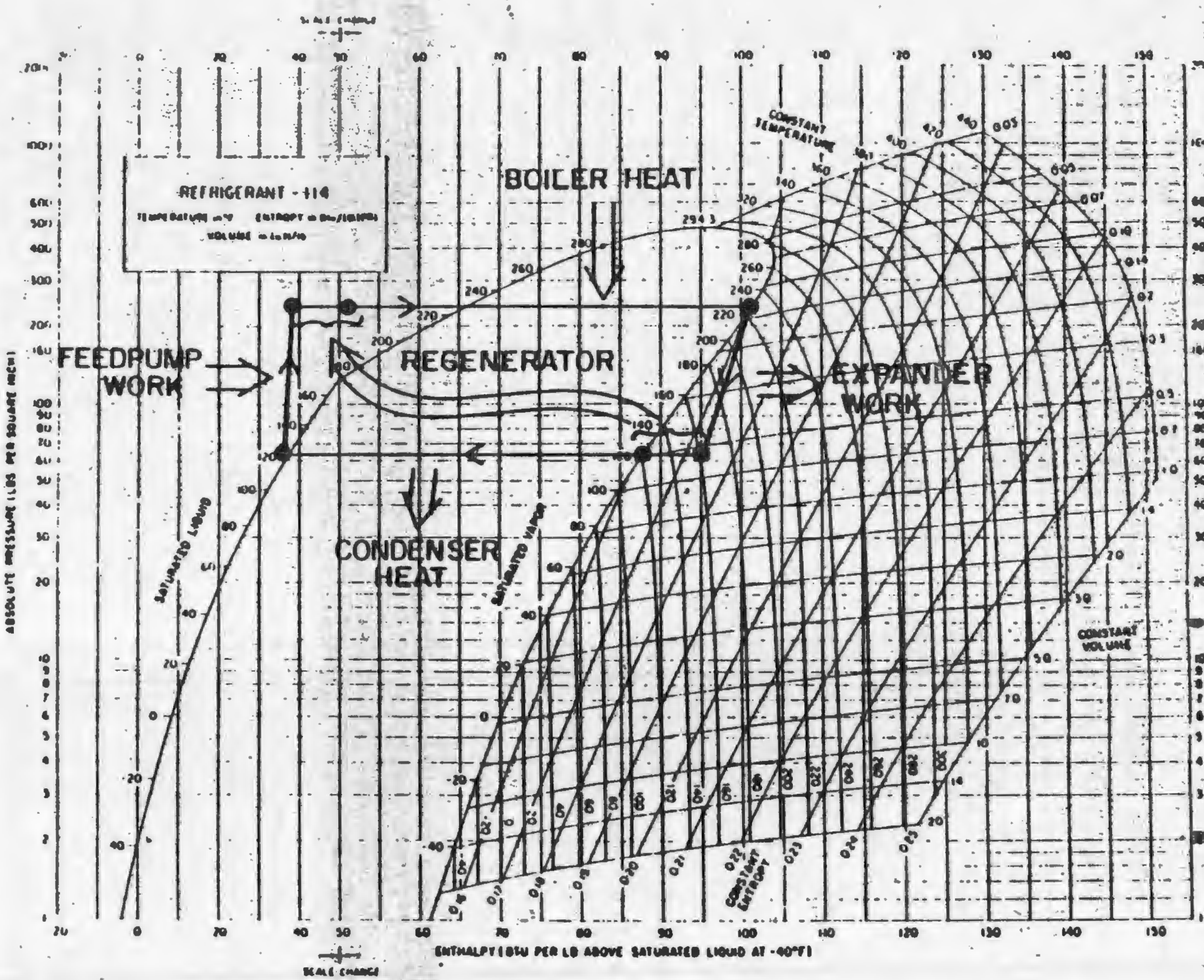


FIGURE II-14 CYCLE DIAGRAM OF RANKINE POWER CYCLE USING R-114

- From the condenser, the liquid working fluid passes into the feedpump, where it is pressurized to the design boiler pressure. The output pressure is controlled by a pressure control valve at the pump outlet.
- The high pressure liquid passes through the liquid side of the generator, where it picks up heat from the superheat expander exhaust.
- The liquid then goes into the "boiler," where it is again vaporized, completing the cycle.

The efficiency of the system depends on both the ideal efficiency of the cycle (assuming 100% component efficiencies and zero pressure drops) under consideration and the actual efficiencies of the major system components, such as the expander and feedpump. The results of testing by a number of firms for a range of component configurations indicate that the following efficiencies can be obtained using existing technology:

- Expander efficiency 70-85%
- Feedpump efficiency 70-93%

The component efficiencies used to calculate the system performance curves shown in Figure II-15 are an expander efficiency of 72% and a feedpump efficiency of 80%. As these figures indicate, system performance tends to improve both as the peak cycle temperature increases and as the condensing temperature decreases.

The temperatures indicated in Figure II-15 are those of the working fluid. The heat source temperature (i.e., collector temperatures) must be higher and the heat sink (cooling water) temperatures lower than those indicated. With proper heat exchanger design, the temperature drops can usually be reduced to the 15-25°F range. Consequently, a working fluid temperature of 200°F would require a collector output temperature of 215-225°F.

The efficiency of the engine is determined primarily by the ideal cycle efficiency for the working fluid and by the expander efficiency. For a solar "fired" system, losses associated with working fluid pressure drops and parasitic loads (controls, etc.) are

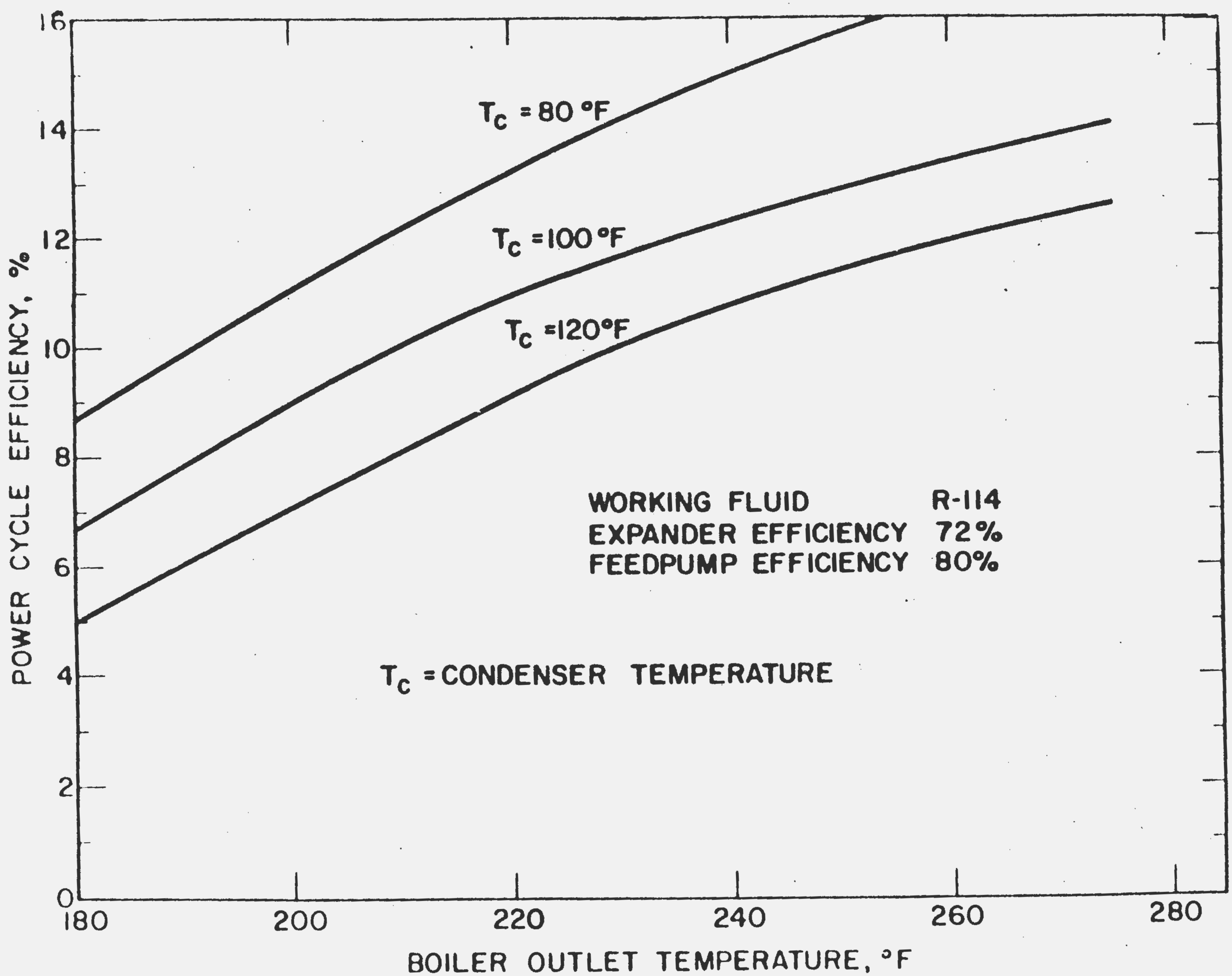


FIGURE II-15 R-114 POWER CYCLE EFFICIENCY - EFFECT OF BOILER AND CONDENSER TEMPERATURES usually small.

The 80°F condenser temperature corresponds to what would be typical of systems using a "once through" water cooled condenser, 100°F to an evaporatively cooled condenser (cooling tower) and 120°F would be consistent with an air cooled condenser.

4. Working Fluid Considerations

A number of working fluids can be considered for use in the engine. The best fluid will depend on a number of factors including:

- peak operating and heat rejection temperatures
- type of expander (turbine or positive displacement)
- power level
- toxicity and/or flammability restrictions

Ideal* cycle efficiencies are indicated for several representative fluids in Figure II-16 assuming a condenser temperature of 90°F. As indicated, there are relatively modest variations in efficiency between working fluid options and, therefore, working fluid choice is usually determined by considerations such as the need for a regenerator, line size requirements, compatibility with expander type, and safety.

To date, R-11 and R-113 have been the fluids most commonly used in those applications using turbine equipment. Lower volume flow rate fluids such as R-22 and R-114 might be better choices if positive displacement equipment is used.

5. Component Configurations

Most of the components comprising the organic Rankine cycle engine would be standard commercial items. For example, the "boiler" and "condenser" could be tube-in-tube counterflow heat exchangers commonly used in the air conditioning industry. The "feedpump" could be hydraulic fluid pump modified to account for the low viscosity of refrigeration fluids.

The key component within the system of a unique nature is the expander used to convert high pressure vapor into mechanical work during its expansion from boiler to condenser pressure. This component

*Assuming 100% efficient components and zero pressure drops.

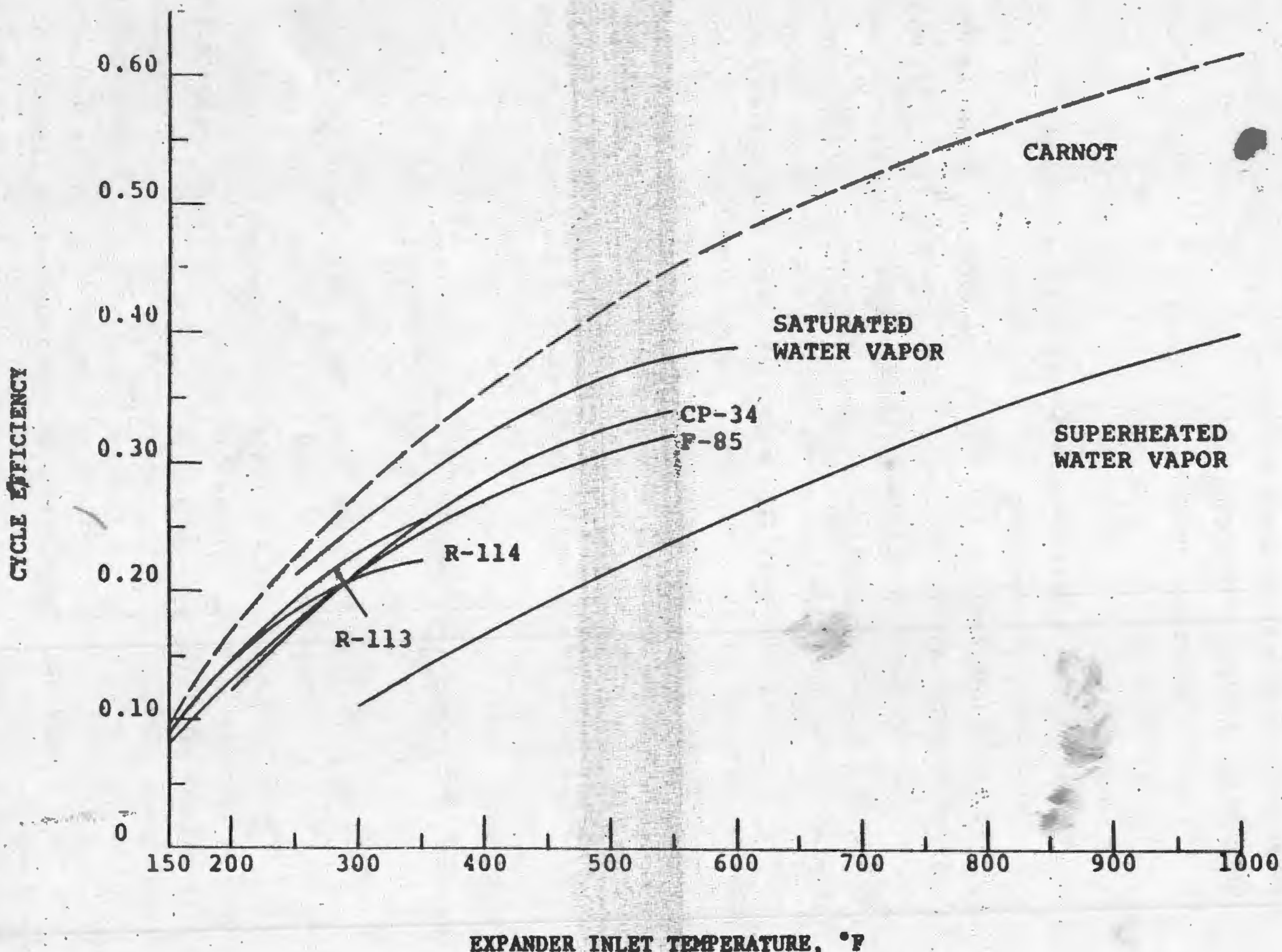


FIGURE II-16 IDEAL RANKINE CYCLE EFFICIENCY VERSUS EXPANDER INLET TEMPERATURE WITH WORKING FLUIDS CONDENSING AT 90°F (REFERENCE 23) determines, to a great extent, the efficiency of the engine. It also strongly influences its operational speed, reliability, and size. There are a number of expander types that can be used in organic Rankine cycle engines including:

- reciprocating equipment
- turbines
- scrolls*
- other positive displacement rotary devices such as rotating vane expanders

For very large power output systems (probably in excess of 250 Hp) turbines are probably the most appropriate expander type. For smaller units, however, positive displacement expanders have many advantages, including low operating speeds and high efficiency. Positive displacement expander configurations include reciprocating equipment as well as rotary configurations such as rotary vanes, screws, and the Scroll.*

6. Cooling System Performance

The organic Rankine cycle engine will directly drive a standard vapor compression air conditioning loop. The performance of this loop is indicated by the cooling coefficient of performance defined as:

$$COP_c = \frac{\text{Cooling Effect}}{\text{Power Input to the Compressor Shaft}}$$

This COP_c is shown in Figure II-17 for a number of working fluids at an evaporator temperature of 45°F and a compressor efficiency of 72%. The refrigeration working fluid most commonly in use today for air conditioners is R-22. Higher COP values are seen to be associated with the use of fluids such as R-142B. One reason this (or a similar) fluid is not used more widely is that its use results in considerably higher compressor displacements and evaporator coil diameters than does the use of R-22. For electrically driven air conditioners

*Developed by ADL.

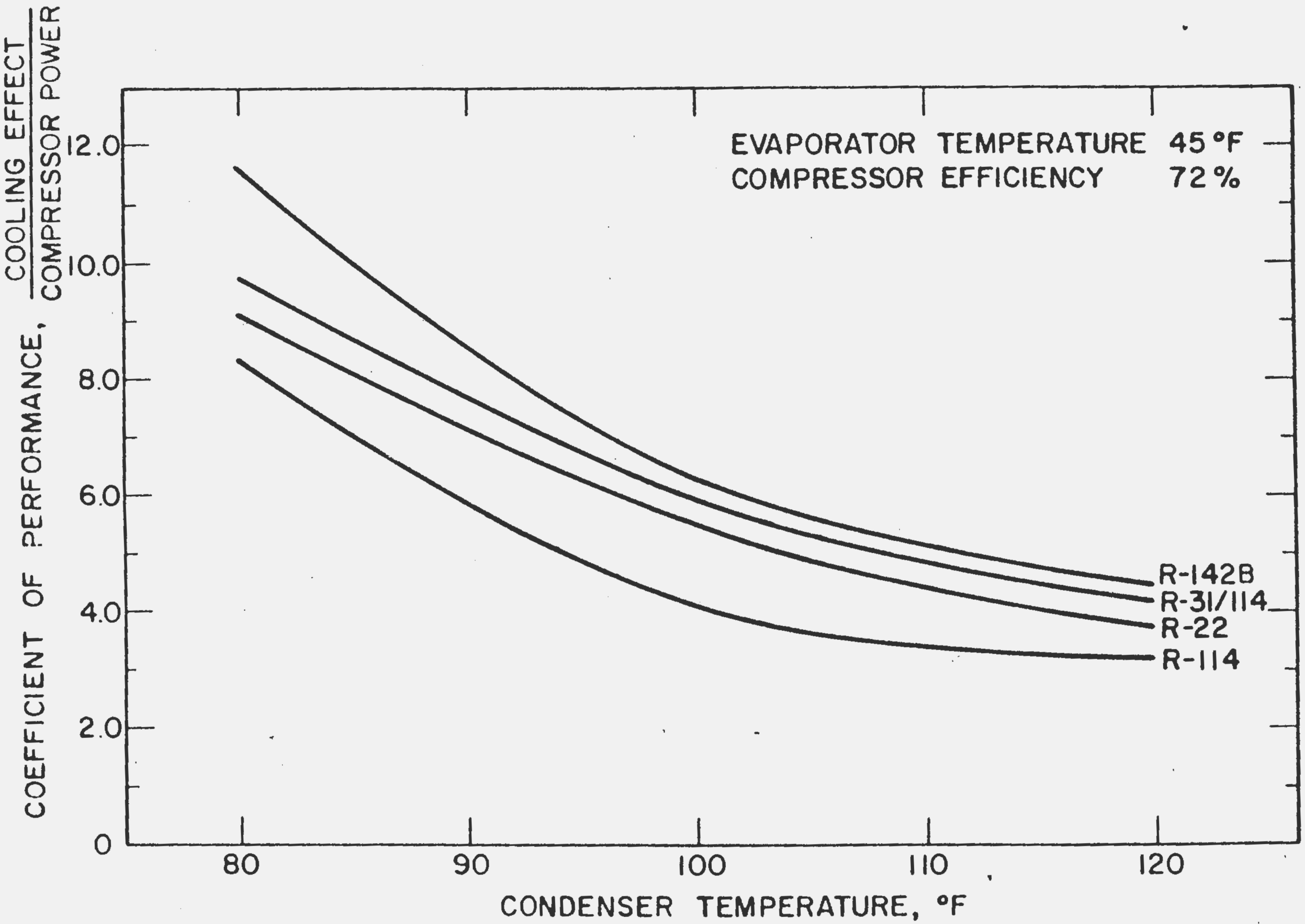


FIGURE II-17 COOLING COEFFICIENT OF PERFORMANCE FOR VARIOUS WORKING FLUIDS - EFFECT OF CONDENSER TEMPERATURE systems, the higher COP levels have not in the past been considered to justify the increased cost of the larger compressor and evaporator coils. As indicated in Figure II-17 COP_c values of about 3-4 are obtainable for a typical condenser temperature of 100-120°F.

7. Combined System Performance

The system efficiency is defined as:

$$COP_s = \eta_e \times COP_c = \frac{\text{Cooling Effect}}{\text{Heat Input to Engine}}$$

where η_e is the engine efficiency. This COP is, therefore, directly comparable to that used to quantify the performance of absorption air conditioning units. Figure II-18 shows the overall system COP as a function of peak cycle temperature for a system with R-114 and R-22 as the working fluids in the power and cooling cycles, respectively. These curves indicate the expected trend of increasing performance with increasing collector temperature as well as the extreme sensitivity of the system performance to the heat rejection temperature.

