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PHASE I REPORT
SOLAR ASSESSMENT STUDY
IN SUPPORT OF THE
INTERNATIONAL EKISTICS PROGRAM

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ENERGY AND ENVIRONMENTAL SYSTEMS

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1.0 INTRODUCTION

Background:

As part of the International Ekistics Program, the Community Systems Branch of ERDA is working with the Greek Government to determine how both conventional and renewable energy sources can be used to reduce the consumption of conventional fuels in a new community being planned for the coast of Crete. This project is being carried out within the ATMES* program which is administered by the Argonne National Laboratory (ANL).

Program Goals:

Arthur D. Little, Inc. (ADL) assisted ANL in this first phase of the project which had as its primary tasks to:

- Identify general approaches for using solar, wind, and biomass energy sources (referred to as gentle energy sources due to their small environmental impacts) in a new community where the energy system can be preplanned to result in minimal energy consumption,
- Provide preliminary assessments of the economic/technical performance of the renewable energy sources as they would function in Crete, and
- Indicate the availability and cost of key components which would be used in the various system arrangements.

The overall program goal was to assist in the community planning process by indicating how unconventional energy sources can be used within a new community context. Those system options of most interest would have to be examined in detail once their possible role in the new community is defined.

The Phase I study described in this report was intended to provide preliminary information on the above subject areas within the scope of a 2-month program effort. The estimated cost of systems and components

* Advanced Technology Mixed Energy Systems.

assumed their use in the United States. The cost of implementing these system options in Greece may be somewhat different due to variations in material and labor costs.

The new community is not intended to be a scientific experiment. Therefore, those subsystems and components for which technical/cost performance characteristics are reasonably well defined were stressed in this Phase I study. However, since the project could take over 10 years to complete, several particularly promising longer range technologies (such as photovoltaics) were identified along with their possible role in the new community energy system.

Within a new community renewable energy sources can be used in two ways:

- On individual buildings such as with small residential solar hot water heaters, and
- Within an integrated energy system serving large segments of the community, for example, through combined power/heating systems (various forms of total energy).

It is expected that both types of energy systems will play a role in the community planning process and, therefore, the analysis of system options covered a corresponding size range. There are, however, several unique advantages associated with utilizing renewable energy sources within the context of a well-planned integrated community energy system concept:

- Many community energy system concepts would utilize centralized electric power and thermal (heating, chilled water, etc.) facilities for high building density areas. These relatively large energy systems will lead to substantial economies of scale for the equipment associated with the renewable energy sources. For example, the cost of wind turbines on a per unit capacity basis decreases substantially as capacity increases.
- Small systems on individual buildings need to be of very simple design and operation to be

practical. It is possible, to utilize more complex system arrangements within a community system context since personnel with special training can operate and maintain the equipment.

- The use of a central energy system makes it practical to combine renewable energy sources in overall system arrangements which combine directly the outputs of both conventional and unconventional means of generating power and heat. For example, hot water for a district heating arrangement can be heated simultaneously by engine exhaust gases and a solar collector array.

Figure 1.1 shows schematically the general program approach. Available climatic, and insolation data from Crete were used to make rough estimates for the heating/cooling loads which will occur on individual buildings and for the community as a whole. These data were considered in preparing "generalized community energy system concepts" with emphasis placed on systems using renewable energy sources. These concepts could include a mix of on-site energy sources for individual (or groups) of buildings as well as integrated energy arrangements serving large segments of the community.

There are myriad different community system concepts which can be considered using different mixes of renewable energy sources (solar, wind, ocean water) and conventional energy arrangements (diesel engines, waste heat boilers, etc.). However, most of these system arrangements use a relatively small number of basic energy subsystems (albeit arranged in a vast number of alternative arrangements and for different applications).

The preparation of specific community energy systems designs was outside the scope of this Phase I program; therefore, the generalized energy system concepts were prepared only in sufficient detail and numbers to identify the key energy subsystem which would be used as the building blocks for most of the energy system concepts.

FIGURE 1.1

GENERALIZED APPROACH FOR DEFINING ENERGY SYSTEMS AND SYSTEM COMPONENTS

The technical/cost performance of selected basic energy subsystem modules using renewable energy sources was estimated using both in-house analysis and outside information sources (manufacturers literature, ERDA reports, etc.).

The status of the important components (collectors, engines, storage units, etc.) comprising the various key energy subsystems was then re-reviewed to provide information on component operation, physical configurations, cost, availability, manufacturers, and maintenance requirements.

2.0 CLIMATIC CONDITIONS ON THE GREEK COAST

The evaluation of both conventional and unconventional (solar, wind, etc.) energy options requires a definition of the climatic conditions in the region of interest. Climatic conditions influence the choice of energy systems in two ways:

- The heating and air conditioning loads which must be met by the energy systems are dependent on climatic conditions,
- The operation of most "gentle" energy sources depend directly on climate (insolation, wind, etc.).

This section indicates the climate that is typical of the coastline of Greece. The data is somewhat incomplete due to the difficulty in obtaining detailed climatic data for Greece in the short time frame of this project.

The Greek coastline experiences a climate typical of the Mediterranean region. The summers are usually warm and dry and the winters are relatively mild, with most of the annual rainfall occurring during the winter months. Three areas for which some data was available that were selected to represent three different climatic regions of Greece were: Salonika, which represents a northern coastal climate; Athens, a central coastal climate; and Crete, a southern coastal climate.

2.1 Air Temperature Variations:

The average air temperature for most of the Greek coastline does not undergo large seasonal variations. This is partially due to the moderating effect of the ocean. The average temperature swing for the northern portion of Greece is typically from a mean low of approximately 49°F to a mean high of 78°F while the southern portion of Greece undergoes an average swing from 58°F to 78°F. Also presented in Table 2.1 is the monthly mean maximum and mean minimum for a location on the southern coastline of Crete. As indicated by the data in Table 2.1, the temperature variations for the Greek coastline are very modest. There are occasions when the temperature does drop below freezing (32°F) in the winter, especially in the northern section, and above 100°F in the summer, but it does not occur very often.

The mildness of the climate of the coastal area of Greece appears to lend itself to the use of heat pumps as a means of heating and cooling buildings. The humidity in Greece is relatively high all year-round ranging from an average of 74% in the winter to 54% in the summer.

2.2 Solar Insolation:

The solar insolation in Greece varies from an annual daily average of approximately 400 langleys in the southern sections (Crete) to 330 langleys in the northern sections (Salonika). The annual solar insolation in the southern sections of Greece is typical of that found in the San Francisco area while the insolation in the northern sections of Greece is typical of that found in the Boston area. The average monthly solar insolation levels for Athens is presented in Table 2.2. The insolation levels in Athens are typical of those found along the coast of West Virginia. The amount of cloudiness an area experiences is also important to the assessment of the potential use of solar energy. The data presented in Table 2.3 indicates the mean cloudiness expected along the southern coast of Crete. This data reflects the relatively high insolation levels which are experienced in the southern sections of Greece (Crete). This area appears to be a desirable location for the utilization of solar radiation as a source of energy.

2.3 Wind Speeds:

Many coastal areas tend to be windy. Greece is no exception with a reasonable amount of wind energy available along its coastline. The data which represents the wind energy available along the coast in the three sections of Greece described earlier is presented in Table 2.4. The intensity of the wind speed appears to remain fairly constant on a seasonal basis, particularly on Crete. Based on these relatively high and constant wind speeds, the use of wind power could be a viable option along the Greek coast.

2.4 Ocean Temperature Variations:

The temperature of the ocean remains fairly stable through the seasons. The mean temperature of the ocean varies by only 20°F from

winter to summer. Even in the winter, surface water temperatures are 60°F or higher off the coast of Crete. Daily variations are negligible due to the large heat capacity of the ocean water. The seasonal ocean surface temperatures are presented in Table 2.5. As indicated in Section 5.5, the year-round availability of warm ocean water could provide for the highly effective use of water source heat pumps for space and water heating in communities located on the coast.

The variation of the ocean temperature with depth is not reflected in Table 2.5. The limited data available on this subject indicates a temperature of about 55°F (15°C) at a depth of 100 meters. At this relatively moderate temperature, it is probably impractical to use seawater directly for space cooling applications.

TABLE 2.1

VARIATION IN AIR TEMPERATURE

<u>MONTH</u>	<u>MEAN AIR TEMPERATURE¹</u>			<u>CRETE AIR TEMPERATURES²</u>		
	<u>SALONIKA</u>	<u>ATHENS</u>	<u>CRETE</u>	<u>MEAN MAX</u>	<u>MEAN MIN</u>	<u>MEAN AVG</u>
	<u>°F (°C)</u>	<u>°F (°C)</u>	<u>°F (°C)</u>	<u>°F (°C)</u>	<u>°F (°C)</u>	<u>°F (°C)</u>
January	49 (9)	54 (12)	58 (14)	60 (16)	50 (10)	55 (13)
February	50 (10)	55 (13)	58 (14)	57 (14)	47 (8)	52 (11)
March	52 (11)	56 (13)	59 (15)	63 (17)	51 (11)	57 (14)
April	60 (16)	61 (16)	62 (17)	68 (20)	56 (13)	62 (17)
May	64 (18)	68 (20)	68 (20)	76 (24)	63 (17)	70 (21)
June	72 (22)	74 (23)	74 (23)	82 (28)	69 (21)	76 (24)
July	78 (26)	78 (26)	78 (26)	85 (24)	73 (23)	80 (27)
August	77 (25)	78 (26)	78 (26)	86 (30)	72 (22)	80 (27)
September	72 (22)	75 (24)	76 (24)	82 (28)	88 (20)	76 (24)
October	65 (18)	68 (20)	72 (22)	77 (25)	64 (18)	70 (21)
November	60 (16)	63 (17)	67 (19)	69 (21)	58 (14)	63 (17)
December	50 (10)	55 (13)	62 (17)	64 (18)	53 (12)	58 (14)

1. Data from "A Climatic Resume of the Mediterranean Sea", NAVAIR 50-16-64, November 1975.

2. Data from "Energy Systems and Design of Communities", by Environmental Design Company, Athens, Greece, E350-66, February 1977.

TABLE 2.2

AVERAGE DAILY SOLAR INSOLATION LEVELS FOR ATHENS

<u>MONTH</u>	<u>BTU/ft²-day (langleys)</u>
Jan	664 (180)
Feb	1018 (276)
Mar	1232 (334)
Apr	1686 (457)
May	1904 (516)
June	2129 (577)
July	2111 (572)
Aug	1838 (498)
Sept	1461 (396)
Oct	1018 (276)
Nov	690 (187)
Dec	594 (161)
Annual	1362 (369)

TABLE 2.3

AVERAGE MONTHLY CLOUDINESS ALONG THE SOUTHERN COAST OF CRETE ¹

<u>MONTH</u>	<u>MEAN CLOUDINESS</u> <u>(Tenths)</u> ²	<u>CLOUDY DAYS</u>
January	5.3	8.6
February	5.3	5.2
March	5.1	6.0
April	4.9	4.4
May	2.8	1.4
June	1.4	1.0
July	0.1	0.0
August	0.3	0.0
September	1.6	0.8
October	4.0	4.2
November	5.9	6.2
December	5.9	6.2
Annual	3.6	44.0

1. Data from "Energy Systems and Design of Communities", by Environmental Design Company, Athens, Greece, E350-66, February 1977.
2. This number when multiplied by ten is the percent of sky covered by clouds of any type at any altitude.

TABLE 2.4

AVERAGE WIND SPEEDS IN GREECE ¹

<u>MONTH</u>	<u>MEAN WIND SPEED (MPH)</u>		
	<u>SALONIKA</u>	<u>ATHENS</u>	<u>CRETE</u>
January	15	15	15
February	6	12	16
March	7	12	16
April	7	9	15
May	6	8	12
June	8	8	13
July	7	10	14
August	10	9	15
September	5	10	14
October	8	12	13
November	9	12	12
December	17	13	15

1. Data from "A Climatic Resume of the Mediterranean Sea", NAVAIR 50-16-64, November 1975.

TABLE 2.5

OCEAN SURFACE TEMPERATURES ALONG THE GREEK COASTLINE ¹

<u>MONTH</u>	<u>SALONIKA</u>	<u>ATHENS</u>	<u>CRETE</u>
	<u>°F (°C)</u>	<u>°F (°C)</u>	<u>°F (°C)</u>
January	55 (13)	58 (14)	62 (17)
February	48 (9)	58 (14)	60 (16)
March	55 (13)	58 (14)	60 (16)
April	58 (14)	60 (16)	62 (17)
May	61 (16)	65 (18)	67 (19)
June	72 (22)	71 (22)	72 (22)
July	76 (24)	76 (24)	75 (24)
August	76 (24)	77 (25)	77 (25)
September	73 (23)	75 (24)	76 (24)
October	69 (21)	71 (22)	73 (23)
November	64 (18)	68 (20)	69 (21)
December	58 (14)	61 (16)	65 (18)

1. Data from "A Climatic Resume of the Mediterranean Sea", NAVAIR 50-16-64, November 1975.

3.0 COMMUNITY ENERGY NEEDS

3.1 Energy Uses:

The energy needs of any community include some combination of heat and electricity to satisfy the following functions:

- **Comfort Conditioning:**
The heating and air conditioning of living and working spaces.
- **Domestic Hot Water:**
Heating of hot water used within residential, institutional and commercial buildings for general washing functions.
- **Electrical Equipment:**
Includes those components normally operated by electricity such as lighting, electric motors for fans, pumps, and machinery, refrigeration equipment, etc.
- **Process Water/Steam:**
Heating of water and steam for use in industrial processes.
- **Transportation:**
Shaft power to operate automobiles, farm equipment and trucks required for the transportation of goods and people.
- **Special Functions:**
 - desalination
 - irrigation pumping

The quantitative discussion of energy system options that satisfy the above functions, particularly those using renewable energy sources, requires making preliminary estimates of the size of the energy loads under consideration and how those loads may vary on a daily and seasonal basis. Since energy systems will be considered over the size range from those used on individual houses (solar hot water units, etc.) to total community systems (central wind turbine systems, etc.), energy loads are estimated both for individual buildings and for the total community. These are very rough estimates at this time since no community design now exists. These preliminary estimates are intended only to indicate which energy functions are likely to be the most

important and the probable magnitude of these loads so that the size range of equipment can be specified.

3.2 Building Loads:

The buildings that have been selected are representative of those which might be required in a new Greek community. The building energy loads are comprised of space and hot water heating, air conditioning, and electricity for lights and appliances. The specification of building construction is indicated in Table 3.1 and the resultant thermal parameters for calculating heating and cooling loads in Table 3.2. The heat loss coefficients used for residences (single and multi-family) are typical of those used for American buildings located in warmer climates where construction is less expensive and insulation is not a primary concern. The use of domestic hot water and electricity were assumed to be considerably less than what would be found in an American home. This is a reflection of the lifestyle and customs of the Greek population.

The hot water and electric needs of the commercial buildings were assumed to be the same as for well-constructed buildings in the United States serving the same function. Occupancy of most buildings is assumed to be typical of an American building of the same type. Except for the hospital, the buildings are not usually fully occupied 100% of the time; therefore, an occupancy factor has been used to reflect the true loads. This occupancy factor is based on the number of hours a week that the building is expected to be fully occupied; i.e., the office building will be used for approximately 8 hours a day, 5 days a week.

The average monthly variation in loads for several building types, assuming the climatic conditions typical of Crete, are shown in Figures 3.1 to 3.9. These loads are the raw energy loads based on building losses, climatic conditions, internal loads, hot water usage, and electric loads. They do not include burner efficiencies or COP's (Coefficient of Performance) of heating or cooling equipment.

These load curves indicate several important points which could affect the choice of energy systems for on-site use.

TABLE 3.1

DEFINITION OF PROTOTYPICAL BUILDINGS

<u>BUILDING TYPE</u>	<u>DESCRIPTION</u>	<u>NO. IN COMMUNITY</u>
Single Family Residence	One-floor dwelling (30'x50'x8'); basically masonry construction with little insulation and 15% fenestration of single pane glass.	375
Multi-Family Residence	Two-floor dwelling (50'x180'x20') consisting of 12 units; construction similar to single family except for 30% fenestration on side walls only of single pane glass.	125
Hotel	Five-floor building (50'x300'x60') consisting of 250 units; construction is typical of that currently used in the U.S. for commercial buildings of this type with 40% fenestration on end walls only of double-pane glass.	2
Hospital	Two-floor building (60'x250'x24); construction similar to hotel except for 15% fenestration all walls of double pane glass.	1
School	One-floor building (100'x400'x12'); construction is similar to hotel with 20% fenestration all walls.	4
Office Building	Two-floor building (90'x150'x20'); construction similar to hotel with 30% fenestration all walls.	3
Plant/Warehouse	One-floor building (200'x300'x18'); construction similar to hotel with 5% fenestration one wall only.	4

TABLE 3.1 (Continued)

<u>BUILDING TYPE</u>	<u>DESCRIPTION</u>	<u>NO. IN COMMUNITY</u>
Shopping Center	One-floor building (260'x260'x15') with central wall; construction is similar to hotel with 20% fenestration on south wall only.	1
Theater	One-floor building (60'x100'x35'); construction is similar to hotel with 40% fenestration one wall only.	1

TABLE 3.2

PARAMETERS¹ USED TO COMPUTE BASIC ENERGY USAGE

<u>BUILDING</u>	<u>HEAT LOSS COEFFICIENTS</u>				<u>OCCUPANCY</u>		<u>AVG. HOT WATER USE</u>		<u>AVERAGE ELECTRIC USE (kWh/day)</u>
	<u>Walls</u>	<u>Roof (BTU/m-°F)</u>	<u>Windows</u>	<u>Avg. Air Changes (Vol/hr)</u>	<u>No.</u>	<u>Factor</u>	<u>Gal/Person/Day</u>	<u>Temp (°F)²</u>	
Single Family	.4	.2	1.1	1	4	.7	20	125	10
Multi-Family	.4	.2	1.1	.25	36	.7	20	125	120
Hotel	.2	.1	.6	.5	500	.5	20	120	2000
Hospital	.2	.1	.6	3	300	1.0	45	160	3000
School	.2	.1	.6	2.3	1000	.24 ³	5	140	2520
Office Building	.2	.1	.6	1.25	270	.24	2	140	1620
Plant/Warehouse	.2	.1	.6	.25	1200	.24	5	140	5700
Shopping Center	.2	.1	.6	1	670	.43	.5	140	3690
Theater	.2	.1	.6	1.5	400	.25	1	120	250

1. Data from Solar Climate Control Project, A. D. Little, Inc., 1974.

2. Inlet Temperature is 55°F (13°C).

3. Only 9 1/2 months of the year.

Residential:

The residential space heating load is of relatively short duration and is rather low. This load could be further reduced from those indicated by use of (a) south facing windows which function as small passive solar heating units or (b) increased insulation. Based on the small magnitude and short duration of space heating loads, it is doubtful if active solar space heating systems would be an attractive option.

The hot water load is seen to be the largest single thermal load on a year-round basis and represents about one-third to one-half of the heating load even in the coldest months. Hot water heating, particularly in multi-family units, may be a good opportunity for solar energy applications.

The cooling loads indicated for residences is based on maintaining comfort conditions of 72°F in the buildings. As a practical matter, residential buildings are rarely air conditioned in Greece and this cooling load will probably not be satisfied (i.e., its a hypothetical load).

Institution, Commercial, Industrial:

The curves of Figures 3.3 through 3.9 indicate that there are no space heating loads in non-residential buildings. This is not quite true in practice since these curves were based on average temperatures. There will be abnormally chilly periods (particularly at night) when small amounts of space heating will probably be required. However, as a practical matter, the space heating loads on non-residential buildings are expected to be so low as to not constitute a significant load on the community energy system.

A substantial cooling load is indicated for all non-residential buildings. In many cases this cooling load will be satisfied by use of air conditioning equipment and, therefore, this air conditioning load could impose a significant peak load problem on the energy system.

Due to high internal loads, cooling is indicated for some buildings (for example, the hospital) even in the winter months. In energy

FIGURE 3.1 MONTHLY ENERGY LOADS FOR A SINGLE FAMILY RESIDENCE IN CRETE

FIGURE 3.2 MONTHLY ENERGY LOADS FOR A MULTI-FAMILY RESIDENCE IN CRETE

FIGURE 3.3 MONTHLY ENERGY LOADS FOR A HOTEL IN CRETE

FIGURE 3.4 MONTHLY ENERGY LOADS FOR A HOSPITAL IN CRETE

FIGURE 3.5 MONTHLY ENERGY LOADS FOR A SCHOOL IN CRETE

FIGURE 3.6 MONTHLY ENERGY LOADS FOR AN OFFICE BUILDING IN CRETE

FIGURE 3.7 MONTHLY ENERGY LOADS FOR A PLANT/WAREHOUSE IN CRETE

FIGURE 3.8 MONTHLY ENERGY LOADS FOR A SHOPPING CENTER IN CRETE

FIGURE 3.9 MONTHLY ENERGY LOADS FOR A THEATER IN CRETE

conserving HVAC design, when the outside temperature is 10-15°F below the desired inside temperature the system would be designed to utilize cool outside air in a ventilation cycle. As a result, the air conditioning loads indicated during cooler months will be greatly reduced from those indicated.

3.3 Community Definition:

The size and type of central power generating facilities, district heating and cooling arrangements, and community layout could depend critically on total energy loads within the community. Preliminary indications of the total community loads were made assuming a permanent population of 2,500 people and the following mix of building types:

- Residences are 20% multi-family and 80% single family*,
- Hotels for 400 tourists
- Facilities to accomodate tourists' needs (shopping, theaters, etc.),
- Schools for approximately 650 pupils, and
- Light industry to support the population.

This building inventory reflects the objective of having a combination of agricultural (with attendant light industry) and tourist related commercial activities.

It is desirable in a community which is being considered for a total energy system concept to locate many of the buildings in close proximity to the power generating plant and to each other to facilitate electrical and heat energy distribution. This would imply having a central section in the community which would consist of the multi-family residences, schools, commercial buildings, and power generating station(s), with outlying areas being used for agricultural and single-family residential purposes.

* Typical of a rural community in Greece. There is, however, an incentive to use a larger fraction of multi-family housing if a community energy system concept is to be considered. The mix of single to multi-family dwellings will not have a large impact on total loads but will effect the cost of an energy distribution system.

It is expected that this type of system layout would be considered for a new community utilizing preplanned community energy systems. Since a seaside community is under consideration, there is also an incentive to have the high load density area located on the coastline to take advantage of the thermal energy which can be extracted from warm seawater.

3.4 Community Energy Loads:

The total energy requirements of the community can be satisfied by supplying one or a combination of two basic types of energy, thermal or electric. Space and hot water heating can be supplied as thermal energy or as electric resistance energy. Air conditioning (cooling) can also be provided by using either thermal energy or electric energy. Whichever form of energy is chosen, there is an efficiency (or COP) which is associated with the load requirements. The COP's that were used for this study are listed in Table 3.3. These are typical of commercially available equipment. The total thermal and electric energy requirements for the community, using the referenced COP's, is shown in Figures 3.10 and 3.11. The drop in load requirements during the summer months is a result of the schools closing for the summer months. The two figures indicate loads based on two substantially different assumptions regarding the nature of the overall energy system.

