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PREFACE

The InterTechnology Corporation (ITC) was one of two companies who received awards to conduct studies and analyses for the Energy Research and Development Administration (ERDA) in support of the National Solar Heating and Cooling Demonstration Program. This report was prepared as part of the requirements under contract E(11-1)2688. It is not intended to reflect an ERDA plan or position. It is expected to be used by ERDA in the development of their plans for implementing the Solar Heating and Cooling Demonstration Program.

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I. INTRODUCTION

The rate at which the sun's radiant energy falls on a square meter of surface outside the earth's atmosphere at the mean earth to sun distance is approximately 1.35 kw/m^2 . Owing to the annual variation of the earth to sun distance this number varies from 1.399 kw/m^2 on January 1 to 1.309 kw/m^2 on July 1. In passing through the earth's atmosphere much of this energy is lost through absorption by water vapor, CO_2 and ozone. In addition, a good deal of scattering takes place leading to a non-direct (diffuse) as well as a direct component of the radiation. The question of how much of the sun's energy actually strikes a unit area of the earth's surface is further complicated by the daily rotation of the earth as well as its annual precession.

In the attempt to capture the sun's radiant energy these geometrical effects are extremely important and the detailed design of solar systems are heavily influenced by geometrical considerations. Of course the relatively low areal energy density of solar energy at the earth's surface requires that large collector systems be utilized, which in turn results in large capital costs for solar systems.

Another factor which is important in the design of solar systems is the intended use of the system. Some of the areas of use have included heating of swimming pools, hot water, space heating and cooling, crop drying, desalinization, production of electricity and special high temperature applications. The most significant variables relating to these applications which impact solar systems design are the energy required, the temperature required, the energy available (insolation) and the demand pattern (both yearly and daily).

In the attempt to tie all of these factors together, manufacturers of solar systems have designed two basic types of collectors; namely, flat plate and concentrating types. Within each of these categories there are significant variations which will be discussed at length in this paper. However, the primary factor which separates these two types is the operating temperatures

which can be achieved in a cost effective manner. To a first approximation flat plate collectors are designed to operate at temperatures of less than 200°F, whereas concentrating collectors are designed to operate above 250°F. For temperatures between 200 and 250°F either collector type may be sufficient but a detailed engineering tradeoff is usually required to determine which is most cost effective.

Of course, the other subsystems making up a solar system are also of great importance. Among these are storage, control, delivery and auxiliary supply subsystems. Each of these will be discussed separately as well as in a section on overall systems performance.

The most important factor in determining whether a solar energy system or any other system should be used for providing the thermal requirements of a particular process is the cost per million Btu of delivered energy. For this reason as much cost data as possible will be given for the various components and subsystems discussed in this report.

Since our basic concern in this report is with the application of solar-thermal systems to heating and cooling of buildings, an attempt will be made to restrict our attentions to solar systems with good potential for these applications.

II. COLLECTORS

A. Flat Plate

1. Non-selective flat absorber with 0, 1, and 2 cover plates.

A non-selective solar absorber has both a high absorptivity for solar radiation and a high emissivity for thermal (infrared) radiation. All commercial black paints are non-selective and typically have absorptances, α , of 0.95-0.98 and emittances, ϵ , of 0.85-0.95. Selective absorber coatings have emittances less than 0.4, with the very best having $\epsilon = 0.08 - 0.1$.

Flat-plate solar collectors may be subdivided into air collectors or liquid collectors depending on the coolant used for energy removal. The majority of collectors available today are of the liquid cooled variety. Air collectors have the advantages of freedom from corrosion and freezing. Their major disadvantages are poorer thermal performance, higher power requirements for coolant transport, and a larger volume required for thermal storage.

The function of a collector cover plate is to reduce the loss of energy from the absorber plate to the outside air. The cover plate must be sufficiently transparent to solar radiation so that a high percentage of the energy striking the cover plate is transmitted to the absorber. The types of materials used for cover plates are glass and a wide range of plastics. The transmittances of these materials for solar energy range from 0.84 to 0.92. The highest transmittances are achieved by glass with a very low iron oxide content. The maximum transmittance is limited to 0.92 because of reflection from the front and back surfaces of the glass. A promising technique, which is still under development, is to etch the surface of the glass with an acid. This treatment modifies the index of refraction of the surface and has resulted in transmittances as high as 0.98 for laboratory samples. At present, the etch treatment is not commercially available.

The cover plate or glazing must also fulfill the mechanical and structural requirement of withstanding wind and snow loads, hailstorms, etc. Plastics generally require more support than glass which leads to an overall reduced transmittance for the collector. Also, the durability and lifetime of many of the plastic covers is not well established for solar energy applications.

The performance of non-selectively coated solar collectors is presented in the next section for 0, 1, and 2 cover plates. Typical uses and price ranges for these collectors are also given.

Unglazed collectors - Unglazed collectors have performance curves which are extremely dependent on wind speed and direction. In Figure 1 the collector heat loss coefficient is plotted as a function of wind speed. The heat loss

FIGURE 1
HEAT-LOSS COEFFICIENT VS. WIND SPEED FOR
FLAT-BLACK UNGLAZED COLLECTOR

coefficient U_L is determined from the Hottel-Whillier-Bliss model of collector performance. In this model, the collector efficiency, η , is given by:

$$\eta = F_R \left(\alpha\tau - \frac{U_L [T_{in} - T_{amb}]}{S} \right)$$

where

F_R = collector heat removal factor

α = solar absorptance of coating

τ = transmittance of glazing

U_L = collector heat loss coefficient, Btu/hr-ft²-°F

T_{in} = fluid inlet temperature, °F

T_{amb} = ambient temperature, °F

S = insolation, Btu/hr-ft²

In Figure 2, the efficiency of an unglazed collector is plotted versus $(T_{in} - T_{amb})/S$ for various wind speeds. For comparison, the efficiency curves for a single and double glazed collectors are also plotted.

The main feature of these curves is the very rapid drop in efficiency of the unglazed collector with both increasing temperature difference and wind speed. This limits the practical uses of unglazed collectors to applications where the temperature difference between inlet and ambient is very small. The major application is for swimming pool heating.

The prices of unglazed collectors range from \$1.50/ft² to \$4/ft². Systems costs are also low because cheap uninsulated plastic pipe can be used and the collectors are easily installed.

Single glazed collectors - In Figure 2, the performance of a single glazed collector is shown. Because of the rapid drop off in efficiency with inlet temperature, these collectors are used where the inlet temperature is not too high (100°F or less above ambient) or where there is ample sunshine. The major use of these collectors is for hot water heating. They are also effective when combined with heat pumps because of the low water temperatures required.

FIGURE 2
PERFORMANCE CURVES FOR NONSELECTIVE FLAT PLATE
COLLECTORS WITH 0, 1 AND 2 COVER PLATES

Single glazed collectors are more expensive than unglazed collectors because of the glazing and its supports, insulation, and a container. In addition, the single glazed collector can reach temperatures above the operating limit of most plastics so a metal absorber plate must be used.

Prices of single glazed collectors range from \$7/ft² to \$11/ft² depending on the materials used for the absorber plate and glazing.

Double glazed collectors - The addition of an extra cover to a single glazed collector has the effect of lowering the collector heat loss coefficient and decreasing the amount of solar energy reaching the absorber plate. This effect is shown in the performance curves in Figure 2.

Double-glazed, non-selective, collectors are used principally for hot water and space heating where the required temperatures are less than 200°F. Commercial solar collectors are available using either liquid or air cooling. Air-cooled collectors suffer from a poorer thermal performance but have the operational advantages of requiring no freeze protection, freedom from corrosion, etc.

A comparison of the performance of double-glazed, flat-black air and liquid cooled collectors is shown in Figure 3. In actual operation using rock bed storage the air collector will generally have lower inlet temperatures due to stratification of the storage medium. The costs of air cooled collectors are generally in the same range as liquid cooled collectors, that is from about \$9 to \$12/ft². An air collector has the potential for being somewhat less costly than a comparable liquid-cooled collector. At the present time there are only a few manufacturers of air collectors.

A summary of the characteristics, costs, and typical applications of non-selective flat-plate collectors is given below.

FIGURE 3

PERFORMANCE COMPARISON OF AIR AND WATER COOLED
COLLECTORS, NON-SELECTIVE COATING

Number of Cover Plates	Cost*/ft ²	$\alpha\tau$	U_L Btu/hr-ft ² -°F	Uses
0	\$1.5-\$4	0.92-0.95	Strong function of wind velocity	Swimming pool heating
1	\$7-\$11	0.82-0.88	1.2 - 1.4	Hot water heat- ing, space heating in moderate climates
2	\$9-\$12	0.69-0.78	0.85 - 0.95	Hot water heating, Space heating Air and water cooled versions available

* Collector cost, uninstalled.

The value of the collector heat removal factor, F_R , depends on the coolant used, its flow rate, the collector heat loss coefficient, and the absorber plate design. For most double glazed collectors of good design F_R will be between 0.9 and 0.96. Some collectors which use tubes clamped to an absorber plate may have lower performance due to poor heat transfer between the plate and the tubes.

2. Selective flat-plate collectors

A selective surface has a high absorptivity for solar radiation and a low emissivity for thermal (long wave length infrared) radiation. When applied to a solar collector, the selective surface decreases the radiative heat loss and thus more energy is transferred to the coolant. At higher temperatures (200°F or above) radiative losses are dominant and the increase in efficiency given by the selective surface is quite pronounced.

In Figure 4, the performances of liquid cooled selectively coated single and double glazed collectors are given. The selective coating is assumed to have $\alpha = 0.92$ and $\epsilon = 0.1$. This represents the best values that have been achieved to date for commercial selective surfaces. As in the case of the single and double glazed non-selective collectors, there exists a range of temperature

FIGURE 4
PERFORMANCE COMPARISON OF SELECTIVE COATED
COLLECTORS WITH 1 AND 2 COVERS

differences and insolation values where the single glazed collector outperforms the double glazed version. The calculations were performed for a low-iron glass cover plate ($\tau = 0.90$). With lower quality glass ($\tau = 0.86$) the differences are more pronounced.

From a comparison of the curves in Figures 2 and 4, it is seen that a selectively-coated single-glazed collector has better performance than one which is non-selective and double glazed. A selective coating can also improve the performance of an unglazed collector somewhat but its use is generally ruled out due to the effects of weathering. There is no data available yet on the performance of a selectively coated air-cooled collector although it is expected that its performance could be significantly improved due to the reduction of the heat loss coefficient, U_L , and the improvement, thereby, of the heat removal factor, F_R .

The chief question that must be addressed with regard to selective coatings is one of durability. Will the improvement in performance due to the selective coating hold up over the 20 year (or greater) lifetime of the solar? Will peeling, fading, or other failure modes cause the performance to actually be poorer over the long term than that of the non-selective coating? There is very little data with which to formulate an answer to the above questions.

Selective coatings have a short history of use in the United States. In Australia and Israel selective coatings have been used for many years in water heating applications but no hard data is available on the extent of any changes in the properties of the coatings. The use of a selective coating can also influence the durability of other components of the collector due to the high temperatures reached under stagnation or no-flow conditions.

There are three selective surfaces commercially available at the present time. These are:

(a) Black chrome - $\alpha = 0.92$, $\epsilon = 0.1$ (typically)

This coating is applied by electrodeposition on a nickel plated substrate.

The base material can be any metal. The durability of the coating with respect to the temperature and moisture has been found to be very good. The coating is available only at a very limited number of plating shops. Current costs of the coating are approximately \$1.50/ft². This cost is expected to fall below \$1/ft² under mass production.

(b) Copper oxide - $\alpha = 0.9$, $\epsilon = 0.15$

This coating can be applied only to copper absorber plates. It is formed by the conversion of the surface of the metal to copper oxide (CuO) by the action of a hot solution of salts and oxidizing agents. It is limited in its durability above temperatures of 400°F. The coating process lends itself to high volume and coating costs are around \$0.25/ft².

(c) Alcoa 655 process - $\alpha = 0.93$, $\epsilon = 0.35$

This is a proprietary process developed by Alcoa for use on aluminum collector panels. This coating will degrade at temperatures above 300°F by losing some of its solar absorptivity. The emissivity does not degrade and may even improve after exposure to high temperatures. The coating is applied by chemical conversion of the surface and like the copper oxide coating is suitable for batch processing. Coating costs at the present time are \$0.50/ft².

In summary, the benefits of using a selective coating are well established and in terms of increased performance are well known. What liabilities there may be in their use is not known. Prior experience in the use of selective coatings in Australia and Israel does exist but does not apply to the conditions found in the United States. The best selective coating available today is black chrome. The only drawback is the high cost of application. This increased cost can be offset by the decrease in collector area necessary to provide a fixed percentage of heat by solar energy. An example from the performance calculations for hot water heating is shown below. Collector type

used in the model is single glazed with a selective surface of emittance 0.4 or 0.1. The output hot water temperature was required to be 200°F. The insolation used was for Lincoln, Nebraska.

	<u>Area for 50% Solar</u>	<u>Area for 75% Solar</u>
$\epsilon = 0.1$	847	1825
$\epsilon = 0.4$	1063	3196

3. Flat Plate Collectors with Planar Reflectors

In the design of SHAC systems, planar reflectors can be used to enhance the insolation falling on the flat plate collectors. Some simple arrangements of collector - reflector arrangements are shown in Figures 5a, 5b, and 5c. When a reflector is employed, the collector area required for a fixed output can be decreased but only at an added cost for the reflectors. The reduction in required collector area with the reflectors for SHAC systems ranges from about ten percent to thirty percent. If the added cost of reflectors is less than the savings derived from the reduced collector area, the reflectors may be desirable.

(a) Some considerations in the design of SHAC systems

Selection or design of the best possible collector module in itself will not yield the most cost effective SHAC system. The optimum design of the system at any geographical location has to take the following into account:

1. The thermal loads and their distribution over the year.
2. The operation characteristics of various system components and their life time.
3. The initial costs and maintenance costs of the system components.
4. The present cost of conventional energy source and the expected fuel price escalation over the system life time.
5. The control strategy for the operation of the system.

FIGURE 5a

FIGURE 5b

FIGURE 5c

The items 1 to 4 mentioned above are the prime factors which determine what fraction (f) of the total annual load should be carried by solar. The thermal loads imposed on the buildings and the load distribution over the year primarily determine the performance optimum collector tilt angle. A typical curve is shown in Figure 6. A performance optimum tilt angle may not be the economic optimum because the cost of support structure is a function of the tilt angle but this dependence usually is very weak.

For a given tilt angle, the load carried by solar increases as the collector area (A_c) is increased for a fixed ratio of the storage to collector area (S/A_c). A typical variation is shown in Figure 7.

As the collector area (A_c) is increased for given S/A_c , each additional collector increment supplies less and less energy. Depending upon the cost of the solar system components and the cost of conventional fuel, we may reach a point when the cost of obtaining extra energy from the solar system by increasing collector area becomes equal to the cost of conventional energy. The collector area at this point is the one that will yield the least life-cycle-cost. In a similar fashion there is an optimum S/A_c ratio as well.

For a SHAC system we can write the following expression for the life-cycle-cost:

$$LCC = (1 - f) \bar{C} L_1 L_2 + (a + bS + c A_c) \quad (1)$$

where

LCC = life cycle cost in \$

f = fraction of annual load carried by SHAC system

$\bar{C}$ = average cost of conventional energy over the system life time in \$/MM Btu

L_1 = average annual load in MM Btu/year

L_2 = SHAC system life time in years

a = fixed initial costs of the SHAC system and the maintenance costs over the system life time in \$

FIGURE 6

FIGURE 7

b = cost of insulated and installed tanks in \$/gallon (including the maintenance costs)

S = storage volume in gallons

C = cost of installed collectors in \$/ft² (including the maintenance costs)

A_c = collector area in ft²

At optimum storage capacity S , $\frac{\partial LCC}{\partial S} = 0$ and therefore

$$\frac{\partial f}{\partial S} = \frac{b}{\bar{C} L_1 L_2} \quad (2)$$

and at optimum collector area A_c , $\frac{\partial LCC}{\partial A_c} = 0$

and therefore

$$\frac{\partial f}{\partial A_c} = \frac{C}{\bar{C} L_1 L_2} \quad (3)$$

Equations (2) and (3) are used to find the optimum combination of storage volume and collector area.

Besides the costs of various components of the SHAC system, the climatic factors also determine whether a SHAC system will be cost effective. Given the same costs of components and the same cost of conventional energy, the optimum fraction of load carried by solar will be different in different regions of the country. In the Phase I study, the country was divided into 12 regions described in Table I. In general, the higher the insolation and HVAC loads, the more cost-effective will the SHAC system be.

TABLE 1
CLIMATIC PARAMETERS FOR CHARACTERISTIC CITIES WITHIN CLIMATIC REGIONS

Region Number	Representative City	Characteristic Location	Heating Degree-Days	Cooling Degree-Days	Hours in Heating Season	Hours in Cooling Season	Average Change in Humidity Ratio*
1	Boston	Boston, Mass.	5634	661	6480	2160	.0037
2	Wash., D.C.	Wash., D.C.	4211	1415	5040	3600	.0024
3	Madison	Madison, Wisc.	7863	460	6480	2160	0
4	Los Angeles	Davis, Calif.	2819	1063	5760	2880	0
5	Charleston	Charleston, S.C.	2033	2078	4320	4320	.0055
6	Bismarck	Bismarck, N.D.	8851	487	7200	1440	.0003
7	Nashville	Oak Ridge, Tenn.	3944	1367	5040	3600	.0035
8	Ft. Worth	Midland, Texas	2621	2250	4320	4320	.0037
9	Omaha	Lincoln, Neb.	6218	1148	5760	2880	.0003
10	Seattle	Seattle, Wash.	5145	200	7200	1440	0
11	Phoenix	Trona, Calif.	2370	2888	4320	4320	0
12	Miami	Miami, Fla.	214	4038	720	7920	.0046

*Relative to indoor design conditions of 78°F dry bulb and 50% relative humidity.

B. The Solar Energy Concentrating Systems

1. Introduction

(a) A concentrating collector is generally defined to be a collector containing mirrors, reflectors, refractors, and/or optical lenses which direct incoming solar radiation onto the absorber whose surface area is smaller than the collector's aperture. The aperture is the projected area of the solar concentrator through which unconcentrated solar energy is admitted. The absorber is that part of the collector which absorbs the radiation directed upon it, usually transferring the absorbed energy to a working fluid. The concentration ratio is generally defined to be the ratio of the aperture area to the absorber area, although care must be taken with each concentrator collector type to insure that a meaningful parameter is selected.

(b) General Describing and Technical Considerations

Solar energy, as it naturally falls onto the earth's surface, is low in intensity on a unit area basis. The simplest form of collector, the flat plate collector, attempts to capture this energy by letting the sunlight fall, unredirected, upon an absorbing surface. Large absorber areas are thus required to collect a given amount of heat. Consequently, heat loss to the surroundings by convection and conduction can be quite high since this heat transfer mechanism is approximately proportional to the absorber area. Therefore, one objective of concentrating collector systems is to significantly reduce this heat loss to the surroundings. This is accomplished by redirecting the incoming radiation onto a smaller area. In addition to reducing the conduction and convection losses, the smaller absorber area will result in reduced radiative losses for the same emissivities and absorber temperature.

The higher solar energy flux per unit area and corresponding lower thermal losses per unit area for a concentrating system compared to a flat plate

system, means that more energy must be transmitted to the working fluid in order to satisfy the first law of thermodynamics. The advantage of the concentrator systems is that this energy transfer can take place at higher temperatures for the same efficiency.

The previous discussion points out the fundamental differences between concentrating and flat plate collectors; it does not point out the losses unique to concentrator systems. Concentrating systems are shackled by the following technical problems and fundamental limitations:

Solar Radiation Beam Divergence

The sun is a finite disk in the sky having an angular divergence of about 0.5° . In addition, intensity of light emitted from the sun varies over the disk, the intensity decreasing from the disk's center. As a consequence, the spot or line upon which the sun beam is focused is finite in size and non-uniform in intensity. This in turn limits the ultimate concentration ratio.

Specular Reflectives

The specular reflectivity (that is, the fraction of the incident beam which is reflected such that the angle of reflection equals the angle of incidence) for real surfaces varies from .85 for a commercial Alzac process aluminum to .96 for fresh electroplated silver. Only a negligible fraction of the non-specularly reflected beam will be captured by the absorber.

Intercept Factor

This parameter is the fraction of the specularly reflected radiation which strikes the absorber. This parameter is important because for a real concentrator design, the radiation intensity in the focal plane has a nonuniform distribution. Thus the size, shape, and position of the absorber in relation to this nonuniform distribution directly influences the fraction of specularly reflected radiation which will strike the absorber. In addition to the solar beam divergence resulting from a finite sun disk, this distribution results

from microscopic surface irregularities (angular dispersion), the macroscopic deviations from ideal reflector shape (dependent upon the accuracy of manufacturing process), misalignment of the absorber relative to the focal spot or line, and inaccurate tracking of the sun. The larger the absorber area, the larger will be the intercept factor, but also larger will be thermoconductive convective and radiative losses. Thus a tradeoff is implied, a tradeoff which ultimately involves cost factors.

Direct Radiation

Although there are a number of nonimaging concentrating collectors having a concentration ratio of less than about ten which will collect a portion of diffused radiation, most concentrators which will be needed to produce high temperatures require direct beam radiation. These systems are unable to produce any significant usable energy under cloudy or very hazy conditions since only the diffuse rays parallel to the focal axis and those directly striking the small absorber tube will be absorbed. Table 2 indicates the fraction of values of total monthly average daily insolation which is direct or beam radiation and is thus suitable for collection by concentrator systems. The quantity $(1 - f_b)$ represents the fraction which is diffuse and represents the additional component of insolation which can be collected by flat plate collectors.

(c) Concentrator Design

Many concentrator designs have been proposed. As of June 1975 at least 26 designs have been patented. Since then, at least three new designs being given serious consideration have been patented or a patent applied for. All

TABLE 2
FRACTION AND VALUES OF INSOLATION WHICH IS DIRECT FOR SEVERAL LOCATIONS

Month	Phoenix		Washington, D.C.		Boston		Madison		Bismarck	
	f_b	H_b	f_b	H_b	f_b	H_b	f_b	H_b	f_b	H_b
January	.75	845	.58	367	.55	278	.62	350	.70	411
February	.78	1181	.60	541	.57	421	.61	495	.73	682
March	.80	1574	.63	791	.58	619	.63	776	.70	930
April	.88	2102	.63	1008	.57	773	.61	887	.67	1118
May	.83	2249	.64	1182	.62	1097	.62	1082	.69	1419
June	.82	2281	.66	1373	.62	1156	.65	1321	.68	1478
July	.76	1862	.65	1254	.62	1163	.67	1371	.74	1706
August	.77	1771	.64	1096	.61	958	.64	1114	.71	1369
September	.81	1726	.65	940	.61	774	.66	953	.69	994
October	.80	1351	.63	682	.59	529	.63	625	.69	702
November	.75	967	.60	458	.52	279	.55	306	.63	378
December	.75	781	.60	356	.54	239	.60	298	.65	302

NOTE: f_b is the fraction of the total monthly average daily radiation which is direct or beam radiation.
 H_b is the monthly average daily direct or beam radiation.

these designs more or less attempt to eliminate or minimize one or more of the losses discussed in Section b. Time will be required to sift through all the possibilities to isolate the most useful few.

Cross sectional sketches and artist renditions of those concentrator designs considered to have the greatest potential for engineering applications are indicated in Figure 8. A little study of these various designs suggest that each concentrator can be characterized in a number of ways. These are indicated with each sketch. A solar concentrator design can be classified as either imaging or non-imaging. An imaging concentrator will attempt to produce an image of the sun in its focal plane whereas a nonimaging collector redirects in a random manner the incoming solar radiation onto an absorber surface. Most solar concentrator designs also attempt to produce either a point focus or line focus. These points or lines are, of course, finite in size. Non-imaging collectors, however, cannot be so readily classified as producing a point or line image (see for example Figure 8b and 8d). Concentrator systems can also be classified as requiring suntracking or they can be nontracking or partially tracking in the sense that collector orientation may need to be altered a finite number of times per year. The focal image is produced by either reflectors, refractors, lenses, or in a few cases a combination of several of these. There are several methods by which the collector may track the sun. One method would leave the absorber such as a copper tube stationery while adjusting the orientation of a matrix of, refractors (Fresnel type lenses), lenses such that the focus remains on the absorber tube. The other basic technique would involve leaving the reflector or concentrating arrangement fixed while adjusting the orientation and geometrical position of the absorber to follow the changing position of the sun. Table 3 presents such a categorization of those generic concentrator designs presented in Figure 2.

Those collector systems illustrated in Figure 8 and Table 3 have been or are currently the subject of serious studies. After a thorough search of the literature and contacts with manufacturers, however, we have found that

figure 8: GENERIC SOLAR CONCENTRATOR CONCEPTS

FIGURE 8 (CONT'D)

(g)

(h)

FIGURE 8 (CONT'D)

(i)

(j)*

(k)

FIGURE 8 (CONT'D)

(1)

FIGURE 8 (CONT'D)

TABLE 3
CATEGORIZATION OF CONCENTRATOR TYPES

Concentrator Design (Refer to Figure 2)		a	b	c	d	e	f	g	h	i	j	k	l
Generic Features													
1. Concentration													
High (>10)		x		x				x	x	x	x	x	x
Medium (3-10)			x			x							
Low (1-3)					x		x						
2. Focus Geometry													
"Line" (Strip)		x	x	x	x	x	x	x	x				
"Point" (Disk)										x	x	x	x
3. Focus Type													
Image		x								x	x	x	
Non-Image			x	x	x	x	x	x	x				x
4. Tracking Requirement													
Tracking		x	x	x									
Non or Partial								x	x	x	x	x	x
Tracking					x	x	x						
5. Tracking Method													
SC/TA*			x					x	x		x		
SA/TC**		x		x						x		x	x
N.A.					x	x	x						
6. Focusing Mechanics													
Mirrors		x	x		x	x	x	x	x	x	x	x	x
Lenses										x			
Refractors				x									
7. Normal Orientation													
Single Axis													
North-South		x		x									
East-West			x		x	x	x	x	x				
Double Axis										x	x	x	x

*SC/TA - Stationary Concentrator/Tracking (Moving) Absorber

**SA/TC - Stationary Absorber/Tracking (Moving) Concentrator

reasonable and reliable performance data over a wide range of parameters is lacking for most of these concepts. The status of these systems range from a gleam in someone's eye to large scale commercial production. Our opinion is that those systems which are currently in production or very near large scale production represent those generic designs which are likely to gain widespread acceptance during the next ten to fifteen years. This assumption by no means implies that the best concentrator designs have been or are being produced; rather the assumption has been made because these represent the only systems for which reasonable cost data is available or can be estimated and for which reasonable performance data can be estimated or is available.

2. Compound Parabolic Collectors

(a) General Description and Characteristics

The compound parabolic solar collector is currently the only concentrating collector being seriously developed which does not produce an image. Instead, this collector focuses the sunlight in a diffuse manner upon the absorber. This collector was invented by Roland Winston, a physicist at the University of Chicago, who was attempting to design light-collecting instruments for physics experiments. The concept has been referred to as the Winston collector.

Using elegant mathematical techniques, Winston was able to show that the maximum concentration of a divergent beam of light allowed by phase space conservation is given by the following equation:

$$\frac{d_1}{d_2} = \frac{n_2}{n_1} \frac{1}{\sin \theta_{\max}}$$

where n_2 is the index of a fraction at the exit, n_1 is the index of a fraction at the entrance, and $\theta_{\max}$ is the maximum beam divergence half angle. This equation applies to cylindrical troughs or the two-dimensional case. Figure 9

Figure 9: Cross-Sectioned View of Cylindrical Compound Parabolic Collector Concept

illustrates these parameters and provides a general idea of the collector design. Only those light rays entering the collector within the acceptance angle, $\theta_{\max}$, will strike the absorber situated at the exit aperture, d_2 . Light entering at angles greater than $\theta_{\max}$ will be reflected back out of the collector. Note also that the determining angle is the angle that the projection of the sun ray onto the plane perpendicular to the trough axes (plane of the paper) makes with the optic axis (see Figure 9). Figure 10 illustrates the relationship between the concentration ratio, C , and the maximum acceptance angle, $\theta_{\max}$.

The amount of diffuse hemispherical radiation accepted by this type of collector is precisely the reciprocal of the concentration ratio. This relationship is obtained from geometric considerations in which the field of view of the collector from any point on the entrance aperture is given by an infinite rectangular strip whose width subtends an angle of $\pm \theta_{\max}$. From this it follows that the solid angle for diffuse light is $4 \theta_{\max}$ steradians.

