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

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

By
Dan S. Ward
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H. S. Oberoi

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



Solar Energy

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FINAL REPORT - EXECUTIVE SUMMARY
FOR PERIOD 1 FEBRUARY 1976-30 SEPTEMBER 1978

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

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PROJECT SUMMARY

Contract EY-76-S-02-2858 provided for the detailed experimental evaluation and analysis of a residential solar heating and cooling system installed in Colorado State University (CSU) Solar House III. During the bulk of the contract (1 February 1976-31 May 1978), the CSU Solar House III system utilized the Owens-Illinois (O-I) liquid-heating evacuated tube solar collector. During the period 1 June 1978-30 September 1978, the Chamberlain liquid-heating, state-of-the-art flat-plate solar collector was evaluated for a complete cooling season. Further evaluation of the Chamberlain collector will be forthcoming, pending follow-on funding.

Significant and important results of the two systems evaluations include:

- The demonstrated non-feasibility of the O-I liquid-heating evacuated tube solar collector in residential solar heating and cooling system applications. This conclusion is based on 20 months of operating experience with the O-I collector, during which time continual breakage of the collector tubes, collector fluid leakage, inadvertent boiling, etc. prevented normal day-to-day operations.
- The O-I collector demonstrated excellent thermal performance characteristics while integrated with the Solar House III heating and cooling system. This performance includes high monthly-averaged daily solar collector efficiencies (approximately 50 percent), low solar operating thresholds (minimum solar radiation intensity at which the collector can collect useful heat -- approximately 125 W/m^2 ; $40 \text{ Btu/hr}\cdot\text{ft}^2$), and the maintenance of good efficiency with simplified control strategies.
- Theoretical considerations have been advanced by the project staff to support the predicted excellent performance of an air-heating evacuated tube solar collector. These arguments have shown the feasibility of maintaining the same excellent performance characteristics described above, in an air-heating collector of similar design, but which can be expected not to experience the same operating difficulties encountered with the liquid-heating design. (Virtually all operational

difficulties experienced over 20 months can be directly related to the use of a liquid as the solar collector heat transfer medium).

- The importance of solar collector heat capacity for liquid-heating collectors integrated with a solar heating and cooling system has been demonstrated. (Previous researchers have tended to ignore heat capacity effects.)
- Evaluation of acquired data on the performance of the CSU Solar House III system has provided the theoretical construction and experimental verification of the importance of solar operating thresholds (i.e., the minimum solar radiation intensity at which the collector can collect useful solar heat). In addition, the effects of collector loop heat losses, variations in control strategies, and parasitic electrical power requirements on the operating threshold have been clearly quantified and their importance in the design process justified.
- The concept of system efficiency has similarly been quantified by results of the project and the critical importance of the concept has been experimentally justified. The demonstrated effect of heat losses from system piping and thermal storage units, control strategy variations, heat capacities in system piping, and parasitic power requirements on the solar system's efficiency has been detailed. In addition, it has been clearly shown that neglect of these critical factors in the design process can easily lead to solar installations which consume more energy than can be usefully acquired by the solar system.
- The importance of thermal storage heat losses to the interior of the heated space has been shown to be a critical factor in determining the feasibility of active solar heating systems.
- CSU Solar House III cooling data, acquired during the 1978 summer, has provided substantial evidence that shows the non-feasibility of solar absorption (LiBr) cooling systems whenever conventional thermal storage units (utilizing water or other similar liquids) are located interior to the conditioned space of the building. The importance of thermal storage heat losses cannot be overemphasized in connection with solar cooling systems.
- Experimental evidence and subsequent analysis has provided a clear basis for the evaluation of solar cooling systems by

relating the useful cooling by solar to the parasitic electrical power requirements of the solar system. The absolutely essential necessity of minimizing electrical power requirements (to the greatest possible degree) has been clearly delineated.

- The marginal feasibility of cool storage subsystems has been shown by reference to interior space requirements and additional (unwanted) parasitic power requirements.
- The importance of temperature differentials between collector and storage during solar cooling operations have been demonstrated. It should be stressed that minimizing these temperature differences is critically important to the solar cooling system performance. In addition it can be anticipated that control strategies for cooling systems will have to be critically re-evaluated in light of these results.

EXECUTIVE SUMMARY

1.0 OBJECTIVE

The objective of the study is to test and evaluate the practicality of an integrated evacuated tube solar collector and absorption cooling system installed on Colorado State University (CSU) Solar House III. This objective was to be accomplished by designing and installing a complete solar heating and cooling system (including appropriate data acquisition equipment and instrumentation), performing a detailed analysis and evaluation of all aspects of the solar system, and comparing the seasonal performance of the system with two other solar heating and cooling systems installed in adjacent buildings with virtually identical thermal characteristics.

On the authority of the program manager, the technical evaluation of the evacuated tube solar collector was terminated and the complete solar collector array was removed from Solar House III in May 1978. In June 1978, a new flat-plate state-of-the-art liquid-heating collector was installed and the objective of the project was modified (by authority of the program manager) to include the testing and evaluation of the practicality of the flat-plate state-of-the-art solar collector integrated with the heating and cooling system and to provide for the future comparison of seasonal performance data with an air-heating solar collector to be installed on CSU Solar House II. (This latter objective of the comparison of air and liquid systems is not part of the present contract, which terminated on 30 September 1978, but is intended as one of the principal objectives of a follow-on proposal now under consideration by the Department of Energy).

2.0 PROGRESS OF WORK

After the award of the contract on 1 February 1976, all design work on the integrated solar heating and cooling system (including a custom designed solid-state control system) was completed and the system was fabricated and/or acquired and subsequently installed. The data acquisition system for the continuous monitoring of the system was also designed and installed. The system became operational in September 1976, and data on the operational performance of the system were acquired and evaluated over a 20-month period. Due to continuing difficulties (particularly with the evacuated tube solar collector array), less than six months of useful month-long, continuous data were acquired during this period.

The installation of a replacement solar collector array (utilizing a flat-plate state-of-the-art liquid-heating collector) was completed in June 1978, and approximately three months of useful cooling data have been acquired on the thermal performance of the system. A preliminary analysis of this thermal performance data has been made and is summarized below under the section "Summary of Technical Results".