Figure 3.10 indicates the total thermal loads assuming all air conditioning is provided by heat actuated air conditioning systems, such as absorption units. The use of absorption units is quite common in total energy systems using hot engine exhausts as the heat source. Their use is also consistent with solar cooling system operation.

Figure 3.11 indicates electric loads assuming that heating and cooling loads are satisfied by electricity. The combined load shown would, therefore, be consistent with an all electric community concept. In these systems, heating was assumed to be electric resistance. The heating loads could be reduced by a factor of 2-4 by use of heat pump arrangements.

TABLE 3.3

COEFFICIENT OF PERFORMANCE (COP) OF SOME CONVENTIONAL ENERGY SYSTEMS

	<u>COP</u>
Electric Hot Water Heater	.9
Absorption A/C (Thermal)	.7
Vapor-Compression A/C (Electric)	3.0

FIGURE 3.10 TOTAL MONTHLY THERMAL LOADS FOR A COMMUNITY IN CRETE

FIGURE 3.11 TOTAL MONTHLY ELECTRIC LOADS FOR A COMMUNITY IN CRETE

3.5 System Sizing:

The community loads shown in Figures 3.10 and 3.11 can be used to roughly estimate the capacity of centralized electric and thermal systems designed to meet a large portion of the community loads. These capacities can, in turn, indicate approximate sizes of solar thermal, photovoltaic, and wind energy systems which might be considered for use in centralized community energy systems.

Electric Load:

The base electrical load indicated in Figure 3.11 is consistent with an average power output of about 0.9 megawatts. Peak power demands could be several times this value. This modest amount of power can be satisfied by combinations of conventional Diesel or gas turbine generators and unconventional energy sources. For example, if solar thermal power systems are used to supplement conventional generating facilities during the day (no storage), the land area requirements for the collector field would be 2-5 acres; depending on the efficiency of the system considered. The land area requirements of a solar thermal power system appears, therefore, to be consistent with the area availability along the coast of Crete. Also, the ERDA program in wind turbines emphasizes the large systems with 100 to 1500 kW of output. A substantial portion of the electric power requirements of the community could be satisfied using 1 or 2 of the larger wind turbine systems once they are developed.

Heating Loads:

The hot water heating load is seen to be the largest thermal load on an annual basis. The average DHW load would require a heating rate of roughly 2×10^6 Btu/hr if a central hot water system is utilized.

Hot water heating is also one of the most promising applications of solar energy. A high percentage of the community hot water needs could be satisfied with a collector array about 1.5 acres in size (requiring approximately 3 acres of land). Again, it appears that these areas are reasonable and could be considered for use in the new community.

4.0 ENERGY SYSTEM OPTIONS

The energy needs indicated in Section 3.0 can be satisfied by a mixture of electrical power supplied by a central utility and the on-site combustion of fuels in water heaters, boilers, and engines. Two alternative extremes are, however, possible:

- There is the possibility of creating an "all electric" society where all functions are performed by electricity produced by large centralized facilities. In such a system even transportation could be electrified by use of battery driven cars and space conditioning by use of heat pump systems.
- The "total energy" approach where all needs are serviced by fuels burnt within the community itself; primarily in engine driven generators with waste heat recovery.

The first option (all electric) could be viable if nuclear energy or some very inexpensive energy (hydropower) source is available which can only be used in large centralized facilities. However, if conventional fuels are being consumed, the low overall thermal efficiencies associated with a large power plant are becoming an increasing source of concern. Most community energy systems employ "total energy" concepts to some degree in order to make better use of the fuels used in the electric power generation process. The reasons for this are shown in Figure 4.1. As indicated, the efficiency of a power plant in converting heat into electric power is usually 30-40%. The remaining 60-70% of the energy value contained in the fuel is rejected (exhaust gases and water jacket heat, in a diesel engine).

This reject heat can be recovered in waste heat boilers and used for:

- space heating,
- absorption air conditioning units,
- domestic and/or process water heating,

FIGURE 4.1

ENERGY FLOW IN A DIESEL POWER PLANT

- special functions,
 - desalination
 - drying

A typical energy system using only conventional subsystems is indicated in Figure 4.2. In this arrangement, electricity is produced by a diesel power plant and the reject heat is used to satisfy the space and hot water needs of a high density portion of the community by means of central distribution systems. Solid wastes from the community are used to operate an incinerator with waste heat recovery, which also serves the heating requirement of specific building clusters. In the arrangement shown, outside utility power is also used to supplement that produced by the diesel generators. Conventional heating systems service the buildings in low density areas.

There are many ways in which renewable energy sources can be introduced into the conventional energy system mix indicated in Figure 4.2. The unconventional energy sources of primary interest in this study are:

- solar
 $\begin{matrix} \swarrow \text{thermal} \\ \searrow \text{photovoltaic} \end{matrix}$
- wind
- ocean (stored heat and "cold")
- Biomass (municipal solid wastes and agricultural wastes)

Figures 4.3 through 4.8 indicate a variety of community system arrangements using a combination of conventional and renewable energy sources. The arrangements shown are intended only to indicate some of the basic subsystem arrangements which might be considered and not necessarily to suggest specific arrangements for the new community in Crete.

In these schematics the energy systems service a "load" which can be some combination of:

- All of the buildings within the community

FIGURE 4.2

COMMUNITY ENERGY SYSTEM CONCEPT USING CONVENTIONAL ENERGY SOURCES

- Clusters of high population density buildings.
- Individual buildings - particularly larger buildings such as schools, hotels, hospitals, etc.

These energy system arrangements are discussed briefly below.

4.1 Combined Solar Thermal/Diesel Electric Power Plant

The schematic of Figure 4.3 indicates that the exhaust heat from a Diesel or Gas Turbine power plant can be used to operate a waste heat boiler which provides heat for the operation of an organic Rankine cycle (ORC) engine. The ORC produces an additional 20-40% of power with no additional fuel consumption. This arrangement is being given considerable attention by ERDA and industrial companies and is referred to as the "bottoming cycle" concept.

The use of solar derived heat fits in very nicely with this arrangement. Heat from a solar collector field could be used to supplement the heat from a Diesel engine in operating the organic Rankine cycle engine. Depending on system sizing, the ORC could be operated with solar heat during sunny periods and by the Diesel engine exhaust during cloudy and/or particularly high load periods. In this way, the use of conventional fuel sources could be minimized without the requirement for large storage capacity.

This arrangement is also consistent with using the condenser waste heat from the ORC to heat water for space and domestic hot water applications which could be distributed to high building density areas in a "district heating" arrangement.

4.2 Combined Wind Turbine/Diesel Power Plant

This arrangement utilizes a Diesel or Gas Turbine power plant with a waste heat recovery boiler as the primary energy source. The heat from the boiler is used for district heating and/or cooling.

FIGURE 4.3

EXAMPLE OF COMBINED CONVENTIONAL/UN-CONVENTIONAL
COMMUNITY ENERGY SYSTEM

FIGURE 4.4

COMBINED WIND TURBINE/DIESEL POWER PLANT

Wind turbine generator systems operate in parallel with the engine driven generator and supply a portion of the electrical energy requirements on an "on-line" basis. In such an arrangement the need for electrical energy storage could be minimized since the Diesel generators can supply power on demand.

The percent of the load satisfied by the wind turbine will depend on the detailed nature of wind regimes, time variations and magnitude of electric loads and requirements for waste heat from the engines.

4.3 Solar Augmented Central Heating/Cooling System

Solar collectors can be used to operate a central heating/cooling system as shown in Figure 4.5. In this arrangement, a thermal storage volume would be heated by solar energy. Heat from storage can be used directly for space and water heating. This heat can also be used to drive absorption air conditioning units.

In the schematic shown, the solar heat is augmented by a solid waste incinerator unit which is provided with a waste heat recovery system.

4.4 Seawater Heat Source Heat Pump System

Space and/or hot water heating can be provided by operation of a heat pump using warm seawater as the heat source. Seawater temperatures off the coast of Crete are sufficiently high that this arrangement requires considerably less energy than more conventional flame fired options. In the system shown, the heat pump is directly driven by a Diesel engine which further minimizes energy consumption since the exhaust heat from the Diesel is also utilized.

4.5 Solar Augmented Desalination

On the Greek Islands, potable water is often a problem. The use of a community arrangement allows for operation of efficient conventional desalination units using a combination of solar and waste heat sources. The system indicated (Figure 4.7) shows a multi-stage flash distillation unit operated with solar heat and the waste heat from an engine driven generator system.

FIGURE 4.5

SOLAR AUGMENTED CENTRAL HEATING/COOLING SYSTEM

FIGURE 4.6

SEAWATER HEAT SOURCE HEAT PUMP SYSTEM

FIGURE 4.7

SOLAR AUGMENTED DESALINATION

4.6 Photovoltaic Electric Power System

The projected cost of solar cells in a 5 to 10 year time frame could make their use competitive in a community energy system. Figure 4.8 shows a photovoltaic power system augmenting the output of a Diesel power unit. The combined use of photovoltaics and a Diesel power plant could reduce storage requirements and insure a better match between thermal and electrical loads on the Diesel generator.

FIGURE 4.8

COMBINED PHOTOVOLTAIC/SOLAR THERMAL ENERGY SYSTEM

5.0 IDENTIFICATION OF KEY SUBSYSTEMS

The system schematics of Section 4.0 indicate various approaches for using thermal and mechanical equipment to satisfy the energy needs of a community. These system arrangements are, in turn, assembled from a relatively small number of key subsystems which can be interconnected in different combinations to create a large number of overall energy system options. The cost/technical performance of these key subsystems allows for estimating the performance of most of the various options for community energy system concepts.

The community energy systems will include many conventional subsystems such as Diesel engines, waste heat recovery boilers, etc. In this program the use of renewable energy sources is being stressed so that the subsystems discussed in this section all utilize solar, wind, or biomass energy forms.

For purposes of this study, the subsystems of Table 5.1 were identified for analysis.

There are a surprisingly small number of basic modules required to assemble a large variety of energy systems. For example, the solar hot water module discussed in Section 5.3 can be used for many functions including:

- domestic hot water,
- space heating,
- air conditioning (with absorption or desiccant units), and
- desalination.

In all these applications, the solar hot water module delivers hot water at an energy cost determined by:

- equipment costs,
- operating temperature,
- climatic conditions,
- load factor,
- interest rates,

- maintenance requirements, and
- depreciation.

This cost can be compared to that of delivering heat with conventional energy systems using gaseous or liquid hydrocarbon fuels. To a first approximation, the economic viability of the solar hot water module should be independent of the specific application in which it is used for equivalent delivery temperature and load factor conditions. For example, the economics of providing solar heat for air conditioning and desalination with a low temperature multi-stage flash distillation system should be roughly equivalent since:

- both applications require hot water at a temperature in the 180-225°F range, and
- both systems could have high load factors.

The community planner can, therefore, consider the hot water module as a "heat source" which delivers hot water to a function of his choosing at a specified cost depending on operating temperature, location, and load factor.

Similar arguments apply to the other system modules. The energy form delivered by each of these modules is indicated in Table 5.1, along with the cost parameter used to indicate its economic characteristics.

All the basic subsystems can be assembled from a variety of components (collectors, storage volume, heat engines, etc.) each with their own technical/cost parameters. In the following section, the cost/performance characteristics of the basic modules are indicated assuming component configurations which are either commercially available or in an advanced commercial development stage. It is expected that the operational characteristics indicated will be representative of those achievable in the near term. The resultant technical/cost parameters should, therefore, be a good guide in determining the viability of the subsystems for use in the new community.

Analyses are provided as a function of system size so that the use of these subsystems to service individual buildings as well as a substantial portion of overall community energy requirements can be ascertained.

TABLE 5.1

KEY ENERGY SUBSYSTEMS

- Solar Thermal Power Module
 - Energy Output Form - Shaft power
 - Performance Rating - ¢/kWh
- Wind Turbine Module
 - Energy Output Form - Shaft power
 - Performance Rating - ¢/kWh
- Hot Water Module
 - Energy Output Form - Hot Water (or Steam)
 - Performance Rating - $\$/10^6$ Btu
- Hot Air Module
 - Energy Output Form - Hot Air
 - Performance Rating - $\$/10^6$ Btu
- Ocean Source Heat Pump Module
 - Energy Output Form - Hot Water
 - Performance Rating - $\$/10^6$ Btu
- Photovoltaic Power Module
 - Energy Output Form - Electricity
 - Performance Rating - ¢/kWh
- Incinerator with Waste Heat Recovery Module
 - Energy Output Form - Hot Water/Steam
 - Performance Rating - $\$/10^6$ Btu

5.1 Solar Thermal Power Module

There are many approaches to solar thermal power being investigated. These include both high temperature systems (600-1800°F) which require high levels of solar concentration and low to medium temperature (200-600°F) systems which can utilize low levels of solar concentration. Several system options under consideration are listed below.

High Levels of Solar Concentration (400-1500X):

- Parabolic Dish with open Brayton cycle engine.
- Parabolic Dish with Stirling engine.
- "Power Tower" system with steam turbine
- "Power Tower" system with open Brayton cycle engine.

Medium Levels of Solar Concentration (10X-400X):

- Parabolic trough with Rankine cycle engine.
- Linear slot concentrator.
- Linear fresnel lense.

Low Levels of Solar Concentration (1X-10X):

- Flat plate collector.
- Compound parabolic concentrator.
- Planar reflector with flat plate collector.

These various collector options are discussed in Section 6.0. For purposes of this study the following system was assumed: (See Figure 5.1.1)

1. A collector array which uses Compound Parabolic Concentrators (CPC) with modest levels of solar concentration (3X to 10X) combined with an evacuated tube receiver;
2. A Rankine cycle engine using an organic working fluid.

Several advantages of this arrangement are:

- The CPC collector can utilize radiation over a relatively wide acceptance angle. This eliminates the need for accurate tracking and allows the

FIGURE 5.1.1

SOLAR THERMAL POWER MODULE SCHEMATIC

utilization of a large fraction of the diffuse radiation. Diffuse radiation represents over 25% of the total on an annual basis, even in desert areas of above average insolation. At coastal sites the humidity levels tend to be quite high, which is further incentive to use a collector which is not particularly sensitive to the spatial distribution of solar radiation.

- Both the collector arrangement and the engines are in an advanced state of development and their potential performance levels and cost parameters are well established. Therefore, the proposed system represents a low-risk approach to solar power generation.
- Both collectors employing seasonal adjustments and those which follow the sun from east to west can be considered. If east-west tracking is used for a more even power output during the day, the accuracy of tracking is not critical; therefore, inexpensive, rugged gears, motor drives, and controls can be utilized.
- The Rankine cycle engine can operate at high efficiency (up to 55-65% of Carnot) in the temperature range of primary interest (300-600°F), resulting in high overall system efficiency levels.
- Compared to more conventional focusing systems, the proposed collector system will have less-stringent requirements on tracking accuracy, surface contour precision and specular reflectivities, and will be less sensitive to the degree of collimation of the insolation. Hence, its potential for achieving and maintaining the required performance, at various geographical locations, appears superior.

- The external appearance of the collector arrangement under consideration is similar to that of a flat plate collector. The highly reflective surfaces are protected from the environment by an outer glass (or clear plastic) cover plate and there are no trough or spherical shapes to collect dust, sand, etc.
- The collector array has a low vertical profile which minimizes structural requirements. Site preparation will be minimal since accurate leveling and adjustment of the collector support structure is not required (as tends to be the case for highly concentrating systems).

The fact that many different system arrangements using various levels of solar concentration are being considered for solar power generation indicates that it is still difficult to tell which systems will prove most viable. However, the system chosen for consideration appears to have significant operational advantages and is sufficiently far advanced that cost estimates can be made with some degree of confidence. The performance of this arrangement should be representative of that obtainable from solar thermal power units of small to medium size (1-5,000 kW) which makes it a reasonable choice to serve as a basic subsystem module in a small community.

System Operation:

During sunny periods solar energy heats a heat transfer fluid (pressurized water or a heat transfer oil) flowing through the collector bank. This fluid, in turn, heats the "boiler" of the organic Rankine cycle engine thereby producing hot, pressurized vapor used to drive the organic Rankine cycle engine. (The operation of these engines is described in more detail in Section 6.4.)

The output of this system is shaft power which can be used to:

- Drive pumps for irrigation and desalination units,
- Drive compressors for air conditioning, and
- Drive generators for electric power generation.

System Efficiency:

A critical technical factor influencing the economics of this system arrangement is the overall efficiency of the system which, in turn, is related to operating temperatures, solar intensity, etc. System efficiency is important because it determines collector area requirements for a given net power output. There are structural plumbing, wiring, and maintenance costs directly related to collector area which significantly affect installed system costs. Efficiency and power output estimates were made for the following solar flux conditions:

- System efficiency was calculated at solar noon assuming a solar flux of 300 Btu/hr-ft^2 . This insolation level is consistent with clear sky conditions. Most systems would be rated under these conditions of high solar flux.
- System power output was calculated for a clear sky during July 21st. The calculation indicates the variation in power output through the day assuming no thermal storage is utilized.
- Preliminary estimates were made of annual power output for load factors of 1.0 and 0.5. The annual power output allows for making an economic evaluation of this system for different applications.

The efficiency of the solar collector configuration assumed for this study is shown in Figure 5.1.2. The evacuated tube collector with concentration maintains high efficiency levels for temperatures in the 300-600°F range. The efficiency of a "good" flat plate collector using a selective absorber coating is also shown for comparison purposes.

Estimates of engine efficiency with peak cycle temperature are shown in Figure 5.1.3 for several candidate working fluids. Engine efficiency calculations were based on the following component efficiency levels:

FIGURE 5.1.2

VARIATION IN COLLECTOR EFFICIENCY WITH TEMPERATURE

FIGURE 5.1.3

RANKINE CYCLE ENGINE EFFICIENCY - VARIATION WITH PEAK CYCLE TEMPERATURE

- Expander Efficiency - 75%.
- Feedpump Efficiency - 75%.

These component efficiencies have been achieved on equipment in the United States and are considered to be representative of those achievable on equipment designed for this application. R-114 was assumed as the working fluid for lower-temperature systems (200-400°F), while higher-temperature systems were assumed to use Benzene or Toluene as the working fluid. In all cases, a condenser temperature of 80°F was used which assumes a once-through water-cooled condenser. This assumption is valid for water pumping applications and may be valid for power generation applications if seawater cooling can be used.

The collector efficiency curves of Figure 5.1.2 and the engine efficiency curves of Figure 5.1.3 can be combined to yield overall system efficiency defined as:

$$\eta_s = \eta_c \times \eta_e$$

where

η_s = system efficiency,

η_c = collector efficiency,

η_e = engine efficiency.

Curves showing the resultant system efficiencies at midday are presented in Figure 5.1.4. The peak noontime efficiency of the system is about 14.1% and occurs at a peak cycle temperature of 450°F.

The efficiency curves of Figure 5.1.4 do not specifically take into account heat exchanger temperature drops, which would occur between the collector (or storage) heat-transfer loop and the engine working fluid. On the other hand, the average temperature at which heat is added to the engine is less than the peak cycle temperature. For example, heat is added over a temperature range from 300°F to 450°F for the engine using Benzene. The reduced average heat input temperature is reflected in a lower average collector temperature. The negative effects of heat-exchanger temperature drops, and the positive effects of a variable

FIGURE 5.1.4

OVERALL SYSTEM EFFICIENCY

engine heat input temperature, are assumed to roughly cancel each other out for purposes of this preliminary comparative analysis.

The daily variation in power output for the specified thermal system option is shown in Figure 5.1.5. For a clear day in July, the net energy production of this unit (with a fixed collector array) would be about 7.1 kWh per kW of peak generating capacity.

System Costs:

The cost of the solar thermal power module as a function of peak capacity is summarized in Figure 5.1.6 assuming collector costs are \$8/ft². This cost number is consistent with that projected by manufacturers should their use for air conditioning, heating, and power applications justify significant production quantities.

Collector costs do not include structural or piping costs required for collector installation. As indicated by Figure 5.1.6, installation is a significant portion of overall system cost.

The larger units of 100 kW or higher are seen to have a cost of about half that of small units which might be used on individual residential structures (1-2 kW). This is a further incentive to consider community concepts when solar power options are included in the energy mix.

Energy Storage:

Energy storage was not considered in the system costs of Figure 5.1.6. The kW indicated is a peak kW which occurs only during a short period in the middle of the day. It is, therefore, not possible to directly compare the costs of Figure 5.1.6 with the cost of a conventional power system capable of 24-hour output.

As a practical matter, most systems would require some amount of storage. For those higher temperature systems, storage could be in the form of a hot heat transfer oil/rock bed combination (See Figure 6.7.1). Based on previous studies on solar thermal power options, the use of storage has little effect on system economics. The increased system

FIGURE 5.1.5

DAILY POWER OUTPUT (On a Clear Day)

FIGURE 5.1.6

COST OF SOLAR THERMAL POWER MODULE

costs resulting from storage (and increased collector area required to charge storage) tend to be balanced by an increased capability to generate power.

System Economics:

The economic performance of the solar power unit is a function of many variables including:

- Initial Cost,
- Operation and Maintenance Costs,
- Interest Rates,
- Depreciation (useful life), and
- Load Factor.

The curves of Figure 5.1.7 assume the initial cost indicated in Figure 5.1.6 and,

- Annual Fixed Charges - 12%,
- Annual O&M Cost - 5%.

Curves are provided for two load factors:

Case A: Load Factor = 1.0

This case assumes that the power unit is used on a year-round basis. Such applications could include electric power generation and desalination.

Case B: Load Factor = 0.5

This case assumes that the system is used only half the year. This may be the case for air conditioning and irrigation pumping applications.

The cost of power produced by the system is seen to be in the 16¢/kWh to 34¢/kWh range for systems with a high load factor. This cost of power is still higher than from Diesel or Gas Turbine engines but may approach that from conventional units if the anticipated increases in fuel costs occur.

FIGURE 5.1.7

COST OF ELECTRIC POWER FROM SOLAR THERMAL POWER SYSTEMS

This cost of power increases dramatically for those systems which are not utilized on a year-round basis. This further indicates the importance of proper choice of application when using solar power units.

Combined Solar/Engine System:

As indicated in Section 4.1, the organic Rankine cycle engine can receive heat from both a solar collector array and the exhaust gases of a Diesel or Gas Turbine engine. When this is done, the cost of the ORC can be charged off against the conventional engine and the only incremental cost associated with the solar unit is that associated with the collectors themselves.

The resultant cost of solar derived power from this combined unit is also shown in Figure 5.1.7. In this case, the use of solar energy results in a power cost of about 10¢/kWh for larger units; a cost number which is becoming increasingly competitive with Diesel engine derived power.

5.2 Wind Power Module

The climatic data of Section 2.0 indicates that the wind conditions of the coast of Crete should be quite favorable for the use of wind energy (with average wind speeds being about 14 MPH). Due to the hilly terrain of the south coast of Crete, there may be local conditions even more favorable than that indicated above. For this reason, the use of wind turbines both on a large scale and for small on-site applications (water pumping, etc.) should be seriously considered.

The basic wind power module is shown schematically in Figure 5.2.1. In this arrangement a horizontal axis wind turbine is used to drive a generator which is directly connected (possibly via a gearbox) to the turbine shaft. The output of the wind turbine system is, therefore, electrical energy. This output can, in turn, be used for many functions, both large and small, within the community including:

- Tie-in with the electric grid to supplement power produced by conventional sources. This arrangement would require

FIGURE 5.2.1

WIND POWER MODULE

careful control of the quality (frequency, waveform, voltage, etc.) of the wind turbine module output power.