Utilizing the same phase space conservation laws, Winston was able to derive an expression for the depth of the collector trough, which is given by the following:

$$H = (d_1 + d_2) \cot (\theta_{\max})/2$$

This is the minimum depth required to satisfy the phase space conservation laws. This relationship normalized to the absorber width, d_2 , is presented in Figure 11. As this figure illustrates, one of the main disadvantages of the compound parabolic concentrator is the extreme depth of the trough required to obtain a given concentration ratio. Thus at a concentration of four, the collector trough depth is 10 times the absorber dimension. The staff at Argonne National Laboratory has investigated this problem and found that fortunately, a large portion of the top of a compound parabolic

Figure 10: Relationship between Concentration Ratio, C , and Maximum Collector Acceptance Half Angle.

Figure 11:

Relationship Between Concentration Ratio, C , and the Ratio of Theoretical Through Depth, H , to Absorber Width, d_2 .

concentrator can be truncated with little loss in performance. A truncation of the collector reduces the concentration ratio but does not affect the angular acceptance. While no complete economic studies have been conducted to date, it is generally believed that most compound parabolic concentrators will be truncated to about one-third of the theoretical height. Figure 12 illustrates the relationship between the concentration ratio, the acceptance half angle and the height-to-aperture ratio.

To be able to model the performance of this collector over the course of a day, a month or a season, it is important to be able to determine whether a given ray of direct radiation will be accepted by the collector. This so-called "view angle" equation has been derived and is given by the following:

$$\cos \theta_v = (\cos \theta_c) / \sqrt{1 - K^2}$$

where $\cos \theta_c = \sin \delta \sin L \cos \Sigma - \sin \delta \cos L \sin \Sigma \cos \psi$
 $+ \cos \delta \cos L \cos \Sigma \cos H + \cos \delta \sin L \sin \Sigma$
 $\cos \psi \cos H + \cos \delta \sin \Sigma \sin \psi \sin H$
 $=$ standard cosine of the angle formed by the ray
with the collector normal

and $K = \sin H \cos \psi \cos \delta - \sin \psi \sin L \cos H \cos \delta + \sin \psi$
 $\cos L \sin \delta$

where δ = solar declination
 L = latitude
 Σ = collector tilt angle from horizontal
 ψ = collector orientation from South, East positive
 H = hour angle

Thus, any solar beam radiation for which θ_v is greater than the maximum acceptance angle, will not be collected. However, one over the concentration ratio of the diffuse portion will be accepted by the collector.

Figure 12: Trough Height/Aperture Area for Full and for Truncated Compound Parabolic Collector (θ =Acceptance Half Angle)

Most of the developmental work on the compound parabolic concentrator has taken place at Argonne National Laboratory. Members of this Lab have extensively studied the collector characteristics, and a number of the major works are referenced at the end of this section. Although this collector is still in the development stage, three manufacturers currently have licenses to produce this collector. They are the Steelcraft Company in Memphis, Tennessee, M-7 Corporation, and Chamberlain Manufacturing.

Steelcraft Company appears to be most seriously intent on manufacturing the CPC collector. They have invested significant corporate monies to test and modify a design which they intend to manufacture in late Fall, 1976. The company considers most of the design details at this stage as proprietary.

The M-7 Corporation is considering an application of the CPC to photovoltaic applications. They are bonding a solid lucite CPC onto a photocell. The design is interesting because, having a higher index of refraction than air, the lucite will accept radiation at angles larger than the acceptance angle. Also all reflections are internal; thus no silvered surfaces are required. No further consideration has been given to this collector since it is not a purely thermal application.

The status of the potential production of this collector by Chamberlain Manufacturing is still unclear. This company constructed two CPC's for Argonne National Laboratory. These collectors have been recently tested by Argonne National Laboratory, and their preliminary results are presented in the subsequent section in this report. More specific details concerning collector construction should be available in a forthcoming progress report from Argonne.

(b) Performance

It has been only within the past year that the CPC collector has graduated from being strictly a laboratory device to being tested and refined to develop

possible production models. Currently, three private firms intent on mass producing this collector are testing and modifying designs, and one government laboratory (Argonne National Laboratory) is also testing a variety of configurations. The three private firms (those discussed in Section A) currently consider their performance data proprietary. The only performance data available to date has been provided by Argonne National Laboratory.

While theoretical and experimental performance data are available for a number of concentration ratios and collector designs, unfortunately a detailed description of the various collector configurations used to develop the performance data will be presented in a forthcoming progress report, and are not yet available.

Performance curves of several compound parabolic collectors currently being studied by Argonne National Laboratory and represented in the National Bureau of Standards format are illustrated in Figure 13. These curves illustrate the wide variation in performance which can be expected from concentrators having substantial design difference. Curve number one illustrates the theoretically expected performance of a times three concentrator whose absorber is surrounded by a vacuum. The absorber in this case is a strip of copper or steel plate coated with black chrome, surrounded by an evacuated glass tube. The glass tube is considered to be cradled at the bottom of the concentrator troughs which are thermo-formed ABS plastic (see Figure 14). Vacuum-deposited aluminum forms the reflective surface for the troughs. Conduction of heat through the glass face between the glass and the absorber tube is assumed to be negligible since the gas pressure is assumed to be below 10^{-5} torr. Also, heat loss through the glass wall at the glass-to-metal interface is assumed to be minimized through the appropriate use of insulation. Thus, the primary mechanism for heat loss in this particular collector is radiation, and this accounts for the rapid drop-off in performance for $\Delta T/I$ above about one. It is also important to point out that this performance curve is based on average absorber temperature and not working-fluid temperatures.

Figure 13: Performance Characteristics of Several Compound Parabolic Collectors Developed at Argonne National Laboratory

Figure 14: Reflector and Receiver Cross Section for CPC Providing Performance Indicated by Curve No. 1 in Figure 5.

Performance data for concentrators having concentration ratios of 5.3 and 10 are illustrated in curve numbers 2 and 3. The significant difference between these curves and curve number 1 is accounted for by the fact that these collectors have nonevacuated absorbers painted with a flat black nonselective paint. The mirror surfaces are of Alzac. Since these absorbers are non-evacuated, conduction and convection losses through the glazing, sides, and the back are significant. Furthermore, heat loss from the absorber through the insulation to the back and sides of the collector is aggravated by the fact that the thermal conductivity of the insulation is greater than would be expected for a flat-plate collector due to the higher operating temperatures of the CPC collector (the thermal conductivities of most insulations increase with temperature).

Furthermore, it appears that the absorbers for these collectors are in thermal contact with the extruded aluminum reflecting troughs. Hence, the troughs act as heat dissipation fins. Except for very high concentration ratios, it does not appear that the nonevacuated CPC-type collector will perform sufficiently better than the flat-plate collector to justify the increased cost associated with the required large reflecting surfaces. Refer to section on costs.

The "optical" efficiency is a useful measure of how well the collector will transmit incoming radiation to the absorber surface. This efficiency is given by the following equation:

$$\eta_o = \gamma \tau \rho \langle n \rangle \alpha$$

In which τ is the transmissivity of the collector cover, α is the absorptivity of the absorber, γ is the fraction of the total solar radiation accepted by the concentrator, ρ is the specular reflectivity and $\langle n \rangle$ is the average number of reflections which the spectrum of radiation within the acceptance angle undergoes. The average number of reflections can be determined from Figure 15.

Fig. 15: Number of Reflections for Full and for Truncated CPC, averaged over all radiation which can get from absorber to aperture. For a truncated CPC, the line $1 - 1/C$ can be taken as lower bound for the number of reflections for radiation within the acceptance angle.

(c) Compound Parabolic Collector/System Costs

To date, there have been no installations of working, nonexperimental systems employing compound parabolic collectors, and thus no actual system costs available. Indeed, to date no collectors have been produced except for prototype developmental models. Private communications with Argonne National Laboratory and engineers associated with firms interested in marketing this collector indicate that the first production models will cost between \$20 and \$30 per square foot of aperture area. This cost is the capital cost of the collector module only and does not include installation or tilt-adjustment systems.

(d) Tracking Requirements

One of the advantages the CPC collector is viewed as having over other concentrator collectors is that of minimal tracking requirements. These collectors require only a relatively small number of tilt adjustments, the exact number depending on the concentration ratio and the desired performance. Table 3 illustrates the number of tilt adjustments required for several concentration ratios such that the minimum collection time per day is seven hours. Generally, performance would improve only slightly for additional tilt adjustments because only the lower-intensity sunlight in the early morning and the later afternoon would be collected. However, fewer tilt adjustments would result in a significant drop in performance since collection time would drop below seven hours per day, thus "topping off" the capture of direct radiation later in the morning and earlier in the afternoon. To date, these effects have not been quantified on a seasonal system performance basis.

It is not clear at this time precisely how best to physically make the tilt adjustments. Factors which will ultimately influence the method include:

1. The number of tilt adjustments required, as the fewer required, the more likely these adjustments can be made manually during periodic maintenance and inspection.

TABLE 3
ADJUSTMENTS OF COLLECTOR TILT

Require minimum collection time 7 hours/day (except for $\theta = 5.5^\circ$ min. collect. time = 6.78 hours/day)

Accept. Half Angle (Conc. of full CPC)	Collection Time Aver. Over Year (hours/day)	Number of Adjust./Year	Shortest Period w/o Adjustment	Average collection Time if Tilt Adjusted Every Day (hours/day)
19.5° (3.0)	9.22	2	180 days	10.72
14° (4.13)	8.76	4	85 days	10.04
11° (5.24)	8.60	6	35 days	9.52
9° (6.39)	8.38	10	24 days	9.08
8° (7.19)	8.22	14	16 days	8.82
7° (8.21)	8.04	20	13 days	8.54
6.5° (8.83)	7.96	26	9 days	8.36
6° (9.57)	7.78	80	1 day	8.18
5.5° (10.43)	7.60	84	1 day	8.00

2. The concentration ratio as it affects the required depth of the collector and thus determines how bulky or awkward the collector may be to install and to alter tilts.

3. The number of reflective troughs per collector module as this influences whether the tilt should be made by only the trough or the entire collector module.

It seems clear, however, that the final method developed for adjusting collector tilt for a times three concentrator will be significantly different from a times 10 concentrator. For example, the daily tilt adjustment required for a times 10 concentrator certainly implies a motor drive. The frequency of tilt adjustments further suggests that a continuous tracking system would be the likely candidate.

(e) Problems and Limitations with the Compound Parabolic Collector

Best estimates for the cost of a high-quality, compound parabolic collector utilizing a selective absorber enclosed by an evacuated tube and plastic, vacuum-deposited aluminum reflector troughs are in the neighborhood of \$20-\$30 per square foot of aperture area. While no experimental data are yet available, theoretical calculations indicate that this collector can significantly outperform a standard flat-plate collector seasonal performance in certain operating regimes. The improved performance does not, however, appear to outweigh the greatly increased cost over flat-plate collectors. It would be preferable if a less sophisticated but still high-performance collector could be developed. However, several performance and materials relationships serve to limit the available design flexibility of this collector.

First, it would be desirable to remove the evacuated tube (a very high-cost, low-reliability item) and simply utilize a flat absorber strip or tube located at the reflecting trough exit aperture. This would reduce capital cost to a

limited extent, because insulation must then be added to the back of the collector where it previously was not needed. Furthermore, the temperature of the materials comprising the reflective troughs (and the whole collector) would then rise substantially. If the reflector components are constructed of plastic, a strong possibility exists for their melting during no-flow conditions. Furthermore, performance will drop substantially (see Figure 5) while the drop in cost may not be sufficient to affect this. Although polished aluminum or silvered glass could conceivably be employed as the reflective troughs, performance would deteriorate (due to greatly increased thermal conductivities) to a level not significantly better than a standard flat-plate-type.

One design option does, however, appear to be worth serious consideration. This design involves the use of a nonevacuated tube surrounding the absorber strip to reduce costs associated with an evacuated system. Plastic reflective troughs could then conceivably still be utilized provided adequate ventilation of the collector module could be maintained. No known investigations of this option have been performed to date.

Second, providing high-quality reflecting troughs is an inherent requirement for the compound parabolic concept. This materials requirement also contributes substantially to collector cost, and relatively little can be done to overcome it. Vacuum-deposited aluminum on extruded plastic troughs appears to be the most likely design at this time, but considerable care must be taken to prevent overheating and subsequent melting. Cast or machined aluminum is another possibility, although materials cost, added weight, and high thermal conductivity do not make this choice obvious.

In addition to the high cost of the collector, the unique periodic tilt adjustments add special requirements for flexible piping connections and provisions for either automatic, motorized or manual, hand-cranked tilt adjustments. As yet no solutions to these requirements have been found.

Considerable additional materials research and development and system studies must be performed before definitive statements regarding the applicability of this collector to various energy supply requirements can be made.

(f) Compound Parabolic System Installations

To date, there are no known installations of this collector type into working, nonexperimental systems. The U. S. Department of Housing and Urban Development has, however, selected one demonstration project (not yet complete) which will incorporate this collector type (to be produced by Steelcraft Company of Memphis, Tennessee). The project will consist of 860 square feet of collector area on eight units of student housing totaling 8,479 square feet. The location will be Boulder, Colorado; but no other details are as yet available.

3. Tracking Parabolic Cylinder

(a) General Description and Characteristics

A parabolic-cylinder collector consists of a specularly reflecting trough whose cross-section is parabolic in shape. The trough length ranges from four to ten feet, although no optimum length is implied by these dimensions. The parabolic trough aperture width varies from one to six feet. In this arrangement, the focus of the parabola forms a line which runs the length of the trough. Along this line is placed the solar energy receiver, usually a copper or stainless steel pipe. To reduce the radiative and convective losses, the absorber is enclosed by a glass tube, the annulus being either evacuated or non-evacuated. Collectors currently available in the market are non-evacuated.

Working fluids for these collectors range from water to several heat transfer fluids (such as Therminol-66) available from the major oil companies. The fluid flow rate through the absorber tube is adjusted to ensure that the flow is turbulent (Reynolds number greater than 2300). Experience reported by Sandia Laboratory indicate erratic and unpredictable results using flow rates in the laminar or transitional regimes.

For these collectors to operate it is essential that they track the sun. By tracking is meant that the sun line and the focal axis must be in the same plane. In this way, the beam radiation will be focused onto the absorber tube situated along the focal axis. This type of tracking is accomplished by leaving the orientation of the absorber fixed with respect to the earth and rotating the reflective trough about the absorber axis.

(b) Performance Characteristics

The concentrating collectors are able to operate efficiently at high temperatures because the surface area where the loss occur is small. Typical efficiency curves for four different parabolic cylinder concentrators are presented in Figure 16. Also presented in this figure is the efficiency curve of a good flat plate collector. At low temperatures there is no advantage in using parabolic cylinders, but as the operating temperature requirements increase, the efficiency of the parabolic-cylinder collectors becomes much higher than that of the flat plate collector.

It should be noted, however, that these collectors will not collect diffuse sunlight. A fraction of the diffuse radiation equal to the reciprocal of the concentration ratio will be absorbed by the absorber; however, this fraction is so small for concentrators of interest as to be negligible. In this regard, flat plate collectors are likely to perform better in regions known for consistent hazy and cloudy skies. However, preliminary in-house studies of a system utilizing parabolic cylinder collectors indicates that even for high diffuse/low direct insolation situations, the parabolic cylinder may be able to make more efficient use of the direct component than the flat plate collector may of both the diffuse and direct components. Much additional work must be done to precisely identify cost effective applications for this collector type.

(c) Comments on Optimum Tracking and the Collector Geometry

In a 1973 report to the NSF by the University of Minnesota and Honeywell,

Figure 16: Performance Characteristics of Several Tracking Parabolic Cylinder Collectors

results of an extensive study on the optimization of collector geometry were presented. The important results of that study are as follows:

1. For single axis tracking the equatorial or polar mount (which has the collector axis of rotation/tracking aligned north-south but tipped up out of the horizontal plane by an angle equal to the local latitude) is the best orientation on the basis of the total solar flux intercepted over a year. In this tracking configuration, the tracking angle varies linearly with time and therefore a "clock-drive" can be used for tracking.
2. The optimal concentration ratio with perfect optics and selective absorber coating is 35. The optical imperfections lower this optimal value to 12. A non-selective black absorber has the same optimal concentration ratio for perfect optics, but optical imperfections have a sharper effect in this case. As a result the optimal concentration ratio here has a value of 20.
3. The optimal rim angle is in the interval from 90 to 130°. The stochastic scatter does not permit the determination of a single optimal value. A rim angle of 115° has an advantage that the collector is mass-balanced about the focal line, assuming a uniformly thin mirror.

(d) Cost of Parabolic Cylinder Concentrators

Cost data for Parabolic Cylinder Concentrators (PCC) is available only from three manufacturers. The DEL Manufacturing Corporation's collector has a concentration ratio of about 24 with perfect optics, and they expect the price to be in the range of \$20 to \$30 per square foot of aperture area. The Aerotherm Acurex Corporation has quoted a price of about \$15.50 per square foot for PCC which has a concentration ratio of about 58 with perfect optics. The prices mentioned above are for the collectors including mirror, absorber, absorber cover, support, and tracking mechanism. Sun Power Systems in Tempe, Arizona is producing a parabolic cylinder collector having a 4' aperture. While no performance data has been made available, published costs are about \$10 per ft² of aperture area. This cost figure apparently includes a simple frame on which are mounted three collector units, insulated manifolding, and the tracking mechanism.

4. Fresnel Refractor Collectors

a. General Characteristics

The only collectors of this type available currently are manufactured by Northrup. The collector incorporates a Fresnel-type lense which concentrates the solar radiation onto an absorber. The Fresnel-lense is made up of many prisms which refract the light passing through them. A view of the collector as being currently manufactured is shown in Figure 17. The sun following capability of the collector enables it to begin collecting solar energy shortly after sunrise and to continue collection until just before sunset. The collector is capable of generating high temperature ($220^{\circ}\text{F}+$) heat at good efficiency but its utilization is limited to the areas which receive relatively larger fraction of beam radiation because its performance is very poor even on hazy days.

b. Performance Characteristics

No theoretical model has yet been developed for the performance evaluation of a thermal collector involving linear Fresnel lens. The only experimental data available for this type of collector was obtained by the University of Texas at Arlington using Northrup collector. The experimental data is shown in Figure 18. According to researchers at the University, the experimental data should not be extrapolated. This collector collects about two percent or less of diffuse radiation and its performance is very poor even on hazy days.

c. Cost of Collectors

The cost of collectors and tracking arrangement is about \$16/ft² and this includes manifolding, framing, insulation, and tracking arrangement.

5. Tubular Collectors

Tubular collectors (see Figure 19) are cylindrical in shape and have a relatively high vacuum ($\sim 10^{-4}$ torr) between the inner collecting surface and

FIGURE 17

FIGURE 18: Based on Total Insolation Which For These Tests Was Assumed to Be All Direct

FIGURE 19

an outer transmitting surface. The high vacuum eliminates a large fraction of the convective/conductive losses usually associated with flat plate collectors. In addition, the inner surface is usually coated with a selective absorber of some type (emissivity ≤ 0.2) thereby reducing the radiative losses considerably. Heat removal from this collector is accomplished by introducing a fluid through a separate tube located inside the collection tube and allowing the fluid to flow out between the space of the collection tube and the fluid inlet tube. Generally these tubes are mounted in a frame as shown in Figure 20 with a gap between adjacent tubes. A white diffuse scattering surface is below the tubes and enhances the collectors overall performance. The performance of this collector can be described using the equation shown in Table 4. A plot of the instantaneous efficiency versus $(T_{in} - T_{amb})/S_p$ is shown in Figure 21. It can be seen from this figure that this type of collector can operate efficiently at relatively high operating temperatures ($T_{in} \leq 250^\circ\text{F}$) and a low incident flux levels, S_p . This collector has the unusual characteristic that its peak performance does not come at solar noon for the geometry shown in Figure 20. Because of this fact, a designer has more flexibility as far as incorporating the collector into a building.

Currently this collector is only manufactured by one firm in the United States, Owens-Illinois, although Corning has done a great deal of research on the concept. The Phillips Research Laboratory in the Netherlands has also manufactured collectors similar to these. The Owens-Illinois collector is currently being or will be installed on the Colorado State University House III, the Terraset Elementary School in Reston, Virginia and on several other installations.

For the remainder of 1976, the collector is being sold for experimental purposes at \$20 per square foot of applied area. It is projected that the prices for 1977 will be in the range of \$15-20 per square foot, depending upon the quantities involved.

FIGURE 20

TABLE 4

- THE EQUATION AND PARAMETERS WHICH DESCRIBE
TUBULAR COLLECTOR PERFORMANCE

$$\dot{Q}_u = A_c F_R [\alpha \tau S_{\text{eff}} - \pi U_L (T_{\text{in}} - T_a)]$$

$$A_c = 0.47 \text{ ft}^2$$

$$F_R = 0.97 - 0.98$$

$$\alpha = 0.85 - 0.9$$

$$\tau = 0.92$$

- $S_{\text{eff}} = 500 \text{ BTU/hr ft}^2$
- $U_L = 0.1 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F}$

FIGURE 21

III. CONTROLS

A. Introduction

The spectrum of control systems for solar heating and cooling installations ranges all the way from use of standard thermostats to custom-designed solid-state electronic controls for highly specialized systems. In this section we shall discuss a number which are believed to span the spectrum, and take note of several others at the "specialized" end of the range.

There are three broad categories within the spectrum. At the low or unsophisticated end are the ordinary thermostats used for residences, and simple systems using differential thermostats with two or three temperature sensors. Midrange are the solid state or relay operated electronic systems, custom designed for particular installations of fixed configuration. At the complex end are the large systems using standardized HVAC control components in various arrangements to suit the current needs. Such systems are usually contracted for in entirety by a major controls supplier, who constructs his own signal/control diagram or truth table for the logic flow process, and builds the system accordingly.

We shall discuss typical controls or systems in each category.

B. Single Thermostat Controls

A single thermostat can provide the simplest direct control of a solar heating system. This control consists of a thermostat (or other temperature sensor) mounted in a configuration similar to that of the solar collectors. As the sun warms the thermostat above a set temperature, the collector pump is activated. As the sun sets (or if clouds obscure the sun), the temperature will drop below the set-point and the pump will shut down. A photo-cell or silicon cell could be used in place of the temperature sensor.

The single thermostat type of control is useful for certain specialized types of applications. One example is the control of a solar system providing heat to a large thermal mass operating near ambient temperature (swimming pool, sewage lagoon, etc.). In these cases the inlet temperature to the collector is approximately constant, allowing a single set-point temperature for solar collection to be used. The solar collectors used for these applications are often unglazed. This is because the operating temperature of the collectors is near ambient temperature and thus the maximum economy is obtained with unglazed, uninsulated collectors. However, in order to operate efficiently, the flow rate through the collectors must be high in order to minimize thermal losses from the collector surface to the outside air.

The recommended flow rate for one type of plastic solar collector used for swimming pool heating is 50 pounds of water per square foot of collector area. Since the energy collection rate would normally be 150 - 200 Btu/hr-ft², the collector is only 3-4°F. This temperature difference is too small to use for control of the system because of the tolerances available in commercial temperature sensors. Thus, a single thermostat control will provide for more efficient operation of the system.

If the ambient temperature and/or the storage temperature varies significantly over the year, the set point temperature must be varied. Otherwise useful energy could be lost by either not operating the collector (set point too high) or operating it when the collector is losing heat (set point too low).

C. Differential Thermostats

The majority of solar systems involve the use of solar collectors to supply energy to a storage tank of limited thermal capacity. The economic optimum for the heat capacity of the storage medium (water for liquid collectors, rock for air collectors) is approximately 10 - 20 Btu/°F per square foot of collector area. A typical day's energy collection might be 600 Btu/ft² which would correspond to a rise in temperature of the storage tank of 30 - 60°F.

A single thermostat control cannot efficiently operate the solar system over such a large span of collector inlet temperatures. The most efficient operation of the system is with a differential type of control with the temperature difference between the collector outlet and the storage tank being used as the controlling parameter. In theory, the collector pump should be on/off whenever the collector temperature is higher/lower than the storage temperature. Such a control would, however, lead to a rapid cycling of the pump under partly cloudy conditions.

Most differential thermostatic controllers manufactured at the present time have a built-in hysteresis in the control points to prevent pump cycling. The turn-on temperature difference is typically 10°F - 20°F while the turn-off differential is 3°F - 5°F . Since the turn-off temperature difference is at a point where the system is still collecting energy, it is important that the collector flow rate be sufficiently small so that the energy collection rate at the turn-off point is small. If, for example, the flow rate were 20 pounds of water per square foot of collector and the turn-off differential were 5°F , the energy collection rate would be 100 Btu/hr-ft^2 at the turn-off point. In the above example, the system would resume operation after shut-down because sufficient energy would be available to warm the collector past the turn-on temperature difference. Useful energy will be lost while the system is off, however.

The location of the collector outlet temperature sensor is very important for efficient system operation. If the sensor is located outside the collector on the outlet pipe, the system will be slow to turn on due to the temperature of the sensor lagging behind the temperature of the collector plate. The ideal location of the sensor is on the collector plate near the outlet. This requires that the temperature sensors be able to withstand the maximum stagnation (no-flow) temperatures obtainable by the collector (in excess of 400°F for high performance models).

Some differential thermostat controls include a third temperature sensor which can be used for freeze protection or oven-temperature protection. Others have options for temperature readout and user adjustable control set points. Some controls are U.L. listed and do not require an electrician for installation.

Most commercial controls use either thermistors or wire-wound resistance temperature sensors. The sensors must be well matched and stable over the operating temperature of the collector.

A partial list of manufacturers of differential thermostat controls is given below in Table 5. The prices of the controllers vary from \$30 to \$80 depending on the number and types of features supplied.

D. Advanced Solid State Control Systems

Two examples of solid state electronics control systems are discussed here. The first is the new custom designed system being installed at the Fauquier High School in Warrenton, Virginia under an ERDA contract. The second is a simpler version of the same system, but designed for a single storage tank instead of two.

Both systems are best described by reference to their specifications. It should be mentioned that a Fauquier High School two-tank control system was built to replace an earlier relay-operated control system which proved to have infant mortality problems of sufficient magnitude to warrant its replacement. Reference (1) is the specification for this new system. It is included here as Appendix A.

The system provides eight automatic modes of operation for two tanks, three pumps, and ten electrically operated valves. Both tanks may now be used simultaneously, one to supply heat to the classrooms, the other to collect heat. The logic determines the cooler tank and circulates that tank to the collector, and the warmer tank to heated space when it is occupied. The

TABLE 5
PARTIAL LISTING OF DIFFERENTIAL THERMOSTAT MANUFACTURERS

<u>Manufacturer</u>	<u>Turn-On $\Delta T(^{\circ}F)$</u>	<u>Turn-Off $\Delta T(^{\circ}F)$</u>	<u>Sensor Type</u>
Rho Sigma, Inc. 15150 Raymer Street Van Nuys, California 91405	20 $\pm$ 3	3 $\pm$ 1	Thermistor
Hawthorne Industries, Inc. Solar Energy Division 1501 South Dixie West Palm Beach, Florida 33401	5 Also makes variable pump speed controls.	5	Thermistor
Robertshaw Controls Company 100 West Victoria Street Long Beach, California 90805	15 $\pm$ 5	5 $\pm$ 3	Thermistor
Penn Division Johnson Controls, Inc. 2221 Camden Court Oak Brook, Illinois 60521	20 $\pm$ 5	5 $\pm$ 3	Nickel, wire wound
Heliotrope General 3733 Kenora Drive Spring Valley, California 92077	3 9 15	1.5 3 5	Thermistor

system differentiates between "daytime" and "night" by means of a system clock, and energizes one of two thermostats in each heated classroom space, depending on the time of day, or whether the room is normally occupied or not. Thresholds for collector operation are set at a low of 100°F (38°C) and a high of 180°F (82°C). Stability criteria are established for changes of mode, such that for instance, a temporary loss of insolation will not shut the system down until the status quo has been interrogated for periods of five minutes each, variable up to an hour. The system maintains consistent tank water levels by always draining the collector or space heater water back into the tank from which it came. The collector is always drained when not in use, to prevent freezing or overheating in case of power failure. A special selector switch allows either tank to be "removed from the system" for repairs if necessary. Manual control is, of course, also provided, with a display board to indicate valve and pump status.

This specification was written entirely by ITC, issued as a complete item, and subcontracted to Roburn Corporation of Fairfax, Virginia. The conditions were specific in that the control system must interface with existing valves and pumps, which had been connected to the old relay-operated controls. For this reason, two complete logic matrices were generated in tabular form and these algorithms became part of the specification. The tabulations, and the schematic diagram were already constructed by ITC for the subcontractor to design. (refer to Appendix A.).

A less complicated control system specification for a solar heating system with a single tank has been generated at ITC, reference (2). As yet undated, it provides the same limitations and safeguards as its predecessor of reference (1). However, there are only eight valves in all (six automatically controlled) and if the tank is above ground, only one pump is required for circulation to the collector. As in reference (1), an auxiliary immersion pump is shown in the schematic, for underground installations, to provide positive suction head to the main pump.