3.0 PROBLEMS ENCOUNTERED

The major problem encountered during the project was the inability of the Owens-Illinois liquid-heating evacuated tube solar collector array to perform on a continuous basis. Continuing and severe collector liquid leakage problems plagued the project during the entire 20 months of performance evaluation. These leakage problems were primarily associated with the interface between the evacuated tube itself and the manifolding, which integrated 24 tubes per module into a single series flow unit. In addition, leakage problems were encountered between adjacent collector module manifolds. A design change in the manifold itself has, however, virtually solved this latter leakage problem.

A more significant problem has been the continuing breakage of the glass tubes due to severe thermal stresses. For example, a collector tube which contains no liquid (because of boil off, draining, etc.) will continue to collect heat and approach temperatures at the absorber surface of 300 to 400°C. Should boiling liquid (at $T \approx 100^\circ\text{C}$) from an adjacent tube reach the dry, hot tube, a severe thermal shock on the glass absorber tube will result and cause, in most cases, complete destruction of the collector tube.

Such breakage of the glass tubes from thermal shock has been observed during boil off of the collector array, during initial filling, and during apparently normal operations. This latter aspect is particularly noteworthy because it suggests an inability of the solar collector to withstand the thermal stresses associated with normal operating conditions. Such operational performance makes it clear that the design is not now practical for residential solar heating and/or cooling applications.

Discussions with Owens-Illinois (O-I), the evacuated tube solar collector manufacturer, tend to support this conclusion, but O-I did emphasize potential success with industrial applications. The key, apparently, is that the industrial heating systems use continuous flow (hoping to utilize every available Btu) and this solves most (if not all) of the breakage problems. According to O-I, the main problem is a lack of

good flow distribution. Once a good flow distribution is achieved and the collection pump runs continuously, then they claim to have few, if any, problems. O-I has also taken our recommendation and designed a 4 ft by 4 ft module with all the tubes above the manifold. This allows easy draining and solves some of the operational problems. However it is the opinion of the project staff that the liquid-heating design tested on CSU Solar House III and similar designs cannot be considered as a viable alternative for residential applications.

It should be noted that, because of numerous difficulties with the initial set of O-I evacuated tubes and manifolds (broken and leaking tubes, freeze burst manifolds, etc.), O-I furnished a complete replacement of all evacuated tubes and eight new manifolds. Most of the difficulties experienced in glass breakage occurred with the initial collector array. The replacement array proved to be of a much higher quality and relatively fewer difficulties were encountered following installation in December 1976 and January 1977.

Other problems encountered with the O-I collectors included:

- (1) Difficulty in obtaining adequate flow distribution through the collector
- (2) Difficulty inserting and maintaining control instrumentation sensors within the liquid flow volume of the evacuated tubes
- (3) Control problems associated with long response times
- (4) Inability to drain the collector without disassembly
- (5) Freezing in the evacuated glass tubes and the solar collector module manifolds
- (6) Frequent boiling resulting in additional loss of collector fluid and evacuated glass tube breakage
- (7) Large collector heat capacity, resulting in substantial overnight heat loss.

These factors are discussed in more detail below under "Summary of Technical Results".

A separate but significant problem was encountered with the automatic data acquisition system. Initially the system located in CSU Solar House II was utilized in conjunction with the data instrumentation sensors in CSU Solar House III. This arrangement proved unsatisfactory in practice and was replaced in the spring of 1977.

The initial operation of the new House III data acquisition system was plagued by problems during the first months of operation, causing a

significant loss of useful data during this critical "shakedown" period. Noise in the data acquisition unit and associated tape recorder, grounding problems, passive parity errors, and sporadic power surges and/or losses have resulted in considerable lost time in the acquisition of system data.

The problems associated with the data acquisition equipment have been continually reduced over the time of operation of the equipment to the extent that the system is now considered reasonably reliable. Partial losses of daily data due to the data subsystem malfunctions have, for example, been reduced to a total of four days out of a collection period of 102 days.

The significance of this experience is the realization that the automatic and continuous acquisition of data from numerous sensors (50 to 80 channels) and in a form suitable for direct computer analysis is a particularly difficult undertaking. It is essential that time be allowed in any proposed project of this nature to properly bring the data acquisition equipment reliability up to an acceptable level.

4.0 FUTURE ACTIVITIES

No major tasks remain on the O-I phase of the project. The O-I collector array was removed from CSU Solar House III in May 1978. No post-contract activities are planned with respect to the O-I collector and this phase of the project must be considered a dead-end. The liquid-heating evacuated tube collector utilized in this project cannot be considered practical for use in residential solar heating and cooling systems.

The second phase of the project utilized the Chamberlain flat-plate solar collector for the acquisition of summer cooling data. Post-contract activities within the framework of a follow-on project include:

- (1) Evaluation of the performance of the solar cooling system utilizing the state-of-the-art flat-plate collector and the Yazaki 2-ton lithium bromide absorption chiller with cool storage during the summer of 1978.
- (2) Operate, maintain, record continuous data, and evaluate the performance of the liquid-heating flat-plate solar collector on CSU Solar House III during the winter of 1978-1979 and compare the performance data with the results from CSU Solar House II's air-heating collector system.
- (3) Install, operate, maintain, record continuous data, and evaluate the performance of the Direct Contact Liquid-Liquid Heat Exchanger project on the state-of-the-art flat-plate solar collector and

solar heating system in CSU Solar House III during selected periods of the 1978-1979 heating season.

It is anticipated that the analysis of the quantitative results of the cooling subsystem performance during the 1978 summer cooling season will provide an important indication of the technical feasibility of cool storage and an assessment of the practicality of this type of cooling subsystem. In addition, the potential for a triple thermal storage system (utilizing one exterior hot storage and two interior cool storage units in summer and two interior (stratified) hot storage units in winter) can be assessed and the quantitative advantages of this control of heat loss/heat gain by the thermal storage units can be addressed. In combination with other DOE cooling experiments, it should then be possible to obtain definitive conclusions on the utilization of cool storage in residential applications.

The comparative performance of solar air-heating and liquid-heating systems during the winter heating season (1978-1979) is of primary importance. While previous efforts have indicated a slight superiority of the air system over the liquid system, these results were based on early designs of solar collector systems and cannot be relied upon as a clear indication of the relative performance of the two types of systems. The utilization of CSU Solar Houses II and III (with identical floor plans and construction characteristics and virtually identical solar and meteorological conditions) with the installed state-of-the-art air and water heating systems can then allow for definitive conclusions of the relative performance of these two types of systems.

