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# RESIDENTIAL SOLAR HEATING AND COOLING USING EVACUATED TUBE SOLAR COLLECTORS—CSU SOLAR HOUSE III

Final Report, Technical Summary, February 1, 1976—September 30, 1978

By
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March 1979

Work Performed Under Contract No. EY-76-S-02-2858

Solar Energy Applications Laboratory
Colorado State University
Fort Collins, Colorado

## U.S. Department of Energy



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# RESIDENTIAL SOLAR HEATING AND COOLING USING EVACUATED TUBE SOLAR COLLECTORS - CSU SOLAR HOUSE III

FINAL REPORT - TECHNICAL
FOR PERIOD 1 FEBRUARY 1976-30 SEPTEMBER 1978

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MARCH 1979

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PREPARED FOR THE
U.S. DEPARTMENT OF ENERGY
SOLAR ENERGY
UNDER CONTRACT EY-76-S-02-2858

TECHNICAL REPORT
TABLE OF CONTENTS

TECHNICAL REPORT
TABLE OF CONTENTS

## TECHNICAL REPORT
## LIST OF TABLES



## TECHNICAL REPORT
## LIST OF FIGURES

FINAL REPORT - TECHNICAL

This report documents the technical results achieved under contract EY-76-S-02-2858 during the period 1 February 1976 through 30 September 1978. The bulk of the technical effort was devoted to the evaluation of a residential solar heating and cooling system utilizing an array of evacuated tube solar collectors. In May 1978, the evacuated tube collectors were removed and replaced with a state-of-the-art liquid-heating flat-plate solar collector array. Because of the distinct nature of the evaluations of the two collector systems, this report will consider each unit separately.

The first section of this report will deal exclusively with the installation, performance, and operating experience of the Owens-Illinois (O-I) evacuated tube solar collector integrated with the CSU Solar House III residential sized solar heating and cooling system. The second, smaller section of this report will discuss the four months of installation, operation, and evaluation of the Chamberlain flat-plate collectors installed on CSU Solar House III.

## PERFORMANCE EVALUATION - OWENS-ILLINOIS

### 1.0 BACKGROUND

The use of a vacuum such that the pressure is kept below $10^{-3}$ torr virtually eliminates conduction and convection heat transfer and limits any heat transport to radiation only [1]. The use of a vacuum between the absorber surface and the cover of a solar collector, therefore, has the advantage of greatly reducing the heat losses from the collector. This is seen by noting that the heat loss rate from an evacuated solar collector ($Q_L$) can be written as:

$$Q_L = \varepsilon' \sigma (T_s^4 - T_g^4) \qquad (1)$$

where $\sigma$ is the Stefan-Boltzman constant, T is the absolute temperature of the absorber surface ($T_s$) and glass cover ($T_g$), and $\varepsilon'$ is the effective emissivity of the absorber. $\varepsilon'$ depends on the geometry of the absorber; for a cylindrical absorber within an only slightly larger concentric glass cylinder, the areas of the two surfaces are approximately equal and can be described by the same area, A. Thus from [2] we obtain:

$$\varepsilon' = \frac{A}{1/\varepsilon_s + 1/\varepsilon_g - 1} \tag{2}$$

where $\varepsilon$ is the emissivity of the absorber surface ($\varepsilon_s$) and glass surface ($\varepsilon_g$).

Using equations (1) and (2), we obtain the heat loss coefficient ($U_L$) of the evacuated solar collector. Calculated values are typically $U_L \approx \varepsilon(8.44\ W/m^2\text{-}°C)$. For absorber emittances of $\varepsilon_s = 0.07$ to 0.10, $U_L$ varies from 0.6 to 0.8 $W/m^2$-°C. Because of the support structure of the absorber surface within the evacuated glass tube and the associated conduction losses, experimental values of $U_L$ are typically 0.85 to 1.15 $W/m^2$-°C.

Transmissivity-absorptivity products ($\tau\alpha$) for tubular glass collectors are equivalent to two glass covers with normal incidence, i.e., ($\tau\alpha$) = 0.80, but the use of anti-reflection coatings and low absorptance glass allows for values of ($\tau\alpha$) to 0.85 to 0.92. Values of ($F_R$), the collector heat removal factor, vary considerably with the design of the evacuated tube solar collector and the heat transfer fluid utilized, but for liquid-heating collectors are generally high.

In general, the combination of very low $U_L$ and high $(\tau\alpha)F_R$ values allow for exceptionally high daily solar collector efficiencies. This is due to the fact that the evacuated tubular collector is able to collect useful solar energy under conditions of limited solar availability. For example, the "Solar Threshold" (the minimum solar radiation required for the useful collection of solar energy) for a typical flat-plate collector operating at a temperature 50°C above ambient is 0.25 to 0.30 $kW/m^2$. For the evacuated tubular collectors, this same solar threshold is only 0.13 to 0.15 $kW/m^2$. The importance of this lower solar threshold is that the solar collector can collect useful energy earlier in the mornings and later in the evenings and under less favorable solar conditions (thus extending each day's operating period, as well as extending the operation of the collector to more days per year). This latter aspect provides for very substantial improvements in average daily collector efficiencies in climatic regions of marginal solar availability.

The increase in the solar operating period enjoyed by the evacuated tubular solar collectors also improves the overall performance of a solar cooling system, which requires high temperatures for operating absorption cooling machines. State-of-the-art lithium bromide absorption units typically require generator input temperatures of 70 to 90°C, while ammonia absorption units require temperatures of 90 to 180°C, depending upon the type of heat rejection equipment. The integration of evacuated tubular collectors with absorption cooling then allows for high efficiencies and higher percentage of cooling by solar per unit area of solar collector installed.

## 2.0 SUMMARY

Performance and pertinent operating experience with the Owens-Illinois evacuated tube liquid-heating solar collector integrated with the CSU Solar House III heating and cooling system has been acquired and evaluated over a 20-month period. The solar collector shown in Figs. 1 and 2 demonstrated excellent performance, achieving 50 percent daily collector efficiencies under relatively adverse weather conditions and low solar insolation. However, electrical parasitic power requirements were excessive and overnight heat losses from the solar collector and associated piping constituted approximately 33 percent of the useful heat delivered during the winter heating months.

Because of the numerous operational difficulties inherent in the design of the collector and experienced over the 20-month evaluation of the collector, the project staff has determined that the O-I liquid-heating evacuated tube solar collector cannot be considered practical for residential solar heating and cooling applications. However, the superior performance of this evacuated tube collector indicates an excellent potential for use as an air-heating solar collector.

## 3.0 TECHNICAL ACCOMPLISHMENTS

- The O-I evacuated tube solar collector array was installed on the roof of CSU Solar House III and integrated with the specially designed solar heating and cooling system. The design integration of the solar collector with the system has been reported previously [3,4,5,6,7] (see section on "Publications/References")
- Performance and pertinent operating experience was acquired over a 20-month period. During this time numerous operational difficulties were encountered and evaluated. These problems include:
  (1) Excessive leakage of the collector liquid
  (2) Consistent breakage of the evacuated glass tubes
  (3) Considerable difficulty in obtaining adequate flow through the collector following a boiling episode
  (4) Difficulty inserting and maintaining control and data sensors within the liquid flow volume of the evacuated glass tubes

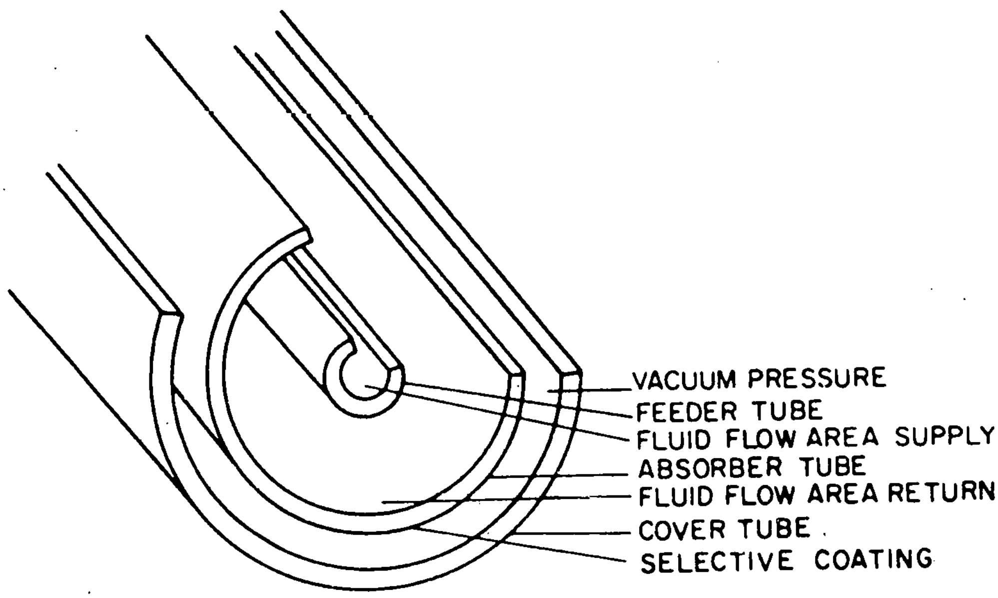


Figure 1. Evacuated Tube Solar Collector Schematic (Owens-Illinois)

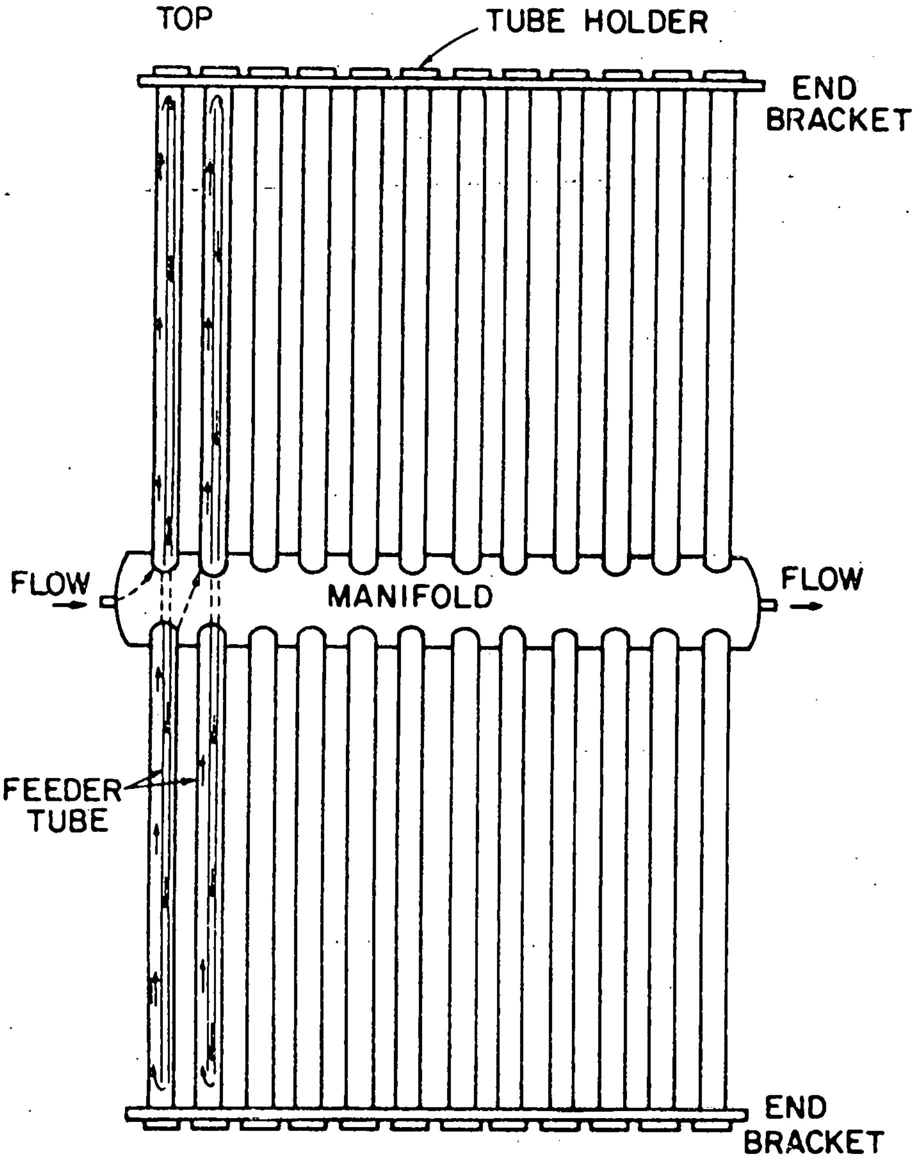


Figure 2. Evacuated Tube Solar Collector Module
(Owens-Illinois)

(5) Control problems associated with long response times
(6) No possibility of draining the collector without disassembly
(7) Freezing in the evacuated glass tubes and the solar collector module manifolds
(8) Frequent boiling resulting in additional liquid loss and evacuated glass tube breakage
(9) Large heat capacity resulting in large overnight heat loss.

The problems have been discussed in detail in references [6], [8], and [9].

- Specific performance characteristics of the solar collector and the system have been evaluated and reported in the literature. These characteristics include:
  (1) Low solar collection thresholds
  (2) No significant advantage in the use of specular reflectors
  (3) High electrical usage in pumping the collector liquid through the collector
  (4) Excellent stability of the absorber tube selective surface
  (5) High equilibrium, no flow (i.e., stagnation) collector temperatures
  (6) High daily collector efficiencies

  Specific results are reported in reference [8] and [9].
- Monthly summaries of continuous data for two specific and operating periods have been acquired and reported (reference [8] and [9]).
- The O-I solar collector array has been integrated with the Direct Contact Liquid-Liquid Heat Exchanger (DCLLHE) for a month-long test. The results are reported in that project's final report.

## 4.0 DESIGN

CSU Solar House III utilizes the Owens-Illinois evacuated tubular solar collector, the Yazaki lithium bromide absorption chiller (WFC-6003), and the integrated solar heating and cooling system design shown schematically in Fig. 3. Two cool storage tanks are utilized to operate the absorption chiller. As the chiller provides chilled water, "Cool Storage II" is filled and "Cool Storage I" is drained, thus the output water temperature from the chiller can be kept as low as possible and not mixed with the warmer water in "Cool Storage I". The cool storage tanks provide a temperature difference of about 5°C (9°F) which is sufficient to chill the recirculated room air. In addition to improving the operating characteristics of the absorption chiller, the use of cool storage in SH III permits a smaller capacity chiller.

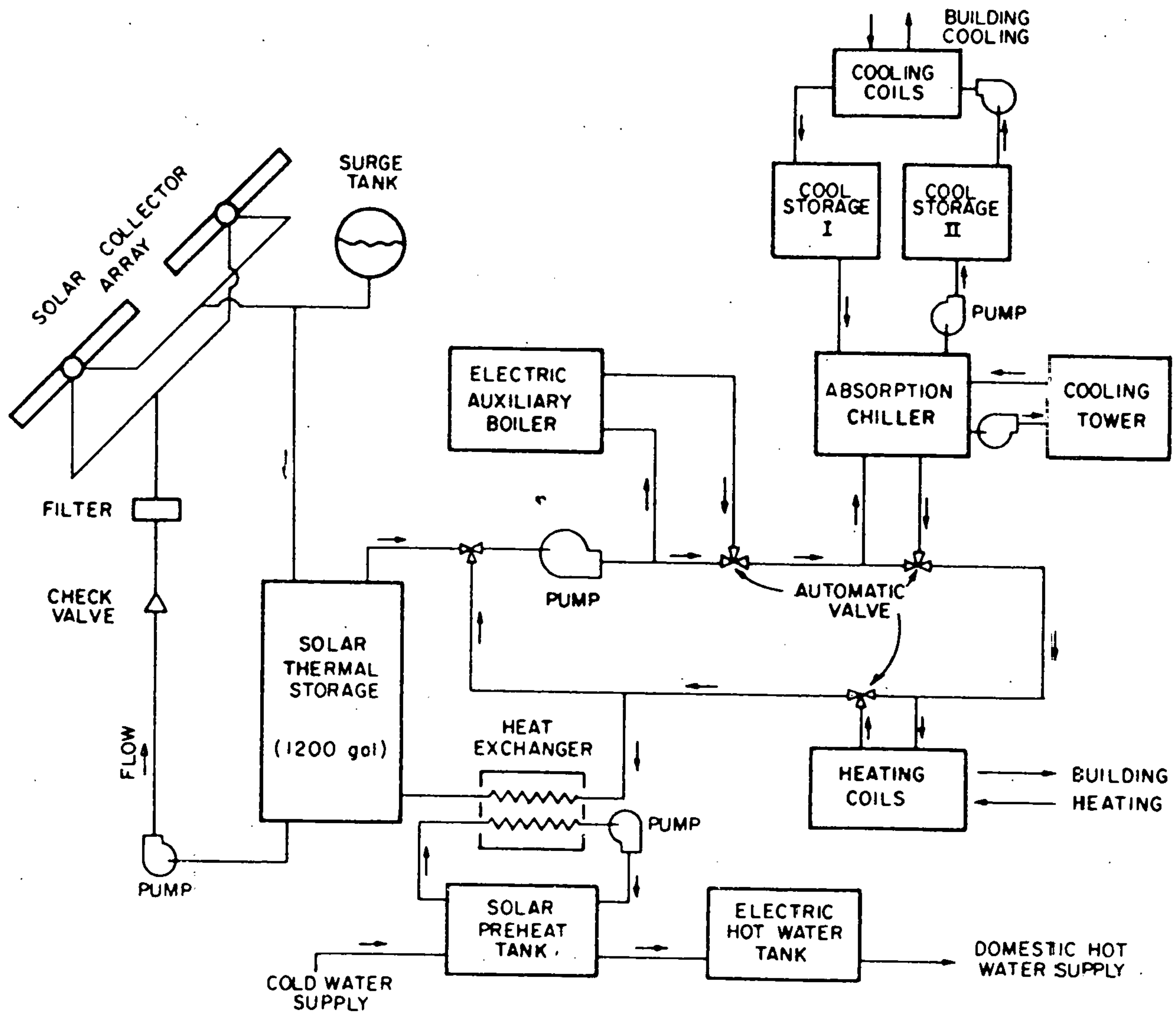


Figure 3. Solar House III System Schematic

Instead of requiring the air conditioner to be large enough to provide the required cooling under a heavy load condition, the cool storage can be used together with the chiller to provide a larger cooling capacity than the chiller alone. Also, with a smaller chiller capacity and storage the unit operates nearly continuously, which results in maximum operating efficiency. Where normally a 3-ton chiller is required, a 2-ton chiller can be used instead (1 ton = 3.5 kW). This is better than a 33 percent reduction in chiller capacity.

The solar collector array on Solar House III consists of 16 modules of evacuated tube collectors arranged in two rows and each module contains 24 evacuated tubes. Each tube has an outer diameter of 5.7 cm (2.25 in.) and a length of 1.15 m (3.8 ft) and are spaced as shown in Fig. 2, so that each collector module covers an area of about 3 $m^2$ (32 $ft^2$). The effective collection area, however, is only 2.55 $m^2$ (27.4 $ft^2$) with an absorber area of 1.05 $m^2$ (11.3 $ft^2$).

The flow pattern of the water through the collector module is indicated in Fig. 2. Each of the 24 evacuated tube collectors is in series so that water passes through all 24 tubes in sequence before returning to storage. It takes about 10 to 15 minutes for water to travel from the inlet of the module manifold to the outlet.

While the tubes in a module are arranged in series, the modules in the array are arranged in parallel. Space between the tubes is important because reflection from the area between and behind the tubes can be directed toward the collector tubes.

## 4.1 Evacuated Tube Collector Array Installation

### 4.1.1 On-Site Assembly

Figure 4 shows the components of a single 4 ft by 8 ft collector module of the O-I design. The actual assembly of all the components on-site (generally a steeply sloping roof) is time consuming and requires careful attention to detail. The parts list for a single module consisted of about 300 separate pieces, thus the Solar House III 16 module array consists of about 4800 pieces. While many of these components can be assembled on the ground (e.g., 12 grommets, 24 O-rings, and 24 end seals can be installed in the manifold), others require individual attention. For example, the tube support cup assemblies consist of five pieces which can be preassembled, but have a tendency to fall apart prior to actual installation. Such features slow the installation process considerably.

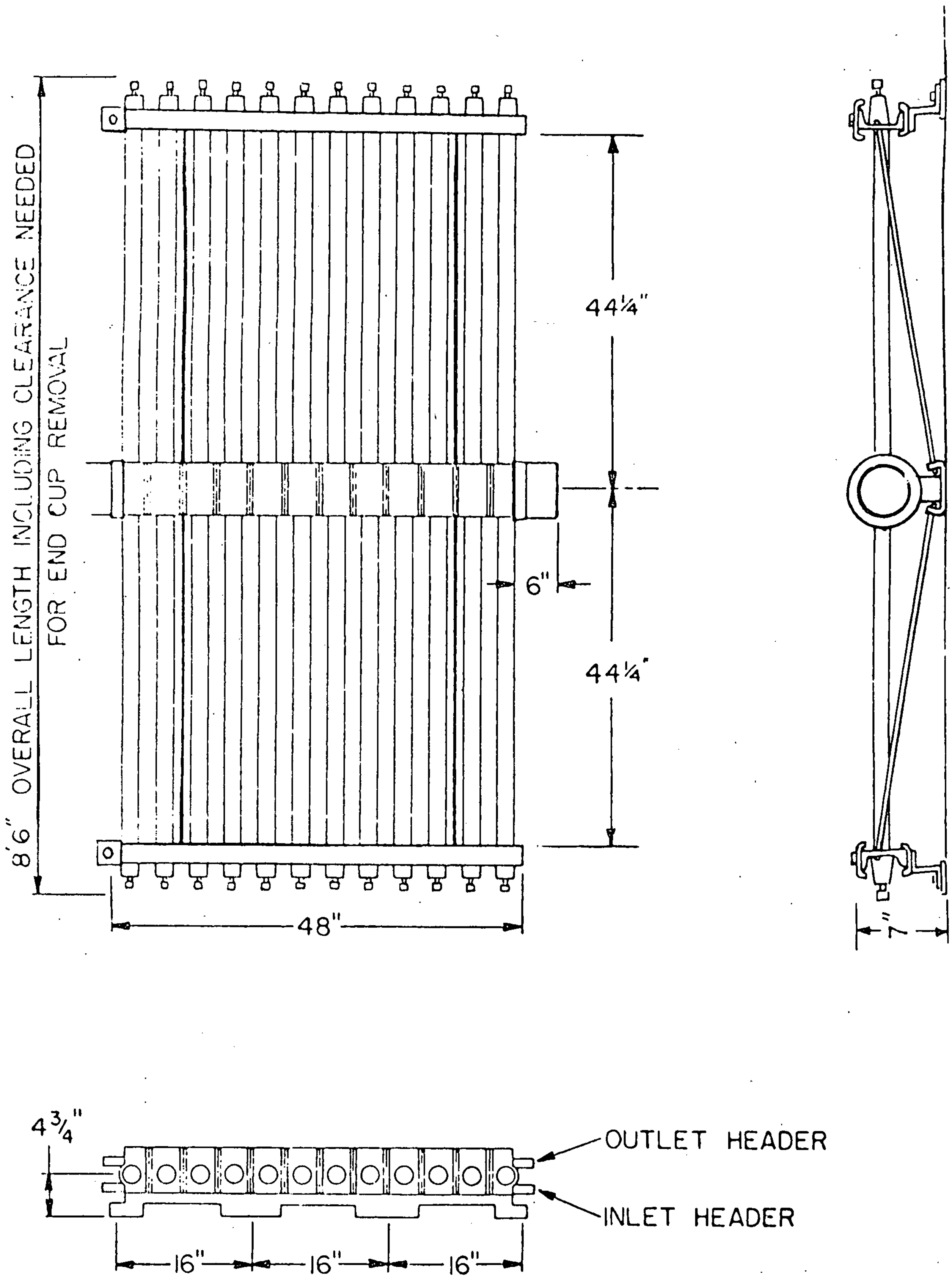


Fig. 4. Owens-Illinois Collector Components

If reflectors are added to the above list, this constitutes 24 reflectors and 96 wire clips per module. The wire clips are particularly hard to work with and require workmen to work without gloves (even in cold weather). The effect of this multitude of pieces (about 6700 total) is to greatly increase the cost of installation and would suggest that a self-contained modular design that could be lowered into place as a single unit would be much preferable.

In this regard it should be noted that the dimensions are critical and must be laid out on the roof with great care. For example, the distance between the center line of the manifold and the end bracket must be exactly 44.25 inches, in order for the evacuated tubes to be able to fit in the space. But, in addition to the difficulty of laying out this dimension and maintaining a perfect square, one must also contend with the tendency of the manifolds and end brackets to lean toward the down slope of the roof (a factor more noteworthy whenever an installer inadvertently uses the manifold as a foothold).

Finally, the collector module requires an additional 15 cm (6 in.) for an overall length of 259 cm (8 ft 6 in.) in order to provide sufficient clearance for end cup removal. In reality even more area between collector arrays is needed in order to provide working space during the actual assembly (and avoid putting weight on the manifolds or end brackets). This additional area must then be charged to the total collector area; the required area for two arrays of 8 modules each might then be a space of 9.75 m by 5.49 m (32 ft by 18 ft), or 53.5 $m^2$ (576 $ft^2$) instead of 47.6 $m^2$ (512 $ft^2$) of which 40.7 $m^2$ (438 $ft^2$) is reflector, 21.75 $m^2$ (234 $ft^2$) is tube, and 16.8 $m^2$ (181 $ft^2$) is absorber tube area.

#### 4.1.2 Leakage Problems

A principal difficulty with an on-site assembly of the solar collector modules is the inability to check for potential leaks until the entire collector has been installed. Once a leak is discovered, a large portion of the complete array of tubes must be removed in order to repair the leak. And, in the process of removing tubes, this can damage other components. For example, because the collector usually sits in a no flow dry condition prior to initial start-up, the rubber connectors for the feeder tubes are effectively baked on the feeder tube. In the process of removing the collector tubes to repair a leak, these rubber connectors must be cut and then replaced with new ones upon reassembly. The collector manufacturer has recognized this problem and is now considering alternative materials for the feeder tube connectors.

Substantial leaks have occurred in the connections between the manifolds and at the junction of the evacuated tubes and the manifold. The latter source of leaking is relatively easy to correct and usually requires only an adjustment of the tube in the manifold cup. The only difficulty arises when an upper tube must be adjusted. In this case it is easy to dump the collector liquid during the adjustment and thus introduce a large air pocket into the module. This is not self-correcting and it may be necessary to drain the entire collector array in order to correct the situation. (The draining of the collector array is discussed below.)

Significant leaks in the manifold connections are generally more difficult to correct and, in the case of the first O-I design, were more common. The first design required soldering of 2.5 cm (1 in.) copper pipes (two per interface between manifolds). Normally two manifolds are soldered together on the ground and then placed on the roof as a unit. This proved completely unsatisfactory, as the subsequent movement stressed the solder joint to the extent that five out of eight connections made in this way developed leaks. Ironically, no soldering work performed on the roof to connect the manifolds had any leaks at all. O-I has since replaced the requirement of copper pipe soldering with specially adapted fittings. These fittings enormously simplify the installation on a roof and are a decided improvement. In the CSU installation, the lower array utilized the improved fittings while the upper array used the earlier soldering technique.

The difficulties associated with being able to check for leaks in the collector manifold assembly and the related piping, prior to the completion of the array's installation, are compounded by the inability to easily drain the collector to an extent which would allow repairs to be made and to subsequently recommence operations. On the other hand, the upper tube on either side of each collector module would self-drain unless specific precautions were taken to prevent this. (In fact, it was necessary in the CSU installation to incorporate a piping loop on the outlet of the collector which could provide back pressure greater than the water head to prevent an unwanted drain-down.) Unfortunately, the self-draining is a very slow and lengthy process. In one case where a copper solder joint on the inlet side of the collector (the piping leading to the collector) developed a leak, it was necessary to drain the collector inlet in order to repair it. (It is extremely difficult to repair a solder leak when there is any water in the pipe as the heat goes toward boiling the water rather than heating the copper pipe.) Because of the slow self-draining, it took three days for the tubes to drain to the extent that repair was possible.

It is, of course, possible to drain the system in a more positive manner. One method is to pull all upper tubes out of the manifold, thereby dumping the collector liquid onto the roof. This is a difficulty, lengthy and somewhat hazardous procedure due to the normally high temperatures of the collector liquid (note that, once the collector flow rate was shut off, the time to empty all tubes in an array could easily allow the last tubes to be boiling by the time they were attended to). While the lower tubes would not have to be emptied in order to repair the leak, they will have to be emptied in order to assure proper refilling (with no air pockets) and subsequent normal operation. Thus this method of draining is time consuming and quite laborious.

A simpler method is to allow the collector tubes to boil the collector liquid until dry. One disadvantage is the loss of any collector liquid or additive other than water (e.g., ethylene glycol). Another disadvantage is the time (again, several days) to complete the process. There is also the disadvantage of possible damage to the collector due to boil off. In addition, the collector manufacturer no longer considers such intentional boil off an acceptable procedure.