There are two basic approaches which can be pursued in the design of solar Rankine cooling system configurations:

- Split System:
  The power loop and the cooling loop can be totally independent and thereby use different working fluid. This arrangement requires the use of both engine and compressor shaft seals to prevent leakage of working fluids. A practical advantage of this arrangement is that a conventional motor can be attached to the compressor shaft via a gear or belt drive to operate the system during low solar flux periods.
- Integrated System:
  If the same working fluid is used on both sides of the system, the expander can be connected directly to the compressor with only minimal sealing required

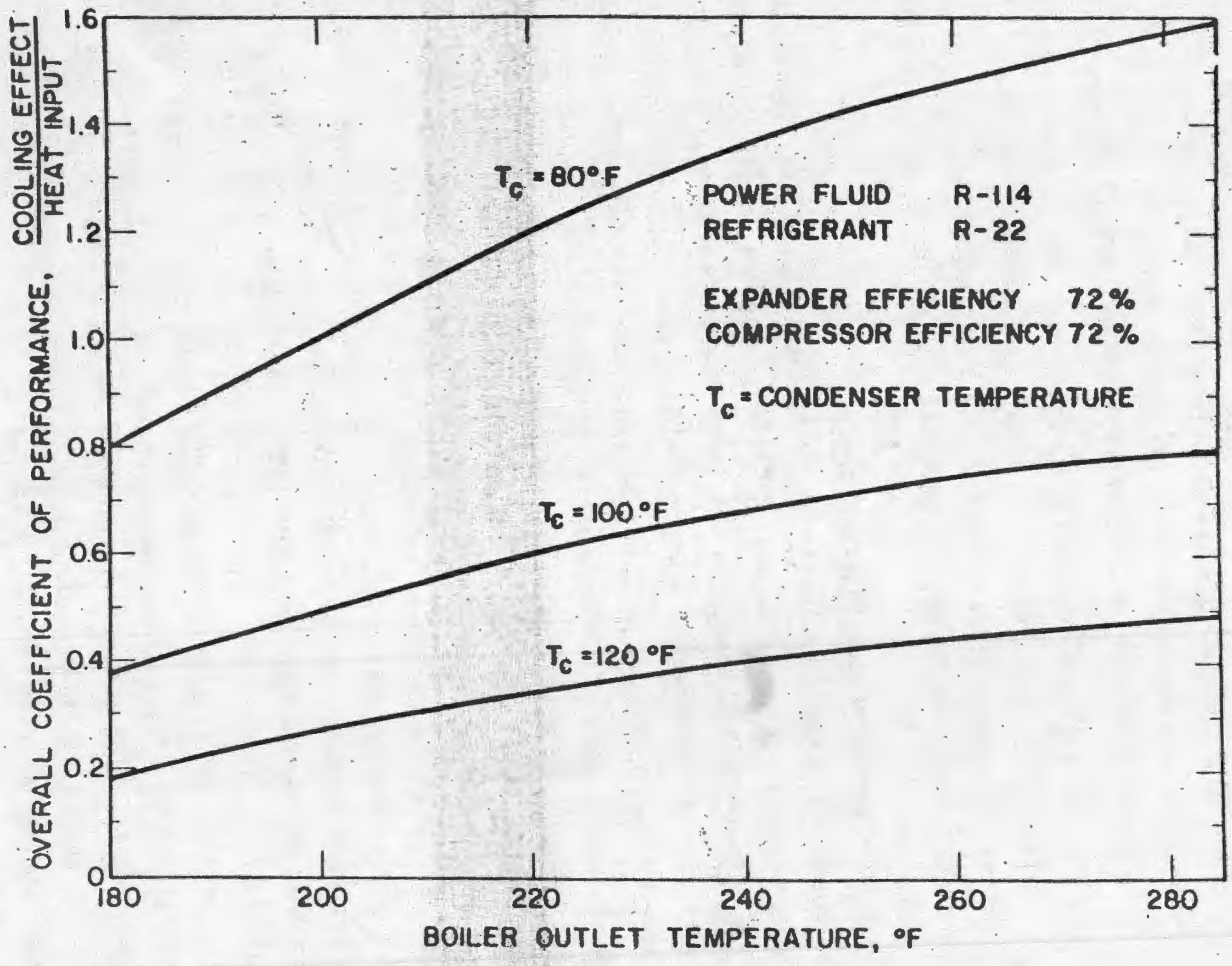


FIGURE II-18 SYSTEM COEFFICIENT OF PERFORMANCE FOR VARIOUS CONDENSER TEMPERATURES between the two sides of the system (to prevent working fluid inventory problems). This facilitates sealing the systems against loss of working fluid.

Which system type is eventually used will depend on system size* and the trade-offs between system efficiency, the ease of connecting an electric parallel drive, cost and reliability. Most test systems fabricated to date have used a split system so that attention could be focused on the engine which represents the new development.

8. Heat Rejection Considerations

The performance estimates shown in Figure II-18 are for systems where negligible power is used to operate the heat rejection systems for either the Rankine cycle engine or the cooling loop. This would approximate the case for a water-cooled system. For many applications, however, water-cooling will not be practical, and air-cooled condensers will be required. With condensers using forced air circulation a fraction of the power output of the Rankine cycle engine may be required to power the fans. The system design becomes a trade-off between the power input to the condenser fan, the condenser size and cost, and the increased collector area resulting from the condenser fan power requirements. Preliminary design studies indicate that acceptable condenser sizes result if the power input to the air circulating fan is between 5 and 10% of the Rankine cycle engine output.

The above considerations combined with the rapidly decreasing system performance with increasing condenser temperature, suggests that solar cooling with a Rankine engine/compressor cooling system will be most practical in situations where cooling water is available (once through or from cooling towers).

9. Status of Heat Engine Driven Solar Air Conditioning

As a practical matter, any firm developing and/or manufacturing

*Open compressors are usually used with systems of greater than 25 tons cooling capacity.

low temperature organic Rankine cycle engines is positioned to develop solar fired air conditioning units.

A number of American firms have made prototype organic Rankine cycle engines for non-solar air conditioning applications which would also be appropriate for use with a solar-heat input. These applications include industrial waste heat recovery, bottoming cycles, and solar irrigation pumping. A list of firms in the field is shown in Table II-4. The effort of several of these firms (United Technologies, Air Research, General Electric) is sponsored by ERDA as part of the solar heating and cooling program.

Several solar powered demonstration organic Rankine engine systems have been assembled in the U.S., primarily as a result of government sponsored programs. These include a 2-1/2 Hp solar driven engine* using R-113 as the working fluid. This engine is directly coupled to the compressor of an air conditioning system.

A 32-kW system was recently delivered to Sandia Labs** for use in their solar "total energy" concept. This system uses toluene as the working fluid at the relatively high temperature of 500-600°F (due to use of focusing collectors).

One firm (Sofretes in France) is now producing a limited number of solar water-pumping systems using a combination of flat-plate solar collectors and a "freon" engine. These systems are primarily for use in remote areas.

Presently no U.S. firms are offering solar driven engines on a strictly commercial basis (although prototype units are available by negotiation) but there are a number of major firms in the U.S. and elsewhere who perceive that solar power organic Rankine cycle engines represent a viable approach to solar on-site power and air conditioning. All indications are that the number of such firms is increasing and that engine units will be available on a commercial basis in the near term (1-3 years).

The firm which has made the most units for solar applications

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*Barber-Nichols Company
**Sunstrand

TABLE II-4 DEVELOPERS AND/OR SUPPLIERS OF ORGANIC RANKINE CYCLE ENGINES

	TYPE MECHANICAL EQUIPMENT	OUTPUT (kW)	WORKING FLUID	APPLICATIONS
Thermo Electro Corporation	Reciprocating and Turbine	3-1000	F-85 & Water	Automotive & Bottoming Cycle
Sunstrand	Turbine	20-200	Toluene	Waste Heat Recovery
Barber-Nichols	Turbine	2-100	R-11, R-113, F-85	Solar Power, Air Conditioning, Waste Heat Recovery
Aerojet General	Turbine	150	Proprietary	Automotive
General Electric	Rotary Vane	3-10	R-11	Solar Air Conditioning
Dupont	Rotary Boiler and Turbines	10-200		Automotive
Authur D. Little, Inc.*	Scroll	1-100		
Sun Power Systems	Gear Expander			
Ormat (Israel)	Turbine	1-5		
IHI (Japan)	Turbine	100-500	R-11	Waste Heat Recovery
United Technologies	Turbine	3	R-114	Solar Air Conditioning
AiResearch	Turbine	10	R-12	
Sofretes (France)	Reciprocating and Turbine	1-50	R-11	Solar Pumping

*Expander development only

is Barber Nichols Company of Denver, Colorado. This firm made a 3-ton cooling system for ERDA and will shortly deliver a 77-ton unit to the Los Alamos Scientific Laboratories (LASL). Design parameters for these systems are:

	3-Ton	77-Ton
Collector Temperature	200°F	200°F
Evaporator Temperature	45°F	45-55°F
Cooling Water Temperature	85°F	65°F
Working Fluid	R-113	R-11
Turbine Type	radial flow	axial
Operating Speed	50,000 RPM	17,000 RPM
Turbine Efficiency	72%	75%
Mechanical Coupling	gearbox	direct drive
Supplemental Energy	parallel electric	steam
COP	0.6*	0.84

The above data would be typical of that from a solar fired organic Rankine cycle engine using turbine equipment. Operating speeds at low power levels are very high, usually necessitating the use of a gearbox when connecting to the compressor system. The availability of relatively cold water at the condenser of the 77-ton unit (cooling tower operating in a low wet bulb environment) greatly enhances its operation.

10. Projected Costs

Costs of the cooling systems are difficult to estimate since commercial units are not now available. However, several firms have made projections as a result of ERDA programs in solar cooling. Preliminary estimates are indicated below for both large scale production (over 10,000 per month) and for low level production (~100-500 per year). These projections assume the sale is to a government agency and do not include dealer mark-up. Costs include

*Projected with an 85% compressor; test installation used compressor with 62% efficiency resulting in a COP ≃0.45.

the power unit, cooling loop, and a cooling tower. The hot water from the collectors is assumed to be provided at a temperature of 200°F. If higher temperature collectors are used, cost will decrease as a result of reduced heat exchanger area requirements.

	3-Ton*	25-Ton*
Large Scale Production	$3,000-$4,000	$15,000-$25,000
Limited Production	$6,000-$8,000	$25,000-$35,000

The cost per unit capacity is seen to decrease as output increases. These economies of scale are quite large and could be an important factor in determining the most attractive solar air conditioning applications.

D. DESICCANT COOLING SYSTEMS

Desiccant dehumidification systems utilize material which have the ability to attract and remove water from an air stream. The desiccant may either be a liquid, such as a glycol compound, or solids, such as lithium chloride or a molecular sieve material. The water removal process may either be by absorption, which involves either a physical or chemical change in the desiccant (as is the case with materials such as glycol or lithium chloride) or by adsorption which involves no physical or chemical change in the desiccant, but generally depends on surface affects.

The basic regenerative dehumidification cycle includes two elements:

- Absorption, or removal of water vapor from the air stream into the desiccant, will generate heat the major portion of which is the heat of condensation of water vapor. Means of heat removal must be provided with either an external coolant, or by accepting the rejected heat as a temperature rise in the process air. The lower the temperature of the absorption process, the lower the attainable air dew-point.

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*Prepared with the asssitance of Barber-Nichols Company, Denver, Colorado.

- Regeneration, or removal of water from the desiccant, must be accomplished by heat addition either in the form of heating coils or as a warm regenerative gas stream. The degree of sorbent drying is proportional to the regeneration temperature.

Desiccant systems can be applied to achieve air-conditioning in the following manners:

- Improve comfort through lowering of the relative humidity.
- Serve as a preconditioning step which lowers latent heat with subsequent refrigeration type sensible cooling. This preconditioning allows a refrigeration type air-conditioner to operate at a lower load and at a higher evaporator temperature, with the potential for higher coefficient of performance.
- Achieve cooling as well as dehumidification by over-drying the air and then rehumidifing to the desired temperature and humidity.
- Accomplish latent, as well as sensible, heat recovery between inlet air and discharge air.

A number of system considerations distinguish the desiccent approach from other solar cooling approaches. Desiccant devices generally require larger air handling equipment and operate at working pressures near atmospheric. Electric power is generally needed only for air circulation with these devices. The only form of storage feasible with solar desiccants is the storage of solar collected heat in a hot storage unit for subsequent activation of the desiccant unit.

1. <u>Liquid Absorbent Cycles</u>

Liquid absorbent desiccants have a high capacity for water removal and are easily circulated and brought into intimate contact with the process and regeneration streams. Liquid absorbents, such as triethylene glycol, have been used in commercial systems for dehumidification of inlet ventilation air, and other process uses. These systems tend to function most effectively in the high humidity region. In order to achieve dewpoints low enough for comfort cooling, the absorber chamber must be cooled with cooling tower water.

Early work on a solar desiccant system using a tri-ethylene glycol absorbent was reported by Löf (Reference 27). Figure II-19 illustrates the concept. The major elements are an absorption chamber in which the room air is dehumidified and a stripping chamber in which the glycol is regenerated by air heated in a solar collector. The process proceeds as follows. Household air is circulated through a dehumidifying chamber into which a cool concentrated solution of tri-ethylene glycol is sprayed. Moisture is removed from the air by the liquid absorbent and the dry air is then either recirculated directly to the building or a portion of it is cooled by partially rehumidifying in an evaporative cooler. The moisture received by the liquid absorbent is then removed by spraying the liquid into a second chamber (stripping column) through which solar heated air is passing. Water is evaporated from the liquid absorbent and vented into the atmosphere in the solar air stream. The reconcentrated liquid absorbent is then returned to the absorber-spray chamber. Heat exchangers are employed and the heat of condensation of moisture is removed by use of cooling water.

The dehumidification capacity of the absorber-striper unit depends on numerous factors such as the volume of the two spray chambers, spray characteristics, glycol concentration, temperatures, air rates and temperatures in the two chambers, and heat exchanger areas (Reference 27). In a unit of a certain size and design, the most important single variable affecting the dehumidification capacity is the temperature of the air supplied to the stripping column from the solar collector. The regeneration temperatures required for liquid systems are moderately low. For non-solar operation, a warm water temperature of 175°F is generally considered adequate. Löf found his system could operate at or somewhat below 175°F. The coefficient of performance of the liquid absorbent unit is about 0.5.

2. Solid Desiccant Cycles

Solid desiccants generally have a lower moisture capacity than liquids (particularly at high humidity conditions), but may have the

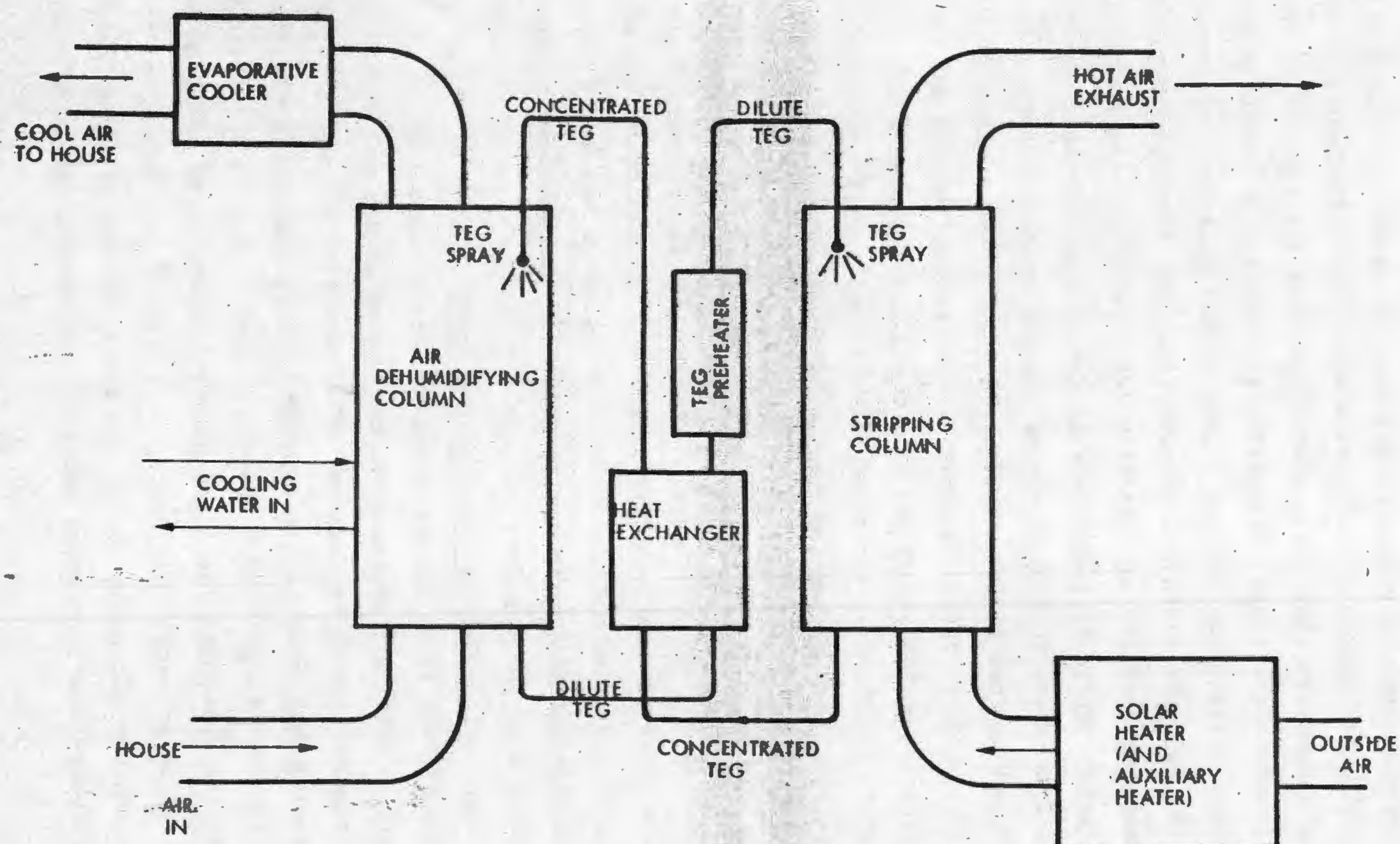


FIGURE II-19 DESICCANT-CYCLE COOLING SYSTEM USING TRIETHYLENE GLYCOL DESICCANT (REFERENCE 4) capability of deeper drying. Two common mechanical forms of regenerative solid desiccant equipment are the rotary wheel and the dual bed arrangement. Rotary equipment usually has a greater capacity per unit equipment volume and generally has been favored for space conditioning applications. However, solar systems may have greatly different design parameters than those associated with existing equipment. For instance, there are indications that silica gel systems designed for solar cooling may achieve higher useful capacity and longer bed life with the fixed bed geometry than the rotary configuration.

Desiccant systems are normally designed to fit the particular application. There is relatively little design information available in the literature to enable one to readily establish parameters of a desiccant system to fit a new application. The feeling in the industry is that systems designed for solar application may be significantly different from those currently existing.