In this connection the incorporation of the DCLLHE in the CSU Solar House III liquid-heating system will provide additional information on the potential performance improvement of the liquid system and its relative performance with the air unit. The use of the DCLLHE provides for a collector heat transfer liquid that is immiscible with water but which transfers its heat to a water storage tank with a minimal temperature difference between the collector and storage loops. The DCLLHE thus eliminates a heat exchanger and additional storage loop pump (with corresponding cost savings) while at the same time eliminating boiling and freezing problems in the collector loop by the appropriate choice of the collector heat transfer liquid. The incorporation of the DCLLHE and a state-of-the-art solar flat-plate collector system will provide a clear indication of the potential for greatly improved system performance using this collector loop/storage design.

In addition the DCLLHE will be incorporated in the testing and evaluation of a solar absorption cooling system. Because of the demonstrated importance of temperature differentials (between collector and storage) [see "Summary of Technical Results - Chamberlain"], the DCLLHE will also allow for very significant improvements in the performance of solar cooling systems.

SUMMARY OF TECHNICAL RESULTS - OWENS-ILLINOIS EVACUATED TUBE

Because of the two separate and distinct phases of the project, it is desirable to separate the reporting of the technical results. The evaluation of the Owens-Illinois liquid-heating evacuated tube solar collector array integrated with the CSU Solar House III system will be discussed first.

1.0 CONCEPT

Component testing of evacuated tube solar collectors shows that, for a given solar radiation level, this advanced collector produces higher output temperatures of the collector heat transfer fluid and at higher efficiency than flat-plate solar collectors. In addition, the evacuated tube collector can collect useful energy at much lower levels of insolation, i.e., the solar radiation threshold for an evacuated tube collector is about 125 W/m^2 (40 Btu/hr-ft^2), but is over 300 W/m^2 (100 Btu/hr-ft^2) for conventional liquid-heating flat-plate collectors. This significantly lowered threshold allows for longer periods of collection (earlier in mornings and later in evenings) and during cloudy periods such that daily collector efficiencies for evacuated tube collectors (total daily useful heat divided by total daily radiation) can be double that of flat-plate collectors.

An essential aspect of evaluating the practicality of this advanced collector is its integration into a complete solar heating and cooling system. Only under the demands imposed by system testing can the practicality of the collector array be adequately assessed.

The evacuated tube solar collector utilized with the solar heating and cooling system in CSU Solar House III is manufactured by Owens-Illinois and is shown schematically in Fig. 1.

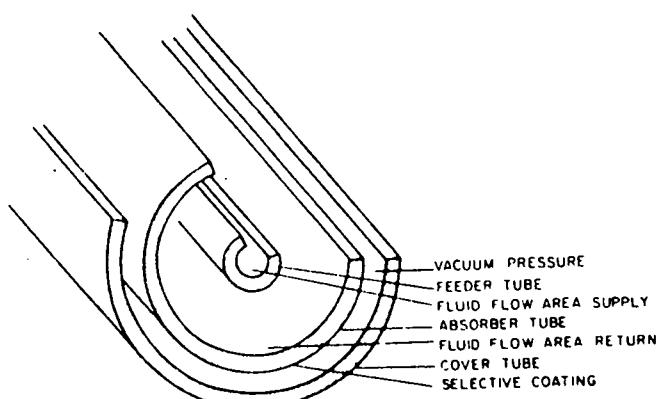


Fig. 1. Evacuated Tube Solar Collector

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 over a 20-month period. The solar collector showed excellent performance, achieving 50 percent daily collector efficiencies under relatively adverse weather conditions and low solar insolation. However, electrical parasitic power requirements were about 15 percent of the useful heat delivered and overnight heat losses from the solar collector contributed approximately 33 percent of the useful heat delivered.

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 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 [1,2, 4,6,7] (see section on "Publications/Reference").
- 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 tube
 - (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.

These problems have been discussed in detail in other reports [3,5, 8].

- 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 (e.g., stagnation) temperatures in the collectors
- (6) High daily collector efficiency.

Specific results are reported in the literature [3,8].

- Monthly summaries of continuous data for two specific operating periods have been acquired and reported [3,8].
- The O-I 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 TECHNICAL DEVELOPMENTS

Significant technical developments and results of the project include:

4.1 Evacuated Tube Solar Collection Threshold

The theoretical minimum solar radiation intensity required to achieve useful heat by the solar collector was computed by O-I to be 465 kJ/hr-m^2 ; 125 W/m^2 (41 Btu/hr-ft^2). Experimentally the solar threshold necessary to turn on the collector pump was observed during the first year of operation to average 560 kJ/hr-m^2 ; 155 W/m^2 (49.3 Btu/hr-ft^2). This higher value, however, is due to the set temperature differential between the collector outlet temperature and the thermal storage temperature, i.e., $\Delta T_{ON} \approx 10^\circ\text{C}$.

Use of a photocell during the second winter allowed for a more accurate determination of the solar threshold. Solar thresholds during this period averaged 694 kJ/hr-m^2 ; 193 W/m^2 (61 Btu/hr-ft^2) in the morning and 339 kJ/hr-m^2 ; 95 W/m^2 (30 Btu/hr-ft^2) in the evenings. This difference in the two

values are indicative of the very large heat capacity of the collector. During February 1978, the collector was able to begin collection of useful heat at solar radiation intensities as low as 291 kJ/hr-m^2 ; 82 W/m^2 (26 Btu hr-ft^2) and to continue collecting useful heat until the solar radiation intensity dropped to as low as 70 kJ/hr-m^2 ; 19 W/m^2 (6 Btu/hr-ft^2).

4.2 Equilibrium No-Flow Conditions

The highest equilibrium no-flow condition (i.e., stagnation temperature) recorded was 280°C (540°F) at a solar radiation intensity of 769 W/m^2 (272 Btu/hr-ft^2).

4.3 Durability of Selective Surface

Numerous observations have been made of significant color changes in the absorber surfaces of the O-I evacuated tubes. These color changes have been carefully evaluated and have been shown to result in no loss of absorptivity. The color changes do result in a shift of the absorptivity versus wavelength curve, but in no way indicate any degradation in the ability of the absorbing surface to absorb incident solar energy.

4.4 Use of Specular Reflectors

The use of specular reflectors located directly behind and attached to the evacuated tubes were expected to yield 25 percent more energy than modules with diffuse reflector backgrounds. An additional 25 percent yield was experimentally observed for clear days, but the use of specular reflectors on more overcast days resulted in net losses. During the month of March 1977, the useful energy output from the collector array equipped with specular reflectors showed no improvement over the collector array utilizing a plain white background.