- To drive heat pump systems for air conditioning and heating.
- To operate pumping systems for irrigation and desalination systems.

These applications are indicated schematically in Figure 5.2.2. If the power is used as a supplement to the electric grid, any storage would probably have to be in the form of batteries which would significantly increase costs. With the other applications indicated (heat pumps and water pumps) no storage of electric energy would be required since storage could be in the form of chilled or hot water, or a water reservoir. These latter applications may, therefore, be more economically attractive.

System Costs and Economics:

The cost of commercially available units (as discussed in Section 6.3) are presented in Figure 5.2.3.

Typical of mechanical systems, this cost per unit capacity tends to decrease as the capacity increases. The largest commercial units are presently in the 14 kW range. Also shown are cost projections made by ERDA for their larger developmental systems (100-200 kW). These costs are seen to be considerably lower than for the smaller systems, which is one reason ERDA is stressing the development of large systems.

The costs indicated are for the turbine, tower, generator and associated controls and, therefore, correspond to the basic module indicated in Figure 5.2.1.

Economic Considerations:

The cost of electric energy from a wind turbine module depends on several factors, including:

- Equipment cost,
- Maintenance requirements,

FIGURE 5.2.2
APPLICATIONS OF THE WIND POWER MODULE

FIGURE 5.2.3

COST OF WIND GENERATORS

FIGURE 5.2.4

COST OF ELECTRIC POWER FROM WIND POWER MODULES

- Load Factor,
- Wind Speed Variations,
- Interest Rates, and
- Useful Life.

These costs were estimated for a wind turbine system operating in Crete, using the wind speed data in Section 2.0 , the equipment costs of Figure 5.2.3, and

- Capital costs (interest, depreciation, etc.)	12%
Operational and Maintenance Costs	3%
Load Factor	1.0

The load factor refers to the percentage of the year during which the output of the wind turbine is effectively utilized. A load factor of 1 is consistent with the year-round generation of electric power or operation of desalination units. Irrigation pumping may, on the other hand, only be necessary for 4 or 5 months of the year which will adversely effect system economics.

The power cost estimates of Figure 5.2.4 indicate that the smaller units would produce electricity for about 10¢/kWh. The larger units are seen to have the potential for reducing this cost to 5-7¢/kWh.

If localized areas are identified with considerably higher average wind speeds than the average 14 MPH assumed in Figure 5.2.4, these cost numbers may be further reduced.

5.3 Solar Hot Water Module

Figure 5.3.1 shows a schematic of a solar water heating module. The basic components of this system are:

- a. Solar collector panels
- b. Piping, valves, pumps, etc.
- c. Differential temperature controller
- d. Storage tank
- e. Heat exchanger (only required in freezing climates)

The solar hot water module is probably the most versatile of the

FIGURE 5.3.1
SOLAR HOT WATER MODULE

renewable energy source building blocks and the most likely to find widespread use in a community energy system in the near term. This system could be used in many functions including:

- Space Heating
- Domestic Hot Water Heating
- Operation of Absorption Air Conditioning Units
- Operation of Multi-stage Flash Distillation Desalination Units
- Process Hot Water
- Operation of Low Temperature Organic Rankine Cycle Engines

As indicated in Section 6.1, there are numerous collector configurations which could be utilized in this system. For low temperature (90-150°F) systems, conventional collectors using two transparent covers and a flat black absorber could be utilized. For higher temperature applications (150-250°F), evacuated tube collectors with selective absorber coating may be more viable. Temperatures higher than 250°F can be attained using low levels of concentration. However, most non-power applications of solar heat can be satisfied at lower temperatures achievable with flat plate collector technology.

The cost and performance of the hot water module will be estimated based on two basic collector configurations which are now available.

Collector A: Conventional Flat Plate Collector

Two Transparent Cover Plates, ($\tau = 0.88$)
Flat Black Absorber, ($\alpha = \epsilon = 0.95$)
Insulation, ($R = 10$)

Collector B: Evacuated Tube Collector

Selective Absorber Coating ($\alpha = 0.92, \epsilon = 0.10$)

The thermal performance of these collector types is shown on Figure 5.3.2. As indicated, the difference in performance is relatively small at the low temperatures associated with space heating or water preheating. At the higher temperatures required of air conditioning (200°F) or industrial process water, the performance of the evacuated tube collector is over twice as high as that of a conventional arrangement.

FIGURE 5.3.2

PERFORMANCE OF SELECTED COLLECTOR OPTIONS

System Costs:

The single largest component in the system cost is the solar collector array. As indicated in Section 6.1, the present costs of flat plate solar collectors average about $\$10/\text{ft}^2$ in the United States. Evacuated tube collectors are still in a developmental stage and their cost on a prototype basis is quite high ($\$20\text{--}\$40/\text{ft}^2$). Based on cost projections by manufacturers, the following costs were assumed for the collector panels.

Flat Plate Collectors - $\$7/\text{ft}^2$

Evacuated Tube Collector - $\$12/\text{ft}^2$

These costs assume that the market for solar heating equipment will grow substantially in the next few years which will lead to improved collector designs and lower per unit area costs.

The resultant system costs are shown in Figure 5.3.3 assuming that 2 gallons of water storage are required for each square foot of collector. Computer studies indicate that this is a reasonable storage volume for most heating and air conditioning applications.

These curves indicate that the cost of small solar hot water modules is relatively high. This is due to the relatively insensitive nature of the cost of controls, pumps, and piping to collector area. Consequently, larger systems requiring in excess of 1000 ft^2 can be installed at lower cost than the $50 \text{ ft}^2\text{--}200 \text{ ft}^2$ systems which might be used for hot water and space heating an individual residence in Crete. This favors the use of systems requiring the use of a solar hot water module within a community system context with the resultant benefit of economies of scale.

System Economics:

The economics of the solar hot water modules are a function of:

- Installed System Costs,
- Average Operating Temperature,
- Load Factor,
- Interest Rates,

FIGURE 5.3.3

INSTALLED COST OF SOLAR HOT WATER MODULE OPTIONS

- Useful Life, and
- Operational and Maintenance Costs.

The cost of energy from the solar hot water modules in $\$/10^6$ Btu is shown in Figure 5.3.4, assuming:

- System Size - 1000 ft²
- Load Factor - 1.0 (year-round operation)
- Capital Costs - 12%
- O & M Costs - 3%

The system using "conventional" flat plate collectors can deliver low temperature hot water at an equivalent energy cost of about $\$/10^6$ Btu. This is roughly equivalent to burning oil at 50¢/gallon if combustion/heat transfer efficiencies are considered.

The effective cost of energy increases rapidly with operating temperature as a result of decreasing collector efficiency levels. An absorption air conditioning system requires hot water at a temperature of about 200°F. At this relatively high temperature, the cost of heat from a solar hot water module using conventional collectors is up to about $\$/10^6$ Btu. At these higher temperatures it would be preferable to use the more expensive evacuated tubular collectors which can deliver heat at a cost of roughly $\$/10^6$ Btu.

The curves of Figure 5.3.4 point out a critical factor in considering solar hot water applications; namely that the economics of low temperature applications (DWH preheating, space heating, etc.) are significantly more attractive than the higher temperature applications (air conditioning, etc.).

Thermosyphon System

The cost numbers of Figure 5.3.3 are based on an "active" system where water is circulated through the collectors by a pump which is, in turn, actuated by a temperature differential controller. In areas such as Crete, where freezing is not a problem, it may be preferable to utilize a thermosyphon arrangement such as shown in Figure 5.3.5.

FIGURE 5.3.4
COST OF ENERGY FROM THE SOLAR HOT WATER MODULE

FIGURE 5.3.5
THERMOSYPHON SYSTEM SCHEMATIC

The thermosyphon water heater combines a flat-plate collector with an insulated water storage tank. The storage tank is mounted high enough above the collector so that the cooler water coming down from the bottom of the storage tank displaces the sun-heated water in the collector tubes. The sun-heated water rises and flows into the storage tank. This process causes a slow but continuous circulation of cooler water downward from the bottom of the tank, upward from the collector, once it is heated by solar energy, and back to the top of the tank. The circulation continues as long as sun shines on the collector. Solar hot water systems based on the thermosyphon arrangement are in widespread use in Australia, Israel, and Japan.

In a sunny location free from shadows the entire day, a 4 by 8 foot collector will give 40 to 50 gallons of hot water a day. This volume will usually be sufficient for residential applications given the relatively modest use of hot water in residences in Greece.

Particularly for small systems, the thermosyphon arrangement has the advantage that it requires no pumps or controllers. In the costs of Figure 5.3.3, the cost associated with these components is appreciable for the small systems. The dotted line indicates the approximate system costs for a thermosyphon system. As indicated, there is a substantial cost reduction in the smaller sizes which makes the use of such systems quite attractive on individual residences.

For larger systems, such as might be used on multi-family, institutional, and commercial buildings, there is relatively little financial incentive to use a thermosyphon arrangement. Also, the requirement to have the storage tank located at a higher position than the collectors could often make it awkward to install large thermosyphon system arrangements.

5.4 Hot Air Module

A schematic of the hot air module is shown in Figure 5.4.1. In this arrangement air is heated directly in the collector panels. This air is then either used directly to satisfy the load or used to heat a storage volume which usually consists of a crushed stone bed.

- AIR FLOW SHOWN IS DIRECT USE MODE
- IF STORAGE USED, DAMPERS SHOWN CAN BE ADJUSTED TO DIRECT AIR FLOW INTO AND OUT OF THE CRUSHED ROCK STORAGE BED AS DIRECTED BY CONTROLLER UNIT.

FIGURE 5.4.1
SOLAR HOT AIR MODULE

Potential applications for the hot air module include:

- Space Heating,
 - Residential
 - Institutional
 - Warehouses
 - Greenhouses
- Industrial drying operations,
- Agricultural drying operations,
- Heat source for heat pump systems, and
- Regeneration of desiccant.

It should be noted that the hot water module can also be used to generate hot air by adding an air to water heat exchanger at its output.

The advantages of the hot air module include:

- No potential leakage problems since only atmospheric pressure air is contained in the system,
- Few corrosion problems, which increases design flexibility and choice of materials,
- Greatly reduced problems associated with the high temperatures which can result under a zero heat withdrawal condition, and
- No problems associated with freeze protection (no problem in Crete - but could be a consideration in other parts of Greece).

The disadvantages are primarily associated with the size and cost of the air ducts required to interconnect the collector panels to each other and the rest of the system. To a first approximation the thermal performance limits of air collectors are similar to those indicated in Section 6.1 for liquid collectors; i.e., the thermal efficiency of liquid and air collectors are roughly comparable. As a result, the overall performance of well designed systems using liquid and air collectors are usually quite similar at any given temperature level. For most applications, the choice between a liquid or air heat source module would depend on practical considerations associated with reliability.

duct sizes, corrosion, and over-temperature protection all of which, in turn, impact on costs.

Distribution System:

As previously indicated, the major disadvantage of air systems are associated with the relatively poor heat transport properties of air (as compared to water). Typical line sizes in a medium size liquid system (500-2000 ft²) would be in the order of 1-2" in diameter. For a comparably sized air system, typical duct sizes would be 12-24" in diameter for a 100 ft run of duct. Even larger ducts would be required if the distance from the collector array to the point of use is longer than 100 ft. As a practical matter, therefore, the size and placement of solar hot air modules will be limited by the size and length of ducting required. The cost estimates of this section assume that the collector array can be placed in close proximity to the load so that ducting lengths external to the collector array are typically 100 ft or less.

With many existing hot air collectors, manifolds are also required to connect collectors to one another. Those ducts internal to the collector array can add significantly to array costs. Several manufacturers are, however, developing air collector systems where the collectors themselves function as the air duct. This will allow for a reduced requirement for internal manifolding with corresponding decreases in cost.

The installed cost of insulated ducting ranges from \$10-\$15 per linear foot in the United States (size range of 1'-2' in diameter). Based on this cost range it is estimated that ducts add \$1.50/ft² to the installed collector array cost.

Collector Costs:

The cost of air collectors tends to be somewhat lower than for liquid collectors for low temperature applications (100-200°F) where a

relatively simple configuration can be used. For the hot air module cost estimate, a collector cost of $\$5.50/\text{ft}^2$ was utilized (as compared to $\$7.00/\text{ft}^2$ for liquid collectors). This cost is about \$2-\$4 lower than that now available and reflects manufacturers projections of costs based on higher production levels.

System Costs:

The ducting costs for air systems tend to be considerably higher than plumbing costs for liquid systems. However, as indicated above, air collector panels for low temperature applications ($\sim 100\text{-}200^\circ\text{F}$) will usually be lower in cost than panels for liquid systems. Costs for controls, storage volumes, blowers, etc. are usually roughly comparable for the two system options.

For purposes of this study, the overall installed systems costs of the hot water and hot air modules are assumed to be roughly equal. It should, however, be emphasized that this assumption will only be valid for those air systems where:

- The collector array is located in close proximity to the load in order to minimize external ducting requirements, and
- Air collector configurations are utilized which do not require extensive manifolding within the collector array itself.

Based on the above assumptions, the economic performance of the hot air module is similar to that of the hot water module as indicated by Figure 5.3.4. To date, air collectors having the performance of evacuated tube collectors have not been fabricated. Therefore, only the cost curve associated with "conventional" collector technology is valid. As a result, the hot air module is primarily suited for application in the $100\text{-}150^\circ\text{F}$ range; such as space heating, desiccant regeneration, and crop drying.

5.5 Seawater Heat Source Heat Pump Module

As indicated in Section 2.0, the surface water temperatures in the Mediterranean are quite high on a year-round basis and vary relatively little by season. Off the coast of Crete these temperatures range from 77°F in the summer to 60° in the winter. In effect, the Mediterranean Sea functions as a giant low temperature solar collector. The surface water temperatures are sufficiently high to make its use practical as a heat source for heat pump systems for space and domestic hot water (DHW) heating. The latter application (DHW) could be particularly favorable on the larger building complexes due to the year-round nature of the load. This application will benefit significantly if the delivery temperature of the water is limited to 120°F rather than the 130-140° often used. In this way, the COP of the heat pump can be increased.

Figure 5.5.1 indicates a schematic of a heat pump system using a seawater heat source. In this arrangement, warm seawater is circulated to a heat exchanger containing the heat pump evaporator coil. The heat pump system extracts heat from the seawater, and transfer it to water contained in a storage volume at a higher temperature (100-130°F). The shaft work for the heat pump compressor can be provided by a number of sources including:

- an electric motor,
- organic Rankine cycle engine, and
- diesel engine.

The parameter which describes the performance of a heat pump system is the coefficient of performance defined as:

$$\text{COP} = \frac{\text{Useful Heat Output in Condenser Coil}}{\text{Compressor Shaft Work Input}}$$

The COP is shown in Figure 5.5.2, assuming a heat pump which operates at 70% of Carnot efficiency (which requires a highly efficient compressor). Curves are provided showing operations using seawater at temperature levels as they vary through the year near Crete. As indicated, the COP

FIGURE 5.5.1

SEAWATER HEAT SOURCE HEAT PUMP MODULE

FIGURE 5.5.2

PERFORMANCE OF HEAT PUMP SYSTEM USING AN OCEAN WATER HEAT SOURCE

is strongly dependent on the condensor coil temperature, which is an incentive to make this temperature as low as practical.

For space heating, the coil temperature is usually about 120°F. At this temperature the COP would be about 6.5 based on shaft input. To deliver 120°F hot water, the coil temperature would probably have to be increased to about 130°F, which still results in a high COP of about 5.5.

For comparison purposes, the COP of a conventional air to air heat pump is shown at rated conditions (45° heat source temperature). In electrically driven commercial equipment using an ambient air heat source, the COP is further reduced due to high losses in the electric motor and parasitic losses in the air handling equipment (fans, etc.).

The relatively high heat source temperatures of the ocean water result in the fuel input required to operate a heat pump system being considerably lower than for heating with conventional arrangements. This effect is shown in Table 5.5.1 which compares the fuel oil required to heat 1000 gallons of water 70°F (50°F to 120°F) using several different techniques, based on the following assumptions;

- Heating value of fuel is 140,000 Btu/gal,
- Flame fired heater efficiency of 65%,
- Electric power plant efficiency of 35%,
- Warm ocean water temperature of 66.5°F,
- Ten degree difference between element and water temperature,
- Efficiency of electric heat pump motor of 90%, and
- Diesel engine efficiency of 35%.

Based on Table 5.5.1, the three heat pump systems are clearly the most fuel efficient. For example, if the heat pump compressor is directly driven by a Diesel engine provided with a waste heat recovery system, the fuel required to heat DHW would only be about 30% that required by a flame fired hot water system.

In an area where fuel oil is scarce and/or expensive, the extra cost of plumbing and equipment might easily be offset by the use of a heat pump in conjunction with a warm seawater source. Table 5.5.2 gives an

TABLE 5.5.1

HOT WATER HEATING OPTIONS
(With no direct solar apparatus)

<u>HEATING SYSTEM</u>	<u>AMOUNT OF FUEL/1000 GAL OF HOT WATER*</u>
Flame Fired Heater	5.94 gallons
Electric Heat Pump	2.35 gallons
Engine Driven Heat Pump w/no Waste Heat Recovery	2.11 gallons
Engine Driven Heat Pump w/Waste Heat Recovery	1.68 gallons
Electric Resistance Immersion Heaters	11.93 gallons

* Heating Value of 140×10^3 BTU/gallon.

TABLE 5.5.2

DESCRIPTION OF HEAT PUMP HOT WATER SYSTEM

Number of People	150
Daily Use of Water	16 gals/person
Temperature Rise of Hot Water	80°F
Ocean Water Pipe Diameter (Inlet & Exhaust)	2 inch
Total Pipe Length	300 Ft.
Ocean Water Pumping Power	1/8 Hp.
Temperature Reduction of Ocean Water	5°F
COP of Heat Pump*	4.5
Compressor Power Requirements	6 Hp.

* Based on:

- Winter Ocean Water Temperature, 58°F
- Compressor Efficiency of 70%

indication of how such a system might perform, for a resort hotel located on the Mediterranean coast. The amount of storage for such a system will depend on how the load varies with time, and whether the unit will be used only during certain hours of the day.

There are several cost factors unique to the arrangement shown in Figure 5.5.1.

- The evaporator heat exchangers will have to be made from a cupro-nickel alloy (or some other salt water resistant metal). This is now common practice with air conditioning units operating with salt water cooling of the condenser coils.
- A length of a pipe must be installed between the point of use and a suitable location in the ocean as close to the load as possible.

This latter expense should be quite modest since the flows of salt water required are low. This is indicated in Table 5.5.2 which shows that the pumping power required is only about 1/8 Hp and line sizes are about 2 inches. The pumping power is seen to be small compared with the 6 Hp required to drive the compressor of the heat pump itself.

The substantial energy savings potentially available by using heat pumps combined with solar energy stored in the sea, should make them one of the prime contenders for applications in a seaside community, particularly when they are used within a community energy system context.

5.6 Photovoltaic Power Module

The present cost of solar cell modules is prohibitively high to make their use viable for electric power production in the new community. However, the possibilities for dramatically reducing the cost of solar cells is considered by many observers to be far greater than for other approaches to unconventional power generation (primarily solar thermal systems and wind turbines).

The ERDA program calls for reducing the cost of solar cell modules from their present price (\$14-\$20 per peak watt) to about \$0.50

per peak watt by the mid 1980's. Even if the cost goal is only partially achieved, solar cell power systems could be viable for use within a community system arrangement in Greece within a 5-10 year time frame. Therefore, a solar cell power module is included for discussion as a basic energy system building block so that the cost/performance parameters required to make their use practical can be defined.

A schematic of a photovoltaic power module is shown in Figure 5.6.1. This system consists of:

- A collector array utilizing silicon solar cell power modules,
- A mounting rack to properly orient the solar cell modules,
- Wiring to connect solar cell panels in the proper series-parallel arrangements, and
- A controller which maintains proper voltage levels in the array (includes any logic required to protect the cells from overloading, and controls the flow of power to the load).

The output of this module is DC power at a voltage determined by the series - parallel arrangement of solar cell modules within the array. The output from this power module can be used for many functions in the community including:

- a. To feed power into the overall electric grid of the community. In this case the power output would have to pass through an inverter system to ensure a waveform and voltage consistent with that available from the conventional power system.
- b. To directly drive pumps (with DC motors) for irrigation or desalination applications.
- c. To directly drive the compressors (with DC motors) of heat pump units for air conditioning and heating applications.

FIGURE 5.6.1

PHOTOVOLTAIC POWER MODULE

- d. To operate refrigeration equipment.
- e. To charge batteries for use in electric vehicles.

The performance, cost, and operation of solar cells are discussed in Section 6.8. One advantage of solar cells is that they generate electricity directly without the need for complex mechanical equipment. This makes their use viable for small systems as well as large ones. Also, their efficiency is almost independent of the solar intensity; i.e., if the solar flux level decreases by half, the output goes down by half. With solar thermal systems, output decreases more rapidly than insolation levels since the collector efficiency also decreases with solar flux.

System Cost:

The system indicated shows a photovoltaic array using low levels (3X) of solar concentration to reduce overall module costs (i.e., replace relatively expensive solar cells with reflector surfaces).

Cost figures are provided for three sets of assumptions regarding the cost of solar cells and concentrators:

Option 1:

- Solar Cell Costs - \$10/peak watt
- Solar Cell Efficiency - 13%
- Concentrator Costs - \$8/ft²
- Panel Cost - \$5.20/peak watt
- Panel Efficiency - 10%

Option 2:

- Solar Cell Costs - \$2/peak watt
- Solar Cell Efficiency - 13%
- Concentrator Costs - \$6/ft²
- Panel Cost - \$1.50/peak watt
- Panel Efficiency - 10%

Option 3:

- Solar Cell Costs - \$0.50/peak watt
- Solar Cell Efficiency - 15%
- No Concentration

The first option corresponds to a cost which will probably be achieved within 1-3 years using fabrication approaches similar to those now used for commercial cells for terrestrial applications.

The second option assumes that both the cost of solar cell grade silicon and cost of processing single crystal silicon wafers is greatly reduced from present levels. It is still, however, consistent with single crystal silicon technology and is a cost which has a good probability of being achieved.

The third option is the intermediate ERDA goal (for 1985) and represents a cost number which will require significant breakthroughs in material cost, fabrication techniques, and production volume levels to achieve.

Installed system costs for the photovoltaic power module are indicated in Figure 5.6.2. The installed cost of the solar cell power module is not very sensitive to power output above a fairly modest power output of about 10 kW. Even at the 1-3 kW level required of an individual residence or small pump, the photovoltaic power system costs are only about 20-50% higher than for large central facilities (100-1000 kW). This makes the use of photovoltaics very flexible within a community since they can be realistically considered for both large and small systems.

Cost of Energy:

Figure 5.6.3 indicates the cost of DC electric power from the photovoltaic power module. For the Case 1 system (existing technology), this cost is still quite high - running at about 45¢/kWh under the favorable case of year-round utilization (load factor = 1).