The difficulties encountered in draining a collector module and in other aspects of the operation of the collector would suggest a modified design. Either of two alternatives is a possibility. One is to design the manifold such that all tubes are below the manifold (the dimensions of the module would then be approximately 1.2 by 1.2 m (4 x 4 ft). This has the advantages of no self-draining, ease in removal with minimum loss of collector liquid, and a simpler design. Alternatively, the tubes could be placed above the manifold (again, 1.2 x 1.2 m or 4 x 4 ft). This could allow for an automatic drain-down system and would simplify the problem of air bubbles trapped in the collector tubes. The essential problem with the present design is the combination of tubes above <u>and</u> below the manifold. Such a combination nullifies many of the advantages of either case and increases the possible disadvantages of both cases. This design recommendation has been communicated to the manufacturer and is now commercially available.

### 4.1.3 Pressure Drops

The pressure drop across an O-I collector module is dependent upon the flow rate and has been given by O-I in the form:

| Flow Rate (gpm) | Flow Rate ($lbs/hr-ft^2$) | Pressure Drop (psi) |
|---|---|---|
| 0.11 | 2.0 | 1.0 |
| 0.22 | 4.0 | 3.5 |
| 0.33 | 6.0 | 7.0 |
| 0.44 | 8.0 | 13.0 |

The CSU installation has a flow rate for 16 modules (in two parallel arrays of eight modules each) of 953 kg/hr (4.2 gpm, 0.26 gpm/module), which would correspond to a pressure drop of about 4.8 psi. The collector pump is a Bell and Gossett Series 60, one-half horsepower pump, designed for a flow rate of 908 kg/hr (4.0 gpm) against a total pressure drop in the collector loop of 15.1 psi. This large pressure drop is due primarily to the pressure drops across the heat exchanger between the collector loop and storage and the O-I collector array. For only one array in the collector loop, the same pump provides for a flow rate of 590 kg/hr (2.6 gpm) (indicating a pressure drop across the eight collector modules of 7.0 psi). Addition of ethylene glycol increases the pressure drops.

These high pressure drops constitute a severe disadvantage of the evacuated tube solar collector because of the potentially high pumping power required and the resulting limit on flow rates of the collector fluid through the collector array. The low flow rates have the disadvantage of potentially increasing the collector operating temperature.

For example, the low heat loss coefficient, $U_L$, of the O-I evacuated tubular solar collector extends the operating temperature range significantly. It is possible to observe very high stagnation temperatures (the stagnation temperature is the temperature of the absorber under an equilibrium no flow condition of the collector heat transfer fluid). In tests of the O-I collector, a stagnation temperature of 280°C (540°F) was observed for a solar intensity of 769 $W/m^2$ (272 $Btu/hr-ft^2$). Under even higher solar conditions, temperatures as high as 350°C (662°F) can be expected. Such conditions emphasize the need for careful design of the solar heating and cooling operating and control systems to prevent boiling of the collector fluid, destruction of the control loop, degradation or destruction of the solar collector or its components, and avoidance of the safety hazards of high pressure steam discharge.

The potential for collector fluid boiling is increased by the control time lag inherent in the CSU Solar House III Owens-Illinois collector. Because of the high pressure drops and low flow rates associated with the O-I collector mentioned above (e.g., the pressure drop per module for water as the collector fluid can be related to the mass flow rate by $\Delta p = (0.278)(\dot{m})^{1.83}$ where $\Delta p$ is the pressure drop per module (psi) and $\dot{m}$ is the mass flow rate (lb/hr-ft$^2$)), it takes about 8 to 15 minutes for the collector liquid entering the first tube of a module until it exits from the last tube of the same module. During this time lag, collector temperature increases as much as 20°C have been observed. During the cooling season when the storage temperature might be 80°C (but still below the minimum operating temperature of an absorption cooling machine), the collector pump, if turned on at the storage temperature plus 2°C, could not prevent boiling of the collector liquid. It has, in fact, been necessary to install a special control instrumentation circuit to activate the collector pump whenever the collector temperature exceeds 75°C regardless of the storage temperature.

In addition, the pressure drop across the O-I collector has been observed to be slightly higher when filling the system. In fact, the one-half horsepower pump described above was unable to fill the system without the back pressure of a head of water from the collector outlet to the pump of about 6 m (20 ft) of water head. While the domestic hot water pressurized water main has been suggested as a simple means of filling the collector, such a tactic is severely limited if it is desired to add ethylene glycol for freeze protection or utilize some other liquid as the collector fluid. It should be noted that O-I has the option of installing enlarged feeder tubes, which significantly reduce the pressure drops across a collector module.

#### 4.1.4 Freeze Protection

It might be anticipated that the very low heat loss characteristic of the evacuated tube might prevent freezing of the collector fluid. This is true to some extent, since the evacuated tube may resist freezing for several days even under cold weather conditions and minimal solar input. However, the collector piping and manifolds are not similarly protected and a water/glycol mixture must be used. This mixture must be sufficient to prevent freezing under all reasonably expected winter conditions. The fact that glycol concentrations of 25 percent by volume in water will yield a slush condition (which will not expand sufficiently to burst a pipe) limits the ability of the collector pump to pump through the collector. It is, in fact, possible that pipe freeze conditions in the collector loop not exposed to the sun could occur when the solar collector itself is being heated by the sun. Under these conditions, the collector will boil, an event already experienced in Solar House III. It is noteworthy that, when the liquid in one tube in an evacuated tubular collector module boils, the resultant vapor lock stops the collector liquid flow and causes eventual boiling in all of the tubes of that module.

4.1.5 Reflectors

The initial design of the O-I evacuated tubular solar collector module required a white reflective surface (as part of the roof or collector support structure) to be located directly behind the evacuated tubes. A modification to this design is the use of a shaped, specular reflector directly behind and attached to the evacuated tubes. This modification was expected to yield 25 percent more energy than a similar module without the specular reflector.

It should be pointed out that the O-I collector must be installed in a north-south orientation at a selected tilt angle of 20 to 90 degrees. This is due to the inability of the glass absorber tube to withstand the thermal stresses imposed on a horizontal tube when partially filled with a liquid. However, while the O-I collectors appear to gain little from the use of reflectors, this is not necessarily true for the use of a reflector on a horizontal tube, since the optical losses on a north-south oriented tube during the course of one day are radically different from those of an east-west oriented tube.

4.1.6 Control Instrumentation

The collector pump is controlled by a differential thermostat between the collector and the thermal storage unit. However, the O-I design of the evacuated tube collector module does not allow for the insertion of a sensor which can measure directly the collector fluid temperature. While the outlet of the collector array could be used, this gives a substantially different reading when there is no flow in the collector itself (in many cases exceeding a difference of 30°C). Thus, unless the collector fluid temperature is directly measured, the ability of the control system to optimally control the collector pump is severely degraded.

In the CSU installation, a control sensor was placed in the last tube of one module and the wiring was run through the manifold piping to a connection on the collector outlet pipe. This was a difficult procedure and, ideally, should not be necessary. In addition, because of numerous boil offs and exceptionally high stagnation temperatures, the sensor required periodic replacement. Such replacement could be greatly facilitated by a specific design feature to allow for easy insertion of control or monitoring instrumentation. (It should be noted that most control sensors require a larger voltage output than is available from thermocouples.)

The provision for the insertion of a temperature sensor in one tube of each module would also allow for a check of adequate flow to all modules piped in parallel flow. On several occasions at CSU when the collector had been operating it was necessary to shut down to prevent boil off and possible damage to the collector tubes and the tubes were manually removed and emptied. In this process, six tubes in the upper array were found to be dry inside, indicating no flow prior to the shut down of the collector pump. A check of the data indicated no detectable change in the flow rate previous to that time. Therefore temperature sensors are deemed essentialy in order to adequately check out the initial operation of the system and for later maintenance checks.

During the heating season of 1977-1978, a photoelectric cell was installed to start the collector and heat exchanger pumps at sunrise and stop them at sunset. Operation of a solar collector in this fashion results in a number of disadvantages but there appeared to be no alternative if interruptions in operation due to frequent boiling were to be reduced to a tolerable level. In addition, the frequent loss of control sensors in the boiling collector provided added impetus for this control modification. The disadvantages of the photocell type of operation are reduced to some extent because of the lower rate of heat loss of an evacuated glass tube solar collector are compared to conventional flat-plate solar collectors. The effects of the photocell control strategy are discussed later under "System Performance".

## 4.2 Solar Heating and Cooling System Installation

### 4.2.1 Equipment

The installation of the remaining components of the solar heating and cooling system was accomplished without major difficulty. However, the fact that the house was completed and occupied complicated the installation and the effort became essentially that of a retrofit although the collector area had already been provided. The major difficulty was in the small area alotted the solar equipment, which included a 4550 ℓ (1200 gallon) horizontal cylindrical storage tank, two 1890 ℓ (500 gallon) cool storage tanks, one 310 ℓ (82 gallon) and one 159 ℓ (42 gallon) hot water tanks, an absorption chiller, an auxiliary boiler, and numerous pumps, heat exchangers, and associated piping.

One particularly difficult area was in the small space between the return air duct and supply air plenum chamber. It was necessary to place two large liquid-to-air heat exchangers plus a house distribution blower in a space measuring less than 16.4 m (5 ft) along the air flow path. This caused the problem of placing the blower too near the building's return air inlet and, consequently, producing an unacceptable noise level. This was eventually corrected by the relocation of the building's return air inlet.

#### 4.2.2 Control Instrumentation

The control instrumentation system was developed by a member of the project staff and utilizes a completely solid-state control design. This system has proven to be reliable, relatively inexpensive, and highly versatile in the incorporation of design changes and in providing additional data information on the status of the system.

## 5.0 COLLECTOR OPERATING EXPERIENCE

### 5.1 Summary

During the 20 months of evaluation of the O-I solar collector installed on CSU Solar House III, several problems have been observed. These include:

(1) Excessive leakage of the collector liquid
(2) Consistent breakage of the evacuated glass tubes
(3) Considerable difficulty in obtaining adequate flow through the collector following a boiling episode
(4) Difficulty inserting and maintaining control and data sensors within the liquid flow volume of the evacuated glass tubes
(5) Control problems associated with long response times
(6) No possibility of draining the collector without disassembly
(7) Freezing in the evacuated glass tubes and the solar collector module manifolds
(8) Frequent boiling resulting in additional liquid loss and evacuated glass tube breakage
(9) Large heat capacity resulting in large overnight heat loss

Based on this experience it is the opinion of the authors that this solar collector can be expected to periodically lose most of its contained liquid due to intermittent glass breakage and leakage. Because of the relatively large quantity of liquid contained in this collector (the capacity of the collector alone on CSU Solar House III is about 570 liters), this loss of collector liquid becomes prohibitively expensive unless the liquid is water. Unfortunately, if the liquid used is water, freezing will be experienced. Because of this and other reasons to be discussed, this collector cannot be considered as a viable alternative for residential applications at the present time. It should be pointed out,however, that the use of this collector as an air-heating solar collector could eliminate most of the problems enumerated above and thus constitute a much more practical design.

While CSU Solar House III experienced frequent electric power failures, many of which caused boiling episodes, there were still other boiling episodes which could not be attributed to power failures. Some of these episodes were due to the long response time involved in starting the collector pump in the morning. The recurrent boiling of the collector liquid caused significant losses of collector liquid and also caused sufficient vapor locking in the flow distribution to significantly degrade the overall performance of the system.

### 5.2 Freezing

In the CSU installation freezing in the piping leading to and from the collector array occured on two separate occasions in November 1976. In January 1977, the lower collector array manifold froze and burst the lower manifold pipe in five places. In this case the collector tubes were not in place and the lower manifold pipe failed to drain. Because of these experiences, the collector liquid was changed to a 24 percent ethylene glycol (by volume) aqueous solution. In reference to the original project work plan, it was anticipated that this concentration would be replaced during the cooling season for an all-water system in an effort to evaluate the effects on the performance of the system due to using an ethylene glycol solution rather than water. This plan, however, was never initiated due to the replacement of the O-I collectors with the Chamberlain flat-plate collectors in June 1978.

### 5.3 Boiling

In the initial installation of the O-I collectors on CSU Solar House III, numerous leaks between the evacuated tubes and the collector manifolds required a shut down of the filling process. It was subsequently decided to allow the collector to boil off any remaining liquid, with plans to refill the system at a later date. On the same day, two tubes destroyed themselves due to what is now believed to have been thermal shock. On the following day, approximately 18 additional tubes were destroyed before the entire collector array was covered with protective black plastic.

The destruction of the tubes took two forms. The first occurred at the juncture of the absorber tube and the outer tube, i.e., the "neck". The second type of breakage was at the opposite end of the absorber tube and was caused either by the inner feeder tube or the coiled spring separating the absorber tube from the outer tube. While the absorber tube was destroyed or broken in every case, the outer tube was broken in about half the cases, with no correlation as to how the absorber tube was destroyed.

Similar breakages were also observed during filling of the collector. For example, in November 1976, project staff began filling the collector at 9:00 am. Because the low flow rate implies a total filling time of about 20 minutes, the last tubes in each module (there are 24 tubes in series for each module) continued to heat up until, at 9:15 am, several tubes near the outlet of each manifold had developed significantly higher temperatures and, when the cool water entered the hot tube, the absorber tubes destroyed themselves.

5.4 Glass Durability

It should be stressed that, while the initial collector installation had numerous and continuing glass breakage problems, the replacement evacuated tubes supplied by O-I, at no additional cost to the project, showed marked improvement. In a test of the ability of the tubes to withstand a boil off condition, all tubes, with the exception of one, were apparently undamaged. The single exception developed a leak between the tube and manifold and required subsequent replacement. At a later date in the operation of the system, a failure of a flow meter interrupted the normal flow and caused an inadvertent boil off condition. In this case the pressure relief valves were automatically actuated and the entire collector array suffered no apparent damage.

The replacement evacuated tubes appeared to be a substantial improvement. During the initial filling of the new tubes virtually no leaks developed between the tubes and manifolds, which was a decided improvement over the earlier tube design. In addition the new support between the absorber and outer tubes was a much more positive support and indications were that the design change was particularly useful. The new tubes also had a getter, which provided for direct indication of a loss of vacuum in the evacuated portion of the tube. Unfortunately, once the tube was installed, the getter was covered by the end cap and hidden from view, thus eliminating an easy check of the collector tubes for vacuum.

Finally, the apparent improvement in the consistency of the absorber's selective surface was noted during the installation of the replacement tubes because the older tubes appeared to have a wide variation in coloring, approaching the appearance of a rainbow. However, after the installation of the replacement tubes, some variation of the collector surface was observed but it was later determined that this discoloration does not affect the thermal performance of the collector even though there had been a slight change in the optical properties of each tube (i.e., a slight shift in the absorption versus wavelength curve).

5.5 Time Lag

Because of the high pressure drops and subsequent low flow rates through the solar collector array, there was a time lag between the time the collector liquid entered the first tube of a module until it exited from the last tube of the same module. In normal operation this time lag had a duration of about 8 to 10 minutes, although during filling operation of one collector at a time, the time lag varied from 15 to 20 minutes. Table 1 gives the temperatures of the collector and thermal storage, as well as the collector flow rate with respect to time for one typical start-up. (The collector pump was first energized at 0844 MST, 16 March 1977 data).

It is noteworthy that the collector temperature rose for about eight minutes until the cooler inlet water reached the temperature sensor in the outlet tube. The temperature rise over this time period (early in the morning) was 14.4°C. At higher initial temperatures the rise was slightly less (e.g., 76.9°C to 89.8°C for a rise of 12.9°C). Obviously, such a $\Delta T$ raises questions as to the validity of such a control system for the collector pump. While the temperature of the collector did return to the region of the initial temperature in each case, there was the distinct possibility of boiling the collector before the temperature rise could be halted.

As an example, the boiling point of water in the Fort Collins area is 95°C and the absorption chiller was designed for temperaurature of 80°C. Thus storage on a summer morning could be expected to be no lower than 80°C, less any heat losses overnight (about a 2°C drop). If the collector/storage temperature differential to turn on the collector pump was 7°C, then the collector pump would turn on when the temperature of the collector reached 85°C.

Table 1. Collector Start-Up Temperatures

| Time | Storage Temperature (°C) | Collector Outlet Temperature (°C) | Collector Flow Rate (gpm) |
|---|---|---|---|
| 0844 | 60.5 | 68.0 | -- |
| 0845 | 60.8 | 68.5 | 3.9 |
| 0846 | 61.2 | 69.0 | 3.5 |
| 0847 | 61.6 | 69.3 | 3.5 |
| 0848 | 62.4 | 69.9 | 3.4 |
| 0849 | 62.6 | 74.9 | 3.4 |
| 0850 | 62.3 | 77.5 | 3.1 |
| 0851 | 62.5 | 81.8 | 3.3 |
| 0852 | 62.3 | 82.4 | 3.7 |
| 0853 | 62.1 | 78.7 | 3.9 |
| 0854 | 62.1 | 74.8 | 3.8 |
| 0855 | 62.0 | 72.2 | 4.0 |
| 0856 | 62.2 | 70.7 | 3.9 |
| 0857 | 61.9 | 69.9 | 4.0 |
| 0858 | 61.7 | 69.2 | 4.1 |
| 0859 | 61.5 | 68.9 | 4.2 |
| 0900 | 61.4 | 68.6 | 4.3 |

If the expected temperature rise is 11 or 12°C, the collector temperature will then reach the boiling temperature before the cooler inlet water is available at the last few tubes of each module. Thus the collector begins to boil.

The collector/storage differential could be lowered and the absorption chiller operated at temperatures as low as 75°C so that, in principle, this problem could be reduced. The addition of ethylene glycol would also raise the boiling point at this location (a 25 percent solution has a boiling point of 102°C in the Fort Collins area), which would ease the difficulty even more. Nevertheless, the problem is likely to persist when storage temperature is high, reflecting an abundance of solar radiation and a light cooling load. Because of this potential difficulty, the control instrumentation was modified to provide boil protection by turning on the collector pump whenever the collector temperature reached a preset value (e.g., 75°C), irrespective of the temperature in storage. The modification eliminated a similar freeze protection mode which, with a glycol/water mixture for collector fluid, was no longer needed. Unfortunately this modification was expected to degrade any optimal control strategy.

### 5.6 Collector Fluid Considerations

The 1977 winter heating season was comparatively mild (about 20 percent fewer heating degree days than normal). The combination of lower heating loads and high collector temperatures, easily obtained by the O-I evacuated tube collectors, provided for a high degree of probability that some boiling would occur over the course of a year. This was particularly true in the spring and fall, when heating/cooling loads are minimal and solar radiation on a 45 degree tilt is at a maximum.

Several alternatives exist to counter the undesirable boiling conditions in liquid-heating evacuated tubular collector systems. The collector array can be undersized for the particular building load and storage can be oversized to account for the spring and fall excess energy. However, any oversizing of storage would have to consider the effects on its ability to meet the temperature requirements of the absorption chiller. A multiple storage tank facility could also be utilized to avoid this problem, but only at a cost of greater system complexity.

Another alternative would be the use of covers for the collector to prevent boil off. This again would mean higher costs and greater complexity without any significant improvement in the performance of the solar system.

A simpler and more desirable alternative is to eliminate the problem altogether by eliminating water as a constituent of the collector fluid. This can be done by the use of a low freezing point/high boiling point heat transfer liquid or by redesigning the evacuated tube solar collector to use air as the heat transfer fluid. Liquids exhibiting the necessary characteristics have been detailed in reference 10 and might include butyl benzyl phthalata (freeze -35°C, -31°F; boil 365°C, 689°F) or diethyl o-phthalate (freeze -41°C, -41°F; boil 289°C, 568°F).

Modification of the evacuated tube solar collectors to utilize air as the heat transfer fluid would have numerous advantages. Difficulties of freezing, boiling, corrosion, and costs of exotic liquids would be eliminated. In addition the thermal performance of solar air-heating systems has been shown to be slightly better than water-heating systems using flat-plate collectors. The only potential disadvantage of a solar air-heating collector has been the inability to obtain temperatures high enough to operate solar cooling machines, but this argument is no longer justified for an evacuated tube air-heating collector (see reference 11). It should be pointed out that both Owens-Illinois and General Electric are presently developing commercial solar air-heating evacuated tube solar collectors.

In summary, the use of an air-heating evacuated tube solar collector instead of a liquid-heating design would:

(1) Eliminate freezing problems

(2) Eliminate boiling problems (which are particularly prevalent in evacuated tube collectors)

(3) Eliminate corrosion problems

(4) Reduce damage to building and system due to leakage problems (although air leaks in an air system would degrade performance of the solar system)

(5) Eliminate costs of antifreeze mixtures, corrosion inhibitors, and/or exotic liquids used as the collector heat transfer fluid

(6) Greatly reduce significant storage heat losses without a heavy cost penalty of greatly improved insulation

(7) Greatly reduce the problem of insertion of control and/or data sensors in the evacuated tubes

(8) Eliminate concern for air pockets occurring in the filling and operation of the collector array, thus eliminating the use of complete modules for collection of solar energy

(9) Reduce large pressure drops in the collector array and associated collector/storage heat exchangers.

5.7 Reflectors

The initial design of the O-I evacuated tube solar collector module called for a white reflective surface (as part of the roof structure) to be located directly behind the evacuated tubes. A modification to this design is the use of a shaped specular reflector directly behind and attached to the evacuated tubes. O-I expected the collector module equipped with the reflectors to yield over 25 percent more energy than a similar module using a diffuse reflector.

The installation at CSU Solar House III had the specular reflectors installed with the solar collector upper array but utilized the diffuse white background reflector on the lower array. Results indicated a lower percentage improvement of the specular reflectors over the diffuse surface over the period of one day. Figures 5,6, and 7 show initial raw data plots. on 16 March 1977 (Fig. 5), the array with specular reflectors (upper array denoted by the temperature, $T_{o,u}$) showed approximately 17 percent improvement. Note, however, that from 1200 to 1500 the percent improvement was about 40. On 19 March 1977 (Fig. 6), we see virtually no improvement due to the reflectors.

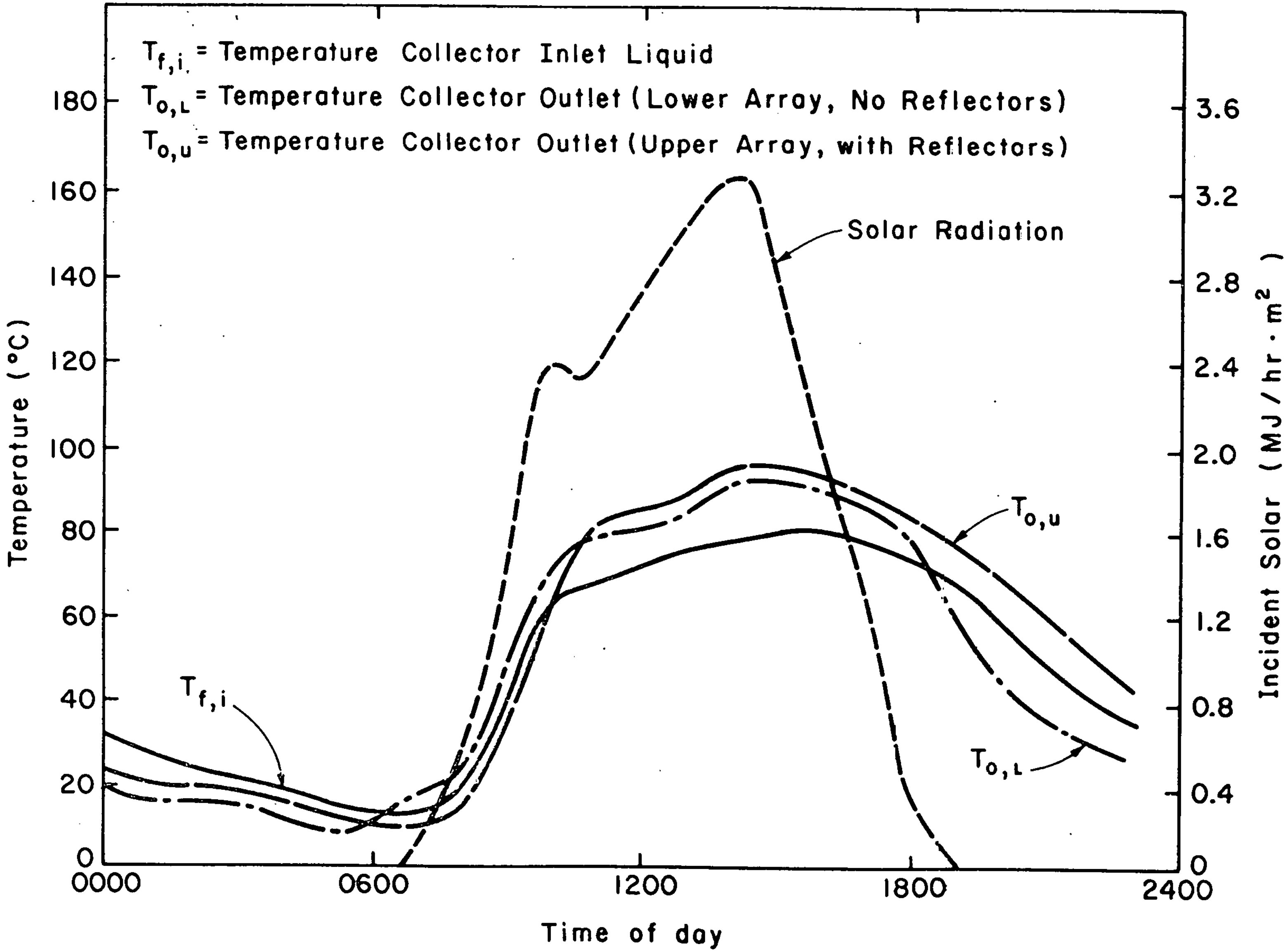


Fig. 5. Comparison of Collector Temperatures With and Without Specular Reflectors, March 16, 1977

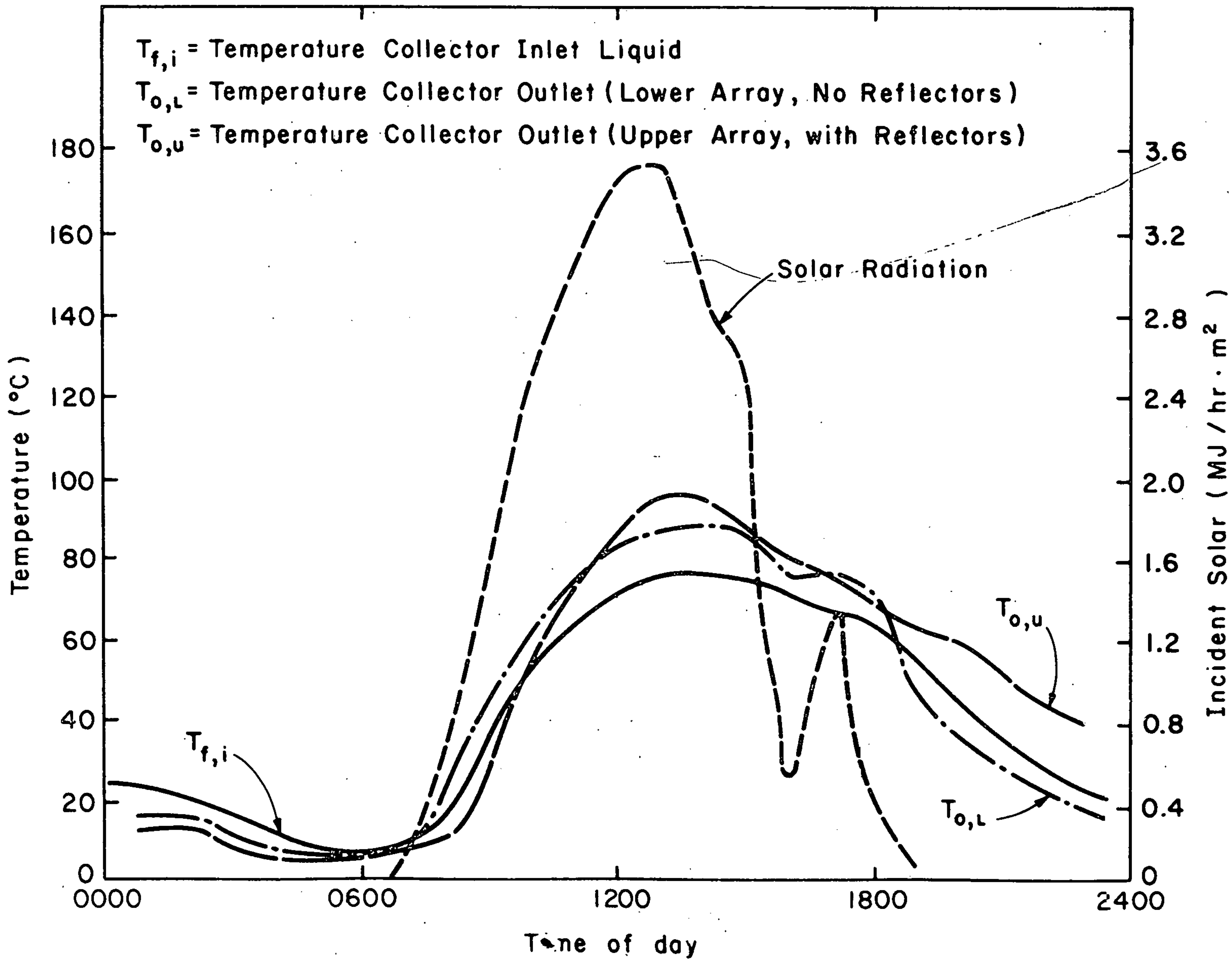


Fig. 6. Comparison of Collector Temperatures With and Without Specular Reflectors, March 19, 1977