Figure II-20 is a conceptual schematic of a solid desiccant system. In this idealized system the warm moist air is first dried isothermally by flow through a desiccant bed--the heat of absorption being removed to keep the process isothermal. Next, the warm dry air undergoes an adiabatic partial rehumidification process which reduces air temperature at the expense of some humidity increase. The desiccant regeneration process is indicated in Figure II-20 by dotted lines where solar heat is added to the desiccant to drive off the water which had previously been removed from the air.

a. Silica Gel Bed Desiccant Systems

Figure II-21 is an illustration of a particular form of solid desiccant cooling system based on the use of silica gel. The system process occurs as follows. The room air is drawn into the air-conditioning system where a regenerative heat exchanger heats the air to approximately 100°F. The air, which is wet to a 50°F dewpoint, contacts the absorbing desiccant material where water is drawn out of the air in an exothermic process. The heat is transferred to the outside and the air comes out of the absorbing bed at about 110°F. A regenerative heat exchanger cools the air down to about 85°F by

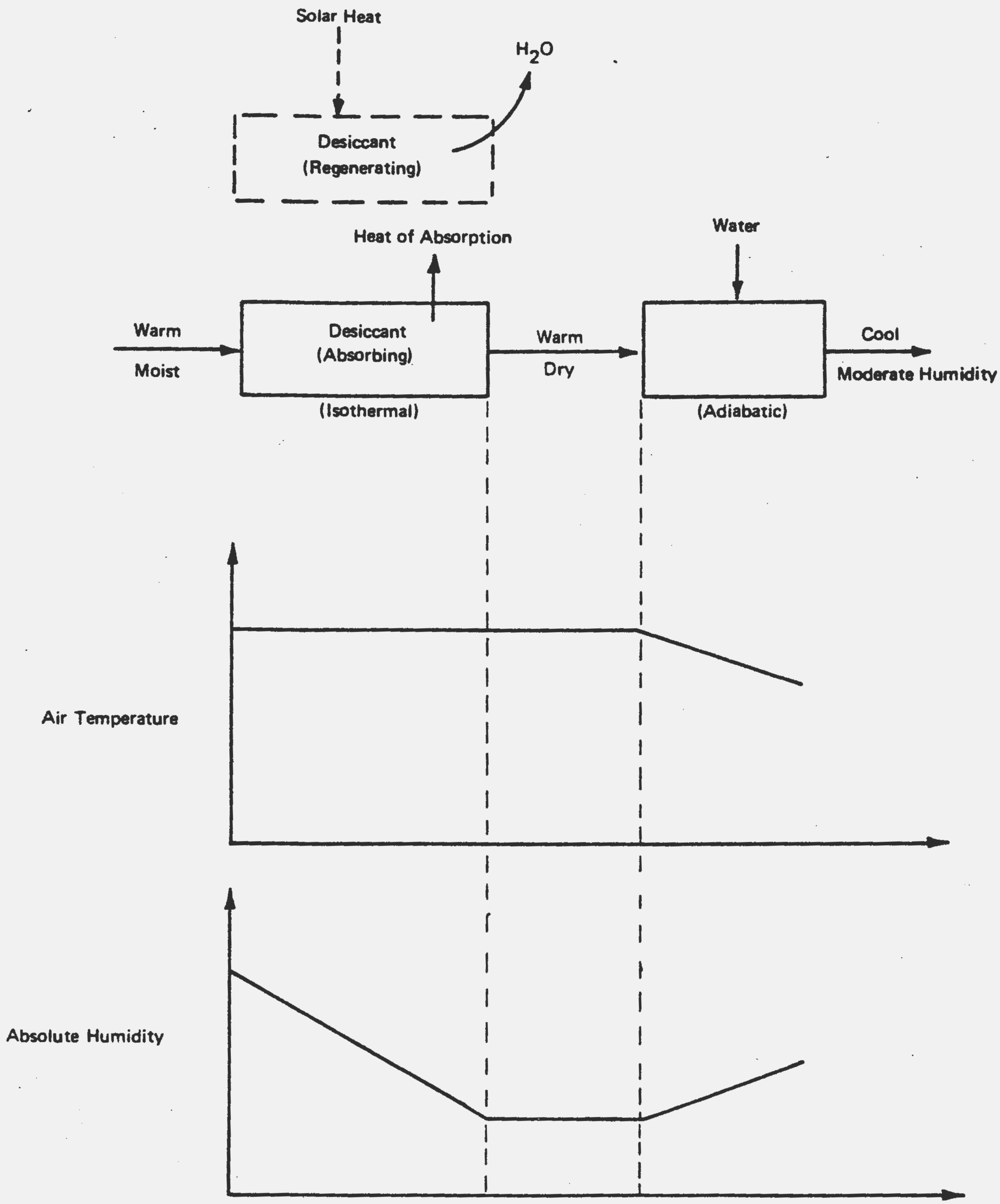


FIGURE II-20 **CONCEPTUAL SCHEMATIC OF DESICCANT SYSTEM**

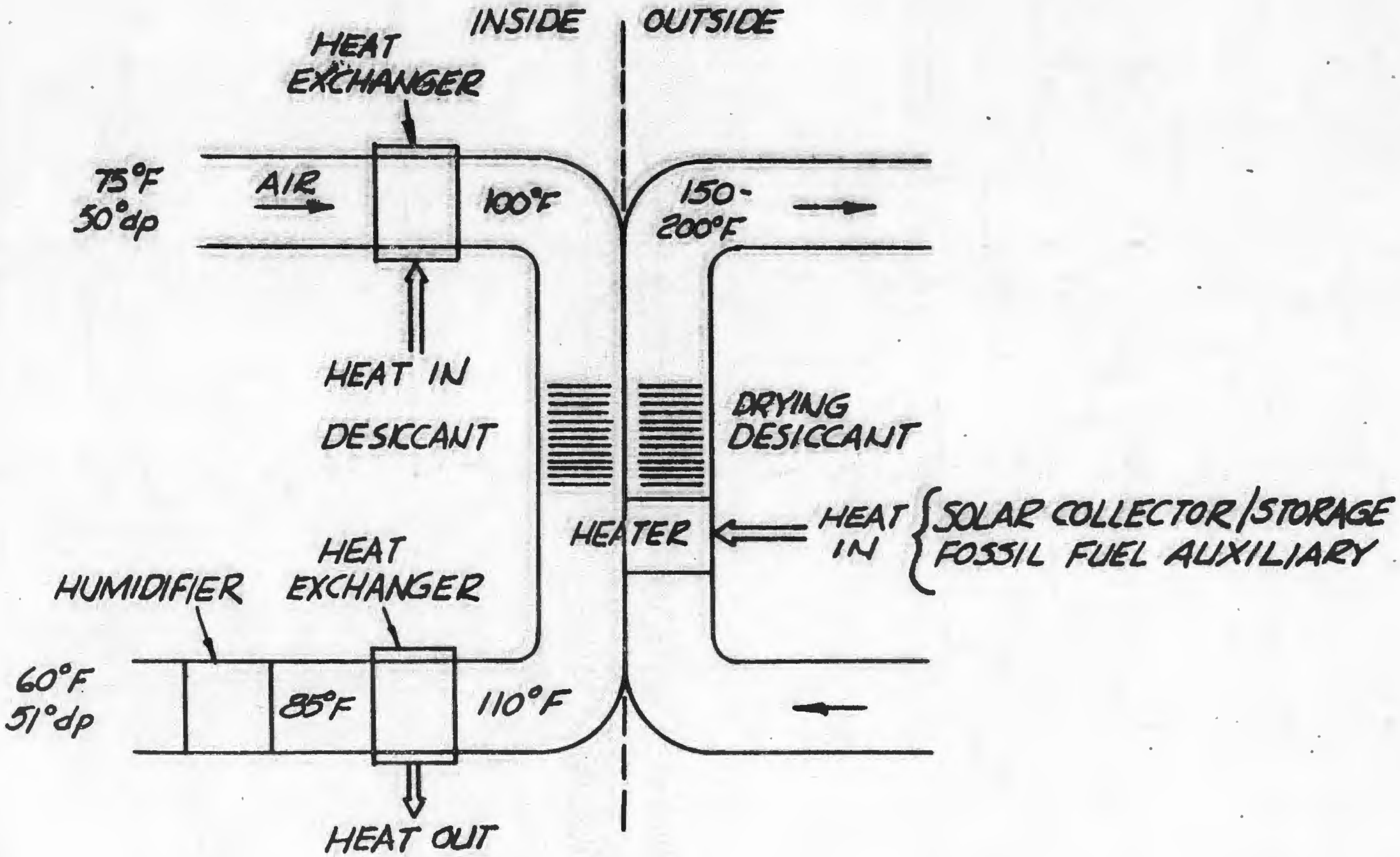
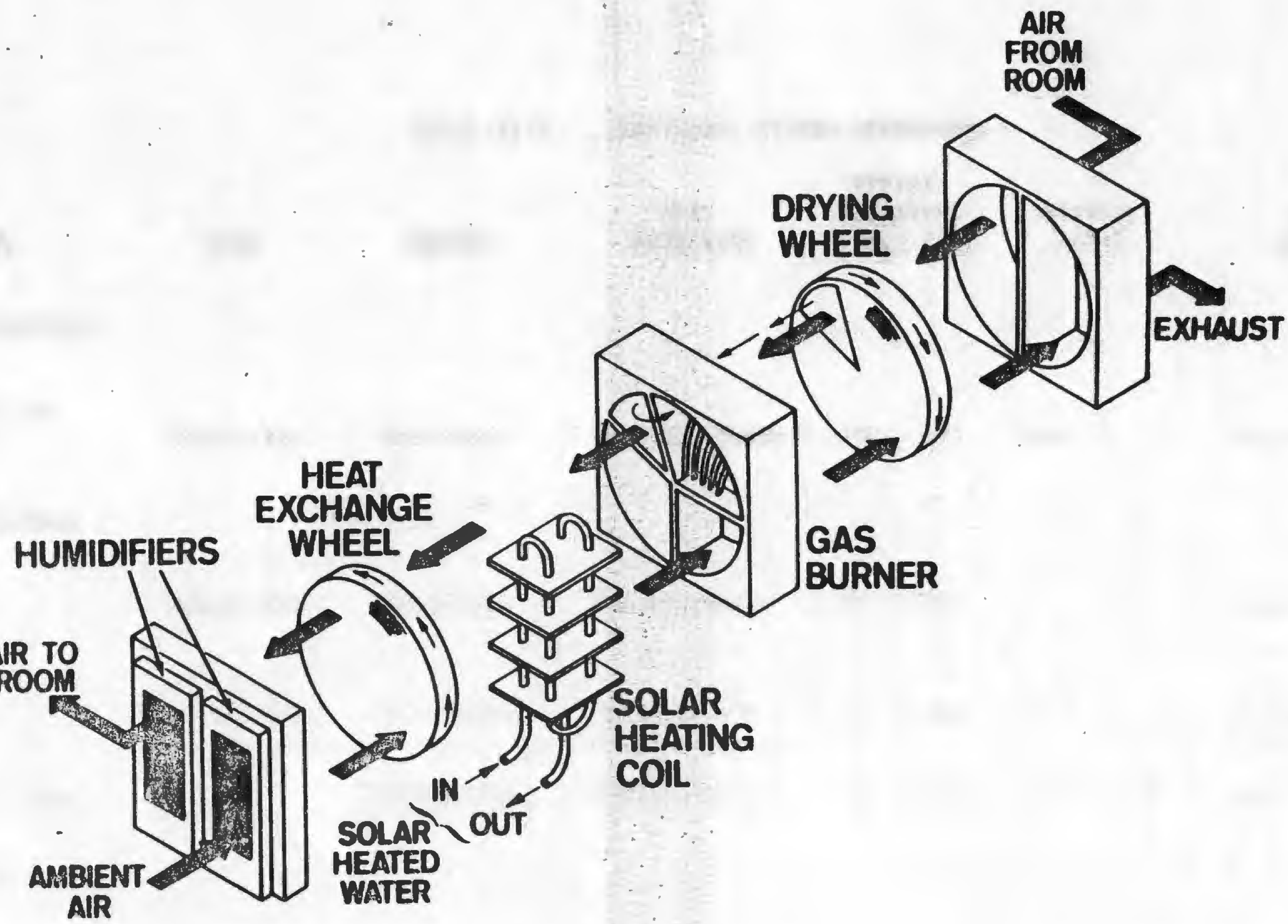


FIGURE II-21 SILICA GEL BED DESICCANT COOLING SYSTEM (REFERENCE 28) transferring some of the heat back to the incoming air. Partial rehumidification lowers the air temperature to 50°F. The system uses a fossil fuel heat source and a collector to dry the silica gel desiccant. Potential cooling COP's of 0.7 to 0.8 have been calculated for this system (Reference 28). The units shown can be operated as a dual bed by appropriate ducting with one desiccant bed being regenerated while the other is used for dehumidification. This type of desiccant cooling system is currently under development by the Center for the Environment and Man (Reference 28). It is anticipated that development activity could lead to a prototype system in three to four years (Reference 2).

b. Rotary Wheel Desiccant

An approach being developed by the Institute of Gas Technology uses a desiccant wheel to provide continuous desiccant cooling (References 29-31). The system is called the Munters Environmental Control (MEC) Unit.

In the cooling mode (Figure II-22), MEC recirculates room air and outside air--each confined to a different side of the unit. One side of the MEC processes room air, which is, in turn, dried, sensibly cooled, and evaporately cooled, to reach the desired end state for room comfort. The other side of the unit processes outside ambient air, which is, in effect, the sink for the dehumification and cooling processes performed on the room air. The room air enters the MEC through a warm molecular-sieve drying wheel where it is dehumidified and heated; it next passes through a heat exchange wheel, where it is sensible cooled, and finally through a humidifying chamber, where it is further cooled and partially dehumidified, resulting in an end state similar to that achieved in a conventional electric air conditioner. On the other side of the unit, outside ambient air enters through a humidifying chamber, where it is cooled by adiabatic saturation; it next flows through the heat exchange wheel, where it absorbs the heat which had been removed from the room air; then it is heated (by a heat source, which may include solar energy); finally it passes through and regenerates the drying wheel before being exhausted outside.



Source: Institute of Gas Technology, Press Release, May 20, 1974.

FIGURE II-22 SCHEMATIC OF SOLAR-MEC SYSTEM WITH HEATING/AIR-CONDITIONING UNIT EXPLODED TO SHOW MAJOR COMPONENTS

With gas firing of the cooling unit, the MEC has produced about three tons of air-conditioning with a COP of about 0.7. Operating at 190°F from solar collectors the solar MEC has a measured COP of 0.5. If a 230°F fluid were available, it has been estimated that fossil fuel requirement could be reduced by 80% (reference 29).

Prototype 3 ton solar desiccant units have been built for actual field testing which was initiated in the Fall of 1975. At present, there is not a commercial base for this type of cooling system. However, discussions with IGT indicates that the solar driven MEC may achieve commercial status within ten years (Reference 45).

3. Summary of State of Art

Table II-5 summarizes the various approaches to solar desiccant cooling. The table gives the status of the systems and operating characteristics, the inlet temperature range and coefficient of performance. While the status of several of the approaches is listed "available," there are at present no commercially available solar type desiccant systems. Extensive commercial based technology does, however, exist for the liquid and rotary wheel desiccant approaches. The desiccant systems have operating temperatures and coefficients of performance similar to those for the continuous single effect heat actuated absorption units.

Table II-6 summarizes the major organizations involved in the development and potential commercialization of various desiccant approaches. The table lists demonstrations of this technology and conventional applications. It is not expected that any solar activated desiccant systems will be commercially available before five to ten years from present.

TABLE II-5 DESICCANT SYSTEM APPROACHES

TYPE	SIZE	STATUS	HEAT REJECTION	TYPICAL OPERATING TEMPS (°F)	TYPICAL COP'S	PRINCIPAL ORGANIZATION
Liquid Absorbent						
Triethylene Glycol	Commercial	Available	Cooling Tower	150 - 175	0.5	Niagara Blower
Solid Absorbent						
Rotary Sieve	Commercial	Available	Regenerative	175 - 250		Cargocaire
	Residential	Development	Regenerative	150 - 250	0.7	IGT (Munters Cycle)
Bed Silica Gel		Development	Regenerative	150 - 250	0.7 - 0.8[a]	CEM

[a]Calculated

TABLE II-6 DEVELOPMENT OF SOLAR DESICCANT SYSTEMS

TYPE	ORGANIZATION	YEARS TO COMMERCIALIZATION[b]	TESTS	COMMERCIAL APPLICATIONS
Liquid-TEG[a]	Niagara Blower	-	Proposed Citi-Corp Building	Large Sized Industrial Drying
Rotary Sieve	IGT	5 - 10	3 Demonstration Houses (3 ton)	-
	Cargociare	-	Proposed Project to NASA	Industrial Dehumidification
Bed Gel	CEM	5 - 10	Laboratory	-

[a] Residential sized experiment carried out in 1952 (Löf)
[b] Stated by organization representative

III. REVIEW OF SOLAR COOLING DEMONSTRATIONS

Demonstations of solar cooling technology have been carried out in the past few years over a range of system capacities from two tons (residential sized units) to close to 100 tons (commercial sized units). The demonstrations provide a test of the practical state of the art of the various solar cooling approaches, and provide useful information relating to the design of solar cooling systems.

Table III-1 provides a summary of the major demonstrations of solar cooling as of mid-1976. The table categorizes the demonstrations by location and type of building, status of experiment, cooling system, collector size and type, storage size and type, operating data and provides comments relating to performance or design conditions. Data on the solar collector and storage subsystems are given because the cooling performance is ultimately related to the design of the entire system rather than to the cooling device alone.

The data summarized in Table III-1 were obtained primarily through discussions with the principal investigators. In many instances, this information was supplemented with reports (see references 32-44).

A. DISCUSSION OF RESULTS

1. Continuous Absorption Solar Cooling

Nearly all of the demonstrations employed commercially available single effect lithium bromide absorption machines, with the preponderance of these being the small tonnage residential size units. For the most part, no modifications were required to the absorption units other than the replacement of the gas burner/generator with a hot water heat exchanger in the three ton Arkla Industries units. The larger size absorption units (greater than 25 tons) normally contain mechanical solution pumps. The units employed in the demonstration were essentially "off-the-shelf" units which were operated under derated conditions (water input temperatures near 200°F as compared to a nominal 240°F input temperature for operation at rated capacity).

	Location	Type of Bldg.	Status	Type	Si
1a.	-	Honeywell Mobil Trailer	Completed	Rankine Turbine Expander	3– 21
1b.	-	Honeywell Mobil Trailer	Completed	Absorption	3–
2a.	Los Alamos, NM	3-story office building ($66,000ft^2$)	In construction	Rankine (turbo-compressor)	77
2b.	Los Alamos, NM	3-story office building ($66,000ft^2$)	In construction	Absorption	77
3a.	Yazaki Corp., Japan	House ($1,370ft^2$)	Operational	Absorption	2-
3b.	Yazaki Corp., Japan	Public assembly hall ($7,000ft^2$)	Operational	Absorption	3
4.	New York City Corp. Building	Office building	Abandoned	Desiccant	5(
5.	Tucson, AZ "Decade 80" Solar House	Single-family ($3,200ft^2$)	Operational	Absorption	2
6.	Ohio State Univ.	Single-family ($2,200ft^2$)	"Operational"	Absorption	3
7.	Timonium Elementary School, Georgia	School ($60,000ft^2$)	Operational	Absorption	5(
8.	Townes Elementary School, Georgia	School ($32,000ft^2$)	Operational	Absorption	6((c 1(
9.	NBS	Model townhouse ($1,200ft^2$)	Completed	Absorption	3
10.	University of Florida	House ($750ft^2$)	Completed	Intermittant Absorption	2 cl
11.	"Little Red School" Bell & Gossett, IL	B & G Training facility	Completed	Absorption	3
12.	Colorado State Univ.	CSU-I House ($3,000ft^2$)	Completed	Absorption	3
13.	NASA MSFC	3 house-trailers	Operational	Absorption	3

A

TABLE III-1 Summary Sheet--Demonstration of Solar Cooling

Cooling System			Collector		Storage		P
ze	Fluids	Manufacturer	Type	Size	Type	Size	COP's
ton @ 5°F	R113/R12	Barber-Nichols	2-pane, selective	571 ft^2	Hot, Baffled	950 gals	0.5 @ 0.25 @
ton	Li Br	Arkla	2-pane, Selective	571 ft^2	Hot, Baffled	950 gals	0.71
-ton	R11\R11	Barber-Nichols	1-pane, selective	8,000ft^2	Hot/Cold	5,000/ 10,000 gals	0.8 (5 chille water)
-ton	Li Br	York	1-pane, selective	8,000ft^2	Hot/Cold	5,000/ 10,000 gals	0.8 (4 chille water
ton	Li Br	Yazaki	2-pane, selective	1,000ft^2	Hot	900 gals	0.4-0.
@ 7.5-ton 2.5 tons	Li Br	Yazaki	plastic pane, selective	4,400ft^2, reflectors	Hot	20 tons	
0-ton	Triethylene glycol		2-pane, black	20,000ft^2	Hot	25,000 gals	0.5
@ 3-ton	Li Br	Arkla	2-pane, black	1,800ft^2	Hot	2,500 gals	0.8
·ton	Li Br	Arkla	2-pane, black	800ft^2	Hot	4,000 gals	
)-ton	Li Br	York	2-pane, black	5,000ft^2, reflectors	Hot	15,000 gals	0.25
)-80-tons lerated from)0 tons)	Li Br	Arkla	2-pane, selective (Alcoa)	10,000ft^2, reflectors	Hot/Cold	30,000/ 15,000 gals	0.6 c water
-ton	Li Br	Arkla	2-pane, black	550ft^2	Hot	500 gals	
.5-ton water iller (55°F)	NH_3/H_2O	Univ. of Florida	1-pane, black	500ft^2	Hot	3,000 gals	0.11 retica
-ton	Li Br	Arkla	2-pane, black	540ft^2	Hot	800 gals	0.65
-ton	Li Br	Arkla	2-pane, black	760ft^2	Hot	1,100 gals	0.7
-ton	Li Br	Arkla	1-pane, selective	1,218ft^2	Hot	3,500 gals	0.59-(

Dra

rformance Data	Operating Temp.	Comments		Si
215 170	170°F-200°F	Rush development of prototype, feed-pump operated by motor (750W), loss in gear box = 0.5hp	Turbine	3- 21
	196°F-210°F		on	3-
)°F 1	185°F up	Very low latent load. High evap. temp. and low condensor temp. Machine derated from nominal 100 ton.	(turbo- or)	77
5°F 4	185°F up	Same comments as above.	on	77
6	176°F-210°F		on	2-
	167°F		on	3 =:
	130°F-150°F	Intended to be used with conventional chiller to eliminate need for reheat & to allow chiller to operate at higher evap. temp. Need disappeared when alternate design developed for conv. HVAC system.	t	5(
	185°F-210°F	Optimum temp. of water to absorption device a function of cooling water temp.	on	[illegible]
	170°F-200°F	No cooling achieved; tanks too large w/ large heat leaks. Direct firing of Arkla attempted w/o result.	on	3-
	180°F	150 ton unit when powered by steam. Chiller oversized for collector. System oversized for peak loads (only 10,000ft² cooled).	on	5(
illed @ 48°F	195°F-200°F 185°F min	No cooling data available.	on	6((c 1(
	190°F	Needed to set generator temp. at absorber temp. +110°F to avoid crystallization because of bubble pump (60% capacity minimum). For collectors (moisture problem) & large storage losses.	on	3-
(theo- l=0.56)	140°F-180°F	Thesis experiment. Collector never above 145°F, leading to low COP's.	tant on	2 cl
	210°F	Little cooling accomplished; collector too small; storage too large (collector losses too high).	on	3-
	175°F min to prevent crystallization	Cold side storage found desirable to reduce cycling losses. Mechanical solution pump added to absorption unit to prevent crystallization.	on	3-
.70	190°F-220°F	Variable flow rate at high temp. to eliminate operation at too high a generation temp. COP higher at lower generation temp; short circuiting problem in storage reservoir.	on	3-

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All of the units had minimum operating temperatures near 180°F-190°F. In the smaller machines the minimum temperature was necessarily to avoid crystallization in the machine (due to lack of sufficient vapor lift in the bubble pump at the low temperatures). Where mechanical solution pumps were employed, the minimum generator temperatures were limited by the reduction in cooling capacity with decreasing input temperature. Maximum temperature limits were also established (for the smaller tonnage units) to: prevent crystallization in the recuperator (between the generator and absorber) due to overconcentration of the absorbent in the generator (the larger units contain dilution chambers to limit concentration of the hot absorbent), or to prevent an excessive boiling rate of the refrigerant, some of which spills back into the regenerative heat exchanger to be reheated in the generator (resulting in a net lowering of the effective cooling COP). These temperature limitations lead to certain constraints on the design of the remainder of the solar system.