The costs of electric power for Case 2 approach being economically viable with engine driven generators while the costs of Case 3 are definitely competitive with small engines. With respect to Case 3, should be noted that the cost goal of \$0.50 per peak watt was originally intended to result in photovoltaic power competitive with utility supplied power in the United States. At this low per unit area cost for

FIGURE 5.6.2
INSTALLED COST OF PHOTOVOLTAIC POWER MODULES

FIGURE 5.6.3
COST OF ELECTRIC POWER FROM THE PHOTOVOLTAIC POWER MODULE OPTIONS

solar cell panels (equivalent to \$5/ft²), the cost of structure, wiring, control, and installation become important, so that the net installed system cost is about \$1 per peak watt.

5.7 Incineration with Waste Heat Recovery

The population of the proposed new community is expected to eventually grow to 2,500 - 3,000 people. It is estimated that the per capita generation of solid waste in a Greek community is 2-2.5 lbs per person per day. This would result in a total generation of solid wastes of 3.5 to 5 tons of solid wastes per day. The commercial functions associated with the tourist trade and agricultural activities (particularly olive pits) could substantially increase the amount of solid wastes; possibly to an average of 10 to 15 tons per day. On the average, the heating value of the solid wastes will be 5,000 to 7,000 Btu/lb*. If the heat content of the solid wastes is effectively utilized, all of the hot water requirements or 15-25% of the electrical requirements of the community could, in principle, be satisfied assuming the energy demands indicated in Section 3.0. The impact of solid waste utilization could be even higher if the net energy needs of the community are reduced from those outlined in Section 3.0 by extensive energy conservation measures.

System Options:

There are several approaches for utilizing the energy content of solid wastes, the most common being:

- Generation of gaseous or liquid fuels with pyrolysis, and
- Incineration with waste heat recovery.

Pyrolysis has the advantage that a fuel form which can be easily transported and stored can be the output of the process. In this way, the output of the solid waste processing plant can be sold through existing channels of energy distribution. Consequently, pyrolysis processes are receiving considerable corporate and government funding in the United States. For a new community, this advantage may not be an important one since the use of steam or hot water produced by direct

* The heating value of olive pits is 7,000 Btu/lb while that of the municipal solid wastes will be 3,500-5,000 Btu/lb.

incineration could be an integral part of the community energy system design.

It appears, therefore, that the most viable near-term approach for using the MSW* in a small community with a well-planned energy system is incineration with waste heat recovery.

Incineration Systems (status):

The increasing costs of conventional approaches for disposing of solid wastes (landfills, etc.) have led to an increasing interest in the use of incineration. Until recently, incineration has usually been used only on a large scale (thousands of tons per day) by large cities. However, there is now an increasing interest in small, modular incinerators (10-1000 tons/day) which can be used by small towns or dispersed throughout larger urban areas in order to reduce transportation costs. To date about 35 small municipal incinerators have been installed in the United States. However, not all these plants are utilizing waste heat recovery. Individual modules have design capacities running from 5-100 tons per day with the most typical being 15 tons in a 10-hour day. This range is well suited for the new community in Crete.

Also, many large industrial and institutional complexes have installed modular incinerators as a means of trash disposal. Several of these are provided with waste heat recovery systems with the steam produced used within their own facilities.

There are a number of companies (6-10) making modular incinerators with waste heat recovery equipment. Several of these firms are listed in Table 5.7.1.

Operation:

Figure 5.7.1 shows a schematic of a commercially available incinerator system with a waste heat recovery boiler. The arrangement shown has two stacks so that the waste heat recovery boiler can be bypassed in which case the system functions as an incinerator only.

Primary combustion occurs in an oxygen deficient atmosphere and

* Municipal Solid Waste.

TABLE 5.7.1

MANUFACTURERS OF "SMALL" SOLID WASTE INCINERATORS

Consumat Systems, Inc.
Richmond, Virginia

Enercon Systems, Inc.
Cleveland, Ohio

Environmental Control Products, Inc.
Charlotte, North Carolina

Farrier Industries, Inc.
York, Pennsylvania

Kelley Company
Milwaukee, Wisconsin

SECONDARY STACK

DUMP STACK

FIGURE 5.7.1

SCHEMATIC OF AN INCINERATOR WITH WASTE HEAT RECOVERY

usually requires some supplemental fuel, particularly if the moisture content of the waste is excessively high. The oxygen deficiency results in a partial pyrolysis of the waste material. The products of combustion are further burnt in secondary combustion chambers which results in a clean exhaust gas which can be directly used in a waste heat boiler (without clogging, etc.) and which satisfies environmental constraints without the need for further gas clean-up operations. The two-stage combustion process is typical of most small incinerator systems. Figure 5.7.2 shows the layout of an operating facility designed for 17 tons per day throughput. The facility is quite simple and requires a minimum of site erection and engineering design, due to the modular construction of the incinerator systems.

The small module systems appropriate for use in a small community usually operate on a batch process with manual collection of the ash. This adds somewhat to labor costs but significantly decreases the complexity of the equipment and initial capital costs.

Equipment Costs:

The cost of small incinerators has recently been studied by the Environmental Protection Agency. These studies indicated that the installed cost of the equipment (including trash collection, building site preparation, etc.) were \$9,000 to \$9,500 per daily ton of capacity for straight incinerator plants. With energy recovery (steam or hot water production) the capital costs rise to about \$16,000-\$18,000 per ton of capacity.

Economics:

Heretofore, the primary incentive for purchasing incinerators was as a means of solid waste disposal which compared favorably on an economic and environmental basis with the available alternatives; i.e., landfill operations, etc. The generation of steam or hot water has usually been a secondary issue. Where there has been the potential for selling the waste heat produced, the revenue so produced has usually not been sufficient to actually make a net profit; i.e., there has still been a net cost associated with waste disposal. Industry spokesmen feel

DESIGNED CAPACITY
17 TONS PER DAY (10 HOURS)

FIGURE 5.7.2
LAYOUT OF MUNICIPAL INCINERATOR PLANT
(17 TONS PER DAY)

this situation is changing as conventional fuel costs increase, so that in the future the incineration of solid wastes will produce steam at a cost below that of using oil and at the same time perform the critical social function of waste disposal in an environmentally acceptable manner.

An example of these favorable economics is provided in Table 5.7.2 which shows the costs associated with operating an incinerator system with waste heat recovery at the rate of 120 tons per day (four incinerator modules of standard design). This example, based on 1976 costs, indicates that incineration with waste heat recovery results in a net income if conventional fuels are valued at $\$3/10^6$ Btu (roughly 40¢/gal).

TABLE 5.7.2

COSTS RELATED TO A MUNICIPAL SOLID WASTE ENERGY PRODUCTION PLANT

120 tons per day design capacity

CAPITAL INVESTMENT

Engineering, Construction Management, Legal Fees, Permits and Emission Testing	\$ 128,800
Land Acquisition, Site Preparation, Sewer System, Roads, Landscaping, Fencing and Signs	84,300
Building Construction	319,000
Incineration—Steam Boiler System with Support Equipment, Instrumentation and Steam Delivery Lines, Installed and Tested	940,400
Truck Scale	17,500
Total Capital Cost	\$1,490,000
Cost per Ton of Design Capacity: \$12,417	

MOBILE EQUIPMENT

Front Loading Tractors	\$22,000
Pick-Up Truck	3,900
Furniture and Supply Storage Facility	3,800
Residue and Waste Containers	7,000
	\$36,700

ANNUAL OPERATING COSTS

Labor, Plant and Office	\$104,570
Fringe Benefits	15,700
Operating Supplies:	
Auxiliary Fuel (#2 Oil)	35,700
Boiler Treatment	9,700
Utilities (Water, Electricity, Phone)	10,900
Plant, Vehicle and Office Supplies	3,800
Vehicle Fuel	1,400
Insurance	15,700
Maintenance Fund	40,300
Residue Disposal (Net Cost)	26,600
Total Operating Cost	\$266,370
Cost per Ton of Waste: \$7.76	
Interest on Investment (@ 9 percent)	\$134,900
Depreciation on Buildings and Plant (Straight Line 15 Years)	99,333
Depreciation on Vehicles (5 Years)	5,180
Total Costs	\$505,783
Cost per Ton of Waste: \$14.74	

INCOME STATEMENT

Sales Revenue:	
Tipping Fees (34,320 tons @ \$5.30)	\$181,896
Steam Sales (206,000,000 lbs. @ \$3.05/1,000 lbs)	628,300
Total Sales	\$810,196
Expenses	\$505,783
Net Income	\$304,413

6.0 COMPONENT DESCRIPTION AND AVAILABILITY

There are numerous component options associated with the assembly of the key subsystem modules discussed in the previous sections. In this section, selected components which could be of particular importance in a new community system are discussed indicating:

- Physical descriptions,
- Technical operating parameters,
- Maintenance requirements,
- Field experience,
- Availability (manufacturers, etc.), and
- Cost information.

Table 6.1 indicates the components considered during this Phase I study. Emphasis was placed on components which are part of the renewable energy source subsystems (collectors, etc.) and those components which interface directly with the unconventional energy sources (heat actuated air conditioners, organic Rankine cycle engines, etc.). Conventional components which are in widespread use, such as waste heat boilers, Diesel and/or gas turbine engines, etc. were not considered at this time since their characteristics are widely known.

TABLE 6.1

"UNCONVENTIONAL" SUBSYSTEMS USED FOR COMMUNITY ENERGY SYSTEMSCOMPONENT IDENTIFICATIONCOMPONENT APPLICATION

(1) Low Temperature Liquid Collectors (80-130°F)	Space Heating, Water Heating, Heat Pump Supplement
(2) Medium Temperature Liquid Collectors (150-225°F)	Air Conditioning, Water Pumping, Desiccant Regeneration, Process Water, Desalination
(3) High Temperature Liquid Collectors (225-600°F)	Process Water & Steam, Air Conditioning, Water Pumping, Desalination, Power Generation
(4) Air Collectors	Space Heating, Crop Drying, Greenhouse Heating
(5) Organic Rankine Cycle Engine	Air Conditioning & Refrigeration, Water Pumping Electric Power Generation, Desalination
(6) Absorption Units	Air Conditioning, Refrigeration
(7) Desiccant Units	Dehumidification, Air Conditioning, Crop Drying
(8) Storage (low to medium temperature)	Space Heating, Water Heating, Air Conditioning, Desalination
(9) Storage (high temperature)	Pumping, Process Water & Steam, Desalination
(10) Wind Turbines	Water Pumping, Desalination, Electric Power Generation, Heat/Cooling (with heat pump)
(11) Heat Pumps	Space Heating/Cooling, Domestic Hot Water, Process Water

TABLE 6.1 (Continued)

"UNCONVENTIONAL" SUBSYSTEMS USED FOR COMMUNITY ENERGY SYSTEMS (Continued)

COMPONENT IDENTIFICATION

COMPONENT APPLICATION

(12) Vapor Compression Units
Multi-Stage Evaporators
Reverse Osmosis Units
Electro dialysis Units

Desalination

(13) Incinerators (with heat recovery)
Pyrolysis Units

Solid Waste Disposal & Utilization

(14) Solar Cell Arrays

Electric Power Generation, Heat Pump Operation,
Water Pumping, Desalination

6.1 Non-Tracking Solar Collectors (Flat Plate and Low Level Focussing):

Most of the solar related applications discussed in Sections 4.0 and 5.0 can be implemented using either flat plate solar collectors (no concentration) or collectors using sufficiently low levels of concentration (up to 10X) that continuous tracking of the sun is not required.

A cross section of a "conventional" flat plate solar collector of the type used for domestic hot water systems is shown in Figure 6.1.1.

The features of a flat plate collector are a blackened absorber plate covered by one or more spaced glass (or plastic) covers. Since the glass is transparent to the incident solar radiation, but opaque to the re-radiated energy, the solar collector, like the greenhouse, serves to trap solar energy. Most flat plate collectors use water (or a non-freezing aqueous solution) flowing in tubes bonded to the absorber plate to remove the absorbed solar heat. For space heating or drying applications, however, air can be used directly to remove heat from the absorber plate.

An important parameter in determining the performance of a solar collector is the thermal efficiency which is defined as the ratio of collected heat (q_c) to incident solar energy (q_i).

$$\eta_c \equiv q_c / q_i$$

The collected heat is equal to the energy transmitted to, and absorbed by, the absorber minus the heat loss from the absorber.

$$q_c = \alpha \tau q_i - U \Delta T$$

Where:

α = solar absorptance (of absorber)

τ = net transmittance of cover pane(s)

U = overall heat loss coefficient (including radiative, convective, and conductive losses)

ΔT = temperature difference between absorber and ambient air.

The efficiency at which a solar collector absorbs solar radiation is important in determining the collector area requirement for any given

FIGURE 6.1.1

CROSS SECTIONAL VIEW OF TYPICAL SOLAR COLLECTORS FOR HEATING WATER

function. As a result, efficiency impacts directly on the installed system costs and is, therefore, important even though the energy source itself is "free".

Approaches being used to improve collector performance, particularly at higher temperature levels include:

- Selective coatings on the absorber plate to reduce reradiation losses,
- Heat traps (honeycomb type structures) to reduce both convection and reradiation losses from the absorber plate,
- 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, and
- Cover plates which have high solar transmission by virtue of reduced absorption and/or anti-reflective surfaces.

Figure 6.1.2 shows performance curves for collectors using various combinations of the above approaches for improving collector performance.

Hot water and space heating both require temperature levels of 80 - 140°F. At these low temperatures, relatively straightforward collector configurations with 1 or 2 transparent covers will operate with adequate efficiency.

The operation of heat actuated air conditioning systems, generation of process hot water, and operation of multi-stage flash desalination units require temperatures of 180 - 225°F ($\Delta T \approx 100 - 150^\circ\text{F}$). At these temperatures those flat plate collectors using either a vacuum or honeycombs to reduce convection losses are required for efficient operation.

$q/A = 250 \text{ BTU/FT}^2\text{-HR}$
 $T_{\text{AIR}} = 50^\circ\text{F}$

FIGURE 6.1.2

PERFORMANCE COMPARISON OF VARIOUS CLASSES OF COLLECTOR

The operation of organic Rankine cycle engines for electric power and water pumping applications can be accomplished with heat input temperatures of 250-400°F. These moderately high temperature levels can be achieved with low levels (3X-5X) of solar concentration which does not require continuous solar tracking. Schematics of these various collector options are shown in Figure 6.1.3.

The curves of Figure 6.1.2 are for a level of solar flux typical of clear day conditions one-hour either side of noontime. 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. For stationary collectors, the daily average efficiency will ordinarily be about 25-40% lower than that obtained near solar noon. The difference between average efficiency and clear day efficiency at noon is accentuated in those areas having less consistent solar flux conditions. As a result, it is critical to calculate the performance of a collector system for any given application on an annual basis using realistic solar insolation input data when evaluating the economics of alternative systems.

Approaches to Achieving High Collection Efficiency

The status of some of the approaches for increasing solar collector efficiency are discussed below.

a. Selective Coatings:

As the collection temperature level increases, the absorber re-radiation 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. should be noted that evacuated collectors require such a coating to be effective.

The existence 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 representative list of selective black coatings which have been used or are under development is given in Table 6.1.1. Several of these coatings are now used on commercially available collectors in the United States (see Table 6.1.2).

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 and nickel.

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, low weight 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.

TABLE 6.1.1
SELECTIVE SURFACE TECHNOLOGY

<u>COATING</u>	<u>ABSORPTION</u>	<u>EMITTANCE</u>	<u>DURABILITY</u>	<u>EXPERIENCE</u>
Black Copper [*]	0.9	0.1	High Temperature Oxidation	Australia, copper substrate
Black Nickel	0.92	0.06	Moisture Destruction	Israel, galvanized iron
Black Chrome	0.95	0.08	Stable	Europe, copper, aluminum, steels
Black Zinc ^{***}	0.9	0.1	Not defined	Development
Aluminum Oxide ^{**}	0.94	0.33	Not defined	

* Provided commercially by Ethane Corporation under Trade Name of Ebonal-C.

** Under development by Alcoa and used in a number of demonstration systems.

*** Provided commercially by Olympic Plating Company.

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 pressure of the cover is 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 employs 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 a "high" vacuum can be established which eliminates both convection and conduction losses. Evacuated tube collectors are being used on several demonstration projects in the United States and their near term commercial availability appears quite certain.

Concentrating Collectors (non-tracking)

Most concentrating collectors such as parabolic dishes, require accurate solar tracking for their operation. The requirement for tracking can significantly complicate system design. Also, such systems only use the direct portion of the solar spectrum which can significantly reduce the effective solar flux under hazy or dusty conditions.

There are, however, several options possible for low level focussing (up to 10X) which do not require any tracking and/or only periodic adjustments in tilt angle. An example of one such arrangement, shown in Figure 6.1.3, is the Compound Parabolic Concentrator (CPC). This collector arrangement is in an advanced development state at the Argonne National Laboratory and several industrial firms¹ have license agreements which will probably lead to its near term commercial availability.

The external appearance of this collector is similar to that of a conventional flat plate collector. The outer glass or plastic cover plate protects the reflective surfaces from dirt and moisture, helping to ensure stable performance characteristics. The CPC optics are such that it will accept solar radiation over a wide acceptance angle (the magnitude depending on concentration ratio). Consequently, the collector need not track the sun on a daily basis and can accept a high percentage of the diffuse radiation. In fact, the tilt angle of 1.5X CPC collectors does not require any adjustment, even on a seasonal basis.

The significant advantages of utilizing low levels of solar concentration such as achievable with this type of collector is indicated in Figure 6.1.2.

Collector Manufacturers and Availability

There are over 100 manufacturers of flat plate solar collectors in the United States. These include several large industrial firms such as Exxon, PPG Industries, Owens-Illinois and Revere Copper and Brass. The majority of the firms are, however, small entrepreneurial companies which specialize in solar products. Table 6.1.2 indicates a representative cross section of these companies in the solar collector field.

1. Chamberlain Manufacturing being the most active.

a. HEAT TRAP (OR HONEYCOMB) COLLECTOR DESIGN

b. LOW LEVEL FOCUSSING (CPC) COLLECTOR DESIGN

c. EVACUATED TUBE COLLECTOR DESIGN.

FIGURE 6.1.3

SCHEMATICS OF HIGH PERFORMANCE COLLECTOR OPTIONS

COLLECTOR MFG	CONCENTRATING	#PANELS & TYPE	ABSORBER MATERIAL	SELECTIVE	AIR OR H ₂ O	FRAME MATERIAL	INSULATION	
Chamberlain	No	2 tempered low iron glass	Steel	Blk paint, Blk chrome	H ₂ O	Aluminum	Fiberglass	Also developing commercial version of the Compound Parabolic Concentrating Collector for higher temperature applications.
CSI	No	1 glass	copper tubes		H ₂ O	Stained plywood	HI Temp Polyethylene	
Energy Converters, Inc.		1 or 2 glass	Al or Cu	Sel or Flt blk	H ₂ O	Steel	Fiberglass	
AMITEK	No	2 low iron glass	copper	Yes $\alpha = 0.98/0.99$ $\epsilon = 0.22/0.4$	H ₂ O	Steel	Fiberglass Mineral wool	
Energy Systems, Inc. (ESI)	No	2 1/8" tempered glass	copper pipe Al fin	No	H ₂ O			
Honeywell/Lennox	No	1 or 2 tempered glass low iron	Cu flow tubes steel plate	Blk chrome on bright Hi $\alpha = 0.94$, $\epsilon = 0.10$	H ₂ O	Aluminum	Fiberglass	
Kalwall	No	2 plastic	Aluminum	Flat black	air	Al/plastic	owner applied	Best use as a passive system
LOF (Ford)	No	2 glass tempered	all copper Cu pipe on Cu plate	Flat black	H ₂ O	Al housing	3" fiberglass	
MIROMIT	No	1 or 2 lime or crystal glass	all steel, steel pipe on plate	$\alpha = 0.92-0.94$ $\epsilon = 0.17-0.25$	H ₂ O	Steel	Rock wool	
NES	No							
Dwens-Illinois SunPak	Slightly	1 tube	Cu or glass	$\alpha = 0.86$, $\epsilon = 0.07$ Evacuated	H ₂ O	Glass	Urethane	Evacuated glass tube collector
PAC	No	2 resin-fiberglass	Fiberglass	Flat black	H ₂ O	Fiberglass	?	
PPG	No	1 or 2 glass tempered	Cu or Al	Flat black or selective	H ₂ O	?	Fiberglass	
Raypak	No	2 glass tempered water white low iron	Cu pipe Al absorber	Flat black	H ₂ O	Steel	Fiberglass	
Revere	No	1 or 2 glass regulator or water white	Cu pipe and Cu absorber	Flat black or selective	H ₂ O	Aluminum	Fiberglass Sides urethane	
Solargenics	No	1 glass	Cu pipe, Al back	Selective	H ₂ O	Aluminum	HI temp foam and fiberglass	
Solar Research	No	2 plastic film (4 mil)	Steel tubes and steel plate	Flat black	Refrigerant	Aluminum		
Sunworks	No	1 or 2 glass	Copper	$\alpha = 0.87/0.92$ $\epsilon = 0.07/0.35$	H ₂ O + Air	Aluminum	Fiberglass and Isocyanurate	Uses plastic heat trap
Daystar	No	1 glass	Copper	Flt blk w/heat trap	H ₂ O	Aluminum	Urethane	
Northrup	Yes	1 Plastic fresnel lens	Copper	Selective	H ₂ O			
Sun Systems	No	Plastic and Teflon	Aluminum (with plastic tubes) Plastic (air system)	Flat black	H ₂ O + Air	Aluminum	Fiberglass	Plastic tubing or liquid collector is to prevent mechanical damage under freezing conditions
Sunsav	No	1 Glass	Copper	Flat black	H ₂ O	Aluminum	Fiberglass	Uses plastic heat trap
Solaron	No	1 or 2 glass	Steel	Flat black	Air	Steel	Fiberglass	

TABLE 6.1.2
REPRESENTATIVE LIST OF COMMERCIALY AVAILABLE SOLAR COLLECTORS

Most collectors now use water (or an anti-freeze mixture) as the heat transfer fluid. However, air collector panels are available from a number of sources.

The use of selective absorber coatings is seen to be quite common as a means of increasing collector efficiency. Usually only one pane of glass is used with a selective coating. As a result the cost of collectors with selective absorber coating are usually comparable to the two cover arrangements with a flat black absorber coating.

Two firms (Sunsav and Daystar*) market a collector with a heat trap (modified honeycomb) arrangement to increase efficiency at higher temperatures. Owens-Illinois has supplied solar collectors using evacuated tube receivers for several demonstration projects. General Electric and Corning Glass are also developing evacuated tube collectors for commercial sales and have allowed prototype units to be used in experimental collector configurations.

Collector Costs

There is a significant variation in the consumer cost of the various collector configurations. Based on product literature and direct contacts with manufacturers the following cost ranges appear to be prevalent for the different collector types.

Conventional Arrangement (liquid and air)

- 2 pane, flat black absorber \$8.00-\$13.00/ft²
- 1 pane, selective black absorber

Heat Trap Collectors

\$12-\$16/ft²

Evacuated Tube Collectors

\$20-\$40/ft² (no truly commercial price established)

Low Level Concentrating Collectors (CPC)

-no commercial experience
-projected by commercial licenses at \$8-\$12/ft²

All the above costs are expected to be reduced as a result of increased experience in collector manufacture and production level increases.

* Owned by Exxon Enterprises.