4.5 Control Sensors

The present O-I solar collector module design does not allow for the insertion of control instrumentation sensors in the evacuated tube itself and thus severely hampers the control function. It is suggested that any evacuated tube design incorporate a mechanism to allow the insertion of a control or data instrumentation sensor. This would allow for improved control methods, as well as the opportunity to check for good parallel flow distribution in the various collector modules. The present design does not allow for a positive check of the operation of the solar collector array.

4.6 Control Time Lag

Because of the low flow rate through the solar collector array 950 l/sec (4.2 gpm through 16 modules), there is an effective time lag from the

time when the collector sensor signals a particular temperature until water entering the collector module can reach the outlet of the collector module. This time lag is typically eight to ten minutes. The effect of this time lag on start-up conditions is to increase the outlet temperature by as much as 15°C before the cooler water being pumped into the collector can reach the outlet of each module. This condition greatly increases the chances of an undesirable boiling of the collector on initial start-up. To prevent such an occurrence, a boil protection circuit was incorporated into the control system to turn on the collector pump whenever the collector liquid reached a preset temperature (e.g., 75°C) sufficiently below boiling. In practice, however, this effort proved to be somewhat futile and occasional boiling episodes continued to be observed. The recurrent boiling of the collector liquid caused significant losses of the collector liquid and, in addition, caused sufficient vapor locking in the flow distribution to significantly degrade the overall performance of the system.

In an effort to prevent this frequent boiling and the corresponding reduction in system performance (as well as interruptions in data collection), a photoelectric cell was installed in January 1978 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 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 as compared to conventional flat-plate solar collectors.

4.7 Collector Array Overnight Heat Loss

Because of the O-I solar collector module design, it could not be drained at night. Consequently, the liquid in the collector cooled overnight and the heat lost in this fashion had to be replaced each morning either by solar energy and/or by heat from storage. As mentioned above, 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/m² or 58,772 kJ/m² for the month of February 1978. This represents about 33 percent of the useful collected solar heat. And, because the collector was operated from sunrise to sunset, almost all of this heat loss came from storage.

4.8 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):

$$q = 0.399 + 0.00403 T_o \quad (1)$$

where

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

T_o = Outlet liquid temperature from the collector array, $^{\circ}\text{C}$

Equation (1) explains 90 percent of the observed variance in flow rate. If 25°C to 100°C is considered the practical solar collector outlet temperature operating range, then it is clear that the flow rate through this solar collector will vary from 0.5 up to $0.8 \text{ m}^3/\text{hr}$, or by a factor of 1.6. For the month of February 1978, the average flow rate through the solar collector was $0.585 \text{ m}^3/\text{hr}$ or $0.0287 \text{ m}^3/\text{hr}\cdot\text{m}^2$ of solar collector.

4.9 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 that:

$$\frac{Q_u}{SA_c} = \eta_o - U_L \left(\frac{T_o - T_a}{S} \right) - cW \left(\frac{\Delta T_o / \Delta t}{S} \right) \quad (2)$$

where

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

S = Solar insolation rate on a tilted surface, $\text{kJ}/\text{hr}\cdot\text{m}^2$

A_c = Solar collector array area, m^2

η_o = Instantaneous solar collector efficiency when $T_o - T_a = 0$, dimensionless

U_L = Solar collector heat loss coefficient, $\text{kJ}/\text{hr}\cdot\text{m}^2\cdot{}^{\circ}\text{C}$

T_o = Solar collector liquid outlet temperature, $^{\circ}\text{C}$

T_a = Ambient (outdoor) air temperature, $^{\circ}\text{C}$

c = Weighted average specific heat of the solar collector materials when the collector is filled with water, $\text{kJ}/\text{kg}\cdot{}^{\circ}\text{C}$ (for the O-I collector, $c = 2.27 \text{ kJ/kg}\cdot{}^{\circ}\text{C}$)

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

Q_u/SA_c = Apparent solar collector efficiency, dimensionless

$$\Delta T = T_o - T_a$$

ΔT_0 = Change in solar collector outlet liquid temperature during the time interval Δt , $^{\circ}\text{C}$

Δt = Time interval, hour

From Equation (2) it is clear that, while the solar collector is warming up during the first part of the day, ΔT_0 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, ΔT_0 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 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 plotted in Fig. 2, where each point is labeled with the hour of the day. ΔT_0 appears to be positive for the 8:00 am through 2:00 pm period and negative for the hours 2:00 pm through 5:00 pm. The intercept for both the morning and evening curves is about the same ($0.6 \pm .04$) and corresponds well with the theoretical values reported by O-I 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.