The results of these demonstrations were mixed, with high COP's and many hours of cooling achieved in some instances, and with no cooling achieved in other instances. The major determinant of the success of the demonstration was the design of the supporting solar collection and storage subsystems. In all instances where the results were reported in detail, significant findings were obtained relating to overall solar system design and control of the units. These findings are discussed below.

a. Solar Collector Subsystem

It is essential that the solar collector either be capable of efficient collection at high temperatures or be oversized in proportion to the capacity of the cooling device. A selective surface absorber is required to obtain moderate-to-high collection efficiency at temperatures near 200°F.

The attempts to generate significant quantities of solar cooling were generally successful when a selective surface collector was employed. The results obtained with the use of black absorber surface collectors were mixed. The relatively poor success of the NBS experiment

(References 39 and 40) was largely attributable to poor collector performance. Undersized collectors using a black absorber surface resulted in poor cooling performance in the Bell and Gossett experiment (Reference 41).

b. Thermal Storage

A number of very important findings emerged from the demonstrations concerning thermal storage. These findings relate to: size of storage relative to the collector size and cooling capacity, energy losses from storage, degrees of stratification achieved in storage, and chilled water storage.

Choice of too large a solar thermal storage unit (for storage of hot liquid) with moderate heat losses may result in poor cooling performance because of the inability of the solar collector to quickly raise the storage temperature to the level needed to drive the absorption units. This effect was experienced in the Ohio State University experiment (Reference 36) where the storage temperature never exceeded 160°F. (Firing of the unit by taking water directly from the collectors was also attempted without success because of fluctuations in insolation.) A large storage reservoir was also used in the NASA experiment (Reference 44) but with reasonable success because of the employment of a selective surface type solar collector which was also oversized in proportion to the cooling system capacity.

Significant heat leaks from thermal storage were reported for the NBS, Colorado State University (CSU) and Bell and Gossett experiments, with significant reductions in system cooling performance. If the thermal storage unit is placed within the conditioned space, such heat leaks not only reduce the effective cooling capacity of the solar systems, but also can contribute directly to the cooling load (Reference 43).

Stratification is desirable in the thermal storage tank to ensure delivery of the maximum temperature fluid to the absorption machine. Baffling arrangements were used in both the Honeywell (Reference 32) and NASA experiments to promote thermal stratification in the storage units. Short circuiting of the return water from the generator to the input side of the generator reduced the effective cooling capacity in the NASA experiment. The degree of short circuiting was found to be directly related to the flow rate of input hot water to the generator of the absorption unit.

It was found in the CSU experiment that a substantial decrease in COP could be experienced during periods of relatively low cooling demand. Under these conditions, the absorption machine would tend to cool down between cooling cycles, resulting in thermal losses associated with warming the machine to its minimum firing temperature before cooling could be obtained. Employment of a cold storage reservoir was suggested as a means of decoupling the solar fired absorption unit from the building cooling load, thereby reducing the cycling of the absorption unit (Reference 42). The experimenters have adopted the use of a cold storage reservoir in the design of a subsequent cooling demonstration system (Reference 43).*

c. Controls

Control of the fluid stream to the cooling unit has emerged as another area requiring careful design. In the NASA experiment, considerable effort was expended to maintain the temperature of the inlet fluid to the generator above 188°F and to reduce the rate of heat input at high temperatures. (Because of the use of a large and efficient solar collector subsystem, excess heating input capacity was frequently available.) The control scheme finally adopted resulted in a reduction by steps of the flow rate to the absorption unit with increasing input temperatures above 195°F. This scheme was adopted to minimize the net heat input to the absorption unit.

Other control functions found desirable to increase the COP of the absorption unit were: (1) a short delay (60 seconds) in start-up to reduce transients in machine performance; (2) full heat input to the absorption unit at a room temperature 3°F higher than the thermostat setting (required to increase the cooling capacity to bring the room temperature to the desired range)**; and (3) shutdown of the cooling unit after prolonged (more than four minutes) operation at an evaporator temperature above 60°F.***

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*Two separate cold storage units are planned to achieve virtually complete thermal stratification.

**Operation at less than full heat input increased the COP.

***Low cooling rates and negligible dehumidification are achieved at 60°F evaporator temperature.

2. Rankine Cycle Solar Cooling

To date, one operating Rankine cycle cooling system has been demonstrated--a three ton turbine expander unit, demonstrated in the Honeywell Mobil Trailer (References 24 and 32). Other demonstrations of this technology are planned. Tests of Rankine cooling using a rotary vane expander have been reported, carried out by General Electric Company, but little information is available on the results.

The demonstration unit was developed and delivered in a relatively short time period (nine months). Because of the time limitation, available hardware was used to the extent possible. However, several major problem areas arose in the design, specifically: the feed-pump power loss was substantially higher than the design value and required external drive by an electric motor; and the gear box loss was approximately 25% of the turbine shaft power output. Other losses were associated with moderate pressure drops in the tubing and motor/generator inefficiencies. The measured system COP (cooling effect/heat input to engine) at 215°F inlet temperature was about 0.5 and at 170°F was about 0.25, excluding the electrical energy expanded to drive the feed-pump.

The demonstration showed that Rankine cycle solar cooling is technically feasible. However, the demonstration was not totally realistic because of the tight time constraints placed on the development of the machine. With more time and development effort, the Rankine system has the potential of approaching or exceeding the technical performance of the absorption unit. The key development areas highlighted by the demonstration are:

- Efficient expanders for low power application;
- Low cost, reliable feed-pumps capable of efficient operation with low viscosity fluids, and
- Control schemes and fluid flow regulation for speed control.

3. Other Approaches

a. Intermittant Cycle Absorption Cooling

Experimental studies with a water-ammonia intermittant cycle absorption unit have been reported by the University of Florida (Reference 18).

There is some question whether the experiment should be classified a "demonstration" in the same sense as the others discussed above. However, the results of this experiment are included in Table III-1 since the unit was incorporated into a solar system and was used to partially cool a test house.

Because of low collection efficiencies (a maximum collection temperature of only 145°F was achieved) the COP was only about 0.11, compared to a theoretical value of about 0.56 at an inlet temperature of 180°F. Because of the low COP and significant electrical power requirements for the water and solution pumps, the EER (cooling output to electrical power dissipation) was only about 4.5. At design conditions, an EER of 14.2 is predicted--a value not large compared to conventional electrically driven vapor compression units.

b. Desiccant Systems

A solar activated liquid desiccant system was planned for installation and test in a large office building. It was intended for use as a supplement to a conventional chiller and was expected to meet roughly 5% of the building air conditioning load. The project was ultimately abandoned when other (more cost effective) energy conservation measures were applied to the conventional HVAC system design.

Demonstrations of the cooling performance of the IGT Munter's dessicant system are reported to be currently underway in small buildings owned by local gas companies in three North American cities* (Reference 45). The demonstration units are reported to provide three tons of cooling. No performance data is at present available to the public.

B. CONCLUSIONS FROM DEMONSTRATIONS

The demonstrations have shown that solar cooling can be achieved by at least several alternative approaches. The goals of these experiments were essentially technical in nature with little emphasis placed upon the cost effectiveness of the various approaches.

Of the cooling devices demonstrated, only the lithium-bromide continuous absorption units can truly be said to be a proven technology.

*City of Industry (California), Dallas, Chicago.

The other approaches used technology that remains in a developmental stage, where the efficiency, reliability and manufacturing costs are as yet not established.

The demonstrations revealed that the keys to an effective solar cooling system are not only the characteristics of the cooling device itself, but the manner in which the device is integrated into the solar system. The major conclusions are:

- Good performance requires matching of collector, storage and cooling device to loads.
- Good collection efficiency at "high" temperatures (200°F) is critical to success of experiment (selective absorber surface required with collector).
- Cold storage is advantageous to reduce warm-up transient heat losses in the cooling unit. Some hot storage is desirable to buffer the collector and cooling unit.
- Heat leaks from storage are critical to cooling system performance.
- Stratification is desirable in both hot and cold storage units. Short circuiting of fluids in tank should be avoided.
- Control of inlet and cooling streams is necessary to optimize system performance.

IV. PERFORMANCE EVALUATIONS

A. OVERALL APPROACH

Figure IV-1 illustrates the overall approach developed for the economic performance evaluation of solar heating and cooling systems. Analyses of system performance was made for single-family residences and for typical design office buildings in the seven cities specified by CEL. The analysis procedure began with an estimate of monthly heating and cooling loads for the two building types in each location. The loads are related to the thermal design of the building--primarily the degree of insulation and type of fenestration used in the building--and to the heating and cooling degree-days. Internal heat generation in the building serves to reduce the annual heating load, and to increase the annual cooling load.

The next step in the analysis was to estimate the monthly solar flux incident on the solar collector. The net solar energy available for use in driving the cooling system is related to the area of the collector, the tilt angle of the solar collector and the monthly insolation. An approximate procedure was developed for estimating the solar flux on the tilted collector from values for monthly insolation on the horizontal surface.

Once the monthly loads and monthly solar insolation were determined, correlations were used to estimate the fractions of the heating and cooling load met by the solar system. The correlations, which will be described in greater detail in the next section, involve the ratio of the solar flux on the collector to the monthly space conditioning load.

Having determined the fraction of the monthly load met by the solar system, the auxiliary electrical energy expenditures were estimated. The energy expenditures consist of pump energies in the solar collector loop and other fluid transfer loops, pump energies expended in the refrigeration machine, and other backup electric energy as required.

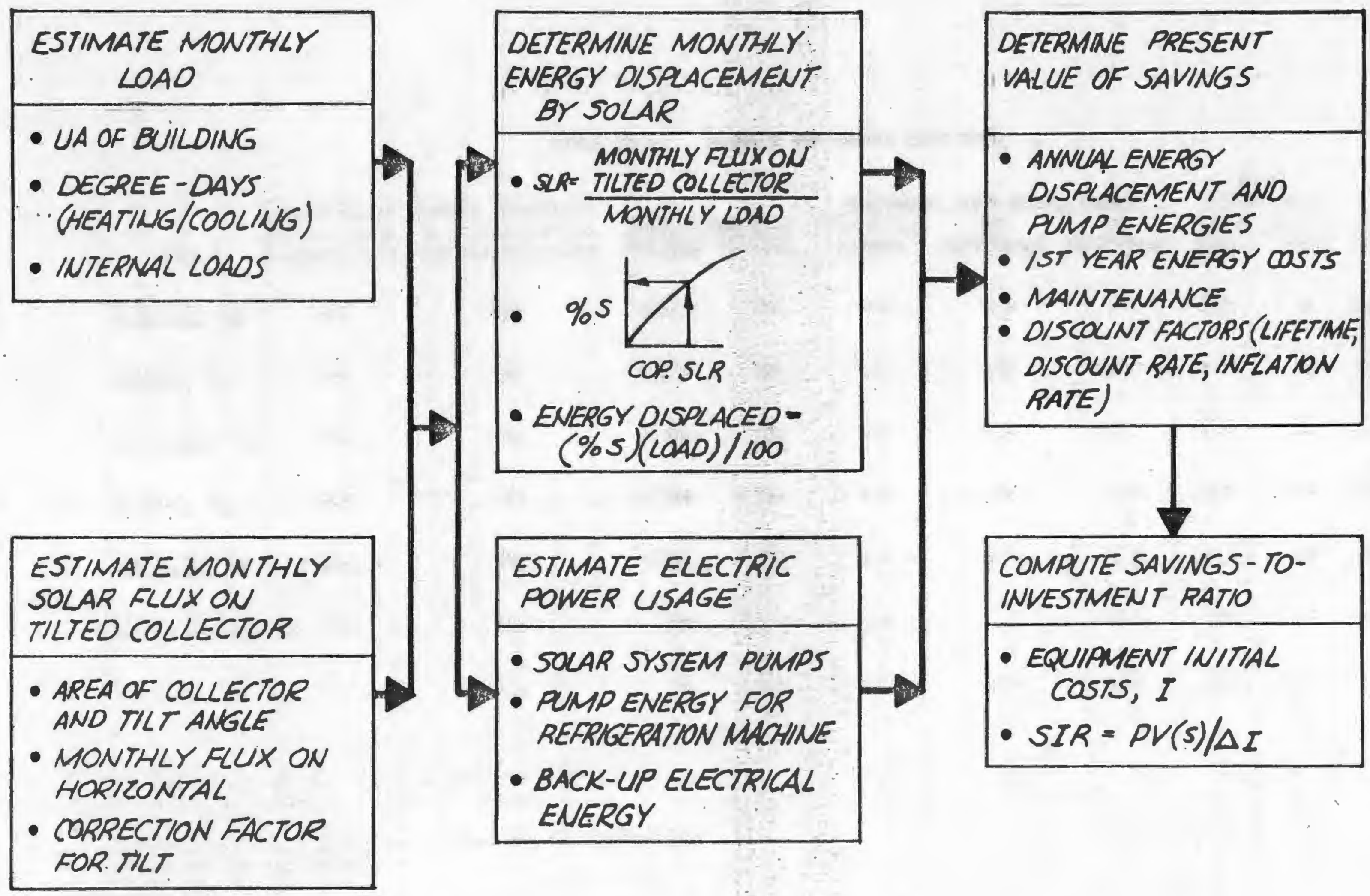


FIGURE IV-1 OVERALL PROCEDURE FOR EVALUATION OF ECONOMIC PERFORMANCE

From the fraction of the monthly loads displaced by the solar system and the auxiliary electrical energy expenditures, estimates were made for the energy cost savings achieved by the solar system. The present value of these savings was determined over the defined lifetime of the system (10 years) by taking into account the appropriate discount factors, inflation rates, energy escalation rates and maintenance. From the present value of the savings and the initial cost of the solar equipment, the savings-to-investment ratio was computed.

1. Building Types and Climatic Conditions

Table IV-1 summarizes climatic and energy cost data for the seven selected cities. The climatic data listed are the annual insolation on a horizontal surface and the annual heating and cooling degree-days. The insolation and degree-day data were provided by the Civil Engineering Laboratory. The estimated annual insolation values for collector surfaces tilted at the latitude angle are also tabulated. September, 1976 energy cost data, obtained from the U.S. Bureau of Labor Statistics, are listed in the table for electrical energy, oil and gas. Data are provided in the table for the expected real escalation in energy costs, in units of percent per year. These nominal values were provided by the Civil Engineering Laboratory and were used for subsequent economic evaluations. From these nominal energy cost escalations, the ten year inflation-discount factors were calculated.

The table reveals a significant variation in climatic conditions between the seven cities selected for analysis. Bremerton and Chicago have low annual solar fluxes and high heating degree-days. At the other extreme, Key West has a high annual insolation and high cooling degree-days. Figure IV-2 shows monthly heating and cooling loads for both a single-family residence in Boston and Miami, and for a typical office building in Boston and Miami.* The

*A computer program was used to calculate the monthly loads in Boston and Miami using weather tapes which were available to us for these two cities.