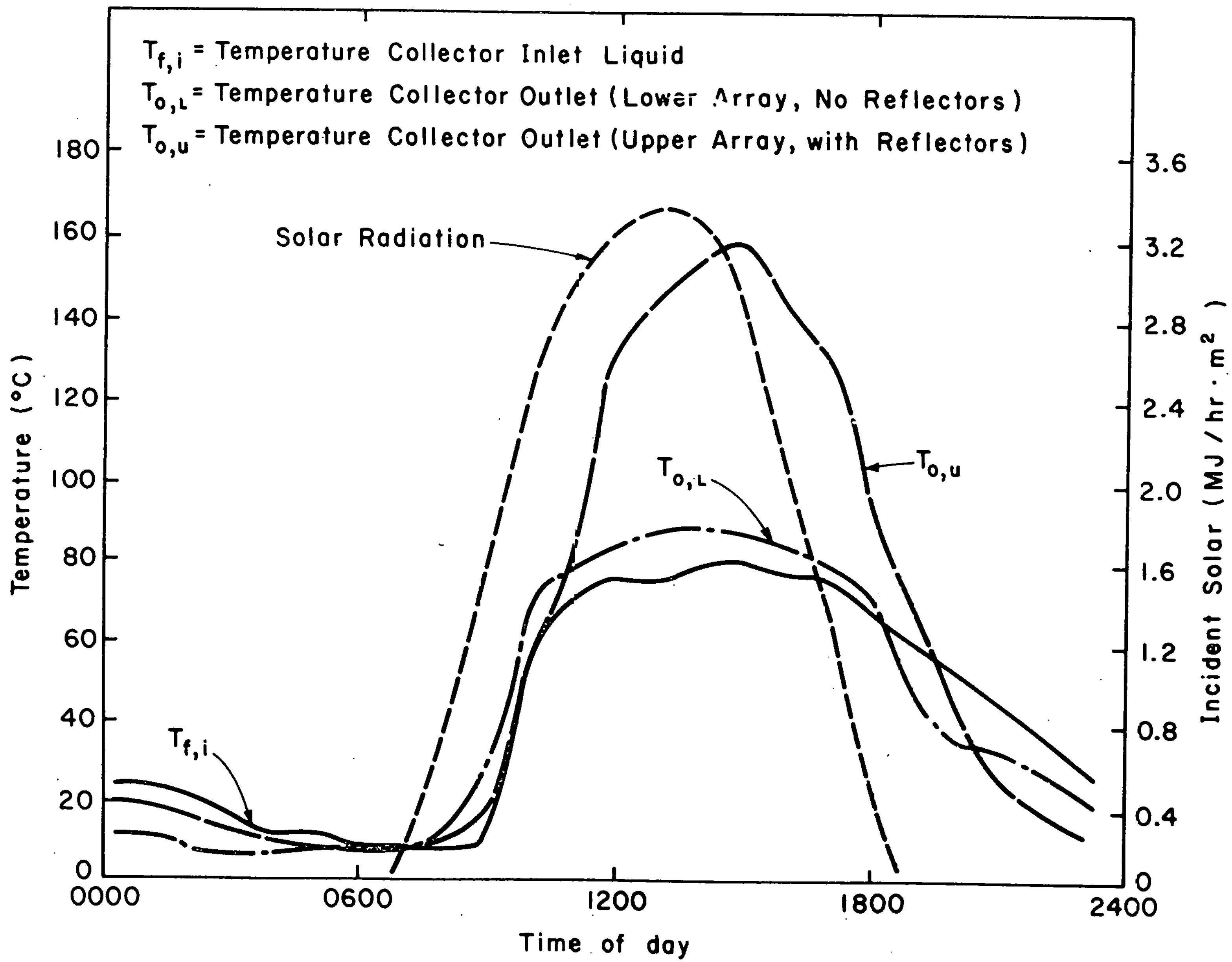


Fig. 7. Comparison of Collector Temperatures With and Without Specular Reflectors, March 22, 1977

5.8 Collector Threshold

According to Owens-Illinois, their evacuated tube solar collector had a threshold of 71.8 W/m$^2$ (25.4 Btu/hr-ft$^2$) beam (i.e., the minimum insolation required to operate the solar collector). This corresponds to a total radiation (beam and diffuse) threshold of about 116 W/m$^2$ (41 Btu/hr-ft$^2$). Experimental data at CSU Solar House III during 1977 indicated a slightly higher threshold of 140 W/m$^2$ (50 Btu/hr-ft$^2$). More detailed results are described below.

5.9 Stagnation Temperature

The highest stagnation temperature recorded to date was 280°C (540°F) at a solar intensity of 769 W/m$^2$ (272 Btu/hr-ft$^2$).

6.0 SYSTEM PERFORMANCE

Table 2 shows the overall system performance during March 1977. Noteworthy results include a high daily average solar collector efficiency, a high fraction of the heat load carried by solar, a low operating threshold for the collector, and a significant heat loss from the thermal storage and domestic hot water units as well as piping, etc.

The solar collector efficiency shown in row 17 is the average daily collector efficiency over the entire month and is obtained by dividing the useful collected solar heat (row 4) by the total monthly solar radiation during collector operations (row 3). Even when the total monthly solar radiation (i.e., sunrise to sunset) is considered (row 2), the collector still collects and delivers to storage 41 percent of the available energy.

One of the reasons for this excellent performance is the low operating threshold, i.e., the minimum solar radiation intensity necessary for useful heat collection. A comparison of rows 2 and 3 indicates that the collector was operating a substantial portion of the time and collected solar energy during the time that 85 percent of the total monthly radiation was available.

The large fraction of the load carried by solar (row 18) can be attributed to a somewhat milder winter (about 20 percent fewer heating degree days) and the excellent performance of the collector. However it should be noted that heat losses from thermal storage and other system components (row 5) constituted over 30 percent of the useful heat (row 4). And, while the bulk of these heat losses were to the conditioned space, thus helping to meet the heating load, they were uncontrolled heat losses.

Table III shows a comparison of the performance of House III with the Löf house data. The Löf house utilizes an obsolete, but still effective,

Table 2. Energy Balances on CSU Solar House III

| Row | Description | | Value |
|---|---|---|---|
| 1 | Month/year | | March 77 |
| 2 | Total monthly solar radiation ($10^6$ kJ/month) | | 23.52 |
| 3 | Total monthly solar radiation during collector operations) ($10^6$ kJ/month | | 19.91 |
| 4 | Useful collected solar heat ($10^6$ kJ/month) | | 9.68 |
| 5 | Heat losses (storage, DHW, piping, etc.)($10^6$ kJ/month) | | 2.97 |
| 6 | Total useful heat delivered by solar ($10^6$ kJ/month) | | 6.71 |
| 7 | Total useful heat delivered by auxiliary ($10^6$ kJ/mon) | | 2.46 |
| 8 | Total heating load ($10^6$ kJ/month) | | 12.14 |
| 9 | Electrical | Collector/exchanger pumps | 0.56 |
| 10 | Energy | Circulating pump | 0.28 |
| 11 | Used | Total solar system pumping power | 0.91 |
| 12 | (parasitic) | Blower | 0.90 |
| 13 | Electrical | Air conditioned space (lights) | 3.96 |
| 14 | Heat Added | Garage | 1.01 |
| 15 | To: | Utility room | 1.62 |
| 16 | Total electricity used in house | | 15.50 |
| 17 | Solar collection efficiency (percent) | | 48.6 |
| 18 | Monthly fraction of load furnished by solar (percent) | | 79.7 |
| 19 | Monthly fraction of total solar radiation delivered as useful heat by solar system (percent) | | 41.1 |

Table 3. Comparison of CSU Solar House III and Löf House

| Row | Description | House III | Löf House |
|---|---|---|---|
| 1 | Month/year | March 77 | March 77 |
| 2 | Total monthly solar radiation | 23.52 | 24.53* |
| 3 | Total monthly solar radiation (during collector operations) | 19.91 | 18.19* |
| 4 | Useful collected solar heat | 9.68 | 7.91* |
| 5 | Heat losses (storage, DHW, piping, etc.) | 2.97 | -- |
| 6 | Total useful heat delivered by solar | 6.71 | 6.79* |
| 7 | Total useful heat delivered by auxiliary | 2.46 | 10.79** |
| 8 | Total heating load | 12.14 | 17.96** |
| 9 | Total solar system power | 0.91 | 1.68* |
| 10 | Solar collector efficiency | 48.6% | 43.45% |
| 11 | Monthly fraction of total solar radiation delivered as useful heat by solar system | 41.1% | 32.2% |

Notes: All energy units are in million kilojoules per month
* Corrected for same collector area
** Different building heating loads

air-heating flat-plate solar collector. The data for the same period with similar radiation levels provide for useful comparisons.

Table 4 shows the system performance for February 1978. These data are distinct from the March 1977 data because of the use of a photocell instead of a collector/storage temperature differential to turn the collector pump on and off. The upper collector array was not used during February because of excessive collector liquid leakage problems. Therefore only half of the installed solar collector area (20.35 $m^2$) was used in February 1978.

This was the coldest February since 1960 (the average air temperature was -2.6°C). While precipitation was only 63 percent of normal, there were an above normal number of days with precipitation. The frequent periods of "upslope" weather conditions which began in mid-January continued during February with a large number of cloudy days, fog days and light wind days. Snow cover was continuous from January 15, but the long period of snow on the ground ended February 23. The heating degree days (base 18.3°C) totaled 581 °C-days. Total precipitation for the month was 0.69 cm (including 12.4 cm of snow). During February there were only four clear days, five partly cloudy days, and 19 cloudy days. Average wind speed was 6.9 kilometers per hour.

From rows 1 and 2 of Table 4 it is clear that the solar collector operated from sunrise to sunset. Because of this, parasitic power requirements were excessive (row 13). The large electric power requirements were due in part to the sizing of the collector and exchanger pumps on the basis of 40.7 $m^2$ of collector area. Because only half the collector area was utilized, it might be considered appropriate to reduce the electrical power used in collecting solar energy to approximately one-half the value given in row 6 of Table 4. However, even with this increased flow rate (approximately 135 percent of the manufacturer's specified design value), some boiling episodes were still experienced. The greater flow rate is therefore desirable, from an operational viewpoint, in reducing the boiling of the collector liquid. The collector pump started operating about two hours earlier and continued to operate about one hour later than it should have. Total hours of useful heat varied from as little as five hours per day up to nine hours per day with an average of eight hours per day. Once the useful heat collection began in the morning, it continued throughout the day. On the average, the collector pump (and heat exchanger pump) ran 11 hours per day, or three hours per day more than they should, so that the electrical power consumption reported in Table 4 was about 27 percent greater than

Table 4. System Performance, February 1978

| Row | Description | | $\frac{\text{Megajoules}}{\text{m}^2\cdot\text{month}}$ |
|---|---|---|---|
| 1 | Total solar radiation on collector area during February | | 361 |
| 2 | | When collector operated | 357 |
| 3 | Useful collected solar heat | | 180 |
| 4 | Heat delivered by auxiliary for house heating | | 835 |
| 5 | Total heating load (solar plus auxiliary) | | 1,015 |
| 6 | Electrical Energy Used | Collector and heat exchanger pumps | 29 |
| 7 | | Circulating pump | 23 |
| 8 | | Air blower | 44 |
| 9 | Electrical heat added to air conditioned space by lights, etc. | | 190 |
| 10 | Total electricity used (row 4 + row 6 + row 7 + row 8 + row 9) | | 1,121 |
| **Row** | **Description** | | **Percentage** |
| 11 | Solar collector efficiency (row 3 ÷ row 2) | | 50.4 |
| 12 | Percent of total solar radiation delivered as useful solar heat (row 3 ÷ row 1) | | 49.9 |
| 13 | Parasitic power requirement (row 6 ÷ row 3) | | 16.1 |

Megajoules per square meter of solar collector used (20.35 $m^2$)

necessary for useful heat collection. Considering this latter factor, a more realistic value for the parasitic power requirements listed in row 6 of Table 4 would be 23 $MJ/m^2$-month.

Assuming an overall average efficiency for electrical generating power plants (including distribution) of 25 percent and a 60 percent average conventional furnace efficiency, it is clear that the parasitic power requirement in terms of fossil fuel consumption is about 50 percent of the useful collected solar heat. With the recommended sizing of collector and heat exchanger pumps, this parasitic power requirement could be reduced but boiling episodes would be more frequent. Thus, if the collector had not been operated with a photocell, the electrical parasitic power requirement would have been 15 percent or, in terms of fossil fuel consumption, about 36 percent of the useful collected solar heat (see equation (7)).

## 7.0 THERMAL STORAGE PERFORMANCE

During the month of February the average temperature of the thermal storage was 43 ± 5°C, that of the domestic hot water solar preheat tank was

39 ± 6°C, and that of the domestic hot water auxiliary (electrically heated) tank was 67 ± 3°C. The corresponding heat loss coefficients of the three water tanks (determined experimentally) were 51.5, 8.15, and 6.97 kJ/hr-°C, respectively. With an average house temperature of 21°C, the total heat lost to the house from these three tanks was 38,400 kJ/day.

Because the storage was designed for a collector area double that used in February, the storage water temperature never exceeded 52°C during the month and averaged only 43°C. Consequently, a maximum of 42 percent of the available storage was used and the average utilization was only 30 percent.

## 8.0 SOLAR COLLECTOR PERFORMANCE

### 8.1 Collector Array Overnight Heat Loss

Because of the O-I solar collector module design, it could not be drained at night. Therefore the liquid in the collector cooled overnight and the heat lost had to be replaced each morning either by solar energy and/or by heat from storage. As mentioned earlier, the O-I solar collector contains a relatively large volume of liquid so that the overnight heat loss is significant. The average overnight heat loss was 2099 kJ/m$^2$ or 58,772 kJ/m$^2$ for the month of February. This represents approximately 33 percent of the useful collected solar heat. Because the collector was operated from sunrise to sunset, almost all of this heat loss came from storage.

### 8.2 Flow Rate Through the Solar Collector

The flow rate through the solar collector was related to the outlet liquid temperature as follows (correlation coefficient = 0.950):

$$q = 0.399 + 0.00403\ T_o \qquad (3)$$

where:

$q$ = Flow rate through the O-I solar collector array, m$^3$/hr

$T_o$ = Outlet liquid temperature from the collector array, °C

Equation (3) explains 90 percent of the observed variance in flow rate. If 25 to 100°C is considered the practical solar collector outlet temperature operating range, it is clear that the flow rate through this solar collector will vary from 0.5 up to 0.8 m$^3$/hr or by a factor of 1.6. For the month of February, the average flow rate through the solar collector was 0.585 m$^3$/hr or 0.0287 m$^3$/hr-m$^2$ of solar collector.

### 8.3 Collector Efficiency and Heat Capacity

The O-I solar collector has a significant heat capacity which directly affects the apparent collector efficiency. It has been shown [5] that:

$$\frac{Q_u}{SA_c} = \eta_o - U_L \left(\frac{T_o - T_a}{S}\right) - cW \left(\frac{\Delta T_o/\Delta t}{S}\right) \tag{4}$$

where

$Q_u$ = Useful heat delivered by the solar collector, kJ/hr

$S$ = Solar insolation rate on a tilted surface, kJ/hr-m$^2$

$A_c$ = Solar collector array area, m$^2$

$\eta_o$ = Instantaneous solar collector efficiency when $T_o - T_a \equiv \Delta T = 0$, dimensionless

$U_L$ = Solar collector heat loss coefficient, kJ/hr-m$^2$-°C

$T_o$ = Solar collector liquid outlet temperature, °C

$T_a$ = Ambient (outdoor) air temperature, °C

$c$ = Weighted average specific heat of the solar collector materials when the collector is filled with water, kJ/kg-°C (for the O-I collector, c = 2.27 kJ/kg-°C)

$W$ = Mass of the solar collector per unit area (with water in the collector), kg/m$^2$ (for the O-I collector, W = 34.1 kg/m$^2$)

$\Delta T$ = $T_o - T_a$

$Q_u/SA_c$ = Apparent solar collector efficiency, dimensionless

$\Delta T_o$ = Change in solar collector outlet liquid temperature during the time interval $\Delta t$, °C

$\Delta t$ = Time interval, hour

From equation (4) it is clear that, while the solar collector is warming up during the first part of the day, $\Delta T_o$ will be positive and consequently the apparent solar collector efficiency will be less than the true collector efficiency for the same value of $\Delta T/S$. Conversely, during the latter part of the day when the solar collector is cooling off, $\Delta T_o$ will be negative and therefore the apparent solar collector efficiency will be greater than the true collector efficiency for the same value of $\Delta T/S$.

In order to demonstrate this behavior, average values and standard deviations of both $Q_u/SA_c$ and $\Delta T/S$ were determined for each hour of the day for several days during the middle of February. These results are given in Table 5. The data in Table 5 are plotted in Fig. 8, where each point is labeled with the hour of the day. $\Delta T_o$ appears to be positive for the hours 8:00 am through 2:00 pm and negative for the hours 2:00 pm through 5:00 pm. The equations for the two straight lines in Figure 8 are for the hours 8:00 am through 2:00 pm (correlation coefficient = -0.972).

Table 5. Average Values and Standard Deviations of $Q_u/SA_c$ and $\Delta T/S$ for Several Days during the Middle of February 1977

| Hour Beginning At | $\Delta T/S$ $°C \cdot m^2 \cdot hr/kJ$ | $Q_u/SA_c$ |
|---|---|---|
| 8 | 0.071 ± 0.038 | 0.347 ± 0.169 |
| 9 | 0.045 ± 0.023 | 0.459 ± 0.081 |
| 10 | 0.032 ± 0.012 | 0.499 ± 0.030 |
| 11 | 0.023 ± 0.010 | 0.495 ± 0.024 |
| 12 | 0.024 ± 0.012 | 0.529 ± 0.036 |
| 13 | 0.029 ± 0.020 | 0.496 ± 0.044 |
| 14 | 0.035 ± 0.012 | 0.549 ± 0.078 |
| 15 | 0.074 ± 0.034 | 0.480 ± 0.065 |
| 16 | 0.182 ± 0.098 | 0.297 ± 0.082 |

$$\frac{Q_u}{SA_c} = 0.599 - 3.43 \frac{\Delta T}{S} \text{ kJ/hr-m}^2\text{-°C} \qquad (5)$$

and for the hours 2:00 pm through 5:00 pm (correlation coefficient = 0.99996):

$$\frac{Q_u}{SA_c} = 0.608 - 1.71 \frac{\Delta T}{S} \text{ kJ/hr-m}^2\text{-°C} \qquad (6)$$

The intercept in both cases is about the same (0.6 ± .04) and corresponds well with the theoretical values, reported by Moan [12] and Beekley [13], of 0.64. It should be noted that this experimental value was obtained under operating conditions which included the effects of dust, snow and ice on the collector tubes.

Between the hours of 9:00 am and 4:00 pm, the apparent collector efficiency was ≥ 0.459. This is clearly important because at this time of year at this location, the vast bulk of the total daily solar radiation on a slope of 45 degrees falls during this time period and this is reflected by the fact that the average collector efficiency for the month was about 50 percent and that about 50 percent of the incident solar radiation was delivered as useful heat by this solar collector.

Perhaps a better measure of the true solar collector efficiency is obtained by subtracting the parasitic power requirements in terms of fossil fuel consumption (i.e., the thermal equivalent of the electrical power used).

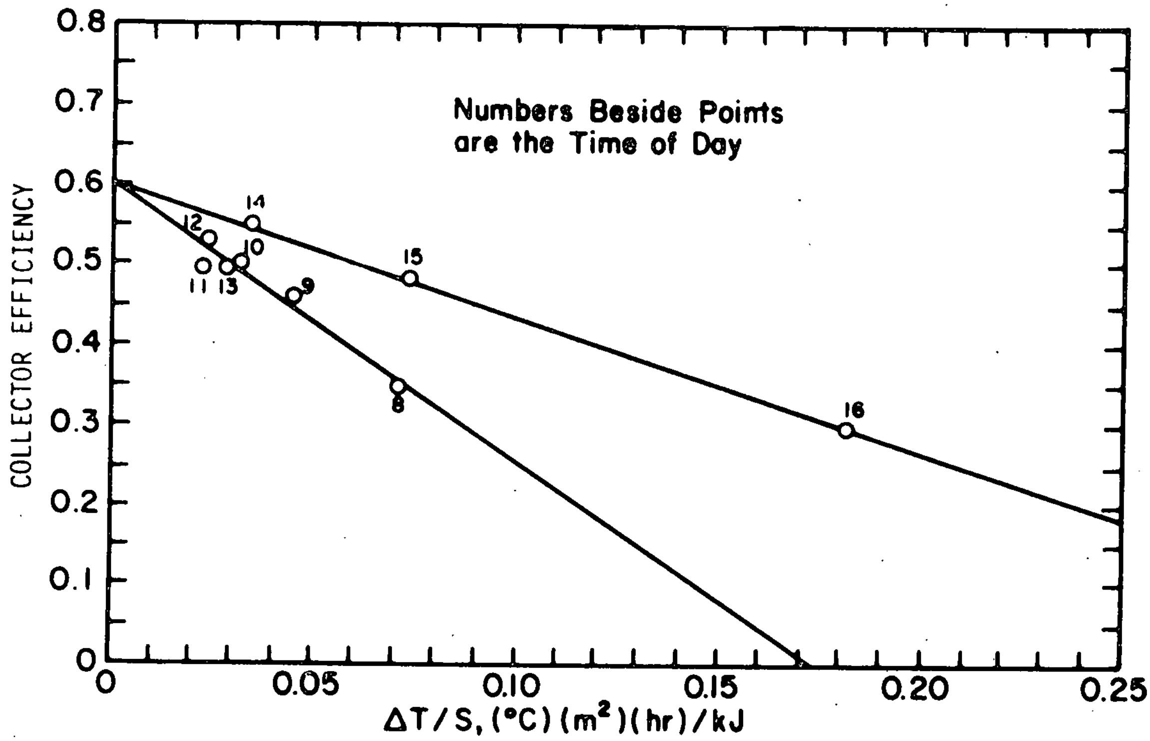


Figure 8. Collector Efficiency as a Function of ΔT/S

This consideration leads to:

$$\eta = \frac{Q_u}{SA_c} - \left(\frac{E_g}{E_e}\right)\left(\frac{E}{SA_c}\right) \tag{7}$$

where

$\eta$ = Solar collector efficiency corrected for parasitic electrical power consumption, dimensionless

$E_g$ = Natural gas furnace efficiency, dimensionless

$E_e$ = Generation, transportation and distribution efficiency of a thermal power plant, dimensionless

$E$ = Parasitic electrical power requirements for the collection of solar energy, kJ/hr

For the 0-I collector installed on CSU Solar House III:

$$\frac{Q_u}{SA_c} = 0.504 \qquad \frac{E}{SA_c} = 0.0777$$

If, for example, $E_g$ = 60% and $E_e$ = 25%, the ratio of $E_g/E_e$ = 2.4, which interestingly enough is roughly the same value of a seasonally averaged coefficient of performance of a heat pump. In other words, a heat pump utilizing electricity from a hydroelectric plant contains the thermal equivalent of 2.4 kJ/hr of 1 kJ/hr(electric).

Utilizing these values in equation (7) we obtain:

$$\eta = 0.504 - (2.4)(.0777) = 0.32$$

## 8.4 Conditions for Useful Solar Heat Collection

The 0-I solar collector started collecting useful solar heat (i.e., $Q_u \geq 0$) at solar radiation intensities as low as 291 kJ/m$^2$-hr and continued collecting useful solar heat until the solar radiation intensity dropped to as low as 70 kJ/m$^2$-hr. On the average, the 0-I solar collector started collecting useful solar heat at solar radiation intensities of 694 kJ/m$^2$-hr and continued to collect useful solar heat until the solar radiation intensity had dropped to 339 kJ/hr-m$^2$.

We may also consider the operating threshold of the collector based on the ratio $(T_o-T_a)/S = \Delta T/S$ where:

$T_o$ = Solar collector outlet liquid temperature, °C

$T_a$ = Outdoor ambient air temperature, °C

$S$ = Solar radiation intensity on a tilted surface (45°), kJ/m$^2$-hr

The 0-I solar collector started collecting useful solar heat when the ratio $\Delta T/S$ was as high as 0.137 °C-m$^2$-hr/kJ. About the middle of February the 0-I solar collector started useful heat collection between 8:00 and 9:00 am and continued until sometime between 4:00 and 5:00 pm.

In considering a realistic operating threshold of the collector (i.e., the lowest solar insolation rate at which the collector will collect useful heat), it is important to consider the additional energy usage expended in operating the collector by the collector pump (i.e., the electrical energy input). Normally such thresholds are computed by assuming zero useful energy collected. For example, equations (5) and (6) may be used to calculate the morning and evening thresholds by setting $Q_u$ equal to zero:

$$0 = 0.599 - 3.43 \frac{\Delta T}{S_m} \quad (\text{kJ/hr-m}^2\text{-°C})$$

$$0 = 0.608 - 1.71 \frac{\Delta T}{S_e} \quad (\text{kJ/hr-m}^2\text{-°C}) \qquad (8)$$

where

$S_m$ = Morning start-up operating threshold

$S_e$ = Evening shut down operating threshold

For a typical $\Delta T$ = 50°C, $S_m$ = 287 kJ/hr-m$^2$ and $S_e$ = 140 kJ/hr-m$^2$. This provides a rough estimate of the solar thresholds.

However, if we require our definition of the operating thresholds $S_m$ and $S_e$ to include a useful heat gain which will account for the electrical energy usage, $Q_u$ can no longer be negative. In fact we may write:

$$\frac{Q_u}{SA_c} \geq \left(\frac{E_g}{E_e}\right)\left(\frac{E}{SA_c}\right) \qquad (9)$$

For the 0-I solar collector on CSU Solar House III this implies that:

$$\frac{Q_u}{SA_c} \geq 0.19$$

for collection of useful heat.