E. Solar Heating and Cooling Control Systems

The last and most complicated of the control system categories is the large HVAC type in which the solar collector system is but one component among many. An example of this is the Page Jackson Elementary School in Charles Town, West Virginia. ITC is prime contractor for the design of the solar heating and cooling system of that school, now under construction.

Specifications for the controls of this solar heating and cooling system are contained in reference (3). They are reproduced in part here as Appendix B which is in itself an appendix to reference (3).

It appears that this type of specification is the rule, rather than the exception, in that a section is devoted to operational control requirements in text form, rather than as an algorithm or truth table as was done in the Fauquier High School control system specification.

One advantage of this method is that the entire section is the responsibility of the controls contractor, who must then construct his own truth tables and logic. He is then responsible for the entire job.

Although this specification is written for a pneumatic control system which is a commonplace method, it is adaptable to electronic controls as well.

Care was taken to use only standard, readily available equipment in the design of the Page Jackson Elementary School's control system. The system will provide:

1. Hot water to the storage tanks automatically, whenever the collectors are warmer than the storage;
2. Heating or cooling to the central air handling unit as called for;
3. Control for the built-in energy conservation methods;
4. Variable ventilation according to building occupancy; and
5. Automatic or manual seasonal selection.

The control system will also have a functional display panel to demonstrate system operation for the benefit of both the student body and visitors. This demonstration panel will be centrally located in the resources area of the building.

Included in the large system category are the other three schools funded by the National Science Foundation in 1973-74 and presently operated under ERDA sponsorship. Physical descriptions of the four school solar heating systems are given in reference (4), but only the briefest of descriptions of the control systems are given therein. However, this reference is useful for a general comparison.

REFERENCES

1. ITC Specification SCS-1001-010575 Solar Control System, revised 23 July 1975. (See Appendix A)
2. ITC Specification SCS-1002-() Solar Control System for Single Tank, See Appendix B.
3. ITC Final Report to Jefferson County School Board under ERDA Contract E(40-1)-4926, September 1975. Appendix G-1, Temperature and Ventilation Controls.
4. Proceedings on the Workshop on Solar Heating and Cooling of Buildings, sponsored by NSF-RANN, June 1974, prepared by Dept. of Mechanical Engineering, School of Engineering and Applied Science, University of Virginia and ASHRAE under NSF Grant GI 43965.

IV. SELECTIVE SURFACES FOR SOLAR HEATING AND COOLING

A. General

A selective surface for solar collection is defined herein as a surface with a solar absorptance, α , greater than 0.8 and with an emittance for thermal radiation, ϵ , of less than 0.5. This definition is used rather than one specifying a minimum value of the ratio α/ϵ alone. This is because the solar absorptance and thermal emittance enter separately into the equations describing the overall heat balance in a solar collector.

Flat plate collectors utilizing selective surfaces were first demonstrated in 1955. Since that time a limited amount of research has been performed on selective surfaces. A large fraction of that work was for power generation in space using solar driven heat engines.

Recent research on selective surfaces has been concentrated in two main areas:

1. Flat-plate collector surfaces for solar heating and cooling of buildings.
2. Collector surfaces for solar power generation.

For the purpose of the present report only selective surfaces suitable for use on flat-plate collectors have been considered. A limit of \$2.00/ft² of collector area has been selected as the maximum allowable cost for a selective surface to be used in solar heating and cooling (SHAC) applications. Vacuum deposited coatings have, therefore, not been included because of their high application costs.

B. Commercial Availability of Selective Surfaces

The number of selective surfaces which are available commercially is quite limited at the present time. These surfaces are listed in Table 6. The

TABLE 6
COMMERCIALY AVAILABLE SELECTIVE COATINGS

Electroplated Coatings

<u>Coating</u>	<u>Substrate</u>	<u>α</u>	<u>ϵ</u>	<u>Cost/ft²</u>	<u>Comments</u>
Black Nickel	Ni-plated steel	0.9 to	~0.1	?	Stability is a question.
	Ni-plated Al	0.94			Used on galvanized steel by Sunsource (Miromit) for flat plate collectors.
Black Chrome	Ni-plated steel	0.9 to	0.1 to	\$1.87 to	Considered to have good stability, durability.
	Ni-plated Al	0.95	0.15	\$1.29	Production facilities limited.

Chemical or Conversion Coatings

CuO	Copper	~0.9	~0.1	\$0.25	Long term stability of ϵ is a question. Coating used by Sunworks, Beasle, Soltex for flat plate collectors.
"Alcoa"	Al	0.92	0.3	\$0.50	Information on stability of α, ϵ is needed.
Chromate	Galvanized steel Zn plated Al	~0.9	~0.1	\$0.70(?)	Still under development.
Black iron oxide	Steel	0.85	0.1	\$0.50	Ebonol S process (chemical dip to form Fe ₃ O ₄ on surface)

grouping of the coatings is by the method of application. It is interesting to note that all of the selective surfaces listed were developed as decorative or protective black finishes for metals. The selective nature of these surfaces and their application to solar energy has not generally been recognized in the metal finishing industry. However, a number of the leading companies are at work on black conversion coatings for aluminum which may have selective properties.

A major problem with the electroplated surfaces is the lack of adequate facilities for mass production of large solar panels. For example, very few electroplaters have the large plating tanks and power supplies to apply black chrome to a 4' x 8' panel. Also, it will be necessary to spot-check the absorptance and emittance as part of a quality control procedure. Visual estimates of the "blackness" of a panel are to be avoided, as approximately half of the solar energy is at wavelengths above the visible region.

There will be no production problem in the case of conversion coatings. This is due to the fact that capital requirements are much lower and the productivity is much higher because of batch processing. In fact, a breakthrough in the development of a good selective conversion coating for aluminum could make obsolete a black chrome production line.

A detailed discussion of the various kinds of selective surfaces will be found at the end of this report.

A number of solar collector systems which use selectively coated absorber plates have recently been installed. These installations, along with the type of selective surface used, are listed in Table 7.

C. Economic Analysis

A computer simulation of the performance of a solar heating system was performed for a number of values of collector plate absorptivity and

TABLE 7
SOLAR COLLECTOR INSTALLATIONS USING SELECTIVE COATINGS

<u>Installation</u>	<u>Collector Area (ft²)</u>	<u>Selective Surface - Substrate</u>
MSFC Solar Test House Huntsville, Alabama	1,300	Black nickel on nickel plated aluminum
North View Junior High School Minneapolis, Minnesota	5,000	Black nickel on nickel plated steel
Honeywell Mobile Solar Lab.	570	Black nickel on nickel plated steel
Barber House Guilford, Connecticut	400	Copper oxide on copper
Pinchot Guest House Guilford, Connecticut	420	Copper oxide on copper
Connecticut Shoreline House Westbrook, Connecticut	450	Copper oxide on copper
Towns Elementary School Atlanta, Georgia	10,000	Alcoa coating on aluminum
General Electric Space Center King of Prussia, Pennsylvania	5,000	Alcoa coating on aluminum
Fauquier High School Warrenton, Virginia	2,400	Alcoa coating on aluminum

emissivity. With the assumption that the rate of return on investment is equal to the annual increase in fuel cost, the optimum collector area was determined for each set of absorptance and emittance values with an assumed cost for back-up fuel of \$13.18/10⁶ Btu's (4.5¢/kWh). The collector was taken to be of double-glazed construction with a glazing transmittance of 0.89 (single sheet).

The optimum collector area for all cases was found to be near a load factor (Heating requirement/collector area) of 142,000 Btu/ft². For this load factor the percentage of the total heating requirement met by the solar system fell between 55 percent and 75 percent for the cases under study. In Figure 22 the loci of points of equal economic worth are plotted. The zero of worth is taken to be a flat black paint with $\alpha=\epsilon=0.95$. For purposes of normalization the worth has been computed based on a back-up fuel cost of \$10/10⁶ Btu's.

For example, a coating with $\alpha = 0.95$ and $\epsilon = 0.25$ is found to lie near the \$2/ft² line. Thus, a coating with these parameters is cost-effective relative to black paint if it can be applied at a cost of less than \$2/ft² above the cost of applying the black paint. If fuel were valued at \$15/10⁶ Btu's one would simply multiply the cost figures by 1.5. It can also be seen that if the emittance of the above coating were reduced to 0.1 and the absorptance fell to 0.91, the worth of the coating would be unchanged.

The figure is also useful in that it shows that small changes in solar absorptance are much more significant economically than small changes in the emittance. Thus, the requirements for accuracy in measuring solar absorptance are quite high. These results, of course, are for double-glazed collectors. For single-glazed collectors the numerical value of the emittance becomes more important.

- Conditions - Winter Heating (5000 DD/Yr.)
- Double Glazed Collector
 - 20 Yr. Amortization
 - Rate of Return = Fuel Inflation Rate
 - Fuel Cost $\$10/10^6$ BTU's. (3.4¢/kWh)

FIGURE 22: WORTH, W, OF A SELECTIVE SURFACE RELATIVE TO BLACK PAINT, $\$/\text{FT}^2$

D. Electrodeposited Coatings

1. Black Nickel

Black nickel selective coatings were first developed by Tabor in 1955. At the present time, the only commercial collectors using black nickel are the Miromit Collectors imported from Israel by Sunsource. The black nickel is applied to a zinc (galvanized steel) substrate and is reported to have $\alpha = 0.9$ or greater and $\epsilon = 0.1 - 0.15$.

In the United States, black nickel coatings have been studied mainly by the NASA Marshall Space Flight Center and by Honeywell, Inc. (1). Values of α as high as 0.94 have been reported with values of ϵ less than 0.1. Substrates used have been nickel-plated steel or aluminum. The main drawback to this coating is its susceptibility to degradation by extremes of temperature and humidity. It is possible that the moisture resistance of black nickel can be improved by overcoating with a thin lacquer or varnish. If the protective lacquer can be applied very thinly (less than 2-3 microns), the low emittance of the surface will not be adversely affected. No current estimates of the cost of a black nickel coating have been found in the literature.

2. Black Chrome

Black chrome has been used for a number of years as a decorative-protective black coating. Consequently, a considerable amount of data has accumulated which shows that black chrome has excellent resistance to the effects of outdoor exposure. However, it has only been within the past year that the solar selective properties of black chrome have been discovered. Thus, the stability of the coating as regards its absorptance and emittance is not known, but is expected to be quite good based on the above mentioned outdoor exposure experience.

Most of the development work on black chrome has been performed at the NASA Lewis Research Center (2, 3). Small panels (4" x 6") of Ni-plated steel have been coated with black chrome to give absorptances as high as 0.96 with emittances around 0.1.

The high values of solar absorptance are obtained with the plating smut undisturbed. Plating smut is a fine, soot-like surface deposit which is non-adherent.

The main drawback to the black chrome process is the large power requirement for electrodeposition (180A/ft² at 24 volts). A 24 ft² panel would require a 100 kW power supply. The requirements of large power supplies and large tanks limit the location of potential vendors to areas where there is a high concentration of heavy industry. Thus, shipping charges may add significantly to the cost of the coating. A possible solution would be to coat a thin metal foil with black chrome and then adhesively bond it to the collector plate. As the adhesive layer would be very thin, the thermal performance of the collector would not be altered. In addition, the selectively coated foil could be produced at a much lower cost due to automation of the electrodeposition process.

At present, there have been no large (20 ft² or over) collector panels plated with black chrome, so it is not known if the good results obtained with the small test panels will be forthcoming under full-scale production. Also, the mass-produced cost of the coating is not accurately known. Estimates range from \$1.29/ft² to \$1.87/ft². Even at the higher figure a coating with $\alpha = 0.96$ and $\epsilon = 0.1$ will be cost effective over 20 years as shown in Figure 22.

E. Chemical or Conversion Coatings

1. Copper Oxide on Copper

Cupric Oxide (CuO) on a copper substrate is one of the oldest known selective surfaces dating back to 1955. It has been used quite extensively in Australia on collectors for water heating. In the United States, Sunworks, Inc. of Guilford, Connecticut markets a single-glazed collector with a CuO surface applied by Enthone, Inc. Corning Glass has also used a CuO selective surface on its prototype collectors. Enthone is the manufacturer of Ebonol C which was the chemical originally used to produce selective surfaces on copper. Now,

Enthone has a proprietary process said to be better than the Ebonol C process. Enthone guarantees solar absorptances greater than 0.90 and emittances less than 0.12. Coating costs are estimated to be \$0.25/ft² for large quantities (4).

Some thermal performance data is available on CuO (on copper) coated collectors. These data were taken at NASA-Lewis using a solar simulator (5). The CuO coated collectors were from Sunworks, Inc. and from Beasley (Australia). Based on their thermal loss coefficients, the collectors did not perform significantly better than black painted collectors. The following argument will show how this can happen.

Since the CuO coating is quite thin, it is very easy for oxygen to diffuse through it and oxidize the copper underneath. This reaction is speeded up to a considerable degree at the temperatures obtained under no-flow or stagnation conditions. Thus, one may expect that the emittance of the surface will approach that of a thick oxide layer ($\epsilon = 0.8$) and that the performance of the collector will be only marginally different from one painted flat black.

Whether the above scenario actually happens remains to be seen. The question of whether copper is suitable as a substrate for a selective coating because of oxidation and moisture induced corrosion problems should be the subject of further research.

2. Black Oxides on Aluminum

The above mentioned problems of deterioration of the substrate are overcome if the black coating is deposited on aluminum. There are three ways of doing this.

One way is to spray the aluminum plate with copper nitrate; the plate is then heated to decomposition temperatures to produce black cupric oxide. This was the process developed by Hottel and Ungar at MIT (6). Their best results were an absorptance of 0.93 with an emittance of 0.12. The optimum coating weight was found to be 0.3 mg/cm² which implies a coating thickness of less than one micron.

A second approach was developed by Daniels and collaborators (7). They plated a thin copper or cobalt film on the substrate and then completely oxidized the coating by heating. It would also be possible to produce the black oxide by chemical means. The advantage of this approach over the one developed by Hottel is that the coating thickness is dependent on the amount of copper or cobalt plated and this can be accurately controlled.

A third method would be to stain the aluminum black in a chemical bath. H. Tabor has developed and patented a process using a heated acidic solution of potassium permanganate and copper nitrate (8). It is quite possible that the Alcoa process is based on a similar formulation. Reynolds Metals is also developing selective coatings for aluminum (9). There is little known about the stability of these coatings.

3. Black Coatings on Zinc

There are a number of black coatings that can be applied to zinc or to galvanized steel. McDonald (10) has recently reported on a chromate conversion coating which has $\alpha = 0.9$, $\epsilon = 0.07$. The stability of this coating is not known. Costs are estimated to be \$0.70/ft².

Enthone, Inc. manufactures two chemicals which can blacken zinc, Ebonol Z and Ebonol Z-80. It is not known whether coatings produced by these chemicals have been tested for their solar selective properties.

4. Black Iron Oxide on Steel

Ebonol S produces a selective black oxide coating on steel, but the solar absorptance is low (0.85). The emittance is approximately 0.1.

5. Selective Paints

At present, no selective paints are commercially available. The problem in developing a selective paint is in finding a binder which has high temperature stability and also very low emittance. The silicone resins have excellent high temperature properties but have poor emittance properties except in very thin coatings.

Honeywell, Inc. is conducting a systematic search of suitable binders and pigments (11). Some of the black oxide pigments such as Cu-CrO_x show promise of providing a selective paint with $\alpha = 0.9$, $\epsilon = 0.3$. A better approach would be to make the substrate for the paint partially selective by a simple chemical dip (to obtain $\alpha = 0.8$, $\epsilon = 0.1$). Then a thinner layer of paint can be applied resulting in a higher absorptance with the same or lower emittance. No additional steps are required since the metal base has to be cleaned prior to painting in any event.

F. Near Term Research and Development (Selective Surfaces Used in SHAC)

There are three main areas in the development of selective surfaces that show a need for further research and development. These areas are:

1. Reliability
2. Cost
3. Performance.

The order of appearance reflects the priorities that should be assigned these areas.

A number of research topics in these three areas have been identified and are discussed below. It is, of course, recognized that in some cases the reliability, cost, and performance of a product are interrelated. In the case of selective surfaces, however, not enough data have been accumulated to make any general statements as to what the correlations are.

1. Reliability

The absorbing surface of a collector must be able to maintain its properties over the lifetime of the system. During this time the coating will be subjected to:

1. Thermal cycles and shocks.
2. Large absorbed quantities of solar radiation (including uv).
3. High temperatures when the collector is "dry" on a sunny day.
4. High humidities if moisture is trapped within the collector.

These factors may act singly or in concert to produce changes in:

1. Solar absorptivity,
2. Thermal emissivity,
3. Adhesion to substrate.

Any one of these changes will result in partial failure of the collector.

Standard tests should be devised to properly assess the lifetimes of coatings. These tests are not only necessary for selective surfaces but also for black paints and other non-selective coatings.

In adopting a set of standardized tests for coatings, consideration should be given to the unique conditions to which they are subject. For instance, coatings used on glazed collectors will not be subject to mechanical abrasion. Thus, a loose or powdery surface (such as the smut produced in electro-deposition) need not be ruled out if it can be shown that adhesion to the substrate can be maintained with respect to temperature cycles, etc.

Changes in the absorptance or emittance of a coating will lead to change in system performance that can be easily misinterpreted. For the National Demonstration Plan it is recommended that all surfaces used be monitored

for their absorptance and emittance at the site. This can be accomplished by the use of portable instruments. The collectors tested would have to have removable glazing. Thus, the evacuated collectors could not be checked.

2. Cost

At present the cost situation for selective surfaces is quite tentative. This is due to the lack of facilities for mass production and EPA and OSHA requirements which have greatly increased production costs. Electroplated coatings such as black chrome have been estimated to have an application cost as low as \$0.50-\$0.60 per square foot. Current quotations, however, run from \$1.29 to \$1.87 per square foot.

For conversion type coatings such as CuO on copper the application costs can be quite low. This is because the processing steps are simple, many panels may be coated simultaneously, and the chemical costs are low (approximately \$0.05/ft²). Overall costs for conversion coatings should be as low or lower than the cost of applying a good quality flat black paint. This is because the base metal must be prepared or primed for painting, and the final coat must be baked in order to prevent volatiles from outgassing and condensing on the collector glazing.

Research and development work should be concentrated on:

1. Increasing productivity of the electroplating process by eliminating steps and/or reducing processing times.
2. Identifying new conversion coatings (especially for aluminum) which show solar selective properties.
3. Basic research on the optical properties of materials and surfaces.

3. Performance

The standard of performance of selective surfaces is now quite high. To date the best selective surfaces have solar absorptances between 0.90 and 0.94 with

thermal emittances from 0.07 to 0.15. Since the emittances of clean metal surfaces are in the range of 0.04 to 0.10 it can be seen that further reductions in the emittance of selective surfaces will be difficult to attain. From Figure 1 it can be seen that these improvements are of marginal cost-effectiveness.

Improvements in the solar absorptance on the other hand can be very cost-effective even if the emittance is increased. For instance a collector surface with $\alpha = 0.96$, $\epsilon = 0.15$, is worth \$1/ft² more than one with $\alpha = 0.90$, $\epsilon = 0.10$. A surface with $\alpha = 0.95$, $\epsilon = 0.30$ is just as cost effective as one with $\alpha = 0.90$, $\epsilon = 0.10$. The worth is dependent on the cost of the backup fuel.

In order to improve the absorptance it will be necessary to produce changes in the surface index of refraction. One method would be to overcoat the surface with a clear, thin, stable material of low index of refraction. This is routinely done in the metal finishing industry to deepen the black colors of conversion coatings or black oxide coatings. The materials used however (oils, waxes) are generally not stable at the temperatures that collectors are likely to reach under no flow (dry) conditions.

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V. THERMAL ENERGY STORAGE

A. Introduction

Solar heating systems require some heat storage to be able to supply heat during part or all of the periods without sun. A minimum heat storage capacity is probably equivalent to a night's supply of heat while, theoretically, there is no maximum to the storage capacity. The optimum storage capacity must be determined for each individual case on the basis of the heat load, climatic conditions, cost of heat supplied by fuels and other relevant factors so as to minimize the overall cost of heating.

Water and rocks have been extensively used as heat storage media (1) (2). The possibility of using materials such as $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ (Glauber salt) relying on their heat of fusion for storage is actively pursued. Systems based on these materials are less advanced because of various technical problems such as supercooling and loss of heat storage capacity upon multiple cycling (3).

Space cooling of buildings generally requires less storage, as the cooling needs are approximately matched in time with periods of high insolation. Some storage is however necessary in some cases for hot and humid periods without sufficient insolation to drive the cooling system. Cooling systems driven by solar heat generally require higher storage temperature than space heating systems (minimum temperatures: about 170°F versus about 100°F for heating).

The design and performance of thermal storage are closely related to the collector and heating and/or cooling loads characteristics. It is thus difficult to compare the relative merits and properties of thermal storage systems per se. The following notes attempt to establish some general criteria of comparison between the three classes of storage systems considered at present, i.e., water, rocks and phase-change systems. The comparison will be restricted to the storage system itself, thus excluding auxiliaries such as blowers, pumps, exchangers, etc. These can only be taken into account within the context of complete systems.

B. General Features of Presently Used Heat Storage Media and Systems

1. Systems Considered

Two classes of heat storage systems are considered: those storing sensible heat as exemplified by water and rock heat storage and those relying on phase changes as their main mechanism for storage, as represented by melting salts systems. The various media used for heat storage are discussed individually below.

2. Water or Liquid Heat Storage

Water is an inexpensive, readily available and practical material for storing sensible heat. The storing medium itself may be used to collect and distribute heat, thus limiting heat transfer problems at the interfaces between the collectors and the load and the storage systems. Water may also, and has been, used in conjunction with forced air heating systems; this combination requires a water-air exchanger which may reduce the efficiency of the system.

Table 8 shows some typical values of parameters of importance in the design of water storage systems. The seventh column " ΔT for heating" gives the range of temperatures between which water can be used practically for space heating purposes: water stored at temperatures below about 100°F would have little value for heating purposes because of poor heat transfer between the source of heat and the ambient. The upper limit is just below the boiling point. The eighth column gives the corresponding values for water stored to drive a cooling system. The lower useful temperature of storage is about 170°F. With heat inputs at a temperature below this value, most present day cooling systems -- absorption, dehumidification, organic Rankine cycle engine -- become very inefficient. Higher upper temperatures can be used with pressurized systems: the upper temperature which can be reached will probably be determined by the characteristics of the collectors used. The storage of 10^6 Btu in a water system between 100 and 200°F for heating purposes would require 10^4 pounds of water or about 160 cubic feet or about 1,200 gallons. For cooling purposes, the storage of 10^6 Btu between 170 and 200°F, would require about 33,000 pounds or about 530 cubic feet or about 4,000 gallons. These values

TABLE 8
GENERAL CHARACTERISTICS OF SOME THERMAL STORAGE MEDIA

Storage media	Specific heat Btu/lb °F	Heat of fusion Btu/lb	Melting point °F	Density lb/ft ³	Cost \$/lb	ΔT for heating °F	ΔT for cooling °F
Water	1	--	32	62.4 (1)	0.0012	100-210	170-210
Rock	0.2	--	--	140	0.0025	100-...	170-...
Na ₂ SO ₄ · 10H ₂ O	~0.4	104	90	91.4	0.02	90-180	--
Na ₂ HPO ₄ · 12H ₂ O	--	120	104	94.9	0.09	104-180	--
AlK(SO ₄) ₂ · 12H ₂ O	--	76	176	109.7	0.10	176-220	176-220
Na ₂ S ₂ O ₃ · 5 H ₂ O	0.35(s) 0.60(l)	90	120	108	0.14	100-180	--
Na ₂ SO ₄ · 10 H ₂ O/NaCl/ NH ₄ Cl (eutectic)	0.35(s) 0.65(l)	78	55	--	--	--	50- 70
CaCl ₂ -Mg Cl ₂ -H ₂ O (eutectic)		75	77	--	--	77-150	--
Mg(NO ₃) ₂ · 6H ₂ O-Al(NO ₃) ₃ · 9H ₂ O (eutectic)		64	142	--	--	142-180	--
Acetamide-stearic acid (eutectic)		94	149	--	--	149-180	--
Urea - NH ₄ NO ₃ (eutectic)		74	115	--	--	115-180	--

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would be reduced to about 14,000 pounds or about 1,700 gallons if a pressurized water system working between 170 and 240°F was used. In its simplest form, a water storage system comprises an insulated tank filled with water. The auxiliaries comprise control systems and means for circulating the water to the collectors and/or the space to be heated. Both functions, space heating and heat collection can be performed simultaneously.

The use of water as a collection and storage medium creates two problems: corrosion and freezing during winter. Corrosion can be limited through the use of inhibitors in solution and standard corrosion protection procedures. Freeze protection can be achieved either by drainage of the freeze-prone sections of the system during nights or periods of poor insolation or by the use of antifreeze mixtures. Drainage of the system by gravity has been shown to be an effective method of freeze protection (4) and can probably be recommended whenever the layout of the piping permits easy, reliable and fast drainage. When complex piping loops creating a risk of trapping pockets of water are involved, antifreeze solutions should be used. Antifreeze solutions are expensive, however. Ethylene glycol costs about \$5/gallon and a 30% by volume mixture of ethylene glycol and water would cost about \$1500 per 1,000 gallons as compared to about \$1 per 1,000 gallon for water alone. Thus, when freeze protection must be provided by some antifreeze mixture, the antifreeze solution should be contained in a closed loop of limited volume used to collect heat which is then transferred to plain water for storage.

The temperature of the storage water varies with time as a result of changes in heat collection rates, in heat loads and eventually heat losses. As a result, the efficiency of the overall system does change: where storage temperature is low, heat transfer to the space to be heated is poor while heat collection from the solar collectors improves and vice versa. This situation may be improved by stratification in the storage unit, i.e. layers of water at different temperatures are formed so that heat is supplied from the "hot" side of the storage for space heating and water from the cool side of the tank is used in the collectors thus improving overall efficiency. To achieve effective and useful stratification, baffles should be used in the water storage tank to reduce turbulence due to pumping. Another approach is to use

several storage tanks with water at different temperatures and program the heat distribution and collection between these tanks so as to maximize overall efficiency (2,4,5).

Heat storage has also been performed in solar ponds. In a typical realization (1), a roof pond about 10 inches deep, accumulates heat during daytime. At night, the pond is covered to reduce losses and the stored heat used for space heating. In summer, the insulating properties of the roof pond reduce internal heating of the house. By letting the pond radiate its heat at night, and covering the pool during daytime, some degree of coolness can be accumulated. Such a system can only be efficient in certain climates and is thus of limited interest. Solar ponds in which a temperature inversion is achieved, i.e. high temperature water layer at the bottom and low temperature at the top, have also been considered for heat storage. The temperature inversion results from the existence of high concentration gradients of dissolved salts within the pond. Temperatures of the order of 170°F at the bottom and 85°F at the surface have been recorded (5).

Although auxiliary systems are not part of the present review, it should be noted that the energy required to activate a water storage system is generally negligible compared to the amounts of heat energy made available through storage.

3. Pebble Bed Storage

Pebble bed storage systems use the heat capacity of a bed of loosely packed pebbles through which a fluid, generally air, is circulated to store or remove heat from the bed. In principle any solid could be used, but rock, because among others of its low cost, is the most widely used material (1,2,5,6).

Values of parameters used in the design of rock heat storage units are shown in Table I. The specific heat of rock is about five times lower than that of water, while its density is a little over twice as high. Thus, storage of a given amount of heat between two specified temperatures will require a mass of rock about five times larger than that of water required to store the same amount of heat under the same conditions. For a void space of about 25%, the

volume occupied by the pebble bed will be between 2.5 and 3 times as large as the corresponding volume of water. Well designed rock beds are characterized by very good heat transfer between air and the solids making the bed. A pebble bed storage unit comprises a mass of pebbles enclosed in an insulated container. The pebble size should be relatively uniform, typically one to two inches in diameter, to obtain homogeneous void distribution and minimize pressure drop. The auxiliaries comprise control systems and means for circulating the heat carrying fluid -- air generally -- to the collectors or the space to be heated.

During heat collection, flow is maintained through the bed in one direction. Flow is reversed during heat removal. Stratification will thus normally be present in the bed thus improving heat transfer. On the other hand, heat collection and space heating cannot be performed at the same time as is the case for water storage systems.

When local conditions are adequate, pebble beds can be used during night hours to store "coolness" to be used during the hot hours of the day. Pebble beds can store heat at temperatures higher than unpressurized water systems: they could thus in principle be used to drive cooling machinery. However, efficient heat transfer from the bed to absorption or other cooling systems appears difficult. Mixed water-rock storage systems have been used, (7): heat collection is performed by water which is stored. The water tank is surrounded by a bed of rocks acting as storage medium and heat exchanger for air space heating.

Although not strictly a pebble bed storage unit, experiments have been performed with building structures acting as collection and storage units: in a typical design, a concrete or brick wall facing south and insulated from the outside by glazing collects heat during day time and radiate the heat towards the living space at night. Such systems are for winter operation only and often referred to as the "French System"; (1,5).