6.2 Organic Rankine Cycle Engines

Many of the system schematics of Section 4.0 indicate a Rankine cycle engine power system which is operated by some combination of solar heat and/or exhaust heat from Diesel or gas turbine engines. Particularly for those systems using relatively low temperature heat (300-800°F) and at small or modest power levels (1-1000 kW), the use of a Rankine cycle engine utilizing organic working fluids is receiving considerable attention by industrial concerns and within various ERDA programs. These engines could, therefore, be a critical component within many energy system options available for use in a community energy system concept.

System Description and Operation:

A schematic of an organic Rankine cycle engine is shown in Figure 6.2.1. This system consists of:

- A "boiler" which is simply a heat exchanger which can transfer heat from solar collectors, engine exhausts, or a direct combustion process into the working fluid,
- An expander (turbine, piston, etc.) which produces shaft power when supplied with a pressurized vapor,
- A condenser which consists of a water-cooled heat exchanger in most applications, and
- A feedpump (piston or centrifugal).

With some working fluids, the vapor leaving the expander is highly superheated. To increase the cycle efficiency a regenerator may be placed at the outlet of the expander to use the superheat energy for preheating the fluid from the feedpump before it flows into the "boiler" (regenerator not shown on the simplified schematic of Figure 6.2.1).

Engine Cycle Description:

Many fluids can be considered for use in these engines. Figure 6.2.2 shows the pressure-enthalpy diagram of a Rankine power cycle using one of the candidate fluids (R-113) operating in a temperature range of interest for solar and engine exhaust heat applications. The cycle indicated would

FIGURE 6.2.1

SCHEMATIC OF AN ORGANIC RANKINE CYCLE ENGINE

Enthalpy (Btu per lb. above saturated liquid at -40°F)

FIGURE 6.2.2

THERMODYNAMIC CYCLE USING R-113

utilize a regenerator. This cycle is described below:

- In a counterflow heat exchanger, heat is transferred to the boiling working fluid. The working fluid leaves the boiler at an elevated temperature and pressure (325°F, 220 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 (6-7 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 flows to the condenser, where it is condensed at a temperature of between 80 to 120°F, depending on the type of configuration used (i.e., water or air cooled).
- 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 regenerator, where it picks up heat from the superheated expander exhaust.
- The liquid then flows 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 drop in the heat exchangers) under consideration and the actual efficiencies of the major system components, such as the expander and feed-pump. 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 6.2.3 are an expander efficiency of 75% and a feedpump efficiency of 75%. As these figures indicate, system performance tends to improve as the peak cycle temperature increases. However, the rate of increase with temperature decreases at higher temperature levels, which reduces the incentive to increase temperatures above the 400-600°F range.

Working Fluid Considerations:

The performance of engines with several working fluids is shown in Figure 6.2.3. R-114 and R-113 are candidates for lower temperature applications (up to 400°F) while fluids with high thermal stability, such as Benzene, Toluene, and F-85, must be used for higher temperature applications.

A large number of working fluids can be considered for use in the engine (other than those indicated in Figure 6.2.3). 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

As indicated in Figure 6.2.3, there are relatively modest variations in efficiency between working fluid options at any given temperature 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, Toluene, and F-85 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.

FIGURE 6.2.3

RANKINE CYCLE ENGINE EFFICIENCY - VARIATION WITH PEAK CYCLE TEMPERATURE

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-shell counterflow heat exchangers commonly used in the air conditioning industry. The feedpump could be hydraulic fluid pumps 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 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^{*}, and
- other positive displacement rotary devices such as rotating vane and screw 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.

For large systems (>100 kW), the heat exchanger comprising the "boiler", regenerator and condenser are, by far, the largest components based on physical size. This is indicated by Figure 6.2.4 which shows the layout of a 2000 kW system corresponding to the thermodynamic cycle of Figure 6.2.2. These heat exchangers will also tend to control system costs, particularly in larger production quantities.

* Under development at Arthur D. Little, Inc.

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Commercial Status:

A number of American firms have made prototype organic Rankine cycle engines for a diversity of applications, including industrial waste heat recovery, bottoming cycles on Diesel engines, solar irrigation pumping, and solar air conditioning. A list of firms in the field is shown in Table 6.2.1. The effort of several of these firms is sponsored by ERDA as part of the solar cooling program and the energy conservation program (waste heat applications).

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 drive 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 and an ORC engine. These systems are primarily for use in remote areas.

As part of the ERDA energy conservation program, three firms^{***} will make demonstration engines in the 500 kW range which will use Diesel engine exhaust heat or industrial waste heat steam as the energy input. These engines are in an advanced development state and installation will, therefore, begin in late 1977 or early 1978.

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 powered organic Rankine cycle engines

^{*} Barber-Nichols Company.

^{**} Sunstrand.

^{***} ThermoElectron, Sunstrand, and Mechanical Technologies, Inc.

TABLE 6.2.1

DEVELOPERS AND/OR SUPPLIERS OF ORGANIC RANKINE CYCLE ENGINES

	<u>TYPE MECHANICAL EQUIPMENT</u>	<u>OUTPUT (kw)</u>	<u>WORKING FLUID</u>	<u>APPLICATIONS</u>
Thermo Electron Corporation	Reciprocating and Turbine	3-1000	F-85 & Water	Automotive & Bottoming Cycle
Sunstrand	Turbine	20-500	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		Solar Air Conditioning
Dupont	Rotary Boiler & Turbines	10-200	Proprietary	Automotive
Arthur D. Little, Inc. *	Scroll	1-100		
Kinetic, Inc.	Gear Expander	1-10		Solar and Waste Heat Recovery
Ormat (Israel)	Turbine	1-5		Remote Power System
IHI (Japan)	Turbine	100-500	R-11	Waste Heat Recovery
United Technologies	Turbine	3		Solid Air Conditioning
Sofretes (France)	Reciprocating & Turbine	1-50	R-11	Solar Pumping
Mechanical Technologies, Inc.	Turbine	~500	R-114 & Water	Waste Heat Recovery

* Expander development only.

FIGURE 6.2.5
COST PROJECTIONS FOR 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).

Projected Costs:

The cost of organic Rankine cycle engines is difficult to estimate since they are still not available on a strictly commercial level. However, several firms have estimated the cost of these units once they are in a commercial production. These costs will depend on several variables, including:

- power output,
- temperature of heat input, and
- production levels.

Figure 6.2.5 shows estimates for the cost of organic Rankine cycle engines based on conversations with several potential manufacturers, Arthur D. Little, Inc. cost estimates, and written material on the subject. The rather wide cost range reflects the uncertain commercial status of these engines at this time.

As indicated, engine system costs decrease as the power level increases. This is due to the fact that the cost of mechanical equipment (expanders, pumps, etc.) and controls tend to vary very slowly with power level. This output dependency of engine costs tends to favor larger scale applications which are consistent with community energy systems.

6.3 Wind Turbines

All the commercially available, non-experimental wind turbines have electrical outputs in the 0.1 to 15 kW range. These units are typically used where utility power is not available. During the 1940's and 1950's several large (100-1000 kW) wind turbine systems were built in the United States and Europe. In general, however, these larger systems were not able to compete with power produced by conventional power generation systems.

With steadily increasing costs for fuel to operate conventional power systems and improving wind turbine technologies, there is a good possibility that wind turbines, both large and small, may be competitive in the near-term in areas of favorable wind regimes, such as Crete.

Technical Discussion:

An important feature about wind turbines is that their power output is proportional to the cube of the wind speed. Figure 6.3.1 shows the resultant power output for an efficient wind turbine system. As indicated, relatively small increases in average wind velocity can significantly affect the performance of a wind turbine. For example, a 16% increase of wind speed from 12 to 14 MPH results in a 50% increase in output. As a result of this strong sensitivity to average wind speed, the viability of wind turbine systems can be highly variable with location. Consequently, particularly in seaside areas, locations only a few miles apart can have significantly different potentials for wind power utilization.

An important consideration in the design of a wind turbine is the hub mechanism, which controls the orientation of the blades. Figure 6.3.1 indicates the operating modes of an actual horizontal axis wind turbine. For this machine, the generator starts to produce power at a wind speed of about 7 MPH, and as the wind speed increases, the power increases at a rate approximately proportionate to the wind speed cubed. However, beyond a certain wind speed, the power output begins to level off, and at around 32 MPH, rated power is reached. From this wind speed on up, no increase in power output is realized, and at 45 MPH wind velocity, the generator output drops to zero. One reason the output curve has these characteristics is due to the fact that the blades do not have a constant efficiency (for any one pitch angle) at all wind speeds; as the rotational speed of the wind turbine increases, the pitch angle of each blade must be decreased by the hub mechanism to derive maximum power out of the available wind. As the wind speed approaches a specified upper limit for the machine, the blades are generally turned to infinite pitch (or feathered) to prevent mechanical damage due to;

FIGURE 6.3.1

.....IATION IN POWER OUTPUT WITH WIND SPEED FOR A COMMERCIALY AVAILABLE WIND GENERATOR

1) rotor and equipment overspeed, and 2) blade failure due to gust-induced gyroscopic forces at high rotational speeds. The necessity of a reliable, fail safe rotor hub, and controls can add significantly to the cost of high performance wind turbine equipment and is a serious practical consideration in reliable wind turbine design.

Also, this sensitivity of output with wind speed should be considered when evaluating alternative wind turbine systems. With most equipment, the "rated" wind speed is 18-35 MPH. This speed range is significantly higher than the average wind speed in most locations (on the coast of Crete it is about 14 MPH). The average output of these wind turbines will, therefore, usually be only one-fifth to one-half the rated output even in a favorable location. This is indicated in the performance curve of a commercially available wind generator shown in Figure 6.3.1. This system is rated at 6 kW but would only produce an average of 1.5 kW in a 14 MPH wind.

Commercially Available Equipment:

There are a number of firms in the United States, Australia, and Europe making small wind generators with outputs in the 0.1-15 kW range at rated wind speeds. The largest of these firms are indicated in Table 6.3.1 and a list of their products (with approximate costs) in Table 6.3.2. To date, the only large United States firm to enter the small wind turbine field is Grumman which makes a 15 kW system.

In order to see how wind turbine size affects output costs, the installed capital cost per kW (as rated by the manufacturer) was determined as a function of the rated capacity for several manufacturers windmills, Figure 6.3.2. The cost per rated kW varies inversely as the rated output of the machine. The cost of a tower will, of course, depend on the height; one company indicates that \$16 per foot of height is representative of the cost of towers capable of supporting the windmills they distribute. However, another manufacturers 45' free standing concrete tower for their 15 kW windmill is \$1965 or \$43.67 per foot.

TABLE 6.3.1

FIRMS PRODUCING WIND TURBINE SYSTEMS

ERDA CONTRACTORS (CORPORATE) IN WIND ENERGY CONVERSION

General Electric
United Technologies (Hamilton Standard)
Koman
Westinghouse
Boeing-Vertol
Lockheed

MANUFACTURERS OF COMMERCIAL WIND GENERATORS

Dave-Dunlite Electrical Company, Australia
Electro G.m.b.H., Switzerland
Dyna Technology, Iowa
Aerowatt, France
Lubing Maschinenfabrik, Germany
Dominion Aluminum Fabricating (DAF), Canada
Grumman

TABLE 6.3.2
COST OF WIND GENERATOR SYSTEMS
PARTIAL LIST OF COMMERCIALLY AVAILABLE WIND MACHINES

<u>MANUFACTURER</u>	<u>MODEL</u>	<u>RATING</u> (kw)	<u>RATED</u> <u>WINDSPEED</u> MPH	<u>AREA</u> (ft ²)	<u>INSTALLED</u> ¹ <u>COST</u> (\$)
AEROWATT	150 FP7	.1		33.8	6,065
WINCHARGER		.2	23	58.5	3,150
AEROWATT	300 FP7	.275		86.6	8,250
AEROWATT	1100 FP7	1.1		211.3	12,800
ELECTRO	WV 15G	1.2	23	76.1	5,540
KEDCO	1200	1.2		113.2	4,300
DUNLITE	2000	2.0	25	113.2	7,135
AEROWATT	4100 FP7	4.1		811.9	25,390
ELECTRO	WV 35G	4.5	24	162.9	8,060
ELECTRO	WV G50G	6.0	30	211.3	8,610
GRUMMAN	WINDSTREAM	15.0	26	490.9	24,360
DAF ²	15'	3.2		176.7	8,350
DAF ²	20'	6		314.2	11,100

1. Assumes \$2,500 installation cost; includes tower, generator, and controls.

2. Vertical Axis Wind Turbine.

Based on late 1974 information

COST OF WIND GENERATORS

FIGURE 6.3.2

Installation costs will also depend on location, terrain, and to a lesser extent, manufacturer and the degree of installation desired (initial survey through start-up and testing or just tower and base installation, etc.). Discussion with distributors indicate a daily installation rate is \$800/working day (4-man crew with equipment) and three day assembly, (assuming no bad weather at \$550/day). The resultant installation costs range between \$2500 and \$3600. These installation costs are probably typical of those associated with smaller systems in the United States.

Vertical Axis Machines:

There is considerable attention being given to the use of vertical axis wind generators (as typified by the Darrieus Rotor). One such machine which is commercially available is shown on Figure 6.3.3^{*}. This type of machine has the advantage that it does not have to face into the wind as do horizontal axis systems and the position of the electric generator is fixed at the base of the rotor, which simplifies the mechanics. However, to achieve efficiency levels comparable to horizontal axis machines (and, therefore, comparable swept area requirements) the blades of a vertical axis machine require high precision which can be quite difficult to achieve due to their long length. Also, they must be placed on towers if they are to operate in a reliable wind regime.

It is, therefore, not clear that the vertical axis machines offer significant advantages over the highly developed and proven horizontal axis equipment.

Development of Large Systems:

As indicated in Figure 6.3.2, the cost per unit capacity of wind generators tend to decrease with increasing size. Based on detailed system analysis by ERDA contractors, the trend is valid up to systems with capacities as large as 100-1500 kW. Consequently, the major programs to develop wind turbine systems in the United States which could eventually be competitive with utility power have stressed large

* Manufactured by DAF and listed in Table 6.3.2.

SOURCE: Rede Corporation

FIGURE 6.3.3
VERTICAL AXIS WIND GENERATOR

FIGURE 6.3.4

100-KILOWATT EXPERIMENTAL WIND POWER GENERATOR

TABLE 6.3.3

**SUMMARY OF COSTS FOR 100 kW EXPERIMENTAL WGPS*
AND SIMILAR FOLLOW-ON WGPS SYSTEMS**

	First 100 kW Experimental WGPS		Follow-on 150 kW WGPS Systems	
Rotor				
Blades	160K	} 50.4%	140K	} 64.0%
Hub, Pitch/Change	95K		50K	
Mechanical				
Gear Box	11.5K	} 10.8%	11.5K	} 10.5%
Bedplate, Shafts, etc.	43K		20K	
Electrical Generator, Controls	68K	13.5%	25K	8.5%
Tower, Foundation	128K	25.3%	50K	17.0%
Total	\$505K		\$296.5K	
	\$ /kW ~ \$5000/kw		~ \$2000/kw	

*Wind Generated Power System

Source: NASA Lewis

systems (as compared to using many smaller systems as they are presently commercially available). Towards this end, ERDA is working with NASA and industry to build and test, first, a two-blade, downwind 100 kW generator at Plum Brook, Ohio (Figure 6.3.4) and later, two 200 kW rated units of the same configuration, one at Clayton, New Mexico and the other at an as yet unannounced site. The 200 kW unit at Clayton scheduled for late 1977 installation, will have the same 125 ft. total diameter as the wind turbine at Plum Brook, the difference in output attributable to a higher rated wind speed. Only a limited amount of operational experience is available on these large systems since one blade of the Plum Brook facility experienced fatigue after a very brief operation period. This failure was attributable to uneven cyclic-loading caused by the blade's passing through the support towers shadow (an area of reduced wind speed directly downwind of the tower). Another difficulty encountered was the directional control system's inability to keep the windmill pointed into the wind with sufficient rigidity. These problem areas (though since corrected) should be kept in mind when considering ERDA's projected economic analysis, Table 6.3.3, for their experimental and follow-on units.

Cost of Large Systems:

The cost projections for these larger systems are also indicated in Figure 6.3.2. They are seen to be approximately \$1000 per kW which is 3-6 times lower than the least expensive of the small units now offered commercially.

6.4 Absorption Cycle System

Heat is often used to produce chilled water for air conditioning applications using absorption air conditioning systems. Such units are commonly used commercially in:

- large institutional buildings (hospitals) where steam is readily available for their operation,
- areas where natural gas costs have been sufficiently low so that the operational costs of direct fired

absorption units are lower than electrically operated vapor compression system,

- industrial applications where hot water or low pressure steam is a waste heat source available to operate absorption systems,
- as part of "total energy" systems using exhaust heat from Diesel or gas turbine engines as the heat source, and
- more recently for solar air conditioning demonstrations.

These units are readily available commercial systems which are well suited for use in a community energy system where heat from solar collectors and engine exhausts will often be available for their operation.

Operational Principles:

a. Cycle Operation

The basic (single-effect) absorption cooling system is shown schematically in Figure 6.4.1. 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

FIGURE 6.4.1
ABSORPTION SYSTEM SCHEMATIC

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.

Since this recombination reaction is exothermic, heat must be removed from the absorbent 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%.

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 6.4.2 presents the theoretical coefficient of performance of ammonia-water and lithium-bromide air conditioners.

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

FIGURE 6.4.2

ABSORPTION SYSTEM PERFORMANCE

theoretical COP for the water-lithium bromide system than for the ammonia-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 recouperator, plus heat to superheat the refrigerant vapor.

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 temperature is decreased is a natural function of the increased velocity and specific volume of the refrigerant through the eliminator separating the generator and condenser, and the reduction of pressure difference between the generator and absorber 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.

Commercial Equipment Availability:

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 3-tons, the 100-ton and 25-ton units being available in water-fired versions. 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 lift 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 6.4.3 illustrates cycle arrangements for two single-effect lithium-bromide-water machines. The direct fired unit is similar to the Arkla 3-ton unit. The unit differs from the simplified cycle diagram shown in Figure 6.4.3 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 or waste heat, the burner is replaced by a hot water heat exchanger.

Figure 6.4.3 also 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.

Water-Ammonia

The water-ammonia absorption cycle is also commercially available 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.

* 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.

Gas Engineers Handbook used with permission of Industrial Press, 200 Madison Avenue, New York, N.Y. 10016

A. Direct Fired Heating and Cooling Unit

Reprinted by Permission from ASHRAE Guide and Data Book Equipment 1972, Chapter 14

B. Steam Fired One-Shell Water Chiller

FIGURE 6.4.3

SINGLE-EFFECT LITHIUM BROMIDE CYCLE

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 6.4.1 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 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.

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 the coefficient of performance. The superheated 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 that generated in the single-effect machine. For fixed absorber, evaporator, and condenser temperatures, the fraction of the refrigerant from the first stage generated that is condensed is determined by the generator temperature. Figure 6.4.4 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 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

TABLE 6.4.1

COMPARISON BETWEEN $\text{H}_2\text{O-LiBr}$ AND $\text{NH}_3\text{-H}_2\text{O}$ SYSTEMS

AREAS OF COMPARISON	$\text{H}_2\text{O-LiBr}$	$\text{NH}_3\text{-H}_2\text{O}$
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

FIGURE 6.4.4

TWO-STAGE ABSORPTION SYSTEM PERFORMANCE

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.

Summary:

Table 6.4.2 shows the firms manufacturing absorption air conditioning systems in the United States while Table 6.4.3 compares the different types of equipment which are commercially available. The advantage of the double-effect machine is their high COP. Their disadvantages are higher costs and a higher input temperature requirement. There should be no difficulty supplying heat at the required temperatures (300-400°F) when engine exhausts are the heat source. When solar collectors are the heat source, the viability of the double-effect machines will depend on the cost/temperature trade-offs associated with the solar collector field.

System Costs:

Commercial costs for absorption systems are shown in Figure 6.4.5. In the lower tonnage range, some of the lithium-bromide equipment is designed specifically for operation with lower temperature (170-200°F)

TABLE 6.4.2

COMMERCIAL STATUS OF CONTINUOUS H₂O - LiBr ABSORPTION MACHINES

<u>Type</u>	<u>Size</u>	<u>Firms Marketing for Solar Application</u>	<u>Solar Modifications</u>
Single- Effect	2 - 25 ton	Arkla, Yazaki	Mechanical Pump, Dilution Chamber, Low temp operation
	100 - 1700 ton	Trane, York (Carrier)	None Run derated
Double- Effect	65 - 1,060 ton	Trane, Yazaki Kisha Seiza Keisha	None

- Little industry emphasis on improving CoP's
- No U.S. manufacturers involved in new design of
NH₃ - N₂O system
- Little industry interest in new fluids

TABLE 6.4.3

ABSORPTION SYSTEM APPROACHES

<u>TYPE</u>	<u>STATUS</u>	<u>HEAT REJECTION</u>	<u>TYPICAL OPERATING TEMPS (°F)</u>	<u>TYPICAL COP'S</u>	<u>ADVANTAGES</u>	<u>DISADVANTAGES</u>
Continuous Single-Effect						
• $\text{H}_2\text{O-LiBr}$	Commercial	Cooling Tower	170 - 240	0.6 - 0.8	Low Pressure	Crystallization
• $\text{NH}_3\text{-H}_2\text{O}$	Commercial in Small Tonnage	Air/Cooling Tower	120 - 240	0.1 - 0.5	Capable of Refrigeration	High Pressure Need for Rectifier
• $\text{NH}_3\text{-N}_a\text{SCN}$	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	
• Intermittant	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

FIGURE 6.4.5

COST OF ABSORPTION COOLING SYSTEMS

solar heat sources as compared to the more usual design temperatures of 210-230°F. The low temperature solar equipment tends to be about 30% more expensive than conventional equipment of the same capacity.

Per unit capacity costs are seen to decrease quite rapidly with output - again indicating the advantage of utilizing equipment of this type within community energy system context.

6.5 Desiccant Dehumidification and Cooling Systems:

The cost of Crete has a relatively high average humidity level (typically 55 to 75%). Therefore, a significant portion of the air conditioning load on institutional and commercial buildings is associated with removing moisture from the air. This function can be performed using desiccant systems that utilize materials which have the ability to attract and remove water from an air stream.

A desiccant may either be a liquid, such as a glycol compound, or solid, 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 dessicant (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.

These desiccant systems are now used to dehumidify air used in the ventilation of large buildings, for maintaining low humidity levels in storage facilities, and for drying goods at low temperatures (instead of drying by the application of heat).

These units can be combined with evaporative cooling of the air to produce cool air at an acceptable humidity level for use as an air conditioning system.

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.

- Regeneration, or removal of water from the dessicant, 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. The regeneration heat can be supplied by solar energy, engine exhaust heat or direct combustion.

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 loads on the subsequent refrigeration type sensible cooling systems. 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 rehumidifying to the desired temperature and humidity,
- Accomplish latent, as well as sensible, heat recovery between inlet air and discharge air.

1. Liquid Absorbent Cycles

Liquid absorbent dessicants 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. Figure 6.5.1 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 heated air (provided by solar, waste heat sources, or a flame fired heater). The process proceeds as follows: Building 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 re-humidifying 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 heated air is passing. The air can be heated by solar energy, waste heat sources, or by an auxiliary flame fired heater. 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 the moisture is removed by the use of cooling water.