TABLE IV-1 CLIMATE AND ENERGY COST DATA

City	Insolation at Surface (Avg. Langleys/Day)	Annual Insolation on Tilted Surface (10^6Btu/ft^2-year)[1]	Nominal Deg-Days Heating	Nominal Deg-Days Cooling	September, 1976 Energy Costs[2] Electricity (¢/kWh)[3]	Oil ($/$10^6$Btu)[4]	Gas ($/$10^6$Btu)[5]	Real Energy Cost Escal. (%/Yr) Elec.	Fuel	10 Year Infl-Disc. Factors[6] Elec.	Fuel
Bremerton, WA	273	.477	4,727	183	1.4	3.2	2.4	7.3	8	8.86	9.14
Chicago, IL	273	.437	6,127	925	4.5	2.8	2.0	5.6	8	8.21	9.14
New London, CT	339	.532	5,700	633	4.3	2.9	2.1	6.9	8	8.70	9.14
Norfolk, VA	382	.573	3,488	1,441	4.7	3.0	2.8	5.8	8	8.29	9.14
San Diego, CA	407	.590	1,507	727	4.4	2.7	2.1	7.3	8	8.86	9.14
Corpus Christi, TX	436	.610	930	3,474	5.8	-	2.0	7.5	8	8.94	9.14
Key West, FL	452	.624	64	4,888	4.1	2.9	1.8	5.8	8	8.29	9.14

[1] Estimated, collector tilted at latitude angle
[2] Source: U.S. Bureau of Labor Statistics
[3] Based on 500 kWh/month
[4] #2 fuel oil; #6 fuel oil costs about 80% of figure shown
[5] Based on 100 therms/month
[6] 10% real discount rate

NOTE: 10^6Btu = 10 therms = 1,000 cubic feet of gas
= 7 gallons #2 fuel oil
= 80 lbs. coal
= 293 kWhs

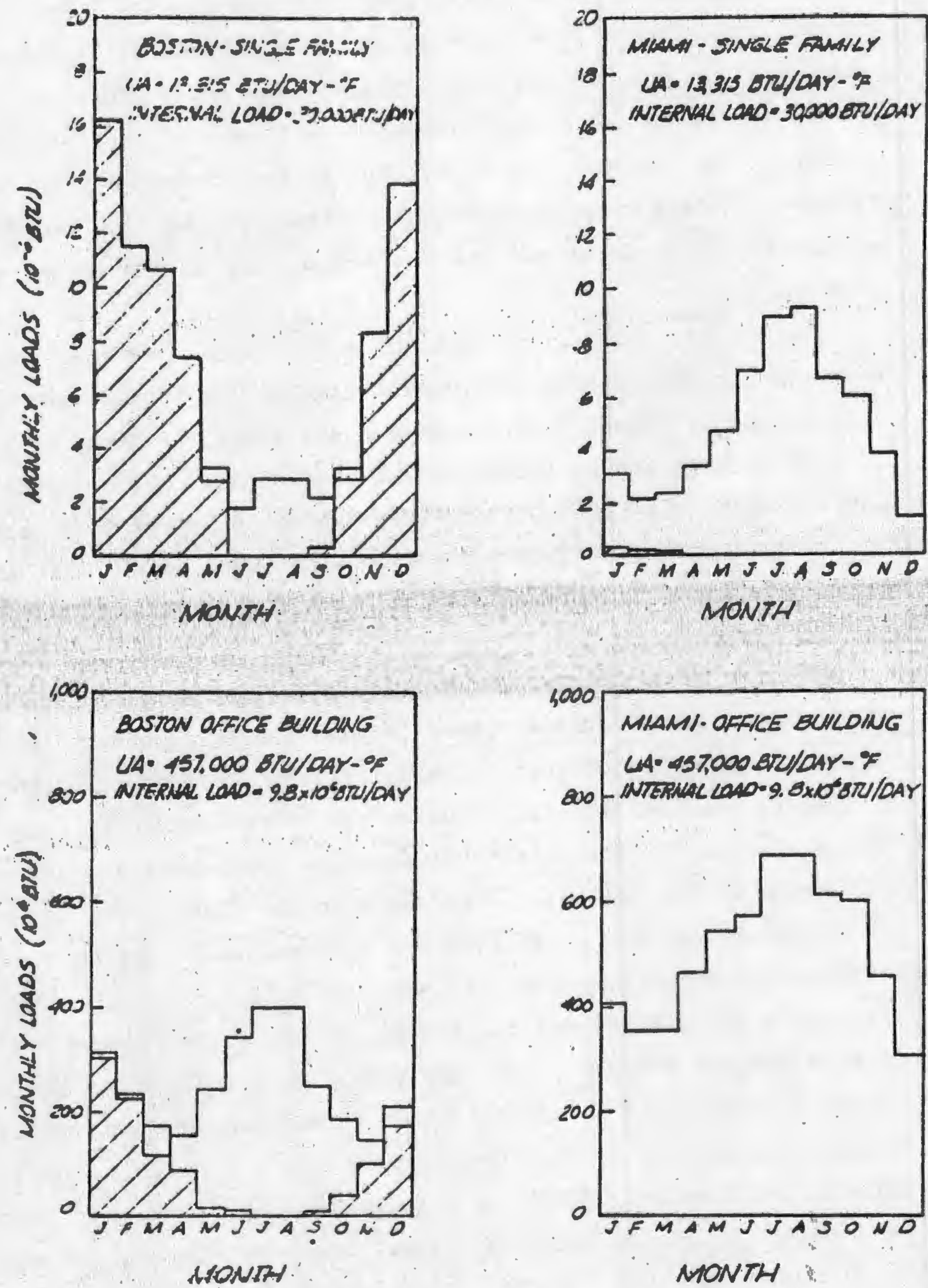


FIGURE IV-2 MONTHLY LOADS FOR TYPICAL BUILDINGS loads shown for Boston should be similar to those expected in cold regions such as Chicago or Bremerton, while the loads shown for Miami are expected to be quite close to those for Key West. The single-family and office building can have substantially different load profiles even in the same location. In particular, the single-family residence has a much higher ratio of annual heating load to cooling load. In the office building active cooling may be required throughout the entire year whereas cooling will be required in the single-family northern climate residences only during the mid-summer.

The monthly heating and cooling load profiles for the seven cities selected for analysis by CEL are expected to lie between the extremes illustrated in Figure IV-2 for Boston and Miami. Figures IV-3 and IV-4 illustrate the estimated annual heating and cooling loads in these cities. The annual loads were estimated by assuming them to be proportional to the heating and cooling degree-days. The annual loads calculated for Boston and Miami are also noted on the figures.

2. Correlations for Energy Displacement by Solar

In order to estimate the percent of the load supplied by the solar system it is necessary to determine the solar energy incident on the tilted solar collector. Figure IV-5 illustrates a method of estimating the monthly solar flux on the tilted collector from the tabulated data for the monthly flux on the horizontal and from the tilt angle of the collector. As shown in the figure, the angle of incidence between the solar flux and a normal vector to the collector surface will vary throughout the year and will be a function of the latitude angle and the declination angle--the angle between the equator and the ecliptic. The approximation formula shown in the figure is similar to an approximation formula recommended by Los Alamos Scientific Laboratory (References 46 and 47). The second term in the equation, 0.004, is a correction factor used to improve the estimation and is valid when the fluxes are expressed in units of 10^6 Btu/ft^2-month. Figure IV-6 illustrates a conversion factor

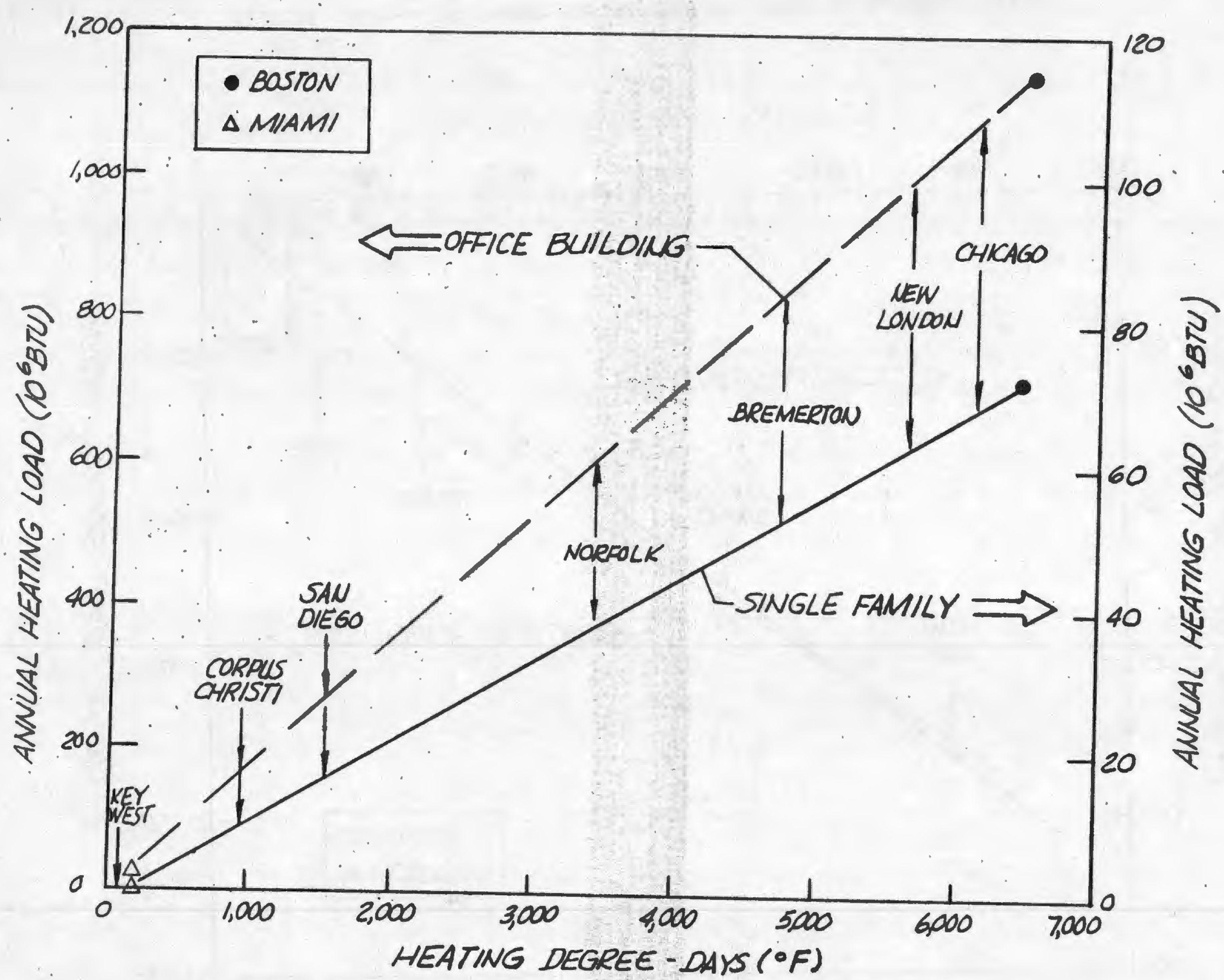


FIGURE IV-3 ESTIMATED ANNUAL HEATING LOADS IN VARIOUS CITIES

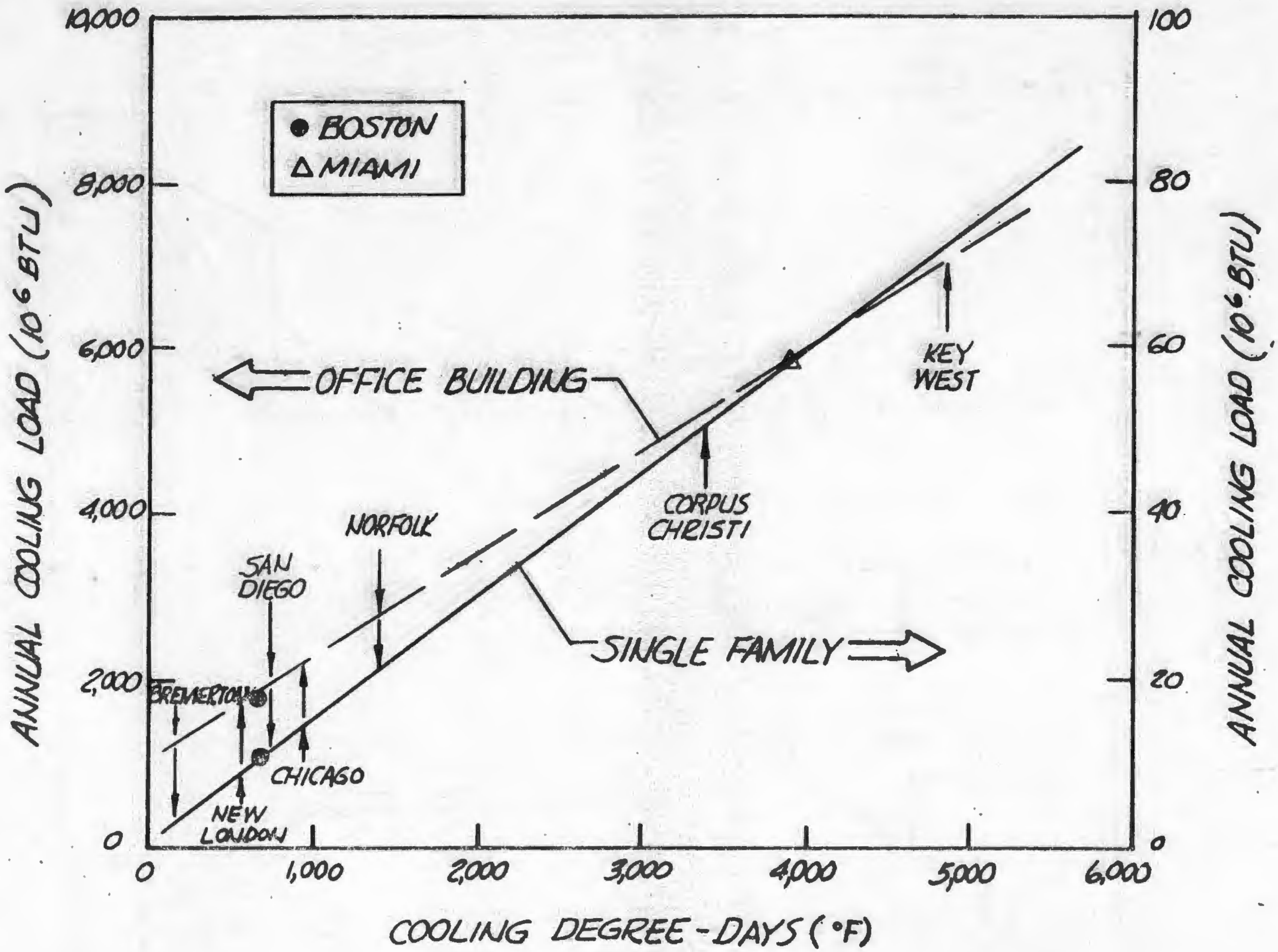
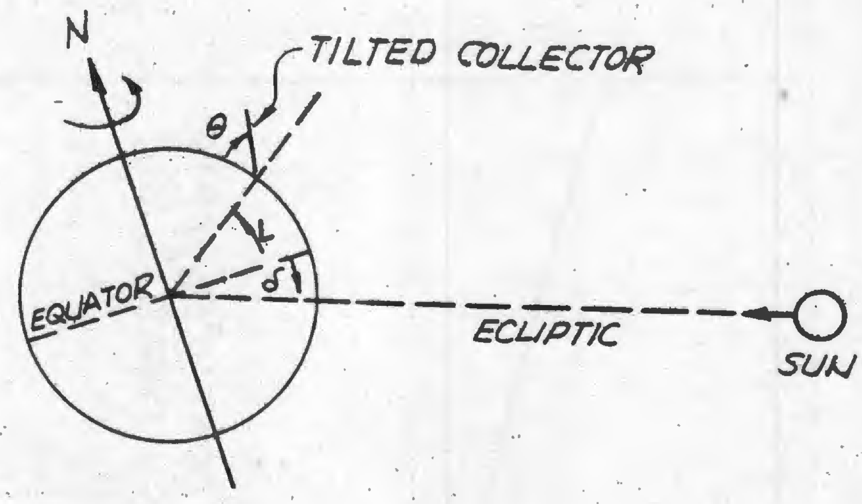


FIGURE IV-4 ESTIMATED ANNUAL COOLING LOADS IN VARIOUS CITIES



θ = TILT ANGLE (FROM LOCAL HORIZONTAL)

L = LATITUDE

δ = DECLINATION - 23.45 cos (30M-187)

M = MONTH NUMBER, 1 = JAN, 2 = FEB, ETC.

- MONTHLY FLUX ON TILTED COLLECTOR*

$$= \frac{\cos(L-\delta-\theta)}{\cos(L-\delta)} \times \text{MONTHLY FLUX ON HORIZONTAL}^{*} - 0.004^{*}$$

- 1 LANGLEY/MONTH = 3.687 BTU/FT^2-MONTH

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* UNITS = 10^6 BTU/FT^2-MONTH

FIGURE IV-5 ESTIMATION OF MONTHLY SOLAR FLUX ON TILTED COLLECTOR

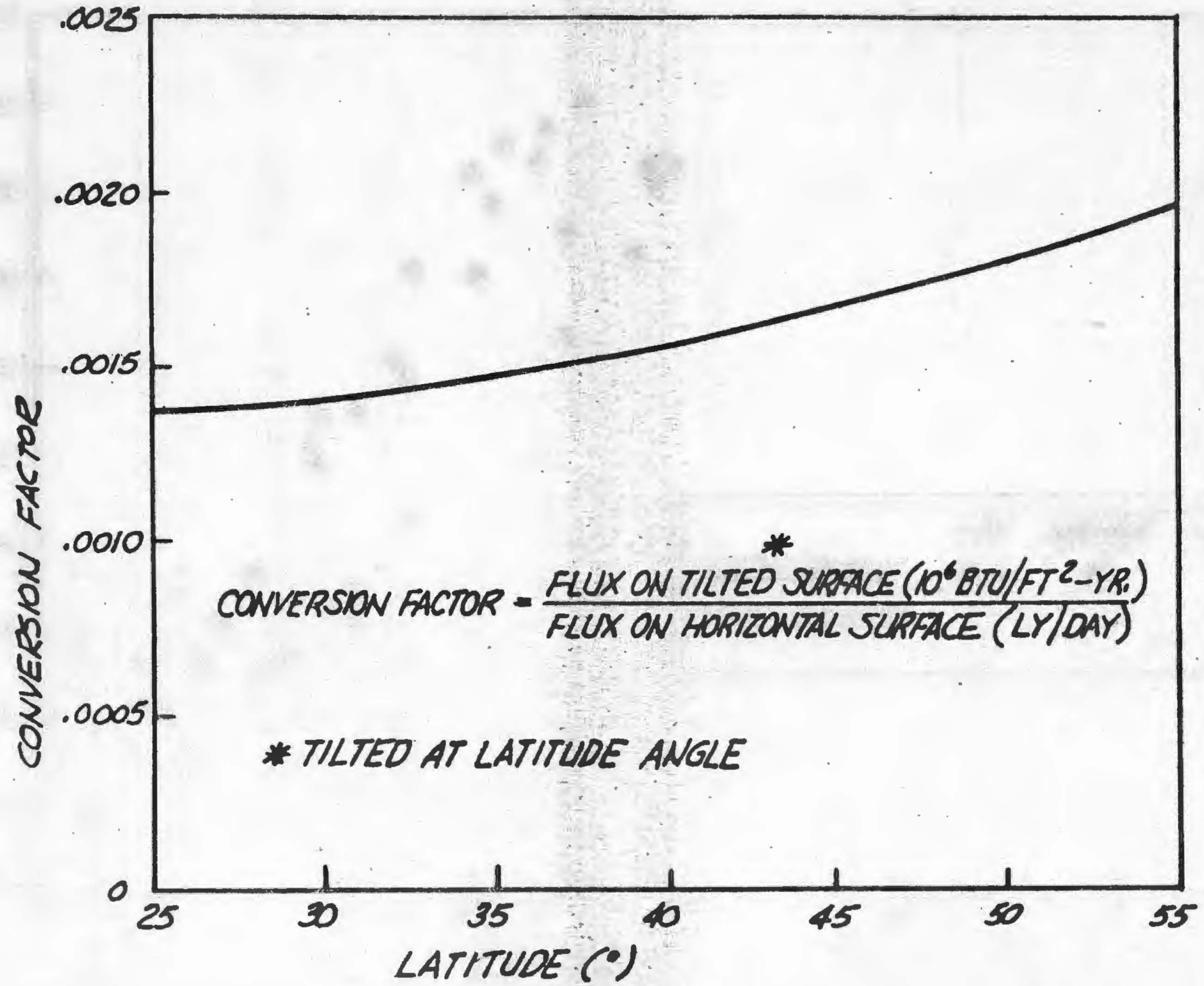


FIGURE IV-6 CONVERSION FACTOR FOR ANNUAL SOLAR FLUX ON TILTED COLLECTOR FROM ANNUAL SOLAR FLUX ON HORIZONTAL SURFACE that can be used to convert the annual solar flux on the horizontal surface, given in units of average langley's per day, to the annual solar flux on the tilted surface, in units of 10^6 Btu/ft^2-yr. The conversion factor is shown as a function of latitude angle. Figure IV-6 was used to convert the annual insolation data provided by the Civil Engineering Laboratory to the data shown in Table IV-1 for a surface tilted at the latitude angle.

From estimated values for the monthly buildings loads and the monthly solar flux incident on the solar collector, approximate values for the fraction of the load met by the solar system were obtained. Figure IV-7 illustrates values for the percent solar--the fraction of the load met by the system--as a function of a correlating parameter: SLR COP, where the term COP represents the coefficient of performance of the system and SLR, called the Solar Load Ratio, is defined by:

Solar Load Ratio (SLR)

$$= \frac{\text{Solar Flux on Tilted Collector}}{\text{Load}}$$

$$= \frac{\text{(Flux/Area)(Collector Area)}}{\text{Load}}$$

where:

Load = thermal energy which must be supplied by HVAC system to satisfy building's comfort requirements

The load is always defined in terms of the energy which must be either added to the building for heating purposes or removed from the building for cooling purposes. The presence of the COP in the correlation

*Note: For tracking collectors multiply SLR by 1.5.