4. Phase-change heat storage

Materials undergoing a change of phase in a convenient temperature range may be used for heat storage if the phase change is accompanied by high latent heat effects and if it is reversible over a large number of cycles without loss of performance. The phase change must occur with little or no supercooling. The material is generally placed in containers through which heat is transferred to and from the storage material (3,5,8 to 11). A typical phase change storage unit thus comprises an insulated structure in which containers filled with the phase-change material are distributed with adequate void space so as to insure good heat transfer between the containers and the heat carrying fluid. The latter is generally air and, thus, the design problems for phase change systems are similar to those for pebble bed systems. Another approach consists in having a bath of phase change material through which tubes or exchangers filled with heat-carrying fluid circulate. This solution however appears to create serious heat transfer problems.

The physico-chemical properties of some candidate materials -- salts and salt mixtures -- for phase change systems are shown in Table 8.

These materials are generally characterized by high heat of fusion and significant specific heat in the liquid and solid states. Heat storage will thus involve phase change as well as specific heat effects. The densities are usually comprised between those of water and of rocks. The most realistic cost is that for $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ (Glauber salt) which can be obtained in bulk quantities in industrial grade. The cost of the other salts refers to packaged material and could probably be lowered to the level of Glauber salt for bulk delivery. Few if any of the salts can be used both for heating and cooling. Salts with melting points in the range of 90 to 140°F about are probably best suited for winter heating but their temperature of storage is too low to drive cooling units efficiently. Higher temperature systems, 170°F and over, would be well suited to drive cooling units in summer when such storage temperatures can easily be obtained from solar collectors. Lower temperature systems could be used to store coolness. The storage of 10^6 Btu in a $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ system between 90 and 180°F would require about 7,100 pounds of material if advantage is taken of the specific heat of the material

in the temperature range of use above the melting point. Assuming a void fraction of 33% for good air flow and heat transfer, such a system would require a volume of about 105 cubic feet or about 780 gallons. Comparison of these values with the corresponding numbers for water and rock storage indicate that under comparable working conditions, phase change systems are the most compact of the three types of storage units considered.

Although the bulk of the heat is theoretically recovered at fixed temperature, i.e. the melting point, because of, among others, the contribution of the specific heat to heat storage, heat will be stored and recovered at variable temperatures. As was the case for pebble bed systems, and because of the similitude of phase change systems with pebble beds, a certain degree of temperature stratification may be expected within the storage bed. As discussed earlier for the pebble bed systems this is a favorable circumstance as far as efficiency of heat transfer for collection and recovery is concerned. Heat transfer to and from the phase-change material is a complex problem. Experimentally, the material is placed in containers -- tubes, plastic bags, trays, metal boxes,... -- which are distributed in the storage space. The heat transfer problem between these containers and the heat carrying fluid--air in most cases--is similar to that encountered in pebble beds. Within the containers, however, partial melting or recrystallization starting at the walls modify the thermal resistance to heat transfer. Also, many of the phase-change materials considered display supercooling effects: various techniques have been considered to remedy this situation.

The encapsulation of phase change materials also raise associated problems such as corrosion, breakage, side reactions, toxicity of products, etc. Tubes, pans and plastic bags are used or tried as containers (10) (11).

C. Present-Day Realizations. State of the Art.

1. Introduction

The most complete listing of solar heating and cooling systems found so far is the survey established by W. A. Shurcliff, (12). It lists, by state, systems presently or previously used as well as some systems in the design and/or construction stages. In the following sections, an attempt is made at summarizing this data. The three types of storage systems will be considered separately, and in each case the devices in operation will be analyzed separately from the devices in planning, design or construction stages. Some of the units described in reference 12 were ignored in the analysis for lack of sufficient information.

2. Water Systems

Table 9 summarizes the data published concerning the water storage systems used or planned.

Examination of the data suggests some general trends:

a. Units In Service

- Metal tanks are used most often, a few units use concrete containers. Plastic bags or bottles have been used in special cases.
- Rock storage is sometimes combined with water storage (Thomason system for instance).
- About equal numbers of the units reported use water-fan-coil and water-air exchangers to carry heat to building.
- Freeze protection is more often achieved by drainage in the more recent units. In other cases, closed water-antifreeze loops, with an exchanger in the storage unit, are often used.

TABLE 9 WATER STORAGE SYSTEMS

A. SYSTEMS PRESENTLY OR PREVIOUSLY IN OPERATION

Location (Sponsor)	Type (1)	Date (2)	Volume Gal.(3)	Temp. °F(4)	Gallons Ft ² (5)	Storage Unit(6)	Treatment (7)	Carry Over Days(8)	Use (9)	Heat. %(10)	Ref. (11)	
Huntsville, Ala. (Nasa)	H	1974	4,700	Up to 260	3.6	T	Deionized	-	H-C	-	p.2	
Tucson, Ariz. (Meinel)	H	1973	1,000	-	3.3	P (1 gal. each)	-	-	H-(C) (air)	20	p.6	
26 Tucson, Ariz. (U. of Ariz.)	H	1959	4,500	-	2.8	T	-	-	H-(C) (water)	85	p.6	ended 1969
Atascadero, Calif. (Hay)	H	1973	6,300	-	5	P	-	4	H-(C)	100	p.8	
Davis, Calif. (Springer)	H	1974	210	-	1	T	-	-	H (water)	65	p.10	
Pueblo, Colo. (Sunray I)	H	1974	2,000	-	2	T	-	-	H (air)	75	p.24	
Snowmass, Colo. (Shore)	H	1974	5,300	138	9.4	C	0	7	H (water)	100	p.25	
Westbrook, Conn. (Barber)	H	1974	2,000	-	4.4	T	-	2	H (water)	45	p.32	
Gainsville, Fla. (U. of Florida)	H	1963	3,000	-	10	T	-	-	H (water)	100	p.36	
		1973	3,000	-	6	T	-	-	H-C (water)	100	p.36	

TABLE 9 WATER STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Gal.(3)	Temp. °F(4)	Gallons Ft ² (5)	Storage Unit(6)	Treatment (7)	Carry Over Days(8)	Use (9)	Heat. %(10)	Ref. (11)	
District Heights, Md. (Thomason)	H	1959	1,600	-	2	T	-	-	H-(C) (air)	85	p.41	Combined with rock storage.
	H	1963	1,600	-	1.8	C	0	-	H (air)	75	p.43	Electric AC
Timonium, Md. (AAI & NSF)	S	1974	15,000	150 (April)	3	T	-	6	H (water)	(91) (Spring)	p.47	Unit & rocks cooling in- stalled
Boston, Mass. (GE & NSF)	S	1974	2,000	100 to ~ 200 (April)	0.45	T	Ethylene glycol	~0	H (air)	(68) projected	p.49 (14,15)	245° max T. recorded
Cambridge, Mass. (Hottel, MIT-I)	H	1939	17,400	195 (max)	44	T	-	winter	H (air)	-	p.50	ended 1941
Cambridge, Mass. (Hottel, MIT-III)	H	1949	1,500	-	3.75	T	-	2	H	75 to 85	p.51	ended 1955
Lexington, Mass. (Hottel, MIT-IV)	H	1959	1,500	-	2.3	T	-	-	H	57	p.52	ended 1962
Waltham, Mass. (Solar Heat Corp.)	H	1974	4,000	-	26.7	C	-	-	H	100	p.55	
Brooklyn Park, Minn. (Honeywell, NSF)	S	1974	3,000	130-150	0.6	T	Ethylene glycol	~1	H (air)	(6)	p.57 (16)	

TABLE 9 WATER STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Gal.(3)	Temp. °F(4)	Gallons Ft ² (5)	Storage Unit(6)	Treatment (7)	Carry Over Days(8)	Use (9)	Heat. %(10)	Ref. (11)	
Albuquerque, N.M. (Bridgers & Paxton)	B	1956- 1962	6,000	-	8	T	0	3	H-C (water)	-	p.65	Water drained.
		1974	6,000	-	8	T	-	-	H-C (air)	65	p.65	
Coos Bay, Oregon (Mathew)	H	1967	8,000	-	11	T	0	-	H (air)	80	p.76	
Mount Holly, Penn. (Slyder)	H	1974	4,000	-	3.3	T	0	6	H	(75)	p.78	Water drained.
Schnecksville, Penn. (PP & L)	H	1974	1,000	-	4	T	-	(3)	H (water)	(30)	p.79	
Warrenton, Va. (ITC - NSF)	S	1974	11,000	-	4.6	C	anti corrosion	5	H (water)	100	p.87	Water drained.
Surrey, D. B., Canada (Hoffman)	H	1971	800	170 max summer	1.7	T	-	1	H (air)	40	p.91	Water drained.

B. SYSTEMS DESIGNED AND/OR UNDER CONSTRUCTION

Tucson, Ariz. (Copper Dev.Ass.)	H	1975	2,500	210-220 (summer)	1.1	T	Ethylene glycol	1-1.5	H-C (water)	100	p.5	Absorption cooling.
Yuma, Ariz. (Hansberger)	B	1975	1,800	180-200 (summer)	3.6	T	0	3	H-C (water)	95	p.7	Water drained.

TABLE 9 WATER STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Gal.(3)	Temp. °F(4)	Gallons Ft ² (5)	Storage Unit(6)	Treatment (7)	Carry Over Days(8)	Use (9)	Heat. %(10)	Ref. (11)	
Del Mar, Calif. (Beckstrnad)	H	1975	2,000	-	4	T	Inhibitor	2	H-(C) (air)	80-90	p.11	
Los Altos, Calif. (Brench)	H	1975	500	-	2.5	T	-	1	H (air)	70-80	p.13	Air collector.
Santa Clara, Calif. (NSF-ERDA)	B	1975	10,000	180-230 (summer)	1.5	T	-	-	H-C (water)	75	p.14	
Valley Center, Calif. (Mills)	H	1975	2,400		4.8	C	0	1.5	H-(C) (air)	70-80	p.15	Heat pump to cool water.
Visalia, Calif. (Cottrell)	H	1974	8,000	-	20	C	0	21	H (air)	80	p.16	Water drained.
Colorado Springs, Colo. (Phoenix)	H	1974	8,000	Up to 200	5	T	-	5	H (water)	80	p.18	
Colorady Springs, Colo. (Wood)	H	1974	1,800	-	2.4	T	0	3	H-(C) (air)	80	p.19	Water drained rocks around tank.
Denver, Colo. (Crowther)	H	1974	1,000	-	1.8	T	0	2	H-(C) (water)	90	p.20	Water drained.

TABLE 9 WATER STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Gal.(3)	Temp. °F(4)	Gallons Ft ² (5)	Storage Unit(6)	Treatment (7)	Carry Over Days(8)	Use (9)	Heat. %(10)	Ref. (11)	
Ft. Collins, Colo. (Col. State Univ. I)	H	1974	1,100	175-200 (summer)	1.5	T	Inhibitor	2	H-C (water)	75	p.22 (20)	
Quilford, Conn. (Barber)	H	1975	2,500	-	6	T	-	-	H (water)	60-70	p.29	
Quilford, Conn. (Pinchot)	H	1975	1,500	-	3.6	T	-	2	H (water)	70	p.30	
Atlanta, Ga. (NSF)	S	1975	45,000	140 200	4.5	T	-	4	H-C	60	p.37	Water drained.
Eureka, Ill. (Sun Systems)	H	1975	1,500	-	2	T	-	-	H	60	p.38	
Bar Harbor, Me. (Davis)	H	1974	1,700	-	3.2	T	-	-	H (air)	large	p.39	Rock added.
Gaithersburg, Md. (NSF)	H	1974	1,500	-	3	T	-	-	H-C (air)	75	p.40	
Acton, Mass. (Acorn Structures)	H	1975	2,000	-	6.7	C	-	1.5	H (water)	45	p.48	
Manchester, Mass. (Merchant)	H	1975	3,000	-	3.3	T	0	2	H (water)	50-75	p.54	
N. Chelmsford, Mass. (New England Tel.)	B	1974	500	200	1.2	T	-	1	H (water)	50	p.55	

TABLE 9 WATER STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Gal.(3)	Temp. °F(4)	Gallons Ft ² (5)	Storage Unit (6)	Treatment (7)	Carry Over Days(8)	Use (9)	Heat. %(10)	Ref. (11)	
Rosemount, Minn. (U. of Minn.)	H	1974	2,000	-	2.7	T	-	-	H (air)	-	p.59	Rocks added.
Etna, N. H. (Sol-R-Tech.)	H	1975	4,500	-	7.5	C	-	2	H (air)	100	p.60	Heat pump.
Las Cruces, N.M. (Dept.of Agriculture)	B	1975	30,000	-	4.3	T	-	-	H-C	100	p.67	
New Palz, N. Y. (Eccli)	H	1974	450	-	1.9	T	-	1.5	H (air)	70	p.72	Some rock storage.
Fairview, N.C. (Woodward)	H	1976	10,000	-	6.8	C	Bactericide	-	H (air)	large	p.73	Water drained.
Columbus, Ohio (Ohio State U.)	H	1974	4,000	-	5	T	-	-	H-C	50-70	p.74	Strati- fication.
Wagoner, Okla. (Engle)	H	1975	1,500	-	2	C	-	1.5	H (water)	75	p.75	
White River Junct., Vt. (Brown)	B	1974	5,000	-	5.5	C	-	-	H-(C) (air)	100	p.83	Heat-pump.
Fairfax, Va. (Miller & Smith)	B	1974	4,000	-	5.3	T	Inhibitor	-	H (water)	30-50	p.84	
Greenway, Va. (Madeira School)	S	1975	10,000	130 180 max	2.5	T	-	2-3	H (water)	-	p.85	

TABLE 9 WATER STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Gal.(3)	Temp. °F(4)	Gallons Ft ² (5)	Storage Unit(6)	Treatment (7)	Carry Over Days(8)	Use (9)	Heat. %(10)	Ref. (11)
Milton Keyes, Great Britain (Milton Keyes Corp.)	H	1975	1,100	-	2.9	T	-	-	H (water)	60	p.95
Baltimore, Md. (Poole & Kent Co.)	B	1974	1,100	-	1.5	T	-	-	H (air)	50	(17) 2 tanks: strati- fication.
Valley Forge, Pa. (G.E.)	B	1975	8,000	210 (max)	1.6	T	Inhibitor	-	H (air)	-	(18)
Washington, D.C. (G.E.-Skyline-NSF)	H	1975	400	190 + (summer)	0.9	T	-	-	H-C (air)	75 80 for cooling	(19) Arkla ab- sorption unit. Mobile home.

Footnotes to Table 9

(1) Type of building: H = house, S = school, B = commercial or office building.

(2) Date: year of start of operation for units used or previously used, projected year for start of operation for units in design or in construction.

(3) Volume: volume of water used for storage.

(4) Temperature: temperature of storage.

(5) Gallons/ft²: ratio of gallons of storage water to area of collector in ft².

(6) Storage unit: T = metal tank, C = concrete container, P = plastic container.

(7) Treatment: treatment or additives to storage water (not to collector water).

(8) Carry-over: number of days of carry-over by storage tank, generally quoted for January days.

(9) Use: H = heating only, H-C = heating and cooling, (C) = cooling by radiation to ambient without cooling equipment. Air between brackets refers to forced air heating system with a water to air heat exchanger while water generally refers to fan-coil heating units.

(10) Heating: percentage of heating load supplied by solar system.

(11) Reference: a page number refers to reference (12) while a standard reference number refers to the general list of references.

- Little or no air cooling except by nocturnal radiation in adequate climate is reported.
- Carry-through periods are generally of the order of 2 to 5 days. In this case the storage volume to solar collector area (gallons/ft²) is generally comprised between 1 and 4, with most units having a ratio of 2 to 3. Under these conditions, from 50 to about 85% of the heat load is supplied by the solar system. Systems having longer carry-through periods or supplying a large fraction of the load have higher storage volume to collector area ratios.
- Most systems are reported to work or have worked well. In some cases the period of use has extended over five years or more.

b. Units in planning, construction or early period of use.

The technical data for these units are either preliminary, i.e. do not include a full heating and cooling season, or predictions. With these limitations, the following comments can be made:

- A majority of the units planned for operation in the near future use metal storage tanks.
- A number of units include heating and cooling. Cooling is achieved in some cases with adsorption units with high (~200°F) temperature storage in pressurized or unpressurized systems. Heating and cooling is also performed by heat pump extracting or rejecting heat from or to the storage unit.
- Planned carry-through periods are generally of the order of 2 to 4 or 5 days. In these cases, the storage volume to collector area ratio (gallons/ft²) is generally comprised between 2 and 4, with most house units having a ratio of 2 to 3. Under these conditions, it is expected in most cases that 60 to about 80% of the heat load will be supplied by the solar system. The average

fraction of the heat load predicted to be supplied by the solar unit in the planned building is usually somewhat lower than the fraction reported for earlier buildings. This may be due to the fact that the design of new buildings does involve economic trade-offs to minimize heat costs, while the purpose of some at least of the earlier buildings was to demonstrate the feasibility of solar systems with relatively little emphasis on economic viability.

The new solar buildings are thus generally characterized by the use of more sophisticated peripheral equipment such as heat pumps and cooling units around the storage units.

3. Rock or Pebble Bed Systems

Table 10 summarizes the data published concerning the pebble bed units used or planned.

a. Units In Service

A limited number of units using pebble bed storage are or have been in use. They all have central air heating systems. A few systems provided some cooling by cooling of the bed of rock through night radiation: this is only applicable in some climates. Reliable service has been recorded over periods of fifteen and more years. Solaron Corporation sells a solar heating system using rock storage: this air heating unit will be complemented by a space cooling system in the near future, (6).

There is great flexibility as far as the container for the rocks or pebbles is concerned: insulated wooden bins, concrete or masonry containers and metal tanks have been used.

A number of units using masonry and concrete sections of buildings as heat storage systems have been built particularly in France. Performance appears good.

TABLE 10. PEBBLE BED STORAGE SYSTEMS

A. SYSTEMS PRESENTLY OR PREVIOUSLY IN OPERATION

Location (Sponsor)	Type (1)	Date (2)	Volume Ft ³ (3)	lbs. ft ² (4)	Storage Unit(5)	Carry Over(6)	Use (7)	Heat. % (8)	Ref. (9)	Ft ³ ft ² (10)	
Amado, Arizona (Bliss)	H	1954	1,300	412	B	-	H-(C) (air)	100	p.3	4	4 in. ϕ rock. ended in 1956.
Boulder, Colo. (Lof)	H	1945	180	35.9	-	-	H (air)	23	p.17	0.4	3/4 in. stone ended.
Denver, Colo. (Lof)	H	1959	254	40	T	-	H (air)	26	p.22	0.4	1 to 1.5 in. rocks still in use.
Weston, Mass. (Saunders)	H	1960	1,500	-	C	-	H (air)	~40	p.56	~3	Walls & floors = storage mass, 15 yrs. of use.
Corrales, N. H. (Davis)	H	1972	1,134	209	B	-	H (air)	75	p.67	2.6	Fist size stones.
Santa Fe, N. M. (van Dresser)	H	1958	-	-	-	-	H (air)	65	p.69	-	Storage in bricks, sand, stones under house.
Solaron Corp.	-	present	-	~50	B or other	~1	H (air)	~75-80	(6)	0.5	Values suggested by Solaron Corp.
Highbett, Australia (CSIRO)	B	1964	1,100	167	T	2	H-(C) (air)	90	p.90	1.8	0.75 in. ϕ stones

TABLE 10 PEBBLE BED STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Ft ³ (3)	lbs. ft ² (4)	Storage Unit(5)	Carry Over(6)	Use (7)	Heat. % (8)	Ref. (9)	Ft ³ ft ² (10)	
Odeillo, France (CNRS)	H	1967	1,017	-	C	-	H-(C) (air)	60-70	p.93	-	Concrete wall = storage and collector.
<u>B. SYSTEMS DESIGNED AND/OR UNDER CONSTRUCTION</u>											
Beulah, Colo. (Bullen Land Devel. C.)	H	1975	-	250	-	-	H (air)	75	p.17	-	Gravel, heat pump booster.
Denver, Colo. (Crowther)	H	1974	540	-	B	2-3	H-(C) (air)	85	p.21	1.4	Retrofit - 3/4 to 1.5 in. rocks.
Ft. Collins, Colo. (Colo. State U.-II)	H	1975	-	42	B	1-2	H-(C) (air)	75	p.23	-	2 in. ϕ rocks.
Wellington, Colo. (Perma)	H	1975	870	54	C	-	H-(C) (air)	85	p.28	0.7	1 in. ϕ stones.
Marlow, N. H. (Total Env. Action)	H	1975	75	160	C	4	H (air)	60-70	p.62	0.3	Storage = stone and masonry.
Santa Fe, N. M. (Allers)	H	1974	576	140	B	1 to 1.5	H (air)	60	p.68	1.15	1 to 3 in. ϕ stones.
Santa Fe, N. M. (Seton Village)	H	1975	704	136	B	1.5	H (air)	65	p.70	1.4	2 in. ϕ stones.
Santa Fe, N. M.	H	1974	-	-	-	-	H (air)	90	p.71	-	Walls & floors = storage.

TABLE 10: PEBBLE BED STORAGE SYSTEMS (Continued)

Location (Sponsor)	Type (1)	Date (2)	Volume Ft ³ (3)	lbs. ft ² (4)	Storage Unit(5)	Carry Over(6)	Use (7)	Heat. % (8)	Ref. (9)	Ft ³ ft ² (10)	
Mifflensburg, Penn. (Knepshield)	H	1975	720	-	B	1-3	H-(C) (air)	large	p.77	-	1-2 in ϕ stones.
Norwich, Vermont (Total Env. Action)	H	1975	2,400	279	B	2	H (air)	large	p.82	2.8	2.5 in ϕ stones.
Padern, Andovia (Padern College)	H	1975	918	134	-	3	H (air)	80	p.89	1.07	Masonry wall = storage.

Footnotes to Table 10

- (1) Type: H = house, B = commercial or office building.
- (2) Date: year of start of operation for units used or previously used, projected year of start of operation for units in design or in construction.
- (3) Volume: volume of storage material (rocks, masonry, ...).
- (4) lbs/ft²: ratio of pounds of storage material to area of collector in ft².
- (5) Storage Unit: B = bin, T = metal tank, C = concrete structure.
- (6) Carry-Over: number of days of carry-over by storage unit, generally quoted for January days.
- (7) Use: H = heating only, (C) = cooling by radiation to ambient without cooling equipment. All systems used forced air as heating system.
- (8) Heat.: percentage of the heat load supplied by the solar system.
- (9) Reference: a page number refers to reference (12) while a standard reference number refers to the general list of references.
- (10) ft³/ft²: ratio of volume of storage unit in ft³ to area of collector in ft².

b. Units in planning, construction or early period of use.

Fewer units using rock storage are planned for the near future than in the case for water storage units. Most of the units planned are located in hot and dry climatic zones where limited air cooling can be achieved by nocturnal cooling of the storage bed for use during the hot hours of the day.

The designs call for about 60 to about 80 percent of the heat load supplied by the solar system with relatively short carry over periods (1 to 3 days in most cases). This probably represents the economic optimum for the climates considered. The required amount of storage material under those circumstances is from about 50 to 150 lbs. per square foot of collector area. Pebbles are generally from about 1 to about 3 inches in diameter.

4. Phase-change Storage Units.

Only a few phase-change storage units have been or are being tested. Apparently no new units are planned for the near future as phase-change storage systems must still be considered as being in the R & D phase.

The only unit presently in use is the one incorporated in the Solar One house at the University of Delaware, Newark, Delaware. The system has performed adequately for at least two years, and its general characteristics described in Table 11 stress the advantages of this system as compared to water or rock systems, i.e. small mass and volume of the storage system required to supply a certain fraction of the heating load. The system also includes a "coolness" storage unit. Some technical problems are still encountered which will have to be solved before these units can be considered operational.

TABLE 11: PHASE-CHANGE STORAGE SYSTEMS

A. SYSTEMS PRESENTLY OR PREVIOUSLY IN OPERATION

Location (Sponsor)	Type (1)	Date (2)	Volume Ft ³ (3)	lbs. ft ² (4)	Storage Unit (5)	Carry Over Days(6)	Use (7)	Heat. %(8)	Ref. (9)	Ft ³ ft ² (10)	T °F(11)	
Newark, Del. (Solar One, U. of Delaware)	H	1973	216(x)	9.4	Bin(x)	-	H (air)	67	p.33 (10)	0.25	120	Na ₂ S ₂ O ₃ .SH ₂ O
			24	1.65	Bin	-				0.03	75	Na ₂ SO ₄ . 10H ₂ O
			240	11.05	-	-	-	-	-	0.28	-	Total for heating.
			Part of (x)	3	Same as (x)	-	C (air)	-	-		55	Heat-pump & elec- tric.
Dover, Mass. (Peabody)	H	1949	470	~60	T (5 gal. each)	6	H (air)	80	p.52	0.65	89	Na ₂ SO ₄ . 10H ₂ O void: 53% ended 1953
Princeton, N. J. (Princeton Solar Lab)	B	1959	275	-	-	-	H (air)	100	p.64	0.46	-	Eutectic ended.
Las Cruces, N. M. (State College)	H	~1955	-	-	-	2	H	50	p.71	-	-	Na ₂ SO ₄ . 10H ₂ O copper collector degraded after 7 years.

Footnotes to Table 11

- (1) Type: H = house, B = building.
- (2) Date: year of start of operation.
- (3) Volume: volume of storage unit.
- (4) lbs/ft²: weight of storage material in lbs per ft² of collector area.
- (5) Storage Unit: B = bin, T = metal tank.
- (6) Carry-Over: number of days of carry-over by storage unit, generally quoted for January days.
- (7) Use: H = heating, C = cooling - all units used forced air.
- (8) Heat: percentage of the heat load supplied by the solar system.
- (9) Reference: a page number refers to reference (12) while a standard reference number refers to the general list of references.
- (10) ft³/ft²: volume of storage unit in ft³ per unit area of collector.
- (11) T: melting point of storage medium used.

No reference of phase change systems being designed and/or under construction.

D. Cost comparison of Storage Units

1. Introduction

The present section attempts to compare the base price of storage units of the three types considered. The comparison deals only with the storage unit per se, i.e. container, storage medium and insulation. None of the auxiliaries such as heat exchangers, control systems and auxiliary power are included as these can only be defined for specific cases. It should thus be kept in mind that the unitary costs estimated below are only base costs to which other cost items will be added when the storage system is integrated in a complete solar unit to be optimized on the basis of some particular criteria.

2. Method

A detailed discussion of the cost comparison method is given in Appendix C.

The main aspects of the cost estimating procedure are given below. Assume Q Btu are stored for Δt_w days for winter heating between temperatures T_1 and T_2 , $T_2 > T_1$, where $T_1 - T_2$ is the useful range of temperature for heat storage.

The first cost, (FC), of storage is

(FC) = cost of container + cost of storage material + cost of insulation.

$$= C_c V + C_m M + A (a + 2 \sqrt{D k_{ins} C_h \alpha b}) \quad (1)$$

where V = Volume of container (ft³ or gal), this volume does include void space when necessary,

C_c = unitary cost of container (\$/ft³ or \$/gal.), this cost is usually a function of the capacity of the container,

M = Weight of storage material required (lbs)

C_m = Unitary cost of storage material (\$/lb).

A = Area of container (ft²). This quantity depends on the shape of the container selected.

$a + 2 \sqrt{Dk_{ins} C_h} ab$ = Optimized cost of insulation (\$/ft²). The thickness of the insulation is chosen so as to minimize the total (insulation + heat loss value) equivalent first cost. (see Appendix I).

a = first cost of insulation (\$/ft²). This is the base cost of any insulating job, essentially irrespective of the size of the insulation job. This contribution will generally be small compared to the cost of installation of insulation material (item b).

b = Cost of insulation (\$/in-ft²).

D = Loss factor (hr-°F) = $24 d_w (T_{Tw} - T_w) + 24 d_s (T_{Ts} - T_s)$.

d_w = Average number of winter days $\approx (HDD)/(65 - T_w)$.

T_w = Average winter temperature (°F).

T_{Tw} = Average storage temperature in winter (°F).

d_s = Average number of summer days $\approx (CDD)/(T_s - 75)$.

T_s = Average summer temperature (°F).

T_{Ts} = Average storage temperature in summer (°F).

k_{ins} = Loss factor of insulating material (Btu-in/hr-ft²-°F).

C_h = Cost of heat (\$/10⁶ Btu).

α = interest factor = $1 - (1 + i)^{-n}/i$ with i = rate of interest.

For $i = 0.06$ and $n = 20$ years, $\alpha = 11.47$.