We can also rewrite equations (8) as:

$$0 = 0.599 - 3.43 \frac{\Delta T}{S_m} \quad (\text{kJ/hr-m}^2\text{-°C}) - 2.4 \left(\frac{E}{S_m A_c}\right)$$

$$0 = 0.608 - 1.71 \frac{\Delta T}{S_m} \quad (\text{kJ/hr-m}^2\text{-°C}) - 2.4 \left(\frac{E}{S_e A_c}\right)$$

or

$$S_m = \frac{4.01E}{A_c} + 5.73\,\Delta T \text{ kJ/hr-m}^2\text{-°C}$$

$$S_e = \frac{3.95E}{A_c} + 2.81\,\Delta T \text{ kJ/hr-m}^2\text{-°C} \qquad (10)$$

For $E/A_C$ = 28 MJ/m$^2$-month and $\Delta T$ = 50°C, we obtain (for a rough approximation:

$$S_m = (167 + 287) \text{ kJ/hr-m}^2 = 454 \text{ kJ/hr-m}^2$$

$$S_e = (165 + 140) \text{ kJ/hr-m}^2 = 305 \text{ kJ/hr-m}^2$$

## 9.0 PUBLICATIONS/REFERENCES - OWENS-ILLINOIS


1. Speyer, E., "Solar Energy Collection with Evacuated Tubes". ASME Paper No. 64-WA/SOL, J. of Engring for Power (1964)
2. Duffie, J.A. and Beckman, W.A., Solar Energy Thermal Processes. (New York: John Wiley and Sons) (1974)
3. Ward, D.S., "Design Aspects of CSU Solar House III". Progress report to ERDA, presented at ERDA Contractors' Workshop, Silver Spring, MD, July (1976)
4. Ward, D.S., Uesaki, T., and Löf, G.O.G., "Cooling Subsystem Design in CSU Solar House III". Proc. ISES Conf., Sharing the Sun, Winnipeg, Canada, August (1976)
5. Ward, D.S. and Ward, J.C., "Design Considerations for Residential Solar Heating and Cooling Systems Utilizing Evacuated Tube Solar Collectors". Proc. 1977 Annual ISES Meeting, Orlando, June (1977)
6. Ward, D.S., Duff, W.S., Ward, J.C. and Löf, G.O.G., "Integration of Evacuated Tubular Solar Collectors with Lithium Bromide Absorption Cooling Systems". Proc. ISES Internat'l Cong., New Delhi, January (1978)
7. Ward, D.S., Löf, G.O.G. and Uesaki, T., "Cooling Subsystem Design in CSU Solar House III". Solar Energy, 20, No. 2 (1978)
8. Ward, D.S. and Ward, J.C., "Residential Solar Heating and Cooling Using an Evacuated Tube Solar Collector". Annual progress report to ERDA (COO/2858-5), April (1977)
9. Ward, J.C. and Ward, D.S., "Performance of an Evacuated Tube Solar Collector in a Solar Heating System". Proc. ISES Meeting, Solar Diversification, Denver, August (1978). Submitted to ASME Journal
10. Buchanan, R.M., Majestic, J.R. and Bilau, R., "Toxicological Evaluation of Liquids Proposed for Use in Direct Contact Liquid-Liquid Heat Exchangers for Solar Heated and Cooled Buildings". Topical report prepared for ERDA (COO/2867-1), September (1978)
11. Ward, D.S., "Solar Air-Heating and Cooling Systems for Residential and Light Commercial Applications". Proc. ISES Meeting, Solar Diversification, Denver, August (1978)
12. Moan, K.L., "Evaluation of an All-Glass, Evacuated, Tubular, Non-Focusing, Non-Tracking Solar Collector Array". First annual progress report to DOE (COO/2919-3), December (1977)

13. Beekley, D.C., et al, "Sunpak Solar Collector by Owens-Illinois". Unpublished report, November (1975)

Other reports (not specifically referenced above) but which are important in evaluation of the CSU Solar House III system and O-I evacuated tube solar collectors include:

14. Beekley, D.C. and Mather, G.R., Jr, "Analysis and Experimental Tests of High-Performance Tubular Solar Collectors". Proc. ISES Cong., 1975, Los Angeles, June (1975)

15. Beekley, D.C. and Mather, G.R., Jr., "Analysis and Experimental Tests of a High-Performance Evacuated Tubular Collector". Unpublished report, Owens-Illinois, Inc., (1976)

16. Simon, F., "Solar Collector Performance Evaluation with the NASA-Lewis Solar Simulator - Results for an All-Glass-Evacuated-Tubular Selectively Coated Collector with a Diffuse Reflector". NASA Tech. Mem. TMX-71695, Lewis Research Center, April (1975)

17. Ward, D.S. and Löf, G.O.G., "Design and Construction of a Residential Solar Heating and Cooling System". Solar Energy, 17, No. 1, (1975)

18. Ward, D.S. and Löf, G.O.G., "Design, Construction and Testing of a Residential Solar Heating and Cooling System". Report to the Committee on the Challenges of Modern Society (CCMS)/ERDA, July (1976)

19. Ward, D.S., Karaki, S., Löf, G.O.G. and Winn, C.B., "Sizing, Installation and Operation of Systems". (Washington; Government Printing Office), S/N 003-011-00085-2, October (1977)

20. Ward, D.S., "Residential Solar Heating and Cooling Using Evacuated Tube Solar Collectors (CSU Solar House III)". Presented at ERDA Contractors' Meeting, Virginia, August (1977)

21. Ward, D.S., "Solar Absorption Cooling Feasibility". Proc. ISES Conf., Solar Diversification, Denver, August (1978)

22. Ward, D.S., "Realistic Sizing of Residential Solar Heating and Cooling Systems". Proc. ISES Conf., Solar Diversification, Denver, August (1978)

# SUMMARY OF TECHNICAL RESULTS - CHAMBERLAIN FLAT-PLATE COLLECTOR

## 1.0 CONCEPT

CSU Solar House III is an integrated solar energy system supplying useful heat from a solar collector to a residential-style building for purposes of space and domestic hot water (DHW) heating and space cooling. Water, or a mixture of water and ethylene glycol, is used as the heat transfer fluid which delivers heat collected by single glazed, selective surface, liquid-heating, state-of-the-art flat-plate solar collectors to a thermal storage tank (approximately 4500 liters; 1200 gallons of water). The heated water from the thermal storage is then pumped (1) through a water-to-air heat exchanger to be utilized for space heating, (2) through a double-walled water-to-water heat exchanger to be utilized for DHW heating, and/or (3) to the generator of a 25,320 kJ/hr (2-ton) lithium bromide absorption chiller which in turn provides for space cooling. The LiBr chiller utilizes the solar heat to operate the unit and accomplishes the cooling by removing heat from water in an evaporator and discharging the heat to the ambient. The resulting chilled water is placed in "cool storage" tanks and is pumped through a water-to-air heat exchanger to effect space cooling. An automatic control system provides for all functions of the system operation without requiring any manual manipulation.

CSU Solar House III is fully instrumented for the purpose of unambiguous evaluation of the performance of the integrated solar heating and cooling system. The intent of the project is to integrate various components (collector, absorption chiller, thermal storage, etc.) with specific, known operating characteristics and then determine the system operational performance. Variations in performance due to different control strategies, degradation in component performance, maintenance and installation practices, etc. can also be investigated.

## 2.0 SUMMARY

The present phase of the project commenced on 1 May 1978. Accomplishments to date include a redesign of the solar heating and cooling system, replacement of the previous solar collector array (Owens-Illinois evacuated tube solar collector) with a state-of-the-art flat-plate solar collector (manufactured by Chamberlain Corporation), system modifications for improved operations with the new collector, and approximately 3.5 months of continuous data on the operation of the new system.

Preliminary performance and pertinent operating experience with the Chamberlain solar collector integrated with the CSU Solar House III heating and cooling system have been acquired and a preliminary analysis accomplished. The initial analysis utilized primarily data from the months of July and August (1978) in addition to an analysis of the operation of the solar cooling system which utilizes the Yazaki 25,300 kJ/hr (2-ton) lithium bromide absorption chiller and a cool storage subsystem.

Results of the analysis provide clear indications of the critical importance of temperature differentials between the collector outlet and the absorption chiller generator inlet, the effects of alternative control strategies, the marginal feasibility of cool storage, the devastating effect on system performance of the heat losses from the thermal storage unit, and the importance of parasitic power requirements on the ultimate feasibility of solar absorption cooling.

## 3.0 TECHNICAL ACCOMPLISHMENTS

- The O-I evacuated tube solar collector array was removed in May 1978 and replaced with a Chamberlain single cover, selective surface liquid-heating flat-plate solar collector array with a gross collector area of approximately 71.1 $m^2$ (765 $ft^2$; 30 each, 3 ft by 7 ft modules plus manifolding). The installation of the Chamberlain collector was coordinated with another DOE-sponsored project, "Cost-Effective Ways for Improving the Fabrication and Installation of Solar Energy Heating and Cooling Systems for Residences". Figure 9 shows CSU Solar House III with the Chamberlain solar collector array.
- The existing solar heating and cooling system has been redesigned in order to incorporate the flat-plate solar collector array on CSU Solar House III. Integration of the Chamberlain collector required higher collector flow rates, redesign of the collector loop piping, and several modifications to the control strategy.
- The existing system (including the control instrumentation) was modified for improved integration with the new solar collector array.
- Three months of continuous data (July through September) on the system's performance have been acquired and a preliminary analysis accomplished on two-thirds of these data as part of this report. Two-week summaries of the acquired data, corresponding to different

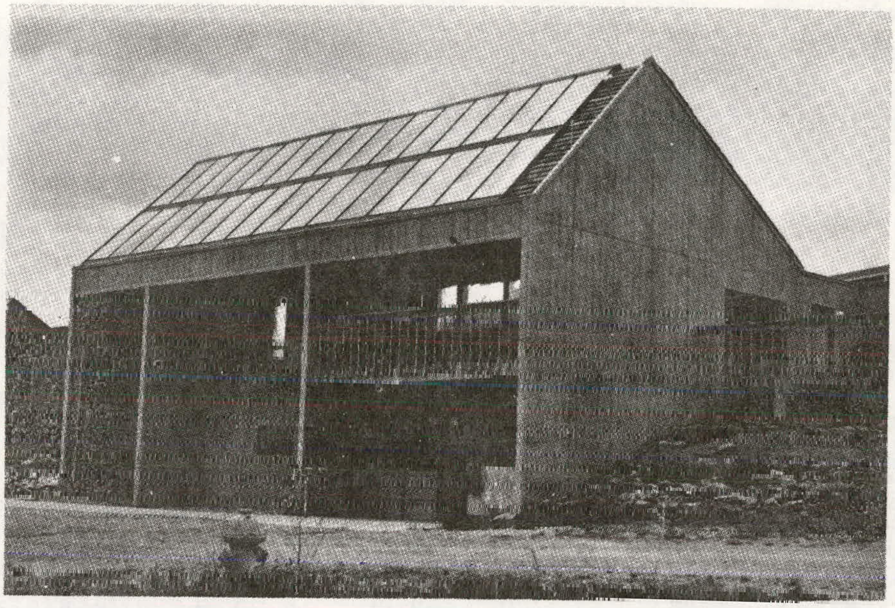

Fig. 9. CSU Solar House III with Chamberlain Solar Collectors

control strategies, are shown in Table 6. A brief analysis of these data is outlined under "System Performance".

- Initial performance data for the Chamberlain collector integrated with the solar cooling system has been reported at the Conference on Solar Heating and Cooling Operational Results (Ref [2]). A summary of this report is included below under "System Performance".
- Operating experience and system performance of both the O-I and Chamberlain collectors have led to generalizations in the utility of operational results of solar heating and cooling systems. These efforts have also been reported at the Solar Heating and Cooling Systems Operational Results Conference (Ref. [3]) and are included in this report as Appendix A. Additional information is included in references [4] and [5] and Appendix B.

## 4.0 TECHNICAL DEVELOPMENTS

The importance of the results of CSU Solar House III during the summer of 1978 is to illustrate the critical importance of minimizing parasitic electrical power requirements in the solar system design, in reducing the minimum temperature to the absorption chiller, and in demonstrating the overriding importance of solar system efficiency as opposed to collector efficiency.

Table 6. Performance of the Solar Cooling System

| Row | Energy Flow (million KJ) | Time Periods July 1-15 | July 16-31 | Aug 1-15 | Aug 16-31 |
|---|---|---|---|---|---|
| 1 | Total solar radiation on collector* | 13.56 | 14.54 | 14.64 | 15.75 |
| 2 | Total solar radiation on collector when collector pump is operating | 7.51 | 9.13 | 9.80 | 8.95 |
| 3 | Total useful heat collected * | 2.21 | 4.36 | 5.00 | 4.78 |
| 4 | Thermal storage heat losses | 1.03 | 0.96 | 0.96 | 0.81 |
| 5 | Piping heat losses ** | 0.17 | 0.33 | 0.38 | 0.38 |
| 6 | Total solar heat delivered to load | 1.00 | 3.07 | 3.66 | 3.59 |
| 7 | Total auxiliary heat delivered to load | 9.19 | 6.82 | 6.41 | 13.06 |
| 8 | Total heat delivered to chiller | 10.19 | 9.89 | 10.07 | 16.65 |
| 9 | Heat removed from chilled water | 4.69 | 4.60 | 5.17 | 7.14 |
| 10 | Heat rejected to cooling tower (row 8 + row 9) | 14.88 | 14.49 | 15.24 | 23.79 |
| 11 | Heat rejected to cooling tower (measured) | 13.75 | 14.97 | 16.36 | 23.45 |
| 12 | Heat gained by cool storage | 0.16 | 0.16 | 0.19 | 0.16 |

*Based on a total absorber area of 53.3$m^2$. The total solar collector area is 58.6 $m^2$. It should be noted that the gross collector area, including manifolds and interconnections between individual modules, is approximately 70 $m^2$.

** Based on energy balances

In design terms this implies:

(1) A minimum of solar collector loop piping and minimal pressure drop through the collector and associated piping (i.e., sufficiently large diameter piping)

(2) Elimination of the heat exchanger between collector and storage (and the heat exchanger pump) which implies a drain-down system or the use of a Direct Contact Liquid-Liquid Heat Exchanger

(3) The use of an exterior hot thermal storage (essential!)

(4) Minimal piping and pressure drops in the circulating loop, which implies a minimal distance between the exterior storage and the absorption chiller

(5) Minimal piping and pressure drops in the cooling tower loop, implying a minimal distance between the exterior cooling tower and the absorption chiller

(6) Elimination of cool storage

(7) The desirability of an air instead of water chiller

(8) The absolutely essential consideration of <u>system</u> efficiency in selecting control strategies, flow rates, etc.

## 5.0 SYSTEM PERFORMANCE

Table 6 is a summary of the system performance during the period 1 July to 31 August 1978. The data are presented for four 15-day periods when control strategy and temperature set points were varied.

During the first two-week period (July 1-15), the control strategy required a minimum temperature for input to the generator of the absorption chiller of 80°C. On 15 July this set point was reduced to 75°C. During the first two-week period the collector boiled frequently, thus reducing the useful heat collection to about half of the useful heat collection of the second two-week period (July 16-31).

During both periods the temperature rise through the collector at the design flow rate was 8°C. Therefore, in order to avoid boiling in the collector when using water, the maximum water storage tank temperature was 87°C (the local boiling point of water at the CSU Solar House III elevation is 95°C). During the first two-week period this limitation allowed a useful operating temperature range of the thermal storage unit of 80 to 87°C, or 7°C. During the second two-week period, however, the effective temperature was increased to 13°C (87-74°C) so that the heat storage capacity was effectively doubled.

Table 7 shows the effects of these storage capacity limitations on the operation of the system for each of the two-week periods. The table is based on hourly data for July 1978, which indicated an average heat collection rate of:

(1) 36,900 kJ/hr for the period 0800-0930

(2) 52,800 kJ/hr for the period 0930-1030

(3) 95,000 kJ/hr for the period 1030-1400

(4) 52,800 kJ/hr for the period 1400-1530

During this same period the average rate of heat input to the chiller was 42,200 kJ/hr, the average heat loss rate from thermal storage was 2740 kJ/hr (corresponding to an overnight temperature decrease in storage of about 1.7°C), and piping heat losses (between the collector and storage) averaged 7.5 percent of the gross heat collection. The collector delivered useful heat from 0800 to 1530 hours. The cooling load was negligible from about 0800 to 0930.

In constructing Table 7, columns 2 and 3 were assumed to be zero after 1230 for the period 1-15 July because of boiling, however heat continued to be supplied to the chiller until 1430. In addition, the numbers in columns 6 and 7 apply to the 1-15 July period only until 1230.

It is evident from column 5 of Table 7 that the doubling of the thermal storage capacity, discussed above, has the effect of doubling the amount of solar energy supplied to the chiller load. Additionally, if a Direct Contact Liquid-Liquid Heat Exchanger [1] were used, the storage temperature could be raised another 8°C because the collector liquid would not boil at 95°C and the maximum storage temperature would be the boiling point of water. This would increase the thermal storage capacity by a factor of 1.6.

At the beginning of the third two-week period (August 1-15), a routine maintenance of the chiller was conducted (a slight loss of vacuum was detected and corrected) and the cooling tower flow rate to the chiller was increased. No changes in control strategy were made during the period August 16-31, but the load increased significantly (see Table 6, row 9).

## 6.0 ANALYSIS

Table 8 summarizes a portion of the data analysis. The effect of the change in control strategy on 15 July is evident in the increases in collector efficiencies (rows 13 and 14) and fractions of solar furnished to load (rows 23 and 25). There is no corresponding decrease in the coefficient of performance of the chiller due to the lower input temperature to the generator.

Table 7. Effect of Collector Boiling on Solar Heat Supplied to Cooling Load

| (1) Time of Day | (2) Storage Heat Collection KJ | (3) Piping Heat Loss * KJ | (4) Storage Heat Loss KJ | (5) Solar Heat Supplied to Cooling Load KJ | (6) Net Heat Gain by Hot Storage KJ | (7) Change in Storage Tank Temp °C | (8) Hot Storage Temp °C | (9) Collector Outlet Temp ** °C | (8) Hot Storage Temp °C | (9) Collector Outlet Temp ** °C |
|---|---|---|---|---|---|---|---|---|---|---|
| At 0800 | | | | | | | 78.3 | 86.1 | 72.2 | 80.0 |
| 0800-0930 | 55,400 | 4,160 | 4,100 | 0 | 47,120 | 2.5 | 80.8 | 88.6 | 74.7 | 82.5 |
| 0930-1030 | 52,800 | 3,960 | 2,740 | 42,000 | 3,850 | 0.2 | 81.0 | 88.8 | 74.9 | 82.7 |
| 1030-1130 | 95,000 | 7,120 | 2,740 | 42,000 | 42,890 | 2.2 | 83.2 | 91.0 | 77.1 | 84.9 |
| 1130-1230 | 95,000 | 7,120 | 2,740 | 42,000 | 42,890 | 2.2 | 85.4 | 93.2 | 79.3 | 87.1 |
| 1230-1330 | 95,000 | 7,120 | 2,740 | 42,000 | 42,890 | 2.2 | 83.2 | Boiling | 81.6 | 89.4 |
| 1330-1400 | 47,500 | 3,560 | 1,370 | 31,000 | 21,444 | 1.1 | 82.1 | | 82.7 | 90.5 |
| 1400-1530 | 79,000 | 5,930 | 4,110 | 63,000 | 5,776 | 0.3 | 81.0 | | 82.9 | 90.7 |
| 1530-1700 | | | 4,110 | 63,000 | -67,418 | -3.6 | 80.5 | Collector | 79.3 | Collector |
| 1700-1915 | | | 6,170 | 95,000 | -101,130 | -5.3 | 80.0 | Cooling | 74.1 | Cooling |
| 1915-0800 | | | 34,980 | | -34,975 | -1.2 | 78.3 | ↓ | 72.4 | ↓ |
| 7/1-7/15 | 298,100 | 22,360 | 65,800 | 220,000 | - 1,150 | 0 | | ↓ | | ↓ |
| 7/16-7/31 | 519,600 | 38,970 | 65,800 | 410,000 | 3,334 | 0 | | ↓ | | ↓ |

* (0.075 x column 2) ** (column 8 + 7.8°C)

Table 8. Evaluation of the Solar Cooling System

| Row | Component Description | Time Period | | | |
|---|---|---|---|---|---|
| | | July 1-15 | July 16-31 | Aug 1-15 | Aug 16-31 |
| 13 | Solar collector efficiency (row 3 ÷ row 2) | 29.4% | 47.8% | 51.0% | 53.4% |
| 14 | Daily collector efficiency (row 3 ÷ row 1) | 16.3% | 30.0% | 34.1% | 30.3% |
| 15 | System efficiency (row 6 ÷ row 2) | 13.3% | 33.6% | 37.4% | 40.1% |
| 16 | Daily system efficiency (row 6 ÷ row 1) | 7.4% | 21.1% | 25.0% | 22.8% |
| 17 | Additional solar for cooling heat losses (million KJ) * | 2.91 | 2.61 | 2.51 | 2.19 |
| 18 | Fraction of useful solar heat lost in storage (row 4 ÷ row 3) | 46.9% | 22.0% | 19.2% | 17.0% |
| 19 | Fraction of cooling by cool storage heat gain (row 12 ÷ row 9) | 3.4% | 3.3% | 3.7% | 2.2% |
| 20 | Average COP of chiller (row 9 ÷ row 8) | 0.46 | 0.47 | 0.51 | 0.43 |
| 21 | Average COP of chiller (solar) | 0.55 | 0.58 | 0.62 | 0.59 |
| 22 | Average COP of chiller (auxiliary) | 0.40 | 0.41 | 0.45 | 0.38 |
| 23 | Fraction of chiller load provided by solar (row 6 ÷ row 8) | 9.8% | 31.1% | 36.3% | 21.5% |
| 24 | Useful cooling by solar ** (million KJ) | -0.48 | 0.82 | 1.31 | 1.31 |
| 25 | Fraction of useful cooling provided by solar *** | Neg | 22.75% | 31.20% | 20.86% |

* Storage heat losses (row 4) multiplied by (1 + 1/COP) where COP is the chiller coefficient of performance (row 21). See Ref. [2].

** Row 24 = (row 6)(row 21) - (row 4)

*** Row 25 - (row 24)/[(row 24) + (row 7)(row 22)]

Rows 15 and 16 give values of two different system efficiencies. These are based on the same logic as the collector efficiencies, i.e., useful heat delivered to load divided by solar radiation during collector operation only and during the full day.

Rows 23 and 25 give two values of the fraction of the load carried by solar. Row 23 is the fraction of solar heat delivered to the chiller divided by the total heat delivered to the chiller. Row 24 calculates the useful cooling by solar but considers as useful cooling only the heat removed by the chiller (when solar heat is being supplied) over and above the cooling necessary to account for thermal storage heat losses into the conditioned space. This is due to the fact that not only do the heat losses to the interior of the conditioned space reduce the availability of solar heat to operate the chiller, they add to the cooling load so that useful heat must be provided at the COP of the chiller just to break even. The effect of the heat losses from thermal storage (and also from collector piping which is exterior to the conditioned space) is to reduce the actual useful cooling by solar to a negative value during the period July 1-15. In the following two-week periods, the actual useful cooling by solar represented only 18.9, 26.2 and 27.4 percent of the useful heat collected (row 24 divided by row 3). Row 25 takes into account these thermal storage heat losses, as well as the lower COP of the auxiliary operation, in calculating the fraction of useful cooling by solar.

It is noteworthy that the amount of heat gain by cool storage is relatively minor in comparison to heat loss by the hot storage. This would suggest that it is better to store "cool" instead of heat. In the four two-week periods, the actual useful solar heat delivered to the chiller represents 45, 70, 73 and 75 percent of the useful heat collected (row 6 divided by row 3). In addition, cool storage heat gains actually reduce the cooling load somewhat, whereas hot storage heat losses contribute to the cooling load. For the periods as a whole, thermal storage heat losses represented 17 percent of the cooling load.

The average COP of the chiller listed in Table 7 does not reflect adequately the operation of the Yazaki chiller. Because of the small heat capacity of the electric auxiliary boiler and the relatively high heat capacity of the generator of the absorption chiller, considerable cycling was observed which greatly reduced the auxiliary/chiller COP. When solar heat from storage was used to operate the chiller (thus eliminating the cycling), typical COP's were 0.52 to 0.65.

## 6.1 Electrical Consumption

Table 9 presents data on the electrical consumption of the solar and auxiliary systems in providing solar cooling to the building. Table 10 presents certain important ratios of energy flows. It should be noted that frequent cycling by the auxiliary boiler caused additional electricity usage (i.e., circulating pump, row 28, and cooling unit power, row 30) during the month of August. The larger power use for collection (row 27) for the period 1-15 July is larger due to frequent boiling of the solar collector. The auxiliary electric use (row 26) is based on the electric meter readings, whereas row 7 of Table 6 is based on flow and temperature measurements (electric meter readings are made weekly). Row 34 considers only that portion of the circulating pump and cooling unit power usage when solar heat is being supplied to the chiller.

In Table 10, row 35 is the ratio of electrical power required to collect useful heat to the total useful heat collected. Row 36 is the ratio of the electric power required to deliver heat to the load (in this case, the absorption chiller) to the total heat delivered to load. Each of these two ratios should ideally be less than 2 percent and, in any case, not greater than 3 percent. The larger values observed are due to a combination of less efficient pumps and high pressure drops in the solar collector.

Row 37 is the ratio of electrical power to operate the cooling unit divided by the actual cooling accomplished. It should be noted that the cooling tower pump was replaced on 1 August, in order to obtain a higher flow rate and thus improve the chiller's performance. The observed resulting chiller improvement, however, had the effect of reducing the overall performance of the cooling subsystem because of the substantially higher parasitic electrical power usage. The inverse of row 37 is an effective COP of the chiller based on cooling achieved and electrical power inputs, i.e., we define $(COP)_{eff}$ by:

$$(COP)_{eff} = \frac{\text{Total useful controlled solar cooling}}{\begin{array}{c}\text{Total electrical parasitic power required}\\ \text{to operate the solar cooling subsystem}\end{array}}$$

These COPs range from 3.33 to 6.12, which cannot be considered competitive with vapor compression systems.

Row 38 is the total cooling (solar and auxiliary) divided by the electrical power usage by the system. When the additional cooling load, caused by solar storage heat loss, is taken into account and the effects of the auxiliary cycling are removed, we obtain the values shown in Row 39 for the

Table 9. Electrical Consumption Data

| Row | Electrical Consumption [million KJ(electric)] | Time Period | | | |
|---|---|---|---|---|---|
| | | July 1-15 | July 16-31 | Aug 1-15 | Aug 16-31 |
| 26 | Auxiliary (electric) boiler (electric meter) | 9.27 | 7.17 | 7.23 | 13.67 |
| 27 | Collector and heat exchanger pumps | 0.184 | 0.154 | 0.121 | 0.113 |
| 28 | Circulating pump | 0.351 | 0.248 | 0.352 | 0.359 |
| 29 | Total system power * | 0.561 | 0.426 | 0.495 | 0.483 |
| 30 | Chilled water, cooling tower pumps/fans | 0.837 | 0.751 | 1.554 | 1.724 |
| 31 | Load pump (from cool storage) | 0.244 | 0.219 | 0.230 | 0.292 |
| 32 | Total cooling subsystem power (row 30 + row 31) | 1.080 | 0.971 | 1.784 | 2.016 |
| 33 | Space distribution system blower | 0.588 | 0.550 | 0.702 | 0.708 |
| 34 | Total solar system power (row 29(s) + row 32 (s)) ** | 0.269 | 0.428 | 0.690 | 0.452 |

* Row 27 + row 28 + control power usage
** (s) implies portion of electrical power for solar only

Table 10. Electrical Consumption Analysis

| Row | Ratio | Time Period | | | |
|---|---|---|---|---|---|
| | | July 1-15 | July 16-31 | Aug 1-15 | Aug 16-31 |
| 35 | Solar heat collection (row 27 ÷ row 3) % | 8.33 | 3.54 | 2.43 | 2.36 |
| 36 | Heat delivery to chiller (row 28 ÷ row 8) % | 3.45 | 2.51 | 3.51 | 2.15 |
| 37 | Cooling (row 30 ÷ row 9) % | 17.82 | 16.33 | 30.06 | 24.14 |
| 38 | System COP * | 2.86 | 3.29 | 2.27 | 2.86 |
| 39 | Solar system COP ** | Neg | 1.92 | 1.90 | 2.90 |

* Row 38 = row 9/(row 29 + row 32)
** Row 39 = row 24/row 34

solar system COP (without auxiliary). The COPs are clearly not competitive with conventional cooling equipment, either.

To emphasize this point, we can evaluate the energy flows in the solar portion of the system during the period 16-31 July (without any use of auxiliary). A total of 4.36 mkJ (million kilojoules) was collected by the solar collector at an energy expense of 0.154 mkJ(electric). Of the useful energy collected, 0.33 mkJ was lost in the collector piping, which is about 7.5 percent and is a fairly conservative value. An additional heat loss by the thermal storage reduced the useful heat total by another 0.96 mkJ so that only 3.07 mkJ was available for solar cooling, which is about 70 percent of the useful heat collected. The circulating pump used an additional 0.056 mkJ(electric) to deliver the solar heat to the absorption chiller (the electrical power used by the circulating pump and chiller subsystem attributed to the auxiliary have been subtracted from the value shown in rows 28 and 32). The chilled water pump and cooling tower pump and fan utilized another 0.169 mkJ(electric) in order to provide 1.78 mkJ of cooling and the solar average COP was 0.58. The load pump used 0.050 mkJ(electric) in cooling the space air with chilled water from solar storage. Thus the solar cooling system provided 1.78 mkJ of space cooling at an energy cost of 0.428 mkJ(electric). However, the thermal storage heat loss of 0.96 mkJ to the conditioned space required 1.66 mkJ of useful solar heat delivered to the chiller in order to remove the additional cooling load. Thus only 0.82 mkJ (1.78-0.96) of useful cooling was accomplished at a cost of 0.428 mkJ of electricity, yielding a system COP of 1.92 (0.82 ÷ 0.428).

Locating the thermal storage exterior to the conditioned space eliminates the additional cooling load penalty of 1.66 mkJ. In addition, the heat losses from an exterior storage would be reduced because the ambient air around the storage would be 3 to 17°C warmer. In the CSU Solar House III system, this would constitute a reduction in thermal storage heat losses of 0.16 mkJ. The result is a solar contribution to the space cooling of 1.87 mkJ. For the electrical requirements of 0.428 mkJ, the effective COP (i.e., cooling divided by electrical usage) is 4.37. Comparing this performance to conventional systems indicates marginal feasibility for the solar system.

Alternatively, a reduction in the electrical power requirements of the solar system could be achieved. Several alternatives exist, including:

(1) Eliminate load pump (and indirectly the cool storage subsystem) and have the chilled water pump deliver cooling directly to the air distribution system

(2) Eliminate the chilled water pump (and load pump) by using an absorption air chiller instead of a water chiller
(3) Reduce circulating power use by optimization of piping
(4) Reduce cooling tower pumping by optimization of piping
(5) Reduce collector pump power by optimization of collector piping
(6) Eliminate exchanger pump (and heat exchanger) by use of a Direct Contact Liquid-Liquid Heat Exchanger [1]

These options are listed with potential energy savings in Table 11. The potential electrical power savings listed in column 3 are based on potential reductions in pump horsepower at CSU Solar House III for each option during the period 16-31 July.