The dehumidification capacity of the absorber-stripper 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. 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. The regeneration temperatures required for liquid systems are moderately low, typically a warm water temperature of 175°F is generally considered adequate. This temperature is achievable with solar collectors of good design and with many waste heat streams. 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 6.5.1

DESICCANT-CYCLE COOLING SYSTEM USING TRIETHYLENE GLYCOL DESICCANT

capacity of deeper drying required of some crop or industrial drying processes. 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 usually 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 6.5.2 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 6.5.2 by dotted lines where solar or waste 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 6.5.3 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

FIGURE 6.5.2 CONCEPTUAL SCHEMATIC OF DESICCANT SYSTEM

FIGURE 6.5.3

SILICA GEL BED DESICCANT COOLING SYSTEM

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 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. 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. It is anticipated that development activity could lead to a prototype system in three to four years.

b. Rotary Wheel Desiccant

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

In the cooling mode (Figure 6.5.4), 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 evaporatively 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 dehumidification 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 sensibly cooled, and finally through a humidifying chamber, where it is further cooled, 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

FIGURE 6.5.4

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

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

<u>TYPE</u>	<u>SIZE</u>	<u>STATUS</u>	<u>HEAT REJECTION</u>	<u>TYPICAL OPERATING TEMPS (°F)</u>	<u>TYPICAL COP'S</u>	<u>PRINCIPAL ORGANIZATION</u>
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 6.5.1
DESICCANT SYSTEM APPROACHES

<u>TYPE</u>	<u>ORGANIZATION</u>	<u>YEARS TO COMMERCIALIZATION^b</u>	<u>TESTS</u>	<u>COMMERCIAL APPLICATIONS</u>
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	-

^aResidential sized experiment carried out in 1952 (L8f)

^bStated by organization representative

TABLE 6.5.2
DEVELOPERS OF SOLAR DESICCANT SYSTEMS

solar energy); finally it passes through and regenerates the drying wheel before being exhausted outside.

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 requirements could be reduced by 80%.

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.

3. Summary of State of Art

Table 6.5.1 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 6.5.2 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.

6.6 Low Temperature Energy Storage

Low temperature storage systems are herein defined as those which operate below a temperature of 300°F. The use of thermal storage systems can be critically important in many of the community system options discussed in Section 3.0. It should be noted, that even those

energy system options which DO NOT use solar related options may require energy storage.. For example, total energy system arrangements often use the exhaust heat from Diesel engines to heat water for space heating, DHW, and air conditioning (with absorption units) applications. Since the demand for electricity from the Diesel generator usually will not match the demand for heat, this system may require heat storage for maximum overall thermal efficiency.

Most heat related functions in the community can be accomplished with low temperature heat. Up to a temperature of about 300°F, there is the option of storing heat in the form of hot water. At this temperature, the vapor pressure of water is about 53 psi (gauge). At these modest pressure levels, steel tanks of relatively light construction can be used. Further increases in temperature results in rapid increases in pressure making the use of water storage increasingly less tenable (for example, at 400°F the vapor pressure of water is increased to 233 psi).

There are several approaches to heat storage being considered at lower temperatures, including

- Sensible heat storage in water,
- Sensible heat storage in rock beds, and
- Latent heat storage in heat of fusion materials.

Of the above, the two sensible heat storage options are by far the furthest advanced and the ones now used regularly on operating systems. This section will, therefore, emphasize these two approaches.

Water Storage:

Water has the highest heat capacity of any of the common fluids (for example, a factor of 2-3 better than the heat transfer oils). Other desirable attributes are its good heat transfer characteristics, non-flammability, and its safety. The only potential drawback to the use of water as a heat storage material is its tendency to corrode many of the common materials of construction. These corrosion problems are, however, well documented and can be controlled by proper choice of materials and the use of coatings (epoxy, glass, etc.) inside the containment vessels.

There is little new technology associated with water storage tanks. They are now fabricated from steel (properly lined), fiberglass, and concrete. This discussion stresses the use of steel since it is still the most widely used material for water tanks and the cost of the other alternatives are not significantly lower, in fact, they are now usually higher.

Often, for esthetic reasons, storage tanks are buried underground. To install a tank in the ground is more expensive than to install the same tank on pedestals above the ground. Underground tanks have a tendency to "bob" to the surface during wet soil conditions and when the tank is empty; therefore, buried tanks often require some form of anchoring.

Also, to reduce heat losses, buried tanks will require thermal insulation. Since this insulation must maintain its effectiveness when operating in contact with wet soil, only water resistant insulations can be used. The effectiveness of the underground insulation systems are often not as high as those used in above ground applications.

The following discussions on low temperature water thermal storage tanks is, therefore, applicable primarily to above ground tanks which would probably be most often used in a new community system.

Costs of Tanks for Hot Water Storage:

The factors contributing to the installed cost of hot water storage tanks are:

- Tank (including manholes and other special attachments or modifications),
- Transportation,
- Insulation,
- Installation, and
- Foundation.

a. Tank Costs

This study assumes that the tanks will be welded construction using mild steel with dished heads. Provision has been made for at least

one 24" manhole* and estimated incremental costs for additional manholes are:

16" manhole - \$150

24" manhole - \$280

30" manhole - \$400

On larger tanks (above 500 gallons) manholes would usually be required for access for repair and to insert tube bundle heat exchangers.

Figure 6.6.1 presents the representative cost that has been developed for steel, epoxy-lined tanks suitable for this application. The tanks are provided with saddles, one 24" manhole, and lining. Included in the price are skids and costs associated with loading the tank on either rail car or truck. The cost of lining the tanks with epoxy has been taken at \$2/ft² of internal surface area.

b. Transportation

Steel storage tanks are quite bulky; therefore transportation costs can be important, particularly if the tanks are not fabricated locally. Tanks are usually fabricated within 200 miles of the point of installation and the average distance would be approximately 100 miles. Delivery costs for smaller tanks, 4' x 10' in diameter, are \$3-\$5/mile and for the larger tanks (12' diameter) about \$10/mile. Assuming that tanks are fabricated within 100 miles of the point of use, transportation can add \$300-\$1000 to the delivered cost.

c. Insulation

Insulation will usually be in the form of a fiberglass blanket with a vapor barrier. Indoor installations would typically use a 2" fiberglass blanket with a vapor barrier and an 8-ounce canvas cover. Installed, this insulation would cost \$3.00/sq ft with a painted exterior. Insulation for tanks installed outdoors will include a 4" fiberglass blanket, covered with an aluminum waterproof jacket. Installed cost of this combination is about \$5.00/sq ft.

* Small tanks for use on individual residences (60-300 gallons) would not require a manhole.

FIGURE 6.6.1

COST OF BARE STEEL TANKS

d. Installation

Installation can be a substantial factor in estimating storage tank costs. Installation cost estimates for a range of tank sizes are indicated in Table 6.6.1. These costs are based on conversations with a number of firms which install oil and water tanks in the United States.

e. Foundation

Tanks installed in a building will usually use the foundation of the building structure to support the tank.

The foundation of the building may have to be made larger to accomodate the hot water storage tank. For example, a 5,000 gallon tank might have a floor area of 10' x 26'; if one assumes the foundation cost, including excavation, at \$10/sq ft, an additional \$2,600 should be added to the installation cost of the tank.

All tanks mounted outdoors must have a concrete slab or pedestal foundation to prevent sinking and resultant stresses on the inter-connecting piping system.

Based on the above considerations, the installed cost of hot water tanks for thermal storage is shown in Figure 6.6.2. As indicated, the cost per unit capacity decreases substantially as the tank volume increases. This factor favors the use of storage in large increments such as would be required for many community energy system arrangements.

Rock Storage:

Solar heating system arrangements using air solar collectors utilize a crushed rock bed as the thermal storage media (see Section 5.4). In these systems, hot air heated by the solar collectors is blown through the rocks during the charging cycle. Cool air is blown through the hot rock bed during the discharge cycle.

The use of these systems is almost entirely for solar applications and little data is available on their performance and cost.

TABLE 6.6.1
INSTALLATION COST

<u>TANK SIZE</u> (gallons)	<u>INDOORS*</u> (<u>\$</u>)	<u>CREW</u>	<u>OUTDOORS</u> (<u>\$</u>)	<u>CREW</u>
100	70	1	70	1
200	70	1	70	1
600	200	2	150	2
1,000	300	2	250	2
2,000	300	2	250	2
4,000	300	2	250	2
6,000	400	2	300	2
8,000	400	2	300	2
10,000	500	3	400	3
12,000	500	3	400	3
20,000	500	3	400	3
30,000 (2/15)	1,000	3	800	3
40,000 (2/20)	1,000	3	800	3
60,000 (3/20)	1,500	3	200	3
60,000 (1/60 VERT)	5,000	\$1,000/day	5,000	\$1,000/day
80,000 (1/80 VERT)	6,000	\$1,000/day	6,000	\$1,000/day
100,000 (1/100 VERT)	7,000	\$1,000/day	7,000	\$1,000/day

* Basement

FIGURE 6.6.2

HOT WATER STORAGE TANK COSTS vs TANK CAPACITY

FIGURE 6.6.3

CRUSHED ROCK STORAGE SYSTEM

Typical Arrangement:

A typical arrangement for such a rock bed storage system is shown in Figure 6.6.3. One manufacturer suggests the use of rocks with a typical dimension of about 1" as being a good size to result in high heat transfer areas, low air side pressure drops, and low void volumes (typically 20%). The effective heat capacity of this arrangement is 24.0 Btu/ft³°F (as compared to that of water at 62.4 Btu/ft³°F) therefore, a rock bed storage must be roughly 260% larger than a water storage volume for comparable heat storage capacity.

The rock bed is usually contained in a concrete block enclosure which must be provided with insulation at roughly the same thickness as a water storage volume (2" to 4" of fiberglass).

Costs:

A crushed rock storage volume is constructed almost in its entirety at the point of use. Often, locally available materials (stone, concrete blocks, etc.) could be used in its fabrication. The cost of a crushed rock storage volume is, therefore, highly dependent on the local availability of appropriate materials and the cost of labor for its installation. In the United States, the cost of crushed rock storage systems on an installed basis appear to be comparable to a hot water unit of equal storage capacity. Crushed rock storage, however, is an excellent match with hot air collectors and has the advantages that it is not subject to corrosion or leakage.

6.7 High Temperature Storage

Many of the community system energy options may require storing heat at elevated temperatures (300-900°F). These could include those systems using solar energy, engine exhaust heat or the heat output of incinerators to operate Rankine cycle engine power systems. The approaches for high temperature energy storage fall into the two basic categories of sensible and latent heat storage, both of which consist of a heat storage medium, energy transfer equipment, and a containment vessel. Sensible heat storage is that in which the energy available is

used to raise the temperature of a single-phase substance and latent heat storage (or heat of fusion (HOF) storage) is where energy is stored in the material by changing its phase.

Sensible Heat Storage:

At low temperatures (75-300°F) water is usually the preferred heat storage media. However, water is not usually a convenient high temperature storage medium for above ground use due to the rapid increase in pressure as the temperature increases above 300°F (underground storage of high pressure water in aquifers or pressurized caverns is being looked at).

The most common approaches to this type of high temperature storage are non-aqueous liquids which have low vapor pressures or solids. These include:

- Petroleum Based Oils,
- Molted Salts, and
- Solids (rocks, metals, etc.)

Table 6.7.1 is a list of some storage medium manufacturers and the properties of the products they sell.

The heat transfer oils are generally limited by thermal stability considerations to temperatures of 350°C (650°F). They have a heat capacity of from 1/3 to 3/4 that of water. Therefore, larger volumes of heat transfer oils are required to store a given amount of thermal energy than if water were being used. These relatively large volumes combined with rather high costs (see Table 6.7.1) results in the heat transfer oil being a high percentage of the energy storage system costs. To reduce the costs associated with the oils, some larger systems use a storage tank filled with crushed rock or metal to reduce the inventory of heat transfer oil necessary to fill the system; Figure 6.7.1 is a diagram of one such set-up being used with a concentrating solar collector system.

Latent Heat Storage:

Latent heat or HOF storage is characterized by an increase in the

TABLE 6.7.1

HIGH TEMPERATURE HEAT TRANSFER FLUIDS

MANUFACTURER	PRODUCT	USEFUL TEMPERATURE RANGE		LIQUID HEAT CAPACITY	LIQUID THERMAL CONDUCTIVITY	APPROXIMATE COST (2,5)
		°F		Btu/lb °F	Btu/HR/Ft ² (°F/ft)	\$/16
Monsanto Industrial Chemicals Company St. Louis, Missouri	Therminol - 88	300-800 ⁽¹⁾		0.47-0.61	0.07-0.06	0.71 ⁽³⁾
	" - 66	0-650		0.32-0.68	0.07-0.06	0.86
	" - 60	- 50-600		0.32-0.67	0.08-0.06	0.83
	" - 55	0-600		0.42-0.72	0.08-0.06	0.33
Dow Chemical, U.S.A. Midland, Michigan	Dowtherm A 4	60-750		0.50-0.63	0.07-0.05	0.77-0.87
	" F	0-500		0.36-0.38	0.06-0.05	0.65
	" G	20-700		0.48-0.56	0.07	0.81-0.905
	" HP	15-550		0.63-0.73	0.07	0.22 ⁽⁶⁾
	" J	-100-575		0.60-0.70	0.07-0.06	0.64-0.73
Resource Technologies Corp. New Britain, Connecticut	Sun-Temp	- 40-500		0.56	0.07	0.50
Dow Corning Corp. Midland, Michigan	Silicone Heat Transfer Fluids	- 50-400		0.37-0.42	0.09-0.08	
Coastal Chemical Abbeville, Louisiana	HITEC	Maximum Temperatures 800	Melting Range 288°	0.37	0.35	0.32-0.65
Comstock & Wescott Cambridge, Massachusetts and	Thermkeep	900	575°-525°	0.66 ⁽⁷⁾	Solid Thermal Conductivity Btu/(HR)(Ft ²)(°F/ft) 1.0	0.20
Hooker Chemicals & Plastics Corp. Niagara Falls, New York	"					

1 Melting Point, 293°F

2 Based on 55 gal. drum size

3 Shipped as a solid in 75lb. bags

4 Now sold under the name Ortho-Dichlorobenzene, Heat Transfer Grade

5 Does not include shipping and/or freight

6 Based on tank truck size lots

7 Based on average of latent and sensible heat between 250°-900°F

FIGURE 6.7.1
Dual-Medium Thermal Storage Concept

temperature of a substance only up to a certain point and any subsequent heat input being absorbed in a transition of the material from one phase to another until the transformation is complete; additional heat input then causes further temperature increases in the form of sensible heat addition. Table 6.7.2 indicates some of the materials which have been considered as HOF storage material.

As indicated, the heat of fusion of these materials is quite high on a Btu/lb or Btu/ft³ basis. As a result, storage units using a HOF material will tend to be considerably smaller than those using the sensible heat contained in a heat transfer oil or a salt. A comparison is given below assuming:

- The HOF material is NaOH melting at 565°F, and
- The heat transfer oil is Therminol-66 which operates through a temperature swing per cycle of 50°F (typical of operating Rankine cycle engine systems from stored heat).

The resulting energy densities of the two energy storage options are:

	$\frac{\text{Btu/ft}^3}{\sim 12,370}$
HOF Storage System *	
Sensible Heat Storage System	$\sim 1,550$

One disadvantage of most HOF materials is their low thermal conductivity, particularly in the solid state. This low thermal conductivity requires large heat transfer areas in order to be able to rapidly charge and discharge the storage volume. To reduce the heat transfer area requirements, systems are being tested where the solidifying material is scraped off the heat transfer surfaces during operation. In any case, a large part of equipment costs are associated with the extensive heat transfer surface and any mechanical equipment used to keep them clean.

* Assuming 20% void volume for heat exchangers.

TABLE 6.7.2

HEATS OF FUSION FOR LATENT HEAT STORAGE MATERIALS

<u>MATERIAL</u>	<u>MELTING POINT</u> <u>°F</u>	<u>HEAT OF FUSION</u> <u>Btu/lb</u>	<u>HEAT OF FUSION</u> <u>Btu/ft³</u>
$\text{Na}_2\text{CO}_3\text{-Na}_2\text{ONaOH}$	505	102	12,000
$\text{NaNO}_2\text{-NaOH}$	460	105	12,000
$\text{NaNO}_3\text{-KNO}_3$	432	50	6,200
$\text{NaNO}_3\text{-NaNOH}$	475	68	8,100
$\text{NaCl-NaNO}_3\text{-Na}_2\text{SO}_4$	532	76	9,200
NaDR-NaOH	500	70	8,800
LiNO_3	490	164	18,200
$\text{Ba(NO}_3)_2\text{-LiNO}_3$	485	158	21,000
$\text{Ca(NO}_3)_2\text{-LiCl}$	509	72	8,400
AlCl_3	378	120	18,200
RE_2O_7	565	59	22,400

SOURCE: Comstock & Wescott, Inc.

FIGURE 6.7.2

CONCEPT OF PHASE CHANGE THERMAL STORAGE UNIT

SOURCE: Comstock & Wescott, Inc.

FIGURE 6.7.3

THERM-BANK® WATER HEATER

Commercially Available HOF Storage Products:

The HOF storage technique closest to commercial availability is a sodium-hydroxide (NaOH) device called Thermkeep^R. It was chosen for testing at the Sandia Laboratory Solar Total Energy Test Facilities at Albuquerque, and while not yet on the market, Thermkeep is undergoing testing for space and hot water heating, and it appears that it will be the first such unit of its type available on anything but an experimental scale. Figure 6.7.2 is a diagram of a possible storage unit. The Sodium Hydroxide is contained in an insulated cylindrical tank. A separate heat transfer fluid is heated (possibly by solar collectors) and enters the top of the tank. The heat transfer fluid warmed by the unit also leaves at the top, maintaining the natural thermocline, or stratification (separation of the heat storage medium into hot and cold zones) that occurs. Heat is exchanged by a series of concentric helical tubes displaced throughout the storage medium. Thus, the device as pictured could provide storage for many functions (electric power generation, shaft power production, and space and domestic hot water heating) through the use of external heat exchangers. Figure 6.7.3 is a representation of how a Thermbank^R hot water heater might appear.

Costs:

One of the few high temperature systems for which costs have been developed is that using thermkeep. The cost of the NaOH itself is about \$0.20/lb. Preliminary estimates indicate that the cost of the salts will be less than 50% of the complete storage unit cost; the remaining costs being in the containment vessel and the heat exchanger surfaces. There are, however, not yet system costs available on a commercial basis.

6.8 Solar Cells

6.8.1 Introduction

No field in solar energy has received more development effort than the direct conversion of sunlight to electricity with solar cells. This interest in photovoltaics is based on a combination of the following

considerations:

- Solar cells directly convert sunlight to electricity without requiring moving parts as in solar thermal power generation. Their demonstrated success in satellite applications gives good reason to believe that their use could be expanded to both small-scale and large-scale terrestrial applications.
- Steady conversion efficiency improvements of several solar cell materials and continued cost reductions indicate that photovoltaic conversion could be a leading candidate for solar energy system applications.
- Photovoltaics is a high technology field which can benefit from the technical and financial resources large corporations can bring to bear on its development.

Numerous photovoltaic material combinations involving several fabrication processes are now being investigated by corporate, government, and academic organizations. The number and variety of options makes judgments as to the best approach to pursue in the solar cell field a complicated process. Therefore, there are still many questions regarding which types of solar cells will eventually have the best cost/performance characteristics for terrestrial applications. Presently, all such applications use solar cells made from single-crystal silicon, primarily as a result of its demonstrated long-term reliability in space applications. However, on the basis of the cost and performance projections being made for alternative materials, such as cadmium sulfide, Gallium arsenide, and Indium phosphide, it appears that cells made from other materials could eventually compete with cells made from silicon. These alternative materials will probably not be an important factor within the time frame of most interest for the Crete project.

The present cost of terrestrial solar cell panels is \$15-\$30 per peak-watt*. In spite of this high cost, solar cells are increasingly being used commercially for remote, unattended, power applications where high reliability is at a premium.

* Power output under high solar flux conditions associated with midday operation on a clear day.

The primary interest of ERDA* and the industrial organizations involved in solar cell development is to reduce costs dramatically to the point where they can produce power on a competitive basis with conventional power plants. On the basis of existing utility rate structures and generating costs, the cost of solar cell power systems will have to be reduced to less than \$0.50 per peak watt to be competitive with the purchased price of electric power in the United States (i.e., consumer price). To be competitive with present utility busbar generating costs in the United States the cost of solar cell power systems would have to be further reduced to the \$0.15-\$0.30 per peak watt range. These cost requirements for solar cells to be competitive with utility power in the United States, are very restrictive and will require over an order of magnitude reduction in the cost of solar cells from their present levels. However, there are potentially viable applications for solar cells over a wide cost range which could see widespread implementation in the near to intermediate term (5-10 years). These applications include remote power systems for electric power and pumping systems which are now serviced by internal combustion engines and/or require costly extensions of the utility grid. These intermediate markets could be serviced by solar cells in the \$4 per peak watt range (roughly \$10-\$50/ft²), a cost goal which appears to be within reach given improvements in several of the technologies under investigation. Photovoltaics electric power systems may, therefore, prove to be viable for the new community in Crete for some applications in a 5-10 year time frame.

6.8.2 Present Development Status

1. Government

The development of photovoltaic power conversion systems is a high-priority program within the Solar Energy Division of ERDA, as evidenced by the steadily increasing funding levels in this field. The projected ERDA budget for 1977 is as follows:

* Energy Research and Development Administration.

1. SYSTEMS & APPLICAT.
ENG. & ANALYSIS
DEMONSTRATIONS

2. LOW-COST SILICON ARRAYS

TECH. DEVELOPMENT

LARGE-SCALE PROD.

3. CONCENTRATOR SYSTEMS

4. TEST & STANDARDS

5. RESEARCH & DEVELOP.

6. STORAGE & POWER COND.

7. ASSMT. OF GOALS

	FY	75	76	77	78	79	80	81	82	83	84	85
T.				Prel. Design Spec.		Detail Design Spec.						
S			45kW	170kW	285kW	600kW	1MW	2MW	4MW	7MW	10MW	
ARRAYS						\$500/kw Feas. Demo.			\$2000/kw Etab.		\$1000/kw Etab.	500/kw Etab.
ENT		170kW Delv.	150kW Delv.	270kW Delv.	500kW Delv.	1MW Delv.	2MW Delv.	3MW Delv.	4MW Delv.			
OD.					\$5000/kw Etab.		\$2000/kw Etab.		\$1000/kw Etab.			\$250/kw Etab.
TEMS		Test Fac. Oper.				Solar Cell Perf. Spec.						
OP.							10% Eff. Thin- Film Demo.				\$100 to \$300/w Thin Film Feas. Demo.	
COND.			Eval. Tasks Init.	Prelim. Subsyet. Tests	Cand. Subsys. Selected			Dev. & Proto- Test Comp.		Hard- ware Devlpd.		
							Plant Owner- ship Deter.					

	<u>\$ Million</u>
Single-Crystal Silicon Solar Cells	35
Other Materials	9
System and Applications Studies and Demonstrations	<u>16</u>
TOTAL	60

Some of the funding for single-crystal silicon is directed toward the low cost production of pure silicon feedstock. The results of these programs would be applicable to amorphous and polycrystalline silicon, which are included under "other materials".