Thus, the first cost per million Btu of storage capacity, $(FC)_m$ is

$$(FC)_m = (FC) \text{ (equation (1))} \times 10^6/Q \quad (\$/10^6 \text{ Btu}) \quad (2)$$

Assuming a straight depreciation over a 20 year period, the depreciation per million Btu-year used is

$$d = \frac{(FC)(\text{equation (1)}) \times 10^6}{20 \times Q_y} \quad (\$/10^6 \text{ Btu-year}) \quad (3)$$

where Q_y = total average Btu stored per year

$$= \frac{Q_w}{\Delta t_w} d_w + \frac{Q_s}{\Delta t_s} d_s - Q_L \quad (\text{Btu}) \quad (4)$$

Q_w = storage capacity (Btu) for winter storage.

Δt_w = average period of storage in winter (days).

Q_s = storage capacity (Btu) for summer storage.

Δt_s = average period of storage in summer (days).

$$Q_L = \text{yearly heat losses} = \frac{D k_{ins} A}{\delta^*} \quad (\text{Btu})$$

$$\delta^* = \text{optimum thickness of insulation (in)} = Dk_{ins} C_h \alpha/b)^{1/2}$$

3. Discussion

The first cost per million Btu of storage capacity depend on a number of factors as can be seen from equation (1).

a. Type of Storage

All other conditions being equal, the three terms contributing to the first cost of storage may be expressed as functions of intrinsic parameters of the storage media as (see Appendix C),

$$(FC)_m = \frac{c_c 10^6}{\rho(h_f + c\Delta T)} + \frac{c_m 10^6}{(h_f + c\Delta T)} + \frac{10^6}{\rho(h_f + c\Delta T)} f (a + 2 \frac{Dk_{ins} c_h^{ab}}{ab}) \quad (5)$$

where ρ = density of storage material (lb/ft³)
 h_f = heat of fusion of material (when applicable) - (Btu)
 c = specific heat of material (Btu/lb-°F)
 $\Delta T = T_2 - T_1$ = limits of storage temperatures
 f = area/volume ratio for storage container

For $\Delta T = 80^\circ\text{F}$ and typical values of the various parameters for water, rocks and salts, the three terms in (5) have values as shown in Table V.

It is apparent that for the same storage capacity, the water and rock containers are approximately equal in price, while the phase change material is much cheaper because of the smaller volume of material needed. On the same basis, the cost of storage material is lowest for water and approximately equal for rocks and salts. However, if the cost of containers for salts is included, the cost of phase change materials becomes the highest.

The area to insulate is the lowest for salts, intermediate for water and the highest for rocks. Combining these contributions to first cost and assuming the same type of insulation for each system, the total first cost is lowest for salts, intermediate for water and highest for rocks. In the case of rocks, however, heat transfer from the bed of pebbles to and through the walls of the container is less favorable than is the case for water, and it is thus expected that the cost of storage for rock systems will be comparable to that of water as less insulation is probably needed for the same overall losses.

TABLE 12

	Container \$/10 ⁶ Btu	Material \$/10 ⁶ Btu	Insulated Area ft ²	Total \$/10 ⁶ Btu	
				@ \$1/ft ²	@ \$2/ft ² (4)
Water	599 ⁽¹⁾	1.5	200 ⁽¹⁾	800	1,000
Rocks	469 ⁽²⁾	156	348 ⁽²⁾	973	1,321
Salts	170 ⁽³⁾	147 ⁽⁵⁾	112 ⁽³⁾	429	541
		350 ⁽⁶⁾		632	744

(1) Cylindrical metal tank @ \$0.40/gal (f≈1)

(2) Cubic wooden bin @ \$7/ft², with 15% void space (f≈0.78)

(3) Cubic wooden bin @ \$7/ft² with 30% void space (f≈1.39)

(4) Assumed cost of insulation.

(5) Materials alone (no containers)

(6) Assumed cost with containers: same as cost of material.

The depreciation for the three systems, equation (3), should follow the same pattern as that for first cost as it was assumed that all parameters except nature of the storage system were the same.

b. Influence of Storage Temperature

Assuming that T_2 , see equation (1), is the average storage temperature equal to T_{TW} , the storage temperature will influence the three terms in equation (1). The two first terms, vary inversely with storage temperature. The third term vary inversely with storage temperature and proportionally to the square root of the storage temperature: the effect of storage temperature is thus less pronounced on the insulation cost term.

c. Influence of Climate

All other conditions being equal, the effect of climate manifests itself in the third, insulation, term through the square root of D . The effect is complex as generally, as T_w decreases, the number of heating degree day increases. The overall effect is small: for locations from Bismarck, N. D. to San Antonio, Texas, the change in first cost per 10^6 Btu is only about 3% of its value for the southernmost location.

d. Period of Storage

The average period of storage influences the depreciation cost, $\$/10^6$ Btu-year. As seen from equations (3) and (4) as the period of storage increases, the total Btu stored per year decreases and the depreciation cost increases.

4. Comparative Storage Cost Estimates

a. Introduction

The present section attempts to compare the costs of storing a given amount of heat under comparable conditions by the three storage systems considered. The method used was described in section B. above.

Table 13 shows the numerical values used in the comparison:

1. The storage capacity is taken as a parameter.
- 2-3. The lowest and highest temperatures of storage are reasonable values which can be obtained during winter and are usable for heating. The range of useful temperatures for summer is much narrower and higher.
- 4-5-6. The number of winter days and the winter temperature are characteristic of northern Virginia climate. The average storage temperature is within the winter storage temperature range, which can be accepted for water, rocks and some phase-change materials.
- 7-8-9. These are the average summer values corresponding to items 4 to 6 for the same type of climate.
10. The insulation material was chosen to be fiberglass matts for all three systems.
- 11-12. Self-explanatory.
13. The first cost of installation of the insulation was estimated from Building Cost File, 1972, Eastern edition, Construction Pub. Co., N. Y., adjusted for 1975.
14. The loss factor D was evaluated for winter storage only or for winter and summer storage combined (i.e. when the stored heat is used for cooling purposes in summer).

TABLE 13
NUMERICAL VALUES USED IN COST ESTIMATES

1. Storage capacity, Q (Btu): 1, 5, 10, 20 and 40 x 10⁶
2. Lowest temperature of storage, T₁ (°F): 100 (winter) 170 (summer)
3. Highest temperature of storage, T₂ (°F): 180 (winter) 210 (summer)
4. Average number of winter days, d_w (days): 170
5. Average winter storage temperature, T_{tw} (°F): 170
6. Average winter temperature, T_w (°F): 35
7. Average number of summer days, d_s (days): 90
8. Average summer storage temperature, T_{ts} (°F): 190
9. Average summer temperature, T_s (°F): 75
10. Loss factor of insulation, k_{ins} (Btu-in/hr-ft²-°F): 0.27 (fiberglass matts)
11. Interest factor, α: 11.47 (6% - 20 years)
12. Cost of insulation, b (\$/in-ft²): 0.035 (with installation)
13. First cost of insulation, a (\$/ft²): 0.15
14. Loss factor, D (Hr-°F): 0.551 x 10⁶ for winter only
0.799 x 10⁶ for winter and summer
15. Cost of heat, C_h (\$/10⁶ Btu): 6.74 for electric heat
3.57 for fuel heat
16. Optimum insulation thickness, δ*(in): 18.1 winter electric
21.8 winter and summer electric
13.2 winter fuel
15.9 winter and summer fuel
17. Optimum cost of insulation:

$\sqrt{Dk_{ins} C_h ab}$ (\$/ft ²)	0.634	winter electric
	0.764	winter and summer electric
	0.462	winter fuel
	0.556	winter and summer fuel

15. Two cases are considered for the cost of heat supplied by the auxiliary (non solar) system:

Electric Heating: $2.3¢/\text{kWh}$ (100% efficiency), i.e. $\$6.74/10^6 \text{ Btu}$.

Oil Fired Heating: $40¢/\text{gal.}$ (80% efficiency), i.e. $\$3.57/10^6 \text{ Btu}$.

The optimum thickness of the insulation will depend on whether the solar system replaces electric or oil heat.

16. The optimum thickness is calculated for winter only or winter and summer operation, both supplemented by electric or oil heating. The optimum thickness is the same for all systems as the climatic conditions, storage capacity and temperatures of operations are assumed to be the same.

17. The same remarks made under 16 apply here.

b. Water Systems

The method of section 2 was first applied to water storage systems. The data is shown in Table 14, the numerical values of Tables 8 and 13 were used in the estimates.

Figure 23 shows the first cost and depreciation cost for winter or winter and summer storage in metal tanks with electric or fuel auxiliary systems. The first cost decreases sharply from about $\$1600/10^6 \text{ Btu}$ of capacity for small units ($\sim 1 \times 10^6 \text{ Btu}$) to about $\$1000/10^6 \text{ Btu}$ for $20 \times 10^6 \text{ Btu}$ capacity and over. The depreciation costs are significantly lower when the storage unit is used in winter and summer.

Figure 24 compares the first cost of storage for metal tanks, concrete and cinder block (waterproofed) containers. For all capacities a waterproofed cinder block container is the cheapest, while metal tanks and concrete units are similar in cost above about $20 \times 10^6 \text{ Btu}$ capacity. Figure 25 shows the corresponding depreciation costs.

TABLE 14

COST ESTIMATES OF WATER SYSTEMS

	1	5	10	20	40
Storage Capacity, Q (10^6 Btu)					
Weight, M (10^5 lbs)	0.125	0.625	1.25	2.50	5.0
Volume of water, V (gallons)	1,500	7,500	15,000	30,000	60,000
Volume of storage, V (ft^3) ¹	220	1,100	2,200	4,400	8,800
(gallons)	1,650	8,250	16,500	33,000	66,000
Loss area A (ft^2) cylinder ²	202	590	936	1,486	2,360
cubic ³	219	639	1,015	1,611	2,557
Cost of container, C_c (\$/gal) Tank ⁴	0.80	0.66	0.58	0.53	0.52
Concrete ⁵	1.45	0.87	0.68	0.54	0.53
Cinder Block ⁶	0.47	0.29	0.21	0.18	0.18
<u>Tank Storage</u>					
Winter-electric ⁷ Tank (\$)	1,320	5,445	9,570	17,490	34,320
Water (\$)	1.5	7.5	15	30	60
Insulation (\$)	286	837	1,327	2,107	3,346
Total (\$)	1,608	6,290	10,912	19,627	37,726
FC/ 10^6 Btu (\$/ 10^6 Btu)	1,610	1,258	1,091	981	943
Q_L (10^6 Btu/year)	1.66	4.85	7.69	12.2	19.4
Δt_{sw} (days)					
Q_y (10^6 Btu/year)	55	278	559	1,121	2,247
d_y (\$/ 10^6 - year)	1.46	1.13	0.98	0.88	0.84
<u>Winter & Summer⁸ Insulation (\$)</u>					
electric	339	990	1,571	2,494	3,960
Total (\$)	1,661	6,443	11,156	20,014	38,340
FC/ 10^6 Btu (\$/ 10^6 Btu)	1,661	1,290	1,116	1,000	959
Q_L (10^6 Btu/year)	2.0	5.84	9.26	14.7	23.4
ΔT_s (days) winter-summer					
Q_s (10^6 Btu)	0.5	2.5	5	10	20
Q_y (10^6 Btu/y)	100	502	1,007	2,019	4,043
d_y (\$/ 10^6 Btu-y)	0.83	0.64	0.55	0.50	0.47

TABLE 14: COST ESTIMATES OF WATER SYSTEMS (continued)

Winter-fuel ⁹ insulation (\$)	217	634	1,005	1,596	2,535
total (\$)	1,539	6,087	10,590	19,116	36,915
FC/10 ⁶ Btu (\$/10 ⁶ Btu)	1,539	1,217	1,059	956	923
Q _L (10 ⁶ Btu/year)	2.28	6.65	10.55	16.75	26.6
Δt_{sw}			3		
Q _y (10 ⁶ Btu/year)	54.4	276.7	556	1,117	2,240
d (\$/10 ⁶ Btu-year)	1.41	1.10	0.95	0.86	0.82
Winter & Summer ¹⁰ Insulation (\$)	255	745	1,181	1,875	2,978
Fuel Total (\$)	1,577	6,198	10,766	19,395	37,358
FC/10 ⁶ Btu (\$/10 ⁶ Btu)	1,577	1,240	1,077	970	934
Q _L (10 ⁶ Btu/year)	2.74	8.00	12.7	20.2	32.0
Δt_{sw}			3-1		
Q _S (10 ⁶ Btu)	0.5	2.5	5	10	20
Q _y (\$/10 ⁶ Btu/year)	99	500	1,004	2,013	4,035
d (\$/10 ⁶ Btu-year)	0.8	0.62	0.54	0.48	0.46
Concrete Storage Tank (\$)	2,393	7,178	11,220	17,820	34,980
Winter Electric ⁶ Water (\$)	1.5	7.5	15	30	60
Insulation (\$)	311	906	1,439	2,284	3,628
Total (\$)	2,706	8,092	12,674	20,134	38,668
FC/10 ⁶ Btu (\$/10 ⁶ Btu)	2,706	1,618	1,267	1,007	967
Q _L (10 ⁶ Btu/year)	1.8	5.25	8.34	13.2	21.0
Δt_{sw} (days)			3		
Q _y (10 ⁶ Btu/year)	54.9	278	558	1,120	2,246
d (\$/10 ⁶ Btu-year)	2.46	1.46	1.14	0.90	0.86
Cinder Blocks Tank (\$)	776	2,393	3,465	5,940	11,880
Winter Electric ⁶ Total (\$)	1,089	3,307	4,919	8,254	15,568
FC/10 ⁶ Btu (\$/10 ⁶ Btu)	1,089	661	492	413	389
d (\$/10 ⁶ Btu-year)	0.99	0.59	0.44	0.37	0.35

TABLE 14: COST ESTIMATES OF WATER SYSTEMS (continued)

Cinder Blocks					
Winter & Summer Electric ⁸					
Insulation (\$)	367	1,072	1,703	2,703	4,291
Total (\$)	1,145	3,473	5,183	8,673	16,231
FC/10 ⁶ Btu (\$/10 ⁶ Btu)	1,145	695	518	434	406
QL (10 ⁶ Btu/year)	2.2	6.3	10.0	15.9	25.3
Δt_s			3-1		
Qy (10 ⁶ Btu/y)	99.5	502	1,007	2,017	4,041
d (\$/10 ⁶ Btu/y)	0.58	0.35	0.26	0.21	0.20

Footnotes to Table 14

1. Includes 10% void space for drainage of collectors and expansion.
2. Cylindrical metal tank with optimized area to volume ratio
3. Cubic concrete or cinder block container
4. See Appendix II for cost of metal tank as function of storage capacity.
5. See Appendix III for cost of concrete tank as function of storage capacity.
6. See Appendix IV for cost of waterproofed cinder block container as function of storage capacity.
7. Winter storage for heating with auxiliary electric heating.
8. Winter (heating) and summer (cooling) storage with auxiliary electric systems.
9. Winter (heating) storage with auxiliary fuel heating.
10. Winter (heating) and summer (cooling) storage with auxiliary fuel systems.

First Cost
\$/10⁶ Btu

FIGURE 23: First Cost and Depreciation Cost of Water Storage
as Function of Storage Capacity

Metal tanks
Concrete Containers
Waterproofed Cinder Block Units

Waterproofed cinder block containers have not been used so far to our knowledge but are probably worth investigating in view of the present estimates.

c. Rock or Pebble Bed Systems

Table 15 shows the numerical data for rock storage systems. Only the cases of winter storage for heating with electric or fuel auxiliaries were considered as the unit is rarely used for cooling. Only the case of wood containers for the rocks was considered as it provides the least expensive storage system.

Figure 26 shows the first cost of rock storage with electric or fuel auxiliary systems as compared to water storage in metal tank or waterproofed cinder block units. The first cost of rock systems are lower than metal tank water systems and comparable to a hypothetical waterproofed cinder block water storage unit.

Figure 27 shows the corresponding depreciation costs about which the same comments as above can be made. It should, however, be stressed that water storage offers the possibility of cooling in summer while the rock system, generally speaking, does not.

d. Phase Change Systems

A phase-change system working within the same temperature range as the water and rock system was considered. Following Table I, $\text{Na}_2\text{HPO}_4 \cdot 12\text{H}_2\text{O}$ could meet these requirements. The cost of the salt material was assumed to be the same as that of Glauber salt which can be obtained in bulk in industrial grade. The base price of the material was doubled to account for a container for the material.

A single salt used for winter heat storage cannot supply storage for summer air cooling. Thus, the case of a storage system including winter storage through one salt and summer storage at higher temperature through another salt was considered: the system was designed to supply the same winter and summer storage capability as was the case for the water system considered in Table 14.

TABLE 15
COST ESTIMATES OF ROCK SYSTEMS

	1	5	10	20	40
Storage Capacity, $Q(10^6 \text{ Btu})$					
Weight, $M(10^5 \text{ lbs})$	0.625	3.125	6.25	12.5	25
Volume of Rocks, $V(\text{ft}^3)$	446	2,232	4,464	8,928	17,857
Volume of Storage, $V(\text{ft}^3)^1$	558	2,790	5,580	11,161	22,321
$V(\text{gal})$	4,174	20,870	41,741	83,482	166,964
Loss Area $A(\text{ft}^2)$ cube	407	1,189	1,888	2,996	4,757
Cost of Container, $C_c(\$/\text{gal})^2$ (wood bins)	0.16	0.08	0.07	0.07	0.07
<u>Winter-electric</u> ³ container (\$)	668	1,670	2,922	5,844	11,687
rocks (\$)	156	781	1,562	3,125	6,250
insulation(\$)	577	1,686	2,677	4,248	6,745
total (\$)	1,401	4,137	7,161	13,217	24,682
$FC/10^6 \text{ Btu} (\$/10^6 \text{ Btu})$	1,401	827	716	661	617
$Q_L(10^6 \text{ Btu/year})$	3.34	9.77	15.5	24.6	39.1
$\Delta t_{sw}(\text{days})$			3		
$Q_y(10^6 \text{ Btu/year})$	53.3	273.6	551	1,109	2,228
$d(\$/10^6 \text{ Btu-year})$	1.31	0.76	0.65	0.60	0.55
<u>Winter-fuel</u> ⁴ insulation (\$)	437	1,277	2,028	3,218	5,109
total (\$)	1,261	3,728	6,512	12,187	23,046
$FC/10^6 \text{ Btu} (\$/10^6 \text{ Btu})$	1,261	746	651	609	576
$Q_L(10^6 \text{ Btu/year})$	4.59	13.4	21.3	33.8	53.6
$\Delta t_{sw}(\text{days})$			3		
$Q_y(10^6 \text{ Btu/year})$	52	270	545	1,099	2,213
$d(\$/10^6 \text{ Btu-year})$	1.21	0.69	0.60	0.55	0.52

Footnotes to Table 15

- (1) Includes 25% void space for air circulation.
- (2) See Appendix V.
- (3) Winter Storage for heating with auxiliary electric heat.
- (4) Winter Storage for heating with auxiliary fuel burner.

FIGURE 26 - First Cost of Rock Storage System as
Function of Storage Capacity

FIGURE 27 Depreciation Cost of Rock Storage System as
Function of Storage Capacity

The numerical data are presented in Table 16.

Figure 28 shows the first cost of various phase-change units as function of storage capacity. The cost of water storage in metal tanks are shown for comparison. It is apparent that:

- Phase-change winter storage with electric or fuel auxiliary is cheaper than water storage in tanks, comparable to water storage in waterproofed cinder block units and slightly lower than rock storage in wood bins (see Figure 26).
- Double phase-change systems designed for heating and cooling are cheaper than water storage in tanks but more expensive than water storage in waterproofed cinder block units. They compare to rock storage but the latter does not offer the possibility of cooling. Storage of phase-change systems in cinder block containers increases somewhat the first cost.

Figure 29 shows the depreciation costs of phase-change systems as function of storage capacity. There is little difference between the various phase-change systems considered: the depreciation costs of these systems are lower than those for water winter storage in tanks and compare with those of water used for winter and summer storage (see Figure 25).

TABLE 16

COST ESTIMATES OF PHASE-CHANGE SYSTEMS

	1	5	10	20	40
Storage Capacity, Q (10^6 Btu)					
Weight, M (10^4 lbs)	0.66	3.3	6.6	13.2	26.3
Volume of Salt, V (ft^3) ¹	69.5	347	695	1,389	2,768
Volume of Storage, V (ft^3) ²	92.4	462	924	1,847	3,681
V (gal)	691	3,452	6,914	13,818	27,537
Loss Area, A (ft^2) ³	123	358	569	903	1,431
Cost of Container, C _c (\$/gal) Bin ⁴	0.30	0.17	0.13	0.09	0.075
<u>Wood Bin</u>					
Winter-electric ⁵ Bin (\$)	207	587	890	1,244	2,065
Salt (\$) ⁶	264	1,320	2,640	5,280	10,520
Insulation (\$)	174	508	807	1,280	2,029
Total (\$)	645	2,415	4,337	7,804	14,614
FC/ 10^6 Btu (\$/ 10^6 Btu)	645	483	434	390	365
Q _L (10^6 Btu/Year)	1.0	2.9	4.7	7.4	11.8
Δt_{sw} (days)			3		
Q _y (10^6 Btu/year)	55.7	280	562	1,126	2,255
d _y (\$/ 10^6 Btu-year)	0.58	0.43	0.39	0.35	0.32
<u>Winter-fuel⁷ insulation (\$)</u>					
total (\$)	132	384	611	970	1,537
FC/ 10^6 Btu (\$/ 10^6 Btu)	603	2,291	4,141	7,494	14,122
Q _L (10^6 Btu/year)	603	458	414	375	353
Δt_{sw} (days)	1.4	4.0	6.4	10.2	16.1
Q _y (10^6 Btu/year)			3		
d _y (\$/ 10^6 Btu-year)	55.3	279	560	1,123	2,251
	0.55	0.41	0.37	0.33	0.31

TABLE 16 COST ESTIMATES OF PHASE-CHANGE SYSTEMS (continued)

Winter-Summer Electric ⁸	1	5	10	20	40
Summer Storage Capacity (10^6 Btu) ³	0.5	2.5	5	10	20
Weight, M (10^4 lbs) ¹⁰	0.55	2.7	5.4	10.9	21.7
Volume of Salt, V (ft^3) ¹¹	50	245	491	991	1,973
Total Volume of Storage, V (ft^3) ¹²	159	788	1,577	3,165	6,305
V (gal)	1,189	5,893	11,796	23,674	47,162
Loss Area, A (ft^2) ¹³	176	512	813	1,293	2,048
Cost of Container, C _c (\$/gal) Bin ⁴	0.25	0.14	0.10	0.077	0.07
Bin (\$)	297	825	1,180	1,823	3,301
Salt (\$)	484	2,400	4,800	9,640	19,200
Insulation (\$)	295	859	1,364	2,170	3,437
Total (\$)	1,076	4,084	7,344	13,633	25,938
FC/ 10^6 Btu (\$/ 10^6 Btu)	1,076	817	734	682	648
Q _L (10^6 Btu/y)	1.74	5.07	8.04	12.8	20.3
Δt_{sw}					
Q _y (10^6 Btu/year)	100	503	1,009	2,021	4,046
d (\$/ 10^6 Btu-year)	0.54	0.41	0.36	0.34	0.32
3-1					
Winter-Summer Fuel ¹⁴					
Insulation (\$)	222	646	1,026	1,632	2,585
Total (\$)	1,003	3,871	7,006	13,095	25,086
FC/ 10^6 Btu (\$/ 10^6 Btu)	1,003	774	700	655	627
Q _L (10^6 Btu/y)	2.4	6.9	11.0	17.5	27.8
Δt_{sw}					
Q _y (10^6 Btu/year)	99.3	501	1,006	2,016	4,039
d (\$/ 10^6 Btu-year)	0.51	0.39	0.35	0.32	0.31
3-1					
Concrete Blocks ¹⁵					
Winter-Summer Electric ¹⁶					
Cost of Container, C _c (\$/gal) ¹⁷	0.49	0.32	0.24	0.18	0.175
Container (\$)	583	1,886	2,831	4,261	8,253
total (\$)	1,362	5,145	8,995	16,071	30,890
FC/ 10^6 Btu (\$/ 10^6 Btu)	1,362	1,029	900	804	772
d (\$/ 10^6 Btu-year)	0.68	0.51	0.45	0.40	0.38

Footnotes to Table 16

1. $\text{Na}_2\text{HPO}_4 \cdot 12\text{H}_2\text{O}$ is assumed to be the phase change material.
2. 33% void space included.
3. Cubic shaped container assumed.
4. See Appendix V for unitary costs.
5. Winter storage with electric auxiliary heat.
6. Cost of salt: twice cost of Glauber salt (see Table I) to take containers into account.
7. Winter Storage with fuel auxiliary burner.
8. Two salt system supplying same heating-cooling capability as water system of Table VII. Winter: $\text{Na}_2\text{HPO}_4 \cdot 12\text{H}_2\text{O}$ as (5) and (7); summer storage: $\text{AlK}(\text{SO}_4)_2 \cdot 12\text{H}_2\text{O}$ - see Table I.
9. Summer storage capacity of water system - (Table VII).
10. Weight of summer storage salt.
11. Volume of summer storage salt.
12. Volume of winter and summer storage combined (see (2) above).
13. For winter and summer systems assumed to be in one container.
14. Two salt system as (8) but with fuel auxiliary.
15. Cinder block container with inside surface finish to improve air flow.
16. Comparable to (8) except for type of container.
17. See Appendix IV.

Metal Tank Water Storage - Electric Auxiliary
 Phase Change Storage - Winter - Electric Auxiliary
 Phase Change Storage - Winter - Fuel Auxiliary
 Phase Change Storage - Winter & Summer - Electric auxiliary
 Phase Change Storage - Winter & Summer - Fuel Auxiliary
 Phase Change Storage - Winter & Summer - Electric Auxiliary
 (Cinder Block Container)

E. Summary and Recommendations

1. Summary

Table 17 summarizes the discussion of the previous sections. Some general comments regarding the various items in the table may be made:

Items 1 and 2: Using water as reference storage material, phase-change systems are about 1/3 lighter and compact, while rock systems are about 5 times heavier and about 3 times larger. Note that in the case of phase-change systems, the comparison is based on a phase-change system capable of performing the same heating and cooling functions as the water reference system. The rock system is only capable of heat storage for winter heating.

Item 3: Water systems can heat a building and store heat simultaneously, while the other systems can perform only one function at a time.

Item 4: Provided the collector system enables high enough temperatures of storage to be reached, unpressurized water systems can be used for heating and cooling. Phase-change systems generally require two materials having adequate temperature of fusion to perform both functions. Rock systems do not appear practical for use in conjunction with cooling machinery such as absorption units.

Items 5 and 6: Water is an easy material to handle but requires some precautions to prevent freezing and corrosion. Rock storage systems are the most trouble-free and don't require any protection devices. Phase-change materials are difficult to handle, can be corrosive, display supercooling and recrystallization effects and reduced performance after thermal cycling, (11). Some of these problems have apparently been solved, (11), but significant R & D work is still needed before these systems may be considered as operational.

TABLE 17

SUMMARY COMPARISON OF STORAGE SYSTEMS

	<u>Water</u>	<u>Rocks</u>	<u>Phase-Change</u>
1. Ratio of weights for storing unit of heat under same conditions	1	5	~0.7
2. Ratio of volumes for storing unit of heat under same conditions (void included)	1	2.5-3	~0.65
3. Storage and heating	Simultaneous	one or the other	
4. Heating and Cooling	Yes	Difficult	Yes, but generally requires 2 systems.
5. Handling of storage material	Easy	Easiest	Difficult so far.
6. Problems	Freezing Corrosion	-	Supercooling Loss of performance on cycling.
7. Heating system	Water or air well known	Air Well known	Air-(water) some unknown.
8. Cost - Investment (\$/10 ⁶ Btu)	1,600 to 1000 could be 1000 to 400	1,400 to 600	1000 to 600 could be higher.
	(See Figure VIII)		
Depreciation (\$/10 ⁶ Btu-y)	1.40 to 0.85 could be 0.15 to 0.2	1.30 to 0.55	0.5 to 0.30 could be higher.
	(See Figure IX)		
9. State of the Art	In use	In use	R & D needed.

Item 7: Water storage can be used in conjunction with forced central air heating (water-air exchanger) or hot water systems (radiators or fan-coil systems). Rock storage is combined with forced air central heating. The technology of these heating systems is well established. Phase-change systems can be used with air or eventually water heating systems. Although the general principles of these heating systems are well-known, there are some uncertainties as far as the actual heat transfer mechanism at, and within, the storage material is concerned.

Item 8: This first cost of water storage units with metal tanks is the highest, \$1000/10 Btu for 20 x 10 Btu capacity and over (see Figure 30). However, this cost could be brought down to \$500/10 Btu or less if water-proofed cinder block tanks were used. Rock and phase-change materials cost approximately the same, i.e. about \$700/10 Btu. These cost refer to systems designed for heating and cooling except in the case of rock storage systems. The cost of the phase change system could be higher, if as was reported in some cases, the storage capacity of the material decreases after a number of heating cycles (3,8). Similar comments may be made about the depreciation costs, see Figure 31. The relatively high depreciation cost of rock systems is due to the fact that these units are used only in winter.