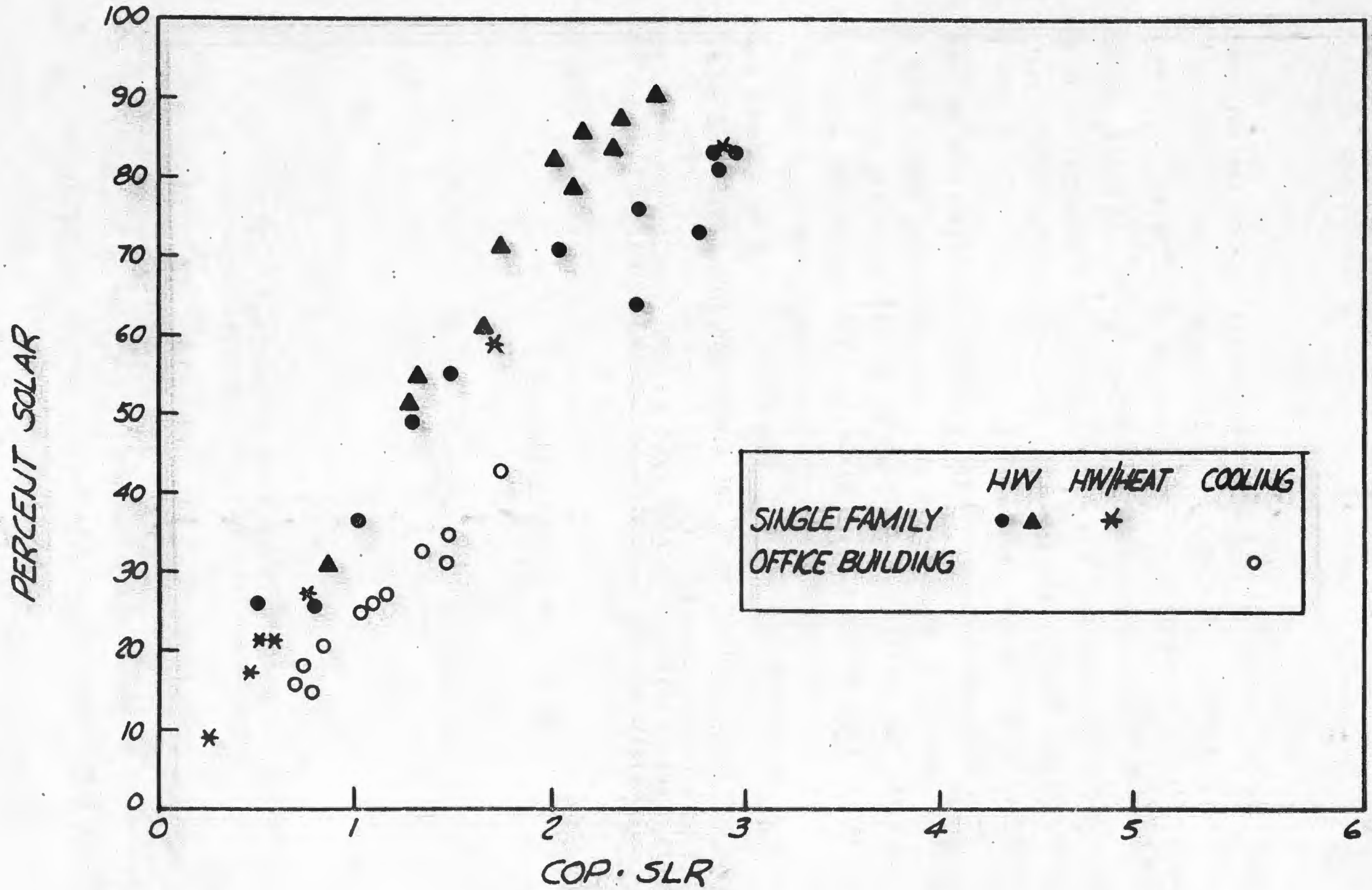


FIGURE IV-7 CORRELATION CURVE FOR 1-PANE, SELECTIVE SURFACE COLLECTOR accounts for the fact that the device performing either the heating or cooling function may require heat input quantities which are different from the quantity of heat added to the building or removed from the building. Hence, the product of the system COP and the Solar Load Ratio yields the ratio of the solar flux incident on the collector over a given period of time to the thermal energy required as input to the HVAC system to meet the building heating or cooling load over the same period of time. The correlation is intended to hold for all HVAC functions: water heating, space heating, and space cooling. For water heating and/or space heating, the COP is generally unity. For space cooling, on the other hand, the COP is generally less than unity (typically 0.7-0.8).

The symbols shown in Figure IV-7 represent actual computer calculations for the monthly percent solar for single-family and office buildings in a number of locations, using a proprietary ADL computer simulation program. The calculations were performed for cities in the U.S., Japan and Europe. The correlation is similar to one developed by Los Alamos Scientific Laboratory (References 46 and 47). The main difference is the explicit use of the COP as a part of the correlation parameter.

Figure IV-8 presents a correlation for a high performance collector (see Appendix A for a discussion of collector technology). The high performance collector utilizes a very low vacuum in addition to a selective absorber surface to reduce the outward losses. As a result, the collection efficiencies will be higher than those for a one-pane selective surface collector.

As an illustration of the use of the correlation curves consider the following example. Assume that a monthly load of 10×10^6 Btu is to be met by a cooling system that has a COP of 0.5. Further assume that a 600 square foot solar collector is used and that over the month the flux incident of the collector is $.05 \times 10^6$ Btu/ft^2. The SLR

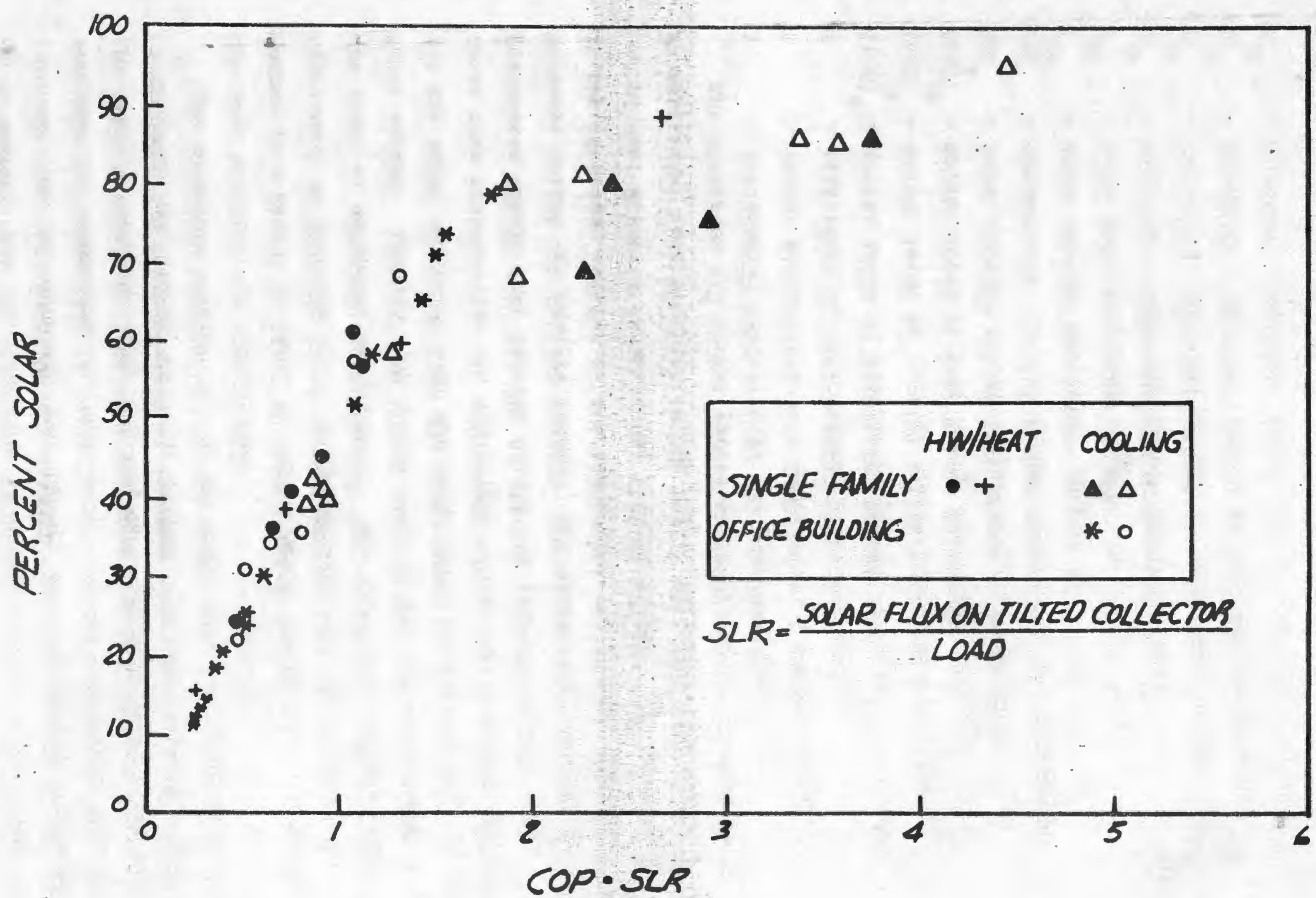


FIGURE IV-8 CORRELATION CURVE FOR HIGH-PERFORMANCE COLLECTOR ratio would then be equal to 600 x .05 ÷ 10 = 3.0. Hence, over the month, COP • SLR = 1.5. For a high performance collector the fraction of the cooling load met would be approximately 62%. If a single pane selective surface absorber collector were used, on the other hand, the percent solar cooling achieved during the time period would be only about 32%.

3. Solar Cooling Economics

a. Present Value of Savings

The net cost savings realized by the solar system consists of: costs saved for the conventional energies displaced by the solar system, costs expended for additional electrical energy required by the solar system (for example to operate the solar collector pump), and costs expended for maintenance required on the solar system. The present value of savings is determined over the lifetime of the system. The inflation - discount factors, the values of which are dependent on assumptions about rates of energy cost escalation and discount factors, have been given in Table IV-1 for the seven cities selected for analysis. Formulas for computing the present value of savings from the discount factors and other solar system performance factors are presented below in Equations IV-1a and IV-1b:

Electric Conventional Cooler:

$$PV\,(s) = DF_h(Q_{hs}/\eta)(V/Q)_h + DF_c(Q_{cs}/COP_c)(V/Q)_c - 0.017\,DF_e[Q_{hs} + Q_{cs}/COP_s](V/Q)_e - DF_m\,\phi\,I \qquad \text{(IV-1a)}$$

Heat Actuated Conventional Cooler:

$$PV\,(s) = DF_h(Q_{hs}/\eta)(V/Q)_h + DF_c(Q_{cs}/COP_c)(V/Q)_c/\eta - 0.017\,DF_e[Q_{hs}+Q_{cs}/COP_s](V/Q)_e - DF_m\,\phi\,I \qquad \text{(IV-1b)}$$

DF_h = inflation - discount factor for heating energy displaced
DF_c = inflation - discount factor for cooling energy displaced
DF_e = inflation - discount factor for electrical energy
DF_m = inflation - discount factor for maintenance
Q_{hs} = solar heat delivered, 10^6Btu
Q_{cs} = solar cooling delivered, 10^6Btu
COP_c = conventional cooling system coefficient of performance
COP_s = solar cooling system coefficient of performance
$(V/Q)_h$ = dollar value of heat energy displaced, $\$/10^6$Btu
$(V/Q)_c$ = dollar value of cooling energy displaced, $\$/10^6$Btu
$(V/Q)_e$ = dollar value of electrical energy, $\$/10^6$Btu
η = efficiency of conventional heating system
ϕ = annual maintenance as a function of equipment costs, %/year
I = incremental capital cost for HVAC equipment, $

The equations are almost identical except for the presence of an additional term relating to the boiler efficiency in the equation for the heat actuated conventional cooling system. The first term in the equations relates to the discounted energy cost savings achieved during the heating season. The second term relates to the discounted energy cost savings during the cooling season. The third term accounts for the additional electrical energies expended for the solar collector pump and other pumps that are a part of the solar system. Finally, the fourth term in each equation relates to the costs of equipment maintenance. The additional term for boiler efficiency in Equation IV-1b arises because fuel energy must be burned in a boiler in order to create steam or hot water to drive the heat actuated air conditioner.

The equations contain all of the terms that are required in order to evaluate the present value of savings over the equipment lifetime. The basic assumptions that were used when carrying out the economic analyses are summarized in Table IV-2. The pump energies have been factored into the equations for present value of savings given above. It is assumed that the remaining electrical energy expenditures are the same for the solar system and the conventional system. For the

TABLE IV-2   ASSUMPTIONS RELATING TO ECONOMIC ANALYSIS

- Assumptions Made in Estimating Savings
  - Pump energies for solar system = 5 kWh/10^6Btu of solar heat collected
  - Absorption machine pump energies = 4 kWh/10^6Btu of cooling
  - Remaining auxiliary electrical energies (cooling tower pump, fan, etc.) same for solar and conventional systems
  - Maintenance = 2%/year of equipment initial cost, escalates at same rate as general inflation

- System Considerations
  - "Conventional" system assumed same as backup to solar system
  - Heating and cooling performed throughout year
  - COP = 3.5, electric vapor compression machine
    = 0.65, absorption machine
  - Burner efficiency = 60%

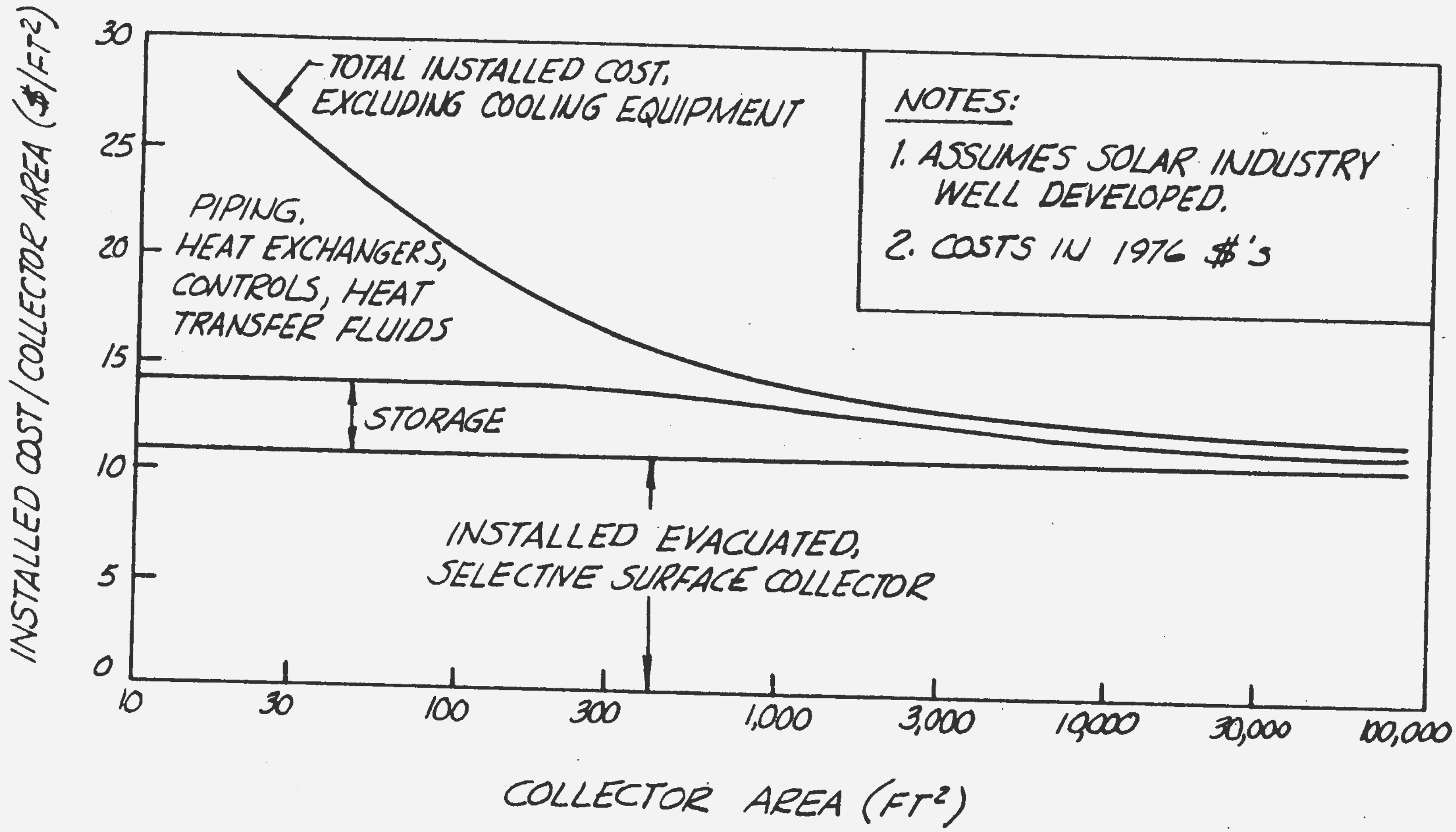
- Nominal Assumptions
  - 10 year life
  - No resale value
  - 10% real discount rate
  - Labor has no real cost growth

base case calculations of present value of savings, we assumed an annual equipment maintenance of 2% per year based on the initial costs of the equipment and that the maintenance rate escalates at the same rate as the general inflation. It is assumed that the conventional HVAC system is the same system that is used as the backup to the solar system--that, the solar heating and cooling equipment is added to the existing or planned conventional HVAC system. When using Equations IV-1a and IV-1b the COP of the conventional cooling equipment was taken to be 3.5 when an electric vapor compression machine was used, and was taken to be 0.65 when a heat actuated absorption machine was used. We assume a burner efficiency of 60%. The remaining assumptions relating to equipment lifetime and discount rates are also shown in the table.

b. Investment Costs for Solar

The investment cost used in calculations for the present value of savings is the incremental cost for the solar system as compared to the HVAC system without solar heating and cooling components. The costs include: solar equipment, delivery, installation, design and check out. Because the solar heating and cooling industry is not yet mature, costs for installed solar equipment cannot be stated with a great deal of precision. At present when a solar heating and cooling system is considered for installation in a building, a large part of the cost is associated with the design of the equipment. In addition, solar components may have to be specially fabricated for the job. In addition, installation costs for solar equipment tend to be fairly high at present because of the relative inexperience on the part of general contractors in dealing with this new equipment and their reluctance to take risks when installing such equipment.

Figure IV-9 illustrates estimates made by Arthur D. Little of installed costs of solar heating equipment as a function of system size assuming a mature solar industry. Costs are given in 1976



SOURCE: ARTHUR D. LITTLE ESTIMATES

FIGURE IV-9 INSTALLED COST OF SOLAR EQUIPMENT (EXCLUDING COOLING EQUIPMENT) dollars in units of installed cost per collector area (dollars per square foot). The data shown in the figure represent costs that are achievable in the 1985-1990 time frame. Some component costs will be achieveable in a shorter time period. The data shown in the figure assume the use of an evacuated selective surface collector. The installed costs for this component are assumed to be $11 per square foot (present component costs for evacuated selective surface collectors are in the range $20-$40 per square foot). The costs of solar storage were computed assuming the use of a steel tank containing approximately two gallons of water per square foot of solar collector area. The data shows that the largest systems have the smallest costs on a dollars per square foot of collector area basis. Figure IV-10 was used when carrying out analyses of savings-to-investment ratios of the various candidate heating and cooling systems.

When solar cooling is performed, the solar system capital cost will be increased by addition of a heat actuated cooling system, if one is not already present in the conventional HVAC system. Figure IV-10 summarizes the unit costs of various types of cooling devices as a function of the cooling capacity. Figure IV-11 shows the installed cost of cooling towers as a function of cooling capacity. The data shown in Figures IV-10 and IV-11 were used in subsequent economic analyses.

B. RESULTS

1. Technical Performance Parameters

Table IV-3 presents results for the percent of the cooling load met by an absorption system and Rankine cooling system for various types of solar collectors. (The case of an office building located in Miami was studied, using the ADL computer model and a weather tape for Miami.) The Rankine cooling system is assume to have an overall cooling COP equal to 36% of Carnot.* Under this assumption,

*The COP's of the heat engine and refrigerator are assumed to be 65% Carnot and 55% Carnot, respectively, based on fluid inlet temperatures.