Clearly the installation of an interior hot thermal storage greatly reduces the feasibility of a solar absorption cooling system. With an exterior storage, the selection of a single option provides for COP's in the range of 4 to 18. Because of the greater cost of absorption systems over conventional cooling units, only the case where all options are considered can be viewed as realistic. The significant result is that solar absorption cooling can be considered feasible only if extreme care is taken in the design of the complete solar system (i.e., an absolute minimum of parasitic electrical power is utilized) and the hot thermal storage is located exterior to the conditioned space. Coefficients of performance less than 10 cannot be considered feasible.

## 6.2 System Effects on Collector Efficiency

From 1-7 September, the solar collector flow rate was maintained at 1.5 $m^3$/hr and from 8-15 September, the flow rate was increased to 2.8 $m^3$/hr. Four days were selected for analysis (two in each period) for which the daily total solar radiation (per day) on the collector surface was 25,000 kJ/$m^3$.

It was found that not only was there a saving in parasitic power due to the lower flow rate (power is related to flow rate by $P \alpha (\dot{m})^3$), but also the efficiency of collection for the lower flow rate was slightly higher (about 3 percent in daily collector efficiency).

This is explained in Fig. 10, which is plotted from data obtained. During the cooling season, the majority of the load occurs during the day and thus most of the heat supplied to the chiller from storage is during the collection period. At the lower flow rate the heat collected by storage is supplied directly to the chiller by what constitutes an effective short

Table 11. Potential Electrical Energy Savings

| Row | Option (see ) (list) | Electrical Power Savings mkJ(elect) | Electrical Requirement After Savings | Effective COP* | |
|---|---|---|---|---|---|
| | | | | Interior Storage | Exterior Storage |
| 1 | 1 | 0.07 | 0.36 | 2.3 | 5.2 |
| 2 | 2 (incl. 1) | 0.12 | 0.32 | 2.6 | 5.9 |
| 3 | 3 | 0.04 | 0.39 | 2.1 | 4.8 |
| 4 | 4 | 0.09 | 0.34 | 2.4 | 5.5 |
| 5 | 5 | 0.04 | 0.39 | 2.1 | 4.8 |
| 6 | 6 | 0.04 | 0.39 | 2.1 | 4.8 |
| 7 | 3,4 | 0.14 | 0.30 | 2.8 | 6.3 |
| 8 | 3,4,5 | 0.17 | 0.26 | 3.1 | 7.1 |
| 9 | 1,6 | 0.11 | 0.33 | 2.5 | 5.7 |
| 10 | 1 thru 6 | 0.33 | 0.11 | 7.8 | 17.7 |

*Effective COP = 0.82/electric consumption (for internal storage)
Effective COP = 1.87/electric consumption (for external storage)

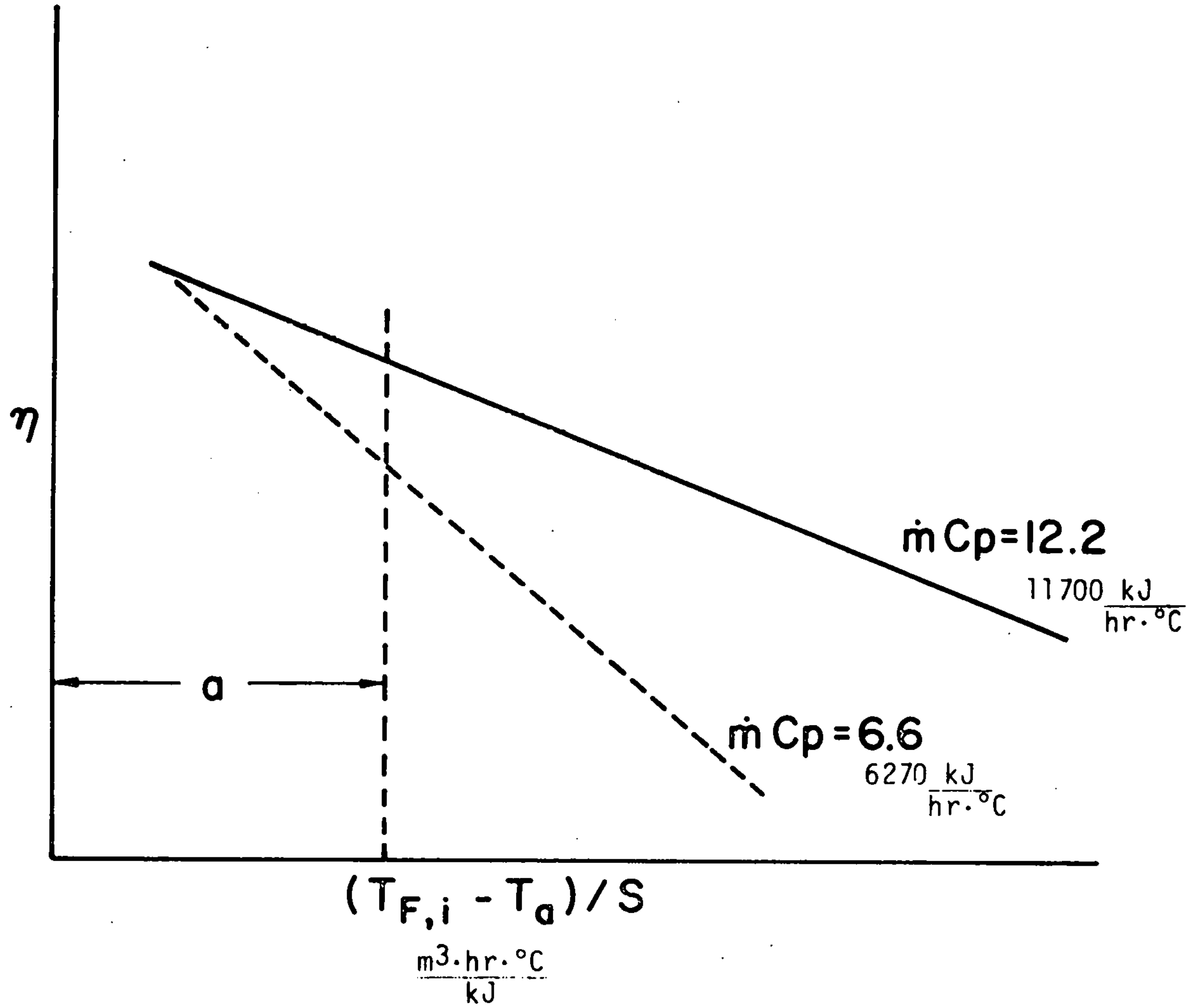


Fig. 10. Collector Efficiency as a Function of Collector Loop Flow Rate

circuit across the top of the storage unit. The return from storage also "short circuits" across the bottom of the tank and a temperature stratification of about 12°C from the top of storage to the bottom is achieved.

At the 1.5 $m^3$/hr flow rate, 6 to 8 hours of collection is taking place in the region indicated in Fig. 10 as (A). However, at the 2.8 $m^3$/hr flow rate, very little stratification is obtained (approximately 4°C) so that most of the collection is taking place in the region indicated by (B) in the figure. This explains the slightly higher efficiencies obtained at the lower flow rate. However, it should be noted that, if this "stratification" were not possible due to constraints of system design such as variations in circulating pump flow rate, the efficiency of collection would reduce to operating in the (B) region of the curve.

A major advantage of operating at lower flow rates (higher collector outlet temperatures) is that the solar supply to chiller can start earlier in the day than with the higher flow rate. This is due to the constraint of the minimum temperature of water that can be supplied to the chiller. Lower flow rates clearly imply a reduction in collector efficiency, but the results of CSU Solar House III clearly indicate substantial improvements in system efficiency, total solar cooling accomplished, and higher COP's (solar cooling divided by electrical parasitic power input) with the lower solar collector flow rate.

The importance of these results lie in the demonstration that system performance (rather than collector efficiency) must be the criterion for selecting flow rates, control strategies, etc. Collector efficiencies are of importance only to the extent of their effect on the efficiency of the overall system.

## 7.0 CONCLUSIONS

The importance of the results of CSU Solar House III during the summer of 1978 is to illustrate the critical importance of minimizing parasitic electrical power requirements in the solar system design, of reducing the minimum temperature to the absorption chiller, and of demonstrating the overriding importance of solar system efficiency (as opposed to collector efficiency).

In design terms this implies:

(1) A minimum of solar collector loop piping and minimal pressure drop through the collector and associated piping (i.e., sufficiently large diameter piping)

(2) Elimination of the heat exchanger between collector and storage (and the heat exchanger pump), which implies a drain-down system or the use of a Direct Contact Liquid-Liquid Heat Exchanger

(3) The use of an exterior hot thermal storage (essential!)

(4) Minimal piping and pressure drops in the circulating loop (implying a minimal distance between the exterior storage and the absorption chiller)

(5) Minimal piping and pressure drops in the cooling tower loop (implying a minimal distance between the exterior cooling tower and the absorption chiller)

(6) Elimination of cool storage

(7) The desirability of an air instead of water chiller

(8) The absolutely essential consideration of system efficiency in selecting control strategies, flow rates, etc.

The results of the performance of the CSU Solar House III system have provided the essential performance data for the generalization of important solar system design considerations. These generalizations have been published in three principal papers [3,4,5] and additional papers are now in preparation (see Appendix D). Because of the extreme importance of the systems design concept, references 3, 4, and 5 are reprinted as Appendices A, B and C.

## 8.0 PUBLICATIONS/REFERENCES - CHAMBERLAIN COLLECTOR

Publications resulting from work under the Chamberlain portion of this contract include:


1. Ward, D.S., "Evaluation of a Residential Solar Heating and Cooling System - CSU Solar House III". Presented at DOE Contractors' Workshop, Washington, D.C., September (1978)

2. Ward, D.S., Ward, J.C. and Oberoi, H.S., "Solar Cooling Performance in CSU Solar House III". Proc. Conf. on Solar Heating and Cooling Systems Oper. Results, Colorado Springs, November (1978)

3. Ward, D.S., "Utilization of Operational Results". Proc. Conf. on Solar Heating and Cooling Systems Oper. Results, Colorado Springs, November (1978)

4. Ward, D.S., "Solar Heating and Cooling System Efficiency as a Function of Design and Installation". Submitted to ASME Journal, September (1978)

5. Ward, D.S., "Operating Thresholds of Solar Collection Systems". Submitted to ASME Journal, September (1978)

A key reference in the literature is:

6. Ward, J.C. and Loss, W.M., "Direct Contact Liquid-Liquid Heat Exchanger: Pilot Plant Results". Proc. 1978 ISES Conf., Solar Diversification, Denver, August (1978)

References pertaining to the Solar House III contract include:

7. Ward, D.S. and Löf, G.O.G., "Design and Construction of a Residential Solar Heating and Cooling System". Solar Energy, 17, No. 1 (1975)

8. Ward, D.S. and Löf, G.O.G., "Design, Construction and Testing of a Residential Solar Heating and Cooling System". Report to ERDA Committee on the Challenges of Modern Society, July (1976)

9. Ward, D.S., "Design Aspects of CSU Solar House III". Progress report to ERDA, presented at Contractors' Workshop, Silver Spring, July (1976)

10. Ward, D.S., Karaki, S., Löf, G.O.G., and Winn, C.B., "Sizing, Installation and Operation of Systems". (Washington; Superintendent of Documents, Government Printing Office), S/N 003-011-0085-2 (1977)

11. Ward, D.S., Löf, G.O.G. and Uesaki, T., "Cooling Subsystem Design in CSU Solar House III". Solar Energy, 20, No. 2 (1978)

12. Ward, D.S., "Solar Absorption Cooling Feasibility". Proc. 1978 ISES Conf., Solar Diversification, Denver, August (1978)

13. Ward, D.S., "Solar Air Heating and Cooling Systems for Residential and Light Commercial Applications". Proc. 1978 ISES Conf., Solar Diversification, Denver, August (1978)

14. Ward, D.S., "Realistic Sizing of Residential Solar Heating and Cooling Systems". Proc. 1978 ISES Conf., Solar Diversification, Denver, August (1978)

APPENDIX A

# UTILIZATION OF OPERATIONAL RESULTS

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# UTILIZATION OF OPERATIONAL RESULTS

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## ABSTRACT

To be of any value the operational results of solar space heating and cooling systems must be viewed as a means of improving the technical and economic advantages of this class of solar energy system. Individual installations should not be viewed as successes or failures, but to the degree that practitioners can learn from them, and ultimately improve their designs, methods, and practices.

Specific attention must be directed toward the concept of daily efficiencies of solar collectors and overall system efficiency before practical conclusions can be reached. In addition, the effects of piping and/or ducting and thermal storage heat losses, installation procedures, choice of control strategies, solar operating thresholds, parasitic power requirements, etc., must also be considered in order to adequately judge the performance of a solar system.

## INTRODUCTION

The proper utilization of the operational results of solar heating and cooling systems, experiments, and demonstrations can provide the essential learning experience necessary for its early, large-scale commercialization. In turn, the increased rate of solar heating and cooling systems commercialization can significantly reduce the deleterious effects of the rapidly decreasing resources of conventional energy.

To be effective, however, the operational results of solar systems must be viewed as a means of improving the technical and economic advantages of solar energy. No reasonable person can question the ability of well-engineered, properly installed solar heating and cooling systems to provide conventional energy savings. It is for the purpose of improving and optimizing the engineering and installation of solar systems that operational results of existing solar installations are presented and discussed. Individual systems should not be viewed as successes or failures, but to the degree that practitioners can learn from them, and ultimately improve their designs and installation procedures.

The purpose of this brief paper is to emphasize the criteria for constructive evaluation of the operational results of solar heating and cooling systems. These criteria include:

1. The thermal performance of complete solar systems as opposed to individual components within the system;
2. The effects on the system thermal performance due to: Heat losses from thermal storage and piping (and/or ducting), installation practices and procedure, and control strategies and sensors; and
3. The effects of the usage of electrical parasitic power to operate and control the solar system.

The economic feasibility of thermal performance improvements as a function of the cost of the improvement and the ultimate economic cost per unit energy usefully provided by the solar system are additional considerations of paramount importance. However, only energy saving potential will be considered in this paper.

## SYSTEM THERMAL PERFORMANCE

### Collector Efficiency

The thermal performance of a solar collector is often based on its efficiency as a function of operating and ambient temperature and the intensity of solar radiation. Typically the collector efficiency is determined by [1]:

$$\eta = F_R(\tau\alpha) - F_R U_L \left(\frac{T_i - T_a}{H_T}\right) \quad (1)$$

where

- $\eta$ = Solar collector instantaneous efficiency, dimensionless
- $F_R$ = Solar collector heat recovery factor, dimensionless
- $(\tau\alpha)$ = Collector transmissivity-absorptivity product, dimensionless
- $U_L$ = Collector heat loss coefficient, Btu/hr·ft$^2$·°F
- $T_i$ = Collector fluid inlet temperature, °F
- $T_a$ = Ambient air temperature, °F
- $H_T$ = Solar insolation on tilted surface of collector, Btu/hr·ft$^2$ of collector

Experimental data on solar collector efficiency provides values for the collector's characteristics, $F_R(\tau\alpha)$ and $F_RU_L$. These two experimentally derived numbers are sufficient to completely characterize the specific collector design and allow for unambiguous comparisons of different collectors when utilized in otherwise identical solar systems.

Unfortunately, the performance of a solar collector (as described by equation 1) does not describe the performance of the collector when integrated with a solar system. Experimental values of collector efficiencies are obtained under idealized conditions, e.g., collector efficiency is normally evaluated within one hour of solar noon (because of heat capacity effects). However, as has previously been noted, solar noon occurs only once a day. Beyond that, it's all downhill for collector efficiency.

System performance testing is therefore essential in evaluating the collector's performance under actual operating conditions. The relevant collector efficiency, which is useful from a practical viewpoint, is the daily collector efficiency. Typically a solar collector with an optimized noontime efficiency (instantaneous efficiency) of 45 to 50 percent will have a daily collector efficiency of 25 to 30 percent. The fact that the collector efficiency is significantly degraded from the value normally quoted is the essential justification for measurement and reporting of system performance.

It should be noted that daily collector efficiency is oft times quoted in two forms. One is given by:

$$\eta'_{daily} = \frac{\text{Useful heat delivered by solar collector}}{\text{Total radiation incident on the solar collector (during the collector operation)}}$$

Thus the radiation incident on the collector prior to the collector pump or blower turning on and later after the pump or blower is turned off, is neglected.

A more reasonable and more useful definition of $\eta_{daily}$ is:

$$\eta_{daily} = \frac{\text{Useful heat delivered by solar collector}}{\text{Total radiation incident on collector}} \tag{2}$$

Solar Threshold

Equation (2) provides for a more realistic evaluation of a collector performance because it incorporates the effects of the operating threshold of the collector, i.e., the minimum value of solar radiation which is necessary before the solar collector can collect useful heat. This operating threshold can be defined by setting ($\eta$) in equation (1) equal to zero. The result is:

$$(H_T)_{OT} = \frac{F_RU_L}{F_R(\tau\alpha)}\,(T_i - T_a) \tag{3}$$

where
$(H_T)_{OT}$ = operating threshold of a collector, Btu/hr·ft²

$(H_T)_{OT}$ is clearly dependent upon the operating temperature of the system (as well as the ambient air temperature), but in effect it is also dependent upon the effect of heat losses in the collector loop piping and/or ducting, on the control strategy, and on the electrical power requirements used in the collection of solar energy.

Such effects on $(H_T)_{OT}$ have been considered by Ward [2] and can be written as:

$$(H_T)_{OT} = \frac{F_RU_L}{F_R(\tau\alpha)}\left\{(T_i - T_a) + \Delta T_c + \frac{Q_L}{A_cF_RU_L} + \frac{E}{A_cF_RU_L}\right\} \tag{4}$$

where
$\Delta T_c$ = Increase in collector operating inlet temperature due to control strategy, °F
$Q_L$ = Heat losses in collector loop piping and/or ducting during collector operations, Btu/hr
$A_c$ = Collector area, ft²
$E$ = Thermal equivalent of electrical energy used in operating solar collector, Btu/hr

The importance of a solar operating threshold as described in equation (4) is that it defines the conditions under which the solar system can collect useful solar energy. The effect of control strategies, collector loop heat losses, and electrical power requirements is to limit the periods of useful heat collection and ultimately to reduce the overall system efficiency.

System Efficiency

In an analogous manner, heat losses and electrical power usage also reduce the overall system efficiency. Ward [3] has shown that:

$$\eta_s = \eta_{daily}\left\{1 - \frac{\overline{Q}_L}{Q_u} - \frac{Q_{SL}}{Q_u} - \frac{\overline{E}}{Q_u}\right\} \tag{5}$$

where
$\eta_s$ = Overall solar system efficiency, dimensionless
$Q_u$ = Useful heat collection by solar collector array, Btu/day
$Q_{SL}$ = Heat losses from thermal storage, Btu/day
$\overline{Q}_L$ = Daily heat losses in collector and system loops piping and/or ducting, Btu/day
$\overline{E}$ = Thermal equivalent of daily electrical energy used in all solar system operations, Btu/day

Equation (5) is based on the same reasoning as equation (4) but in calculating system efficiency, we must also consider the heat losses from the thermal storage unit over a 24 hour period. In addition we should note that there are heat losses from the collector and system loops, both during the operation of the collector and at other times as well; and that electrical power will be required for system operation as well as collector operation.

Sample Results

We can better see the significance of equations (4) and (5) by performing a few sample equations. Let us assume a rather typical solar installation which uses a liquid-heating solar collector with the characteristics:

$F_R(\tau\alpha) = 0.75 \qquad F_RU_L = 0.825\ \text{Btu/hr}\cdot\text{ft}^2\cdot{}^\circ\text{F}$.

These are rather excellent values for a flat-plate solar collector but they will serve to illustrate our point.

Using equation (3) we would expect a solar operating threshold of $(H_T)_{OT} = 1.1\ \text{Btu/hr}\cdot\text{ft}^2\cdot{}^\circ\text{F}\ (T_i - T_a)$. For January we may assume conditions of $T_a = 30°F$ (average daytime temperature) and a thermal storage temperature of 115°F. With a heat exchanger between storage and the collector (with a corresponding temperature difference of 5°F), we obtain $T_i - T_a = 90°F$. Thus $(H_T)_{OT}$ is just less than 100 Btu/hr·ft².

If, however, we utilize equation (4), we obtain a significant variation. The control strategy would typically turn the collector on when the collector/storage temperature differential exceeded 20°F and turn the collector off when it dropped below 5°F. On the average, therefore, we might expect $T_{collector} - T_{storage} = 10°F$, average, or $\Delta T_c = 5°F$ (the heat exchanger ΔT having already been accounted for).

With two inch fiberglass insulation on the collector loop piping, we can except a heat loss of about 2,000 Btu/hr (see Ref. [3]), which is relatively independent of the solar collector area. For the collector and exchanger pumps, the electrical energy usage would be about 1/2 hp, or about 1,270 Btu (electric)/hr for a 500 square foot collector. The thermal equivalent of this energy is obtained by dividing the efficiency of a conventional fuel-fired furnace (e.g., 60 percent) by the efficiency of converting the fuel to electricity in a power plant and distributing this energy to the system (i.e., about 25 percent). Thus:

$E = (2.4)(1{,}270\ \text{Btu (electric)/hr})$

$E = 3{,}048\ \text{Btu (thermal)/hr}$

Under these conditions equation (4) becomes:

$$(H_T)_{OT} = 1.1\ \text{Btu/hr}\cdot\text{ft}^2\ \{\ 90°F + 5°F + \frac{2{,}000\ \text{Btu/hr} + 3{,}048\ \text{Btu/hr}}{(500\ \text{ft}^2)(0.825\ \text{Btu/hr}\cdot\text{ft}^2\cdot°F)}\}$$

or

$$(H_T)_{OT} = 1.1\ \text{Btu/hr}\cdot\text{ft}^2\cdot°F\ \{\ 90°F + 5°F + 4.9°F + 7.4°F\ \} = 118\ \text{Btu/hr}\cdot\text{ft}^2 \qquad (6)$$

This represents an 18 percent increase in the solar operating threshold from the simplified equation (3).

The effects of heat losses and power usage on the system efficiency is even more pronounced. Here the heat loss from the collector loop includes not only the heat loss during collector operations but the heat loss at other times. If we assume that the collector operates only once a day and that all heat in the collector loop (from shut down in the evening until start-up the next morning) is lost overnight, then this additional heat loss is just the heat capacity of the collector loop. For a 500 square foot solar collector array, we would expect a typical installation to have about 250 feet of 1.5 inch pipe in the collector loop and perhaps 100 feet in the system loops. These heat capacities would then approximate 20,000 Btu and 10,000 Btu in the two major solar loops.

It is noteworthy that the collector loop may be exterior to the heated space and that the system loops may be interior. The relative temperature differentials would be $\Delta T_{ext} = T_i - T_a = 125°F - 30°F = 95°F$ and $\Delta T_{int} = T_{storage} - T_{room} = 115°F - 70°F = 45°F$. For a collector array operating six hours per day and the solar storage delivering heat to load for eight hours per day, piping heat losses are:

$$\overline{Q}_L = 32{,}000\ \text{Btu/day} + \delta 16{,}000\ \text{Btu/day}$$

where (δ) indicates interior heat losses.

Storage and domestic hot water preheat tank heat losses are simpler to calculate. For example, R-30 insulation would typically result in a heat loss of 500 Btu/hr from these water tanks or a daily heat loss of $Q_{SL} = \delta 12{,}000$ Btu/day.

Because of the use of electricity in the system loops (delivering solar heat to load), the electrical energy requirements are larger than the 1/2 hp for the collector pumps. We will assume a value of about 1/4 hp for the system, so that the total power requirements are $\overline{E} = 30{,}000$ Btu (thermal)/day.

Finally we consider $Q_u$, the useful heat collection. $Q_u$ is given by:

$$Q_u = A_c\ [\ H_T F_R(\tau\alpha) - F_R U_L\ (T_i - T_a)\ ] \qquad (7)$$

Recognizing that our operating threshold (from equation 6) is 118 Btu/hr·ft², we might consider two values of the average daily solar radiation during the operation of the collector. If $H_T$ = 150 Btu/hr·ft²:

$$Q_u = 500\ \text{ft}^2\ [\ 150\ \text{Btu/hr}\cdot\text{ft}^2\ (0.75) - (0.825\ \text{Btu/hr}\cdot\text{ft}^2\cdot°F)(95°F)\ ]$$

$$Q_u = 17{,}060\ \text{Btu/hr} \qquad \text{or} \qquad Q_u = 102{,}500\ \text{Btu/day}$$

Using these values in equation (5), we obtain:

$$\eta_s = \eta_{daily}\ \{1 - \frac{32{,}000 + \delta 16{,}000}{102{,}500} - \frac{\delta 12{,}000}{102{,}500} - \frac{30{,}000}{102{,}500}$$

$$\eta_s = \eta_{daily}\ \{1 - .312 - \delta .156 - \delta .117 - .293\}$$

$$\eta_s = \eta_{daily}\ \{.395 - .273\delta\} \qquad (8a)$$

This implies that 39.5 percent of the daily collector output is actually delivered as useful heat to the heating load and 27.3 percent of the collector output or 69 percent of the useful heat is delivered to the heating load as uncontrolled heat losses from the system. Had the storage and solar system been entirely exterior to the building, the heat losses from storage and the system would have been greater because of a greater ΔT and the useful heat to load would have constituted a negative 18 percent of the solar collector's output; i.e., the solar system utilized more energy than it provided.

At higher solar insolation levels (e.g., $H_T$ = 250 Btu/hr·ft$^2$), $Q_u$ becomes 330,000 Btu/day and we obtain:

$$\eta_s = \eta_{daily} \{0.812 - .085\delta\} \quad (8b)$$

The effect of heat losses and electrical power usage on system performance is thus evident, particularly for lower average solar insolation rates.

## INSTALLATION EFFECTS

It is noteworthy that specific installation procedures and other factors can further degrade the system efficiency. For example, an improper setting on the control system, such that the temperature differential between the collector and storage was 5°F greater than anticipated in the design would increase the solar operating threshold from 118 Btu/hr·ft$^2$ to 124 Btu/hr·ft$^2$ and decrease the system efficiency (equation 8a) from:

$$\eta_s/\eta_{daily} = .395 - .273\delta$$

to

$$\eta_s/\eta_{daily} = .311 - .311\delta$$

Thus the control "error" constitutes an additional 8 to 12 percent loss of the collector output at the lower insolation rate and the uncontrolled heat losses to the building equal 100 percent of the useful heat collected!

Even more important is the effect of minimal or zero piping insulation on the collector/system loops and/or thermal storage. For example, if R-19 insulation is used on the thermal storage (instead of R-30), $Q_{SL}$ = 18,000 Btu/day (50 percent increase) and

$$\eta_s/\eta_{daily} = .395 - .332\delta \quad [(H_T) = 150 \text{ Btu/hr·ft}^2]$$

or an additional increase in uncontrolled heat losses of about six percent of the useful collector output.

No insulation on the collector loop piping increases the operating heat loss from 2,000 to 6,000 Btu/hr, such that $\bar{Q}_L$ is increased from 32,000 Btu/day to 96,000 Btu/day. The system efficiency/daily collector efficiency ratio is reduced to:

$$\eta_s/\eta_{daily} = -.229 - .273\delta \ !$$

This devastating effect on the system efficiency implies that a negative 23 percent of the useful collected heat is delivered to load and that more conventional energy will be used with a solar system than without.

## CONCLUSIONS

These numbers and similar calculations provide a clear indication that the results of operating experience of solar heating and cooling systems are heavily dependent upon installation procedures, choice of control strategies, collector solar operating thresholds, and parasitic power requirements. The design of systems must therefore consider these factors. In addition, operating results from existing solar systems must be evaluated with these factors in mind if the analyses are to be of practical value.

The previous lack of appreciation by designers on the effects of these factors on system performance indicates that definitive conclusions on the existing systems may require reevaluation. Earlier failures of certain systems designs may, in fact, be due to a lack of optimization of specific factors which are correctable. It is imperative that operational results of solar systems be utilized to improve and optimize the designs of solar heating and cooling systems, and not as a tool to demonstrate or not demonstrate solar feasibility.

## REFERENCES

[1] J.A. Duffie and W.A. Beckman, Solar Energy Thermal Processes (Wiley and Sons; New York), 1974.

[2] D.S. Ward, "Operating Thresholds of Solar Collection Systems", Submitted to ASME Journal, September 1978.

[3] D.S. Ward, "Solar Heating and Cooling System Efficiency as a Function of Design and Installation", Submitted to ASME Journal, September 1978.

APPENDIX B

# SOLAR HEATING AND COOLING SYSTEM EFFICIENCY AS A FUNCTION OF DESIGN AND INSTALLATION

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SEPTEMBER 1978

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# SOLAR HEATING AND COOLING SYSTEM EFFICIENCY AS A FUNCTION OF DESIGN AND INSTALLATION

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## ABSTRACT

The effects of design and installation features on the overall efficiency of solar heating and cooling systems are evaluated. Solar system piping and thermal storage heat losses and parasitic power requirements are quantitatively evaluated by considering different degrees of insulation and design configurations.

It is demonstrated that these system variables have a very strong effect on the technical feasibility of solar heating and/or cooling systems.