Item 9: Both water and rock systems are presently in use. The phase-change systems, although promising from several aspects, need more R & D work before they can be realistically evaluated.

2. Recommendations

In view of its versatility, flexibility and ease of handling, water storage, should be further developed. Two areas appear worth investigating: lower cost storage containers (waterproofed cinder block units and above ground swimming pool techniques for instance) and use of two or more containers to take advantage of temperature stratification.

Research and development work should be pursued on phase-change systems in order to evaluate their potential.

FIGURE 30 - First Costs of Storage Systems as Function of Storage Capacity

FIGURE 31 - Depreciation Cost of Storage Systems as Function of
Storage Capacity

W = Water in Metal Tank

R = Rocks in Bin

W' = Water in Water Proofed Cinder Block Unit

S = Phase Change Material in Bin

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VI. SOLAR-ASSISTED HEAT PUMPS

A. Summary

A heat-rejection process is inherent in the operation of any cooling machine, hence a cooling machine can also be used for heating. When such a machine is employed for the purpose of heating, it is called a heat pump. A heat pump is nearly always designed for the dual function of heating and cooling, hence the term "heat pump" generally implies a combination heating and cooling machine.

In either mode of operation, a heat pump receives heat from a relatively low-temperature heat source, and rejects heat to a higher temperature heat sink. The amount rejected is always higher than the amount received from the low-temperature source, the difference being equal to the amount of energy required to operate the machine. In the cooling mode, the low-temperature heat source is the cooled space, and the heat sink is some available external medium, such as ambient air or cooling water. In the heating mode, the external medium becomes the low-temperature heat source, and the heated space serves as the heat sink.

The temperature of the low-temperature heat source is fixed by design for cooling, but not for heating. In fact, performance improves as this temperature increases when the machine is used for heating. This permits the use of a solar-heated fluid (usually a liquid) as the external low-temperature heat source. A system employing solar energy in this manner is termed a solar-assisted heat pump.

Over a full heating season, an electrically powered heat pump, using ambient air as the external heat source, can provide about twice as much heat per unit of electricity as a "conventional" electric resistance heating system. Electrically powered solar-assisted heat pumps can roughly redouble this gain, i.e. they can provide about 4 units of heat energy-output for every unit of electrical energy input.

There is no analog to solar assistance for cooling, i.e. "solar-assisted cooling" is not a viable concept. In contrast, both the cooling and heating functions can be "solar-powered", as discussed in the report on solar-powered cooling systems.

Seven solar-assisted heat pump installations were reported on at the June 1975 ERDA sponsored workshop on Solar Energy Heat Pump Systems for the Heating and Cooling of Buildings. The primary reasons for this relatively low number are thought to be: i) the inability to utilize the collected solar energy year-around causes the installation of solar equipment to become cost-prohibitive and ii) the use of heat pumps in general is still relatively uncommon.

The early heat pump installations were not properly designed for the operating conditions, and hence experienced a very high failure rate. Although current heat pumps are reportedly as reliable as fossil fuel heating systems, the bad experiences of the past seem to be still exerting a large negative influence on their acceptance. As this influence recedes, and as the scarcity and cost of oil, gas and electricity increases, it is anticipated that heat pumps will become one of the most widely used types of heating system.

Near-term research and development consists primarily of continuing efforts toward improvement of heat pump performance and the design of solar-assisted installations. One area of concentration should be concerned with determining the optimum manner of using the storage system in these installations. It is also suggested that availability analyses, rather than energy analyses, be employed as the most appropriate method for identifying those components for which design improvements can yield the greatest gains.

B. Heat Pumps

1. Principle of Operation

As explained in (Section VII., "Solar-Powered Cooling Systems"), a cooling machine performs the function of maintaining a given space at a temperature below the prevailing ambient temperature. This is done by judicious location of the components of the cooling machine, i.e. the "Cooling-load-input space" of the machine is placed in thermal contact with the space to be cooled and the "heat-rejection space" of the machine is placed in thermal contact with the ambient "heat sink". If the location of these two components were switched, the machine would still operate, but a different phenomenon would occur, viz., there would be an "ambient heat input" into the machine and heat rejection from the machine to the given space, with an accompanying rise in the space's temperature. When employed in this configuration, the machine serves as a heater rather than a cooler, and it is called a heat pump. The cooling-load-input space and the heat-rejection space of the cooling configuration will be called the ambient-heat-input space and the heating-load-output space, respectively, of the heat pump configuration. The rate of the heat transfer from the heating-load-output space is called the heating capacity of the heat pump.

2. Coefficient of Performance

There must be a power energy input for both the cooling and heating configuration of a particular machine, and its function is the same for both, viz., to provide the "driving energy" for the machine. The ratio of heating capacity to power input is the coefficient of performance (COP) of a heat pump. Note that this definition is different than that for the COP of a cooling machine. Using the energy balance, it can easily be shown that for any given machine the value of heat pump COP is equal to the value of cooling COP plus one, under equal conditions of operations for both configurations.

The energy balance also shows that the COP of a heat pump is always greater than one. This applies to heat-powered heat pumps as well as work-powered heat pumps, (although COPs for work-powered heat pumps are inherently much higher, just as they are for work-powered cooling machines). Hence any source of space heat can be augmented by using the source indirectly as the heat input to a heat pump, rather than directly as the source of space heat. In particular, in the case of electrical heating, the heat pump offers a decided advantage over the "conventional" resistance heaters, as resistance heaters inherently operate at a COP of one.

3. Heat Pumps With Solar Energy Input

Energy input to a heat pump occurs at two distinct locations within the machine, just as it does for a cooling machine. The heat pump application, however, permits the low-temperature heat input to be at ambient temperature or higher; this is not permitted in a cooling application. It is therefore possible to collect heat in a solar collector and use it as the ambient-heat input to either a work-powered heat pump or a heat-powered heat pump. A heat pump/solar collector system assembled in this manner is called a solar-assisted heat pump. It is also possible to use collected solar heat as the heat power input to a heat-powered heat pump. A heat pump/solar collector system assembled in this manner is called a solar-powered heat pump.

4. Heating vs. Cooling

The "solar assistance" in solar-assisted heat pumps is limited to the heating function of the machine. "Solar-assisted cooling" is not a viable concept because, as mentioned above, one of the heat inputs to a cooling machine must be at a lower-than-ambient temperature. In order to utilize solar energy for cooling, the solar energy must be used as the power input, and the machine must therefore be of the heat powered variety. To power an electrically operated

cooling machine with solar energy would require the addition of a heat engine to convert the solar energy to mechanical (or electrical) energy, which would in turn be used to power the cooling machine. The Rankine Cycle-Vapor Compression (RCVC) solar-powered cooling system, as discussed in section VII.C.3.c. of "Solar-Powered Cooling Systems", is of this type. When this type of system is used in the heat-pump mode, it can be either solar powered or solar assisted. For example, Battelle Columbus Laboratory's pivoting-tip vane RCVC system is designed to operate as a solar-powered heat pump as well as a solar-powered air-conditioner.

5. Solar-Assisted Heat Pump System Configurations

a. Heating Units

It is important to note the restriction that a heat-powered cooling machine is required for "solar power" but not for "solar assistance". Hence, e.g., electrically powered heat pumps can be solar assisted, but not solar powered. Solar-assisted heat pumps can be used as a primary heating system, or as the auxiliary heat source in a solar heating system. As a primary system, the heat pump would be used to furnish all of the required heating, with ambient heat input from a solar-collector-heated fluid (air or liquid). As an auxiliary heat source, the heat pump would be used with solar ambient heat input only when the collected or stored solar heat was not sufficient to provide the required heating. The "primary" use is preferred, because the collector temperature can be kept lower. At lower temperatures, the collector is more efficient, hence the contribution of solar energy to the total seasonal heat load is increased. In practice, it is most likely that some combination of these two types of uses would be employed, with the heat pump being by-passed under conditions of light heating demands and high insolation, or when an abundance of stored heat is available.

b. Collectors

Solar-assisted heat pumps are well adapted to low solar-energy-collection temperatures. As such they present perhaps the optimum application for the state-of-the-art flat-plate collectors. In contrast to solar powered applications, no gain in the overall system performance of a solar-assisted heat pump can be realized from higher collector temperatures.

c. Storage Systems

Solar-assisted heat pump systems can utilize high temperature storage, "ambient" temperature storage, or some combination of the two. As one example, consider high temperature storage serving as the source of space heat. The solar collector could provide heat directly to storage under conditions of sufficient insolation. Under slightly worse conditions, the solar-assisted heat pump could be used to maintain the storage temperature, i.e. the heat pump would be turned on and the collected solar heat would be re-routed from the collector to the ambient heat input portion of the heat pump. Under even more adverse conditions, with no significant insolation, the storage could be "switched over" to serve as the ambient heat source for the heat pump, with space heating supplied directly by the heat pump. As a final precaution, an additional heat source could be provided to insure maintenance of the storage temperature above some minimum acceptable value.

CI. State-of-the-Art Review

1. Commercially Available Heat Pumps

a. Electrically Powered Heat Pumps

The advantage afforded by heat pumps can be most easily exploited in those cases where a building is electrically heated and cooled, by designing the central air-conditioning system to perform as a heat pump also. The machines

currently being marketed as heat pumps are dual-function machines of this type. They are available from a diversity of manufacturers including General Electric, Westinghouse, Singer, Lennox, Carrier, Fedders and Trane. In comparison to an air-conditioning only system, heat pumps generally require heavier duty compressors, higher capacity heat exchangers, more complicated control systems and additional components, such as a reversing valve, check valves and an accumulator. As a result, the price of a heat pump system is comparable to the price of a more conventional furnace heating/electrical central air-conditioning system [1].

b. Energy Sources and Sinks

Heat pump installations employ both water and air in all possible combinations as ambient-heat-input and heating-load-output media. Typical residential systems are "air-to-air", while typical commercial systems are "water-to-water". The remaining combinations, i.e. "air-to-water" and "water-to-air", can be used in special circumstances.

A disadvantage encountered in ambient-air-source heat pumps is that a decrease in the ambient air temperature causes a simultaneous increase in the heating load and decrease in the heating capacity of a given heat pump. Therefore, a heat pump capable of meeting the maximum heating load is considered an uneconomical design in most cases. The usual practice is to size the dual-function machine to provide full air-conditioning during the cooling season, and to install either electrical resistance type "strip heaters" or a fossil-fuel furnace to supplement the heat pump's output as required.

A further problem encountered in ambient-air-source heat pumps is that of coil freeze-up, i.e. the condensing and freezing of moisture from the air onto the exposed surfaces of the evaporator, when the outside air temperature is near or below 32°F. Some sort of automatic defrosting mechanism must be provided to alleviate this problem. Defrosting is most commonly accomplished by a short-time reversal of operation from heat pump to cooling mode, causing a rapid internal heating of the coils, thereby melting the ice.

c. Dual-Function Design

It is not necessary to physically interchange the location of the evaporator and condenser in order to switch a given machine between the heating and cooling functions. Instead, the same piece of equipment that functions as a condenser in the air-conditioning mode can be made to function as an evaporator in the heat-pump mode, and vice versa, by merely reversing the direction of flow through the condenser, expansion valve, and evaporator. In this manner, a central air-conditioning system can also be employed as a heat pump by providing the necessary switching valves and by incorporating heat exchangers (condenser and evaporator) that are capable of handling the higher capacity of the two functions for which they are used.

An alternate method of switching between the heating and cooling functions without switching the physical location of the evaporator and condenser is to switch the "external energy flows" rather than the refrigerant flow. This type of switching is accomplished most easily in water-to-water installations, but it can be applied to other types also. For example, when heating is required in a given space, air from the space can be ducted to the condenser while outside air is ducted to the evaporator. Switching the duct flow, so that outside air flows to the condenser and space air flows to the evaporator, will cause the machine to function as a cooler for the space.

d. Typical Performance Characteristics

The variation of heating capacity and COP with outdoor air temperatures for a typical heat pump system [2] is shown in Figure 32. The drop in performance at 40°F is caused by the need for defrosting of the outdoor coils that occurs below this temperature. During defrosting there is power input with no heat output, causing a drop in the time-average total heat output.

As the outdoor air temperature increases, both the heating capacity and COP increase. For a given change in outdoor air temperature there is a greater

FIGURE 32: HEATING CAPACITY AND COEFFICIENT OF PERFORMANCE VS OUTDOOR AIR TEMPERATURE FOR LENNOX HP8-511V AND HP8-513V (CB3-65V AND CBH3-65V) HEAT PUMP ASSEMBLY

percentage increase in the heating capacity than in the COP. This is due to the effect of a constant volume air flow rate across the condenser coil, which causes the refrigerant condensing temperature to increase from 80°F to 111°F as the outdoor-air-temperature increases from 15°F to 65°F.

e. System Reliability

The first heat pumps were manufactured and marketed in the early 1950's, and proved to be very unreliable. The main reason for their failure is that their design followed closely that of air-conditioning only systems, whereas their operating conditions imposed much higher stresses on the compressors. The result was a high rate of compressor failure. Other problems occurred in the operation of the switchover valves and the automatic defrosting systems. Today's heat pump systems have reportedly been improved to the point of being very reliable, in fact as reliable as fossil fuel heating systems [3].

2. Solar-Assisted Heat Pumps

A solar-assisted heat pump will operate at "effective outdoor temperatures" that are significantly higher than the actual outdoor air temperatures. Under these conditions, the average COP for the entire heating season can be greatly improved. One prediction for a typical case [4], is that the season-average COP can be raised from 2 to 3.5 by adding solar assistance. This could result in an even more significant increase in the total amount of heat provided by the heat pump, e.g. the "typical" performance curves of Figure 32 show that the heating capacity triples as the COP increases from 2 to 3.5. Solar assistance can provide other advantages also, e.g. if enough solar assistance is provided, the need for auxiliary resistance heaters and coil defrosting can be eliminated. Also, because of the higher ambient-heat-input temperatures provided by solar assistance, the compressor operates in a "lower stress" region [4], thereby reducing the chance for compressor failure.

To date, few solar-assisted heat pump installations have been designed or developed. At the June 1975 ERDA sponsored workshop on Solar Energy Heat Pump Systems for the Heating and Cooling of Buildings held at Pennsylvania State University, seven of thirty-one papers presented solar-assisted heat

pump installations. The other papers covered related topics, such as solar-powered heat pumps, heat pump computer models and the general characteristics of heat pumps.

D. Near-Term Research and Development Potential

Perhaps the main reason that solar-assisted heat pumps have received so little use is that heat pumps in general have not become a popular type of heating system. This general lack of acceptance is probably due partly to the relatively recent development of the electrical heating concept in general, but primarily to a lingering mistrust of the reliability of heat pumps caused by the high failure rate of the early systems. Assuming that heat pump systems have matured to the point of "full reliability" as reported [4], it is suggested that "public relations" efforts be undertaken to dispell the old fears and promote heat pumps as the "energy conservative", and also less costly, method of electrical heating.

It must also be remembered that the solar-assisted heat pump system can utilize solar energy only for heating. The current prevailing trend seems to emphasize the use of solar energy in cooling systems also, in order to get more use from the investment made on the solar collectors. Thus considerable design and development work on solar-powered cooling systems is underway, as discussed in section C.3.c.(4) of "Solar-Powered Cooling Systems." Any solar-powered cooling system can be switched to operate as either a solar-powered heat pump or a solar-assisted heat pump. Perhaps some economic trade-off studies should be undertaken to determine if there are any conditions under which a solar-powered heat pump could give better performance if it were operated as a solar-assisted heat pump instead.

Efforts should be undertaken to determine the optimum manner of component utilization . This includes choices such as when to switch from direct solar

heating to solar-assisted heat pump heating, and whether to use high temperature or "ambient temperature" storage.

Availability analyses, as discussed in section C.4.j of "Solar-Powered Cooling Systems", can be applied to solar-assisted heat pumps for the same purposes as they can be applied to solar-powered cooling systems. It is suggested that these "second law" analyses be undertaken as the only appropriate method of assessing the thermodynamic performance of these systems, and of identifying those components having the greatest potential for payback of investments made toward improved performance.

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VII. SOLAR POWERED COOLING SYSTEMS

A. Summary

Cooling machines can be broadly classified as either work-powered or heat-powered. The distinction is important because heat-powered cooling machines necessarily require more energy input than work-powered cooling machines for the same cooling load. Hence, the coefficient of performance, i.e. the ratio of cooling load to energy input, is inherently lower for heat-powered cooling machines. In addition to the amount of energy supplied, the temperature at which this energy is received is of importance for heat-powered cooling machines, as the theoretical coefficient of performance increases with an increase in the supply temperature.

A solar-powered cooling system combines a solar collector with a cooling machine. Since the solar energy is received and collected as heat, the cooling machine must be of the heat-powered variety. The currently operational solar-powered cooling systems employ either absorption or Rankine cycle-vapor compression cooling machines. These systems have coefficients of performance of about 0.6 with heat supplied from flat-plate collectors at temperatures in the range of 170°F to 215°F. Under these limitations, the practical state-of-the-art application of solar-powered cooling systems is limited to air-conditioning with water cooling.

Arkla Industries manufactures a three-ton "Solaire model 501-WF" absorption system designed for heat input from flat plate solar collectors. The current selling price is \$3000. They also plan to manufacture a "Solaire model WF-300", which is a twenty-five-ton capacity system for similar applications. Rankine cycle-vapor compression systems have not as yet reached the marketing stage, but it is anticipated that they can be cost-competitive with absorption systems.

The economics of both types improves considerably in larger sizes.

Other types of heat-powered cooling machines are being investigated for use with solar power. These include a Vuilleumier cycle system, vapor-jet ejector systems and intermittent cycle systems. Also, a desiccant drying type of air-conditioning system that uses solar energy as part of the heat input needed to dry the desiccant is being developed and tested.

The following recommendations for near-term research and development of solar-powered cooling systems are made:

1. Rather than designing for a specific "operating point", designs should be aimed at providing optimum performance over the complete range of conditions provided by solar collectors.
2. Provide a feedback control to allow the system to seek out and operate at the condition of maximum output under the prevailing input conditions.
3. Re-investigate the characteristics of various absorbent/refrigerant combinations to determine if there are better ones than those currently being used at the relatively low heat-input temperatures provided in solar-powered cooling systems.
4. Continue work on the development of cost-effective collectors that can supply energy in amounts equal to that supplied by state-of-the-art flat plate collectors, but at significantly higher temperatures.
5. Continue development work on gas-cycle, vapor-jet, and intermittent-cycle cooling systems.
6. Analyze the operation of the systems from an availability viewpoint, rather than an energy viewpoint, in order to identify those components from which the greatest gains can be achieved through design improvements.

B. DESCRIPTION OF COOLING MACHINES

1. General

Whenever two adjacent spaces are at different temperatures, there is a spontaneous flow of energy from the hotter space to the colder space. The terms "heat" and "heat transfer" are often used to denote the energy and the transfer process, respectively when this phenomenon occurs, and that usage will be followed herein. If there are no other energy transfers, heat transfer will tend to raise the temperature of the colder space and lower the temperature of the hotter space. If the colder temperature is to be maintained, it is necessary that an amount of heat equal to that entering the cold space be removed.

The purpose of a cooling machine (freezer, refrigerator, air-conditioner) is to accomplish this heat removal from the cold space. This cooling of the cold space is accomplished by generating, within the cooling machine, a space having a temperature lower than that to be maintained in the cold space, thereby effecting heat transfer from the cold space to the cooling machine. The rate of heat transfer from the cold space to the cooling machine is called the cooling load of the machine; that portion of the machine where the cooling load enters is called the cooling-load-input space of the machine.

Within the cooling machine, there are one or more fluids (depending upon the design of the particular machine) which participate in the energy exchanges to and from the machine. These fluids are called working fluids, and are made to undergo various processes in a proper sequence to effect the cooling process. The working fluid that absorbs the cooling load from the cold space is called the refrigerant. The processes undergone by the working fluids invariably constitute cycles, i.e., they return the working fluids to their initial states, permitting the use of the same fluid over and over again in the cooling process.

In addition to the cooling load energy input, an energy input is also needed to power the cooling machine, i.e. to enable the machine to operate and perform the cooling function for which it is designed. The rate of this energy input will be called the power input to the machine. The ratio of cooling load to power input is called the coefficient of performance (COP) of the machine. The power input need not be a heat input, but it can be. If it is not a heat input, it is a work input; hence the terms heat-powered cooling machine (HPCM) and work-powered cooling machine (WPCM) will be used to distinguish these two types of machines. This distinction is very important, because it necessarily takes more energy to power an HPCM than it does to power a WPCM. This difference is equivalent to the "unavailable" portion of heat, i.e., that portion which cannot be converted into another form (i.e., "work form") of energy.

Over a period of one cycle of the working fluids, the same amount of energy that enters a cooling machine must leave the machine. This energy removal is accomplished in a manner similar to that for the cooling load input, i.e., by generating, within the machine, a space having a temperature higher than that of an available "ambient heat sink", thereby effecting heat transfer from the machine to the heat sink. That portion of the cooling machine from which this energy leaves is called the heat-rejection space of the machine. With few if any exceptions, the medium used for the heat sink is either the ambient air or "cooling water". Cooling water can be supplied from an external source (such as a lake or a river) if one is convenient, or otherwise circulated through a cooling tower. With water cooling the temperature in the heat-rejection space, or heat-rejection temperature, can be lower than with air cooling, resulting in a larger cooling load for a given cooling machine.

2. Heat-Powered Cooling Machines

If a cooling machine is to be powered by solar energy, it must necessarily be an HPCM, since solar energy is inherently received and collected as heat. The use of solar cells to convert the solar energy directly into electricity to power a WPCM could be considered a possible exception, but even this system is a HPCM if the solar cells are considered to be part of the cooling machine.

In addition to the cooling-load-input temperature (T_L) and the heat-rejection temperature (T_R), the temperature at which the heat power input is received by the machine, or heat-power-input temperature (T_H), is of significance to the performance of the machine. The theoretical maximum value of COP for a heat-powered cooler is dependent upon these three temperatures according to the expression $T_L (T_H - T_R) / T_H (T_R - T_L)$, when the temperature values are expressed in absolute units (degrees Rankine or degrees Kelvin). Figure 33 is a plot of this maximum COP as a function of heat-power-input temperature for a cooling-load-input temperature of 45°F and heat-rejection temperatures of 80°F and 120°F. The 80°F and 120°F are considered representative of the heat-rejection temperatures maintainable with water and air cooling, respectively. The 45°F cooling-load-input temperature is representative of that required for an air-conditioning system. The maximum theoretical COP increases as either the heat-power-input temperature or the cooling-load-input temperature increases, and as the heat-rejection temperature decreases. The COP of real heat-powered cooling machines varies in the same general manner as the theoretical maximum, with actual values being in the range of 30% to 50% of the theoretical maximum.

FIGURE 33 THEORETICAL MAXIMUM COEFFICIENT OF PERFORMANCE OF A HEAT-POWERED COOLING MACHINE VS. HEAT-POWER-INPUT TEMPERATURE FOR A COOLING-LOAD-INPUT TEMPERATURE OF 45°F AND HEAT-REJECTION TEMPERATURES OF 80°F AND 120°F .

C. SOLAR-POWERED COOLING SYSTEMS

1. General

A solar-powered cooling system (SPCS) includes a cooling machine and the associated solar components, which are primarily a collector and one or more energy storage components.

2. Energy Transfer

The energy transfers in a solar-powered cooling system are shown schematically in Figure 34. The arrow widths are proportional to the respective heat transfer rates, and reflect a representative COP value of 0.6 for solar-power air-conditioning systems currently in use. The solar heat power input to the cooling machine can be either directly from the collector or indirectly from a high-temperature energy storage system. A low-temperature energy storage system can also be provided, which can be cooled when the cooling machine is providing excess capacity, then used later to provide cooling when solar energy is no longer available in sufficient quantities to operate the cooling machine during periods of insufficient insolation.

3. State-of-the-Art Descriptions

a. Collector

The cooling capacity of an SPCS is dependent upon the characteristics of the collector as well as those of the cooling machine. Whereas the COP of an HPCM increases with increasing heat-power-input temperature, the heat energy available from a solar collector decreases with increasing collector temperature. There is, therefore, for any given collector/HPCM combination, a collector temperature at which the cooling capacity is a maximum.

FIGURE 34 ENERGY FLOW DIAGRAM FOR A SOLAR-POWERED COOLING SYSTEM.

The state-of-the-art in collector technology provides only flat-plate collectors on a commercial scale. The heat output available from a representative flat-plate collector [1] can be expressed as $0.64S - 0.6[T_C - T_A]$, where S is the insolation flux in Btu/hr ft², T_C is the average temperature of the collector absorbing plate in °F (or °R) and T_A is the ambient temperature in °F (or °R). If this collector is used to power a perfect cooling machine (i.e., a cooling machine whose COP is the theoretical maximum) with perfect heat-input heat exchange (i.e., with $T_H = T_C$), the cooling load will vary as shown in Figure 3. In this figure, the heat-rejection temperature is assumed to be equal to the ambient air temperature. Curves are presented for ambient air/heat-rejection temperatures of 80°F and 120°F, and for insolation fluxes of 100 Btu/hr ft², 200 Btu/hr ft² and 300 Btu/hr ft² at each of these temperatures. All curves are for a cooling-load-input temperature of 45°F.

The dashed curves in Figure 35 are drawn through the maxima of the constant S curves for each value of ambient air/heat-rejection temperature. These show that the heat-power-input temperature must change with solar insolation flux if maximum cooling is to be achieved at all times.

Present HPCMs are capable of no more than 50% of the theoretical maximum cooling capacity. Also, as previously mentioned, the cooling capacity of a given HPCM, and hence a given SPCS, decreases with cooling-load-input temperature. These characteristics, along with the theoretical maximum performance capabilities shown in Figure 35 limit the practical application of SPCSs to air-conditioning with water cooling at the present time. Other

FIGURE 35 THEORETICAL MAXIMUM COOLING CAPACITY VS. HEAT-POWER-INPUT TEMPERATURE WITH INSOLATION FLUX AS A PARAMETER FOR A REPRESENTATIVE FLAT PLATE COLLECTOR.

applications can become practical only with improvements in HPCM and/or collector performance.

b. Absorption Cooling Machines

(1) Description of Operation

The type of HPCM to receive the widest application in SPCSs is the absorption cooling machine. A schematic diagram of the component arrangement in a basic absorption cycle is shown in Figure 36. Refrigerant enters the condenser as a vapor at the heat-rejection temperature and a relatively high pressure. In the condenser, heat is transferred from the refrigerant to the external heat sink (ambient air or cooling water), causing the refrigerant to condense, i.e. change to a liquid. The liquid refrigerant then flows through an expansion valve; as a result the pressure and temperature are reduced and the refrigerant partially vaporizes. The cold refrigerant enters the evaporator, where the cooling-load-input is received by the machine, causing the refrigerant to vaporize. This refrigerant vapor then enters the absorber, where it enters into solution in an absorbent which has a high affinity for the refrigerant in the low temperature low pressure state. The absorption process releases energy, and hence cooling is required to maintain the absorber temperature at the heat-rejection value. The absorbent-refrigerant liquid solution then flows through a pump, which raises its pressure and forces the flow into the generator. The heat-power input is received in the generator, where it heats the absorbent-refrigerant solution. The effect of this heating process is to force some of the refrigerant out of

FIGURE 36: SCHEMATIC OF BASIC ABSORPTION COOLING MACHINE

solution in the high pressure vapor state at which it can re-enter the condenser and repeat the cooling cycle. The remaining solution in the generator is returned to the absorber where it can re-absorb the refrigerant. Because of the pressure differences, the returning solution must expand before entering the absorber. This is shown schematically as an expansion valve, but is normally accomplished with spray nozzles located within the absorber. One additional component of the system is a liquid-liquid heat exchanger to provide heat flow from the solution leaving the generator to the solution entering the generator. This reduces the heat power input and the absorber cooling required, resulting in a higher COP for the machine.

(2) Performance Characteristics

Although many absorbent/refrigerant combinations have been investigated for use in absorption machines, the only ones to receive commercial application are lithium bromide/water and water/ammonia. COPs for the best performance possible from these combinations have been determined [2]. The results are reproduced in Figure 37 for the same conditions presented in Figure 34, permitting valid comparison with the theoretical maximum COP. This comparison shows that the highest possible COP attainable from an absorption cycle is about 65% to 70% of the theoretical maximum. The absorption machines currently in use operate with COPs of about 70% of those shown in Figure 37.

FIGURE 37: MAXIMUM ATTAINABLE COEFFICIENT OF PERFORMANCE VS. HEAT-POWER-INPUT TEMPERATURE FOR WATER/AMMONIA AND LITHIUM BROMIDE/WATER ABSORPTION COOLING CYCLES.