ERDA's program has two major goals:

- To demonstrate photovoltaic power system operation in large (several kW) terrestrial applications using existing technology, and
- To support R&D activities which will result in improving the performance and lowering the cost of solar cells so that their use can be economically competitive on a large scale with conventional power sources.

The ERDA near-term goal is \$0.50 per peak watt by 1985 -- probably based on silicon technology. The longer-term goal of \$0.15 per peak watt is based on "thin film" technology which may or may not include silicon.

2. Commercial

Eight firms now sell photovoltaic power packs for terrestrial applications (Table 6.8.2). The most established of these firms are Solar Power Corporation, Solarex, and Spectrolab. All the producers listed in Table 6.8.2, except Solar Energy Systems, use relatively conventional silicon technology as the basis for their cell fabrication. Consequently, their products are quite similar in appearance and electrical performance. The cost of small power units is now about \$20-\$30 per peak watt. This cost can be reduced

TABLE 6.8.2

FIRMS ENGAGED IN MANUFACTURE AND DEVELOPMENT OF SOLAR CELLS

(A) Manufacturers of Solar Cell Power Units (Terrestrial Applications)

<u>COMPANY</u>	<u>SOLAR CELL MATERIAL</u>
Solar Power Corporation (Exxon Enterprises)	Silicon
Solar Energy Systems (Shell Oil)	Cadmium Sulfide
Solarex Corporation	Silicon
Spectrolab (Hughes)	Silicon
Optical Coating Laboratories	Silicon
Sensor Technology	Silicon
International Rectifier	Silicon
M7 International	Silicon

(B) Firms with R&D Programs in Photovoltaics:

Tyco Laboratories (Mobil Oil)	Silicon Ribbon and Tubes
Varian Associates	Gallium Arsenide
Westinghouse	Cadmium Sulfide
Comsat	Silicon
I.B.M.	Silicon, Gallium Arsenide, Organic Semiconductors
Exxon	Organic Semiconductors
Eastman Kodak	Organic Semiconductors
AT&T	Silicon, Indium Phosphide, Gallium Arsenide
Texas Instruments	Silicon
Hughes Aircraft	Gallium Arsenide
Dow Corning	Production of Low Cost
Monsanto	Silicon Raw Stock
Union Carbide }	Silicon (amorphous)
RCA	Silicon
Motorola	Gallium Arsenide,
Rockwell International	Indium Phosphide

somewhat for large-scale purchases, as indicated by the results of a recent JPL procurement in which the winning bids were as follows:

	<u>\$/peak watt</u>
M7 International (k kW)	27.80
Sensor Technology (40 kW)	12.80
Solarex (30 kW)	19.70
Solar Power Corporation (15 kW)	23.26
Spectrolab (40 kW)	17.95

6.8.3 Technology

1. Solar Cell Panels/Arrays

Figure 6.8.1 is an example of a solar cell array configuration for terrestrial applications using single-crystal silicon. The grid structure evident on top of the solar cell wafers functions as one of the electrodes. Typically, this grid is made from copper or silver alloys laid down by a vapor deposition or "silk screen" methods. The spacing of the grid is intended to reduce internal electric resistances to an acceptable limit. Typical cell thickness (using single-crystal silicon) is 0.015-0.025 inch for space applications and somewhat thicker for terrestrial applications. The substrate material, which also functions as one of the electrodes, often is vapor-deposited copper, or aluminum. The front of the array (containing the electric grid structure) is protected from the environment by sealing it in with glass or clear plastic.

Solar cell diameters for terrestrial applications are typically 2-3 inches although cells up to 4 inches in diameter are now being produced. Such a cell produces about 0.5 watt at 0.5 volt under noontime solar insolation conditions.

The production of large amounts of power requires joining many cells in series-parallel combinations. Because of the large number of electrical contacts required, each contact must have a very high

SOURCE: Solarex Corporation

SOURCE: Solar Power Corporation

FIGURE 6.8.1

SILICON SOLAR CELL WAFERS

reliability. For example, 1 kW of peak output requires up to 2,000 individual cells. There are substantial cost factors associated with the wiring of such a large number of individual cells into a complete array, particularly since the wiring must be capable of long-term, reliable operation, under fluctuating weather conditions.

2. Solar Cell Materials

Many semiconductor materials exhibit a photovoltaic effect. Of those materials which have received primary attention for power generation (Table 6.8.2), silicon, gallium arsenide, and the cadmium sulfide-copper sulfide heterojunction are the major contenders. Several of these solar cells in single-crystal form already perform almost as well as can be expected in terms of efficiency (Table 6.8.3).

All solar cells now in use for terrestrial and space applications use single-crystal silicon wafers as the photovoltaic material. The most active R&D area for both government and industrially-sponsored development efforts is the improvement of silicon solar cell fabrication methods. Since solar cells made from silicon will probably predominate in the commercial marketplace for the foreseeable future (5-10 years), the following discussion is limited to this material.

The approaches to silicon solar cell fabrication receiving the most attention are:

- Single Crystal Form
 - Czochralski Boules (single-crystal rods)
 - Ribbons
- Polycrystalline Form
- Amorphous Form

All commercial cells now use rods of single crystal silicon (boules) made by the Czochralski process. For terrestrial applications these circular boules are cut into thin (~15-30 mils) wafers which are the "raw" stock used to make silicon solar cells. These

TABLE 6.8.3

SOLAR CELL EFFICIENCIES (%)
(WITHOUT CONCENTRATION)

	<u>Achieved</u>	<u>Goal</u>	<u>Theoretical</u>
SILICON			
Single Crystal	16	18	22
Polycrystalline	10	15	22
Amorphous	5	10 ⁺	?
GALLIUM ARSENIDE			
Single Crystal	18	20	26
Polycrystalline	-	-	-
CADMIUM SULFIDE			
Vacuum Deposition	8	12	14-16% ⁽¹⁾
Spray	5 ⁺	10	? ⁽²⁾

(1) Theoretical understanding is still in formative stages

(2) Should approach that of vacuum deposital cells

wafers are typically 2 to 4 inches in diameter (see Figure 6.8.1).

In solar cell panels as now fabricated, the silicon wafer constitutes about 30-40% of the final production cost (the remainder being accounted for by the processing of the wafer into a solar cell, the grid structure, the electrical leads, and the encapsulation into a panel).

Two approaches being taken to substantially reduce the cost of the wafers while still using the basic Czochralski single-crystal growing technique are as follows:

- Reducing the cost of the silicon raw material (now about \$60/kg) by improved silicon production techniques and by lowering the purity requirements of the silicon from the semiconductor grade now used to a level closer to metallurgical grade, and
- Developing wafer cutting and polishing methods which involve much less wastage than is now the case (up to 50% is now wasted) and are more amenable to large-scale production.

A combination of the above two approaches could reduce the cost of silicon wafers by 30-50% without requiring a basic change in technology.

(1) Ribbon Processes. Several approaches to forming single-crystal silicon ribbons are being investigated in the United States and Japan (Table 6.8.4). The farthest advanced of these is the "Edge Defined Growth" process being developed by Mobil/Tyco. In the process, the single-crystal silicon is grown in ribbons of a size which can be directly used for solar cell production without suffering any appreciable losses due to slicing and cutting of the solar cell blanks. The efficiency of solar cells produced from these ribbons is about 10%, versus 12-15% for silicon solar cells produced from wafers cut from Czochralski Boules.

TABLE 6.8.4

ERDA-SPONSORED RIBBON GROWTH PROCESSES

<u>Laboratory</u>	<u>Research and Development Area</u>
Mobil-Tyco Waltham, Massachusetts	Edge-defined Film-fed Growth (EFG)
IBM Hopewell Junction, New York	Capillary Action Shaping Technique (CAST)
University of S. Carolina Columbia, S. Carolina	Web-dendritic growth
RCA Princeton, New Jersey	Inverted Stephanov growth
Motorola Phoenix, Arizona	Laser zone crystallization

(2) Polycrystalline Silicon. There is an increasing interest in the use of polycrystalline silicon as the basic stock for producing solar cells. One of the most promising developments in this area is that of the Wacke-Telefunken team in Germany. In this process, rectangular rods of polycrystalline silicon are formed which have a preferred crystal structure along the axis. The rods are then cut into wafers. Crystal sizes are of the same order (several mils) as the wafer thickness so that crystal boundary effects are reduced; i.e., it is almost like having many single-crystal cells packed close together.. The wafers are processed in a conventional manner to produce cells which have efficiencies exceeding 10%. This performance level from cells made with polycrystalline silicon has also been achieved by Solarex Corporation using a proprietary fabrication technique.

(3) Amorphous Silicon. Silicon can be deposited in thin layers on a substrate by using sputtering or glow discharge techniques. Silicon so deposited does not have a crystalline structure and is referred to as "amorphous". Possibly the most advanced effort in amorphous silicon technology is being undertaken by RCA where Schottky barrier cells with 5.5% efficiency have been fabricated. Amorphous silicon solar cells could have a dramatic effect on the cost of solar cells because:

- Only 1-2 μ of silicon need be used, which is less than 1% of the thickness usually used with single-crystal cells;
- The lead structure required to collect the current is an integral part of forming the solar cells; and
- The process appears to be readily amenable to mass production techniques.

RCA projects that such cells could be an order-of-magnitude cheaper to produce than current silicon solar cells, particularly if the expected improvements in efficiency materialize.

6.8.4 System Considerations

A solar cell power system (Figure 6.8.2) consists of four major subsystems.

1. A Solar Cell Array to convert solar energy into D.C. electric power.
2. A Power Controller to control the solar cell array power output distribution between direct use by the load and to the energy storage subsystem.
3. An Energy Storage System (usually batteries) to store excess power produced during peak solar radiation periods for later use.
4. A Power Conditioning System to convert the solar array output to AC power or possibly to higher-voltage DC power.

Most (60-70%) of the cost of solar cell power systems is now associated with the solar cell array itself. As the cost of the solar cell arrays is reduced to the point where their use could become practical for widespread applications, the effect of the power conditioning and energy storage subsystem on overall system cost will become increasingly important. For example, the present cost of inverters is about \$0.40/watt, which is comparable to the ERDA goals for the solar cell panels.

Collector Array Area Requirements

The area of solar cells required to generate one watt under typical noontime solar flux conditions (0.1 watt/cm^2 , or 316 Btu/Hr-ft^2) is referred to as a "peak watt" of capacity. The cost of solar cells is usually given in terms of "dollars per peak watt", or peak kilowatt. The area required to generate a peak kilowatt of power is a function of the efficiency of the solar cell type (Table 6.8.5).

The amount of electric energy (kW-hr) supplied by a solar cell array is a function of the hours of sunlight available for power production. On a sunny day approximately 2000 Btu/ft^2 falls on a solar cell array facing south and tilted at an angle equal to latitude. The amount

FIGURE 6.8.2

GENERALIZED SCHEMATIC OF A PHOTOVOLTAIC ELECTRIC POWER SYSTEM

TABLE 6.8.5

AREA OF SOLAR CELLS TO PRODUCE 1 PEAK KILOWATT

<u>Cell Efficiency (%)</u>	<u>Solar Cell Panel Area (Ft²)</u>
20	54
15	73
10	108
5	216

of energy produced by one peak watt of capacity under such highly favorable conditions is 6 watt-hours per day.

6.8.5 Effect of Solar Concentration

One approach to reducing the cost of terrestrial photovoltaic panels is to use solar concentration. By so doing, relatively high-cost solar panel area can be replaced with "inexpensive" reflector area (or Fresnel lenses). The economic benefits of concentration are most evident when the cost of solar cells is high. For example, if the ERDA goal of \$0.50 per peak watt is achieved, the solar cell panel would be similar in cost per unit area to the reflector system (including tracking and support), and there would be no economic benefit resulting from concentration.

The effect of solar concentration is indicated by Figure 6.8.3, which shows system cost per peak watt for various concentration ratios, assuming:

- A concentrator cost of \$8.00 per square foot of aperture area,
- An optical efficiency of the concentrator of 80% to account for reflection losses and mechanical errors in the system, and
- A solar cell panel efficiency of 10% (consistent with 14% cells with an 85% density and 10% series resistance losses).

For solar cells having the present and projected (near-term) cost of silicon (\$2-\$20 per peak watt), only a relatively modest degree of solar concentration is required to make the array cost (cells and concentrators) insensitive to the cost of the solar cells. In fact, most of the potential cost reductions would result from a concentration factor of 10 or less. This degree of solar concentration is achievable without a high degree of precision in the concentrator optics. It is also consistent with the use of silicon solar cells with passive cooling; i.e., the cells will not suffer from an overtemperature condition, even with natural convective cooling.

FIGURE 6.8.3

EFFECT OF SOLAR CONCENTRATION ON ARRAY COST

To date, terrestrial solar cell power systems have not utilized solar concentration to reduce costs. This is, in part, due to the nature of existing remote, unattended panel applications which require very high reliability with a minimum of system complications (such as might be introduced by a concentrating system).

For more-cost-sensitive applications, it seems certain that solar concentration (probably at low levels) will play a significant role in the near-term cost reduction process. In particular, solar cells combined with modest levels of solar concentration could prove a viable alternative on Crete in a near-term time frame.

6.9 Concentrating Collectors

General Description:

A concentrating solar collector is a device that focusses incident solar radiation and directs the resulting high density solar flux onto some type of energy absorbing surface. Typically, a concentrating system will use a set of lenses or high reflectivity mirrors to perform the focussing function. Flat-plate collectors can only transmit to the absorber surface that amount of energy that is incident upon the outermost light collecting (called the aperture) area. As a result, the absorber area is as great as the aperture area, leading to considerable heat losses by conduction, reradiation, and convection from the large, higher-than-ambient temperature absorber area. With a concentrating collector the absorber area is only a fraction that of the aperture. The thermal losses from this small area are much more easily managed, even at a temperature substantially higher than those for a flat plate collector. Thus, a most attractive feature of concentrating collector systems is their ability to operate at higher temperatures than conventional flat-plate solar collectors.

Lower levels of concentration can be achieved without the need for continuously tracking the sun. One such collector concept (CPC) is described in Section 6.1. The performance of flat plate collectors can also be augmented by the use of planar reflectors which require only periodic adjustments. This section stresses those higher level

concentrator systems which require tracking and are commonly considered for use in solar thermal power systems. The disadvantages of concentrating systems include:

(a) Tracking:

Concentrating collectors must continuously track the sun. For linear collectors (such as parabolic troughs), tracking need only be in one plane, while point concentrators (such as parabolic dishes) require two axis tracking. In either case, the tracking mechanisms, and associated controls add to system cost and complexity.

(b) Acceptance Angle:

The concentrating collectors discussed in this section will only accept that radiation subtended by the solar disc ($.5^\circ$). They do not utilize the diffuse portion of the solar flux as do flat plate or low level concentrators. Even in arid regions, the diffuse radiation can be 15-20% of the annual incident flux. In humid areas, this ratio may be even higher.

This fundamental limitation in the performance of concentrating collectors must be carefully evaluated when comparing their performance with flat plate or low level focussing systems.

(c) Precision/Reliability:

Concentrating solar collectors require high precision for effective operation. This precision is required in:

- The contours of the reflecting surfaces or lense systems,
- The placement of the receiver relative to the focal plane.
- The accuracy of tracking, and
- The surface finish on the reflector or lense systems.

These requirements for machining, forming, and tracking accuracy have initial cost ramifications which are still not well-defined. Also, the requirement to maintain the precision requirements for long periods of time could negatively impact on reliability.

Description of Different Concepts:

There are two different categories of concentrating systems being investigated today:

- Central Receiver, and
- Distributed Collector.

A. A central receiver system (also known as the tower of power concept) utilizes a large number of reflectors placed in an array at the base of a tall tower. The incident solar energy is reflected from the mirrors to the apex of the tower, where it is absorbed and transferred to the heat transfer medium (usually steam). Figure 6.9.1 is a sketch of a baseline design for such a system proposed by ERDA. The major cost item in such a system is currently the heliostats (tracking mirror assemblies, an example of which is shown in Figure 6.9.2). The cost of the heliostats is estimated at \$7/ft² - \$10/ft² by the various ERDA contractors working on this concept.

The "power tower" system is most often considered for use at high power levels (100 megawatts and higher) in very arid regions where a particularly high percentage of the insolation is direct. However, ERDA is now studying the use of a mini-power tower for individual pumping systems at the 150 kW power level.

B. Distributed collector systems can be described as those systems in which the solar energy is reflected and received in each individual module, rather than at a single surface, as in the central receiver concept. There are many types of distributed concentrator collectors being investigated. Four of the more common configurations are:

- Parabolic Trough,
- Parabolic Dish,
- Slat Type Configurations, and
- Fresnel Lenses (point and linear concentrators).

1. A Parabolic Trough Collector, shown in Figure 6.9.3, is also known as a Linear Parabolic Concentrator because the cross section of the reflector perpendicular to the major axis of the collector

NAMEPLATE CAPACITY
AVG OVER 4000 HRS
TOWER HEIGHT
MIRROR FIELD SIZE

215 MW(e)
147 MW(e)
1500 FT
6200 FT OD
1160 FT ID

208

FIGURE 6.9.1
CENTRAL RECEIVER CONCEPT

LOW-PROFILE MULTI-FACETED HELIOSTAT

FIGURE 6.9.2

HELIOSTAT CONFIGURATION

1. Reflecting Lighting Sheet
2. Shadow Band Tracker
3. Absorbing Receiver Tube
4. Motor Drive

FIGURE 6.9.3
PARABOLIC TROUGH COLLECTOR

is a constant paraboloid and because energy is focussed on a line, not a point. The receiver is rigidly located at the focus of this parabola and its configuration varies between designs from an exposed flat-black pipe to an evacuated glass outer tube and an inner selective surface absorber. (Evacuating the space between the absorber and the glass tube minimizes convection and conduction losses.)

Concentration ratios are typically in the 20X-40X range. Higher concentration ratios require excessive contour and tracking accuracies to be practical in this or any linear concentrator configurations. These single-axis tracking systems (rotation about the major axis of the collector to keep the incoming energy focussed on the receiver) require a pointing error of 0.10° or less, to maintain the higher end of the concentration range (30-40X).

The major losses from the more sophisticated collector designs (for example, those using a special or evacuated receiver configuration) tend to be optical and not thermal in nature. Reflector losses are a function of the following:

- Surface Quality. A highly polished surface is necessary to obtain the reflectance needed for operation.
- Manufacturing Accuracy. Even with a good specular reflectivity, manufacturing errors that deflect light in the wrong direction will hamper performance.
- Dust. A layer of dust, etc., on the reflector surface also reduces the light transmitted to the receiver.

There may also be optical losses associated with transmitting the concentrated sunlight into or onto the receiver. These losses take the form of transmission losses through glass tubes (with evacuated tube receivers) and non-ideal absorption on the absorber plate. The efficiency of a 25X parabolic trough concentrator is shown in Figure 6.9.4. Even at noon, the optical losses referred to above reduce the efficiency of the collector to about 70% even at low temperature levels.

ANNUAL COLLECTOR PERFORMANCE

FIGURE 6.9.4
EFFICIENCY OF PARABOLIC TROUGH COLLECTOR

2. A Parabolic Dish Concentrator, Figure 6.9.5, also has a parabolic cross-section, but is a figure of revolution. This type of collector is characterized by high concentration ratios (300-1500X) and the necessity for 2-axis tracking, since the sun, the receiver and the apex of the dish must all lie on the same line, during operation. Due to the relatively high concentration ratio and the subsequent need for continuous 2-axis tracking, the tracking system mechanisms and structural supports tend to be a comparatively complex, which impacts unfavorably on costs.

Advantages of parabolic dishes include their ability to generate very high temperature levels (Figure 6.9.6) and their flatter output curve during the day resulting from following the sun with a 2-axis tracking system.

3. A Slat Type Concentrating Collector is comprised of a receiver assembly and a series of parallel mirrors that reflect the incident energy onto the receiver in much the same fashion as the parabolic trough collectors. Two different configurations are applicable and are illustrated in Figures 6.9.7 and 6.9.8. In the first, the receiver is affixed to the basic frame of the assembly and the mirrors are rotated about their major axis to direct the light into the stationary receiver. In the second concept, the mirrors are fixed in place and the receiver swings over head to account for the daily apparent movement of the sun (FMSC, Fixed Mirror Solar Collector).

One motivation for such designs is to reduce the amount of equipment necessary to provide tracking. In theory, it requires a small, simpler, mechanism to move several small mirrors or one pipe then to move a comparatively large, rigid, trough and receiver assembly. In addition, making the mirror slats to the required tolerance should be less complex than making a large trough to corresponding specifications. Repairs to the reflector surface are also simplified, since only the damaged slat(s) need be replaced. For the rotating slat design, protection for the

PARABOLOID-OF-REVOLUTION DISH SOLAR COLLECTOR

FIGURE 6.9.5
PARABOLIC DISH COLLECTOR

FIGURE 6.9.6

VARIATION OF PARABOLIC DISH COLLECTOR EFFICIENCY WITH TEMPERATURE

SOURCE: Sheldahl

FIGURE 6.9.7
SLATSTM COLLECTOR CONFIGURATION

FIGURE 6.9.8

FIXED MIRROR SOLAR COLLECTOR (FMSC) CONFIGURATION

reflective surfaces is provided by downward rotation during night, cloudy or no-demand periods.

All slat type configurations are linear concentrators. Therefore, the comments on the optical and thermal losses associated with the parabolic trough designs apply here also. Dust on the mirrors of either slat type collector will have the same affect on performance it had in the parabolic trough. However, due to the large focal lengths (distance from reflector to receiver) common to these designs losses due to end effects, especially during off-noon operation, can be serious unless collector segments are quite long (over 50').

4. A Fresnel Concentrator can be either a linear or a point concentrator design. The distinguishing feature in these devices, is that light is refracted and not reflected onto the receiver. As illustrated in Figure 6.9.9, light energy is incident on the top surface of the Fresnel lens. This energy passes through the material (either glass or plastic) and is refracted by small ridges on grooves on the underside of the lens, which, though shown flat in the figure, can be either flat or curved. A curved Fresnel concentrator has a shorter focal length than a flat lens system, but is correspondingly more difficult to manufacture. The same receiver assembly used in the slat or parabolic trough could be used for a linear Fresnel concentrator, and similarly, a point concentrator could use a receiver similar to that in a parabolic dish. Tracking requirements are similar to those of reflector type concentrators for equal levels of solar concentration. Durability of plastic lenses and the inability to manufacture linear glass lenses are, however, major challenges which have limited the use of lense systems. In addition, maintaining clean surfaces under actual use (the grooves tend to become clogged with dust) is of real concern.

FIGURE 6.9.9

LINEAR FRESNEL LENSE CONCENTRATOR

Currently Available Equipment:

There are a number of companies offering solar collectors which utilize concentration. Four of the firms which have supplied concentrator collectors for various demonstration projects are listed in Table 6.9.1 along with their general specifications. The collector systems indicated have usually been provided on a demonstration system basis and, therefore, cost numbers are rather tentative at this time. Communications with the firms involved indicate costs now are in the \$15-\$30/ft² range on an uninstalled basis but that these costs are expected to drop substantially as the market develops.