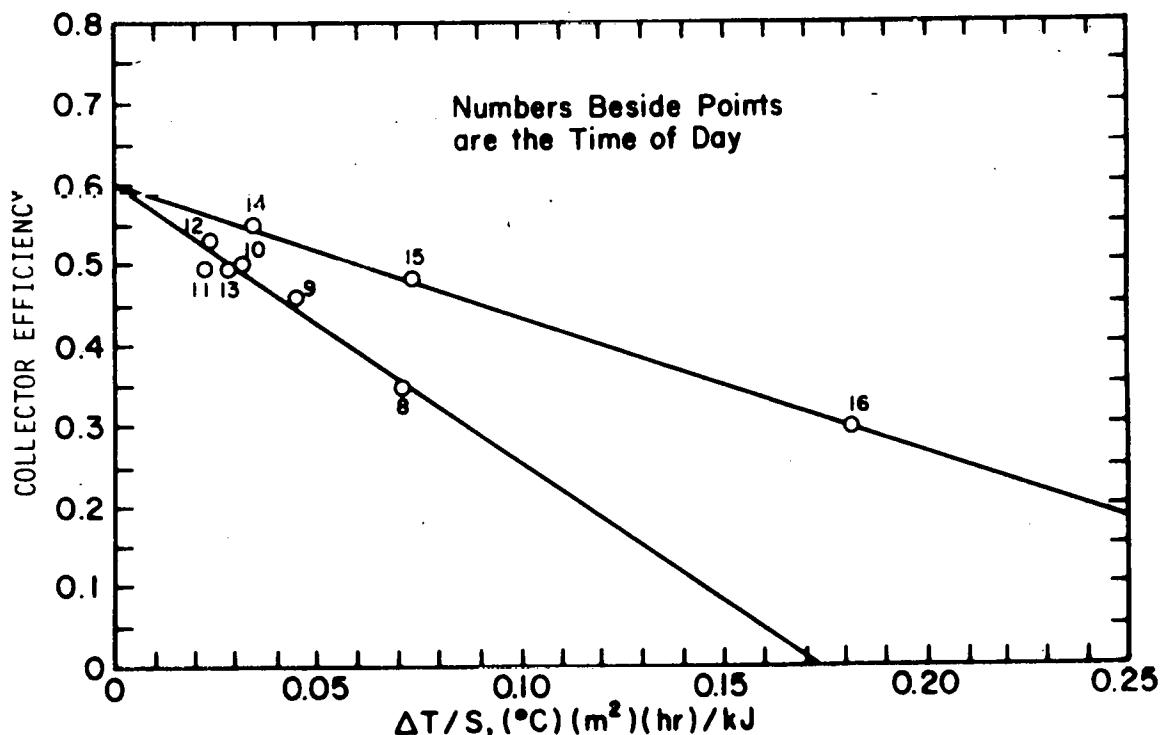


Fig. 2. Collector Efficiency as a Function of $\Delta T/S$

Between the hours of 9:00 am and 4:00 pm, the apparent collector efficiency was ≥ 0.459 . This is clearly important because 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 winter months was about 50 percent and that about 50 percent of the incident solar radiation was delivered as useful heat by this solar collector.

4.10 Electric Usage

Due to significant pressure drops in the O-I evacuated tube solar collector arrays (4.8 psi at a flow rate of 4.2 gpm), electrical power requirements during the first year of operation for the collector and exchanger pumps have been about 0.37 kW. When combined with the space and domestic hot water heating, the power requirements have been observed at a 0.5 kW level. Solar collection and operation of the space cooling subsystem required an electrical power level of 0.9 kW.

4.11 Collector Liquid

The use of pure water as the collector liquid has been shown to be inadequate because of freezing problems. The addition of ethylene glycol to the water has prevented freezing, but has other disadvantages. These disadvantages concern themselves principally with the additional difficulty of filling the O-I collectors with the collector liquid mixture and in potential boil off of the collector liquid.

4.12 Recommended Design Changes of the O-I Collector

It is recommended that the O-I collector be manufactured as a complete modular unit in order to prevent excessive installation costs. In addition, redesigning the manifold to connect tubes on only one side (with new module dimensions of 4 ft by 4 ft) provides for several advantages in the initial and continuing operation of the solar collector array. This recommended design change was previously communicated to the manufacturer.

5.0 SYSTEM PERFORMANCE

Table 1 lists the important parameters of the solar system's operation. Four sets of values are given. The first column (Feb. 1978) is the actual reduced data obtained and represents the use of the photocell on the collector pump control strategy. The second column is the data corrected to eliminate the adverse effect of the photocell (obtained by neglecting those periods of operation which yielded a net loss because of the running of the collector

Table 1. Energy Balances on CSU Solar House III
(monthly totals)

Row	Description	Column			
		(1) 1 Array	(2) 1 Array	(3) 2 Arrays	(4) 2 Arrays
1	Month/year	Feb 78	Feb 78*	Feb 78* (31 days)	March 77
2	Total monthly solar radiation (10 ⁶ kJ/month)	7.35	7.35	16.27@	23.52
3	Total monthly solar radiation during collector operations (10 ⁶ kJ/month)	7.27	6.93	15.35@	19.91
4	Useful collected solar heat (10 ⁶ kJ/month) †	2.78	3.66	8.12@	9.68
5	Heat losses (storage, solar DHW) (10 ⁶ kJ/month)	1.08	1.08	2.75	2.97
6	Total useful heat delivered to load by solar (10 ⁶ kJ/month)	1.70	2.58	5.37	6.71
7	Total useful heat delivered to load by auxiliary (10 ⁶ kJ/month)	16.99	16.99	14.74	2.46
8	Total heating load (10 ⁶ kJ/month) (rows 5,6,7)	19.77	20.65	22.86	12.14
9	Electrical	Collector/exchanger pumps	0.75	0.58	0.64
10	Energy	Circulating pump	0.46	0.46	0.51
11	Used	Total solar system power	1.21	1.04	1.15
12	(parasitic)	Blower	0.89	0.89	0.99
13	Total electricity used in house	22.32	22.82	21.29	15.50
14	Solar collector efficiency (%) (row 4 ÷ row 3)	38.20	52.80	52.90	48.60
15	Monthly fraction of load furnished by solar (row 4 ÷ row 8)	14.10	17.70	35.52	79.70
16	Monthly fraction of total solar radiation delivered to useful heat by solar system (row 4 ÷ row 2)	37.80	49.80	49.91	41.10

* Adverse effect of photocell eliminated by neglecting those periods of operation which yielded a net loss in the operation of the collector pump

@ Multiplied by two to consider both collector arrays

† Collector piping heat losses to ambient constitute about one half-million kilojoules

pump), and column 3 is column 2 multiplied by 31/28, to allow comparison with the March 1977 data in column 4 (and, in some cases, multiplied by two to account for the use of two arrays in March 1977 and one array in Feb 1978).

The effect of the photocell control strategy is seen to be devastating. Not only does the useful collected heat drop from 3.66 to 2.78 million kJ and the collector efficiency drop from 53 to 38 percent, but the parasitic power requirements required to operate the collector for about 2.5 hours more each day (and to pump at higher pressure drops when the collector liquid is cold in the mornings) jumps from 0.58 to 0.75 million kJ. The parasitic power problem is clearly of overriding importance because of the high percentage of electrical use to useful solar energy collected (27 percent for the photocell and 15.9 percent for the differential temperature strategy).

The higher level of power consumption of the circulating pumps can be attributed to the substantially higher heating load experienced in Feb. 1978 over that of March 1977. It should be pointed out that blower power was not greatly heavier in Feb. 1978 than in March 1977.

From rows 2 and 3 of Table 1, it is clear that the solar collector operated from sunrise to sunset during February. Because of this, parasitic power requirements were excessive (row 11). 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 9 of Table 1. 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 1 was about 27 percent greater than necessary for useful heat collection. Considering this latter factor, a more realistic value for the parasitic power requirements listed in row 9 of Table 1 would be $0.47 \times 10^6 \text{ kJ/month}$.