Figure 37 shows that, for a given pair of heat-rejection and cooling-load-input temperature values, there is a definite value of heat-power-input temperature below which the system will not function. As the heat-power-input temperature is increased from this lower limit, there is a sharp rise in COP to a value which remains relatively constant with further increases in heat-power-input temperature. In lithium bromide/water systems crystallization will occur if the heat-power-input temperature gets too high, so that this temperature is limited to a useful range of about 60°F. Ammonia/water systems can operate with heat-power-input temperatures up to about 400°F, at which point dissociation begins.

The amount of refrigerant that can be generated from each unit of absorbent/refrigerant solution increases quite rapidly with increasing heat-power-input temperature, hence the cooling capacity continues to rise even though the COP does not. In a typical case [3] the capacity doubles as the heat-power-input temperature increases from 175°F to 220°F. Since heat-power-input temperatures are generally below 200°F in an SPCS, solar-powered absorption systems must be quite a bit larger than equal capacity systems that are powered by higher temperature hot water or steam.

(3) Operating Systems

Lithium bromide/water is the most common working fluid combination to be used commercially in

absorption air-conditioners. At one time Arkla Industries manufactured a "residential size" lithium bromide/water absorption system which was designed for heat power input from a gas burner and a cooling load capacity of 3.5 tons.* Production of these units was ultimately discontinued as they proved to be noncompetitive with electrically-powered systems. With the recent growth of interest in solar energy applications, interest in these absorption cooling systems was revived. As a result, Arkla modified their seventeen remaining 3.5 ton gas-fired systems for hot water heat power input. Under this modification, the rating has been reduced to 3 tons at a hot-water-supply temperature of 210°F. All of these units have been placed in service in various demonstration buildings, and Arkla has begun manufacturing more of them. These are now on the market as the "Solaire model 501-WF" at a selling price of \$3000. Arkla is also planning the production of a "Solaire model WF-300", which is designed to deliver a cooling capacity of 25 tons when supplied with water at a rate of 60 gpm and a temperature of 225°F.

Larger size lithium bromide/water absorption cooling systems, having cooling capacities of 50 tons or more and using hot water or (low grade) steam as the heat power source, have been commercially available for some time. U. S. manufacturers include the Trane Co., Carrier Corp., York Division and Arkla Industries. These systems are generally designed for generator (i.e., heat-power-input) temperatures in the range of 220°F to 240°F. Current solar-powered installations of these systems include a York model ES absorption chiller already installed and operating

*The ton is a unit of energy transfer rate, or power, and is equal to 12,000 Btu/hr. It is the standard (U.S.) unit of cooling capacity, having its historical roots as the nominal rate of heat transfer needed to melt one ton (2,000 lb) of ice in one day.

in the Timonium elementary school in Timonium, Maryland, and an Arkla model WF1200 planned for installation in the George A. Towns elementary school in Atlanta, Georgia. The York unit is designed to provide 135 tons of cooling when supplied with hot water at 240°F, and is re-rated to provide 50 tons of cooling when supplied with solar heated water at 187°F. The Arkla unit is rated at 100 tons for a 220°F supply temperature, and 50 tons for a 175°F supply temperature.

c. Rankine Cycle - Vapor Compression Cooling Machines

(1) Description of Operation

In Rankine cycle - vapor compression (RCVC) cooling machines, the heat energy input is used to generate mechanical power, which in turn is used to drive a WPCM. The method used to generate the mechanical power is known as the Rankine cycle; it is the same cycle used by the baseline steam generating units of central power plants. The method used to provide cooling is known as the vapor - compression cycle; it is the same cycle used in "conventional" electrically powered freezers, refrigerators and air-conditioners; it is also essentially a reverse Rankine cycle.

A schematic of the component arrangement in an RCVC cooling machine is shown in Figure 38. The function of the condenser, expansion valve and evaporator are the same as in the absorption cycle. The compressor replaces the absorber/generator and serves to raise the temperature and pressure of the refrigerant to the required condenser inlet values. In the Rankine power cycle, the heat power input is used to evaporate the working fluid at a relatively high pressure and temperature. The working fluid then flows through an expander (turbine, rotary expander

FIGURE 38: SCHEMATIC OF BASIC RANKINE CYCLE-VAPOR COMPRESSION COOLING MACHINE.

or reciprocating expander) in which the mechanical energy is produced. The working fluid leaves the expander in the vapor state at a low pressure and temperature. It is then condensed to a liquid, after which it flows through a pump which raises its pressure to the value required by the boiler, so that the cycle can be repeated. In some instances, depending upon the type of fluid used and the operating state points of the cycle, a heat exchanger, or regenerator, is included to heat the liquid leaving the pump with the vapor leaving the expander, thereby raising the efficiency of the cycle. This can be done only when the working fluid is a "drying fluid", and leaves the expander as a superheated vapor [4]. Comparable efficiencies can be achieved with wetting fluids without a regenerator, hence their use has the advantage of fewer component requirements in the cycle. It is also possible in an RCVC system to combine both condensing operations into one condenser by using the same working fluid in both cycles, thereby simplifying their manufacture.

(2) Working Fluids

Organic fluids have been found to be the most suitable working fluids for operation at the relatively low boiler temperatures and power output rates of solar-powered Rankine cycles. Water is not a suitable working fluid because its low vapor density necessitates the use of multiple stage, or very large single stage, expanders.

(3) Comparison with Absorption Systems

RCVC systems have several apparent advantages over absorption systems:

1. They can be readily used as a source of mechanical (or electrical) power when the solar energy supply rate is higher than necessary to provide heating or cooling.
2. They are readily adaptable to the use of electricity as the auxiliary energy input by coupling an electric motor directly to the compressor. In this manner the cooling capacity per unit of electrical energy input is greatly increased over what it would be if the electric energy were converted to a heat input in the Rankine cycle boiler.
3. They have no inherent limits on the range of heat-power-input temperature at which they will operate. Furthermore, their COP will continue to increase with an increase in heat-power-input temperature.
4. They possess the potential for greater improvements in performance (i.e., higher COP) with further design refinements.

(4) Operating Systems

An RCVC cooling system has been built and tested by Barber-Nichols Engineering Company [5]. The measured performance of this unit is summarized in Figure 7. The design cooling capacity was 3 tons, to be achieved at a heat-power-input temperature of 215°F. As this temperature decreases to 175°F, the capacity drops to 1 ton. Figure 39 shows a performance comparable to that from an absorption system, especially when it is noted that the heat-rejection temperature for the curves of Figure 7 is 5°F higher than that for the absorption system curves in Figure 5. Reducing the heat-rejection temperature to 80°F for the RCVC system would improve the performance by approximately 20%. This system is currently being operated by solar energy from a flat-plate collector in the Honeywell Mobile Van [6].

FIGURE 39: PERFORMANCE OF THREE TON BARBER-NICHOLS RANKINE CYCLE-VAPOR COMPRESSION AIR-CONDITIONER.

Three ton capacity RCVC systems are also under development at United Technology Research Laboratory [7], General Electric Co. [8], and Battelle Memorial Institute [9]. The first two units are in the breadboard testing stage, and the third has completed the design stage. The United Technology and Battelle units employ a single condensor, made possible by utilizing the same working fluid in both cycles.

Of the four RCVC systems discussed, only the Battelle system is designed specifically for the solar power application; the other units are assembled from existing components whose main uses are in other applications, with a few exceptions. The Battelle unit offers much promise; its main features are: i) a low friction pivoting-tip vane rotary expander and compressor, ii) a hermetically sealed expander-compressor unit on a single shaft, iii) a provision for using the vapor compression cycle as a heat pump during the heating season.

In addition to these 3 ton systems, Barber-Nichols Engineering Co. has designed a 77 ton RCVC system for the Los Alamos Scientific Lab [10]. This will use an existing York vapor-compression cooler having a design cooling capacity of 100 tons, which has been de-rated to 77 tons at the rated speed of the powering Rankine cycle. The working fluid is Freon 11 for both loops.

The 3 ton Barber-Nichols system is reported to have a manufacturing cost of \$2,500 to \$3,000, with a projected selling price of \$1,100 in production quantities [5]. This compares favorably with the cost of the Arkla 501WF absorption system reported earlier. Also, the economics of larger systems becomes much more favorable; it is estimated that increasing the cooling capacity of an RCVC system from 3 tons to 100 tons, a 33-fold increase, would result in

only a 10-fold increase in price [5].

d. Vuilleumier Cycle Cooling Machine

The Vuilleumier cycle cooling system utilizes a gas as the working fluid, and consists essentially of a Stirling power cycle driving a Stirling refrigeration cycle. The Stirling cycle is a gas cycle; the working fluid never changes state from vapor to liquid as it does in the Rankine cycle. Vuilleumier cycle systems have proven successful as cryogenic refrigerators. Studies are underway to develop a solar powered Vuilleumier air-conditioner having a cooling capacity of 3 tons [14]. The reported advantages of this type of HPCM are i) it can be operated with no mechanical energy input, ii) it is a relatively simple device with only two moving parts (displacers) and close tolerances are not required, iii) the engine-refrigerator combination can be hermetically sealed, eliminating sealing problems and iv) the predicted COP is high (approaching 1 when solar powered). The major deterrent to the rapid development of the system is the need for a redesign of the regenerators, because of the close-to-ambient temperatures encountered in the solar powered air-conditioning application.

e. Adsorption Air-Conditioning Machines

(1) Description of Operation

Another type of heat-powered air-conditioning system that is receiving some attention for solar power is the adsorption, or desiccant, system. A schematic of this type of system is shown in Figure 8. This system employs moisture exchange and heat exchange between the air entering and leaving the cooled space in a combination that results in cooled and de-humidified entering air. The moisture exchange is accomplished through the use of a desiccant

Note: Numbers in Parentheses are typical dry bulb over wet bulb temperatures, °F

FIGURE 40: SCHEMATIC OF DESICCANT TYPE HEAT-POWERED AIR-CONDITIONING SYSTEM.

wheel. The heat power input is used to regenerate the desiccant by heating it, which drives out the moisture adsorbed from the incoming or recirculating air stream.

(2) Operating Systems

A prototype 3 ton model using a molecular sieve as the desiccant is being developed by the Institute of Gas Technology and is currently being tested in Los Angeles [11]. This type of desiccant requires higher regenerating temperatures than can be furnished by a flat-plate solar collector, hence the solar energy is used only as a supplement to natural gas heating. A similar system using silica gel as the desiccant is reported under development in Australia [12]. Silica gel can be regenerated at a lower temperature than molecular sieves, hence the Australian system is designed to operate completely on solar energy.

f. Vapor-Jet Cooling System

A schematic of a vapor-jet air-conditioning system is shown in Figure 41. In this system, compression of the refrigerant is accomplished with a vapor-jet-ejector pump; a portion of the condensed refrigerant forms the working fluid for the ejector. A preliminary study [13] reports that a vapor-jet system using n-butane as the working fluid is nearly as efficient as an RCVC system, and can operate at lower heat-power-input temperatures than an absorption system. The advantage of this system is its simplicity, hence low initial cost and low maintenance. No solar-powered vapor-jet systems have been built and operated.

FIGURE 41: SCHEMATIC OF VAPOR-JET-COOLING SYSTEM.

g. Intermittent Systems

The absorption system cycle can be operated intermittently, permitting the combining of the absorber and generator and of the evaporator and condensor [15]. Even though storage must be provided for condensed refrigerant in this scheme, the net result is a system of simpler hardware and manufacture.

A variation of this is under study [16], in which a molecular sieve (solid adsorbent) is used in the collector, and water is used as the refrigerant. During the day, the collector serves as a generator, using the insolation to drive water out of the adsorbent, from which it flows through a condenser, expansion valve and evaporator as in the "conventional" refrigeration cycle. The vapor produced in the evaporator is adsorbed in a container of silica gel, where it is stored until nightfall. When the collector cools at night, a valve is opened to permit the refrigerant (water) to flow from the silica gel storage to the molecular sieve in the collector. This flow occurs naturally because of the molecular sieve's greater affinity for water under the prevailing temperature and pressure conditions. The heat of adsorption is transferred through the collector by convection and radiation to the cooler air and sky. The reported advantages of the system are that it is capable of satisfactory operation with air cooling, and that it is economically competitive with other solar-powered cooling systems, having projected costs of \$1,000 to \$2,000 per ton of cooling capacity.

4. Near Term Research and Development Potential

a. Design of Present Systems

The solar-powered absorption and RCVC cooling systems currently in

use are adaptations of systems that were originally designed for other types of energy input. The absorption systems were designed for heat input from hot water, steam, or the burning of a fuel. The RCVC systems use heat engines that were originally conceived to be powered from the exhaust gases of internal combustion engines, coupled to vapor compression refrigeration systems that are either a standard system designed for electrical power input or an assembly of existing components that were originally designed for other applications. It would seem probable, therefore, that improvements in the performance of solar-powered cooling systems could be made if the unique characteristics of solar energy input were given more consideration in their design. These characteristics are primarily a relatively low temperature and a very unsteady supply rate.

b. **Variable State Design**

Typical engineering system designs have been oriented around conditions that remain time-invariant, or at "steady state", for example the Barber-Nichols 3 ton RCVC system was designed for hot water supplied at a rate of 11 gpm and a temperature of 215°F. In view of the highly variable solar energy supply rate, it is suggested that the long-term performance of a SPCS could be improved if the design concept were altered from one of steady state performance to one of "variable state" performance. The final design should be selected on the basis of that one which gives the best "time integrated" performance over the range of input conditions (i.e., solar energy supply rates) that are likely to be encountered.

c. Maximization of Output

As noted in Figure 35 the collector temperature at which an SPCS gives the best performance is quite highly dependent upon the insolation rate to the collector. The collector temperature is controlled by the flow rate of the collecting fluid through the collector; the faster the flow, the lower the collector temperature. An SPCS could be made to constantly seek its level of maximum cooling capacity by employing some sort of feedback control system which senses the output and regulates the collector fluid flow rate until the output is a maximum. The development of such a control system is suggested as a contribution to the improved operation of SPCSs. It is also suggested that design considerations should incorporate the collector performance in a manner similar to that used to calculate the curves of Figure 3, except that the characteristics of the real HPCM should be used in place of those of a thermodynamically perfect one.

d. Storage Systems

The effects of different types of storage should also be included in design and performance studies of SPCSs. Coupled with this would probably be some allowable variation in cooling-load-input temperature to accomodate low-temperature storage. The types of storage considered should include high-temperature storage, low-temperature storage, and storage of condensed refrigerant at the heat-rejection temperature. The latter type should involve essentially no heat losses because it is at ambient temperature, and might be most appropriate for absorption systems where water is used as the refrigerant.

e. Working Fluids

The working fluids used in present SPCSs were selected for use under the original design conditions of the systems, particularly in the case of absorption systems. Lithium bromide/water has been selected by the industry as the best all-around absorber/refrigerant combination for heat-input temperatures of 220°F or higher, whereas the practical temperature range of state-of-the-art solar collectors is below 200°F most of the time. It would therefore seem appropriate for the industry to reexamine the characteristics of other possible absorbent/refrigerant pairs with the view of determining the best all-around combination for use with heat-input temperatures in this lower range. It seems quite probable that some pair other than lithium bromide/water might satisfy this criterion.

f. Rankine Cycle Modification

The efficiency of a Rankine cycle is dependent not only upon the heat-input and heat-rejection temperatures, but also on the working fluid and its operating state points. As a practical matter, these cycles are usually designed so that the working fluid is in the vapor state during the expansion process in order to avoid erosion of the internal surfaces of the expander by impinging liquid droplets. However, the efficiency of a Rankine cycle can be increased if the working fluid is a liquid-vapor mixture during expansion. Therefore, efforts toward developing expanders that can tolerate two-phase working fluids might well contribute toward the development of more efficient Rankine cycle engines.

g. Gas Cycle Systems

The COP of a cooling machine consisting of a Stirling power cycle and a Stirling refrigeration cycle is equal to the theoretical maximum, whereas the COPs of absorption and RCVC systems are inherently below the theoretical maximum. The Vuilleumier cycle is a combination of the Stirling power and refrigeration cycles operating as interacting open cycles [14]. Hence, the Vuilleumier cycle cooling machine has a greater performance potential than any of the cooling machines being currently used in SPCSs. For this reason, and also because these types of machines have been successfully designed and operated for other applications, it would seem highly desirable to continue their development for the solar powered air-conditioning application. Such systems could also prove to be of simpler construction than either absorption or RCVC systems. The same conclusion also applies to the closed cycle Stirling engine coupled to a closed cycle Stirling refrigerator.

h. Other Systems

The other types of SPCSs reported on in sections C.3.e.-g. each have their own merits, as mentioned, and should receive further development work.

i. Collectors

It is probable that the greatest potential for improvement in the performance of SPCSs lies in the solar collectors rather than the cooling systems. Figure 33 shows that at a cooling-load-input temperature of 45°F and a heat rejection temperature of 80°F, the theoretical maximum COP doubles as the heat-input temperature

increases from 180°F to 325°F. Thus, a collector that is capable of supplying the same amount of energy at 325°F as another collector supplies at 180°F will double the cooling capacity of an SPCS if the COP of the cooling machine remains at some fixed fraction of the theoretical maximum. On the other hand, trying to design a cooling machine that can provide twice the capacity of a state-of-the-art absorption or RCVC system at a heat-power-input temperature of 180°F is essentially impossible, as this performance would be approaching the theoretical maximum.

To demonstrate that the above cited collector performance is not impossible to achieve, curves analogous to those presented in Figure 35 have been calculated for the Owens-Illinois evacuated tube collector, which has a reported heat output curve given by $0.58S - 0.19 (T_C - T_A)$. The results are shown in Figure 42 and show that the theoretical maximum cooling capacity is a maximum at a collector temperature of 422°F when the insolation flux is 300 Btu/hr ft² and the heat-rejection temperature is 80°F.

There would be little advantage in using such a high performance collector with the absorption machines currently employed in SPCSs, because of the upper limit of heat-power-input temperature imposed by crystallization of the lithium bromide. However, there are "double effect" absorption cooling machines available, which accomodate these higher heat-power-input temperatures, with a corresponding increase in COP [17].

FIGURE 42: THEORETICAL MAXIMUM COOLING VS. HEAT-POWER-INPUT TEMPERATURE WITH INSULATION FLUX AS A PARAMETER FOR AN EVACUATED TUBE COLLECTOR.

j. Economic Gains from Improved Component Design

It appears highly probable that the cost per unit of cooling from SPCSs can be reduced by improving the performance of each of the components present in the system. However, design and development should be concentrated on those components which offer the most "economic potential" for improvement. It is suggested that the most rational basis for selecting these components rests in a "Second Law" availability* based analysis, rather than on the more usual "First Law" energy based analysis.

The availability of a system is the maximum amount of energy that can be obtained by work transfer from the system and reference atmosphere** Thus availability is fundamentally concerned only with work transfers of energy, whereas energy is concerned with both work and heat transfers.

One way to state the Second Law of Thermodynamics is "Availability is continuously destroyed in every real process". Thus, availability, unlike energy, is inherently non-conserved. In reversible processes, which represent the best thermodynamically possible, availability is conserved - the availability destruction rate is zero. Hence, a measure of how "ideal", or how close a process is to being reversible, is the amount (or rate) of availability destruction. Thus, instead of thinking in terms of "energy destruction" (which has to be zero), it is more meaningful to think in terms of availability destruction (the less the better).

*Availability is also called "essergy", "exergy", "available energy". It might be considered the complement to entropy, to the extent that entropy represents the "unavailable energy" in a system.

**The reference atmosphere is a theoretical system that is large in extent compared to the system in question, large enough so that there is no change in its intensive properties (p , T , ρ ...) when it interacts with any other system.

The availability transferred with a heat transfer of energy is equal to the theoretical maximum amount of work that could be produced by delivering the heat energy to a power cycle that has a heat-rejection temperature equal to the temperature of the reference atmosphere. This permits the direct comparison of heat and work transfers of availability. For example, as noted earlier, the energy input to an HPCM must be higher than that to a WPCM if they have equal cooling capacities and equal "losses". However, the availability inputs would be the same, therefore availability inputs form a valid basis of comparison. Furthermore, the losses can be directly quantified as availability destruction rates within the system. In a thermodynamically perfect, i.e., reversible, cooling machine, there is no availability transferred in the heat-rejection process, and availability is transferred from the machine to the cold space (the opposite direction of the energy transfer) in the cooling load input process. Furthermore, the availability transferred to the cold space must equal the availability input to the cooling machine, since there is no destruction of availability in a reversible process.

The cold space has availability because it could be used as a "heat sink" for a power cycle with heat input from the reference atmosphere (at a higher temperature than that of the cold space). The work energy delivered by this cycle originates in the reference atmosphere, whereas the availability resides in the cold space. The cold space loses availability as its temperature increases from the heat input of the cycle. This example emphasizes that availability and energy are separate and independent properties; the availability contained in a system is not a part of the energy contained in the system.

"Availability economics" assigns costs to the availability destruction rates and the availability delivery rates of each of the components in a thermodynamic system [18]. The component with the best "potential for improvement" is that for which a unit decrease in availability destruction could be achieved for the least cost.

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APPENDIX A

ITC SPECIFICATION SCS-1001-010575

SOLAR CONTROL SYSTEM

REVISED 23 JULY 1975

- 1.0 Scope - This specification is intended to provide the performance requirements for the design, fabrication, assembly, installation and testing of an electronic controller which is capable of performing the various functions necessary in the operation of the Fauquier High School Solar Heating System. The requirements are described below in both prose and tabular form, either one being sufficient for constructing the other.
- 2.0 Background
- 2.1 The system consists of a solar collector array, thermal storage and heating capacity for five detached classrooms at the Fauquier High School in Warrenton, Virginia. Thermal storage for the system is accomplished via two 5,500 gallon tanks of water. Operation of the system is keyed to the relative temperatures of the collector, storage tanks and classrooms. Continuous monitoring of these temperature relationships permits several modes of operation in which the water is selectively routed through the system by activating various pumps and valves.
- 2.2 Recent modification of plumbing configuration to allow for greater flexibility of operation requires that a new controller be procured, since the old controller is incompatible with the new system. Also, it is desired that the new controller demonstrate a higher degree of reliability than the old system. Accordingly, the primary design emphasis should be placed on operational performance, reliability and maintainability, in that order.

3.0 Description

- 3.1 A detailed description and schematic diagram of the old system are contained in ITC Report No. 090974, dated 4 December 1974, Solar Energy School Heating Augmentation Experiment, Design Construction and Initial Operation. The modified plumbing configuration and temperature control points are schematically represented in Figure 1.
- 3.2 The controller will be capable of monitoring a number of input temperatures and differences in temperature from temperature control points. Based on these inputs, the controller will logically determine which combination of electromechanical devices (pumps, valves, etc.) should be activated. The system will be capable of fully automatic operation unless overridden by manual control. The manual control mode will provide the means for individually activating any valve or pump via the System Status Display Board which contains status lamps and toggle switches for each controllable device, arranged in a manner that depicts the system configuration schematically.

4.0 System Requirements.

- 4.1 General -The general requirements for daytime and nighttime controller operation are summarized in this section. For this section "daytime" and "nighttime" are defined as follows: "daytime" is defined as the period of time beginning one hour before the normal school starting time and ending at school closing time during any school day. All other times are considered to be "nighttime", including weekends and holidays.
- 4.2 Operation.
- 4.2.1 The general principles governing the decisional logic are as follows:
- 4.2.1.1 For most efficient collection, the cooler tank is circulated to the collector;
- 4.2.1.2 For quickest space heating during "daytime" operation when the room is occupied, the warmer tank circulates to space;

- 4.2.1.3 During "nighttime" when the room is not occupied the lower grade heat in the lower temperature tank is consumed which prepares that tank for more efficient collection during "daytime".
- 4.2.1.4 These general principles apply as long as at least one tank is above a threshold temperature for effective space heating, which threshold is higher for the "daytime" than for the "nighttime" mode. When both tanks are at temperatures less than this threshold, the warmer tank is circulated to the collector on demand, and only then is a space demand responded to. These thresholds are set at 21°C (70°F) and 38°C (100°F), for "night" and "day", respectively.
- 4.2.2 Three simultaneous conditions trigger collector demand for water circulation.
- 4.2.2.1 No water is circulated to the collector from a tank with temperature above an upper limit at which it is no longer necessary to collect heat. Thus, to respond to collector demand at least one tank must have temperature below this limit, which is set at 82°C (180°F).
- 4.2.2.2 Since there is little benefit in circulating cooler water through a warmer collector when that temperature difference is due only to ambient temperature (for instance on a warm day with low insolation), it is necessary to discriminate for insolation heating of the collector. This threshold insolation query can be implemented by requiring collector temperature to be greater than ambient, by a certain amount. (The exact temperature difference in deciding to initiate flow would be set optimally by comparison, through the energies involved, to the temperature difference between tank outlet and return). In view of the small increase in heat collection to be gained, this temperature difference $\Delta 1$ is set at 22°C (40°F).
- 4.2.2.3 The third simultaneous test for initiating flow is the relative temperature difference $\Delta 2$ between the collector and the supply tank, which is the cooler tank except where both tanks are below threshold. For circula-

tion, the collector must be the hotter by an amount variable between 6°C (10°F) and 28°C (50°F) in increments of 6°C (10°F).

4.2.3 After collector flow is initiated by these conditions, it is halted when:

4.2.3.1 Both tank temperatures are above 82°C (180°F) or,

4.2.3.2 The temperature increase between the tank outlet and return approaches some small value of marginal benefit. This is set at $\Delta 3 = 1.0^{\circ}\text{C}$ (2°F) to 5°C (9°F) in adjustable increments of 1.0°C (2°F). A 0.6°C (1°F) rise with system flow corresponds to storing at a rate of approximately 4 Btu/sec or 4 kw, and this flow is driven by 1.5 hp or 1 kW. The total tradeoff might occur below 0.3°C (0.5°F). Thus, a control range of 0.1 to 1°C in 0.1°C steps would appear more proper, but circuitry and cost constraints obviate such tight limits. The extra heat that might be collected is in any case a small part of the whole.

4.2.4 Three characteristic time intervals arise in determining desired response to changes in the operating conditions envisioned above.

4.2.4.1 When the space thermostat is activated and demands heat, it shall be supplied immediately. ($\Delta\tau_1 = 0^{\text{m}}$).

4.2.4.2 Collector demand shall be responded to within a longer time interval on determination of some stability of the condition. For the system to accommodate dynamic operation characteristics such as intermittent cloud cover and temperature equilibration of piping with circulating water, it is required that the collector not be involved in a mode change until all input parameters to collector demand have remained static for a specified interval of time. This time interval shall be easily changed in 5 minute increments from an interval of 5 minutes to an interval of 60 minutes. If any inconsistency arises in four evenly spaced samples in the interval, then the period for change verification shall be restarted as of the time of that inconsistency. ($\Delta\tau_2 = 5 \rightarrow 60^{\text{m}}$).

4.2.4.3 The third characteristic time involves the decision to switch tanks when supplying either space or collector because of change of their temperatures. Since their rate of temperature change is typically 10°C per 10 hours, and because the difference in efficiency of space heating or of heat collection is insignificant over a water temperature interval of a few degrees, it is not necessary to change often between tanks. Because tank temperature changes slowly and because temperatures are rounded to 10°C for logic decisions, there is an intrinsic delay of perhaps one hour in switching tanks. Thus, an additional delay of 0 to 3 hours in 1 hour increments shall be incorporated, ($\Delta\tau_3 = 0 \rightarrow 3^{\text{h}}$).

4.2.5 An important additional requirement is that when the collector demand ceases, the circulating water shall drain back into the tank from which it came. This will automatically happen if the logic of Table I is implemented, including the "no change" entries, except for V3 and V4 during transitions from states A to S and B to P respectively. As indicated in the table, these require a temporary value of 1 during the period required for drainage. This transient valve position does not interfere with continuing response to space demand, but it is necessary that the drain period be shorter than the verification period $\Delta\tau_2$ for collector turn-on given in section 4.2.4.2. This drainage period $\Delta\tau_4$ shall range from 5 to 9 minutes in one minute increments, ($\Delta\tau_4 = 5 \rightarrow 9^{\text{m}}$). The temperature limits and time delays given in the above paragraphs are summarized in Table III.

4.2.6 Provision shall be incorporated for manually bypassing either of the storage tanks by selector switch so that either tank may be in effect "removed from the system". The method of system operation in this single tank mode shall be essentially the same as the requirements outlined above.

4.2.7 It is desired that the controller be designed to minimize any non-essential electromechanical cycling and to also minimize power requirements.