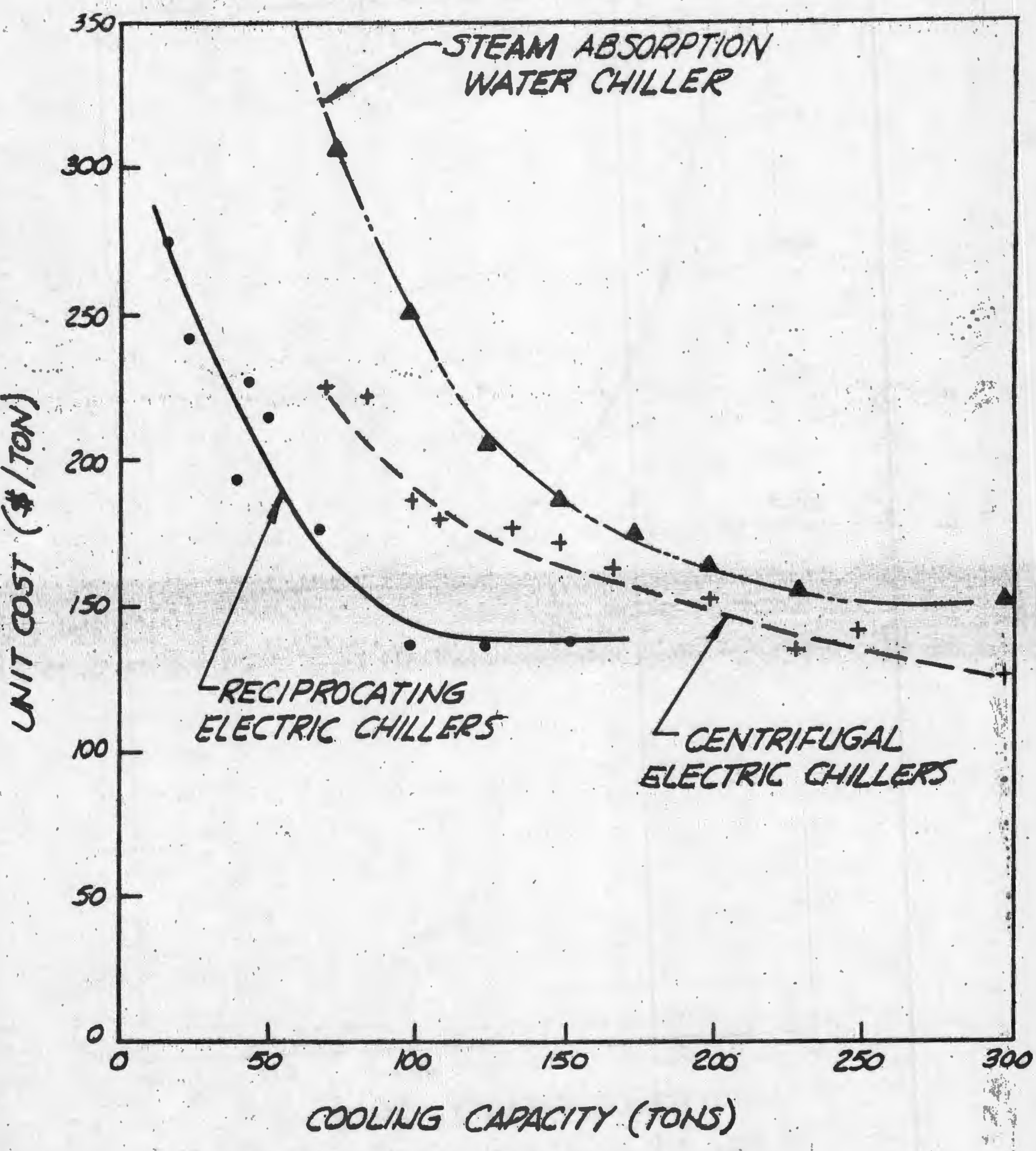
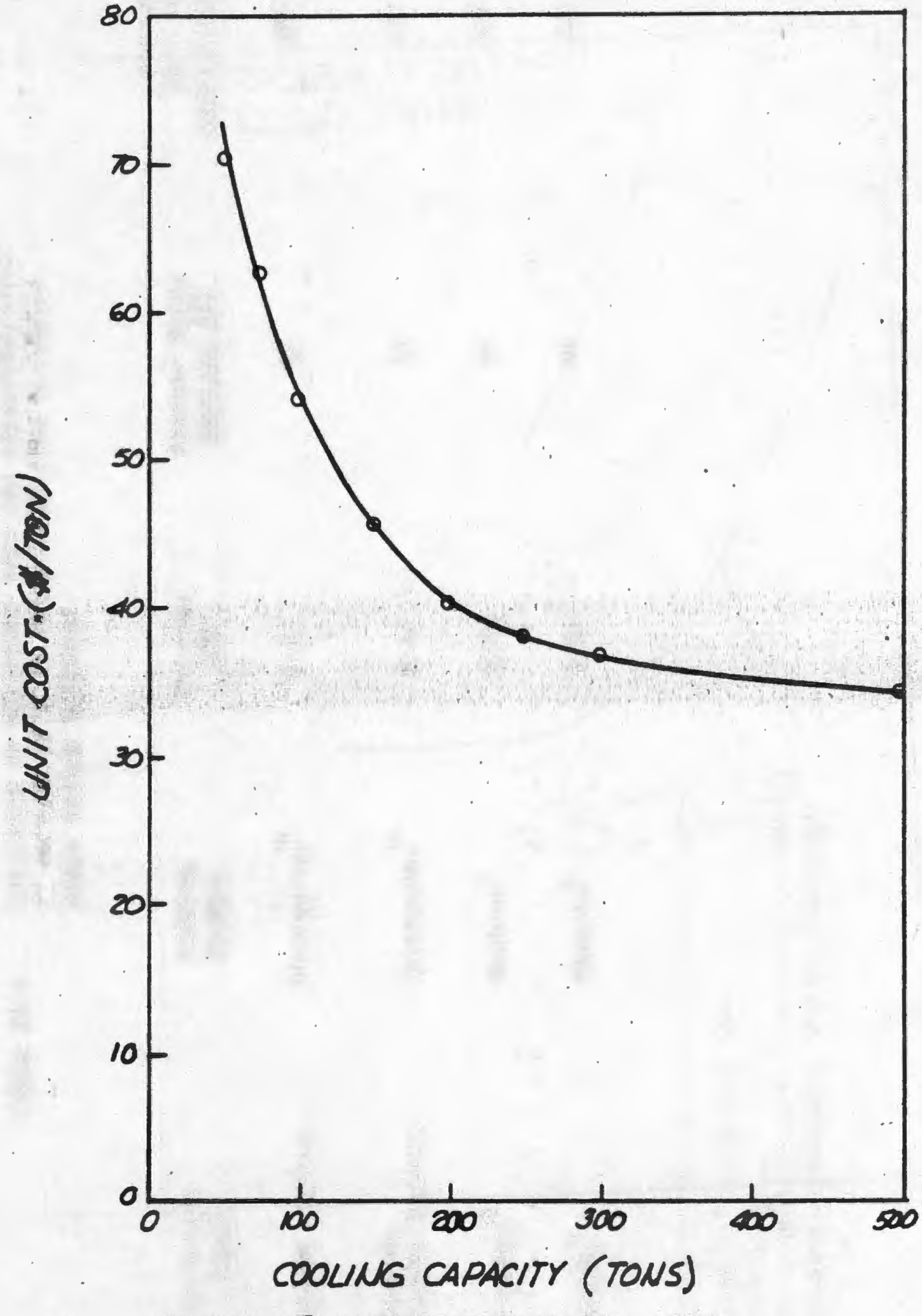


FIGURE IV-10 INSTALLED COSTS OF CHILLERS



SOURCE: BUILDING COST FILE, 1976

FIGURE IV-11 INSTALLED COSTS OF COOLING TOWERS the Rankine system achieves a COP of 0.65--the value assumed for the absorption unit--at an inlet temperature to the boiler of 175°F, cooling water temperature of 85°F and an evaporator (chilled water) temperature of 45°F. (It should be noted that significant development efforts on the Rankine system are required to achieve such a high COP, while the COP assumed for the absorption unit is based on current technology. Some of the same design features applied to the Rankine system to achieve the assumed COP--e.g., large heat exchanger surfaces--could be applied to the current technology absorption units to achieve higher operating COP's, as discussed in Section II-B.)

The results in Table IV-3 highlight the importance of using highly efficient solar collectors when solar cooling is to be accomplished. Use of an evacuated solar collector will yield about twice the solar cooling as accomplished when an unevacuated flat plate collector is used. (Other approaches to achieving high efficiency collection at high temperatures are discussed in Appendix A.) Tracking by the solar collector can yield further significant advantages (compare the two results shown for a Rankine cooling system). Tracking increases the solar flux incident on the collector, and increases the collector efficiency at a given solar flux, by reducing the glancing angle reflection losses.

The average COP's shown for the model Rankine cooling system are higher than those for current technology absorption machines. According to the model for the Rankine system, the COP increases with inlet temperature, whereas COP is relatively insensitive to inlet temperature with an absorption device. Because of this variation in COP with inlet temperature, the average COP of the Rankine cooling system driven by a tracking collector was about 0.9. (As discussed in Section II-B, the COP of a double-effect absorption machine will be in the range of 1.0-1.1 at inlet temperatures near 300°F.) Combination of high COP and increased solar flux on the tracking collector results in a substantial increase (about 40%) in the fraction of the cooling

TABLE IV-3 INFLUENCE OF COLLECTOR TYPE AND CHARACTERISTICS OF COOLING UNIT ON SYSTEM OPERATING ECONOMICS, MIAMI OFFICE BUILDING

Collector Type[a]	Cooling Device	Average COP	Percent Solar Cooling (%)	Average Heat Input Temperature (°F)
Selective Surface	Absorption[b]	0.65	24	187
Evacuated, Selective Surface	Absorption[b]	0.65	51	195
Stationary[d]	Rankine[c]	0.74	57	189
Tracking[d]	Rankine[c]	0.90	80	225

[a] 15,000 ft^2 collector area
[b] COP = 0.65
[c] COP = 35% Carnot (= 0.65 at 175°F inlet)
[d] Evacuated, selective surface collector

load met by the solar system.

2. Savings-to-Investment Ratios

a. Example of Water Heater

The savings-to-investment ratio achieved by the solar system is dependent not only on the cost of the energy displaced and the capital cost of the solar system but also on the character of the load displaced. As an example, we first consider the case of a solar water heating system located in New London, Connecticut. A solar water heating system enjoys the advantage of displacing a relatively constant load thereby utilizing the solar equipment throughout the year.

Table IV-4 defines the case analyzed. The example solar water heating system contains a 50 square foot, 1-pane selective surface solar collector, and is used to displace electricial energy. The installed cost of the solar system would be about $1,000 under the assumption that the solar industry is well developed. With an annual water heating load of 20 x 10^6 Btu, the system will displace approximately 49% of the electrical energy expended to meet the annual load. Assuming a ten year life and a 10% per year (real) discount rate, the present value of savings is $1,195, yielding a savings-to-investment ratio of 1.2. If we had assumed that the conventional energy displaced were oil costing $2.90 per million Btu's, with a heating efficiency of 60%, the savings-to-investment ratio would be about 0.29. This case example serves to illustrate several important points. First the savings-to-investment ratio is highly sensitive to the cost of the conventional energy displaced. Secondly, the order of magnitude of savings-to-investment ratio is approximately 1. This is an important finding and should be kept in mind when reviewing the subsequent results for solar heating and space cooling equipment.

b. Performance Results for Cooling Systems

Using the methodology described earlier, calculations were made for the fractions of the heating and cooling loads met in a single family residence and in an office building in each of the seven cities.

TABLE IV-4 EXAMPLE OF SOLAR WATER HEATER

- Definition of Example Case:
  - New London, annual flux on tilted collector = 0.53×10^6 Btu/ft^2
  - 50 ft^2 solar collector, 1-pane selective
  - Solar system cost = \$1,000 installed (assumes solar market well developed)
  - Energy displaced is electricity at 4.3¢/kWh
  - Annual load = 20×10^6 Btu
  - Conventional electric water heating efficiency = 80%

- Economic Assumptions:
  - 10 year life, 10%/year discount rate
  - Energy cost escalation = 6.9%/year, inflation-discount factor = 8.7
  - Maintenance = 2%/year of capital cost

- Performance Results:*
  - From correlation curve, %S = 40%
  - Energy displaced = 9.8×10^6 Btu
  - PV (s) = \$1,195
  - SIR = 1.20

---

*If conventional fuel is oil at \$2.9/10^6Btu, with heater efficiency = 60%, SIR = 0.29.

The analyses were carried out assuming the use of a stationary, evacuated selective surface collector sized to meet roughly 50% of the annual space conditioning load. The cooling unit used for these analyses was assumed to be a single stage lithium bromide absorption system.*

Analyses were carried out for two types of conventional systems, one containing an electrically driven vapor compression cooling system, and the other containing an absorption type cooling system. For the single family residence the system was sized at 200 square feet of collector area, with an investment cost of $6,100. It was assumed that the conventional cooling unit was an electrically driven compression unit. For the office building the solar system was sized at 5,000 square feet of collector area, with an investment cost of $72,500 when the convention cooling system was an absorption unit and a cost of $97,500 when the conventional cooling system was an electric chiller ($25,000 higher cost because of the requirement for an absorption machine as part of the solar system.

Table IV-5 summarizes the calculated savings-to-investment ratios in the seven cities. For the single family residence the calculations were carried out only for systems containing a conventional electric vapor compression unit. SIR's are shown both for systems meeting a space heating and cooling load and for systems which are used only to meet the cooling part of the load. Using the base case economic assumptions (defined in Table IV-2) the savings-to-investment ratios for a single family residence are of the order of 0.1. For the office building the calculated savings-to-investment ratios range from about 0.2 to 1 depending on the type of cooling system conventionally used in the building. The highest savings-to-investment ratios are found when the conventional system uses an absorption

*Performance predictions were not carried out for the Rankine or desiccant systems because of large uncertainties in COP and capital costs for these devices.

unit, because the cost of the absorption unit it not considered part of the investment cost of the solar system, and because the operating economics of a fuel fired absorption unit are frequently poorer than that of an electric chiller.* (The absorption machine has a lower coefficient of performance and requires the expenditure of fuel energies in a boiler, which itself, has an efficiency of less than 100%.)

When comparing savings-to-investment ratios for buildings in the various cities, it was found that the highest ratios are obtained for those cities having the highest conventional energy costs (Table IV-1). Other factors favoring high savings-to-investment ratio are high annual solar flux and high annual space conditioning loads.

Table IV-6 illustrates the results of a sensitivity analysis to determine the influence of various economic parameters on the calculated savings-to-investment ratio. The analyses were carried out for an office building in Key West. The parameters varied were the system lifetime, discount rate, energy escalation rate, annual maintenance costs, and base year energy costs. The most sensitive parameters are the system lifetime and the base year energy cost. With a lifetime of 20 years, the savings-to-investment ratios nearly double over those found for a system with a 10 year life. Using 1985 as the base year for energy costs (equivalent to installing the system in 1985) the savings-to-investment ratio is more than doubled, and is approximately 2 when the conventional cooling system is an absorption machine. (The energy cost data for 1985 were established from the 1976 energy cost data and escalation factors given in Table IV-1.) Another significant influence on the savings-to-investment ratio is the assumed discount rate. Use of only a 5% discount rate (expressed in real terms) yields an increase in the SIR of about 30%. Ordinarily, life-cycle cost analyses carried out for solar systems are based on a real discount rate of less than 5%/year and a system lifetime of 15-20 years. While the base case economic assumptions lend conservatism to the analysis, they appear to be somewhat restrictive in terms of weighing the potential cost

*Savings-to-investment ratio is determined relative to the conventional system, in this case a fuel-fired absorption unit.

TABLE IV-5 SUMMARY PERFORMANCE INDICES FOR SINGLE-STAGE LITHIUM BROMIDE ABSORPTION SYSTEM WITH EVACUATED, SELECTIVE SURFACE COLLECTOR - NOMINAL VALUES FOR ECONOMIC PARAMETERS

		Conventional: Electric Compressor[a] S.I.R.		Conventional: Absorption Machine[b] S.I.R.	
Building	Location	Heat and Cool	Cool-Only	Heat and Cool	Cool Only
Single-Family	Bremerton	0.14	--		
	Chicago	0.12	--		
	New London	0.13	--		
	Norfolk	0.14	--		
	San Diego	0.02	--		
	Corpus Christi	0.10	0.06		
	Key West	0.03	0.03		
Office-Building	Bremerton	0.19	--	0.65	0.27
	Chicago	0.19	--	0.49	0.21
	New London	0.28	--	0.65	0.28
	Norfolk	0.25	0.13	0.85	0.65
	San Diego	0.19	0.12	0.65	0.55
	Corpus Christi	0.27	0.23	0.45	0.39
	Key West	0.19	0.19	0.94	0.94

[a] Conventional Compressor COP = 3.5
[b] COP = 0.65

TABLE IV-6 SENSITIVITY ANALYSIS ON ECONOMIC PARAMETERS*

Parameter	Inflation - Discount Factors			S.I.R.	
	Labor	Electric	Fuel	Elec Compressor Conventional	Absorption Conventional
Nominal	6.45	8.29	9.14	0.19	0.94
20-Year, 10% Discount Rate	8.94	13.90	16.74	0.35	1.79
10-Year, 5% Discount Rate	7.91	10.39	11.55	0.24	1.20
Energy Escalation 25% Faster than Nominal	6.45	8.83	10.0	0.21	1.05
Maintenance = 3%/Year	6.45	8.29	9.14	0.12	0.88
1985 Energy Costs	6.45	8.29	9.14	0.39	2.03

*Key West Office Building, COP Absorption System = 0.65.

savings achievable with a solar space conditioning system.

The calculations for the savings-to-investment ratio were based on the use of a solar driven heat actuated cooling unit with a COP of 0.65. With increases in component sizes and controls, higher average COP's can be obtained from absorption type units (see Section II-B). Table IV-7 illustrates the effect of coefficient of performance on savings-to-investment ratio for a cooling system in an office building in Chicago and Key West. Although the calculations were carried out assuming the use of an absorption type cooling system, the calculations span the range of COP's achievable with the desiccant cooling system, and the Rankine cooling system. As before, the savings-to-investment ratios reflect the use of the 1976 energy cost data and the baseline economic assumptions.

TABLE IV-7 INFLUENCE OF COP ON SAVINGS-TO-INVESTMENT RATIO FOR ABSORPTION COOLING SYSTEM IN OFFICE BUILDING (BASE LINE ECONOMIC ASSUMPTIONS)

		Percent Solar (%)		S.I.R.*	
Location	COP	Heating	Cooling	Elec. Compressor Conventional	Absorption Conventional
Chicago	0.4	47	11	0.15	0.36
	0.5	47	14	0.17	0.41
	0.6	47	17	0.19	0.46
	0.7	47	19	0.20	0.52
	0.8	47	22	0.22	0.57
	0.9	47	25	0.24	0.63
	1.0	47	28	0.25	0.68
Key West	0.4	100	10	0.06	0.56
	0.5	100	13	0.11	0.73
	0.6	100	16	0.16	0.90
	0.7	100	18	0.21	1.07
	0.8	100	21	0.26	1.25
	0.9	100	24	0.31	1.42
	1.0	100	26	0.36	1.59

*Assumes capital cost independent of COP

REFERENCES

REFERENCES

A. GENERAL REVIEWS OF SOLAR COOLING TECHNOLOGY

1. J. A. Duffie and W. A. Beckman, "A Review of Solar Cooling", Proceedings of Joint Conference 1976 of the ISES and Solar Energy Society of Canada, Vol. 3, Solar Heating and Cooling of Buildings, pp. 13-26.

2. Technology Evaluation Report, Commercial Buildings, Report to ERDA by General Electric Company, COO/2683-76/1, April, 1976.

3. Solar Heating and Cooling of Buildings, Phase Zero Report, Westinghouse Electric Corporation to the National Science Foundation, NSF-RA-N-74-023, 1974.

4. Solar Heating and Cooling of Buildings, Phase Zero Report, TRW System Group Report to National Science Foundation, NSF-RA-N-74-022, 1974.

5. Solar Heating and Cooling of Buildings, Phase Zero Report, General Electric Company report to National Science Foundation, NSF-RA-N-74-021, 1974.

6. "Assessment of Solar-Powered Cooling of Buildings", Hittman Associates Report #HIT-600, RANN Document No. NSF-RA-N-75-012, April, 1975.

7. "Solar Air Conditioning", Task Report 3, A.D. Little, Inc. Solar Climate Control Project, September, 1974; also, summary Final Report of the Solar Climate Control Project, Phase I: Identification of Opportunities.

B. ABSORPTION CYCLE COOLING

8. "Optimization Studies of Solar Absorption Air-Conditioning System", prepared by University of Maryland for NSF, Report #NSF/RANN/SE/GI-39117/PR/76/2, January, 1976.

9. P. J. Wilbur and C. E. Mitchell, "Solar Absorption Air Conditioning Alternatives, Solar Energy, Vol. 17, 193-199 (1975).

10. "Engineering Design, Construction and Testing of a Salt-Water Absorption Cooling Unit Optimized for Use with a Solar Collector Heat Source", Progress Report to NSF prepared by Arkla Industries, Inc., Report # NSF/RANN/SE/GI-44095/PR/74/4, January, 1975.

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11. J. S. Rauch and B. D. Wood, "Steady-State and Transient Performance Limitation of the Arkla Solair Absorption Cooling System", Proceedings of Joint Conference 1976 of ISES and Solar Energy Society of Canada, Vol. 3, Solar Heating and Cooling of Buildings, pp. 387-405.

12. L. A. Stump, "Operation of Absorption Water Chillers for Seasonal Minimum Energy Requirements", Paper No. H3, Central Chilled Water Conference Proceedings, Editors: J. J. Pearson and D. L. Weast, Purdue University, 1976.