## INTRODUCTION

The prediction of system performance on the basis of collector efficiencies does not take into account the very real design and installation variables encountered in the field. Such factors as quality of installation, adequacy of thermal insulation on pipes and storage units, and the adherence of control instrumentation performance to design specifications have enormous effects on the operating performance of a solar system.

In general the effects of these variables and other factors encountered in the field result in degrading the system's performance. In a simple residential solar space heating system, many of the problems are forgiven. For example, any thermal storage heat losses to the interior of the conditioned space help to meet the heating demand, and thus constitute "useful heat delivered to load". An alternative is to simply ignore features such as parasitic power requirements.

Unfortunately, such simplistic viewpoints can no longer be tolerated. In a solar cooling system, the problems threaten the very feasibility of the solar system. Even in heating systems the performance is substantially modified in many cases. The purpose of this paper is to investigate the effects of certain common design and installation features which degrade the solar system's performance.

## SYSTEM EFFICIENCY

The daily efficiency of a solar collector, $\eta_{day}$, is given by Duffie and Beckman [1], i.e.,

$$\eta_{day} = \frac{\Sigma Q_u}{A_c \Sigma HR} \qquad (1)$$

The summations are for the different energy flow rates and the different periods of time for solar collection and solar availability. More explicity, we can write:

$$\eta_{day} = \frac{\overline{Q}_u \Delta t}{A_c \overline{HR} \Delta t'} \tag{2}$$

Equations (1) and (2), unfortunately, do not account for the effects of the system on the operation of the collector. These effects include: Heat losses from the thermal storage unit and domestic hot water (DHW) preheat tank, heat losses from the collector/storage loops and other piping and/or ducting particular to the solar system (both exterior and interior to the conditioned space), and the parasitic electrical power requirements of collecting and distributing to load the solar energy. To account for these losses, we must rewrite equation (2) in the form:

$$\eta_s = \frac{1}{A_c \overline{HR} \Delta t'} \{\overline{Q}_u \Delta t - \overline{Q}_L \Delta t_L - \overline{E} \Delta t_e - \overline{Q}_{SL} \Delta t_d\} \tag{3}$$

Three of the terms in brackets in equation (3) require further elaboration.

The daily average values of heat loss from the piping and/or ducting of the solar system, $\overline{Q}_L \Delta t_L$, must account for heat losses to the exterior and interior of the conditioned (heated and/or cooled) space as well as the leakage of the heat transfer fluid from the pipes and/or ducts. That is:

$$\overline{Q}_L = \overline{Q}_L' + \overline{Q}_L'' \delta (1 + 1/COP) \tag{4}$$

The kronecker delta, $\delta$, is zero for space heating applications because the interior heat losses can be considered as useful heat to the conditioned space. For cooling applications, $\delta = 1$; interior heat losses not only reduce the availability of solar heat for cooling purposes, but also add to the cooling load. The factor (1 + 1/COP) accounts for this fact by requiring useful heat collection delivered to the cooling machine to result in an amount of space cooling to offset the conditioned space's heat gain.

Both $\overline{Q}_L'$ and $\overline{Q}_L''$ can be further detailed, in terms of heat losses during collector and/or system operation, heat losses during periods when the system and collector are not operating, and physical leakage of the heat transfer fluid from the pipes or ducts, during collector operation and otherwise. For simplicity, this is detailed below and under Results.

The third term in the brackets of equation (3), $\overline{E}\Delta t_e$, is the thermal equivalent of the parasitic electrical power requirements of the solar system. These power requirements are essentially the power used by blowers and/or pumps (we can ignore the small electrical usage by the control instrumentation). Therefore:

$$\overline{E} = (\overline{\eta}_f/\overline{\eta}_e)\,\overline{E}_e \tag{5}$$

$(\overline{\eta}_f/\overline{\eta}_e)$ is the ratio of conventional fossil fuel furnace efficiency to the electrical generation and distribution efficiency, and converts the electrical power usage, $\overline{E}_e$, to its thermal equivalent. In this way the solar energy requirements to offset the energy to operate the solar system are compared in the same units as the auxiliary fuel. Alternatively we may replace $(\overline{\eta}_f/\overline{\eta}_e)$ with the seasonal average COP of a conventional cooling unit and/or heat pump heating and cooling system.

The last term in brackets in equation (3) is the thermal storage and DHW preheat tank heat losses. Using reasoning similar to that employed in deriving equation (4), we obtain:

$$\overline{Q}_{SL} = \overline{Q}_{SL}' + \overline{Q}_{SL}''\,\delta\,(1 + 1/COP) \tag{6}$$

With the use of equations (4), (5), and (6), we can quantitatively evaluate equation (3). It is, however, preferable to rewrite equation (3) in the form:

$$\eta_s = \frac{\overline{Q}_u\,\Delta t}{A_c\,\overline{HR}\,\Delta t'}\left[1 - \frac{\overline{Q}_L\Delta t_L}{\overline{Q}_u\Delta t} - \frac{\overline{E}\Delta t_e}{\overline{Q}_u\Delta t} - \frac{\overline{Q}_{SL}\Delta t_d}{\overline{Q}_u\Delta t}\right] \tag{7}$$

The term on the right hand side of equation (7), outside the brackets, is just the daily collector efficiency, $\eta_{day}$. The heat loss and parasitic power terms thus represent percentage degradations of the daily collector efficiency.

The importance of equation (7) cannot be overemphasized. The degradations in performance of a solar system by particular design and installation practices directly affect the potential operating efficiency and, in many cases, directly threaten the technical feasibility of a system design, i.e., $\eta_s/\eta_{day} < 0$. To illustrate this fact it is useful to consider several examples. For completeness we will choose three systems; a liquid heating system, an air heating system, and a liquid heating and cooling system. Details of the three systems are described in Appendix A.

## RESULTS

### Liquid Space and DHW Heating System

The application of Space heating causes $\delta$ in equations (4) and (6) to be zero, so that:

$$\overline{Q}_L = \overline{Q}_L'; \qquad \overline{Q}_{SL} = \overline{Q}_{SL}' \tag{8}$$

Assuming that there is no physical leakage of the collector of system liquid, $Q_L'$ includes heat losses from the collector loop during periods of solar collection, as well as periods when the collector is not operating and, in some cases, losses from the collector/storage heat exchanger when the thermal storage is located exterior to the conditioned space. We will assume that the remainder of the solar system is inside the conditioned space. Thus:

$$\overline{Q}_L'\Delta t_L = (UA)_{L_c} \Delta T_{L_c} \Delta t_L + (UA)_{L_s} \Delta T_{L_s} \Delta t_L \,\xi$$

Heat losses from each loop occur during operation of the collector (during time $\Delta t$) and when the solar collector is not operating, $(\Delta t_d - \Delta t)$, where we have assumed $\Delta t_d = 24$ hours. If the collector operates continuously from start-up in the morning until shut down in the evening, and we assume that all heat in the collector loop is lost to the ambient overnight, then the heat loss during the period, $\Delta t_d - \Delta t$, is just the heat capacity of the collector loop. The same applies for the storage side of the exterior ($\delta$ = 1) heat exchanger loop. Thus:

$$\bar{Q}_L' \Delta t_L = (UA)_{L_c} \Delta T_{L_c} \Delta t + (UA)_{L_s} \Delta T_{L_s} \Delta t \, \xi + (cW)_c \Delta T_{L_c} + (cW)_s \, \xi \, \Delta T_{L_s} \qquad (9)$$

It is advantageous to evaluate equation (9) under three conditions. We will assume the collector and exchanger loops have: (1) 5.08 cm (2 in.) fiberglass insulation; (2) 1.27 cm (1/2 in.) fiberglass insulation; and (3) no insulation. Utilizing the data in Appendix A, we obtain the heat loss values given in Table 1.

Table 1. Liquid Heating System Piping Heat Losses

| | $(UA)_{Lc}$ (kJ/hr·°C) | $\Delta T_{Lc}$ (°C) | $(UA)_{Ls}$ (kJ/hr·°C) | $\Delta T_{Ls}$ (°C) |
|---|---|---|---|---|
| 2" insulation | 30.4 | 56 | 15.4 | 48 |
| 1/2" insulation | 43.4 | 56 | 22.5 | 48 |
| 0 insulation | 95.7 | 56 | 50.6 | 48 |

It is noteworthy that the heavier pipe insulation substantially increases the surface area of the pipe as well as the resistance to heat flow. In addition, $(cW)_c \Delta T_{L_c}$ = 31,200 kJ and $(cW)_s \Delta T_{L_s}$ = 14,900 kJ.

The appearance of the heat capacity terms in equation(9) must be given special consideration. For example it can be argued that the solar heating of the collector in the morning will heat the collector prior to the active operation of the system. Unfortunately, the heating of the collector does not correspond to heating the collector loop. In fact, experience [2] has indicated that 60 to 75 percent of the initial heating of a collector loop comes directly from the thermal storage. With appropriate control methods (the exchanger pump remains off when the collector first turns on), the heating of the collector loop from storage will be greatly reduced. Of course, electrical power is used during this period and no useful heat is delivered to storage. The unavoidable fact is that the overnight heat loss from the collector loop (not including the collector array itself), must be made up by active collection of solar heat.

Using equations (8) and (9) and Table 1, we can now calculate values for $\overline{Q}_L \Delta t_L$. Then knowing $\overline{Q}_u \Delta t$ (see below), we can calculate the percentage degradation, i.e., $\overline{Q}_L \Delta t_L / \overline{Q}_u \Delta t$. These latter values are listed in Table 2.

In considering the parasitic power requirements, we note that $\overline{\eta}_e \approx 0.25$ [3]. If $\overline{\eta}_f = 0.60$, then

$$\overline{\eta}_f / \overline{\eta}_e = 2.4 \tag{10}$$

which, coincidentally, is approximately the seasonal average COP of a commercial heat pump (which might be used as an auxiliary to the solar system).

The electrical energy requirements listed in Appendix A must be considered minimal. A more conservative (and possibly more likely) value would include a slight increase in sizing for each pump. For the two cases, $\overline{E}_e$ = 1,675 kJ(electric)/hr (5/8 hp) to 2,685 kJ/hr (1.0 hp). And, while the collector and exchanger pumps run only a period, $\Delta t$, during the day the circulating and DHW pumps run for longer periods. These time factors are included in the results in Table 2.

For the thermal storage and DHW preheat tank heat losses, we calculate the heat loss coefficient times tank surface area, $(UA)_L$, and then calculate the heat loss on the basis of the temperature difference between the storage medium and the surrounding temperature. In addition we note that storage losses occur 24 hours per day, i.e., $\Delta t_d$ = 24 hrs.

For a thermal storage interior to the building (we assume that the DHW preheat tank is always inside the conditioned space), $\overline{Q}_{SL} = \overline{Q}_{SL}' = 0$. Two other cases of interest are an exterior tank with R-30 insulation ($1/U = 1.47$ hr·m$^2$·°C/kJ; 30 hr·ft$^2$·°F/Btu) and one with R-19 insulation. A fourth alternative is a buried storage tank with R-19 insulation. The results for the four cases are:

$$\overline{Q}_{SL}' = \left\{ \begin{array}{ll} 0 & \text{(interior thermal storage)} \\ 476.6 \text{ kJ/hr} & \text{(exterior R-30 insulation)} \\ 752.7 \text{ kJ/hr} & \text{(exterior R-19 insulation)} \\ 602.1 \text{ kJ/hr} & \text{(buried R-19 insulation)} \end{array} \right\} \quad (11)$$

A noteworthy aspect of these results is the effect of a buried storage unit. Clearly improved insulation is preferred over the cost and difficulty of a buried thermal storage unit. In addition, it should also be noted that Jacobsen [4], San Martin [5] and Ward [2] have reported heat losses where observed values were 150 percent of the calculated values.

In order to complete our evaluation of equation (7), it is necessary to calculate the useful energy collected by a solar collector, $\overline{Q}_u$, using the relationship:

$$\overline{Q}_u = A_c \, [\overline{HR} \, F_R \, (\tau\alpha) - F_R U_L (\overline{T_i - T_a})] \quad (12)$$

We will assume two solar radiation levels, $\overline{HR}$ = 2,300 kJ/hr·m$^2$ and $\overline{HR}$ = 1,700 kJ/hr·m$^2$. For the collectors and system described in Appendix A, the corresponding $\overline{Q}_u$ values are: 40,700 kJ/hr and 14,650 kJ/hr, respectively. Note that $\overline{Q}_u$ is the <u>average</u> solar collection over a day, when the collector is actively collecting heat. We will assume the average time of collection is $\Delta t$ = 6 hrs.

Table 2 lists the results for the liquid space heating system. Discussion is contained in the section on Conclusions (below).

Table 2. Design and Installation Effects on Solar System Efficiencies (Liquid System - Space and DHW Heating)

| | Condition† | $\frac{\bar{Q}_L \Delta t_L}{\bar{Q}_u \Delta t}$ | | $\frac{\bar{E} \Delta t_e}{\bar{Q}_u \Delta t}$ | | $\frac{\bar{Q}_{SL} \Delta t_d}{\bar{Q}_u \Delta t}$ | | $\eta_s/\eta_{day}$ | |
|---|---|---|---|---|---|---|---|---|---|
| Row | $\overline{HR}$ (kJ/hr·m$^2$)= | 2300 | 1700 | 2300 | 1700 | 2300 | 1700 | 2300 | 1700 |
| 1 | 1-A-α | .170 | .471 | .138 | .384 | .000 | .000 | .692 | .145 |
| 2 | 2-A-α | .187 | .521 | .138 | .384 | .000 | .000 | .675 | .095 |
| 3 | 3-A-α | .259 | .721 | .138 | .384 | .000 | .000 | .603 | -.105 |
| 4 | 1-A-β * | .249 | .691 | .138 | .384 | .047 | .130 | .566 | -.205 |
| 5 | 2-A-γ * | .275 | .764 | .138 | .384 | .074 | .206 | .513 | -.354 |
| 6 | 3-A-ε * | .380 | 1.056 | .138 | .384 | .059 | .164 | .423 | -.604 |
| 7 | 1-B-α | .170 | .471 | .218 | .605 | .000 | .000 | .612 | -.076 |
| 8 | 2-B-α | .187 | .521 | .218 | .605 | .000 | .000 | .595 | -.126 |
| 9 | 3-B-α | .259 | .721 | .218 | .605 | .000 | .000 | .523 | -.326 |
| 10 | 2-B-β * | .275 | .764 | .218 | .605 | .047 | .130 | .460 | -.499 |
| 11 | 2-B-γ * | .275 | .764 | .218 | .605 | .074 | .206 | .433 | -.575 |
| 12 | 3-B-ε * | .380 | 1.056 | .218 | .605 | .059 | .164 | .343 | -.825 |
| 13 | 3-B-γ * | .380 | 1.056 | .218 | .605 | .074 | .206 | .328 | -.867 |

† Conditions: * $\xi = 1$

1. 5.08 cm (2") insulation on all system piping
2. 1.27 cm (1/2") insulation on all system piping
3. No insulation on all system piping

A. Minimal power requirements (1,675 kJ(electric)/hr, 5/8 hp)

B. Conservative power requirements (2,685 kJ (electric)/hr, 1 hp)

α. Internal thermal storage

β. External thermal storage with R-30 insulation

γ. External thermal storage with R-19 insulation

ε. Buried thermal storage with R-19 insulation

AIR SPACE AND DHW HEATING SYSTEM

For heating purposes, equation (4) reduces to:

$$\overline{Q}_L = \overline{Q}_L'$$

where $\overline{Q}_L'$ includes heat loss from ducting as well as the physical loss of collector fluid along the ducts. The air leakage in and out of the air heating solar collectors and associated ductwork is an important feature of an air system. One the one hand, ductwork can be sealed to the extent that air leakage from the entire system constitutes less than four percent of the useful heat collected [6].

On the other hand, air leakage in and out of a solar collector array is a major factor (e.g., air flow rates to the collector of 600 CFM may result in an air flow rate from the collector of 1000 CFM). Close and Yusoff [7] have shown in detail the effects of air leakage in and out of the solar collector array. Rather than repeat this excellent discussion, we will not consider this particular aspect, other than to note the results of Close and Yusoff, that air leaks into the collector (in so much as they provide necessary ventilation to the conditioned space) increase the collector efficiency. For our purposes this increased efficiency is already incorporated into the collector characteristics, $F_R(\tau\alpha)$ and $F_RU_L$.

In addition to the four percent air leakage in the ducts, we have the heat losses during collector operation and the heat capacity loss discussed above. Thus:

$$\overline{Q}_L'\Delta t_L = (UA)_{L_C}\Delta T_{L_C}\Delta t + (UA)_{L_S}\Delta T_{L_S}\Delta t\,\xi + (cW)_C\Delta T_{L_C} + (cW)_S\Delta T_{L_S}\,\xi + .04\,\overline{Q}_u\Delta t$$

Air ducts have a poorer heat transfer coefficient and thus higher resistance to heat loss, but have a larger heat transfer area. When both factors are taken into account, the heat loss coefficient times the area for two cases of insulation on the ducts (2.54 cm, 1 in. fiberglass and zero insulation), we obtain Table 3.

Table 3. Air Heating System Ducting Heat Losses

| | $(UA)_{Lc}$ (kJ.hr·°C) | $\Delta T_{Lc}$ (°C) | $(UA)_{Ls}$ (kJ/hr·°C) | $\Delta T_{Ls}$ (°C) |
|---|---|---|---|---|
| 1" insulation | 200 | 30 | 85.4 | 30 |
| No insulation | 1,281 | 30 | 546.0 | 30 |

In addition, $(cW)_c \Delta T_{L_c} = 177$ kJ and $(cW)_s \Delta T_{L_s} = 84$ kJ.

The parasitic power requirements are limited to one blower and one pump, which operate only during collector operations (the auxiliary blower is considered part of the conventional system -- just as in the liquid system). Minimal power requirements total 2,350 kJ/hr (7/8 hp), while the conservative power requirements are 3,350 kJ/hr (1-1/4 hp).

For the thermal storage (exterior), we note the experimental value of Karaki, et al [8], which yields a value of $(UA)_L$ for the pebble-bed storage of 10.2 kJ/hr·°C. If we consider three storage modes, i.e., interior, exterior, and buried (all with R-11 insulation), we obtain:

$$\bar{Q}_{SL} = (UA)_L(T_s - T_a) \approx \begin{Bmatrix} 0 \text{ interior} \\ 460 \text{ kJ/hr exterior} \\ 360 \text{ kJ/hr buried} \end{Bmatrix}$$

The useful heat delivered, $\bar{Q}_u$, is calculated as before. The result for the system in Appendix A is $\bar{Q}_u$ at $\overline{HR} = 2{,}300$ kj/hr·m$^2$ is 55,200 kJ/hr and at $\overline{HR} = 1{,}700$ kJ/hr·m$^2$ is 35,050 kJ/hr. The various calculated values of the components of equation (7) are shown in Table 4.

Table 4. Design and Installation Effects on Solar System Efficiencies (Air System - Space and DHW Heating)

| Row | Condition @ $\overline{HR}$ (kJ/hr·m$^2$)= | $\frac{\overline{Q}_L \Delta t_L}{\overline{Q}_u \Delta t}$ 2300 | 1700 | $\frac{\overline{E} \Delta t_e}{\overline{Q}_u \Delta t}$ 2300 | 1700 | $\frac{\overline{Q}_{SL} \Delta t_d}{\overline{Q}_u \Delta t}$ 2300 | 1700 | $\eta_s/\eta_{day}$ 2300 | 1700 |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1-A-α | .149 | .212 | .102 | .161 | .000 | .000 | .749 | .627 |
| 2 | 1-A-β * | .196 | .285 | .102 | .161 | .033 | .052 | .669 | .502 |
| 3 | 1-A-γ * | .196 | .285 | .102 | .161 | .026 | .041 | .676 | .513 |
| 4 | 2-A-α | .737 | 1.137 | .102 | .161 | .000 | .000 | .161 | -.298 |
| 5 | 2-A-β * | 1.034 | 1.605 | .102 | .161 | .033 | .052 | -.169 | -.818 |
| 6 | 1-B-α | .149 | .212 | .146 | .230 | .000 | .000 | .705 | .559 |
| 7 | 1-B-β * | .196 | .285 | .146 | .230 | .033 | .052 | .625 | .434 |
| 8 | 2-B-α | .737 | 1.137 | .146 | .230 | .000 | .000 | .117 | -.366 |
| 9 | 2-B-γ * | 1.034 | 1.605 | .146 | .230 | .026 | .041 | -.206 | -.875 |
| 10 | 2-B-β * | 1.034 | 1.605 | .146 | .230 | .033 | .052 | -.213 | -.886 |
| 11† | 1-A-α | .309 | .372 | .102 | .161 | .000 | .000 | .589 | .536 |
| 12† | 2-B-β | 1.194 | 1.765 | .146 | .230 | .033 | .052 | -.373 | -1.046 |

†(plus 20% leakage of air from ducts)

@ Condition: *$\xi$ = 1

1. 2.54 cm (1") fiberglass duct insulation on all ducts
2. No duct insulation

A. Minimal power requirements (2,350 kJ (electric)/hr, 7/8 hp)

B. Conservative power requirements (3,350 kj(electric)/hr, 1-1/4 hp)

α. Interior pebble-bed storage (R-11 insulation)

β. Exterior pebble-bed storage (R-11 insulation)

γ. Buried pebble-bed storage (R-11 insulation)

## LIQUID SYSTEM - SPACE COOLING AND DHW HEATING

For the cooling season we must account for the interior heat losses and the different operating and ambient temperatures. We again consider three types of piping insulation: No insulation, 1.27 cm (1/2 in.) insulation, and 5.08 cm (2 in.) insulation. Equation (4) is:

$$\overline{Q}_L = \overline{Q}_L' + \overline{Q}_L'' (1 + 1/COP)$$

where the first term is again given by equation (9), i.e.,

$$\overline{Q}_L'\Delta t_L = (UA)_{L_C}\Delta T_{L_C}\Delta t + (UA)_{L_S}\Delta T_{L_S}\Delta t\,\xi + (cW)_C\Delta T_{L_C} + (cW)_S\xi\Delta T_{L_S}$$

$\overline{Q}_L''$ is the same type of equation but considers the interior piping instead, i.e.:

$$\overline{Q}_L''\Delta t_L = (UA)_{L_I}\Delta T_{L_I}\Delta t_I + (cW)_I\Delta T_{L_I} + (UA)_{L_S}\Delta T_{L_S}\Delta t(1-\xi) + (cW)_S\Delta T_{L_S}(1-\xi)$$

Again, $\xi = 1$ for exterior storage and 0 for interior storage. The results for the design in Appendix A are given in Table 5, where we have assumed that $\Delta t = 8$ hrs and $\Delta t(\text{system}) = 12$ hrs.

Table 5. Liquid Cooling System Piping Heat Losses

| | $\Delta t_L\overline{Q}_L'$ (kJ) | $\overline{Q}_L''\Delta t_L$ (kJ) |
|---|---|---|
| 2" insulation | 52,813 + $\xi$23,386 | 12,732 + $(1-\xi)$27,418 |
| 1/2" insulation | 59,677 + $\xi$26,680 | 14,791 + $(1-\xi)$31,280 |
| No insulation | 87,292 + $\xi$39,718 | 23,005 + $(1-\xi)$46,566 |

The cooling system requires two additonal pumps, one for the cooling tower and a chilled water pump, if a water chiller is used. The "minimal" and "conservative" parasitic power requirement are now 2,014 kJ/hr (3/4 hp) and 4,028 kJ/hr (1.5 hp), respectively. The times of operation are again different, so that we obtain the results:

$$\bar{E}_e \Delta t_e = \begin{cases} 20{,}140 \text{ kJ minimal power requirements} \\ 40{,}280 \text{ kJ conservative power requirements} \end{cases} \quad (13)$$

For the thermal storage we will consider four cases: Interior with R-30 insulation and R-19 insulation and exterior with R-30 insulation and R-19 insulation. In all cases we assume an interior DHW preheat tank with heat losses of about 200 kJ/hr. The results are:

$$\bar{Q}_{SL}' = \begin{cases} \xi \; 572 \text{ kJ/hr} & \text{R-30 insulation} \\ \xi \; 903 \text{ kJ/hr} & \text{R-19 insulation} \end{cases}$$

$$\bar{Q}_{SL}'' = \begin{cases} [200 + (1-\xi)620] \text{ kJ/hr} & \text{R-30 insulation} \\ [200 + (1-\xi)979] \text{ kJ/hr} & \text{R-19 insulation} \end{cases} \quad (14)$$

To complete our evaluation, we again use equation (12) to calculate $\bar{Q}_u$. For the cooling season, we will assume $\overline{HR}$ values of 3,500 kJ/hr·m$^2$ (308 Btu/hr·ft$^2$) and 2,800 kJ/hr·m$^2$ (246 Btu/hr·ft$^2$). The results are $\bar{Q}_u$ = 81,200 kJ/hr and 50,800 kJ/hr, respectively.

Using equations (4), (5), (6), (7), (10), (13), and (14), and Table 5, we obtain the results shown in Table 6.

## CONCLUSIONS

A brief analysis of Tables 2, 4, and 6 leads to several important conclusions. For example, from Table 2, we note that a simple lack of collector piping insulation reduces the useful heat collection by 8.9 percent and 25 percent for the two assumed values of insolation (rows 1 and 3). When this loss is combined with the larger parasitic power requirements, the combined reductions of useful heat collection are 16.9 percent and 47.1 percent, respectively (rows 1 and 9). The additional parasitic power requirement (3/8 hp) alone reduces the useful heat collection by eight percent for the higher insolation rate (rows 1 and 7). Burying the

Table 6. Design and Installation Effects on Solar System Efficiencies (Liquid System - Space Cooling and DHW Heating)

| Condition @ | | $\frac{\overline{Q}_L \Delta t_L}{\overline{Q}_u \Delta t}$ | | $\frac{\overline{E} \Delta t_e}{\overline{Q}_u \Delta t}$ | | $\frac{\overline{Q}_{SL} \Delta t_d}{\overline{Q}_u \Delta t}$ | | $\eta_s/\eta_{day}$ | |
|---|---|---|---|---|---|---|---|---|---|
| Row | $\overline{HR}$ (kJ/hr·m$^2$)= | 3500 | 2800 | 3500 | 2800 | 3500 | 2800 | 3500 | 2800 |
| 1 | 1-A-α * | .170 | .271 | .074 | .119 | .041 | .065 | .715 | .545 |
| 2 | 2-A-α * | .194 | .310 | .074 | .119 | .041 | .065 | .691 | .506 |
| 3 | 3-A-α * | .300 | .463 | .074 | .119 | .041 | .065 | .585 | .353 |
| 4 | 1-A-β * | .170 | .271 | .074 | .119 | .053 | .085 | .703 | .525 |
| 5 | 2-A-γ | .281 | .449 | .074 | .119 | .081 | .129 | .564 | .303 |
| 6 | 3-A-ε | .420 | .671 | .074 | .119 | .116 | .186 | .390 | .024 |
| 7 | 1-B-α * | .170 | .271 | .154 | .246 | .041 | .065 | .635 | .418 |
| 8 | 2-B-α * | .194 | .310 | .154 | .246 | .041 | .065 | .611 | .379 |
| 9 | 3-B-α * | .300 | .463 | .154 | .246 | .041 | .065 | .505 | .226 |
| 10 | 2-B-β * | .194 | .310 | .154 | .246 | .053 | .085 | .599 | .359 |
| 11 | 1-B-γ | .246 | .393 | .154 | .246 | .081 | .129 | .519 | .232 |
| 12 | 3-B-γ | .420 | .671 | .154 | .246 | .081 | .129 | .345 | -.046 |
| 13 | 3-B-ε | .420 | .671 | .154 | .246 | .116 | .186 | .310 | -.103 |

@ Conditions: $*\xi = 1$

1. 5.08 cm (2") insulation of all system piping
2. 1.27 cm (1/2") insulation of all system piping
3. No insulation on system piping

A. Minimal power requirements (2,014 kJ(electric)/hr, 3/4 hp)

B. Conservative power requirements (4,028 kJ(electric)/hr, 1.5 hp)

α. Exterior storage with R-30 insulation

β. Exterior storage with R-19 insulation

γ. Interior storage with R-30 insulation

ε. Interior storage with R-19 insulation thermal storage tank results in reductions of useful heat collections of 5.9 percent and 16.4 percent (respectively), from that of an interior tank (rows 1 and 6). Note also the extra losses (rows 1 and 4) when the storage is taken out of the conditioned space (with R-30 insulation). This factor becomes important when a space cooling capability is added to the solar system.

It is particularly noteworthy that it is not possible to collect useful heat at a solar insolation of $\overline{HR}$ = 1,700 kJ/hr·m$^2$ (150 Btu/hr·ft$^2$) average daily solar radiation for almost all variations of the liquid solar system. The calculation of the minimum solar insolation rate on an instantaneous basis for the various configurations is discussed in more detail by Ward [9].

From Table 4, it is readily apparent that the lack of duct insulation is much more disastrous than for the liquid heating system. From rows 1 and 4 and rows 2 and 5, we see percentage decreases of useful collected energy of 58.8 percent and 83.8 percent, respectively, at the higher insolation rate. On the other hand, additional parasitic power requirements cause only 4.4 percent and 6.9 percent reductions in the useful heat collections for the two insolation rates.