TABLE 6.9.1

MANUFACTURERS OF CONCENTRATING SOLAR COLLECTORS

<u>MANUFACTURER</u>	<u>MODEL</u>	<u>COLLECTOR TYPE</u>	<u>DIMENSIONS</u> (of one unit)	<u>GEOMETRIC CONCENTRATION RATIO</u>	<u>TEMP. RANGE</u> °F	<u>COST</u> \$/ft ²	<u>COMMENTS</u>
Acurex-Aerotherm 485 Clyde Ave. Mountain View CA. 94042 (415)964-3200	3001	Parabolic Trough	10' x 6'	35:1	250°-600°		Glass Outer Receiver Tube
	3002-01	Parabolic Trough	10' x 4'	40:1	140°-250°	\$15*	Exposed Absorber
221	3002-02	Parabolic Trough	10' x 4'	40:1	250°-350°		Exposed Absorber
Hexcel Corp. 11711 Dublin Blvd. Dublin, CA 94566 (415)828-4200		Parabolic Trough	20'X(up to 9')	50:1	Air to 600° Water to 350° Oil to 600°		Receiver Contained in Box w/aperture Window
Sheldahl, Inc. Northfield, MINN, 55057 (507)645-5633	Slats TM	Movable Mirror, Fixed Receiver, Slat	10' x 11'	40:1	140°-350°**		All mirrors controlled by common motor and linkage
Northrup, Inc. 302 Nichols Drive		Fresnel	1' x 10'	10-15:1	up to 600°		Mounted in groups of 4 in a 1 axis (North-South) Tracking System

* Based on 5,000 ft² to 10,000 ft².

** Higher temperature system under development.

7.0 SPECIAL FUNCTIONS

There are energy related systems which do not fit into the categories discussed in previous sections but which could be quite important to consider for new communities on the Greek coast. Three of these are discussed in this section.

Section 7.1 discusses the various commercial cost approaches to desalination of seawater with emphasis on how these desalination techniques can utilize solar energy and waste heat streams for their operation. In Section 7.2, the energy balance in greenhouses located on Crete are discussed and approaches suggested for reducing the winter heat loads. Section 7.3 outlines composting techniques and equipment for municipal solid waste disposal.

7.1 Desalination:

Ocean water desalination might well be the most practical method of creating a potable water supply for a seaside community, particularly if it is situated on an island or in an area of sparse rainfall. There are, in theory, perhaps a dozen different ways to desalinate water; in practice, however, there are only a few methods widely employed using commercially available equipment, the rest requiring further development or large amounts of complex equipment or chemicals. These methods fall into the two basic categories of distillation processes and membrane processes.

A distillation process can be described as one in which heat is added to, or a vacuum is drawn over, the solution to be distilled, in order to change the phase of the water. The vaporized water is then drawn off and condensed, leaving a concentrated brine behind and fresh water as the product.

Membrane processes are characterized as those which utilize a membrane or partition permeable only to certain substances. In these devices, a pressure or electrical differential is created across the membrane materials and in the case of brine the water molecules are absorbed through the partition leaving the sodium and chloride ions

behind. The magnitude of the potential is a function of both the total amount of dissolved solids and the degree of purification required.

The three most common salt water distillation methods are: 1) multi-stage flash distillation, 2) vapor-compression distillation and on a much smaller scale 3) solar humidification. Reverse osmosis is also a very common approach to desalination. However, this method is usually used with brackish water with salt concentrations well below that of seawater.

The most common approaches to desalination are described briefly below.

Multi-Stage Flash Distillation:

In a multi-stage flash distillation unit (shown schematically in Figure 7.1.1) the incoming seawater is heated, and upon introduction into a chamber of slightly below atmospheric pressure, some of the water boils instantly or flashes to a vapor, and in so doing cools the remaining salt solution. This salt solution is then pumped into another chamber of lower pressure and temperature where, again, some of the water flashes to vapor and is drawn-off. Several stages are used at successively lower pressures to permit the needed boiling at lower temperatures. The condensation of the flashed vapor occurs when it is passed through the other side of the preheaters for the incoming ocean water.

Vapor Compression:

A vapor compression cycle, Figure 7.1.2, is similar to other distillation processes, except in the manner in which heat is added to the brine.

The salt water enters the containment vessel after passing through a preheater. The salt water solution is heated to the point of vaporization by absorbing the heat given off from a condensing coil in which higher pressure, product water vapor is flowing. The vapor created in this evaporation process is drawn off, compressed (to increase the temperature) and is then circulated back through the coil

Multistage flash distillation process.

FIGURE 7.1.1

FIGURE 7.1.2
SCHEMATIC OF VAPOR COMPRESSION PROCESS

to supply heat to the brine solution. In giving up heat to the brine, the fresh water vapor is condensed. The salt water preheater uses heat from the waste, concentrated brine (which must be removed from the main vessel) and the condensed product water. In a pure vapor-compression process, heat is added only during transient periods (start-up, flow increase, etc.).

Solar Humidification (Solar Stills):

Solar humidification is technologically the simplest of the distillation processes; one possible setup is illustrated in Figure 7.1.3. The sun's rays pass through the glazing without losing a significant amount of energy, and the black surface at the bottom of the water becomes warmer and, in turn, heats the adjacent water. Some of the water evaporates and upon rising to the glazing, which is not as hot as the vapor, condenses on the cooler surface. The condensate then trickles down the glazing and is collected at the sides, leaving a more concentrated brine solution in the bottom of the still. The rate of evaporation of water is largely dependent on two parameters; the temperature of the water and the relative humidity of the airspace above the water will both effect performance, so control of heat and vapor leaks is quite important. Unfortunately, even though the technique is simple, large collector areas make this method unattractive on a large scale.

Membrane Processes:

The two membrane desalination techniques most commonly encountered are reverse osmosis and electro dialysis, both of which are comparatively recent technologies. In a reverse osmosis procedure, one side of a thin membrane is supplied with water at a high pressure to be desalted. The film absorbs water at a higher rate than suspended or dissolved solids and ions, and as a result water is drawn-off the other side of the membrane, as in Figure 7.1.4. The capacity of these devices is increased not by raising the pressure, but by adding more membrane area. The membranes have a predictable life span which is a function of pressure, feed water impurity, and product water standards, and the equipment only requires the input of energy to run the instrumentation and

FIGURE 7.1.3
A SIMPLE SOLAR STILL

FIGURE 7.1.4

AN ILLUSTRATION OF TUBULAR REVERSE OSMOSIS

the high pressure pumps. (These units generally have more than 1 stage.)

In an electrodialytic process, Figure 7.1.5, one electrode is placed on each side of the outside of a passage formed by 2 semipermeable membranes. When a direct electric current is applied to the electrodes, the positive ions migrate to the negative plate and the negative ions move towards the positive electrode, leaving an ion depleted medium between the membranes. Since these devices depend on the charged nature of the dissociated salt molecules for their operation, sediment and uncharged particles must be filtered out separately from the resulting fluid between the membranes.

Ongoing research is aimed at producing viable cost competitive chemical (where the ions are chemically combined with other substances that settle out of solution) and freezing techniques (where brine is chilled and the pure ice crystals that result are then separated and melted) and improving feed water pretreatment and membrane characteristics for both reverse osmosis and electrodialysis desalinators. Also being investigated are new ways of combining current techniques for providing potable water from brackish or salt water in a safe and inexpensive fashion.

The single most important operational cost of all desalination systems is now that associated with energy. Therefore, a critical factor in determining the relative viability of different desalination options is the energy input required per pound of water distilled. For fuel fired systems, this impact of this parameter is obvious. Also, desalination is considered one of the prime applications for solar energy. The size and cost of the collector array will depend critically on the relative energy requirements of the different processes.

A calculation of the total energy input per pound of product water is not particularly meaningful for comparing these systems, on the basis of solar assistance potential, since the form of the required energy is not the same for each type of desalination technique. For example, the reverse osmosis systems will use almost entirely shaft

FIGURE 7.1.5
AN ELECTRODIALYSIS CELL

input (to create a pressure difference) where a solar still will utilize almost entirely solar heat input.

Table 7.1.1 is an estimation of the energy input for various desalination schemes based on manufacturers information and, in the case of the freezing process, experimental and projected data.

Table 7.1.1 indicates several important factors when considering desalination:

- The simple solar still is by far the least efficient approach to desalination. Its operation requires about 1000 Btu/lb of water purified while a multi-stage unit requires about 1/10 - 1/5 this amount of heat. Therefore, such arrangements may be appropriate for producing small amounts of fresh water but are probably not the best choice if solar energy is to be used to produce larger amounts of water for the whole community.
- Although the energy input to membrane or freezing equipment is low on a Btu/lb basis, this advantage decreases substantially if the energy required to produce the mechanical and/or electrical power is considered.

TABLE 7.1.1

COMPARISON OF ENERGY INPUT FOR DESALINATION PROCESSES

<u>PROCESS</u>	<u>ENERGY INPUT FORM</u>			
	<u>Electricity</u> <u>Btu/lb</u>	<u>Shaft</u> <u>Btu/lb</u>	<u>Heat</u> <u>Btu/lb</u>	<u>Temp</u> <u>°F</u>
Freezing	--	17	--	--
Hybrid				
Vapor Compression	--	17	15	180-220
Multi-Stage Flash (18 Stage)	--	5 ¹	150	175-250
Simple Distillation ²	--	--	1110	120-212
Reverse Osmosis	--	20	--	--
Electrod Dialysis	62	--	--	--

¹ Most processes require some shaft input for pumping, vacuums, etc.

² Such as is used in a solar humidification process.

TABLE 7.1.2

DESALINATION EQUIPMENT MANUFACTURERS

<u>NAME</u>	<u>PROCESS TYPE</u>
Aquachem Milwaukee, Wisconsin	Multistage Flash Vapor-Compression Vertical Tube Evap. Multiple Effect Spray Film
Ryley-Beard, Inc. Shreveport, Louisiana	Multistage Flash
Mechanical Equipment Company New Orleans, Louisiana	Vapor-Compression Hybro Vapor-Compression Multi Stage Flash
Purification Techniques Avon-By-The Sea, New Jersey	Reverse Osmosis
Osmonics Hopkins, Minnesota	Reverse Osmosis
Ionics Watertown, Massachusetts	Electrodialysis
Concentration Specialists Andover, Massachusetts	Freezing (R & D)

Manufacturers and Equipment Availability:

There are a number of manufacturers of desalination equipment in the United States which cover a range of sizes from 250 gallons per day to 1,000,000 gallons per day.* For a small community of 7,500 people, about 200,000 GPD** would be required for potable water use. These manufacturers and their product line are indicated in Table 7.1.2.

7.2 Greenhouses:

There is the possibility that winter vegetables could serve as an agricultural base for a community on Crete. Even with the moderate climate on Crete, greenhouses would be needed to optimize winter growing conditions, and the cost of fuel required to heat a large number of greenhouses could be significant.

Figure 7.2.1 shows the daily solar insolation or heat gain and the heat loss per unit area for a typical greenhouse (single polyethylene film glazing), for each month of the year. As indicated, even in the coldest months, the daily solar heat gain exceeds the heat loss. The problem is that, as indicated in Figure 7.2.2, the heat gain is during the daylight hours, and the majority of heat loss occurs at night. The result is that these greenhouses will tend to overheat during the day and heat must be dumped (i.e., they must be ventilated), while at night, heat must be added to maintain optimum growing conditions.

Greenhouse Design Considerations:

Recent Canadian and North American facilities have confronted this same problem, and use the following two approaches (some ideas for which are listed in Table 1) in greenhouse design:

- Minimize nighttime losses; and
- Optimize retention of available heat.

* Larger output plants use multiple units.

** Assuming 25 GPD per person

MONTH
FIGURE 7.2.1

MONTHLY HEAT GAINS AND LOSSES FOR A TYPICAL GREENHOUSE

FIGURE 7.2.2

DAILY HEAT BALANCE FOR A TYPICAL GREENHOUSE

In a climate such as that in Greece, some combination of the passive design features listed in Table 7.2.1 and described below should easily reduce the heating load to a point where a small auxiliary heater can handle the few periods of excess load. Therefore, active solar heating arrangements, such as those using separately mounted air collectors and crushed rock storage, would probably not be justified in a temperate climate such as that found on the Greek coast.

Reduction of Night Losses:

One of the more common features on large scale North American greenhouses (possibly not desirable in this climate due to excessive daytime heat buildup) is the use of a second layer of polyethylene, with an air space between it and the first, to reduce the heat losses. Another large part of the load is attributable to reradiation to the cooler night sky; a special curtain, suspended just inside the glazing(s), is drawn at night to trap this radiant energy inside the structure. This same curtain can be used to provide partial shading from the intense summer sun. A more exotic (and much less common) technique is to inject foam between the layers of polyethylene at night. The foam, which is opaque to the wavelength of the reradiated energy, keeps the heat inside in much the same fashion as the curtain. Some growers have found that unattended polyethylene film, while not itself opaque to infrared energy, on a cool night, will condense water vapor (which is infrared opaque) on the cooler interior surfaces of the film, thereby reducing heat loss from the structure. As the sun heats the water droplets the following morning, they evaporate, leaving the film unobstructed.

Passive Heat Storage:

Another area commonly mentioned by greenhouse designers as needing attention is the low thermal mass of these structures; i.e., their inability to store the excess solar heat which accrues during the day. Various methods are suggested for achieving this result; insulated concrete floors, concrete benches filled with crushed rock and soil for produce growing, and insulated, north facing walls of concrete (Figure

TABLE 7.2.1

HEAT LOAD REDUCTION TECHNIQUES

Techniques for Nighttime Heat Loss Reduction

- 2nd Layer of Polyethylene Film
- Nightly Use of Curtain to Prevent Reradiation Losses
- Blown Foam Between Polyethylene Layers

Methods for Increasing Heat Capacity

- Insulation of Surfaces Not Used for Heat Gain (Northwall Floor, One or Both Ends)
- Use of Concrete for Surfaces Not Used for Heat Gain
- Use of Concrete for Structures Inside the Green House
- Water-Filled Containers Lining One Unused Surface of the Structure

FIGURE 7.2.3

SOME POSSIBLE HEATING AND COOLING LOAD REDUCTION APPROACHES

TABLE 7.2.2

MATERIALS AND PROPERTIES

<u>MATERIAL (SUPPLIER)</u>	<u>WEIGHT Oz/Ft²</u>	<u>SOLAR ENERGY TRANSMISSION %</u>	<u>COST¹ ¢/Ft²</u>
<u>Polyethylene Film</u>	.15	92	.25 - .35
Cadillac Plastic and Chemical Company	.30	90	.95 - 1.35
Detroit, Michigan	.45	88	1.45 - 2.00
<u>Fiberglass Reinforced Plastic</u>			
Filon Hawthorne, California	4	86	55 - 85
Kalwall Corporation Manchester, New Hampshire	5	85	
<u>Glass</u>	20.0	86	35 - 60

¹ Does Not Include Installation

7.2.3) or water-filled 55-gallon drums are all ways of retaining heat received in the daylight hours for use at night, as well as reducing the amount of heat to be dumped during the day.

Greenhouse Materials:

There are 3 basic types of materials commonly found as glazings in modern greenhouses (listed in Table 7.2.2). These are glass, polyethylene film and fiber-glass reinforced plastics. By far, the most popular among large-scale commercial growers is the polyethylene film, since a 2-glazing cover can be had for an installed cost of approximately $10¢/\text{ft}^2$, where a single glazing glass cover (still used for smaller scale floral applications) is around $\$1/\text{ft}^2$. Fiber-glass reinforced plastic sheets cost between $50¢$ to $\$1+\text{ft}^2$ installed depending on the manufacturer, the size, and the grade (some are coated with tedlar for longevity and are warranted for as long as 20 years). Problems with polyethylene film include its relatively short lifespan of 20-24 months (commercial growers simply replace it annually or every other year) and its lack of rigidity; the material adds little or no strength to the structure.

There is a small amount of work being done with Acrylic (Plexiglas) sheets, but so far, it is quite expensive ($\sim \$2/\text{ft}^2$). The Japanese are working with a polyvinyl chloride film, which has a longer life than the polyethylene film (due to decreased ultra-violet degradation), but becomes electrostatically charged and often partially obscured by dust. It appears that polyethylene film or fiber-glass reinforced plastic panels are still the two primary choices.

7.3 Composting:

This section discusses the conversion of municipal solid wastes and agricultural wastes into useful thermal energy in small incinerators provided with heat recovery units. An alternative for converting these wastes into a functional product of potential value to the community is composting.

Composting is a biochemical degradation of organics by micro-organisms and can be a sanitary means of turning municipal wastes and sewage sludges into humus. There are many different specific processes presently in use which generally follow a basic procedure.

First, refuse is prepared for digestion by the sorting out of non-compostables such as plastic, metals, rubber, wood, glass, etc. The sorted refuse is then ground in small particles to facilitate handling, digestion and mixing of materials. Sewage sludge can now be added and this resultant mix must have a 50% to 60% moisture content by weight for effective digestion. That percentage is achieved by either adding water or in the case of most sewage sludge, the dewatering of the sludge prior to mixing. Also important is the carbon to nitrogen ratio. Refuse has much carbon due to paper, and sewage sludge has nitrogen in the form of ammonia. The optimum rotting ratio is 30 parts carbon to one of nitrogen and can be achieved with the addition of an appropriate carrier. Typical nitrogen sources are green grass clippings, ammonium sulphate, urea, or blood meal. Some carbon carriers used are sawdust, chopped straw, turf, and old leaves.

The actual digestion or decomposition can take place in oblong piles called windrows or in aerated tanks for a faster composting time. The degrading microorganisms need oxygen for decomposition and this requirement is fulfilled by turning and aerating the windrows and pumping air into the enclosed systems. Sufficient aeration will prevent foul odors from occurring during this process. Active decomposition heats the compost to about 150°F (65°C), temperatures which destroy pathogenic organisms, weed seeds, fly ova, etc. Composting time depends upon the makeup of the waste and the specific process, but generally takes a few months in windrows and a few weeks in aerated enclosed systems.

The compost can be used immediately for hotbed applications but will cause nitrogen loss to soils unless allowed to stabilize for an additional period of time. This curing time usually lasts a few weeks, depending upon the decomposition process, and then the humus

can be screened or ground as required by its intended use.

This compost is not a fertilizer but a sterile, dark brown, high organic content, soil conditioner which is desirable for its consistency and its ability to till well and hold both water and nutrients. It can be fortified with fertilizer or pellitized to satisfy other applications.

Economics determining the net cost of composting are difficult due to variations in plant size, land costs, labor costs, number of shifts, composting methods used, and the value attached to the humus by-product.

A U.S. Public Health Service - TVA Composting Project, Johnson City, Tennessee provides some cost observations. This windrow plant, built in 1967, could process 50 tons of refuse a day, with about 5,000 gallons of sewage sludge. The 1967 cost of buildings was \$368,338; equipment, \$463,251; site improvement, \$126,786; land cost, \$7,600; Total - \$965,980. This cost is typical of that associated with composting facilities, which is roughly \$15,000 - \$20,000 per ton of daily capacity. The operating cost of this Johnson City plant at full capacity is \$13.65 per ton of refuse processed, with labor expenses constituting about 75% of this figure. An additional expense is that of disposing the 30% of the refuse which is uncompostable. If disposed of in a landfill, the extra cost per ton of refuse processed could be as high as \$.50.

The sale of the compost would serve to bring these costs down. The problem is its high weight and resultant high cost of shipping. In the United States, the value of the humus as a soil conditioner varies greatly, with a typical range being \$7 - \$20 per ton. The composting systems, however, are generally installed as an environmentally acceptable means of disposing of municipal solid wastes and/or sewage sludges. Any revenue gained by selling the humus by-product is used to offset the operating costs of the system; i.e., the production of humus is usually not the primary function of the facility.

There are a limited number of firms which specialize in the installation of composting systems. One firm fabricates commercial sized aerated tanks (Figure 7.3.1) which are used in smaller towns in Europe and are now being introduced into the United States (with limited success to date).

Composting systems using windrows use a combination of shredding, sorting, grinding, and screening in their operation. This equipment is not specially designed for use in composting systems (i.e., is available commercially for a variety of applications). Several firms, some of which are listed in Table 7.3.1 are engaged in the design and installation of composting systems for municipal and industrial applications.

A list of some of the composting systems installed in the United States is presented in Table 7.3.2.

For a small community of 2,500 people and hotel space for 400 tourists, a plant of maximum processing capacity of about 7 tons of refuse a day would be appropriate. The cost per ton of processing on such a limited scale may be quite high for a plant having all the equipment for receiving, sorting, grinding, adding sewage sludge, composting, final grinding and screening, and curing and storage.

A simplified system involving more labor and less equipment might be feasible if the cost of labor is low enough and if the compost can be treated with a commercial fertilizer and sold to bring in some revenue.

SOURCE: American Bioreactor Corporation

FIGURE 7.3.1
SCHEMATIC OF AERATED TANK COMPOSTING SYSTEM

TABLE 7.3.1

COMPANIES BUILDING COMPOSTING SYSTEMS

All American Environmental Control Corporation
Wilmington, Delaware

American Bioreactor Corporation
Reynoldsburg, Ohio

Ecology, Incorporated
Brooklyn, New York

Fairfield Engineering Company
Marion, Ohio

Zurn Industries
Erie, Pennsylvania

TABLE 7.3.2
COMPOSTING PLANTS IN THE UNITED STATES

<u>LOCATION</u>	<u>COMPANY</u>	<u>CAPACITY (ton/day)</u>
Altoona, Pennsylvania	Altoona FAM, Inc.	45
Boulder, Colorado	Harry Gorby	100
Gainesville, Florida	Gainesville Municipal Waste Conversion Authority	150
Houston, Texas	Metropolitan Waste Conversion Corp	360
Johnson City, Tennessee	Joint USPHA-TVA	52
Mobile, Alabama	City of Mobile	300
New York, New York	Ecology, Inc.	150
San Juan, Puerto Rico	Fairfield Engineering Co.	150
St. Petersburg, Florida	Westinghouse Corp.	105

CONVERSION FACTORS

UNITED STATES ENGINEERING UNIT

S.I. UNIT

1 ft	0.3048
1 sq ft	0.0929 m ²
1 cu ft	0.0283 m ³
1 gallon	0.004546 m ³
1 pound mass (lbm)	0.4536 kg
1 lbm/cu ft	16.02 kgm ⁻³
1 pound force	4.448 N
1 psi	6894.8 Nm ⁻²
1 Btu	1055.1 J
1 cal	4.186 J
1 Btu/hr	0.293 w
1 Hp	745.7 w
1 Btu/hr ft ²	3.152 wm ⁻²
1 Btu/lb°F	4.184 J kg ⁻¹ K ⁻¹
1 Btu/hr ft°F	1.731 wm ⁻¹ K ⁻¹

MISCELLANEOUS CONVERSIONS ²

$$1 \text{ LANGLEY} = \text{cal/cm}^2 = 4.186 \text{ J/cm}^2$$

$$1 \text{ Btu/sq ft} = .271 \text{ cal/cm}^2$$

$$^{\circ}\text{F} = (^{\circ}\text{K} - 273) 9/5 + 32$$

1. SOURCE: "Principles of Heat Transfer", Kreith, 3rd Edition, Intext Publications, 1973.
2. SOURCE: "Solar Energy Thermal Processes", Duffie and Beckman, John Wiley and Sons, 1974.