13. D. K. Miller, "The Performance of Water Cooled Lithium Bromide Absorption Units for Solar Energy Application," Heating/Piping/Air Conditioning , pp. 45-51, January, 1976.

14. "Solar Application of ES Absorption Chillers", York Division of Borg-Warner Corporation Publication Form 155.16-TD (575).

15. K. Dao, et al., "Performance of an Air-Cooled Ammonia-Water Absorption Air Conditioner of Low Generator Temperatures", Proceedings of Joint Conference 1976 of ISES and Solar Energy Society of Canada, Vol. 3, Solar Heating and Cooling of Buildings, 46-52.

16. R. K. Swartman, et. al., "Comparison of Ammonia-Water and Ammonia-Sodium Thiocyanate as the Refrigerant-Absorbent in a Solar Refrigeration System", Solar Energy, Vol. 17, 123-127 (1975).

17. J. A. Clark, "An Investigation of a Solar Powered Continuous Ammonia-Water Absorption Air Conditioner for Residential Dwellings", Masters of Science Thesis, University of Florida, 1976.

18. E. S. Gonzalez, "Design and Evaluation of a Solar Powered Intermittent Ammonia/Water Absorption Air Conditioning System", Master of Engineering Thesis, University of Florida, 1975.

19. S. L. Grassie and N. R. Sheridan, "Modelling of a Solar-Operated Absorption Air Conditioner System with Refrigerant Storage", Proceedings of Joint Conference 1976 ISES and Solar Energy Society of Canada, Vol. 3, Solar Heating and Coolings of Buildings, pp. 53-67.

20. E. H. Perry, "The Theoretical Performance of the Lithium Bromide-Water Intermittant Absorption Refrigeration Cycle", Solar Energy, Vol. 17, 321-323 (1975).

21. F. A. Costello, "Hybrid Solar Air Conditioning System", Solar Energy, Vol. 18, 149-152 (1976).

C. RANKINE-CYCLE COOLING

22. S. L. Sargent and W. P. Teagan, "Compression Refrigeration from a Solar Powered Organic Rankine-Cycle Engine", ASME Paper 73-WA/Sol-8, 1973.

23. "Assessment of the Rankine-Cycle for Potential Application to Solar-Powered Cooling of Buildings, Hittman Associates, Inc. Report HIT-581, prepared for NSF.

24. D. Prigmore and R. Barber, "Cooling with the Sun's Heat, Design Consideration and Test Data for a Rankine-Cycle Prototype," Solar Energy, Vol. 17, 185-192 (1975).

25. L. I. Stiel et. al., "Optimum Properties of Working Fluids for Solar Powered Heat Pumps" IECEC '75 Record, 171-177 (1975).

26. F. R. Biancardi, "Design and Operation of a Solar-Powered Turbo-compressor Air-Conditioning and Heating System", IECEC "75 Record, 186-194 (1975).

D. DESICCANTS

27. G. O. G. Löf, "Cooling with Solar Energy", Proceedings of the World Symposium on Applied Solar Energy, Phoenix, Arizona, November 1-5, 1955.

28. P. Lunde, "Solar Desiccant System Analysis and Materials", Workshop Proceedings, Solar Cooling of Building, February 6-8, 1974, Los Angeles, California, 139-144 (1974).

29. W. Rush, "Solar Desiccant Systems", Workshop Proceedings, Solar Cooling of Buildings, February 6-8, 1974, Los Angeles, California, 151-155 (1974).

30. Institute of Gas Technology Press Release, May 20, 1974.

31. W. F. Rush and R. A. Macriss, "Munters Environmental Control System", Appliance Engineer, Vol. 3 (3), 1969.

E. SOLAR COOLING DEMONSTRATIONS

32. "Design and Test Report for Transportable Solar Laboratory Program", Honeywell, Inc., Report Prepared for NSF Under Grant PTP74-01555, October, 1974.

33. J. C. Hedstrom, et. al., "Comparative Analysis of LiBr Absorption, Dehumidification and Solar-Rankine Cooling System Options for a 66,000 ft^2 Building in Los Alamos, New Mexico", LA-UR-74-1062.

34. "Yazaki Experimental Solar House One", Report by Yazaki Buhin Co., Ltd., Kosai, Shizuoka, Japan.

35. "Solar-Energy System Works Well at Numazu Assembly Hall", JARN, pg. 7, June 25, 1976.

36. "Solar Energy Recovery System, Progress Reports 1 and 2", Department of Mechanical Engineering of Ohio State University, 1975.

37. "Timonium Elementary School Solar Air Conditioning Experiment, Intirim Report for 1975 Cooling System", AAI Corporation Report ER-8566 to ERDA.

38. "Solar Heating and Cooling Experiment for a School in Atlanta", Westinghouse Electric Corporation #NSF-RA-N-74-122 (1974).

39. J. E. Hill and T. E. Richtmyer, "Retrofitting for Solar Heating and Cooling: The Design and Construction of the System", NBS Technical Note 892 (1975).

40. J. E. Hill et. al., "Initial Test Results for a Solar-Cooled Townhouse in the Mid-Atlantic Region", ASHRAE Transaction, Vol. 82, Part II, 1976.

41. "Solar Heating Systems Design Manual", ITT Fluid Handling Division, Bulletin # TESE-576, (1976).

42. D. S. Ward and G. O. G. Löf, "Performance of a Residential Heating and Cooling System", Colorado State University Report #NSF-RA-N-75-053, (1975).

43. D. S. Ward, et. al., "Cooling Subsystem Design in CSU Solar House III", Proceedings of Joint Conference 1976 of ISES and Solar Energy Society of Canada, Solar Heating and Cooling of Buildings, pp. 68-69.

44. W. R. Humphries and D. E. Melton, "Solar Residential Heating and Cooling System Development Test Program", NASA TM-X-64924, (1974).

F. OTHER REFERENCES

45. Phone conversation with William Rush, Institute of Gas Technology, November, 1976.

46. "Pacific Regional Solar Heating Handbook", Los Alamos Scientific Laboratory Report LA-6242-MS.

47. J. D. Balcomb and J. C. Hedstrom", Sizing Collectors for Space Heating", Sunworld, Number 1, July, 1976.

48. Phone conversation with Thurman Skibount, The Trane Company, November 16, 1976.

49. Phone conversation with Ken Cooper, York Division of Borg Warner, November 17, 1976.

APPENDIX A

SOLAR COLLECTOR TECHNOLOGY

1. INTRODUCTION

All the approaches to solar air conditioning require average heat input temperatures considerably higher than for space and water heating applications. As a practical matter, minimum temperatures of 180 °F - 220 °F are required for reliable system operation. Several approaches are being considered and, in some cases, used to improve collector performance at these higher temperatures:

- Selective coatings on the absorber plate to reduce reradiation losses.
- Heat traps (honeycomb type structures) to reduce both convection and reradiation losses.
- Partial vacuums in the space between the absorber and the transparent cover to reduce and/or eliminate convection and conduction losses.
- Solar concentration to increase the heat flux on the absorber.
- Cover plates which have high solar transmission by virtue of reduced absorption and/or anti-reflective surfaces.

Figure A-1 shows performance curves for collectors using various combinations of the above approaches for improving collector performance. The collector types most commonly selected for space and water heating (1 or 2 pane black) have very low efficiency at those temperatures required for air conditioning applications. On the other hand, collectors using combinations of concentration and/or convection suppression maintain a reasonably high efficiency at the higher temperature levels.

The curves of Figure A-1 are for a level of solar flux typical of clear day conditions one hour either side of noon time. The average efficiency of the solar collectors under actual operating conditions must account for the lower levels of solar insolation prevalent during early morning and late afternoon hours as well as partial cloud cover.

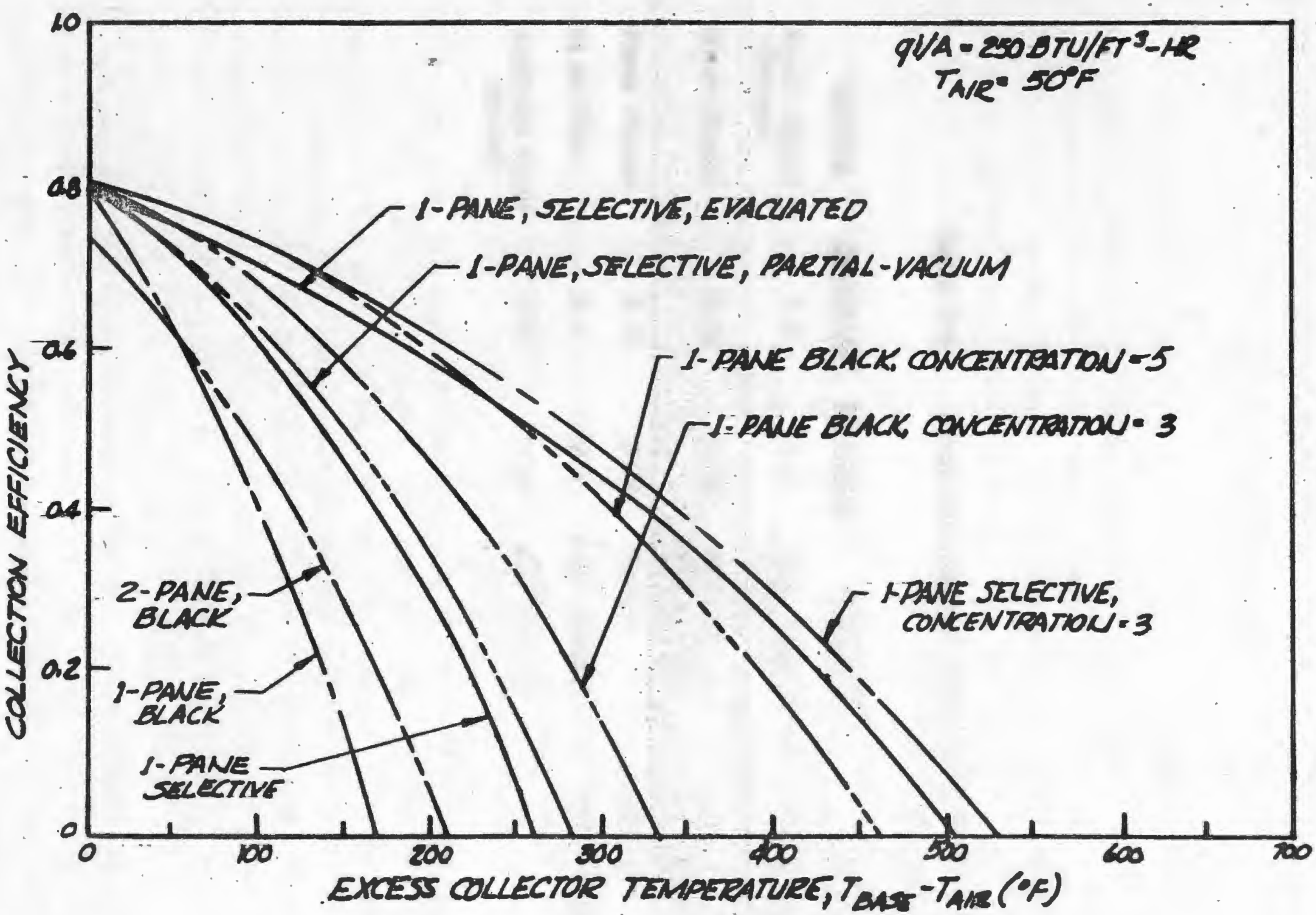

FIGURE A-1 PERFORMANCE COMPARISON OF VARIOUS CLASSES OF COLLECTORS

For stationary collectors, the daily average efficiency will ordinarily be about 30 - 50% lower than that obtained near solar noon. The difference between average efficiency and clear day efficiency at noon is accentuated in those areas of the country having less consistent solar flux conditions. As a result, it is critical to calculate the performance of a collector-cooling system combination on an annual basis using realistic solar insolation input data when evaluating solar cooling system options.

2. APPROACHES TO ACHIEVING HIGH COLLECTION EFFICIENCY

It is possible using relatively straightforward technologies to reliably achieve the temperature levels required for solar air conditioning using flat plate collectors and low level focusing collectors which don't require solar tracking. The present status of high performance collector configurations is outlined below:

a. Selective Coatings

As the collection temperature level increases, the absorber reradiation losses increase rapidly. Reradiation losses can be dramatically reduced by using a selective black absorber coating which has the characteristic of being a good absorber of short wave length solar radiation and a poor emitter of long wave length infrared energy. It should be noted that even evacuated collectors require such a coating to be effective.

The existance of selective coatings has been known for over 15 years, largely as the result of work in Israel and Australia where solar hot water heating has been used for some time. The development of selective coatings in the United States has progressed rapidly in the last few years primarily as a result of the increasing interest on the part of the government and industry in solar heating and air conditioning. A list of selective black coatings which have been used or are under development is given in Table A-1.

TABLE A-1 SELECTIVE SURFACE TECHNOLOGY

Coating	Absorption	Emittance	Durability	Experience
Black Copper (Ethene)	0.9	0.1	High Temperature Oxidation	Australia, copper substrate
Black Nickel	0.92	0.06	Moisture Destruction	Israel, galvanize iron
Black Chrome	0.95	0.26	Stable	Limited
Black Zinc	0.9	0.1	Not defined	Development
Aluminum Oxide (Alcoa)	0.94	0.33	Not defined	

The coatings which are of practical interest (i.e., low cost) are usually made by using chemical or electrochemical deposition techniques to form a thin layer of a semiconductor material on a smooth, low emissivity substrate. These layers are often metallic oxides such as those of copper, nickel, and titanium.

b. Heat Traps

One of the primary heat loss mechanisms from the absorber plate is by convection. Convection can be suppressed and/or eliminated by using a cellular structure placed over the absorber plate where the depth of the cells is deep compared to their width (5-10:1 aspect ratio). These cells have often been referred to as "honeycombs" due to a similarity in shape. In many of the more recent configurations the cells have been aligned in the horizontal plane only and take the shape of long east-west oriented slots.

Many materials have been used to fabricate cell walls including glass, plastics, and thin metal foils. The most common approach at this time appears to be the use of highly transparent plastic films which have the potential advantages of low cost, ease of fabrication, lightweight, and high transmission of solar energy. In addition to suppressing convection, heat traps also tend to reduce reradiation losses from the absorber plate.

The performance of a heat trap collector is expected to be intermediate between those using partial vacuum to eliminate convection and those in which no special steps are taken to reduce convection (see Figure A-1). Collectors using heat traps are now available from several manufacturers.

c. Vacuum

The most effective way to eliminate convection losses is to evacuate the space between the absorber and the transparent cover. There are two levels of vacuum which can be used:

Partial Vacuum:

A moderate vacuum level of 10^{-2} Torr is sufficient to eliminate free convection losses. However, there is still sufficient air available for conduction to occur in the air gap. Such collectors would have performance levels similar to those using heat traps.

High Vacuum:

If a relatively high vacuum of 10^{-4} Torr is used, all convection and conduction losses across the gap between the absorber and the cover are reduced to a level insignificant compared to other losses.

Several groups have worked on collectors using partial vacuum. The configurations involved typically used a flat plate configuration with spaced support rods between the absorber and the cover (usually plexiglass). The support rods were necessary to prevent collapse of the structure under vacuum conditions when the forces on the cover are over 14 psi. To date, none of the efforts on this type of configuration have resulted in commercially available collectors.

By far the most common approach to evacuation employes glass tubes as the vacuum housing. These tubes are similar to those mass produced for lighting fixtures and have sufficient strength to withstand an internal vacuum condition. Due to the "clean" structure involved and the relative ease of sealing the tube, evacuated tube collectors utilize a "high" vacuum which eliminate both convection and conduction losses.

3. TRACKING COLLECTORS

One of the major advantages of flat plate (or low level focusing) collectors is their ability to function with acceptable performance when mounted in a fixed position on the roof or wall of a building. This greatly simplifies the integration of the collectors with the building and reduces the cost of mounting the system over those arrangements requiring movement of the collectors.

However, there are significant advantages to using an arrangement whereby the collector can track the sun - at least in one plane:

- Tracking yields a significant increase (usually about 30-50%) in incident solar flux on each square foot of collector through the day over that associated with a collector mounted in a fixed position. The following additional advantages are obtained from tracking:

- reflection losses are decreased since the solar radiation is more normal to the transparent covers, and
- the average efficiency of collection is higher as a result of the higher flux availability.

• Solar tracking makes possible the use of higher levels of concentration than is possible using collectors mounted in a fixed position. For example, concentration levels of 5-20:1 would be readily obtainable with a north-south linear concentrator with east-west tracking.

Despite the clear cut thermal advantages of using solar tracking, few systems designed for heating or cooling now employ tracking. This is due to several factors:

• It is difficult to integrate a tracking arrangement into a building in an aesthetically acceptable manner.
• The structure required to support a tracking system is more complex (expensive) than that required for fixed systems.
• There are questions regarding the reliability of the motors, gears, and associated drive trains required to rotate the collectors. The reliability issue is complicated by the requirement for these mechanical components to operate under severe environmental conditions (dust, snow, etc.) in exposed locations.

4. MANUFACTURE OF ADVANCED COLLECTORS

There are over 100 firms now manufacturing solar collectors in the United States. Most of these collectors are designed for use at the relatively low temperatures required for domestic hot water and space heating. As, such their performance levels at higher temperatures are usually not adequate for effective solar air conditioning applications. However, several firms are making more "advanced" solar collectors using one or more of the approaches previously mentioned to improve high temperature performance. A list of several of these firms is shown in

Table A-2. Inclusion of a collector on this list is not, in itself, an assurance that the performance of the collector will be adequate for solar air conditioning applications. However, the claimed thermal performance of these collectors is better than that of the more conventional arrangements typified by the use of one or two layers of glass and a flat back absorber.

5. COLLECTOR SPECIFICATION FOR SYSTEM ANALYSIS

As previously indicated there are numerous technical approaches which can be pursued to make solar collectors with thermal performance levels consistent with operating heat actuated air conditioning systems. However, Figure A-1 indicates that the performance levels of most advanced collectors tend to be lower than that achievable with evacuated tube collectors such as those now manufactured (at least on a demonstration basis) by several firms. Based on the considerations of availability and excellent performance levels, evacuated tube collectors have been chosen as the baseline collector for purposes of the solar cooling system analysis. The detailed collectors specifications are:

Type - High Vacuum (10^{-4} Torr)

Glass Tube Characteristics:

- Absorption - 0.04
- Refraction Index - 1.5

Absorber Characteristics:

- Absorption - 0.90
- Emissivity - 0.10

TABLE A-2 MANUFACTURERS OF HIGH PERFORMANCE COLLECTORS

Company Name	Thermal Improvement
Sunworks	Selective Black Absorber
Daystar	Heat Trap
Solargenics	Selective Black Absorber
Northrup	Fresnel Lens Concentrator
Owens-Illinois	Evacuated Tube
Philipps	Evacuated Tube
Corning Glass	Evacuated Tube
SunSav	Heat Trap

TABLE IV-1 CLIMATE AND ENERGY COST DATA

City	Insolation at Surface (Avg. Langleys/Day)	Annual Insolation on Tilted Surface (10^6Btu/ft^2-year)[1]	Nominal Deg-Days		September, 1976 Energy Costs[2]			Real Energy Cost Escal. (%/Yr)		10 Year Infl-Disc. Factors[6]	
			Heating	Cooling	Electricity (¢/kWh)[3]	Oil ($/$10^6$Btu)[4]	Gas ($/$10^6$Btu)[5]	Elec.	Fuel	Elec.	Fuel
Bremerton, WA	273	.477	4,727	183	1.4	3.2	2.4	7.3	8	8.86	9.14
Chicago, IL	273	.437	6,127	925	4.5	2.8	2.0	5.6	8	8.21	9.14
New London, CT	339	.532	5,700	633	4.3	2.9	2.1	6.9	8	8.70	9.14
Norfolk, VA	382	.573	3,488	1,441	4.7	3.0	2.8	5.8	8	8.29	9.14
San Diego, CA	407	.590	1,507	727	4.4	2.7	2.1	7.3	8	8.86	9.14
Corpus Christi, TX	436	.610	930	3,474	5.8	-	2.0	7.5	8	8.94	9.14
Key West, FL	452	.624	64	4,888	4.1	2.9	1.8	5.8	8	8.29	9.14

[1] Estimated, collector tilted at latitude angle
[2] Source: U.S. Bureau of Labor Statistics
[3] Based on 500 kWh/month
[4] #2 fuel oil; #6 fuel oil costs about 80% of figure shown
[5] Based on 100 therms/month
[6] 10% real discount rate

NOTE: 10^6Btu = 10 therms = 1,000 cubic feet of gas
= 7 gallons #2 fuel oil
= 80 lbs. coal
= 293 kWhs

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