Another feature of the air system is the substantial 16 percent reduction in useful heat collection due to poorly installed and leaky ducts. While the air system has the advantage of very low collector loop heat capacity losses, poor installation and design practices help to nullify this advantage. Finally, the location of the pebble-bed storage causes relatively small reductions in useful heat collection.

A comparison of Tables 2 and 4 suggest that the air and liquid systems have comparable efficiencies at the higher insolation level for the conditions of good design and installation. However, at the lower insolation rates the air system is substantially better. Even with leaky ducts the air system performs better at lower insolation rates, but rapidly approaches the point where the system is only marginally feasible.

Because of the disappointing results of the liquid system, it is worth considering what improvements could be obtained from substantial design changes. For example, if the daily average insolation rate (during collector operations of 6 hours/day) is 1,700 kJ/hr·m$^2$, our 1-A-$\alpha$ design indicates that the system losses account for 85.5 percent of the useful heat collected. We can consider two ways of reducing these losses.

First, let us use the collectory array as the south wall (sloped at the same angle), where the manifolds and all system piping heat losses go toward meeting the heating load. In this case our only losses are due to parasitic power requirements. Thus our system losses are only 38.4 percent of the useful heat collected. It is noteworthy, however, that 62.2 percent of the useful heat collected is delivered to the load as uncontrolled heat losses. Thus the system effectively delivers 37.8 percent of the useful heat collected in the form of an active heating system and 62.2 percent as a passive system. Because of the parasitic power, the active system barely breaks even and it would be preferable to use a totally passive system.

To reduce the effects of the collector loop's heat capacity heat loss, the liquid system could utilize a drain down system. This would eliminate the exchanger pump, but the additional power required by the collector to lift the collector liquid to the top of the collector would offset this gain. However, the reduction of the collector loop's heat capacity heat losses would be on the order of 12.8 percent and 35.5 percent, respectively, for the two solar insolation rates. This gives new values of $\eta_s/\eta_{day}$ of .820 and 0.50, respectively. Provided that the system drains down properly and corrosion is not a factor, this would appear to be a better design and is clearly more competitive with the air system.

Another possibility of reducing the collector loop heat losses is to reduce the size of the piping. This reduces the heat loss during collector operations, due to smaller surface area, as well as the overnight heat losses (due to smaller volume). Unfortunately, it increases the parasitic power requirements. An optimum pipe sizing therefore should take into account both considerations.

In Table 6 we see again the importance of good insulation on the piping and, to a lesser extent, on the storage unit. In addition we can compare Tables 2 and 6 for the case of an integrated solar heating and cooling system. It is risky to make absolute comparisons from this data because of the dependence on the assumed solar insolation rate. However, the location of the thermal storage unit may be critical. For a well insulated (R-30) tank, an interior tank enjoys a five to thirteen percent improvement during the winter heating season, but suffers during the summer cooling season and causes a reduction in performance of four to six and one-half percent. Again, the optimization for a particular case must consider both aspects.

It becomes apparent that insulation and parasitic power are critical factors in the liquid system, and that tight and well insulated ducts are equally important in air systems. This basic assessment is not entirely new, but the quantitative basis and the extent to which these factors are important has not been previously addressed in an adequate fashion. Equation (7) and the supporting criterion, therefore, assume substantial importance in the design and installation of solar heating and cooling systems. In addition, any adequate assessment of operating systems must consider the equation as well if the assessment is to be of any value.

Finally, when the heat losses and parasitic power requirements are taken into consideration, we note that $\eta_s/\eta_{day}$ is relatively sensitive to the collector efficiency. For the same flow rates, pipe sizes, and thermal storage size, the effect of an improved collector is to increase $\overline{Q}_u$ without any changes in the loss terms. Thus the percentage losses are reduced. For example, if the collector had the characteristics $F_R(\tau\alpha) = 0.80$ and $F_R U_L$ = 16.35 kJ/hr·m$^2$·°C (0.80 Btu/hr·ft$^2$·°F), the 1-A-$\alpha$ design of the liquid collector (at a daily average solar insolation rate of 1,700 kJ/hr·m$^2$) would result in a value of $\eta_s/\eta_{day}$ = .603; a substantial improvement over .145. Collector efficiency therefore becomes very important, a significant result of the use of equation (7).

## APPENDIX A -- EXAMPLE SYSTEM DESIGNS

Liquid System (Figures 1 and 2)

Solar collector

$A_c = 60\ m^2$ ($645.6\ ft^2$)

$F_R(\tau\alpha) = 0.724$

$F_R U_L = 19.35\ kJ/hr \cdot m^2 \cdot {}^\circ C$ ($0.947\ Btu/hr \cdot ft^2 \cdot {}^\circ F$)

Flow rate = 50 - 65 $kg/hr \cdot m^2$ of collector (0.02 - 0.025 $gpm/ft^2$)

$\dot{m}$ = 3,640 kg/hr (15 gpm)

$C_p$ (water-glycol) = 3.77 kJ/kg·°C (.9 Btu/lb·°F)

$\rho$ (water-glycol) = 1070 $kg/m^3$ (8.92 lbs/gal)

Exchanger (collector/storage)

e.g., Young [10], Tube-in-shell, Model 603-DY

$\Delta p$ (collector side) = 46,200 $nt/m^2$ (6.7 psi)

$\Delta p$ (storage side) = 2,800 $nt/m^2$ (0.4 psi)

LMTD = 6°C (10.8°F)

$\dot{m}$ (storage side of exchanger) = 6,810 kg/hr (30 gpm)

$C_p$ (storage liquid-water) = 4.19 kJ/hr·°C (1 Btu/lb·°F)

$\rho$ (storage liquid) = 1,000 $kg/m^3$ (8.34 lbs/gal)

Storage Capacity

60 - 80 liters of water/$m^2$ of collector (1.47 - 1.96 gal/$ft^2$)

4,000 liters (1,057 gallons)

4,000 kilograms of water (8,820 lbs)

Chiller - Lithium bromide absorption cooling unit

Capacity = 38,000 kJ/hr (3-tons)

COP = 0.60

Minimum temperature to generator = 80°C (176°F)

Other Temperatures

Minimum temperature to heating load = 40°C (104°F)

Average storage temperature - heating = 50°C (122°F)

Average storage temperature - cooling = 85°C (185°F)

Average 24 hour ambient temperature - heating = 0°C (32°F)

Average 24 hour ambient temperature - cooling = 25°C (77°F)

Average daytime ambient temperature - heating = 5°C (41°F)

Average daytime ambient temperature - cooling = 30°C (86°F)

Pumps

| | | |
|---|---|---|
| Collector | 670 kJ/hr | (1/4 hr) |
| Exchanger | 335 kJ/hr | (1/8 hp) |
| Circulating | 335 kJ/hr | (1/8 hp) |
| DHW (2) | 335 kJ/hr | (1/8 hp) |
| Cooling tower | 335 kJ/hr | (1/8 hp) |
| Chilled water | 670 kJ/hr | (1/4 hp) |

Piping Lengths

Collector loop (heat loss purposes)

10 meter of 5.08 cm (2") manifolds

10 meter of 1.91 cm (3/4") interconnections

10 meter of 3.81 cm (1-1/2") run (exterior to conditioned space)

10 meter of 3.81 cm (1-1/2") run (interior $\xi$ to conditioned space)

Exchanger loop (interior $\xi$)

5 meter of 3.81 cm (1-1/2") run

Circulating loop (interior)

10 meter of 2.54 cm (1") run

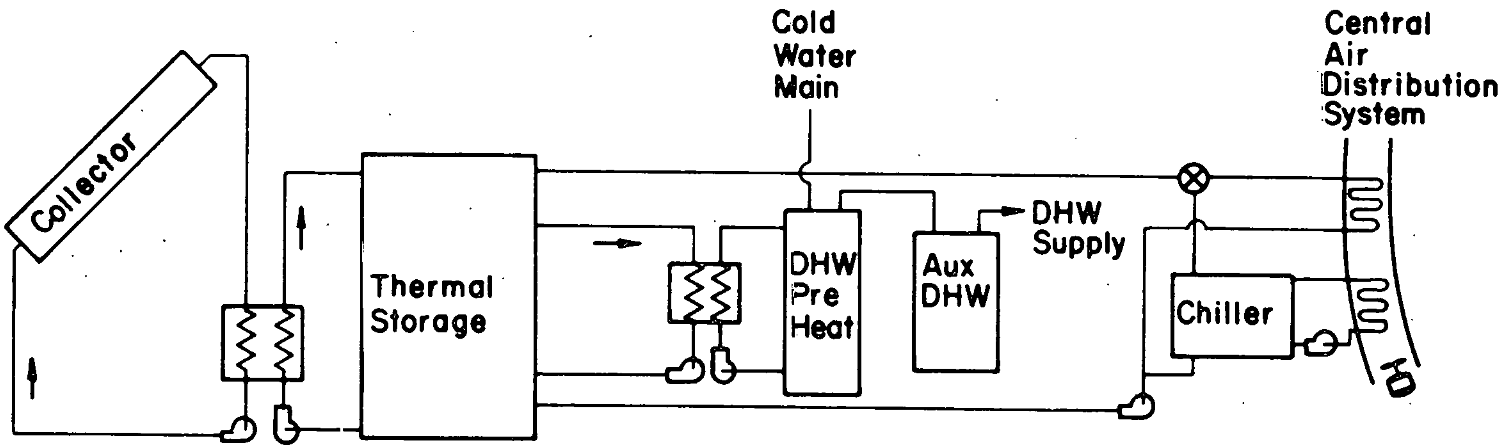



Figure 1. Schematic of Liquid System

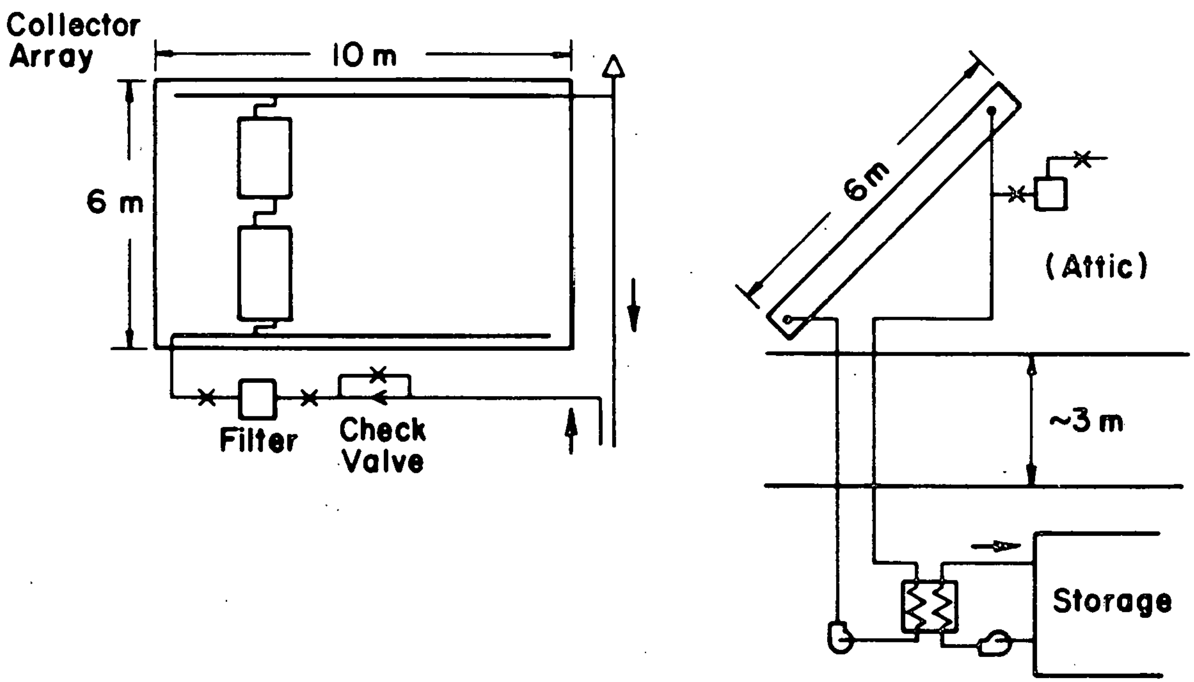



Figure 2. Schematic of Collector Piping Layout

Air System (Figure 3)

Solar Collector

$A_c$ = 60 $m^2$ (645.6 $ft^2$)

$F_R(\tau\alpha)$ = 0.56

$F_RU_L$ = 12.26 kJ/hr·$m^2$·°C (0.60 Btu/hr·$ft^2$·°F)

Flow rate = 44 - 55 kj/hr·$m^2$ of collector (2 - 2.5 CFM/$ft^2$)

$C_p$(air) = 1.006 kJ/kg·°C (0.24 Btu/lb·°F)

$\rho$(air) = 1.202 kg/$m^3$ (0.075 lb/$ft^3$)

$\dot{m}$ = 2,982 kg/hr

$\dot{m}C_p$ = 3,000 kJ/hr·°C

Storage

.75 to 1.5 inch pebble-bed

2 m (high) x 2 m (wide) x 4 m (long), V = 16 $m^3$

Heat capacity ≈ 21,500 kJ/°C (11,300 Btu/°F)

Temperatures

Minimum temperature to heating load = 40°C (104°F)

Average inlet temperature to collector = 20°C (68°F)

Average 24 hour ambient temperature - heating = 0°C (41°F)

Average daytime ambient temperature - heating = 5°C (50°F)

Blower

Flow rate = 2,481 $m^3$/hr (1460 CFM)

Rated at 2,014 kJ/hr (3/4 hp)

DHW pump = 672 kJ(electric)/hr (1/8 hp)

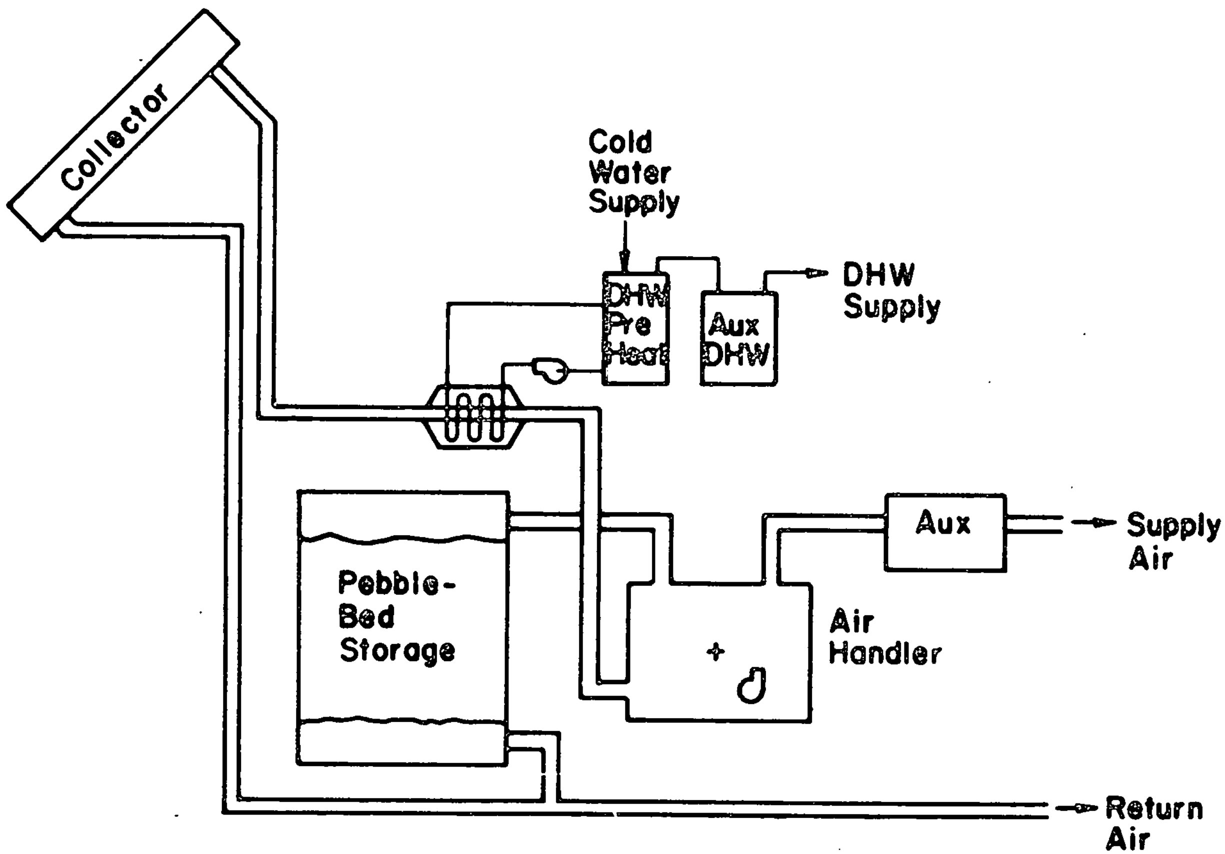
Ducting

Collector loop (exterior) 20 m of manifolding (2' x 1')
5 m of run (2' x 15")

Collector loop (interior) 10 m of run (2' x 15")

Circulating loop (interior) 10 m of run (2' x 15")
(does not include air distribution system ducting)

Figure 3. Air System Schematic



NOMENCLATURE

| | |
|---|---|
| $A_c$ | Gross area of solar collector array, $m^2$ |
| $C_p$ | Specific heat of collector fluid, $kJ/hr \cdot m^2$ |
| COP | Coefficient of performance, dimensionless |
| $(cW)_c$ | Heat capacity of collector loop (exterior), kJ/°C |
| $(cW)_I$ | Heat capacity of interior circulating loop, kJ/°C |
| $(cW)_s$ | Heat capacity of collector and exchanger loop (exterior or interior), kJ/°C |
| $\overline{E}$ | The thermal equivalent of average daily electrical usage in the collection and circulation to load of solar energy, kJ(thermal)/hr |
| $\overline{E}_e$ | The average daily electrical usage by the solar system, kJ(electric)/hr |
| $F_R$ | Collector heat recovery factor, dimensionless |
| HR | Solar insolation on a tilted surface, $kJ/hr \cdot m^2$ |
| $\overline{HR}$ | Daily average value of HR |
| LMTD | Log mean temperature difference, °C |
| $\dot{m}$ | Mass flow rate through collector, kg/hr |
| $\overline{Q}_L$ | Average daily heat losses from the solar system piping and/or ducting, kJ/hr |
| $\overline{Q}_L'$ | Piping and/or ducting heat losses to the exterior of the conditioned space, kJ/hr |
| $\overline{Q}_L''$ | Piping and/or ducting heat losses to the interior of the conditioned space, kJ/hr |
| $Q_u$ | Useful energy obtained from a collector, kJ/hr ($Q_u = \dot{m} C_p \Delta T$) |
| $\overline{Q}_u$ | Daily average value of $Q_u$ |
| $\overline{Q}_{SL}$ | Average daily heat losses from the thermal storage and DHW preheat, kJ/hr |
| $\overline{Q}_{SL}'$ | Thermal storage and DHW preheat tank heat losses to the exterior of the conditioned space, kJ/hr |
| $\overline{Q}_{SL}''$ | Thermal storage and DHW preheat tank heat losses to the interior of the conditioned space, kJ/hr |
| R | Resistance to heat flow (commonly used in English units of $hr \cdot ft^2 \cdot °F/Btu$) |

| | |
|---|---|
| $T_a$ | Ambient temperature, °C |
| $(\overline{T_i - T_a})$ | Average temperature difference between collector inlet and ambient, °C |
| $T_s$ | Temperature of thermal storage, °C |
| $U$ | Heat loss coefficient of insulation, $kJ/hr \cdot m^2 \cdot °C$ |
| $(UA)_L$ | Heat loss rate from thermal storage, $kJ/hr \cdot °C$ |
| $(UA)_{L_c}$ | Heat loss rate from collector loop, $kJ/hr \cdot °C$ |
| $(UA)_{L_D}$ | Heat loss rate from DHW preheat tank, $kJ/hr \cdot °C$ |
| $(UA)_{L_I}$ | Heat loss rate from circulating loop of system, $kJ/hr \cdot °C$ |
| $(UA)_{L_s}$ | Heat loss rate from storage side of exchanger loop, $kJ/hr \cdot °C$ |
| $U_L$ | Collector heat loss coefficient, $kJ/hr \cdot m^2 \cdot °C$ |
| $\delta$ | Knonecker delta, $= \begin{cases} 1 \text{ for space cooling applications, dimensionless} \\ 0 \text{ for space heating applications, dimensionless} \end{cases}$ |
| $\Delta p$ | Pressure drop, $nt/m^2$ |
| $\Delta t$ | Average number of hours of collector operation per day, hours |
| $\Delta t'$ | Average number of hours of solar availability per day, hours |
| $\Delta t_d$ | Average number of hours of effective heat losses from thermal storage and DHW preheat tanks per day, hours |
| $\Delta t_e$ | Average number of hours of parasitic power requirements per day, hours |
| $\Delta t_I$ | Average number of hours of circulation to load per day, hours |
| $\Delta t_L$ | Average number of hours of effective heat losses from piping/ducting per day, hours |
| $\Delta T$ | Outlet temperature of collector minus inlet temperature of collector, °C |
| $\Delta T_{L_c}$ | Temperature difference between collector fluid and ambient during collector operations, °C |
| $\Delta T_{L_I}$ | Temperature difference between circulating fluid and conditioned space, °C |
| $\Delta T_{L_s}$ | Temperature difference between storage/exchange loop fluid and ambient during collector operations, °C |
| $\eta_{day}$ | Daily efficiency of a solar collector array, dimensionless |

| | |
|---|---|
| $\bar{\eta}_e$ | Efficiency of the electric power plant generation and distribution, dimensionless |
| $\bar{\eta}_f$ | Average operating efficiency of a conventional fossil fuel fired furnace, dimensionless |
| $\eta_s$ | Daily efficiency of a solar collector array taking into account heat losses and parasitic power usage, dimensionless |
| $\xi$ | Kronecker delta, $= \begin{cases} 1 & \text{exterior thermal storage} \\ 0 & \text{interior thermal storage} \end{cases}$ |
| $\rho$ | Density of fluid, $kg/m^3$ |
| $(\tau\alpha)$ | Effective transmissivity-absorptivity product, dimensionless |

## REFERENCES

[1] Duffie, J.A. and Beckman, W.A., Solar Energy Thermal Processes (New York: Wiley Interscience), 1974.

[2] Ward, D.S. and Löf, G.O.G., "Design, Construction, and Testing of a Residential Solar Heating and Cooling System". Report prepared for the Committee on the Challenges of Modern Society, Department of Energy, July 1976.

[3] Public Service of Colorado, Private Communication, 1977.

[4] Jacobsen, A.S., "Solar Heating and Cooling of Mobile Homes, Test Results". Proc. 1977 Annual Meeting of Amer. Sec. of International Solar Energy Society, Orlando, Florida, Vol. 1, 1977.

[5] San Martin, R.L., LaPlante, D., Packard, C., and Shaw, H., "Twenty Months Operating Experience with a Solar Heated and Cooled Office Building". Proc. 1977 Annual Meeting of the Amer. Sec. of International Solar Energy Society, Orlando, Florida, Vol. 1, 1977.

[6] Karaki, S., CSU Solar House II results, Private communication, 1978.

[7] Close, D.J. and Yusoff, M.B., "The Effects of Air Leaks on Solar Air Collector Behavior", Solar Energy, Vol. 20, pp. 459-463, 1978.

[8] Karaki, S., Armstrong, P.R. and Bechtel, T.N., "Evaluation of a Residential Solar Air Heating and Nocturnal Cooling System". Report prepared for the U.S. Department of Energy (COO-2868-3), December 1977.

[9] Ward, D.S., "Operating Thresholds of Solar Collection Systems". Submitted to Solar Energy Journal, 1978.

[10] Young Radiator Company, Fixed Tube Bundle Heat Exchangers Catalog 1976, Racine, Wisconsin.

APPENDIX C

# OPERATING THRESHOLDS OF SOLAR COLLECTION SYSTEMS

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# OPERATING THRESHOLDS OF SOLAR COLLECTION SYSTEMS

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## ABSTRACT

A solar collector subsystem operating threshold is defined as the minimum solar insolation rate at which useful heat can be collected when collecting loop piping heat losses and parasitic power requirements are considered. This paper discusses the quantitative effect of these energy flows on the operating threshold of the solar subsystem (and thus on the overall system efficiency), and briefly considers the effects of different installation and design procedures.

This paper should be considered a companion to the paper entitled, "Solar Heating and Cooling System Efficiency as a Function of Design and Installation", by the same author.

## INTRODUCTION

Increasing importance is being placed on the operating threshold of solar collectors, i.e., the required conditions for collection of useful solar energy. Normally an operating threshold refers to the minimum solar radiation intensity, which is necessary before a solar collector can absorb more energy than is lost from the collector through radiation, conduction, and convection.

Such a condition can be illustrated by the use of the relationship [1]:

$$Q_u = A_c \{ HR\, F_R(\tau\alpha) - F_R U_L (T_i - T_a) \} \tag{1}$$

Using equation (1), we can define the operating threshold, i.e., the minimum solar insolation rate necessary to allow for the collection of useful solar energy, by setting $Q_u = 0$. Thus equation (1) becomes:

$$(HR)_c = \frac{F_R U_L}{F_R(\tau\alpha)} (T_i - T_a) \tag{2}$$

where

$(HR)_c$ = Solar insolation operating threshold for a solar collector to collect useful solar energy

The operating threshold in equation (2) can no longer be considered practical, because of the effects of a solar system on the operation of the solar collector. This paper will discuss a revised definition of the solar insolation operating threshold, which accounts more realistically for the practical operation of a solar space and domestic hot water (DHW) heating and space cooling system. Particular emphasis will be directed toward solar space cooling systems.

## OPERATING THRESHOLD

The efficiency of a solar collector can be written as:

$$\eta_c = \frac{Q_u}{HR\, A_c} \tag{3}$$

where

$\eta_c$ = solar collector efficiency, (dimensionless)

and

$$Q_u = \dot{m}C_p(T_o - T_i) = \dot{m}C_p\Delta T_c \tag{4}$$

The collector's efficiency when operating in conjunction with a complete system, however, must take into account the solar energy which is actually delivered to the thermal storage or directly to the load and the parasitic power requirements necessary to collect and transfer the collected heat. For this purpose, we define a solar collector's operating efficiency within a solar system as:

$$\eta_s = \frac{Q_u - Q_L}{HR\,A_c} - \left(\frac{\eta_f}{\eta_e}\right)\frac{E}{HR\,A_c} \tag{5}$$

The disadvantage of equation (5) is that observed efficiencies ($\eta_s$) from one solar installation are not readily transferred to a second installation. In other words, $Q_L$ and E are functions of a particular design or installation and variations in $\eta_s$ between the two different types of solar collectors in different installations are not readily comparable. The practical reality of operating solar heating and cooling systems, however, requires the concept of a solar collector operating efficiency if realistic technical and economic feasibilities are to be considered. It is meaningless to quote collector efficiencies of $\eta_c$ = 30% when the system's collector efficiency is $\eta_s$ = 10%. And such variations in efficiency are not uncommon.

$\eta_s$ can also be used for particular solar collector module designs and basic system layouts, as a comparison point for different installations, i.e., the engineering design can attempt to optimize the thermal performance by reducing $Q_L$ and E as much as possible, consistent with the thermal performance and economic practicality.

A word is appropriate here concerning the term, $\eta_f/\eta_e$. This represents the ratio of the combustion efficiency of a fossil fuel furnace to the electrical generating efficiency of the parasitic power used to operate the solar collection subsystem. The intent of the usage of this term in equation (5) is to convert the electrical power usage, E, to a thermal energy equivalent. Thus $(\eta_f/\eta_e)$ is the ratio of the useful thermal energy delivered by a conventional fossil fuel unit in a non-solar system, to the thermal energy expended in providing the electrical power, E.

If we take the average combustion efficiency of a fossil fuel unit under typical operating conditions, we find that $\eta_f \simeq 60\%$. The electrical generating efficiency of a fossil fuel (or nuclear) power plant combined with the efficiencies of transportaiton and distribution of the electricity to the user imply that $\eta_e \simeq 25\%$ [2]. The ratio of these efficiencies is then:

$$\eta_f/\eta_e \simeq 2.4 \tag{6}$$

It is noteworthy that the average coefficient of performance of state-of-the-art heat pumps is on the order of 2.4. Had the electrical power been derived from a hydroelectric plant, the thermal equivalent or potential useful heat delivery is still about 2.4. With substantial improvements in seasonally average COP's of heat pumps, it may be necessary to increase the value of $\eta_f/\eta_e$.

Another view of understanding the use of $\eta_f/\eta_e$ in equations (5) and (6) is to recognize the greater usefulness of electrical energy to thermal energy and thus penalize the solar system accordingly.