- 4.2.8 These general statements of operating principles are realized by the logic circuit properties indicated in Tables I and II. The temperature differences $\Delta 1$ and $\Delta 2$ of Table II have been set at above 22°C (40°F) and at 6°C (10°F) to 28°C (50°F), respectively.
- 4.3 Electrical
- 4.3.1 In order to provide the maximum degree of reliability, the controller will be of all solid-state design using plug-in components or modules wherever possible.
- 4.3.2 Use of electromechanical components such as relays is discouraged. Specific test points and self-test features shall be provided to the greatest extent practicable to facilitate testing and maintenance. Switching of AC voltages shall be done using solid-state relays employing synchronous/zero voltage turn-on that are capable of handling AC loads presented by the existing pumps and valves. It should be noted that once voltage has been applied to a motorized valve, this voltage must be maintained (latched) until the valve has completed its cycle.
- 4.3.3 It is desired that all numerical displays and indicators, as provided, be solid-state devices, e.g. light-emitting diodes (LED's). Provision shall be made for simultaneously illuminating all indicators as an independent "lamp test" mode.
- 4.3.4 In the event of a power failure, the system shall restore itself automatically to the previous operating mode when power is re-applied. The controller will not require any specific sequence of events in applying/removing power from the system.
- 4.3.5 The proposed controller shall utilize a time-of-day clock. It shall be provided with a numerical readout device, and be capable of being easily reset. Battery operation shall also be provided to power the clock during power outages.

- 4.3.6 To ensure that circulating water does not freeze in the collector, due to any unexpected condition an independent temperature sensor and thermostatic control will interrupt power to P3 and to VS10 whenever collector temperature drops below 40°F.
- 4.4 Mechanical
- 4.4.1 The controller will be constructed to provide ease of maintenance, accessibility and replacement of all modules and parts. All controls will be marked and located in such manner that the control position can be readily identified. It is desired that the controller be mounted within a standard, wall-mounted type, NEMA-12 enclosure. All exposed metal parts that are subject to rusting will be finished with some type of protective coating.
- 4.5 Display Board Interface (DELETED)
- 4.5.1 All valve and pump control signal outputs from the controller shall be compatible with the existing display board, which will be revised to reflect the modified plumbing schematic of Figure 1.
- DELETED
- 4.5.2 In addition, in the "Manual" mode of system operation, all valves and pumps shall be individually controllable from the existing switches and indicator lamps in the modified display board.
- DELETED
- 4.6 Environmental
- 4.6.1 The controller will be required to operate satisfactorily over an ambient temperature range of 0°F. to 160°F at level of relative humidity up to 95%.
- 4.7 Documentation
- 4.7.1 The supplier will furnish, as a minimum, a system manual which includes detailed operating instructions for the controller, a description of the various modes of operation, maintenance procedures, logic diagrams, wiring diagrams, and applicable schematics. Also to be included in the manual are a list of materials and part location diagrams. The manual will be delivered at the time of equipment installation. Any changes

to the equipment made necessary as a result of the installation will be furnished by the supplier in the form of revisions of the manual. A complete set of drawings defining the equipment shall be furnished by the seller. The drawings shall meet as a minimum requirement the following industrial standards:

USAS Y 32.16-1968	Reference designations for electric and electronic parts and equipment.
USAS Y14.15-1966	Electrical and electronic diagrams.
USAS Y32.2-1967	Electrical and electronic diagrams, graphic symbols for
USAS Y32.14-1962	Logic diagrams, graphic symbols for

4.8 Installation and Testing

- 4.8.1 The supplier will furnish, no later than 30 days prior to starting the installation, a proposed installation schedule and a list of equipment and/or services required of ITC prior to or during the installation period. The supplier will also furnish at this time a proposed plan for demonstrating compliance with this Definition of Requirements.

5.0 Additional Conditions

- 5.1 To the extent that the detailed system description contained in Tables I and II of this specification fails to be compatible with the operating principles stated in prose form, or in the event that the detailed description fails to be compatible with the constraints of system hardware, resulting in system malfunction, the contractor assumes full responsibility for informing ITC of such inconsistencies prior to contract agreement, and of his intent to modify circuitry to realize the system generally described.

The technical considerations stated in the Roburn Corporation letter to InterTechnology Corporation dated July 17, 1975 constitute a part of this specification. This letter is furnished as an appendix.

6.0 Acceptance

- 6.1 Prior to starting equipment fabrication, the supplier will furnish to ITC the details of his design approach, which may include such things as a sketch of the proposed control panel layout, logic diagrams and flow diagrams, for final approval.

Table II, Definition of function c, the collector demand variable

Case Number	Variable and Function	Inputs			
		if $T1 = 0$ and $T2 > 82^{\circ}\text{C}$	if $TC - TA > \Delta 1$ or if $P3 = 1$	if $TC - \text{cool tank} > \Delta 2$ or if $P3 = 1$	if $TR - TQ > \Delta 3$
0		0	0	0	0
1		0	0	0	1
2		0	0	1	0
3		0	0	1	1
4		0	1	0	0
5		0	1	0	1
6		0	1	1	0
7		0	1	1	1
8		1	0	0	0
9		1	0	0	1
10		1	0	1	0
11		1	0	1	1
12		1	1	0	0
13		1	1	0	1
14		1	1	1	0
15		1	1	1	1

TABLE III

<u>Spec Paragraph</u>	<u>Symbol</u>	<u>Condition</u>
<u>To Start Collector</u>		
4.2.2.1	T1	Tank 1 Less than 82°C (180°F)
4.2.2.1	T2	Tank 2 Less than 82°C (180°F)
4.2.2.2	$\Delta 1$	$(TC-TA)^* > 22^\circ C (40^\circ F)$
4.2.2.3	$\Delta 2$	One tank > threshold $(TC-T_{\text{cooler tank}}) > \begin{cases} 6^\circ C \rightarrow 30^\circ C \\ 10^\circ F \rightarrow 54^\circ F \end{cases}$ in 60 steps
		Both tanks < threshold $(TC-T_{\text{warmer tank}}) > \begin{cases} 10^\circ F \rightarrow 54^\circ F \\ 11^\circ \text{ steps} \end{cases}$
<u>To Stop Collector</u>		
4.2.3.1	T1	Tank 1 Greater than 82°C (180°F)
4.2.3.1	and T2	Tank 2 Greater than 82°C (180°F)
4.2.3.2	$\Delta 3$	$(TR-T_0) \geq 0.6^\circ C \rightarrow 3^\circ C$ in 0.60 steps $(1.0^\circ F \rightarrow 5^\circ F)$ in 1.00 steps
<u>Mode Criteria</u>		
4.2.1.4	T1	Tank 1 Greater than 21°C "night" (70°F)
4.2.1.4	T1	Tank 1 Greater than 38°C "day" (100°F)
4.2.1.4	T2	Tank 2 Greater than 21°C "night" (70°F)
4.2.1.4	T2	Tank 2 Greater than 38°C "day" (100°F)
<u>Time Intervals</u>		
4.2.4.1	$\Delta \tau 1$	Space demand = 0 ^m
4.2.4.2	$\Delta \tau 2$	Collector Status change 5 ^m → 60 ^m in 5 ^m steps.
4.2.4.3	$\Delta \tau 3$	Tank Switching 0 ^h → 3 ^h in 1 ^h steps.
4.2.5	$\Delta \tau 4$	Collector drain hold period 5 ^m → 9 ^m in 1 ^m steps.

*TC Collector Dry-Plate Temperature
 TA Ambient Temperature
 TR Return Temperature (to tanks)
 T0 Outlet Temperature (from tanks)

Figure 1
Plumbing Schematic and Temperature Control Points

ROBLURN CORPORATION

3432 BROOKWOOD DRIVE

FAIRFAX, VIRGINIA 22030

July 17, 1975

Mr. Norris L. Beard
Director of Marketing
InterTechnology Corporation

Sir:

In response to your request for quotation, the ammended specification (draft) of 10 July 1975 and following our discussions with your people, Roburn has arrived at the following quotation.

Specifications:

Temperature Measurement:

0 to 99°C, $\pm 1.5^\circ\text{C}$ error band max for linearity plus tolerance

System clock:

Real time, battery backup, ± 25 parts per million. Clock provides for setting day/night division plus holiday/normal divisions

Temperature Delta:

The following delta sets are provided for system purposes

T1 30°C in 6°C increments

T2 5°C in 1°C steps

T3 22°C fixed

Temperature Set Points:

The following temperature set points are provided for in system operation

T1 = 4.0°C

T2 = 21.0°C

T3 = 38.0°C

T4 = 82.0°C

Time Deltas:

The following time increments are provided for in system operation

- TM-1 0 to 3 hours in 1 hour increments
- TM-2 5 to 60 minutes in 5 minute increments
- TM-3 5 to 9 minutes in 1 minute increments

Temperature Measurement Points:

Temperature shall be measured at the following system points

Collector-surface mounted unit to one cell (under surface)

Return-return line unit mounted at entrance to tank area in common line

Tank-tank 1 and tank 2 mounted internal to each tank on the tank wall 1/3 down from the top

System Logic:

The system shall be configured for each mode of operation according to table 1 and table 11 of the specification. The system programming shall be according to these tables for automatic operation only. If a particular tank is inhibited for a given reason (switch actuated) it's commands shall be diverted to the remaining tank.

Status, Display, Manual Panel:

Internal to the controller is mounted a status, display, manual panel to provide for the following:

- (1) Indicating lamps to show which pumps and valves have been opened or closed
- (2) A digital display to indicate time or temperature as selected:
 - a. Temperature to nearest degree
 - b. Time to the nearest 6 minutes
 - c. Manual control switches to manually energize each valve or pumps and a self test switch to allow for testing of certain critical units in the system

Valve and Pump Controllers:

All valve and pump controllers shall be solid state units derated by a nominal of 50%.

Construction:

The unit shall be all solid state where feasible, except for certain manual switch functions. The digital unit shall be primarily of wire wrap construction using sockets for plug in components. The analog portion shall be p.c. board construction and encapsulated. All long lines shall employ EIA-RS-232C type transmitters and receivers for noise immunity.

All housings shall be of NEMA 3 and NEMA 12 type units and all exposed metal parts shall be appropriately finished for fungus and humidity. All wiring external to the cases shall use the appropriately rated cables per NFPA, volume 5, where required.

Acceptance Test:

Acceptance test shall be performed on ITC/Fauquire High School premises. Temperatures shall be simulated by using resistance boxes and also stimulated by application of heat and cold.

Deliverable Items:

(1) Main Controller:

NEMA 12 case - contains power supplies, logic unit, system clock, status-display-manual panel and temperature time set switches

(2) Temperature Units:

NEMA 3 case - contains analog to digital conversion circuits for temperature sensors

(3) Pump - Valve Units:

NEMA 3 case - contains solid state switches for driving valves and pumps

(4) Cabling:

Necessary cables and connectors to route signals through the system

(5) Drawings:

Engineering sketches and parts lists as required for repair and maintenance

(6) Manual:

Contains descriptions as to theory of system operation and calibration procedures as necessary

(7) Installation:

Two man days of engineering to assist in installation and acceptance test

Scheduling Milestones:

- (1) Third week after ARO design review and subsequent release of all P.O. and fabrication drawings
- (2) Sixth week after ARO installation instructions and drawings
- (3) Ninth week after ARO - items 1 thru 6

Note: 3 above is based on parts delivery information acquired during bid preparation

Terms of Payment:

- 40% on contract initiation
- 30% at 30 days following design review
- 30% at final installation and acceptance testing

Cost:

Total cost including latest modification is \$4990.00

This quote is valid for 30 days.

William H. Reading III
President
Roburn Corporation

InterTechnology Corporation

InterTechnology Corporation

WHR/br
c.c. Ted Collins

APPENDIX B

TEMPERATURE AND VENTILATION CONTROL SYSTEM

Introduction

The system designs for the tank as well as the mechanical equipment associated with the solar portion of this project, have used only standard manufacturers' readily available equipment. The system has been designed to provide hot water to the storage tanks whenever the water in the tank is cooler than the collector plate temperature. The mechanical system is designed to provide heating or cooling to the central air handling system in response to a manual or automatic season selection. The corresponding heating or cooling which may be required will be provided by energy conservation methods discussed in other portions of this work. These designs have received tentative approval by the product manufacturers associated with the generation of heating and cooling as well as a number of recognized temperature control manufacturers. In addition, a control system is included to reduce the required amount of ventilation as the people load varies within the conditioned space. This system operates in conjunction with other damper operators which control the amount of heating or cooling from the outside air rather than by mechanical generation. A discussion entitled "Economizer Cycle" in another section of this report provides a narrative description of the operation and costs.

SEQUENCE OF OPERATION FOR THE MECHANICAL GENERATION
OF HEATING AND COOLING AS SHOWN ON DETAIL M-2

A. General: The system shall be provided with an outdoor air sensor (T-7) and a manual override switch located in the equipment room. This combination shall index the solar collection system from the heating to the cooling mode as well as start the centrifugal chiller as follows:

1. When the outside air temperature falls below 60°F, outdoor air thermostat (T-7) shall de-energize the chilled water pump P-3 (which in turn shall deactivate the chillers, the tower, and the condenser water pumps P-1 and 2). In addition T-7 shall energize the system pump (P-5) and position 3-way valve V-2 to allow flow, from the pump, to the reheat coil of the air handling system. This shall place the system in the "winter" mode as hereinafter described.

2. When the outside air temperature rises above 63°F, outdoor air thermostat (T-7) shall energize the chilled water pump P-3 which in turn shall energize the condenser water pump and, when water flow has been electrically and mechanically established, the centrifugal chiller shall energize and operate under its own control system. The condenser water pump shall activate the cooling tower controls as shown in detail M-3. In addition, two position, 3-way valve (V-2) shall allow flow to the absorption machine and the system pump (P-5) shall de-energize. This shall place the system in the "summer" mode as hereinafter described.

3. The manual bypass switch shall be located in the equipment room and shall be labeled "summer-auto-winter". This switch will bypass the automatic operation of the system as described in 1. and 2. above and lock the system into the mode selected regardless of outside-air temperature. Valve V-5 shall modulate (through T-5) to maintain 85°F condenser water supply temperature regardless of the manual bypass switch setting.

B. Summer: With V-2 open to the absorption machine, P-5 de-energized, P-3, P-1 and P-2 activated, water flow established, the cooling tower operating and V-5 providing 85°F water to the condenser, the system will operate as follows: when the storage tank temperature (T-6) rises above 180°F the system pump (P-5) shall be energized. Once flow is established through the generator of the absorption machine, the absorption machine's controls shall energize the machine. As the inlet generator water temperature increases above 180°F the machine's control system shall decrease the setting of the evaporators leaving water temperature. As the temperature decreases from 180°F the reverse shall occur. When the inlet generator water temperature (T-3) falls below 171°F, the system pump (P-5) shall de-energize (deactivating the absorption machine). P-5 shall remain inoperative until the tank temperature (T-6) again reaches 180°F. T-2 shall modulate V-4 to maintain cold deck temperature.

C. Winter: With P-5 energized, valve V-2 open to the reheat coils, and the cooling system deactivated, the system shall operate as follows: hot water from the storage tank shall be provided to the reheat coils through V-3. T-1 shall control V-3 and the variable volume box room temperature. Whenever the return air temperature (T-9) falls below 68°F, 2-position 3-way valve V-1 shall allow flow to the boiler. Once flow is established through the boiler, the boiler shall maintain a discharge temperature of 110°F (T-8). When the tank temperature reaches 113°F (T-4) valve V-1 shall reverse and allow flow directly from the tank to the reheat coils. The system shall remain in this condition until the return air temperature again falls below 68°F.

SINGLE TANK COLLECTOR OPERATION

Whenever the collector plate temperature (T-1) rises above the storage tank temperature (T-2), the comparator (C-1) energizes timer (TM). This will be called the "start up cycle". The timer then shall operate for a period of five (5) minutes irrespective of C-1 with its contactor closed, thereby energizing the collector pump for a minimum of five (5) minutes. At the end of the five minute run the timer shall keep the pump energized unless C-1 indicates that the tank temperature is warmer than the plate temperature. This will be called the "shut down cycle". At that time the timer shall reverse its operation and, at the end of another five minutes (assuming C-1 does not call for collection), the timer shall deactivate the collector pump. If C-1 should call for collection during the "shut down cycle", the timer shall then begin to operate on the previous or start up cycle with the exception of rigid adherence to the five (5) minute delay.

Whenever the pump is in operation dump valves V_1 and V_2 shall close. Whenever the pump is de-energized, V_1 and V_2 shall open-draining the collector to the tank.

The pump starter shall be connected to a hand-off auto switch located on the starter and shall be interlocked with any required fire protection devices.

SEQUENCE OF OPERATION FOR AIR HANDLING UNIT DAMPERS

A. With the fan in operation the E-P switch (EP-1) will provide main system air pressure to valve V-1. Controller C-4 through sensing element T-4 will position valve V-1 into the summer or winter position. If the outside air temperature is above 60°F the valve V-1 shall be positioned to summer.

B. Summer Operation:

With the fan running, and V-1 in the summer position V-1 shall supply controller C-1 with main system air pressure. Controller C-1 through sensing elements T-1 and T-2 shall operate pilot positioned dampers D-1, D-2, and D-3 in the following manner:

1. When temperature T-1 is greater than temperature T-2, the exhaust damper D-1 and the outside air damper D-3 shall open. At the same time, return damper D-2 shall close.
2. When temperature T-1 is less than temperature T-2 the reverse of 1. above shall occur - i.e., D-1 and D-3 closed, D-2 open.

C. Winter Operation:

With the fan running, and V-1 in the winter position, V-1 shall supply controller C-3 with main system air pressure. Controller C-3, through sensing element T-3, shall modulate dampers D-1, D-2, and D-3 to maintain 60°F temperature in the cold deck.

D. Minimum Outside Air Damper Control.

1. With AH-1's fan energized, the central communications console shall indicate to electric-pneumatic switches (EP-2) when the resources and administration lights are on. Upon the transmission of a signal to the EP switch that the lights are on, the EP switch shall send EP

air to an averaging cumulator (AH-1 sequence not shown in diagram).

If only one area's lights are on the averaging cumulator will position minimum outside air damper D-4 to its mid position. If both of the area's lights are on, D-4 shall (through the averaging cumulator), be 100 percent open. If both lights are off, the damper shall close. If D-4 is closed and AH-1 continues to operate, dampers D-1, D-2, and D-3 shall return to their normal positions in the winter through V-2.

2. With AH-2's fan energized, the central communications console shall indicate to electric-pneumatic switches EP-2 (typical for six) when the classroom lights are on. A similar indication shall be given to another EP-2 to indicate when the lunchroom lights are operating. The signals from the classroom EP-2 switches shall be transmitted to averaging cumulators A-1, A-2 and A-3 and the average of the total shall then be sent to another cumulator's H-S. This cumulator (H-S) shall add the classroom occupancy to the lunchroom occupancy to provide the proper signal to damper D-4. The system shall operate basically as described in 1. above.

DISPLAY PANEL

A. General: Provide an indicating display panel as shown on detail M-4 and locate it in the resources area of the school. The face of the panel shall be tan linen formica, lucite, or other approved material. There shall be a detailed graphic representation of all water systems as indicated. Graphic representation to be face engraved on formica or reverse engraved on lucite. All symbols, lines, etc. shall be in durable, contrasting color and clearly outlined in black. Flow lines to be color filled to conform with standard color coding. Mechanical equipment shall be represented in outline form so as to be readily recognizable. Pilot lights shall, in addition, be located at each piece of operating equipment on the graphic representation.

B. Indication: Provide an electrically operated, 3 inch scale meter, designed for flush mounting in the panel, for each of the following indication points:

Indication Point	Scale Range
Solar Collector Water Supply	40-240°F
Solar Collector Water Return	40-240°F
Tank Temperatures	40-240°F
Hot Water Supply	40-240°F
Hot Water Return	40-240°F
Boiler Supply	40-240°F
Chilled Water Supply	40-90°F
Chilled Water Return	40-90°F
Reheat Entering	40-150°F
Reheat Leaving	40-150°F
Cold Deck Entering	40-90°F
Cold Deck Leaving	40-90°F
Chilled Water Intermediate	40-90°F
Classroom	40-90°F
Outside Air	40-150°F

All associated devices for the accurate display of the indicated temperatures shall be provided. The electronic sensing elements, where located in pipe shall be provided with a separable well which shall be permanently inserted into the pipe. This contractor, in addition to the items mentioned above, shall provide indication from the pyranometer (installed by others) and it shall be this contractor's obligation to provide the proper interface with the pyranometer.

C. Panel Construction: The panel shall be factory prewired and tested by the controls manufacturer and furnished complete with an integral labelled terminal strip ready for field wiring connections. All panel wiring is to be for 120 volts or less. Provide matching formica nameplates engraved with name/function of each item on panel.

Appendix C. Comparison of Thermal Storage Systems on the Basis of First Cost

Method

Assume Q Btu are stored for winter heating between temperatures T_2 and T_1 , $T_2 > T_1$, for Δt_s days. The difference $T_2 - T_1$ is the useful range of temperature for heat storage.

For water and rocks, the mass M of storage material is

$$M = Q/c (T_2 - T_1) \quad (1b) \quad (1)$$

where c = specific heat = Btu/lb-°F = assumed constant between T_1 and T_2 .

For salts, or other heat of fusion systems, M is

$$M = Q/h_f + c (T_2 - T_1) \quad (1b) \quad (2)$$

where h_f = heat of fusion at fixed temperature, i.e. T_1 = Btu/lb,
 c = specific heat in liquid state over useful range of temperature for the material in the liquid state.

The volume V required to store Q Btu is

$$V = M/\rho \quad (ft^3) \quad (3)$$

where ρ = density = lb/ft³.

The area of the storage media container to be insulated can be determined from (3), once the geometry of the container has been selected.

For cylindrical tanks,

$$A = 2\pi R(R + L) \quad (ft^2) \quad (4)$$

where R = radius of tank (ft)
 L = length of tank (ft)
 Radius and length must satisfy $vV = \pi R^2 L$
 v = void space factor if required by storage media (as in air systems with rock storage for instance).

For parallelepipedic tanks,

$$A = 2(HW + HL + LW) \quad (\text{ft}^2) \quad (5)$$

where

L = length of container (ft)
 W = width of container (ft)
 H = height of container (ft)
 and $vV = L \times W \times H$
 v = void space factor if required by storage media

For cubic tanks,

$$A = 6 L^2 \quad (\text{ft}^2) \quad (6)$$

where L = length of side of the cubic structure
 $vV = L^3$, v = void space factor.

The first cost, FC , of storage is

$$\begin{aligned} FC &= \text{cost of container} + \text{cost of storage material} + \text{cost of insulation} \\ &= C_c V + MC_m + A (a + 2 \sqrt{Dk_{ins} C_h ab}) \end{aligned} \quad (7)$$

where C_c = unitary cost of container (\$/ft³ or \$/gal.)

C_m = unitary cost of storage material (\$/lb)

$$a + 2 \sqrt{D k_{ins} C_h \alpha b} = \text{optimized cost of insulation (21), } (\$/ft^2)$$

$$D = 24 d_w (T_{Tw} - T_w) + 24 d_s (T_{Ts} - T_s)$$

d_w = average number of winter days

d_s = average number of summer days.

T_w = average winter temperature ($^{\circ}F$)

T_{Tw} = average storage temperature in winter ($^{\circ}F$)

T_s = average summer temperature ($^{\circ}F$)

T_{Ts} = average storage temperature in summer ($^{\circ}F$)

k_{ins} = loss factor for chosen insulating material

$$= \text{Btu-in/hr-ft}^2\text{-}^{\circ}F$$

C_h = cost of heat = $\$/10^6$ Btu

α = interest factor = 11.47 for 6% and 20 years

b = cost of insulation = $\$/in\text{-ft}^2$

a = first cost of insulation = $\$/ft^2$

The optimized insulation thickness, δ^* , is

$$\delta^* = (D k_{ins} C_h \alpha / b)^{1/2} \quad (\text{in}) \quad (8)$$

The optimized insulation thickness δ^* is determined by minimizing the total (insulation cost + heat loss value) equivalent first cost(21).

The first cost per million Btu of storage capacity is

$$(FC)_m = FC \text{ (equation (7))} \times 10^{-6}/Q \quad (\$/10^6 \text{ Btu}) \quad (9)$$

Assuming a straight depreciation over a 20 year period, the depreciation per million Btu-year used is

$$d = \frac{FC(\text{equation (7)}) \times 10^{-6}}{20 \times Q_y} \quad (\$/10^6 \text{ Btu}) \quad (10)$$

where Q_y = total average Btu stored per year

$$= \frac{Q_w}{\Delta t_{sw}} \times d_w + \frac{Q_s}{\Delta t_{ss}} \times d_s - Q_L \quad (\text{Btu/y}) \quad (11)$$

Q_w = storage capacity (Btu) for winter storage = Q above if system is designed on basis of winter storage capacity

Δt_{sw} = average period of storage in winter (days) = Δt_s above if system is designed on basis of winter storage capacity

d_w = average number of winter days

$$= (\text{number of heating degree-days}) / (65 - T_w)$$

Q_s , Δt_{ss} and d_s are the quantities corresponding to Q_w , Δt_{sw} and d_w for summer storage

Q_L = yearly heat losses

The total yearly heat losses Q_L are given by

$$\begin{aligned} Q_L &\approx \frac{24 k_{ins} A}{\delta^*} \left[(T_{sw} - T_w) d_w + (T_{ss} - T_s) d_s \right] = \frac{D k_{ins}}{\delta^*} A \\ &= 24 A (b k_{ins} / DC_h \alpha)^{\frac{1}{2}} \left[(T_{Tw} - T_w) d_w + (T_{Ts} - T_s) d_s \right] (\text{Btu/y}) \quad (12) \end{aligned}$$

where use has been made of (8).

Appendix D. Cost of Metal Storage Tanks

Figure D-1 shows the unitary cost (\$/gal) of steel tanks versus tank capacity. These costs include the tank itself, legs, brackets, manholes and openings for piping. The cost of installation ($\$50 + \$0.03/\text{gal}$) is also included.

These cost estimates were made on the basis of quotations from Maphis Chapman Corporation, Harrisonburg, Va. 22801, August 1974. These costs are consistent with other quotations.

Pressurized tanks are expected to be double the prices of Figure D-1.

FIGURE D-1 - Unitary Cost of Steel Tanks as Function of Tank Capacity

(Costs include tank, legs, brackets, manholes and installation.)

Appendix E. Cost of Concrete Tanks

Figure E-1 shows the unitary cost (\$/gal) of concrete tanks versus tank capacity. These costs include the tank itself, manholes and openings, waterproofing finishing layer inside, excavation for underground installation and grading. The tanks are assumed to be cubical in shape. It should be noted that the excavation and grading costs are small compared to the overall costs, so that essentially the same numbers are a good approximation for above ground units. These cost estimates were made on the basis of quotations from "Building Cost File", 1972 Eastern Edition, Construction Publishing Company, Inc., N. Y., adjusted for 1975.

The costs estimated in the figure are consistent with quotations from manufacturers.

FIGURE E-1 - Unitary Cost of Concrete Tanks as Function of Tank Capacity

Appendix F. Cost of Cinder Block Containers

Figure F-1 shows the unitary cost (\$/gal) of cinder block tanks versus tank capacity. These costs include material, labor, duct openings, a wooden cover, inside surface finishing, excavation and grading. The containers are assumed to be cubical as this geometry appears to be preferred for rock storage systems,(6). Excavation and grading costs are small compared to the total costs which can thus also be used for above ground units. These containers are mostly envisioned for use with rock storage in which case the purpose of the inside surface finish is to improve air flow by suppressing turbulence at the rough concrete block surface. In principle a waterproofing finish could alternatively be applied in which case the unit could be used for water storage.

The costs were estimated on the basis of "Building Cost File" - 1972, Eastern Edition, Construction Publishing Co., Inc., N. Y., adjusted for 1975.

The costs on Figure F-1 compare with typical costs of above ground swimming pools (metal frame, plastic liner) i.e. \$0.05/gal to \$0.10/gal for capacities in the range of 5,000 to 15,000 gallons.

FIGURE F-1 - Unitary Costs of Cinder Block Containers
As Function of Capacity of Container

Cost include materials, labor, wood cover, inside surface finishing,
excavation and grading and duct openings.

Appendix G. Cost of Wooden Storage Bins

Figure G-1 shows the unitary costs (\$/gal) of wooden bins for rocks or phase-change materials storage. These costs include a wooden frame, panelling inside and outside (outside grade panels), duct openings, inside finishing if needed to prevent air leaks and excavation and grading required to accommodate the bin in a basement space. The inside finishing were assumed to be moldings around the inside edges of the container to prevent leaks at the panel junctions. Here again excavation is a relatively minor item compared to other costs and the estimates apply as well for outside, independent structures attached or not to a building. As recommended for rock storage units, (6), the bins were assumed to be cubical in shape. The costs were estimated on the basis of "Building Cost File" - 1972, Eastern Edition, Construction Publishing Co., Inc., N. Y., adjusted for 1975.

The estimated costs are comparable to those quoted by independent contractors for similar wooden structures.

FIGURE G-1 - Unitary Costs of Wooden Storage Bins as Function of Bin Capacity

Cost include frame, double panelling, openings, inside finishing, excavation and grading required to accommodate bin.