6.0 THERMAL STORAGE PERFORMANCE

During the month of Feb. 1978, the average temperature of the thermal storage was $43 \pm 5^{\circ}\text{C}$, that of the domestic hot water solar preheat tank was $39 \pm 6^{\circ}\text{C}$, and that of the domestic hot water auxiliary (electrically heated) tank was $67 \pm 3^{\circ}\text{C}$. The corresponding heat loss coefficients of the three water tanks (determined experimentally) were 51.5, 8.15, and 6.97 $\text{kJ}/\text{hr} \cdot ^{\circ}\text{C}$, 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 that 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.

7.0 PUBLICATIONS/REFERENCE (O-I COLLECTOR)

Publications resulting from work under the O-I portion of this contract include:

1. Ward, D.S., "Design Aspects of CSU Solar House III". Progress report to the Energy Research and Development Administration. Presented at the ERDA Contractors' Workshop, Silver Spring, MD, July (1976)
2. 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)
3. Ward, D.S. and Ward, J.C., "Residential Solar Heating and Cooling Using an Evacuated Tube Solar Collector". Annual progress report to ERDA, April (1977)
4. Ward, D.S. and Ward, J.C., "Design Considerations for Residential Solar Heating and Cooling Systems Utilizing Evacuated Tube Solar Collectors". Proc. ISES, Conf, Orlando, FL, June (1977)
5. Ward, D.S. and Ward, J.C., "Performance of Evacuated Tube Solar Collectors on CSU Solar House III". Presented at ERDA Contractors' Workshop, Reston, VA, August (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 Congress, New Delhi, India, January (1978)
7. Ward, D.S., Löf, G.O.G., and Uesaki, T., "Cooling Subsystem Design in CSU Solar House III". Solar Energy, 20, 2 (1978)
8. Ward, J.C. and Ward, D.S., "Performance of an Evacuated Tube Solar Collector in a Solar Heating System. Proc. ISES Conf; Solar Diversification, Denver, CO, August (1978)
9. Ward, D.S., "Solar Absorption Cooling Feasibility". Proc. ISES Conf; Solar Diversification, Denver, CO, August (1978)

10. Ward, D.S., "Realistic Sizing of Residential Solar Heating and Cooling Systems". Proc. ISES Conf; Solar Diversification, Denver, CO, August (1978)
11. Ward, D.S., "Solar Air-Heating and Cooling Systems for Residential and Light Commercial Applications". Proc. ISES Conf; Solar Diversification, Denver, CO, August (1978)
12. Ward, D.S., "Evaluation of a Residential Solar Heating and Cooling System - CSU Solar House III". Presented at DOE Contractors' Workshop, Washington, DC, September (1978)

Key references from the literature include:

13. Speyer, E., "Solar Energy Collection with Evacuated Tubes". ASME paper No. 64-WA/SOL, J. of Engring for Power (1964)
14. Ward, D.S. and Løf, G.O.G., "Design and Construction of a Residential Solar Heating and Cooling System". Solar Energy, 17, 1 (1975)
15. 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 Technical Memo TMX-71695, Cleveland, April (1975)
16. Beekley, D.C. and Mather, G.R., Jr., "Analysis and Experimental Tests of High-Performance Tubular Solar Collectors". Proc. ISES Solar Energy Cong. and Exposition, Los Angeles, June (1975)
17. Beekley, D.C., et al. "Sunpak Solar Collector by Owens-Illinois". Unpublished report, November (1975)
18. Beekley, D.C. and Mather, G.R., Jr., "Analysis and Experimental Tests of a High-Performance Evacuated Tubular Collector". Owens-Illinois, Inc., unpublished report (1976)
19. Ward, D.S. and Løf, 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)
20. Ward, D.S., Karaki, S., Løf, G.O.G., and Winn, C.B., "Sizing, Installation and Operation of Systems". Gov't Printing Ofc, Number 003-011-00085-2, October (1977)
21. Moan, K.L., "Evaluation of an All-Glass, Evacuated, Tubular, Non-Focusing, Non-Tracking Solar Collector Array". First annual progress report to the Department of Energy, Report COO-2919-3, December (1977)

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 flat-plate 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 liquid with a single cover selective surface flat-plate solar collector and thermal storage tank (approximately 4,500 liters (1200 gallons) of water). The heated water from thermal storage is pumped (1) through a water-to-air heat exchanger for space heating, (2) through a double-walled water-to-water heat exchanger for DHW heating, and/or (3) to the generator of a 25,320 kJ/hr (2-ton) lithium bromide absorption chiller for space cooling. The LiBr chiller uses solar heat to operate the unit, which cools by removing heat from water in an evaporator and discharging this heat to the exterior of the building. The resulting chilled water is placed in cool storage tanks and is pumped through a water-to-air heat exchanger for space cooling. An automatic control system provides for all functions of the system operation.

CSU Solar House III is fully instrumented for the purpose of 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

Preliminary performance and pertinent operating experience with the Chamberlain liquid-heating flat-plate solar collector integrated with the CSU Solar House III heating and cooling system has been acquired and a preliminary analysis accomplished. The initial analysis utilized primarily data from the months of July and August (1978) along with an analysis of the operation of the solar cooling system which utilizes the Yazaki 2-ton (25,300 kJ/hr) 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 765 ft² (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 3 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 Chamberlain 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 (July through Sept.) of continuous data on the system's performance had been acquired and a preliminary analysis accomplished on two-thirds of these data as of this report. Two week summaries of the acquired data (corresponding to different control strategies) are shown in Table 2. 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 Systems Operational Results (Ref. [2]).

Table 2. 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 53.3m². The total solar collector area is 58.6 m². It should be noted that the gross collector area, including manifolds and interconnections between individual modules, is approximately 70 m².

** Based on energy balances



Fig. 3. CSU Solar House III with Chamberlain Collectors

A summary of this report is included below under the section entitled "System Performance".

- Operating experience and system performance of both the O-I and Chamberlain collectors have led to generalizations on the utility of operational results of solar heating and cooling systems. These efforts have also been reported at the Conference on Solar Heating and Cooling Systems Operational Results (Ref. [3]) and are included in this report as Appendix A. Additional information is included in Ref. [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).

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 cooling tower loop (implying a minimal distance between the exterior cooling tower 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.