The heat losses in the solar system derive from heat losses in the pipes (or ducts) and should differentiate between heat losses exterior to and inside the heated/cooled space. In addition, $Q_L$ may include leakage of the collector heat transfer fluid (e.g., some air leakage from ducts is fairly common). Thus:

$$Q_L = Q_L' + Q_L''\delta\ (1 + 1/COP) \tag{7}$$

The Knonecker Delta, $\delta$, is utilized because heat losses to the interior of the heated/cool space still contribute to meeting the load in a heating application; but in a cooling application, not only is the heat lost for purposes of utilizing it to operate the solar cooling unit, but the solar heat actually adds to the cooling load.

We are now in a position to evaluate equation (5) in terms of an operating threshold. We can define the operating threshold of a system as the point when $\eta_s = 0$. In this case:

$$(Q_u)_s = Q_L + (\eta_f/\eta_e)\,E \tag{8}$$

where

$(Q_u)_s$ = The required rate of useful solar heat collection at the operating threshold of a system, (kJ/hr)

Previous considerations of solar collector efficiencies (i.e., equation (1)) would imply that $(Q_u)_s = 0$. However, from a system's viewpoint the non-zero value of $(Q_u)_s$ becomes very significant as will be seen below.

To obtain a system equivalent of equation (2), we must rewrite equations (1) and (3) using the same considerations applied in deriving equation (5), i.e.:

$$\eta_s = F_R(\tau\alpha) - F_R U_L \frac{(T_i - T_a)}{HR} - \frac{Q_L}{HR\,A_c} - \frac{(\eta_f/\eta_e)\,E}{HR\,A_c} \tag{9}$$

Again, we can obtain the Solar Operating Threshold, i.e., the minimum solar insolation rate necessary for the collection of useful solar energy <u>and its delivery to thermal storage</u> (<u>or load</u>), $(HR)_s$ by setting $\eta_s$ in equation (9) equal to zero. The result is:

$$(HR)_s F_R(\tau\alpha) = F_R U_L \Delta T + Q_L/A_c + (\frac{\eta_f}{\eta_e})\,E/A_c$$

where

$$\Delta T \equiv T_i - T_a$$

This can be rewritten in the form:

$$(HR)_s = \frac{F_R U_L}{F_R(\tau\alpha)} \{ \Delta T + \frac{Q_L}{A_c F_R U_L} + (\frac{n_f}{n_e}) \frac{E}{A_c F_R U_L} \} \qquad (10)$$

The significance of equation (10) cannot be overstressed if realistic considerations of technical and economic feasibility of particular solar installations are to be made. $(HR)_s$ represents the lower limit of solar insolation which can be effectively utilized for the collection and delivery of useful solar energy.

The term outside the brackets, $F_R U_L / F_R(\tau\alpha)$, represents the effects of the solar collector module design. It is noteworthy that a design modification which reduces $U_L$ by 20%, at the expense of reducing $(\tau\alpha)$ by 20%, will accomplish nothing; the thermal performance will not be improved (assuming no other change in the solar system design or operation).

For example, two collector designs reported by Johnson, et al [3] included one with characteristics $F_R(\tau\alpha) = 0.816$ and $F_R U_L = 24.6$ kJ/hr·m$^2$·°C. A second design had a reduced $F_R U_L$ (= 19.0 kJ/hr·m$^2$·°C), which constituted a substantial 23% decrease in the collector heat loss term. However, the second collector also had a reduced $F_R(\tau\alpha)$ (= .632), which constituted a 23% decrease as well. The second collector, therefore, will not enjoy a reduced operating threshold.

The first term in brackets, $\Delta T$, is a variable of the solar system's design and operation. A significant reduction in $\Delta T$ will have a major effect on lowering $(HR)_s$ and can substantially improve the thermal performance of a system. For example a solar liquid-heating collector might have inlet temperatures to the collector ranging from 40°C to 70°C (with an average of about 50°C) over the course of a winter heating season. Because of the temperature stratification of a pebble-bed storage unit, a solar air-heating collector can expect inlet temperatures averaging about 20°C.

If the liquid and air collectors are similar from the absorber plate up, and the last two terms in brackets of equation (10) can be neglected, it is clear that the air collector enjoys the advantage of a very significantly reduced solar threshold, $(HR)_c$. In fact, for ambient conditions of -10°C, $(HR)_c$ for the air collector is half the corresponding value for the liquid collector.

However, as will be shown, these last two parameters should not be neglected. For the comparison of air and liquid collectors, the appearance of $F_R$ in both denominators does not bode well for the air collector.

## CALCULATIONS

For simplicity we will utilize the three example solar system designs given by Ward [4]. We will assume three types of insulation on the collector loop piping: (1) 5.08 cm (2") fiberglass insulation; (2) 1.27 cm (1/2") fiberglass insulation; and (3) no insulation. In addition, we will consider the case of storage interior to and exterior to the storage space (when the storage is exterior there is a larger area of collector loop piping exposed to the ambient, as well as the storage itself).

Using the results of Ward [4], we obtain the heat loss transfer rates for the liquid system:

Collector Loop (exterior)

$$(UA)_L = \begin{cases} 30.4 \text{ kJ/hr}\cdot°\text{C} & 2\text{" insulation} \\ 43.4 \text{ kJ/hr}\cdot°\text{C} & 1/2\text{" insulation} \\ 95.7 \text{ kJ/hr}\cdot°\text{C} & \text{No insulation} \end{cases}$$

Collector/Exchanger Loops (exterior or interior)

$$(UA)_L = \begin{cases} 15.4 \text{ kJ/hr}\cdot°\text{C} & 2\text{" insulation} \\ 22.5 \text{ kJ/hr}\cdot°\text{C} & 1/2\text{" insulation} \\ 50.6 \text{ kJ/hr}\cdot°\text{C} & \text{No insulation} \end{cases}$$

Circulating Loop (interior)

$$(UA)_L = \begin{cases} 9.34 \text{ kJ/hr}\cdot°\text{C} & 2\text{" insulation} \\ 11.98 \text{ kJ/hr}\cdot°\text{C} & 1/2\text{" insulation} \\ 22.51 \text{ kJ/hr}\cdot°\text{C} & \text{No insulation} \end{cases}$$

Thermal Storage and DHW Preheat (exterior or interior)

$$(UA)_L = \begin{cases} 9.534\ \text{kJ/hr}\cdot{}^\circ\text{C} & \text{R-30 insulation} \\ 15.054\ \text{kJ/hr}\cdot{}^\circ\text{C} & \text{R-19 insulation} \end{cases}$$

For the heating season, we utilize the appropriate temperature differences to calculate:

Collector Loop (exterior)

$$Q_L = \begin{cases} 1{,}702\ \text{kJ/hr} & \text{2" insulation} \\ 2{,}430\ \text{kJ/hr} & \text{1/2" insulation} \\ 5{,}359\ \text{kJ/hr} & \text{No insulation} \end{cases}$$

Collector/Exchanger Loop (if exterior)

$$Q_L = \begin{cases} 739\ \text{kJ/hr} & \text{2" insulation} \\ 1{,}080\ \text{kJ/hr} & \text{1/2" insulation} \\ 2{,}430\ \text{kJ/hr} & \text{No insulation} \end{cases}$$

Thermal Storage (if exterior)

$$Q_L = \begin{cases} 477\ \text{kJ/hr} & \text{R-30 insulation} \\ 753\ \text{kJ/hr} & \text{R-19 insulation} \end{cases}$$

The parasitic power requirements are considered for two cases: A "minimal" power requirement and a "conservative" power requirement. We will assume values of E = 1,675 kJ (electric)/hr (5/8 hp) and 2,685 kJ (electric)/hr (1 hp). Using equation (10), we obtain the results shown in Table 1.

The heat losses and power requirements described above are only the instantaneous losses, i.e., the losses occuring during the collection operation. This is essentially the case where the threshold we calculate provides just enough useful heat to replace the losses occuring during the collector operation itself. A more stringent requirement is the minimum solar insolation that accounts for the daily heat losses and power requirements, including those occurring overnight. Thus the collector must collect enough useful heat not only to balance its losses, but to balance the average overnight heat losses as well. Table 2 gives the value of $(HR)_S$ for several of the design cases in Table 1.

Table 1. Operating Thresholds (Instantaneous) for Design and Installation Variations in a Liquid Solar Heating System

| Row | Conditions* | $\Delta T$ (°C) | $\frac{Q_L}{A_c F_R U_L}$ (°C) | $(\frac{\eta_f}{\eta_e}) \frac{E}{A_c F_R U_L}$ (°C) | $(HR)_s$ (kJ/hr·m²) | $(HR)_s$ Btuh/ft² |
|---|---|---|---|---|---|---|
| 0 | Base | 50 | -- | -- | 1,336 | 118 |
| 1 | 1 - A - α | 50 | 1.5 | 3.5 | 1,470 | 129 |
| 2 | 1 - A - γ | 50 | 2.5 | 3.5 | 1,497 | 132 |
| 3 | 2 - A - γ | 50 | 3.4 | 3.5 | 1,521 | 134 |
| 4 | 3 - A - ε | 50 | 7.4 | 3.5 | 1,628 | 143 |
| 5 | 3 - A - α | 50 | 4.6 | 3.5 | 1,553 | 137 |
| 6 | 1 - B - α | 50 | 1.5 | 5.6 | 1,526 | 134 |
| 7 | 1 - B - γ | 50 | 2.5 | 5.6 | 1,553 | 137 |
| 8 | 2 - B - γ | 50 | 3.4 | 5.6 | 1,577 | 139 |
| 9 | 3 - B - β | 50 | 4.6 | 5.6 | 1,609 | 142 |
| 10 | 3 - B - ε | 50 | 7.4 | 5.6 | 1,684 | 148 |
| 11 | 1 - A - α | 40 | 1.2 | 3.5 | 1,195 | 105 |
| 12 | 3 - B - ε | 40 | 6.1 | 5.6 | 1,382 | 122 |

Conditions:

1. 5.08 cm (2") insulation on piping
2. 1.27 cm (1/2") insulation on piping
3. No insulation on piping

A. Minimal power requirements (1,675 kJ(electric)/hr, 5/8 hp)
B. Conservative power requirements (2,685 kJ(electric)/hr, 1 hp)
γ. Storage located on exterior with R-30 insulation
α. Storage located on interior with R-30 insualtion
ε. Storage located on exterior of conditioned space with R-19 insulation
β. Storage located on interior of conditioned space with R-19 insulation

Table 2. Operating Threshold (Average) for a Liquid Heating System

| Condition | $\Delta T$ (°C) | $(HR)_s$ (kJ/hr·m²) | $(HR)_s$ (Btuh/ft²) |
|---|---|---|---|
| 1 - A - α | 50 | 1,625 | 143 |
| 1 - A - γ | 50 | 1,757 | 155 |
| 2 - B - α | 50 | 1,716 | 151 |
| 3 - B - α | 50 | 1,784 | 157 |
| 3 - B - ε | 50 | 1,980 | 174 |

For the air heating system described in Appendix A of reference [4], we obtain similar results. In this case we consider two types of duct insulation: (1) 2.54 cm (1") fiberglass duct insulation and (2) no insulation. The ducts are assumed to have four percent air leakage. Parasitic power requirements are 2,350 kJ(electric)/hr (7/8 hp) and 3,350 kJ(electric)/hr (1-1/4 hp). We will also consider the two cases for interior and exterior storage with R-11 insulation. These results are shown in Table 3.

The third system considered by Ward [4] is a liquid space cooling and DHW system. An important aspect is that the heat losses to the conditioned space do not go toward meeting the load, but in fact add to the load. This gives us an additional factor of (1 + 1/COP) for all interior heat losses.

We will utilize the same assumptions for the cooling system that were used in the liquid heating system, except for the parasitic power requirements which are slightly higher (i.e., 2,014 and 4,028 kJ(electric)/hr, respectively). The $(UA)_L$ values given above are still valid, but we will utilize different $\Delta T$'s and will need to include the interior heat losses as well. The results of the calculations are given in Table 4. Table 5 gives the solar threshold for the average operating conditions.

## RESULTS

From Table 1, we see that the design variations considered cause an increase in the solar radiation threshold of 348 kJ/hr·m$^2$ (30 Btu/hr·ft$^2$). This is a very significant increase and could, in many locations, make the difference in whether or not the solar system is feasible. When the complete system is considered (Table 2), it becomes apparent that a minimum solar radiation intensity of 1,980 kJ/hr·m$^2$ (174 Btu/hr·ft$^2$), over a period of several hours, will be difficult to come by.

Table 3. Operating Thresholds for Design and Installations Variations in an Air Solar Heating System

| Row | Conditions* | $\Delta T$ (°C) | $\frac{Q_L}{A_c F_R U_L}$ (°C) | $(\frac{\eta_f}{\eta_e}) \frac{E}{A_c F_R U_L}$ (°C) | $(HR)_s$ (kJ/hr·m$^2$) | $(HR)_s$ Btuh/ft$^2$ |
|---|---|---|---|---|---|---|
| 0 | Base | 10 | -- | -- | 219 | 19 |
| 1 | 1 - A - α | 10 | 7.1 | 7.7 | 543 | 48 |
| 2 | 2 - A - α | 10 | 45.7 | 7.7 | 1,388 | 122 |
| 3 | 1 - A - β | 10 | 10.2 | 7.7 | 611 | 54 |
| 4 | 2 - A - β | 10 | 65.2 | 7.7 | 1,815 | 160 |
| 5 | 1 - B - α | 10 | 7.1 | 10.9 | 613 | 54 |
| 6 | 2 - B - α | 10 | 45.7 | 10.9 | 1,458 | 128 |
| 7 | 1 - B - β | 10 | 10.2 | 10.9 | 681 | 60 |
| 8 | 2 - B - β | 10 | 65.2 | 10.9 | 1,885 | 166 |
| 9 | 1 - A - α † | 10 | 8.2 | 7.7 | 567 | 50 |
| 10 | 2 - B - β † | 10 | 74.5 | 10.9 | 2,089 | 184 |

†20% leakage in ducts

*Conditions:

1. 2.54 cm (1") fiberglass insulation on all ducts
2. No insulation on all ducts

A. Minimal power requirements (2,350 kJ(electric)/hr, 7/8 hp)
B. Conservative power requirements (3,350 kJ(electric)/hr, 1-1/4 hp)

α. Interior pebble-bed storage
β. Exterior pebble-bed storage with R-11 insulation

Table 4. Operating Instantaneous Thresholds for Design and Installation Variations in a Liquid Solar Cooling System

| Row | Conditions* | $\Delta T$ (°C) | $\frac{Q_L}{A_c F_R U_L}$ (°C) | $(\frac{n_f}{n_e}) \frac{E}{A_c F_R U_L}$ (°C) | $(HR)_s$ | |
|---|---|---|---|---|---|---|
| | | | | | (kJ/hr·m²) | (Btuh/ft²) |
| 0 | Base | 61 | -- | -- | 1,630 | 143 |
| 1 | 1 - A - α | 61 | 4.4 | 4.2 | 1,860 | 164 |
| 2 | 1 - A - γ | 61 | 7.0 | 4.2 | 1,930 | 170 |
| 3 | 2 - A - γ | 61 | 9.2 | 4.2 | 1,988 | 175 |
| 4 | 3 - A - ε | 61 | 18.9 | 4.2 | 2,248 | 198 |
| 5 | 3 - A - α | 61 | 11.6 | 4.2 | 2,053 | 181 |
| 6 | 1 - B - α | 61 | 4.4 | 8.3 | 1,970 | 173 |
| 7 | 1 - B - γ | 61 | 7.0 | 8.3 | 2,039 | 179 |
| 8 | 2 - B - γ | 61 | 9.2 | 8.3 | 2,098 | 185 |
| 9 | 3 - B - β | 61 | 11.8 | 8.3 | 2,168 | 191 |
| 10 | 3 - B - ε | 61 | 18.9 | 8.3 | 2,357 | 207 |
| 11 | 1 - A - α | 51 | 3.8 | 4.2 | 1,577 | 139 |
| 12 | 2 - B - ε | 51 | 16.5 | 8.3 | 2,026 | 178 |

*Conditions:

1. 5.08 cm (2") fiberglass insulation on all system piping
2. 1.27 cm (1/2") fiberglass insulation on all system piping
3. No insulation on all system piping
A. Minimal power requirements (2,014 kJ(electric)/hr, 3/4 hp)
B. Conservative power requirements (4,028 kJ(electric)/hr, 1-1/2 hp)
α. Storage located on exterior with R-30 insulation
β. Storage located on exterior with R-19 insulation
γ. Storage located on interior with R-30 insulation
ε. Storage located on interior with R-19 insulation

Table 5. Operating Threshold (Average) for a Liquid Cooling System

| Row | Conditions* | $\Delta T$ (°C) | $\frac{Q_L}{A_c F_R U_L}$ (°C) | $(\frac{\eta_f}{\eta_e}) \frac{E}{A_c F_R U_L}$ (°C) | $(HR)_s$ | |
|---|---|---|---|---|---|---|
| | | | | | (kJ/hr·m$^2$) | (Btuh/ft$^2$) |
| 0 | Base | 61 | -- | -- | 1,630 | 143 |
| 1 | 1 - A - α | 61 | 11.7 | 4.2 | 2,055 | 181 |
| 2 | 1 - A - γ | 61 | 17.9 | 4.2 | 2,221 | 195 |
| 3 | 2 - A - γ | 61 | 20.2 | 4.2 | 2,282 | 201 |
| 4 | 3 - A - ε | 61 | 29.9 | 4.2 | 2,542 | 224 |
| 5 | 3 - A - α | 61 | 19.1 | 4.2 | 2,253 | 198 |
| 6 | 1 - B - α | 61 | 11.7 | 8.3 | 2,165 | 191 |
| 7 | 1 - B - γ | 61 | 17.9 | 8.3 | 2,331 | 205 |
| 8 | 2 - B - γ | 61 | 20.2 | 8.3 | 2,392 | 210 |
| 9 | 3 - B - β | 61 | 19.4 | 8.3 | 2,371 | 209 |
| 10 | 3 - B - ε | 61 | 29.9 | 8.3 | 2,651 | 233 |
| 11 | 1 - A - α | 51 | 9.8 | 4.2 | 1,737 | 153 |
| 12 | 3 - B - ε | 51 | 25.4 | 8.3 | 2,264 | 199 |

*Conditions:

1. 5.08 cm (2") fiberglass insulation on all system piping
2. 1.27 cm (1/2") fiberglass insulation on all system piping
3. No insulation on all system piping

A. Minimal power requirements (2,014 kJ(electric)/hr, 3/4 hp)
B. Conservative power requirements (4,028 kJ(electric)/hr, 1-1/2 hp)
α. Storage located on exterior with R-30 insulation
β. Storage located on exterior with R-19 insulation
γ. Storage located on interior with R-30 insulation
ε. Storage located on interior with R-19 insulation

An important aspect of Table 1 is the effect on an inadequately insulated exterior storage. Not only is heat lost from the storage unit itself, but from the additional piping on the exterior of the conditioned space. And yet this may be an acceptable penalty in a solar heating and cooling system because of the even more significant losses when the storage is located on the interior of a cooled space (see Tables 4 and 5).

The values in Tables 2 and 5 were based on the solar collection subsystem operating at a minimum level of solar insolation, over a period of six hours per day, in an effort to account for collector loop heat capacity overnight heat losses and heat losses from thermal storage over a 24 hour period. If the minimum solar insolation threshold is reached on the average for shorter periods of time, then the useful heat collected is still inadequate to account for the heat losses and parasitic power requirements.

The results for the air system in Table 3 indicate that one can expect improved performance from a well designed system, but that the lack of insulation on the ducts cause disastrous increases in the solar threshold. In effect, the lack of duct insulation is sufficient to eliminate the very feasibility of the solar system.

## Temperature Differentials

A very important result of the solar threshold analysis is its effects on control systems. If we set $\eta_s = 0$, in equation (5), we obtain the threshold for $Q_u$, i.e.:

$$(Qu)_s = Q_L + (\eta_f/\eta_e)E$$

We can consider this in the form of a temperature difference by noting that $Q_u = \dot{m}C_p(T_o-T_i)$, i.e.:

$$(T_o-T_i)_s = \frac{Q_L}{\dot{m}C_p} + (\eta_f/\eta_e)\frac{E}{\dot{m}C_p}$$

For the design consideration (1-A-$\alpha$) in the cooling mode, $(T_o - T_i)_s = 1.36°C$. For the (3-B-$\varepsilon$) condition (also in the cooling mode), $(T_o - T_i)_s = 3.3°C$.

If the minimum temperature for delivery to the cooling unit is $T_s = 80°C$, and the LMTD across the heat exchanger is 6°C, then the minimum inlet temperature to the collector, $T_i$, for the 3-B-$\varepsilon$ design is:

$$T_i = T_s + \text{LMTD} + (T_o - T_i)_s = 89.3°C$$

For a $(T_o - T_i)_s = 3.3°C$, the $(T_o - T_i)$ for useful heat collection is about 6°C. Thus $T_o = T_i + 6°C = 95.3°C$.

The control sensor to turn a collector on and off is based on the temperature difference between storage and collector outlet, i.e., $T_o - T_s$. Clearly the turn off signal occurs when:

$$(T_o - T_s)_{off} = 95.3°C - 80°C = 15.3°C$$

To prevent cycling of the collector pump, the turn on signal should be:

$$(T_o - T_s)_{on} = (T_o - T_s)_{off} + \Delta T_{DB} = 15.3°C + 10°C = 25.3°C$$

Thus the collector will not turn on in the morning until $T_o = 105.3°C$. For a 50-50 water glycol mixture the boiling point (unpressurized) is 108.9°C. This allows a storage temperature range of only 3.6°C. For the 4,000 kilograms of water, this amounts to a storage heat capacity of only 60,336 kJ; slightly less than one hour's operation of a 38,000 kJ/hr (3-ton) chiller. Had we used the (1-A-$\alpha$) design, the storage temperature range would be increased about 1.94°C, or 54% more storage capacity for cooling.

Another effect of the control strategy is that the actual turn on of the collector requires an even greater solar threshold. For the (1-A-$\alpha$) design (liquid heating system), the effective $\Delta T$ of 55°C is increased to 65°C, and the solar threshold is increased from 1,470 kJ/hr·m$^2$ (129 Btu/hr·ft$^2$) to 1,737 kJ/hr·m$^2$ (153 Btu/hr·ft$^2$).

## SYSTEM EFFICIENCY

Finally we should consider the effects on the system efficiency of the solar threshold. If the mean solar insolation data for January on the collector is 15 MJ/day·$m^2$ [5], then from Liu and Jordan [6], we can estimate the hourly radiation for the time of year. On this basis, therefore, we can determine the average number of operating hours when the solar insolation is above threshold. Using Table 2, the (1-A-$\alpha$) design is seen to operate about 5 hours per day and the (3-B-$\varepsilon$) design for about 3.5 hours per day. The total solar radiation available for the two designs is thus approximately 11 MJ and 8 MJ. Thus a bad design and installation reduces the potential for solar collection.

In addition, it also reduces the useful heat collected. For example if the collector daily efficiency is 35% (useful heat divided by solar during collector operation), then the useful heat gains from the collector are 3.85 and 2.8 MJ, respectively. This implies daily efficiencies of 25.7% and 18.7%, respectively. But from Ward [4], the ratio of system efficiency to collector daily efficiency for the two designs is .692 and .343. Thus the system efficiency for the (1-A-$\alpha$) design is 17.8% and the useful heat collected is 2.66 MJ. For the (3-B-$\varepsilon$) design the system efficiency is 6.4% and the useful heat collected is .96 MJ. The (1-A-$\alpha$) design therefore collects 275% more energy in January.

NOMENCLATURE

| | |
|---|---|
| $A_c$ | Gross area of solar collector, ($m^2$) |
| COP | Coefficient of performance of the solar cooling unit, (dimensionless) |
| $C_p$ | Specific heat of the solar collector heat transfer fluid, (kJ/kg·°C) |
| E | Parasitic electrical power requirements necessary to collect and transfer the solar collected heat to thermal storage (or load), (kJ of electricity/hr) |
| $F_R$ | Solar collector heat recovery factor, (dimensionless) |
| HR | Solar radiation incident upon the tilted collector surface, ($kJ/m^2 \cdot hr$) |
| $(HR)_c$ | Solar insolation operating threshold for a collector, ($kJ/hr \cdot m^2$) |
| $(HR)_s$ | Solar insolation operating threshold for a collection subsystem, ($kJ/hr \cdot m^2$) |
| LMTD | Log mean temperature difference across collector/storage heat exchanger, (°C) |
| $\dot{m}$ | Mass flow rate of the collector heat transfer fluid through the solar collector, (kg/hr) |
| $Q_L$ | Heat losses in the collector and circulating loops to storage (or load), not already accounted for by the collector heat loss coefficient, $U_L$, (kJ/hr) |
| $Q_L'$ | Heat losses from collector and/or circulating loops (exterior to heated/cooled space), includes leakage of heat transfer fluid, (kJ/hr) |
| $Q_L''$ | Heat losses from collector and/or circulating loops (interior to heated/cooled space), includes leakage of heat transfer fluid, (kJ/hr) |
| $Q_u$ | Useful energy collected by the solar collector, (kJ/hr) |
| $(Q_u)_s$ | Required rate of useful heat collection at the operating threshold of a system, (kJ/hr) |
| $T_a$ | Ambient (outdoor) air temperature, (°C) |
| $T_i$ | Inlet fluid temperature to collector, (°C) |
| $T_o$ | Outlet fluid temperature from collector, (°C) |
| $U_L$ | Collector heat loss coefficient ($kJ/hr \cdot m^2 \cdot °C$) |

| | |
|---|---|
| $\sigma$ | $\begin{cases} 0 \text{ for heating, (dimensionless)} \\ 1 \text{ for cooling, (dimensionless)} \end{cases}$ |
| $\Delta T$ | $T_i - T_a$ |
| $\Delta T_c$ | $T_o - T_i$ = Increase in collector fluid temperature through the solar collector, (°C) |
| $\Delta T_{DB}$ | Control dead band temperature difference, (°C) |
| $\eta_c$ | Efficiency of a solar collector, (dimensionless) |
| $\eta_e$ | Average operating efficiency of an electrical generating power plant including distribution efficiency, (dimensionless) |
| $\eta_f$ | Average operating furnace efficiency of a conventional fossil fuel unit, (dimensionless) |
| $\eta_s$ | The solar collector array's operating efficiency when the collector is integrated with a solar system, (dimensionless) |
| $(\tau\alpha)$ | Effective transmissivity-absorptivity product, (dimensionless) |

## REFERENCES

[1] Duffie, J.A. and Beckman, W.A., Solar Energy Thermal Processes, John Wiley and Sons; New York, 1974.

[2] Public Service Company of Colorado, Private Communication, 1977.

[3] Johnson, S.M. and Simons, F.F., "Comparison of Flat-Plate Collector Performance Obtained Under Controlled Conditions in a Solar Simulator", Presented at ISES Conference, Winnipeg, 1976.

[4] Ward, D.S., "Solar Heating and Cooling System Efficiency as a Function of Design and Installation", Submitted to Solar Energy Journal, 1978.

[5] Karaki, S., Armstrong, P.R., and Bechtel, T.N., "Evaluation of a Residential Solar Air Heating and Nocturnal Cooling System" (COO-2858-3), Report to the Department of Energy, December 1977.

[6] Liu, B.Y.H. and Jordan, R.C., "The Interrelationship and Characteristic Distribution of Direct, Diffuse and Total Solar Radiation". Solar Energy, Vol. 4, 1960.

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APPENDIX D

Abstracts of Papers Submitted to the
International Solar Energy Society Congress
to be held in
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# ABSTRACT

## Solar Heating and Cooling Performance in CSU Solar House III

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Performance and operating experience with a liquid-heating flat-plate solar collector integrated with a residential solar heating and cooling system is presented. Cooling data for the period June through September 1978 and Heating data for October 1978 through April 1979 are included, along with an analysis of the operation of the solar heating and cooling system.

Results of the cooling system analysis provide clear indications of the critical importance of temperature differentials between the collector outlet and the absorption chiller generator inlet, the effects of alternative control strategies, the marginal feasibility of cool storage, the devastating effect on system performance due to heat losses from the thermal storage unit, and the importance of minimizing electrical parasitic power requirements in obtaining feasibility for solar absorption cooling systems.

Performance data and analysis of the current winter heating season will also be presented.

## ABSTRACT

# A Modified $\bar{\phi}$-f Chart

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A modified $\bar{\phi}$-f chart [1] is presented which includes the effects on system performance due to heat losses from the thermal storage and other solar system components (including the collector and system loops piping and/or ducting); use of electrical parasitic power to operate the solar system, solar operating thresholds, system heat capacity effects, and other parameters affecting the solar system efficiency. The solar operating threshold is also modified to include the effects of different control strategies, parasitic power requirements, storage temperature stratification, and collector loop heat losses and heat capacities.

Comparisons to actual data from CSU Solar House III are included for validation of the original design method of Klein and Beckman [1], and for the modified $\bar{\phi}$-f chart. The critical importance of the suggested modifications in determining realistic values of f, is demonstrated for each modification.

[1] S.A. Klein and W.A. Beckman, "A General Design Method for Closed-Loop Solar Energy Systems," Proceedings AS/ISES Annual Meeting, Orlando, June 1977

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