5.0 SYSTEM PERFORMANCE

Table 2 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 (176°F). On 15 July, this set point was reduced to 74°C (165°F). During the first two-week period, the collector boiled frequently, 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 (14°F). Therefore, in order to avoid boiling in the collector when using water, the maximum water storage tank temperature is 87°C (189°F) (the local boiling point of water at the elevation of CSU Solar House III is 95°C, (203°F)). During the first two-week period these limitations allowed a useful operating temperature range of the thermal storage unit of 80 to 87°C (176 to 189°F), or 7°C (13°F). During the second two-week period, however, the effective temperature was increased to 13°C (24°F, (189-165°F)), so that the heat storage capacity was effectively doubled.

At the beginning of the third two-week period (Aug. 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 Aug 16-31, however the load increased significantly (see Table 2, row 9).

Table 3 summarizes a portion of the data analysis. The effect of the change in control strategy on 15 July is clearly 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 COP of the chiller due to the lower input temperature to the generator.

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. The first (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, but 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 represents only 18.9, 26.2 and 27.4 percent of the useful heat collected (row 24 ÷ 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 ÷ 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.

Table 3. 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 $(1 + 1/\text{COP})$ 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)]

The average COP of the chiller listed in Table 3 does not reflect adequately the operation of the Yazaki 2-ton 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.

5.1 Electrical Consumption

Table 4 presents data on the electrical consumption of the solar and auxiliary systems in providing solar cooling to the building. Table 5 presents certain important ratios of energy flows. It should be noted that frequent cycling by the auxiliary boiler caused additional usage of electricity (i.e., circulating pump - row 28 and cooling unit power - row 30) during the month of August. The larger power usage for collection (row 27) for the period 1-15 July is larger due to frequent boiling of the solar collector. The auxiliary electric usage (row 26) is based on the electric meter readings, whereas row 7 of Table 2 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 5, 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 two percent and, in any case, not greater than three 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:

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

Table 4. 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 5. 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

These COP's 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 total 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 solar system COP (without auxiliary). These COP's 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, .33 mkJ was lost in the collector piping, about 7.5 percent, a fairly conservative value. An additional heat loss by the thermal storage reduced the useful heat another 0.96 mkJ so that only 3.07 mkJ was available for solar cooling (about 70 percent of the useful heat collected). The circulating pump used an additional .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, the solar average COP was 0.58. The load pump then 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 the 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 4.36. 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 usage by optimization of piping
- (4) Reduce cooling tower pumping power 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 Exchnager (see Ref [6]).

These options are listed with potential energy savings in Table 6. 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 15-31 July.

Table 6. 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)

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 considered 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. COP's less than 10 cannot be considered feasible.

5.2 System Effects on Collector Efficiency

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

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 $P \propto (\dot{m})^3$), but also the efficiency of collection for the lower flow rate was slightly higher (about three percent in daily collector efficiency).

This is explained in Fig. 4, which is plotted from data obtained. During the cooling season, the majority of the load occurs during the day. 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 directly supplied to the chiller by what constitutes an effective short 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 $1.5 \text{ m}^3/\text{hr}$ flow rate, 6 to 8 hours of collection is taking place in the region indicated in Fig. 4 as (A). However, at the $2.8 \text{ m}^3/\text{hr}$ 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 Fig. 4. 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 (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 (high collector outlet temperatures) is that the solar supply to chiller can start earlier

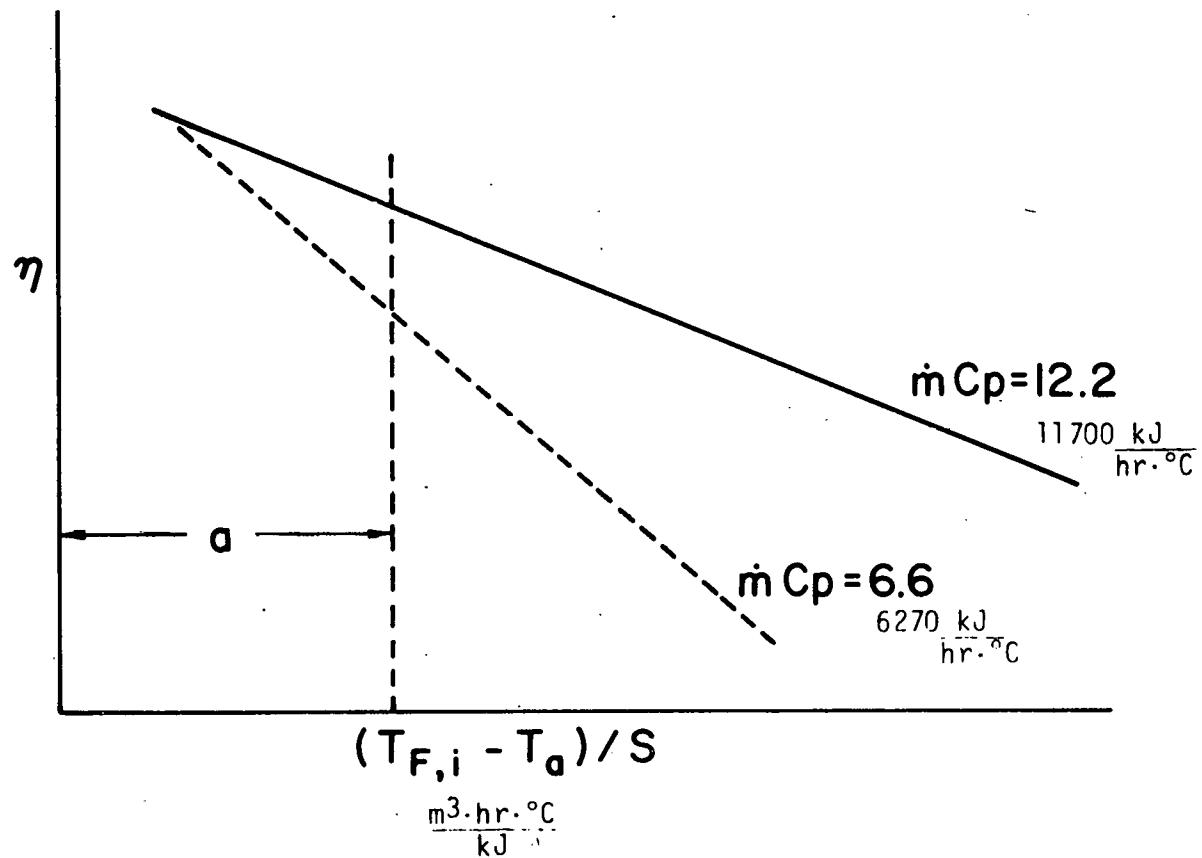


Fig. 4. Collector Efficiency as a Function of Collector Loop Flow Rates

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 in a higher COP (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.

6.0 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, in reducing the minimum temperature to the absorption chiller, and in 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.

7.0 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 the DOE Contractors' Workshop, Washington, DC, Sept. (1978)
2. Ward, D.S., Ward, J.C. and Oberoi, H.S., "Solar Cooling Performance in CSU Solar House III". Proc. Conf. Solar Heating and Cooling Sys. Oper. Results, Colorado Springs, November (1978)
3. Ward, D.S., "Utilization of Operational Results". Proc. Conf. Solar Heating and Cooling Sys. 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. ISES Conf; Solar Diversification, Denver, August (1978)

APPENDIX A
"Utilization of Operational Results"

Presented at the
Solar Heating and Cooling Systems
Operational Results Conference
Colorado Springs, Colorado
November 1978

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]:

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

where

η = Solar collector instantaneous efficiency, dimensionless

F_R = Solar collector heat recovery factor, dimensionless

(τa) = Collector transmissivity-absorptivity product, dimensionless

U_L = Collector heat loss coefficient, Btu/hr·ft²·°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·ft² of collector

Experimental data on solar collector efficiency provides values for the collector's characteristics, $F_R(\tau_a)$ and $F_R U_L$. 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:

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

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 η_{daily} is:

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

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 (η) in equation (1) equal to zero. The result is:

$$(H_T)_{\text{OT}} = \frac{F_R U_L}{F_R(\tau_a)} (T_i - T_a) \quad (3)$$

where

$$(H_T)_{\text{OT}} = \text{operating threshold of a collector, Btu/hr.ft}^2$$

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

$$(H_T)_{\text{OT}} = \frac{F_R U_L}{F_R(\tau_a)} \left((T_i - T_a) + \Delta T_c + \frac{Q_L}{A_c F_R U_L} + \frac{E}{A_c F_R U_L} \right) \quad (4)$$

where

$$\Delta T_c = \text{increase in collector operating inlet temperature due to control strategy, } ^\circ\text{F}$$

$$Q_L = \text{Heat losses in collector loop piping and/or ducting during collector operations, Btu/hr}$$

$$A_c = \text{Collector area, ft}^2$$

$$E = \text{Thermal equivalent of electrical energy used in operating solar collector, Btu/hr}$$

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:

$$\eta_s = \eta_{\text{daily}} \left(1 - \frac{Q_L}{Q_u} - \frac{Q_{SL}}{Q_u} - \frac{E}{Q_u} \right) \quad (5)$$

where

$$\eta_s = \text{Overall solar system efficiency, dimensionless}$$

$$Q_u = \text{Useful heat collection by solar collector array, Btu/day}$$

$$Q_{SL} = \text{Heat losses from thermal storage, Btu/day}$$

$$\bar{Q}_L = \text{Daily heat losses in collector and system loops piping and/or ducting, Btu/day}$$

$$E = \text{Thermal equivalent of daily electrical energy used in all solar system operations, Btu/day}$$

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:

$F_R(\tau_a) = 0.75$ $F_{RUL} = 0.825 \text{ Btu/hr.ft}^2 \cdot {}^\circ\text{F}$
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 $(H_T)_{OT} = 1.1 \text{ Btu/hr.ft}^2 \cdot {}^\circ\text{F} (T_i - T_a)$. For January we may assume conditions of $T_a = 30^\circ\text{F}$ (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 $T_i - T_a = 90^\circ\text{F}$. Thus $(H_T)_{OT}$ is just less than 100 Btu/hr.ft^2 .

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 $T_{\text{collector}} - T_{\text{storage}} = 10^\circ\text{F}$, average, or $\Delta T_c = 5^\circ\text{F}$ (the heat exchanger ΔT having already been accounted for).

With two inch fiberglass insulation on the collector loop piping, we can expect a heat loss of about $2,000 \text{ Btu/hr}$ (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 $1/2 \text{ hp}$, or about $1,270 \text{ Btu} (\text{electric})/\text{hr}$ 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:

$$E = (2.4)(1,270 \text{ Btu} (\text{electric})/\text{hr}) \\ E = 3,048 \text{ Btu} (\text{thermal})/\text{hr}$$

Under these conditions equation (4) becomes:

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

or

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

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 $\Delta T_{ext} = T_i - T_a = 125^\circ\text{F} - 30^\circ\text{F} = 95^\circ\text{F}$ and $\Delta T_{int} = T_{\text{storage}} - T_{\text{room}} = 115^\circ\text{F} - 70^\circ\text{F} = 45^\circ\text{F}$. 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:

$$\bar{Q}_L = 32,000 \text{ Btu/day} + 616,000 \text{ Btu/day}$$

where (6) 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 $Q_{SL} = 812,000 \text{ Btu/day}$.

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

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

$$Q_u = A_c [H_T F_R(\tau_a) - F_{RUL} (T_i - T_a)] \quad (7)$$

Recognizing that our operating threshold (from equation 6) is 118 Btu/hr.ft^2 , we might consider two values of the average daily solar radiation during the operation of the collector. If $H_T = 150 \text{ Btu.hr.ft}^2$:

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

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

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

$$\eta_s = \eta_{\text{daily}} \{ 1 - \frac{32,000 + 616,000}{102,500} - \frac{812,000}{102,500} - \frac{30,000}{102,500} \}$$

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

$$\eta_s = \eta_{\text{daily}} \{ .395 - .2736 \} \quad (8a)$$

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., $H_T = 250 \text{ Btu/hr}\cdot\text{ft}^2$), Q_u becomes 330,000 Btu/day and we obtain:

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

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·ft² to 124 Btu/hr·ft² and decrease the system efficiency (equation 8a) from:

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

to

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

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), $Q_{SL} = 18,000 \text{ Btu/day}$ (50 percent increase) and

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

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 Q_L is increased from 32,000 Btu/day to 96,000 Btu/day. The system efficiency/daily collector efficiency ratio is reduced to:

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

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

Abstracts of Papers to be Submitted to the
International Solar Energy Society Congress

to be held in
Atlanta, Georgia
August 1979

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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